Generic Letter 96-06 Waterhammer Issues Resolution

User's Manual - Non Proprietary

Technical Report

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User's Manual – Non Proprietary

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Final Report, April 2002

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REPORT SUMMARY

The U.S. Nuclear Regulatory Commission (NRC) Generic Letter 96-06 identified potential issues for waterhammer effects during postulated events that can cause potential damage to service water systems. The User's Manual (Volume 1) provides methods recommended for evaluating the impact of potential waterhammer loads on plant service water system components. This Technical Basis Report (Volume 2) provides detailed background information and the technical basis for the methods defined in the User's Manual.

Background
Following either a Loss of Coolant Accident (LOCA) or a Main Steam Line Break (MSLB) concurrent with a Loss of Offsite Power (LOOP), pumps that supply cooling water to fan cooler units (FCUs) and fans that supply air to FCUs will temporarily lose power. Cooling water flow will stop due to the loss of pump head. Boiling may occur in FCU tubes, causing steam bubbles to form in FCUs and pass into the attached piping, creating steam voids. As service water pumps restart, accumulated steam in the fan cooler tubes and piping will condense and pumped water can produce a waterhammer when the void closes. Hydrodynamic loads introduced by such a waterhammer event could potentially challenge the integrity and function of FCUs and associated cooling water system components, as well as pose a potential challenge to containment integrity.

Objective
To provide a technical approach, benchmarked against test data, for evaluating potential waterhammer loads and their impact on service water systems anticipated during event scenarios identified in NRC Generic Letter 96-06.

Approach
A testing and analysis program was undertaken to develop methods for realistic evaluation of waterhammer loads. The program's scope was designed and validated using a Phenomena Identification and Ranking Table (PIRT) assessment. The program focused on fluid condition characteristics for this specific event. Conditions included low system pressure, non-condensables in the water, a low and controlled velocity of column closure, and a thermal layer (a hot layer of water in contact with accumulated steam). The testing program included several types of tests: column closure tests designed to simulate plant conditions, condensation-induced waterhammer tests, and gas release rate tests. The tests characterized the waterhammer pressure pulse, pressure wave propagation in the pipe, and reaction of the pressure transient in prototypical piping supports.

Results

The primary conclusions from the project are the following:

- Waterhammers produced by a LOOP-only event will be more severe than those produced by a LOOP/LOCA or LOOP/MSLB event if voiding occurs during a LOOP alone and if the system valve and pump operation is the same as the LOOP/LOCA or LOOP/MSLB.
- Condensation-induced waterhammer events (CIWH) are limited in magnitude and duration in low-pressure service water systems and do not create a limiting transient, particularly in systems that experience CCWH.
- Non-condensables (air or nitrogen) in the void diminish the severity of the waterhammer events.
- Column Closure Waterhammer (CCWH) events can be evaluated using Method of Characteristics or Rigid Body Model methods considering steam and non-condensables pressurization in the void.
- A simplified, trapezoidal characterization of the pressure-time history can be used to produce a conservative structural loading of the piping system.

EPRI Perspective

The testing and analyses performed within the framework of this program demonstrate that waterhammers following a LOOP and LOCA (or MSLB) are not as severe as originally believed. Methods are provided for evaluating and qualifying systems that contain fan coolers. These methods, and supporting verification, provide realistic predictions that can assure plant safety and minimize plant modifications. Example applications to both an open-loop and a closed-loop service water system are included to help facilitate applications by utility engineers. Although the report's focus has been on issues identified in GL96-06, this technical approach can be used to evaluate similar waterhammers that could occur from water column closures in other than service water systems.

When used for resolution of Generic letter 96-06 issues on a specific plant basis, the information contained in this report must be used in a manner that is consistent with the requirements specified by NRC staff. A copy of the NRC Safety Evaluation Report and supporting EPRI correspondence to NRC are included in the appendices to this report. Additional guidance or restrictions required by NRC staff are not included in this document.

Keywords

Reactor safety and licensing
Waterhammer
Piping loads

ABSTRACT

A testing and analysis program was undertaken to develop a technical approach, benchmarked against test data, for evaluating potential waterhammer loads and their impact on service water systems anticipated during event scenarios identified in NRC Generic Letter 96-06. The program's scope was designed and validated using a Phenomena Identification and Ranking Table (PIRT) assessment. Conditions that were evaluated included low system pressure, non-condensables in the water, a low and controlled velocity of column closure, and a thermal layer (a hot layer of water in contact with accumulated steam). The testing and analyses performed within the framework of this program demonstrate that waterhammers following a LOOP and LOCA (or MSLB) are not as severe as originally believed. Methods are provided for evaluating and qualifying systems that contain fan coolers. These methods, and supporting verification, provide realistic predictions that can assure plant safety and minimize plant modifications. Example applications to both an open-loop and a closed-loop service water system are included to help facilitate applications by utility engineers. Although the report's focus has been on issues identified in GL96-06, this technical approach can be used to evaluate similar waterhammers that could occur from water column closures in other than service water systems.

When used for resolution of Generic letter 96-06 issues on a specific plant basis, the information contained in this report must be used in a manner that is consistent with the requirements specified by NRC staff. Additional guidance or restrictions required by NRC staff are not included in this document.

ACKNOWLEDGMENTS

The technical work described in this report was performed with valuable guidance and inputs from the Waterhammer Expert Panel consisting of Professor Peter Griffith of MIT, Dr. Fred Moody (Consultant), and Dr. Ben Wylie of the University of Michigan. Dr. Robert Henry and Mr. Robert Hammersley of Fauske and Associates performed work on the project, and provided valuable input and data that were used to develop the positions. The overall project direction and funding was provided by the Waterhammer Utility Advisory Group consisting of the following:

Wagoner, Vaughn, Chairman	Carolina Power & Light
Azzarello, Leonard	Duke Energy
Brown, Timothy	Duke Energy
Chang, Hsin-Yung	New York Power Authority
Connor, Todd	Baltimore Gas & Electric
Conry, Marlin	Wisconsin Electric
Cozens, Kurt	Nuclear Energy Institute
Fish, Hamilton	New York Power Authority
Ginsberg, Art	Consolidated Edison
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Husaini, Mahmood	Southern California Edison
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Randels, Raub	ComEd
Ray, Dillip	Southern Nuclear Operating Company
Riat, Dharam S.	Consumers Energy
Robinson, Mike	Duke Energy
Rochino, Lee	Rochester Gas & Electric
Thomas, Steve	Northern States Power
Webb, Thomas	Wisconsin Public Service

In addition, interim report reviews and comments provided by Nuclear Regulatory Commission staff and consultants are gratefully acknowledged.

EXPERT PANEL REVIEW OF THE EPRI REPORTS

The undersigned have worked independently as an "Expert Panel" with the team that has prepared this report. The objective of the Expert Panel efforts was to perform an independent review of the Technical Basis Report (TBR) and User's Manual to support their technical completeness and adequacy to resolve the questions raised by NRC's Generic Letter 96-06.

The Expert Panel consisted of three members – Dr. Peter Griffith, Chairman, Dr. Fred Moody, and Dr. Benjamin Wylie. The Expert Panel participated in four review meetings attended by sponsoring utilities, EPRI, and EPRI's contractor Altran Corporation. The NRC staff and their consultants participated in two of these review meetings. In addition, the Expert Panel members were involved in periodic individual reviews of the draft contents of the TBR during its development. Guidance was provided as requested throughout the project.

The panel performed a review of the overall project plan, the approach for resolving individual issues raised by the NRC, the test plans and the tests performed, data analysis, analysis method development, and conclusions drawn in the Technical Basis Report (TBR).

The Expert Panel has completed its review of the TBR and associated User's Manual. We provide the following conclusions:

- The Expert Panel agrees that the PIRT analysis performed has identified all important phenomena and processes that may impact the waterhammer loads during the transients identified in Generic Letter 96-06.
- The Expert Panel agrees that the technical approach documented in the TBR and User's Manual is technically justifiable and validated against appropriate data for plant application.
- The Expert Panel agrees that, considering the lack of credible threat to the safety functions and the low probability of the potential waterhammer events identified in Generic Letter 96-06, the proposed approach is conservative and uncertainties have been adequately addressed in application to plants.
- The Expert Panel agrees that the guidelines for application by a utility user are sufficiently detailed and example applications provided are representative of real plant applications.

We have been mindful during our review of the stochastic behavior of the events being considered and the overall risk to the safety of the plant. It is judged to be advantageous to be as realistic as possible, but to not compromise the evaluation of the adequacy of the systems being considered. The conservative quantification of every variable that may affect either the waterhammer or the response of the piping was not considered to be the objective of the study or our review. The overall acceptability of the system, considering the known conservatisms, led the review team to conclude that the overall conclusions were acceptable.

In summary, the Expert Panel members endorse the TBR and User's Manual as a credible technical basis for providing high assurance that the integrity of the service water systems can be maintained.

Signed by: Peter Griffith (Chairman)
Frederick J. Moody}
E. Benjamin Wylie

EXECUTIVE SUMMARY

ABSTRACT

The United States Nuclear Regulatory Commission (NRC) Generic Letter 96-06 identified potential issues with cooling water systems following either a Loss of Coolant Accident (LOCA) or a Main Steam Line Break (MSLB) concurrent with a Loss of Offsite Power (LOOP). The potential for damage due to waterhammer associated with the postulated event was included in the Generic Letter.

Column closure and condensation induced waterhammers may occur in the fan cooler system during this transient. Piping loads produced by waterhammer must be evaluated to assure that the integrity of the system is maintained.

SYSTEM CHARACTERISTICS

The fan cooler systems include pumps, piping, fan cooler units (FCUs), and other in-line components such as valves and orifices. The systems in all plants are categorized as either open loop or closed loop. Open loop systems are typically found at fresh water sites. Closed loop systems are typically found at salt-water sites and BWR units. The systems are designed to remove heat from containment during a Loss of Coolant Accident (LOCA) or Main Steam Line Break (MSLB). With normal service water flow into the FCUs, the system will remain water solid.

The systems have great similarity from plant-to-plant. Open loop systems have FCUs that are generally at higher elevations in containment. The pumps are outside containment, near the raw-water source, and the FCUs, supply and discharge pipes in containment form a containment boundary. The closed loop system design is very similar to open loop plants, but there is a heat exchanger in the system and an expansion tank.

BACKGROUND

If a steam void forms in a pipe containing service water, sudden collapse of the void can generate waterhammer pressure loads. Steam voids can form by depressurization *(flashing)* if a loss of offsite power (LOOP) causes pump coast down, and the corresponding loss of pump head reduces local water pressure to the saturated state at elevated points in the flow path. Voids also can be formed by heat transfer *(boiling)* if a loss of coolant accident (LOCA) or main steam line break (MSLB) raises the external temperature enough to heat the water in the FCUs to its saturation temperature.

Void formation by flashing and boiling are considered in service water scenarios associated with containment fan cooler units (FCUs) in nuclear plants. One scenario consists of a LOOP, whereby pumps coast down, causing flow velocity reduction to heat exchangers at a high elevation. So-called "open loop" plants have FCUs that discharge to a low elevation receiver at approximately atmospheric pressure, P_{atm}. If the water temperature is T, with a corresponding saturation pressure $P_{sat}(T)$, the receiver pressure can only support a water column of height corresponding to the static head H. If the heat exchanger elevation is higher than H, which is normally the case, a steam void will form from flashing. A negligible amount of non-condensable gas is released from the small amount of water that flashes in a LOOP-only scenario.

Boiling also can supply steam to the void if the LOOP is concurrent with a LOCA or MSLB. Significant quantities of non-condensable gas can be released from the larger amount of water that boils, relative to the amount of water that flashes during a depressurization. Most of the released gas will occupy the steam void, although some gas may remain in the water as small bubbles.

Open loop plants have FCUs that discharge to a low elevation receiver at atmospheric pressure, and can void due to LOOP alone. In a closed loop plants, the FCUs are in a loop with a volume control tank at a higher elevation than the FCU, so that if the pumps lose power during a LOOP, no void by flashing will form. Voids can form only from boiling in closed loop systems following a LOOP with a LOCA or MSLB.

Once voids are formed in service water piping, they can collapse once the pumps restart and the moving water column closes onto the stationary column, referred to as column closure waterhammer (CCWH). If the void has formed from flashing (LOOP-only), pump restart causes CCWH proportional to the velocity of the closing water column. If the void has formed from boiling (LOOP with LOCA or MSLB), the CCWH that will occur is influenced by the rate of steam condensation and presence of non-condensable gases in the void. These can reduce the relative velocity of impacting water columns, and consequently reduce the resulting waterhammer disturbances.

While the void is forming (prior to pump restart), void collapse can also occur if steam rapidly condensing on cooler water or pipe surfaces causes a transition to slug flow. This can occur only in horizontal pipes and is referred to as a condensation induced waterhammer (CIWH). The severity of this waterhammer is related to the pressure of the system and is reduced by non-condensable gas in the pipe at the time of the waterhammer.

Waterhammer caused by a void collapse – either a CCWH or a CIWH – generates a transient pressure disturbance in the pipe. The pressure disturbance moves in both directions from the location of the closure and causes unbalanced loads in the piping system that loads the pipe and the pipe supports.

The objective of this study is to present a technical approach for evaluating the loads on the piping and pipe supports from waterhammer caused by CCWH and CIWH. This technical approach can be used to demonstrate that the integrity of the system will be maintained in the event that these waterhammer events occur.

CONDENSATION INDUCED WATERHAMMER

The condensation induced waterhammer (CIWH) pressure magnitude is related to the pressure in the pipe at the time of the waterhammer and is affected by the sonic velocity of the water local to the impact location, non condensables in the water and steam, and the size of the void. In the fan cooler system following a LOOP and LOCA or MSLB, non-condensables are present in the water and steam and the system pressure is low, approximately atmospheric pressure.

Analytical methods are inadequate to accurately calculate the pressure magnitude of the CIWH in the specific conditions of the service water system. Because of this, a testing program was used to determine the severity of the CIWH. Eighty-two CIWH tests were conducted, including thirty-seven tests with normally aerated water and forty-five tests with deaerated water. A summary of the tests performed in normal water, prototypical of the water expected to be in the service water system, is shown below. Pressure calculated using the Joukowski equation with an equal bubble and liquid volume (pipe half full) and a sonic velocity of 4,600 feet per second is also shown.

Waterhammer Peak Pressure vs. Driving Steam Pressure – Normal Water

All the CIWH data had a relatively constant pressure impulse, determined as the area under the pressure-time curve. Higher pressure pulses had very short durations and the low magnitude pulses had longer durations. A plot of pressure magnitude versus the time duration of the pulse is shown in the figure below. A line showing constant impulse behavior is also provided. This curve also includes deaerated water test results. It can be seen that the impulse is independent of whether the water is aerated or deaerated.

Waterhammer Peak Pressure vs. Duration – Normal and Deaerated Water

The effect of the constant impulse behavior is evident in the support load response. The support loads were measured at the end of the horizontal test section and are plotted versus the peak waterhammer pressure for both normal water and deaerated water. The pipe support force is limited for waterhammer pressures above approximately 150 psig. The diminished structural response at high pressures is caused by the very brief pulse duration relative to the structural response.

Support Load vs. Waterhammer Peak Pressure – Normal and Deaerated Water

The impulse for the CIWH has been compared to the pressure impulse for a typical range of column closure waterhammers. The CIWH pressure impulse is significantly less than the pressure impulse that occurs for a CCWH in the same system. It has also been shown during the tests that the waterhammer pressures are independent of the draining flow velocity and the pipe length. A scaling analysis concluded that both the waterhammer pressure rise duration and the measured absolute pressure spike in the smaller experimental pipes would be higher than the pressures expected in larger pipes.

The conclusion from the CIWH testing program was that the CIWH waterhammers, for low-pressure service water systems, are limited in magnitude and duration and do not create a limiting transient for the piping system, particularly in systems that also experience a CCWH.

COLUMN CLOSURE WATERHAMMER

The column closure waterhammer (CCWH) occurs after the pumps are restarted and the final closure occurs. An accepted method of evaluation for CCWH is the method of characteristics (MOC). The method of characteristics includes closure velocity reduction due to potential pressurization of non-condensables and steam in the void and wave propagation effects in the fluid.

The method of characteristics method is capable of accurately analyzing the cushioning that will result from the pressurization of non-condensable gas that is accumulated in the void. Testing was required to provide information necessary to determine the steam condensation rate in the void. Condensation rate was determined by comparison of the model results to test data. Once the steam condensation rate was determined, very close correlation between the calculation of the waterhammer using the method of characteristics and the test results was achieved.

Data was also required to determine the amount of gas that would be released into the void by boiling. Gas release tests were performed that showed that approximately 50% of the gas in solution in water that is exposed to boiling would be rapidly released by boiling conditions similar to those that would exist in the FCUs during a LOCA or MSLB. A smaller amount of gas – approximately 24% – is released from the water that steam passes through.

Analyses were performed for the column closure waterhammer using both the method of characteristics and a simplified rigid body model (RBM). The rigid body model differs from the MOC in that the water column closing the void is treated as a solid mass with no wave propagation effects. The RBM was used to develop solution sets for various input parameters that were tabularized for use in plant applications. The rigid body model was shown to be conservative relative to the method of characteristics when wave reflection was taken into account. A comparison between the rigid body model and the method of characteristics is shown below.

RBM/MOC Pressure Comparison (psig)

Both the method of characteristics and the rigid body model were correlated to test results. The following table provides results from the modeling of a specific set of test conditions using the method of characteristics, the rigid body model, and the test results.

RBM and MOC Comparison Against Test Data			
Test Case	**MOC Peak Pressure (psig)**	**RBM Peak Pressure (psig)**	**Test Data: Peak Pressure Range (psig)**
20 psig driving pressure; 20 ft column; Normal Water	[ ]	[ ]	[-]
70 psig driving pressure; 20 ft column; Normal Water	[ ]	[ ]	[-]
45 psig driving pressure; 3 ft column; Normal Water	[ ]	[ ]	[-]
45 psig driving pressure; 17 ft column, Deaerated Water	[ ]	[ ]	[-]

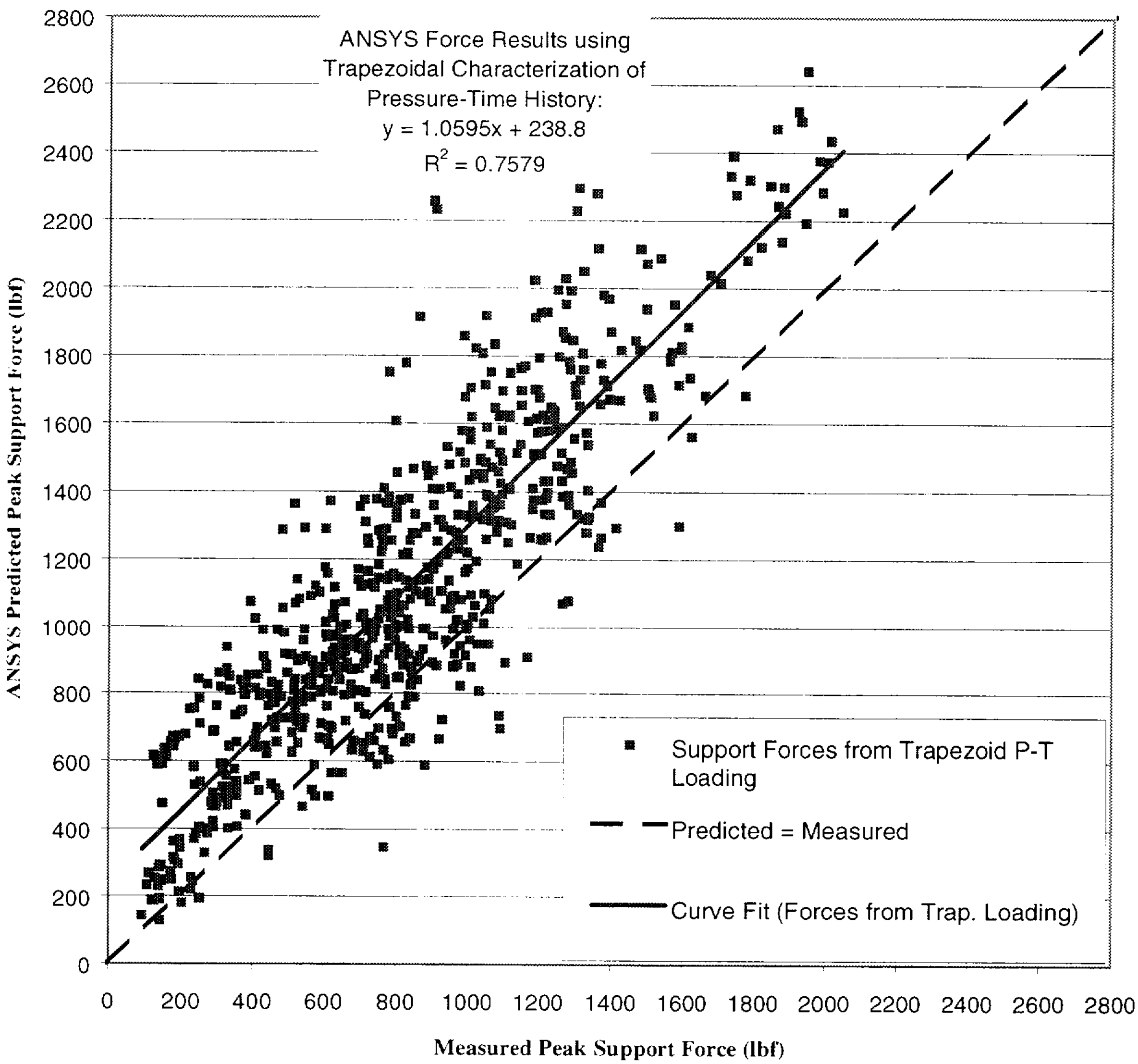
The primary conclusions from the CCWH analyses are the following:

- Pressures from CCWH are mitigated by gas and steam pressurization in the void.
- The method of characteristics provides a means of accurately simulating all aspects of pump startup in a system with vapor pockets of steam plus non-condensable gas, and it predicts the peak pressure and rise times of CCWH events.
- The reduction in velocity and the mitigation of the waterhammer can also be calculated by simplified rigid body model.

SUPPORT LOADS

The support loads were measured in CCWH tests that were performed. It was determined that a simplified trapezoidal model was effective to characterize the actual pressure pulses. This trapezoidal loading was developed to reflect fundamental theory, capture the pulse magnitude, rise time, and duration, and simplify the transient pressure response into a set of defined pressure-time (P-T) points for use in a structural calculation. The effectiveness of the trapezoid model was tested by comparing 1) the response of a finite element analysis model with 2) loading from the idealized trapezoids and 3) loading with actual pressure-time histories to the measured force response from the tests.

The following plot shows the results from the trapezoidal characterization of the pulses for <u>all tests</u> analyzed using the same analytical model. These force responses are plotted versus the measured force data for three piping supports restraints. It can be seen that the trapezoidal characterization of the pressure time pulse shapes is conservative for structural modeling.



Trapezoid Characterization (All Tests)

USER'S MANUAL AND TECHNICAL BASIS REPORT

The User's Manual (UM) (Volume 1) and the Technical Basis Report (TBR) (Volume 2) provide guidance for the evaluation of potential waterhammer events resulting from postulated concurrent LOOP/LOCA or LOOP/ MSLB scenarios. These reports are not intended to replace individual plant analyses, but to provide a methodology that can assist in the calculation of the waterhammer characteristics and be used to evaluate the potential effects of waterhammer arising from the conditions described in GL96-06. The Technical Basis Report supports the methods described in the User's Manual for the evaluation of the service water system loads. The TBR presents test data, theory, and background information intended to justify the methodology and inputs provided in the User's Manual.

NOMENCLATURE

Standard Abbreviations

CCWH	Column Closure Waterhammer
CIWH	Condensation Induced Waterhammer
FCU	Fan Cooler Unit
FSI	Fluid Structural Interaction
LOCA	Loss of Coolant Accident
LOOP	Loss of Offsite Power
LOOP/LOCA	Generic reference to the combined LOOP/LOCA or LOOP/MSLB event
MSLB	Main Steam Line Break
PIRT	Phenomenon Identification and Ranking Table
RAI	USNRC Request for Additional Information
SW	Service Water
TBR	Technical Basis Report
UM	User's Manual

Symbols, Units, and Typical Values for Constant Terms

1. A — internal cross sectional area of pipe (flow area) (in^2)
2. A_{or} — area of an orifice or flow restriction (in^2)
3. a — acceleration (d^2x/dt^2) (ft/sec^2)
4. C — sonic velocity (ft/sec); also used as orifice flow coefficient
5. C_f — sonic velocity based on fluid compressibility (ft/sec)
6. C_{pipe} — sonic velocity based on pipe flexibility (ft/sec)
7. CON_{air} — mass concentration of air in water (mg/liter)
8. CON_{O2} — mass concentration of oxygen in water (mg/liter)
9. D — pipe inside diameter (in)
10. dO_2 — amount of dissolved oxygen in water (mg/L)
11. f — pipe friction factor
12. Fr — Froude number
13. g — gravitational acceleration (32.2 ft/sec^2)
14. h_{fg} — latent heat of vaporization per unit mass (BTU/lbm)
15. h_{cds} — condensing heat transfer coefficient ($BTU/hr \cdot ft^2 \cdot {}^\circ F$)

16.	h	pressure head (ft of water)
17.	K	resistance coefficient from: head loss = $K \cdot V^2/2g$
18.	K_{imp}	constant representing condensation induced waterhammer impulse (psi·sec)
19.	K_R	constant used in curve fit for determining rise time (1/ft)
20.	k	constant used in Joukowski equation to describe closure type (k = 0.5 for water on water closure and k = 1.0 for water on hard surface closure)
21.	L	length of piping system (ft)
22.	L_w	length of water column (ft)
23.	Lwo	length of water column at start of void closure (ft)
24.	Lao	length of void at start of void closure (ft)
25.	Le	length from closure point to a flow area expansion (ft)
26.	m	mass (lb unless units of slugs are identified)
27.	m_g	gas mass (lb)
28.	m_{stm}	steam mass (lb)
29.	p	pressure, can be absolute (psia) or gage (psig)
30.	P_{steam}	steam pressure driving CIWH event (psia or psig)
31.	P_o	system pressure (psia or psig)
32.	P_d	driving pressure (psia or psig)
33.	P_{pa}	partial pressure of air (psia)
34.	P_{ps}	partial pressure of steam (psia)
35.	P_{sat}	saturation pressure at a specific temperature (psia or psig)
36.	P_{atm}	atmospheric pressure (psia or psig)
37.	P_v	void pressure (psia or psig)
38.	P_{inc}	incident pressure (psia or psig)
39.	P_{sys}	system pressure (psia or psig)
40.	P_{tran}	transmitted pressure (psia or psig)
41.	P_{ref}	reflected pressure (psia or psig)
42.	ΔP	waterhammer pressure pulse (change in system pressure due to WH) (psi)
43.	R	gas constant (53.3 ft·lbf/lbm·°R for air); (also used as curve fit regression coefficient and as multiplier when considering constant impulse)
44.	t	time (sec)
45.	t_d	time duration of pressure pulse (sec)
46.	t_r	rise time for pressure pulse (sec)
47.	T_s	condensing surface temperature (°F)
48.	T_{stm}	steam temperature (°F)
49.	T_{pipe_o}	piping temperature at start of transient (°F)
50.	V	fluid velocity (ft/sec) or void closure velocity (ft/sec)

51. V_i impact velocity (ft/sec)

52. V_o steady state velocity (ft/sec)

53. Vol volume (ft^3)

54. ΔV change in velocity (ft/sec)

55. x axial position (ft)

Greek Symbols

1. α void fraction
2. β fluid bulk modulus (psi); also used as orifice diameter ratio
3. δ pipe wall thickness (in)
4. Δ change in specific term
5. γ ratio of specific heat at constant pressure to specific heat at constant volume
6. τ transmission coefficient of pressure
7. ρ density (lbm/ft^3)
8. ρ_g gas density (lbm/ft^3)
9. ν Poisson's ratio

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1 INTRODUCTION

1.1 Purpose

The United States Nuclear Regulatory Commission (NRC) Generic Letter (GL) 96-06 [1] identified potential issues with cooling water systems following either a Loss of Coolant Accident (LOCA) or a Main Steam Line Break (MSLB) concurrent with a Loss of Offsite Power (LOOP). The potential for the effects of waterhammer during the postulated event to damage the system was included in the Generic Letter. Generic Letter 96-06 (GL96-06) was issued following the issuance of LER 1-96-005 [2] and Westinghouse NSAL-96-003 [3], each of which identified similar potential safety issues. Subsequent to the issuance of the Generic Letter and receipt of utility initial submittals, Request for Additional Information letters (RAIs) were sent to many utilities to clarify technical details about their response.

The components of primary interest are the containment air coolers and associated piping. The containment coolers are generally referred to as the containment fan coolers (CFCs), fan cooler units (FCUs), or reactor building cooling units (RBCUs). These components will be referred to generically as fan cooler units or FCUs in this report. The systems that contain the FCUs are referred to as the component cooling water (CCW) or service water (SW) systems. In this report, the system will be referred to as the service water (SW) system but will apply to either system.

During a postulated LOCA (or MSLB) with a concurrent LOOP, the pumps that supply cooling water to the FCUs and the fans that supply air to the FCUs will temporarily lose power. The cooling water flow will lose pressure and stop. The high temperature steam in the containment atmosphere will pass over the FCU tubing with no forced cooling water flowing through the tubing. Boiling may occur in the FCU tubes causing steam bubbles to form in the FCUs and pass into the attached piping creating steam voids. Prior to pump restart, the presence of steam and subcooled water creates the potential for waterhammer. As the service water pumps restart, the accumulated steam will condense and the pumped water can produce a waterhammer when the void closes. The hydrodynamic loads introduced by such a waterhammer event could challenge the integrity and function of the fan cooler units and associated cooling water system, as well as pose a challenge to containment integrity.

This User's Manual (Volume 1) and accompanying Technical Basis Report (TBR) (Volume 2) provide guidance for the evaluation of potential waterhammer events resulting from postulated concurrent LOOP/LOCA or LOOP/MSLB scenarios. These reports are not intended to replace individual plant analyses, but to provide a methodology that can be used to evaluate the potential effects of waterhammer arising from the conditions described in GL96-06.

1.2 Scope

The scope of the User's Manual and Technical Basis Report are limited to waterhammer events in the containment cooling systems due to combined LOOP/LOCA or LOOP/MSLB events. The information contained herein may be useful for evaluation of column closure and condensation induced waterhammers in other plant systems, but any discussions regarding these other potential applications are not included in the scope of this report.

Generic Letter 96-06 requested an assessment of the following three issues. The issues identified in the Generic Letter are as follows:

> "...the stagnant component cooling water in the containment air coolers may boil and create a substantial steam volume in the component cooling water system. As the component cooling water pumps restart, the pumped liquid may rapidly condense this steam volume and produce a water-hammer. The hydrodynamic loads introduced by such a waterhammer event could be substantial, challenging the integrity and function of the containment air coolers and the associated component cooling water system, as well as posing a challenge to containment integrity."

> "Cooling water systems serving the containment air coolers may experience two-phase flow conditions during postulated LOCA and MSLB scenarios. The heat removal assumptions for design-basis accident scenarios were based on single-phase flow conditions. Corrective actions may be needed to satisfy system design and operability requirements."

> "Thermally induced overpressurization of isolated water-filled piping sections in containment could jeopardize the ability of accident-mitigating systems to perform their safety functions and could lead to a breach of containment integrity via bypass leakage. Corrective actions may be needed to satisfy system operability requirements."

Only the first of these three issues is included in the scope of this report.

1.3 Risk Considerations

Risk to plant safety is an important consideration in the development and recommended application of the waterhammer modeling approach described in this document and justified in the accompanying technical basis report. It is shown that the probability of the combined initiating events (LOOP/LOCA or LOOP/MSLB) is very low {3}. Waterhammers that could result from these combined events will not lead to pipe failure due to internal pressure, since the pressure magnitudes are approximately an order of magnitude below pipe burst pressure. Therefore, the waterhammer loading of concern from the LOOP/LOCA event results primarily from the unbalanced forces produced by traveling pressure waves in the piping system.

An engineering approach to determine waterhammer loading on the piping and support system is presented. This analysis approach takes advantage of system characteristics such as low system pressure and dissolved non-condensables. Bounding conservatism in each step of the analysis is not required to achieve an appropriate engineering solution. The overall solution is shown to provide a very high assurance that the pressure boundary of the piping and components inside containment will meet plant design basis requirements. The approach is consistent with the low probability of the initiating events.

1.4 Organization of the User's Manual and Technical Basis Report

The two volumes consist of a User's Manual (Volume 1) and a Technical Basis Report (Volume 2). The User's Manual presents a methodology for analyzing the events described in GL96-06 to determine the effects of potential waterhammer in the containment fan cooler unit piping. The Technical Basis Report (TBR) presents detailed background information, the analytical modeling and test data, and other method justifications that support the approach provided in the User's Manual. In the User's Manual, cross-references to specific sections of the TBR are identified with braces {xx}. References to the items in the reference section of the User's Manual are identified using brackets [xx]. Items that have been determined to be proprietary are identified in the proprietary version of the report with bold brackets [xx]. These items are deleted in the non-proprietary version of the report.

The events expected to occur during the combined LOOP/LOCA or LOOP/MSLB, as described in the Generic Letter, can result in thermodynamic and hydrodynamic transients. This User's Manual includes many of the analytical methods required to determine the magnitude of the waterhammer and the effect on the individual plant. While all of the specific technical steps that should be followed to determine the individual plant response are identified, the methodology for some steps must be defined and utilized in the individual plant analysis. Similarly, some of these items are included in the example problems, but are best addressed on an individual plant basis. A complete approach and methodology are described in Section 2 of this report.

2
METHOD OF ANALYSIS

This section describes a general methodology for the analysis of waterhammer in the containment fan cooler piping due to the conditions described in GL96-06. It is provided as an engineering approach to evaluating GL96-06 waterhammers and their effects, which also takes advantage of the specific characteristics of the service water system and LOOP or MSLB and LOCA events. Other methods can be used at the discretion of the analyst.

2.1 Characteristics of the Event and the Anticipated Waterhammers

The postulated combination of events (LOCA or MSLB plus LOOP) may produce waterhammers in the FCU piping. The combined LOOP/LOCA or LOOP/MSLB transient event will be referred to generically as "LOOP/LOCA" in this report. The specific fluid conditions that follow the postulated LOCA or MSLB plus LOOP event will differ from plant to plant, but there are common features that control the character of the resulting waterhammers. These features cause the resulting waterhammers to be different from waterhammers described in other documents, such as NUREG/CR-5220 [4]. Several of the specific FCU characteristics are described below.

Low Pressure - The system pressure during the LOOP/LOCA transient is expected to be approximately atmospheric pressure, this can significantly reduce the magnitude of any waterhammers caused by mass and energy transfer at steam and water interfaces.

Presence of Non-Condensables - Steam voids in the pipe are formed by boiling water from the service water or component cooling system. The boiling and subsequent condensation of steam releases non-condensable gases. The presence of non-condensables and the low system pressure reduces the severity of the waterhammer from void collapse.

Low Velocity of Column Closure - CCWH is created by closing of the voided regions in the pipe. The velocity of closure is controlled by the characteristics of the pumps that supply water to the system. These are similar among plants and the velocity is generally limited to approximately of 10-20 feet per second. This velocity will result in a limited waterhammer that is also reduced by non-condensables and steam in the void.

Thermal Layer - A hot thermal layer will form in the water column adjacent to steam in the void that forms. The thermal layer develops from heating that occurs in the fan cooler and condensation of steam on both the pipe wall and the water column interface. The higher temperature thermal layer reduces both the rate of steam condensation at the steam/water interface, and severity of the waterhammer.

It is the combination of low pressure and consequent low energy, the presence of non-condensables and a thermal layer, and a limited closure velocity that reduces waterhammer magnitude and the potential for causing damage in the plant. This report describes a methodology to conservatively predict the specific features of the waterhammer events in the FCU piping.

2.2 Suggested Analysis Approach

This User's Manual is intended to provide specific guidance on the evaluation of waterhammers that can occur as a result of the conditions described in Generic Letter 96-06 and to estimate their effects on the service water system. The following is a "road map" for the tasks needed to evaluate the waterhammer issue.

<u>**Tasks**</u>

1) **Evaluate System**: Develop an Event Time Line defining the events and timing of steps during the transient. While this is a plant specific task, general guidance on this subject is provided in Section 3. The development of an Event Time Line should include the following sub-tasks.

   a) **Gather Plant Data:** Gather information relevant to the evaluation of the system. This will include P&IDs, pipe geometry, LOCA and MSLB temperature plots, FCU data, and pump data.

   b) **Determine Limiting Plant Configuration:** Considering the position and operation of valves, pumps, heat exchangers, and other equipment, the limiting configuration of the plant for both voiding phase waterhammer and refilling phase waterhammers should be determined. Single failure of active equipment should be included in this determination. Guidance on this subject is provided in Section 3.2 of this report.

2) **Model System Hydraulics**: The flow, pressure, and potential paths for water to move and voids to form in the service water system should be determined for the duration of the transient. This will specifically include the time from the loss of power to the time of closure of the void. The system hydraulic model should include the following sub-tasks.

   a) **Determine Fan Cooler Unit Performance:** The heat transfer across the coolers and the contribution this heat makes to the generation of steam voids should be determined. This effort should be performed on an individual plant basis – specific guidance is not provided in this User's Manual.

   b) **Determine System Voiding:** Using the FCU heat input, piping elevation and system resistances, the system pressure and voiding should be determined. This effort should be performed on an individual plant basis – specific guidance is not provided in this User's Manual.

   c) **Determine System Refill**: The flow rates and velocities of the refilling water should be determined from the pump curves and system hydraulic model. Determine anticipated location(s) of closure. This effort should be performed on an individual plant basis – specific guidance is not provided in this User's Manual.

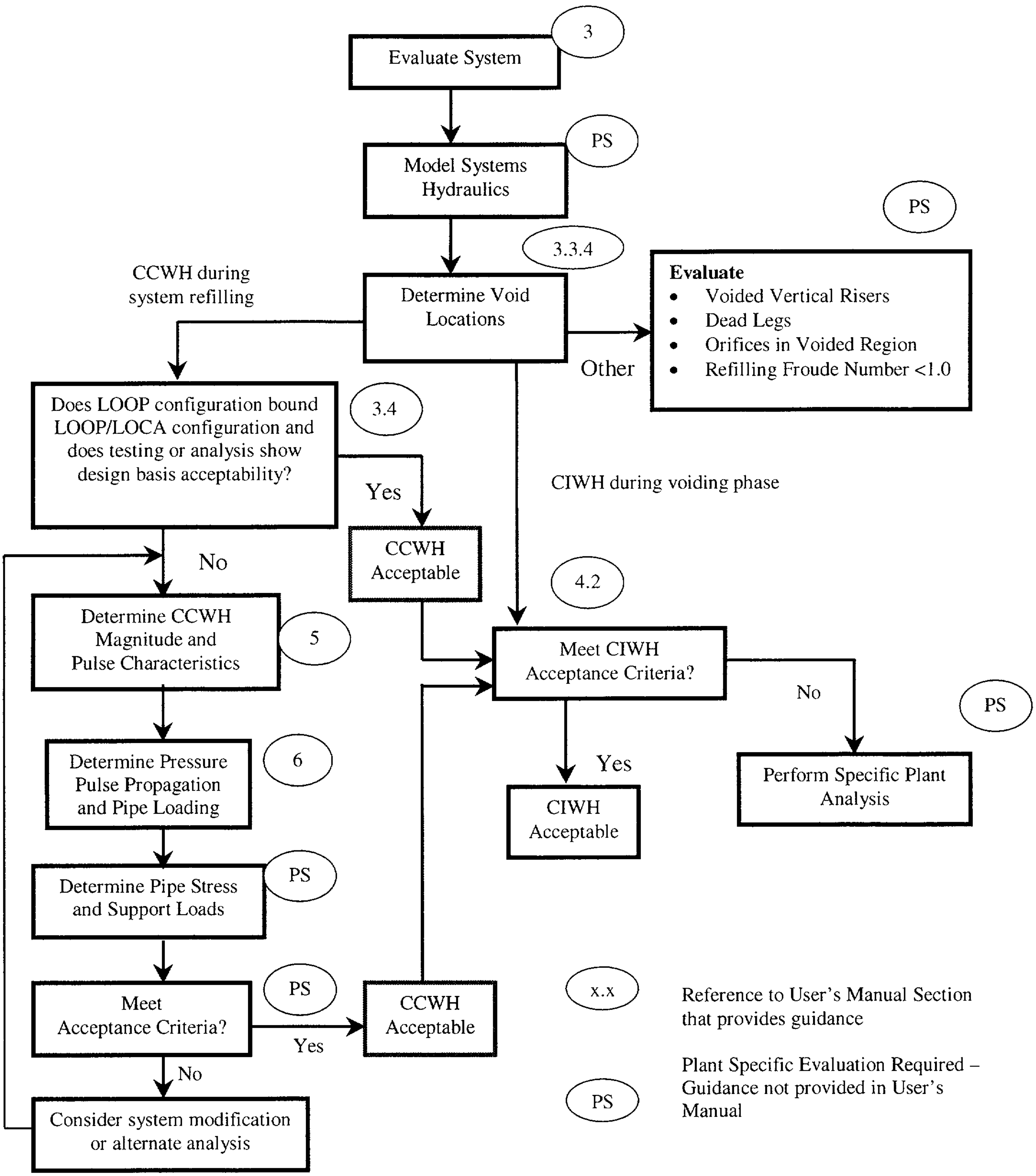


Figure 2-1
Analysis Process Flow Chart

3) **Determine Condensation Induced Waterhammer (CIWH) Magnitude**: If voiding occurs in horizontal piping segments, this assessment may be necessary. Section 4 provides conditions for the system performance parameters that, if met, will eliminate the need to explicitly calculate the CIWH event and the impact on the system.

4) **Determine Potential Closure Locations**: Areas in the voided region that could experience column closure during refilling should be identified. These areas include dead legs, orifices, partially closed valves, and the return side water column.

5) **Determine Column Closure Waterhammer (CCWH) Magnitude and Pulse Characteristics**: If voids form in the system during the transient, re-closure after pump restart should be evaluated to determine the potential CCWH magnitude and time characteristics. This step can be achieved with the following subtasks.

    a) **Determine Column Closure Velocity Limited by Inertia:** A model for determining the velocity of the fluid mass is presented in Section 5.1.

    b) **Determine Released Non-Condensables:** The amount of dissolved gas is important for determining the amount of cushioning, sonic velocity, and column closure waterhammer magnitude. Section 5.2 provides computational details.

    c) **Determine Refilling Velocity Limited by Cushioning:** Non-condensable gas and steam cushioning will help to mitigate the waterhammer generated from column closure. A model for this process is presented in Section 5.2.

    d) **Determine CCWH Magnitude and Pulse Shape:** The CCWH waterhammer magnitude can be determined from the above inputs. The pulse shape, including rise time, duration, and the effects of reflected waves can be estimated. Further discussion is provided in Section 5.3.

6) **Determine Pressure Pulse Propagation and Pipe Loading:** The generated waterhammer pulse will travel from its point of initiation through the piping system. Its propagation properties can be determined from the following subtasks.

    a) **Determine Loading Functions (Force-Time Histories):** The piping along the path of the traveling pulse will be subjected to loads due to differential pressures within each section of pipe. This can be modeled as a piping force-time history, which subsequently can be applied to a piping structural model. Further discussion is provided in Section 6.

    b) **Determine Pulse Amplification and/or Attenuation:** As a waterhammer pressure pulse travels through area changes, the magnitude of the pressure wave will change. Pressure attenuation or amplification may also occur due to fluid structural interaction. Section 6 provides details for predicting amplification and attenuation.

7) **Determine Pipe Stress and Support System Loads**: Dynamic piping response can be obtained from the force-time history. Then the piping system can be qualified to the plant specific design criteria. This effort should be performed on an individual plant basis. Justification for specific methods are developed and correlated to test data in Volume 2, Technical Basis Report.

3
SYSTEM EVALUATION

A typical fan cooler system and the postulated combined LOOP/LOCA event is described in the TBR {2}. This section provides a procedure for determining a conservative system configuration, consistent with the plant's design basis, for the hydraulic analysis of the FCU piping system that will be used in subsequent waterhammer analysis. It is the objective of this section to describe the issues that must be considered in the evaluation of the most limiting operating configuration for the various plant input parameters. For example, column closure waterhammer is maximized by conservatively high refilling velocities, and the valve positions and pump operation that maximize refilling rate should be considered to determine the limiting plant configuration for this portion of the transient. The evaluation should be performed on an individual plant basis.

Information needed for plant analysis includes:

1. A detailed system description and drawings, sufficient for hydraulic calculations and for the determination of limiting system valve and equipment operating positions

2. Detailed information regarding the fan cooler design

3. Pressure and temperature curves for LOCA and MSLB inside containment

4. Operating procedures

5. Event time line (more information and a typical time line are provided below)

3.1 Event Time Line

In order to describe the changes that occur in the service water system during the postulated transient, the event is broken into discrete time periods. The exact times at which the different events occur will be plant specific, but typical times have been selected. A time line is presented below. Plant specific time lines should be developed based on individual plant data.

- Time = 0 sec. Concurrent LOOP and LOCA or LOOP and MSLB events
- Time = 5 sec. Pumps coast down, fans slow gradually
- Time = 10 sec. FCUs boil and void formation occurs
- Time = 30 sec. Pumps restart
- Time = 37 sec. Water columns rejoin, closing the void

3.1.1 Time = 0 to 5 Seconds

To initiate the event, the LOOP occurs concurrently with a LOCA or MSLB. The LOOP causes the pumps to lose power and the pressure in the SW system begins to drop. During the first few seconds, the pumps will coast down. The water will stagnate in the FCUs except for movement due to the voiding of the system. The decrease in water flow will allow the temperature to rise due to the hot steam condensation on the outside of the fan cooler tubes. The water may begin to boil and generate steam in the FCUs since cool water from the pump is not available to replace heated water in the tubes. During this period, the fans will also coast down. Although fan coast down will decrease the heat transfer rate, the fans may take up to 25 seconds or longer to stop, depending upon the specific fan design characteristics. Column separation will occur if the pressure in the high points of the system drops below the saturation point for the service water in the system. Column separation may occur for the LOOP event, even without the addition of heat from a LOCA or MSLB, if the loss of power to the pumps and system geometry cause the high elevation pressures to become sub-atmospheric.

Steam from the LOCA or MSLB environment inside containment will heat the water in the FCUs. While the pressure in the fan cooler tubes is above the saturation pressure corresponding to the water temperature, boiling will not occur. As the pressure drops and the water temperature increases, boiling may be initiated.

3.1.2 Time = 5 to 30 Seconds

Steam may continue to be formed in the FCUs as heat is absorbed from containment. The containment temperature will peak during this time period, as pressure set-points are reached and spray systems start to quench steam in the containment.

Steam generated in the FCUs may move into the piping and the steam voids will grow larger. In an open loop system, the void will grow due to gravity and heat addition. In a closed loop system, gravity typically does not cause void growth since a head tank maintains system pressure. Voids in closed loop systems will expand into the piping only if sufficient heat is added to cause boiling.

One waterhammer mechanism that could occur during this void formation period is Condensation Induced Waterhammer (CIWH). This waterhammer may occur if the steam void in the piping passes into horizontal segments. If the piping contains a stratified layer of hot steam over subcooled water, steam bubbles may become trapped and rapidly condense, causing waterhammer.

3.1.3 Time = 30 to 37 Seconds

Electrical power to the buses feeding the service water pumps is first restored typically between 20 to 33 seconds, depending on the specific plant sequencing. The fans generally start later. The pumps restart rapidly and the system begins to refill. The service water pumps may operate at near run-out conditions initially since the voided system would provide very little resistance. The system resistance increases as the system is filled, particularly once the fan cooler is filled, and the refilling velocity would be expected to decrease. Eventually, all the voids are closed and the system is returned to a water solid condition.

A second waterhammer mechanism, Column Closure Waterhammer (CCWH) can result from the system refilling. Voids created in the piping during the event will close as pumped water fills the system and impacts stationary water in the system. The closing water column can also impact obstructions such as valves, orifices, and other in-line devices that may be present in the voided region, causing additional waterhammer loads. The magnitude of these waterhammers is primarily dependent upon impact velocity.

3.2 Limiting Plant Configuration

Generic Letter 96-06 states that all potential operating conditions brought about by normal operation or failure of a single active component are to be considered in order to ensure that the worst credible conditions have been evaluated. An evaluation of all potential operating and failure scenarios in the plant should therefore be performed to consider the parameters influencing waterhammer occurrence and magnitude.

A discussion of the parameters influencing waterhammer is presented in the following sections. In general, the primary system influences on CIWH will be the void pressure during the voiding phase of the transient. The void pressure will drive the water slug into the trapped steam bubble during the CIWH event. The primary influence on the CCWH will be the refilling velocity after pumps restart. Using these factors as a basis, valves, pumps, and other equipment can be evaluated to determine a conservative plant configuration. For example, since CCWH magnitude is dependent on the refilling velocity, which is directly affected by the performance of the SW pumps, a typical multiple pump system will have a greater refilling velocity with more pumps running. Therefore, systems having multiple pumps should consider the worst condition of the greatest possible number of pumps running.

Effects to be considered should include the items listed below. The system influences on CIWH and CCWH phenomena should be understood before this evaluation is performed. A brief description is provided below. More details are provided in Table 3-1 and Sections 4 and 5.

- Voiding flow rate/regime: During void formation, system alignments that increase system resistance will limit voiding flow rate and potentially increase void pressure. This has the potential to increase CIWH magnitude, which is dependent on system pressure.
- Voiding thermodynamics: During void formation, condensation of steam in the void has the potential to lower void pressure and reduce CIWH magnitude
- FCU thermodynamics: During void formation, the steam generated by the FCUs will increase void pressure if it is greater than the condensation and drain rate. This can increase CIWH magnitude.
- Refilling flow rate/regime: During the refilling phase, CIWH can occur if the pipes do not run full. CCWH can occur when the voids close, and CCWH magnitudes will be dependent upon refilling flow rate.

Table 3-1 presents an example of the evaluation of specific parameters and their effect on the potential waterhammers. These parameters can be used as a guide for the evaluation of specific failure and operating modes described in the following sections. Table 3-2 and Table 3-3 present components that will influence the parameters that affect waterhammer occurrence and magnitude, and these tables should be used in conjunction with Table 3-1 in the review of plant operating parameters to determine the limiting configuration.

Table 3-1
Parametric Effects on Waterhammer

Condensation Induced Waterhammer (CIWH) During Voiding Phase		
Parameter	**System Effect**	**Waterhammer Effect**
Increased system resistance	Lower drain rate. If drainage rate decreases, the local steam pressure will increase in the void.	CIWH dependent on steam pressure, thus CIWH more severe.
Decreased system resistance	Higher drain rate will increase void volume relative to the steam generation, lowering void pressure but potentially exposing more piping to the void.	CIWH dependent on steam pressure, thus CIWH less severe. However, there is the potential that CIWH can occur at more locations.
Increased steam generation rate	Increased steam generation relative to void growth will increase local void pressure.	CIWH dependent on steam pressure, thus CIWH more severe.
Column Closure Waterhammer (CCWH) During Refilling Phase		
Parameter	**System Effect**	**Waterhammer Effect**
Increased pump pressure	Higher refilling rate.	CCWH dependent on closure velocity, thus CCWH more severe.
Increased pump flow	Higher refilling rate.	CCWH dependent on closure velocity, thus CCWH more severe.
Decreased system resistance	Higher refilling rate.	CCWH dependent on closure velocity, thus CCWH more severe.
Increased steam generation rate	Increased local void pressure, gas release.	Closure velocity may be limited by steam and gas cushioning, thus CCWH less severe.

3.2.1 Single Active Failure

To review the potential impact of a postulated single active failure, each of the components that operate during the event need to be considered, along with the time expected for the component to operate. For example, a valve that receives a closure signal upon recovery of power is unlikely to influence the event if the stroke time is greater than 30 seconds. A list of all components activated as the buses re-energize can be used as a basis for this evaluation.

A summary of typical information required for a single active failure evaluation is provided in Table 3-2. The results of Table 3-2 can be compared to the effects of these parameters on waterhammer magnitude, stated in Table 3-1, to determine the overall effect. It should be noted that items such as valve position may have one effect on CIWH and a different effect on CCWH magnitude. From this evaluation, the most conservative system configuration, considering one single failure in the system, can be used as the basis for the evaluation of system hydraulics and potential waterhammer.

Table 3-2
Example Table for Single Active Failure Evaluation

Component	Position before Transient	Time of Signal	Required Action	Time of Operation	Failure Position	Effect on CIWH Parameters	Effect on CCWH Parameters
Control valve	Throttled	LOOP + 25 sec	Open	30 sec	Throttled	None	None
Pump	On	LOOP + 25 sec	Restart	< 5 sec	Off	None	Lower Refilling Velocity
Isolation valve	Open	LOOP	Open	< 5 sec	Closed	Increased System Resistance	Increased System Resistance

3.2.2 Operating Modes Evaluation

The second portion of the evaluation should include any operating or design conditions that may have been implemented during operation prior to the postulated event that would differ from the normal conditions expected at the start of the LOOP/LOCA or LOOP/MSLB transient. For example, a valve to an FCU could be closed to isolate an FCU due to tube leakage. Isolation of an FCU could result in higher flows to the remaining coolers during system refilling.

An assessment of all possible operating or design conditions should be developed along with single active failure. An effective assessment can be performed by tabulating all operating modes for the components evaluated in the single active failure evaluation and assessing each condition. The position of each component and its influence on the CIWH or CCWH may be evaluated. It should be noted that operating modes do not constitute a failure mode, and placing a plant in a specific operating configuration may need to be considered in combination with single failure.

Table 3-3 is an example table that can be used to list all the components and operating and design modes for the SW system. Each should be evaluated to determine the effects on potential waterhammer occurrence and magnitude. This list can become complex when considering all the components and potential operating configurations for the system, but a comprehensive list will ensure that items affecting the waterhammer occurrence or magnitude are not overlooked. In cases where the system configuration presents conflicting CIWH and CCWH parameters, evaluations for each system configuration may be required.

Table 3-3
Example Table for Operating Modes Assessment

Component	Variable	Effect on CIWH Parameters	Effect on CCWH Parameters
Control valve	Throttled	Increased system resistance	Increased system resistance
Pump	Multiple operating	None	Increased pumping pressure
Isolation valve	Closed	None	Increased flow on parallel trains Potential closure against valve
Fan cooler	Tubes plugged	Decreased steam generation rate	Increased system resistance
Fan cooler	Tubes clean	Increased steam generation rate	Decreased system resistance
LOCA profile	Higher	Increased steam generation rate	None

3.3 Plant Evaluation

Based on the system characteristics, an evaluation of the transient thermodynamics and hydraulics of the system should be performed. The evaluation should consider heat transfer from the containment environment to the SW system, including the transient performance of the FCUs as air fans and SW flow coast down. The evaluation should also determine the potential void formation in the SW system based on the heat input from the FCUs and the transient system pressure during the voiding phase. The location, pressure, and size of the voids should be determined. A brief discussion of the voiding phenomena is discussed below.

3.3.1 MSLB/LOCA Comparison

The MSLB and LOCA events should be compared to determine the conservative transient for the LOOP/LOCA or LOOP/MSLB combined event. The heat transfer from either LOCA or MSLB will increase steam void temperature and pressure during the void formation stage. As shown for the system evaluation, higher void pressure generally will increase CIWH magnitude. Initial void pressure will have a minor effect on CCWH. LOCA events generally transmit more heat due to the higher moisture content and increased condensation on the FCUs. MSLB events often have a higher peak temperature. Comparison of these events relative to the influence on void formation is left to the individual plant analysis.

3.3.2 FCU Performance

The Fan Cooler Units should be evaluated in conjunction with the system hydraulic model to determine steam generation during the period when pump pressure is lost. Specific consideration should be given to the elevation of the supply and return piping relative to the FCU. For example, an FCU that is positioned high in containment and has low attached supply and return piping will drain very quickly when the system depressurizes. Any initial steam formed in the cooler will expand to fill the cooler and piping, given the very low steam density at sub-atmospheric pressure. Conversely, a cooler with high attached piping will remain full and require significant heat addition to raise the temperature of the total mass of water before significant steam voids will form. This analysis is highly dependent on plant geometry and should be addressed on an individual plant basis.

3.3.3 Voiding

Voiding occurs in a line when the fluid exceeds the saturation conditions corresponding to the fluid temperature and pressure. Either the pressure of the fluid falls below the boiling point, or the temperature rises, or both. Plants should evaluate the hydraulics of the SW system to determine the growth of the void during the period when pump pressure is lost. This analysis is highly dependent upon plant geometry and should be addressed on an individual plant basis. It should also be noted that multiple regions of the SW system can void simultaneously, and the changing void regions will influence the static pressure on the remainder of the system.

3.3.4 Void Locations

Considering the growth of the steam void and the generation of steam in the FCUs as described above, an assessment of the extent of the voided piping should be made to determine potential locations for column re-closure. The refilling water may impact and create waterhammer loads at various locations. Each of the following should be evaluated for potential waterhammer:

- Draining or stagnant water
- Vertical risers that branch in the voided portion of the pipe.
- Dead legs.
- Orifices or partially closed control valves in the voided portion of the pipe.

Additionally, during the refilling phase, the potential exists for condensation induced waterhammer to occur if the refilling velocity is not sufficient to keep the pipes full. This will not generally occur for pump run-out conditions in most pipe sizes. However, slowly filling headers or other pipes with low velocities should be checked and, if the Froude number for refilling is less than 1, the potential for CIWH during refill should be evaluated.

3.4 LOOP Event

Another event, not specifically addressed in the Generic Letter, but important to the performance of the service water system, is the LOOP event without LOCA or MSLB. In many cases, the elevation changes between the portions of the system exposed to atmospheric pressure (the lake, river, or other ultimate heat sink for open loop plants) and the high points in the system are sufficient to produce voiding during the LOOP event. These voids will cause column closure waterhammer events upon pump restart that can bound the waterhammers produced by the combined LOOP/LOCA or LOOP/MSLB event.

The following two criteria should be considered:

1. Can the system void due to LOOP conditions?

2. Is the pump and system line-up the same as that of the LOOP/LOCA or LOOP/MSLB events?

If the above are true, then the LOOP-only waterhammers will bound those of the combined events {11}. The reason for this is that, without the LOCA or MSLB, the LOOP event will not have any heat addition at the FCUs. With the LOCA or MSLB, boiling will occur in the FCU and the water will release non-condensable gas into the void and will add steam to the void. The steam and gas pressure will resist system refilling, the gas and steam will cushion the final closure, and the subsequent waterhammer magnitude will be lower. The LOOP without LOCA or MSLB will have less cushioning and therefore a larger waterhammer. Plants may be able to take advantage of this fact to demonstrate system acceptability.

4

CONDENSATION INDUCED WATERHAMMER

A horizontal pipe containing stratified layers of cold water and hot steam creates the potential for Condensation Induced Waterhammer (CIWH). This condition can occur in horizontal pipe runs as voids form and the water moves away from the FCU or, in some cases, during refilling after pump restart if the pipe fills in a stratified manner.

Condensation induced events have often been associated with counter-current flow, in which the steam and water are travelling in opposite directions. However, CIWH can occur during the voiding phase of the LOOP/LOCA or LOOP/MSLB event with the water and steam travelling in the same direction due to the vastly different velocities of the steam and water. Key parameters required to cause CIWH are dependent on flow regime, pipe geometry, condensation rate, and bubble collapse. A CIWH description is presented in the TBR {7}. To evaluate CIWH, evaluations of the system hydraulics, heat transfer from the containment environment, steam void formation, and steam void pressure should be completed.

4.1 CIWH Mechanism

During the void formation transient, horizontal portions of the lines become exposed to steam which preferentially enters the top of the pipe. Large amounts of steam can rapidly condense on the pipe and exposed water, this increases the steam velocity over the water. The high velocity steam can form waves that trap steam bubbles that condense and collapse. Essentially, this is a transition from a stratified flow condition to a slug flow condition due to the shear forces on the water from the high velocity steam. Rapid condensation of a trapped steam bubble creates low void pressure that causes the surrounding water to accelerate into the void. Closing the void causes a waterhammer pressure pulse. This scenario is shown in Figure 4-1.

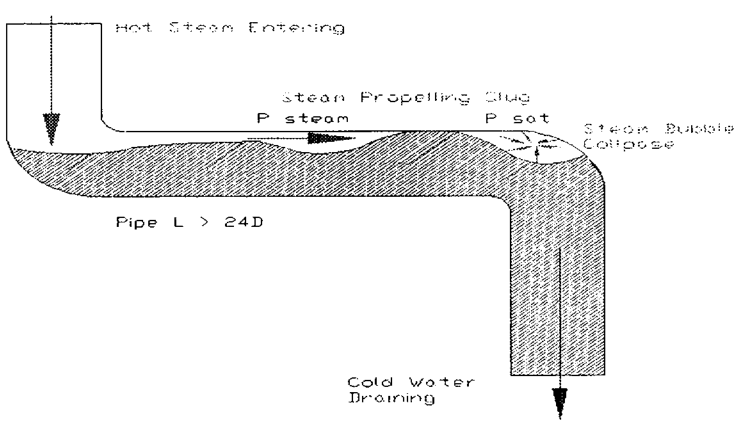


Figure 4-1
Condensation Induced Waterhammer

4.1.1 Flow Regime

During the LOOP/LOCA transient, stratified flow in a horizontal pipe can occur during both voiding and refilling phases. A pipe with stratified, subcooled water and steam creates the potential for transition to slug flow and CIWH. High steam velocities are required for a transition to slug flow. Steam velocity decreases as void fraction increases, and slug flow, and therefore CIWH is not likely at void fractions greater than 0.5 {7.1.1}. CIWH during the voiding phase can occur at very high fluid draining velocities if sufficient steam is produced in the FCU to maintain pressure in the void. CIWH is not mitigated by high fluid draining velocity.

For refilling a pipe, CIWH could occur if a horizontal pipe is not running full, i.e. it fills in a "stratified manner" {7.1.1}. A pipe will not stratify if it is filled at a rate greater than a "keep full" velocity determined using a Froude number of 1.0. At such velocities, the potential of CIWH is negligible. The Froude number is

$$Fr = \frac{V}{\sqrt{g \cdot D}}$$

Equation 4-1

4.1.2 Subcooling

Vigorous condensation in the void is required for CIWH to occur. The rapid decrease of pressure in the steam void (void collapse) is a direct result of the condensation rate. For this rapid condensation to occur, the draining fluid must have sufficient subcooling relative to the steam and there must be sufficient surface exposed on which the steam can condense.

The subcooling of the fluid is the temperature below the saturation point for a given pressure. According to Griffith [5], if the draining fluid is not subcooled more than 36°F subcooling, the condensation rate is insufficient to produce significant condensation induced waterhammers.

4.1.3 Pipe Geometry

Condensation induced waterhammers occur in long horizontal or near horizontal piping in which a stratified flow regime exists at the onset of transition to slug flow. There are two geometric constraints provided in the TBR {7.1.3} that determine susceptibility to CIWH:

- The pipe should be declined (sloped down relative to drain direction) no more than 2.4° to the horizontal for waterhammer to occur.
- The length to diameter ratio (*L/D*) of a horizontal pipe must be at least 24 for CIWH to occur.

4.2 Recommended Evaluation Approach

Based on the testing and evaluation developed in the TBR, the effects of the CIWH due to LOOP/LOCA can be shown to be less than the effects of CCWH occurring in the same piping system. Therefore, proving piping acceptability for the CCWH events will also demonstrate acceptability for CIWH. The criteria for using this evaluation are presented below. If these criteria are not met, CIWH will require explicit evaluation.

Bounding Waterhammer

Condensation induced waterhammers that may occur in low pressure service water systems can be limited in magnitude and/or duration such that they are not a threat to pressure boundary integrity {7}. This conclusion is applicable only in systems that meet the following conditions:

- the system steam pressure at the time of the postulated CIWH is less than 20 psig,
- the piping has been shown by test or analysis to be capable of withstanding a CCWH following LOOP, LOOP/LOCA, or LOOP/MSLB.

If these conditions are met, explicit calculation of CIWH magnitude is not required.

5 COLUMN CLOSURE WATERHAMMER

Column closure waterhammer (CCWH) is a well documented phenomenon in nuclear power plants [6, 7]. This waterhammer event can occur when a voided pipe is refilled and the refilling water column impacts stationary objects such as valves, orifices, and standing water columns. The dynamics of the column closure event and resulting waterhammer are described in this section.

The inputs to the evaluation of CCWH magnitude are similar to the ones for CIWH, but differ because the water velocity is driven by the pumps and system hydraulics and not by rapid condensation of a trapped steam bubble. To evaluate CCWH, analyses of the system hydraulics, providing the dynamics of voiding the system and flow rates upon pump restart, should be completed.

5.1 Re-closure Velocity Limited by Inertia

The closure velocity following pump restart is driven by the pump characteristics. The velocity is limited by the acceleration of the mass of fluid from rest and the system frictional losses. This is particularly important for small voids in which the volume of water accelerated is much greater than the volume of steam void. The problem is essentially: "to what velocity can the fluid accelerate in a limited length of pipe?" To answer this question a differential equation for the motion of the fluid column was developed in Section 6.8 of Reference [6]. It is conservatively assumed that the pump head is instantly available, and the pump pressure forces are resisted by the pipe wall shear stresses and the acceleration of the liquid mass. The gravity effects on the fluid are neglected.

The differential equation of motion for a liquid mass refilling a voided pipe was solved for a range of initial axial water lengths to total pipe length ratios X_o/L in a pipe [6, 7], producing a set of graphs relating the impact velocity to the pipe losses, as shown in figure 5-1. The impact velocity V_i is expressed as a multiple times the steady state velocity V_o. V_o is the steady state velocity of a liquid column of length L, i.e. the steady state velocity at impact. It can be seen that the velocity tends towards steady state as the pipe fills; however, the initial lengths can fill faster than steady state velocity for some cases. Note that the curves representing the solution set for this problem are conservative in that they do not include nozzle losses at the inlet. These additional losses would limit the incoming flow velocity to approximately the steady state velocity. Also note that the source document for Figure 5-1 (Reference [6]) utilizes H and F for head and loss coefficient, designated h and f, respectively, in this report.

For typical systems with long voids, the combination of X_o/L and $fL/2D$ terms in Figure 5-1 will be in a region of the curve where V_i/V_o will be approximately 1.0. This means that the impact velocity will be approximately equal to the steady state velocity as determined by the pump characteristics and system losses. These equations conservatively presume that the void does not provide any resisting pressure that would tend to reduce the impact velocity. The accumulation of non-condensables or steam that does not condense rapidly enough in the void can slow down the impact velocity at closure. The potential for cushioning the final impact will be pursued further in this report.

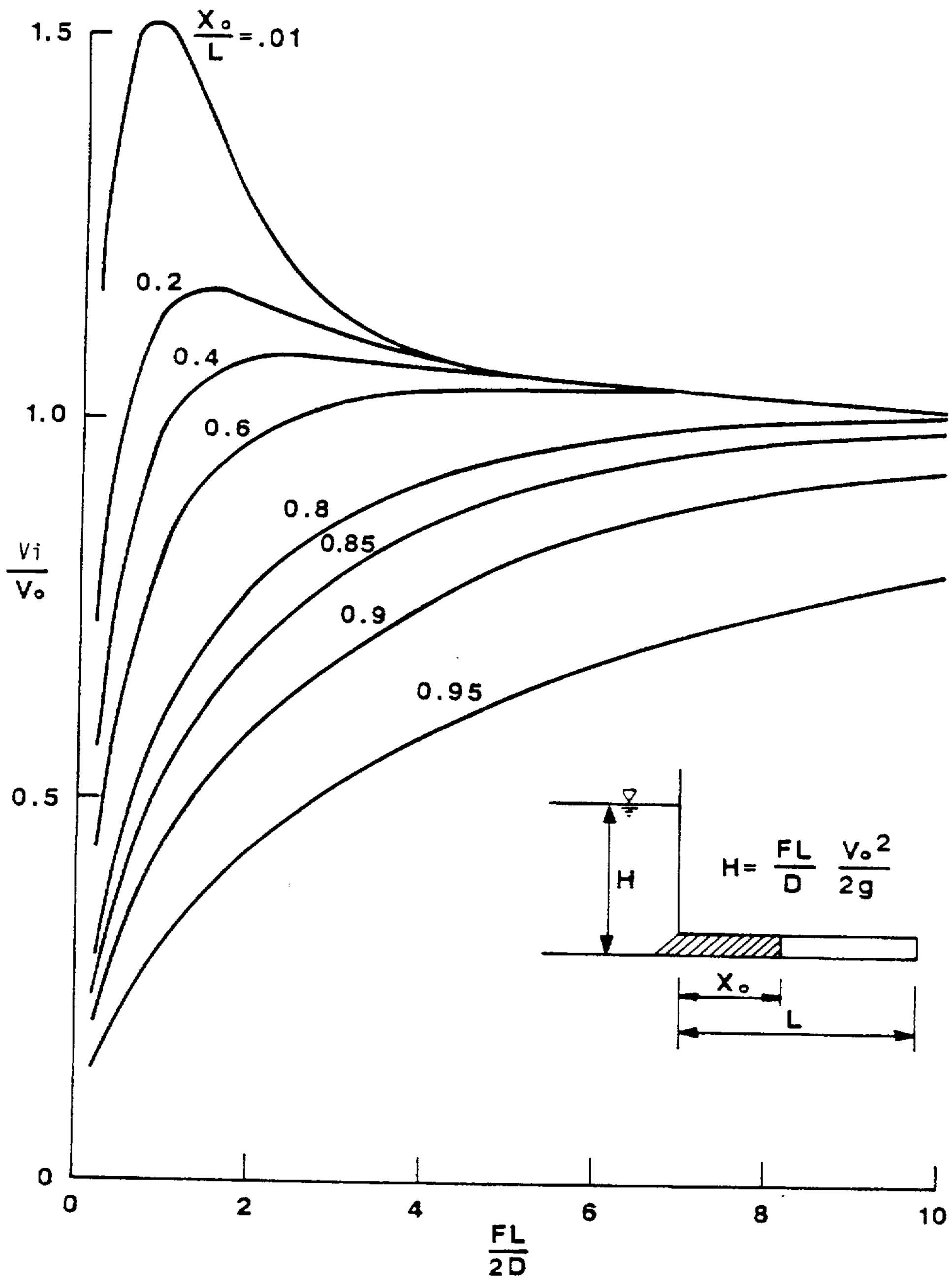


Figure 5-1
Velocity of a Refilling System

It can be seen from the figure that voids extending into less than approximately 30% of the piping ($X_o/L > 0.7$) will produce a condition in which the impact velocity is limited by the acceleration of the fluid mass. This velocity reduction can be used to limit the waterhammer impact velocity.

5.2 Re-closure Velocity Limited by Cushioning

The advancing water column velocity can also be limited by deceleration as the void compresses, pressurizes, and resists the oncoming water column. Void pressurization can reduce waterhammer magnitude since pulse magnitude rises slower, which can allow time for reflections from free surfaces to relieve pressure buildup. Void pressurization will both slow the oncoming water column and accelerate the downstream column. This effect is referred to as "cushioning." The relative velocity is reduced and the peak waterhammer pressure buildup is slower. Additionally, the "slower" increase in the pressure as the void compresses provides significant additional rise time to the pulse shape. In some cases, sufficient air mass can result in the refilling water columns not coming in contact and not creating a true waterhammer. In these cases, a pressure pulse caused only by the gas compression will occur, and this pulse will travel through the piping system.

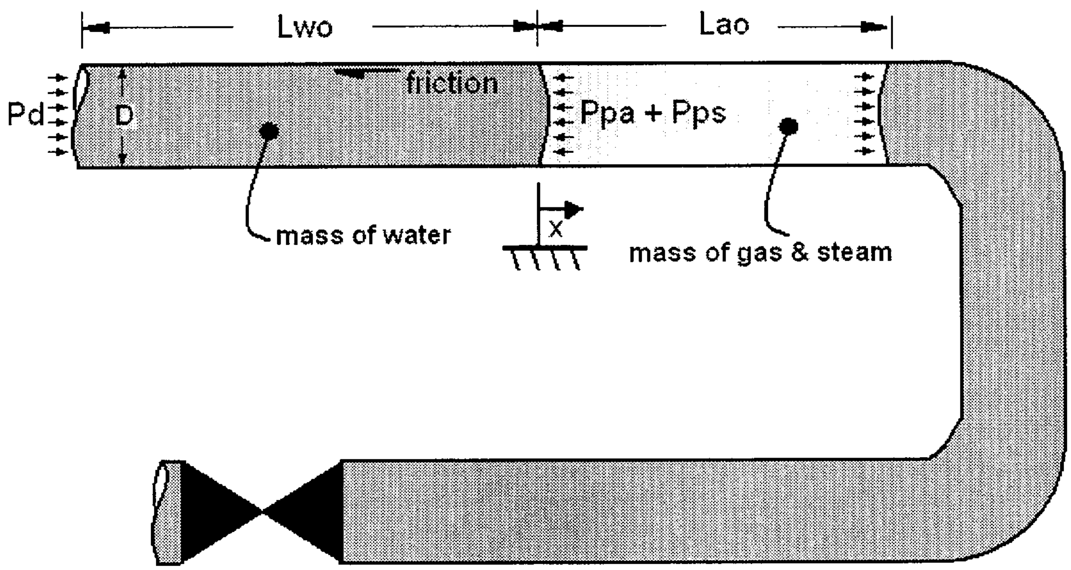


Figure 5-2
Gas Cushioning of Waterhammer

Figure 5-2 depicts an example of cushioning. When a water column closes a void (e.g. due to pump restart), non-condensable gas and steam will act as a cushion. It will slow the velocity of the advancing column, increase the velocity of the stagnant or slowly moving column, and decrease the column closure waterhammer magnitude. The final impact velocity is lower than the impact velocity that would be expected without non-condensables or steam in the void.

Pressurization of the void can be caused by two primary effects. These are:

1. compression of released non-condensables that have accumulated in the void, and
2. slow condensation of steam that has accumulated in the void.

Non-condensable gases will pressurize with decreasing volume as the water advances and the void volume is reduced. Steam will compress like a gas except for the steam mass removed by condensation. The temperature of the fluid and pipe surrounding the void will influence condensation rates. Void pressure will increase if the pressure rise due to the volume reduction is greater than the pressure decrease that is caused by condensation of the steam.

Condensation of the steam carrying released non-condensables will increase the concentration of gas in the void region. Condensation will occur as the steam void leaves the FCU and transfers heat to the adjacent piping.

The released non-condensable gas can also affect the waterhammer magnitude by reducing the sonic velocity. Gas that becomes entrained in the closing water column can cause a sonic velocity reduction that will reduce the waterhammer pressure.

A model for predicting the velocity reduction due to cushioning is described in the following section. A model for predicting the sonic velocity will also be provided in a later section. In order to use this model, an evaluation of the released non-condensable gas in the void is required. This is also described in the following sections.

5.2.1 Rigid Body Model Description

To determine the closure velocity limited by cushioning, a conservative analytical model predicting the inertia of the decelerating water column and pressurization of the gas/steam void has been developed. This model is referred to as a "Rigid Body Model" (RBM) since it treats the fluid as an incompressible mass. In this model, the steam and non-condensable gas in the void are allowed to pressurize as the volume decreases, and steam is removed from the volume due to condensation. Wave propagation effects are not included in the model and are addressed separately. This model was benchmarked against a method of characteristics (MOC) analysis and against test data. A set of solutions to the RBM is included in graphical form in Appendix A. These solutions can be used to characterize the impact velocity for use in determining CCWH magnitude.

5.2.2 Dissolved Gas Concentration

To apply the RBM, a prediction of non-condensable gas concentration in the void is required. At the initiation of the transient, non-condensable gases will be concentrated in the water at the saturation point for the temperature of the fluid.

Figure 5-3 shows the saturation point for air, nitrogen, and oxygen dissolved in water at atmospheric pressure [8]. Some of the gas will evolve due to the drop in pump pressure and boiling process during the LOOP/LOCA transient. A portion of this gas will concentrate in the void. To calculate the amount of dissolved gas in the fluid, a conservatively high temperature and a conservatively low pressure should be used, since the amount of dissolved gas decreases with increased temperature and decreased pressure. The amount of dissolved gas in the fluid should be determined and then be used to determine the amount of gas released during the transient.

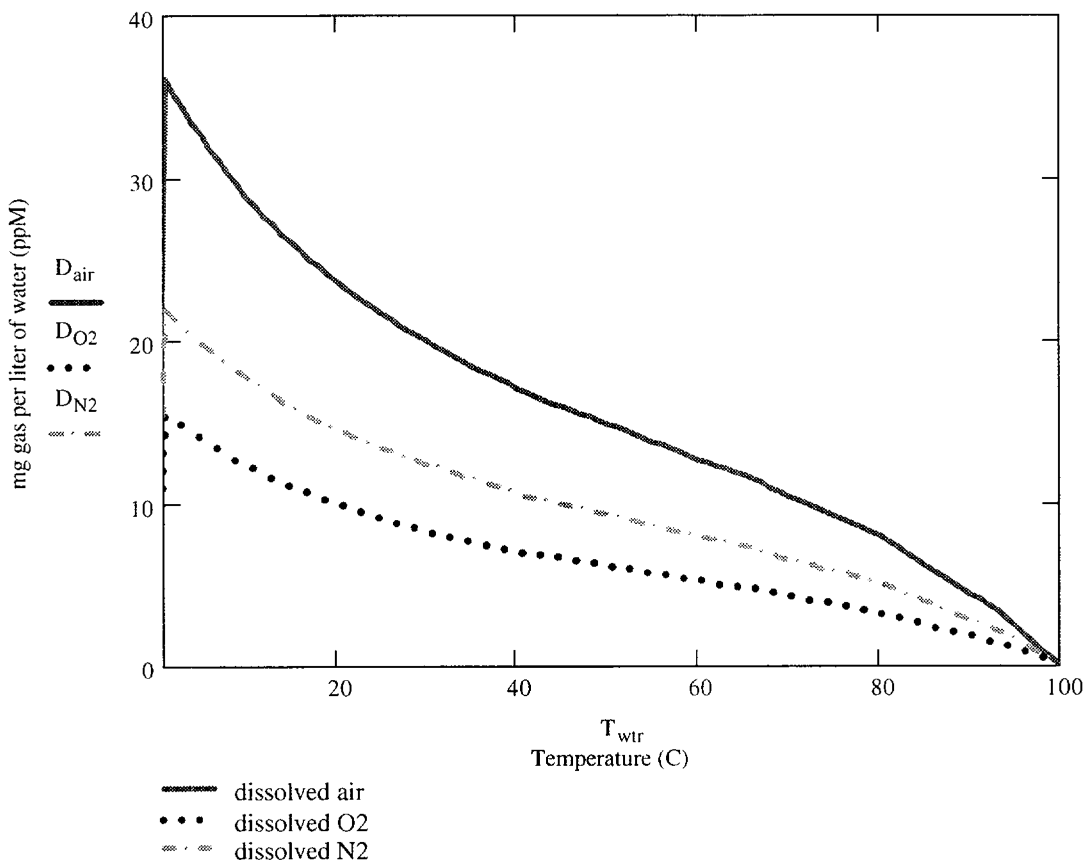


Figure 5-3
Solubility of Air, Nitrogen, and Oxygen in Water at Atmospheric Pressure

5.2.3 Gas Release Modeling

Gas evolved during the transient will reside in two places at the end of the transient. Some of the evolved gas will be bubbles that will stay in the re-closing water columns. The bubble rise times are slow for small bubbles [9]. This entrained gas will affect the sonic velocity in the region near the collapse. Other gas will be liberated by boiling and will travel with the steam into the void.

The model considers both systems that maintain a head on the FCUs during the voiding phase of the transient, and FCUs that can drain. Typically, FCUs with supply and return piping attached at higher elevations relative to the FCU elevation will remain full except for the mass driven out as steam. In these FCUs, it has been determined by testing that a significant portion of the dissolved non-condensable gas is driven out of the participating water mass.

Boiling must occur for credit to be taken for non-condensable gas release. Credit for gas release should be calculated if the water in the FCU is exposed to temperatures above the boiling point corresponding to the pressure. All the water does not boil, but it has to be exposed to boiling that occurs in the region for the gas to be removed. In order to assure that the water is exposed to boiling, credit for gas release should not be taken unless the temperature of the tubes in the FCU reaches a temperature that is 10°F above the saturation point. This will assure that the water in the tubes is exposed to boiling and that "gas stripping due to boiling" will occur.

If boiling occurs and sufficient margin exists in the temperature reached, tests have shown that 50% of the dissolved gas is evolved from the total mass of water in the cooler and 24% of the dissolved gas is evolved from the total mass of water that is "trapped" in the headers – the local piping through which steam passes during the transient {6}. If boiling does not occur with the 10°F margin, no credit shall be taken for gas release for the region of the FCU in which boiling does not occur.

This model is also applicable to systems that do not maintain head on the FCUs during the transient. Typically, FCUs with supply and return piping attached low relative to the FCU will "drain from the bottom" and will not remain water filled during the transient. However, the water in the tubes was also shown by testing to release a significant percentage of its non-condensable gas. Tests showed that 50% of the dissolved non-condensable gas was release from the draining water in the tubes during the time of the transient {6}. Again, credit for gas release should not be taken unless the water in the FCU is exposed to temperatures that are at least 10°F above the saturation point. The steps to determine the amount of gas in the void are presented with a simplified figure below.

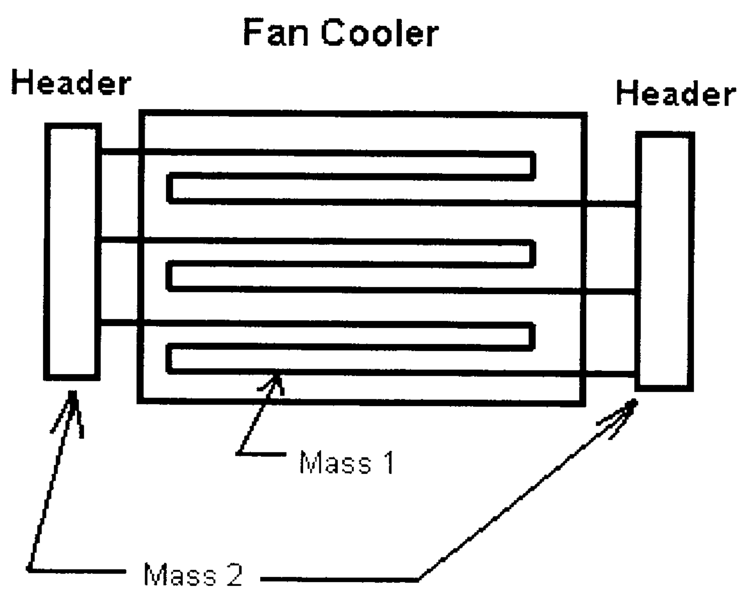


Figure 5-4
Simplified FCU Showing Water Masses Releasing Gas

Gas Evolution Analysis Steps

1. Determine initial gas concentration for the water in the heat exchanger tubes and header. This should be based on plant specific design basis maximum temperature and dissolved gas data as shown in Figure 5-3 adjusted for pressure.

2. Determine mass of water in the heat exchanger tubes (Mass 1 in Figure 5-4).

3. Determine the amount of this water that is exposed to temperatures that is at least 10°F above the saturation point.

4. Gas released into the void is 50% of the gas in this water mass.

If the heat exchanger has headers that drain, then the calculation is finished.

If the heat exchanger has headers that remain full:

5. Determine mass of water in the heat exchanger headers and attached piping through which steam passes (Mass 2 in Figure 5-4).

6. Gas evolved from Mass 2 is 24% of the gas in this mass of water.

7. Total gas evolved is the sum of the gas released from Mass 1 and Mass 2.

5.2.4 Sonic Velocity

The sonic velocity is the speed at which a pressure wave travels through the fluid medium. This term is important for determining both the magnitude of the pressure pulse and the time that the wave travels through the piping system. This transit time is important to the dynamic loading of the structure. Sonic velocity is related to the fluid compressibility β and density ρ as follows:

$$C_f = \sqrt{\frac{\beta}{\rho / g}} \qquad \textbf{Equation 5-1}$$

Using the following values produces a sonic velocity of 4,870 ft/sec.

β = 319,000 lbf/in^2 (approximate)

ρ = 62.4 lb/ft^3

Moody [10] presents an equation for the estimation of the effects of thin walled pipe deformation on the speed of sound. Pipe flexibility "softens" the fluid it contains. The following relationship revises the sonic velocity to consider pipe flexibility.

$$C_{pipe} = C_f \frac{1}{\sqrt{1+\dfrac{D \cdot \beta}{\delta \cdot E}}}$$ **Equation 5-2**

Modifying the sonic velocity calculated above for a small, stiff pipe such as a 2" schedule 80 pipe produces a sonic velocity of 4580 ft/sec. A larger, thin walled pipe of the same material reduces the sonic velocity to 3510 ft/sec. These values are shown in Table 5-1.

Table 5-1
Sonic Velocity In Flexible Pipe

	2" sch 80 pipe	12" std pipe
Pipe Outside Diameter	D = 2.375"	D = 12.75
Wall Thickness	δ = 0.218"	δ = 0.375"
Elastic Modulus	$E = 28 \cdot 10^6$ psi	$E = 28 \cdot 10^6$ psi
Sonic Velocity	4,580 ft/sec	3,510 ft/sec

5.3 Pressure Pulse Modeling

This section describes how the RBM solution can be used to determine final closure velocity of the refilling water column in a power plant service water system. The steps described in the evaluation include the following:

- The velocity that is reached by the fluid closing the void should be determined from plant flow models and/or test data. This analysis should be performed on an individual plant basis and should consider the inertia and friction limits described in Section 5.1.
- The amount of non-condensable gas in the void should be determined (Section 5.2.3) and the sonic velocity should be determined (Section 5.2.4).
- A simplified characterization of the piping system is presented for use in the RBM. This characterization should be used with the amount of non-condensable gas to determine a reduced closure velocity that includes the effect of cushioning.
- This reduced closure velocity should be input to the Joukowski equation given below in Equation 5-3 to determine waterhammer pressure pulse magnitude.
- The pressure pulse rise time and duration should be calculated to complete the characterization of the pressure pulse. Methods are provided in Section 5.3.4 and Section 5.3.5

Waterhammer magnitude can be calculated using the Joukowski equation noted below [11]. The pressure pulse magnitude, ΔP, is a function of the water density ρ, the sonic velocity C, the change in velocity of the water slug ΔV, and the impact characteristic k. For water impacting a fixed end, $k = 1.0$; for water impacting water, $k = 0.5$. For most CCWH in the FCU systems, $k = 0.5$.

$$\Delta P = k \cdot \rho \cdot C \cdot \Delta V \qquad \textbf{Equation 5-3}$$

5.3.1 Plant Characterization

The rigid body model was solved for different cooling water system conditions. Details of the solution are provided in the TBR {9} and output plots are provided in Appendix A. In order to use the data in the plots, the actual plant conditions must be correlated to a set of the prototypical conditions used in the analysis. This correlation to an actual plant configuration is demonstrated in Figure 5-5 and described below.

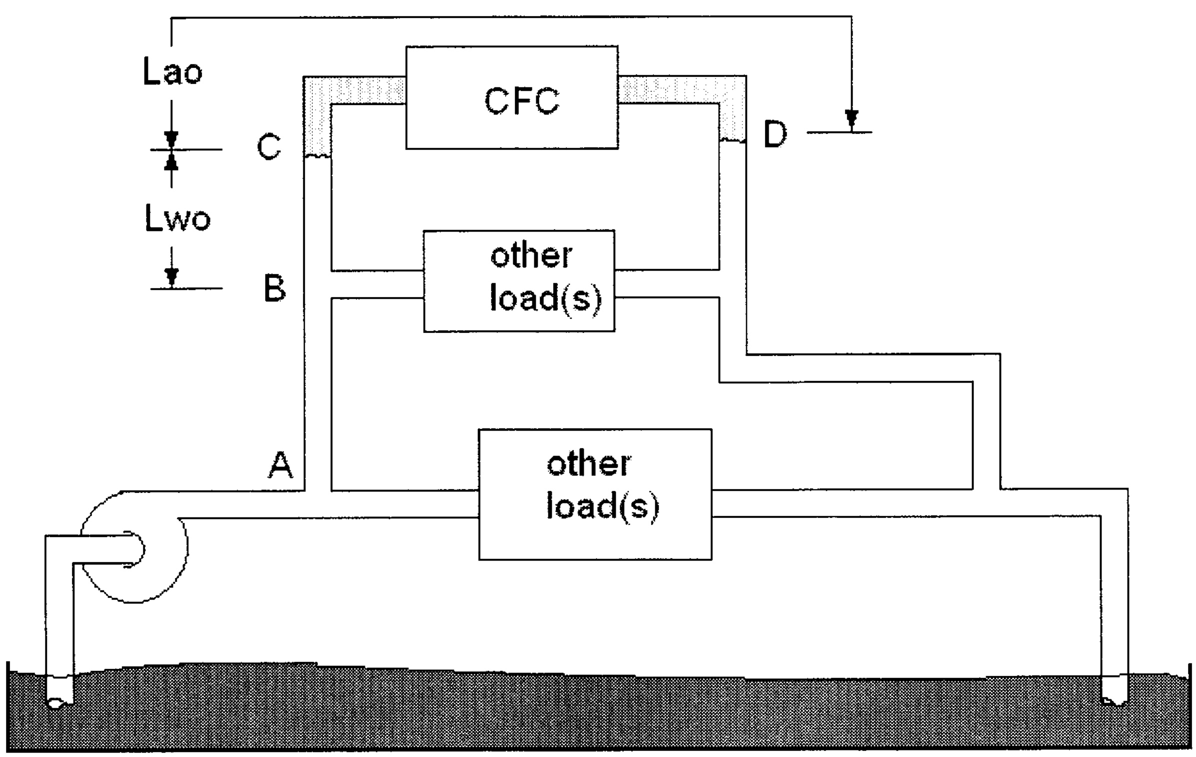


Figure 5-5
Plant Modeling

To correlate the actual plant configuration to the rigid body model, the following information is needed:

- The initial water slug mass is a portion of the mass of water from the void to the pump(s) that will be accelerating during the void closure. It is not always necessary to consider the mass all the way to the pump, particularly if the flow to the void is a relatively small percentage of the total flow. The first major flow branch may be an acceptable point (i.e. point "B" in Figure 5-5) if the pressure at this point does not change dramatically during void closure. It is recommended that the area changes be used to estimate the boundary of the participating mass. The transmission coefficients (see Section 5.3.6) should be calculated upstream of the closure location until 80% reflection ($\tau = 0.20$) is achieved. All mass from this location to the void will form the initial water slug.
- The volume of the void is correlated to an equivalent length of piping at the void closure diameter size (*Lao*).
- A total flow coefficient K is needed for the system from point "B" to the void closure point "D". This flow coefficient needs to be an equivalent coefficient corresponding to flow in a pipe with a diameter equal to the diameter in which the final closure occurs.

An initial approximation of the void closure velocity should be made. The void closure velocity should be determined considering the acceleration limit described in Section 5.1. The closure velocity should consider the driving pressure at point "B", the equivalent flow coefficient K, and the elevation difference between point "B" and the void closure location. If reverse flow from point "D" to point "C" is expected, then the relative velocity should be calculated and applied as the first order velocity approximation.

5.3.2 Pressure Pulse Magnitude

The terms "Initial Velocity" and "Cushioned Velocity" are introduced here.

- Initial Velocity: the velocity of the columns at the location of final closure approximated by the inertia and friction limits described in Section 5.1 ($V_{initial}$).
- Cushioned Velocity: the impact velocity that is calculated after considering the cushioning effects of steam condensation and/or gas compression in the void ($V_{cushioned}$).

The plots shown in Figure 5-6 and Figure 5-7 demonstrate two sets of solutions of the RBM, providing the cushioned velocity $V_{cushioned}$ as a ratio of the initial velocity $V_{initial}$. The initial impact velocity is calculated using steady state friction or inertia limited methods as described previously. The first solution set in Figure 5-6 is for 4" piping with negligible steam cushioning effects (i.e. only gas cushioning is considered). The second solution set in Figure 5-7 is for 4" piping with steam and gas cushioning.

The flow coefficient, length, velocity, gas mass, and condensation limitations are shown on the plots. Multiple simulations were performed with different variables. Similar plots for the 10" and 16" pipe sizes are included in Appendix A. Details of the analysis are included in the TBR {9}.

Figure 5-6
4" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo* = 100 ft

Figure 5-7
4" Pipe, Gas and Steam Cushioning, Initial Velocity 10 fps, *Lwo* = 100 ft

Plant conditions that fall within the following limits are bounded by the RBM runs performed. The basis for these limitations is discussed in detail in the TBR {9.4.3}.

Table 5-2
Rigid Body Model Analysis Limits

Variable	Requirement	Basis
Water Column Size	$Lwo_{RBM} \geq Lwo_{plant}$	The RBM runs bound shorter slug lengths.
Void Length	$Lao_{RBM} \geq Lao_{plant}$	The RBM runs bound shorter void lengths.
Initial Velocity	$V <$ ~20 fps	The RBM was verified up to 30 ft/sec. Curves are provided for velocity 10, 15, and 20 ft/sec.
Gas Content	$Mgas > 60$ mg $(ID/2)^2$ (ID in inches)	A minimum amount of gas is required to apply the fixed hA determined by the test data. This amount is related to the square of the pipe diameter {8.3.3}.
Void & Interface Temperature	$Tvoid = Tsurf \geq 200°F$ Void Temp < 200°F	Consider gas and steam cushioning. Consider gas cushioning only.

The following steps are recommended when crediting gas and/or steam cushioning:

1. Determine the initial velocity using the friction or inertia limited method described previously.

2. Verify that the criteria of Table 5-2 are satisfied.

3. Determine the cushioned velocity and corresponding pressure pulse using the "Gas Cushion Only" plots contained in Appendix A. (Select the appropriate plot based on pipe size and initial velocity).

4. If it is desired to further reduce the cushioned velocity by considering steam cushioning, determine the cushioned velocity using the "Steam and Gas Cushion" plots contained in Appendix A.

The waterhammer magnitude may also be affected by reflected pressure waves and attenuation due to rarefaction waves may need to be considered. This is discussed in the following sections (Section 5.3.7). This may be considered with or without crediting gas or steam cushioning.

5.3.3 Pressure Pulse Shape

Pressure magnitude only defines part of the dynamic character of a waterhammer or pressure pulse event. The rise time and duration of the pulse are equally important. The essential elements of a pressure pulse can be represented with a simplified, trapezoidal shape. This shape captures the pulse magnitude, rise time, and duration.

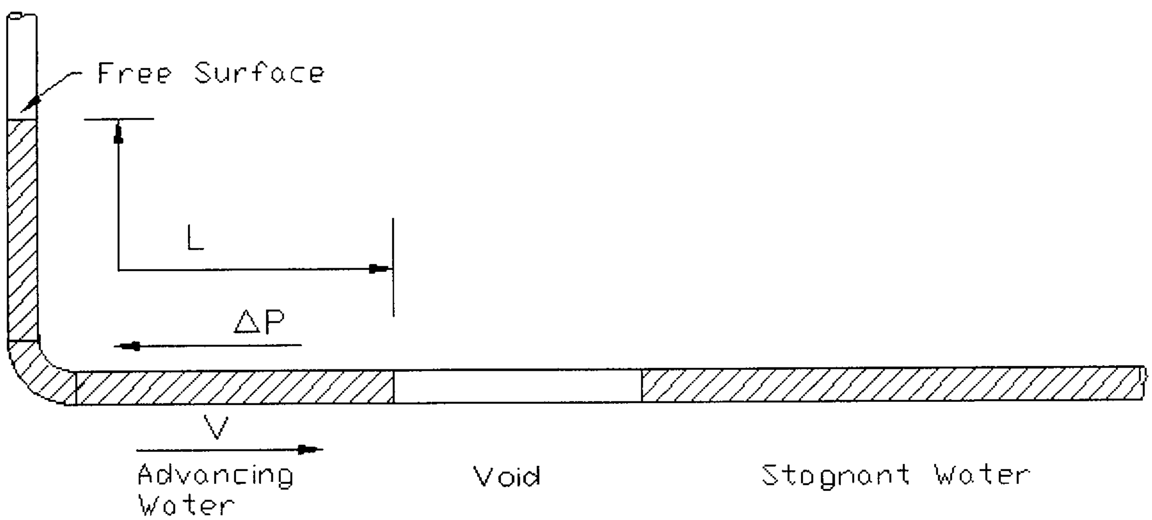


Figure 5-8
Example Column Closure Waterhammer Event

A diagram of a refilling system is presented in Figure 5-8. Theoretically, this waterhammer event produces a square wave with a duration t_d based on the water-solid length L through which the wave travels to a reflecting surface and back. For a wave traveling at the sonic velocity C, the duration can be calculated as follows and is shown in Figure 5-9 below:

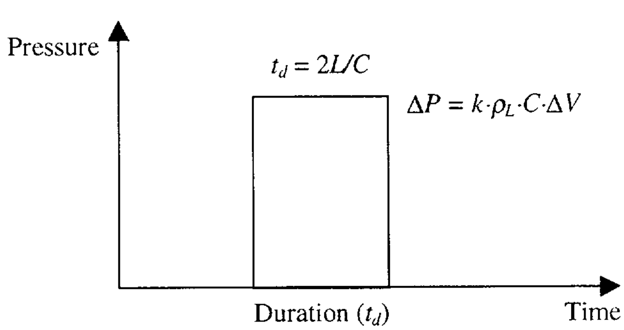


Figure 5-9
Idealized (Square) Pressure Wave

The length of the water column L used to calculate duration is usually taken as the length back to the nearest free surface as shown in Figure 5-8. The free surface is not able to maintain an elevated pressure, and a rarefaction wave is reflected back through the fluid, reducing the pressure to its value before the transient.

In reality, the analytical square wave model is modified due to several phenomena that are displayed graphically in Figure 5-10. The leading edge of the waveform does not have an instantaneous rise, but has a finite "rise time" over which the pressure magnitude increases from the steady state value to the elevated transient pressure. A gradual pressure increase can be attributed to physical phenomena such as an irregular shape of the water face as it closes the void. More importantly, it is also due to the increase in void pressure as the void compresses. Non-condensables or slow steam condensation rates can contribute to the increase of pressure in the void before column closure. The rise time is particularly important to the structural loading of the piping, since loads are dependent on the slope of the rise.

The leading and trailing edges of the wave may also be influenced by partial reflections from changes in direction and small voids in the water. The reflection effects may even reduce the maximum pressure attained by the pulse, if rarefaction waves arrive back at the pulse initiation location before the pressure rise is completed. This is referred to as "peak clipping" in this report and described in more detail in Section 5.3.7.

The combination of these effects can produce a wave that is more trapezoidal than the theoretical square wave. In some cases, a relatively long rise time and short duration will create a triangular pulse.

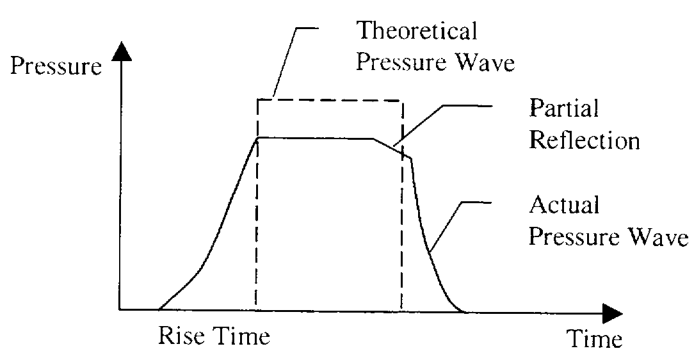


Figure 5-10
Actual Pressure Wave Comparison

An appropriate analytical representation for the pressure pulse can be based on a trapezoidal shape. This simplified form captures the pressure pulse magnitude, duration, and rise time in a more realistic manner than a simple square wave.

5.3.4 Rise Time Prediction

The rate of pressure increase during the waterhammer event can have a significant effect on the peak pressure pulse magnitude and on the unbalanced loads in piping segments. The peak pressure will be reduced by reflections of rarefaction waves from area expansions if the rise time is long relative to the reflection time. The unbalanced loads will be reduced if the distance occupied by the rising pressure wave is long relative to the length of a piping segment.

Pressure pulse rise time is affected by the driving pressure, relative lengths of the water column and void, rate of condensation, and amount of gas in the void. The rigid body model was used to predict the pressure pulse rise times and was compared to test data. A conservative bounding function was devised to fit the predicted and measured rise time based on the impact velocity {9.3.2}. The impact velocity was chosen as the independent variable because it integrates the net effects of the driving pressure, lengths, condensation, and gas.

A bounding function was developed based on the rise time being inversely proportional to the polytropic compression of the void. The constant for the function was developed from comparison to RBM results for a range of cases and to actual test data {10.8.1.3}. The RBM results apply to a range of pipe diameters and the provided function bounds the results from 2" to 18" pipe. For closure velocity V in ft/sec and rise time t_r in seconds:

[ ] **Equation 5-4**

5.3.5 Duration

A pressure wave will travel from the origination point, through the piping system, and reflect as a partial rarefaction wave as it passes area changes and free surfaces. The net effect of many partial reflections from area changes will be to eventually return the pressure pulse to the system pressure. The time it takes for the pressure pulse to return to the system pressure is the pulse duration.

Transmission of the pressure wave can be expressed in terms of a transmission coefficient (*Tau* or τ), which is the ratio of the transmitted pulse to the incident pulse. Formulae for the calculation of transmission coefficients based on piping geometry are provided in Section 5.3.6.

The approach described herein uses a value of 90% of the pulse reflected back to the origination point to define the return of the pressure to its undisturbed state. Stated in terms of the transmission coefficient τ, 90% reflection would produce a transmission coefficient of 10%. The distance the pulse travels to reduce the transmitted pulse by 90% can be converted to pulse duration knowing the velocity of the traveling pulse, which is the sonic velocity. The value of 10% transmission has been selected based on the test program modeling. The lower 10% of the pressure pulse was found to not significantly affect the pulse duration {12}.

To find the pulse duration, τ is calculated for each successive area change moving away from the initiation point until τ is less than 0.1. The distance L_{dur} is defined as the length of piping from the closure location to the point that the pulse is reduced by 90%. The pulse duration t_d is then calculated from:

$$t_d = 2 \cdot \frac{L_{dur}}{C} + t_r$$

Equation 5-5

The entire pulse is shown in Figure 5-11 below. The total duration is calculated as the time is takes the pulse to travel to the $\tau = 10\%$ location and back ($2L_{dur}/C$) plus the rise time (half the rise time on the front end and half on the back end).

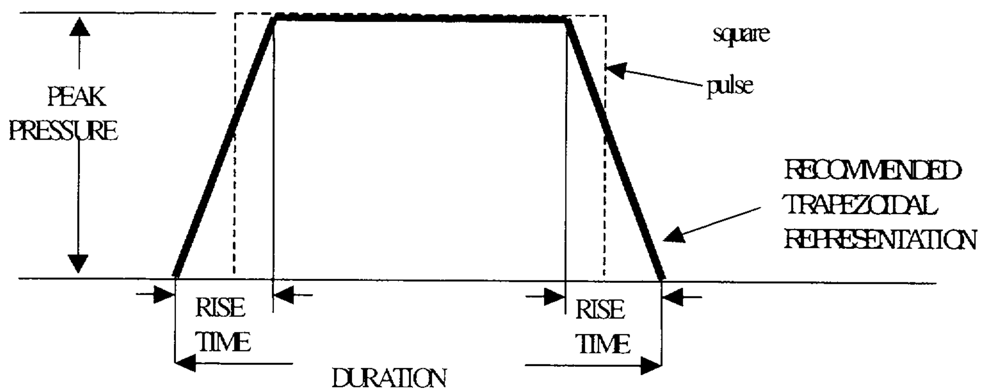


Figure 5-11
Duration Definition and Trapezoidal Representation

5.3.6 Transmission Coefficients

Flow area effects on pressure waves are normally characterized by transmission coefficients. A transmission coefficient (*Tau* or τ) is the ratio of the transmitted pulse to the incident pulse, as provided by Equation 5-6. Transmission coefficient definitions and the means of calculating the transmission coefficients are defined for branches/reducers and for throttle devices in this section.

$$\tau = \frac{P_{tran} - P_{sys}}{P_{inc} - P_{sys}}$$

Equation 5-6

5.3.6.1 Branch/Reducer Transmission Coefficients

A pressure pulse is amplified as it travels from a large flow area into a smaller flow area. A pressure pulse is attenuated as it travels from a small flow area into a large flow area. The pressure that approaches the area change is referred to as the incident pressure P_{inc}. The steady state system pressure is referred to as P_{sys}. The pressure that travels downstream of the flow area change is referred to as the transmitted pressure P_{tran}. Pressure also reflects at an area change. Pressure that returns in the direction opposite to the transmitted pressure is referred to as the reflected pressure P_{ref}. The reflected pressure is referred to as a rarefaction wave if it is lower pressure.

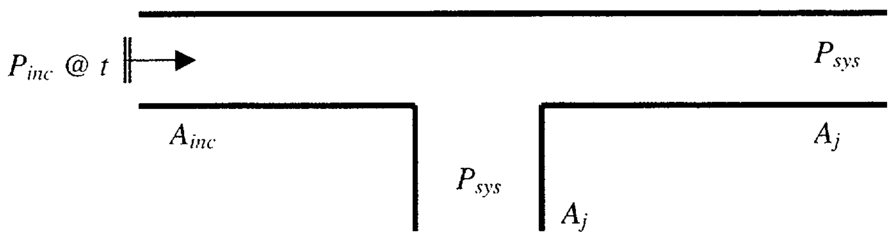


Figure 5-12
Pressure Pulse Approaching Area Change at Time = *t*

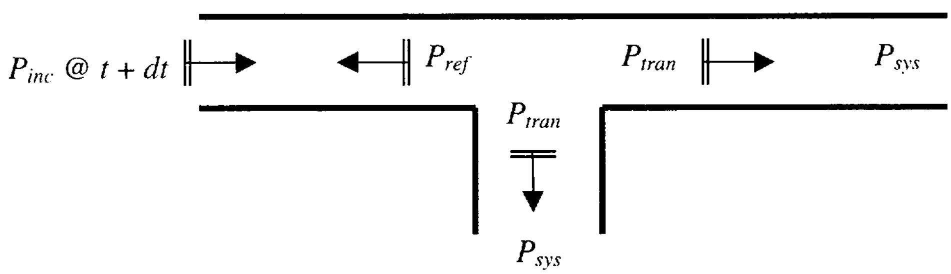


Figure 5-13
Transmitted, Reflected, and Incident Pressure at Time = *t + dt*

A diagram of an incident wave is shown in Figure 5-12, showing a pressure increase from P_{sys} to P_{inc}. A diagram of the resulting transmitted and reflected waves is shown in Figure 5-13.

The transmitted and reflected pressures are related to the incident pressure by the flow areas and sonic velocities in each flow path. The relationship between the incident pressure and the transmitted pressure is defined by the transmission coefficient τ. The coefficient τ is defined as [11]:

$$\tau = \frac{P_{tran} - P_{sys}}{P_{inc} - P_{sys}} = \frac{2 \cdot \dfrac{A_{inc}}{C_{inc}}}{\dfrac{A_{inc}}{C_{inc}} + \sum_j \dfrac{A_j}{C_j}}$$

Equation 5-7

In most cases, the change in sonic velocity at the area transition is negligible and the transmission coefficient is a function of only the flow areas.

$$\tau = \frac{P_{tran} - P_{sys}}{P_{inc} - P_{sys}} = \frac{2 \cdot A_{inc}}{A_{inc} + \sum_j A_j} \qquad \text{Equation 5-8}$$

The transmitted and reflected pressure pulses are calculated from:

$$P_{tran} = \tau \cdot (P_{inc} - P_{sys}) + P_{sys} \qquad \text{Equation 5-9}$$

$$P_{ref} = (\tau - 1.0) \cdot (P_{inc} - P_{sys}) + P_{sys} + P_{inc} \qquad \text{Equation 5-10}$$

If the flow area change is from large to small, then τ is greater than 1.0 and the transmitted pressure is amplified. The reflected pressure will also be greater than the incident pressure.

If the flow area change is from small to large, then τ is less than 1.0 and the transmitted pressure is attenuated. The reflected pressure will be less than the incident pressure if the flow area is from small to large.

5.3.6.2 Throttle Devices Transmission Coefficients

An additional flow area change of interest is an orifice or throttle device. Wylie and Streeter [11] developed equations for the transmitted and reflected pressures as a result of a pressure pulse incident upon an orifice. The effects of control valves or other throttling valves are analogous. Here, the steady state pressures upstream and downstream of the throttle device are referred to as P_{up} and P_{dwn} respectively. The configuration with the incident pressure approaching the orifice is represented in Figure 5-14 below. The transmitted and reflected pressure conditions are shown in Figure 5-15 below.

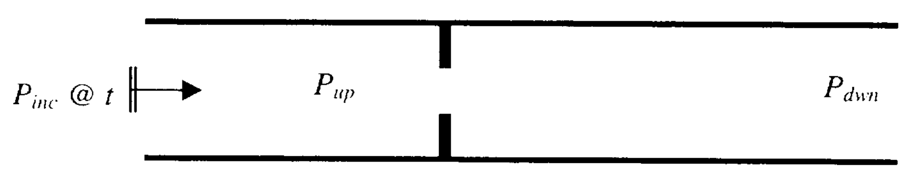


Figure 5-14
Pressure Pulse Approaching Throttle Device at Time = *t*

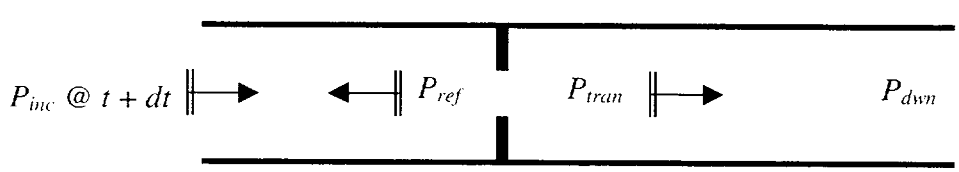


Figure 5-15
Transmitted and Reflected Pressures at Time = *t* + *dt*

The transmission coefficient for a throttle device is defined as [11]:

$$\tau_o = \frac{P_{tran} - P_{dwn}}{P_{inc} - P_{up}} = \frac{-\left(1+\frac{P_{up}-P_{dwn}}{\rho \cdot C \cdot V}\right)+\sqrt{\left(1+\frac{P_{up}-P_{dwn}}{\rho \cdot C \cdot V}\right)^2+\frac{K}{C^2}\cdot\left(\frac{P_{inc}-P_{up}}{\rho}\right)}}{\frac{K}{2 \cdot C^2}\cdot\left(\frac{P_{inc}-P_{up}}{\rho}\right)}$$ **Equation 5-11**

The transmission and reflection coefficients for a throttle device are defined as the ratio of the transmitted or reflected pulse to the incident pulse. The transmitted and reflected pulses are then [11]:

$$P_{tran} = \tau_o \cdot (P_{inc} - P_{up}) + P_{dwn}$$ **Equation 5-12**

$$P_{ref} = (2.0 - \tau_o) \cdot (P_{inc} - P_{up}) + P_{up}$$ **Equation 5-13**

In general, throttle devices must have a significant area reduction before any reflection or transmission of interest occurs. This is demonstrated by plotting the reflection and transmission coefficients against the β ratio (orifice diameter to pipe diameter). This is shown in Figure 5-16. In development of this figure, the following conditions were imposed:

- The orifice flow coefficient C_o was assumed to be constant and equal to 0.6.
- The flow rate was assumed to be zero.
- The velocity head coefficient K_o for the orifice was calculated from:

$$K_o = \left(\frac{1}{C_o \cdot \beta^2} - 1\right)^2$$ **Equation 5-14**

It is apparent from Figure 5-16 that the ratio of orifice to pipe diameter must be less than 40% to have appreciable reflection or transmission effects on the pressure pulse. As a result, in many service water systems, the effects of orifices and control valves are negligible when evaluating the pressure pulse propagation effects.

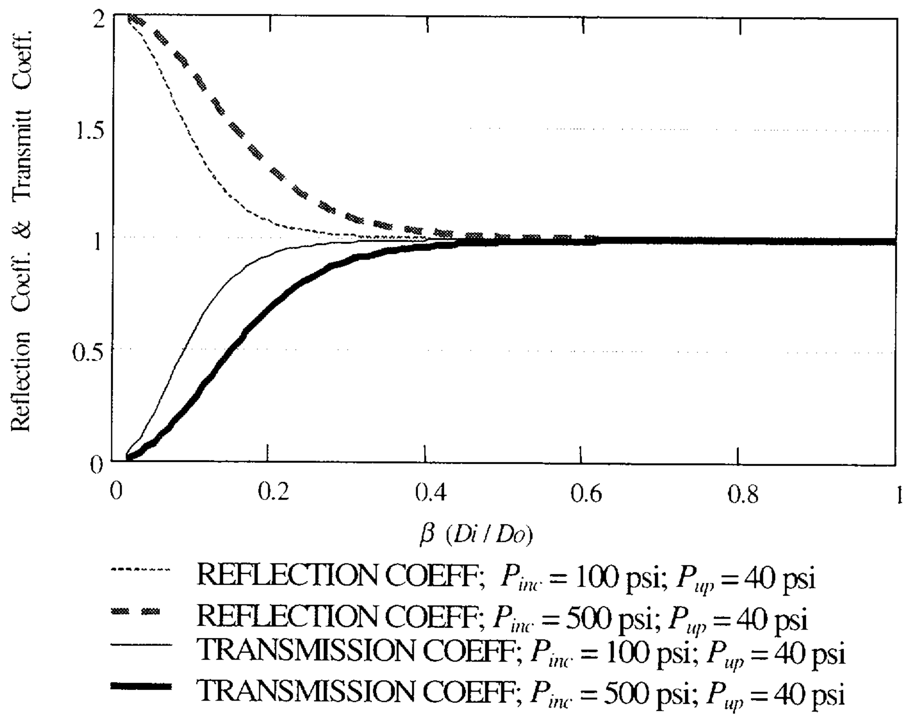


Figure 5-16
Orifice Reflection and Transmission Coefficients

Waterhammers that occur due to direct impact on an orifice or partially closed valve can be determined by the following equation:

$$\frac{\Delta P}{\rho \cdot C \cdot V_o / g} = 1 + \frac{1 - \sqrt{1 + 2\left[\left(\frac{A}{A_{or}}\right)^2 - 1\right]\frac{V_o}{C}}}{\left[\left(\frac{A}{A_{or}}\right)^2 - 1\right]\frac{V_o}{C}}$$ **Equation 5-15**

where: A_{or} = area of the orifice or restriction

V_o = impact velocity

5.3.7 Peak Pulse "Clipping"

The peak pressure pulse may be affected by reflections. A method of evaluating the effects of reflections on the pressure pulse magnitude is described in this section.

During a column closure event, the pressure rises as the void closes. This rising pressure travels upstream and downstream from the closure location. As the pulse encounters area changes, a reflected wave travels back toward the closure location. The reflected wave will add to the pressure it encounters in a positive or negative manner. If the reflection comes from an expansion, then it will have a negative magnitude and cause the oncoming pressure to be reduced. The peak pressure will be "clipped" if the reflection reaches the closure location before the pressure peaks. Similarly, if the pressure wave encounters a reduction in area while it is rising in magnitude, then a positive wave is reflected and the peak pressure may be increased.

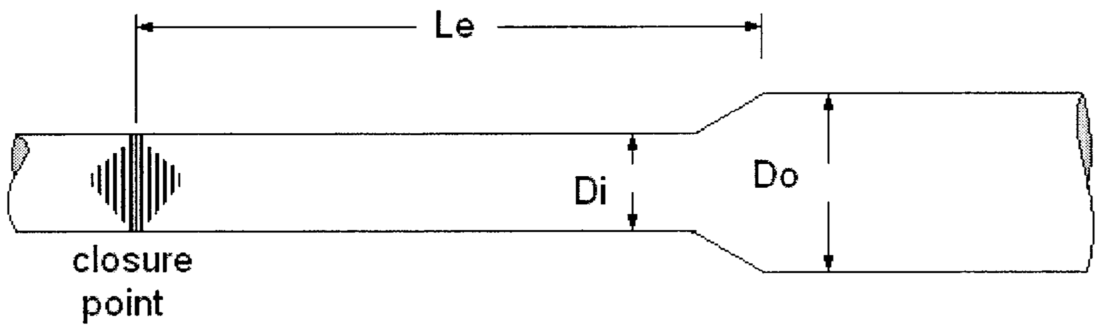


Figure 5-17
Pressure Peak Clipping Due to Reflection

When the reflected pressure reaches the point of pulse initiation then the new pressure that propagates from the closure point will be (written as a function of time):

$$P_{new}(t) = P_{inc}(t) + P_{ref}(t - 2L_{ref} / C)$$ **Equation 5-16**

where:

$P_{new}(t)$	=	the new pressure at the closure point at time t
$P_{inc}(t)$	=	the pressure at the closure point without considering reflections at time t
$P_{ref}(t - 2L_{ref}/C)$	=	the reflected pressure from the area change at time $t - 2L_{ref}/C$
L_{ref}	=	the distance from the initiation point to the reflection of concern

In terms of only the incident pressure and the transmission coefficient, the new pressure at the initiation point is:

$$P_{new}(t) = (\tau - 1) \cdot P_{inc}(t - 2L_{ref} / C) + P_{inc}(t)$$ **Equation 5-17**

This equation may be applied to determine the peak pressure pulse as a result of reflections. It is recommended that this equation not be applied to shorten or lengthen the pulse duration as a result of reflections. It is recommended that the duration associated with the initiating trapezoidal pulse be maintained unless more detailed pulse propagation analyses are required.

For trapezoidal shaped pulses, the equation for P_{new} was solved for a series of transmission coefficients and a 10-ft/sec closure velocity. The ratio of the peak interaction pressure P_{new} to the peak incident pressure P_{inc} was plotted as a function of the length between the closure point and the reflection point. The plot is shown in Figure 5-18.

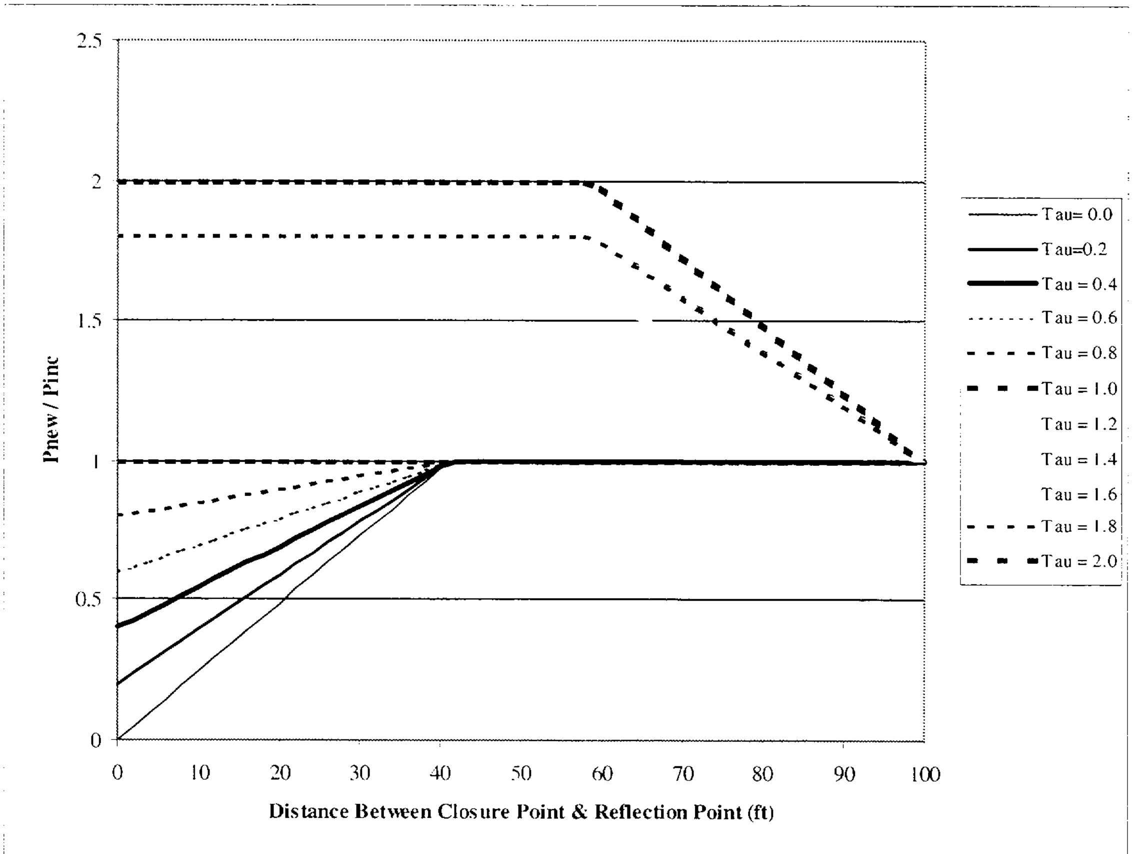


Figure 5-18
Peak Clipping

(Note: this figure applies to 10" pipe, V = 10 fps, 100 ft = (duration × $C/2$), trapezoidal shape; equations in this section may be used to determine pressure interaction ratio for other trapezoidal shaped pulses.)

As expected, P_{new} is less than P_{inc} for cases where:

- The pulse is incident upon a larger area ($\tau < 1$).
- The distance between the reflection point and the initiation point is less than the distance the pulse can travel in half the rise time.

Also as expected, P_{new} is greater than P_{inc} for cases where:

- The pulse is incident upon a smaller area ($\tau > 1$)
- The distance between the reflection point and the initiation point is less than the distance the pulse can travel over half its duration.

The relationships provided in Table 5-3 may be used to determine the increase or decrease in the pressure pulse as a result of reflections from area changes for any trapezoidal pulse. In this table, t_r is the pulse rise time and t_d is the pulse duration.

Table 5-3
Reflection Effects

Transmission Coefficient & Reflection Length	Pressure Interaction Ratio
$\tau < 1$ and $0 < Le < t_r \cdot C/2$ reflections will reduced pressure and occur before the full rise is achieved	$\frac{P_{new}}{P_{inc}} = \frac{1-\tau}{t_r \cdot C/2} \cdot Le + \tau$ Reduce pressure
$\tau < 1$ and $t_r \cdot C/2 < Le$ reflections will reduced pressure but occur after the full rise is achieved	$\frac{P_{new}}{P_{inc}} = 1$ Pressure unchanged
$\tau > 1$ and $(t_d - t_r) \cdot C/2 < Le < t_d \cdot C/2$ reflections will increase pressure and occur during the rise time	$\frac{P_{new}}{P_{inc}} = \frac{1-\tau}{t_r} \cdot (2Le/C - t_d) + 1$ Increase pressure
$\tau > 1$ and $Le < (t_d - t_r) \cdot C/2$ reflections will increase pressure and occur after the rise and during the duration	$\frac{P_{new}}{P_{inc}} = \tau$ Increase pressure
$\tau > 1$ and $(t_d) \cdot C/2 < Le$ reflections will increase pressure but occur after the pulse duration	$\frac{P_{new}}{P_{inc}} = 1$ Pressure unchanged

To calculate the effect of reflections on the peak pressure pulse magnitude the following steps are recommended:

- Calculate the peak potential pressure pulse that could occur considering cushioning from gas and/or steam.
- Calculate the rise time for the peak potential pressure pulse.
- Calculate the length from the reflection point of concern to the closure point.
- Calculate the transmission coefficient as described above.
- Calculate the ratio of the new pressure to the incident pressure using the appropriate equation listed above.

6
PULSE PROPAGATION AND SYSTEM LOADING

The pressure pulse generated by the column closure event will progress through the piping system. It will travel away from the initiation point while the loss of energy diminishes the pulse magnitude. In order to determine the propagation of the pressure pulse through the system and the loading on the structure, the evaluation of the CCWH pulse magnitude and pulse characteristics should be completed.

As a pressure pulse propagates through a water system, the wave is principally influenced by three phenomena – flow area, piping movement, and friction. These issues are discussed in this section and summarized briefly below:

- Flow area attenuation: The pulse will be affected by area changes in the pipe through which it travels. These effects should be determined to evaluate pipe loading.
- Fluid structural interaction (FSI): The movement of the structure will remove energy from the pressure pulse, but potentially combine to amplify the pulse. The potential amplification is small relative to the reduction, and can therefore be conservatively ignored.
- Friction: The effect of friction on a traveling pressure pulse is small and can be ignored.

6.1 Flow Area Attenuation/Amplification

The pressure pulse magnitude is attenuated or amplified as the travelling pulse encounters flow area changes. As the pressure pulse interacts at area changes, a transmitted pulse continues past the area change and a reflected pulse returns in the opposite direction. In Section 5, the influence of the reflected pressure wave on the pulse magnitude was discussed. In this Section, the influence on the transmitted pressure wave is discussed. Considering both of these phenomena, a pipe area change will influence the pressure wave in three ways:

1. Reflections from flow area changes help define the pressure pulse duration as discussed previously (see "Duration").

2. Reflections from flow area changes help define the magnitude of the initiating pressure pulse as discussed previously (see "Peak Clipping").

3. Transmitted pressure pulses can be attenuated or amplified by pipe cross sectional area change.

To determine the change to the transmitted pressure pulse as it travels through the system, a table or list of each of the pipe area changes being encountered in a specific system should be developed and utilized. The transmission coefficient of the transmitted pulse should be calculated, and the resulting transmitted pressure pulse magnitude should be determined. This amplified or reduced pulse will then continue into the next piping segments. These effects are cumulative, and the pressure pulse will eventually be dispersed due to the effects of area change. In a manner similar to the duration calculation, a reasonable lower bound for transmitted pressure pulse is 10% of the original pressure pulse magnitude, i.e. $\tau = 0.1$ or 90% of the pressure pulse has been reflected.

Since the transmission coefficients present a repeated yet simple calculation, a table or spreadsheet approach will simplify this calculation. An example is presented below for a pulse initiating at 100 psi above the system pressure.

Table 6-1
Example Transmission Coefficient Calculation

Seg. Number	Size Sch. (in)	Component	Incident Area, A_{inc} (in^2)	Transmitted Area, A_{tran} (in^2)	Transmission Coefficient			Pressure Pulse (psi)
					Formula	τ	Cum τ	
1	4" Sch 40	Pipe	12.73	12.73	$2A_{inc}/(A_{inc}+A_{tran})$	1.0	1.0	100
2	4" Sch 40, 3" Sch 40	Reducer	12.73	7.393	$2A_{inc}/(A_{inc}+A_{tran})$	1.265	1.265	126.5
3	3" Sch 40	Tee	7.393	7.393 x 2	$2A_{inc}/(A_{inc}+A_{tran})$	0.667	0.844	84.4
4	3" Sch 40, 6" Sch 40	Expansion	7.393	28.89	$2A_{inc}/(A_{inc}+A_{tran})$	0.408	0.344	34.4

6.2 Fluid Structural Interaction

The pressure pulse magnitude is attenuated or amplified by piping movement. Piping movement results when a pressure causes an unbalanced load on a piping segment. Piping movement can compress the fluid and generate a new pressure pulse. As the initiating pulse and new pulse pass each other, they are additive and can cause a localized increase in pressure amplitude. This is an example of fluid structure interaction that is referred to as strain related coupling.

Piping movement is a result of a transfer of energy. The pressure pulse loses some of its energy during this transfer. As a result, the pressure pulse magnitude can be attenuated. This is another type of fluid structure interaction. Both the amplification and attenuation of the pressure pulse will be described. In the cases considered in this report, the amplification is generally less significant and the attenuation can be relatively important.

Each of these – strain related coupling and pulse attenuation – will be described below.

6.2.1 Strain Related Coupling

Strain related coupling is also referred to as Poisson coupling. Strain related or Poisson coupling results from the transformation of circumferential strain caused by internal pressure to axial strain. The internal pressure associated with the incident waterhammer pressure pulse causes a circumferential deformation (or strain) in the pipe wall, and a tension wave is formed in the pipe wall. The pipe itself is not treated as a rigid body but deforms in a wave manner ahead of the waterhammer pressure pulse. The tension wave that travels in the pipe wall has two effects.

1. First, a contraction of the pipe wall is induced ahead of the incident pressure. This contraction causes a pressure in the fluid, but this effect is not significant and can be neglected when evaluating SW systems for GL96-06.

2. Second, the tension wave causes axial movement of the piping. The piping movement can induce a secondary pressure pulse in the fluid. The tension wave travels through the pipe wall at the speed of sound in the piping (approximately four times the sonic velocity in the fluid [12]). The pipe will move toward the incident pressure when the tension wave arrives at a change of direction. This pipe movement has a piston effect on the fluid and can generate a pressure in the fluid.

Several factors should be considered relative to this potential load increase. First, conservation of energy indicates that we are not creating new pressure pulses, but are redistributing pressure loads. The energy taken to move the pipe is removed from the incident pressure wave, reducing its impact on the system. Second, the conservatism inherent in the differential pressure loading model should envelop the predicted small load increases and low probability of this event. Last, column closure waterhammers associated with GL96-06 are not instantaneous. The non-instantaneous rise time reduces the Poisson pressure since the fluid has more time to move with the pipe and avoid being pressurized (less piston effect is expected).

Based on a study of the potential amplification due to strain related coupling, it is determined that most amplification is of low significance compared to the initial pressure pulse. An upper bound on amplification of 15% in typical SW systems due to FSI is shown in the TBR {12.3.2}.

6.2.2 Attenuation Resulting from Pipe Movement

When a pressure pulse is incident upon a change in direction in the piping, an unbalanced load occurs and some pipe movement results. The pressure pulse exerts work on the piping system, and some energy is removed. This is represented in Figure 6-1. Attenuation at direction changes can be greater than 10% of the incident pulse. The amount of attenuation is dependent on many variables (the pulse shape, the mass being moved, the stiffness of the system, etc.).

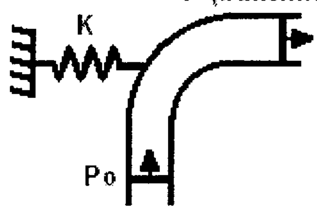


Figure 6-1
FSI Pressure Pulse Attenuation

The modeling of this type of attenuation needs to consider the length of the pressure pulse relative to the length of the line being entered. Long pulses entering short pipes may not result in a full peak pressure loading on the segment. The amount of attenuation may then be reduced. However, short pipes have less mass and are more likely to experience movement compared to longer pipes. Moody [10] presents the following equation for determining pressure pulse attenuation due to fluid structural interaction (FSI).

$$\frac{d^2}{dt^2}P(t)+\frac{\rho\cdot C\cdot A}{2\cdot m}\cdot\frac{d}{dt}P(t)+\frac{K\cdot g}{m}\cdot P(t)=\frac{d^2}{dt^2}P_o(t)+\frac{K\cdot g}{m}\cdot P_o(t)$$

where: t = time

P = pressure disturbance leaving the elbow (transmitted)

ρ = water density

C = sonic velocity in water

m = mass of system

A = flow area

K = axial restraint stiffness

g = gravitation acceleration constant

P_o = pressure disturbance entering the elbow (incoming)

If piping attenuation is to be taken into account, plant specific piping stiffnesses, segment lengths, and pulse magnitudes should be used to determine solutions to this differential equation to determine the specific attenuation to be applied at a particular plant. Example solutions are also provided in the TBR {12}.

6.2.3 FSI Recommendations

Both attenuation and amplification as a result of fluid structure interaction can occur. It is more likely that attenuation will be of significance. This is because the attenuation as a result of segment movement affects the pulse in an additive manner. That is, the pressure can be attenuated at each change in direction as a result of piping movement. If the pressure is attenuated 10% at each change in direction due to piping movement then it only takes 8 changes in direction to reduce the pressure by 50%. In contrast to this, amplification as a result of strain related coupling is a local effect. That is, it results in a local increase in pressure but the change does not propagate through the system. A 15% increase in pressure due to strain related coupling does not cause the pressure to get progressively higher.

It is overly conservative to consider amplification without crediting attenuation. Similarly, it is not appropriate to credit attenuation without considering potential amplification. If FSI is to be considered to reduce pressure magnitude in the piping evaluation, then either the effect of amplification needs to be specifically calculated or a bounding increase in the initial peak pressure of 15% should be added. FSI attenuation should then be calculated based on the system specific geometry, stiffness, and pressure pulse.

6.3 Friction

A pressure pulse traveling through a piping system will be attenuated by friction. The frictional effect on the pressure pulse is most significant in a high viscosity fluid with high frequency pressure pulses. The CCWHs being evaluated under GL96-06 are in low viscosity fluid and are relatively low frequency pulses. As a result, the column closure waterhammer pressure pulses experience insignificant frictional attenuation. It is conservative to ignore this energy loss mechanism. Additional discussion regarding frictional attenuation may be found in Wylie and Streeter [11].

6.4 Structural Loading

Structural loads are produced in a piping system by unbalanced pressure forces that occur as a pressure wave travels through a segment of piping. In the complete solution of the momentum equation, a momentum flux term will also contribute to the structural loading, but this is negligible in comparison to the pressure loading in this case. A representation of this phenomenon is presented in Figure 6-2 for a simplified trapezoidal pressure wave. This figure presents two different trapezoidal pressure waves each superimposed on a length of pipe running from point 1 to point 2. The wave enters the pipe from the left and travels across the span at the sonic velocity of the fluid. Based on the length of the piping segment compared to the length of the pressure wave, different shaped forcing functions can occur as shown in figure 6-2.

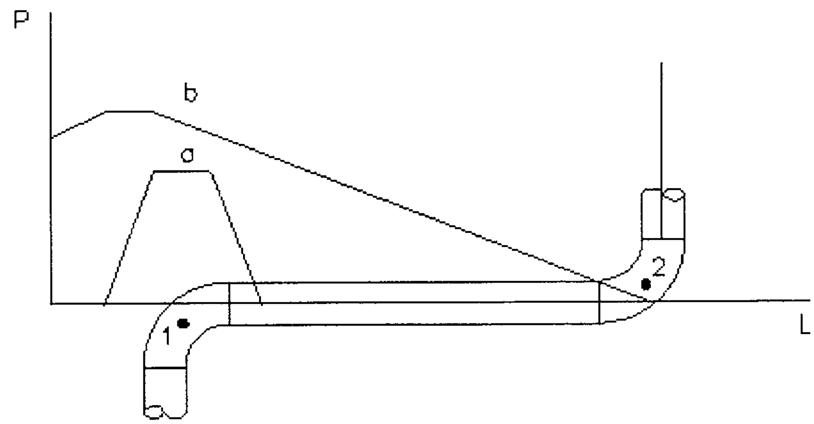
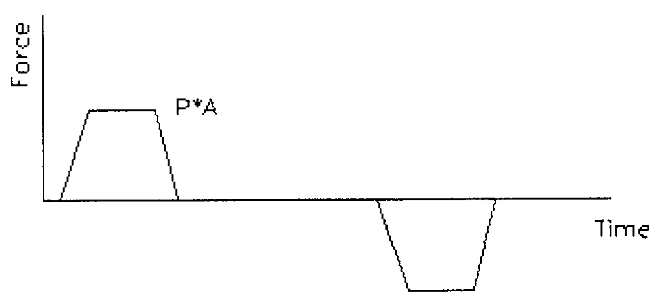


Figure 6-2
Differential Pressure Loading

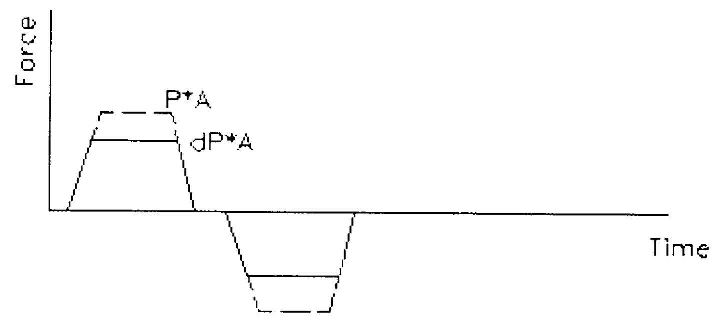
If the pressure wave is much shorter than the length of the pipe segment, then the pressure wave becomes "fully developed" within the pipe segment. This situation is depicted by the pressure wave labeled "a". Theoretically, the maximum unbalanced force on the pipe will be the pressure times the cross sectional area A of the pipe. As the wave moves down the pipe, the force on the segment increases to a value of $A \times P_{max}$ and then decreases linearly. The wave travels through the pipe and loads the other end in the same manner, producing a pair of positive and negative net forces on the segment as shown in figure 6-3 below.



Full Wave: L (segment length) > $t_d{\cdot}C$ (duration × sonic velocity)

Figure 6-3
Full Pressure Wave

If the pressure wave is longer than the pipe segment length, then the forcing function developed from the pressure wave will be truncated at the maximum differential pressure between the two end points. This is demonstrated by wave "b" in Figure 6-2. Since the pressure wave is longer than the piping segment, there is some pressure at both points "1" and "2" at the same time. The force on the segment is from the difference dP between the pressure at the two points, and is therefore truncated below the maximum. This condition is shown in Figure 6-4.



Truncated Wave: $L < t_r \cdot C$ (rise time x sonic velocity)

Figure 6-4
Truncated Pressure Wave

This example demonstrates that the net force on each straight pipe run is based on the differential pressure at each elbow for a simple pulse characterization. Loads can become more complicated for other wave approximations, and it is often easier to use the pressure-time history directly at each elbow. Of course, consideration must be made for the time offset between loads as the wave traverses a given pipe system.

The trapezoidal pulse shapes do not take into consideration any reflected waves resulting from direction changes, which can be as much as 10% to 15% of the original wave magnitude [13]. In the cases tested, reflected waves at direction changes were found to be insignificant {9.3.4}.

The following example presents the calculation of the forces and timing at each elbow in a sample system. A pressure pulse having the shape shown in Figure 6-5 loads an example piping system shown in Figure 6-6. The characteristics of the pressure pulse can be defined in terms of the six points shown in the figure.

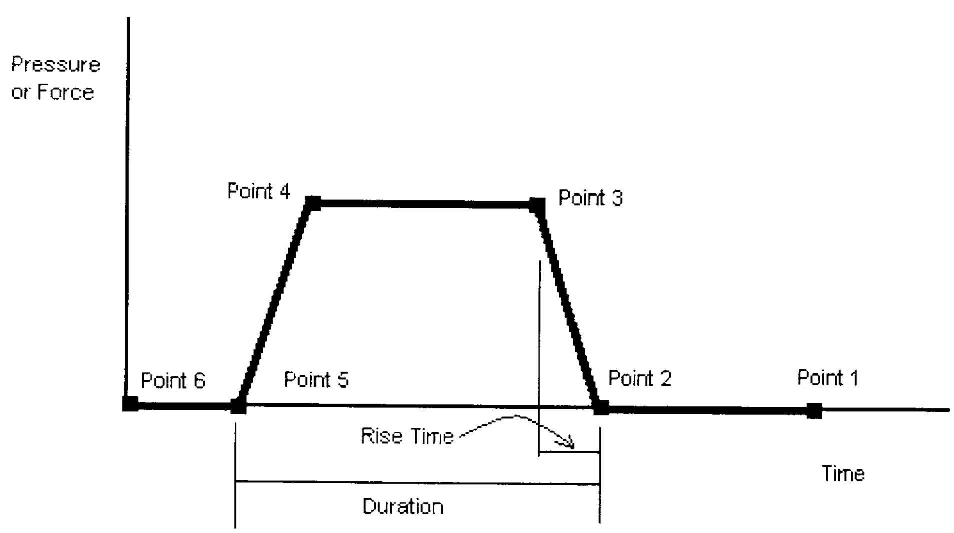


Figure 6-5
Example Pressure Pulse

The pressure pulse loads each node (elbow) as it passes, producing a force pair acting on each elbow as shown in Figure 6-6.

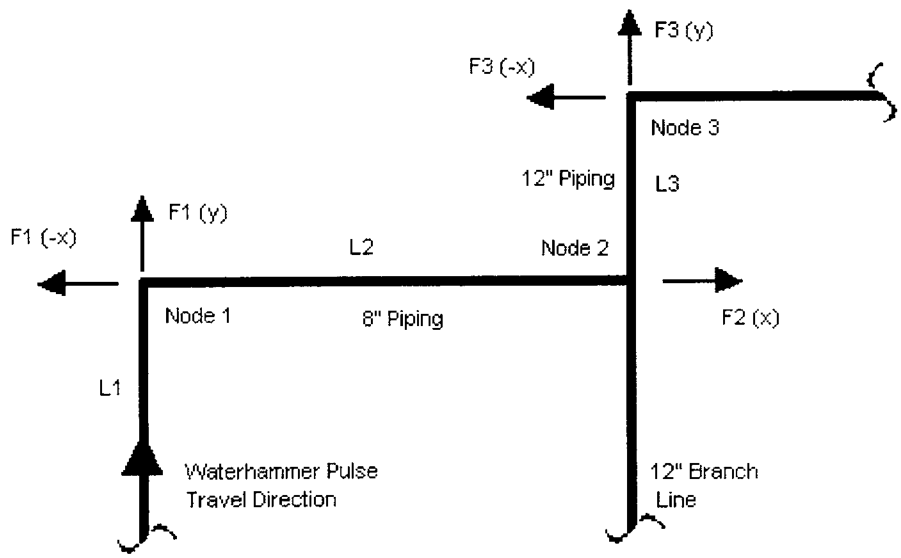


Figure 6-6
Example System Loads

The magnitude and timing of the forces are calculated in Table 6-2. This table contains the definition of the pressure wave (see "Pressure Wave" in the top half of the table) and the development of the force-time pairs for each node (see "Forces" in the bottom half of the table). The equations for the time and force at each node are provided below the calculated values. The symbols for the pressure wave definition (C for sonic velocity, t_r for rise time, etc.) are constant throughout. The time values ($t1$, $t2$, etc.) are node specific. Note that downstream of node 2, the pressure wave will travel in two directions, but only the forces at node 3 are presented in the example.

Common time start and stop points (point 1 and point 6 in Figure 6-5 and in Table 6-2) are often required for input into a piping structural program. They are assumed as point 1 = 0 sec (start) and point 6 = 0.1 sec (stop).

The forces calculated in Table 6-2 need to be applied in the direction of the attached pipe segments. Typically the direction cosines for each piping segment are used to apply these forces. In the application of the force-time pairs to Figure 6-6:

At node 1, apply $F1$ in (-x) and (+y) directions.

At node 2, apply $F2$ in (+x) direction.

At node 3, apply $F3$ in (-x) and (+y) directions.

Table 6-2
Force-Time History for Example System

Pressure Wave			
	Symbol	Value	Units
Wave Travel Speed	*C*	4500	ft/sec
Rise Time	t_r	0.005	sec
Duration	t_d	0.05	sec
Peak Pressure	*P*	200	psi

Forces																
			Point 1		Point 2		Point 3		Point 4		Point 5		Point 6		τ	
Node	Pipe Area	Seg Length	Time ($t1$)	Force	Time ($t2$)	Force	Time ($t3$)	Force	Time ($t4$)	Force	Time ($t5$)	Force	Time ($t6$)	Force	Seg	Cum
(units)	(in^2)	(ft)	(sec)	(lbs)	(sec)	(lbs)	(sec)	(lbs)	(sec)	(lbs)	(sec)	(lbs)	(sec)	(lbs)		
1	50	100	0	0	0.022	0	0.027	10000	0.067	10000	0.072	0	0.1	0	1.0	1.0
equation			0	0	$L1/C$	0	$t2+t_r$	PxA	$t2+t_d-t_r$	PxA	$t2+t_d$	0	TF	0		
2	113.10	50	0	0	0.033	0	0.038	8194	0.078	8194	0.083	0	0.1	0	0.362	0.362
equation			0	0	$(L1+L2)/C$	0	$t2+t_r$	$PxAx\tau$	$t2+t_d-t_r$	$PxAx\tau$	$t2+t_d$	0	TF	0	(1)	=Seg×Cum
3	113.10	10	0	0	0.036	0	0.041	8194	0.081	8194	0.086	0	0.1	0	1.0	0.362
equation			0	0	$(L1+L2+L3)/C$	0	$t2+t_r$	$PxAx\tau$	$t2+t_d-t_r$	$PxAx\tau$	$t2+t_d$	0	TF	0		=Seg×Cum

Note (1) From Equation 5-7, $\tau = 2 \cdot A_{inc}/(A_{inc}+\Sigma A_{br}) = 2 \cdot (50.0)/(50.0+2\times 113.1) = 0.362$

7
EXAMPLE PROBLEMS

7.1 Objective

The objective of this section is to provide two example problems that show how to calculate a SW system column closure waterhammer pressure pulse using the methods developed herein. Prototypical open and closed loop systems are simulated.

7.2 Sample Problem Inputs & Configuration

This evaluation requires:

- Knowledge of system hydraulics. System losses and pump characteristics.
- Calculation of voiding stage heat transfer and fluid dynamics.

The system configuration for the open loop simulation is shown in Figure 7-1. The closed loop configuration is shown in Figure 7-2. Table 7-1 summarizes the inputs to the evaluation for each case.

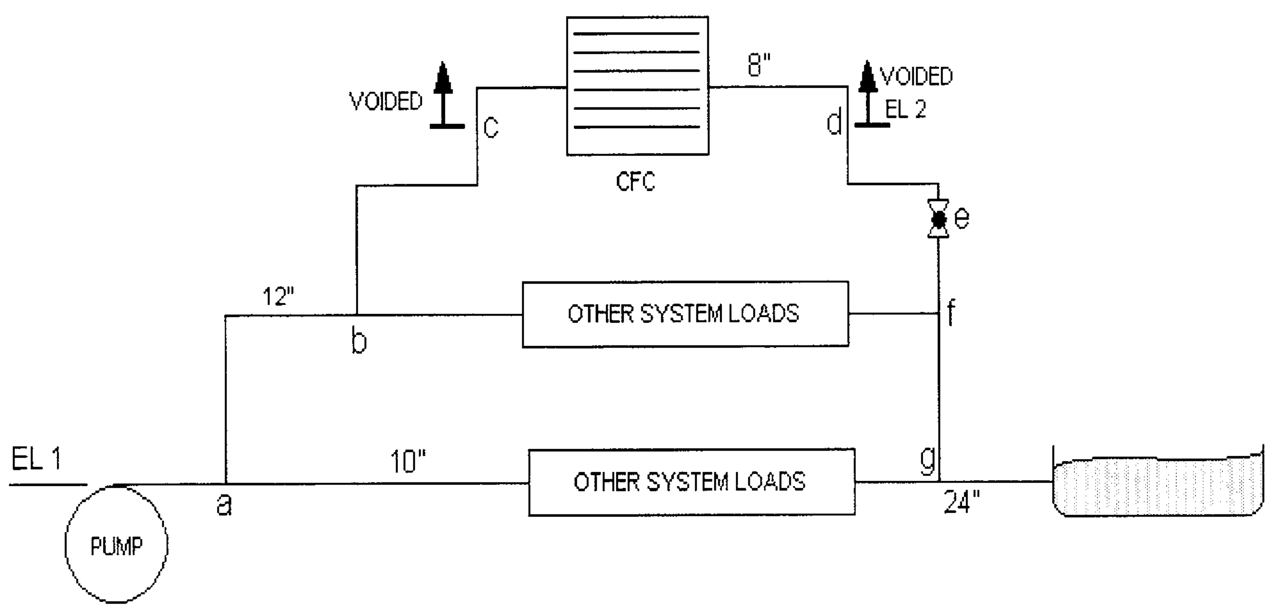


Figure 7-1
Open Loop Configuration

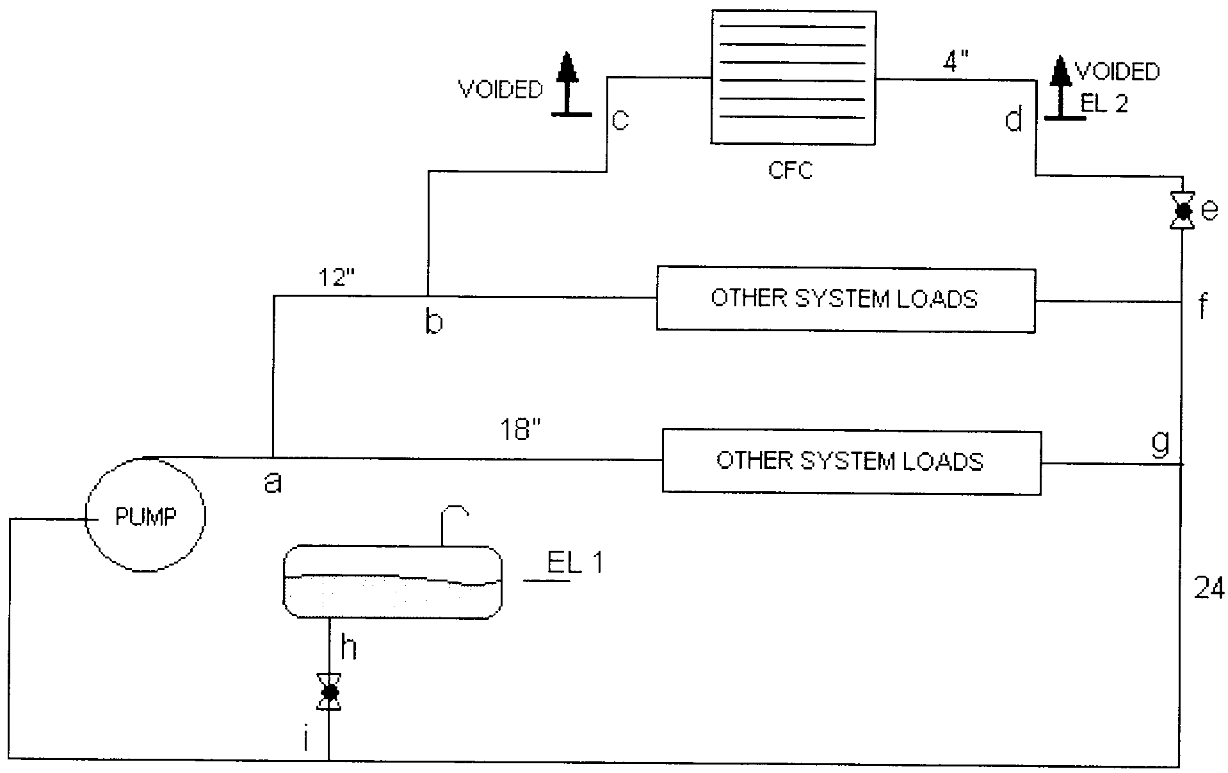


Figure 7-2
Closed Loop Configuration

Table 7-1
Example Problem Inputs

INPUT	OPEN LOOP VALUE	CLOSED LOOP VALUE	DESCRIPTION
EL 1	100 ft	250 ft	Elevation of point "a" and elevation of heat sink
EL 2	150 ft	220 ft	Elevation of points "c" and "d" (rear and front of void)
Patm	14.7 psia	14.7 psia	Atmospheric pressure
$H(Q)$	$300 + 0.3Q - 0.3Q^2$	$350 + 0.35Q - 0.35Q^2$	Pump discharge head as a function of flow rate. Head in units of feet of H_2O and flow in ft^3/sec.
Tvoid	210°F	245.5°F	*Void temperature when pumps restart; calculated from separate heat transfer/hydraulics evaluation*
Tpipe_initial	80°F	95°F	Piping temperature before the transient started.
Lab	50 ft	60 ft	Linear feet of piping from point "a" to point "b"
Lbc	40 ft	40 ft	Linear feet of piping from point "b" to point "c"
Lcd	40 ft	40 ft	Linear feet of 8" piping between point "c" and point "d"; this does not include the fan cooler.
Lde	90 ft	120 ft	Linear feet from front of void at point "d" to control valve at point "e"
Lef	5 ft	20 ft	Linear feet from point "e" to point "f"
Lfg	5 ft	20 ft	Linear feet from point "f" to point "g"
Qag	7000 gpm	4000 gpm	Flow from point "a" to "g"
Qabf	500 gpm	2000 gpm	Flow from point "b" to "f" through the other system load (not through the fan cooler).
Qbcd	950 gpm	200 gpm	Flow through the fan cooler
Kvlv	250 (8" throttled globe valve)	0.765 (4" open butterfly valve)	Equivalent resistance coefficient for valve at point "e" from $h = K \cdot V^2/2g$
Schedule	40	40	All piping is schedule 40, 12" piping is std.
Vol_{wtr_fcu}	0.5 ft^3	*0.5* ft^3	Volume of water that is left in the FCU when the pumps restart.
Vol_{wtr_2phase}	6 ft^3	16 ft^3	*Volume of water that flows into the cooler after voiding has started and before the pumps restart. This volume experiences two phase conditions.*
Ltube	15 ft	10 ft	Length of tubing in fan cooler
ID_tube	0.625"	0.625"	Fan cooler tube ID
N_tube	600	240	Number of fan cooler tubes
Tdes	95°F	100°F	System design temperature

7.3 Method Description

The analysis steps are described below.

7.3.1 Initial Velocity

A calculation of the initial closure velocity will be made. This will involve:

- Determining the flow coefficients for the flow paths.
- Writing the flow balance equations.
- Solving the set of simultaneous equations for re-closure flow rate (this step may be performed by using commercially available steady state software for hydraulic modeling of a system. If a hydraulic model of the system is already available, then it may be performed by creating another run for the voided configuration).

  The steady state flow through the FCU path will not be the same as the flow rate when voided. As will be shown in the open loop example, an equivalent flow coefficient for the steady state flow can be determined from the steady state model. This loss coefficient is reduced for the voided state by removing the effects of the pipe downstream of the void. The revised model is then run to determine re-closure velocity. The velocity produced in this manner is the V_o term described in Section 5.1, for which inertial limits of the accelerating refilling water can then be determined (V_i).

  For a closed loop plant, the system flow characteristics are controlled by flow from the head tank and movement of the upstream and downstream steam/water interfaces. The closed loop example shows the evaluation of the closure velocity based on constant void pressure and dissimilar flow into and out of the void. The difference between the upstream and downstream flows provides the closure velocity. This analysis must be performed on an individual plant basis.

7.3.2 Accelerating Column and Void Lengths

The lengths of the accelerating water column and gas volume will be calculated.

7.3.3 Mass of Gas

The mass of gas that becomes concentrated in the void as a result of boiling/two-phase flow before the pumps restart will be calculated.

7.3.4 Cushioned Velocity

Given the initial velocity, pipe size, void and water column lengths, the cushioned velocity can be determined using the charts of Appendix A.

7.3.5 Sonic Velocity

The sonic velocity will be calculated using Equation 5-1 and Equation 5-2 from Section 5.2.

7.3.6 Peak Pulse with No Clipping

The peak waterhammer pressure pulse without any clipping will be calculated using the cushioned velocity, sonic velocity, and Joukowski equation.

7.3.7 Rise Time

Knowing the cushioned velocity, the rise time will be calculated using Equation 5-4.

7.3.8 Transmission Coefficients

The transmission coefficients will be calculated using the methods presented in Section 5.3.

7.3.9 Duration

The pulse duration will be calculated using the guidance in Section 5.3.

7.3.10 Peak Pressure Clipping

The peak pressure pulse will be calculated considering clipping as described in Section 0.

7.3.11 Pressure Pulse Shape

The pressure pulse will be plotted as a function of time.

7.3.12 Flow Area Attenuation

Using the transmission coefficients defended in Section 5.3, the attenuation of the pulse as it travels through the system will be calculated.

The calculation is performed for the open and closed loop cases in Sections 7.4 and 7.5. A sensitivity evaluation is performed in Section 7.4.13, it considers alternate inputs and a RBM simulation of the specific configuration.

7.4 OPEN LOOP EXAMPLE PROBLEM

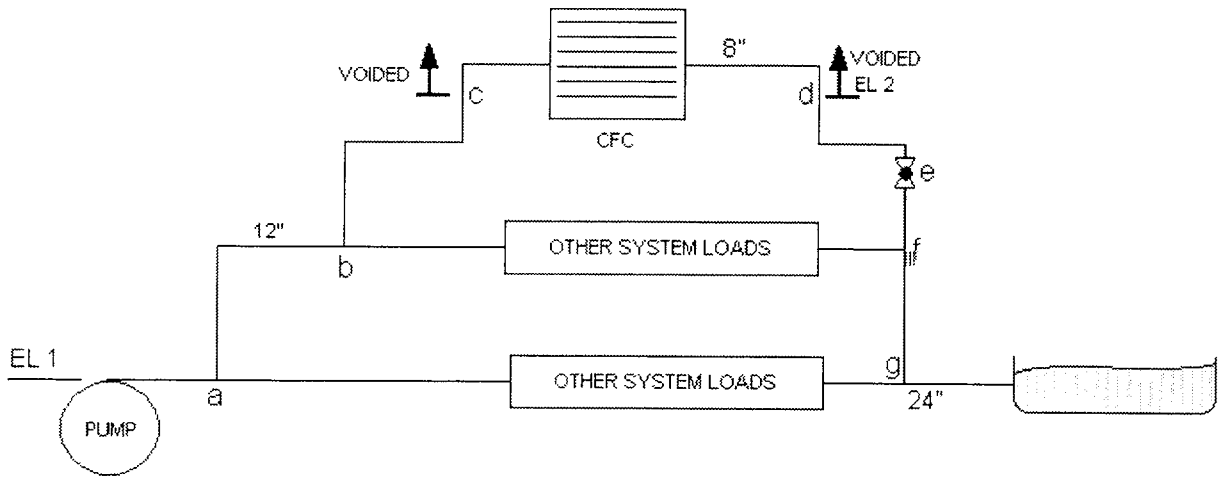


Figure 7-3: Open Loop Configuration

<u>Pressure & Temperature</u>

Note, pressures listed as "psi" are absolute (psia) or differential (psid) unless otherwise stated

$P_{atm} := 14.7 \cdot psi$	Pressure above reservoir and above sink (absolute)
$T_{void} := 210 \cdot F$	Temperature in the void when the pumps restart (ie surface temperature of piping)
$T_{pipe\_initial} := 80 \cdot F$	Temperature of the fluid and piping when the transient starts

<u>Pipe Geometry</u>

$EL_1 := 100 \cdot ft$	Elevation of node "1"
$EL_2 := 150 \cdot ft$	Elevation of node "2"
$L_{ab} := 50 \cdot ft$	Length from node "a" to node "b"
$L_{bc} := 40 \cdot ft$	Length from node "b" to node "c"
$L_{cd} := 40 \cdot ft$	Length from node "c" to node "d"
$L_{de} := 90 \cdot ft$	Length from node "d" to node "e"
$L_{ef} := 5 \cdot ft$	Length from node "e" to node "f"
$L_{fg} := 5 \cdot ft$	Length from node "f" to node "g"
$L_{g\_sinK} := 400 \cdot ft$	Length from node "g" to the ultimate heat sink.
$ID_{abf} := 12.00 \cdot in$	Inside diameter of piping along path from "a" to "b" to "f"
$ID_{bcd} := 7.981 \cdot in$	Inside diameter of piping along path from "b" to "c" to "d"
$ID_{ag} := 22.624 \cdot in$	Inside diameter of piping along remaining path from "a" to "g"
$OD_{bcd} := 8.625 \cdot in$	Outside diameter of piping along path from "b" to "c" to "d"

Flows

$Q_{abf} := 500 \cdot gpm$ Flow along path from "a" to "b" to "f" during steady state condition without voiding

$Q_{bcd} := 950 \cdot gpm$ Flow along path from "b" to "c" to "d" during steady state condition without voiding

$Q_{ag} := 7000 \cdot gpm$ Flow along remaining path from "a" to "g" during steady state condition without voiding

FCU Characteristics

$N_{tube} := 600$ Number of tubes in cooler

$ID_{tube} := .625 \cdot in$ Internal diameter of tubes

$L_{tube} := 15 \cdot ft$ Length of tubes

Pump Characteristics

$H_s := 300 \cdot ft$ Pump shutoff head

$A1 := .3 \frac{sec}{ft^2}$ 1st order pump curve coefficient

$A2 := -.3 \cdot \frac{sec^2}{ft^5}$ 2nd order pump curve coefficient

$Hpump(Qp) := A2 \cdot Qp^2 + A1 \cdot Qp + H_s$ Pump curve equation

Other Inputs

$K_{vlv} := 250$ Valve frictional flow coefficient for throttled globe valve

$V_{wtr\_fcu} := .5 \cdot ft^3$ Volume of water that is left in the FCU when the pumps restart

$V_{wtr\_2phase} := 6 \cdot ft^3$ Volume of water flows into the cooler after voiding has started and before the pumps restart. This volume of water is exposed to two phase flow conditions.

$\rho_{wtr} := 62 \cdot \frac{lb}{ft^3}$ Water density

$T_{des} := 95 \cdot F$ Design temp of the system

$R_{gas} := 1717 \cdot \frac{ft^2}{sec^2 \cdot R}$ Gas Constant

Pump Flow Rate Equation

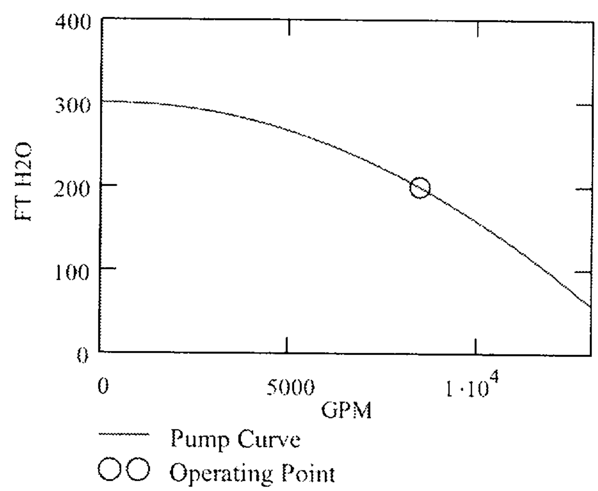
$Qtot_{normal} := Q_{ag} + Q_{bcd} + Q_{abf}$ $\qquad H_{norm} := Hpump(Qtot_{normal})$

$Qtot_{normal} = 8.45 \times 10^{3}\ gpm$ $\qquad H_{norm} = 199\ ft$

The total system flow rate is solved at any pump operating point using:

$$Qpump(Hd) := \frac{-A1 - \sqrt{A1^{2} - 4 \cdot A2 \cdot (H_s - Hd)}}{2 \cdot A2}$$

$$Qpump(H_{norm}) = 8.45 \times 10^{3}\ gpm$$



PUMP CURVE & OPERATING POINT

Figure 7-4

7.4.1 Initial Velocity & FLOW COEFFICIENT PREDICTION

The water at the front of the void (point "d") is assumed to not move for simplification in this problem. More detailed hydraulic modeling may be performed to determine the reverse or forward flow at point "d". In many cases this flow is less than 10% of the incoming flow.

After combining parallel paths the system is then simplified to:

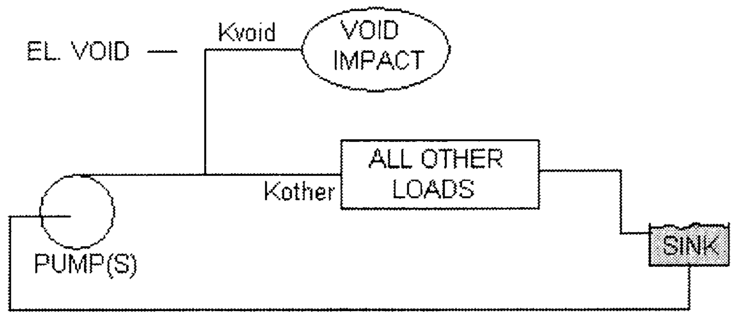


Figure 7-5: Simplified Open Loop Model

In terms of the initial flow diagram (Figure 7.3), the flow area for each path is calculated:

$A_{abf} := \frac{\pi}{4} \cdot ID_{abf}^{2}$ $\quad A_{bcd} := \frac{\pi}{4} \cdot ID_{bcd}^{2}$ $\quad A_{ag} := \frac{\pi}{4} \cdot ID_{ag}^{2}$

$A_{abf} = 0.785\,ft^2$ $\quad A_{bcd} = 0.347\,ft^2$ $\quad A_{ag} = 2.792\,ft^2$

The velocity for each path is calculated:

$V_{abf} := \frac{Q_{abf}}{A_{abf}}$ $\quad V_{bcd} := \frac{Q_{bcd}}{A_{bcd}}$ $\quad V_{ag} := \frac{Q_{ag}}{A_{ag}}$

$V_{abf} = 1.4\,fps$ $\quad V_{bcd} = 6.1\,fps$ $\quad V_{ag} = 5.6\,fps$

The flow coefficient for each flow path is calculated.

The flow resistance from point "a" to point "b" and from point "f" to point "g" are assumed to have a negligible effect on the flow split to the different paths. *In an actual plant system, the engineer may choose to use values from a previously qualified system hydraulic model to determine a more accurate initial velocity.*

$$h_f = K \cdot \frac{V^2}{2 \cdot g} \qquad \Rightarrow \qquad K = \frac{2 \cdot g \cdot h_f}{V^2}$$

$$K_{abf} := \frac{2 \cdot g \cdot H_{norm}}{V_{abf}^2} \qquad K_{bcd} := \frac{2 \cdot g \cdot H_{norm}}{V_{bcd}^2} \qquad K_{ag} := \frac{2 \cdot g \cdot H_{norm}}{V_{ag}^2}$$

$$K_{abf} = 6.375 \times 10^3 \qquad K_{bcd} = 346 \qquad K_{ag} = 411$$

An equivalent flow coefficient for the "other loads" path (Figure 7.3) is calculated from:

$$K_{other} := \left(\frac{A_{abf}}{\frac{A_{abf}}{\sqrt{K_{abf}}} + \frac{A_{ag}}{\sqrt{K_{ag}}}} \right)^2 \qquad K_{other} = 28 \qquad A_{other} := A_{abf} \qquad ID_{other} := ID_{abf}$$

The flow coefficient for the path to the void is calculated by subtracting the flow coefficient downstream of the void along this path. To simplify this sample problem only the valve resistance downstream of the void is considered:

$$K_{void} := K_{bcd} - K_{vlv} \qquad K_{void} = 96$$

The pressure in the void is assumed to correspond to the saturation pressure for the void temperature.

$P_{void} := 14 \cdot psi$ Absolute

The pump total developed head (TDH) is written by using Bernoulli's equation:

$$H_{atm} + EL_1 + TDH = H_{void} + EL_2 + H_f$$

where the following terms are defined in terms of feet H2O:
H_{atm} = atmospheric pressure head
EL_1 = elevation of node "1"
TDH = total developed head from pump
EL_2 = elevation of node "2"
H_f = frictional losses from point "1" to "2"

The frictional losses are written using Darcy's formula with an appropriate units conversion factor:

$$H_f = 0.00259 \cdot K_{loss} \cdot \frac{Q^2}{ID^4}$$

where
K_{loss} = loss coefficient
Q = flow rate in GPM
ID = pipe inside diameter in inches

Two equations for the total head developed (TDH) by the pump are written with a corresponding flow balance:

Given

$$TDH = 0.00259 \cdot K_{other} \cdot \frac{Q_{other}^2}{\left(\frac{ID_{abf}}{in}\right)^4}$$ frictional losses along "other" path equal the total developed head

$$TDH = 0.00259 \cdot K_{void} \cdot \frac{Q_{void}^2}{\left(\frac{ID_{bcd}}{in}\right)^4} + \left(EL_2 - EL_1 - \frac{P_{atm}}{\rho_{wtr} \cdot g} + \frac{P_{void}}{\rho_{wtr} \cdot g}\right) \cdot ft^{-1}$$ Bernoulli's along the "void" path

$$Q_{other} + Q_{void} = Qpump(TDH \cdot ft) \cdot gpm^{-1}$$ pump curve

The solution to the simultaneous equations is solved and defined as "Results".

$$Results := Find(TDH, Q_{other}, Q_{void})$$

$TDH := Results_0 \cdot ft$ $\qquad$ $TDH = 189.538\,ft$

$Q_{other} := Results_1 \cdot gpm$ $\qquad$ $Q_{other} = 7.318 \times 10^3\,gpm$

$Q_{void} := Results_2 \cdot gpm$ $\qquad$ $Q_{void} = 1.521 \times 10^3\,gpm$

The initial velocity is then:

$$V_{initial} := \frac{Q_{void}}{A_{bcd}} \qquad V_{initial} = 9.8\,ft\,sec^{-1}$$

The total resistance for this path is:

$$K_{void} = 96$$

check : is the velocity within the RBM bounds ?

$$V_{initial} < 20 \frac{ft}{sec}$$ ===> yes, velocity is within bounds of RBM runs

7.4.2 VOID & WATER COLUMN LENGTHS

The volume of piping that is voided is calculated:

$$V_{pipe\_voided} := L_{cd} \cdot \frac{\pi}{4} \cdot ID_{bcd}^2 \qquad V_{pipe\_voided} = 14\,ft^3$$

The void of the fan cooler unit is calculated:

$$V_{fcu} := N_{tube} \cdot L_{tube} \cdot \frac{\pi}{4} \cdot ID_{tube}^2 \qquad V_{fcu} = 19.2\,ft^3$$

The equivalent void length is then:

$$Lao := \frac{V_{pipe\_voided} + V_{fcu}}{A_{bcd}} \qquad Lao = 95\,ft$$

The initial water column length is assumed to be the distance from point "a" to point "c". The discussion that follows explains why point "a" was chosen:

Ignoring the FCU, the flow area changes from the closure point to node "a" are the same as the area changes from the closure point to node "g" on the return side. The transmission coefficients calculated for the return side demonstrate that less than 10% of the pressure pulse propagates to the header. Because of the similar flow area changes, less than 10% of any pressure would propagate into the supply header upstream of point "a". In general, this indicates that the header acts like a large pressurized reservoir during the void closure process and water in the supply header does not add to the inertia of the decelerating water column.

Note: if desired, a plant could select a length all the way back to the pumps. However, this is considered excessively conservative.

The length of the accelerating water column is then:

$$Lwo := L_{ab} + L_{bc} \qquad Lwo = 90\,ft$$

check : are the lengths within the bounds of the RBM ?

$Lao < 100ft$

$Lwo < 100ft$ ===>> yes lengths are within bounds of RBM runs

7.4.3 GAS RELEASE AND MASS OF AIR CONCENTRATED IN VOID

The mass of air concentrated in the void during the void phase of the transient is calculated by assuming that the water that has experienced boiling and subsequent condensation releases its air as described in Section 5 of the User's Manual.

For this problem, the tube volume only will be credited, assuming a draining FCU in which the headers do not remain full. This mass of water will release 50% of its non-condensable gas.

$V_{fcu} = 19.175\,ft^3$ or $V_{fcu} = 543\,liter$ From 7.4.2

$m_{fcu} := V_{fcu}\cdot\rho_{wtr}$ $m_{fcu} = 1189\,lb$

This represents the mass of water in the tubes which will lose 50% of its non-condensable gas. The concentration of gas is obtained from Figure 5-3.

$T_{des} = 95\,F$

$\frac{(T_{des} - 32\cdot F)}{1.8\cdot F} = 35$ deg C

$CON_{air} := 18\cdot\frac{mg}{liter}$

$m_{air} := 0.50 CON_{air}\cdot V_{fcu}$

$m_{air} = 4887\,mg$

check : is mass of air within bounds of UM ?

for void closure in 8" piping there should be at least 900 mg of air per Table 5.2.

===> yes, mass of air is within RBM run bounds.

7.4.4 Cushioned VELOCITY

The graphs presented in Appendix A for the velocity ratios are solutions to the simultaneous differential equations that capture the acceleration of the advancing column and pressurization of the void.

In order to determine the cushioned velocity the following terms that are needed are repeated:

$V_{initial} = 9.8\,ft\,sec^{-1}$

$K_{void} = 96$

$Lao = 95\,ft$

$Lwo = 90\,ft$

$m_{air} = 4887\,mg$

$T_{void} = 210\,F$

check : Is the temperature within the bounds of the RBM ?

$T_{void} > 200F$ ===> yes, the temperature is within the RBM run bounds

7.4.4.1 Air Cushioning

If only credit for air cushioning is considered then Figure A-10 from Appendix A is selected. This figure corresponds to 10" piping while the sample problem has 8" piping. 10" piping bounds the 8" piping since the inertia modeled in the 10" piping runs is greater than that in the 8" piping runs and the velocity has reached a steady state until the final void closure occurs. This is apparent by comparison of the 4" and 10" RBM run results for the same gas mass; the velocity is reduced more in the smaller pipe case. ***If the pipe size at a given plant is not shown then the Velocity Ratio chart for the next larger size pipe will always be bounding.***

Figure A-10 corresponds to a initial velocity of 10 fps. The initial velocity calculated in this sample problem is less. The higher velocity chart is selected because the higher momentum associated with the higher velocity bounds the lower velocity. ***If the initial velocity at a plant is not shown then the Velocity Ratio chart for the next larger velocity will always be bounding.***

For a K of 96 as calculated in the sample problem, from Figure A-10 the ratio of the second to initial velocity is:

[ ] only air cushion credited

Therefore, the final closure velocity will be reduced by 20% just considering air in this sample problem. Pressure "clipping" is not included here and is calculated later.

7.4.4.2 Air and Steam Cushioning

The velocity that results by considering steam cushioning is found using Figure A-37 from Appendix A. Note that the condensing surface temperature was verified being within the bounds of the RBM run limitations so steam condensation cushioning may be credited. The steam and air cushioning result in a ratio of cushioned to initial velocity of:

[ ] air and steam cushioning credited

The cushioned velocity is then:

[ ] [ ]

7.4.5 SONIC VELOCITY

The sonic velocity is calculated from Equation 5-1 and 5-2 in the main body of the User's Manual.

$p_{void} := 14 \cdot psi$

$$C = \sqrt{\frac{\frac{B}{\rho}}{\left(1 + \frac{B \cdot OD}{E \cdot t}\right)}}$$

where
B= bulk modulus of water $B := 319000 psi$
E = Young's modulus for steel $E := 28 \cdot 10^6 \cdot psi$
OD = outside diameter of pipe
t = wall thickness of pipe

$$C := \sqrt{\frac{\frac{B}{\rho_{wtr}}}{1 + \frac{B \cdot OD_{bcd}}{E \cdot \left(\frac{OD_{bcd} - ID_{bcd}}{2}\right)}}}$$

$$C = 4274 \frac{ft}{sec}$$

7.4.6 PEAK PRESSURE PULSE WITH NO "CLIPPING"

The peak waterhammer pressure is calculated using the Joukowski equation with a coefficient of 1/2 for a water on water closure:

$$\Delta P_{no\_clipping} := \frac{1}{2} \cdot \rho_{wtr} \cdot C \cdot V_{cushion}$$

$$\Delta P_{no\_clipping} = 218\,\text{psi}$$

7.4.7 RISE TIME

The rise time is calculated by using equation 5-4 from the UM.

[ ] ms = milliseconds

7.4.8 TRANSMISSION COEFFICIENTS

The pressure pulse may be affected by rarefaction waves as it is developing and the peak may be "clipped". In addition, the pressure may be attenuated as it propagates through the system as a result of area changes. In order to calculate each of these effects, the transmission coefficients at junctions is required. The transmission coefficients are calculated consistent with section 5.3 of the UM.

At points "f" and "g" the transmission coefficients are calculated using Equation 5-8 from the UM; for simplification here the sonic velocity is assumed to be constant up and downstream of the junction:

$$\tau = \frac{2 \cdot A_{incident}}{A_{incident} + \sum_j A_j}$$

$$\tau_f := \frac{2 \cdot A_{bcd}}{A_{bcd} + A_{abf} + A_{abf}} \qquad \tau_f = 0.362$$

=> 36% of the incident pulse continues past point "f" and 64% of the incident pulse returns towards the initiation point.

$$\tau_g := \frac{2 \cdot A_{abf}}{A_{abf} + A_{ag} + A_{ag}} \qquad \tau_g = 0.247$$

=> 25% of the pulse that is incident upon point "g" continues past point "g" and 75% returns towards the initiation point.

$$\tau_{total} := \tau_g \cdot \tau_f$$

$$\tau_{total} = 0.089$$

When the pressure pulse travels past point "g" only 10% of the pulse will continue on. 64% of the incident pulse was reflected as a negative pulse at point "f" and then 75% of the pulse that was incident upon point "g" was reflected back as a negative pulse. The net reflection effect is:

$$P_{ref} = P_{inc} \cdot (-64\%) + (36\% \cdot P_{inc}) \cdot (-75\%)$$

$$P_{ref} = P_{inc}(-64\% - 36\% \cdot 75\%)$$

$$P_{ref} = -91\%$$

This reflection travels back to the initiation point. The pulse at the initiation point is 9% of its original value when this reflection arrives. For simplicity, the compounding effect of the "f" node transmission coefficient on the reflected wave from node "g" is ignored.

The transmission coefficient evaluation needs to consider the control valve.

The transmission coefficient at the control valve is calculated by assuming the valve acts like an orifice as the pressure pulse propagates through it. Equation 5-14 provides a simple relationship for an orifice flow coefficient in terms of its diameter ratio (β). This equation is used to back calculate an equivalent β ratio for the control valve knowing its flow coefficient and assuming Co=0.6.

$$\beta := \mathrm{root}\left[\left(\frac{1}{0.6 \cdot \beta^2} - 1\right)^2 - K_{vlv}, \beta\right] \qquad \beta = 0.315$$

For this β ratio and for the approximate waterhammer pressure already solved, the control valve will have a slight effect on the pressure pulse propagation by inspection of Figure 5-15 The reflection from this interaction will add approximately 10% to the incident pulse.

In general what this means is that 10% of the pulse magnitude is reflected in a positive sense back towards the initiation point. To account for this effect, the peak pressure pulse is conservatively increased by 10%.

7.4.9 DURATION

The pressure pulse is reduced to approximately 10% of its peak value as a result of the reflections from the area changes at points "f" and "g". As a result, the time that it takes the pressure pulse to travel to point "g" and back may be used to calculate the pressure pulse duration.

$TD_{eg} := \frac{(L_{de} + L_{ef} + L_{fg}) \cdot 2}{C}$ $\quad TD_{eg} = 46.8\,\mathrm{ms}$ time for pulse to travel to and from point "g"
note that reflections from "a" & "b" are not credited.

The total duration is conservatively increased by adding the rise time.

$TD := TD_{eg} + TR$

$TD = 83\,\mathrm{ms}$

7.4.10 PRESSURE CLIPPING

The peak pressure is checked for "clipping" using Table 5-3.

$L_e := L_{de} + L_{ef} + L_{fg}$ $\quad L_e = 100\,\mathrm{ft}$ $\quad TR \cdot \frac{C}{2} = 76\,\mathrm{ft}$ $\quad \tau_{total} = 0.089$

This corresponds to the conditions in row two of the table referenced and no pressure clipping is expected.

$\Delta P := (1.1) \cdot \Delta P_{no\_clipping}$ 1.1 is from the control valve

$\Delta P = 239\,\mathrm{psi}$

7.4.11 PRESSURE PULSE SHAPE

The pulse shape is then characterized by four points.

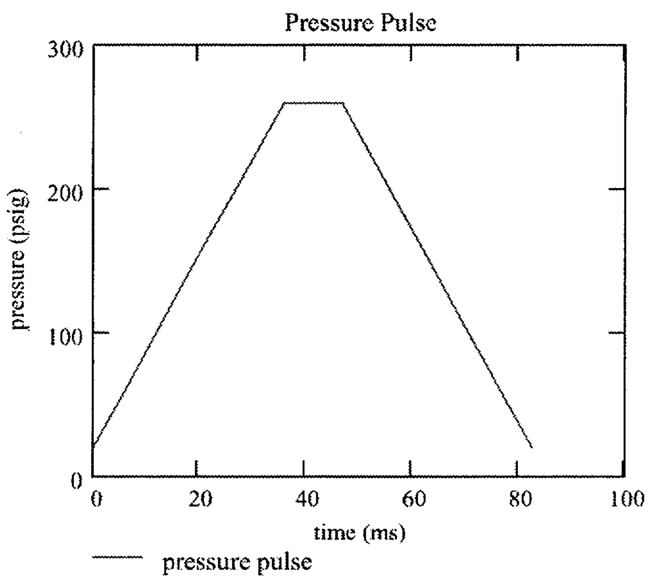
$Psys := 20 \cdot psi$ this is the steady state system pressure

Using an index, i = 0, 1, 2, 3 $\quad i := 0..3$

$time_i :=$	$pressure_i :=$
$0 \cdot ms$	Psys
TR	$\Delta P + Psys$
TD − TR	$\Delta P + Psys$
TD	Psys

This provides the following values, which are plotted below

$$time = \begin{pmatrix} 0 \\ 0.036 \\ 0.047 \\ 0.083 \end{pmatrix} sec \qquad pressure = \begin{pmatrix} 20 \\ 259 \\ 259 \\ 20 \end{pmatrix} psi$$



7.4.12 FLOW AREA ATTENUATION

To simplify the analysis of the SW structures, the approach suggested here is to take the initiating pressure pulse and propagate the pulse through the system. For this example problem, the duration of the pulse is assumed to remain unchanged as it travels. In reality, the duration of the pulse is shortened as it approaches negative reflection sites. Maintaining the duration conservatively increases the impulse.

As the pressure pulse propagates through the system it will be attenuated/amplified by flow area changes. For this example only the downstream propagation is considered. The pulse will be attenuated by the increase in area at "f" and "g". The transmission coefficients were previously calculated.

incident pulse	pulse transmission	transmitted pulse
$\Delta P = 239\,\mathrm{psi}$	$\Delta P_f := \tau_f \cdot \Delta P$	$\Delta P_f = 87\,\mathrm{psi}$
$\Delta P_f = 87\,\mathrm{psi}$	$\Delta P_g := \tau_g \cdot \Delta P_f$	$\Delta P_g = 21\,\mathrm{psi}$

Downstream of point "g" only the following pulse magnitude will remain: $\Delta P_g = 21\,\mathrm{psi}$

7.4.13 SENSITIVITY

Two approaches were taken to show sensitivity of the results.

First, the inputs for this problem were input directly into the Rigid Body Model to evaluate the affect of running the code versus using the simplified tables.

Second, inputs were changed and the results discussed using the tabulated data of Appendix A.

RBM Run Directly

The rigid body model was run for the air only and for the air/steam cushion case. Using the same inputs resulted in the following:

AIR CUSHION ONLY:	$\frac{\text{Vcushion}}{\text{Vinitial}} = .73$	there was more cushioning
	$\text{Rise\_Time} := 120 \cdot \text{ms}$	This rise time based on review of RBM is longer than calculated using eq. 5-3
STM & AIR CUSHION:	$\frac{\text{Vcushion}}{\text{Vinitial}} = 0.71$	there was more cushioning
	$\text{Rise\_Time} := 137 \cdot \text{ms}$	the rise time was even longer

It is apparent that there are implicit conservatisms in using the tabulated data. A direct RBM run for the particular configuration indicates the peak pressure is smaller and that the rise time is longer. Both of these should contribute to less structural loading.

Input Changes

Selected inputs were adjusted to demonstrate the sensitivity of the results to the particular configuration. The tabulated data from Appendix A was applied (RBM was not re-run here).

The following input changes are discussed:
1) Mass of air in the void
(2) Length of the void and accelerating column
(3) Transit time to major expansion (clipping)

The results of these variations are summarized and discussed below.

(1) Increased Air Mass

In order to achieve an increase in air mass, the volume of water that is heated was increased. A larger heat exchanger or larger risers near the cooler may contribute to more heated water.

The Air Mass was increased to 7800 mg and all other inputs remained as original.

AIR ONLY: [From 7800 mg of air on Fig A-10.]

AIR & STM: [From 7800 mg air on Figure A-37.]

[ ]

(2) Increase Water Column Length

If the water column length or the void length were increased beyond 100 ft then the appropriate values would be chosen from Figure A-13 (200 ft) or A-16 (400') for air cushion and A-40 (200') and A-43 (400').

[ ]

If the void length was decreased then the RBM runs could be used and conservative predictions of closure velocity would be expected.

(3) Reduced Transit Time

If the distance from the closure point to point "g" were reduced significantly then the peak pressure may be clipped. For example:

$L_{de} := 10 \cdot ft$ If this input is changed then the closure velocities are not affected but the pressure pulse is clipped.

DURATION

The pressure pulse is reduced to approximately 10% of its peak value as a result of the reflections from the area changes at points "f" and "g". As a result, the time that it takes the pressure pulse to travel to point "g" and back may be used to calculate the pressure pulse duration.

$$TD_{eg} := \frac{\left(L_{de} + L_{ef} + L_{fg}\right) \cdot 2}{C} \qquad TD_{eg} = 9.36\,\mathrm{ms}$$ time for pulse to travel to and from point "g"

This time is less than the rise time so some clipping is expected. $TR = 35.735\,\mathrm{ms}$

PRESSURE CLIPPING

The peak pressure is checked for "clipping" using the Table 5-3.

$$L_e := L_{de} + L_{ef} + L_{fg} \qquad L_e = 20\,\mathrm{ft} \qquad TR \cdot \frac{C}{2} = 76.36\,\mathrm{ft} \qquad \tau_{total} = 0.089$$

This corresponds to the conditions in row one of the table referenced and some pressure clipping is expected.

$\Delta P := (1.1) \cdot \Delta P_{no\_clipping}$ 1.1 is from the control valve

$\Delta P = 239\,\mathrm{psi}$ no clipping

$$Pnew := \left(\frac{1 - \tau_{total}}{TR \cdot \frac{C}{2}} \cdot L_e + \tau_{total} \right) \cdot \Delta P \qquad Pnew = 78\,\mathrm{psi}$$ with clipping

The pulse shape would then change as well. This pulse would be characterized as a triangular pulse with the new peak pressure and the new duration.

7.5 CLOSED LOOP EXAMPLE PROBLEM

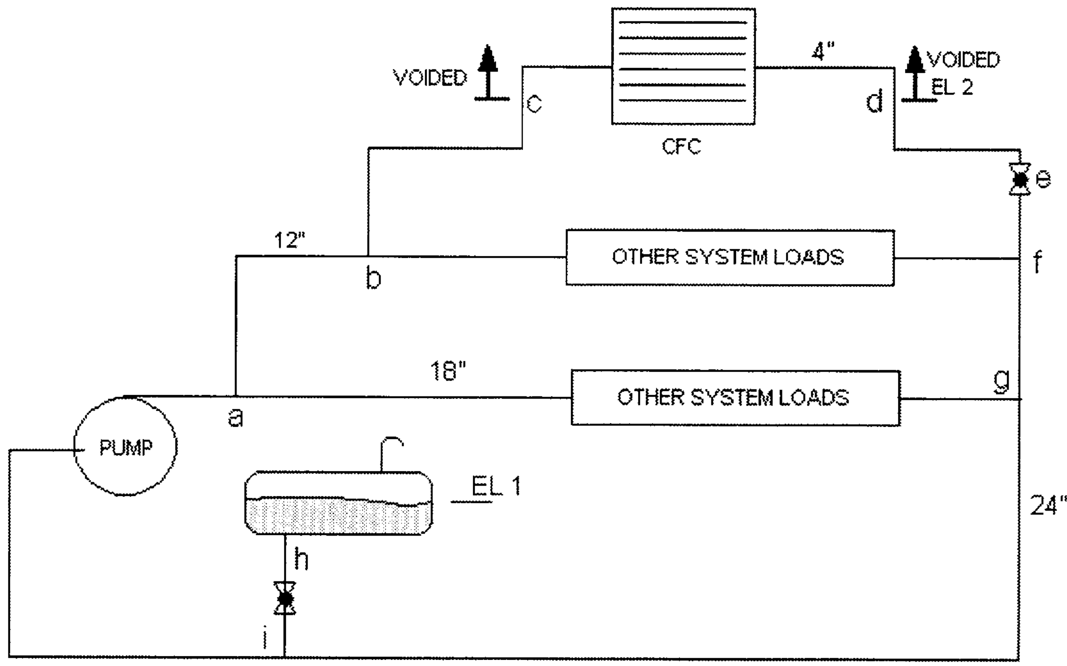


Figure 7-6: Closed Loop Configuration

<u>Pressure & Temperature</u>

Note, pressures listed as "psi" are absolute (psia) or differential (psid) unless otherwaise stated

$P_{atm} := 14.7 \cdot psi$	Pressure above reservoir and above sink (absolute)
$T_{void} := 245.5 \cdot F$	Temperature in the void when the pumps restart (ie surface temperature of piping)
$T_{pipe\_initial} := 95 \cdot F$	Temperature of the piping when the transient first started

<u>Pipe Geometry</u>

$EL_1 := 250 \cdot ft$	Elevation of node "1"
$EL_2 := 220 \cdot ft$	Elevation of node "2"
$EL_i := 200$	Elevation of node "i"
$EL_{all} := 200$	Elevation of all other nodes = 200 ft
$L_{ab} := 60 \cdot ft$	Length from node "a" to node "b"
$L_{bc} := 40 \cdot ft$	Length from node "b" to node "c"
$L_{cd} := 40 \cdot ft$	Length from node "c" to node "d"
$L_{de} := 120 \cdot ft$	Length from node "d" to node "e"
$L_{ef} := 20 \cdot ft$	Length from node "e" to node "f"
$L_{fg} := 20 \cdot ft$	Length from node "f" to node "g"

Geometry, continued

$ID_{abf} := 12.00 \cdot in$ Inside diameter of piping along path from "a" to "b" to "f" to "g"

$ID_{bcd} := 4.026 \cdot in$ Inside diameter of piping along path from "b" to "c" to "d"

$ID_{ag} := 16.876 \cdot in$ Inside diameter of piping along remaining path from "a" to "g"

$ID_{ga} := 22.624 \cdot in$ Inside diameter of piping along path from "g" to "a" (to suction of pump).

$ID_{hi} := 6.065 \cdot in$ Inside diameter of piping along path from "h" to "i" (head tank).

$OD_{bcd} := 4.5 \cdot in$ Outside diameter of piping along path from "b" to "c" to "d"

$L_{hi} := 100 \cdot ft$ Length from node "h" to node "i"

$K_{vlv\_hi} := 340 \cdot (.015)$ Open globe valve loss coefficient in path from node "h" to node "i"

$K_{vlv\_hi} = 5.1$

Flows (gpm)

$Q_{bf} := 2000 \cdot gpm$ (gpm) Flow along path from "a" to "b" to "f" during steady state condition without voiding

$Q_{bd} := 200 \cdot gpm$ Flow along path from "b" to "c" to "d" during steady state condition without voiding

$Q_{ag} := 4000 \cdot gpm$ Flow along remaining path from "a" to "g" during steady state condition without voiding

FCU Characteristics

$N_{tube} := 240$ Number of tubes in cooler

$ID_{tube} := .625 \cdot in$ Internal diameter of tubes

$L_{tube} := 10 \cdot ft$ Length of tubes

Pump Characteristics

$H_s := 350 \cdot ft$ Pump shutoff head (ft)

$A1 := .35 \cdot \frac{sec}{ft^2}$ 1st order pump curve coefficient $\frac{sec}{ft^2}$

$A2 := -.35 \cdot \frac{sec^2}{ft^5}$ 2nd order pump curve coefficient $\frac{sec^2}{ft^5}$

$Hpump(Qp) := \left(A2 \cdot Qp^2 + A1 \cdot Qp + H_s\right)$ Pump curve equation

Other Inputs

$\rho_{wtr} := 62 \cdot \frac{lb}{ft^3}$ Water density

$T_{des} := 100 \cdot F$ Design temperature of the system

$K_{vlv} := 0.765$ Valve frictional flow coefficient from Crane for 4" open butterfly valve

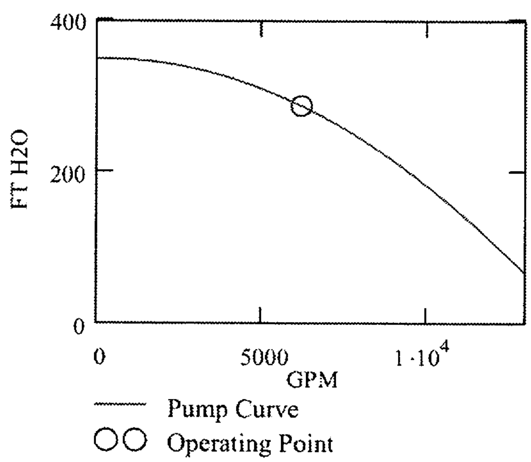
Pump Flow Rate Equation

$Qtot_{normal} := Q_{ag} + Q_{bd} + Q_{bf}$ $\qquad$ $H_{norm} := Hpump(Qtot_{normal})$

$Qtot_{normal} = 6.2 \times 10^3\ gpm$ $\qquad$ $H_{norm} = 288\ ft$

The total system flow rate is solved at any pump operating point using:

$$Qpump(Hd) := \frac{-A1 - \sqrt{A1^2 - 4 \cdot A2 \cdot (H_s - Hd)}}{2 \cdot A2} \qquad Qpump(H_{norm}) = 6.2 \times 10^3\ gpm$$



PUMP CURVE & OPERATING POINT

Non - Dimensionalizing

$H_s := \frac{H_s}{ft}$ $\qquad$ $H_s = 350$ $\qquad$ Pump shutoff head (ft)

$A1 := A1 \cdot \frac{gpm}{ft}$ $\qquad$ $A1 = 7.798 \times 10^{-4}$ $\qquad$ 1st order pump curve coefficient $\frac{sec}{ft^2}$

$A2 := A2 \cdot \frac{gpm^2}{ft}$ $\qquad$ $A2 = -1.737 \times 10^{-6}$ $\qquad$ 2nd order pump curve coefficient $\frac{sec^2}{ft^5}$

$Hpump(Qp) := \left(A2 \cdot Qp^2 + A1\,Qp + H_s\right)$

$Hpump(10000) = 184.057$ $\qquad$ Check OK

$Hpump(0) = 350$ $\qquad$ Check OK

System Elevations:

$EL_a := EL_{all}$

$EL_b := EL_{all}$ $\quad EL_b = 200$

$EL_e := EL_{all}$

$EL_i = 200$

$EL_f := EL_{all}$

$EL_g := EL_{all}$ $\quad EL_h := 250 \quad EL_d := 220$

The flow area for the 4" path (A_{bcd}) is calculated and used as a common reference diameter. Other flow areas are also determined:

$$A_{bcd} := \frac{\pi}{4} \cdot ID_{bcd}^{\ 2} \qquad A_{abf} := \frac{\pi}{4} \cdot ID_{abf}^{\ 2} \qquad A_{ag} := \frac{\pi}{4} \cdot ID_{ag}^{\ 2}$$

$$A_{bcd} = 0.088\,\text{ft}^2 \qquad A_{abf} = 0.785\,\text{ft}^2 \qquad A_{ag} = 1.553\,\text{ft}^2$$

The velocity for each path is calculated (normalized to the reference 4" pipe):

$$V_{bf} := \frac{Q_{bf}}{A_{bcd} \cdot \frac{ft}{sec}} \qquad V_{bd} := \frac{Q_{bd}}{A_{bcd} \cdot \frac{ft}{sec}} \qquad V_{ag} := \frac{Q_{ag}}{A_{bcd} \cdot \frac{ft}{sec}} \qquad V_{gi} := \frac{\left(Q_{bf} + Q_{bd} + Q_{ag}\right)}{A_{bcd} \cdot \frac{ft}{sec}}$$

$$V_{bf} = 50.405 \qquad V_{bd} = 5.04 \qquad V_{ag} = 100.81 \qquad V_{gi} = 156.255$$

Other known values at steady state:

$f := 0.015$

$h_h := 34.25$

$V_{hi} := 0$

$$K_{hi} := \left(\frac{f \cdot L_{hi}}{ID_{hi}} + K_{vlv\_hi}\right) \cdot \left(\frac{ID_{bcd}}{ID_{hi}}\right)^4 \qquad K_{hi} = 1.566$$

$$A := \frac{A_{bcd}}{ft^2} \qquad A = 0.088$$

7.5.1 Initial Velocity & FLOW COEFFICIENT PREDICTION

A system flow balance calculation is performed to determine the heads, velocities and loss coefficients for the system. This step may be performed using standard plant models and commercially available software. In this example, a simple model using simultaneous linear equations for head and flow at each node is solved using MathCad's "solve function". This function requires initial seed values for the unknown variables.

The steps used in this solution approach are as follows:

1) Define seed values

2) Define linear equations for head (Bernoulli equation) and flow (continuity)

3) Solve for head, velocity and loss coefficients under known, steady state conditions

4) Re-define the linear equations for void refill using the known use the loss coefficients. In this case, a dissimilar flow into and out of a constant pressure void will provide refill velocity.

5) Solve for the void refill velocity. The water that refills the void will quickly accelerate to a steady state velocity for most void sizes. It is conservative to use this velocity or reduce it by the methods described in Section 5.1.

Normal Condition

Seed Values

The following flow rates, heads, and "K"'s are seed (initial guess) values for the MCAD solve block

$h_a := 250$

$h_b := 250$

$h_d := 220$

$h_f := 250$

$h_g := 250$

$h_i := 250$

$V_{ab} := 5$

$V_{fg} := 5$

$V_{df} := 5$

$V_{ia} := 5$

$K_{ab} := 1$

$K_{bd} := 1$

$K_{df} := 1$

$K_{bf} := 1$

$K_{ag} := 1$

$K_{fg} := 1$

$K_{gi} := .5$

Normal Condition

Simultaneous Equations

(1) The flow is balanced at each junction.
(2) Bernoulli's equation is written to balance the pressure across each element using the following function:

$$h_{dwn}(EL_{dwn}, h_{up}, EL_{up}, K_{flow}, V) := h_{up} + EL_{up} - EL_{dwn} - K_{flow} \cdot \frac{V^2 \cdot \frac{V}{|V|}}{2 \cdot \left(g \cdot \frac{sec^2}{ft} \right)}$$

Given

Pressure Balance Equations

$$h_b = h_{dwn}(EL_b, h_a, EL_a, K_{ab}, V_{ab})$$

$$h_d = h_{dwn}(EL_d, h_b, EL_b, K_{bd}, V_{bd})$$

$$h_f = h_{dwn}(EL_f, h_d, EL_d, K_{df}, V_{df})$$

$$h_f = h_{dwn}(EL_f, h_b, EL_b, K_{bf}, V_{bf})$$

$$h_g = h_{dwn}(EL_g, h_a, EL_a, K_{ag}, V_{ag})$$

$$h_g = h_{dwn}(EL_g, h_f, EL_f, K_{fg}, V_{fg})$$

$$h_i = h_{dwn}(EL_i, h_g, EL_g, K_{gi}, V_{gi})$$

$$h_i = h_{dwn}(EL_i, h_h, EL_h, K_{hi}, V_{hi})$$

$$h_i + EL_i + Hpump(V_{ia} \cdot A \cdot 7.48 \cdot 60) = h_a + EL_a$$

Flow Balance Equations

$$V_{gi} = V_{fg} + V_{ag}$$

$$V_{fg} = V_{bf} + V_{df}$$

$$V_{ia} = V_{hi} + V_{gi}$$

$$V_{ab} = V_{bd} + V_{bf}$$

$$V_{ia} = V_{ab} + V_{ag}$$

$$V_{df} = V_{bd}$$

(Conservatively assume resistance upstream of the void is small relative to the downstream piping to determine K_{bd})

$$K_{bd} = .1K_{df}$$

Mathcad's "Find" function is used to solve for an exact solution to the set of simultaneous equations:

$$soln := Find(h_a, h_b, h_d, h_f, h_g, h_i, V_{ab}, V_{fg}, V_{df}, V_{ia}, K_{ab}, K_{bd}, K_{df}, K_{bf}, K_{ag}, K_{fg}, K_{gi})$$

Normal Condition

The results of the solve block are assigned:

$h_a := soln_0$ $V_{ab} := soln_6$ $K_{ab} := soln_{10}$

$h_b := soln_1$ $V_{fg} := soln_7$ $K_{bd} := soln_{11}$

$h_d := soln_2$ $V_{df} := soln_8$ $K_{df} := soln_{12}$

$h_f := soln_3$ $V_{ia} := soln_9$ $K_{bf} := soln_{13}$

$h_g := soln_4$ $K_{ag} := soln_{14}$

$h_i := soln_5$ $K_{fg} := soln_{15}$

$K_{gi} := soln_{16}$

$h_a = 372.308$ $V_{ab} = 55.445$ $K_{ab} = 1.005$

$h_b = 324.314$ $V_{fg} = 55.445$ $K_{bd} = 0.11$

$h_d = 304.27$ $V_{df} = 5.04$ $K_{df} = 1.104$

$h_f = 323.834$ $V_{ia} = 156.255$ $K_{bf} = 0.012$

$h_g = 275.82$ $K_{ag} = 0.611$

$h_i = 84.25$ $K_{fg} = 1.005$

$K_{gi} = 0.505$

$V_{ag} = 100.81$

$V_{ag} + V_{fg} = 156.255$

$V_{hi} = 0$

Normal Condition Outputs

Flow Rate Summary

	flow in gpm
$Q_{ab} := V_{ab} \cdot A \cdot 7.48 \cdot 60$	$Q_{ab} = 2199.8$
$Q_{ag} := V_{ag} \cdot A \cdot 7.48 \cdot 60$	$Q_{ag} = 3999.7$
$Q_{bd} := V_{bd} \cdot A \cdot 7.48 \cdot 60$	$Q_{bd} = 200$
$Q_{bf} := V_{bf} \cdot A \cdot 7.48 \cdot 60$	$Q_{bf} = 1999.9$
$Q_{fg} := V_{fg} \cdot A \cdot 7.48 \cdot 60$	$Q_{fg} = 2199.8$
$Q_{gi} := V_{gi} \cdot A \cdot 7.48 \cdot 60$	$Q_{gi} = 6199.6$
$Q_{hi} := V_{hi} \cdot A \cdot 7.48 \cdot 60$	$Q_{hi} = 0$
$Q_{ia} := V_{ia} \cdot A \cdot 7.48 \cdot 60$	$Q_{ia} = 6199.6$

Pressure Summary

	Pressure in psia
$P_a := h_a \cdot \rho_{wtr} \cdot g \cdot ft$	$P_a = 160.299$ psi
$P_b := h_b \cdot \rho_{wtr} \cdot g \cdot ft$	$P_b = 139.635$ psi
$P_d := h_d \cdot \rho_{wtr} \cdot g \cdot ft$	$P_d = 131.005$ psi
$P_f := h_f \cdot \rho_{wtr} \cdot g \cdot ft$	$P_f = 139.428$ psi
$P_g := h_g \cdot \rho_{wtr} \cdot g \cdot ft$	$P_g = 118.757$ psi
$P_h := h_h \cdot \rho_{wtr} \cdot g \cdot ft$	$P_h = 14.747$ psi
$P_i := h_i \cdot \rho_{wtr} \cdot g \cdot ft$	$P_i = 36.274$ psi

Void Condition

Known conditions

void pressure = saturation pressure at void temp (245.5F) = 27.6 psia

$h_c := 284.3$ $K_{ab} = 1.005$ $K_{bf} = 0.012$ $K_{hi} = 1.566$ $EL_c := EL_d$ Guess Value

$h_d := 284.3$ $K_{bd} = 0.11$ $K_{ag} = 0.611$ $K_{gi} = 0.505$ $EL_c = 220$ $V_{bc} := 5$

$h_h = 34.25$ $K_{df} = 1.104$ $K_{fg} = 1.005$

Simultaneous Equations for void case

(1) The flow is balanced at each junction.
(2) Bernoulli's equation is written to balance the pressure across each element using the following function:

$$h_{dwn}(EL_{dwn}, h_{up}, EL_{up}, K_{flow}, V) := h_{up} + EL_{up} - EL_{dwn} - K_{flow} \cdot \frac{V^2 \cdot \frac{V}{|V|}}{2 \cdot \left(g \cdot \frac{sec^2}{ft} \right)}$$

Given

Pressure Balance Equations

$$h_b = h_{dwn}(EL_b, h_a, EL_a, K_{ab}, V_{ab})$$

$$h_c = h_{dwn}(EL_c, h_b, EL_b, K_{bd}, V_{bc})$$

$$h_f = h_{dwn}(EL_f, h_d, EL_d, K_{df}, V_{df})$$

$$h_f = h_{dwn}(EL_f, h_b, EL_b, K_{bf}, V_{bf})$$

$$h_g = h_{dwn}(EL_g, h_a, EL_a, K_{ag}, V_{ag})$$

$$h_g = h_{dwn}(EL_g, h_f, EL_f, K_{fg}, V_{fg})$$

$$h_i = h_{dwn}(EL_i, h_g, EL_g, K_{gi}, V_{gi})$$

$$h_i = h_{dwn}(EL_i, h_h, EL_h, K_{hi}, V_{hi})$$

$$h_i + EL_i + Hpump(V_{ia} \cdot A \cdot 7.48 \cdot 60) = h_a + EL_a$$

Flow Balance Equations

$$V_{gi} = V_{fg} + V_{ag}$$

$$V_{fg} = V_{bf} + V_{df}$$

$$V_{ia} = V_{hi} + V_{gi}$$

$$V_{ab} = V_{bc} + V_{bf}$$

$$V_{ia} = V_{ab} + V_{ag}$$

Mathcad's "Find" function is used to solve for an exact solution to the set of simultaneous equations:

$$Vsoln := Find(h_a, h_b, h_f, h_g, h_i, V_{ab}, V_{bc}, V_{df}, V_{bf}, V_{ag}, V_{fg}, V_{gi}, V_{ia}, V_{hi})$$

Void Condition

The results of the solve block are assigned:

$h_a := Vsoln_0$ $\quad V_{ab} := Vsoln_5$

$h_b := Vsoln_1$ $\quad V_{bc} := Vsoln_6$

$h_f := Vsoln_2$ $\quad V_{df} := Vsoln_7$

$h_g := Vsoln_3$ $\quad V_{bf} := Vsoln_8$

$h_i := Vsoln_4$ $\quad V_{ag} := Vsoln_9$

$V_{fg} := Vsoln_{10}$

$V_{gi} := Vsoln_{11}$

$V_{ia} := Vsoln_{12}$

$V_{hi} := Vsoln_{13}$

$h_a = 363.359$ $\quad V_{ab} = 61.343$

$h_b = 304.611$ $\quad V_{bc} = 13.471$

$h_f = 304.179$ $\quad V_{df} = 2.658$

$h_g = 264.302$ $\quad V_{bf} = 47.873$

$h_g = 264.3$ $\quad V_{ag} = 102.145$

$h_i = 81.404$ $\quad V_{fg} = 50.531$

$V_{gi} = 152.676$

$V_{ia} = 163.489$

$V_{hi} = 10.813$

$V_{bc} - V_{df} = 10.813$

Void Condition Outputs

Flow Rate Summary

flow in gpm

$Q_{ab} := V_{ab} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{ab} = 2433.9$

$Q_{ag} := V_{ag} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{ag} = 4052.7$

$Q_{bc} := V_{bc} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{bc} = 534.458$

$Q_{df} := V_{df} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{df} = 105.454$

$Q_{bc} - Q_{df} = 429.004$

$Q_{bf} := V_{bf} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{bf} = 1899.4$

$Q_{fg} := V_{fg} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{fg} = 2004.9$

$Q_{gi} := V_{gi} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{gi} = 6057.6$

$Q_{hi} := V_{hi} \cdot A \cdot 7.48 \cdot 60$ $\quad Q_{hi} = 429.004$

Pressure Summary

Pressure in psig

$P_a := h_a \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_a = 156.446 \text{ psi}$

$P_b := h_b \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_b = 131.152 \text{ psi}$

$P_d := h_d \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_d = 122.407 \text{ psi}$

$P_f := h_f \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_f = 130.966 \text{ psi}$

$P_g := h_g \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_g = 113.797 \text{ psi}$

$P_h := h_h \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_h = 14.747 \text{ psi}$

$P_i := h_i \cdot \rho_{wtr} \cdot g \cdot ft$ $\quad P_i = 35.049 \text{ psi}$

The initial velocity is then:

$$V_{initial} := \left(V_{bc} - V_{df}\right) \cdot \frac{ft}{sec}$$

$$V_{initial} = 10.8 \frac{ft}{sec}$$

The total resistance for this path is:

$$K_{void} := K_{ab} + K_{bd}$$

$$K_{void} = 1$$

check : is the velocity within the RBM bounds (Table 5-1)?

$V_{initial} < 20 \frac{ft}{sec}$ ===> yes, velocity is within bounds of RBM runs

7.5.2 VOID & WATER COLUMN LENGTHS

The volume of piping that is voided is calculated:

$$V_{pipe\_voided} := L_{cd} \cdot \frac{\pi}{4} \cdot ID_{bcd}^{2} \qquad V_{pipe\_voided} = 4\ ft^{3}$$

The void of the fan cooler unit is calculated:

$$V_{fcu} := N_{tube} \cdot L_{tube} \cdot \frac{\pi}{4} \cdot ID_{tube}^{2} \qquad V_{fcu} = 5.1\ ft^{3}$$

The equivalent void length is then:

$$Lao := \frac{V_{pipe\_voided} + V_{fcu}}{A_{bcd}} \qquad Lao = 98\ ft$$

The initial water column length is assumed to be the distance from point "a" to point "c". The discussion that follows explains why point "a" was chosen:

Ignoring the FCU, the flow area changes from the closure point to node "a" are the same as the area changes from the closure point to node "f" on the return side. The transmission coefficients calculated in section 7.5.8 for the return side demonstrate that less than 10% of the pressure pulse propagates to the header. Because of the similar flow area changes, less than 10% of any pressure would propagate into the supply header upstream of point "a". In general, this indicates that the header acts like a large pressurized reservoir during the void closure process and water in the supply header does not add to the inertia of the decelerating water column.

Note: if desired, a plant could select a length all the way back to the pumps. However, this is considered excessively conservative.

The length of the accelerating water column is then:

$$Lwo := L_{ab} + L_{bc} \qquad Lwo = 100\ ft$$

check : are the lengths within the bounds of the RBM ?

$Lao < 100ft$

$Lwo < 100ft$ ===>> yes lengths are within bounds of RBM runs

7.5.3 GAS RELEASE AND MASS OF AIR CONCENTRATED IN VOID

The mass of air concentrated in the void during the void phase of the transient is calculated by assuming that the water that has experienced boiling and subsequent condensation releases its air as described in Section 5 of the User's Manual.

For this problem, the tube volume only will be credited, assuming a draining FCU in which the headers do not remain full. This mass of water will release 50% of its non-condensable gas.

$V_{fcu} = 5.113\,ft^3$ or $V_{fcu} = 145\,liter$ From 7.5.2

$m_{fcu} := V_{fcu} \cdot \rho_{wtr}$ $m_{fcu} = 317\,lb$

This represents the mass of water which will lose 50% of its non-condensable gas. The concentration of gas is obtained from Figure 5-3.

$T_{des} = 100\,F$

$$\frac{(T_{des} - 32 \cdot F)}{1.8 \cdot F} = 37.8 \quad \text{deg C}$$

$$CON_{air} := 16 \cdot \frac{mg}{liter}$$

$$m_{air} := 0.50 CON_{air} \cdot V_{fcu}$$

$m_{air} = 1158\,mg$

check : is mass of air within bounds of UM ?

for void closure in 4" piping there should be at least 240 mg of air per Table 5-2.

===> yes, mass of air is within RBM run bounds.

7.5.4 CUSHIONED VELOCITY

The graphs presented in Appendix A for the velocity ratios are solutions to the simultaneous differential equations that capture the acceleration of the advancing column and pressurization of the void.

In order to determine the cushioned velocity the following terms that are needed are repeated:

$V_{initial} = 10.8\,ft\,sec^{-1}$

$K_{void} = 1$

$Lao = 98\,ft$

$Lwo = 100\,ft$

$m_{air} = 1158\,mg$

$T_{void} = 245.5\,F$

check : Is the temperature within the bounds of the RBM (Table 5-2)?

$T_{void} > 200F$ ===> yes, the temperature is within the RBM run bounds

Air Cushioning

If only credit for air cushioning is considered then Figure A-2 from Appendix A is selected. This figure corresponds to 4" piping. ***If the pipe size at a given plant is not shown then the Velocity Ratio chart for the next larger size pipe will be bounding.***

Figure A-2 corresponds to an initial velocity of 15 fps which bounds the sample problem initial velocity. ***If the initial velocity at a plant is not shown then the Velocity Ratio chart for the next larger velocity will be bounding.***

For a K of 1 as calculated in the sample problem, from Figure A-2 the ratio of the cushioned to initial velocity is:

[ ] only air cushion credited

The final closure velocity will be reduced by 12% as a result of just considering air in this sample problem. Pressure "clipping" is not included here and is calculated later.

Air and Steam Cushioning

Steam cushioning is then credited. The cushioned velocity that results by considering steam cushioning is found using Figure A-29 from Appendix A. The steam and air cushioning result in a ratio of cushioned to initial velocity of:

[ ] air and steam cushioning credited

The cushioned velocity is then:

[ ] [ ]

7.5.5 SONIC VELOCITY

The sonic velocity is calculated from Equations 5-1 and 5-2 in the UM.

$$C = \sqrt{\frac{\frac{B}{\rho}}{\left(1 + \frac{B \cdot OD}{E \cdot t}\right)}}$$

where
B= bulk modulus of water
E = Young's modulus for steel
OD = outside diameter of pipe
t = wall thickness of pipe

$B := 319000\,psi$

$E := 28 \cdot 10^{6} \cdot psi$

$$C := \sqrt{\frac{\frac{B}{\rho_{wtr}}}{1 + \frac{B \cdot OD_{bcd}}{E \cdot \left(\frac{OD_{bcd} - ID_{bcd}}{2}\right)}}}$$

$$C = 4427 \frac{ft}{sec}$$

7.5.6 PEAK PRESSURE PULSE WITH NO "CLIPPING"

The peak waterhammer pressure is calculated using the Joukowski equation with a coefficient of 1/2 for a water on water closure:

$$\Delta P_{no\_clipping} := \frac{1}{2} \cdot \rho_{wtr} \cdot C \cdot V_{cushion}$$

$$\Delta P_{no\_clipping} = 218\,\text{psi}$$

7.5.7 RISE TIME

The rise time is calculated by using equation 5-4 of the UM.

[ ] ms = milliseconds

7.5.8 TRANSMISSION COEFFICIENTS

The pressure pulse may be affected by rarefaction waves as it is developing and the peak may be "clipped". In addition, the pressure may be attenuated as it propagates through the system as a result of area changes. In order to calculate each of these effects, the transmission coefficients at junctions is required. The transmission coefficients are calculated consistent with section 5.

At point "f" the transmission coefficient is calculated using Equation 5-8; for simplification here the sonic velocity is assumed to be constant up and downstream of the junction:

$$\tau = \frac{2 \cdot A_{incident}}{A_{incident} + \sum_j A_j}$$

$$\tau_f := \frac{2 \cdot A_{bcd}}{A_{bcd} + 2A_{abf}}$$

$\tau_f = 0.107$ => 10.7% of the incident pulse continues past point "f" and 90.3% of the incident pulse returns towards the initiation point.

$$\tau_{total} := \tau_f$$

$$\tau_{total} = 0.107$$

When the pressure pulse travels past point "f" only 10.7% of the pulse will continue on. 90.3% of the incident pulse was reflected as a negative pulse at point "f".

This reflection travels back to the initiation point. The pulse at the initiation point is 10.7% of its original value when this reflection arrives.

The transmission coefficient evaluation needs to consider the control valve.

The transmission coefficient at the control valve is calculated by assuming the valve acts like an orifice as the pressure pulse propagates through it. Equation 5-14 provides a simple relationship for an orifice flow coefficient in terms of its diameter ratio (β). This equation is used to back calculate an equivalent β ratio for the control valve knowing its flow coefficient and assuming Co=0.6.

$$\beta := \mathrm{root}\left[\left(\frac{1}{0.6 \cdot \beta^2} - 1\right)^2 - K_{vlv}, \beta\right] \qquad \beta = 0.943$$

For this β ratio and for the approximate waterhammer pressure already solved, the control valve will have a negligible effect on the pressure pulse propagation by inspection of Figure 5-15.

7.5.9 DURATION

The pressure pulse is reduced to approximately 10.7% of its peak value as a result of the reflections from the area changes at point "f". As a result, the time that it takes the pressure pulse to travel to point "f" and back may be used to calculate the pressure pulse duration.

$$TD_{ef} := \frac{(L_{de} + L_{ef}) \cdot 2}{C} \qquad TD_{ef} = 63\,\text{ms}$$

time for pulse to travel to and from point "f"

The total duration is conservatively increased by adding the rise time.

$$TD := TD_{ef} + TR$$

$$TD = 101\,\text{ms}$$

7.5.10 PRESSURE CLIPPING

The peak pressure is checked for "clipping" using Table 5-3.

$$L_e := L_{de} + L_{ef} \qquad L_e = 140\,\text{ft} \qquad TR \cdot \frac{C}{2} = 82.732\,\text{ft} \qquad \tau_{total} = 0.107$$

This corresponds to the conditions in row two of the table referenced and no pressure clipping is expected.

$$\Delta P := \Delta P_{no\_clipping}$$

$$\Delta P = 218\,\text{psi}$$

7.5.11 PRESSURE PULSE SHAPE

The pulse shape is then characterized by four points:

$P_d = 122.407\,\text{psi}$ this is the system pressure at location "d" at the time of the transient

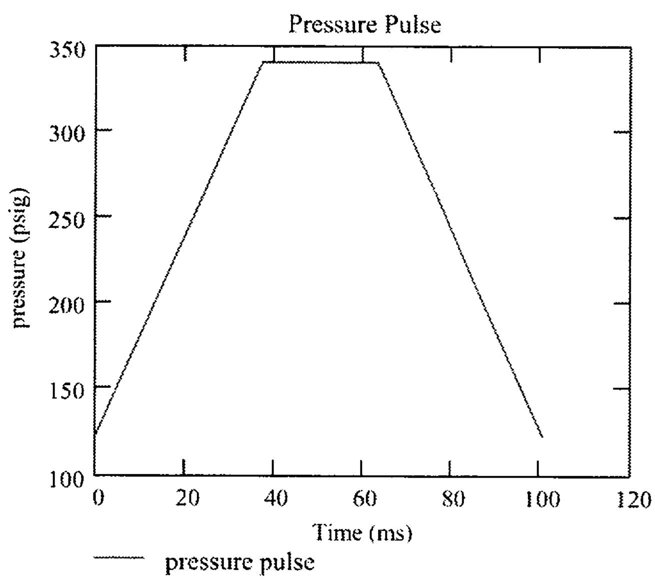
Using index i= 0,1,2,3: $i := 0..3$

$Time_i :=$ $pressure_i :=$

$Time_i$	$pressure_i$
$0 \cdot ms$	P_d
TR	$\Delta P + P_d$
$TD - TR$	$\Delta P + P_d$
TD	P_d

This provides the following values, which are plotted below

$$time = \begin{pmatrix} 0 \\ 0.036 \\ 0.047 \\ 0.083 \end{pmatrix} sec \qquad pressure = \begin{pmatrix} 122.407 \\ 340.201 \\ 340.201 \\ 122.407 \end{pmatrix} psi$$



7.5.12 FLOW AREA ATTENUATION

To simplify the analysis of the SW structures, the approach suggested here is to take the initiating pressure pulse and propagate the pulse through the system. For this example problem, the duration of the pulse is assumed to remain unchanged as it travels. In reality, the duration of the pulse is shortened as it approaches negative reflection sites. Maintaining the duration conservatively increases the impulse.

As the pressure pulse propagates through the system it will be attenuated/amplified by flow area changes. For this example only the downstream propagation is considered. The pulse will be attenuated by the increase in area at "f". The transmission coefficients were previously calculated.

incident pulse	pulse transmission	transmitted pulse
$\Delta P = 218\,\text{psi}$	$\Delta P_f := \tau_f \cdot \Delta P$	$\Delta P_f = 23\,\text{psi}$

Downstream of point "f" only the following pulse magnitude will remain: $\Delta P_f = 23\,\text{psi}$

8 REFERENCES

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4. NUREG/CR-5220, "Diagnosis of Condensation Induced Waterhammer," August 1988.

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A
APPENDIX A: RBM SOLUTIONS FOR CUSHIONING

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Notes Regarding Application of the RBM to the Curve Development

The method used to develop the curves included in the following figures was based on using fixed initial closure velocities (V = 10, 15 and 20 ft/sec). To provide a constant closure velocity for a varying set of resistance values (K = 10, 40, 70, 100), the driving pressure was varied within in curve. From an initial steady state velocity, the reduced (cushioned) velocity at impact was determined, and the ratio of these velocities is presented on each figure.

The range of gas mass used in the curves was varied based on pipe diameter. The range of gas used in the plots was based on the minimum amount of gas required to use gas and steam cushioning as provided in Table 5-2 and based on the limited benefit of gas cushioning at low gas mass and large pipe diameter. The following minimum gas in the plots was used:

4" Diameter Pipe 1000 mg

10" Diameter Pipe2000 mg

16" Diameter Pipe3000 mg

For the gas-only cushioning, it is possible to extrapolate the curves to lower gas mass, if desired. For gas and steam cushioning, the limits of Table 5-2 should be followed for low gas limits. It should be noted that in no cases is it valid for $V_{cushioned}/V_{initial}$ to be greater than 1.0. Within the accuracy of the RBM method, extrapolation of the provided values beyond 1.0 is not possible.

Based on the method used to generate the initial velocity, some models did not reach a steady state velocity in the available gas length. For these cases, a longer (200 ft) void length was used or the K = 10 values were not included. It is always conservative for a plant with a shorter void length or lower K value to use the higher values in the provided curves.

For steam and gas cushioning, a heat transfer coefficient of 64,000 BTU/hr·ft^2·°F was determined in the MOC validation as the highest value to match the test data {8}. This heat transfer coefficient was conservatively increased 12.5% to 72,000 BTU/ hr·ft^2·°F for the development of the RBM curves. This higher coefficient consumes steam faster and provides less cushioning.

Figure A-1: 4" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo*=100 ft

Figure A-2: 4" Pipe, Gas Cushioning, Initial Velocity 15 fps, *Lwo*=100 ft

Figure A-3: 4" Pipe, Gas Cushioning, Initial Velocity 20 fps, *Lwo*=100 ft

Figure A-4: 4" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo*=200 ft

Figure A-5: 4" Pipe, Gas Cushioning, Initial Velocity 15 fps, *Lwo*=200 ft

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Figure A-7: 4" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo*=400 ft

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Figure A-9: 4" Pipe, Gas Cushioning, Initial Velocity 20 fps, *Lwo*=400 ft

Figure A-10: 10" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo*=100 ft

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Figure A-13: 10" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo*=200 ft

Figure A-14: 10" Pipe, Gas Cushioning, Initial Velocity 15 fps, *Lwo*=200 ft

Figure A-15: 10" Pipe, Gas Cushioning, Initial Velocity 20 fps, *Lwo*=200 ft

Figure A-16: 10" Pipe, Gas Cushioning, Initial Velocity 10 fps, *Lwo*=400 ft

Figure A-17: 10" Pipe, Gas Cushioning, Initial Velocity 15 fps, *Lwo*=400 ft

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Figure A-33: 4" Pipe, Gas and Steam Cushioning, Initial Velocity 20 fps, *Lwo*=200 ft

Figure A-34: 4" Pipe, Gas and Steam Cushioning, Initial Velocity 10 fps, *Lwo*=400 ft

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Figure A-50: 16" Pipe, Gas and Steam Cushioning, Initial Velocity 15 fps, *Lwo*=200 ft

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Figure A-54: 16" Pipe, Gas and Steam Cushioning, Initial Velocity 20 fps, *Lwo*=400 ft

B
NRC SAFETY EVALUATION REPORT

UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D.C. 20555-0001

April 3, 2002

Mr. Vaughn Wagoner, Chairman
EPRI Waterhammer Project Utility Advisory Group
Carolina Power and Light Company
411 S. Wilmington Street, CPB 6A1
Raleigh, NC 27601

SUBJECT: NRC ACCEPTANCE OF EPRI REPORT TR-113594, "RESOLUTION OF GENERIC LETTER 96-06 WATERHAMMER ISSUES," VOLUMES 1 AND 2

Dear Mr. Wagoner,

The NRC staff has completed its review of the subject Electric Power Research Institute (EPRI) report that was submitted by letter dated December 15, 2000, as supplemented by letters dated July 10, 2001, August 9, 2001, September 17, 2001, and February 1, 2002. The staff finds this report acceptable for performing evaluations addressing GL 96-06 waterhammer concerns to the extent specified and within the limitations delineated in the EPRI report and in the associated NRC safety evaluation (enclosed). The safety evaluation presents the bases for our acceptance of the EPRI report.

In accordance with procedures established in NUREG-0390, the NRC requests that EPRI publish accepted versions of the submittal (as modified by the supplementary information that was provided), proprietary (-P) and non-proprietary (-NP), within 3 months of receipt of this letter. The accepted versions shall incorporate (1) this letter and the enclosed safety evaluation between the title page and the abstract and (2) an "-A" (designating "accepted") following the report identification symbol. The supplementary information that was submitted by letters dated July 10, 2001 (without the enclosures), August 9, 2001, September 17, 2001, and February 1, 2002, shall be appended to the approved version of the EPRI report.

Pursuant to 10 CFR 2.790, the staff has determined that the enclosed safety evaluation does not contain proprietary information. However, the staff will delay placing the safety evaluation in the NRC Public Document Room for 15 calendar days from the date of this letter to allow you the opportunity to comment on the proprietary aspects only. If, after that time, you do not request that all or portions of the safety evaluation be withheld from public disclosure in accordance with 10 CFR 2.790, the safety evaluation will be placed in the NRC Public Document Room.

If the NRC's criteria or regulations change so that its conclusion that the submittal is acceptable are invalidated, EPRI or the applicant making use of the EPRI report, or both, will be expected to revise and resubmit the respective documentation, or to submit justification for the continued applicability of the EPRI report without revision of the respective documentation.

OGC-02- 001057

Should you have any questions or want further clarification, please contact James Tatum at 301-415-2805.

Sincerely,

John N. Hannon, Chief
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UNITED STATES
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Enclosure

EVALUATION OF ELECTRIC POWER RESEARCH INSTITUTE

REPORT TR-113594, "RESOLUTION OF GENERIC LETTER 96-06

WATERHAMMER ISSUES," VOLUMES 1 AND 2

DATED DECEMBER 2000

1 INTRODUCTION

Generic Letter (GL) 96-06 required (among other things) that licensees evaluate containment air cooling systems to confirm that they are not vulnerable to waterhammer transients that might occur during a loss-of-coolant accident (LOCA) or a main steamline break (MSLB) event, concurrent with a loss of offsite power (LOOP). A LOOP during the postulated LOCA or MSLB would result in a temporary loss of pumping power to systems supplying cooling water to the air coolers. The heat that is released into containment during the event might cause the stagnant water in the cooling coils of the containment air coolers to boil and form steam. A waterhammer transient that exceeds the design limits of the cooling water system piping could occur when the containment air coolers are draining during the LOOP condition as hot steam comes in contact with cold water, or upon restoration of pumping power following the LOOP condition as cooling water is forced to flow into a steam pocket. Failure of cooling water systems that penetrate containment and remove heat from the containment air coolers could provide a pathway for radioactive gases to escape through the containment protective boundary to the environment. Failure of these cooling water systems could also result in a loss of heat sink for other safety-related components that are cooled by the failed cooling water system, as well as an unanalyzed increase in the amount of water that is discharged into the containment during the accident. If systems are found to be vulnerable to these conditions, licensees are expected to assess the operability of affected systems and take corrective action as appropriate.

Containment cooling systems are categorized as either open loop or closed loop systems. Open loop systems are typically found at fresh water sites. Closed loop systems are typically found at salt water sites. During a LOOP, the containment air coolers of open loop systems will typically drain, producing low-pressure voiding within the cooling coils without any additional heat being provided via a LOCA or MSLB. Closed loop systems typically do not drain during a LOOP and require additional heat from the containment atmosphere to produce voiding.

GL 96-06 did not prescribe how potential waterhammer events should be analyzed, and licensees were expected to use assumptions and methods that are appropriate and suitable for the plant-specific conditions involved. However, the GL suggested that licensees may find the information contained in NUREG/CR-5220, "Diagnosis of Condensation-Induced Waterhammer" (Ref. 1), to be informative and useful in evaluating potential waterhammer conditions. In order to assure that evaluations addressing the GL 96-06 waterhammer concerns were suitable and appropriate, responses to GL 96-06 were reviewed by the NRC and detailed requests for additional information (RAIs) were issued asking that licensees describe and justify the analytical methodology and assumptions that were used.

Waterhammer is a complex phenomenon, involving many variables that make it difficult to model and analyze. Bounding analytical methods use conservative assumptions and tend to yield conservative results. NUREG/CR-5220 estimates that these analyses can be conservative by a factor of 2 to 10, depending on the specific conditions involved. In order to address the RAIs that were issued, and to develop and justify a method that would be suitable for evaluating the waterhammer concerns discussed in GL 96-06, a group of utilities initiated a waterhammer testing and analysis program. By working as a group, licensees were able to minimize costs and avoid multiplication of effort in addressing the plant-specific RAIs that were issued. The scope of the program was designed and validated using a Phenomena Identification and Ranking Table (PIRT) assessment. The initiative was sponsored by 14 utilities, and the technical work was coordinated by the Electric Power Research Institute (EPRI) and performed by Altran Corporation. Independent program oversight and review was provided by a panel of industry experts, consisting of Dr. Peter Griffith of the Massachusetts Institute of Technology, Dr. Benjamin Wylie of the University of Michigan, and Dr. Fred Moody, an independent consultant. This evaluation will simply refer to EPRI as the organization responsible for completing this initiative. The NRC was also an active participant, monitoring and critiquing the work that was being performed by EPRI as the initiative progressed.

EPRI has completed its work on this initiative, and the results have been documented in EPRI Report TR-113594, "Resolution of Generic Letter 96-06 Waterhammer Issues," dated December 2000. The report consists of two volumes; Volume 1 is the User's Manual (UM), and Volume 2 is the Technical Basis Report (TBR). The UM establishes a methodology for participating utilities to use for evaluating the GL 96-06 waterhammer concerns, and the TBR provides the technical justification for the proposed methodology. The EPRI report was submitted for NRC review and approval by letter dated December 15, 2000. If approved, the EPRI report could be used by licensees in responding to the GL 96-06 plant-specific RAIs that were issued by the staff.

2 DISCUSSION

The testing and analysis program that was completed by EPRI investigated the two waterhammer phenomena that might occur in the low-pressure cooling water systems that are connected to the containment air coolers. These are column closure waterhammer, produced by the sudden start of a pump that causes water to flow into a partially voided section of piping; and condensation induced waterhammer, produced by unstable condensation in partially filled horizontal pipes.

2.1 Discussion of Column Closure Waterhammer (CCWH)

A CCWH, which results when two water columns join together within a pipe, can be evaluated by using the Joukowski equation for calculating the maximum pressure at column closure:

$$\Delta P = K\rho C(\Delta V)$$

In this equation when consistent units are used:

ΔP= the resulting pressure rise.

K= a dimensionless coefficient which is 0.5 for a column closure of liquid.

C= the velocity of sound in the medium.

ΔV= the velocity in the medium just prior to impact.

ρ= the density of the flowing liquid.

When the sonic velocity of cold water is used in the Joukowski equation, the resulting calculated pressure pulse is maximized. This methodology tends to provide results that are very conservative, with actual waterhammer loads being reduced by a factor of 2 to 10 from those predicted by the Joukowski equation due to reductions caused by air and steam cushioning, flexibility of the piping system, geometry of the water slug, frictional effects, and reductions that can occur in the length of the water slug (Ref. 1).

2.1.1 Air Release Due to System Depressurization

Small amounts of non-condensable gas bubbles suspended in the converging water columns will act to slow the speed of sound and have a direct effect on the resulting pressure rise. The solubility of air in water is proportional to the absolute pressure (Henry's law). When the pressure is reduced for water saturated with dissolved air, the excess air begins to leave the solution. Experiments by Zielke (Ref. 3) have shown that following a depressurization approximately 10 percent of the excess dissolved air contained in liquid water rapidly leaves solution, forming minute bubbles which dramatically reduce the speed of sound. The remainder of the excess dissolved air was found to leave solution more slowly depending on the presence of nucleation sites. Pressure reduction in containment cooling systems that result from the loss of pumping power in open cooling systems is expected to produce reductions in the speed of sound similar to that observed in the experiments. Reduction in the speed of sound will proportionally reduce the peak waterhammer pressure on impact.

2.1.2 Air Release Due to Boiling

If the containment atmosphere becomes heated as the result of MSLB or LOCA, the water contained in containment cooler units may boil, even in closed systems which do not experience a large pressure reduction on the loss of pumping power. Heating of water reduces the solubility of air. Boiling provides nucleation sites for the excess air, which cause the excess air to be stripped away with the steam.

To determine the air content in the void that might form within the low-pressure cooling water piping following a LOCA or MSLB at operating reactors, EPRI performed a series of tests. The test apparatus included a 5/8-inch ID copper tube that was 10 feet long to simulate one tube of a containment air cooler unit. The 5/8-inch tube was connected to a 2-inch vertical pipe to simulate a header. The 5/8-inch tube was sheaved in a 2-inch diameter steam jacket to simulate the containment environment during a LOCA or MSLB event. The tube was filled with water at approximately 70 °F. The tests were initiated by reducing the pressure within the copper tube to ½ atmospheric pressure to simulate loss of pumping power during a LOOP event. Heating by the containment atmosphere was simulated by supplying heated steam within the 2-inch diameter steam jacket. After 30 seconds, the test was terminated and water samples were collected. The oxygen content of the water was measured. This was compared to the original oxygen content. Since oxygen content should be proportional to air content, the difference should give a measure of the air evolved. In a first set of tests, only the copper test section was filled with water. EPRI determined that an excess of 50 percent of the original air in the test section came out of solution as the water was heated, boiled, and ejected from the test section.

In a second series of tests, the header pipe was filled with water. In these tests the water and steam from the test section were permitted to flow into the header. At the end of the test, EPRI measured the oxygen content in the header and in the water that flowed out of the header into an overflow tube. From these measurements, EPRI concluded that more than 24 percent of the air in a header to which fan cooler tubes are connected would be released if boiling occurred in a containment cooling unit during a LOCA or MSLB. In both series of tests, two temperatures of steam were used to vary the boiling rate. The air release fraction was found to be insensitive to the boiling rates that were used.

EPRI presented arguments that the test configuration was conservative in comparison to an actual containment air cooler unit in an operating plant. This is because in the test some of the water drained out of the test section without boiling, thus retaining much of its dissolved air. In an actual containment air cooler, the tube length is 3 to 4 times longer than the tube used in the test, which would permit more time for water in the tubes to boil. The water exiting the air cooler would be the hottest water in the air cooler and more likely to boil before draining into the header, compared to the test where the water was all the same temperature. Since a larger fraction of water in an actual air cooler would boil compared to the water in the test apparatus, a larger fraction than 50 percent of the air would be expected to be released.

2.1.3 Steam and Air Cushioning

Upon restoration of pumping power, the steam and air bubble located in the low-pressure cooling water piping for the containment air coolers will undergo compression and the steam will begin to condense. As the steam bubble condenses and begins to collapse, the air and steam contained in the void between two converging water columns will tend to cushion the resulting impact, thereby reducing the magnitude of the waterhammer pressure pulse.

CCWH tests were performed to measure the effects of air and steam cushioning. The effect of air cushioning was confirmed by performing experimental CCWH tests using water containing normal amounts of air (steam + air cushioning), and using de-aerated water (steam cushioning only). In both sets of tests, a steam bubble was formed in the test section by external heating and water was then accelerated into the steam bubble. Using de-aerated water resulted in waterhammer pressures that compared closely to the pressures predicted by the Joukowski equation, indicating that steam cushioning without the benefit of air was rather inconsequential. Using water with normal air content, the effect of steam and air cushioning together caused the resulting waterhammer pressures to be much less than predicted by the Joukowski equation.

The steam and air cushioning tests utilized a pipe section that was 2 inches in diameter and filled with water of normal air content. The test section was surrounded by an 8-inch diameter pipe section. Steam was admitted into the 8-inch diameter pipe section to cause the water in the 2-inch pipe to boil and form steam. A slug of water was then accelerated into the steam bubble, causing a waterhammer to occur. Various steam bubble lengths, dissolved air concentrations, and water column accelerations were investigated.

2.1.4 Correlation of CCWH Test Data

The data were correlated by two methods. The first method utilized the method of characteristics developed by B. Wylie (Ref. 4). The second method is a simplified calculational approach that was developed by Altran Corporation under contract to EPRI, which is referred to as the rigid body model.

2.1.4.1 Method of Characteristics (MOC)

The MOC allows the partial differential equations for momentum and continuity to be converted into ordinary differential equations. Pipelines are divided into equal length segments. The segment length is the distance that an acoustic wave will travel in one time interval. The MOC methodology is widely used for solving one dimensional, single phase, waterhammer problems. Wylie extended use of the MOC methodology to piping systems that have a discrete free-gas cavity, and this is the solution method used in the EPRI report. The method was benchmarked against test data from the CCWH tests. The tests had a known void size when the process of column closure was begun. The void was a mixture of steam and air which had left solution as a result of heating and boiling the water in the test section. As the columns converged, some of the steam was recondensed. The condensation rate was retarded by the presence of non-condensable air, resulting in a cushioning effect during column closure.

Although the initial dissolved air mass was measured for the CCWH tests, the final air mass was not. EPRI had to estimate the amount of air released based on the boiling time in the 2-inch test section. These estimates were refined by examining the waterhammer pressure pulse shapes. EPRI varied the heat transfer coefficient for condensing steam and the assumed air content until both the experimentally measured waterhammer pulse shapes and pulse magnitudes were reproduced. A constant heat transfer coefficient was derived that best correlated all of the CCWH test data. A heat transfer coefficient slightly larger in magnitude was proposed for plant evaluation purposes since a larger coefficient will produce a conservatively larger waterhammer pressure pulse.

2.1.4.2 Rigid Body Model (RBM)

The RBM methodology for waterhammer pressure pulse evaluation does not require a detailed noding of the piping system as does the MOC. Instead, a conservatively high waterhammer peak pressure pulse value is calculated using the Joukowski equation. A bounding pulse shape is calculated based on the time required for a pressure wave to travel to a reflecting surface and back. The calculated pulse magnitude and shape are then modified based on comparisons with the experimental data and nomographs that include the effects of steam and air cushioning. The nomographs were calculated by solving the differential equations for the acceleration of a water slug that is resisted by the compression of a gas bubble. The conservatively high heat transfer coefficient proposed for performing plant analyses from the MOC evaluations was used to account for the condensing of steam within the steam bubble. Information that is required for using of the nomographs are mass of air in the void, the frictional flow loss coefficient between the pump and the void, and the water slug length. The nomographs are provided in the UM. Table 5-2 in the UM provides analysis limits for use of the RBM. Applications lying outside these limits require plant specific analysis.

The CCWH test data indicated that the resulting waterhammer pulses were not rectangular in shape, but incrementally increasing in magnitude as the gas bubble was compressed. From this data, EPRI produced a correlation predicting the rate of increase for a pressure pulse using the RBM. Pressure pulse rise times were derived from a linearized treatment of the experimental pressure pulses. These rise times, in the form of a ramp, are applied to the peak pulses calculated using the RBM. The resulting pulse is trapezoidal or triangular in shape, depending on the geometry of the system. As a pressure pulse is transmitted through the piping system, the pulse will be reflected or attenuated by changes in piping geometry. Reflected pulses may affect the overall pressure in a positive or negative manner. EPRI provided a table and a nomograph to aid in determining the effect of reflected pulses.

The RBM was benchmarked against the MOC and found to be conservative. The difference between the two results is attributed to the treatment of reflected pressure pulses. The RBM was also compared to the experimental data from the CCWH experiments and was found to be conservative. The differences between measured and predicted values of pressure were attributed to the uncertainty in the amount of air present.

2.1.5 Scaling Considerations

All of the CCWH test data is from a 2-inch diameter test section. The diameter of the piping that supplies water to the containment air coolers in operating reactor plants is substantially larger than 2 inches. EPRI performed a scaling analysis showing that the condensing heat transfer coefficient was independent of the pipe flow area.

2.1.6 LOOP-Only Event

A CCWH in an open loop cooling water system that occurs during a LOOP-only event is expected to be more severe than one that occurs concurrently with a LOCA or MSLB. This is because the cooling water is not heated during the LOOP-only event. This results in much less steam and air being released into the void, which causes a reduction in air and steam cushioning, which causes the CCWH pressure to be higher.

Operating plants have conducted tests for LOOP, and some have experienced inadvertent LOOP events. These tests and events have resulted in CCWH occurrences in plants that have open loop containment air cooler cooling water systems. While minor piping support damage was noted in some instances, there were no piping failures.

One of these tests was conducted in November of 1991. The test was representative of a LOOP event with subsequent restart of the cooling water pumps. A peak pressure pulse of 205 psig was measured. The measured pulse is approximately ½ of that which would be obtained using the Joukowski equation without consideration of the non-condensable gas that would have come out of solution, and caused the resulting pressure peak to be reduced. The experience in operating plants tends to complement the analytical and experimental conclusions that have been reached, indicating that the release of non-condensable gas during system depressurization reduces the severity of CCWH.

2.1.7 Limitations and Restrictions Associated with CCWH Analyses

Use of either the MOC or RBM methodology for performing CCWH analyses of containment air cooler cooling water systems requires that licensees first perform an evaluation that is sufficient to obtain the necessary analytical inputs discussed in Section 2.2 of the UM (system design characteristics, worst-case conditions, steam bubble geometry, refill velocity, etc.). The following conditions must also be met in order for the analyses to remain within the scope of the test data:

- The calculated air mass must be greater than (60 mg)$(ID/2)^2$, where ID is the pipe inside diameter in inches at the location of the potential waterhammer.
- The calculated void temperature must be greater than 200 °F for the heat transfer coefficient derived by EPRI for steam and air cushioning to be valid.
- Additional limitations on use of the RBM are given in Table 5-2 of the UM. These limitations relate to water column size, void length, void and interface temperature, and initial closure velocity.

2.2 Discussion of Condensation Induced Waterhammer (CIWH)

This type of waterhammer might occur as containment cooling systems are drained during a LOOP condition, or during initiation of cooling water flow. CIWH typically occurs in horizontal pipe segments partially filled with cold water overlaid by steam that is supplied by a steam source. Condensation of steam on the surface of the water can cause tongues of water to be whipped up in a wave like manner. If steam condensation is vigorous enough, a water tongue can extend to the top of the pipe forming a plug. The steam bubble trapped between two plugs of cold water would experience a drop in pressure from continued condensation. The pressure difference then causes the two plugs to be driven together. CIWH occurs when the trapped steam bubble condenses and the plugs of water converge. Potential damage from this type of waterhammer is a function of the steam pressure that acts to drive the water plugs together. Piping damage from this type of waterhammer typically occurs in feedwater and other high pressure piping systems where steam pockets might form. In NUREG/CR 6519, "Screening Reactor Steam/Water Piping Systems for Water Hammer," dated September 1997, Griffith reports that in most cases system pressure must be greater than 100 psia before any significant damage will occur due to CIWH (Ref. 2). Cooling water systems that are used to remove heat from containment air coolers usually operate well below this pressure threshold.

2.2.1 Test Results

EPRI conducted a series of tests in order to evaluate the potential severity of CIWH on low-pressure cooling water systems, such as those that provide cooling for the containment air coolers. The test section was 4 inches in diameter and approximately 22 feet in length, providing a long horizontal pipe section for CIWH to take place. Steam pressures were limited to 15 psig. Tests were initiated by simultaneously opening valves to permit steam to enter one end of the horizontal pipe and water to exit the other end. Horizontal stratified flow conditions were produced, and from this CIWH occurred. The resulting pressure pulses were recorded by pressure transducers located throughout the system. The waterhammer pressures were found to be relatively low (less than 200 psig). Those pulses having the highest peak pressures were found to have the shortest duration and consequently, the impulse loads were approximately constant for all of the CIWH pressure pulses that were generated. As a result of the testing and analyses that were completed, EPRI concluded that a CIWH in most low-pressure cooling water systems for the containment air coolers would not be severe enough to cause any significant damage. EPRI also concluded that the CCWH would typically be more severe in magnitude and duration than the CIWH, and generally would form the more limiting case.

2.2.2 Scaling Considerations

Cooling water systems supplying the containment air cooler units for actual plants are generally larger than the 4-inch diameter that were used in the EPRI tests. Based on the results of a scaling analysis, EPRI concluded that lower CIWH pressures would be expected for larger pipe sizes.

2.2.3 Limitations and Restrictions Associated with CIWH Analyses

The CCWH tests performed by EPRI produced much higher pressures and loadings than those of the CIWH tests. EPRI concluded that for plant conditions that are within the bounds of the test data, CCWH will be bounding and plant-specific CIWH analyses need not be performed. As specified Section 4.2.1 of the UM, this conclusion is applicable only in systems that meet the following conditions:

- The system pressure at the time of the postulated CIWH must be less than 20 psig.
- The system water has not been degassed.
- The piping has been shown by test, analysis, or operating experience to be able to withstand a CCWH following LOOP, LOOP with LOCA, or LOOP with MSLB, without significant degradation.

2.3 Advisory Committee for Reactor Safeguards (ACRS) Review and Comment

EPRI met with the ACRS Thermal-Hydraulic Phenomena Subcommittee to present and discuss EPRI Report TR-113594, Volumes 1 and 2, during the meetings that were held on November 17, 1999, January 16-17, 2001, and August 22-23, 2001. The ACRS discussed this matter with EPRI and the NRC staff during its 485th meeting on September 5-7, 2001, and completed its review of this matter during the 486th meeting of the ACRS on October 4-6, 2001. The results of the ACRS review was provided to Dr. William D. Travers in a letter dated October 23, 2001. The ACRS recommended that the 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. The ACRS review focused on the prototypicality of the EPRI experiments, the adequacy of the scaling model, and the appropriateness of the condensation and air-release models. ACRS member Dr. Graham Wallis provided a detailed discussion of these areas in an attachment to the October 23, 2001, letter. The ACRS concluded that EPRI's conceptual model was oversimplified, and that it was not clear how the model could be applied to plant-specific scenarios and configurations. However, recognizing that the waterhammer concerns discussed in GL 96-06 may not be risk significant, the ACRS indicated that it would be supportive of risk-informed approaches for addressing the GL 96-06 waterhammer issue.

2.4 EPRI Response to the ACRS Comments and Discussion of Risk Considerations

By letter dated December 3, 2001, the NRC staff requested that EPRI respond to the comments contained in the October 23, 2001, ACRS letter, including an assessment of the risk significance of the GL 96-06 waterhammer issue. EPRI provided its response to the staff's request in a letter dated February 1, 2002.

2.4.1 Response to ACRS Comments

With respect to the ACRS comments that were raised, EPRI pointed out that many of the ACRS comments [discussed in the attachment to the ACRS letter] were focused on the complexities associated with plant-specific evaluation of heat transfer in the containment air coolers and the hydraulics of the voiding and draining process. EPRI clarified that details such as these were not included within the scope of the EPRI initiative. As discussed in the EPRI report (e.g., Figure 2-1 and Section 2.2 of the UM), these complexities must be evaluated on a plant-specific basis. Once these analyses have been completed, the EPRI report provides a methodology for determining waterhammer loads at the time of final refill and column closure following pump restart. The EPRI report also provides methods for calculating loads in pipe supports and stresses in piping.

2.4.2 Risk Considerations

Recognizing that the ACRS does not fully agree with the methodology being proposed for evaluating the GL 96-06 waterhammer issue, EPRI felt that consideration of the risk associated with the events and the results that would be achieved if the EPRI methodology is used may be an important factor. In responding to the staff's December 3, 2001 letter, EPRI proposed that if the EPRI methodology does not significantly increase the risk of unacceptable plant performance nor lead to an unacceptable risk to the plant, use of the EPRI methodology can be done safely without compromising the integrity or safety of the systems being evaluated. In order to assess the risk to the plant of using the proposed EPRI methodology, a review of the progression of events that could lead to an unacceptable condition was performed. The "unacceptable condition" was defined as a breach of the containment air cooler cooling water system boundary (i.e., pipe failure). EPRI outlined the following progression of events:

- Occurrence of a LOCA or MSLB

  From NUREG/CR-5750, the mean frequency of occurrence of a large-break LOCA for a PWR is listed as 5E(-6) per year, a medium LOCA is 4E(-5) per year, and a MSLB inside containment is 1E(-3) per year.

- Occurrence of a LOOP following a LOCA or MSLB

  NUREG/CR-6538 and subsequent NRC work indicates that the dependent probability of a LOOP event following a LOCA at a PWR is approximately 1.4E(-2) per demand.

- Occurrence of Simultaneous LOCA/LOOP Event

  The required design-basis consideration is for the simultaneous occurrences of a LOCA or MSLB and a LOOP. Combining the probabilities for occurrence of a LOCA or MSLB with the probability of a dependent LOOP yields a probability of a simultaneous occurrence of a LOCA or MSLB with LOOP of 1.4E(-5), or about 1E(-5) per year. With best estimate probabilities, this event likelihood of occurrence could be expected to be even lower.

- Void Formation

  Given a LOCA/LOOP event, a void will form in open loop cooling water systems with certainty. In a closed loop plant, void formation will depend on the specific plant characteristics and a void may or may not form.

- Pump Restart

  The cooling water pumps will restart with certainty and the velocity in the pipe, immediately prior to closing the void, will be defined by the pressure in the void, the piping geometry, and the pump characteristics. This velocity will not be higher than the rate at which the pumps, once restarted, can pump water. This is a plant specific analysis that can be conservatively performed.

- Column Closure

  The water columns will refill the void and the velocity at closure cannot be larger than the largest calculated differential velocity for the upstream and downstream water columns.

- Maximum Waterhammer Pressure

  An upper bound on the waterhammer pressure can be calculated by the Joukowski relationship with the uncushioned closure velocity that corresponds to the pipe in which the closure will occur. Based on the tests that were performed by EPRI and others, it is unlikely that the Joukowski pressure will be attained due to the variations that can exist in void distribution just prior to final void closure. For example, at a velocity range of interest, the EPRI test data indicates that the maximum pressure attained was around 400 psig and the minimum was around 200 psig, with the theoretical maximum pressure (Joukowski) calculated to be around 775 psig. The EPRI methodology uses the highest calculated (Joukowski) pressure as the starting point for calculating the cushioned velocity, even though it is very likely that the pressure will be lower as seen in the tests that were conducted.

- Cushioned Waterhammer

  The EPRI methodology predicts (for open loop cooling water systems) a cushioned velocity that is on the order of 30 percent to 40 percent less than the maximum velocity that could be attained due to the cushioning effects of gas and steam in the void just prior to column closure. The cushioned velocity is expected to be on the order of 10 percent to 15 percent less for closed loop cooling water systems because less boiling and air release is expected to occur (i.e., closed-loop systems do not drain and therefore the saturation pressure should be higher than for open-loop systems). Consequently, the peak pressure from an actual waterhammer could be on the order of 30 percent to 40 percent less than what would be predicted by the Joukowski equation (10 percent to 15 percent less for closed-loop systems). The risk impact of the potentially higher (Joukowski) stress was considered in two ways.

  a. EPRI pointed out that the occurrence of waterhammer following a LOOP event has occurred many times, either during LOOP tests or following actual LOOP events that have occurred. The total number of LOOP-only events was estimated to be on the order of at least several hundred based on a review of available plant data. These occurrences have all been in open loop cooling water systems (closed loop systems do not drain during a LOOP), and are more severe than a waterhammer following a LOCA/LOOP event would be because the final column closure is not cushioned during a LOOP-only event. No piping failures have occurred in any of these events, which would indicate that the probability of pipe failure is on the order of 1E(-2) or less.

b. Alternatively, EPRI estimated the probability of piping failure if the ASME Code limits were exceeded by 40 percent. For this evaluation, EPRI assumed that the ASME Code stresses in the pipe are at the faulted condition limit when the cushioned waterhammer occurred (i.e., includes the waterhammer stresses). The probability of A106, Gr. B piping failure was calculated based on an assumed stress distribution in the pipe that was 40 percent greater than the ASME Code faulted allowable. Based on the actual margins that are available in the ASME Code, EPRI estimated that the probability of pipe failure is on the order of 1E(-4) or less.

For the purposes of this risk assessment, EPRI used the more conservative value of 1E(-2) for the probability of pipe failure if the cushioned waterhammer pressure was exceeded.

Combining the probabilities from the event progression, 1E(-5) per year for the initiating event frequency and 1E(-2) for pipe failure, yields a probability of an unacceptable event that is on the order of 1E(-7) per year. EPRI concluded that this is below the threshold for significant risk to the plant.

2.5 Additional Considerations

Following preliminary review of the final version of the EPRI report, dated December 2000, the NRC staff requested additional information regarding the proposed methodology. EPRI responded to the staff's request in letters dated August 9, and September 17, 2001. In addition to minor editorial comments, EPRI provided the following additional information and clarifications:

- A comparison was provided of pressure rise times between the test configuration that included air in the steam bubble versus test configurations that did not include air in the steam bubble. EPRI concluded that the pressure rise times are similar when the closure velocities are similar.

- Additional air release testing was performed using test apparatus that more closely modeled the situation that would exist in the containment air coolers. The EPRI report was revised to describe the testing that was done and to reflect the test results that were obtained from these more recent tests.

- Additional information was provided concerning the characterization and validation of the pressure pulse as a trapezoid. The trapezoidal model was developed to reflect fundamental theory, and to capture the pulse magnitude, rise time, and duration in order to simplify the transient pressure response into a set of defined pressure-time (P-T) points for use in a structural calculation. The effectiveness of the trapezoidal model was evaluated by comparing the response of an ANSYS model with loading from the idealized trapezoids and loading with actual P-T histories to the measured force response from the tests. Most of the plotted points fell above the "predicted = measured" line (a favorable comparison), and those points that fell below the line were located close to the line and represented a small percentage of the total data. The comparison indicated that on average, the ANSYS analysis using the actual P-T loading was conservative by about 15 percent, while the ANSYS analysis using the idealized trapezoidal loading was conservative by about 30 percent. EPRI also explained that the pulse duration will change as the pressure magnitude is cushioned to satisfy conservation of momentum, and that there was no recommendation to increase the pressure pulse duration beyond this value. More detailed information about the pressure pulse can be found in Sections 9.2, 9.3, and 13.5 of the TBR.

EPRI also evaluated what effect increasing the assumed structural damping value from 0.1 percent (TBR Figures 13-7 and 13-8) to 2 percent would have on the comparison of calculated versus measured pipe support loads. The resulting dynamic load factors decreased by 1.9 percent to 7.7 percent with the increased damping. Based on a translation of these results to those presented in Figures 13-7 and 13-8 of the TBR, EPRI concluded that: a) the increased damping would not significantly impact the comparison that was made between calculated and measured forces, and b) the conclusion that the trapezoidal representation of the pulse is an appropriate modeling method remains valid.

- Additional justification was provided for using a single pressure pulse instead of multiple pulses. In essence, EPRI concluded that any subsequent pressure pulse after the initial pressure rise is caused by reflected waves passing through the system, and the reflected waves will be significantly smaller in magnitude than the initial pulse.

- EPRI clarified that the simplified method of attenuation described in TBR Section 12.4 is not provided for generic plant analysis, but it is simply used to show that the attenuation (which is cumulative) will quickly surpass any potential amplification due to fluid-structure interaction (FSI). The analysis is used as a basis for the recommendation that the potential amplification from FSI can be conservatively ignored.

- EPRI clarified that credit for air release will not be taken unless the cooling water is exposed to temperatures that are above the boiling point corresponding to the pressure. As an additional conservative measure, the TBR and UM will eliminate credit for air release unless the temperature of the containment air cooler tubes is at least 10 °F above the temperature at which the cooling water will boil.

- In extrapolating the 15 psig CIWH test data to include system pressures up to 20 psig, EPRI investigated the CIWH occurrence mechanism in detail and identified no aspects that would result in a change of the behavior of CIWH in a system that has a pressure that is only 5 psia higher than what was tested. Therefore EPRI believes that the conclusions stated in the TBR for CIWH are valid for system pressures up to 20 psig.

3 EVALUATION

As mentioned in Section 1 of this evaluation, the NRC staff was an active participant in the EPRI initiative, monitoring and critiquing the work that was being performed by EPRI. We considered the PIRT process to be appropriate and adequate for defining the scope of work that needed to be done, and found the outcome of the PIRT evaluation to be acceptable. Because waterhammer is a very complex phenomenon, we also considered EPRI's use of an expert panel to be appropriate and necessary in assuring the completeness and adequacy of the methodology that was being developed. We found the individual panel members to be recognized authorities in their areas of expertise, and well qualified to review the work that was being performed. On several occasions, we observed the work being performed by EPRI, including working level meetings, interaction with the expert panel members, and tests that were being performed. The work appeared to be well planned and orchestrated in all respects, and the individuals performing the work appeared to be very knowledgeable and conscientious in the conduct of these activities.

We have reviewed the information contained in the UM and TBR, as supplemented by letters dated July 10, August 9, September 17, 2001, and February 1, 2002, including interpretation and correlation of the test results, analytical modeling, assumptions, and conclusions that were reached. During our review of the EPRI report to determine whether or not use of the proposed methodology is adequately justified for use in evaluating GL 96-06 waterhammer events, we found the following areas to be of primary interest:

- CIWH Versus CCWH

  Based on the tests that were conducted and review of CIWH information that was available, EPRI determined that CIWH events in the low-pressure cooling water systems for the containment fan coolers will be bounded by CCWH for most situations. Section 4.2.1 of the UM lists the specific criteria that systems must satisfy in order to apply this conclusion. We agree that the information contained in Section 7 of the TBR adequately supports this conclusion and consequently, we focused our evaluation on the proposed methodology for analyzing CCWH events.

- Air Release Model

  During the event scenarios referred to in GL 96-06, air will most certainly be released from solution for any situation that involves the possibility of waterhammer. The testing that was done to determine the amount of air released from the horizontal tubes during boiling most closely modeled the cooling water system for containment air coolers that are located at the high points of the cooling water system, and would bound the case where the containment air coolers are located at a low point in the system. This is because more air would actually be released than what is assumed from the water that is draining back into the containment air cooler tubes from the vertical header and boils during the event. Another shortcoming of the test was that the water in the horizontal tubes mixed with the water in the vertical header, making it difficult to determine how much air was actually released from the water in the vertical header. In order to establish conservative results, EPRI chose the lower bound value of the air released from the water that spilled over into the overflow tube. As an additional measure of conservatism, the EPRI report will only allow credit for air cushioning if the water in the containment air coolers is exposed to containment air cooler tube temperatures that exceed the boiling point for the fluid conditions by at least 10 °F. While the accuracy of the test results can be questioned, we believe that the proposed methodology with respect to air release is conservative and acceptable for the specific application that is described in the UM.

- Scaling Considerations

Section 7.6 of the TBR discusses the "segment scaling" approach that was used for the CIWH experiments, identifying and evaluating scale distortion phenomena. EPRI determined that both the waterhammer pressure rise and the absolute impact pressure in the smaller experimental pipes would be higher than expected in larger pipes, thus yielding conservative results. Section 8.3.6 of the TBR uses analytical methods to show that pipe diameter is not a relevant factor for scaling the CCWH test results. On the basis of these scaling evaluations, EPRI concluded that the proposed analytical method would be applicable to the larger-scale plant cooling water systems for the containment air coolers. EPRI's approach for addressing scaling considerations appeared to be reasonable for this particular application.

- Steam Condensation

Sections 8 and 9 of the TBR discuss the MOC and RBM analytical methods, and describe how they were validated using the test results. The pressurization of the steam bubble due to the presence of air and the condensing of steam as the steam bubble is collapsing tend to cushion the waterhammer pressure pulse by slowing the closure velocity of the water column. Based on the test data and arbitrarily assuming the fluid surface area to be twice the available flow area, EPRI established a conservative condensing heat transfer coefficient for use in determining the steam condensation rate. The test data is well represented by the proposed methodology, and we believe that EPRI's approach is reasonable for this specific application.

- Pulse Shape

In addition to lowering the peak waterhammer pressure, the steam and air cushioning effect results in a longer pressure pulse rise time in order to conserve momentum. Structural loading of the pipes and pipe supports is dependent on the slope of the rise, and a longer rise time results in reduced structural loading. For analytical purposes, EPRI modeled the waterhammer pressure pulse as a trapezoid. Section 9 of the TBR discusses the waterhammer pressure pulse rise times that were observed during the tests that were run, and establishes a method for predicting rise time that is inversely proportional to the water column impact closure velocity.

The selection of a rise time defines the shape of the trapezoidal force-time pulse to be applied to a point in the piping system. The staff identified issues regarding the selection of rise times and the use of a single pulse to represent the loading function on the piping structure. While the rise time does exhibit an inverse relationship with closure velocity, the values selected to represent the test data do not appear to sufficiently bound all cases when considering the steepest part of the pressurization. Shorter rise times could result in additional structural response. In addition, the loading function for a waterhammer event is characteristically a series of pulses which eventually attenuate after several cycles. The consideration of additional pulses could also result in additional structural response due to additional energy input.

To examine the effects of both varying rise times and additional pulses, the staff performed several parametric calculations using a single degree-of-freedom spring and mass model. Using this technique, it may be observed that for a repeating pulse input force at resonant or harmonic frequencies, significant additional energy may be input into the structure. However, it is also noted that actual waterhammer forcing functions have pulses which are not exactly repeating in magnitude or period, as evidenced by viewing plots of actual pressure histories. Therefore, it is expected that additional pulses can result in additional structural response, but not as much as the single degree-of-freedom model would predict. A comparison of the proposed analysis method with actual test data indicates that only a few of the analyzed loads did not bound the test loads, and that for most cases, the analysis method over predicted the actual test loads. Therefore, considering the margin available in the piping system when using the allowable stress for the piping material, the proposed rise times and single pulse methodology are acceptable for defining the forces on the piping due to a waterhammer event in the containment fan cooler service water system.

The work that was performed by EPRI was well focused and has substantially improved our knowledge and understanding of the waterhammer phenomenon, especially in low pressure fluid systems. We agree that the proposed EPRI methodology is generally well supported by the TBR and representative of the test results that were obtained. We also consider the guidance provided in the UM, including limitations and restrictions, to be an acceptable representation of the information developed in the TBR, and that use of the UM will assure proper application of the proposed methodology.

3.1 Risk Perspective

While the ACRS did not fully agree with the technical adequacy of the methodology that was proposed by EPRI, they indicated that a risk-informed approach could be a viable option. We would also agree that a risk-informed approach could be used to further justify the proposed EPRI methodology, given the convincing nature of the work that was done. In response to our request, EPRI provided its perspective on the risk associated with the GL 96-06 waterhammer issue. As discussed in Section 2 of this evaluation, EPRI concluded that the likelihood of an unacceptable GL 96-06 waterhammer event is on the order of 1E(-7) or less. EPRI's conclusion is based in part on plant experience with LOOP-only events. EPRI reasoned that LOOP-only events tend to bound the events of concern that are discussed in GL 96-06 (LOCA/MSLB with LOOP) for open-loop cooling water systems. Many LOOP-only events have occurred and no system failures have resulted. Based on this, EPRI concluded that the likelihood of pipe failure given a waterhammer event is on the order of 1E(-2) or less. We agree with EPRI's approach for evaluating risk, and concur with the risk perspective that was provided.

3.2 NRC Acceptance

A substantial amount of time and resources have been invested by EPRI and the NRC staff on completing this initiative. In order to address the concerns that were expressed by the ACRS, much more time and expenditure of resources would be required. However, given the following considerations, we do not believe that any further effort is warranted:

- The proposed methodology has been reviewed and endorsed by notable industry experts who are recognized authorities in their areas of expertise.

- The proposed methodology is consistent with the guidance provided in NUREG/CR-5220 (Ref. 1).

- Although uncertainties remain, the work that was completed by EPRI is very convincing and well supported by a substantial amount of test data and analytical information.

- The probability that a rupture will occur in the containment air cooler cooling water systems due to the waterhammer scenarios discussed in GL 96-06 is considered to be very low; on the order of 1E(-7) or less.

Therefore, we agree that adequate justification exists for using the proposed methodology in EPRI Report TR-113594, Volumes 1 and 2, for evaluating GL 96-06 waterhammer concerns and in responding to plant-specific RAIs that were issued. We agree that the limitations and restrictions established by EPRI, including clarifications provided by the supplementary information that was submitted, are sufficient to assure proper application of the proposed methodology. This includes the position that any potential pressure amplifications due to fluid/structure interaction (FSI) can be conservatively ignored, and that no pressure reductions due to FSI attenuations will be assumed (see EPRI letter dated August 9, 2001). Our approval of the proposed methodology is limited to use by licensees for addressing the GL 96-06 waterhammer issue, and our approval does not extend to any other regulatory application.

3.3 Licensee Responses to GL 96-06

Licensees who choose to use the methodology in TR-113594, Volumes 1 and 2, for addressing the GL 96-06 waterhammer issue, may do so by supplementing their response to include:

- Certification that the EPRI methodology, including clarifications, was properly applied, and that plant-specific risk considerations are consistent with the risk perspective that was provided in the EPRI letter dated February 1, 2002. If the uncushioned velocity and pressure are more than 40 percent greater than the cushioned values, also certify that the pipe failure probability assumption remains bounding. Any questions that were asked previously by the staff with respect to the GL 96-06 waterhammer issue should be disregarded.

- The additional information that was requested in RAIs that were issued by the NRC staff with respect to the GL 96-06 two-phase flow issue (as applicable).

- A brief summary of the results and conclusions that were reached with respect to the waterhammer and two-phase flow issues, including problems that were identified along with corrective actions that were taken. If corrective actions are planned but have not been completed, confirm that the affected systems remain operable and provide the schedule for completing any remaining corrective actions.

Licensees are reminded that their evaluations and responses to address the GL 96-06 issues may be subject to future NRC audit and inspection activities.

4 CONCLUSIONS

As discussed in Section 3 of this evaluation, we consider the use of EPRI Report TR-113594, Volumes 1 and 2, to be acceptable for performing evaluations to address the GL 96-06 waterhammer concerns. Licensees who choose to use this report may respond to the plant-specific RAIs that were issued as discussed in Section 3.3 of this evaluation. NRC approval of the EPRI methodology is limited to use by licensees for addressing the GL 96-06 waterhammer issue, and does not extend to any other regulatory applications.

5 REFERENCES

1. M. G. Izenson, P. H. Rothe and, G. B. Wallis, "Diagnosis of Condensation-Induced Waterhammer," NUREG/CR-5220 Vol. 1, October 1988.

2. P. Griffith, "Screening Reactor Steam/Water Piping Systems for Water Hammer," NUREG/CR-6519, Massachusetts Institute of Technology, September 1997.

3. W. Zielke, H-D Perko and A. Keller, "Gas Release in Transient Pipe Flow," Proc. 6th International Conference on Pressure Surges, BHRA, Cambridge, England October 4-6, 1989.

4. E. B. Wylie and V. L. Streeter, "Fluid Transients in Systems," Prentice-Hall, Inc., 1993.

C
EPRI CORRESPONDENCE TO NRC

EPRI Correspondence to NRC:

July 10, 2001	"Resolution of Generic Letter GL96-06 Waterhammer Issues" EPRI Report 113594 - V1 & V2, Revised Sections
August 9, 2001	Response to Questions on Generic Letter 96-06
September 17, 2001	Additional Responses to Questions on Generic Letter 96-06
February 1, 2002	Response to ACRS Comments (letter dated 10/23/01) on the EPRI Report on Resolution of NRC GL96-06 Waterhammer Issues

July 10, 2001

Document Control Desk
U.S. Nuclear Regulatory Commission
11555 Rockville Pike
Rockville, MD 20852

Attention: Mr. Jim Tatum

Subject: "Resolution of Generic Letter 96-06 Waterhammer Issues", EPRI Report TR-113594-V1 & V2, Revised Sections

Enclosed are twelve (12) copies of the revised sections of the document "Resolution of Generic Letter 96-06 Waterhammer Issues", EPRI Interim Report TR-113594--V1 & V2, December 2000. Specifically, Section 5.2.2 and 5.2.3 of Volume 1 (User's Manual) and Sections 6 and 8 of Volume 2 (the Technical Basis Report) are enclosed. The other sections of the report are essentially unchanged from the earlier transmittal and will only have editorial changes made prior to final submittal.

This information is being submitted as a means of exchanging information with the NRC for the purpose of supporting industry resolution of GL 96-06 Waterhammer Issues. The specific modifications to the sections noted were made to address issues raised by the ACRS Thermal Hydraulic Subcommittee at our meeting on January 16, 2001. The questions specifically raised were:

1. Limit of test apparatus for determination of air release fraction.
2. Determination of the "h" in the "hA" term (this was backed out of the test data; perhaps a Table listing the various values of "h" for the tests would be useful).
3. Scale-up of the test data (the suggestion here was to conduct some sort of sensitivity study).

The report sections that are modified include the results of additional air-release testing that has been performed to determine the amount of air released during the GL96-06 transient. The test methods and results are documented in the modification to the Volume 2 Section 6. The modified method to calculate air release for a specific plant application is provided in Volume 1 Section 5.2.3.

The "h" values for the test data are documented in the revised Volume 2 Section 8. A sensitivity study of both the Method of Characteristics (MOC) and Rigid Body Model (RBM) to the "h" values was performed and is documented in the revised Volume 2 Section 8. Additional work also was performed on scaling for pipe diameter (see Section 8.3.6).

Please note that the enclosed document contains proprietary information. Therefore, a letter requesting the report be withheld from public disclosure and an affidavit describing the basis for withholding this information is provided as Attachment 1.

We will plan to meet with the NRR Staff and the ACRS on August 23, 2001 to discuss the material that is attached. If you have any questions on the enclosed document or the general subject it addresses, please call me at 919-546-7959 or Avtar Singh at 650-855-2384.

Sincerely,

Avtar Singh for V Wagoner

Vaughn Wagoner
Carolina Power & Light Company
Chairman, EPRI Waterhammer Project Utility Advisory Group

E L E C T R I F Y   T H E   W O R L D

EPRI

August 9, 2001

James Tatum
U.S. Nuclear Regulatory Commission
11555 Rockville Pike, M/S O-11A11
Rockville, MD 20852

Attention: Mr. Jim Tatum

SUBJECT: Response to Questions on Generic Letter 96-06

Enclosed are responses to questions raised on the document "Resolution of Generic Letter 96-06 Waterhammer Issues", EPRI Interim Report TR-113594--V1 & V2, December 2000. We have previously transmitted revisions to Section 5.2.2 and 5.2.3 of Volume 1 (User's Manual) and Sections 6 and 8 of Volume 2 (the Technical Basis Report). The other sections of the report are essentially unchanged from the earlier transmittal and will only have editorial changes made prior to final submittal. The attachment to this letter includes responses to specific questions raised by the NRC. This information will be addressed as applicable in the final revision to the Technical Basis Report.

The enclosed document does not contain any proprietary information.

If you have any questions on the enclosed information or the general subject it addresses, please call me at 919-546-7959 or Avtar Singh at 650-855-2384.

Sincerely,

Avtar Singh for

Vaughn Wagoner
Carolina Power & Light Company
Chairman, EPRI Waterhammer Project Utility Advisory Group

Attachment

Questions from Walt Jensen and Jim Tatum (NRR staff):

1. **The relationship of pressure rise time to impact velocity is given only for test configuration No. 1 which did not include air in the steam void. Please provide a comparison of the pressure rise time relationship with the data from test configuration 2 which did include air.**

   Response:

   The individual rise times for the Configuration 2a and 2b tests have been calculated. This data is provided in Figure 10-8, attached. This figure also includes the Configuration 1 test results. The comparison shows that the rise times for Configuration 2a and 2b tests are similar to the rise times for Configuration 1 when the closure velocities are similar.

2. **Please provide figure 10-9 which was missing from the "Technical Basis Report".**

   Response:

   A copy of Figure 10-9 is attached.

3. **We understand that burst tests have been performed for representative fan cooler tubing and piping which showed failure only at very elevated pressures. Please provide documentation for these tests.**

   Response:

   The burst test data discussed during the January 16, 2001 meeting is industry data that had been previously developed by EPRI. A copy of EPRI report TR-108812, dated December, 1997, describing the burst test program has been provided.

4. **The NRC staff shares the same concerns as the ACRS Subcommittee on T/H's regarding noncondensable gas generation during system draining and steam condensation during column closure. In responding to the ACRS T/H subcommittee on this issue, please also address configuration differences that exist between the test apparatus and the actual plant. For example, the heat exchanger tubes in the FCUs are generally horizontal, while the test apparatus modeled a vertical configuration. It would seem that there could be significant differences in the test results if steam bubbles are rising through a vertical tube (as in the test apparatus) vs. the plant configuration where the steam bubbles form in the tube and must expand to a vertical header that is usually at the high point (but could also be at the low point) of the system. It is not clear how the test results apply to the actual plant configuration.**

Attachment

Response:

Additional testing has been performed to determine the amount of air released during the transient. The test results and a modified approach to evaluating air release are documented in the TBR Section 6 and User's Manual (UM) Section 5.2.3, respectively.

Section 6.3.2 of the revised TBR described the test configuration and compared the test configuration and conditions to the prototypical configurations and conditions in nuclear plant applications. Specifically, the additional air release testing utilized a horizontal tube attached to a vertical header. Tests were run both with the header full and with the header empty to simulate a variety of plant conditions. These tests were more prototypical of actual plant geometry and conditions.

Questions from Gary Hammer (NRR staff):

1. **The rigid body model involves defining the waterhammer pulse as a trapezoid function having a recommended rise time. However, the recommended method for choosing rise times does not appear to be conservative when considering the steepest part of the pressure-time data plots. Also, the pulse duration is recommended to be lengthened by a factor to preserve the area under the trapezoid shape. However, this also does not appear to be conservative since it could result in less structural response than for the actually expected duration.**

   Response:

   The characterization of the pressure pulse as a trapezoid was selected to simplify the complexity of the actual pressure pulses that were seen in the tests. The trapezoidal model used to characterize the pressure pulse was developed to reflect fundamental theory, capture the pulse magnitude, rise time, and duration to simplify the transient pressure response into a set of defined pressure time (P-T) points for use in a structural calculation.

   The selection of the pulse was described in Section 9.2 and 9.3 of the TBR. The adequacy of the trapezoidal representation was evaluated and the results were reported in Section 13.5 of the TBR.

   The effectiveness of the trapezoid model was tested by comparing the response of an ANSYS model with loading from the idealized trapezoids and loading with actual pressure-time histories to the measured force response from the tests. Support loads at three locations were measured in the tests. A set of 44 test measured pressure traces from the tests was used as the "actual" pressure-time input. The test traces were accurately input to ANSYS in detail. These pulses were also characterized as trapezoids using the methods recommended in the User's Manual and then used to load the ANSYS model. The results of these two load sets (idealized trapezoid versus the actual pressure history) compared to the support forces measured in the tests is provided below (Figure 13-7 of

Attachment

the TBR). Most data fell above the "predicted = measured" line at 45° in the figure. The points that fell below the 45° line were a small percentage of the total and these points were located close to the line.

Figure 13-8 in the TBR, also provided below, showed the results for the trapezoidal characterization of the pulses for all tests analyzed using the same ANSYS model. These force responses are plotted versus the measured force data for all three restraints (F1, F2, and F3). The 45° dashed line (predicted = measured) represents exact matching of the measured response. This plot further demonstrates the accuracy of the trapezoidal modeling technique as a means of predicting real support forces.

These two comparisons show that both the actual pressure trace and the trapezoidal representation provide accurate methods to capture the response of the structure when compared to the test data. Figures 13-7 and 13-8 further show that the "curve fit" for the trapezoidal pulses provide a prediction of a higher support load than the actual pressure pulse. This indicates that on average, the trapezoidal pressure pulse is more conservative. The statistical nature of the testing, particularly for events like waterhammer, does provide a small number of calculated loads that are lower than the test results. The number of calculated points that fall below the test data is considered to be typical of what would be expected for this number of tests for a phenomena that has as much scatter as waterhammer testing.

On average, the analysis with the actual pressure time loading is conservative versus the test data by approximately 15% (percentage calculated at a load of 1,000 pounds) and the analysis with the trapezoidal pressure time loading is conservative versus the test data by approximately 30%. The trends from Figure 13-8 are the same.

The margins that exist in the calculation of the pressure magnitude and in the design and qualification of the supports is considered adequate with this trapezoidal representation to assure that a conservative basis for qualification of supports is provided. The trapezoidal representation gives higher loads than the actual pressure time curve.

The question also asked about the "lengthening" of the pressure pulse. The pulse duration will change as the pressure magnitude is cushioned to satisfy conservation of momentum. This accurately represents how the pressure pulse changes with cushioning. This is discussed in Section 9.2.2 of the Technical Basis Report. Calculation of the pulse duration is provided in the User's Manual, Section 5.3.5. The pressure pulse duration to be used is calculated based on the time of reflection. There is no recommendation to increase pressure pulse duration beyond this value.

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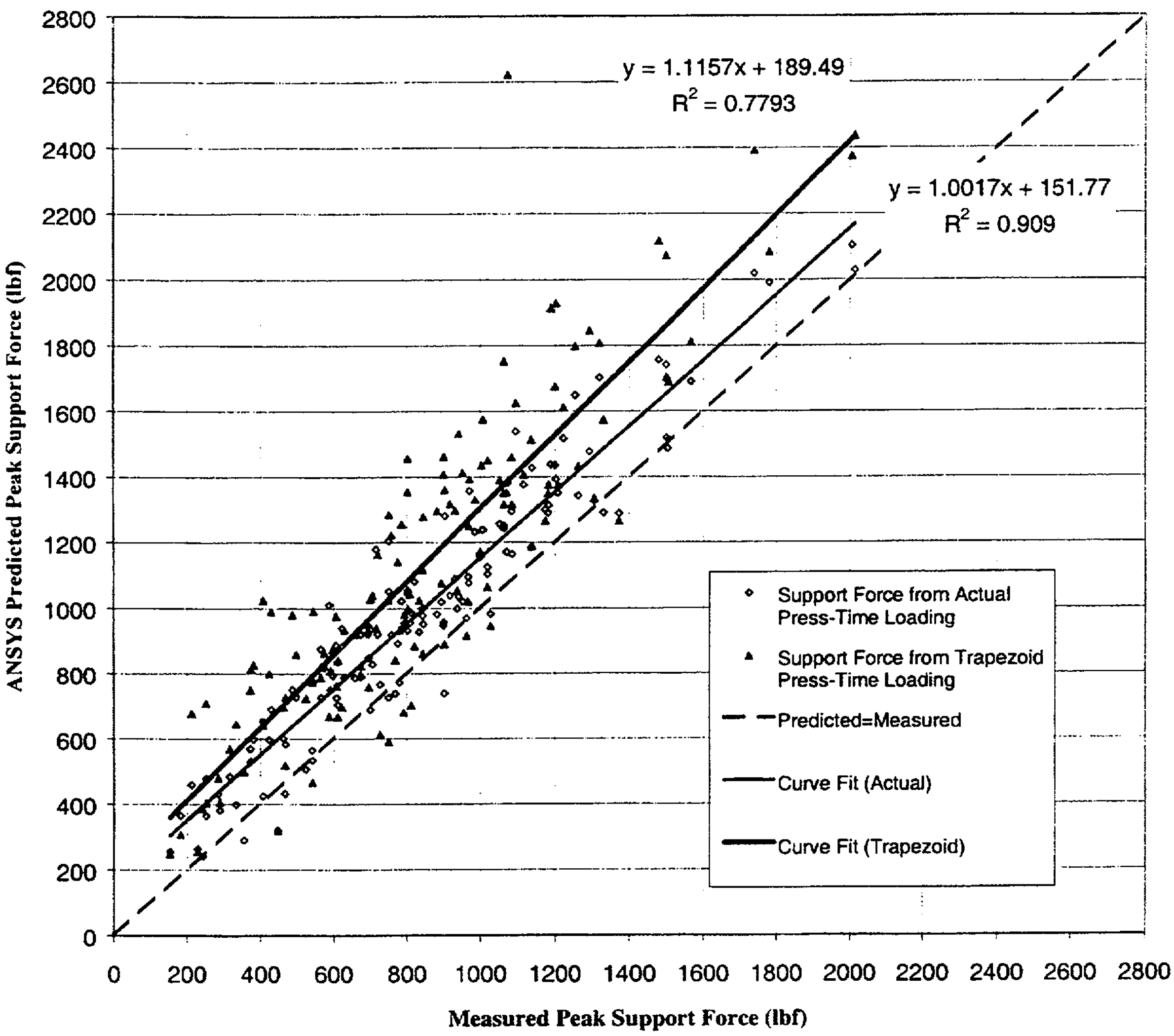

Figure 13-7: Trapezoid Characterization/Actual Data Comparison (44 Tests)

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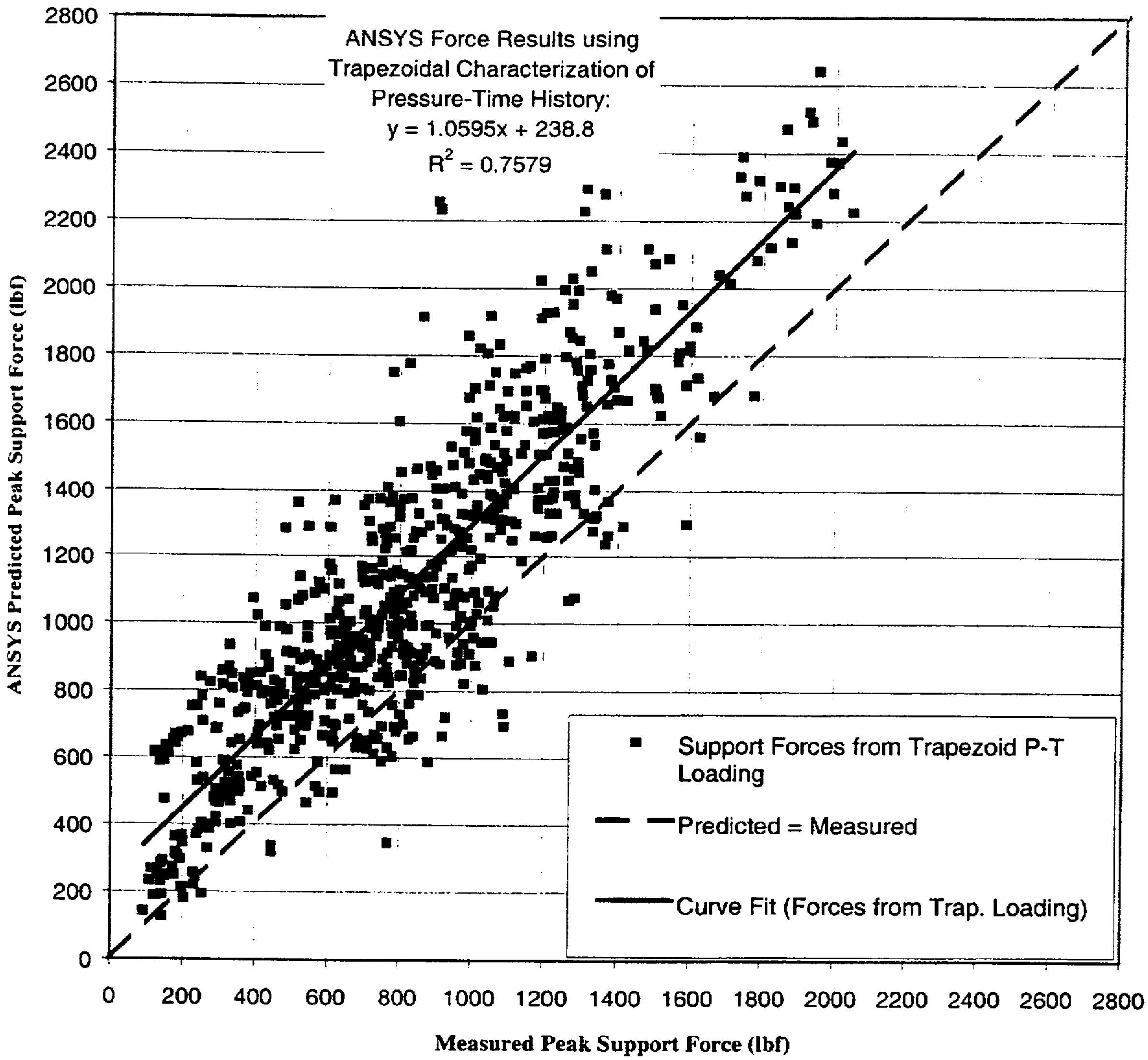


Figure 13-8: Trapezoid Characterization (All Tests)

2. **The report recommends using only one waterhammer pulse in evaluating system piping. However, waterhammer pressure loads are composed of several reversing cyclic pulses (Examples are shown in Figures 5-3 and 7-3). Figure 13-8 indicates that for the test configuration, the use of a single trapezoidal pulse is conservative in most cases evaluated. However, there are a few cases shown where this method is not conservative. Also, the structural and forcing function frequencies for plant piping configurations will differ from the tests. Therefore, a longer pressure history involving several cycles**

Attachment

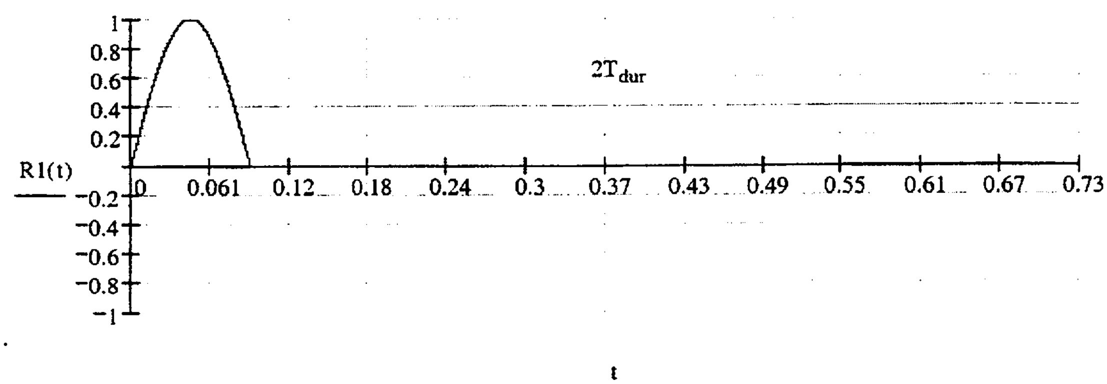
should be included in analysis of plant piping systems since this could result in additional energy being added to the structures.

<u>Response:</u>

The column closure event is essentially a single pulse phenomenon. Any subsequent pressure pulse after the initial pressure rise is caused by reflected waves passing through the system. The reflected waves will be significantly smaller in magnitude than the initial pulse.

To investigate the accuracy of using a single pulse, a single degree of freedom model was selected as typical of a single segment of piping that experiences an axial load caused by a passing pressure wave. This model was loaded with a repeating, decaying pressure pulse that occurs at the <u>precise</u> natural frequency of the structure. This was compared to a single pressure pulse load of the same initial duration. The degree of decay from the first peak to the second peak was approximately 75% as would be expected with a reflected wave and as was seen in Figure 5-3 of the TBR. The two loads are provided in the following figures.

Single Pulse Loading:



Repeated Pulse Loading:

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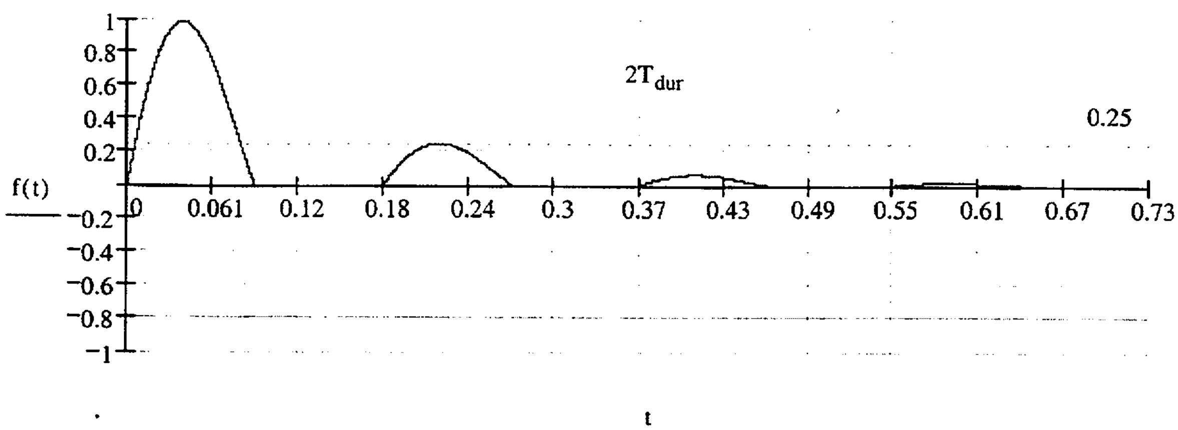

Using 2% damping, the resulting displacements for the multiple loading are within 10% of that produced by a single load. This was for the case where the loading frequency was precisely equal to the natural frequency of the system.

The response of a complex system is dependent not on the response of a single axial segment, but on the combined response of many segments to a passing pressure wave. Further, the load in any individual support is a combination of loads from many parts of a piping system. The likelihood of any individual segment having the precise natural frequency of repeated loading is very low. The likelihood of multiple segments contributing to the loading of a support and having the precise natural frequency as a repeated loading is much lower. Even a small difference between the natural frequency and the driving frequency will dramatically change the response to multiple loading. In other words, the repeated smaller loads do not have the potential to significantly affect the structural response in actual systems.

As shown for the trapezoidal load, the margins that exist in the calculation of the pressure magnitude and in the design and qualification of the supports is considered adequate to allow a single pulse to be used to assure that a conservative basis for qualification of supports is provided.

3. **The report outlines a simple method of incorporating Poisson coupling and junction coupling type fluid-structure interaction based on a study of a very simple configuration. There are significant uncertainties involved in making such predictions, and if fluid-structure interaction is to be considered in attenuating the waterhammer loads, it should be based on a more detailed plant-specific analysis.**

Response:

The analytical evaluation of potential pulse amplification by fluid structural interaction (FSI) is based on the detailed methods defined by Wood as described in reference 33 of the TBR. It was further investigated in references 34 through 38 as described in the TBR.

Attachment

The simplified method of attenuation described in TBR Section 12.4 is not provided for generic plant analysis. It is used to show that the attenuation, being cumulative, will quickly surpass any potential amplification due to FSI. This analysis is used as a basis for the recommendation that the potential amplification from FSI can be conservatively ignored.

At the discretion of the individual licensee(s), fluid structure interaction may be used only if both attenuation and amplification are employed. The degree to which attenuation or amplification dominates fluid structure interaction will be a function of the stiffness of a piping system and its supporting elements. These are plant specific elements, and thus should be addressed in the plant specific responses to the generic letter.

4. **The report does not indicate the structural damping value used in the comparison of analyzed loads vs measured loads. This information needs to be provided as part of the basis for the comparison.**

<u>Response:</u>

The damping used in the analysis for comparison to test data was 0.1% of critical damping. Specific damping values to be used would be plant-specific and would be in accordance with the plant's licensing documents.

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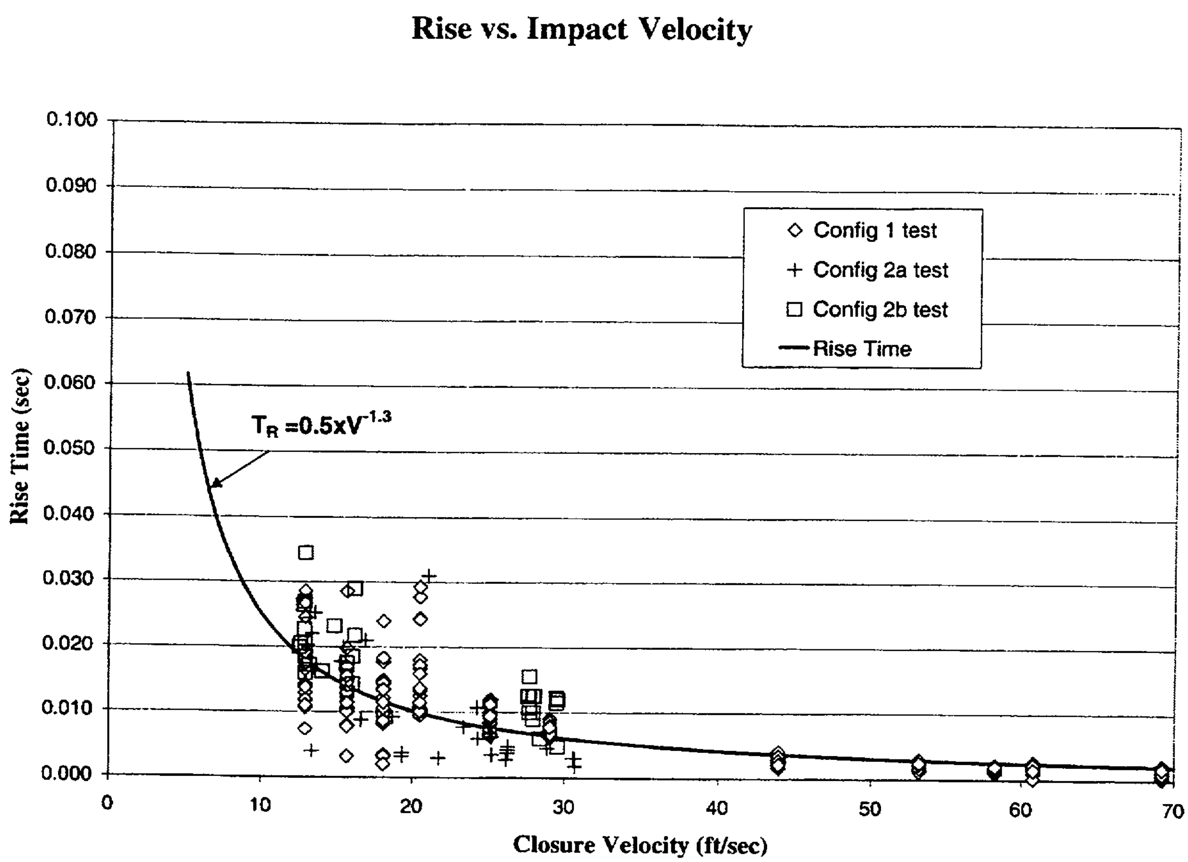

Figure 10-8: Rise Time vs. Impact Velocity – Configuration 1, 2a, and 2b

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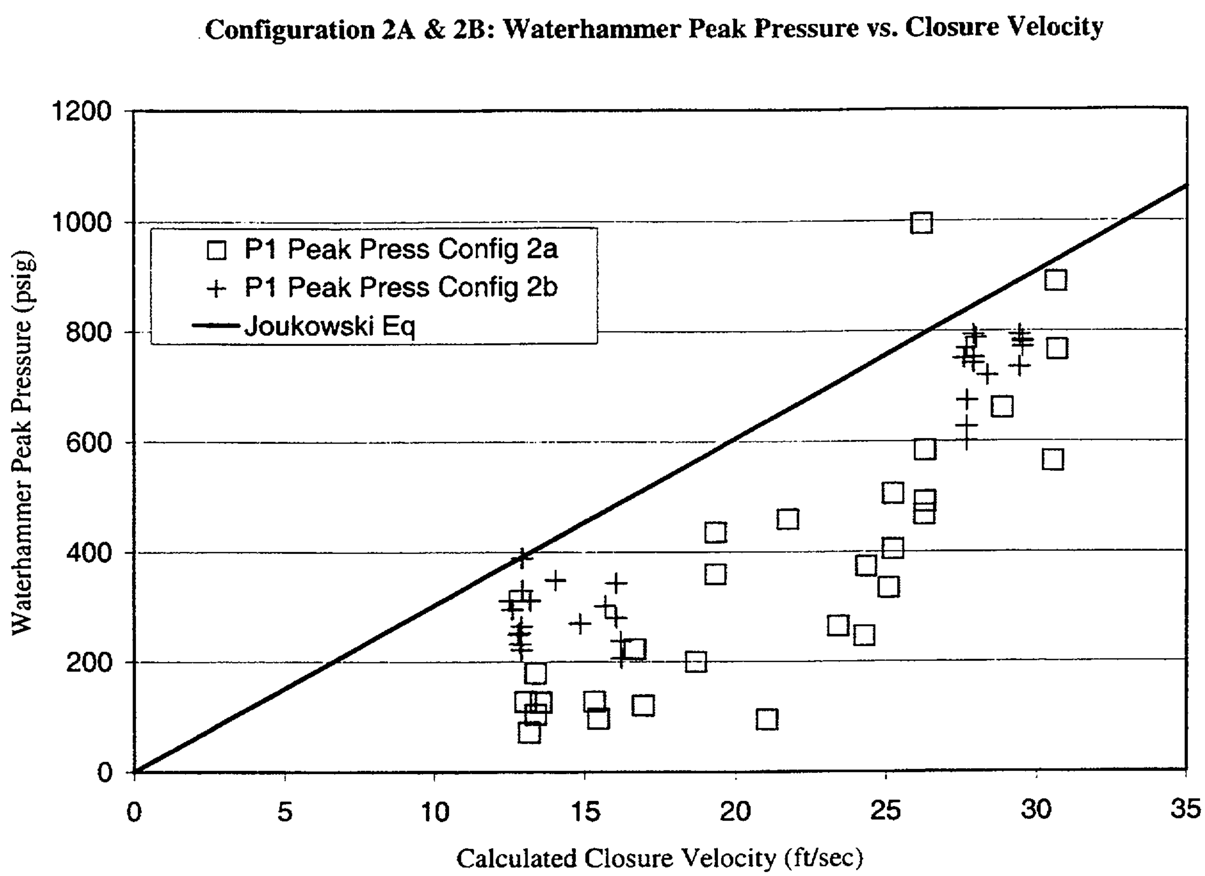

Figure 10-9: Configuration 2a and 2b Peak Pressure vs. Closure Velocity

ELECTRIFY THE WORLD

EPRI

September 17, 2001

Document Control Desk
U.S. Nuclear Regulatory Commission
11555 Rockville Pike
Rockville, MD 20852

Attention: Mr. Jim Tatum

Reference: Letter from Mr. Vaughn Wagoner to Mr. Jim Tatum, Response to Questions on Generic Letter 96-06, August 9, 2001

SUBJECT: Additional Response to Questions on Generic Letter 96-06

Enclosed are additional responses and clarification to our previous letter that responded to questions raised on the document "Resolution of Generic Letter 96-06 Waterhammer Issues", EPRI Interim Report TR-113594--V1 & V2, December 2000. The attachment to this letter includes responses to specific questions raised by the NRC. This information will be added to the final revision of the Technical Basis Report.

Please note that the enclosed document does not contain any proprietary information.

If you have any questions on the enclosed information or the general subject it addresses, please call me at 919-546-7959 or Avtar Singh at 650-855-2384.

Sincerely,

Avtar Singh for

Vaughn Wagoner
Carolina Power & Light Company
Chairman, EPRI Waterhammer Project Utility Advisory Group

Attachment

1. **The relationship of pressure rise time to impact velocity is given only for test configuration No. 1 which did not include air in the steam void. Please provide a comparison of the pressure rise time relationship with the data from test configuration 2 which did include air.**

Response:

The individual rise times for the Configuration 2a and 2b tests have been calculated. This data is provided in Figure 10-8 in the referenced letter. That figure utilized calculated "cushioned" velocity for the abscissa. The ordinate is test data; the abscissa is calculated. The rise time test data has also been plotted against the "uncushioned velocity". A figure showing the rise time versus the "uncushioned" velocity is attached to this letter and is referred to as Figure 10-8a. The uncushioned velocity is a parameter that is much more accurately calculated since it is dependent only on the system hydraulics and not the dynamics of void closure. The uncushioned velocity is readily known for the plant configurations. Note that in Figure 10-8a the rise time equation generally falls on or under the Configuration 2 test results. This is conservative since using a shorter rise time produces higher differential pressures across a pipe segment, and therefore, higher loads. Configuration 2b best represents the plant configuration as it has a longer column that test 2a. Referring to Figure 10-8a, it can be seen that the Configuration 2b test results are all conservative (have longer rise times) relative to the rise time equation recommended for use.

2. **For all the air release tests, the system pressure was decreased to approximately 7.4 psia (15" Hg). We cannot determine how much of the air release was due to the depressurization and how much was due to heat addition. Would the data apply to plants that have closed containment cooling systems and don't depressurize on LOOP?**

Response:

The amount of air released due to depressurization alone, without the agitation and increased nucleation sites developed during boiling, was investigated as part of this project. Research performed by Schweitzer et. al. and Zielke et. al. was reported in the TBR, Section 6.1. Figure 6-1 shows that the amount of dissolved gas released due to depressurization alone reaches a maximum value of approximately 0.031 gm/m^3 during the 30 second transient. The testing performed in the development of the data used to create Figure 6-1 was based on an agitated sample of water that was super-saturated with gas. The agitation was caused either by flow of the water through a pipe or through simple shaking. No boiling occurred in these tests.

Tests performed as part of the TBR development showed that water in the tubes, when exposed to boiling, would release approximately 50% of their dissolved non-condensable gas. Solubility curves were presented (revised TBR sections, Figure 6-7) that show the concentration of dissolved gas in saturated water to be approximately 20 mg/L or 20 gm/m^3. If 50% of the gas is released, the non-condensable gas mass is 10 gm/m^3.

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Therefore, the effect of pressure alone is approximately 0.3% of the mass released by boiling.

Since pressure alone accounts for little gas release, the effect of applying the data to closed cooling water systems where boiling occurs is expected to be insignificant.

3. **The header air content was determined to be reduced by at least 25%. How much of the air reduction was from de-aerating of the original header water and how much of the reduction was from mixing with the water from the test section that was discharged into the header?**

Response:

The water that was selected to be a conservative representative was water that was in the moisture separator region of the test. This water was carried with steam that was being expelled from the tube out of the header region into the moisture separator region. This water had the least amount of air released and had the least amount of mixing in the header. Very little of the air that was released was due to mixing with water from the tube that could have occurred in the header region.

4. **EPRI provided a theoretical argument that pressure in a gas volume between closing water columns is independent of pipe area and the heat transfer coefficient is not a function of pipe diameter. The argument appears correct provided the void shape remains cylindrical so that the heat transfer area does not change shape with piping diameter. Would the heat transfer area change shape for pipe sizes larger than the 2 inch size in the CCWH tests?**

Response:

Several potential physical phenomena could account for a change in surface area, but they can be shown to not occur or they are not credible reasons for making the test data unconservative.

First, the flow area could be postulated to slope away from its assumed planar shape. Calculations show that the "keep-full" velocity for piping is related to the Froude number (TBR Equation 7-1). For pipes up to 18" diameter, the keep-full velocity is between 4 and 7 ft/sec. Since the refill velocity to close a void will be greater than this, the pipe will not fill in a stratified manner, and the shape will remain nearly planar as assumed.

Second, the surface roughness of the exposed water in the closing column could be postulated to increase and thereby increase the surface area. Surface roughness is related by Taylor instability as primarily a function of acceleration (or deceleration), and not of pipe diameter (see TBR section 8.3.3). The tests performed as a part of the TBR preparation were run for closure velocities ranging from 10 to 45 ft/sec. The actual closure velocities in the plant are on the order of 15 to 20 ft/second. The higher tested

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closure velocity would lead to increased deceleration and increased surface roughness compared to the plant. Therefore, the water surface area available for condensation of steam would be larger in the tests than in the plant configuration, relative to the pipe diameter. As described in the TBR, the water surface area change per unit flow area due to surface roughness is pipe diameter independent. Because of the larger relative water surface area in the tests, the steam condensation rate in the plant configuration will be *lower* than those derived from the tests. This would provide more cushioning in the plant than measured in the tests.

5. **CIWH tests were performed at low pressures, less than 15 psig. The TBR states that the report's conclusions should not be utilized above 15 psig. The UM states that the results can be utilized up to 20 psig. What will be the basis for approving analyses for plants where CIWH might occur above 15 psig?**

Response:

The mechanism of the occurrence of condensation induced waterhammer (CIWH) was investigated in detail following the receipt of the test data. No mechanism exists that would result in a change of the behavior of the CIWH in a system that has a pressure that is five psia higher than that tested. The TBR concluded that the CIWH event would not impact the integrity of the system. This conclusion is not different for a modest increase in the system pressure from 15 to 20 psig.

6. **EPRI stated that the comparison of analyzed vs. test data loads was based on a structural damping value of 0.1%. A better comparison would be for typically assumed damping values (2-3%).**

Response:

The calculated responses shown in the predicted to measured support load comparisons provided in TBR Figures 13-7 and 13-8 were made from the results of a structural model with 0.1% damping. The analytical model was loaded by trapezoidal idealizations of the pressure pulses. These calculated loads were compared to actual support load data from the tests.

To determine the effect of increasing the damping to 2%, a single degree of freedom (SDOF) model was made for each pipe segment in the test, using measured geometry and structural frequencies. This is an appropriate model as there was very little participation between the legs of the test model. These SDOF models were loaded with forces of equivalent rise time and duration as developed by the tests and previously used. The resulting dynamic load factors decreased 1.9% to 7.7% with the increased damping. The magnitude of the change depended upon the pulse duration, specific pipe leg being loaded, and stiffness of that pipe leg. Translating these results to Figures 13-7 and 13-8, it can be seen that lowering the predicted forces 1.9% to 7.7% would not significantly impact the comparison between the measured and calculated forces. The conclusion that

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the trapezoidal representation of the pulse is an appropriate modeling method would not change.

7. **The air release experiment was performed at a pressure of approximately ½ atmosphere.**

   a. **What would be the expected amount of air released for other (especially increased) initial pressure conditions?**
   b. **For the actual plant situation, wouldn't pressure be expected to actually increase and not remain at ½ atmosphere? What effect would this have on the amount of air that is released?**

Response:

The pressure change had very little to do with the amount of air released as this is a secondary effect in comparison to boiling (see response to question 2). If boiling does not occur but a pressure reduction did occur, no credit would be taken for air release. The pressure in the test was selected to assure that there was very little residual air in the test fixture prior to the initiation of the test and so that boiling would occur quickly following the initiation of the test. Typically, open loop plants will have pressures that are initially lower than ½ atmosphere and then the pressures will increase to a pressure that is on the order of 1 atmosphere at the time that the pumps are restarted. Closed loop plants will typically have a higher initial pressure (around atmospheric) and the pressure will increase during the event. The test configuration is approximately in the middle of the ranges of pressure expected. Given the small impact of pressure on air release, the amount of air would be independent of the precise pressures reached in the plants.

9. **If the fan coolers are at a low point and the water in the supply and return headers maintain enough pressure on the water in the fan cooler tubes, boiling will not occur and no credit is allowed for air release. Now suppose that the height of water in the supply and return headers is slightly reduced to the point where nucleate boiling occurs in the fan cooler tubes. How much air would be released? Continuing with this thought, as the height of water is gradually decreased, what amount of air would be released for the various boiling intensities that are experienced?**

Response:

Credit for air will not be taken unless the water is exposed to temperatures above the boiling point corresponding to the pressure. All the water does not boil, but it has to be exposed to boiling that occurs in the region for the air to be removed. In order to assure that the water is exposed to boiling, a condition will be added to the TBR and the User's Manual that will eliminate credit for air release unless the temperature of the tubes reaches a temperature that is 10°F above the temperature at which boiling would occur. This will assure that the water in the tubes is exposed to boiling and that air release due to "air stripping due to boiling" will occur.

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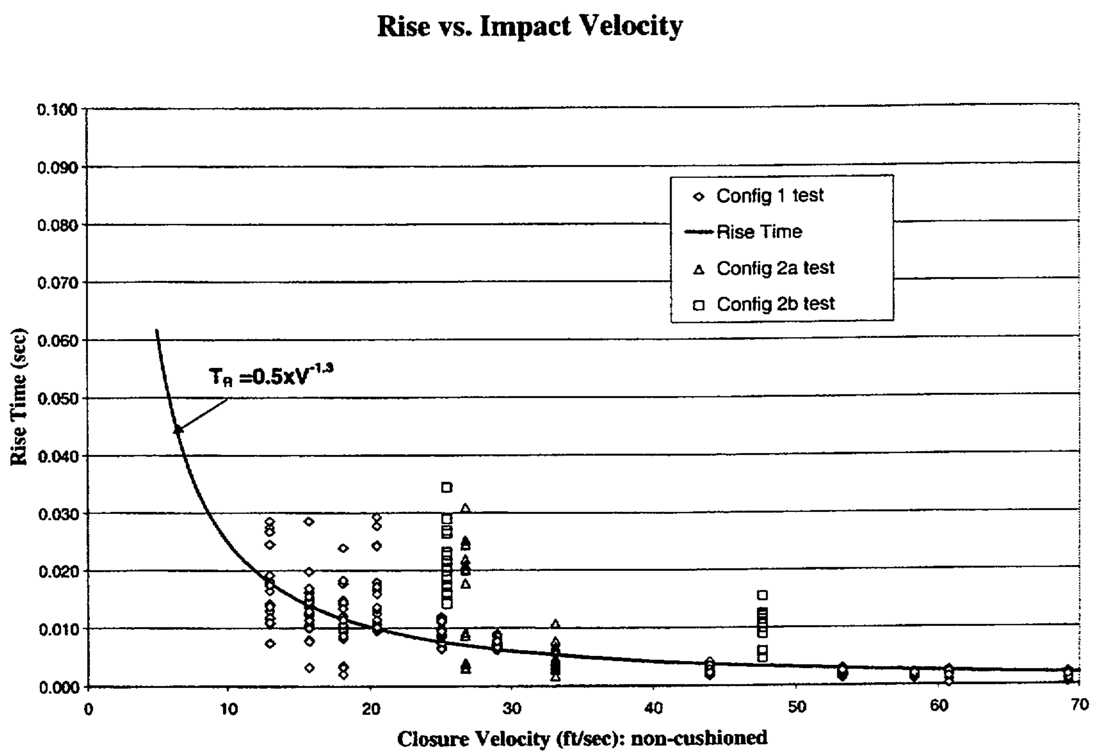

Figure 10-8a: Rise Time vs. Non-cushioned Impact Velocity

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EPRI

February 1, 2002

Document Control Desk
U.S. Nuclear Regulatory Commission
11555 Rockville Pike
Rockville, MD 20852

Attention: Mr. Jim Tatum

SUBJECT: Response to ACRS Comments (letter dated 10/23/01) on the EPRI Report on Resolution of NRC GL96-06 Waterhammer Issues

Dear Mr. Tatum:

EPRI has reviewed the input from the ACRS Thermal Hydraulics Subcommittee and Full Committee relative to Generic Letter 96-06 waterhammer issues as provided in the subject letter. We would like to provide some additional information on the specific issues that are raised in the ACRS letter, and we would also like to provide, as suggested in the letter, a perspective on the risk associated with the use of the methods recommended by the EPRI reports.

ACRS Comments

Many of the ACRS comments address the complexity associated with plant-specific evaluation of heat transfer in the fan coolers and the hydraulics of the voiding and draining process. The application of the EPRI work to plant-specific scenarios and configurations depends upon the plant-specific definition and subsequent evaluation of the plant-specific scenarios and configurations. Although the plants are generally similar in the design and performance of these systems, the plant specific differences require that individual, plant specific analyses be performed to evaluate the behavior of the fan cooler, the draining characteristics, and several other system related parameters. These specific plant calculations must be performed and reviewed on a plant-by-plant basis. The scope of the EPRI Technical Basis Report (TBR) was to provide methodology for evaluation of waterhammer loads at the time of final refill and column closure following pump restart.

The EPRI User's Manual (UM) provided plant specific analysis steps in Figure 2-1 and describes the application in Section 2.2. The following text that was extracted from the User's Manual describes the requirement for plant specific evaluation for heat transfer in the fan cooler, system voiding, and system refill.

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Model System Hydraulics: The flow, pressure, and potential paths for water to move and voids to form in the service water system should be determined for the duration of the transient. This will specifically include the time from the loss of power to the time of closure of the void. The system hydraulic model should include the following sub-tasks.

a) **Determine Fan Cooler Unit Performance:** The heat transfer across the coolers and the contribution this heat makes to the generation of steam voids should be determined. This effort should be performed on an individual plant basis – specific guidance is not provided in this User's Manual.

b) **Determine System Voiding:** Using the FCU heat input, piping elevation and system resistances, the system pressure and voiding should be determined. This effort should be performed on an individual plant basis – specific guidance is not provided in this User's Manual.

c) **Determine System Refill**: The flow rates and velocities of the refilling water should be determined from the pump curves and system hydraulic model. Determine anticipated location(s) of closure. This effort should be performed on an individual plant basis – specific guidance is not provided in this User's Manual.

On some issues, such as consideration of single active failure, guidance was provided in the User's Manual to assist in the development of a plant specific analysis.

Once the specific plant analysis was complete, the EPRI User's Manual and Technical Basis Report (TBR) would then be used to provide methodology for the determination of waterhammer loads at the time of final refill and column closure following pump restart. Limitations of application of the EPRI methodology to a specific plant were also provided. In Section 5.3.2 of the User's Manual, for example, limitations on column length, void length, water velocity, gas content, and void and interface temperature were provided. If these limits are not met by the results of the specific plant analysis, the results presented in the EPRI User's Manual are not applicable.

This is a complex problem that requires the analysis of system hydraulics, fan cooler boiling, void formation, and steam and gas behavior within the system. The EPRI project does not provide this analysis for the plants nor does it recommend methods to perform this analysis. These issues are the responsibility of the plant analysts.

The User's Manual provides guidance for the calculation of waterhammer characteristics after the pumps restart and final closure of the columns occurs. This is where cushioning for steam and other non-condensables in the void is calculated. The User's Manual also provides a basis for determining some of the final closure input (such as the amount of gas release) and methods for applying these dynamic pressures to a piping system to calculate loads in the pipe supports and stresses in the piping.

Risk Consideration

Given that we have not come to a conclusion with the ACRS on the conservatism of the technical approach presented in the User's Manual and the Technical Basis Report, a consideration of the risk associated with the events and the results that will be achieved if the report methods are used may be important. If the methods proposed in the EPRI Reports do not significantly increase the risk of unacceptable plant performance nor lead to an unacceptable risk to the plant, it is proposed that the methods of the EPRI report may be safely implemented without compromising the integrity or safety of the piping for plant application.

In order to assess the risk to the plant of application of the EPRI method, a review of the "progression" of events that could lead to an unacceptable condition should be performed. For the purposes of this evaluation, the "unacceptable condition" following a LOOP/LOCA event will be defined as a breach of the service water system pressure boundary. The events are as follows:

1. **Occurrence of a LOCA or MSLB** – The probabilities of occurrence of LOCA and MSLB events are provided in NUREG/CR-5750. From that document, the mean frequency of occurrence of a large LOCA is $5 \cdot 10^{-6}$/year, a medium LOCA is $4 \cdot 10^{-5}$/year, and a MSLB is $1 \cdot 10^{-3}$/year. The LOCA probabilities are represented in NUREG/CR-5750 as "reasonable but conservative" estimates of the frequency of occurrence.

2. **Occurrence of a LOOP following a LOCA or MSLB** – Studies provided in NUREG/CR-6538 and subsequent NRC work indicate that the dependent probability of a Loss of Offsite Power event following a LOCA event is approximately $1.4 \cdot 10^{-2}$/demand.

3. **Occurrence of a Simultaneous LOCA/LOOP Event** – The required design basis consideration is for the simultaneous occurrence of a LOCA or MSLB and a LOOP. The frequency of the combined event depends upon the probability of the LOCA and the MSLB and the dependent probability of the LOOP given that the LOCA has occurred. Using the values defined in each of the NUREGs referenced above gives a probability of the combined event on the order of $1.5 \cdot 10^{-5}$/year. For our purposes here, the value of probability of the design basis event (LOCA or MSLB occurring simultaneously with a LOOP) will be taken as 10^{-5}/year. With best estimate probabilities, this event likelihood of occurrence could be expected to be even lower.

4. **Void Formation** - If we have a LOCA/LOOP event, a void will form in an open loop plant with certainty. In a closed loop plant, void formation will depend on the specific plant characteristics and a void may or may not form. If a void does not form, a waterhammer will not occur.

5. **Pump Restart** – The pumps will restart with certainty and the velocity of the fluid in the pipe, immediately prior to closing the void, will be defined by the pressure in the void, the piping geometry, and the pump characteristics. This uncushioned closure velocity can be

reliably calculated. This velocity will not be higher than the rate at which the pumps, once restarted, can pump water. The calculation of the water velocity prior to closure is a plant specific analysis that can be conservatively performed.

6. **Column Closure -** The water columns will refill the void and the velocity at closure cannot be larger than the largest calculated differential velocity for the upstream and downstream water columns.

7. **Maximum Waterhammer Pressure** - An upper bound on the water hammer pressure can be calculated by the Joukowski relationship with the uncushioned closure velocity that corresponds to the pipe in which the closure will occur. The waterhammer pressure cannot be larger. With a probability of one, the waterhammer pressure will be equal to or less than the Joukowski pressure. The actual waterhammer pressure that will occur is stochastic and will have a wide variation. This variation is due to variations in the void distribution in the system immediately prior to final closure. This variation appears in all the integral system level experiments. The variation in the test data has been reviewed and, in the velocity range of interest, it varies from 50% to 100% of the maximum (for example, in the Configuration 2a tests, at a velocity of approximately 25 feet per second, the maximum pressure measured from the test was approximately 400 psig, the minimum pressure was approximately 200 psig, the Joukowski pressure for this velocity of closure is 775 psig -- see Figure 10-9 in the TBR). The variation in the test data that has been seen as part of the EPRI project is typical of many other waterhammer tests that have been previously performed and it indicates that it is unlikely that the Joukowski pressure will be attained given the scatter in the results of measured waterhammers compared to those predicted. It is assumed in the EPRI reports that the largest (Joukowski) pressure is attained for the calculated cushioned velocity, although it is very likely that the pressure less than the maximum seen in a test will be experienced.

8. **Cushioned Waterhammer** - With the cushioning that is predicted to occur due to gas and steam, the cushioned velocity will be on the order of approximately 30% to 40% lower than the maximum velocity (see User's Manual appendix – this depends on many parameters, including the amount of gas and steam). For closed loop plants, this value may be only 10-15%. The waterhammer that is predicted, then, will be on the order of 30% to 40% less than the pressure calculated by Joukowski, as the relationship between pressure and velocity is linear. If the cushioning did not occur, the waterhammer pressure and the stresses in the piping would be equivalent to the uncushioned waterhammer that would not have the 30% to 40% adjustment. There are two ways to consider the impact of this potentially higher stress:

   - The first is to consider actual plant performance. The occurrence of the waterhammer following a LOOP event – either simulated in a test or real – is known to have occurred many times in the industry. The waterhammer following a LOOP-only event is not cushioned by gas and steam in the void. The total number of occurrences of LOOP-only events are estimated to be on the order of at least several hundred, based on a review of the available plant data. These occurrences have all been in open loop plants and are more severe than a waterhammer that would occur following a LOOP/LOCA event.

Without any cushioning, the LOOP waterhammer is more severe than that following a LOOP/LOCA. No piping failures have occurred in any of these events. This would indicate that the probability of failure for a more severe waterhammer (an uncushioned waterhammer) is of the order of 10^{-2} or lower.

- The other method is to take the ASME Code limits and to calculate the probability of failure if the code limits were to be exceeded by approximately 40%. For the purpose of this evaluation, it will be assumed that the piping system is designed so that all the ASME code stresses in the piping were at the faulted condition limit when the cushioned waterhammer occurred – that is, the EPRI methodology is used and that the pipe was designed up to the code acceptable limit for that load. To determine probability of failure, an assumed stress distribution is used around a stress that is 40% larger than the faulted allowable ($2.4S_h$) and compared to the actual tested material strengths for A106-Gr B piping. Based on the actual margins available in the ASME code (see NUREG/CR-2137), the probability of the stress exceeding the strength can be shown to be on the order of 10^{-4} or less.

For the purpose of continuing the "event progression", a probability of failure in the pipe if the cushioned waterhammer were exceeded will be taken to be on the order of 10^{-2}. It is probably much less likely.

9. **Likelihood of an Unacceptable Event** - Given the low probability (10^{-5}/year) of the initiating events and the low probability (10^{-2}) of piping failure, the use of the methodology in the User's Manual and the Technical Basis Report will lead to a likelihood of an unacceptable event that is on the order of 10^{-7}. Again, for the purposes of this evaluation, the "unacceptable event" following a LOOP/LOCA event is taken as a breach of the service water system pressure boundary. The probability of 10^{-7} for this event is below the threshold for significant risk to the plant. Use of the methods in the User's Manual, therefore, will not compromise the safety of the plant for the systems within the bound provided in the User's Manual and Technical Basis Report. The methodology should be accepted as recommended in the report.

The most important consideration in the behavior of the waterhammer following pump restart is that there is an upper bound on the waterhammer pressure that can be attained -- the waterhammer without cushioning -- and that the waterhammer without cushioning has occurred many times in simulated LOOP events. The methods proposed in the EPRI TBR use the physics of gas compression to calculate a reduced closure velocity and waterhammer magnitude. The change in risk introduced by the use of these methods is not significant and the methods do not lead to an unacceptable plant risk following a LOOP/LOCA event. Hence, from the Risk-informed perspective, the methods proposed in the submitted EPRI TBR and UM are adequate for plant-specific application for resolution of the Generic Letter 96-06 issues.

The methods provided in NUREG 5220 were considered acceptable for conservatively analyzing waterhammer events. The NUREG uses the Joukowski relationship with the uncushioned velocity. NUREG-5220 acknowledges that the calculated results using their methods could be 2-10 times higher than reality. All seem to agree that if a void forms following a LOOP/LOCA event, it will have non-condensable gas and some steam in it, and that some cushioning will occur. The emphasis of the work that was performed by EPRI is to define the amount of cushioning that would be expected. The cushioning calculated will provide, in general, between zero and approximately 40% velocity reduction and subsequent pressure reduction. The risk discussion of this letter shows that given the low probability of the events, the limited energy available from this event, and the low probability of pipe failure, that the methodology proposed in the User's Manual is reasonable for those cases that fit within the parameters of the User's Manual and Technical Basis Report.

We hope the information provided herein is helpful. If you have any questions on the enclosed information or the general subject it addresses, please call me at 919-546-7959 or Avtar Singh at 650-855-2384.

Sincerely,

Avtar Singh for

Vaughn Wagoner
Carolina Power & Light Company
Chairman, EPRI Waterhammer Project Utility Advisory Group

Target:
Nuclear Power

About EPRI

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