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AUTH. NAME AUTHOR AFFILIATION
KOBER, R. W. Rochester Gas & Electric Corp.
RECIP. NAME RECIPIENT AFFILIATION
LEAR, G. E. PWR Project Directorate 1
LEAR, G. E. Document Control Branch (Document Control Desk)

SUBJECT: Forwards addl info re NUREG-0737, Item II.D.1, "Performance Testing of Relief & Safety Valves," in response to 861211 request.

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U.S. Nuclear Regulatory Commission
Document Control Desk
Attn: Mr. George E. Lear, Chief
PWR Project Directorate No. 1
Washington, D.C. 20555

Subject: NUREG-0737 Item II.D.1, Performance Testing of Relief and Safety Valves
R. E. Ginna Nuclear Power Plant
Docket No. 50-244

Dear Mr. Lear:

Your letter dated December 11, 1986 requested additional information regarding NUREG-0737 Item II.D.1. The attachment to this letter provides the requested information.

Very truly yours,

Roger W. Kober

Roger W. Kober

Attachments

A046
1/1

RESPONSE TO NRC REQUEST FOR ADDITIONAL INFORMATION DATED DECEMBER 11, 1986

NUREG-0737 ITEM II.D.1

Question 1

The appendix of RG&E's March 4, 1983 (Reference 1) submittal showed the calculation of the inlet piping pressure drop for the Ginna safety valves. The pipe length used in the calculation of the acoustic wave pressue drop was 3.3 ft (from Table A-1 of the appendix). Figure 2-2 in the main body of the same report, however, shows the safety valve inlet piping length to be at least 4.93 ft. This is not the total piping length since the length of at least one portion of the piping was not shown. The additional length could make a significant difference in the calculated acoustic wave pressure drop. Also the calculated flow pressure drop in the same appendix did not account for the 90° bend when calculating the L/D term. Provide the corrected pressure drop with this bend accounted for using the equation found in Rev. 2 of the EPRI Submittal Guide (Reference 2).

If the recalculated inlet pressure drop for the Ginna safety valves exceed the pressure drop for the 3K6 test valves, it will be necessary for the ring settings of the Ginna valves to be adjusted so that the valves operate stably.

Response

A review of the pressure drop calculation did uncover discrepancies between the calculation model and the as-built piping configuration. A revised pressure drop calculation has been performed in Attachment 1, using the procedure found in Appendix B of the "EPRI PWR Safety and Relief Valve Test Program Guide for Application of Valve Test Program Results to Plant-Specific Evaluations", Revision 2.

Each inlet line is analyzed separately, as the pipe lengths and consequently the pressure drop results differ between the valves.

The results of the analysis show that Safety Valve 434 has inlet piping pressure drops below the pressure drop for the 3K6 EPRI test valve for closing and is 14.3% above the test valve pressure on opening. Safety valve 435 has inlet piping pressure drops below the pressure drop for the 3K6 valve on closing, and is 14.6% above the 3K6 valve on opening.

The Ginna Station Safety Valves are therefore, bounded by the EPRI 3K6 valve for closing and exceed the 3K6 pressure drop for opening. Exceeding the pressure drop on opening is acceptable because opening operation of the 3K6 valve during

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

the EPRI tests was stable with minor fluctuations during loop seal discharge. These fluctuations are expected and are considered normal operation for this type valve. No tests were terminated due to problems on opening. For this reason, the Ginna 4K26 valves are expected to operate stably without adjustment and with no operational consequences resulting from the less than 15% increase in inlet piping pressure drop.

Question 2

The response to question 7 of our request for information (Reference 3) on the qualification of the PORV control circuitry was not sufficient to qualify the control circuitry under NUREG-0737. The response also stated the control circuitry was not included in the 10CFR50.49 review. It is the staff position that the PORV control circuitry be qualified to perform its required function for any potential environment that it may be exposed to. If any exception is taken with respect to qualification to the harsh environment for equipment important to safety, the licensee must demonstrate that the equipment is not required to perform a safety function to mitigate the effects of any design basis accident when exposed to the environment caused by the accident, and any equipment failure in any mode in the harsh environment will not adversely impact safety function or mislead the operator.

Response

The pressurizer PORVs are not required to perform any safety function to mitigate the effects of any design basis accident when exposed to a harsh environment. The transients that do cause a harsh environment, such as LOCAs, steam line breaks, feedwater line breaks or other high energy line breaks do not require operation of the PORVs to terminate the transient or mitigate the consequences of the event. Therefore, the PORVs are not required to be qualified for operation in a harsh environment.

The PORVs must also not inadvertently actuate when exposed to a harsh environment. The PORV is a spring closed valve that requires air to open. If no air is supplied to the PORV the PORV will not open, even when exposed to a harsh environment. Air is supplied to the PORV by an ASCO solenoid valve. When the solenoid valve is energized, air is supplied to the PORV causing the PORV to open. When the solenoid valve is deenergized air is vented and the PORV closes. The ASCO valve is an environmentally qualified valve, which should not be caused to be inadvertently energized by a harsh environment. The control system connected to the ASCO solenoid

X 120

valve is a two-wire ungrounded DC system. It has been established and agreed to by the NRC (Reference 4) that there exists no credible "hot short" failure which need to be postulated for the two-wire ungrounded DC system used for Ginna control circuitry, such as the solenoid valves. Without a hot short the solenoid valve cannot be energized. Since a harsh environment cannot cause the PORV itself to inadvertently operate, the ASCO solenoid which is qualified for a harsh environment will not inadvertently operate, and there is no potential for environmental qualification-related failures in the electrical system portion of the solenoid valves, it has been concluded that the PORVs will not inadvertently open due to a harsh environment.

Question 3

In the Pressurizer Safety and Relief Line Evaluation Report, which was included as a part of the submittal (Reference 1), the two piping discharge conditions used in the thermal hydraulic analysis were identified as the simultaneous opening of either the two safety valves or the two PORVs. The report did not provide specific information which is needed to define the loading cases. Please give the assumed fluid parameters used for the analysis of each piping discharge case such as the fluid state, peak pressure, temperature, pressurization rate, and valve flow areas and justify that these loading conditions would produce the maximum loads on the piping system. In particular, provide additional information demonstrating the steam discharge case analyzed for the PORV results in piping forces that bound the liquid discharge case. Also give the analysis parameters such as the valve opening time, node spacing, and time step size used for the calculations.

Response

Various fluid transient analyses were performed for the pressurizer safety and relief valve piping system. Operation of the safety valves at power operation and actuation of the relief valves to mitigate cold overpressurization were cases evaluated. In general, the two safety valves opening simultaneously and discharging without PORV flow and the two PORVs opening simultaneously without safety valve flow are the limiting design cases. Typically, the worst case valve discharge case (SOT_F) is the double safety valve discharge transient for the safety valve piping, including the inlet, outlet and common region piping and the double relief valve discharge transient for the relief valve inlet and outlet piping. The initial conditions for the safety valve water slug discharge event included:

[illegible]

[illegible]

[illegible]

P (Upstream) = 2575 psia

h (Steam, Upstream) = 1110 Btu/lb

h (Water, Upstream) = Enthalpy based upon a temperature profile consistent with EPRI safety valve discharge test #917, i.e., approximately 300^{o}F at the valve inlet and saturation temperature at the steam-water interface

P (Downstream) = 14.7 psia

The pressurizer conditions were held constant for the transient at 2575 psia and 1110 Btu/lb.

According to the results of the EPRI tests, high frequency pressure oscillations of 170-260 HZ typically occur in the piping system upstream of the safety valve while the loop seal water passes through the valve. No significant bending moment during this "simmering" phase of the transient will, however, occur (see Rochester Gas and Electric Corporation letter from R.W. Kober to J.A. Zwolinski of the U.S. Nuclear Regulatory Commission dated May 24, 1985 for a discussion of this issue). The safety valve slug discharge event generates limiting loadings for the safety valve inlet, outlet and common region piping.

The initial conditions for the relief valve slug discharge case included:

P (Upstream) = 2350 psia

h (Steam, Upstream) = 1162.4 Btu/lb.

T (Water, Upstream) = 150^{o}F

P (Downstream) = 14.7 psia

For this case, small cold loop seals, each 1 foot long, were assumed to exist upstream of the valves. This is conservative as the piping layout is such that no or very little condensate will remain in the upstream relief valve piping.

The pressurizer conditions were held constant for the entire transient at 2350 psia and 1162.4 Btu/lb.

The initial conditions utilized for the water discharge cold overpressurization case included:

P (Upstream) = 550 psia

T (Water, Upstream) = $100^{o}F$

P (Downstream) = 14.7 psia

The pressurizer conditions were held constant for the transient at 550 psia and $100^{o}F$.

The peak forces in all discharge piping segments were at least 2 times larger for the relief valve slug discharge case when compared to the cold overpressurization case. The slug discharge event generates limiting loadings for the relief valve discharge piping.

The adequacy of the thermal hydraulic analyses can be verified by the comparison of analytical and test results for thermal-hydraulic loadings in safety valve discharge piping for EPRI Tests 908 and 917. In that evaluation, node spacing and time-step size were selected on the basis of stable solutions of the characteristic equations and matching of test data. The safety valve full open flow area of 0.022 ft^2 was used in the model. This area is slightly smaller than the Crosby M-orifice area of 0.025 ft^2 for the tested valve, but results in a good analytical match of the tested fully open valve flow rate. Appropriate water temperatures were used. All pertinent data, including friction factors, loss factors and flow areas were based upon representative calculations and the system layout. Modeling of the water was conducted with the water seal upstream of the valve prior to transient initiation. At time = 0+, the transient was initiated and the slug position was analytically calculated during and subsequent to valve opening.

The Ginna plant specific thermal-hydraulic analysis was conducted taking the same approach as was taken for the comparison to test data. Node spacing was picked consistent with the comparison and varied with pipe size and location. Time-step sizes were utilized consistent with values utilized in the comparison, (0.004 seconds for both slug discharge cases). Valve opening times, 0.040 seconds for the safety valves and 1.00 seconds for the relief valves, were based upon actual data. Valve flow areas (0.0193 ft^2 for the safety valves and 0.0174 ft^2 for the relief valves) were selected based upon actual valve data with appropriate margins applied to account for flow rate uncertainties. All pertinent data, including friction factors, loss factors and flow areas were based upon representative calculations and the system layout. Modeling of the water slug from a temperature profile, considering initial location and movement post-transient initiation, was consistent with the comparison study. The pressurizer pressure was held constant

[illegible]

[illegible]

through the transient at initial values. Choked flow is checked internally and automatically every time-step to ensure the proper formulation is applied at every flow path. The highest pressure at the respective valve inlet was less than or equal to the pressurizer pressure for the two slug discharge cases and the cold overpressurization valve opening case.

The valve flow areas were adjusted prior to finalizing the thermal hydraulic analyses to account for all uncertainties and tolerances in the valve flow rate. The ASME steam flow rating for the Crosby safety valves (orifice size K_2) at 2575 psia is 306,430 lb/hr. The minimum analytically determined steam flow through each of the safety valves in this analysis is greater than 375,000 lb/hr. This is equivalent to a flow of 122 percent of rated. The maximum expected steam flow through the Ginna Copes Vulcan PORVs is 210,000 lb/hr. Values greater than 275,000 lb/hr. were analytically determined. Flows greater than 130 percent of expected values were, therefore, used.

Question 4

Provide the following information on the piping and support stress analysis:

a. The submittal (Reference 1) indicated that the piping supports were analyzed in accordance with the ASME Boiler and Pressure Vessel Code, Section III, Subsection NF, but did not specify the allowable stresses used for the stress evaluation. Provide the allowable stress limits applicable to each load combination for the normal, upset, emergency, and faulted conditions.

b. Provide a comparison of the worst support stresses (or loads) with the applicable allowable stresses (including those modified) to demonstrate that the piping and supports are adequate to withstand the imposed loads.

c. Give the input parameters used in the pipe dynamic analysis such as the lumped mass spacing, solution time step, damping, and cut-off frequency, etc.

Response A

The pressurizer relief and safety valve piping supports were evaluated in accordance with the requirements set forth in ASME Section III, Subsection NF. The supports were evaluated using "Design of Linear Type Supports by Elastic Analysis" described in NF-3231.1. This subarticle refers to Appendix XVII, Article XVII-2000 for the stress limits.

[illegible]
[illegible]

[illegible]

[illegible]
[illegible]

[illegible]

The stress limits as given in this article are:

F_t = Allowable stress in tension
= 0.60 Sy eqn (1)

F_b = Allowable bending stress for square and rectangular section bars bent about their weak axis.
= 0.75 Sy eqn (14)

F_v = Allowable shear stress
= 0.4 Sy

Sy = Minimum yield strength, PSI

From Subarticle NF-3231.1, the allowable stress limits for Normal and Upset conditions are identical and as given in Appendix XVII (which are given above).

For emergency conditions, the stress limits given in XVII-2000 can be increased by one-third.

Based on 36,000 psi minimum yield strength (A-36 material), the allowable stresses are tabulated below.

Condition	F_b/(PSI)	F_t (PSI)	F_v (PSI)
Normal Upset	27,000	21,600	14,400
Emergency	35,910	28,728	19,152

For welds, the allowable stress values given in Table NF-3292.1-1 are used.

Anchor Bolts were evaluated using allowable loads based on the specific size, type and embedment depth. Expansion anchor bolts governed by IE Bulletin 79-02 used a minimum factor of safety of four, while bolts subject to ductile failure used a minimum factor of safety of two.

All supports were evaluated using "Emergency" load combinations and comparing to "Normal" allowables. One support (N-628) required use of Emergency allowables with the Emergency load case, all others did not exceed Normal allowable loads.

Response B

The worst support stresses are given in the table below. This table identifies the support, the stress ratio (actual stress/allowable stress), the limiting component with the support and the allowable used. Note that for supports where the limiting components are the anchor bolts; the stress ratio is based on the allowable after application of the applicable factor of safety.

Support	Stress Ratio	Limiting Item	Allowable Limit
N-628	0.97	Base Plate	Emergency
N-625	0.95	Weld	Normal
N-608	0.78	Anchor Bolts	Normal
PS-10	0.78	Anchor Bolts	Normal
PS-5	0.74	Anchor Bolts	Normal
PS-4	0.69	Anchor Bolts	Normal
PS-2	0.68	Anchor Bolts	Normal
PS-13	0.68	Anchor Bolts	Normal
N-629	0.68	Clamp	Normal
N-601	0.67	Anchor Bolts	Normal

Response C

The modeling approach used for the benchmarking effort was also used for the Ginna specific structural analysis. For a given segment, a mass point was chosen such that the hydrodynamic force would be applied along the axial centerline of the segment. In general, the mass point nearest the center of the segment was utilized. following is a discussion of key parameters used in the structural analyses of the dynamic events:

1. Damping - A conservative system damping of 1 percent was utilized for OBE. 2 percent was utilized for SSE and the thermal hydraulic analyses. This is much lower than the actual expected value and is below the 10 percent damping used in the structural comparison to EPRI Test 908 and 917.

2. Lumping - Lumped mass spacing was determined to ensure that all appropriate mode shapes were accurately represented as shown on Figures 6-1 and 6-2.

3. Supports - The structural supports were modeled in sufficient detail to analytically represent the system. The shock suppressors and struts were modeled by inputting a stiffness in series with the piping. Specifically calculated stiffness values were utilized. All supports were linear and a linear overall system analysis was conducted.

4. Time-Step - The integration time-step is internally determined within the structural program and is based upon convergence criteria that results in stable solutions. The largest time-step ever used could be 0.0001 second. The time-step is automatically adjusted such that the relative error of each modal coefficient is at least less than 10^{-2}.

5. Cut-off Frequency - A cut-off frequency was used to ensure that all appropriate frequencies were included. For the relief line analysis, a cut-off frequency greater than 700 HZ was used. A cut-off frequency greater than 1000 HZ was used for the safety valve line thermal hydraulic analysis.

The structural analysis approach is substantiated by comparisons of analytical results to test data. A discussion of the methodology utilized in performing a safety valve discharge structural analysis and comparison of analytical results to structural test results is presented in the following article:

L.C. Smith and T.M. Adams, "Comparison of Analytically Determined Structural Solutions with EPRI Safety Valve Test Results", 4th National Congress on Pressure Vessel and Piping Technology, Portland Oregon, June 19-24, 1983 PVP-Volume 74, pp. 193-199.

As noted in the response to Question 3, high frequency pressure oscillations of 170-260 HZ could occur in the piping system upstream of the safety valve, while the loop seal water passes through the valve. The safety valve slug discharge event (which occurs after the "simmering" phase), however, generates limiting system responses for the safety valve inlet, outlet, and common region piping.

REFERENCES

1. J.E. Maier, Rochester Gas and Electric Corp. letter to D.M. Crutchfield, NRC, "Post-TMI Requirements, NUREG-0737, Item II.D.1, R.E. Ginna Nuclear Power Plant", March 4, 1983

2. EPRI PWR Safety and Relief Valve Test Program Guide for Application of Valve Test Program Results to Plant-Specific Evaluations, Revision 2, Interim Report, July 1982

3. R.W. Kober, Rochester Gas and Electric Corp. letter to J.A. Zwolinski, NRC, "NUREG-0737, Item II.D.1 - Performance Testing of Relief and Safety Valves", May 24, 1985.

4. J.A. Zwolinksi, NRC, letter to R.W. Kober, Rochester Gas and Electric Corp., "Safety Evaluation for Appendix R to 10CFR Part 50, Items III.G.3 and III.L"

ATTACHMENT 1

GINNA STATION
PRESSURIZER SAFETY VALVE
INLET PIPING PRESSURE DROP
CALCULATION

V-EC-248

Prepared By: Susan R Shaw

S. R. Shaw, Engineer
Pump and Valve Engineering
W Electric Corp.

Reviewed By: M L Lacey

M. L. Lacey, Engineer
Pump and Valve Engineering
W Electric Corp.

Approved By: H A Sepp

H. A. Sepp, Manager
Pump and Valve Engineering
W Electric Corp.

January 12, 1987

I. INTRODUCTION:

This calculation was prepared by Westinghouse Electric Corporation at the request of Rochester Gas and Electric Corporation in support of their response to the Nuclear Regulatory Commission, regarding NUREG-0737 Item II.D.1, Performance Testing of Relief and Safety Valves.

II. SCOPE:

The lines analyzed herein are for the Ginna Station pressurizer safety valve inlets, valve locations 434 and 435.

III. PURPOSE:

The intent of this calculation is to provide plant-specific pressurizer safety valve inlet piping pressure drops, for opening and closing. These values are to be compared with the pressure drops, calculated by the same methods, found for pressurizer safety valves tested as part of the EPRI Safety and Relief Valve Test Program.

IV. CALCULATION:

A. Ginna Pipe Configuration

Several quantities must be determined from the actual physical configuration of the inlet lines for use later in the calculation. These are: total length of pipe, length of straight pipe, and L/D. The configurations, from Ref. 1 and 6 are shown in Figure 1 for valve 434, and Figure 2 for valve 435.

1. Valve 434

a. Inlet piping consists of:

1 3" x 4" Expander	4" Schedule 160 pipe
1 45° Elbow	Pressurizer Nozzle
1 90° Elbow	
1 180° Elbow	

b. Total Length of Pipe:

.8490 + .3333 + .3927 + .366 + .7854 + 1.250 + 1.5708 + .427 = 5.974 ft.

c. Length of Straight Pipe:

.8490 + .3333 + .366 + 1.250 + .427 = 3.225 ft.

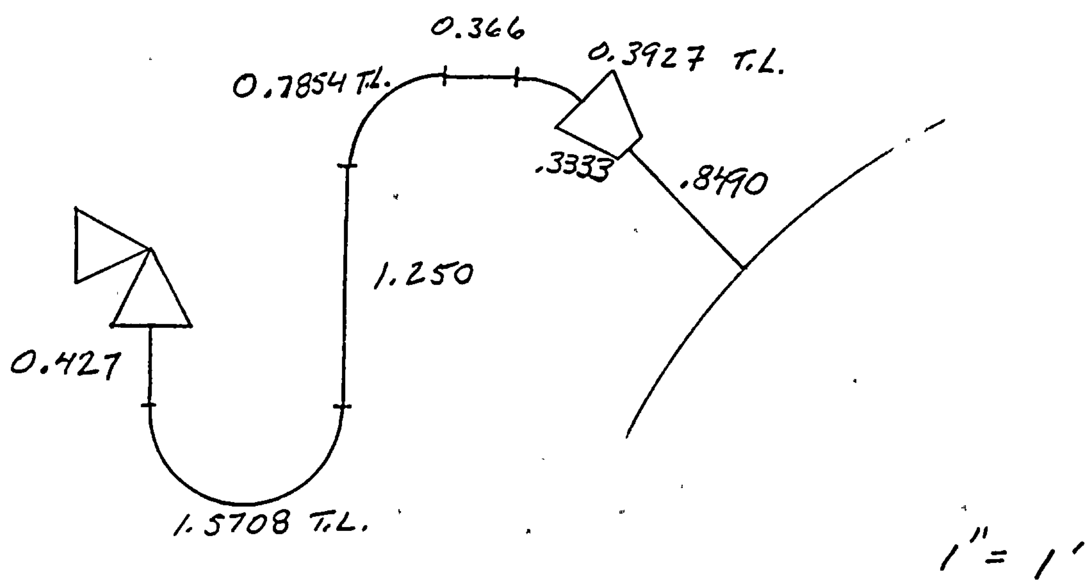

FIGURE 1: VALVE 434
(SHOWN ROTATED INTO PLANE)

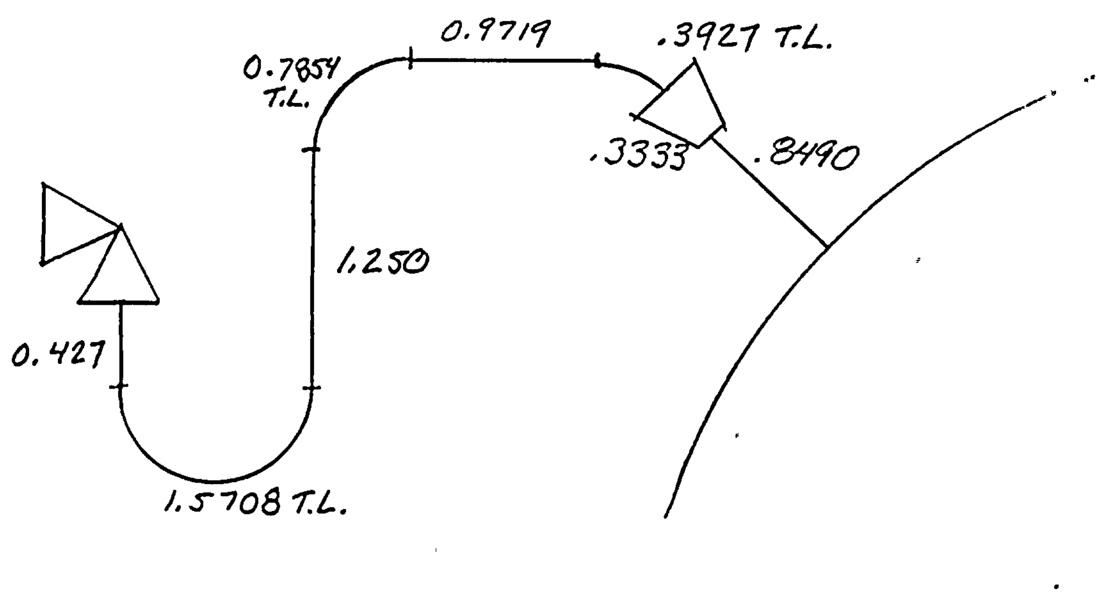

FIGURE 2: VALVE 435
(SHOWN ROTATED INTO PLANE)

d. L/D

1. From Reference 3, Appendix A

Standard 45° Elbow	L/D = 16
Standard 90° Elbow	L/D = 30
Standard 180° Return	L/D = 50
Long Radius 90° Elbow	L/D = 20

All elbows used in this Ginna Station pipe are long radius, 6 inch radius for 4 inch pipe. Since the long radius L/D's are not directly available for the 45° elbow and the 180° returns, they will be found by using the ratio of the long radius to standard 90° values.

Long Radius 45° elbow	L/D = 16 (20/30)=10.67
Long radius 180° elbow	L/D = 50 (20/30)=33.33

Note: Other sources availabe for calculation of the 180° elbow, such as <u>Fundamentals of Pipe Flow</u>, Robert P. Benedict, and from Reference 3, "Resistance of Bends", show the use of this method is conservative.

2. From Reference 4, for 4" schedule 160 pipe

D = 3.438 inches = .287 ft.

3. $L/D = \frac{3.225}{.287} + 10.67 + 20 + 33.33 = 75.24$
(dimensionless)

2. Valve 435

a. Inlet piping consists of:

Pressurizer Nozzle
1 3" x 4" Expander
1 45° Elbow
1 90° Elbow
1 180° Elbow
4" schedule 160 pipe

b. Total Length of Pipe:

.8490 + .3333 + .3927 + .9719 + .7854 + 1.250 + 1.5708 + .427 = 6.580 ft.

c. Length of Straight Pipe:

$.8490 + .3333 + .9719 + 1.250 + .427 = 3.831$ ft.

d. L/D

(Same variables used as in Valve 434 above)

$$L/D = \frac{3.831}{.287} + 10.67 + 20 + 33.33 = 77.35$$

(dimensionless)

B. Calculation for Pressure Drop (Reference 2)

1. For opening and again for closing for each valve a transient, and steady state pressure drop is calculated. For each case, the larger of the transient and steady state values is used as the controlling value for that case. Each transient pressure drop is the sum of the flow pressure difference due to pipe friction and fittings plus the pressure difference due to acoustic wave amplitude. The steady state pressure drop is due to pipe friction and fittings.

Thus there are 4 cases:

Case 1: Valve 434 Opening
Case 2: Valve 435 Opening
Case 3: Valve 434 Closing
Case 4: Valve 435 Closing

And for each case the greater of (Delta P_F + Delta P_{AW}) or Delta P_{ss} is used.

The equations and definitions used from Reference 2 are as follows:

<u>Transient Flow Pressure Difference (Delta P_F) Calculation</u>

The flow pressure difference due to pipe friction and fittings is given by:

- If $T \leq 2L/a$,

$$\text{Delta } P_F = \frac{(K + \frac{fL}{D})(C\dot{M})^2}{2g_c \text{Rho}^2 A^2}$$

- If $T > 2L/a$,

$$\text{Delta } P_F = \frac{\left(K + \frac{fL}{D}\right)(C\dot{M})^2}{2g_c \text{Rho } A^2} \left(\frac{2L}{aT}\right)$$

where,

K = summation of expansion and contraction loss coefficients corrected if required to correspond to the inlet piping flow area. (NOTE: The contraction from the pressurizer to the inlet pipe can be assumed to be smooth and, therefore, the loss coefficient can be assumed to be zero) (dimensionless)

f = friction factor (dimensionless)

$\frac{L}{D}$ = piping equivalent length/diameter considering effects of elbows and friction (dimensionless)

$\dot{M}$ = rated valve flow rate for steam as specified in Table B-1 of Ref. 2 (lb./sec.)

g_c = gravitational constant (32.2 lb-ft/lb-sec^2)

Rho = steam density at nominal valve set pressure (lb/ft^3)

A = inlet piping flow area (ft^2)

a = steam sonic velocity (ft/sec) - 1100 ft/sec. for all calculations

L = inlet piping length (ft)

T = valve opening or closing time for steam inlet conditions as specified in Table B-2 of Ref. 2 (sec.)

C = flow rate constant for valve opening or closing as specified in Table B-2 of Reference 2.

<u>Transient Acoustic Wave Amplitude (Delta P_{AW}) Calculation</u>

There are two situations to consider:

- If $T \leq 2L/a$,

$$\text{Delta } P_{AW} = \frac{a(C\dot{M})}{g_c A} + \frac{(C\dot{M})^2}{2g_c \text{Rho } A^2}$$

- If T > 2L/a,

$$\text{Delta } P_{AW} = \frac{2L\,(C\dot{M})}{g_c AT} + \frac{(C\dot{M})^2 \left(\frac{2L}{at}\right)^2}{2g_c \text{Rho } A^2}$$

All parameters are defined above.

Steady-State Flow Pressure Difference Calculation

The steady-state flow pressure difference associated with valve opening or closing is given by:

$$\text{Delta } P_F = \frac{\left(K + \frac{fL}{D}\right)(C\dot{M})^2}{2g_c \text{Rho } A^2}$$

All parameters are defined above. The values of the flow rate constant, C, are different for valve opening and closing and are provided in Table B-2, Ref. 2.

For this calculation, the parameters used are as follows:

a: K = K expander
From Reference 4, for gradual enlargements this is:

$$\text{K expander} = \frac{2.6 \operatorname{Sin}(\text{theta}/2)(1 - \text{Beta}^2)^2}{\text{Beta}^4}$$

$$\text{where Beta}^2 = \frac{a1}{a2} + \frac{d1^2}{d2^2} + \frac{(2.624)^2}{(3.438)^2} = .58253$$

theta = 23.01°, derived from Reference 5 dimensions

therefore

$$K = \frac{2.6 \operatorname{Sin}\left(\frac{23.01}{2}\right)(1-.58253)^2}{(.58253)^2}$$

$$= .266$$

b: f = .017 (Reference 4)

c: L/D = Calculated above Valve 434 L/D = 75.24
Valve 435 L/D = 77.35

d: $\dot{M}$ = 320,000 lb/hr. (Reference 2) = 88.89 Lb/sec.

e: gc = 32.2 ft-lb/lb-sec^2

f: Rho = 7.65 lb/ft^3 (Reference 4)

g: A = .0645 ft^2 (Reference 4)

h: a = 1100 ft/sec.

i: L = Calculated above
Valve 434 L = 5.974 ft.
Valve 435 L = 6.580 ft

j: T = Valve opening .010 sec. (Reference 2)
Valve closing .016 sec.

k: c = Transient:
Valve opening C = 1.11 (Reference 2)
Valve closing C = .69
Steady State C = 1.11

2. Case 1 Valve 434 Opening

a. Transient Calculation - Pressure drop due to friction

$$\frac{2L}{a} = \frac{2(5.974)}{1100} = .011$$

$$T = .010 < .011 = \frac{2L}{a}$$

Therefore, the transient pressure difference is

$$\text{Delta } P_f = \frac{[.266 + .017(75.24)]\,[(1.11)(88.89)]^2}{(2)(32.2)(7.65)(.0645)^2(144)}$$

$$\text{Delta } P_f = 50.97 \text{ psi}$$

Pressure drop due to acoustic wave amplitude

$$T < \frac{2L}{a}$$

therefore,

$$\text{Delta } P_{aw} = \frac{(1100)(1.11)(88.89)}{(32.2)(.0645)(144)} + \frac{[(1.11)(88.89)]^2}{(2)(32.2)(7.65)(144)(.0645)^2}$$

$$= 362.90 + 32.99 = 395.89 \text{ psi}$$

The total transient pressure drop is then

$$\text{Delta } P_T = \text{Delta } P_f + \text{Delta } P_{aw}$$

$$= 50.97 + 395.89 = 446.86 \text{ psi}$$

b. Steady state calculation

$$\text{Delta } P_{ss} = \frac{[(.266) + (.017)(75.24)][(1.11)(88.89)]^2}{(2)(32.2)(7.65)(.0645)^2(144)}$$

$$= 50.97 \text{ psi}$$

c. Since the transient pressure drop is greater than the steady state pressure drop for Valve 434 on opening, the controlling value is the transient one, which is

$$\text{Delta } P = 446.86 \text{ psi}$$

3. Case 2 Valve 435 Opening

a. Transient calculation

Pressure drop due to friction

$$\frac{2L}{a} = \frac{2(6.580)}{1100} = .012$$

$$T = .010 < .012 = \frac{2L}{a}$$

Therefore, the transient pressure difference is:

$$\text{Delta } P_f = \frac{[.266 + .017\ (77.35)][(1.11)\ (88.89)]^2}{(2)(32.2)(7.65)(144)(.0645)^2}$$

$$= 52.15 \text{ psi}$$

Pressure drop due to acoustic wave amplitude

$$T < \frac{2L}{a}$$

therefore

$$\text{Delta } P_{aw} = \frac{(1100)(1.11)(88.89)}{(32.2)(.0645)(144)} + \frac{[(1.11)(88.89)]^2}{(2)(32.2)(7.65)(144)(.0645)^2}$$

$$= 362.90 + 32.99 = 395.89 \text{ psi}$$

The total transient pressure drop is then

$$\text{Delta } P_T = \text{Delta } P_f + \text{Delta } P_{aw}$$

$$= 52.15 + 395.89$$

$$= 448.04 \text{ psi}$$

b. Steady State Calculation

$$\text{Delta } P_{ss} = \frac{[(.266)+(.017)(77.35)][(1.11)(88.89)]^2}{(2)(32.2)(7.65)(144)(.0645)^2}$$

$$= 52.15 \text{ psi}$$

c. Since the transient pressure drop is greater than the steady state pressure drop for Valve 435 on opening, the controlling value is the transient one, which is

$$\text{Delta P} = 448.04 \text{ psi}$$

4. Case 3 Valve 434 Closing

a. Transient calculation, pressure drop due to friction Since $T = .016 > .011 = \frac{2L}{a}$

then

$$\text{Delta } P_f = \frac{[.266+.017\ (75.24)][(.69)(88.89)]^2\left[\frac{(2)(5.974)}{(1100)(.016)}\right]^2}{(2)(32.2)(7.65)(144)(.0645)^2}$$

$$\text{Delta } P_f = 9.08$$

Pressure drop due to acoustic wave amplitude

$$T > \frac{2L}{a}$$

therefore

$$\text{Delta } P_{aw} = \frac{(2)(5.974)(.69)(88.89)}{(32.2)(.0645)(.016)(144)} + \frac{[(.69)(88.89)]^2\left[\frac{(2)(5.974)}{(1100)(.016)}\right]^2}{(2)(32.2)(7.65)(.0645)^2(144)}$$

$$= 153.14 + 5.87$$

$$= 159.01 \text{ psi}$$

The total transient pressure drop is then

$$\text{Delta } P_T = \text{Delta } P_f + \text{Delta } P_{aw}$$

$$= 9.08 + 159.01 = 168.09 \text{ psi}$$

b. Steady State Calculation

$$\text{Delta } P_{ss} = \frac{[.266 + (.017)(75.24)][(1.11)(88.89)]^2}{(2)(32.2)(7.65)(.0645)^2(144)}$$

$$= 50.97 \text{ psi}$$

c. Since the transient pressure drop is greater than the steady state pressure drop for Valve 434 on closing, the controlling value is the transient one, which is 168.09 psi

5. Case 4 Valve 435 Closing

a. Transient calculation, pressure drop due to friction

Since $T = .016 > .012 = \frac{2L}{a}$

then

$$\text{Delta } P_f = \frac{[.266 + (.017)(77.35)][(.69)(88.89)]^2 \left[\frac{(2)(6.580)}{(1100)(.016)}\right]^2}{(2)(32.2)(7.65)(144)(.0645)^2}$$

$$= 11.27 \text{ psi}$$

Pressure drop due to acoustic wave amplitude

$$T > \frac{2L}{a}$$

therefore

$$\text{Delta } P_{aw} = \frac{(2)(6.580)(.69)(88.89)}{(32.2)(.0645)(.016)(144)} + \frac{[(.69)(88.89)]^2 \left[\frac{(2)(6.580)}{(1100)(.016)}\right]^2}{(2)(32.2)(7.65)(.0645)^2(144)}$$

$$= 168.68 + 7.13$$

$$= 175.81 \text{ psi}$$

The total transient pressure drop is then

$$\text{Delta } P_T = \text{Delta } P_f + \text{Delta } P_{aw}$$

$$= 11.27 + 175.81$$

$$= 187.08 \text{ psi}$$

b. Steady State Calculation

$$\text{Delta } P_{ss} = \frac{[.266 + (.017)(77.35)][(1.11)(88.89)]^2}{(2)(32.2)(7.65)(.0645)^2(144)}$$

$$= 52.15 \text{ psi}$$

c. Since the transient pressure drop is greater than the steady state pressure drop for Valve 435 on closing, the controlling value is the transient one which is 187.08 psi.

Using the values calculated for Ginna Station, and the pressure drop information from the EPRI test from reference 2, the tabulation can be made:

	Ginna 434 Pressure Drop	Ginna 435 Pressure Drop	3K6 "F" Pressure Drop	6M6 "G" Pressure Drop
Valve Opening	446.86 psi	448.04 psi	391 psi	263 psi
Valve Closing	168.09 psi	187.08 psi	194 psi	181 psi

V. REFERENCES

1. "Piping As-Built Analysis Diagram Pressurizer Relief from Pressurizer to Relief Manifold", Gilbert Associates, Inc. Drawing C-381-353, Revision B, Sheets 6 and 7.

2. "EPRI PWR Safety and Relief Valve Test Program Guide for Application of Valve Test Program Results to Plant-Specific Evaluations", Interim Report, July 1982, Revision 2.

3. Crane Technical paper No. 410, "Flow of Fluids Through Valves, Fittings, and Piping", 1969.

4. Ibid., 1978

5. General Catalog No. 55, Ladish Co., 1954

6. "Pressurizer Upper Head Assembly", Westinghouse Electric Corp., Drawing 681J253.