72.57

CONTAINMENT FAN COOLER RESPONSE TO A SIMULTANEOUS LOCA & LOOP EVENT

Technical Report No. 96227-TR-01
Revision 3

prepared for:

Wolf Creek Nuclear Operating Corporation
Wolf Creek Nuclear Plant

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altran

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Report Record

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EXECUTIVE SUMMARY

An evaluation of the effects of a concurrent Loss of Coolant Accident (LOCA) and Loss-of-Offsite-Power (LOOP) on the Service Water (SW) system at the Wolf Creek Nuclear Plant has been completed. The objective of this evaluation was to determine if the effects of a LOCA occurring with a LOOP would create unacceptable waterhammer loads. The system was previously qualified for waterhammer resulting from LOOP and LOCA conditions. The results of our evaluation indicate that the LOOP and LOCA event will not result in waterhammer conditions that cause the Service Water System to exceed faulted allowables.

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TABLE OF CONTENTS

LIST OF FIGURES

The following figures are included as part of this evaluation:

1.0 SUMMARY

The evaluation reported herein considers a LOCA that is assumed to occur concurrently with a LOOP. The LOCA analyzed in this analysis is a double ended guillotine break of the reactor coolant loop as defined in the FSAR [1]. This is the enveloping condition for this analysis because it provides the greatest source of heat in containment. The effects of main steam line break was reviewed and found to be less limiting than the LOCA.

The LOCA fills the containment with saturated steam at a pressure that rapidly rises to 47.3 psig and the peak containment temperature reaches 306.1°F. During this same time all power is assumed to be lost to the Emergency Service Water (ESW) pumps and fans in the containment fan coolers. The water flow and air flow both coast down.

This results in a condition where heat is absorbed out of the containment atmosphere and deposited into the service water in the fan cooler. This analysis considers the potential for steam to be generated in the fan cooler and carefully examines each phase of the LOCA and LOOP event as it effects the SW system.

In addition to the analytical evaluation of the effects of a LOOP/LOCA, a test was performed in 1991 to determine the magnitude of a SW system Waterhammer resulting from a LOOP. This test is compared to the analysis to gage the accuracy of the analytical results.

2.0 OBJECTIVE

The objective of this evaluation was to develop differential peak pressure pulses which result from a LOCA and LOOP event.

3.0 LOOP/LOCA DESIGN BASIS AND ASSUMPTIONS

SW Water

T_{SWi} = Maximum water inlet temperature to coolers=95°F [FSAR 9.2.1.2.2.1]
Q_{LOCA} = flow rate requirement to each cooler during LOCA=1000 gpm [1].
Q_{NORM} = flow rate to each cooler during normal operation=925 gpm [1] Lake level is 1988' maximum [5].

Containment

T_C = Containment temperature following LOOP with LOCA is as shown on attached Figure 1 and as listed in FSAR table 6.1.

Equipment Positions

Following LOOP with LOCA the following times apply:

time (sec)	event
0	-LOOP
	-LOCA
	-SW Pumps, ESW Pumps, Fans, and Valves loose power
12	-D/G's start
	-Valves begin stroking closed [4]:
	HV-23 & 25 ('A' SW supply isolation) - full open to closed
	HV-24 & 26 ('B' SW supply isolation) - full open to closed
	HV-39 & 41 ('A' SW return isolation) - throttled to closed
	HV-40 & 42 ('B' SW return isolation) - throttled to closed
	- Valves begin stroking open:
	HV-37 ('A' train ESW return to UHS) - throttled to open
	HV-38 ('B' train ESW return to UHS) - throttled to open
18	-HV-37 full open (max 6 sec open time)
25.5	-HV-38 full open (max 13.5 sec open time)
32	-"A" ESW Pump starts
37	-"B" ESW Pump starts
42	-HV-23,25,24,26,39,41,40, & 42 full closed (max 30 sec closures)

"A" and "B" train Containment Cooler discharge throttling valves HV-45 and HV-46 are throttled to 21% and 22% open to achieve 2470 gpm and 2730 gpm flow rates respectively [5]. Delayed pump start and closed containment isolation valves are considered to be beyond design basis. Although waterhammer loads may increase marginally under these scenarios, loads are not expected to increase beyond yield.

4.0 DISCUSSION OF THE LOOP/LOCA

4.1 Description of the System

At the Wolf Creek plant there are four containment fan coolers (two coolers per train). The containment fan cooler characteristics are described below [6,7]:

4 coolers
12 coils per cooler
32 tubes per coil
6 passes per tube
9 ft per pass
or 384 qty 54' long tubes per cooler
5/8" OD, 0.035" wall tubing

A fan cooler is shown schematically in Figure 2. The piping configuration for each train is different. The A and B train piping configurations inside containment are shown in Figures 3A, 3B, and 3C. The differences in piping configurations will require each train to be individually analyzed to find the largest waterhammer pressure pulse.

The heat sink for Wolf Creek is a manmade lake. A portion of the lake has a seismically qualified partial height dam that serves as the ESW ultimate heat sink (UHS) in the event of a LOCA. The two ESW trains are independent. Flow for each train is provided by a single ESW pump or by the Service Water (SW) system. Discharge can be either to the SW system return header or to the ESW system return header. The SW system return header releases into the Circulating Water (CW) system discharge tunnel and the ESW system return header releases to the UHS. During normal operation discharge is to both the CW system and UHS.

The ESW and SW pumps are equipped with discharge check valves. A single 14" pipe branches to two 10" pipes inside containment to supply the two coolers on a train. The discharge rejoins in a common 14" pipe before exiting containment. Flow and back pressure to the coolers are controlled with a butterfly valve and orifice on each train located outside containment.

During a LOOP with a LOCA, power is lost to the pumps, fans, and valves until the Diesel Generators are started and the loads are sequenced as shown previously in section 3.0.

4.2 Limiting Break

An analysis of water boiling in the fan coolers under LOCA accident conditions is presented in this report. The LOCA accident is more severe than the main steam line break (MSLB) environment, even though higher containment temperatures are attained in the latter case. The reason why the LOCA is more severe, even at a lower temperature, is due to several heat transfer considerations.

Heat transfer rates are a function of the following three factors:

1. The nature of the fluid
2. The temperature driving the heat transfer
3. The heat transfer coefficient

Each of these factors will be discussed.

1. In the LOCA, the fluid is a saturated mixture of air and water vapor. It is at its "dew point" and will begin to condense as soon as it comes in contact with a cold surface. In the MSLB the steam is superheated, and the entire mixture has to cool as a gas until the vapor reaches the saturation, or condensing, temperature. Even though the temperatures are high, the heat transfer rates are low in this environment.

2. The temperature driving force for condensing, where the latent heat of the steam is transferred to the heat exchangers, is the saturation temperature corresponding to the pressure of the steam. Since there is less steam in the containment vessel in the MSLB than in the LOCA accident, the pressure is lower and the saturation temperature is lower.

3. The heat transfer coefficient during condensing is proportional to its vapor to air ratio. Also, since the volume of liquid released during a LOCA is significantly greater than as a result of an MSLB, the heat transfer coefficient will be larger.

Since all conditions relating to heat release rate give lower rates for the MSLB accident, analysis of the LOCA provides the worst case conditions.

4.3 Sequence of Events

The postulated event is initiated by the simultaneous occurrence of a LOCA and a LOOP. The service water pumps and the containment cooler fans shut down due to the LOOP. Both the pumps and the fans coastdown. The temperature in the containment will rise as shown in Figure 1. The pump coasts down to nominally 14 psig discharge pressure in approximately 2 seconds which is typical of ESW systems.

The A & B ESW pumps restart 32 and 37 seconds respectively after initiation of a LOOP.

While the pumps are coasting down, the water in the cooler tubes will be heated. The heating soon causes boiling in the tubes as the saturation pressure is reached. The boiling expels the water in the cooler and creates a steam void in the cooler. Steaming does not continue in the cooler because the piping configuration at Wolf Creek allows complete draindown of the coolers. As there is no inventory of water to feed the boiling process, steam pressures do not rise after the cooler is voided. The steam in the coolers quickly reaches a superheated condition as the containment temperature continues to rise. The behavior of the steam in the piping adjacent to the coolers is governed by the expanding void space in the piping system. Once steam generation ceases, the pressure in the coolers and the piping will decrease as the void space in the piping system increases.

Figure 4A shows the cooler pressure that corresponds to this sequence of events (this curve corresponds to Case 1 of Appendix A). This pressure curve assumes that the pressure between the initiation of boiling (approximately 2 seconds) and the time that the cooler is empty (approximately 9 seconds) is at the saturation pressure that corresponds to the containment temperature. The actual cooler pressures will rise at a slower rate during this period than shown in Figure 4A due to actual heat transfer characteristics of the coolers during the LOOP conditions. The evaluation is conservative since heat capacitance of the water prior to boiling and fan coastdown time is not included. Following the time that the cooler is empty with no means of additional steam generation in the cooler, the pressure drops to accommodate the steam expansion. The steam expansion is treated isentropically, and the void pressure decreases as the volume increases. The pressure drop is also conservative, since steam condensation which is expected on pipe walls and at water/steam interfaces has not been taken into account.

The water upstream and downstream of the coolers drains at a rate defined by the frictional losses, piping elevation changes, and cooler pressure. As the pressure rises during boiling, the cooler is voided and the water column travels further down the piping system. As the column advances, the length of the void increases. As the void exposes horizontal pipes, condensation induced waterhammers may occur. When the columns rejoin after pump restart, column closure waterhammer will occur.

4.4 Methodology

This analysis will evaluate the system for the occurrence of waterhammers and calculate the magnitude of credible waterhammer pressure pulses. The occurrence of a column closure waterhammer has been previously analyzed and qualified [2]. The previous load qualifications were based on upset condition allowable stresses. As the LOOP with LOCA condition follows the occurrence of a faulted plant condition, faulted condition allowable loads will be used as a basis for acceptability in this analysis.

The objective of the analysis will be to demonstrate that the occurrence of the column closure waterhammer for the LOOP with a LOCA is not more severe than a LOOP without a LOCA. If this can be shown, the column closure event following a LOOP with a LOCA will be enveloped by analyses already completed.

The occurrence of other waterhammer types will also be evaluated. Condensation induced waterhammers are expected to be feasible in the system and their occurrence will be evaluated.

The details of the sequence of events and waterhammer calculations are described in the following sections:

- 5.1 Repressurization Curve Development
- 5.2 System Resistance Development
- 5.3 Volume and dh/dV Determinations
- 5.4 Condensation Induced Waterhammer Susceptibility
- 5.5 Condensation Induced Waterhammer Pressure Pulses
- 5.6 Column Closure Waterhammer Prediction
- 5.7 Flashing Flow Assessment

5.0 LOOP/LOCA ANALYSIS

5.1 Repressurization Curve Development

The ESW system pressure following the LOOP with LOCA event was calculated using a spreadsheet program. Copies of the spreadsheet are shown in Appendix A to this analysis. The development of the spreadsheet is described below:

- An isentropic expansion of the steam following draining of the cooler is assumed. An isentropic exponent of 1.13 is conservatively used. This results in pressures higher than a typical exponent of 1.3 for steam. It is also conservative since pressure reductions due to condensing of the steam in the downstream water is neglected.
- The system resistance is input from section 5.2 below.
- Volumes are input from section 5.3 below.
- The change in height as a function of a change in volume are from section 5.3 below.
- The containment temperatures are from Figure 1.
- The pressure in the cooler while it is draining is conservatively assumed to follow the saturation pressure corresponding to the containment temperature.
- Quattro Pro is used for the spreadsheet.
- Three cases are run with the spreadsheet to conservatively calculate condensation induced waterhammers and column closure waterhammers. Case 1 will predict maximum condensation induced waterhammer pressure pulses on the A train. Case 2 will predict maximum column closure waterhammer pressure pulse. Case 3 will predict the maximum condensation induced waterhammer pressure pulse on the B train.

5.2 System Resistance Development

As the coolers drain during the LOOP, the flow rate of the water out of the coolers will be a function of the system resistance between the coolers and the lake. The supply and discharge piping will drain. The flow paths are parallel and were combined into an equivalent parallel resistance. The draining water can take three paths out to the lake.

The path to the UHS via valves HV-37 and HV-38 is available throughout the transient. The path to the CW discharge tunnel via valves HV-39, 41, 40, & 42 is available throughout the transient but the valves are stroking closed and the resistance in this path is increasing during the transient. A third path is via the SW supply to ESW. While valves HV-23, 25, 24, and 26 go closed, failure of the SW pump discharge check valves is assumed.

The cooler pressure will be lower at the time the cooler empties if the drainage rate is fast versus slow. This is because the water temperature follows the increasing containment temperature. Once the cooler is empty, the pressure decreases as the steam bubble expands isentropically. The pressure when the void expansion starts will have a significant effect on the magnitude of condensation induced waterhammer pressure pulses. It is conservative to not include the water drainage paths to the SW system in the model for condensation induced waterhammer predictions. The condensation induced waterhammer prediction will be referred to as Case 1 for train A and Case 3 for train B.

The void size will be larger with decreased resistance from the SW cross connect being open. This may affect the magnitude of column closure waterhammers. It is therefore necessary to assess the effect of decreased resistance from the SW path and determine if the column travels significantly further with the reduced resistance. This model will be referred to as Case 2.

The system resistances for supply and discharge piping paths were calculated in Appendix B. The resistances are summarized as a simplified circuit diagram in Figure 5.

The resistance in the "B" train is different than the "A" train due to the different piping configurations. The "B" train system resistance will be calculated by comparing HV-45 flow rate with HV-46 flow rate.

The resistances are normalized to the equivalent lengths of 14" piping. The piping lengths and configurations in Appendix B are from isometric drawings listed under reference [8]. The friction factors and resistance coefficients are taken directly from reference [9] except for those components identified below:

'A' Train Return Piping
"F1" Butterfly Valve (HV-45) is throttled 21%:
14" valve
interpolating reference [10] between 20 & 30%
C_v=467 @ 21%
$K=891d^4/C_v^2=891(13.124)^4/(467)^2=121$

"F1" Flow Orifice (FO005)
bore size from reference [11]
d_1=6.9"
d=13.124"
$d_1/d=\beta=0.53$
then from reference [9] page A-20
C=.63 and $K=(1-\beta^2)/C^2\beta4$
this K was then programmed in Appendix B

'A' Train Supply Piping
"COM5" CCW HX
from reference [3]:
ΔP=5 psi
Q=8800 gpm
then $C_v=Q/(\Delta P)^{1/2}=8800/(5)^{1/2}=3935$
then $K=891d^4/C_v^2=891(13.124)^4/(3935)^2=1.71$

Combined Return Piping
during normal operation lineup from reference [3]:
"R2" discharge to CW/SW system
ΔP=30 psi
Q=4376 gpm
then $C_v=Q/(\Delta P)^{1/2}=4376/(30)^{1/2}=799$
then $K=891d^4/C_v^2=891(13.124)^4/(799)^2=41$

"R3" discharge to UHS
ΔP=30 psi
Q=8500 gpm
then $C_v=Q/(\Delta P)^{1/2}=8500/(30)^{1/2}=1552$
then $K=891d^4/C_v^2=891(13.124)^4/(1552)^2=11$
these resistances for R2 and R3 are programmed in Appendix B for Case 2.

after discharge valves have re-aligned for LOCA from reference [3]:

"R2" discharge to CW/SW system
no flow to this path (valves shut)

"R3" discharge to UHS
ΔP=72.8 psi
Q=13576 gpm
then $C_v=Q/(\Delta P)^{1/2}=13576/(72.8)^{1/2}=1591$
then $K=891d^4/C_v^2=891(13.124)^4/(1591)^2=10.4$
This resistance is programmed in Appendix B for Case 1 & 3.

"B" Train Resistance Adjustment
The flow rate from the "B" train containment coolers is set at a higher rate than the "A" train which implies less system resistance in the "B" train than in the "A" train. Assuming the total pressure drop in each train is equivalent, the resistance in the return piping may be adjusted as follows:

Q_A=2470 gpm
Q_B=2730 gpm [5]
$K_B=K_A(Q_A/Q_B)^2$

This resistance is programmed in Appendix B for Case 2 & 3.

SW Supply Resistance Path
The drainage rate will be increased with water flowing to the normal SW supply. The resistance associated with this path is assumed to be the same as the resistance to the UHS. This assumption is considered appropriate given that the water must flow through more piping and valves and back flow through the SW pumps. This path will therefore have more resistance than the UHS path and use of the UHS resistance is considered conservative. This resistance is programmed in Appendix B for Case 2.

CASE 1 TOTAL RESISTANCE:
Case 1 is a model of the "A" train with the following characteristics:

- No reverse flow through the SW supply piping ($K_{s1}=\infty$).
- No flow to the CW/SW system return ($K_{r2}=\infty$).
- "A" train geometry/volumes are used for calculating the repressurization curve.

Case 1 allows prediction of maximum condensation induced waterhammer pressure pulses in the "A" train piping.

K_{CASE1}=28 FROM APPENDIX B

CASE 2 TOTAL RESISTANCE:

Case 2 is a model of the "B" train with the following characteristics:

- System resistance from "A" train is used with an adjustment for decreased B train resistance on the normal return side of coolers.
- Flow to the SW supply is allowed during the entire transient.
- Flow to the CW/SW return is allowed during the entire transient.
- "B" train geometry/volumes are used for calculating the repressurization curve.

Case 2 provides the least system resistance and allows the void to move the furthest possible distance. This allows the most conservative prediction of column closure waterhammers.

K_{CASE2}=22 FROM APPENDIX B

CASE 3 TOTAL RESISTANCE:

Case 3 is a model of the "B" train with the following characteristics:

- No reverse flow through the SW supply piping ($K_{s1}=\infty$).
- No flow to the CW/SW system return ($K_{r2}=\infty$).
- "B" train geometry/volumes are used for calculating the repressurization curve.

Case 3 allows prediction of maximum condensation induced waterhammer pressure pulses in the "B" train piping.

K_{CASE3}= 27 FROM APPENDIX B

5.3 Volume and dh/dV Determinations

The volume and change in height (h) per change in volume (v) are necessary to model the expansion and progression of the steam bubble.

COOLER

$A=\pi(.625-2(.035))^2/4(144)=1.68(10)^{-3}ft^2$
L=110"(6pass)(32tubes)(12coils)(ft/12")=21120ft
$V_1=21120(1.68)(10)^{-3}ft^3=35.48ft^3$

Cooler manifolds [12]
2 qty 3"OD 24" long pipes per coil
$V_2=2(12coils)(2ft)(.0513ft^2)=2.45ft^3$

2 qty 8" OD 11.5' long header per cooler
$V_3=2(.3474ft^2)(11.5ft)=7.99ft^3$

2 qty 6" OD 11.5' long header per cooler
$V_4=2(.2006ft^2)(11.5)=4.61ft^3$

~26' of 3" OD piping from headers to 3" coil pipes
$V_5=26(.0513)ft^3=1.33\ ft^3$

$V_{cooler}=35.48+2.45+7.99+4.61+1.33=51.86\ ft^3$
2 coolers therefore: $V_{2\ coolers}=103.7\ ft^3$
$dh/dV=(11.5ft)/2(51.86ft^3)=0.11\ ft/ft^3$ from 2080'6" to 2060' EL.

"A" TRAIN PIPING [8]

The supply and discharge piping configurations are nearly equivalent so just the discharge piping volumes will be calculated and doubled to account for the supply side volumes.

from EL. 2069' to 2064'9":
$V_8=8.5(.3474)=2.95ft^3$
$V_6=8.5(.2)=1.7ft^3$
$V=2(2.95+1.7)=9.3\ ft^3$
$dh/dV=4.25/9.3=0.457ft/ft^3$

from EL. 2064'9" until leg drained:
$V_8=24.5(.3474)=8.5ft^3$
$V_6=15(.2)=3ft^3$
$V_{10}=9(.5475)=4.9ft^3$

$V=2(16.4)=32.8ft^3$
$dh/dV=0$

<u>from EL 2064'9" to 2058'9"</u>
$V_8=6(.3474)=2.1ft^3$
$V_{10}=6(.5475)=3.29ft^3$
$V=2(5.4)=10.8ft^3$
$dh/dV=6/10.8=.556ft/ft^3$

<u>from EL 2058'9" until leg drained</u>
$V_8=5.25(.3474)=1.8ft^3$
$V_{10}=27.75(.5475)=15.2ft^3$
$V_{14}=16(.9394)=15ft^3$
$V=2(32.1)=64.2ft^3$
$dh/dV=0$

<u>from EL 2058'9" to 2018'8"</u>
$V_{14}=2(40)(.9394)=75.2ft^3$
$dh/dV=40/75.2=.53\ ft/ft^3$

<u>from EL 2018'8" until leg drained</u>
$V=2(75.75)(.9374)=142.3ft^3$
$dh/dV=0$

"B" TRAIN PIPING [8]

The "B" train supply and discharge piping configurations are not as symmetric as in the "A" train so both supply and discharge piping volumes will be calculated. The B and D cooler piping configurations have differences which may make the water columns drain at different rates around EL 2027' 6" where the B cooler supply/discharge piping turns to a horizontal run while the D cooler discharge piping continues as a vertical run. The drainage rates are assumed to be similar up to this point. The calculations will show that there is not significant void expansion beyond this point so that this approach is appropriate.

<u>from EL. 2069' to 2063'6":</u>
$V_6=(10.75+9+3.5+9.5)ft(.2ft^2)=6.55ft^3$
$V_8=(10+9+16)ft(.3474ft^2)=12.16ft^3$
$V=18.71ft^3$
$dh/dV=5.5/18.71=.29\ ft/ft^3$

<u>from EL 2063'6" until drained:</u>
$V_8=(14.75+12+3)ft(.3474ft^3)=10.3ft^3$
$V_6=(18.75ft)(.2ft^2)=3.75ft^3$

V_{10}=(23.75+22+21.75+20)ft(.5475ft^2)=47.91ft^3
V=61.96ft^3
dh/dV=0

<u>from 2063'6" to 2027'6":</u>
dh=2063.5-2027.5=36ft
dV=4(36ft)(.5475ft^2)=78.84ft^3
dh/dV=36/78.84=.457ft/ft^3

<u>from 2027'6" until drained:</u>
V_b=.5475(62+64.5+5.75)=144.8ft^3
Note that real available volume is more at this elevation since D cooler lines are not accounted for here. The void progression will be assessed in section 5.6 to determine if the D cooler lines would be emptied.
dh/dV=0

5.4 Condensation Induced Waterhammer Susceptibility

The uncovering of horizontal runs of pipe during the draindown creates the potential for condensation induced waterhammer. As horizontal portions of the lines are exposed, steam will enter the space formed at the top of the pipe. The space between the top of the pipe and the exposed water can allow condensation of steam and trapping of steam bubbles. The rapid condensation of the trapped steam and the subsequent closing of the void by water causes a condensation induced waterhammer pressure pulse [14,15].

There is piping in both the "A" and "B" trains that is susceptible to condensation induced waterhammers. The magnitude of the waterhammer is proportional to the steam pressure at the time of the occurrence. Since the steam pressure is decreasing as the void expands, the first susceptible pipe will have the largest pressure pulse in each line.

The following criteria will be imposed to determine what piping is susceptible per reference [13]:

- Horizontal or near horizontal piping
- Subcooling greater than 36°F.
- L/D > 24.

The following assumptions are made to screen for susceptible piping:

- It is conservatively assumed that during draindown, horizontal pipes will drain from the top down as opposed to being "piston driven" from one end.
- Water temperatures correspond to the containment temperatures of Appendix A.
- The difference between the coldest water in the header and the hottest steam will be used to evaluate subcooling margin. This conservatively neglects mixing in the headers.
- Once the cooler is drained, the steam temperature remains constant. Since no credit is being taken for condensation during the pressure transient, this assumption is appropriate.

Screen "B" Train for limiting pipe:

The B cooler discharge piping is the section of piping on the B train that is most susceptible to condensation induced waterhammer loads since it has the longest sections of horizontal piping near the coolers. Therefore only the B cooler will be assessed:

Checking L/D:

<u>6" piping at elevation 2065' 6":</u>

L=5.25' D=0.5ft L/D=10.5 < 24 therefore not susceptible

<u>8" piping at elevation 2065'6":</u>

L=4.5' D=0.665ft L/D=6.8 < 24 therefore not susceptible

<u>8" and 10" piping at elevation 2063'6":</u>

L=38.5' D=0.833ft L/D=46.2 > 24 therefore may be susceptible

<u>10" piping at elevation 2027'6":</u>

L=62' D=0.833ft L/D=74.4 > 24 therefore may be susceptible

Checking subcooling:

<u>piping at elevation 2063'6":</u>

Volume of water in the piping at this elevation is $V=61.961ft^3$
$V_{cooler}=103.7ft3$

$(V_{cooler} - V)=41.74ft^3 = V_{out}$

The water in the cooler when V_{out} is 41.74 ft^3 will be the coldest water in the header when the void reaches the header. When V_{out} from Appendix A case 3 equals 41.74 the time is 5 seconds and the water temperature is 227°F.

Subcooling: 243-227=16°F < 36°F therefore not susceptible

<u>piping at elevation 2027'6":</u>

The piping at elevation 2027'6" has a volume of 144.8 ft^3 which is greater than the cooler volume. A portion of this header, then, may be near the cooler outlet temperature prior to the transient of 100°F. This would allow sufficient subcooling margin for condensation induced waterhammer. Therefore, this piping is susceptible.

Screen "A" Train for Limiting Pipe

The discharge pipes from the A and C coolers have the longest lengths of horizontal piping near the coolers and are therefore most susceptible to condensation induced waterhammers.

Checking L/D:

<u>A cooler 6" & 10" piping at EL 2064'9"</u>
$L=L_6+L_8+L_{10}=15+1+8=24'$ D=.5' L/D=48>24 therefore may be susceptible

<u>A & C piping at EL 2058'9"</u>
$L=L_6+L_8+L_{10}+L14$
$L=3+5.25+26+17=51.25'$ D=.665' L/D=77>24 therefore may be susceptible

Checking subcooling:

<u>6" and 10" piping at elevation 2064'9" subcooling:</u>

Volume of water in the piping at this elevation is $V=32.81ft^3$
$V_{cooler}=103.7ft3$
$(V_{cooler} - V)=70.9ft^3$

When Vout from Appendix A Case 1 equals $70.9ft^3$, the time is 7 seconds. The water that exited the cooler at 7 seconds is the water that is at the downstream end of the header when the steam starts to create a void in the header. The water coming out of the cooler at 7 seconds is 234°F.

Subcooling: 243-234=9°F < 36°F therefore not susceptible

<u>A & C piping at EL 2058'9"subcooling:</u>

Volume of water in the piping at this elevation is $V=64.2ft3$
$V_{cooler}=103.7ft3$
$(V_{cooler} - V)=39.5ft^3$

When Vout from Appendix A Case 1 equals $39.5ft^3$, the time is 5 seconds. The water that exited the cooler at 5 seconds is the water that is at the downstream end of the header when the steam starts to create a void in the header. The water coming out of the cooler at 5 seconds is 227°F.

Subcooling: 243-227=16°F < 36°F therefore not susceptible

A & C piping at EL 2018'8":

The piping at elevation 2018'6" has a volume of 142.3 ft^3 which is greater than the cooler tubing volume. A portion of this header, then may be near the cooler outlet temperature prior to the transient of 100°F. This would allow sufficient subcooling margin for condensation induced waterhammer. Therefore, this piping is susceptible.

5.5 Condensation Induced Waterhammer Pressure Pulses

Condensation induced waterhammer is evaluated by calculating the system pressure that will exist when each horizontal line is exposed. This system pressure is then used to calculate the pressure pulse that would result from the waterhammer. The equation to be used which is derived from the Joukowski equation and an energy balance [14] is:

$$\Delta P = 0.707C\sqrt{(P_o - P_v)\ \rho_l \frac{\alpha}{1-\alpha}}$$

where C=sonic velocity
P_o=system pressure (steam pressure)
ρ_l=water density
α=void fraction
P_v=void pressure (water saturation pressure)

A maximum ratio of steam to liquid of 0.35 is used in the analysis ($\alpha/(1-\alpha)$= 0.35). Since steam velocities are expected to be low with the isentropic steam expansion, 0.35 is considered conservative for the steam to water volume ratio. It is more likely that condensation induced waterhammers will occur at lower void ratios with a resultant lower pressure pulse. A sonic velocity of 2300 ft/sec is used in the calculation based on References [16, 13]. The highest system pressure in the most susceptible horizontal pipes in the "A" & "B" trains is 8.2 psia and 7.9 psia respectively. For the "A" train, a void ratio of 0.35 is not reached prior to pump restart. The largest void ratio in the "A" train is 0.22. The corresponding condensation induced waterhammer loads are:

$$\Delta P_A = 0.707(2300\frac{ft}{sec})\sqrt{(8.2-1.3)\frac{lb}{inch^2} 61.7\frac{lb}{ft^3}(.22)\frac{sec^2}{32.2ft}\frac{ft^2}{144inch^2}}$$

ΔP_A = 231.1 psi
P_{pulseA}= (231.1-14.7) psig
P_{pulseA}= 216 psig

$$\Delta P_B = 0.707(2300\frac{ft}{\sec})\sqrt{(7.9-1.3)\frac{lb}{inch^2}61.7\frac{lb}{ft^3}(.35)\frac{\sec^2}{32.2ft}\frac{ft^2}{144inch^2}}$$

ΔP_B = 285.1 psi
P_{pulseB}= (285.1-14.7)psig
P_{pulseB}= 270.4 psig

These pressure pulses are conservative considering that 2300 ft/sec is used for the sonic velocity. With typical SW air concentrations of 8 ppm the sonic velocity will be less than 2300 ft/sec. The sonic velocity decreases significantly if free air is present in the water. Normal air concentrations in open loop service water systems are typically 8 ppm. As the temperature is increased following a LOCA, the air will be released and significantly lower the sonic velocity. The effect of temperature increase is shown in Figure 6. This figure was developed from the spreadsheet shown in Appendix C. With water temperatures greater than 200°F in the horizontal pipes where the condensation induced waterhammer will occur, the sonic velocity predicted would be less than 1000 ft/sec. Using 2300 ft/sec results in conservative loads.

5.6 Column Closure Waterhammer Prediction

Pressure pulses can be caused in systems where voids form due to elevation differences between the equipment or piping and their suction or discharge reservoir. This is the case with the fan coolers where a void will form in the discharge piping any time that the service water pumps are shut down. The velocity of the closing water will determine the magnitude of the pressure pulse.

When the ESW pump starts, the water advances towards the cooler from the normal supply side and at a lesser rate from the normal discharge side. The advancing water columns will eventually meet in the discharge piping and a column closure waterhammer will occur. The impact velocity of the LOOP with LOCA waterhammer will be less than the LOOP without LOCA waterhammer. The reasons for this are defined below:

(1) The frictional resistance in the two cases are the same. The <u>total</u> system resistance associated with a LOOP with LOCA is greater than a LOOP without LOCA because the void is initially at a pressure greater than 0 psia (LOOP without LOCA void is at 0 psia). Since the system resistance is greater with a LOCA, the closure velocity in the LOCA case is less than or equal to the closure velocity in the no LOCA case.

(2) Since the system resistance in the "B" train is less than the "A" train, a large void is possible on the "B" train. The ratio of impact velocity to closure velocity is dependent upon the void size. The larger the void size the greater the potential for increased impact velocity. The "B" train was selected to evaluate the impact velocity to closure velocity. Reference [17] provides a relationship of impact velocity to closure velocity as a function of void size and piping lengths. The length of piping from the pumps to the coolers is approximately 430 ft on the B train to the D cooler. The void progresses into the long horizontal runs below the B and D coolers per Appendix A Case 2.

The total void volume is 338.5 ft^3 from Appendix A Case 2 upon pump restart. The volume accounted for at elevation 2027'6" was just the piping serving the B cooler (145ft^3) for the spreadsheet model. 76 ft^3 of the void is below 2027'6". Approximately half this volume is in the B cooler piping and half in the D cooler piping and the void progresses the following distance past 2027'6" in each line:

$l = .25(76 ft^3)/(.5475 ft^3/ft) = 35$ ft

Modeling the system with just the B piping volume at elevation 2027'6" in Appendix A Case 2 & 3 is acceptable since the void will not progress into the 14" piping downstream of the D cooler by inspection of the piping lengths available. The void length on each side of the cooler is calculated as 35 feet plus the distance to the cooler from 2027'6" (66 feet). For the D cooler:

Void size on each side of the cooler = 35 + 66 = 101 ft.
L=430 + 101 ft = 531 ft.
X_o=430 - 101 = 329 ft.
X_o/L=329/531 = 0.62 for LOOP with LOCA

For a LOOP without LOCA the void progresses less and the ratio X_o/L will be slightly larger. A figure showing the relationship between the impact velocity and X_o/L for different FL/2D ratios [17] is shown in Figure 7. For a large FL/2D ratio, the difference in impact velocity V_i for small differences in X_o/L is small. The ratio of impact velocity to closure velocity approaches unity for all fL/2D ratios greater than 20.

Following a LOOP without a LOCA the supply side column will close at a rate of 10.32 Ft/sec. in a 10" Sch40 pipe per Reference 21. The system resistance corresponding to this closure is found from pump runout conditions. With pump runout at 24000 GPM the system resistance is 250 ft. from the pump curve [22]. The void pressure at pump restart is 0 PSIA and 5.9 PSIA for LOOP without and with LOCA respectively.

The LOOP with LOCA void size is larger than LOOP without LOCA since the void is pressurized in the LOCA case. LOOP without LOCA will reach the horizontal header at elevation 2027'-6" The frictional resistance during closure is defined by:

$h_f = P_1 - P_o - \Delta h$

Where
P_1 = Pump dynamic head
P_o = Void pressure
Δh = Elevation change from reservoir to closure point

Then
Δh = 2027'6" - 1988' = 39.5'
$P_{o\text{-}LOOP}$ = 0 PSIA = 0 Ft H_2O
P_1 = 250 Ft H_2O
$h_{f\text{-}LOOP}$ = 250 - 0 - 39.5 = 210.5 Ft H_2O

From [21] $V_{O\text{-}LOOP}$ = 10.32 Ft/Sec for 10" SCH 40 pipe then:

$$h_{f\text{-}LOOP} = \frac{fL}{2D}\,\frac{V_{LOOP}^{2}}{g}$$

$$\frac{fL}{2D} = \frac{176.5(32.2)}{(10.32)^2} = 53 >> 20$$

Therefore V_f/V_o will approach unity for LOOP with and without LOCA.

The impact velocity for the LOCA case will be less than or equal to $V_{O\text{-}LOOP}$ since the void is pressurized in the LOCA case.

In the LOOP with LOCA case, the sonic velocity at closure will be lower because the water is heated in the cooler releasing free air in the water prior to closure. The magnitude of the column closure pressure pulse will be lower for the LOOP with a LOCA and the limiting column closure waterhammer is the LOOP without LOCA.

During refilling, bubble collapse type waterhammers similar to those that occur in the horizontal lines during draining will not occur because the refill velocity exceeds the velocity required to keep the pipe full. A velocity of approximately 5 ft/sec is needed to keep a 10" pipe full [18]. The refilling velocity exceeds this and will preclude the occurrence of condensation induced waterhammer in the horizontal lines during refill.

The "B" train column closure waterhammer pressure pulse is calculated as 193 psig as shown in Appendix E. The "A" train column closure waterhammer pressure pulse is calculated as 225 psig as shown in Appendix E.

5.7 Flashing Flow Assessment

Design flows are required to be established upon starting of the ESW pumps to ensure design heat removal. If flashing occurs in the ESW system, flow may be reduced. Two phase flow increases frictional losses and creates the potential for choked flow conditions. Wolf Creek was evaluated for flow limitation at (1) restrictions upstream of coolers, (2) the coolers themselves, (3) restrictions downstream of the coolers, and (4) restrictions in the 30" header. For all cases evaluated, no flow limiting condition was found. Each of these are evaluated below:

(1) Upstream of the coolers the water is not significantly heated since the cooler voids quickly and the water is not heated. There are also no significant flow restrictions upstream of the coolers. The water will not flash upstream of the coolers after starting of the ESW pumps based on the following:

Determination of water temperature in horizontal headers:

The water in the horizontal piping susceptible to condensation induced waterhammer is assumed to be well mixed. This is appropriate since the horizontal pipes will drain from the top down and the agitation from the condensation induced waterhammers encourages mixing.

To determine the header mixed temperature:

1) The initial 50% of the cooler volume which empties is assumed to be at the average water temperature from the beginning of the transient to the time when half the cooler is drained:

$$T_{half1} = 0.5\,(T_{time=0} + T_{time\ 1/2\ drained})$$

This assumption simplifies the analysis. Since the flow rate does not change significantly during draining of the cooler and the pressurization is modeled as nearly linear, this assumption is appropriate.

2) The last 50% of the cooler volume which empties is assumed to be at the average water temperature from the time when half the cooler is drained to the time when the entire cooler is drained:

$$T_{half2} = 0.5\,(T_{time\ 1/2\ drained} + T_{time\ empty})$$

3) The volume of water in the header not displaced by hot water, is at a temperature of 100°F (normal cooler outlet temperature).

<u>"A" Train</u>

@ EL. 2018'8"

$V_{hdr} = 142.3\ ft^3$

$T_{time=0} = 100°F$

$V_{time\ 1/2\ drained} = 51.85\ ft^3$

$T_{time\ 1/2\ drained} = 230.6°F$

$T_{half1} = .5(100+230.6) = 165.3°F$

$V_{time\ empty} = 103.7\ ft^3$

$T_{time\ empty} = 243°F$

$T_{half2} = .5(230.6 + 243) = 236.8°F$

$V_{hx} = 35.48\ ft^3$

$x_{half1} = x_{half2}$

$= 35.48/142.3/4 = .062$

(divided by four since half the cooler goes to supply side and half to discharge side)

$x_{hdr} = (142.3-.5(35.48))/142.3$

$x_{hdr} = 0.875$

$T_{mix} = x_{half1}T_{half1} + x_{half2}T_{half2} + x_{hdr}T_{hdr}$

$T_{mix} = .062(165.3 + 236.8) + .875(100)$

$T_{mix} = 112°F$

<u>"B" Train</u>

@ EL. 2027'6"

$V_{hdr} = 144.8\ ft^3$

$T_{time=0} = 100°F$

$V_{time\ 1/2\ drained} = 51.85\ ft^3$

$T_{time\ 1/2\ drained} = 230.6°F$

$T_{half1} = .5(100 + 230.6) = 165.3°F$

$V_{time\ empty} = 103.7\ ft^3$

$T_{time\ empty} = 243°F$

$T_{half2}\ .5(230.6 + 243) = 236.8°F$

$V_{hx} = 35.48\ ft^3$

$x_{half1} = x_{half2}$

$= 35.48/144.8/4 = .061$

$x_{hdr} = (144.8-.5(35.48))/144.8$

$x_{hdr} = 0.877$

$T_{mix} = x_{half1}T_{half1} + x_{half2}T_{half2} + x_{hdr}T_{hdr}$

$T_{mix} = .061(165.3 + 236.8) + .877(100)$

$T_{mix} = 112°F$

For conservatism add 10°F

$T_{mix} = 122°F$ for each train

The lowest system pressure upon pump restart is 5.7 psia from Train B Case 2.

The saturation temperature at 5.7 psia is 168°F. The saturation pressure at $T_{mix} = 122°F$ is 1.8 psia. There is no pressure drop of significance between the column and the heater to cause the pressure to fall to 1.8 psia and so flashing will not occur in the piping upstream of the cooler.

(2) The leading edge of the water entering the tubes from the supply side following pump restart will flash to steam since the tubes will be empty and the tubes will be at the containment temperature. The two phase flow downstream of the advancing water will cause increased resistance in the cooler. The water pressure will increase to accommodate the increased resistance and continually displace steam in the cooler as the tubes are filled. If a choked steam flow condition exists in the tubes, the water pressure will increase and compress the steam void. The water will progress through the tubes as steaming occurs at the leading edge of the water. The progression of water through the tubes will not be significantly affected by choked steam flow in the tubes based on the following:

Reference [21] indicates that the advancing water column closes at a velocity of 10.32 feet/sec. This water will flash when it enters the hot tubes. Assuming this water immediately flashes to steam at the void pressure then the resultant volumetric flow rate would be:

$m = 10.32 ft/sec(60 lb/ft^3)(.5475 ft^2) = 339 lb/sec$
$V_{stm} = 339 lb/sec(65 ft^3/lb) = 22035\ ft^3/sec$
since an area of only $0.65 ft^2$ (384 qty 5/8" .035" wall tubes) is available for flow and the sonic velocity for steam at these conditions is approximately 1400 ft/sec [23], the flow would choke. The system resistance would increase and cause the system pressure to increase. The increase in system pressure would cause the steam specific volume to decrease and allow more water to progress into the cooler and fill the cooler. This process would happen in a fraction of second more than the amount of time it would take the cooler to fill without choking:

$V_{hx} = 35.5\ ft^3$
at $339\ lb/sec(ft^3/60 lb) = 6\ ft^3/sec = 2536 gpm$
time = 35.5/6.0 = 5.9 seconds without steam flashing

with flow choked at 1400 ft/sec
$V_{stm} = 1400\ ft/sec(.65 ft^2)(lb/65\ ft^3) = 14\ lb/sec$
$Q = 14\ lb/sec(7.48)(60)(ft^3/60 lb) = 104.5\ gpm$
Since the system pressure will increase and reduce the steam specific volume, filling of the cooler will occur. The maximum rate at which the filling would be reduced corresponds to the choking flow rate:
$Q_{new} = 2536 gpm - 104.5 gpm = 2432 gpm = 5.42\ ft^3/sec$
time = 35.5/5.42 = 6.6 seconds with steam flashing

The 7/10's of a second difference is insignificant and flow limitation is therefore not a concern.

(3) The significant resistance in the piping downstream of the coolers prior to the 30" header is the 14" orifices. System and local pressures in the piping downstream of the coolers stays above its saturation point upon restart of the pumps and two phase flow conditions are avoided in the piping upstream of the 30" header based on the following:

Flow orifice throat = d_1 = 6.9" [11]
Pipe I.D. = d_2 = 13.124"
Pipe velocity = V_2 = 2(10.32) = 20.46 ft/sec [21]

$$\textit{Therefore } V_1 = 20.46\ \frac{(13.124)^2}{(6.9)^2} = 74.02\ ft/\text{sec}$$

$$\frac{V_1^2}{2g} = 86.4\ ft\ H_2O$$

$$\frac{V_2^2}{2g} = 1.65\ ft\ H_2O$$

Pressure drop due to velocity change:

ΔP_V = 86.4 - 1.65 = 85 ft H_2O = 36.7 psi

Pressure in 30" header downstream is P_{HDR} = 72.8 PSIG [3]
30" HDR EL. = 1979'6" [8]
ORIFICE EL. = 2001'6" [8]

Conservatively neglecting frictional losses, the orifice discharge pressure is:

$$P_{OR} = 72.8 - \frac{(2001.5\ -\ 1979.5)}{2.31} = 62\ PSIG = 77\ PSIA$$

The pressure at the throat is then P_{TH} = P_{OR} - ΔP_V = 77 - 36.7 = 40.3 PSIA.

Conservatively neglecting any mixing, the hottest water is 243°F with a saturation pressure of $P_{SAT} = 27$ PSIA.

$P_{SAT} < P_{TH}$ therefore the flow will not flash.

There are no other flow restrictions as significant as the orifice and two phase flow conditions will not occur downstream of the coolers.

(4) The significant resistance in the 30" return header is the UHS restricting orifices. System and local pressures stay above the saturation pressure and two phase flow is avoided in the 30" header based on the following:

Flow orifice throat = d_1 = 11.375" [27]
Pipe ID = 29.25"
Pipe flow rate = 13576 GPM = 30.25 Ft^3/Sec [3]
Pipe flow area = 4.67 ft^2 [9]
Pipe velocity = V_2 = 30.25/4.67 = 6.48 ft/sec

$$V_1 = 6.48 \frac{(29.25)^2}{(11.375)^2} = 42.8\ ft/\text{sec}$$

$$\frac{V_1^2}{2g} = 28.5\ ft\ H_2O$$

$$\frac{V_2^2}{2g} = 0.7\ ft\ H_2O$$

Pressure drop due to velocity change:

$\Delta P_V = 28.5 - 0.7 = 27.8$ ft H_2O = 12 psi

Pressure in 30" HDR downstream is P_{HDR} = 15.2 PSIG = 29.9 PSIA.

The pressure at the orifice throat is then $P_{TH} = P_{HDR} - \Delta P_V = 29.9 - 12 = 17.9$ PSIA.

The maximum LOCA temp in the 30" HDR is 170°F per [3]. This temperature is conservatively increased to account for the hot containment cooler discharge. Using a maximum temperature of 243°F and a flow rate of 2000 GPM, the header temperature is then:

$$T_{HDR} = \frac{2000(243) + 11576(170)}{13576}$$

$$T_{HDR} = 181^{\circ}F \textit{ with a saturation pressure, } P_{SAT} = 7.7 \; PSIA$$

$P_{SAT} << P_{TH}$ therefore the flow will not flash.

There are no other flow restrictions as significant as the orifice and two phase flow conditions will not occur in the 30" return header.

Since flashing flow conditions will not occur and limit flow, only void closure is required to establish design flow rates. The time required to close the void is calculated as follows:

Cooler Volume = V_{HX} = 103.7 ft^3
2027'6" Header Void Volume = V_{HDR} = 380-265 = 115 ft^3
(For train "B" case 2 which has largest void)

Piping Volume between Coolers and Header
V_{other} = 18.71 + 61.96 + 78.84 = 159.5 ft^3
(From Section 5.3)

Void Volume = $V_{VOID} = V_{HX} + V_{HDR} + V_{OTHER}$ = 378.2 ft^3

From Reference [21] the forward and reverse direction closure velocities for Wolf Creek are:

V_{FOR} = 10.32 ft/sec for 10" SCH 40 pipe
V_{REV} = 1.9 ft/sec for 10" SCH 40 pipe

The flow area for 10" SCH 40 piping is 0.5475 ft^2 per [9].

These velocities are for each cooler, the total flows for the train are then:

Q_{FOR} = 2(10.32)(.5475) = 11.3 ft^3/sec.

Q_{REV} = 2(1.9)(.5475) = 2.1 ft^3/sec.

The time to fill the coolers is then:

$t_{fill} = t_{restart} + (1/Q_{FOR})(1/2V_{HDR} + 1/2\ V_{OTHER} + V_{HX})$
Where $t_{restart}$ = 37 seconds when the pumps restart.

$t_{fill} = 37 + (1/11.3)(1/2(115) + 1/2(159.5) + 103.7)$

$t_{fill} = 58.3$ sec

The time to close the void completely is:

$$t_{CLOSE} = t_{RESTART} + \frac{V_{VOID}}{Q_{FOR} + Q_{REV}} = 37 + \frac{378.3}{11.3 + 2.1} = 65.2\ sec$$

Once the coolers are filled (prior to the closing of the void in the discharge piping), the heat removal rate will be initially higher than normal. This is due to higher service water flow velo ity that will exist until the full void is closed on the discharge side and the initially higher tubing temperatures. The water is flowing at a rate greater than normal because there is less system resistance while the void is closing. The flow velocity through the tubes is 8.8 ft/sec prior to final void closure (11.3 ft^3/sec through two coolers, 5.65 ft^3/sec through one cooler with a flow area of 0.645 ft^2) versus a normal flow velocity of 3.5 ft/sec (1000 gpm, or 2.3 ft^3/sec per cooler). The increased flow rate results in a higher tubeside velocity and greater heat removal capability. The tubing temperatures are greater than normal because the coolers have been exposed to a hot containment atmosphere without service water flowing to remove heat. The heat transfer will exceed the normal heat transfer until the void is closed. When the void is closed (at 65.2 seconds), the heat transfer will become the normal heat transfer since the velocity will go down to the normal system flow and the tubes will have reached a steady state temperature.

The basic heat transfer equation demonstrates the improved heat removal capability further:

$$Q = m*Cp*dT$$

where m = mass flow rate
Cp = specific heat of water
dT = water temperature change

When the cooler is first filled, the mass flow rate and temperature rise is greater than design basis conditions. Changes in the specific heat of water are insignificant for the temperature ranges of concern. As a result, the heat transfer rate Q (in BTU/hr) will be greater prior to final column closure than for design basis conditions.

Although final column closure will occurs at 65.2 seconds, the CFCs are capable of providing design basis heat removal prior to 60 seconds.

6.0 LOOP TEST EVALUATION

The 1991 tests of the SW system were conducted from 11/12/91 to 11/14/91. These tests included multiple system line-ups to simulate several operating conditions. The individual tests are identified by the step number of the test procedures [STS KJ-001B] utilized to obtain the data. A copy of the memo describing the test and the test results is included in Appendix E.

The primary difference between the portions of the test pertains to the cross tie from the non-safety and essential service water (ESW) pumps. During a LOOP, the ESW pumps lose power and the system depressurizes, causing the fan coolers to void which, in turn, causes column closure waterhammer. If the non-safety service water pumps are cross-tied to the ESW pump discharge and remain on, the system does not depressurize when the ESW pumps are shut off, thereby preventing voids in the fan coolers.

The data obtained during test step 5.2.19 provides a simulation of the LOOP with an SI Signal generated during a LOCA. This test most closely represents the system configuration for which this analysis is performed.

This representative test provides a peak pressure pulse of approximately 205 psig for column reclosure waterhammer. This is in excellent agreement with predicted column closure waterhammer pulses of 193 psig and 225 psig for the "B" and "A" trains respectively.

7.0 FLUID STRUCTURE INTERACTION

Fluid/Structure interactions have been demonstrated to increase piping/support loads in some test environments [30]. These effects are predominant in thinner walled pipes and is not expected to be a concern considering the assessment shown in Appendix F.

8.0 LOOP/LOCA ANALYSIS CONCLUSIONS & RECOMMENDATIONS

As described in this report, two kinds of waterhammer are anticipated. One is the column closure which will not be more severe than the LOOP without LOCA waterhammer. The other is the trapping and condensing of steam during the draining phase. Calculations indicate that both waterhammer pressure pulses are acceptable as reported in Altran Technical Report 96227-TR-03 [29].

9.0 REFERENCES

[1] FSAR-Wolf Creek

[2] Altran Corporation Report No. 94100-C-02, "Structural Dynamic Analysis of ESW System Waterhammer", Rev. 1.

[3] WOLF CREEK SYSTEM FLOW DIAGRAM ESSENTIAL SERVICE WATER, M-11EF01, Rev. 6.

[4] IST Valve data sheets - included in Appendix D

[5] Telecon Matt W. Zweigle with Bill Selbe, 11/21/96, included in Appendix D,

[6] Cooling Coil Data Sheet - included in Appendix D

[7] WCNOC calculation, "Engineered Safeguards Parameters For MAAP Input File", SA-90-018, Rev. 0.

[8] Wolf Creek Piping Isometric Drawings

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M-13EF01 Rev. 6
M-13EF02 Rev. 8
M-13EF03 Rev. 13
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[19] WCNOC Calculation EF-S-011 Rev.0, "Qualify the Existing ESW Piping and Its Supports for Waterhammer Loads".

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[22] Byron Jackson Pump Curve for WCNOC ESW Pumps, included in Appendix D.

[23] ASME Steam Tables, 5th Edition.

[24] ADLPIPE, Static & Dynamic Pipe Stress Analysis, Version 4F7B, Research Engineers, Yorba Linda CA.

[25] ASME, Section III, Subsection NC Div. 1, 1974 with Addendum through 1976.

[26] ASME, Section III, Subsection NF 1974 Addendum through 1976.

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[28] Altran Calculation No. 96227-C-01, "Structural Qualification of CFC Discharge Line to a simultaneous LOCA and LOOP Event". Rev. 0

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[30] Wiggert, Otwell, and Hatfield, "The Effect of Elbow Restraint on Pressure Transients," Transactions of the ASME, Volume 107, September 1985.

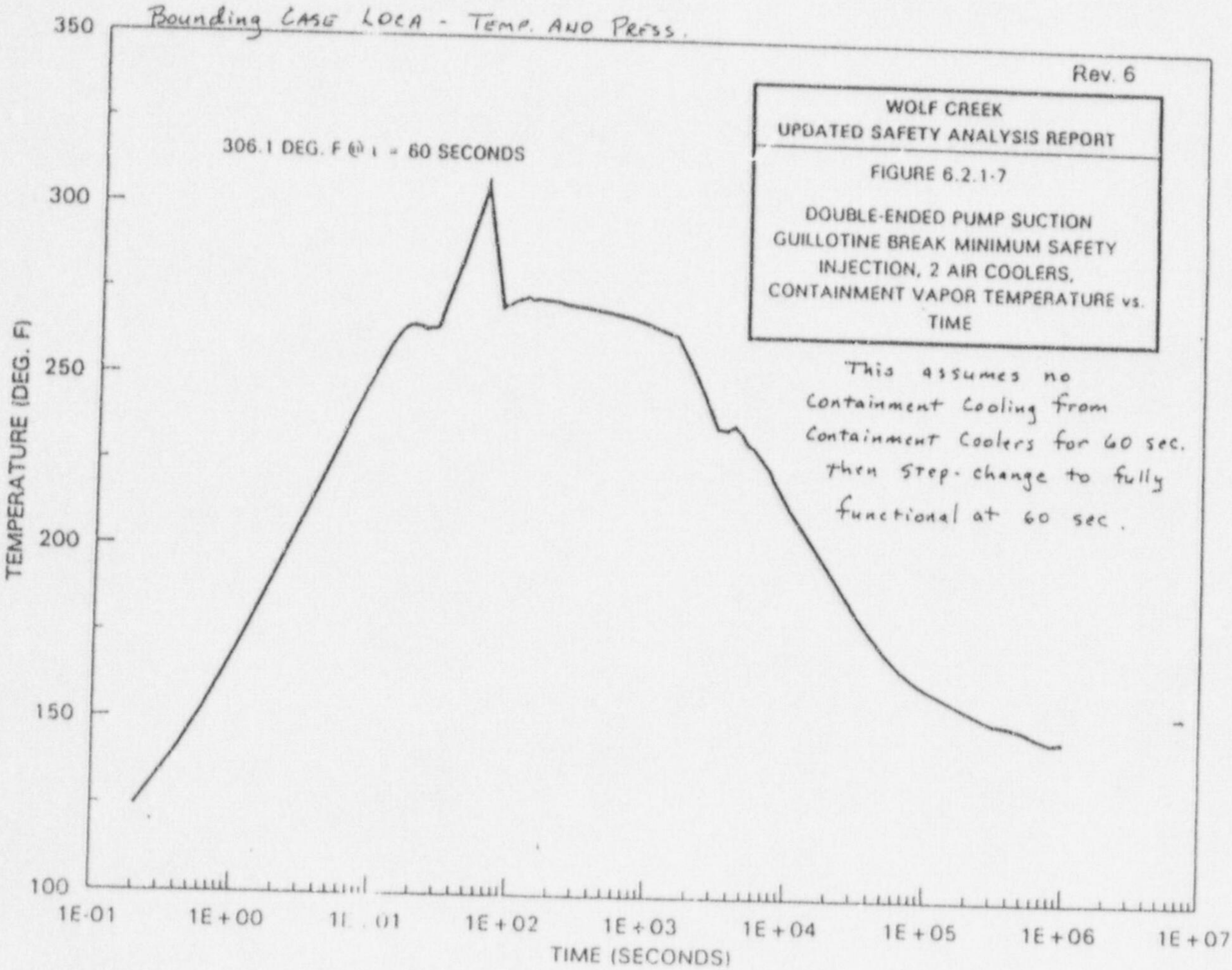

Figure 1 - Containment Temperature Profile

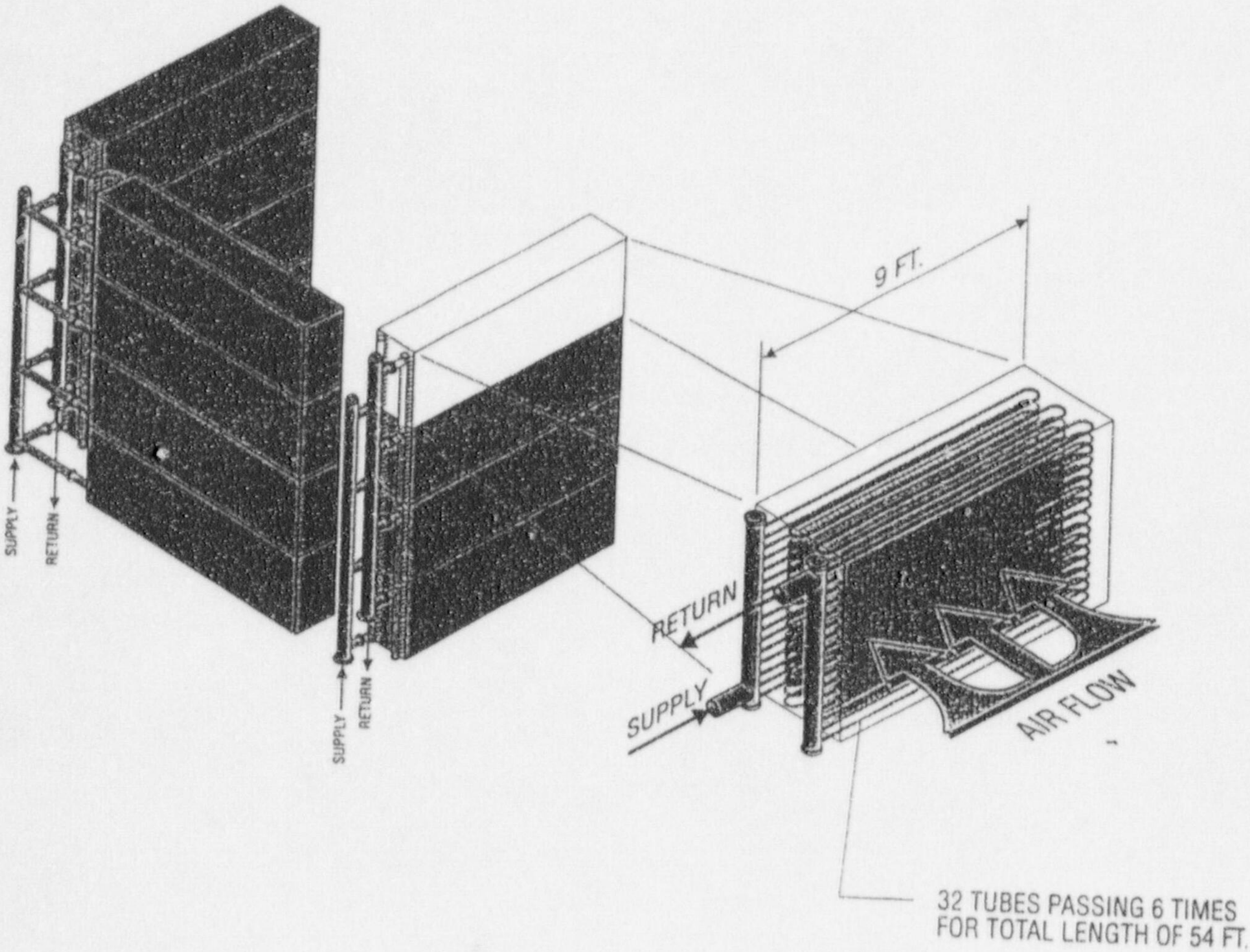

Figure 2 - Containment Cooler Configuration

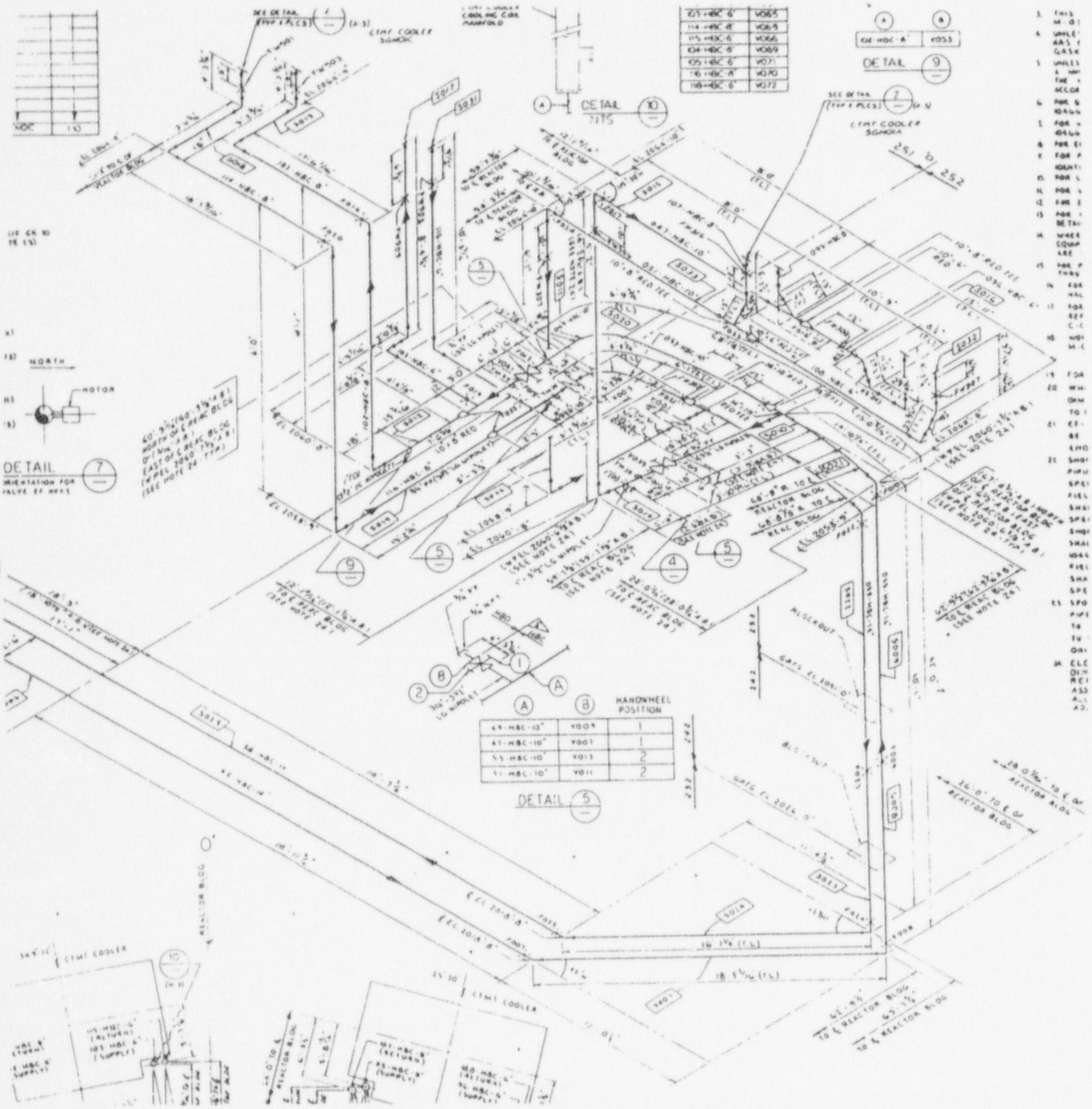

Figure 3A - A Train, A & C Cooler, Piping Configuration Inside Containment

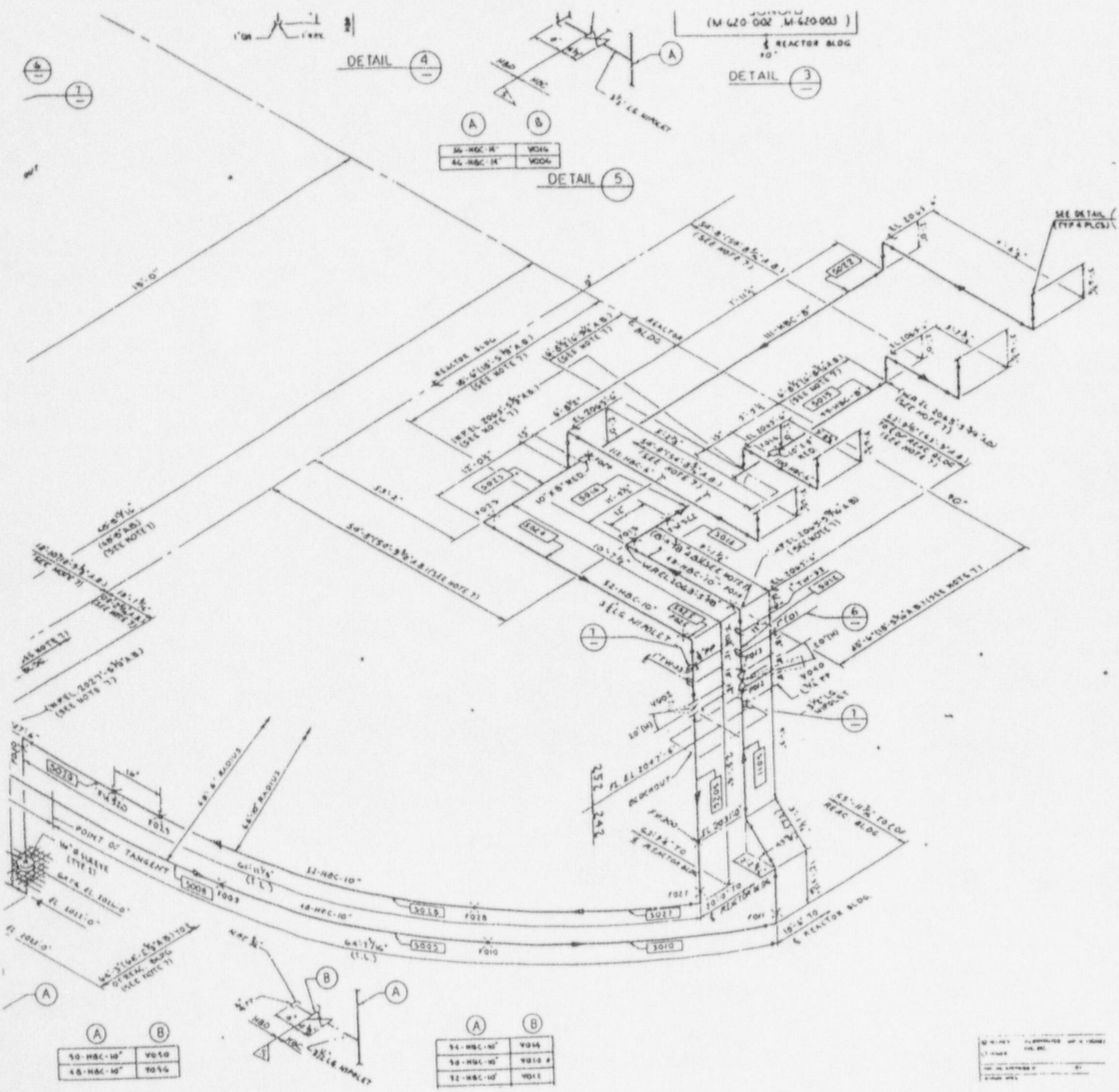

Figure 3B - B Train, B Cooler, Piping Configuration Inside Containment

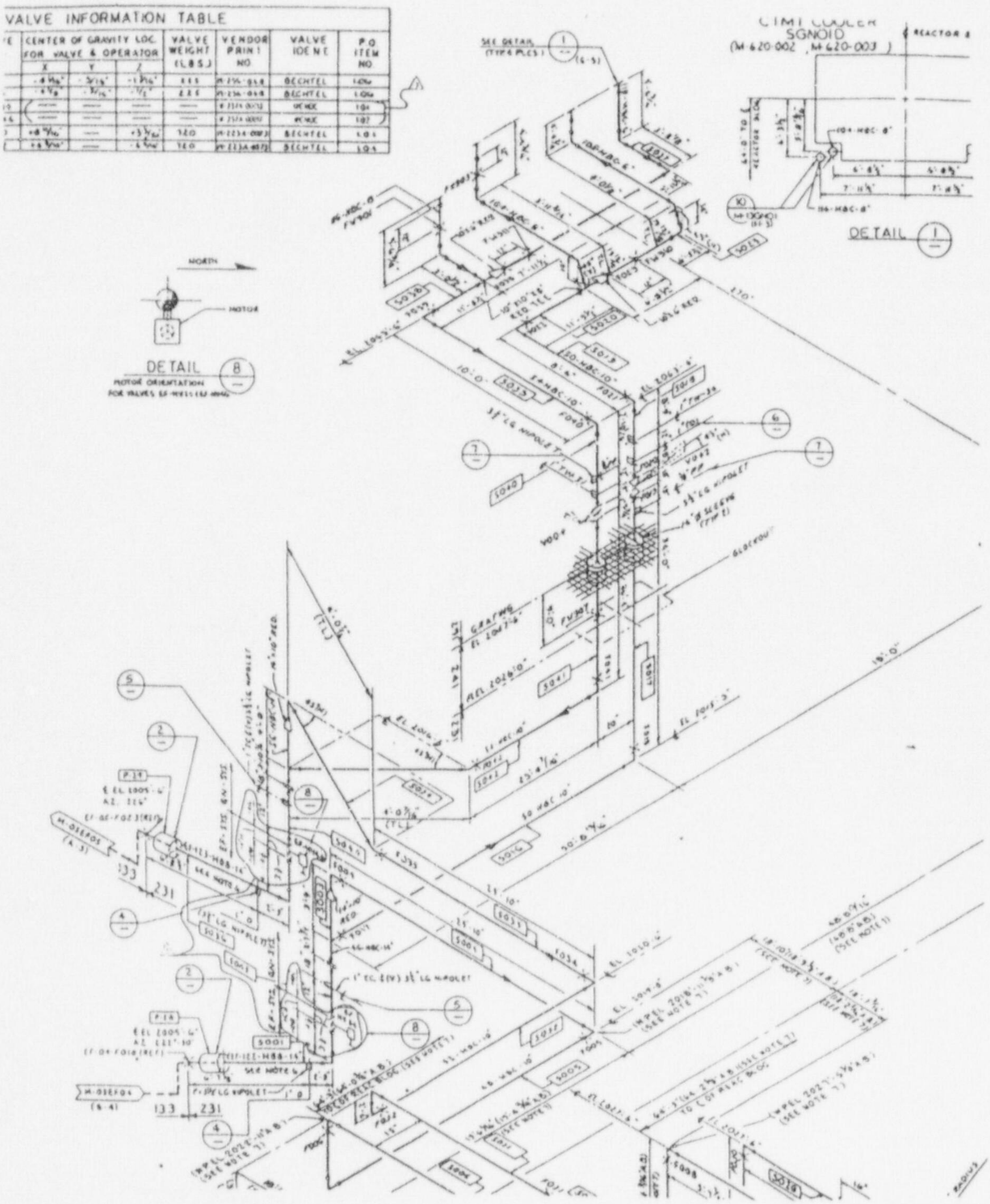

Figure 3C - B Train, D Cooler, Piping Configuration Inside Containment

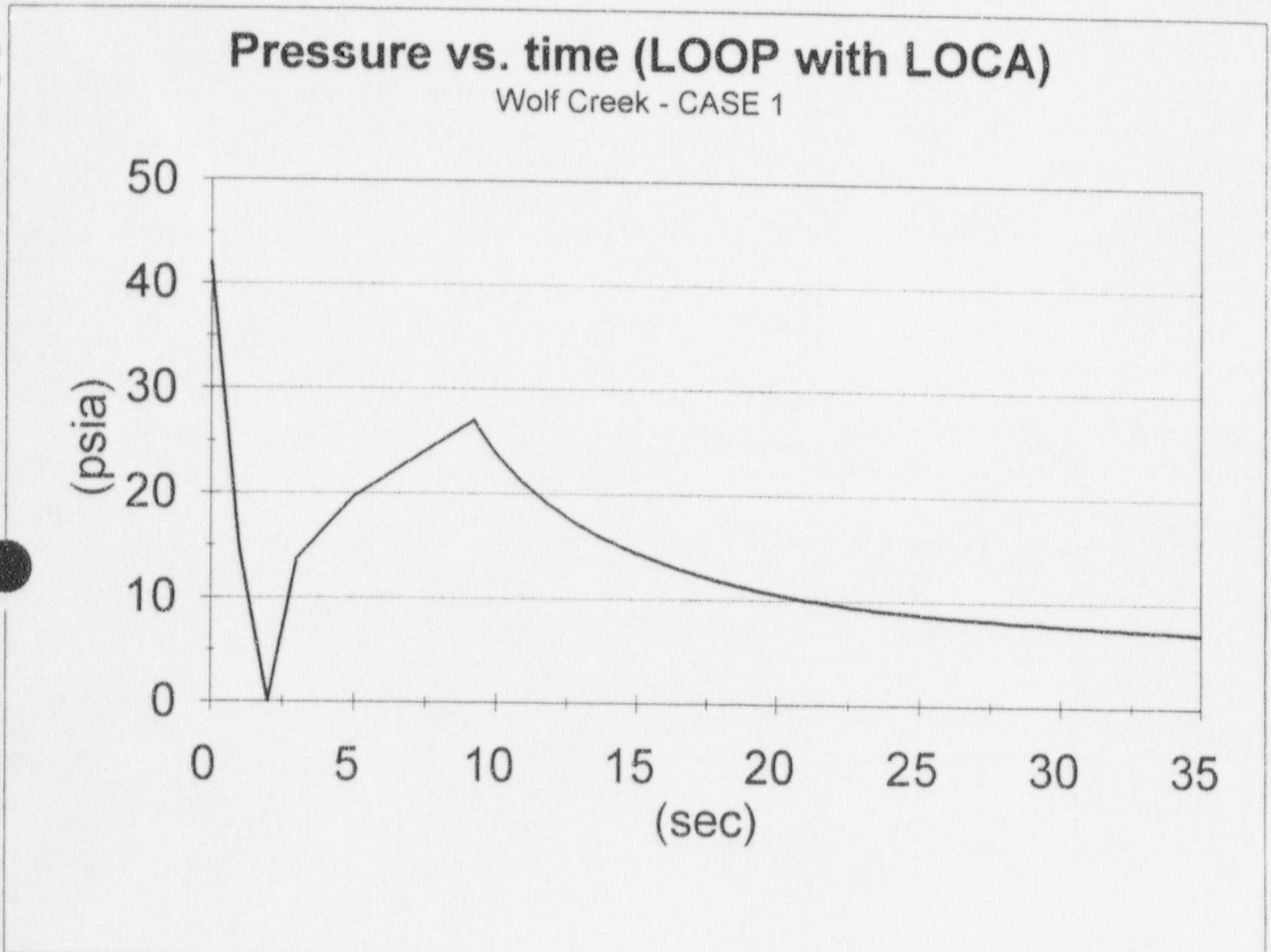

Figure 4A - Case 1 - Train A Pressure (SW supply isolated)

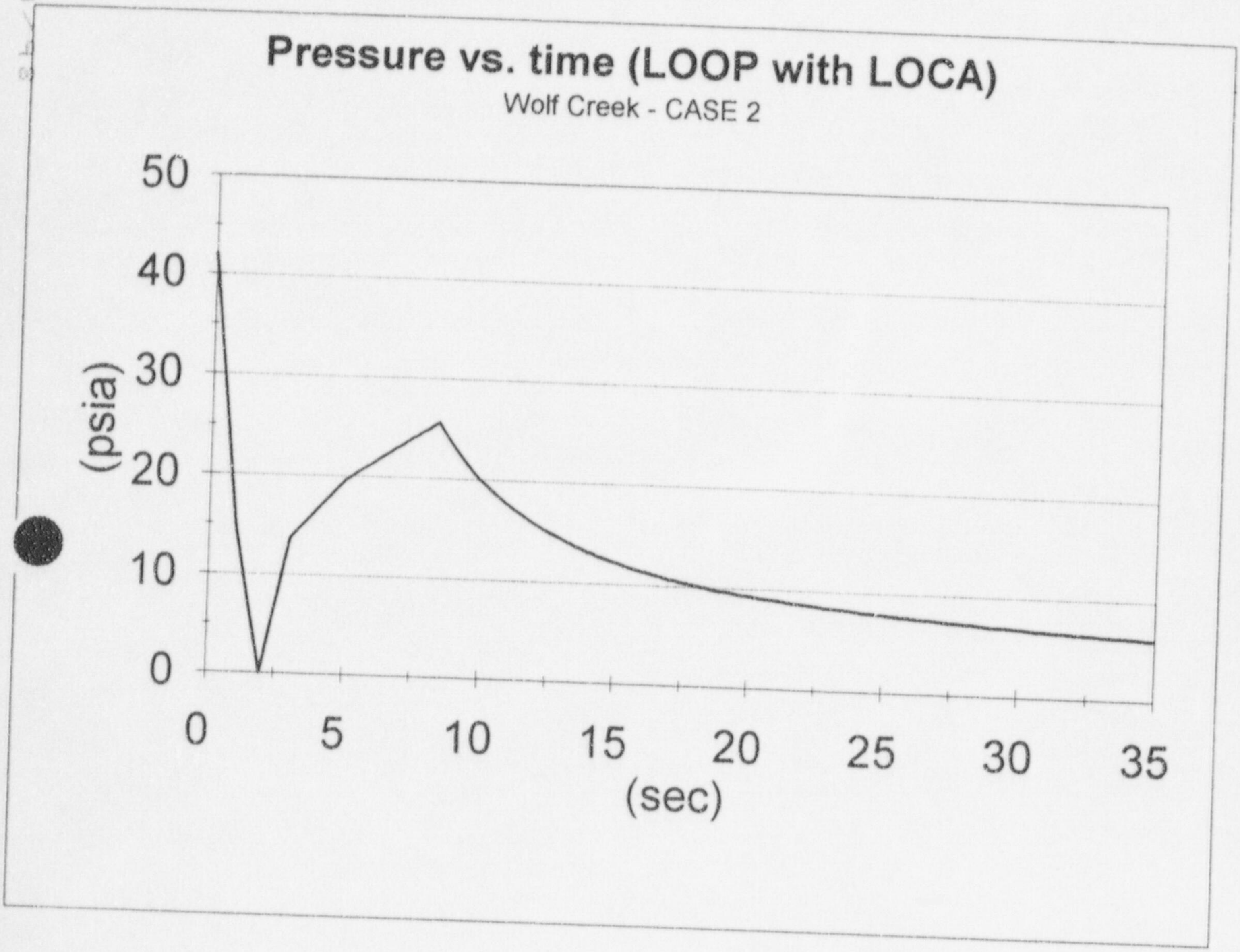

Figure 4B - Case 2 - Train B Pressure (SW supply open)

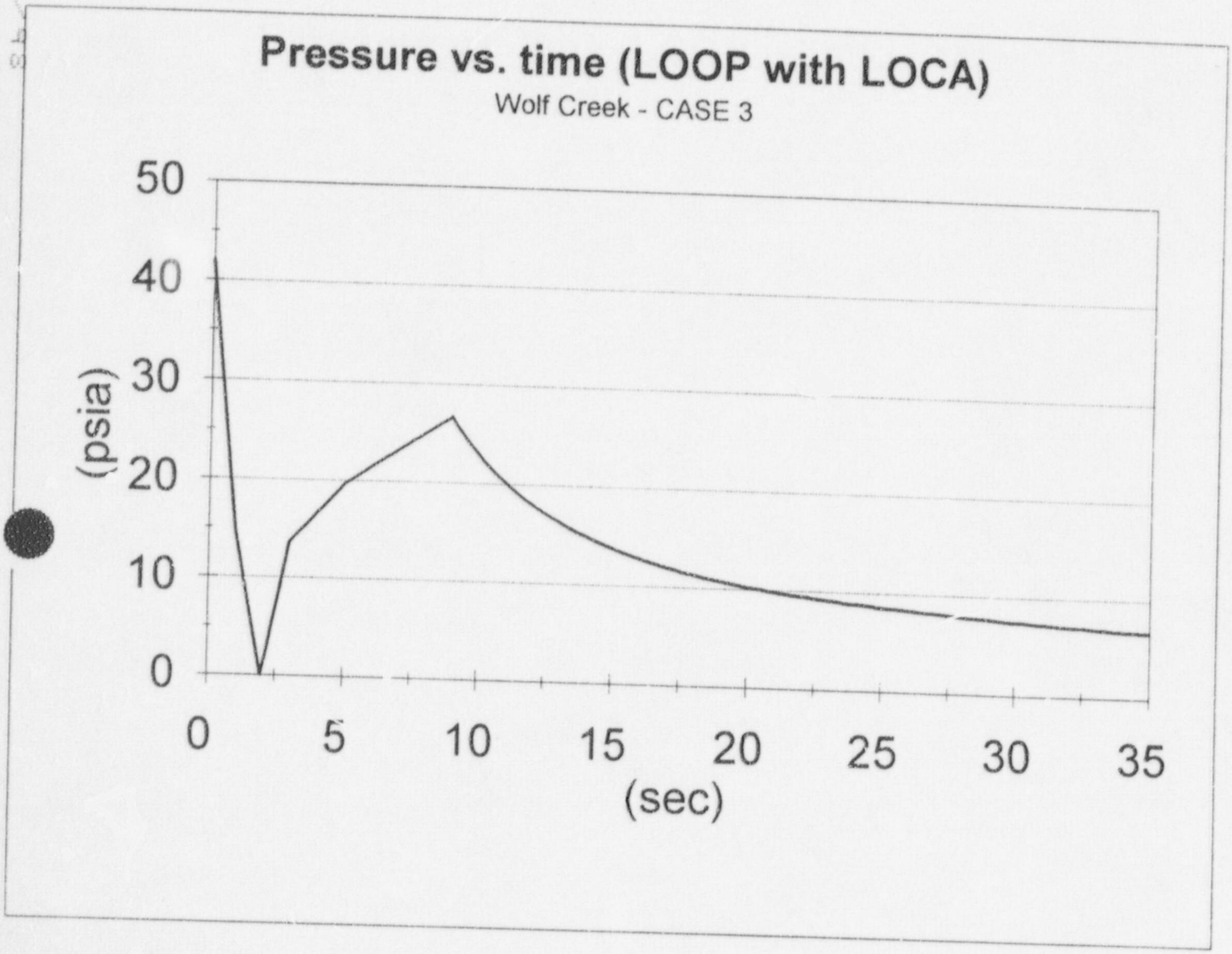

Figure 4C - Case 3 - Train B Pressure (SW supply isolated)

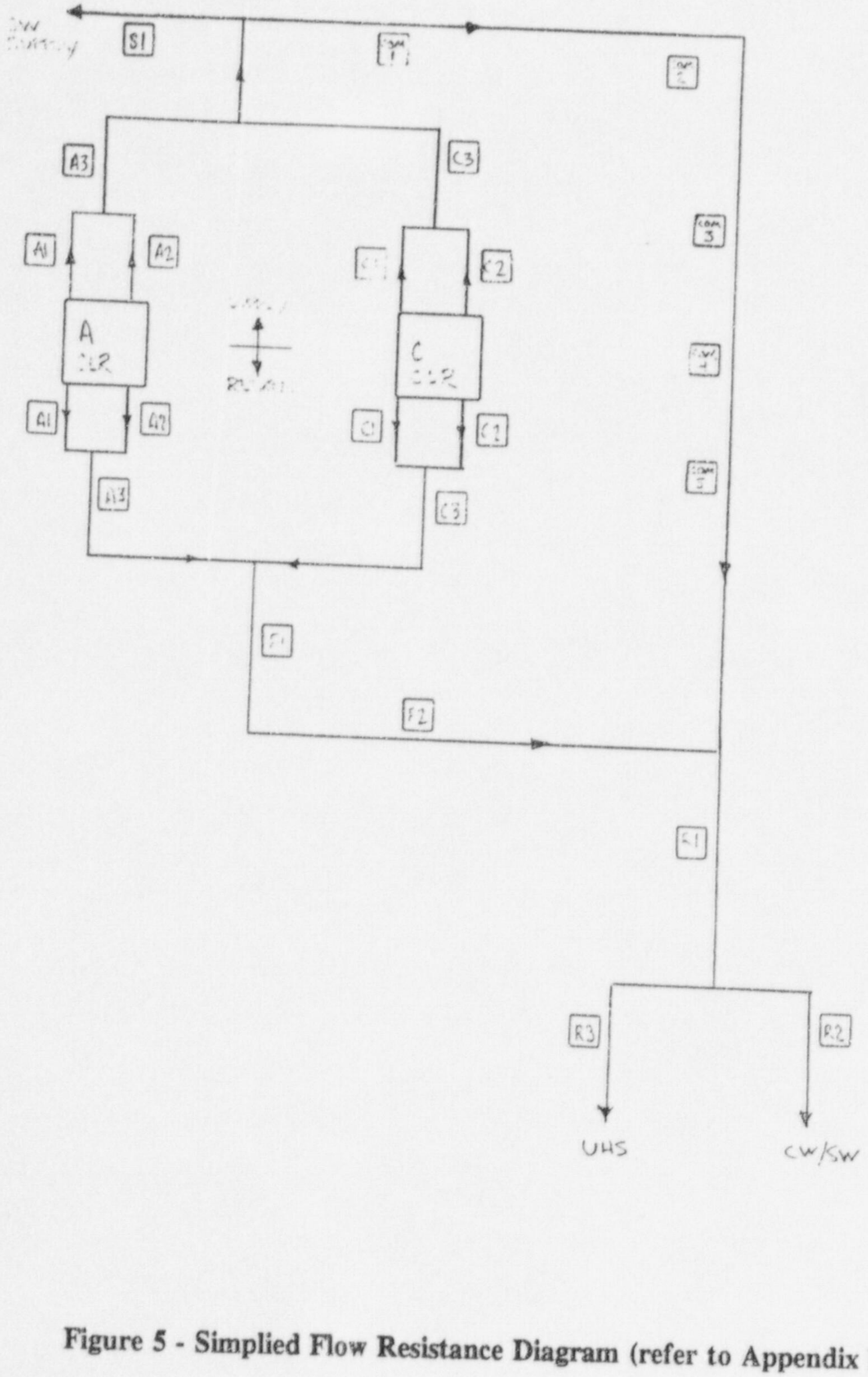

Figure 5 - Simplied Flow Resistance Diagram (refer to Appendix B)

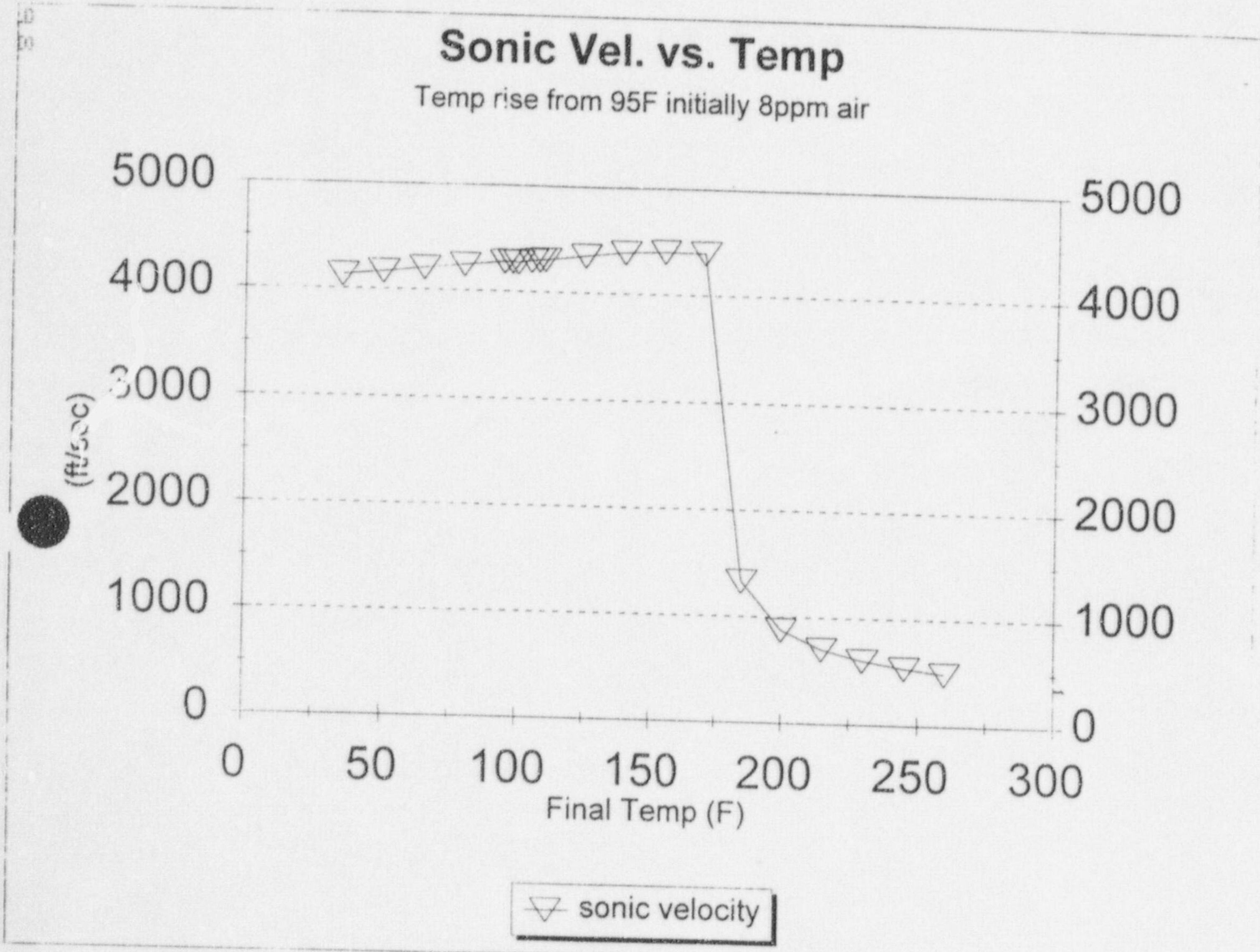

Figure 6 - Sonic Velocity Change With Temperature Rise

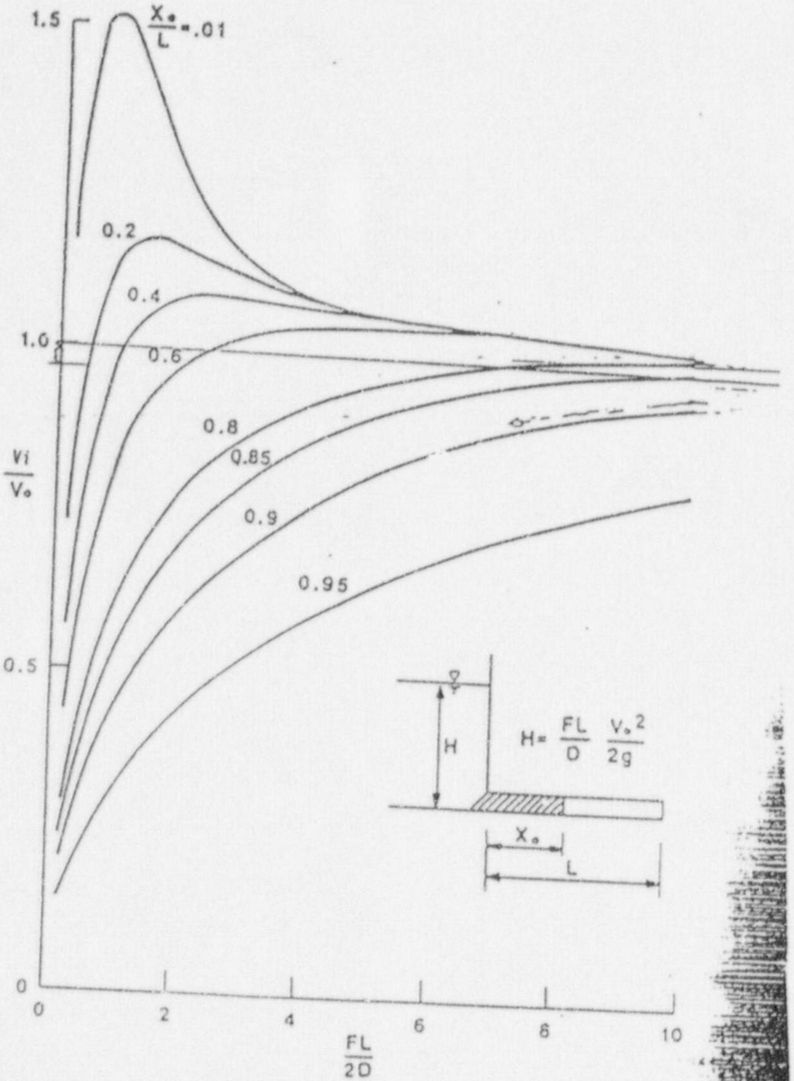

Figure 9-2 Impact Velocity for Filling a Voided Line Liquid Column Originating from Reservoir

Figure 7 - Impact Velocity Curve from Reference 17

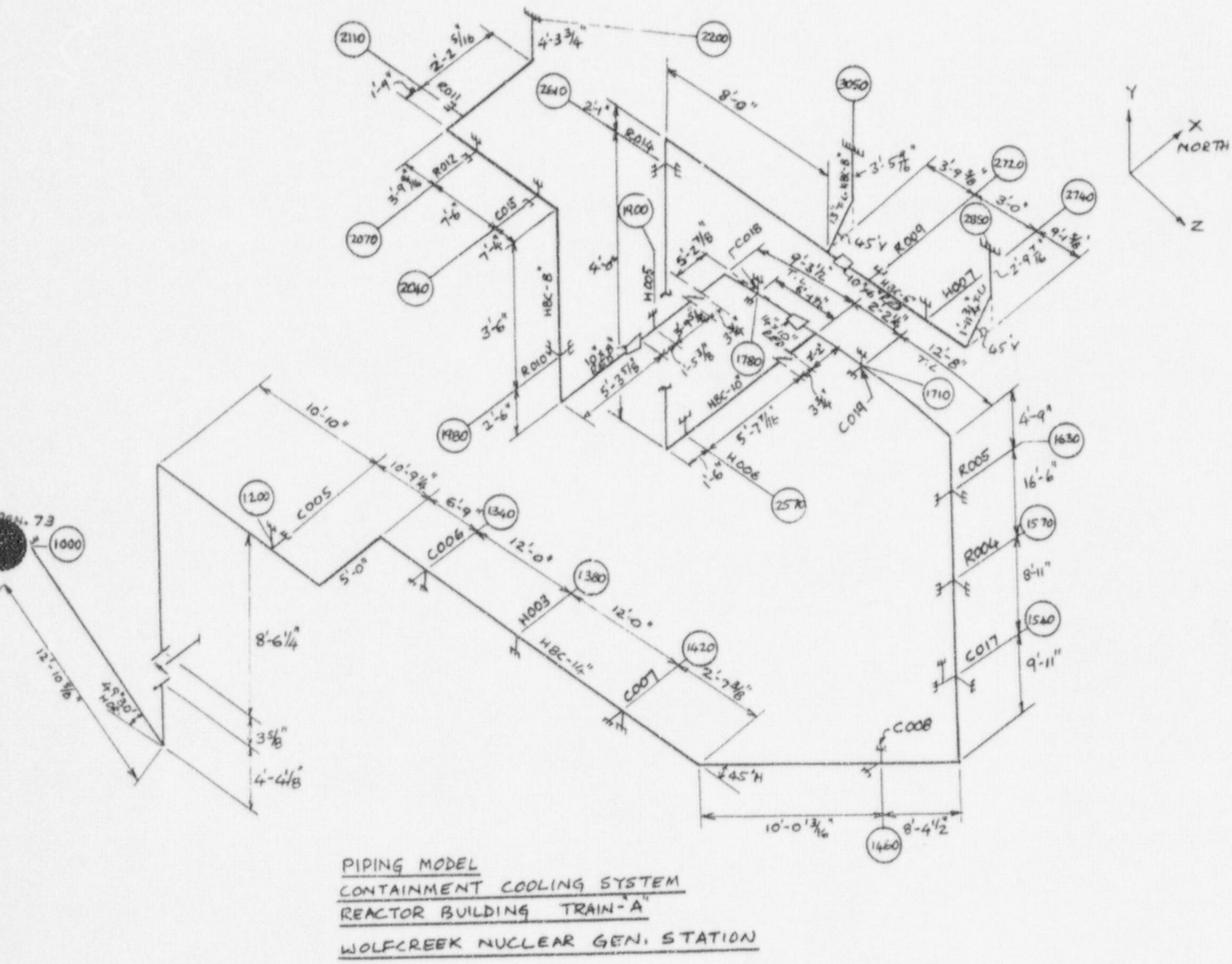

Figure 8 - Train A ADLPIPE Structural Assessment Model

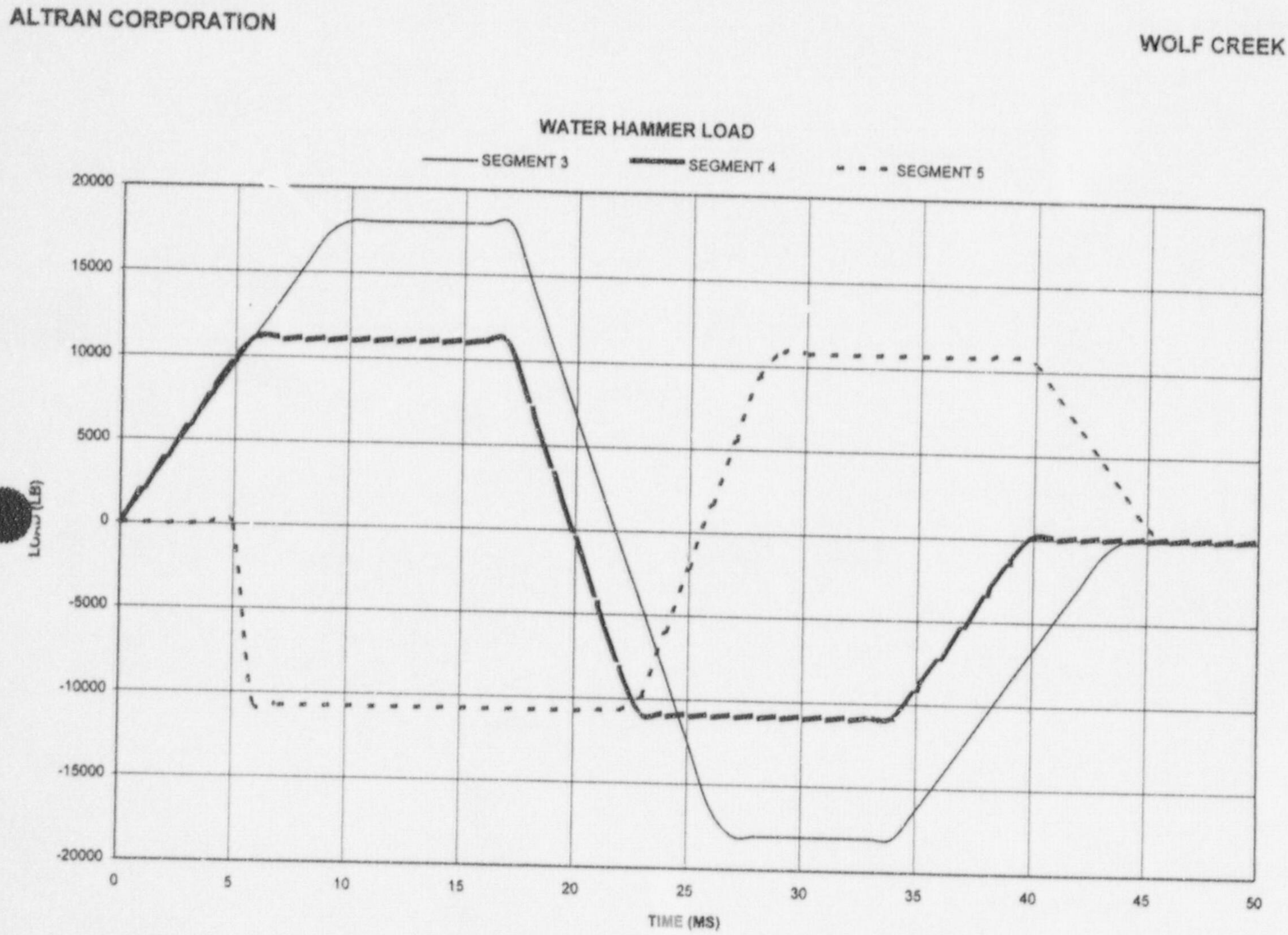

Figure 9 - ADLPIPE Structural Model Segment Forcing Functions Condensate Waterhammer

APPENDICES

The following Appendices are included as part of this evaluation:

Appendix A - Pressurization Spreadsheet for Cases 1, 2, & 3
Appendix B - System Resistance Spreadsheet for Cases 1, 2, & 3
Appendix C - Sonic Velocity Spreadsheet
Appendix D - Plant Data Including:

- IST Valve Data Sheets
- 11/21/96 Telecon: Bill Selbe (WCNOC) with Matt Zweigle (Altran)
- Containment Cooler Data Sheet

Appendix E - Test Results
Appdenix F - Fluid Structure Interaction