OYSTER CREEK STATION NO. 1
SAFETY AND RELIEF VALVE
PIPING DESIGN REPORT

August 1972

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I. SUMMARY

A. Background

In the past two years, there have been incidents of steam plant safety relief valves breaking off their inlet piping at nuclear power plants: specifically, one at the Robinson Plant and one at Turkey Point. Each occurred during the plant initial test program. In each case the failures were attributed to the reaction forces generated by the steam flow transient.

In view of these incidents, the designs of the safety and relief valve installations at Oyster Creek were re-evaluated. The re-evaluation included the review of the existing calculations and performance of additional analyses as required to provide a complete set of stress calculations. This report presents the main conclusions and a description of the system modifications performed as a result of this evaluation.

B. Conclusions

1. Safety Valves

The Oyster Creek main steam system includes sixteen spring loaded safety valves, eight on each of the north and south steam headers. The review of the main steam safety valve system design concludes that the safety valve installation is satisfactory, based on USAS B31.1 design criteria. USAS

B31.1 was used for the original design and is invoked for similar systems in nuclear plants currently under construction.

The results of the review indicate that the reaction loads were properly considered in the original analyses: however, since the time the analysis was performed, the valves were changed from Crosby to Dresser valves. In view of this, the analysis was updated to reflect the installed valve characteristics and to include the stresses resulting from pressure, seismic and deadweight effects as well as the effect of the flow reaction loadings. The results of this analysis are summarized in Chapter II of this report and show that the safety valve installation is satisfactory.

2. Electromatic Relief Valves

The Oyster Creek steam system includes five electromatic relief valves, three on the south main steam header and two on the north main steam header.

The review of the original design analyses of the electromatic relief valve installation showed that relief valve reaction loadings had not been considered. For this reason, additional analyses were performed to calculate these loads and determine whether the existing relief valve piping installation is satisfactory. The results of the analyses, based on USAS B31.1 design criteria, indicates that four of the five valve

inlet connections and associated discharge piping would be overstressed. These stresses were calculated assuming that all of the valves in either header lift simultaneously as would be the case in the event their set point is reached during an actual plant transient.

It should be noted that the plant operator reports that to date only one electromatic relief valve has been lifted at a time as a part of the required periodic tests of these valves. Since the stresses are reduced roughly proportional to the number of valves lifting on any one header, the calculated stresses were below yield with only one valve lifting.

In order that the relief valve piping meet the USAS B31.1 allowable stress requirements during simultaneous discharge of all the electromatic relief valves, snubber type pipe supports have been designed and added to the piping system to absorb the reaction loads. This has satisfactorily reduced the stresses in both the valve inlet connections and in the discharge piping in the drywell to comply with USAS B31.1. The analyses performed to determine the relief valve reaction loads, to design the necessary piping supports, and to determine the resulting stresses during valve relieving and during heatup and cooldown of the piping in the drywell are summarized in Chapter III of this report.

As a result of the analysis of the adequacy of the safety and relief valve connections, it was concluded that the design of the discharge tail pipe within the torus should also be reviewed. It was found that additional supports are required to support the loads which would occur in the event all valves in a header relieve simultaneously. Therefore, support were designed and installed to assure that failure of this piping would not occur. Further review of this piping in the torus and its supports is underway to assure the long term adequacy of the design. The analyses performed and the supports added for the two tailpipes are described in Chapter IV of this report.

II. SAFETY VALVES

The main steam safety valve installation at Oyster Creek consists of 16 spring loaded safety valves which are divided equally between the north and south headers as shown on Figure II-1. The safety valves were manufactured by Dresser Industrial Valve and Instrument Division and are described in reference 2.

The safety valves are supported directly from the main steam headers by the inlet connections and, if opened, discharge steam directly to the drywell through an open ended tee as shown schematically in Figure II-2. Use of the tee in the discharge piping prevents large thrust forces from occurring during the steady-state discharge of the safety valves.

Transient thrust forces occur, however, during the initial opening transient of the valves. These forces occur because the sonic time delay between the valve and the tee causes a higher steam mass flow to occur at the valve than at the tee at any instant of time. This gradient in flow causes an unbalance in the momentum and pressure forces between the valve and the tee.

The magnitude of the thrust force is directly related to the rate of acceleration of the valve disk as it opens. A valve that opens quickly causes a rapid increase in steam flow which in turn results in a large thrust force. Therefore, the valve disk opening position versus time was assessed to estimate the reaction forces which occur.

The Oyster Creek safety valve opening characteristic transient was determined by contacting valve vendors and reviewing previous calculations, as follows:

- The valve manufacturer (Dresser Industrial Valve and Instrument Division) was contacted regarding the opening transient of the Oyster Creek valves. Dresser indicated that these valves pop open to about 70% of full flow in less than 60 msec and follow the shape of a (1-cos wt) curve during the transient.

- The Crosby Valve Company was also contacted regarding the opening transient of Crosby valves. Crosby stated that their valves open linearly from initial popping to about 70% flow in a minimum time of 40 msec.

- The original safety valve calculations show a catenary curve for valve opening to 60% flow in 40 msec.

The opening characteristic used in the original analysis has the steepest slope and results in the largest thrust forces; therefore, it was used for the updated analysis. This transient develops a thrust force of 1660 pounds force. However, to account for impact loadings which may occur during relieving, the transient force was doubled to 3320 pounds for the analysis. This force acts approximately along the center line of the valve discharge nozzle in a direction opposite the discharge flow.

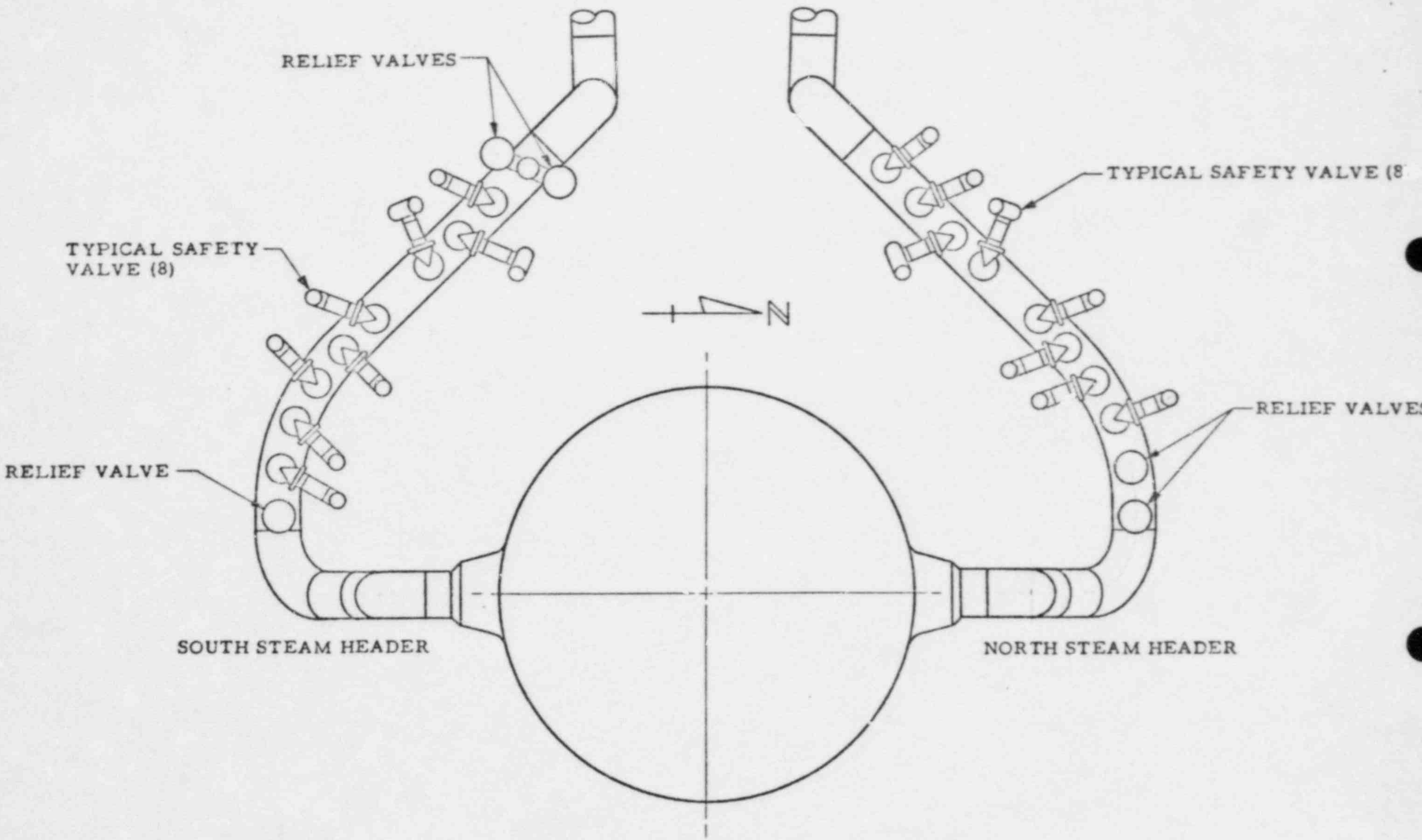
The stresses resulting from the effects of seismic, deadweight and internal pressure forces as well as the opening transient impact loadings were calcu-

lated at the following locations:

- header to inlet nozzle connection
- valve inlet piping
- valve discharge piping

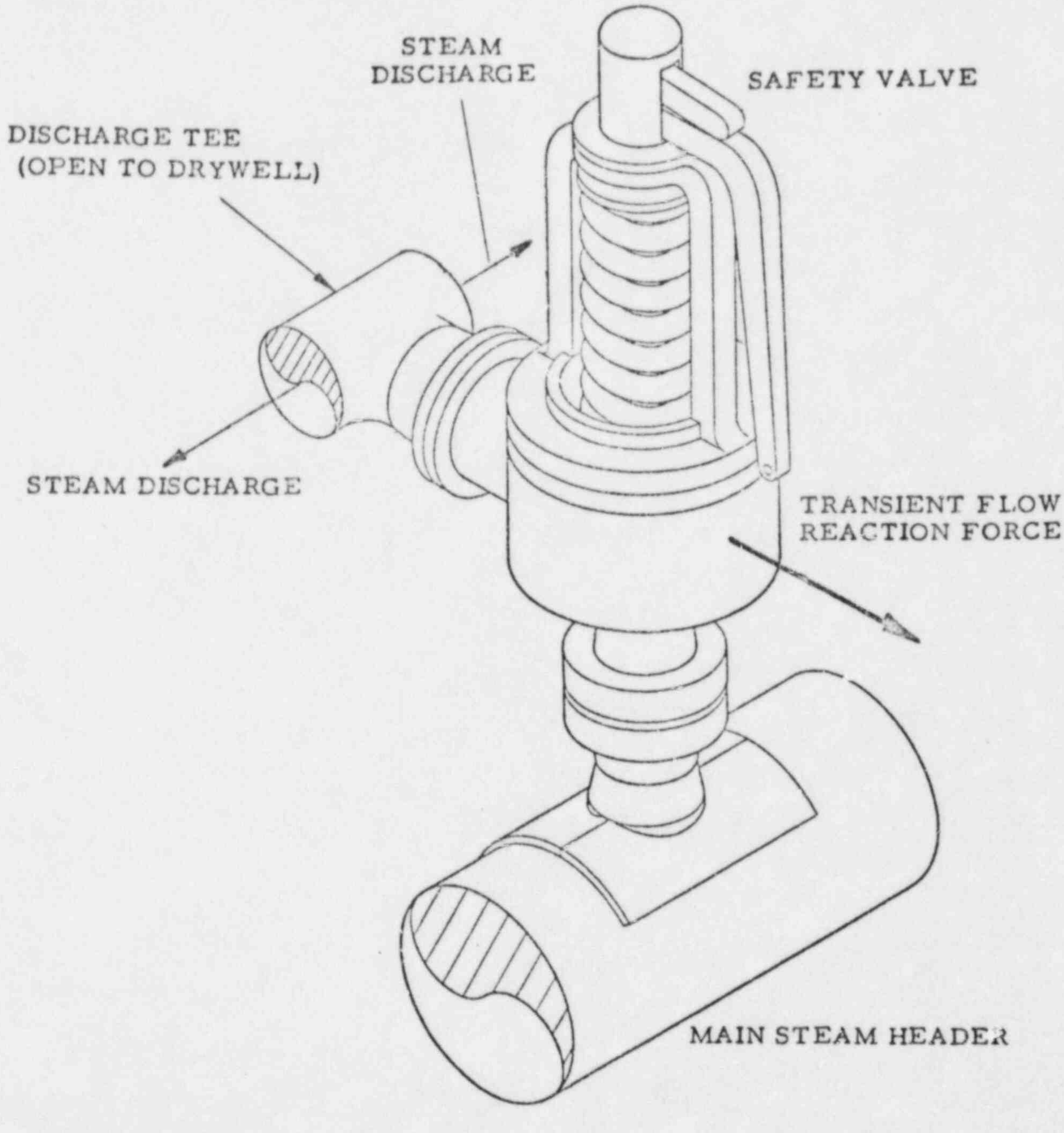
The stresses at the header to inlet nozzle connection were determined using the method described in the Welding Research Council Bulletin (WRC) 107 (reference 3). This method permits calculation of the secondary bending stresses as well as the primary stresses. The primary stresses were compared to USAS B31.1 allowable stresses. The sum of primary plus secondary stresses were compared to ASME Code, Section III, criteria since USAS B31.1 does not provide acceptance criteria for secondary bending stresses.

A summary of the resulting stresses and a comparison to the USAS B31.1 and ASME Section III allowable stresses are included in Table II-1. The results show that the safety valve piping stresses are satisfactory and well below the allowables.



LOCATION OF MAIN STEAM SAFETY VALVES AND RELIEF VALVES

FIGURE II-1



MAIN STEAM SAFETY VALVE

FIGURE II-2

TABLE II-1

SUMMARY OF STRESS ANALYSIS OF SAFETY VALVE PIPING

Location	Type of Stress	Stress (psi)	Allowable Stress (psi)	Code and Formula*
°Discharge Piping to tee	Hoop	1840	17,500	1.0Sm(B31.1)
	Longitudinal	1970	21,000	1.2Sm(B31.1)
°Valve Inlet Piping	Hoop	8260	17,500	1.0Sm(B31.1)
	Longitudinal	8460	21,000	1.2Sm(B31.1)
°Inlet Nozzle Connection to Steam Header	Local Membrane			
	- Hoop	12,822	21,000	1.2Sm(B31.1)
	- Longitudinal	7,441		
	Local Membrane & Secondary Bending			
	- Hoop	24,763	59,100	3.0Sm(Sec III)
	- Longitudinal	15,548		

*For A-106, Gr. C: B31.1 Sm=17,500 psi; Sec III Sm= 19,700 psi

III. ELECTROMATIC RELIEF VALVE PIPING WITHIN THE DRYWELL

The electromatic relief valve installation at Oyster Creek consists of five electrically actuated valves and associated inlet and discharge piping. Two of these valves are located on the north main steam header and three are located on the south header as shown in Figure II-1. These valves discharge steam to the suppression chamber (torus) when the pressure in the main steam header exceeds the set pressure of the valves. Figures III-1 and III-2 are isometric drawings of the electromatic relief valve piping from the valve inlet connections at the main steam headers to the discharge in the torus.

Evaluation of the Loads on the Piping

Large forces occur along the axes of the relief valve piping when discharging steam from the relief valves to the suppression chamber. These forces are similar to those described for the safety valves. They result from momentum changes as the steam changes direction through the piping elbows and from pressure changes as the steam expands moving down the piping. In the steady-state, these momentum and pressure forces cancel each other between upstream and downstream elbows.

Unbalances in these forces occur, however, during the initial opening transient of the valve. Since the valves open quickly, the steam mass flow in the upstream piping is higher than it is in the downstream piping. This unbalance in flow causes the momentum forces in upstream elbows to be

larger than the momentum forces in the downstream elbows. Similarly, because of the time required for the pressure wave to pass through the discharge piping, the pressure in the upstream piping and elbows will be greater than in the downstream portions during the transient. Consequently, the momentum and steam expansion pressure forces do not cancel. Further, an additional pressure exists which occurs from accelerating the steam in the piping from zero flow to full flow. The resulting force from this pressure adds to the force unbalance.

The magnitude of the transient forces which occur on the Oyster Creek relief valve piping were determined using a steam blowdown computer program (reference 4) which calculates the flow rates and pressures versus time during the steam discharge. In this approach, the system is divided into a number of nodes and connectors. Each node is assigned the proper volume, mass and energy. Mass is transferred from node to node through the connectors accounting for the flow resistances, acceleration, momentum, expansion, and choked flow effects during relieving.

The momentum and energy equations for each of the nodes and the related connectors are solved for individual, small time increments using the conditions at the beginning of the increment. The result of this calculation provides a set of conditions at the end of the time increment which are then used as the initial conditions for the following time increment. The solution proceeds in this manner.

The properties of the steam for each time increment are determined from a set of steam tables included within the program. These tables include properties of pure liquid, pure vapor or two phase mixtures. The data used to determine choked flow conditions were taken from the ASME Steam Tables.

The sequence of events considered in the analysis is as follows:

1. The piping is initially at atmospheric pressure. That portion of the piping which is below the water level in the torus is filled with water.

2. The relief valves are considered to open linearly from zero to full flow in 0.15 seconds. This 0.15 second time constant was provided by the valve manufacturer based on tests of a similar valve.

3. The transient continues until after the relief valves reach full flow and the water is discharged from the piping permitting the steam flow to reach steady state.

The pressures and flows versus time for each node and connector were used to calculate the forces which occur at each piping elbow during the transient. The force at each elbow is equal to the sum of (1) the force resulting from changing the steam momentum as it turns through the elbow and (2) the force due to the expansion and acceleration pressure acting on the pipe area at each elbow. Since the steam momentum and pressures at each elbow are not equal during the transient, a net force occurs on each straight run of piping.

The largest gradient in steam momentum and pressure in the piping in the drywell occurs at the time all of the mass of water in the piping submerged

in the torus is discharged. Up to this time, the pressure in the piping builds up while the water is accelerating. After the water leaves the piping in the torus, the steam is permitted to discharge to the torus causing a rapid decrease in pressure and increase in flow.

The results of the above analysis are shown as force vectors in Figure III-1 for the north header piping and Figure III-2 for the south header piping.

Description and Evaluation of the Modified System

The loads applied during the electromatic relief valve opening transient are sufficient to overstress the relief valve inlet and discharge piping if unrestrained. Therefore, five hydraulic snubber pipe supports rated at 10,000 pounds each have been added to the system and two existing spring type hangers were modified. These supports are identified in Figures III-1 and III-2 as N1 thru N3 on the north header and S1 thru S4 on the south header. A description of the pipe support designs and a summary of the stresses in the modified system are described below:

A. Pipe Support Design

The hydraulic snubber pipe supports were manufactured by Bergen Patterson Pipe Support Corporation, and are described in detail in Bergen Patterson Catalog No. 66, part number HSSA-10. These snubbers transmit the reaction loadings to the structural members attached to the biological shield during the valve opening transient. At other times the piping is free to move unrestricted. Consequently, these hangers do not impose large stresses on the piping during normal plant heatup and cooldown.

The forces which occur, the support design loads, and the maximum stresses in the supports are presented in Table III-1. The resulting stresses in these supports are considered acceptable. These stresses were calculated by conservatively using the design loads rather than the actual loads calculated using the blowdown computer program. This additional design margin is shown in Table III-1.

B. Piping Analysis

The addition of the five hydraulic snubbers and the modification of the two existing spring hangers changed the piping stresses that occur during plant heatup and valve relieving. In addition, several of the reaction loadings that occur on the piping are not restrained. Consequently, it was considered necessary to perform a piping flexibility analysis to supplement the original analyses. This analysis was performed for the as-modified system, including the additional five hydraulic snubbers and two modified spring hangers, and also considered the following:

- The rate of piping thermal growth at the time the valves relieve is sufficient to cause some of the hydraulic snubbers to lock up and prevent motion. This increases the pipe stresses due to thermal growth of the piping and was, therefore, included in the analysis.

- The pipe support flexibility, linkage play, and compressibility of the hydraulic oil in the snubbers will permit the piping to move

to a limited extent when the reaction forces occur. This motion may increase the piping stresses and was modeled in the flexibility analysis.

The flexibility analysis was performed for the relief valve discharge piping system for the two main steam lines using a piping flexibility analysis computer program (reference 5).

The cases which were analyzed included:

Case 1. Plant Heatup

The movement of the relief valve inlet nozzles which occurs during normal plant heatup was imposed on the relief valve piping systems. This case includes the effects of the piping weight and the restraining effects of the spring hangers and rigid supports and the attachment of the discharge pipe to the 6'-6" vent line as it enters the torus.

Case 2. Relief Valve Discharge Flow Reaction Forces

The piping motions and resulting stresses which occur as a result of the reaction forces during the valve opening transient were determined. The snubbers support the reaction forces but permit some piping motion due to hanger foundation resilience, manufacturing tolerances on the snubber connecting linkages and the compressibility of the snubber hydraulic oil. The effects of piping deadweight and plant heatup motions were also included in this case. The reaction forces used in the calculations were the design loads in Table III-1. Since

the reaction forces actually calculated using the steam blowdown computer program and shown in Figure III-1 and III-2 are all lower than these design loads, the piping stresses calculated are on the conservative side.

Case 3. Relief Valve Line Heatup During Valve Discharge

The effects of thermal heatup of the relief valve discharge piping during valve relieving, including the effects of pressure, deadweight and plant heatup were determined as a third case. This condition occurs at some time after steady-state flow in the relief valve piping is reached.

The results of the analyses of these three cases are summarized in Table III-2. In addition, the maximum combined longitudinal stresses due to internal pressure, piping motions during plant heatup, flow reaction forces during the relief valve discharge transient, and piping weight are presented in Table III-2. As can be seen from the table, all the stresses are below the allowable. The isometrics of the piping system shown in Figure III-1 for the north header and Figure III-2 for the south header identify the locations of the highest stressed portions of the system which are listed in Table III-2.

TABLE III-1 - PIPING SUPPORT SUMMARY

Identification Symbol	Design Load (lbf)	Maximum Stress in Supports Due to Design Loads (psi)	Actual Calculated Load (lbf)	Design Margin [Design Load / Actual Load]
N1	10,000	9,250	4100	2.44
N2	3,000	333	770	3.90
N3	10,000	*	5380	1.86
S1	10,000	*	2800	3.57
S2	3,000	400	1530	1.96
S3	3,000	*	1150	2.61
S4	10,000	17,000**	9450	1.06

*Standard Bergen Patterson Components rated for 10,000 lbs.

** This stress was conservatively calculated by assuming the full design load was applied to only one of the two supports for this rigid hanger.

TABLE III-2

SUMMARY OF STRESS ANALYSIS OF RELIEF VALVE PIPING

Location	Loadings Considered	Point of Maximum Stress[a]	Maximum Combined Stress (psi)[b]	Allowable Stress (psi)	Code and Formula
Valve Discharge Piping					
Case 1 - Plant Heatup	Plant Heatup Thermal Expansion +Piping Weight	(1)-North (1)-South	13,815 6,228	22,500	Sa=1.5Sm[c] (B31.1)
Case 2 - Relieving Transient	Plant Heatup Thermal Expansion +Piping Weight +Flow Reaction Force	(2)-North (2)-South	13,259 8,016	22,500	Sa=1.5Sm (B31.1)
Case 3 - Discharge Pipe Heatup	Plant Heatup Thermal Expansion +Piping Weight +Discharge Piping Heatup +Pressure	(3)-North (3)-South	6,752 10,358	22,500	Sa=1.5Sm (B31.1)
Maximum Combined Longitudinal Stresses	Internal Pressure +Case 2[f]	(2)-North (2)-South	17,539(f) 11,256	18,000	Sa=1.2Sm (B31.1)

Table III-2 (continued)

Location	Loadings Considered	Point of Maximum Stress[a]	Maximum Combined Stress (psi)[b]	Allowable Stress (psi)	Code and Formula
Valve Inlet Piping					
Case 1 - Plant Heatup	Plant Heatup Thermal Expansion +Piping Weight	(4)-North (4)-South	979 5,369	26,200	Sa=1.5Sm[d] (B31.1)
Case 2 - Relieving Transient	Plant Heatup Thermal Expansion +Piping Weight +Flow Reaction Force	(5)-North (5)-South	5,992 7,677	26,200	Sa=1.5Sm (B31.1)
Case 3 - Discharge Pipe Heatup	Plant Heatup Thermal Expansion +Piping Weight +Discharge Piping Heatup +Pressure	(6)-North (6)-South	1,703 6,744	26,200	Sa=1.5Sm (B31.1)
Maximum Combined Longitudinal Stresses	Internal Pressure +Case 2[f]	(5)-North (5)-South	7,702 9,387	21,000	Sa=1.2Sm (B31.1)

Table III-2 (continued)

Location	Type of Stress	Point of Maximum Stress	Maximum Combined Stress (psi)[b]	Allowable Stress (psi)	Code and Formula
Inlet Nozzle Connection to Steam Header	Local Membrane				
	Hoop	10" Nozzle in South Header[g]	14,861	21,000	1.2Sm[d] (B31.1)
	Longitudinal		9,725		
	Local Membrane and Secondary Bending				
	Hoop	10" Nozzle in South Header	25,607	59,100	3.0Sm[e] (Sec III)
	Longitudinal		17,287		

(a) Points are identified on Figures III-1 and III-2

(b) These stresses are the result of applying the design loads which are greater than the actual loads shown in Figures III-1 and III-2.

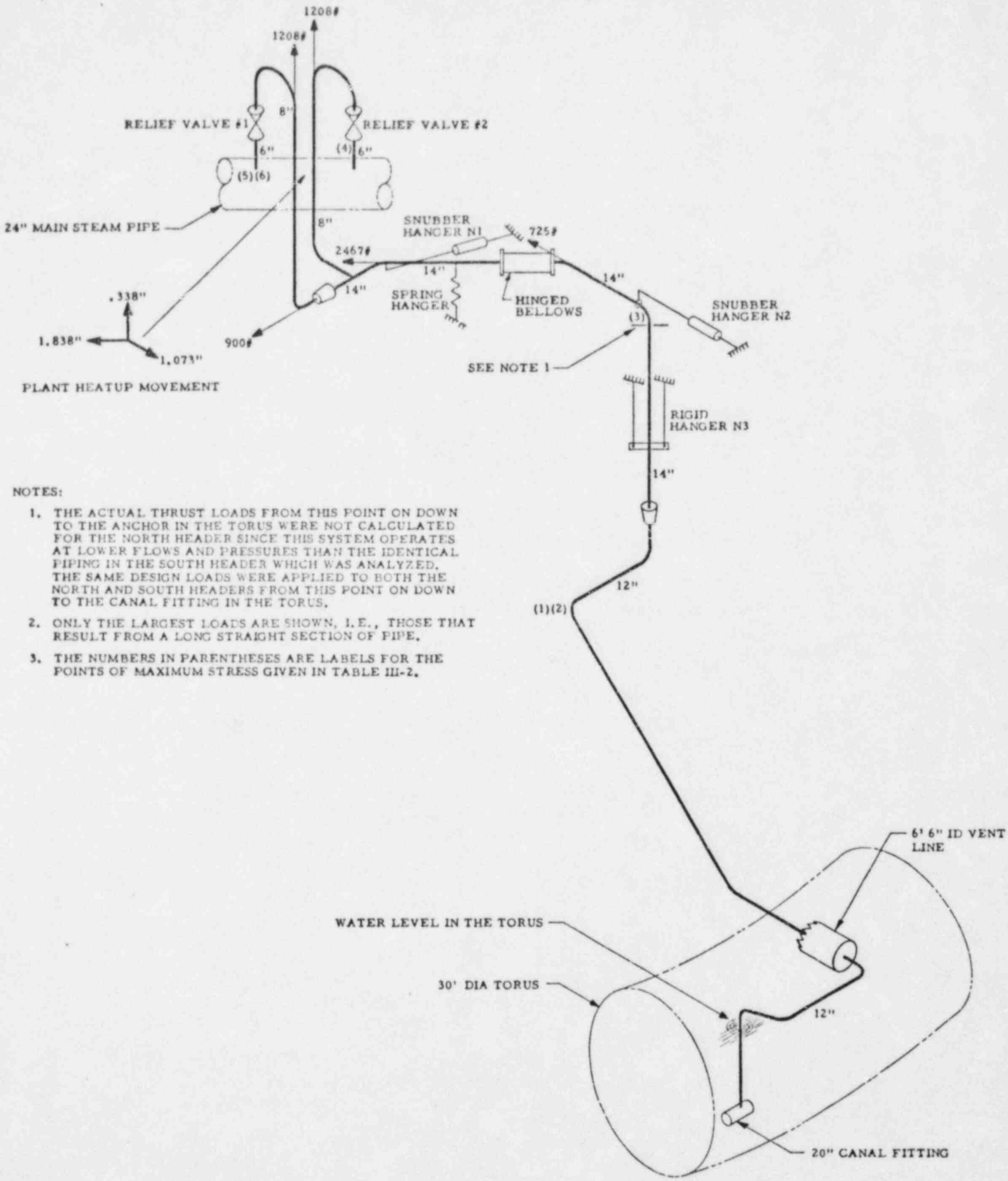
(c) A-106, Gr. B: B31.1 Sm=15,000 psi

(d) A-106, Gr. C: B31.1 Sm=17,500 psi

(e) A-106, Gr. C: Section III Sm=19,700 psi

(f) This stress includes thermal expansion stresses due to plant heatup which are not required by B31.1 to be included in the 1.2Sm category. Therefore, the maximum combined longitudinal stresses tabulated above are conservative.

(g) The stresses in the 10" nozzle are higher than the 6" nozzles in north or south header.

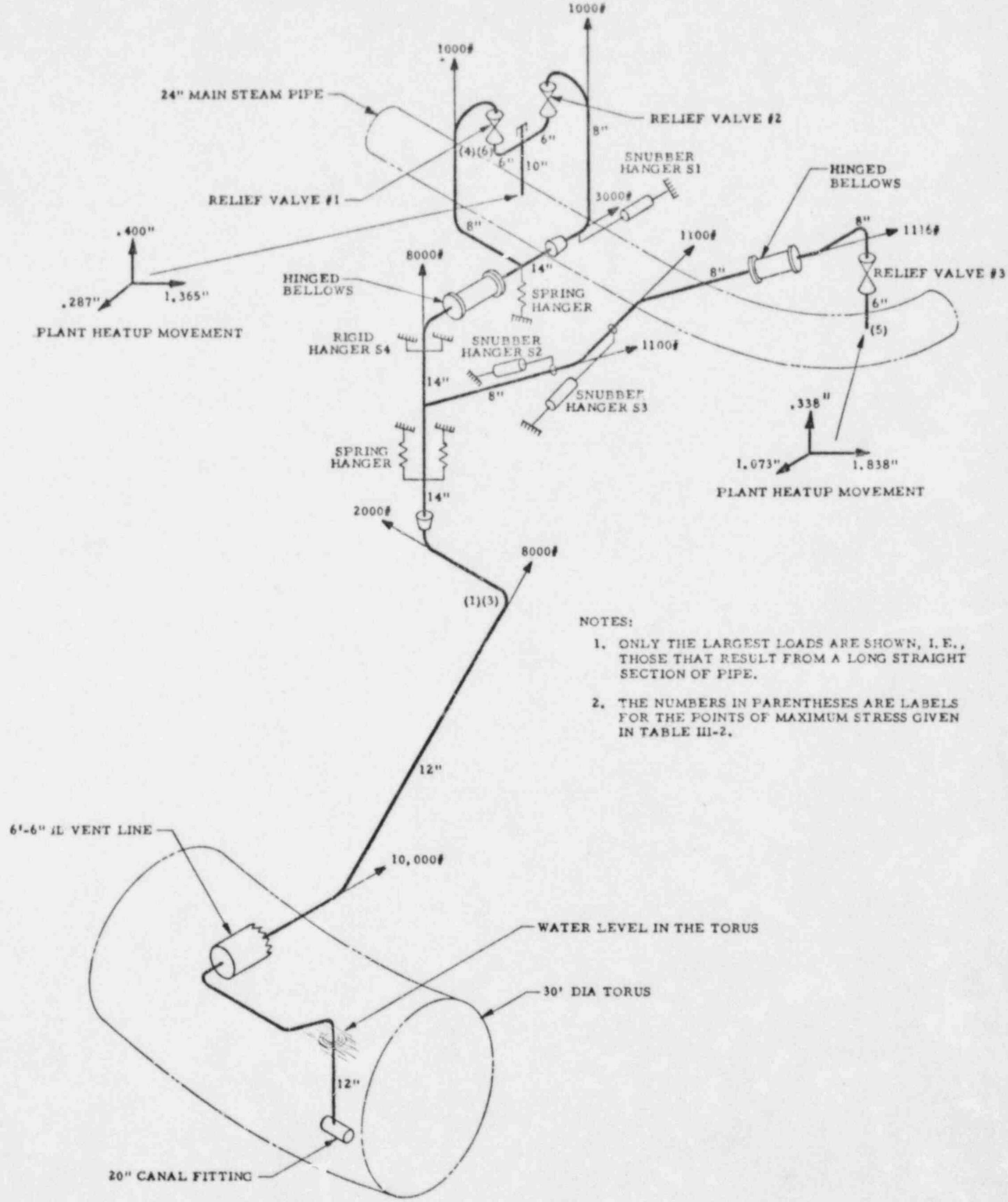


NOTES:

1. THE ACTUAL THRUST LOADS FROM THIS POINT ON DOWN TO THE ANCHOR IN THE TORUS WERE NOT CALCULATED FOR THE NORTH HEADER SINCE THIS SYSTEM OPERATES AT LOWER FLOWS AND PRESSURES THAN THE IDENTICAL PIPING IN THE SOUTH HEADER WHICH WAS ANALYZED. THE SAME DESIGN LOADS WERE APPLIED TO BOTH THE NORTH AND SOUTH HEADERS FROM THIS POINT ON DOWN TO THE CANAL FITTING IN THE TORUS.
2. ONLY THE LARGEST LOADS ARE SHOWN, I.E., THOSE THAT RESULT FROM A LONG STRAIGHT SECTION OF PIPE.
3. THE NUMBERS IN PARENTHESES ARE LABELS FOR THE POINTS OF MAXIMUM STRESS GIVEN IN TABLE III-2.

NORTH HEADER RELIEF VALVE DISCHARGE PIPING

FIGURE III-1



NOTES:

1. ONLY THE LARGEST LOADS ARE SHOWN, I.E., THOSE THAT RESULT FROM A LONG STRAIGHT SECTION OF PIPE.

2. THE NUMBERS IN PARENTHESES ARE LABELS FOR THE POINTS OF MAXIMUM STRESS GIVEN IN TABLE III-2.

SOUTH HEADER RELIEF VALVE DISCHARGE PIPING

FIGURE III-2

IV. ELECTROMATIC RELIEF VALVE TAILPIPING WITHIN THE SUPPRESSION CHAMBER

The discharge tailpiping within the suppression chamber (torus) is an extension of the electromatic relief valve discharge pipe in the drywell and functions to carry the steam underwater to discharge at a point near the bottom of the torus. Unbalanced forces occur in this piping during the relief valve opening transient in the same manner as in the piping in the drywell as discussed in Chapter III. In addition, a large unbalanced load occurs at the pipe exit due to the expulsion of the slug of water initially in the pipe.

The loads and stresses in this piping were calculated using the same computer program discussed in Chapter III. It was found that additional supports are required to support the loads which would occur in the event all valves in one line relieve simultaneously. Therefore, additional supports have been designed and installed which will prevent failure of this piping. The arrangement of these supports is shown in Figure IV-1. An isometric of the piping within the torus is shown in Figure IV-2.

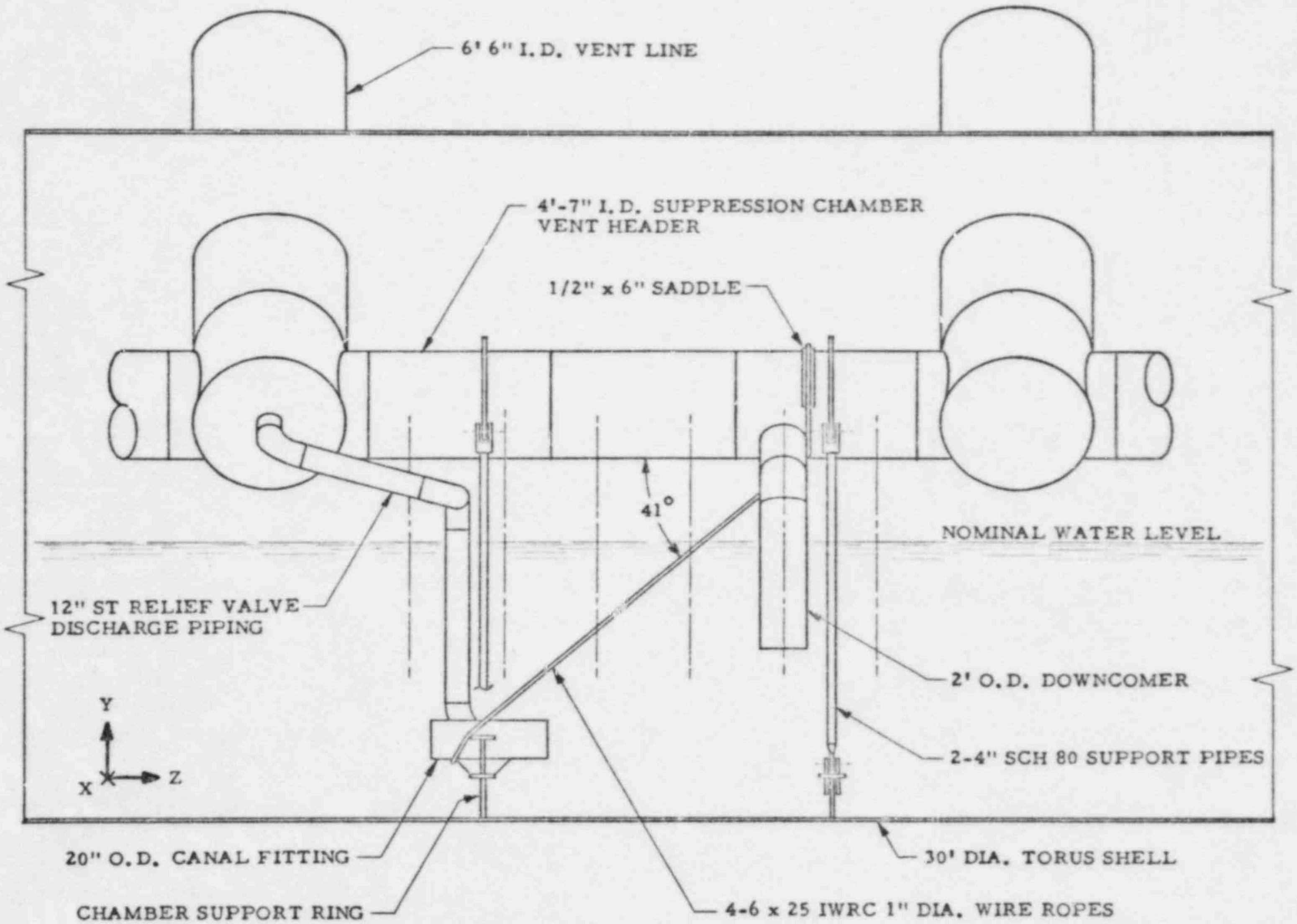
The added supports consist of a 1" diameter wire rope wound twice around the discharge end of each tailpipe and the vent header. The suppression chamber vent header is supported by existing pipe stanchions.

The results of the analysis of the discharge tailpiping and supports are summarized in Table IV-1. The location of the maximum stress in the tailpiping is shown in Figure IV-2.

As can be seen from Table IV-1, the stresses are satisfactory. The stress in the discharge pipe near the canal fitting exceeds the allowable; however, it is less than the minimum yield strength (about 80% of yield). The calculation of this stress was done in a conservative manner, in that the horizontal deflection of the canal fitting (see Figures IV-1 and IV-2) was determined by applying the total steady-state discharge force of 56,200 lbf or transient force of 94,000 lbf to the wire rope. No credit was taken for the lateral support of the chamber support ring or the 12" discharge piping itself in determining the displacement of the canal fitting. In addition, the entire pipe was assumed to heat from room temperature to 550°F even though much of it is submerged in the suppression water. Further, the stress results from the deflection caused by the stretch in the cables, therefore, only a limited strain can occur.

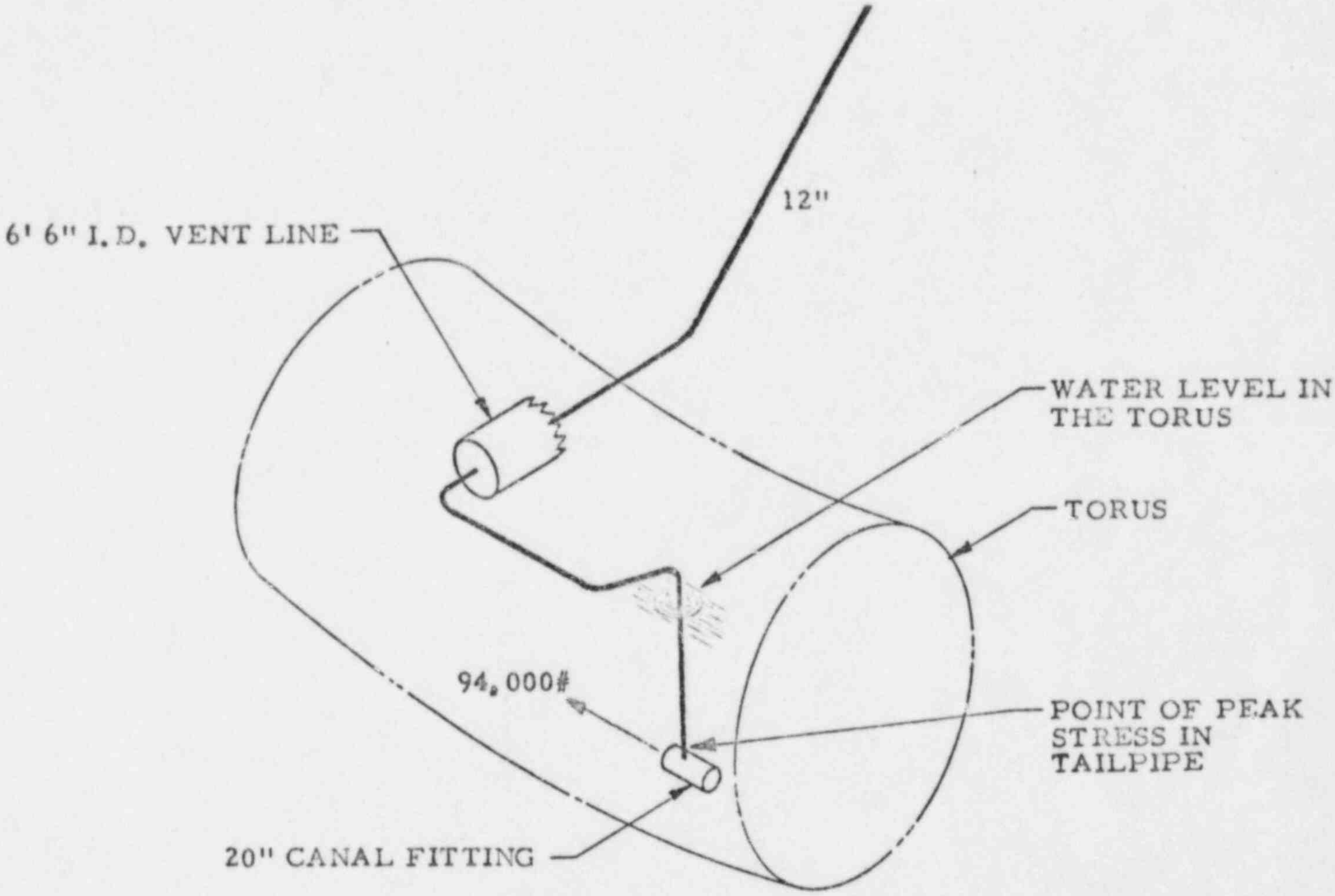
Table IV-1 also shows the conservatism in the design of the cables. The cables have a factor of safety of 4.8 for the steady-state load and a factor of safety of 2.9 on the peak load based on their minimum breaking strength.

The design of the supports for the tailpiping in the torus is being reviewed to assure its long-term adequacy; however, it is concluded that the design as installed will prevent failure of the system.



RELIEF VALVE DISCHARGE TAILPIPING AND SUPPORT STRUCTURE WITHIN THE TORUS

FIGURE IV-1



NOTE:

ONLY THE LARGEST LOAD IS SHOWN SINCE THE OTHERS ARE NEGLIGIBLE BY COMPARISON.

RELIEF VALVE DISCHARGE TAILPIPING WITHIN THE TORUS

FIGURE IV-2

TABLE IV-1
SUMMARY OF STRESS ANALYSIS
VALVE DISCHARGE TAILPIPING IN THE TORUS

Location	Load (lbf)	Stress (psi)	Allowable (psi)*	Code	Comment
12" Tailpipe in the Canal Fitting Transient	94,000	28,053	1.5Sm = 22,500 S_y = 35,000	B31.1 Section III	Occurs at .250 seconds into the transient just as the last slug of water leaves the canal fitting.
Steady-State	56,200 + Thermal Heatup	28,229	1.5Sm = 22,500 S_y = 35,000	B31.1 Section III	During steady state discharge when the pipe heats up.
12" Pipe Bend Inside the Canal Fitting	94,000	21,600	1.5Sm = 22,500	B31.1	At .250 seconds
Shear Between Canal Fitting and Support Foot	94,000	2,600	.6Sm = 9,000	B31.1	At .250 seconds
Tensile Between Canal Fitting and Support Foot	94,000	3,640	Sm = 15,000	B31.1	At .250 seconds
Tensile on Wire Rope					
Transient	125,000	--	Factor of Safety = 2.9		At .250 seconds
Steady-State	74,500	--	Factor of Safety = 4.8		During steady-state

*For A-106, Gr B

V. REFERENCES

1. Drawing 2103-10, "Steam Piping-Plants, Section and Details-Reactor Building." Revision 10, Oyster Creek Unit #1, Burns and Roe, Inc.

2. Drawing 6-3777 QA-RT-21-0S100, "Maxiflow Safety Valve," Dresser Industrial Valve and Instrument Division.

3. Welding Research Council Bulletin 107, "Local Stresses in Spherical and Cylindrical Shells due to External Loadings," August 1965.

4. MPR Computer Program based on "FLASH-4: A Fully Implicity Fortran IV Program for the Digital Simulation of Transients in a Reactor Plant," March 1969, WAPD-TM-840, Bettis Atomic Power Laboratory.

5. "Pipe Flexibility Analysis Program MEL 21", Report No. 10-66, Rev. 1, A Modification of Program MEC 21S, San Francisco Bay Naval Shipyard, Mare Island, dated August 1967.