CLASS 3 NON-PROPRIETARY

WCAP-10105

REVIEW OF PRESSURIZER SAFETY VALVE PERFORMANCE AS OBSERVED IN THE EPRI SAFETY AND RELIEF VALVE TEST PROGRAM

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1.0 INTRODUCTION

1.1 BACKGROUND

Following the Three Mile Island event, the Nuclear Regulatory Commission presented their position that PWR and BWR reactor licensees and applicants conduct testing to qualify reactor coolant system relief and safety valves for their expected operating conditions[1]. It was recognized that ASME Code certification requirements did not address power-operated relief valves (PORVs), as they were not credited in safety analyses and only addressed safety valves insofar as saturated steam conditions, i.e. for safety valves designed for steam service, capacity certification for water-solid and two-phase flow conditions was not required.

As such, the NRC specified that the valves be tested under the full range of expected operating fluid conditions. Testing under anticipated transient without scram (ATWS) conditions was not included.

Clarification of the NRC requirements appeared in several letters and NUREG reports[2,3]. With clarification came expansion of the qualification program to include testing of the PORV isolation (block) valves and analysis of the piping support system. The information request as revised by the NRC is[4]:

"Licensees and applicants shall determine the expected valve operating conditions through the use of analyses of accidents and anticipated operational occurrences referenced in Regulatory Guide 1.70, Revision 2. The single failures applied to these analyses shall be chosen so that the dynamic forces on the safety and relief valves are maximized. Test pressure shall be the highest predicted by conventional safety analysis procedures. Reactor coolant system relief and safety valve qualifications shall include qualification of associated control circuitry, piping, and supports, as well as the valves themselves.

A. Performance Testing of Relief and Safety Valves -- The following information must be provided in report form.

   1. Evidence supported by test of safety and relief valve functionability for expected operating and accident (non-ATWS) conditions must be provided to NRC. The testing should demonstrate that the valves will open and reclose under the expected flow conditions."

Under the direction of a PWR utility group, the Electric Power Research Institute (EPRI) conducted full flow tests on pressurizer safety and relief valves[5]. These tests, recently completed, involved 1 1/2 years of testing at three test sites.

Pressurizer PORVs were tested at both the Marshall Steam Station and Wyle Laboratories test site. Ten PORVs were tested (steam, steam-to-water transition, nitrogen-to-water transition, and water) with all displaying generally acceptable performance.[6,7,8]

Safety valve testing was conducted at the Combustion Engineering test site in Connecticut. Approximately 116 tests on 9 valves (with and without loop seals; steam, steam-to-water transition, and water) were conducted.[8] The tests confirmed the ability of the safety valves to open and close under their expected operating fluid conditions. However, other concerns related to safety valve performance were identified during the tests. These concerns included system overpressure protection, valve chatter (steam conditions), and inlet piping pressure oscillations (water conditions).

The purpose of this report is to address these specific safety valve performance concerns uncovered during the EPRI valve tests while generically addressing the overall issue of pressurizer safety valve qualification.

1.2 APPROACH

Insofar as the operability concerns originated due to valve observed performance differing from valve assumed performance, the report is divided into sections discussing the assumed performance, the observed performance, and the differences between them. Only the potential impact on non-ATWS transients are considered.

In addressing observed valve performance, one must be careful to differentiate between those valves and fluid conditions tested and actual valves and fluid conditions for Westinghouse designed Nuclear Power Plants.[9,10,11] While the valves and fluid conditions tested were representative, in effect enveloping, there were noticeable differences in performance between the valves and fluid conditions that are not in all cases applicable to Westinghouse designed plants.

Terminology used in this report is consistent with that given in ANSI B95.1-1977, "Terminology for Pressure Relief Devices".

As EPRI final data plots were not available prior to publication of this report, EPRI data plots marked preliminary are used.

2.0 ASSUMED VALVE PERFORMANCE

Safety valve performance characteristics are incorporated in reactor coolant system transient models as well as specific valve dynamics models used in generating fluid loads on the piping and support system. Typical models used by Westinghouse are depicted in Figure 2-1.

The system transient and valve dynamics model are not coupled. Consequently, assumptions used in these models are not necessiarily comparable and must be analyzed strictly in conjunction with the model in use.

2.1 FSAR TRANSIENT ANALYSES

The safety valves are presently modeled in FSAR transient analyses (i.e. Chapter 15 safety analyses) by prescribing an opening pressure, a linear opening characteristic with full flow being achieved at some accumulation, and a closing pressure. See Figure 2-2. The safety valve opening pressure is assumed to be 2500 psia with full capacity achieved at 2575 psia (allowing 3 percent accumulation). The safety valve closing pressure is assumed to be 2500 psia. No blowdown is assumed (conservative) as this leads to higher pressures during subsequent pressurization cycles.[12]

Valve opening time is not considered as the time steps of iteration in the transient analyses are long compared to the opening time.

Steam relief flowrates are derived by approximating the Moody curve for dry saturated steam flow. The water discharge flowrate is assumed to be 40 percent of the steam relief flowrate; this approximates the saturated volumetric liquid discharge (versus pressure) for critical two-phase flow.

Specific information detailing the methodology for determining the required pressurizer safety valve relief capacity is given in WCAP-7769.[13] By combining the number of safety valves with the flow assumptions stated above, the assumed maximum safety valve flowrate out of the pressurizer may be determined. Table 2-1 lists the assumed maximum steam and water flowrates used in the FSAR analyses. Actual safety valve design parameters for the EPRI test valves are listed in Table 2-2.

Once the assumed valve parameters are stated, the expected fluid conditions occurring at the valve inlet may be determined.

Documentation of the expected range of pressurizer safety valve fluid conditions is provided in Reference 10. In this report, the transients that result in steam and/or water discharge through the power operated relief valves (PORVs) and safety valves are discussed.

For safety valves, these transients that result in steam discharge are:

- Loss of Load
- Loss of Offsite Power
- Loss of Normal Feedwater

- Rod Withdrawal at Power
- Locked Rotor
- Rod Ejection
- Feedline Break

Within these, the loss of load and locked rotor transients are the pressure enveloping Condition II and Condition IV events, respectively. A summary of the inlet conditions for these events is given in Table 2-3.

Liquid discharge through the safety valves is predicted for only one FSAR event, the feedline break accident. Liquid temperatures and surge rates for this event range from 553 to 672 degrees Fahrenheit and 224 to 2989 gallons per minute, respectively. Pressurization rates are from 1.6 to 12 psi/sec.

An inadvertant or spurious actuation of the safety injection system at power may challenge the safety valves (depending on the pressure-head characteristics of the safety injection system). Valves, if actuated, lift on steam and, following extended operation of the safety injection system, subcooled water discharge may be observed.

In general, the valves open on steam and no liquid discharge is observed until the pressurizer becomes water solid. This is plant dependent and can vary anywhere from approximately 20 minutes to more than six hours. Consequently, the design specification for pressurizer safety valves in Westinghouse designed nuclear power plants is for steam service only.

2.2 STRUCTURAL ANALYSES

Assumptions used in structural analyses are conservatively chosen to maximize the fluid loads on the piping, valves, and supports.

For water slug (loop seal) discharge analyses, water discharge followed by steam relief is assumed. Instantaneous mass flowrates are determined from the orifice equation for subsonic flow and from the steam tables for sonic flow for the instantaneous conditions upstream of and through the valves. The upstream conditions are controlled by such factors as piping layout and loop seal size.

The assumed safety valve opening time is 0.04 second occurring at an opening pressure of 2575 psia. At this opening pressure, typical maximum flowrates for the water followed by steam discharge case are 1057 lbm/sec and 137 lbm/sec, respectively.(14)

TABLE 2-1

ASSUMED MAXIMUM SAFETY VALVE FLOWRATES*

Number of Plants	Power (MWT)	Number of Safety Valves	Safety Valve Capacity [Per Valve] (lb/hr)	Total Safety Valve Flowrate (ft^3/sec)	
				Steam	Water
2-loop					
3	1520	2	288000	19.75	7.30
2	1650	2	345000	23.66	9.47
1	1655	2	350000	24.00	9.60
3-loop					
1	1351	2	240000	16.46	6.58
3	2200	3	288800	29.63	11.85
2	2441	3	293330	30.14	12.06
4	2660	3	345000	35.49	14.20
14	2785	3	345000	35.49	14.20
1	2785	3	420000	43.21	17.28
4-loop					
1	2756	3	408000	41.97	16.79
1	3025	3	420000	43.21	17.28
2	3250	3	420000	43.21	17.28
2	3350	3	420000	43.21	17.28
1	3403	3	420000	43.21	17.28
37	3425	3	420000	43.21	17.28
2	3817	3	501700	51.61	20.65

*Assumed flowrates are linear (see Figure 2-2); values stated are for 2575 psia.

Source: Reference 12, Table I.3.

TABLE 2-2

PRESSURIZER SAFETY VALVE DESIGN PARAMETERS

Valve	Orifice Area (in^2)	Rated Lift (in)	Rated Steam Flow (lbm/hr)
Crosby 3K6	1.841	0.382	212,182
Crosby 6M6	3.644	0.538	420,000
Crosby 6N8	4.381	0.591	504,952
Dresser 31739A	2.545	0.45	297,845
Dresser 31709NA	4.34	0.588	507,918
Target Rock 69C	3.513	-	345,000

TABLE 2-3

SAFETY VALVE INLET CONDITIONS FOR LIMITING
FSAR EVENTS RESULTING IN STEAM DISCHARGE

Reference Plant	Valve Opening Pressure (psia)	Maximum Pressurizer Pressure (psia)/ Limiting Event	Maximum Pressure Rate (psi/sec)/ Limiting Event
2-Loop	2500	2682/Locked Rotor	240/Locked Rotor
3-Loop	2500	2592/Locked Rotor	216/Locked Rotor
4-Loop	2500	2555/Loss of Load	144/Locked Rotor

Source: Reference 10, Table 5-1

000364-1

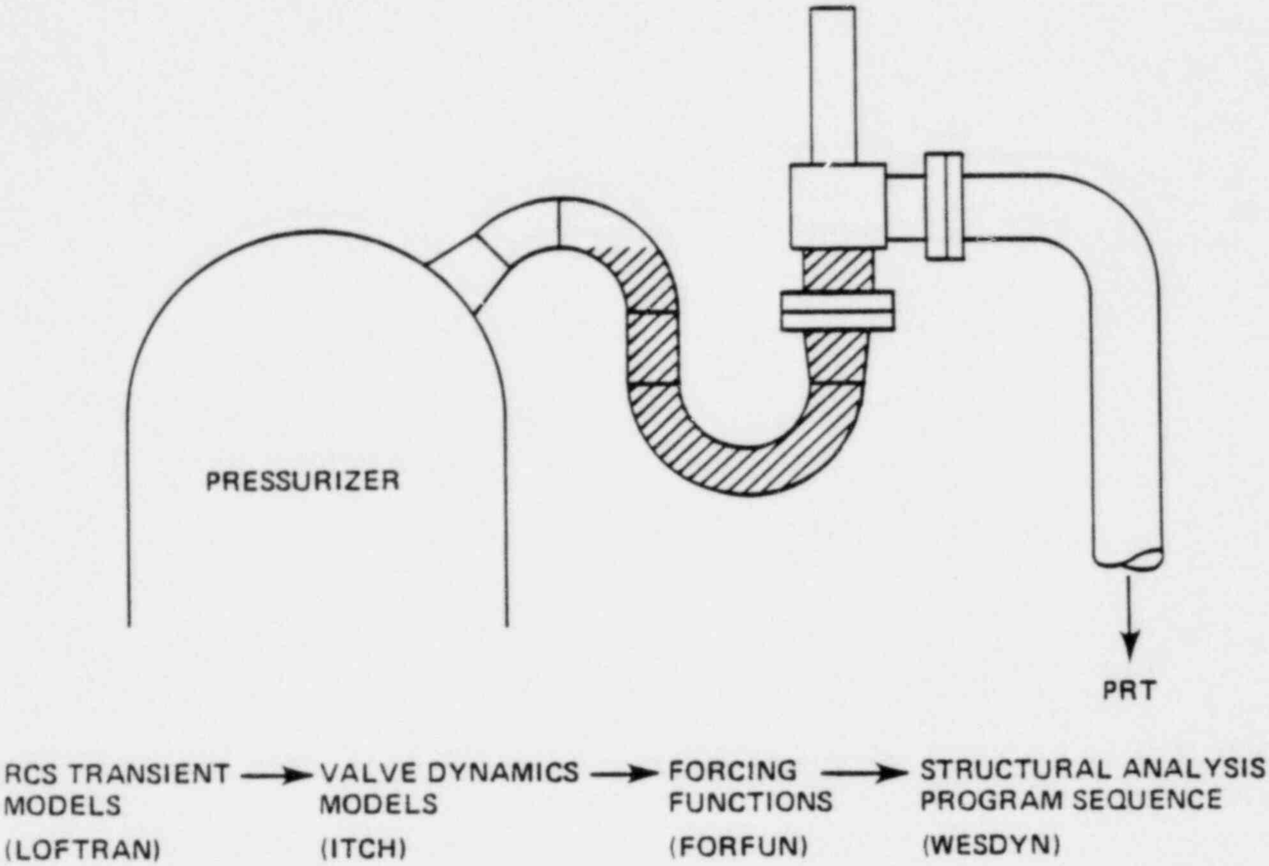


FIGURE 2-1
PRESSURIZER SAFETY VALVE MODELS

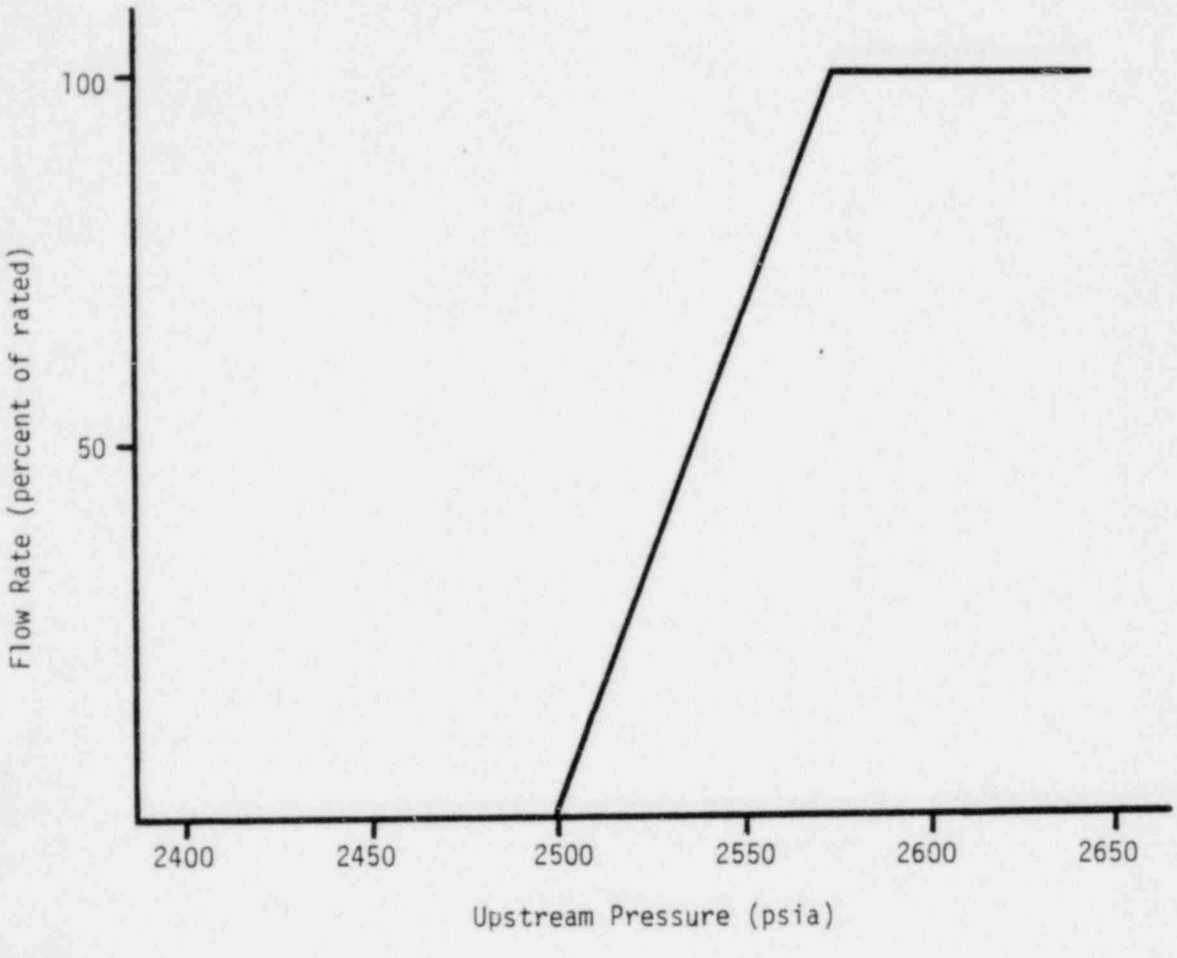


FIGURE 2-2
ASSUMED STEAM FLOW RATE FOR PRESSURIZER SAFETY VALVE

3.0 OBSERVED VALVE PERFORMANCE

Safety valve operation may be characterized by segmenting into stages the opening, blowdown and closing histories of disc travel as depicted in Figures 3-1 and 3-2 for loop seal and non-loop seal discharge cases, respectively.

For the loop seal case, the disc exhibits measureable lift at some opening pressure, oscillatory (or flutter) behavior during the loop seal water bleedoff, simmer, and finally rapid opening (popping) on steam. Non-loop seal behavior is essentially the same with the exception of no water bleedoff. Following opening, the system blows down until the valve reaches some closing pressure. If system inertia is such that the system repressurizes sufficiently, the valve may experience another cycle of operation.

3.1 VALVE OPENING CHARACTERISTIC

A summary of the number of tests for each valve, piping configuration, and fluid condition is given in Table 3-1.

Of the 117 test runs, 63 initial openings occurred with steam at the valve seat and 54 initial openings occurred with water. Within the latter category, 33 loop seal runs were conducted. Multiple openings (on both steam and water) were also recorded. A summary of the EPRI test data for loop seal, steam, and water conditions is given in Tables 3-2, 3-3 and 3-4, respectively.

3.1.1 LOOP SEAL OPENING CONDITIONS

Loop seal configurations were used for five test valves: three Crosby, one Dresser and one Target Rock. All of the valves underwent some initial lift followed by a delay to bleed off the loop seal water.

Characteristic plots of valve stem position during the delay period are shown in Figure 3-3. Figure 3-3a shows the characteristic high frequently oscillatory behavior typical of water flow through the valve. Figures 3-3b and c depict smoother oscillations. Such oscillations could accompany two-phase "flashing" flow through the valve. Each slug of steam mixed with the water is attempting to open the valve. Figures 3-3d and e show a smooth loop seal water bleedoff followed by steam simmer and valve "pop". Finally, Figure 3-3f shows a typical opening for the Crosby 6M6 valve with assist device; the opening is stepped and smooth.

All of the loop seal opening runs exhibited an opening characteristic similar to those shown in Figure 3-3.

Figure 3-4 compares the initial opening of the valves under loop seal conditions to their assumed opening. The openings are plotted against a relative opening time where an initial opening that occurs at the FSAR analysis assumption of 2500 psia prescribes the reference or zero opening point. Initial openings at pressures below 2500 psia are plotted to the left of the zero opening point while those that occur in excess of 2500 psia are plotted to the right.

Tank pressure at valve initial opening is used as pressure measurements at the valve inlet were unavailable. The pressure at the valve seat prior to opening would be slightly less ($\sim$ 1 to 2 psi) due to elevation head and compressibility losses in the inlet piping. Following opening, inlet piping and entrance losses would increase this pressure differential to approximately 20 to 35 psi.

Only the high pressurization rate openings are shown as the low pressurization rate openings occurred at some time considerably removed (tens of seconds) from the zero opening point. Such long delays in opening should not be construed as unacceptable, however, as the pressurization rate is the deciding factor: for a transient with a pressurization rate of 3 psi/sec, a delay in valve opening of ten seconds would result in a pressure rise of approximately 30 psi. This is relatively minor when compared to a one second delay in opening for a transient with a pressurization rate of 300 psi/sec.

Constant pressurization lines are shown in Figure 3-4. That the test points lie in a reasonably linear line about 300-350 psi/sec reflects the capability of the test rig: The facility was designed to provide continuous steam flow throughout the test range of 2300-2700 psia. Hence, if the valve doesn't open, the system continues to pressurize at a fairly constant rate.

The comparison to FSAR transients is twofold: First the pressurization rate does not remain linear in the absence of valve opening. Reactor coolant system pressurization is dictated mainly by the heat removal (steam generators) and core feedback models. As a result, the pressurization rate decreases appreciably with time to the point that the transient turns around even with no valve opening.

Secondly, the pressurization rates tested are in excess of those for Westinghouse designed plants. As listed in Table 2-2, the maximum expected pressurization rates for two, three, and four loop plants would be 240, 216, and 144 psi/sec, respectively. An equivalent valve opening delay in Westinghouse designed plants would, therefore, result in a lesser system pressure rise than that observed in comparable EPRI tests.

Table 3-5 lists the mean tank pressure for each valve at its initial opening. Again, the pressure at the valve seat prior to opening would be slightly less.

Mean loop seal water bleed times are given in Table 3-6. No method is available for determining the end of the water passage and the beginning of the steam simmer. Therefore, the penalty in valve opening delay time due only to the passage of the loop seal water cannot be numerically stated.

The water passage times observed in the EPRI tests are comparable to those observed in tests conducted by Westinghouse and Crosby.[15]

3.1.2 STEAM OPENING CONDITIONS

Test runs conducted with steam at the valve seat are summarized in Table 3-3. Steam opening conditions were used for both short and long (loop seal) inlet piping.

A comparison of the popping pressures observed for the high pressurization rate tests is given in Table 3-7. In the table, the popping pressures for loop seal opening for each of the valves can be seen to be greater than that for non-loop seal (steam) opening. For some of the valves, the pressure difference is significant.

A comparison of other factors that affect the popping pressure must be made.

Table 3-8 compares the effect of inlet piping and pressurization rate for steam tests on the Dresser 31739A valve. For low pressurization rates, inlet piping was observed to have a minor effect on the popping pressure: the valve popped at a lower pressure (approximately 25 psi) for a long inlet than for a short inlet. The high pressurization rate tests produced like results but with a greater pressure difference. The valve on long inlet piping popped approximately 80 psi lower than that for short inlet piping.

This suggests that the factors which effect the valve popping pressure are not limited to loop seal water alone: the inlet piping length and transient pressurization rate must also be considered.

Following the rapid "pop" opening, oscillations characteristic of a spring mass system rebounding against a stop were observed (see Figures 3-3a, e, or 3-12c). Disc movement during these oscillations was typically one or two percent of maximum travel. Additional discussion of this characteristic is presented in Section 3.4.

3.2 FLOW RATES

The steam flow rates during the discharge cycles for each of the test valves are given in Figures 3-5 through 3-10. No figure is given for the Crosby 6N8 valve as flow data was not available.

For each of the discharge cycles, the initial steam spike that accompanied valve "pop" was ignored. The ensuing steam flow rate for stable (no flutter) valve performance is then given as a function of decreasing tank pressure. For each cycle the curve starts at the right and progresses to the left as tank pressure decays.

The Crosby and Target Rock valves exhibited linear flow above some closing "knee." In this linear section, the slope of the flow curve was slightly in excess of P/P_o where the reference pressure is taken as the rated relieving pressure (2575 psia):

Crosby 6M6, $\text{Flow} = 285.7P + W_o - 7.357 \times 10^5$ lbm/hr

Crosby 3K6, $\text{Flow} = 163.6P + W_o - 4.214 \times 10^5$ lbm/hr

Target Rock 69C, $\text{Flow} = 276.4P + W_o - 7.116 \times 10^5$ lbm/hr where W_0 is the measured steam flow rate for a tank pressure of 2575 psia and P is the tank pressure in psia.

The Crosby 6M6 valve with assist (Figure 3-10) exhibited the same linear flow characteristic as that of the Crosby 6M6 valve without assist (Figure 3-5).

All three Crosby valves closed abruptly when the steam flow decreased to approximately 60 to 80 percent of rated steam flow.

As expected, the valves were not sensitive to back pressure.

In contrast to the Crosby valves, the Dresser valves drifted downwards in steam flow until a closing knee was reached at approximately 35 percent of rated steam flow.

Ring adjustments were initially required for the Dresser 31739A valve to enable the valve to reach its rated steam flow rate. Following this, all of the valves achieved their rated values.

3.3 VALVE CLOSING CHARACTERISTIC

Each valve design series has a unique closing characteristic: The Crosby safety valves maintain lift throughout most of the closing cycle (decreasing tank pressure), then close abruptly at some closing pressure. In contrast to the abrupt closing, the Dresser safety valves close gradually in identifiable steps. Figure 3-11 displays typical closing traces for the Crosby and Dresser safety valves.

The main disc position for the Target Rock valve was not measured because it was located inside the pressure boundary and the instrumentation needed was not available during conduct of the tests.

Table 3-9 lists the range of observed blowdown values for each valve. Values are given for blowdown compared to the set pressure (2500 psia) and compared to the actual popping pressure (ANSI 95.1-1977 definition).

The majority of the tests were conducted with valve blowdowns in excess of the default value stated in the ASME Code[16]:

> "Safety valves shall be adjusted to close after blowing down to a pressure not lower than 95 percent of the set pressure unless a different percentage is specified in the safety valve design specification and the basis for the setting is covered in the Overpressure Protection Report (NB-7200)."

Like the abrupt opening, the valves exhibited the same oscillating spring mass behavior on closing. Pressure pulses independent of disc motion were observed in the inlet piping with the pulses usually converging (dampening) in approximately one-half second. Increasing the valve blowdown did not affect this characteristic.

3.4 VALVE STABILITY - STEAM CONDITIONS

Ninety-six steam tests were conducted, 33 of which were initiated with filled loop seals. Of the 96 tests, four resulted in valve chatter occurring during steam flow. An additional three tests experienced flutter which did not increase to chatter.

The four tests that resulted in valve chatter on steam involved four different valves, all with long inlet piping (Tests 201, 508, 920 and 1005). For the first of these, chatter occurred on opening while for the remaining three, the chatter occurred on closing.

In Test 201, the Dresser 31709NA valve opened with a "pop" time of approximately 16 msec. With opening, pressure oscillations of approximately 160 psi peak-to-peak at 175 Hz were generated in the inlet piping. The valve, however, did not initially respond to these oscillations but remained at full lift for approximately 30 msec following which it closed, then reopened, initiating chatter. That the valve did not respond initially to the upstream pressure conditions indicates that the downstream pressure conditions (i.e. backpressure compensation) were dominant.

For the Dresser valves, the lower adjusting ring prescribes the huddle chamber and secondary orifice size. In test 201, this adjusting ring was set at -20 notches relative to the nozzle plane. This setting was unique and not used for other Dresser valve tests. It was noted, however, that negative lower adjusting ring positions for the Dresser 31739A valve (tests 302-314) resulted in incomplete lift and less than rated flow. Lower ring positions at or above the zero plane were required for satisfactory performance (in conjunction with middle ring adjustments).

The three cases of chatter following closing involved the same phenomena, except in reverse: the valve closed, generating pressure oscillations in the inlet piping, and then, following a delay, reopened and initiated chatter.

Figure 3-12 shows the inlet piping pressure oscillations and valve stem positions for test 920 (Crosby 6M6). These measurements are superimposed in Figure 3-13. Here, it is readily seen that the valve did not follow the initial pressure pulses. These initial pulses (3) are seen to be converging (dampening) while the valve makes two attempts at simmering. With the second opening attempt, the pressure pulse increases reflecting the corresponding valve closing. This suggests that the valve is dominating the pressure oscillation cycle rather than following it. By the fourth cycle, both the valve and pressure oscillations are in agreement with valve stem movement occurring approximately 1 to 2 msec after the corresponding pressure oscillation. The absence of any valve stem delay and the 1 or 2 msec phase synchronization that occurs from this point onwards suggest that the valve is now responding to the inlet piping pressure.

The explanation for valve chatter may lie in this reversal phenomena. Pressure oscillations in the inlet piping have a cycle time dependent upon the length of the inlet piping and the sonic velocity for steam. The valve opening time (simmer and pop) is dependent on valve characteristics, i.e. adjusting ring positions. For the Dresser 31739A valve, the 300 series of tests were conducted with a lower adjusting ring position of zero or negative (below the zero reference plane).

During this test series, the total valve opening time as reported in the EPRI data sheets was in the hundreds of milliseconds. This reflects the valve response time (simmer and "pop") to inlet piping pressure events. When the lower ring was raised above the zero plane during the 1000 series of tests, the total valve opening time, as expected, decreased sharply to 40-50 msec. This valve response time now approaches the pressure wave transit time for the long inlet pipe lengths tested.

The three cases of chatter following closing involved blowdown valves of 5.6 percent (Test 508, Crosby 3K6), 8.0 percent (Test 920, Crosby 6M6), and 9.4 percent (Test 1005, Dresser 31739A). For each of these, the valve performed stably (no chatter) at blowdown values less than that observed for the test resulting in chatter. Indeed, 16 of 65 steam tests with the three valves involved blowdown values less than that of the chatter tests.

Adjustments for varying valve blowdown, therefore, do not directly lead to a greater likelihood of valve chatter occurring. If, however, the adjustments decrease the valve response (opening) time then the likelihood of valve chatter may increase.

3.5 INLET PIPING PRESSURE OSCILLATIONS - WATER CONDITIONS

High frequency oscillatory behavior was observed during water passage for all of the valves. With the exception of the Target Rock valve, the oscillation produced significant pressure pulses in the inlet piping.

Figure 3-14 depicts valve stem position, inlet piping pressure, and tank pressure for a typical loop seal discharge case. Due to the compressibility of steam, the high pressure pulses are dampened considerably such that the tank (pressurizer) sees a smooth pressure trace.

Figure 3-15 depicts the pressure oscillations that occur during a typical water solid test. The lack of a steam cushion results in the pressure pulses in the tank being reduced only by the area ratio between the tank and piping.

Evaluation of the Crosby 6M6 loop seal discharge case indicates that the inlet piping was responding to internal pressure oscillations of ± 2450 psia about a steady state pressure of 2650 psia. A few individual pressure pulses exceeded 5100 psia with peak presssure as high as 6300 psia.(17) The length of the loop seal water column was approximately 8.3 feet.

Evaluation of the peak pressures occurring in the inlet piping for valves other than the Crosby 6M6 was not possible due to limitations of the pressure transducers in the piping (limited to 3400 psig).

TABLE 3-1

SAFETY VALVE TEST SUMMARY

Valve	Short Inlet Pipe			Long Inlet Pipe (loop Seal)			
	Steam	Transition	Water	Steam (Loop Seal Drained)	Steam	Transition	Water
Dresser 31739A	15	1	3	6	3	1	2
Dresser 31709NA	8	2	4	1			
Crosby 3K6	11	1	3	6	4	1	
Crosby 6M6				2	11	2, 1*	1
Crosby 6N8	5	1	2				
Target Rock 69C				2	2	1	4
Framatone/Crosby 6M6				1	5	3	2

*Drained loop seal.

TABLE 3-2a

CROSBY SAFETY VALVE TEST DATA, LOOP SEAL CONDITIONS

Valve	Test Number	Test Type[a]	Ring Positions[b] Upper	Ring Positions[b] Lower	Pressurization Rate[c] (psi/sec)	Tank Pressure (psia) at Initial Lift	Tank Pressure (psia) at Valve "Pop"	Opening Delay Time[d] (sec)	Back Pressure[e] (psia)	Steam Flow Rate[e] (lbm/hr)	Tank Pressure at Closing (psia)	Blowdown[f] (percent)	Comments
Crosby 6M6	906	S	-140	-68	3.18	2582	2580	0.86	550	426,700	2294	8.2	
	908	S			297.	2565	2688	0.92	649	441,000	2294	8.2	
	910	S			375.	2480	2628	1.15	227	447,500	2313	7.5	
	913	S	-48	-66	375.	2550	2732	0.81	242	452,000	2316	7.4	
	914	S/W			1.1	2510	2512	2.8	336	390,000	2309	7.6	Flutter during 2d cycle actuation
	917	S	-140	-68	291.	2460	2655	0.67	238	425,000	2276	9.0	
	920	S			297.	2495	2692	0.71	240	460,000	2126	15.0	Valve chattered on closing
	923	S	-190		283.	2649	2732	0.90	650	446,000	2308	7.7	
	929	S	-75	-18	319.	2600	2717	1.18	700	472,000	2373	5.1	
	931	S/W			2.5	2577	2577	0.76	-	-	-	-	
	1406	S			325.	2592	2680	0.35	245	475,000	2266	9.4	
	1415	S			360.	2553	2756	1.10	245	470,000	2346	6.2	
	1419	S			360.	2510	2675	0.89	240	467,000	2285	8.6	
Crosby 6M6 with assist	806	S	-187	-63	3.0	2435	2435	2.7	168	320,000	2245	10.2	(g)
	808	S			2.1	2433	2430	1.47	197	380,000	2276	9.0	(g)
	811	S			267.	2468	2622	0.63	390	450,000	2080	16.8	(g)
	814	S			268.	2450	2625	0.58	491	453,000	2288	8.5	
	817	S/W			2.18	2424	2423	1.93	208	380,000	2282	8.7	
	822	S			175.	2630	2708	0.94	452	369,000	2320	7.2	Assist device not used
	826	S/W			2.3	2423	2420	1.10	-	-	2275	9.0	Chpolansky assist device
	831	S/W			2.3	2420	2413	-	-	-	2278	8.9	Chpolansky assist device
Crosby 3K6	525	S	-115	-14	3.4	2536	2532	12.	445	172,500	2034	18.6	
	526	S			200.	2595	2706	0.93	520	198,000	2031	18.8	
	529	S			18.	2602	2636	1.97	385	202,500	2048	18.1	
	532	S/W			3.1	2573	2568	0.39	384	208,000	-	-	Stabilized by rope pull
	536	S			8.3	-	2677	~3.4	432	205,000	2010	19.6	

TABLE 3-2b

DRESSER SAFETY VALVE TEST DATA, LOOP SEAL CONDITIONS

Valve	Test Number	Test Type[a]	Ring Positions Upper	Middle	Lower	Pressurization Rate[c] (psi/sec)	Tank Pressure (psia) at Initial Lift	Valve "Pop"	Opening Delay Time[d] (sec)	Back Pressure[e] (psia)	Steam Flow Rate[e] (lbm/hr)	Tank Pressure at Closing (psia)	Blowdown[f] (percent)	Comments
Dresser 31739A	1016	S	-48	-40	+3	3.38	2451	2591	-	180	349,000	2266	10.0	(h)
	1017	S				3.15	2530	2676	0.62	191	375,000	2168	13.8	
	1021	S			+11	328	2582	2650	0.26	190	380,000	2177	13.5	
	1025	S/W				1.95	2524	2536	5.3	-	-	2166	13.9	

TABLE 3-2c

TARGET SAFETY VALVE TEST DATA, LOOP SEAL CONDITIONS

Valve	Test Number	Test Type[a]	Pressurization Rate[c] (psi/sec)	Tank Pressure (psia) at Initial Lift	Valve "Pop"	Opening Delay Time[j] (sec)	Back Pressure[e] (psia)	Steam Flow Rate[e] (lbm/hr)	Tank Pressure at Closing (psia)	Blowdown[f] (percent)	Comments
Target Rock 69C	703	S	2.67	2542	2543	0.21	310	382,500	2380	4.8	Valve cycled 8 times with system repressurization.
	706	S	300	2650	2700	0.30	482	434,000	2290	8.4	
	709	S/W	1.97	2508	2508	-	-	-	2390	4.4	Valve cycled open twice on steam.

NOTES TO TABLE 3-2

a. Test type: S - Steam
S/W - Steam followed by water

b. Ring positions referenced to level position.

c. Pressurization rate just prior to valve opening.

d. Time between initial lift and valve "pop".

e. Backpressure and steam flow rates are taken from EPRI data sheets identifying stable flow.

f. Blowdown is listed here as the difference between set pressure and closing pressure stated as a percentage of set pressure.

g. The opening assist device was interfering with the sensor tubing configuration, consequently valve closing was delayed.

h. The valve opened at 2451 psia and partially discharged the loop seal. It closed quickly, remained closed for approximately 54 seconds, then exhibited a characteristic steam opening at 2594 psia.

i. Initial lift for this valve is taken as the initial lift of the pilot valve.

j. Elapsed time between pilot valve opening and main disc opening (doubtful accuracy).

TABLE 3-3a

CROSBY SAFETY VALVE TEST DATA, STEAM OPENING CONDITIONS

Valve	Test Number	Test Type[a]	Inlet Piping Configuration	Inlet Piping Length[b] (feet)	Ring Positions[c] Upper	Ring Positions[c] Lower	Pressurization Rate[d] (psi/sec)	Tank Pressure At Valve "Pop" (psia)	Back Pressure[e] (psia)	Steam Flow Rate[e] (lbm/hr)	Tank Pressure At Closing (psia)	Blowdown[f] (percent)	Comments
Crosby 6M6	903	S	Loop Seal	15.71	-140	-68	291	~2490	-	-	2268	9.3	
	926	S/W			-190		1.96	2389	-	-	2267	9.3	Valve cycled open 3 times on steam, once on water.
	1411	S			-77	-18	300	2415	240	460,000	2297	8.1	
Crosby 6M6 With Assist	803	S	Loop Seal	18.38	-187	-63	283	2455	230	452,000	2295	8.2	Assist device not used
Crosby 3K6	403	S	Straight	6.83	-55	-14	-	-	-	-	2249	10.0	
	406	S					2.7	2455	680	199,000	2250	10.0	
	408	S					2.50	2461	-	-	2243	10.3	
	411	S					286	2502	616	257,000	2226	11.0	
	415	S			-35		300	2570	850	250,000	-	-	Valve fluttered during closing, manually stabilized
	416	S			-45		311	2487	708	255,000	2298	8.1	
	419	S			-38		270	2510	700	253,000	2370	5.2	
	422	S					340	2510	700	250,000	2408	3.7	Valve cycled 3 times
	425	S			-45		325	2510	150	260,000	2290	8.4	
	428	S/W					2.7	2548	-	-	2300	8.0	
	441	S					273	2490	624	260,000	2407	3.7	
	442	S			-55		308	2480	618	250,000	2240	10.4	
	506	S	Loop Seal	14.14			4.2	2707[g]	460	240,000	2330	6.8	Valve fluttered during closing.
	508	S					2.6	2507	250	242,000	-	-	Valve fluttered, then chattered during closing, manually stabilized
	516	S			-115		2.4	2435	468	187,700	2108	15.7	
	517	S					255	2463	595	217,000	2105	15.8	
	535	S					87.5	2500	550	200,000	2000	20.0	
	537	S			-95		280	2520	528	220,000	~2110	15.6	
Crosby 6N8	1202	S	Straight	9.58	-110	-18	2.0	2487	(h)	(h)	2124	15.0	
	1203	S					286	2460			2090	16.4	
	1205	S			-75		317	~2460			2144	14.2	
	1207	S			-40		317	2484			2260	9.6	
	1208	S					325	~2450			2256	9.8	
	1209	S/W					2.6	2466			2288	8.5	Valve cycled 4 times

TABLE 3-3b

DRESSER SAFETY VALVE TEST DATA, STEAM OPENING CONDITIONS

Valve	Test Number	Test Type[a]	Inlet Piping Configuration	Inlet Piping Length[b] (feet)	Ring Positions[c] Upper	Ring Positions[c] Mid.	Ring Positions[c] Lower	Pressurization Rate[d] (psi/sec)	Tank Pressure At Valve "Pop" (psia)	Back Pressure[e] (psia)	Steam Flow Rate[e] (lbm/hr)	Tank Pressure At Closing (psia)	Blowdown[f] (percent)	Comments
Dresser	302	S	Stra'ght	6.83	-48	0	-13	3.75	2483	90	165,000	2338	6.5	(i)
31739A	304	S						300	2526	132	260,000	2370	5.2	(i)
	306	S						320	2557	166	320,000	2350	6.0	(i)
	308	S						330	2547	230	300,000	2393	4.1	(i)
	310	S				-1	-9	343	2557	167	340,000	2337	6.5	(i)
	312	S				0	-13	360	2513	246	290,000	2395	8.2	(i)
	314	S				-20	-6	333	2537	183	357,000	2320	7.2	(i)
	316	S				-40	0	320	2590	200	390,000	2188	12.5	
	318	S						273	2483	200	383,000	2164	13.4	
	320	S						316	2580	293	377,000	2340	6.4	
	322	S						311	2530	250	383,000	2237	10.5	
	324	S				-60		325	2570	337	393,000	2200	12.0	
	32[illegible]	S						333	2500	204	343,000	2085	16.6	
	328	S			48			311	2527	363	390,000	2260	9.6	
	1003	S	Loop Seal	14.97	-48			2.3	2460	140	260,000	2323	7.1	
	1005	S						248	2425	546	296,700	-	-	Valve chattered on closing
	1008	S				-30	+11	275	2447	475	370,000	2160	13.6	
	1011	S				-60	+5	286	2478	410	372,000	2190	12.4	
	1012	S				-40	+3	309	2490	420	362,500	2250	10.0	
	1018	S					+11	308	2455	287	370,000	2211	11.6	
	1104	S	Straight	6.83				316	2550	560	370,000	-	-	Valve cycled twice
	1107	S/W						2.8	2488	-	-	2038	18.5	
Dresser	201	S	Loop Seal	15.22	-48	-34	-20	340-425	2488	-	-	-	-	Valve chattered on opening.
31709NA	603	S	Straight	6.50		-60	0	2.9	2505	185	560,000	2159	13.6	
	606	S						246	2504	204	633,400	2168	13.3	
	611	S						322	2535	374	641,650	2280	8.4	
	614	S				-40		317	2546	312	675,000	2294	8.2	
	615	S				-20		317	2570	339	603,000	2330	6.8	
	618	S				-60		236	2487	525	618,750	2238	10.5	
	620	S				-20		317	2540	205	642,000	2227	10.9	
	623	S/W				-60		2.46	2545	-	-	2054	17.8	
	628	S/W				-20		2.56	2530	-	-	2090	16.4	
	1305	S						308	2530	286	640,000	2302	7.9	

TABLE 3-3c

TARGET ROCK SAFETY VALVE TEST DATA, STEAM OPENING CONDITIONS

Valve	Test Number	Test Type[a]	Inlet Piping Configuration	Inlet Piping Length[b] (feet)	Pressurization Rate[d] (psi/sec)	Tank Pressure At Valve "Pop" (psia)	Back Pressure[e] (psia)	Steam Flow Rate[e] (lbm/hr)	Tank Pressure At Closing (psia)	Blowdown[f] (percent)	Comments
Target Rock	722	S	Loop Seal	18.13	311	2612	430	400,000	2490	0.4	
69C	733	S			307	2643	63	446,000	2410	3.6	

NOTES TO TABLE 3-3

a. Test type: S-Steam

S/W - Steam followed by water

b. Center line length

c. Ring positions referenced to level position

d. Pressurization rate just prior to valve opening

e. Backpressure and steam flow rates are taken from EPRI data sheets identifying stable flow.

f. Blowdown is given here as the difference between set pressure and closing pressure stated as a percentage of set pressure.

g. Considered to be in error; valve set pressure readjusted following run.

h. Due to inlet piping configuration, valve flow data is unavailable for this test series.

i. Valve did reach rated lift

TABLE 3-4a

CROSBY SAFETY VALVE TEST DATA, WATER SOLID CONDITIONS

Valve	Test Number	Inlet Piping Configuration	Ring Positions[a] Upper	Lower	Water Temperature (°F)	Pressurization Rate[b] (psi/sec)	Tank Pressure at Initial Lift (psia)	Water Flow Rate[c] (lbm/hr)	Tank Pressure at Closing (psia)	Comments
Crosby 6M6	932	Loop Seal	-75	-18	535	3.0	2501	-	(d)	Valve Chattered on opening.
Crosby 6M6 w/assist	819	Loop Seal	-187	-63	545	2.8	2425	360,000	2292	
	825				388	4.0	2416	418,000	(e)	
Crosby 3K6	431	Straight	-45	-14	Sat	1.8	2342	441,000	2177	Valve underwent two stable discharge cycles.
	435				522	1.7	2454	187,500	(e)	
	438				550	2.3	2450	-	(e)	Valve chattered on opening.
Crosby 6N8	1211	Straight	-40	-18	Sat	4.6	2450	(f)	1980	
	1213				536	3.1	2526	(f)	(d)	Valve chattered on opening

TABLE 3-4b

DRESSER SAFETY VALVE TEST DATA, WATER SOLID CONDITIONS

Valve	Test Number	Inlet Piping Configuration	Ring Positions Upper	Middle	Lower	Water Temperature (°F)	Pressurization Rate[b] (psi/sec)	Tank Pressure at Initial Lift (psia)	Water Flow Rate[c] (lbm/hr)	Tank Pressure at Closing (psia)	Comments
Dresser 31739A	1027	Loop Seal	-48	-40	+11	626	3.2	2350	775,000	(e)	
	1939					518	1.8	2408	-	(d)	Valve chattered on opening
	1110	Straight				Sat	2.3	2521	470,000	2096	
	1112					538	3.1	2387	373,000	2208	
	1114					421	3.2	2470	370,000	(e)	
Dresser 31709NA	625	Straight	-48	-60	0	Sat	3.0	2412	679,000	2108	
	630			-20		Sat	2.5	2394	1,092,700	1950	
	1308					562	1.8	2503	410,000	2416	
	1311					415	2.6	2558	-	(d)	Valve cycled 5 times followed by chatter.

TABLE 3-4c

TARGET ROCK SAFETY VALVE TEST DATA, WATER SOLID CONDITIONS

Valve	Test Number	Inlet Piping Configuration	Water Temperature (°F)	Pressurization Rate [b] (psi/sec)	Tank Pressure at Initial Lift (psia)	Water Flow Rate[c] (lbm/hr)	Tank Pressure at Closing (psia)	Comments
Target Rock 69C	712	Loop Seal	Sat	2.8	2485	635,000	2191	
	714		565	2.2	2462	850,000	2424	Valve Cycled 8 times
	717		410	2.6	2488	810,000	1910	
	719		394	0.7	2487	581,800	2235	

NOTES TO TABLE 3-4

a. Ring positions referenced to level position.

b. Pressurization rate just prior to valve opening.

c. Water flow rates are taken from EPRI data sheets identifying stable flow.

d. Test terminated by rope pull.

e. Valve reseat data was unavailable.

f. Due to inlet piping configuration, valve flow data is unavailable for this test series.

TABLE 3-5

MEAN INITIAL OPENING PRESSURE-LOOP SEAL CONDITIONS

Valve	Sample Size	Opening Pressure* (psia)
Crosby 6M6	13	2547.9 ± 54.2
Crosby 6M6 W/Assist	6	2438.8 ± 17.3
Crosby 3K6	4	2576.5 ± 29.7
Dresser 31739A	4	2521.8 ± 53.9
Target Rock 69C	3	2566.7 ± 74.1

*Tank pressure

TABLE 3-6

MEAN LOOP SEAL BLEED TIME[1]

	Valve	Sample Size	Bleed Time (sec)
Low pressurization rate[2]:	Crosby 6M6	3	1.47 ± 1.15
	Crosby 6M6 W/Assist	4	1.80 ± 0.69
	Crosby 3K6	4	4.44 ± 5.19
	Dresser 38739A	1	5.3
	Target Rock 69C	1	0.21
High pressurization rate:	Crosby 6M6	10	0.87 ± 0.25
	Crosby 6M6 w/Assist	3	0.72 ± 0.20
	Crosby 3K6	1	0.93
	Dresser 31739A	2	0.44 ± 0.25
	Target Rock 69C	1	0.30

1. Time from initial opening to valve "pop"

2. Tests listed under low pressurization rates involved rates of 18 psi/sec or less. High pressurization rate tests were 175 psi/sec or greater. No tests with pressurization rates between these values were conducted.

TABLE 3-7

MEAN POPPING PRESSURE FOR HIGH PRESSURIZATION RATE TESTS*

Valve	Loop Seal Conditions Sample Size	Loop Seal Conditions Popping Pressure (psia)	Steam Conditions Sample Size	Steam Conditions Popping Pressure (psia)
Crosby 6M6	10	2695.5 ± 39.0	2	2452.5 ± 53.0
Crosby 6M6 w/Assist	2	2623.5 ± 2.1	1	2455
Crosby 3K6	1	2706	10	2504.2 ± 28.8
Crosby 6N8	-	-	4	2463.5 ± 14.5
Dresser 31739A	1	2650	19	2519.0 ± 46.5
Dresser 31709NA	-	-	7	2530.3 ± 27.4
Target Rock 69C	1	2700	2	2627.5 ± 21.9

*Tank pressure

TABLE 3-8

COMPARISON OF MEAN POPPING PRESSURE FOR
DRESSER 31739A UNDER STEAM CONDITIONS*

Configuration	Sample Size	Popping Pressure (psia)
Short inlet piping		
Low pressurization rate	2	2485.5 ± 3.5
High pressurization rate	14	2540.5 ± 30.1
Long inlet piping		
Low pressurization rate	1	2460
High pressurization rate	5	2459.0 ± 25.7

*Tank pressure

TABLE 3-9

RANGE OF OBSERVED VALVE BLOWDOWN VALUES

Valve	Range of Blowdown Values (percent) Related to Set Pressure	Related to Popping Pressure
Crosby 3K6	3.7 - 19.6	4.1 - 27.0
Crosby 6M6	5.1 - 9.4a	4.9 - 14.8
Crosby 6M6 W/Assist	8.5 - 9.0b	5.4 - 13.5
Crosby 6N8	8.5 - 16.4	7.1 - 14.8
Dresser 31739A	4.1 - 13.8	5.5 - 20.3
Dresser 31709NA	6.8 - 13.6	9.1 - 19.6
Target Rock 69C	0.4 - 8.4	4.7 - 16.4

a. Run 920 not used

b. Runs 806, 808, 811 not used due to assist device malfunction

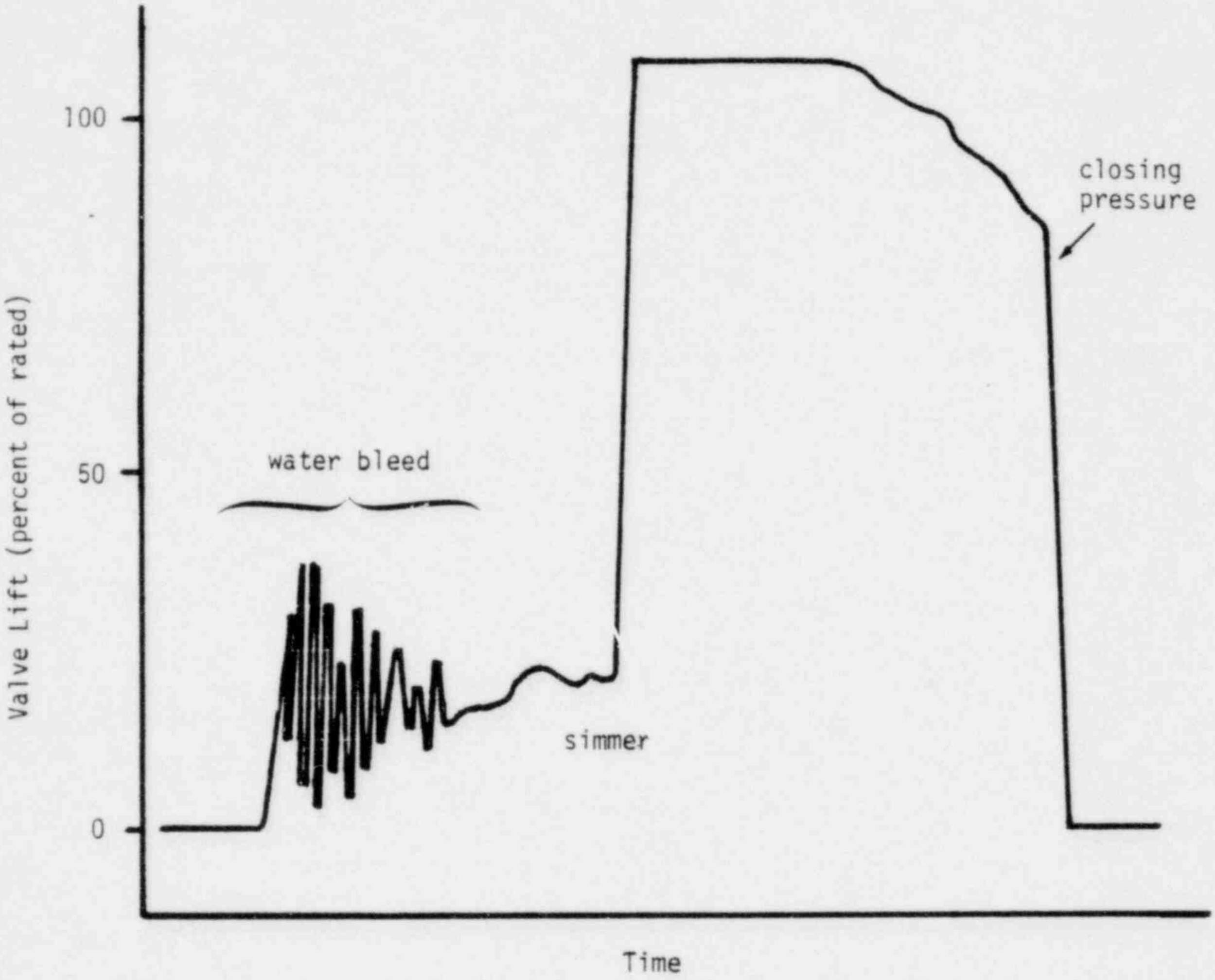


FIGURE 3-1
TYPICAL LOOP SEAL DISCHARGE CYCLE

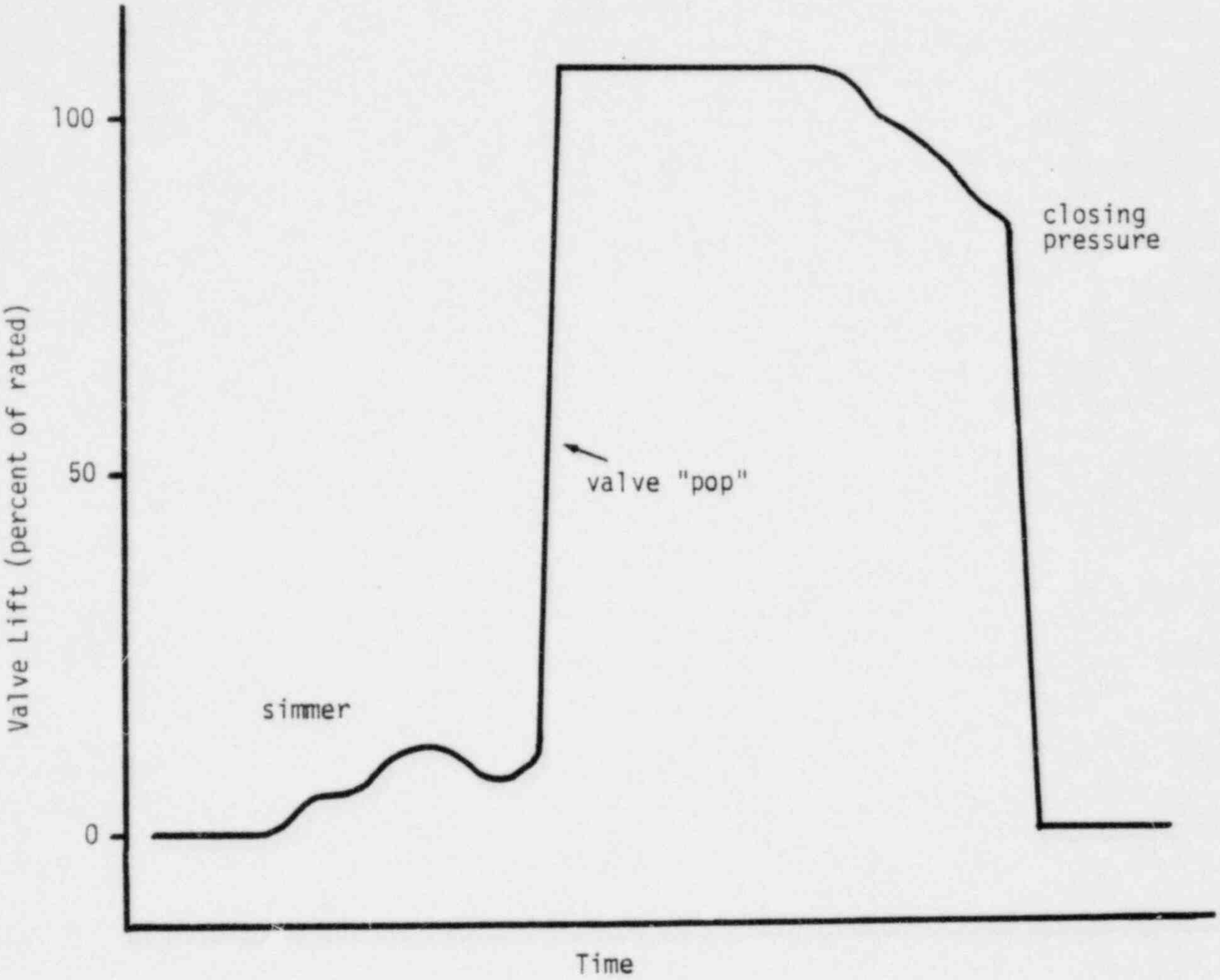
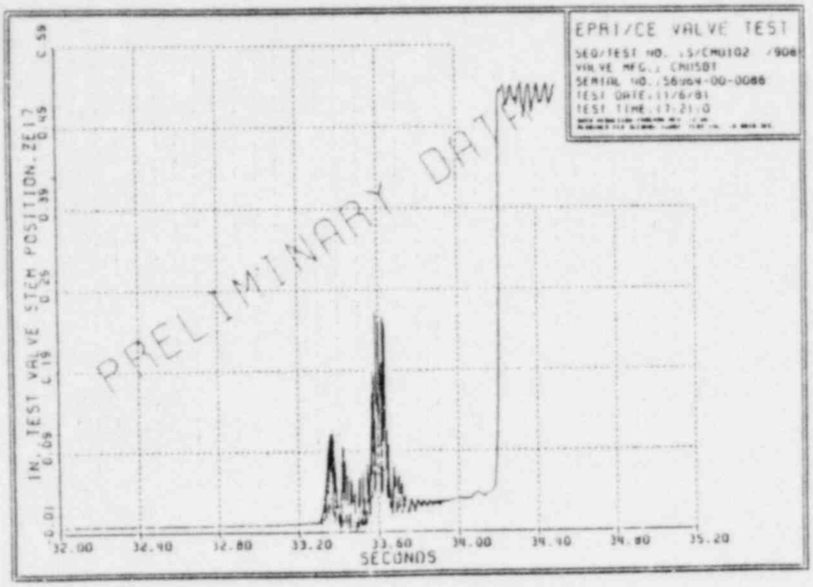


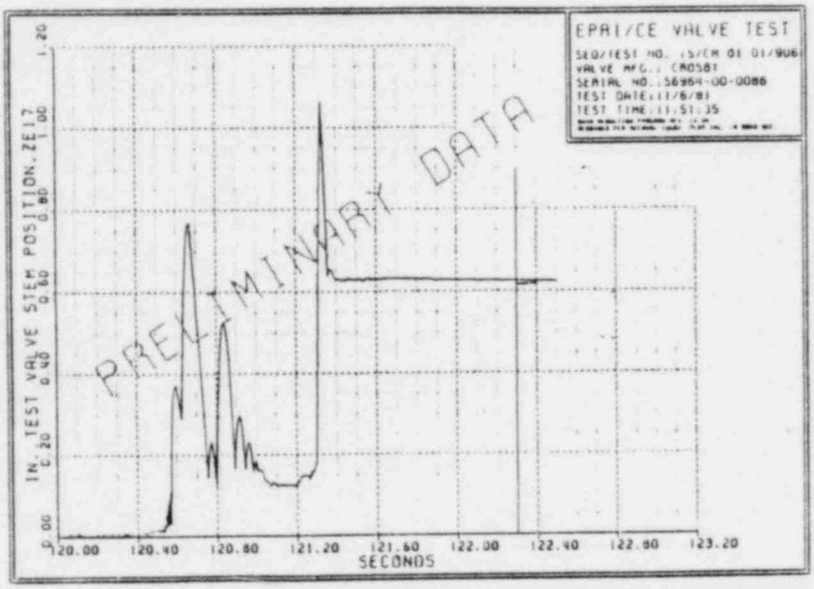
FIGURE 3-2
TYPICAL STEAM DISCHARGE CYCLE

FIGURE 3-3

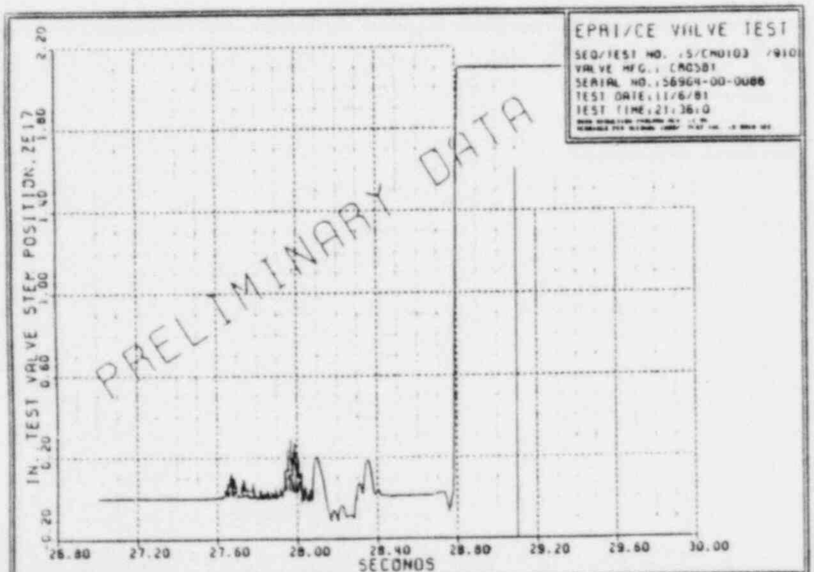
TYPICAL LOOP SEAL OPENING CHARACTERISTICS



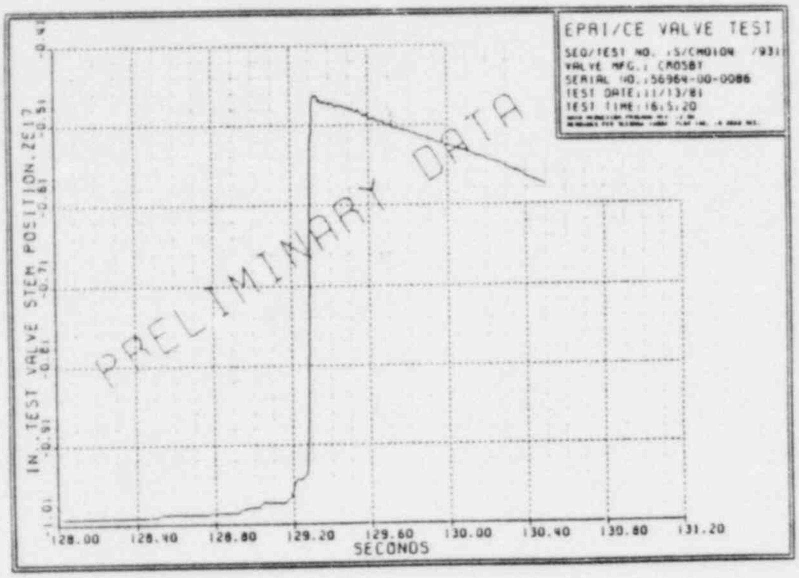
a. high frequency water oscillations



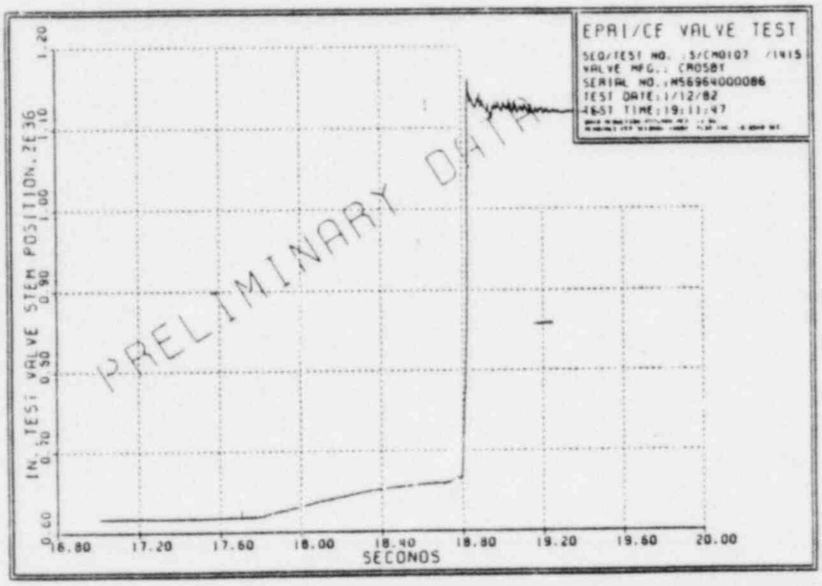
b. steam slugging



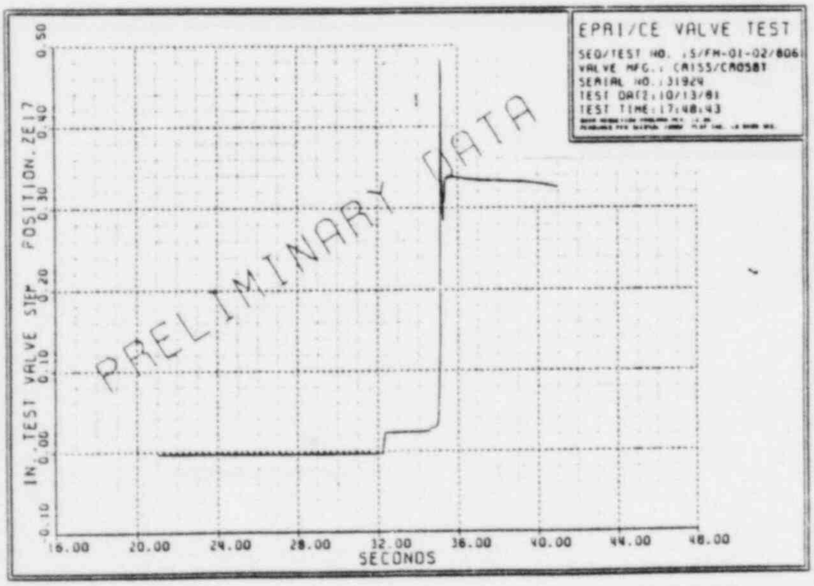
c. water oscillations followed by slugging



d. smooth water bleedoff



e. smooth water bleedoff



f. smooth water bleedoff by use of assist device

FIGURE 3-4

RELATIVE OPENING OF SAFETY VALVES UNDER HIGH PRESSURIZATION RATE, LOOP SEAL CONDITIONS

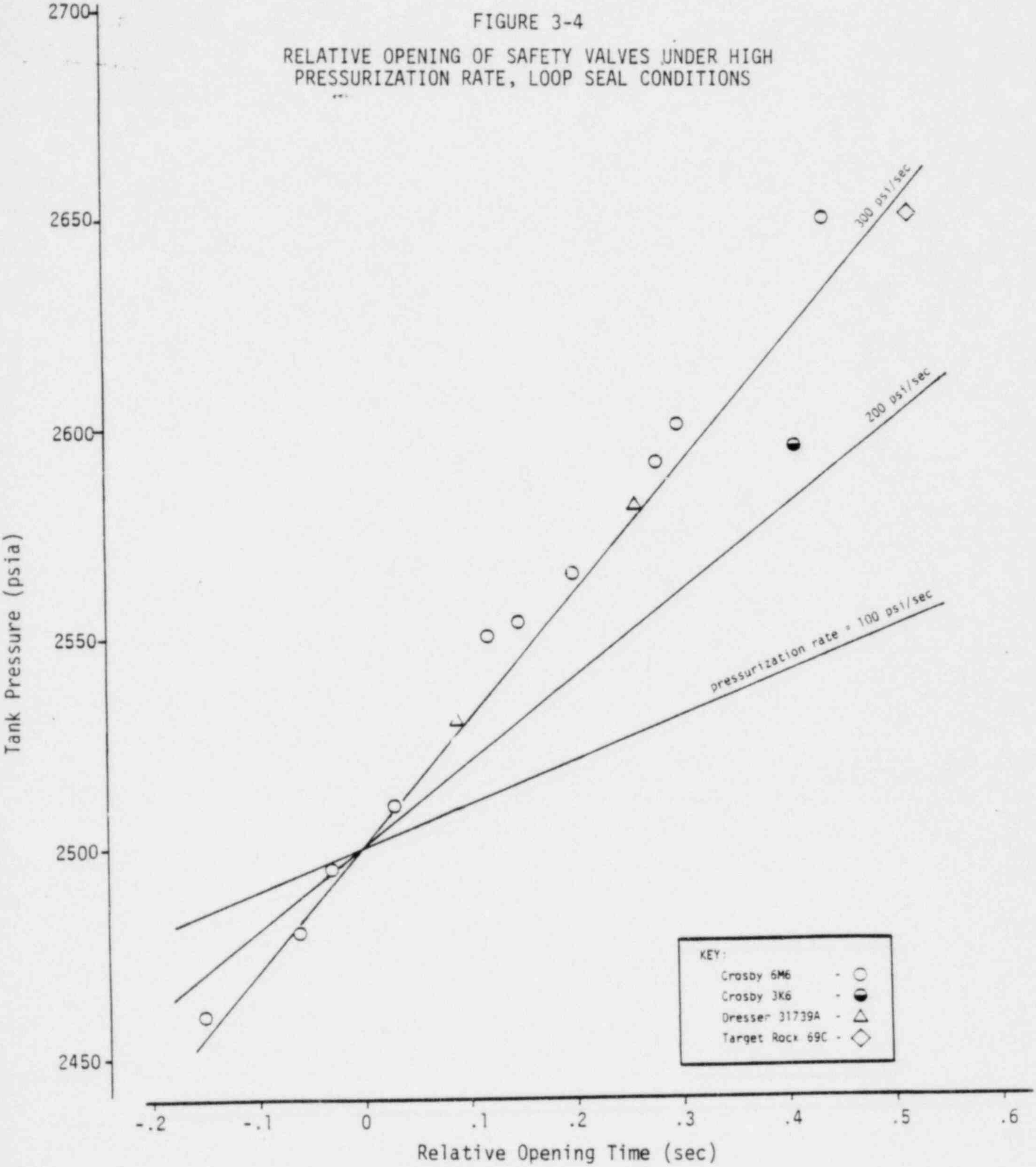


FIGURE 3-5
STEAM DISCHARGE CYCLE (STABLE)
CROSBY 6M6

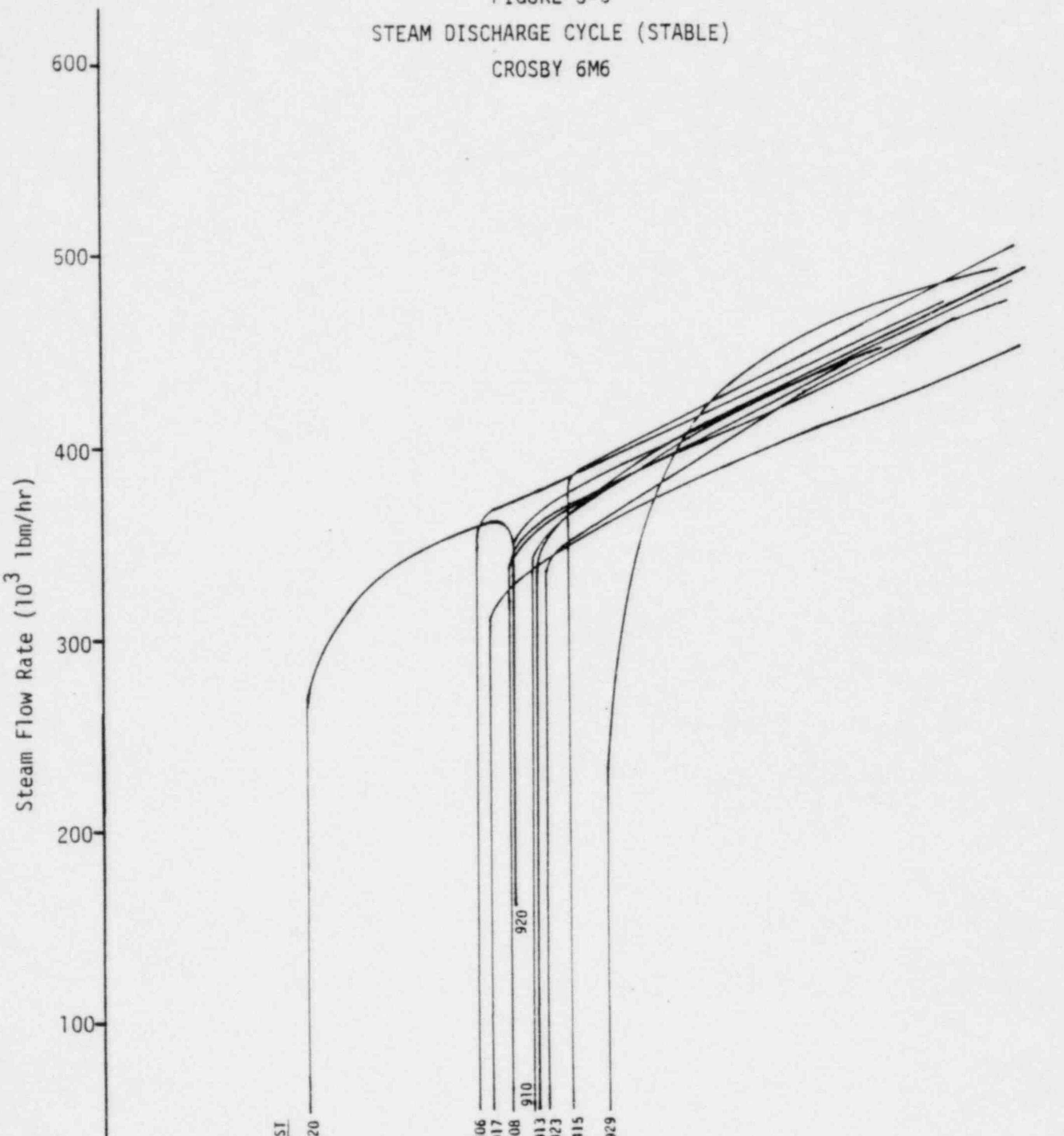


FIGURE 3-6
STEAM DISCHARGE CYCLE (STABLE)
CROSBY 3K6

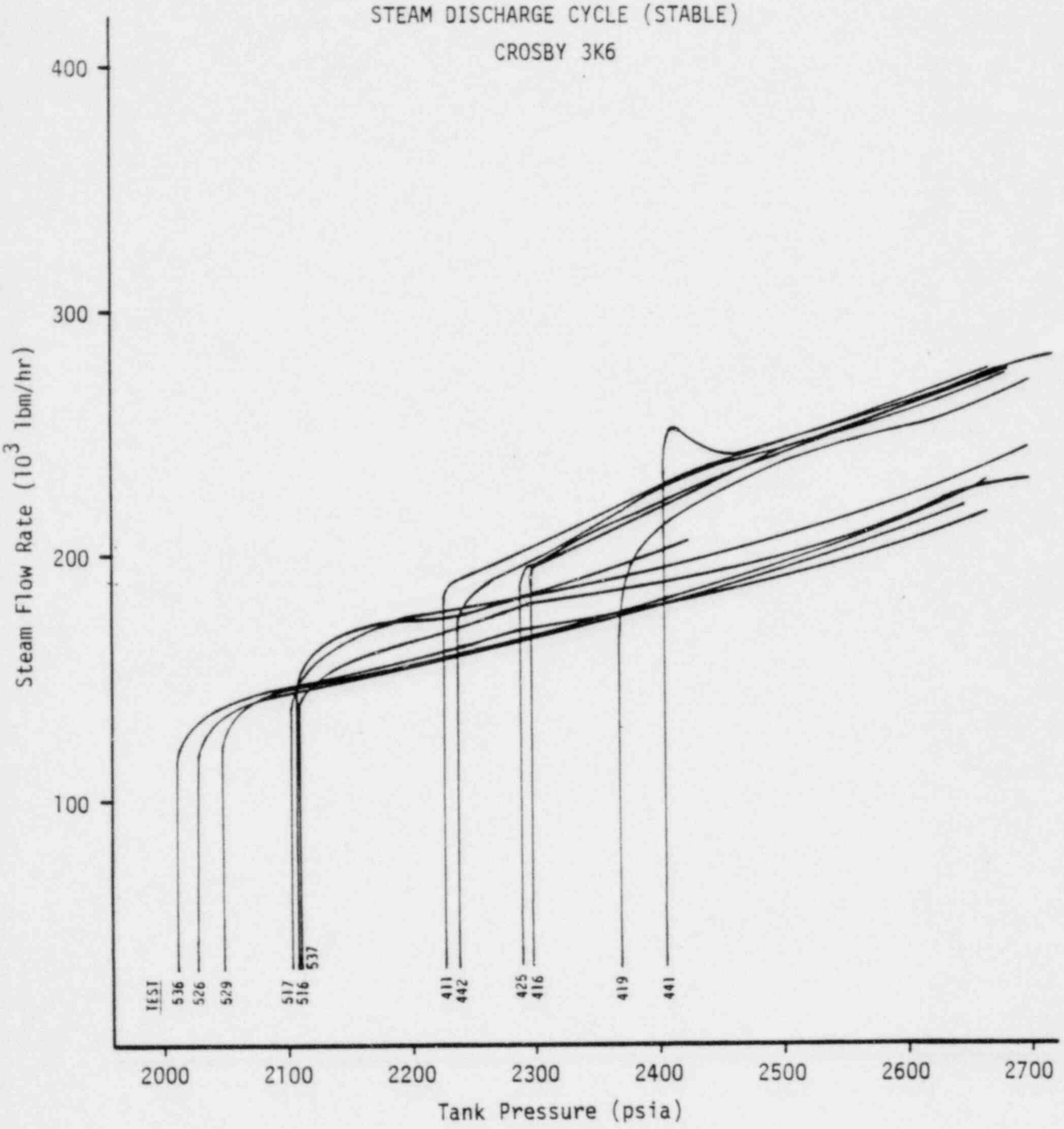
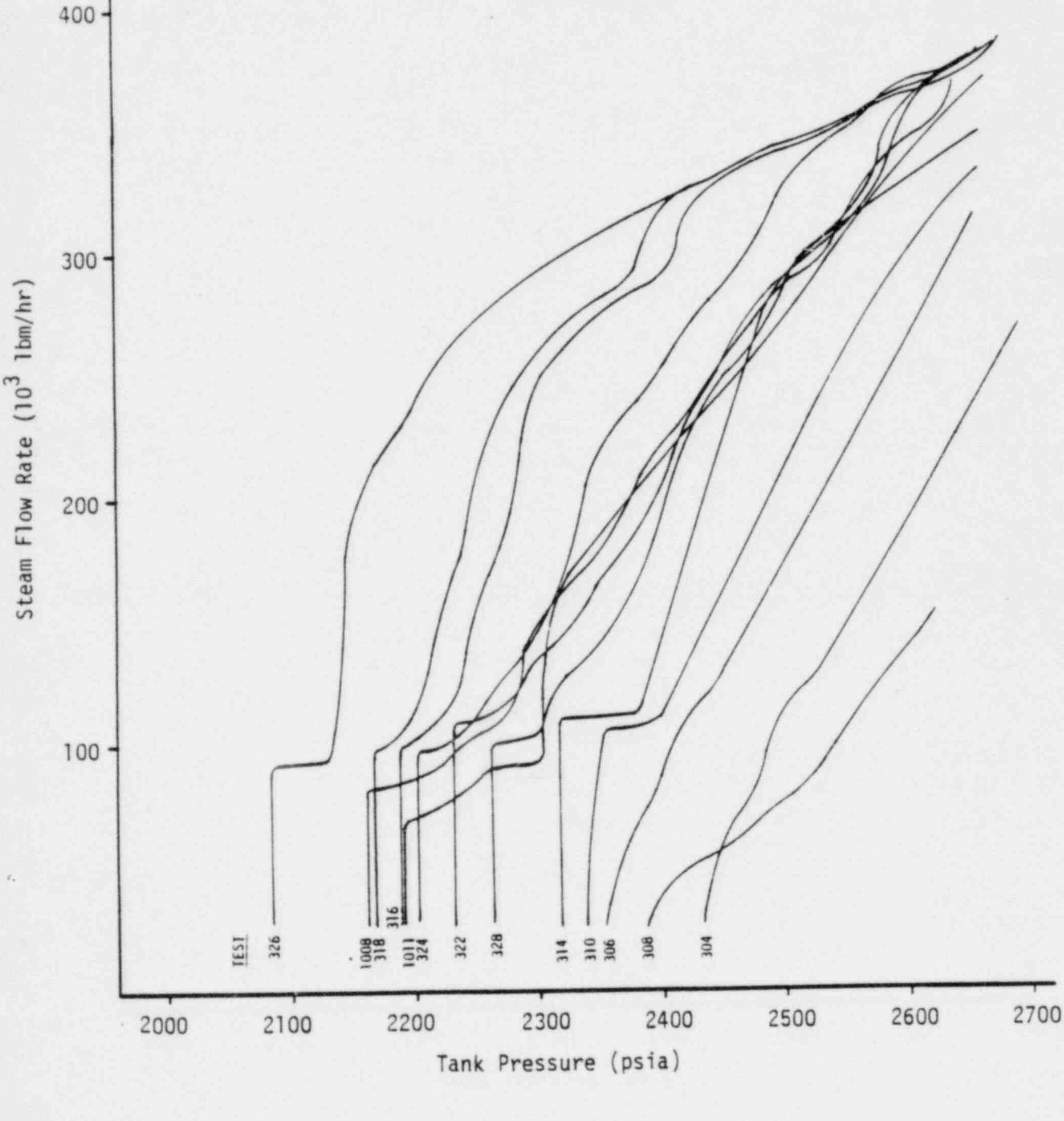


FIGURE 3-7
STEAM DISCHARGE CYCLE (STABLE)
DRESSER 31739A



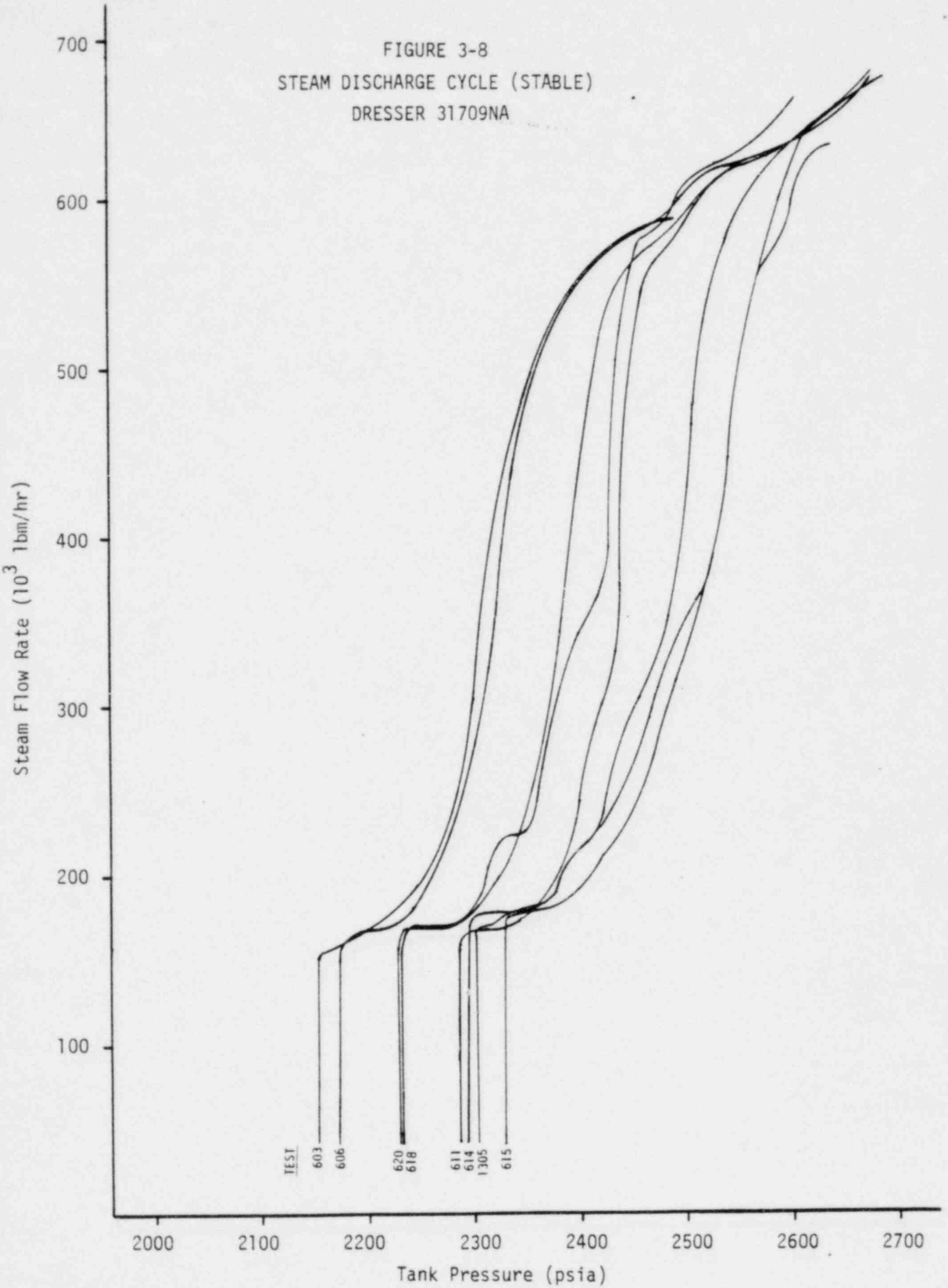


FIGURE 3-8
STEAM DISCHARGE CYCLE (STABLE)
DRESSER 31709NA
Steam Flow Rate (10^3 lbm/hr)
700
600
500
400
300
200
100
TEST
603
606
620
618
611
614
1305
615
2000
2100
2200
2300
2400
2500
2600
2700
Tank Pressure (psia)

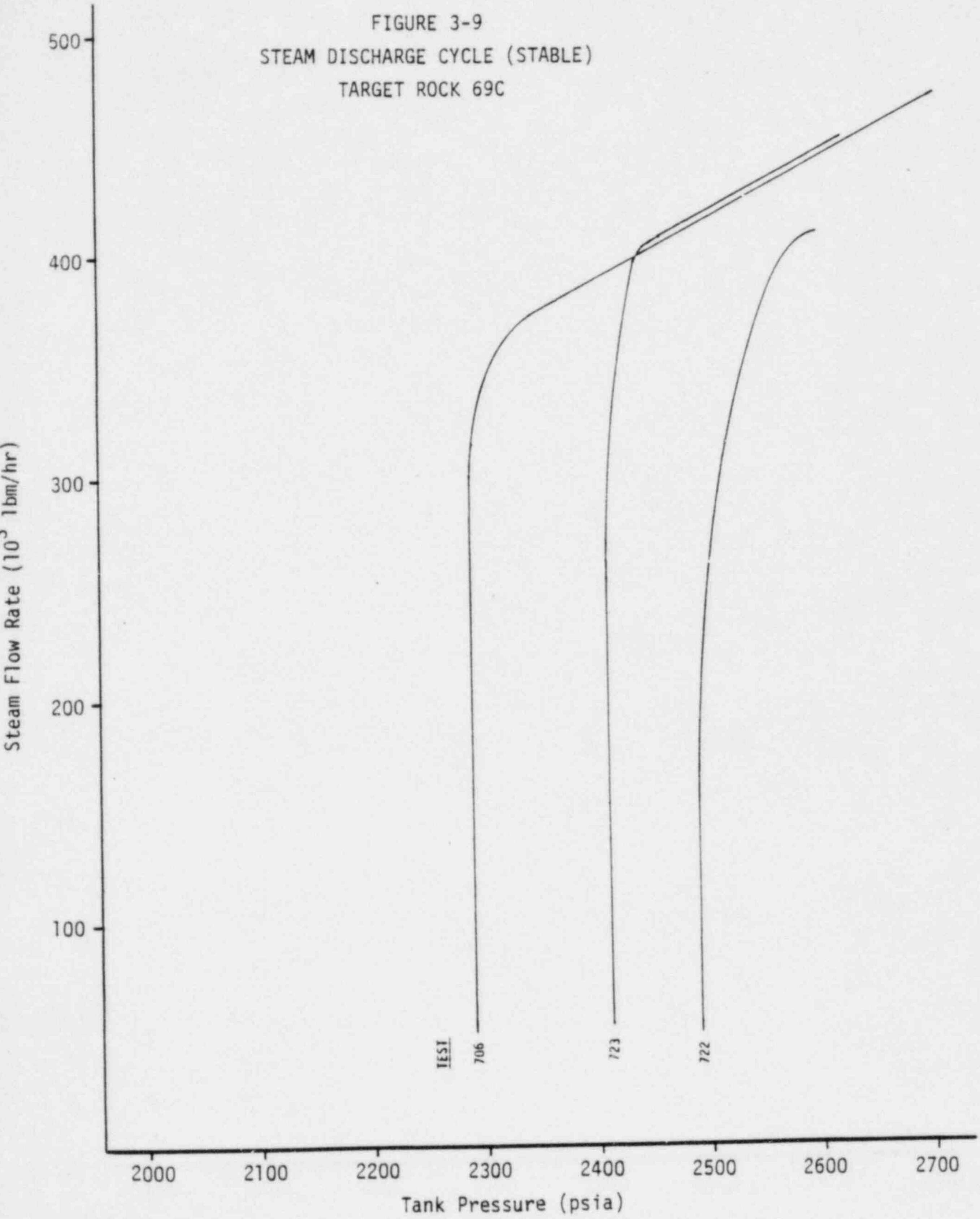


FIGURE 3-9
STEAM DISCHARGE CYCLE (STABLE)
TARGET ROCK 69C
500
400
300
200
100
Steam Flow Rate (10^3 lbm/hr)
TEST
706
723
722
2000
2100
2200
2300
2400
2500
2600
2700
Tank Pressure (psia)

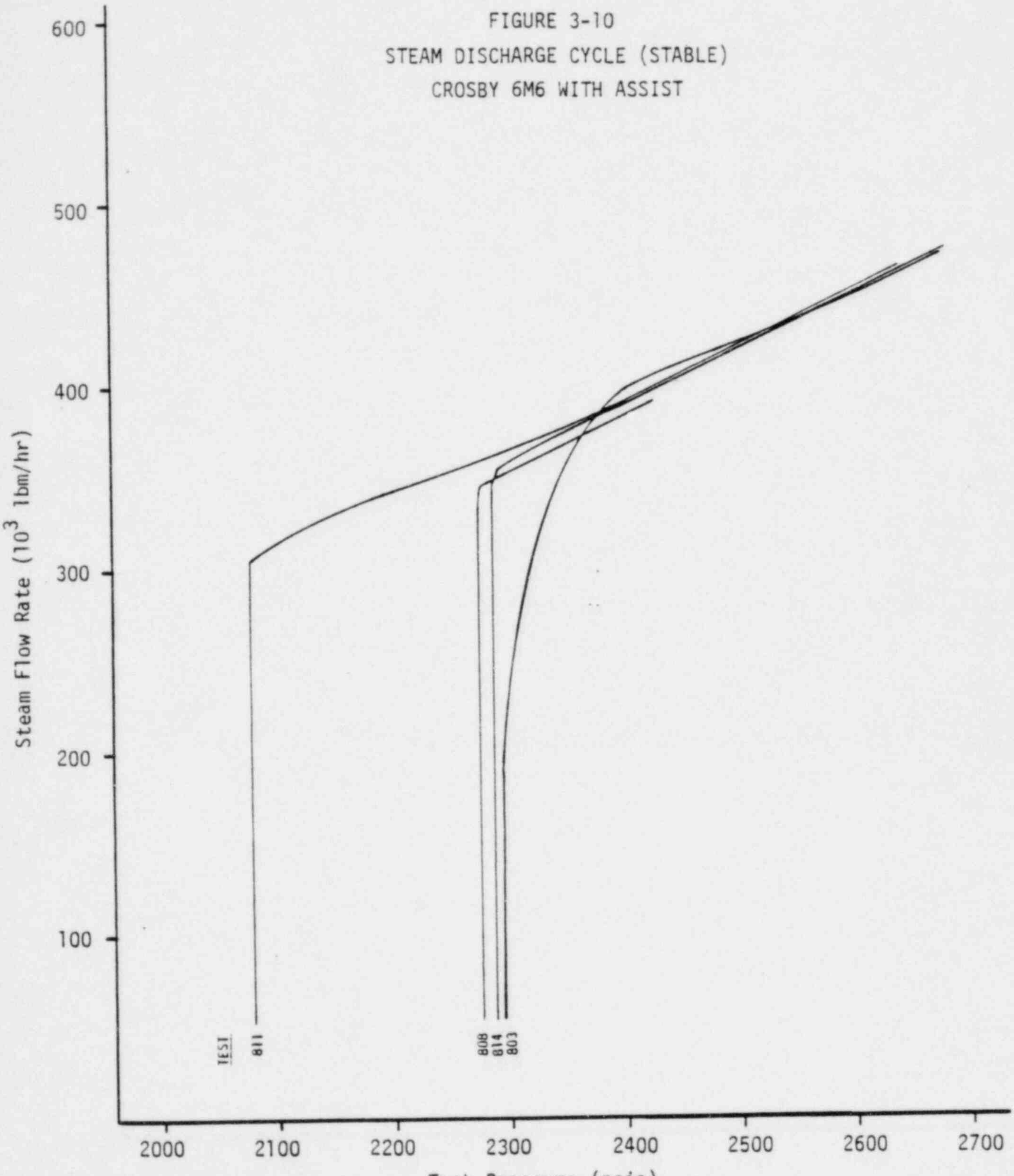
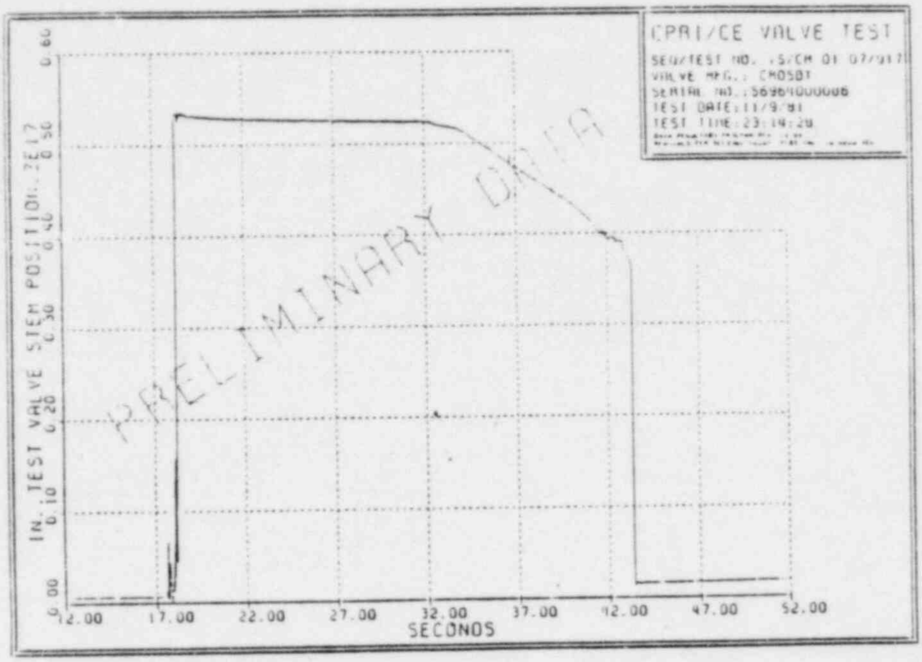


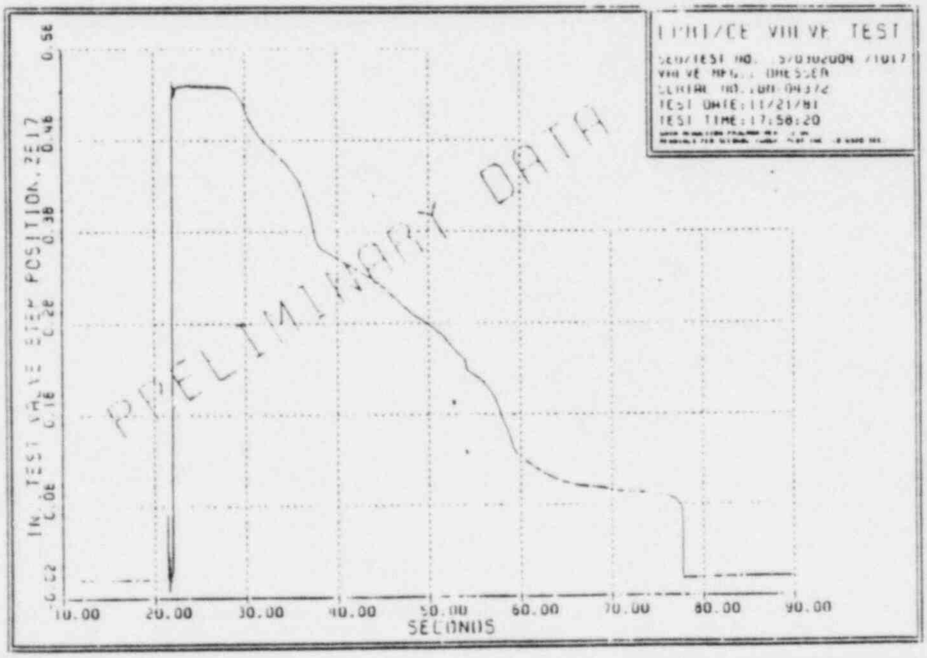
FIGURE 3-10
STEAM DISCHARGE CYCLE (STABLE)
CROSBY 6M6 WITH ASSIST
Steam Flow Rate (10^3 lbm/hr)
600
500
400
300
200
100
TEST
811
808
814
803
2000
2100
2200
2300
2400
2500
2600
2700
Tank Pressure (psia)

FIGURE 3-11

TYPICAL SAFETY VALVE CLOSING CHARACTERISTICS



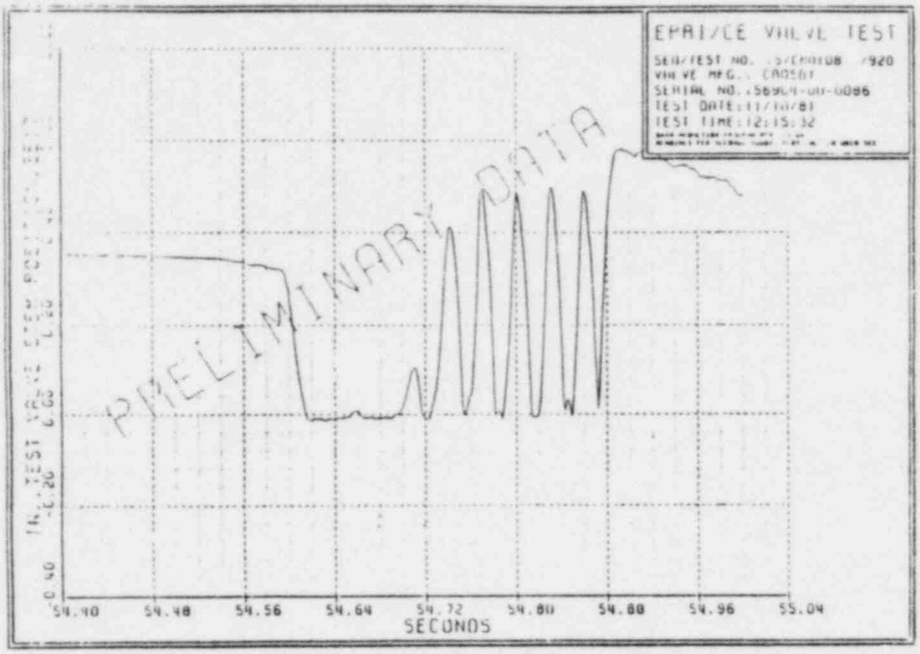
a. Crosby safety valve



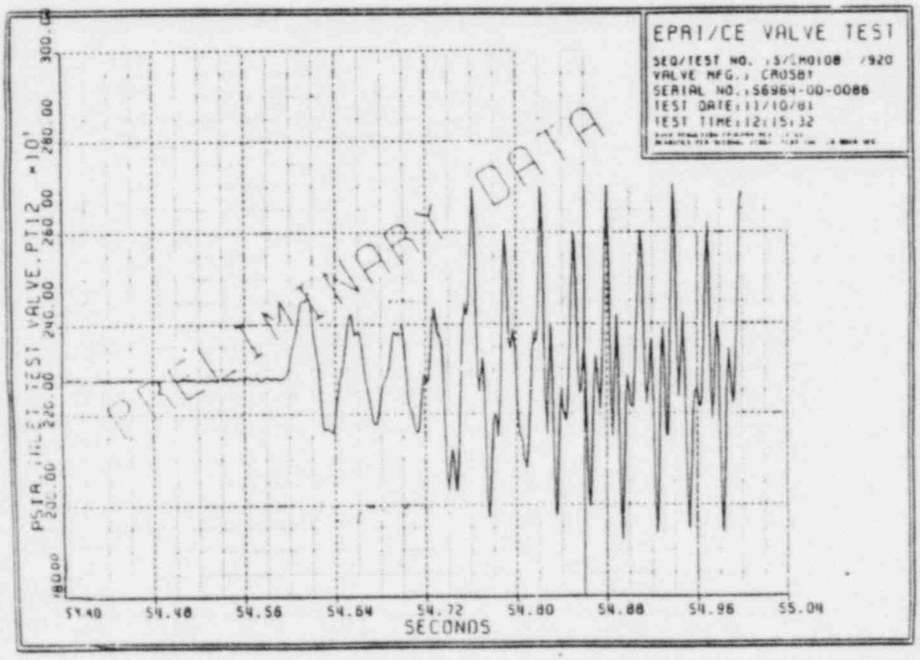
b. Dresser safety valve

FIGURE 3-12

ONSET OF CHATTER FOLLOWING CLOSING
FOR CROSBY 6M6 VALVE



a. Valve stem position



b. Inlet piping pressure

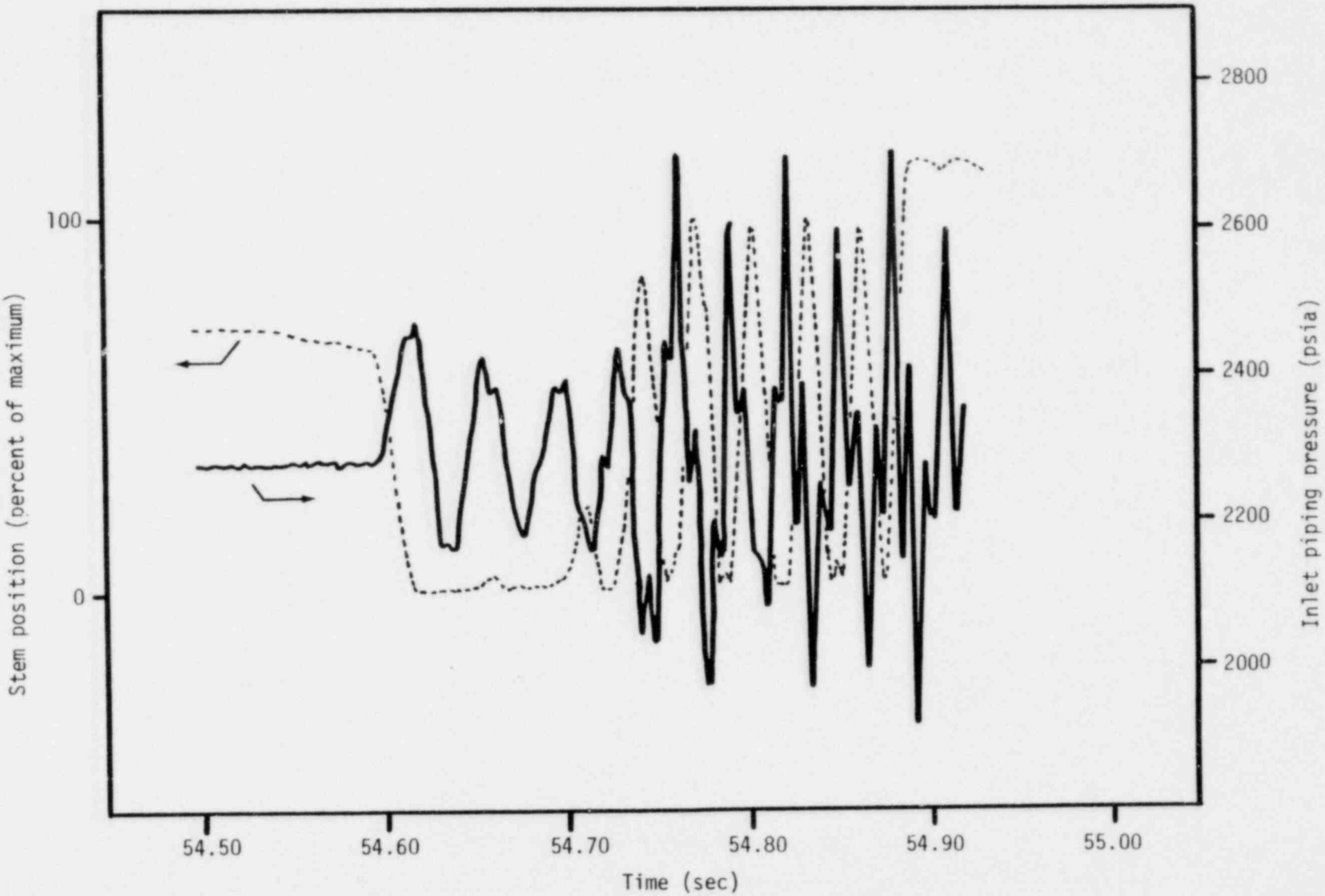


FIGURE 3-13. ONSET OF CHATTER FOLLOWING CLOSING FOR CROSBY 6M6 VALVE

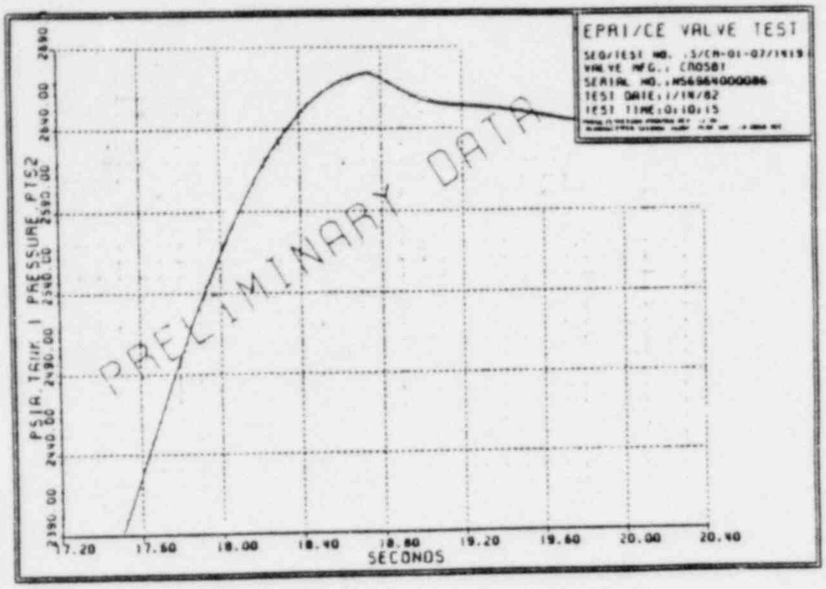
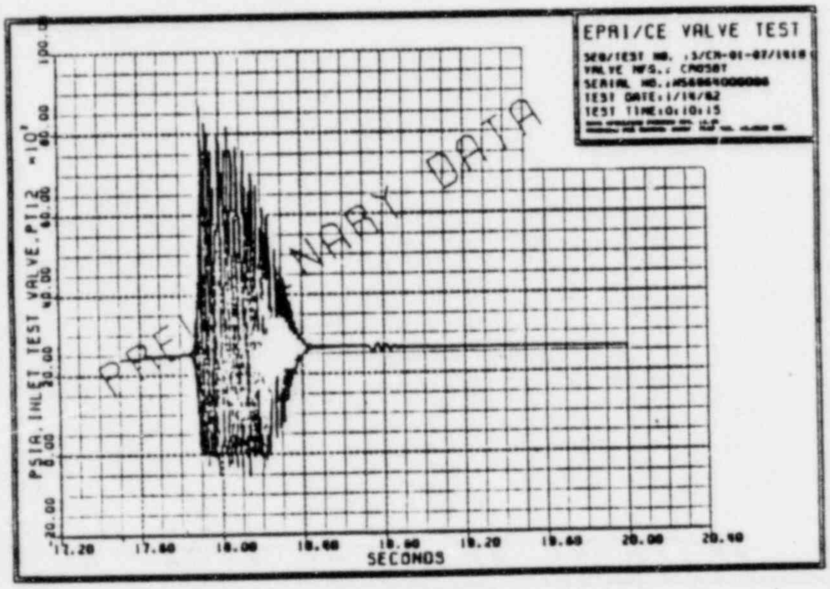


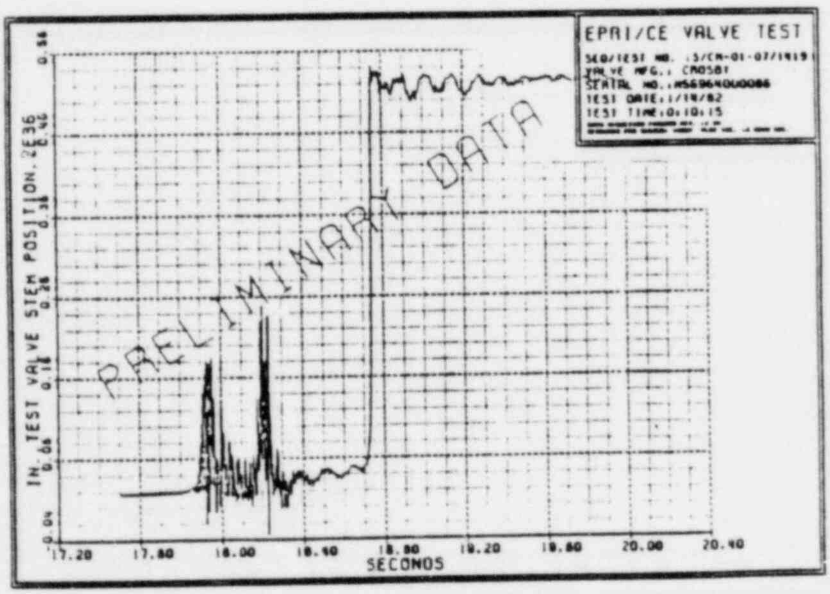
FIGURE 3-14

INLET PIPING PRESSURE OSCILLATIONS DURING LOOP SEAL DISCHARGE

a. tank pressure



b. inlet piping pressure



c. valve stem position

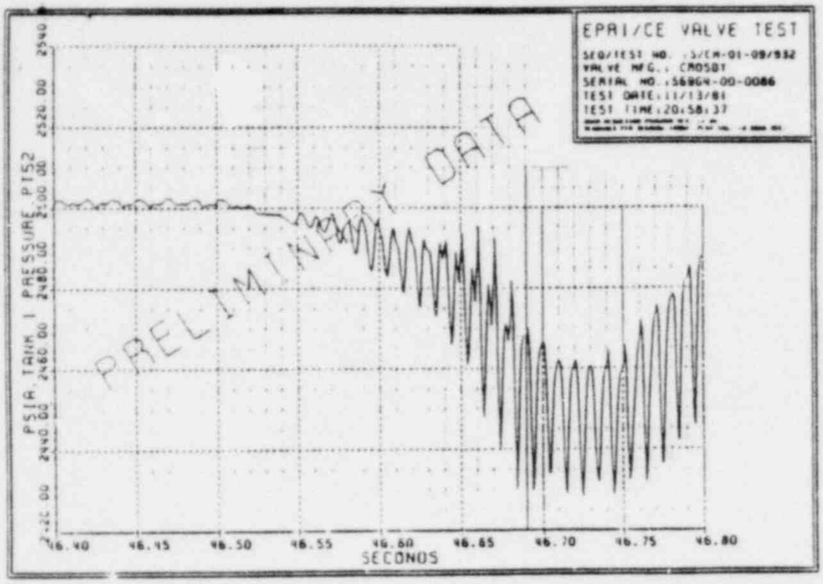
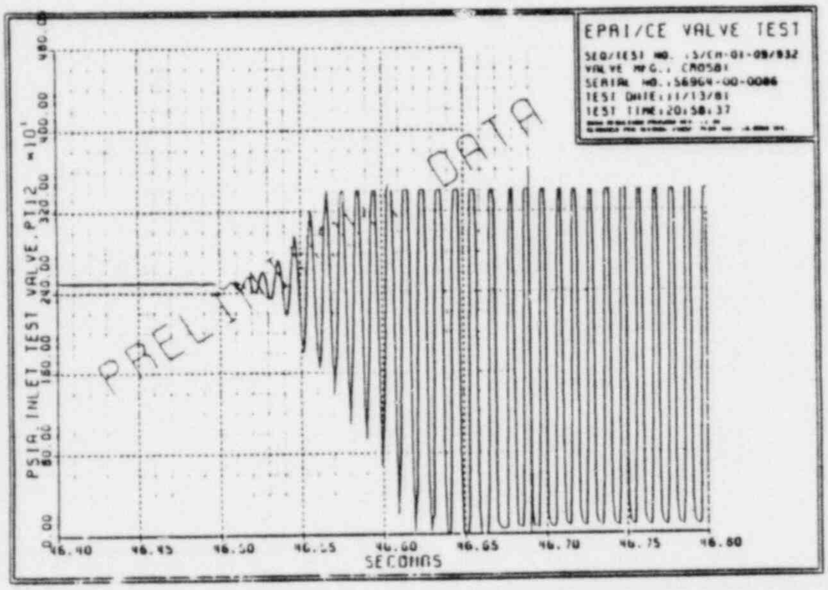


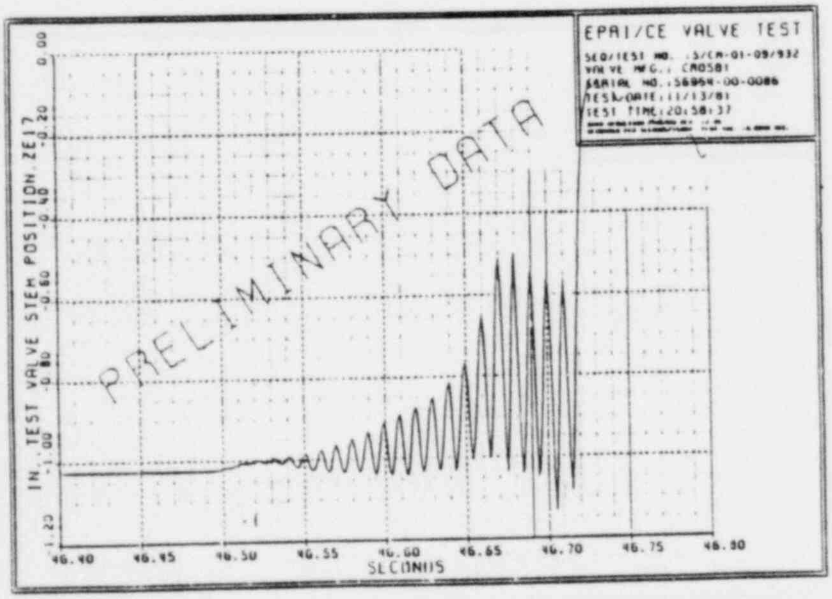
FIGURE 3-15

INLET PIPING PRESSURE OSCILLATIONS DURING WATER SOLID DISCHARGE

a. tank pressure



b. inlet piping pressure



c. valve stem position

4.0 PERFORMANCE ANALYSES

4.1 REACTOR COOLANT SYSTEM OVERPRESSURE

As discussed in Section 2.1, present FSAR transient analyses model the pressurizer safety valve opening by assuming that the valve starts to open at the design set pressure and achieves rated flow at the accumulation pressure. Since valve opening has such an abrupt effect in turning around the overpressure transient, any delay in opening or degradation in flow rate would have a marked effect on the transient. Concurrently, early opening or flow in excess of rated would have an effect on the transient.

The EPRI test data shows that steam flow rates in excess of rated are attainable, thereby, reducing the expected system pressure. However, the data also shows that these flow rates are delayed some period of time following the assumed valve opening point.

To assess the effect on reactor coolant system pressure due to valve opening delay, a series of overpressure transients were run with various time delays inserted for the valve opening. These analyses utilized the limiting overpressure transients for Condition II (loss of load) and Condition IV (locked rotor) events. Other overpressurization events are expected to be conservatively bounded by these two transients.[18]

The reference plants used for these studies were determined on the basis of a parametric study of critical parameters for overpressurization. These parameters included power rating, fuel stored energy, RCS temperature and volume, safety valve capacity, and rod drop (SCRAM) time. The ratio of asymptotic reactor coolant surge rate to safety valve capacity was determined for each plant (see Table 4-1). From this table, it is readily determined that the four loop plant exhibits the highest ratio for surge rate to valve capacity and would be most affected by any perturbations in the valve capacity.

The four loop plant described in Table 4-2 was, therefore, selected as the reference plant for purposes of studying the loss of load transient.

A locked rotor transient is more pronounced in two loop plants as the loss of one reactor coolant pump reduces coolant circulation by 50 percent, while coolant circulation in three and four loop plants is reduced by 33 and 25 percent, respectively. From Table 4-1, the two loop plant with thermal power of 1520 MWt exhibits the maximum surge rate to safety valve capacity ratio for the six two loop plants listed. However, a foreign plant of 1882 MWt (described in Table 4-3) operates at a higher thermal power and exhibits a larger surge rate to safety valve capacity ratio (equal to 1.665). For additional conservatism, this foreign plant was selected for studying the locked rotor transient.

4.1.1 CONDITION II - LIMITING TRANSIENT

The limiting overpressurization transient evaluated for Condition II events is the loss of external electrical load and/or turbine trip without immediate reactor trip (also referred to as loss of load).

In the event of a loss of external electrical load without steam bypass and without reactor trip, a sudden reduction in steam flow results in an increase in pressure and temperature in the steam generator shell. As a result, the heat transfer rate in the steam generator is reduced, causing the reactor coolant temperature to rise. The reactor coolant expands causing an insurge to the pressurizer and reactor coolant system pressure rises.

Both pressurizer safety valves and main steam safety valves open for the loss of load event.

Reference 18 discusses in detail the basis for the limiting transients and the selection of the reference plant (See Table 4-2). This information provided the basis for a generic analysis of the safety impact of the safety valve operating performance for the EPRI test program.

As the analysis is intended to envelope Condition II overpressurization transients, the list of assumptions in Table 4-4 was selected to maximize overpressurization for conservation per standard FSAR analysis methods. Among the assumptions applied to the analysis was the available reactor protection trip actuations. For the loss of load event the reactor is protected by the following potential trip actuations:

a) high pressurizer pressure (2400 psia + 25 psia for conservatism)

b) overtemperature ΔT

c) high pressurizer water level (92 percent span + 5 percent for conservatism)

d) lo-lo steam generator water level (0 percent Narrow Range Span)

Per standard FSAR analysis both pressurizer pressure trip and overtemperature ΔT trip were evaluated. Five cases were evaluated:

Case I	-	Overtemperature ΔT trip with safety valve actuation
Case II	-	High pressurizer pressure trip; 1.2 sec delay in safety valve actuation
Case III	-	Overtemperature ΔT trip; 1.2 sec delay
Case IV	-	Overtemperature ΔT trip; 1.6 sec delay
Case V	-	Overtemperature ΔT trip; 2.0 sec delay

Case I was an overtemperature ΔT trip event with safety valve opening as the reference case. The results demonstrated that high pressure trip preceded high temperature ΔT trip by approximately 2.0 seconds. Figure 4-1 depicts a pressure ramp rate at the safety valve setpressure of 2500 psia of approximately 70 psi/sec. The figure also illustrates that for standard FSAR analysis and assumptions the maximum RCS pressure peaks at 2674 psia for the loss of load transient. (The maximum RCS pressure is defined as the pressure at the bottom of the reactor vessel.)

The results of Case II and III are shown in Figures 4-2 and 4-3. These figures illustrate that the peak RCS pressures for both cases are approximately 2700 psia.

A sensitivity of maximum RCS pressure to valve delay is presented in Figures 4-3, 4-4 and 4-5. These curves show the peak RCS pressure for safety valve opening time delays of 1.2, 1.6 and 2.0 seconds. The figures show that maximum RCS pressure exceed the 110 percent design pressure limit for an opening time delay greater than approximately 2.0 seconds in conjunction with the second reactor protection grade trip.

4.1.2 CONDITION IV - LIMITING TRANSIENT

The locked rotor transient is postulated to result from a sudden locking of one reactor coolant pump rotor. This causes a rapid reduction in core flow rate, reducing the heat transfer rate in the affected steam generator and increasing the temperature of the coolant, resulting in a severe reactor coolant pressure excursion. Departure from nucleate boiling may occur due to flow reduction and the resultant power-coolant mismatch. The safety valves are challenged and are required to flow steam.

Sensitivity analyses were performed to determine the effect of valve opening delay on this transient.

Initially, an analysis was performed assuming the safety valves remained closed. From the analysis, pressure was plotted as a function of time and the time of design valve opening noted (if the valves had opened at 2500 psia). Pressures corresponding to 0.2 second intervals were then selected as the opening setpoints for the sensitivity study.

Figure 4-6 shows pressurizer pressure as a function of valve opening delay time for delays of 0.2 to 1.0 seconds. The pressure transient peaks and turns around in a very short period of time. With no safety valve opening the pressurizer pressure peaks at approximately 2800 psia and decreases as the rods are inserted. A one second delay in opening results in the pressurizer pressure peaking at approximately 2750 psia.

Nuclear power and core flow were determined to be insensitive to opening delays.

4.2 VALVE STABILITY - STEAM CONDITIONS

As observed, a spring loaded safety valve may undergo a rapid opening and closing cyclic behavior during steam discharge. Should the reciprocating motion continue to a point that valve chatter occurs (in which the disc contacts the seat) damage to the valve may result.

An investigation was undertaken to determine those parameters which are critical to the onset of valve chatter under steam discharge conditions. The occurrence of chatter is dictated by valve geometry, spring stiffness, adjustment ring position, and upstream piping length.

4.2.1 APPROACH

An analysis of the phenomenon was conducted using a version of ITCH-1D possessing a dynamic valve model appropriate for steam flow. See Figure 4-7.

The dynamic equatins for the valve motion were inserted in a subroutine of ITCH-1D and solved at each time step.

The equation of motion for the valve stem is

$$m\ddot{y} + \beta\dot{y} + ky = [P_I - P_{set} + \rho_I(V_I - \dot{y})^2]\, A_I + \rho_e V_e^2\, A_e \sin\theta(y).$$

y - displacement of valve stem

m - effective mass of moving parts of valve

β - coefficient of damping, $\beta = 0$ in these analyses

k - spring constant

P_I - inlet pressure

P_{set} - set pressure of valve

$\theta(y)$ - the angle at which fluid exits from the valve. This is a function of y as well as the ring settings.

ρ_I, ρ_e - fluid density at valve inlet and exit, respectively

V_I, V_e - fluid velocity at valve inlet and exit, respectively

The valve stem displacement is coupled to ITCH-1D by means of a ratio defined as y/y_{max}. The steam is assumed to flow isentropically to the point of minimum area and to choke at this point. In this fashion, the upstream valve boundary condition is established. The downstream valve boundary condition is found by iterating on the downstream pressure until the continuity equation is satisfied.

Valve parameters used in the analyses were those approximated for a Crosby 6M6 safety valve. The inlet piping configuration was modeled to approximate that of the 900 series tests.

4.2.2 RESULTS

In order to analyze the valve opening characteristic, the pressurizer was modeled as a constant pressure boundary with a value three percent above the valve set pressure. Following opening, the pressurizer pressure was permitted to decrease linearly until the valve closed. A typical run is shown in Figure 4-8. Pressure oscillations occurring on valve opening and closing as well as the linear pressurizer pressure model are clearly seen. Magnification of the opening pressure oscillations is depicted in Figure 4-9. Figure 4-10 shows the same type of pressure oscillations in the EPRI test data.

Valve step position during opening is shown in Figure 4-11. The valve "pop" and subsequent oscillations may be contrasted with the test date shown in Figure 4-12 for the time period following 18.70 seconds. The predicted oscillations are greater than observed due to the absence of damping (β=0) in the model.

Valve opening time as a function of the upper (adjusting) ring position is shown in Figure 4-13. Raising the ring increases the time required for the valve to open fully.

4.3 INLET PIPING PRESSURE OSCILLATION WATER CONDITIONS

4.3.1 APPROACH

As observed in the loop seal discharge experiments, oscillations occur upstream of a spring loaded safety valve while water is flowing through the valve. This form of oscillation is an acoustical phenomena, analogous to the oscillations in a reed musical instrument.

The analysis of these oscillations was carried out using ITCH-1D, a hydraulic code using the method of characteristics. This method is superior to other commonly used methods for analyzing wave propagation because the wave fronts move with the characteristic lines.

The dynamic equations for the valve motion were inserted in a subroutine of ITCH-1D and solved at each time step. The equation of motion for the valve stem is the same as that given in section 4.2.1.

The fluid is assumed to undergo an isentropic flow down to the point of minimum area. This corresponds to the cylindrical surface between the outer edge of the disc insert and the valve seat.

The velocity of the water at the node corresponding to the end of the nozzle is computed to be

$$V_I = \frac{y}{y_{max}} \sqrt{\frac{P_I - P_e}{\rho_I C_D}}$$

Each time step is iterated until values of V_I, P_I and y converge.

The main concern about the oscillations was whether the peak pressures that result will satisfy the ASME code requirements. The first runs were done using a model which is equivalent to the upstream portion of the EPRI Series 900/1400 test configuration. The downstream portion was not modeled exactly since it has only a small effect on the oscillation.

The upstream noding and the timestep are selected so that the program ran with one characteristic computed implicitly and the other characteristic explicitly. This minimizes numerical damping. The timestep was also selected so that it was much smaller than the period of oscillation of the column of water at resonance.

The remaining runs were carried out with loop seal water lengths covering the range present in Westinghouse plants. The overall length of the pipe was not varied.

4.3.2 RESULTS

Typical inlet piping pressure and valve stem plots are shown in Figures 4-14 and 4-15, respectively. The predicted oscillations start at approximately 170 Hz and increase to approximately 300 Hz as the water bleeds through the valve (and the water column length shortens. This same characteristic was seen in the test. Figure 4-16 shows a typical pressure plot for run 1419 while the corresponding stem position plot is shown in Figure 4-12.

Expanded plots of the predicted inlet piping pressure pulses and valve stem oscillations are shown in Figures 4-17 and 4-18, respectively.

The major parameter analyzed that affects the magnitude of the pressure pulses in the inlet piping is the amount of water present in the loop seal as modeled in water column length. Figure 4-19 shows the results of analyses made with varying the initial water column length in the inlet piping. The analyses show that the peak pressure increases with water column length.

Analyses were also performed to determine the effect of ring position on the peak pressure. Figure 4-20 shows the results of these analyses. It can be seen that varying the adjusting ring position varies the peak pressure in the inlet piping by only a few hundred psi. This has only a minor effect when compared to the initial water column length.

The results of these analyses are in good agreement with the observed data. A peak pressure of 5100 psia was observed for an initial loop seal water column length of approximately 8.3 feet during the test (Section 3.5).

4.3.3 INLET PIPING STRESS ANALYSIS

A. Load Combination and Acceptance Criteria

Load combinations and acceptance criteria based on industrial standards and the EPRI piping sub-committee recommendations for the pressurizer safety and relief valve piping system, including supports, for the piping between the pressurizer and the valves, are shown in Table 4-5. This criteria considers that the relief valve discharge case is an upset transient and combines with OBE loads using the SRSS method. Safety valve discharge is considered as an emergency transient. The intent of this method of combination is to meet the requirements of Standard Review Plan 3.9.3.[20] To be consistent with the load combinations and service limits applicable to piping stress analysis, any combination that includes safety valve discharge is considered a Level C event.

Although certain nuclear steam supply system design transients (for example, loss of load), which are classified as service Level B conditions, may actuate the safety valves, the extremely low number of actual safety valve actuations in operating pressurizer water reactors justifies the service Level C condition from the ASME design philosophy and a stress analysis viewpoint.[22]

B. Piping Adequacy Equations and Limits

To verify the piping adequacy for the peak pressures, as observed in the tests and determined analytically, two areas must be addressed, namely, the permissible pressure limits and combined primary loadings.

For design purposes, the minimum thickness of a pipe wall required for internal design pressure can be determined from[19]:

$$t_m = \frac{PD_o}{2(S_m+Py)} + A \qquad (1)$$

where:

t_m = the minimum required wall thickness, in.

P = internal design pressure, psi

D_o = outside diameter of the pipe, in. (For design calculations, the specified outside diameter of pipe disregarding outside tolerances shall be used to obtain the value of t_m)

S_m = maximum allowable stress intensity for the material, psi

A = an additional thickness to provide for material removed in threading, corrosion or erosion allowance, and material required for structural strength of the pipe during erection, as appropriate, in.

y = 0.4

The allowable working pressure of the pipe can be determined from the following equation:

$$P_a = \frac{2S_m t}{D_o - 2yt} \qquad (2)$$

where:

t = the specified or actual wall thickness as appropriate, minus, as appropriate, material removed in threading, corrosion or erosion allowance, material manufacturing tolerances, bending allowance or material to be removed by counterboring, in.

P_a = the calculated maximum allowable internal pressure for a straight pipe which shall, at least, equal the design pressure, psi.

The permissible pressure may not exceed the pressure P_A calculated in accordance with equation (2), by more than 50 percent when Level C service limits are specified. The maximum permissible pressure for Level C service limits is, thus, given by:

$$P_{permissible} = 1.5\ P_A = \frac{3.0 S_m t}{D_o - 2yt} \tag{3}$$

The primary stress intensity limit requirements are met for design purposes, if equation (4) is met,

$$B_1 \frac{PD_o}{2t} + B_2 \frac{D_o}{2I} M_I \leq 1.5\ S_m \tag{4}$$

where:

B_1, B_2 = primary stress indices for the specific product under investigation

I = moment of inertia, in^4

M_I = resultant moment due to a combination of design mechanical loads, in-lb.

P , D_o, t, S_m = defined previously for equations (1) and (2)

Under any potential service loadings for which Level C service limits are designated, the conditions of equation (4) shall be met using service Level C coincident pressure and moments. These should be selected in such a manner as to result in the maximum calculated stress. An allowable stress intensity to be used for this condition is 2.25 S_m, but not greater than 1.8 S_y, where S_y is defined as the yield strength of the material. Substituting into equation (4) and rearranging results in:

$$M_I \leq \frac{2I}{B_2 D_o} \left[(\text{min of } 1.8\ S_y \text{ and } 2.25\ S_m) - \frac{B_1 P D_o}{2t} \right] \quad (5)$$

Equation (5) defines the maximum allowable resultant moment due to the appropriate combination of pertinent mechanical loads.

D. Results and Comments

Load combinations and acceptance criteria for the pressurizer safety and relief valve piping system are given in Table 4-5. Maximum permissible pressures for pressurizer safety valve inlet piping sizes and schedules representative of Westinghouse pressurized water reactors are given in Table 4-6. For comparison purposes, maximum permissilbe pressures for Level B service limits are also given in the table. Based on tests and analytical work, all accoustic pressures observed or calculated prior to and during safety valve discharge are below the maximum permissible pressure. It should be noted that higher maximum permissible pressure can be determined by use of actual wall thicknesses. Table 4-7 presents the maximum allowable resultant moment due to mechanical loads as a function of piping size, schedule and maximum transient pressure.

TABLE 4-1

PLANT DISTRIBUTION AND SURGE RATE TO
SAFETY VALVE CAPACITY RATIOS

Number of Plants	Number of Safety Valves	Capacity/Valve (lb/hr)	Power (MWt)	Surge Rate (Ft^3/Sec)	Ratio of Surge Rate to Valve Capacity
2 Loops					
3	2	288000	1520	32.62	1.563
2	2	345000	1650	31.54	1.259
1	2	350000	1655	31.61	1.244
3 Loops					
1	2	240000	1351		
3	3	288000	2200	44.69	1.425
2	3	293330	2441	48.71	1.517
4	3	345000	2660	53.18	1.415
14	3	345000	2785/2787	58.96	1.569
1	3	420000	2785	58.96	1.289
4 Loop					
1	3	408000	2758	54.24	1.221
1	3	420000	3025	61.00	1.334
2	3	420000	3250	61.84	1.352
2	3	420000	3350		
1	3	420000	3403	67.84	1.483
37	3	420000	3423/3425	74.03	1.619
2	3	501700	3817	81.22	1.487

Source: Reference 19.

TABLE 4-2

FOUR LOOP REFERENCE PLANT

NSSS Power (MWt)	3425
Thermal Design Flow (gpm)	94400
Reactor Coolant Pressure (psia)	2250
Reactor Coolant Temperature (°F)	
Core Outlet	621.1
Vessel Outlet	617.8
Core Average	591.1
Vessel Average	587.7
Vessel/Core Inlet	557.6
Steam Generator Outlet	557.3
Steam Generator	
Type	Model F
Steam Temperature (°F)	343.3
Steam Presssure (psia)	990
Steam Flow, total (lb/hr)	15.13×10^6
Feed Temperature (°F)	440
Zero Load Temperature (°F)	557
Pressurizer Safety Valves	
Number	3
Capacity/valve (lb/hr)	420000

TABLE 4-3

TWO LOOP REFERENCE PLANT

Number of Loops	2
NSSS, Power (MWt)	1882
Thermal Design Flow, (gpm) (per loop)	94500
Reactor Coolant Pressure (psia)	2250
Reactor Coolant Temperature, (°F)	
Core Outlet	618.9
Vessel Outlet	616.1
Core Average	586.3
Vessel Average	583.0
Vessel/Core Inlet	549.9
Steam Generator Outlet	549.7
Steam Generator	
Type	Model F
Steam Temperature (°F)	534.6
Steam Pressure, (psia)	920
Steam Flow, total (lb/hr)	8.17×10^6
Feed Temperature, (°F)	430
Zero Load Temperature, (°F)	557
Pressurizer Safety Valves	
Number	2
Capacity/valve (lb/hr)	380000
Asymptotic Surge Rate, W_Σ (ft^3/sec)	34.75
Ratio of Surge Rate to Total Safety Valve Capacity	1.665

TABLE 4-4

CONDITION II EVENT ANALYSIS ASSUMPTIONS

- o Standard 412 Plant
- o Model F SG
- o Loss of Load from 102 percent Initial Power
- o Parameters Tavg, Prcp are Initially at Nominal Values
- o No Pressurizer Spray and PORV's
- o No Pressurizer Level Control
- o Pressurizer Safety Valves are Operable
- o No Steam Dump, Safety Valves Operable at 103 percent of Shell Design Pressure
- o No Rod Control
- o Main Feedwater is Lost Simultaneously With Load
  No Auxiliary Feedwater for 60 Seconds
- o BOL Moderator Coefficient, Minimum Doppler
- o 7.5 percent Δk Shutdown
- o Maximum Overall UA's For Fuel To Coolant Heat Transfer
- o Delay Time of 2 Sec From Trip To Rod Motion
- o No Decay Heat For Fast Cooldown
- o Initial Steady State of 10 Seconds Prior To Loss Of Load

TABLE 4-5

LOAD COMBINATIONS AND ACCEPTANCE CRITERIA FOR PRESSURIZER SAFETY AND RELIEF VALVE PIPING AND SUPPORTS - CLASS 1 PORTION

Combination	Plant/System Operating Condition	Load Combination	Service Stress Limit
1	Normal	N	A
2	Upset	N + OBE + SOT_U	B
3	Emergency	N + SOT_E	C
4	Faulted	N + MS/FWPB or DBPB + SSE + SOT_F	D
5	Faulted	N + LOCA + SSE + SOT_F	D

NOTES: 1) Plants without an FSAR may use the proposed criteria contained in the table. Plants with an FSAR may use their original design basis in conjunction with the appropriate system operating transient definitions or they may use the proposed criteria contained in the table.

2) The bounding number of valves (and discharge sequence if setpoints are significantly different) for the applicable system operating transient should be used.

3) Verification of functional capability is not required, but allowable loads and accelerations for the safety-relief valves must be met.

4) Use SRSS for combining dynamic load responses.

TABLE 4-5 (Continued)

DEFINITIONS OF LOAD ABBREVIATIONS

N	=	Sustained Loads During Normal Plant Operation
SOT	=	System Operating Transient
SOT_U	=	Relief Valve Discharge Transient(*)
SOT_E	=	Safety Valve Discharge Transient(*)
SOT_F	=	Max (SOT_U; SOT_E); or Transition Flow
OBE	=	Operating Basis Earthquake
SSE	=	Safe Shutdown Earthquake
MS/FWPB	=	Main Steam or Feedwater Pipe Break
DBPB	=	Design Basis Pipe Break
LOCA	=	Loss of Coolant Accident

* May also include transient flow, if determined that required operating procedures could lead to this condition.

TABLE 4-6

MAXIMUM PERMISSIBLE PRESSURE FOR
PRESSURIZER SAFETY VALVE INLET PIPING*

Pipe Size	Outside Diameter (in)	Nominal Thickness (in)	Permissible Pressure (psi)	
			Level B	Level C
6-inch Sch. 160	6.625	0.719	5229	7131
6-inch Sch. 120	6.625	0.562	4004	5460
4-inch Sch. 160	4.500	0.531	5733	7818
4-inch Sch. 120	4.500	0.438	4644	6333
3-inch Sch. 160	3.500	0.438	6119	8344

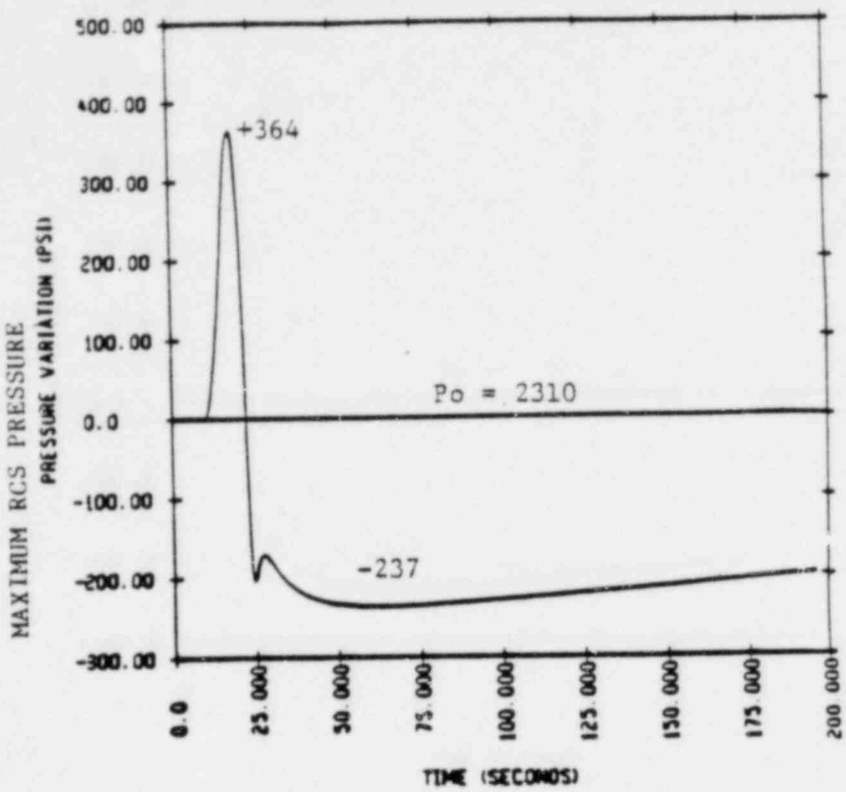
(*) Applicable for temperatures below 300°F.

TABLE 4-7

MAXIMUM ALLOWABLE MOMENT FOR PRESSURIZER SAFETY VALVE INLET PIPING*

Pipe Size	Internal Pressure (psi)	B_2M_I Maximum Allowable Moment (in-kips)
6-inch Sch. 160	5000	516
	6000	475
6-inch Sch. 120	5000	386
	6000	342
4-inch Sch. 160	5000	176
	6000	164
4-inch Sch. 120	5000	143
	6000	130
3-inch Sch. 160	5000	88
	6000	82

*Applicable for temperature below 300°F



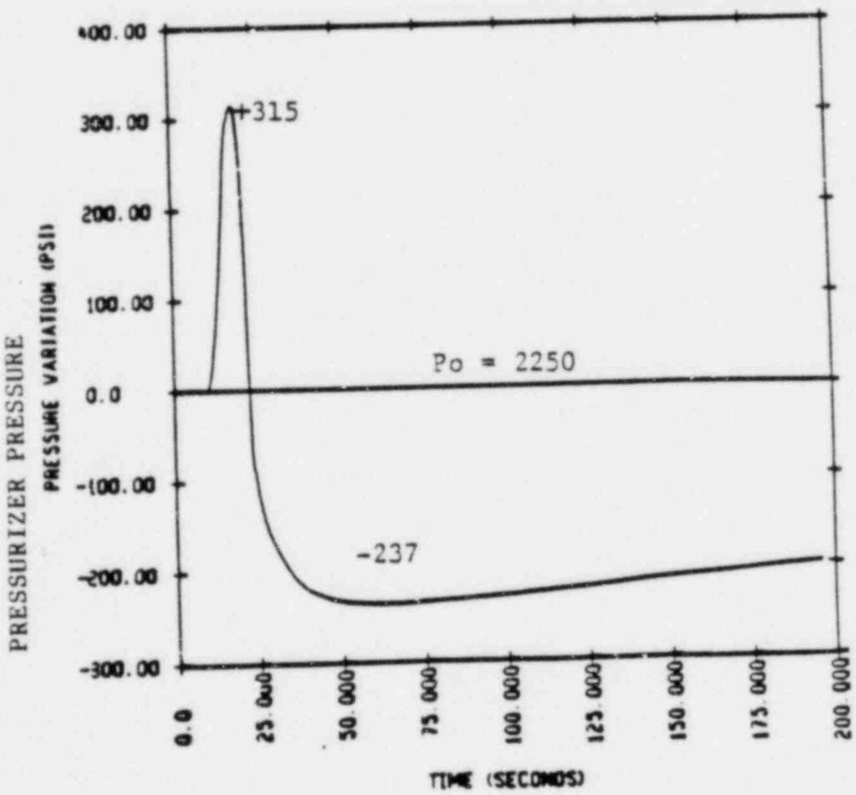
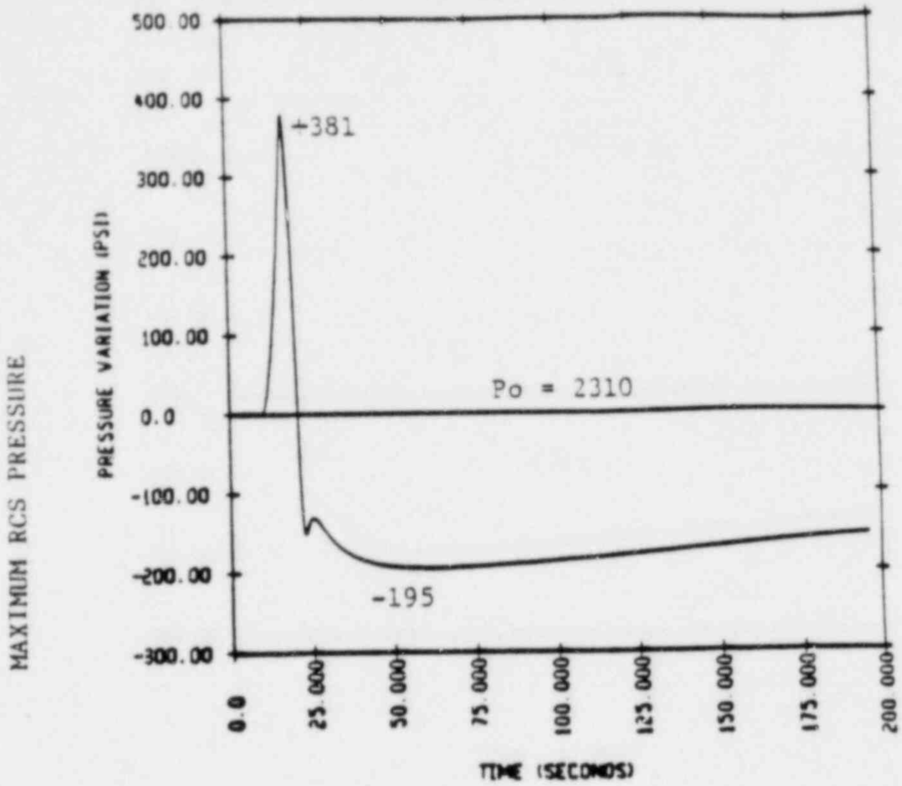


Figure 4-1 Overtemperature ΔT Trip - Reference Case
RCS Pressures vs. Time



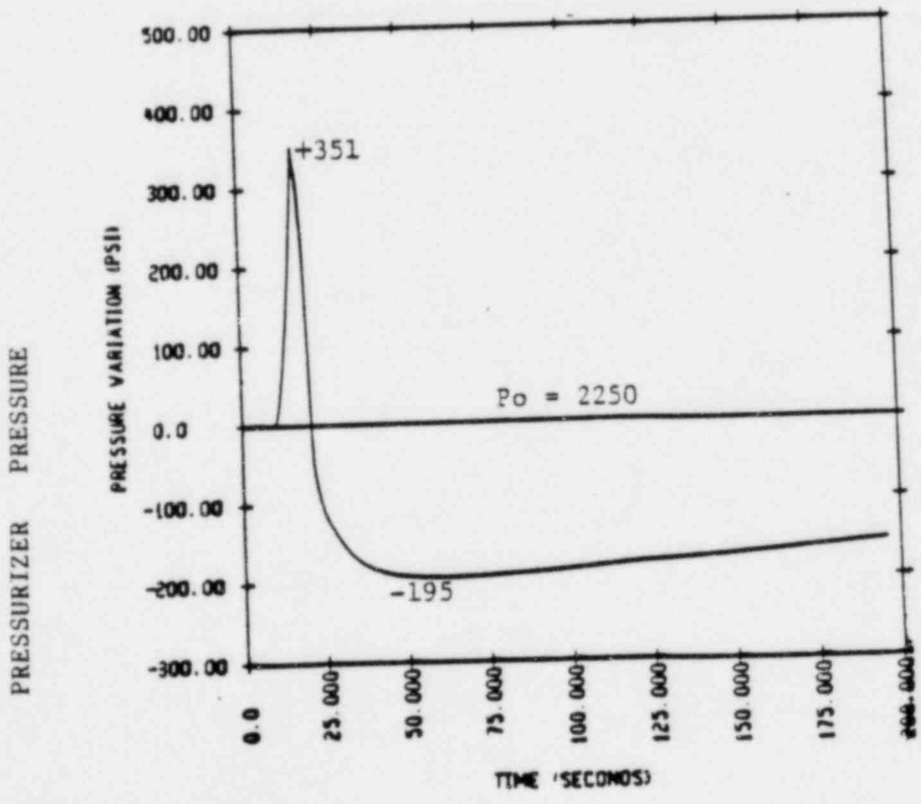
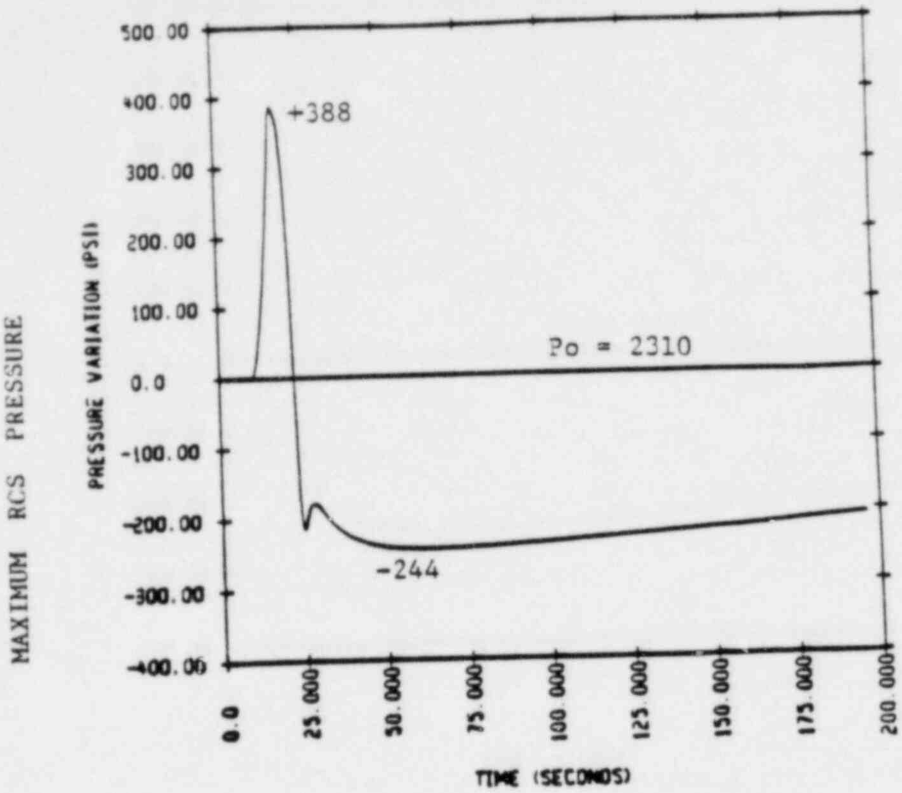


Figure 4-2 High Pressurizer Pressurer Trip - 1.2 Sec Delay
RCS Pressures vs. Time



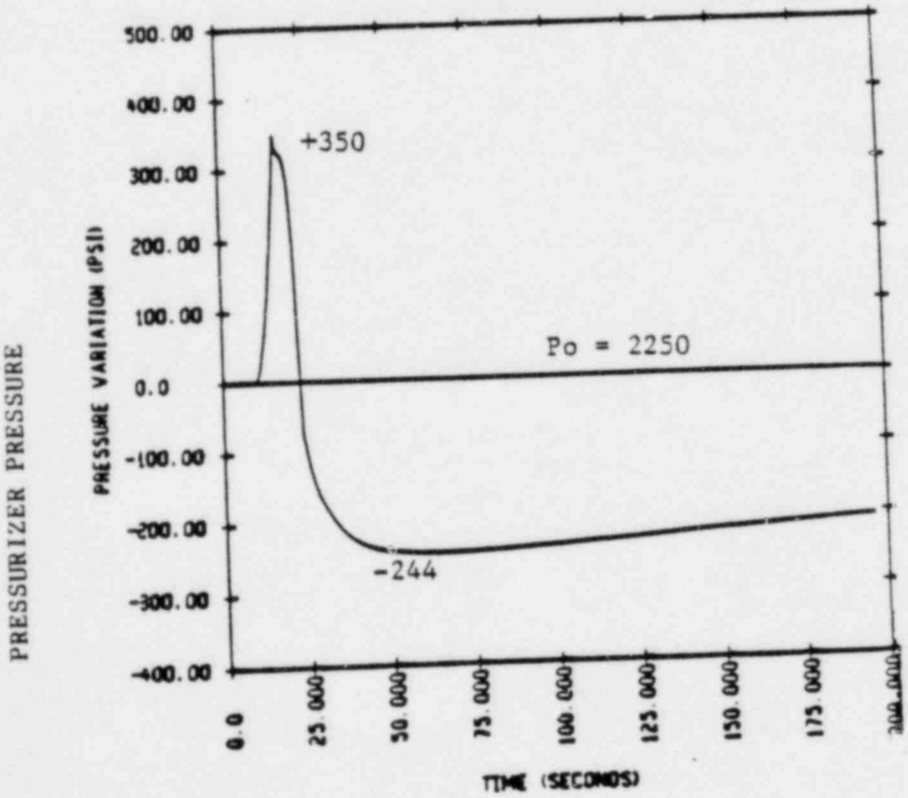
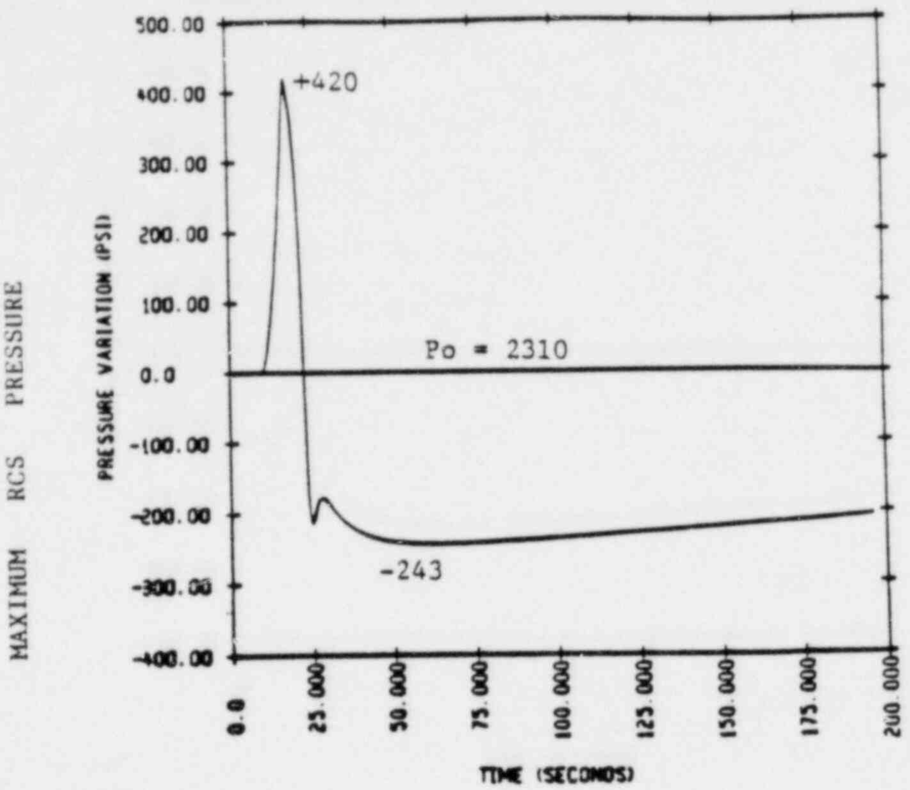


Figure 4-3 Overtemperature ΔT Trip - 1.2 Sec Delay
RCS Pressures vs Time



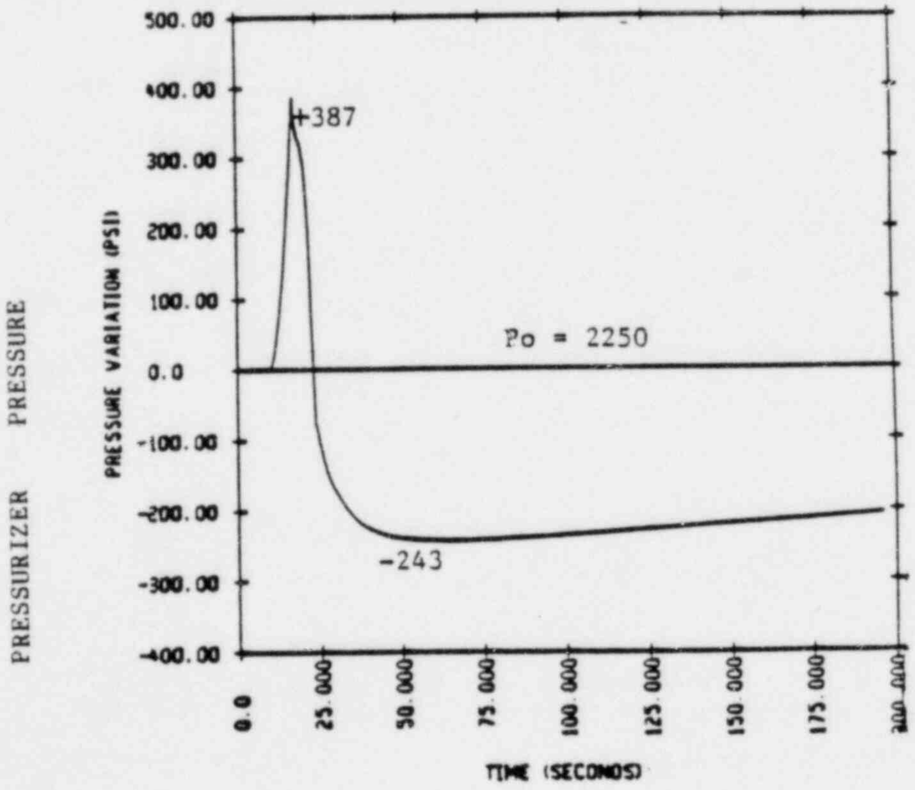


Figure 4-4 Overtemperature ΔT Trip - 1.6 Sec Delay
RCS Pressures vs Time

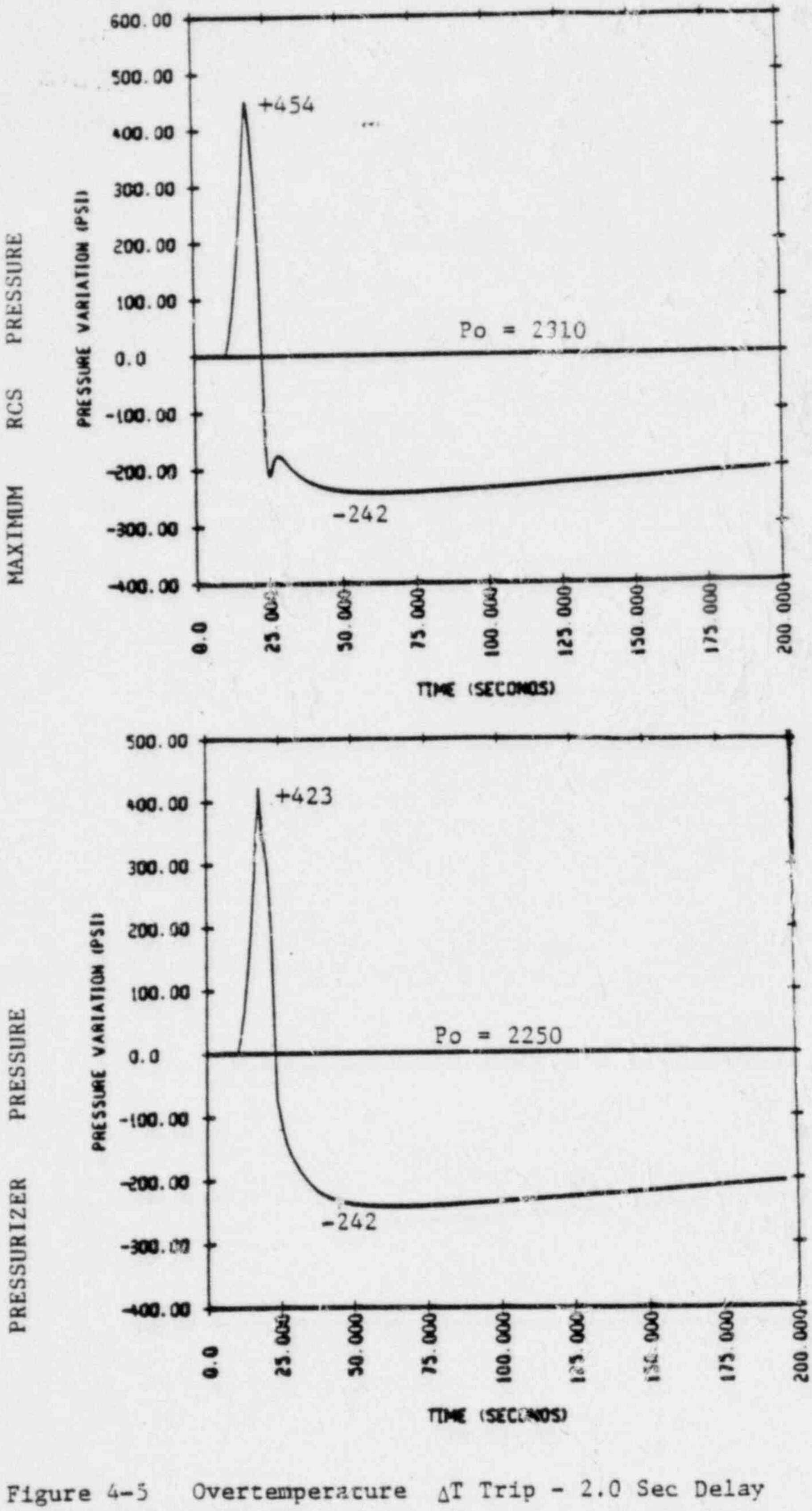


Figure 4-5 Overtemperature ΔT Trip - 2.0 Sec Delay
RCS Pressures vs. Time

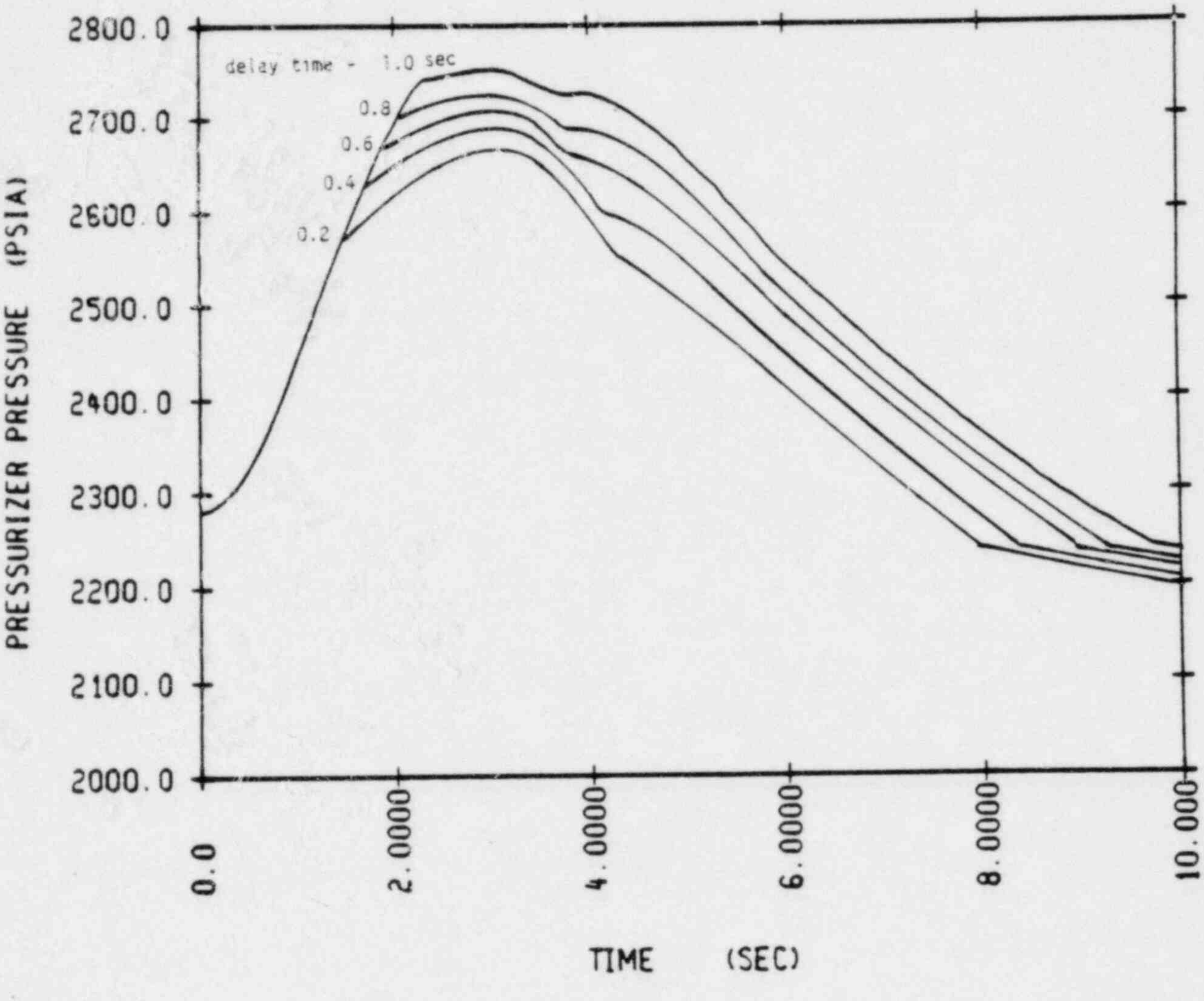


FIGURE 4-6

EFFECT OF VALVE OPENING DELAY ON LOCKED ROTOR EVENT

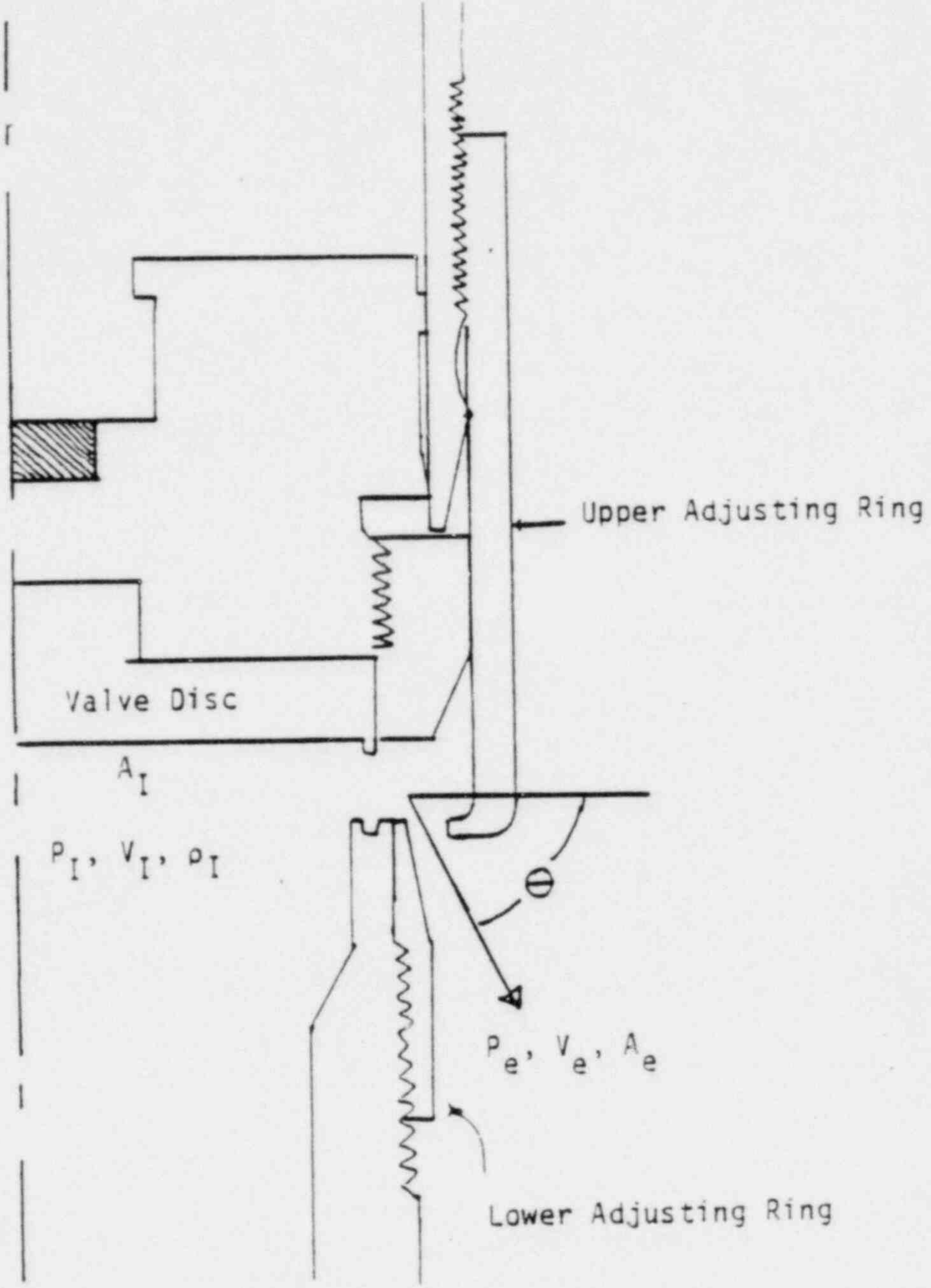


FIGURE 4-7
CROSBY SAFETY VALVE MODEL

FIGURE 4-8

PREDICTED INLET PIPING PRESSURE

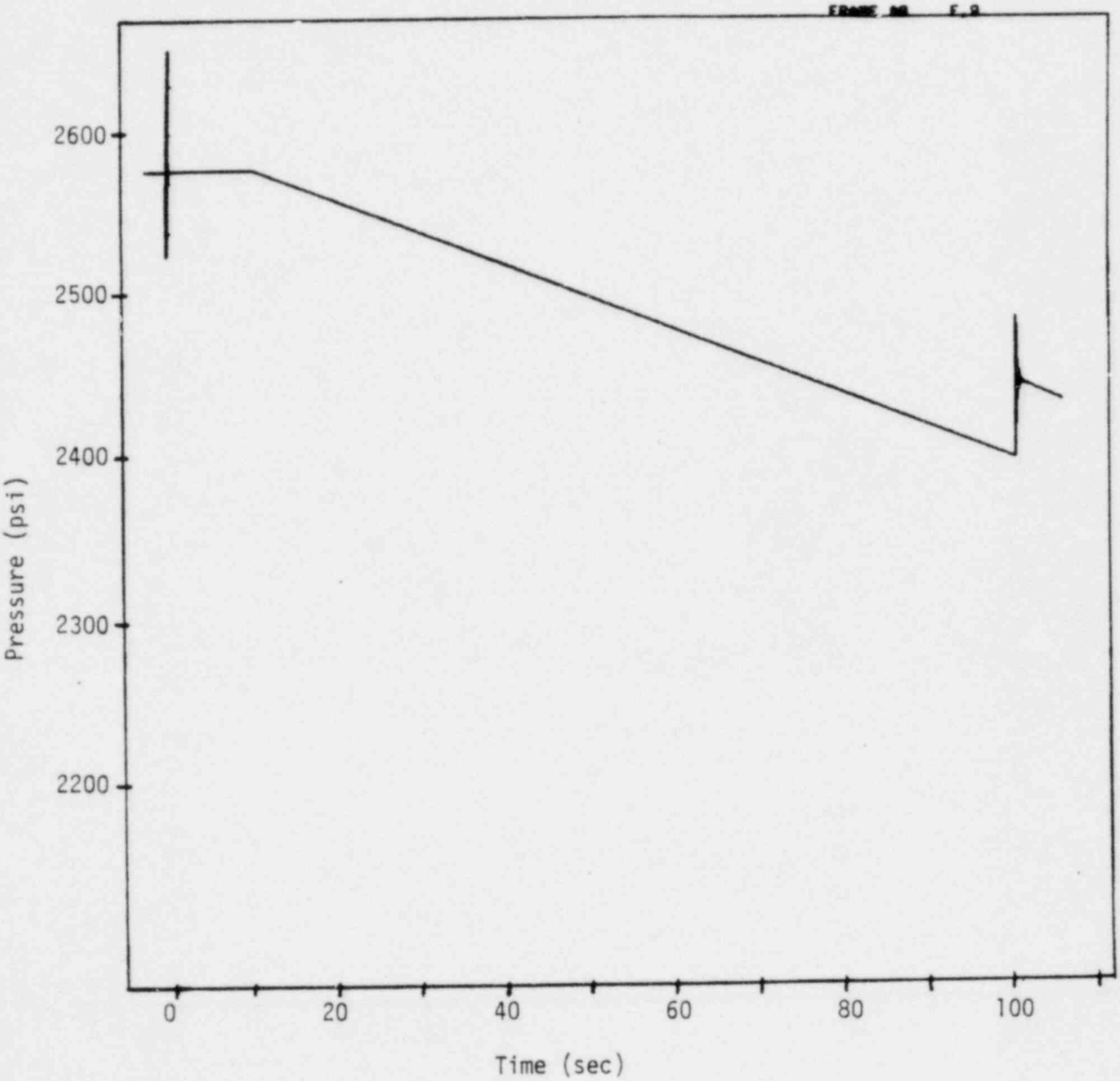
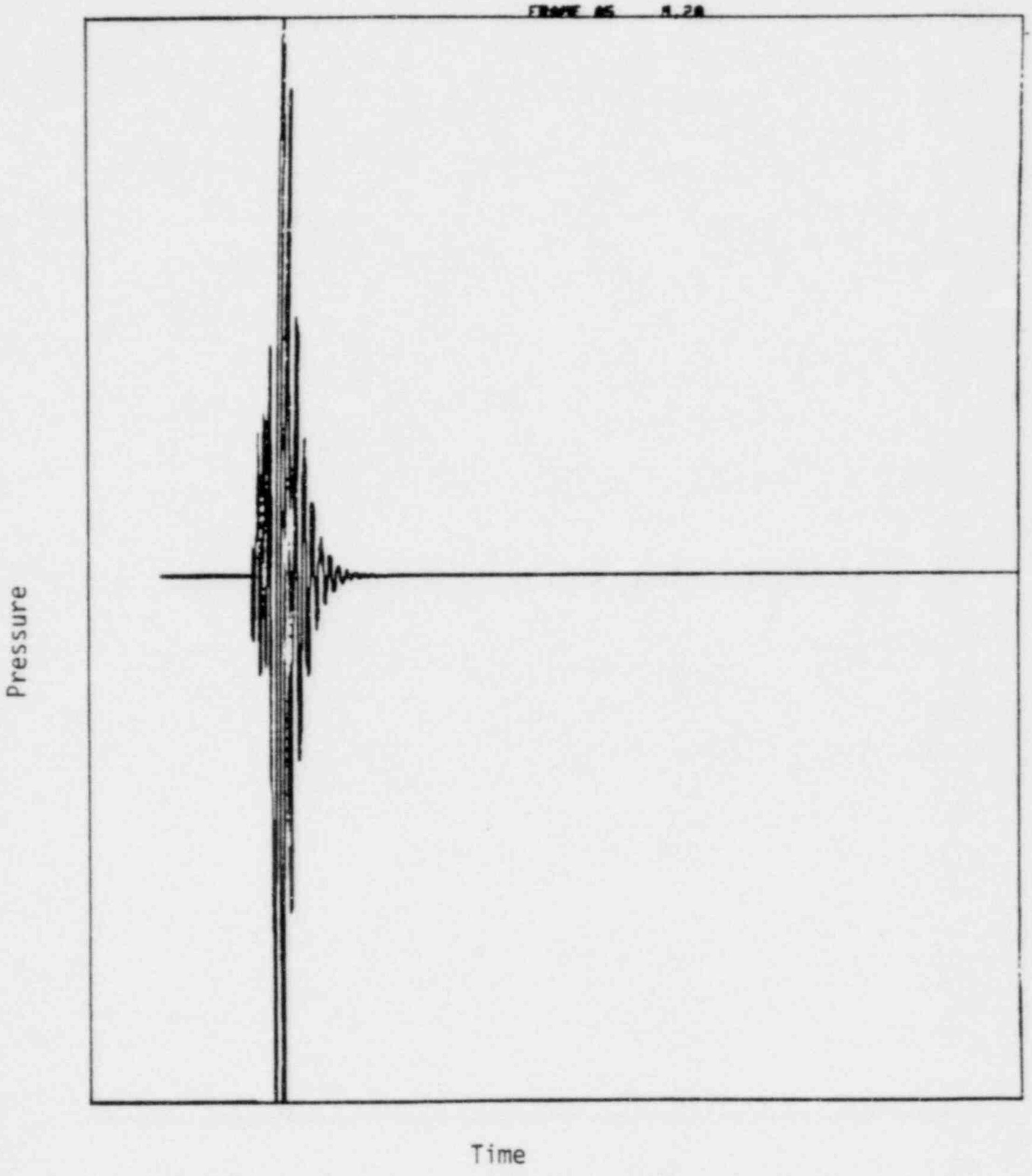


FIGURE 4-9

MAGNIFIED (20X) INLET PIPING PRESSURE OSCILLATIONS FOLLOWING VALVE OPENING



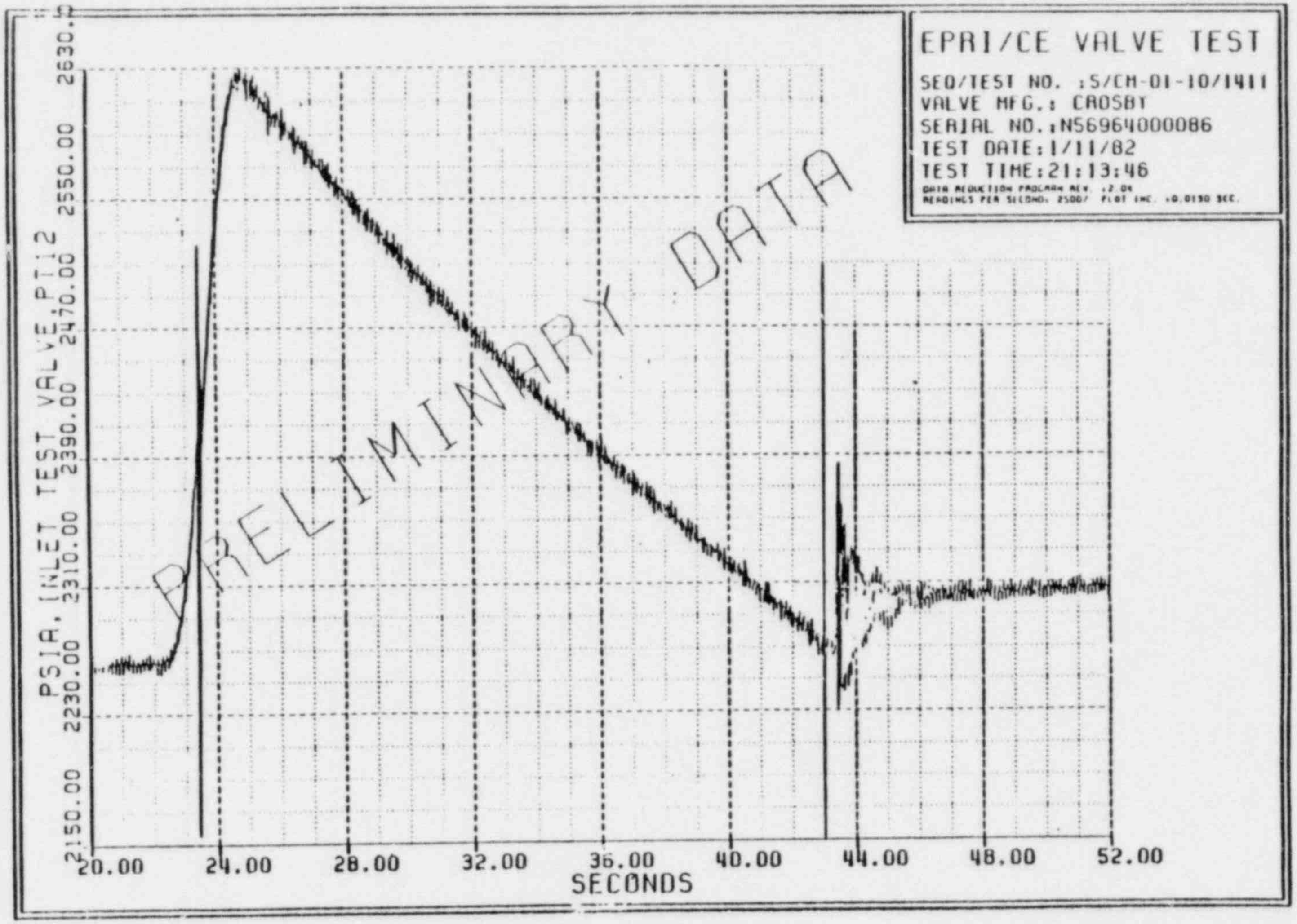
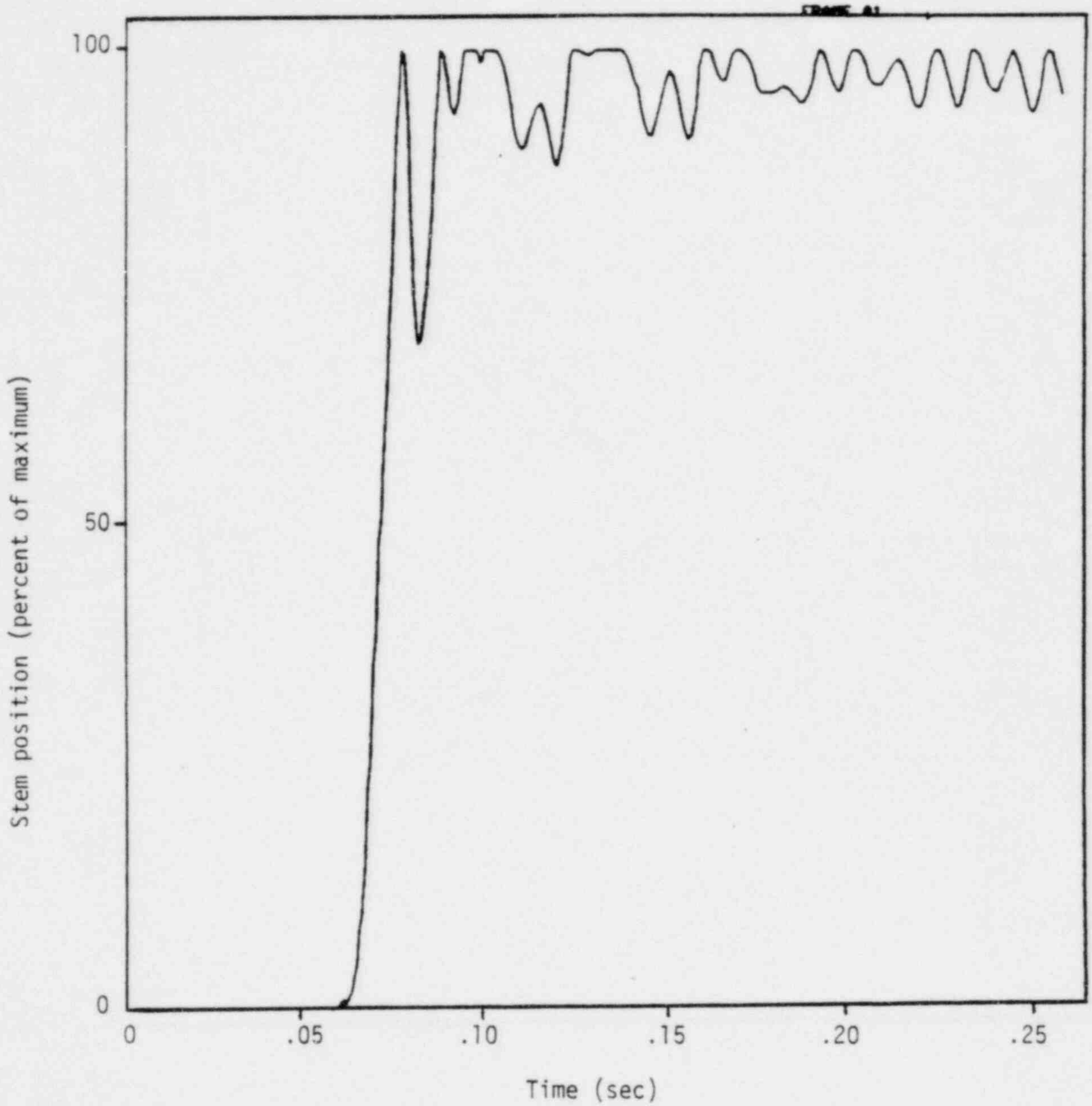


FIGURE 4-10

TYPICAL INLET PIPING PRESSURE CURVE
FOR CROSBY 6M6 VALVE

FIGURE 4-11

PREDICTED VALVE STEM POSITION FOR STEAM
OPENING OF CROSBY 6M6 VALVE



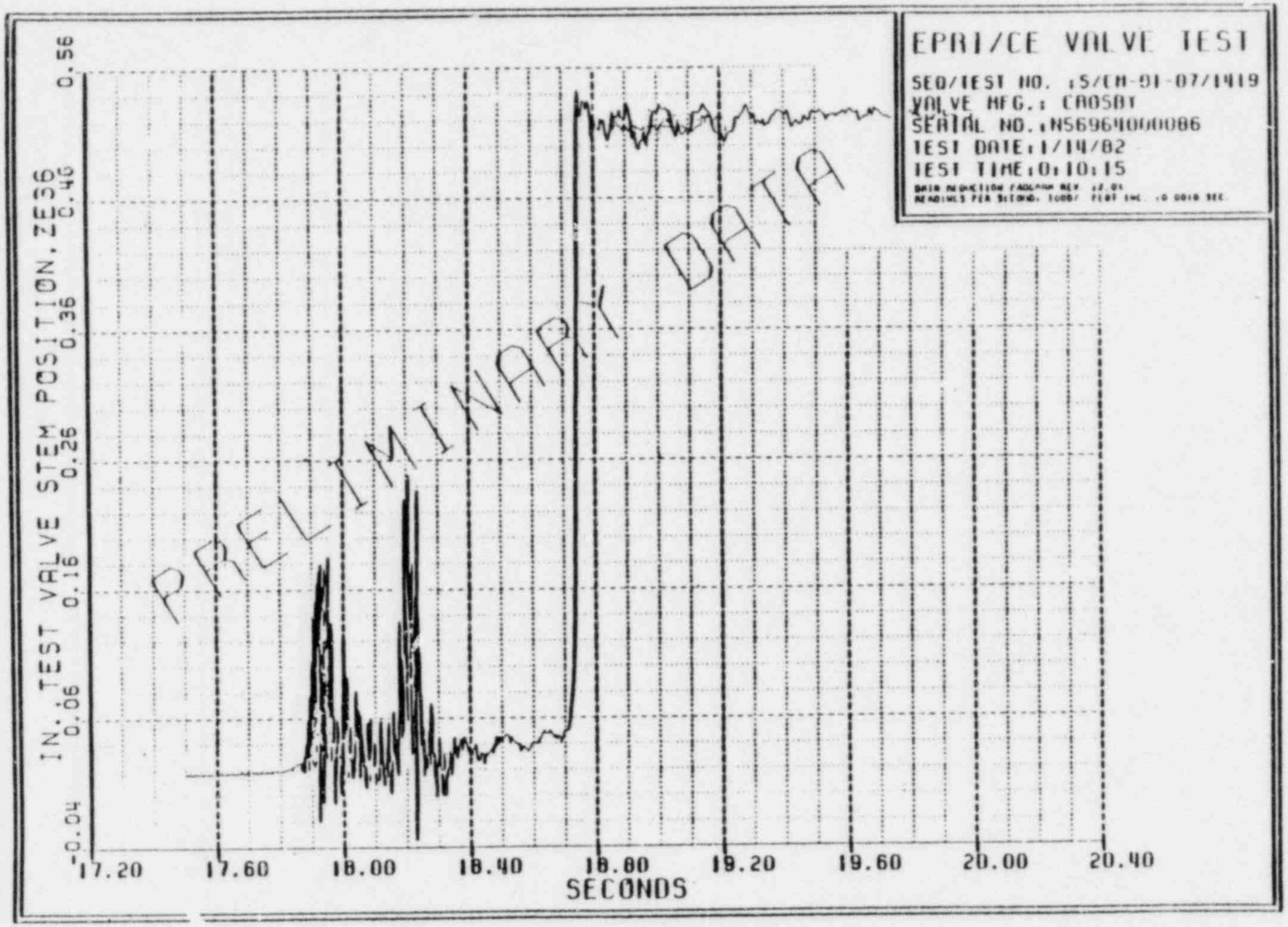


FIGURE 4-12

OBSERVED STEM POSITION DURING TYPICAL
LOOP SEAL OPENING OF CROSBY 6M6 VALVE

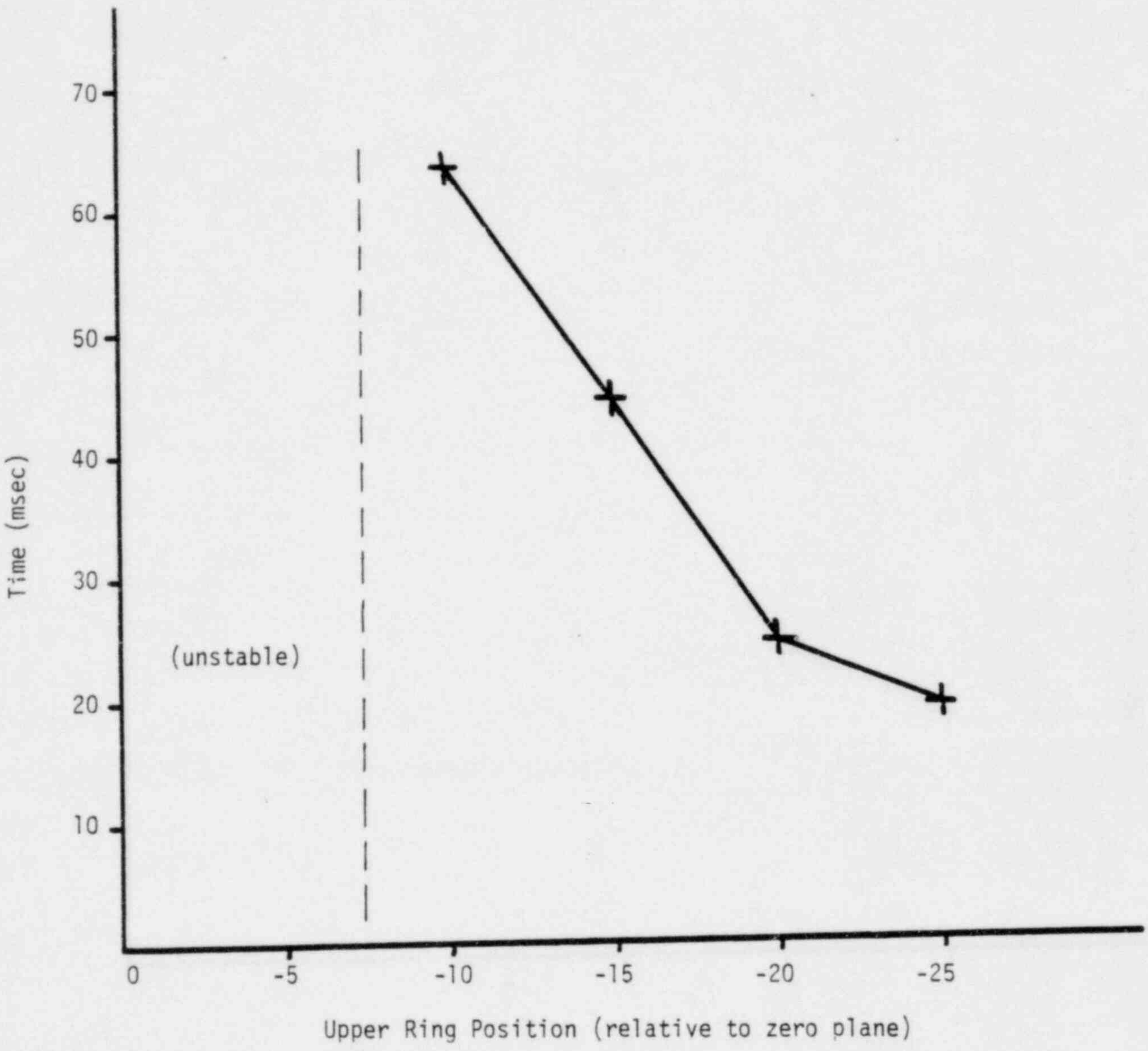


FIGURE 4-13

CROSBY 6M6 VALVE OPENING TIME PREDICTED
AS A FUNCTION OF UPPER RING POSITION

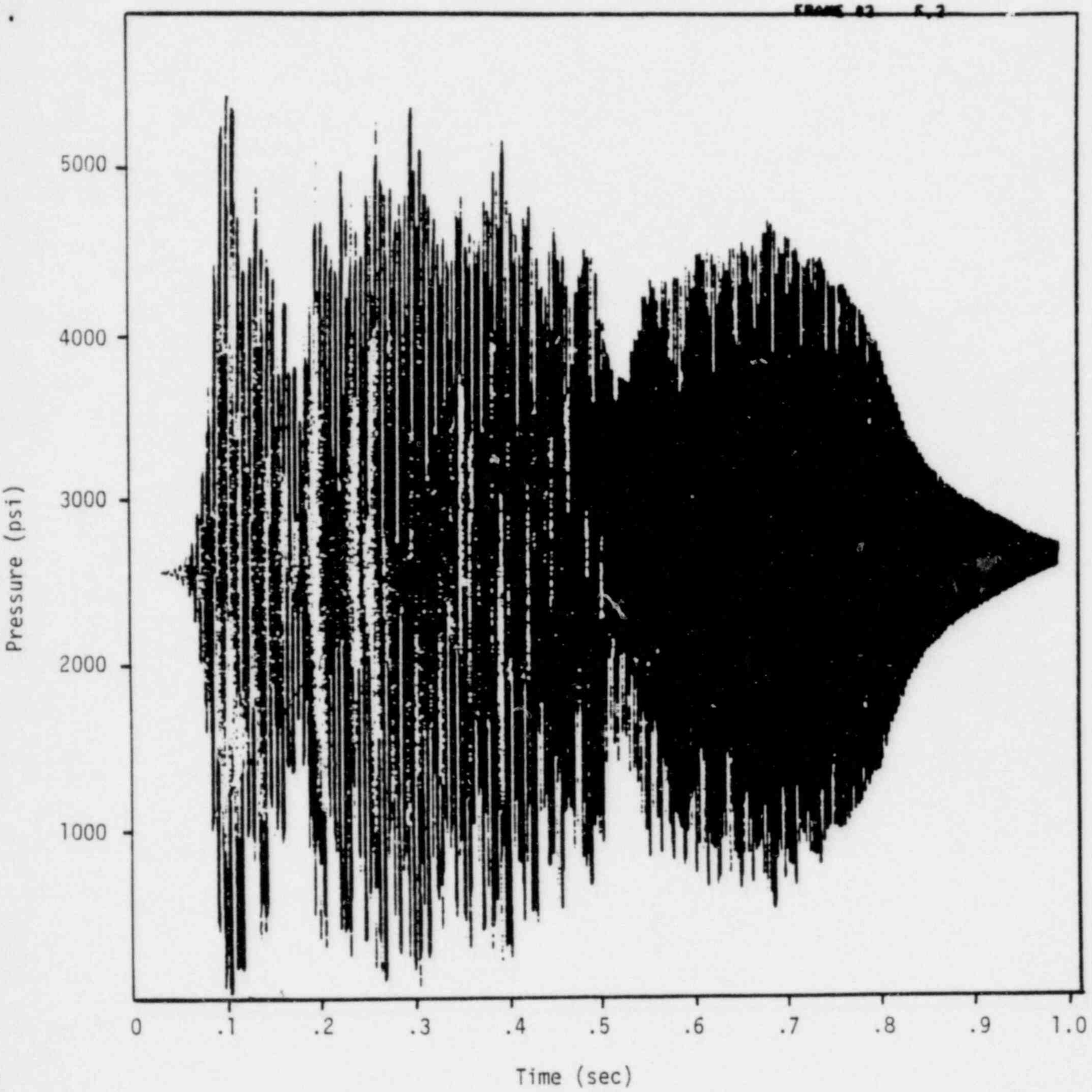


FIGURE 4-14

INLET PIPING PRESSURE OSCILLATIONS PREDICTED
DURING LOOP SEAL WATER DISCHARGE

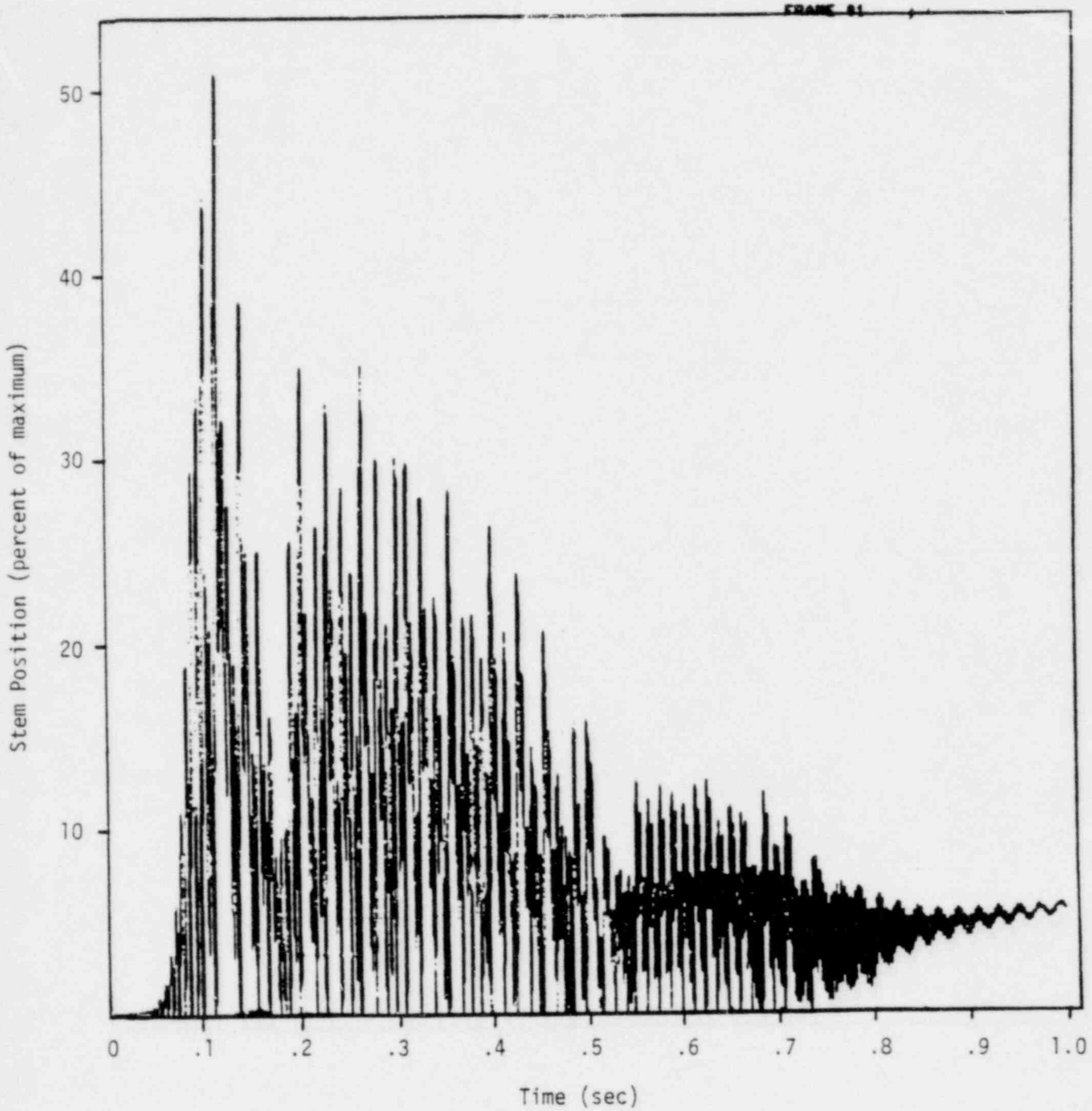


FIGURE 4-15

VALVE STEM POSITION PREDICTED DURING LOOP SEAL WATER DISCHARGE

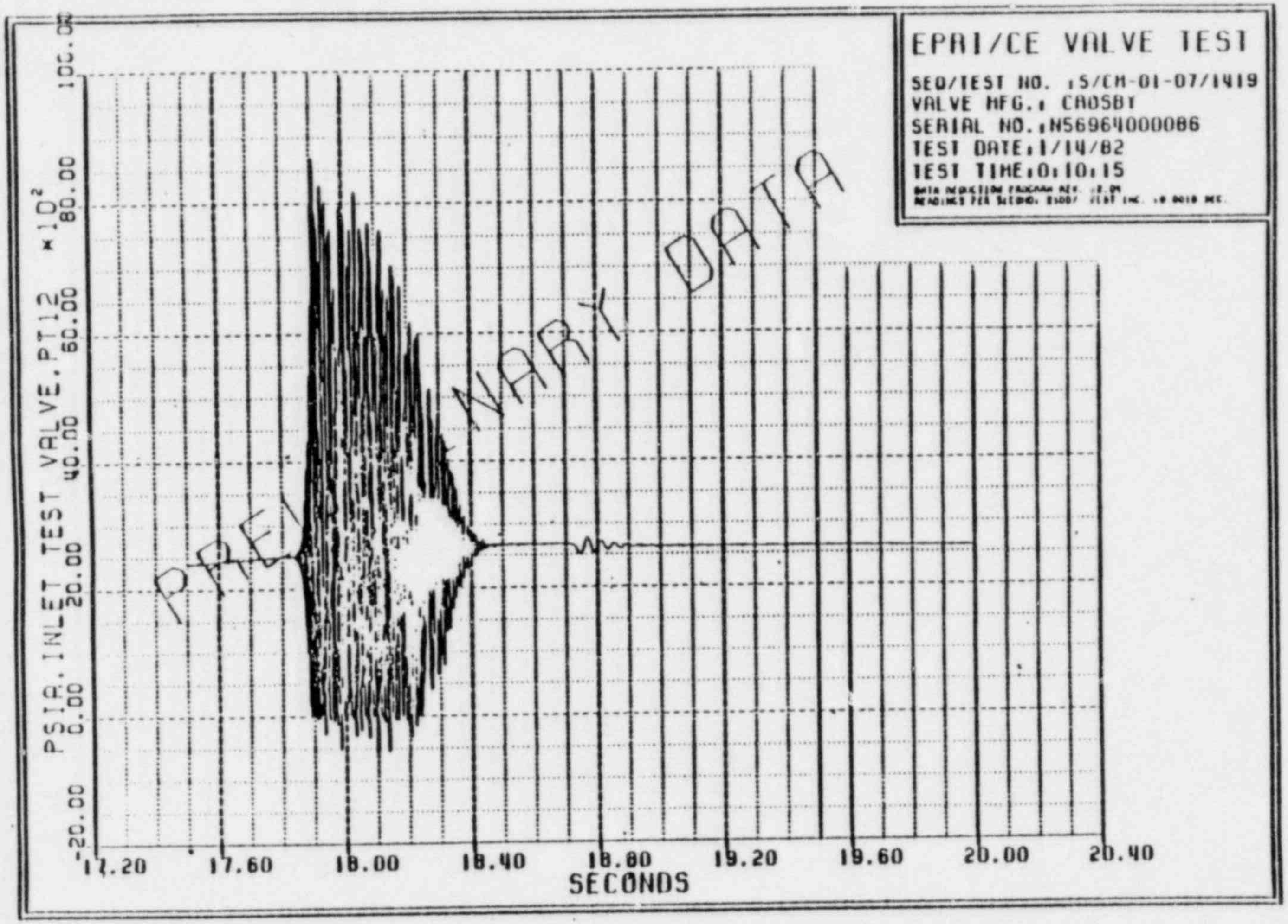


FIGURE 4-16

INLET PIPING PRESSURE OSCILLATIONS DURING
LOOP SEAL WATER DISCHARGE FOR CROSBY 6M6 VALVE

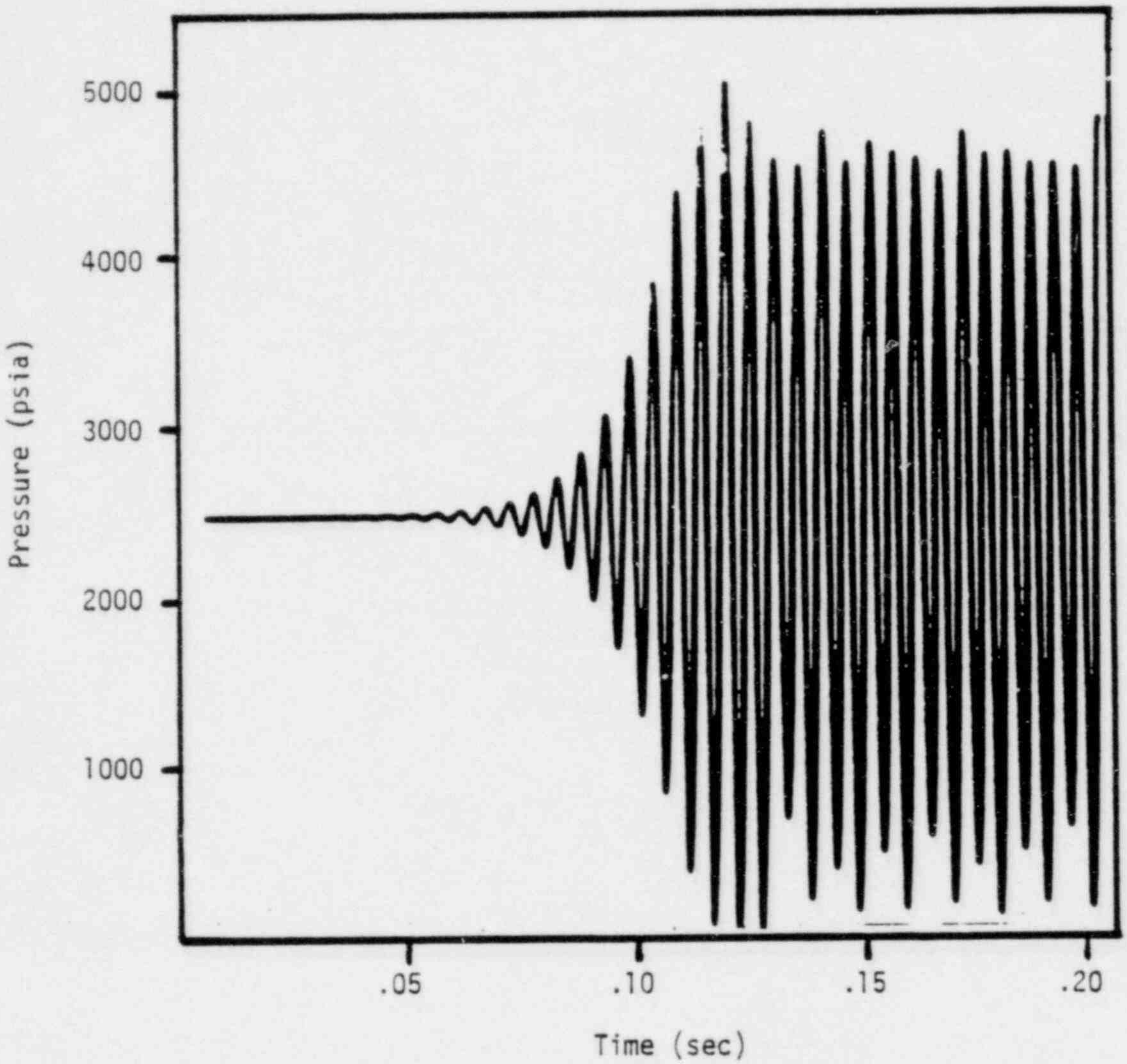


FIGURE 4-17

PREDICTED INLET PIPING PRESSURE AT BEGINNING OF LOOP SEAL BLEEDOFF

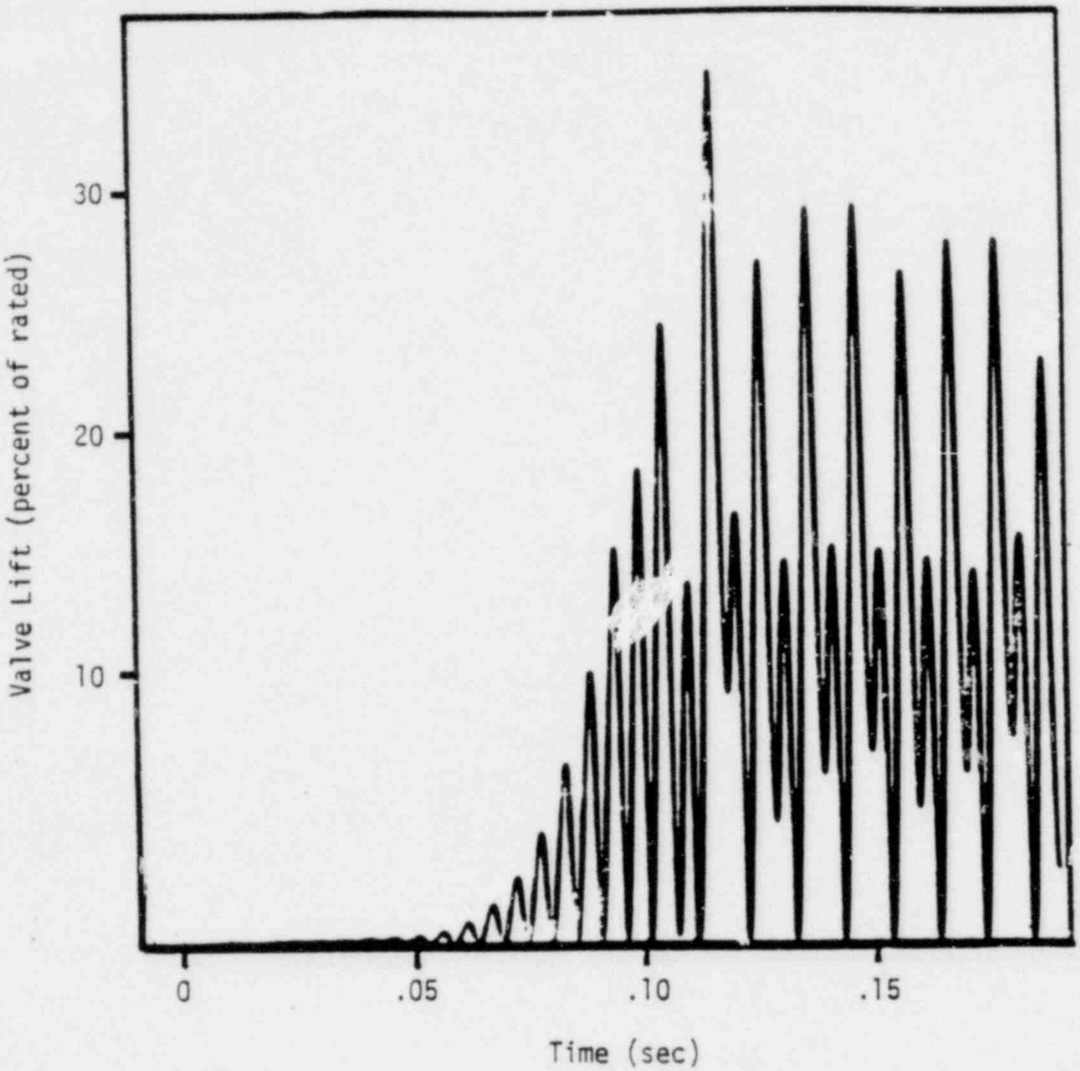


FIGURE 4-18

PREDICTED VALVE LIFT AT BEGINNING OF LOOP SEAL BLEEDOFF

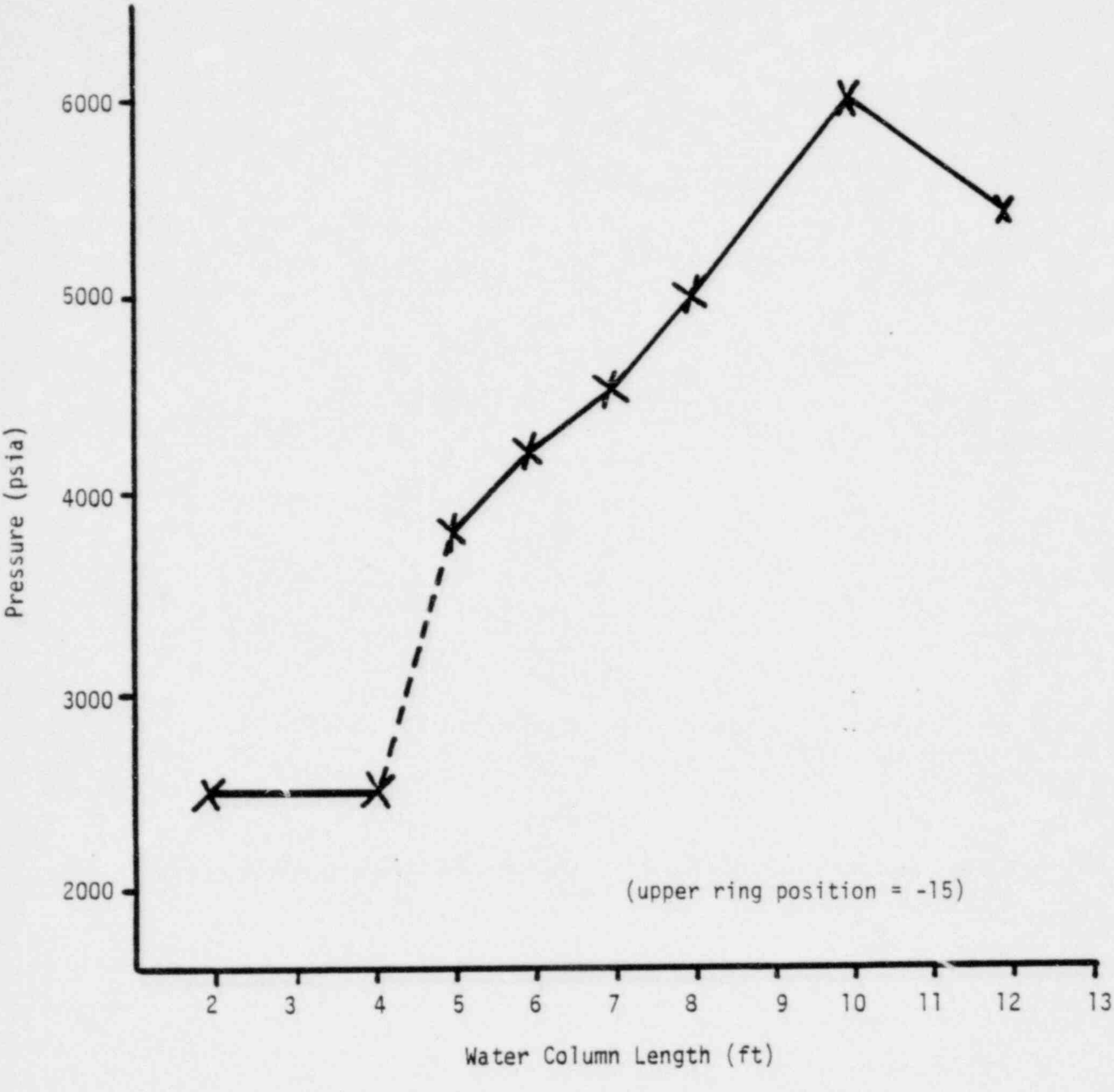


FIGURE 4-19

MAXIMUM PEAK PRESSURE IN INLET PIPING PREDICTED AS A FUNCTION OF LENGTH OF INITIAL WATER COLUMN

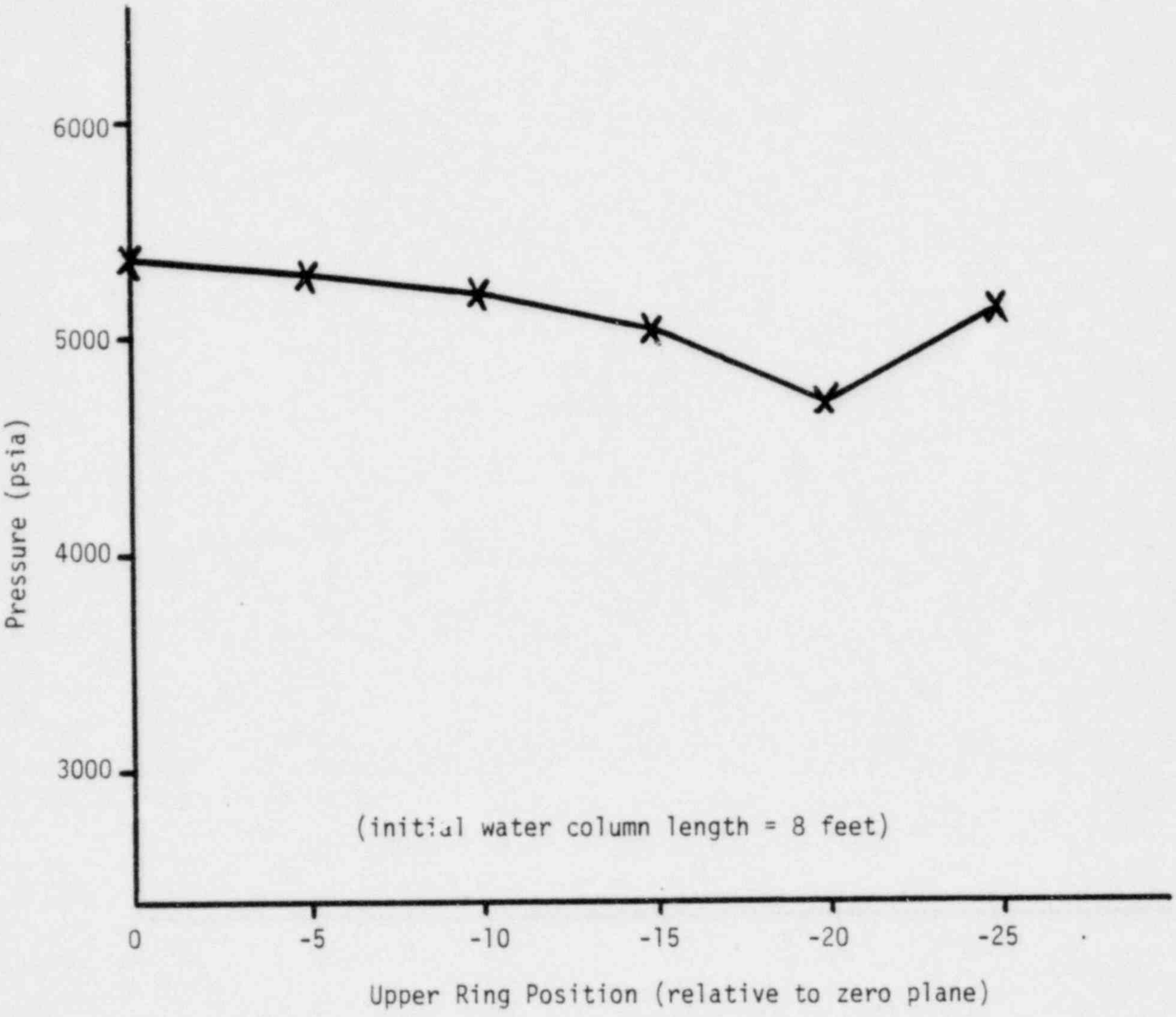


FIGURE 4-20

MAXIMUM PEAK PRESSURE IN INLET PIPING PREDICTED
AS A FUNCTION OF UPPER RING POSITION

5.0 CONCLUSIONS

The functionability of pressurizer safety valves within Westinghouse-designed nuclear power plants is evidenced by the valves' ability to provide adequate pressure relief for the reactor coolant system during anticipated overpressurization transients. Such adequacy is defined by valve opening pressure and steam flow rate.

Valve opening pressures in excess of assumed were evaluated for the limiting Condition II and IV events. For the limiting Condition II events, safety valve functioning is not required if the reactor trips on high pressurizer pressure. If the reactor does not trip until the second protection grade trip (overtemperature ΔT), a valve opening delay time of approximately two seconds would still provide acceptable overpressure protection for the reactor coolant system: all components would be exposed to a pressure within 110 percent of the system design pressure.

Evaluation of the limiting Condition IV event shows that all components of the reactor coolant system would remain wihin 120 percent of the system design pressure in the event of no safety valve opening, assuming reactor trip.

Steam flow rates in excess of rated were measured for all of the test valves. Additionally, stable performance during steam discharge and the repeatability of such performance was demonstrated during the tests.

Valve chatter occurred in four of ninety-six steam tests. For each of these four cases, it was observed that chatter was not directly initiated by the valve closing characteristic, i.e. blowdown, but rather by the valve opening characteristic. Chatter occurred due to a valve's short opening time in relation to the timing of the compression waves in the inlet piping. Ring position adjustments that lengthened the valve opening time were observed to have a positive effect in preventing valve chatter.

Pressure pulses in the inlet piping due to an acoustic water hammer were observed. Both test measurements and analyses show that these pulses may range into several thousands of pounds per square feet. The analyses also show that the magnitude and frequency of these pulses are directly dependent on the length of water upstream of the valve. When compared with ASME Code allowables, it was determined that the pressure pusles, while high, are still within acceptable limits.

The importance of ring position adjustments was observed during the tests. With appropriate adjustments, each of the valves was determined to provide acceptable performance over the range of fluid conditions tested.

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