Water Hammer Transients In Two-Phase Flow

by

Edmund J. Sweeney

Submitted to the Department of
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in Partial Fulfillment of the
Requirements for the degrees of

Master of Science

and

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in Mechanical Engineering

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Abstract

Experiments on two-phase flow water hammer transients due to rapid valve closure were conducted using a laboratory test apparatus. The effect of fluid enthalpy on pressure surge magnitude was investigated. It was determined that although there was a great deal of scatter in the data there was a tendency that the lower the fluid enthalpy the greater the magnitude of the pressure surge. The effect of the length of the pipe conveying the flow on the magnitude of the initial pressure surge could not be determined with any certainty. A simple, conservative, hand calculation for predicting the initial pressure surge magnitude was developed. Experiments were simulated using RELAP5, a transient thermal hydraulic analysis computer program. RELAP5 made conservative predictions of the surge magnitude, which were in good agreement with the values predicted by the developed hand calculation method.

Thesis Supervisor: Peter Griffith

Title: Professor of Mechanical Engineering

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I would like to especially thank James Rooney, P.E., of Stone & Webster Engineering Corporation. Without his guidance the RELAP5 portion of this thesis would not exist. His engineering knowledge and suggestions were of great help in many other sections of this thesis, besides the RELAP5 portion.

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Table of Contents

Abstract	page 2
Acknowledgements	page 3
Table of Contents	page 4
List of Figures	page 5
List of Symbols	page 8
Introduction	page 9
Experimental Apparatus	page 10
Experimental Procedure	page 12
Experimental Results	page 13
Single-Phase Water Hammer	page 24
Two-Phase Water Hammer	page 25
Pressure Surge Duration	page 27
Model Development	page 28
RELAP5 Runs	page 31
Void Fraction Measurements	page 45
Conclusions	page 51
References	page 54
Appendix A	page 55
Appendix B	page 58

List of Figures

Figure 1. Drawing of Experimental Apparatus page 11

Figure 2. (Pressure vs. Time Trace, 210 F, 40 ft pipe) page 14

Figure 3. (Pressure vs. Time Trace, 216 F, 40 ft pipe) page 14

Figure 4. (Pressure vs. Time Trace, 224 F, 40 ft pipe) page 15

Figure 5. (Pressure vs. Time Trace, 235 F, 40 ft pipe) page 15

Figure 6. (40 ft pipe data) page 17

Figure 7. (Pressure vs. Time Trace, 216 F, 24 ft pipe) page 19

Figure 8. (Pressure vs. Time Trace, 225 F, 24 ft pipe) page 20

Figure 9. (Pressure vs. Time Trace, 235 F, 24 ft pipe) page 20

Figure 10. (24 ft pipe data) page 21

Figure 11. Composite of Data for 24 ft & 40 ft Pipes page 22

Figure 12. Illustration of Actual Flashing Pipe Flow page 29

Figure 13. Relation of Void to Pipe Length page 29

Figure 14. Model Assumption of Flashing Flow page 30

Figure 15. Assumed Relation of Void to Pipe Length page 30

Figure 16. Composite of 24 ft &40 ft Pipe Calc. Values page 32

Figure 17. Composite of 24 ft & 40 ft Pipe Calc. and Exp. Data page 33

Figure 18. (RELAP5 Plot, 210 F, 40 ft Pipe) page 35

Figure 19. (RELAP5 Plot, 216 F, 40 ft Pipe) page 35

List of Figures (continued)

Figure 20. (RELAP5 Plot, 224 F, 40 ft Pipe) page 36

Figure 21. (RELAP5 Plot, 235 F, 40 ft Pipe) page 36

Figure 22. (RELAP5 Plot, 210 F, 24 ft Pipe) page 37

Figure 23. (RELAP5 Plot, 216 F, 24 ft Pipe) page 37

Figure 24. (RELAP5 Plot, 224 F, 24 ft Pipe) page 38

Figure 25. (RELAP5 Plot, 210 F, 24 ft Pipe) page 40

Figure 26. (RELAP5 Plot, 216 F, 8 ft Pipe) page 40

Figure 27. (RELAP5 Plot, 224 F, 8 ft Pipe) page 41

Figure 28. (RELAP5 Plot, 235 F, 8 ft Pipe) page 41

Figure 29. Composite of RELAP5 Runs for the Three Pipe Lengths page 42

Figure 30. Composite of RELAP5 Runs & Data (24 & 40 ft Pipes) page 43

Figure 31. Comp. of RELAP5 Runs & Calc. Values (24 & 40 ft) page 44

Figure 32. Drawing of Exp. Apparatus With Void Meas. Section page 46

Figure 33. Composite of Calc. Exit Void & Measurements page 47

Figure 34. (RELAP5 Exit Void vs. Time Plot, 210 F, 24 ft Pipe) page 48

Figure 35. (RELAP5 Exit Void vs. Time Plot, 216 F, 24 ft Pipe) page 48

Figure 36. (RELAP5 Exit Void vs. Time Plot, 224 F, 24 ft Pipe) page 49

Figure 37. (RELAP5 Exit Void vs. Time Plot, 210 F, 40 ft Pipe) page 49

Figure 38. (RELAP5 Exit Void vs. Time Plot, 216 F, 40 ft Pipe) page 50

List of Figures (continued)

Figure 39. (RELAP5 Exit Void vs. Time Plot, 235 F, 40 ft Pipe) page 50

Figure A1. Computer Model of Experimental Apparatus page 56

Figure B1. Experimental Data For 8 ft Pipe page 58

Table of Symbols

a	propagation velocity of pressure pulse
c	isentropic wave velocity
D	pipe diameter
e	thickness of pipe wall
K	bulk modulus of elasticity
L	pipe length
P_{surge}	pressure surge
t_{surge}	duration of the pressure surge
ΔV	fluid velocity change
ρ	density
ρ_a	density of air
μ	Poisson's ratio
α	void fraction

Introduction

In piping systems of power plants, rapid valve closure can cause very large pressure rises and subsequent pressure pulsations throughout the piping systems. This phenomenon is known as water hammer and is a cause of concern because of the significant damage that it can cause to the pipes or their supports. In the experiments reported here, we have simulated the flow in a pipe conveying an almost saturated liquid to a pressure relief valve and have observed the water hammer that occurs when the resulting flashing flow is abruptly stopped by rapid valve closure.

One objective of this study was to obtain experimental data of the water hammer phenomenon and investigate the effects of fluid enthalpy on maximum pressure surge after valve closure, as well as investigate the effects of varying the length of the pipe conveying the fluid.

A second objective of this study was to develop a simple, hand calculation method to predict the magnitude of the pressure surge.

A third objective was to investigate how effectively RELAP5, which is among other things a transient thermal hydraulic analysis computer code for pipe networks, could be employed to predict the pressure surges. Both the hand calculated and computer generated results are compared to the actual experimental results.

Void fraction measurements were also made and the experimental values are compared to the values obtained through several calculation methods and the Relap5 calculated values of exit void fraction.

Experimental Apparatus

Figure 1 is a drawing of the experimental apparatus with a pipe length of 24 feet. The experimental apparatus consists of a 30 gallon cylindrical tank that has several lines entering it to admit air, steam and water. Flows through these lines are controlled by valves at the entrances to the tank. There is one cold water line that is used to fill the tank with water. There is a steam line that enters the tank along its bottom which is used to heat the water in the tank. There is a second steam line that enters the tank through its top and is used to pressurize the tank. There is also an air line into the top of the tank which could be used to pressurize the tank. There is a relief line at the top of the tank to help regulate the tank's presssure. A pressure gauge and a thermocouple are also hooked up to the tank to determine the tank's internal pressure and temperature.

Leading out of the tank is a one inch I.D. insulated copper pipe. The length of this exit pipe is varied in the experiments. Runs were made for pipe lengths of 40 ft, 24 ft and 8 ft. For the pipe lengths of 24 ft and 40 ft the pipe exits of the tank and runs straight for seven feet then bends around 180 degrees with a radius of 1.5 feet and heads in the opposite direction. The pipe has two ball valves throughout its runs (when the void test section is not attached). The first valve (valve 1) is right near the tank exit. The second valve is either at 40 feet, 24 feet or 8 feet downstream, depending on the pipe length in the experiment. For all three of the pipe lengths, eight inches upstream of valve 2 is a pressure tap connected to a pressure transducer with an 800 psi diaphragm. The pressure transducer

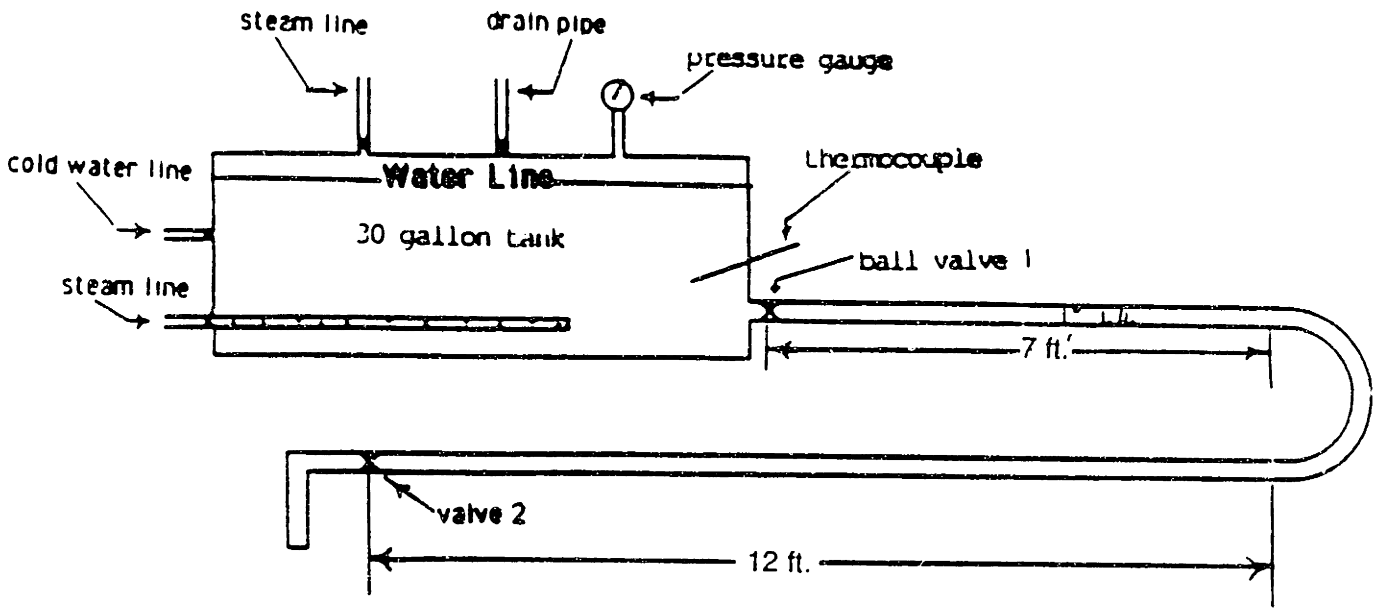

Figure 1. Drawing of Experimental Apparatus

is connected to a carrier signal demodulator which then relays the demodulated signal to an IBM P.C. The P.C. employs the Unkelscope software. Through use of the Unkelscope software we are able to obtain voltage vs. time traces that are converted to a pressure vs. time traces through use of a calibration curve relating pressure to voltage. The calibration curve was developed from calibration tests made with a dead weight tester.

Experimental Procedure

For the experiment, the tank is filled with cold water. The cold water is then heated by bubbling steam in from the sparger at the bottom of the tank. Once the tank's internal temperature reaches the desired level, the bottom steam line is shut off and the tank is pressurized to 10 psig by introducing steam through the top of the tank.

Once the tank's internal pressure reaches 10 psig the valves 1 and 2 along the exit line are opened. After sufficient time has elapsed to allow the flow through the pipe to fully develop, valve 2 is manually slammed shut. The tank's internal pressure is maintained at 10 psig throughout the run. A pressure vs. time trace is ultimately obtained for the location 8 inches upstream of valve 2.

To assure that there was no air entrained at the pipe's entrance during a run, only two runs were made before the tank was refilled with water and reheated. Two more runs would then be made. One run usually filled a four gallon bucket.

Experimental Results

40 Foot Pipe Data

Figures 2, 3, 4 and 5 are pressure vs. time traces for experimental runs with the 40 foot pipe. Tank temperatures for these runs were 210 F, 216 F, 224 F, and 235 F. Figure 6 is a graph of maximum pressure surge magnitude vs. tank temperature for all the 40 ft pipe runs.

Figure 4, the pressure vs. time trace for a run where the tank temperature was 224 F, is considered a normal two-phase flashing flow run. This pressure vs. time trace shows that sometime after valve closure there was an initial pressure surge with a magnitude of 453 psig. We are not sure exactly at what time on the plot the valve is slammed. After the initial pressure surge the pressure dropped below 0 psig. The pressure then began to gradually rise. Approximately .75 seconds after the initial pressure surge there is a second pressure spike with a magnitude of approximately 144 psig. After the second spike the pressure again dropped below 0 psig and gradually began to rise until approximately .36 seconds after the second pressure surge. At that point there was a third pressure surge with a magnitude of approximately 21 psig. After the third pressure surge there is evidence of much smaller negligible pressure rises. For all runs the initial pressure surge was always the maximum pressure surge. This was to be expected. The pressure vs. time trace for the two-phase run with a tank temperature of 224 F (figure 4) was similar to the single-

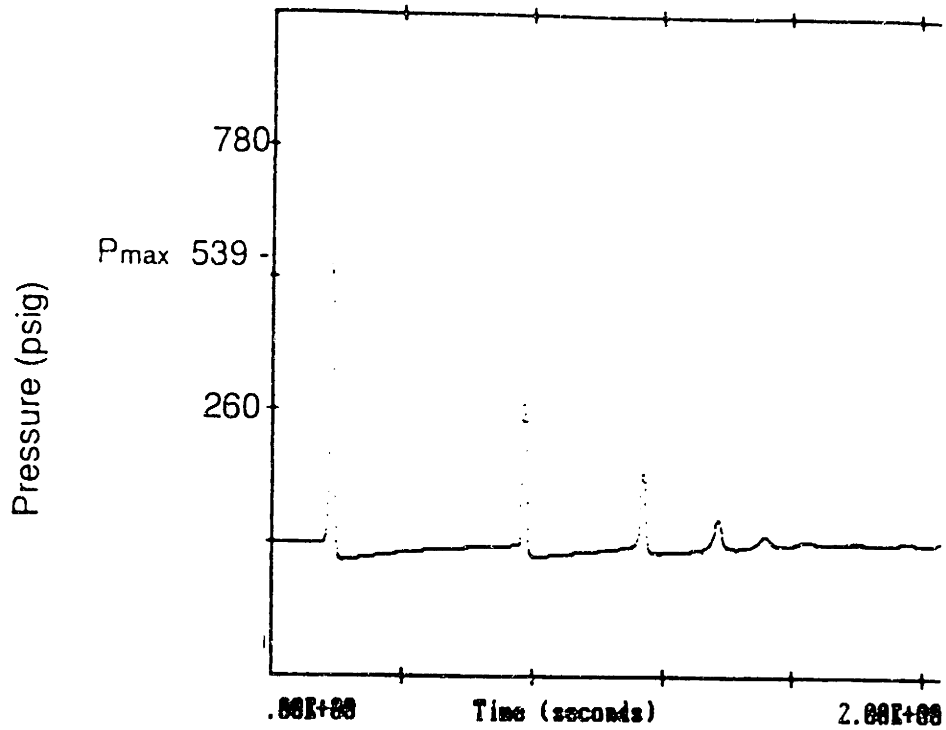

Figure 2 (Pressure v. Time Trace, 210 F, 40 ft Pipe)

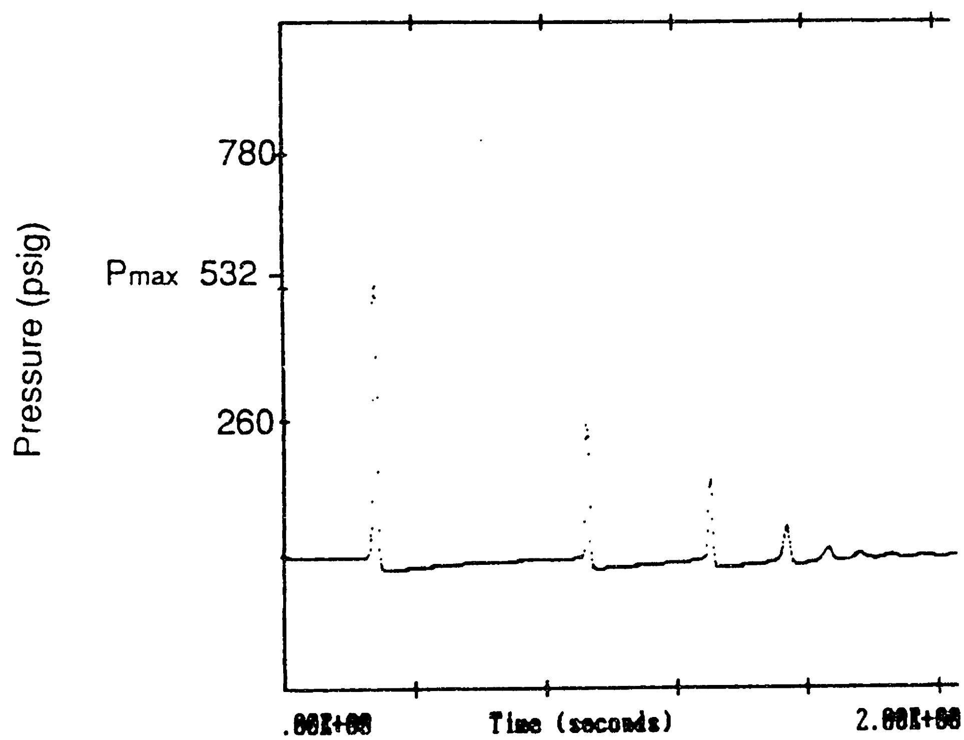

Figure 3 (Pressure v. Time Trace, 216 F, 40 ft. Pipe)

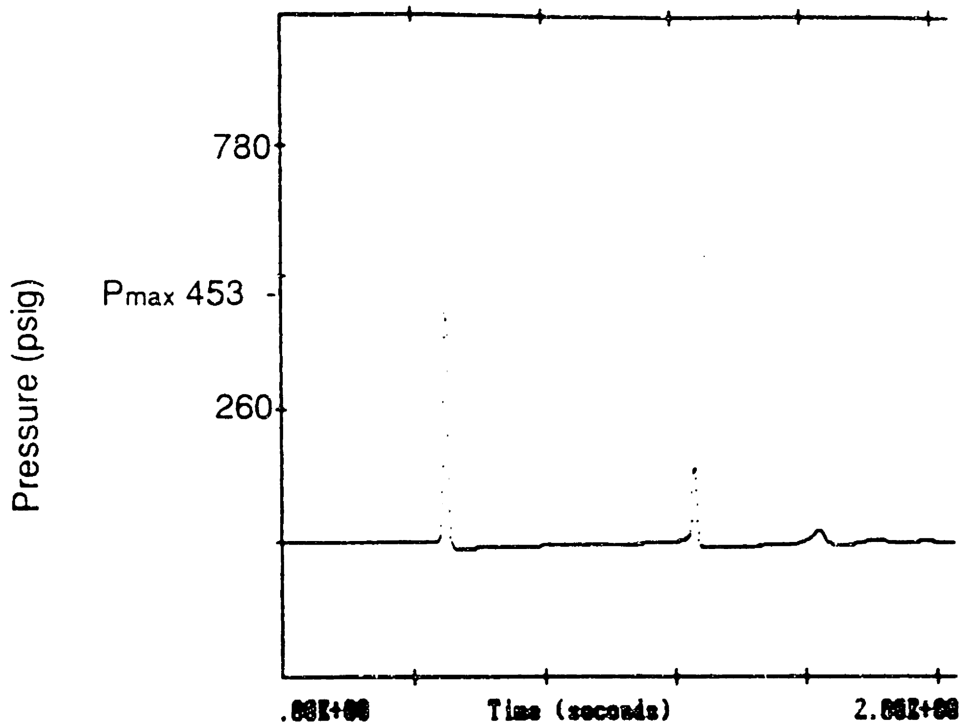

Figure 4 (Pressure v. Time Trace, 224, 40 ft. Pipe)

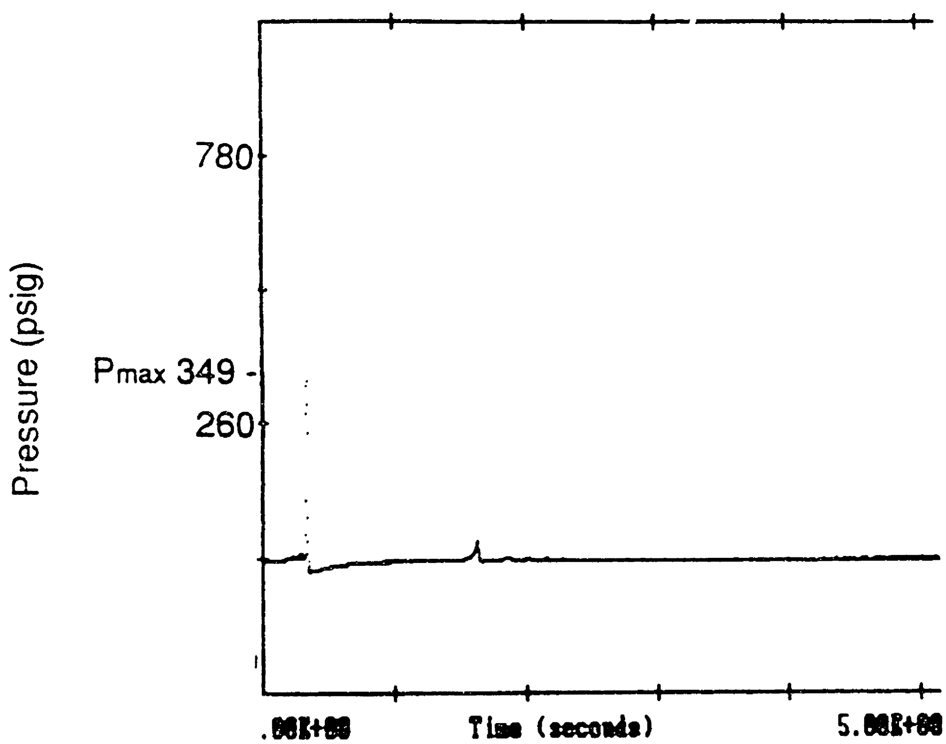

Figure 5 (Pressure v. Time Trace, 235, 40 ft. Pipe)

phase run with a temperature of 210 F (figure 2). For this hot water single-phase run there was an initial high magnitude pressure surge of 513 psig. There were some discrepancies between the transients for the 210 F and the 224 F runs, such as the magnitudes of the inital pressure surges as well as the number and magnitudes of subsequent pressure surges. Figure 2 shows that there are five substantial pressure surges within the sampling time of the 210 F run. However, there is evidence of smaller pressure surges whose magnitudes could be considered negligible. Figure 4 shows that there are only four pressure surges that are distinguishable for the 224 F run.

As can be seen in Figure 6 the highest pressure surge observe was 550 psig at a tank temperature of 65 degrees F. The maximum pressure surge observed for a tank temperature above 200 degrees F was for a tank temperature of 216 degrees F and an observed pressure surge of 532 psig. Figure 6 shows that there is a general tendency for the magnitude of the pressure surge to decrease as the fluid temperature increases above 216 degrees F. The greatest variations in pressure surge magnitude appears to occur at fluid temperatures above 235 degrees. In the temperature range from 234 to 236 F the initial pressure surge magnitudes observed range fom 349 psig to 15 psig. At temperatures of 240 degrees and above the pressure surges were practically negligible.

24 Foot Pipe Data

Figures 7, 8, 9 are pressure vs. time traces for experimental runs using a 24 foot exit pipe at tank temperatures of 216 F, 224 F, and 235 F.

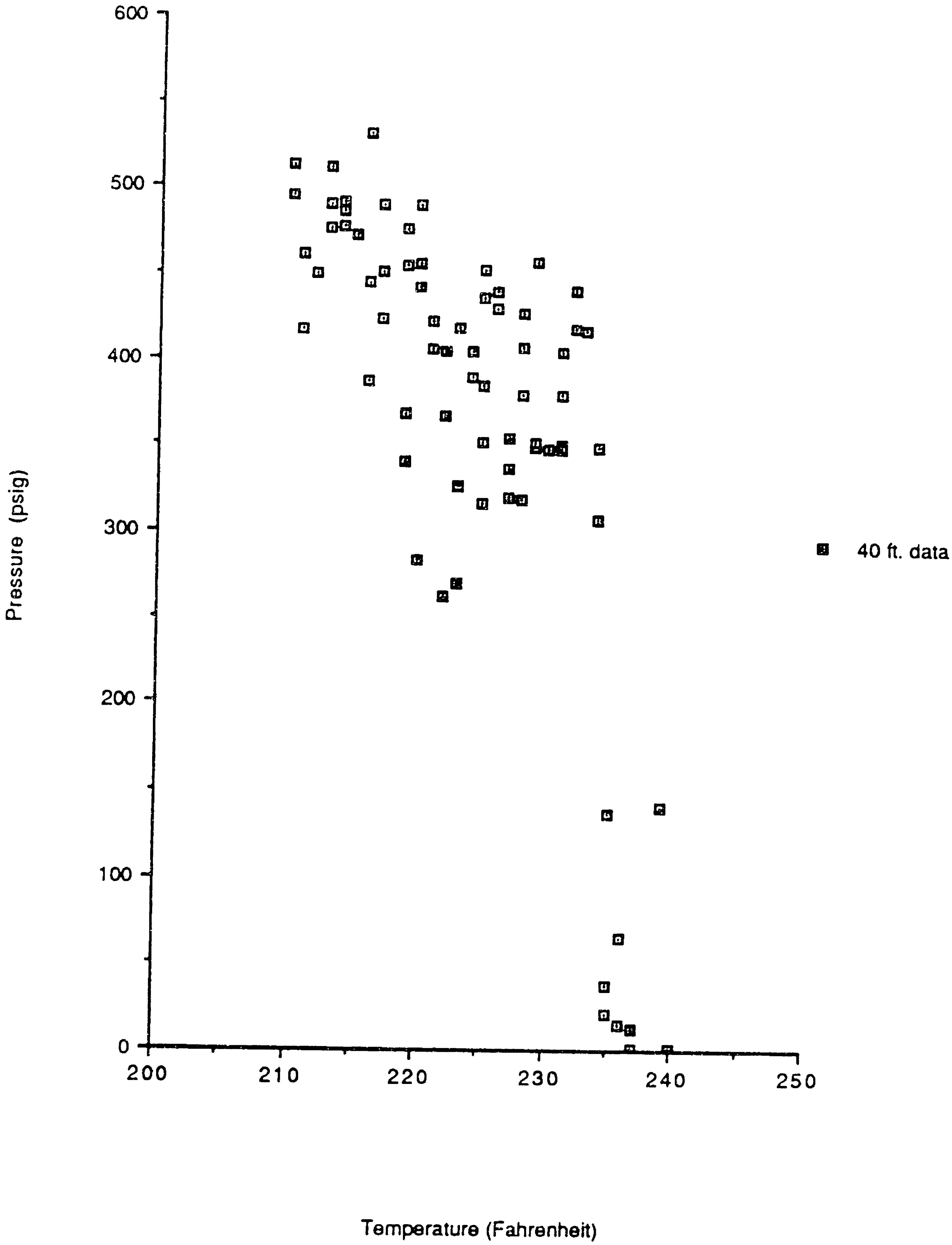

Figure 6. (40 ft pipe data)

Figure 10 shows the data for 24 foot pipe runs. Figure 11 is a graph that includes the 24 foot and 40 foot pipe data. As can be seen from this graph, the data for the 24 foot pipe runs were similar to the data of the 40 foot pipe runs, although the 40 foot pipe data were generally higher in magnitudes. For the 24 foot pipe runs, the highest pressure surge observed was 490 psig at a tank temperature of 227 F, however in general the higher pressure surges were observed for experimental runs with temperatures around 215 F. The magnitudes of the pressure surges followed a similar pattern in that at temperatures higher than 215 F there was generally a decrease in pressure surge magnitude as temeperature increased. It appears that the threshold of the biggest drop in the magnitude of the pressure surge was for runs with an initial tank temperatures within the range of 237 F to 239 F. The magnitude of the pressure surges ranged from 301 psig to 8 psig in this temperature range. The biggest discrepancy between the 40 foot and 24 foot pipe data was at the temperatures in the 210 F to 215 F range. The magnitudes of the pressure surges for the 24 foot pipe runs were considerably lower than the pressure surge magnitudes for the 40 foot pipe runs in this temperature range. However, it should be pointed out that with the initial setup for the 24 foot pipe runs the testing apparatus had only a steam inlet at the top of the tank to pressurize it. This made it difficult to obtain data below 214 F, because the water in the tank would heat up before the tank would reach an internal pressure of 10 psig. Whereas, for the 40 ft pipe runs an air inlet to the top of the tank was added so that taking cold water data would not be a problem.

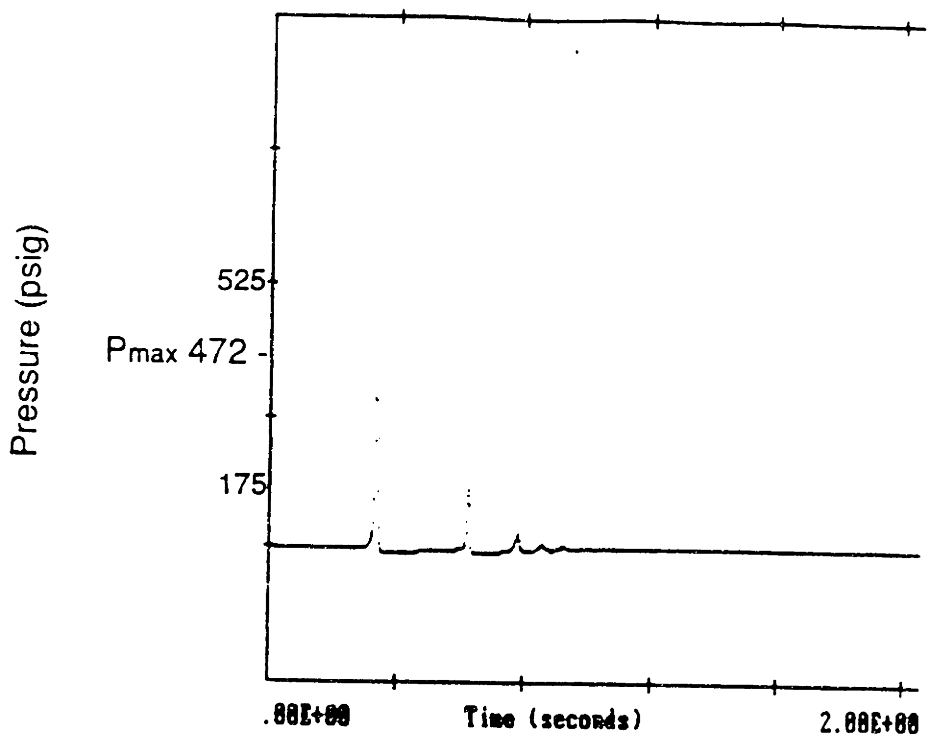

Figure 7. (Pressure vs. Time Trace, 216 F, 24 ft pipe)

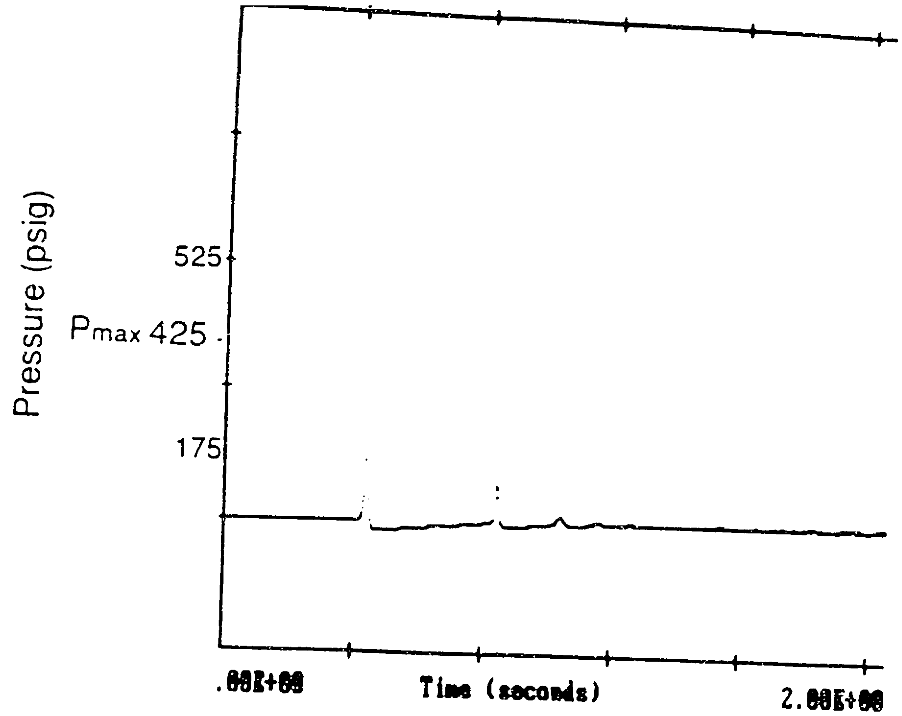

Figure 8. (Pressure vs. Time Trace, 225 F, 24 ft pipe)

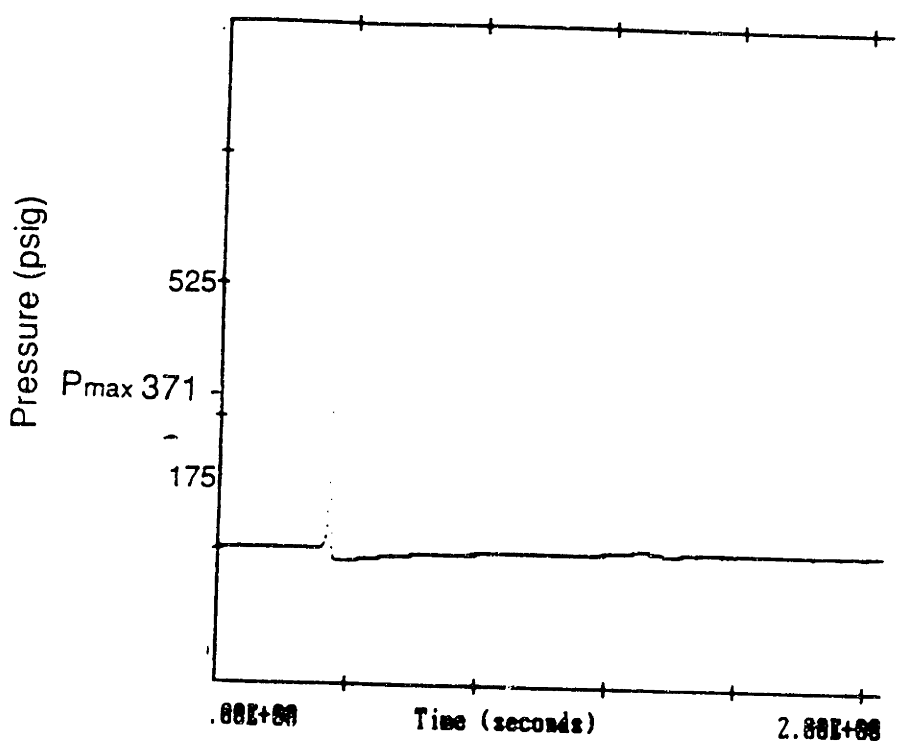

Figure 9. (Pressure v. Time Trace, 235, 24 ft. pipe)

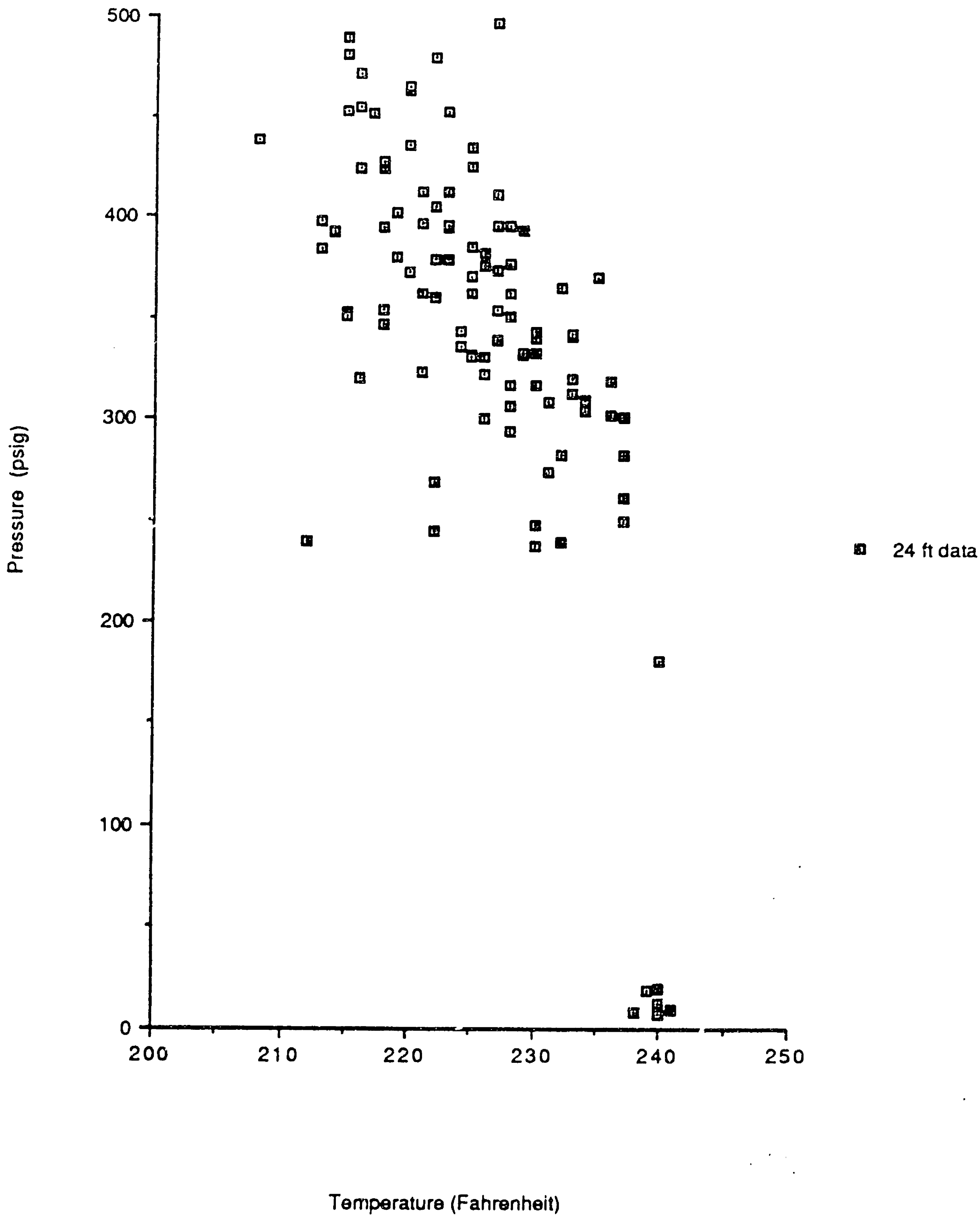

Figure 10. (24 ft pipe data)

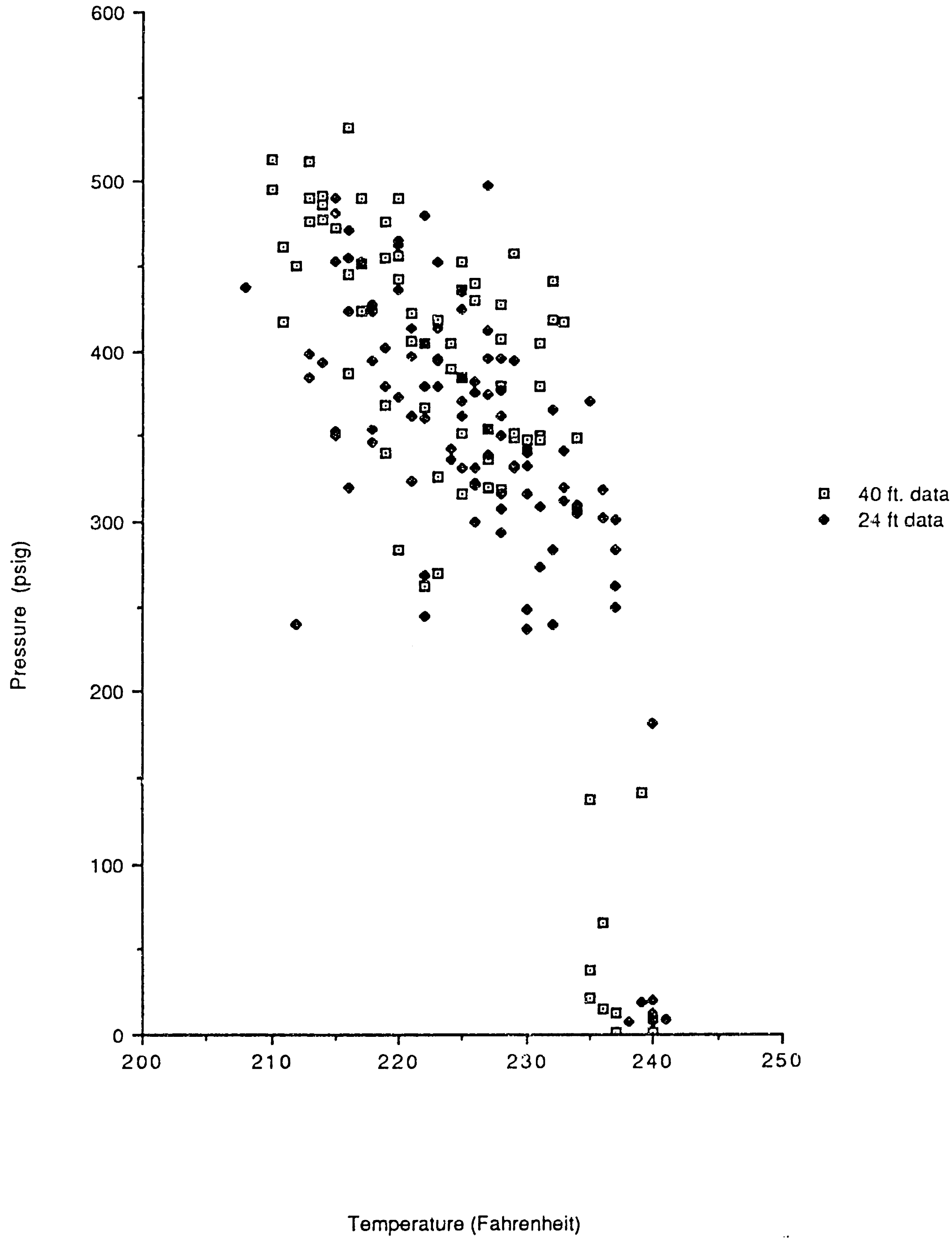

Figure 11. Composite of Data For 24 ft & 40 ft Pipes

8 Foot Pipe Data

Runs were also made using an 8 foot exit pipe. The observed magnitudes of the pressure surges were considerably smaller in magnitude than the 24 foot and 40 foot pipe. Figure B1 (Appendix B) is a graph of the 8 foot pipe data. The smaller pressure surges for the 8 foot pipe runs appear to be a result of the closing time of the manually operated ball valve. The 8 foot pipe appears to have been too short to get accurate readings because the pressure wave travels to the tank and back too fast in relation to the valve closing time. A solenoid valve was then employed instead of the ball valve, however the fluid resistance of the solenoid valve was much higher than the ball valve so that the effective exit pressure was greater than one atmosphere. Therefore, the pressure surges observed with the solenoid valve were lower than those observed with the ball valve. Thus the maximum pressure surge observed was 550 psig for a 40 foot pipe run at a tank temperature of 65 degrees F. The measured fluid velocity for this run was 10.5 ft/s. The calculated pressure surge for water at this velocity and temperature (accounting for pipe elasticity) was 600 psig. The following equation was used in determining the magnitude of this cold water pressure surge

$$P_{surge} = \rho c \Delta V, \qquad \text{Eqn. 1}$$

Considering pipe elasticity , the acoustic speed, a, is determined by the following relation taken [1]

$$a = \sqrt{(K/\rho)} / \sqrt{(1 + [(K/E)(D/e)](1 - \mu/2))} \qquad \text{Eqn. 2}$$

Single-Phase Water Hammer

In our experiments we have observed water hammer due to rapid valve closure for runs that had either single-phase or flashing two-phase flows. The water hammer that occurs in the single-phase cold water runs due to rapid valve closure is described by E. Benjamin Wylie and Victor L. Streeter [1]. The description in [1] will be used to explain the water hammer phenomenon that occurs in the single-phase cold water runs.

When the valve is is initally closed the fluid at the valve interface is immediately brought to rest. A high pressure develops at the valve-fluid interface. This high pressure compresses the fluid layer at the valve interface and causes the pipe to expand. The high pressure then travels as a wave upstream bringing the fluid upstream of the valve interface to rest and causing the pipe to expand. This high pressure wave travels upstream, at the sound velocity, a, which has been previously described. Fluid continues to flow into the pipe from the tank until the high pressure wave reaches the tank. Since the pressure in the tank remains at 10 psig and the pressure in the pipe is higher than the tank's pressure, the fluid in the pipe begins to flow backwards into the tank. This backward flow returns the pipe pressure to about that which existed before valve closure. At the time 2 L/a seconds after valve closure the whole pipe is back to the pressures that existed before valve closure with the fluid flowing backwards toward the tank exit. However, at this instant a low pressure develops at the valve since the backward flow cannot be maintained once the fluid at the valve interface begins to flow backwards.

The low pressure travels as a wave upstream towards the tank bringing the fluid along the way to rest. The low pressure wave reaches the tank exit at 3 L/a seconds after valve closure. At this instant the whole pipe is at the same negative pressure with the fluid at rest. Since the tanks pressure (10 psig) is higher than the pipe's pressure, the fluid begins to flow into the pipe, returning the pipe to the pressures it had before valve closure. At the time 4 L/a seconds after valve closure the wave reaches the valve. At this instant, the fluid velocity and pressures along the pipe are the same as at the instant of valve closure. As claimed in reference [1] fluid friction and the imperfect elasicity of the pipe wall and fluid damp out the vibrations causing the subsequent pressure rises to decrease in magnitude and eventually the fluid comes to rest.

Two-Phase Water Hammer

The above description of water hammer has been documented for single-phase flow. However, most of the observed runs were flashing two-phase flow runs or hot water single-phase flow runs. Investigations of the fluid transients that occur in these two-phase flow runs, suggest that these runs, particularly those runs with lower void fraction, result in transients that have single-phase water hammer characteristics. This could be observed by looking at figures 224 F and 225 F (figures 4 and 8). These figures show data from flashing two-phase flow runs, however the magnitudes of the initial and subsequent pressure surges and the time between subsequent surges suggest that we are looking at a single-phase water hammer phenomenon. Therefore, we will attempt to explain the water hammer for these two-phase flashing flow runs employing the above description of single-phase water hammer.

These higher temperature flashing two-phase runs initially have fluid flowing into the tank at a certain velocity. When the valve is suddenly closed the fluid at the valve interface is brought to a rest. There is no large immediate pressure rise, because unlike the single-phase flow run, there is a certain amount of void in the pipe. This void is compressed and condensed. Fluid continues to flow into the pipe from the tank. As the void in the pipe is being filled with liquid there is a gradual pressure rise in the pipe. Once all the void in the pipe is filled with liquid water, essentially there is a single-phase water hammer. The valve interface experiences a large sudden pressure surge. A pressure wave travels at the speed of sound in the liquid upstream of the valve bringing the liquid throughout the pipe to rest.

Once the high pressure wave reaches the tank, fluid begins to flow backward into the tank from the pipe because the pressure in the pipe is higher than the pressure in the tank. As previously described this backward flow conversion occurs throughout the pipe at the sound velocity. At the instant the fluid at the valve interface begins to flow backwards, a low pressure wave develops at the valve and travels upstream as a wave bringing the fluid along the way to rest. As this low pressure wave travels upstream some of the liquid flashes since the pressure in the pipe is now below the vapor presssure of the fluid. The pipe walls contract due to the low pressure. Once the low pressure wave reaches the tank, the pipe is at same uniform negative pressure with the fluid at rest. At this instant, fluid begins to flow into the pipe from the tank. At the instant the wave reaches the valve, the fluid velocity and the pressure distribution throughout the pipe are the same as at the instant of valve closure and the whole process is repeated. As stated before, fluid friction and the "imperfect elasticity " of the pipe wall and fluid damp out the vibrations, so the subsequent pressure surges are attentuated.

Pressure Surge Duration

Since the pressure surge acts over only a brief period, its duration, as well as its magnitude must be known in order to determine the impulse involved in the water hammer. From the above description of the water hammer phenomenon, the pressure surge duration can be predicted through the following relation:

$$t_{surge} = 2\ L/a \qquad \text{Eqn.3}$$

From the above description the sonic velocity, a, should be that for liquid water in an elastic pipe approximately 4,700 ft/s. However, the experimental data shows that the sound velocity during the pressure surge is approximately 1500 ft/s, approximately a third of the sound velocity for liquid water with no void present in the pipe. This suggests that not all the void had been collapsed. It is possible that there were a small amount of air bubbles present in the pipe. Using 1500 ft/s as the value of the two-phase (air and water) wave velocity we were able to approximate the air void fraction in the pipe through the following relation: [2]

$$a_{tp} = [\ (\alpha\bar{\rho}/\rho_v a_v^2\ )\ (\ (1-\alpha)\bar{\rho}/\rho_l a_l^2\ ]^{-1} \qquad \text{Eqn.4}$$

The void present in order to have a sonic velocity of 1500 ft/s is approximately 10^ -5.

Model Development

Figure 12 is an illustration of the actual flashing two-phase flow throughout the exit pipe run. For the flashing two-phase flow runs the further one goes downstream in the pipe, the greater will be the void fraction. Figure 13 shows a qualitative relation of void to pipe length. Keeping in mind the single-phase characteristics of the two-phase flow water hammer, in developing a hand calculation model, several assumptions were made. All the void was clustered at the valve. Figures 14 and 15 illustrate these model assumptions. We took the vapor bubbles to be clustered downstream of the pipe at the valve at the instant of closure. We assumed that the vapor bubbles were at the saturation pressure for the temperature in the tank. It was assumed that the liquid existed in the pipe upstream of the vapor bubble cluster at the valve. Therefore, since we know the tank pressure and assumed the pressure near the valve to be the vapor pressure in the bubble, we were able to determine the liquid velocity, V_{impact}. From equation 5 we are able to determine the magnitude of the initial pressure surge.

$$P_{surge} = \rho_l * V_{impact} * c \qquad \text{Eqn. 5}$$

is the density of the liquid for the tank temperature. V_{impact} is the velocity of the fluid at the instant of valve closure. The sound speed, c,

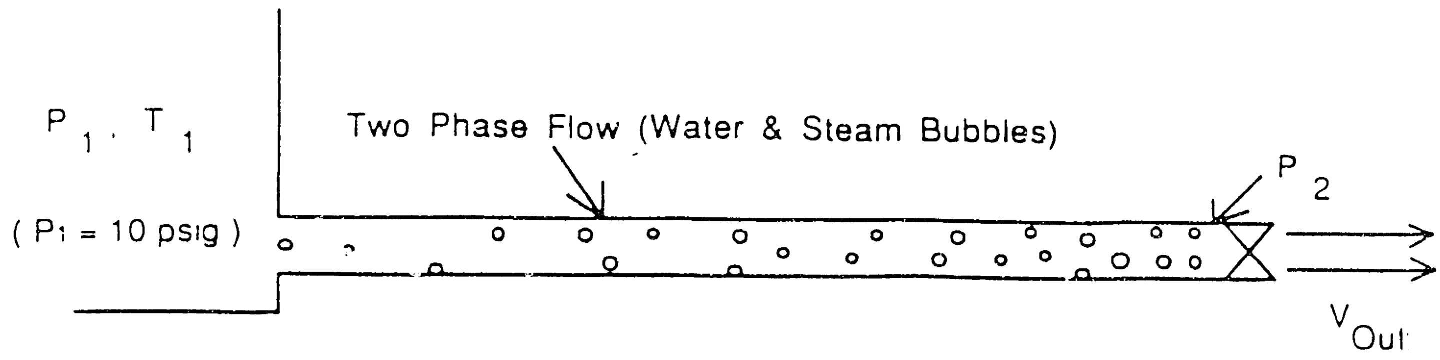

Figure 12 Illustration Of Actual Flashing Pipe Flow

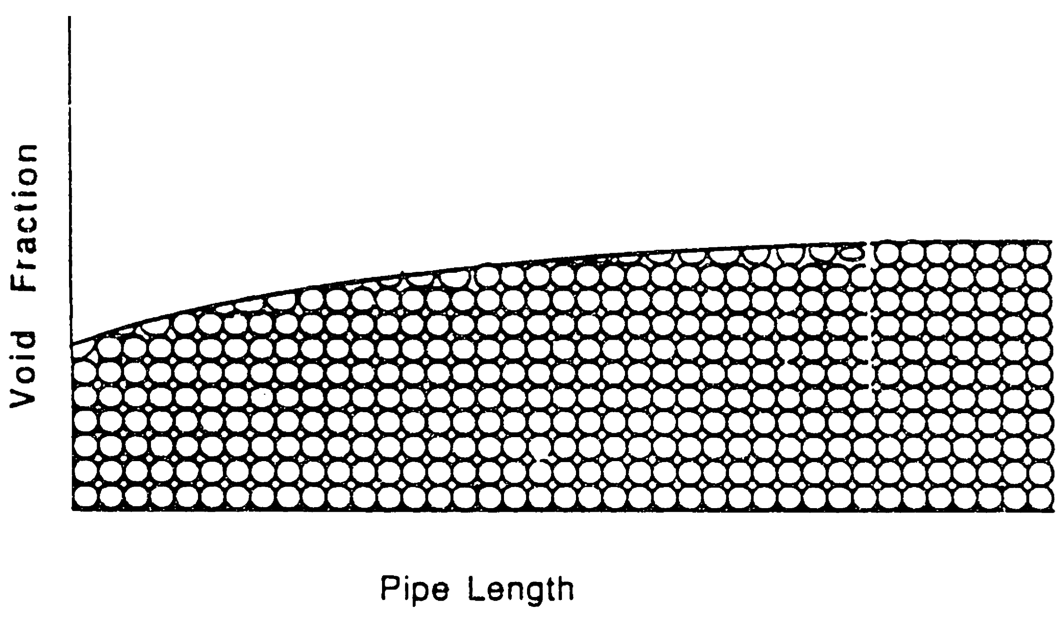

Figure 13 Relation of void to pipe length

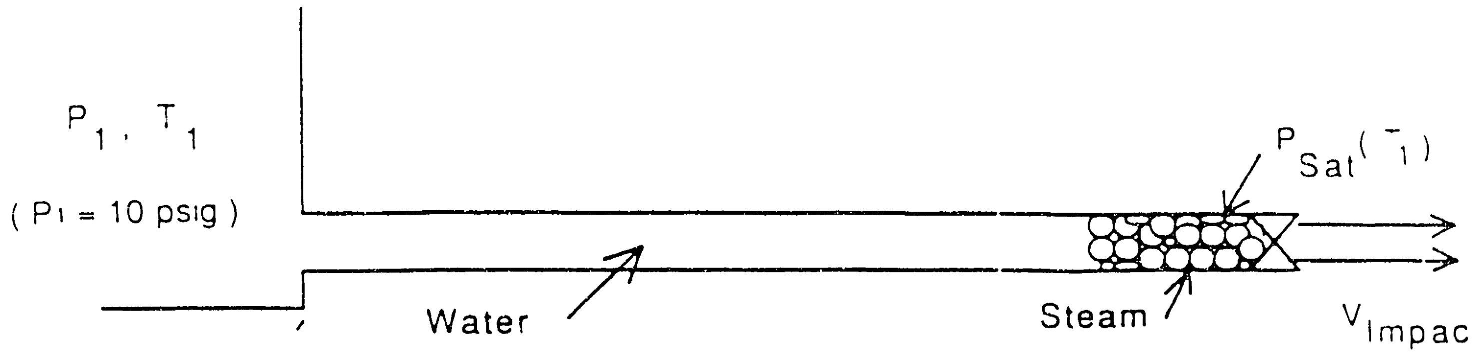

Figure 14 Model Assumption of Flashing Flow

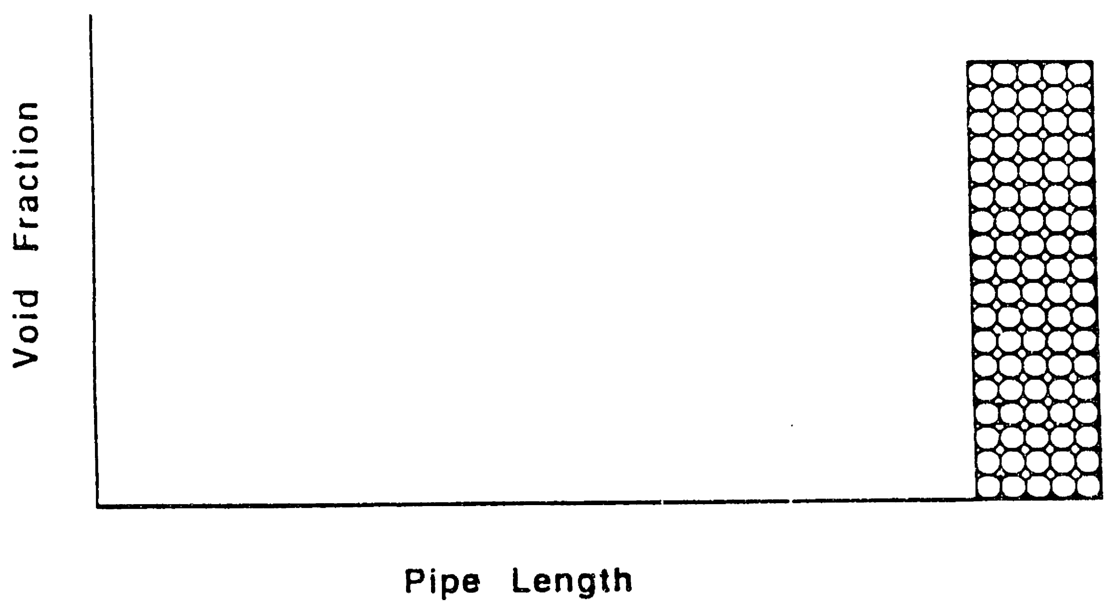

Figure 15 Assumed relation of void to pipe length

used in the calculation was that of cold water in a small pipe 4,680 ft/s. Even though the liquid in the pipe is at higher temeperatures, we used the cold water sound speed to predict the initial pressure surge magnitudes for runs at any temperature, as the answer is conservative.

Figure 16 is the plot comparing the calculated values for both the 24 foot pipe and the 40 foot pipe. As can be seen from the plot, the lower single-phase run temperatures around 210 degrees F, the calculated values for the 24 foot pipe were up to 20 percent higher than the calculated values for the 40 foot pipe. At higher tank temperatures, where there would be a greater presence of void the discrepancy is not as great.

Figure 17 is a composite plot containing the experimental data and the calculated values of the pressure surge. The plot shows that the calculation method offers a reasonable tool for predicting the magnitude of the initial pressure surge. The model generally is conservative especially at tank temperatures lower than 230 F.

RELAP5 Runs

Some of the experimental runs were simulated using RELAP5. These RELAP5 runs simulated actual experimental runs for tank temperatures of 210 F, 216 F, 224 F and 235 F for all three pipe lengths of 40 foot, 24 foot and 8 foot. All the RELAP5 runs we made had a time step of 1 millisecond.

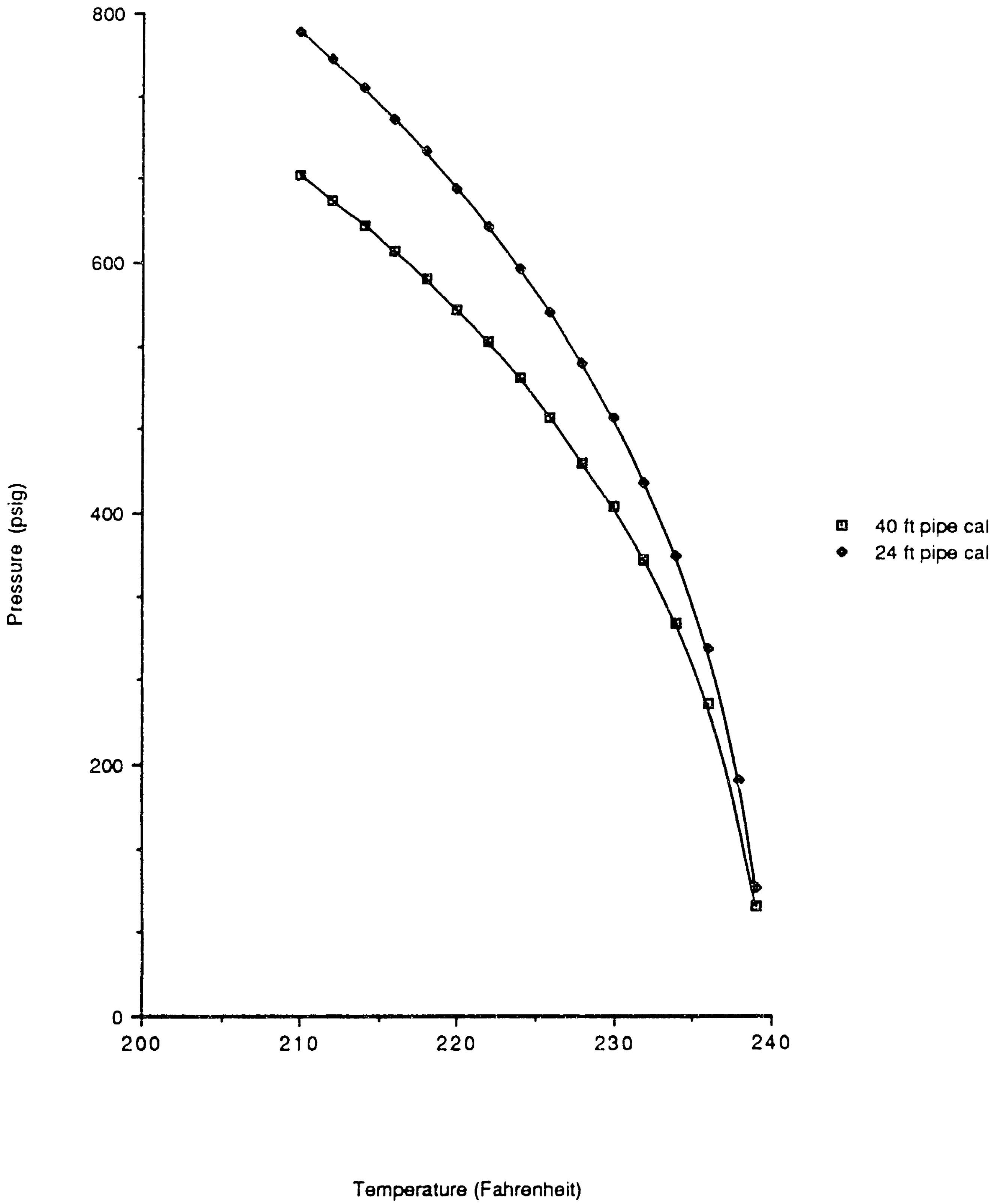

Figure 16 Composite of 24 ft & 40 ft pipe calc. values

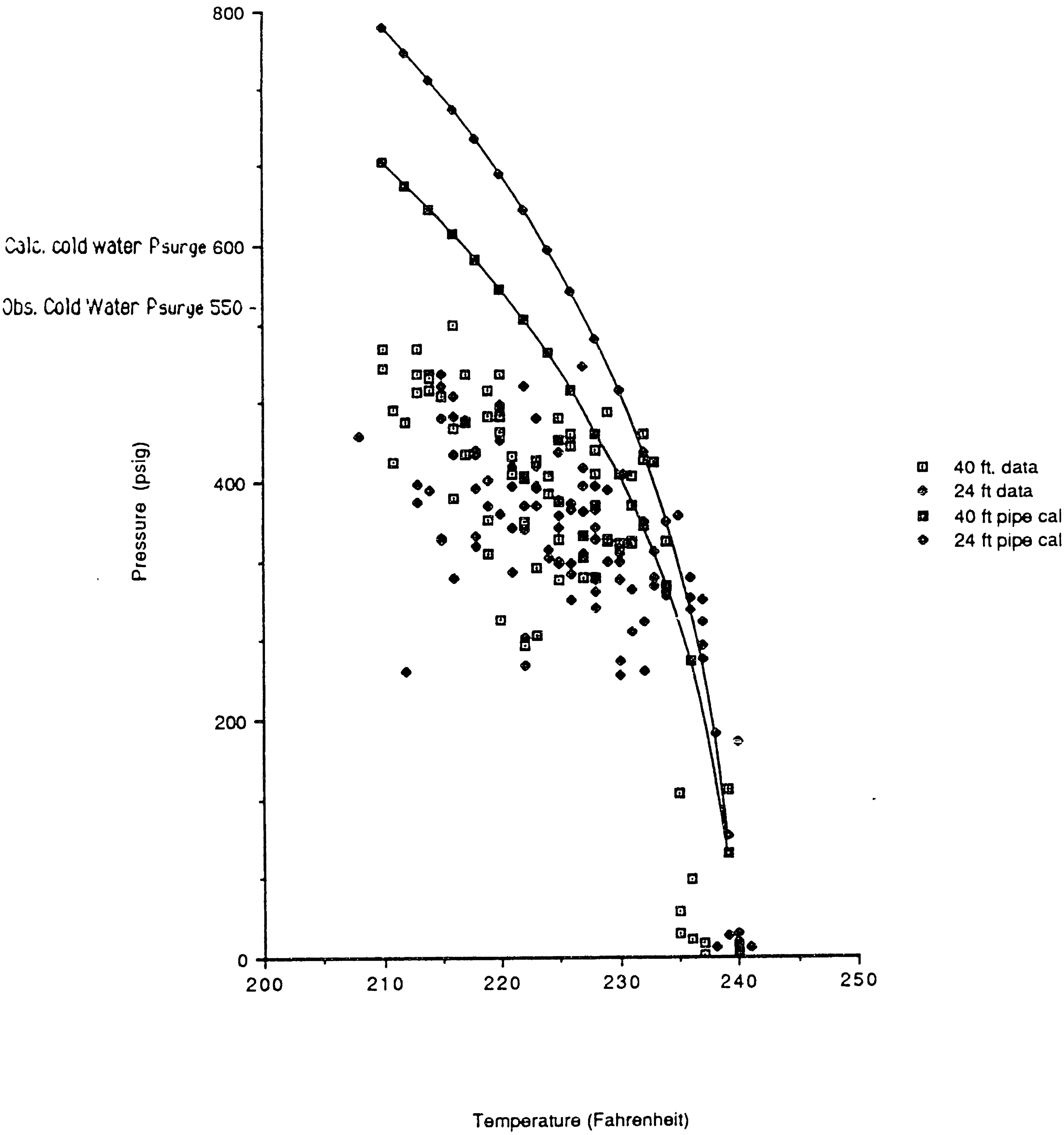

Figure 17 Composite of 24 & 40 ft pipe calc and exp. data

The 40 Foot Pipe RELAP5 Runs

Figures 18, 19, 20 and 21 are pressure vs. time traces from simulations of the 40 foot pipe runs. In simulating the experimental runs for tank temperatures of 210 F, 216 F, 224 F the RELAP5 program was run for 1 second before valve closure and 1 second after the valve closure. The inputted valve closure time was 1 millisecond. The calculated steady state mass flow rate was inputted in the cards so that the program did not have to run too long before valve closure to assure that steady state had been reached. The plots for these simulated runs show that there was just one initial pressure surge after valve closure and no subsequent pressure surges. Looking at the wave travelling time, we assume that if the program had been run longer than 1 second after valve closure we might have seen some subsequent pressure surges after the initial pressure surge.

Figure 21 is the pressure vs. time trace for the 235 F simulated run. The program was run for 1 second before valve closure and 3 seconds after valve closure. Figure 21 shows that there is an initial pressure surge of 360 psia 2.2 seconds after valve closure and there is a subsequent pressure surge of magnitue 200 psia 2.4 seconds after valve closure.

The 24 foot pipe RELAP5 runs

Figures 22, 23, 24 are RELAP5 generated pressure vs. time traces that simulated the 24 foot pipe runs with tank temperatures of 210 F, 216 F and 224 F respectively. The pressure surge magnitudes predicted by RELAP5 for these simulated runs were 914 psia, 911 psia and 780 psia,

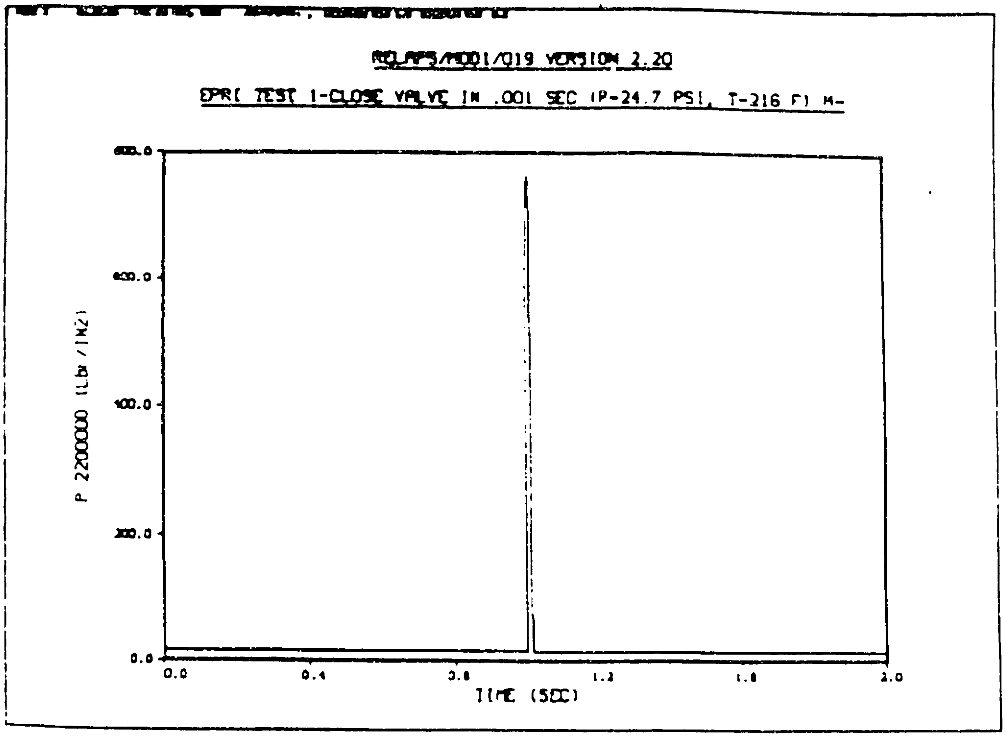

Figure 18. (RELAP5 Plot, 210 F, 40 ft Pipe)

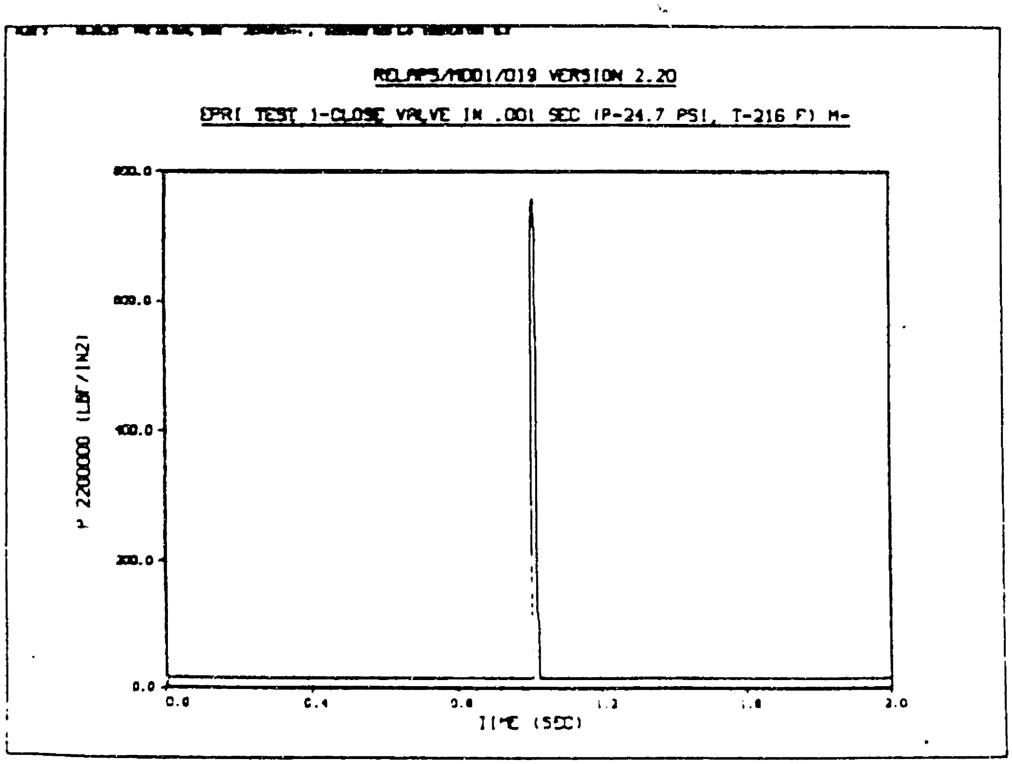

Figure 19. (RELAP5 Plot, 216 F, 40 ft Pipe)

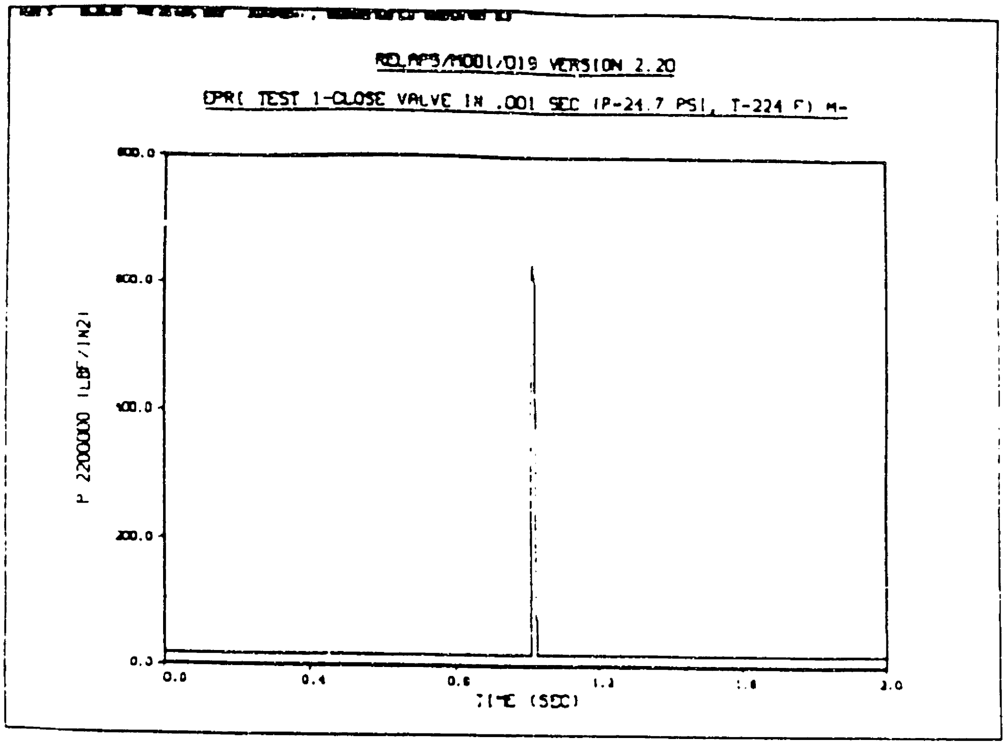

Figure 20. (RELAP5 Plot, 224 F, 40 ft Pipe)

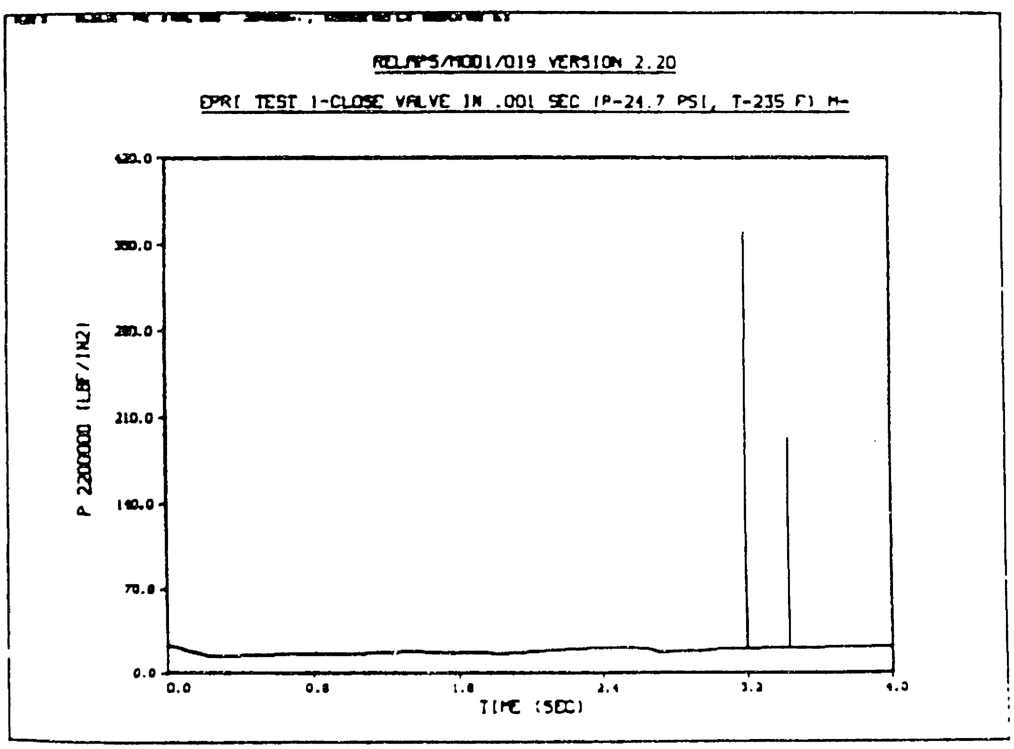

Figure 21. (RELAP5 Plot, 235 F, 40 ft Pipe)

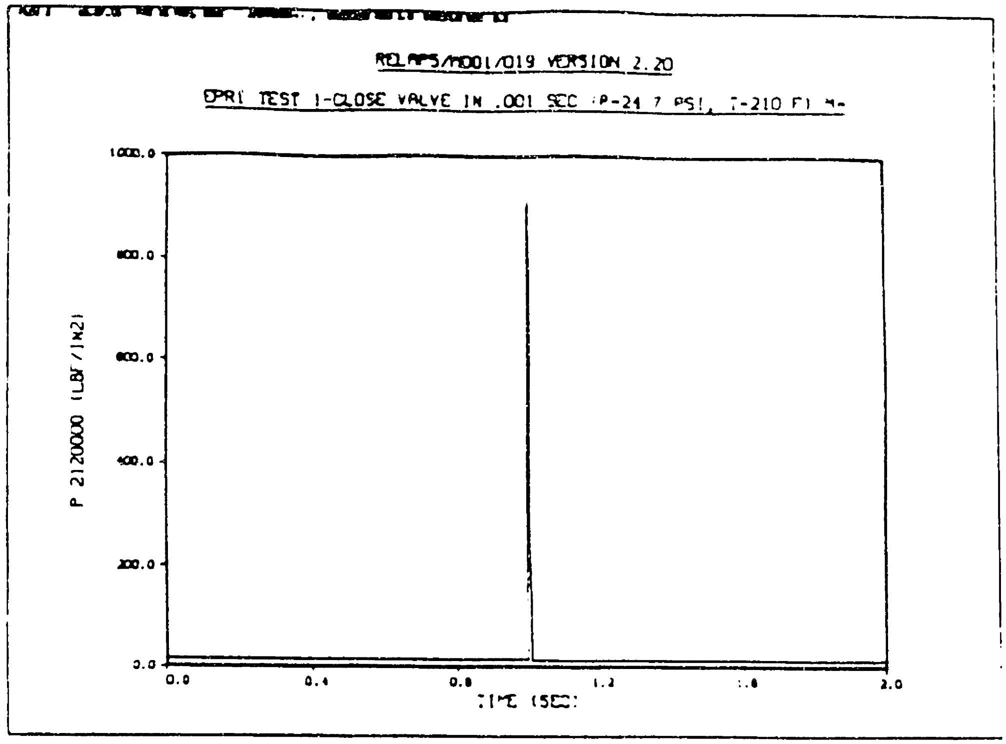

Figure 22. (RELAP5 Plot, 210 F, 24 ft Pipe)

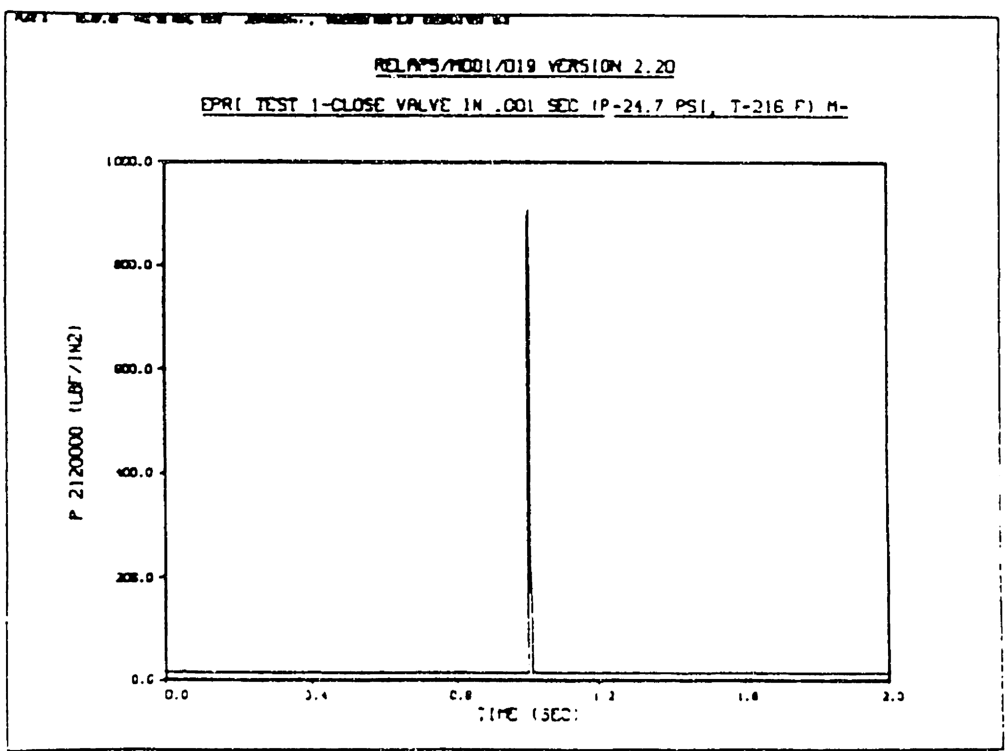

Figure 23. (RELAP5 Plot, 216 F, 24 ft Pipe)

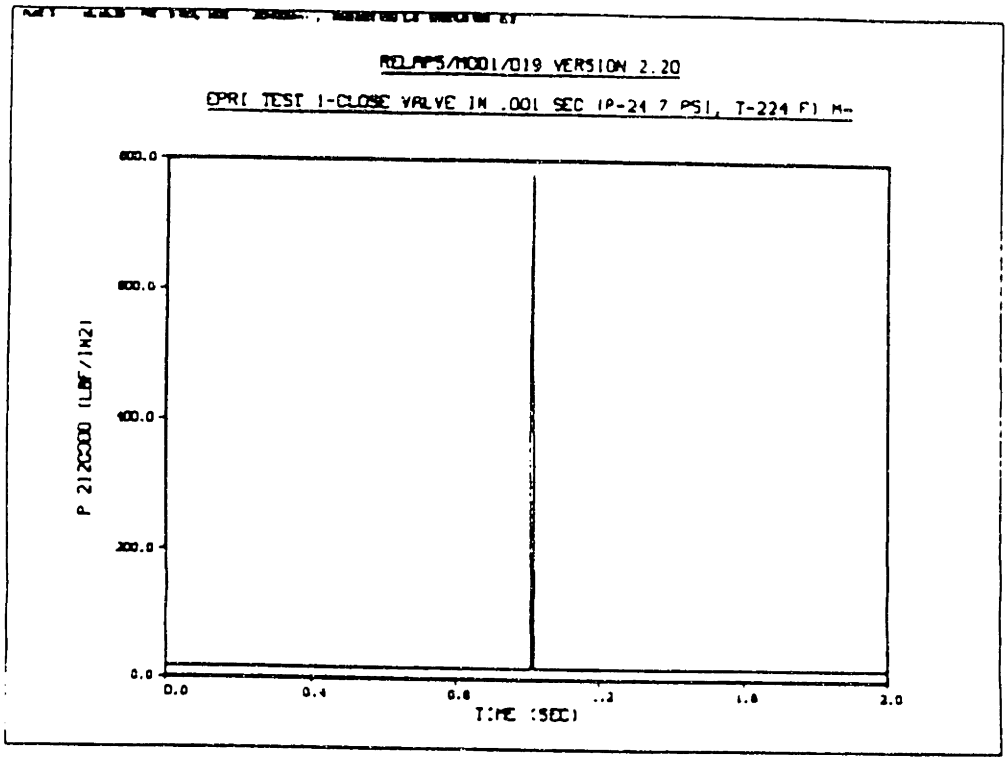

Figure 24. (RELAP5 Plot, 224 F, 24 ft Pipe)

respectively. As with the 40 foot pipe runs there were no subsequent pressure surges after the initial pressure surge within the runtime of the program.

We also simulated an experimental run with a tank temperature of 235 F but we did not obtain a complete pressure vs. time trace for this run. The magnitude of the initial pressure observed for this run was 102 psia.

Figures 25, 26, 27 and 28 are the pressure vs. time traces for the RELAP5 runs that simulated the experimental 8 foot pipe runs. The magnitudes of the initial pressure surges predicted by RELAP5 for 8 foot pipe runs at tank temperatures of 210 F, 216 F, 224 F, and 235 F were 928 psia, 925 psia, 946 psia and 296 psia, respectively. As can be seen in figures 25 through 28, there were several subsequent pressure surges after the initial pressure surges for all four RELAP5 simulation runs.

Figure 29 is a composite pressure vs. temperature plot for all the RELAP5 runs. Figure 30 is a plot of maximum (initial) pressure surge magnitude vs. temperature comparing the 24 foot and 40 foot pipe experimental data as well as the reults of the RELAP5 runs that simulated the experimental runs. Figure 30 shows that RELAP5 renders a conservative prediction of the initial pressure surge due to rapid valve closure. Figure 31 is a composite pressure vs. temperature plot for the RELAP5 runs and the values determined by the hand calculation method that has been previously discussed. RELAP5 rendered somewhat higher predictions of the magnitudes of the initial pressure surges compared to those predicted by the hand calculation but overall they were in good agreement.

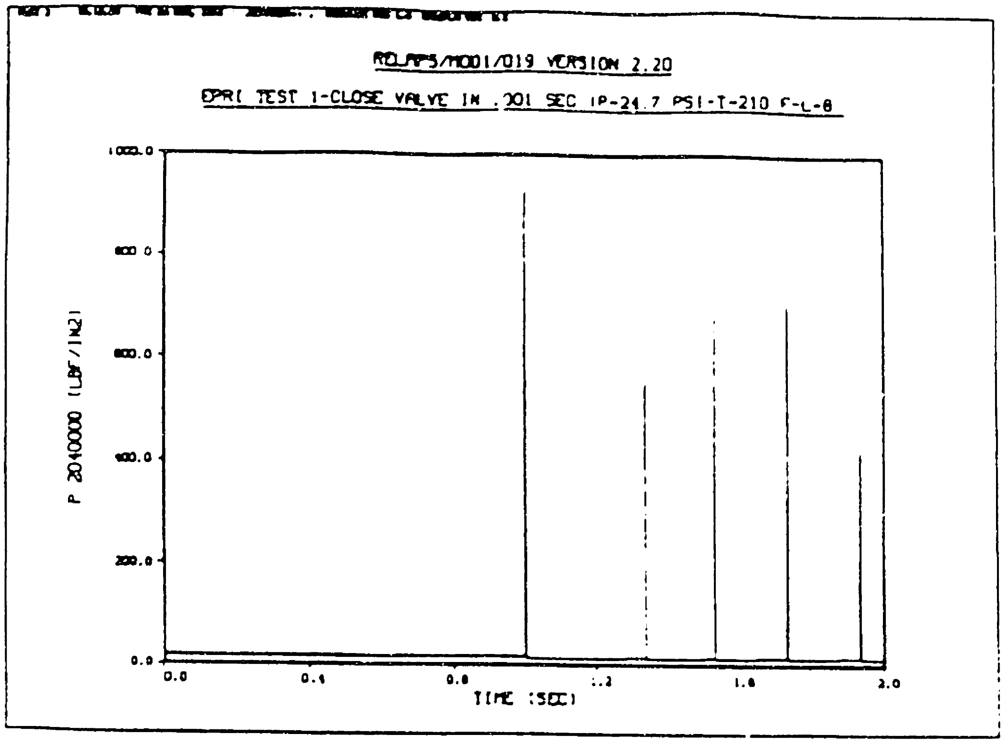

Figure 25. (RELAP5 Plot, 210 F, 24 ft Pipe)

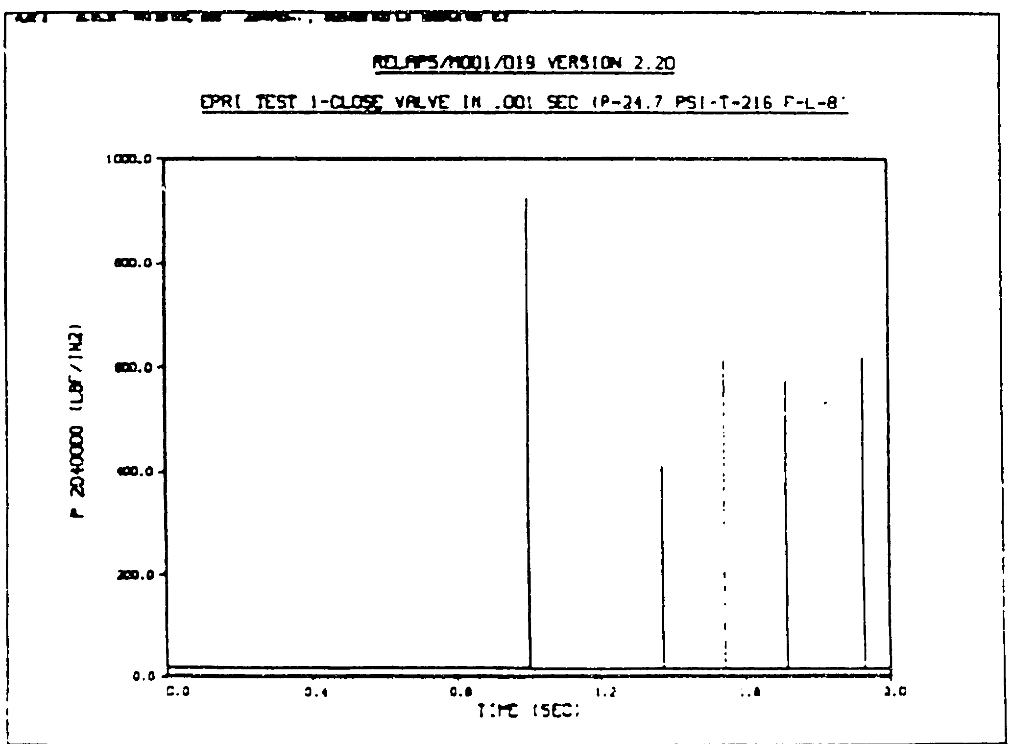

Figure 26. (RELAP5 Plot, 216 F, 8 ft Pipe)

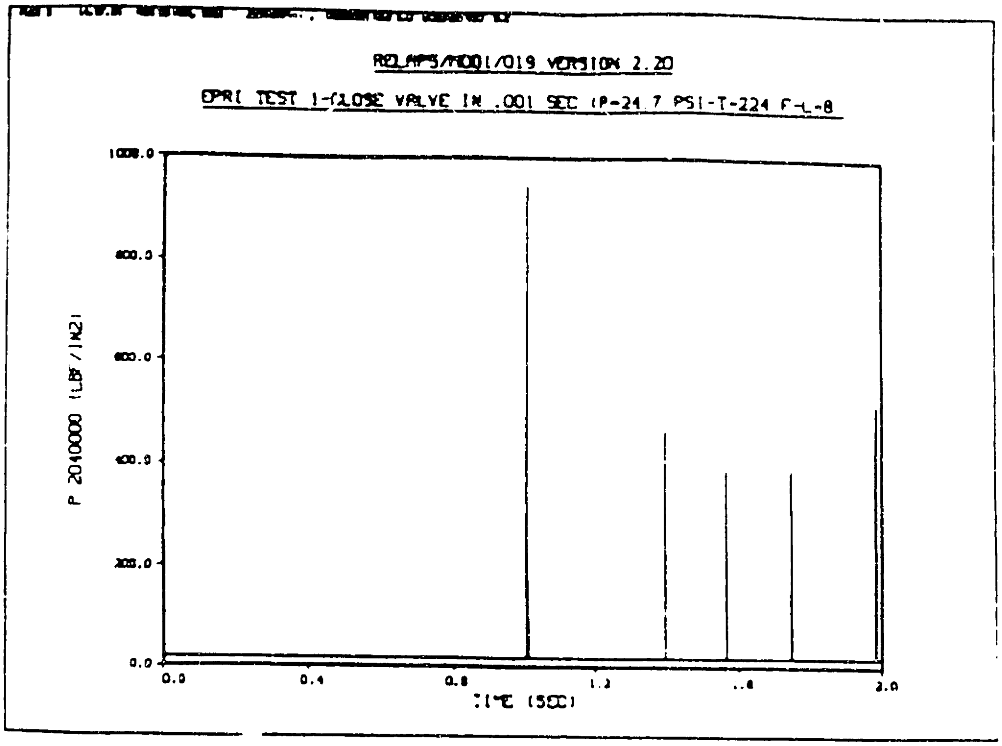

Figure 27. (RELAP5 Plot, 224 F, 8 ft Pipe)

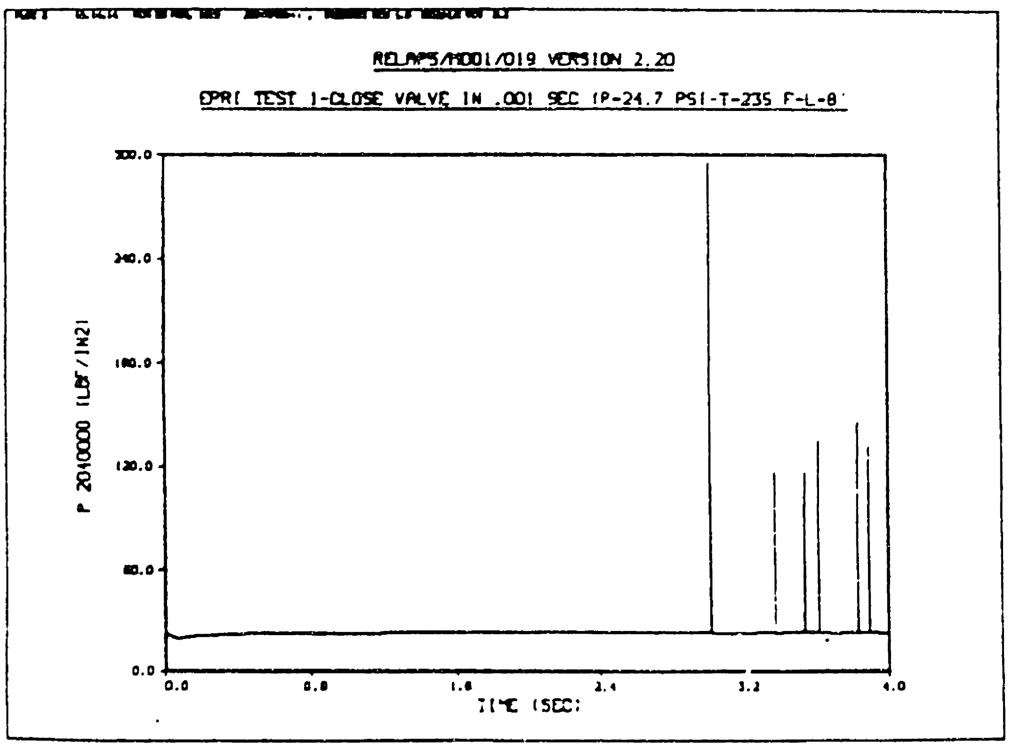

Figure 28. (RELAP5 Plot, 235 F, 8 ft Pipe)

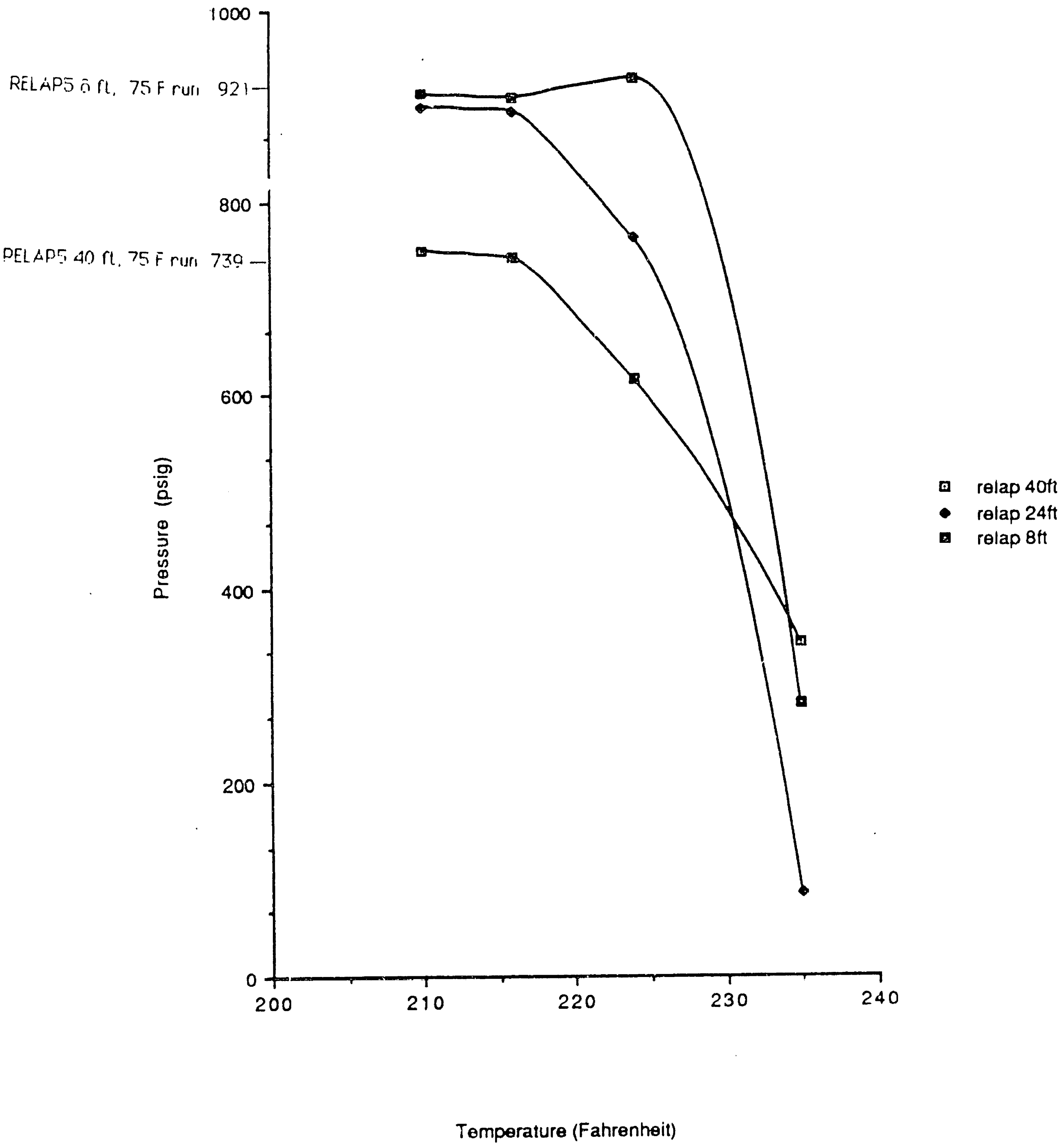

Figure 29. Composite of RELAP5 Runs for the 3 Pipe Lengths

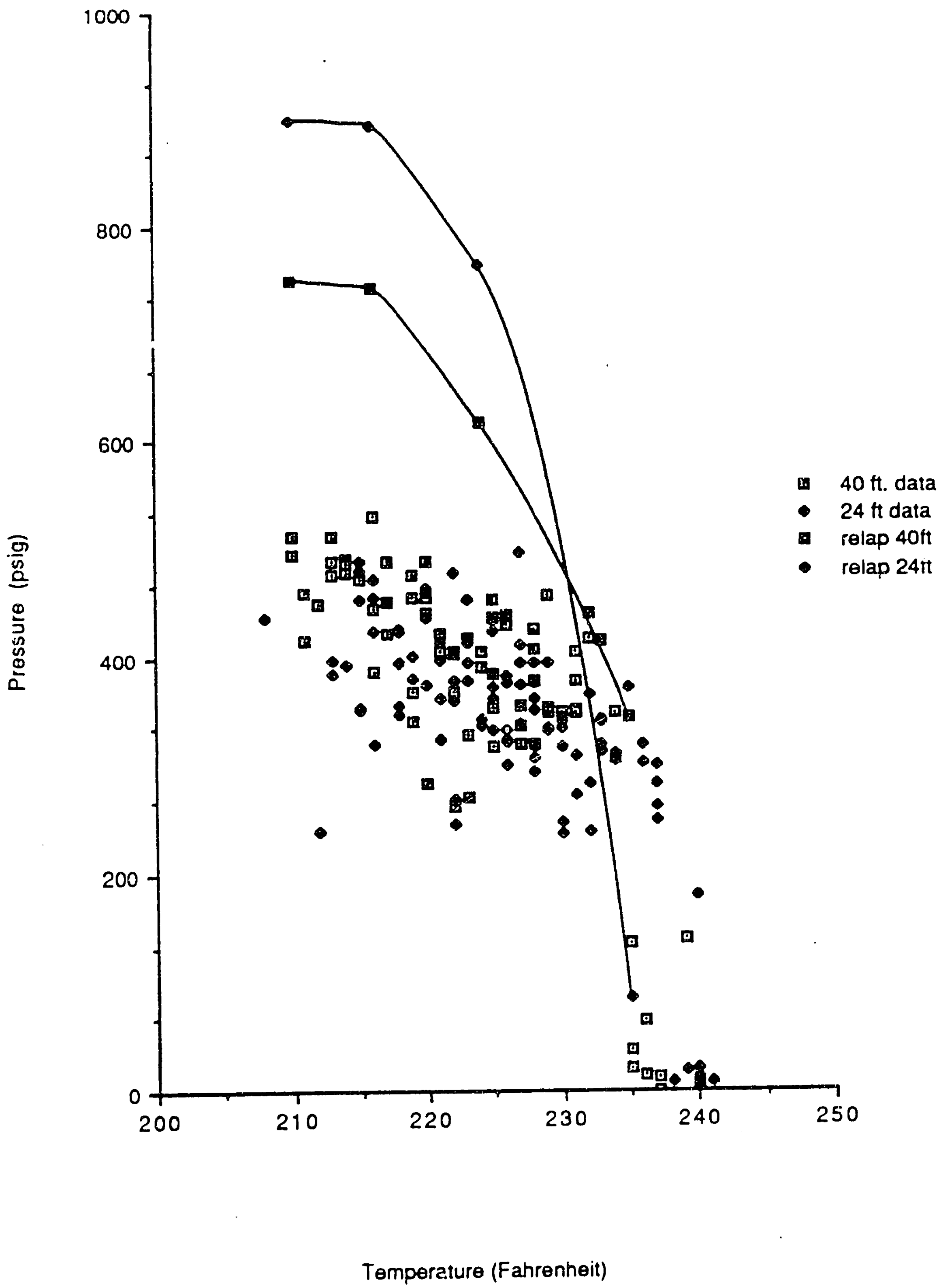

Figure 30. Composite of RELAP5 Runs & Data (24 & 40 ft Pipes)

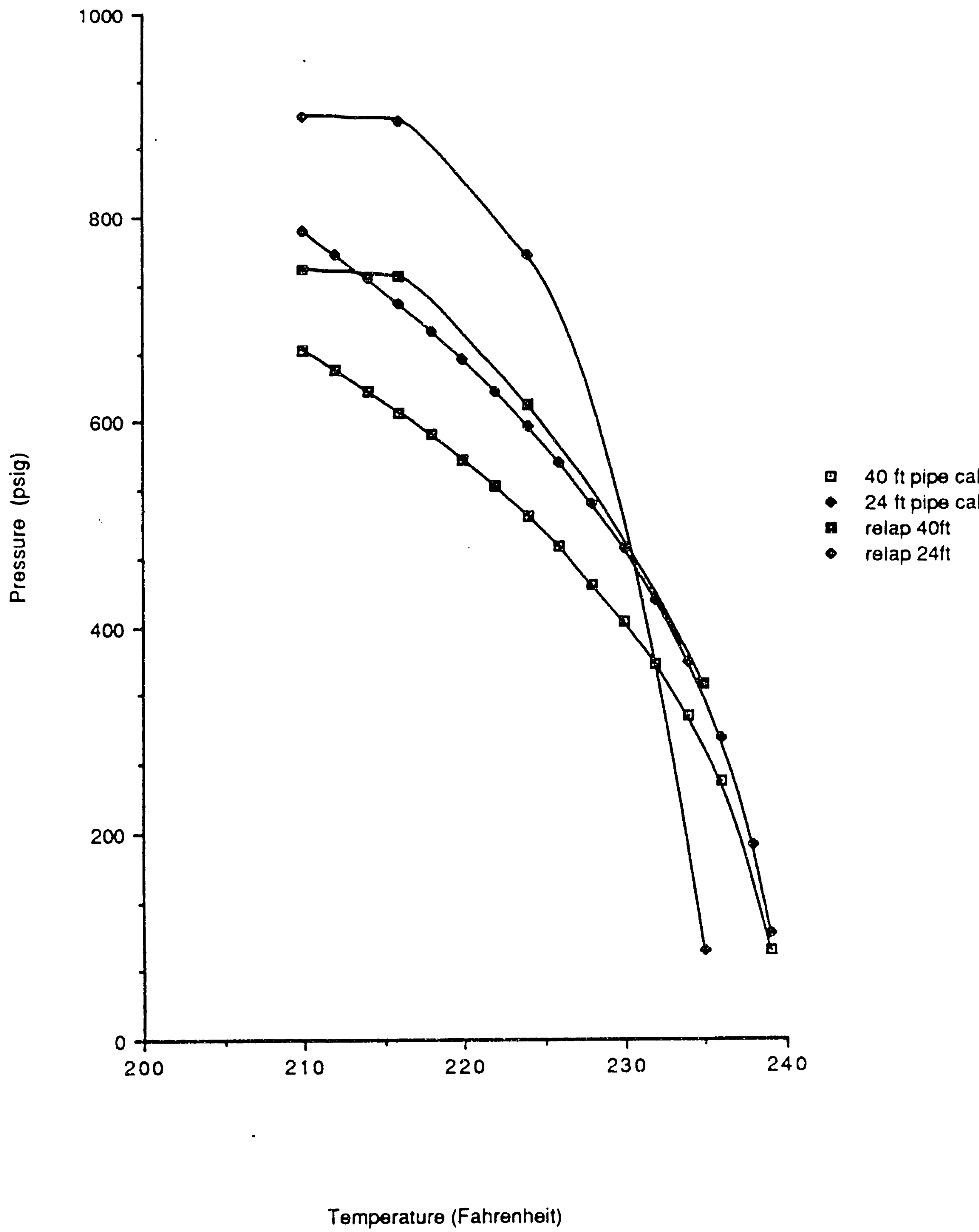

Figure 31. Comp. of RELAP5 Runs & Calc. Values (24 & 40 ft)

Void fraction Measurements

To determine the exit void fraction for the 24 foot pipe experimental runs, an 18 inch long, 1 inch I.D. pipe with two ball valves at each end was added just downstream of valve 2. Figure 32 is a drawing of the experimental apparatus with the void measurement section attached to the end of the exit pipe. A connecting rod was attached to both ball valves (3 and 4) so that the both valves could be closed simultaneously, thereby trapping the flow in the 18 inch pipe section. The test section would then be removed and the volume of water trapped in the section would then be measured and recorded. From this data we were able to determine the vapor volume fraction in the section.

Figure 33 is a plot of exit vapor void fraction vs. tank temperature. Included in the plot are the calculated values of void using the Homogenuous Method [3] and the Thom Slip Method [3] both of which overpredict the void fraction. This suggests that there is a significant departure from thermal equilibrium for the experimental runs.

From the RELAP5 runs we were able to obtain values of void fraction in the last two feet of pipe before valve 2. Although there is some difference between the RELAP5 model and the actual experimental apparatus, RELAP5's predictions of vapor void fraction are reasonably useful. Figures 34 through 39 are plots of vapor void fraction vs. RELAP5 run time for the last 2 foot volume before valve 2. These RELAP5 void plots were obtained for both the 24 foot and 40 foot pipe runs. However, the experimental void fraction data was taken for the 24 foot pipe only. The calculated values of

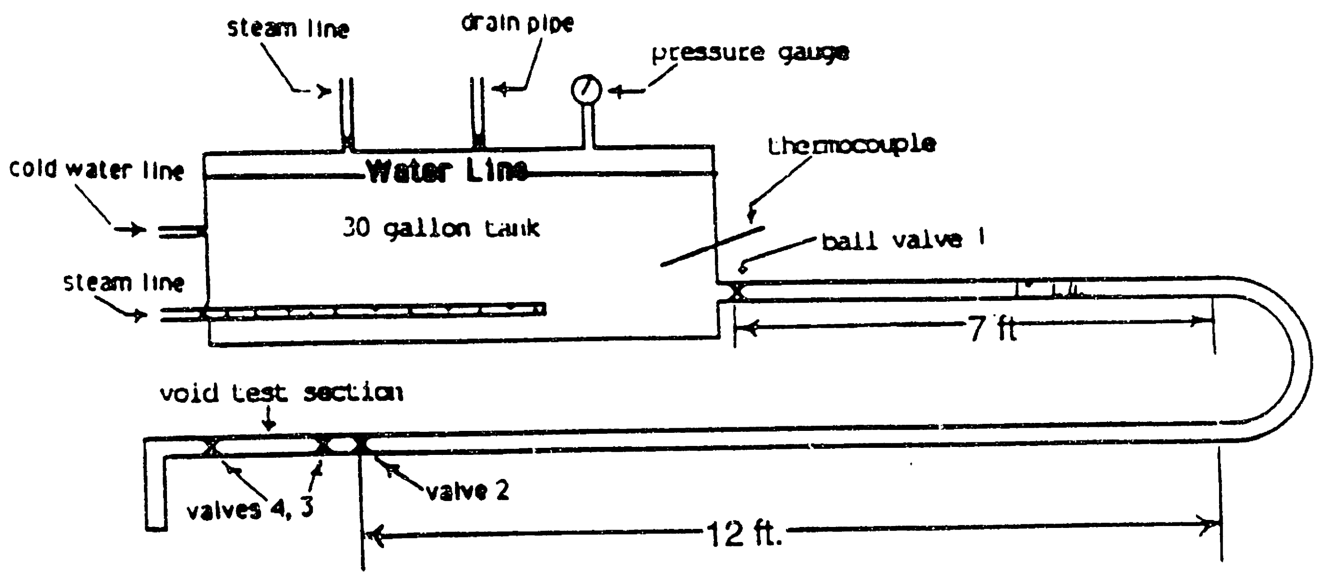

Figure 32. Drawing of Exp. Apparatus With Void Meas. Section

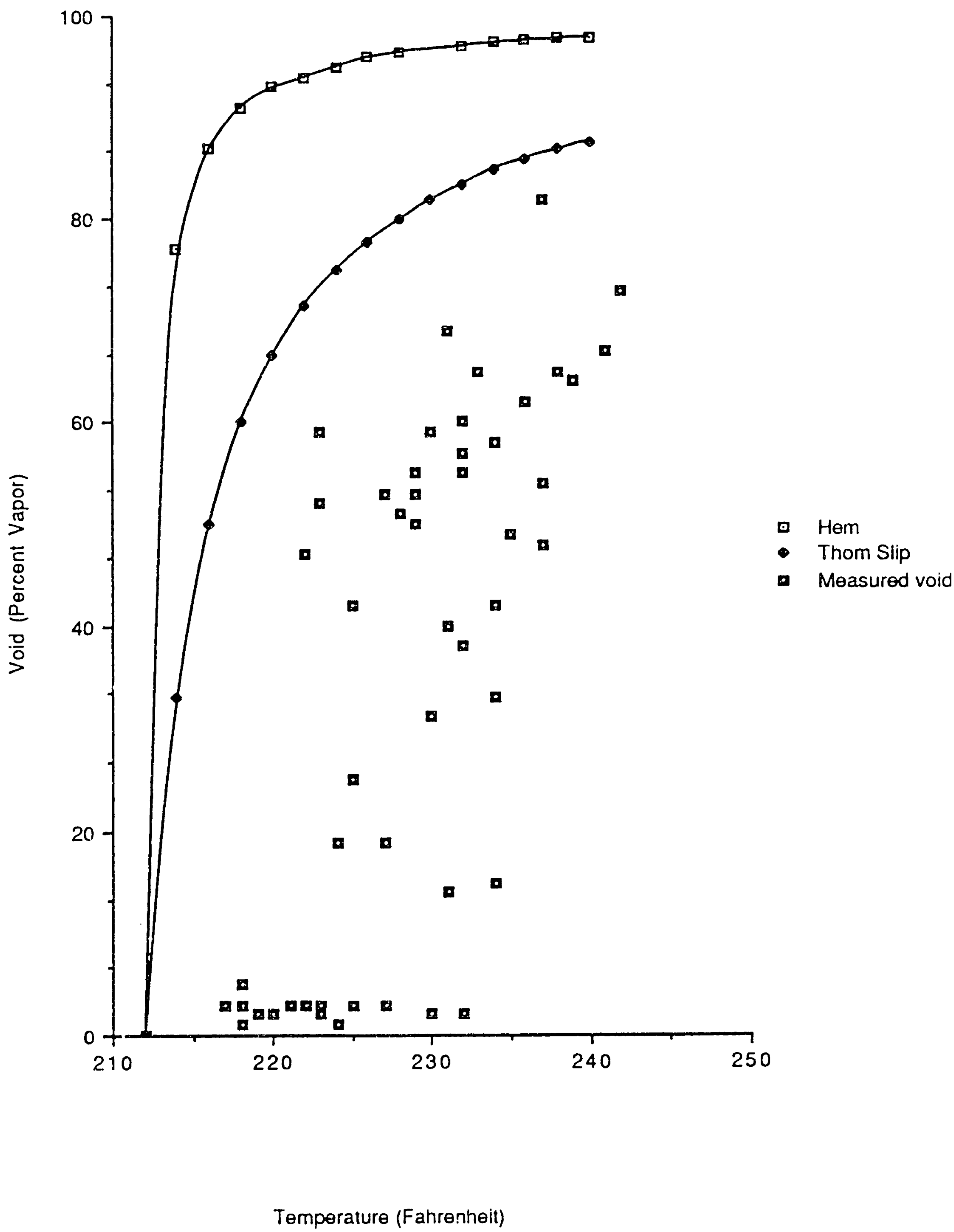

Figure 33. Composite of Calc. Exit Void & Measurements

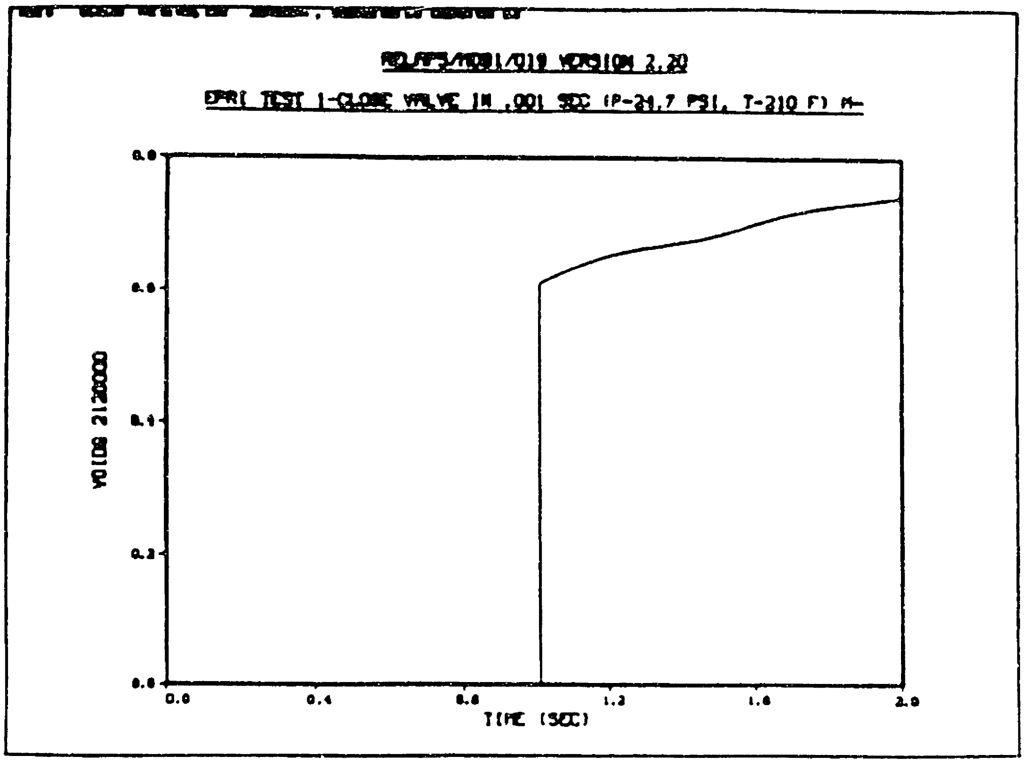

Figure 34. (RELAP5 Exit Void vs. Time Plot, 210 F, 24 ft Pipe)

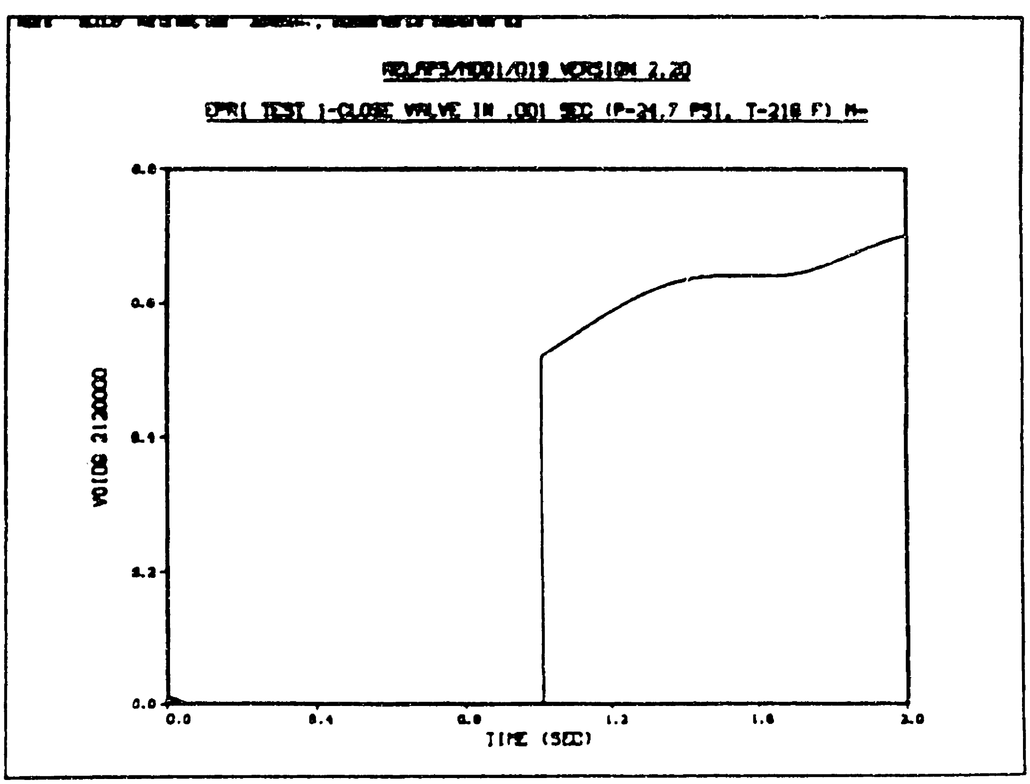

Figure 35. (RELAP5 Exit Void vs. Time Plot, 216 F, 24 ft Pipe)

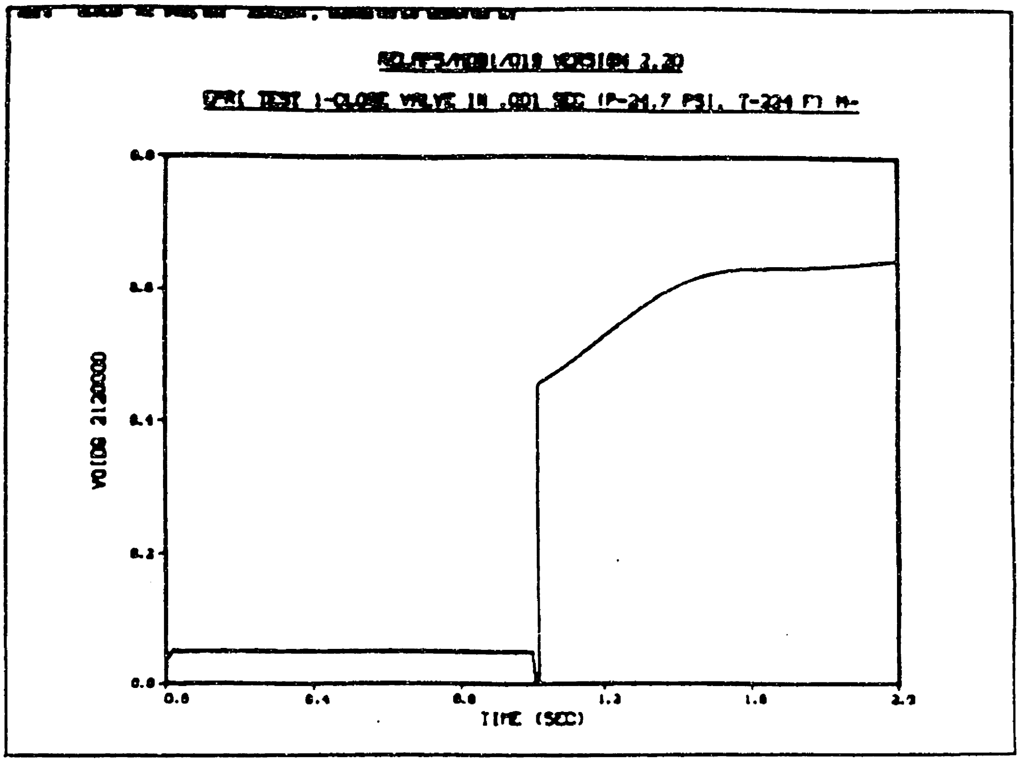

Figure 36. (RELAP5 Exit Void vs. Time Plot, 224 F, 24 ft Pipe)

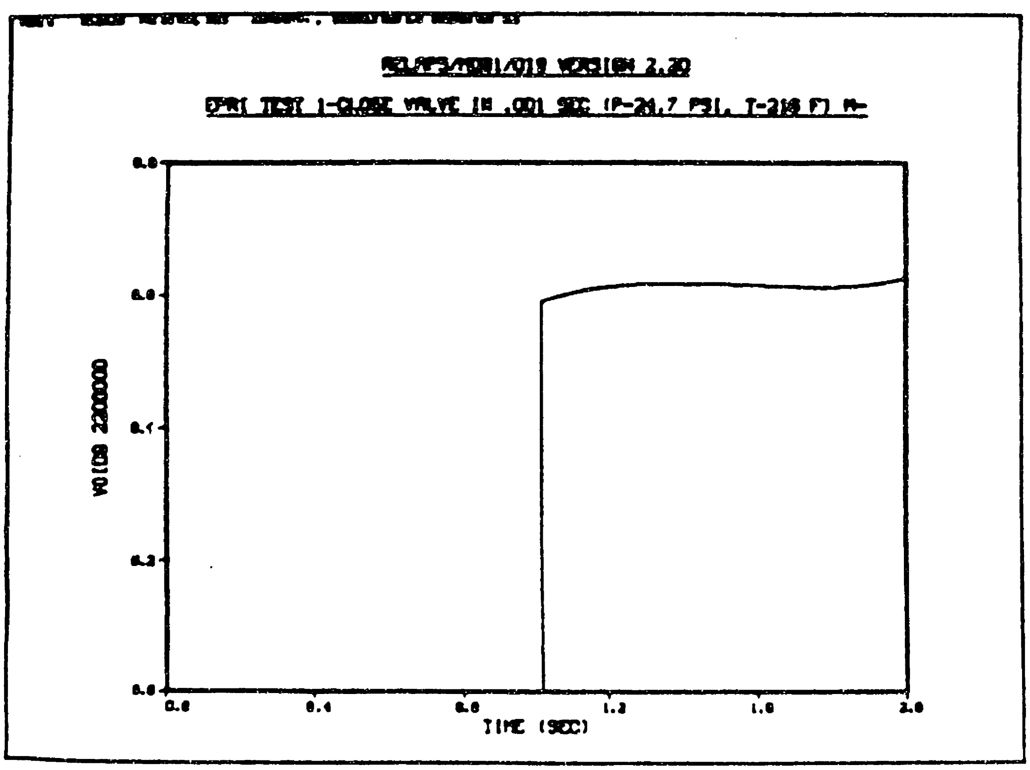

Figure 37. (RELAP5 Exit Void vs. Time Plot, 210 F, 40 ft Pipe)

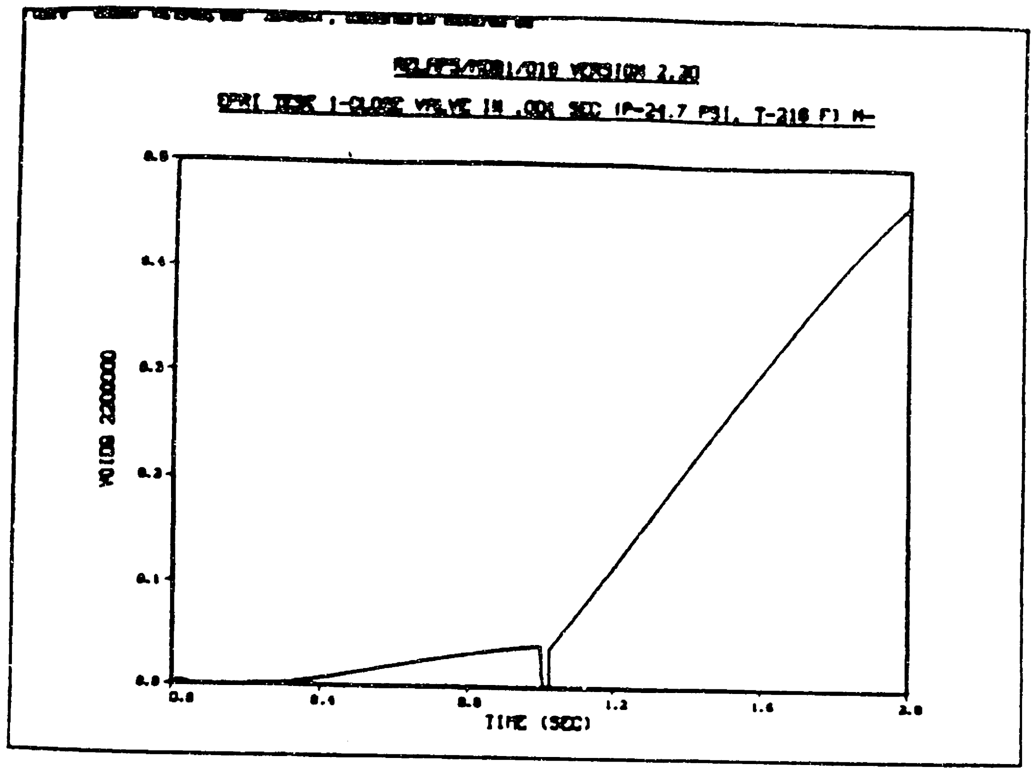

Figure 38. (RELAP5 Exit Void vs. Time Plot, 216 F, 40 ft Pipe)

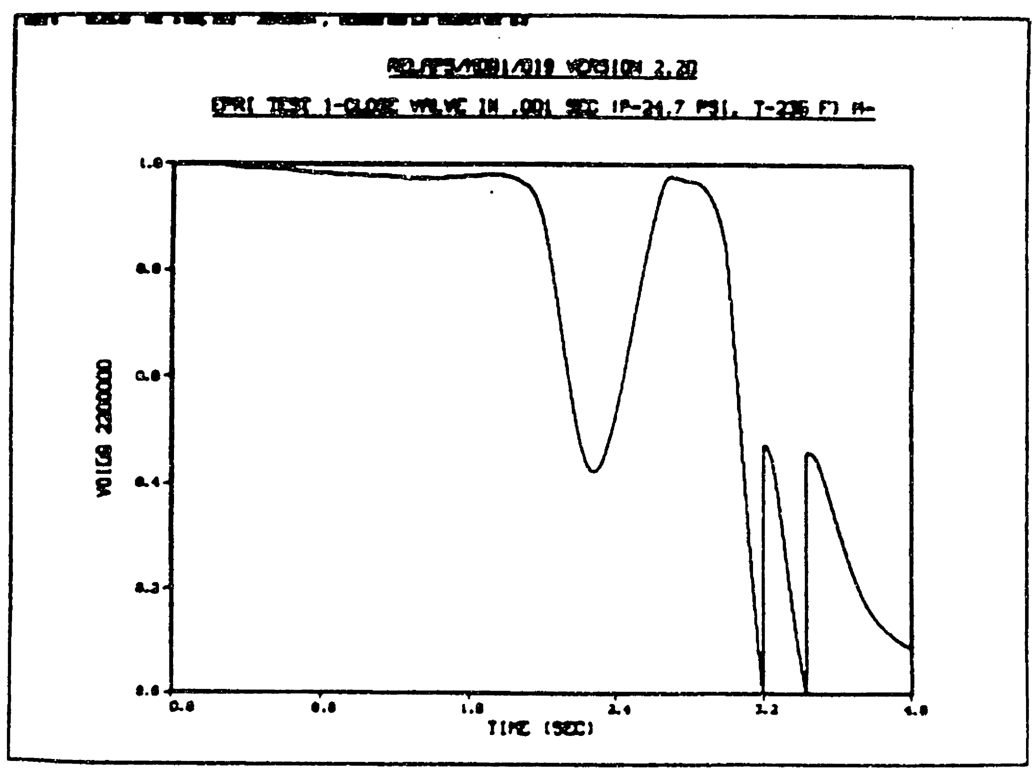

Figure 39. (RELAP5 Exit Void vs. Time Plot, 235 F, 40 ft Pipe)

void fraction were also made only for the 24 foot pipe setup.

Conclusions

The maximum pressure surge due to rapid valve closure in one component, flashing two-phase flow was experimentally investigated. The fluid used in the these experiements was near saturated water. A hand calculation method for predicting the maximum pressure surge was also developed. The experiments were also simulated on RELAP5 computer code. The following conclusions could be drawn from this work.

1. Of primary concern in these investigations was the effect of fluid enthalpy on the maximum pressure surge observed for flashing or near flashing flow. Although there was a great deal of scatter in the data there appears to have been a definite trend that the higher the fluid enthalpy, the lower the maximum observed pressure surge would be in the almost flashing and flashing flow. However, the pressure surges observed did not exceed the expected pressure surges for cold water flow. The highest pressure observed was for a cold water run with a fluid temperature of 70 F. However, some of the pressure surges observed for the flashing or near flashing flows were of higher magnitude than the pressure surges observed for some of the cold water runs.

2. The effect of the length of the pipe conveying the flow on maximum pressure surge is uncertain. The experimentally observed pressure surges

for the 40 foot pipe runs were comparable to, but slightly higher than the pressure surges observed for the 24 foot pipe runs. The observed pressure surges for the 8 foot pipe were considerably lower than those observed for both the 40 foot runs and the 24 foot runs. This is unexpected from a physical standpoint and could not be supported by either hand calculations or computer generated predictions. It appears that the closing time of the manually operated ball valve was too long in comparison to the wave travelling time from the valve to the tank and back.

3. In developing a hand calculation model we clustered all the void at the valve. This simple hand calculation method did a very a good job in yielding a slightly conservative prediction of the maximum pressure surge due to rapid valve closure in both single-phase and two-phase flow.

4. RELAP5 a transient thermal analysis computer code was used to simulate some of the experimental runs. RELAP5 predicted conservative values of the pressure surge magnitudes for the experimental runs it simulated. The RELAP5 predictions were all within a factor of two of the experimentally observed values. Compared to the predictions of pressure surge magnitude obtained through the developed hand calculation method the RELAP5 predicted values were more conservative. However, the predicted magnitudes by both methods were in good agreement with each other. Therefore it can be concluded that both RELAP5 and the developed hand calculation method are useful tools in predicting the maximum pressure surges for both single-phase and two-phase flows. However, it

should be pointed out that from experience the chance of error in predicting the pressure surge magnitudes using RELAP5 is probably high. Although RELAP5 is supposedly "engineer oriented", the inexperienced user will probably have some problems using it.

5. Exit void fraction measurements were also made using a void measurement section. The measured exit void fraction was considerably less than the calculated void using both the Homogeneous Equilibrium Method and the Thom Slip Method. This suggests that there is significant departure from thermal equilibrium for the experimental runs.

References

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[2] Grolmes, M. A. and H. K. Fauske, "Comparison of the Propagation Characteristics of Compression and Rarefaction Pressure Pulses in Two-Phase, One Component Bubble Flow," ANS Trans. Vol. 11, No. 2 (1968)

[3] Griffith, Peter, Handbook of Heat Transfer, Section 14 (Two-Phase Flow), McGraw-Hill (1973)

[4] Grolmes, M. A. and H. K. Fauske, "Pressure-Pulse Propagation in Two-Phase One- and Two-Component Mixtures", ANL-7792 (1971)

{5} J. W. Murdock, Marks' Standard Handbook For Mechanical Engineers, (Fluid Mechanics Section), Mcgraw-Hill (1978)

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[7] Crane's Technical Paper No. 410, Crane Co. (1974)

Appendix A: RELAP5 Discussion

Figure A1 is the nodalization diagram of the experimental apparatus as used in the RELAP5 runs. This model was used for the 24 ft pipe run simulations. The pipe is actually 24 ft long and has a 1 in. I.D. The pipe was nodded into 12 volumes, each with a length of 2 ft. This same volume size selection was used in the 40 ft and 8 ft pipe run simulations. Cost considerations were the primary criteria for this volume size selection. All totaled the cost for the RELAP5 runs made was approximately $ 5,600. Approximately $2,100 of this was for the test runs that either did not work or were terminated.

We encountered a considerable of difficulty using RELAP5. Below are a few areas that we experienced some confusion.

1. Physical vs. Computer Model

As can be seen from figure A1, the computer model is somewhat different from the actual experimental apparatus. Initially the model we used had two pipe components. One that was upstream of the valve and one that was downstream of the valve. This model was more like the actual experimental apparatus. However, after several test runs this downstream pipe was eliminated because the runs were prematurely terminated due to what appeared to be excessive column separation in this downstream pipe. Initially we also made the two reservoirs too small and this lead to problems in achieving steady state.

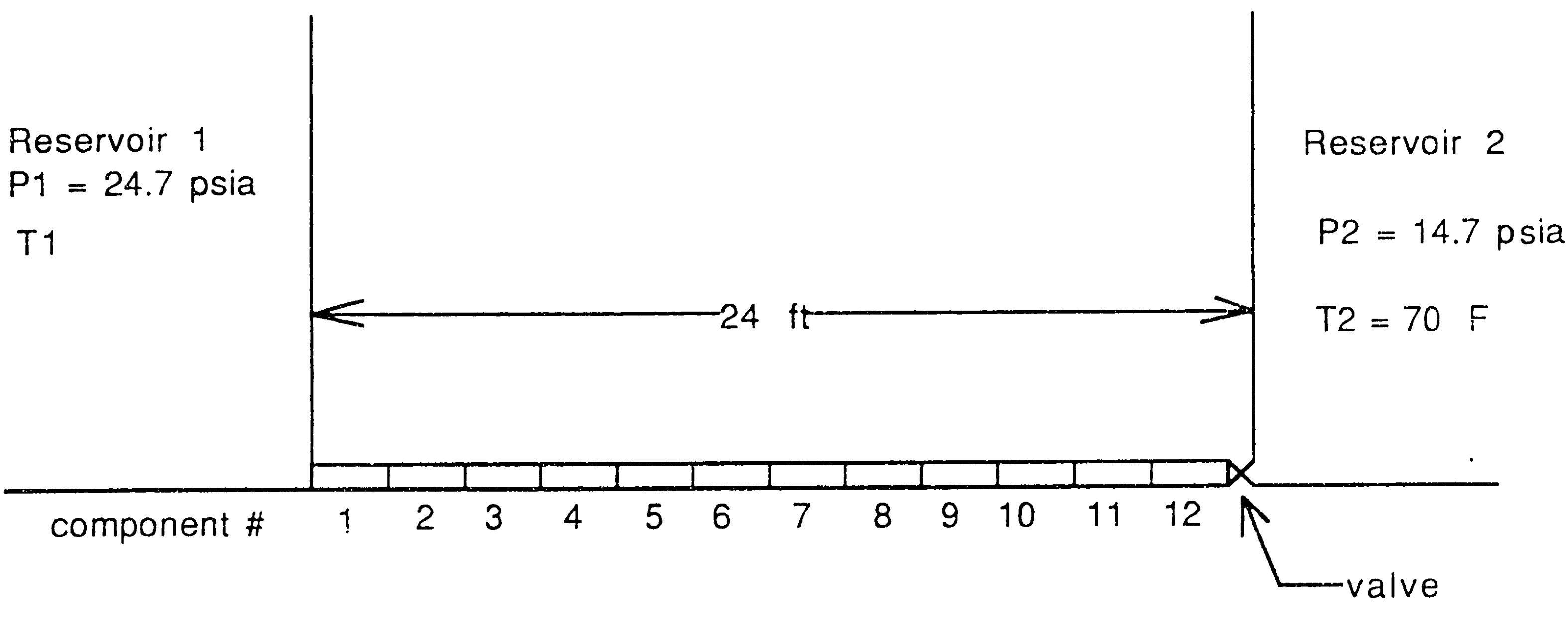

FIGURE A1. RELAP5 MODEL OF EXPERIMENTAL APPARATUS

Appendix A (continued)

2. Volume and Time Step Selection

There is considerable ambiguity in the manual on volume size and time step selection. We feel that it should be better documented on how to optimize these inputs with regards to cost constraints. Actual examples on the effect of volume size and time step selection on the run time cost would be of help.

3. Steady State

We did encounter some problems such as obtaining artificial transients due to the fact that steady state flow had not been reached at the instant of valve closure. Therefore, we had to make longer runs.

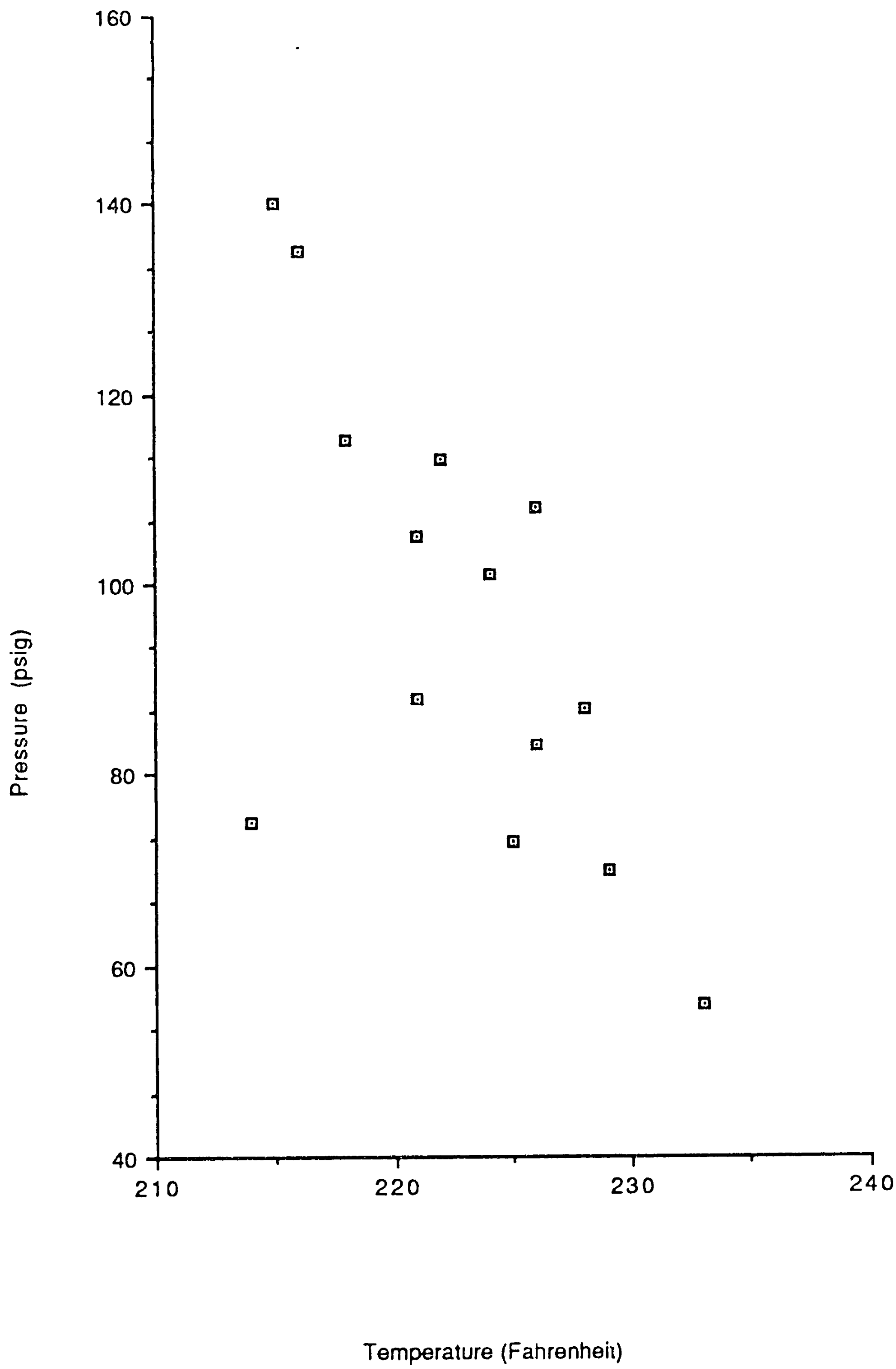

Figure B1. Experimental Data For 8 ft Pipe