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EXPERIMENTAL DETERMINATION OF WATER HAMMER PRESSURE TRANSIENTS DURING VAPOR COLLAPSE

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ABSTRACT

Pressures were measured during water hammer in a steam condensate system. Condensate is formed through condensation of steam upstream of the pump used to pump the condensate. The water hammer mechanism was identified as a vapor collapse in the piping as a pump started, and the mechanism was physically verified by an ultrasonic measurement of water level in one of the pipes. Before the pump started, an overhead pipe was partially full of water. When the pump started, the pipe became full in less than a hundredth of a second. The vapor collapse created audible water hammers and resultant shock waves in the piping. The shock waves were eliminated by controlling the pump start up using a variable frequency drive (VFD) to operate the pump. A slow start prevents the sudden collapse of the vapor space and thus eliminates the water hammer.

NOMENCLATURE

a	wave speed, ft/second
E	elastic modulus, psi
ft	feet
g	gravitational constant, in/second^2 , ft/second^2
gpm	gallons per minute
ID	inside diameter, inch
k	bulk modulus, psi
Ls, Lv	length, ft
NPS	national pipe schedule
OD	outside diameter, inch
psi	pounds per square inch
p, p_{pump}	pressure, psi
R	radius
t	thickness, inch
Vs	velocity, ft / second
VFD	variable frequency drive
ρ	weight density, pounds per cubic foot
SRS	Savannah River Site
ν	Poisson's ratio

INTRODUCTION

This is the first in a set of two papers describing measurements of pipe strains and pressure transients, which result from water hammer. This paper focuses on the pressure transients due to the collapse of a vapor void in a piping system and the actions taken to correct this problem. In particular, water hammer occurred each time a condensate pump was turned on in a condensate system. The water hammer created measured 925 psi pressure transients in a system designed to typically operate at 21 psi. Each time the pump started, an audible water hammer visibly shook piping throughout the system, and the piping deflected ¼ to ¾ of an inch at various locations.

When the water hammer occurred, the pressure gauge on the pump discharge piping typically indicated a pressure spike up to 55 psi. The low frequency response of a typical pressure gauge is incapable of detecting the high frequency pressure spikes associated with the water hammer observed in this system. To measure the pressure spikes, high resolution pressure transducers were required.

Measured pressures are discussed herein with respect to both the initial problem and the solution to minimize the pressure transients. The transients were minimized using a VFD, which slowly brought the pump up to operating speed whenever pump operation was required. To understand the transients, a system description is required.

SYSTEM DESCRIPTION AND TEST SETUP

The piping system of concern to this study consists of steam condensate piping, which discharges from a nuclear waste evaporator at Savannah River Site (SRS). The affected piping connects the flash tank to the pump and the condensate pump to the discharge piping, which exits the evaporator building and crosses several roadways before disposition of the condensate.

Several figures describe the condensate system. A schematic of the overall piping layout is shown in Fig. 1. An aerial view of the evaporator building and outside piping is shown in Fig. 2, and the configuration inside the building is shown on Fig. 3. The pump is actuated by flash tank level transmitters. The pump turns on when the tank level lowers to 4-1/2 inches and turns off when the level decreases to 12-1/2 inches. This actuation causes the pump to start every 10 – 11 minutes and operate for 2 minutes. The figures depict the flow path of the condensate and the various high points in the piping at points A, B, and C, where the pressure transients were initiated. Water condenses in the flash tank; pumps through the condensate pump to points A through E; and then discharges into a large diameter pipe. Typically, the condensate drains at 22 psi, 50 gpm into the nearly empty pipe at atmospheric pressure. To measure the transients, a pressure transducer and data processor were installed about 340 feet from the pump at point E. Calibration of the transducer is discussed in App. A.

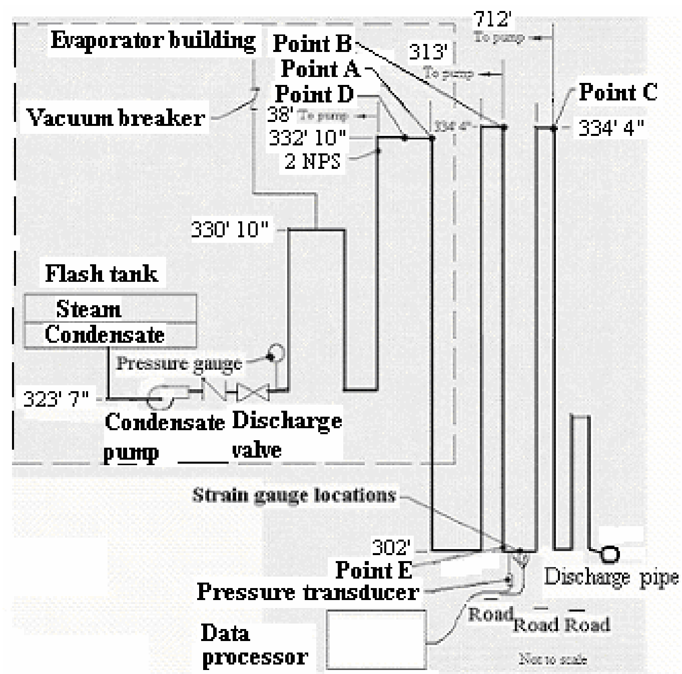

Figure 1: Condensate System Schematic

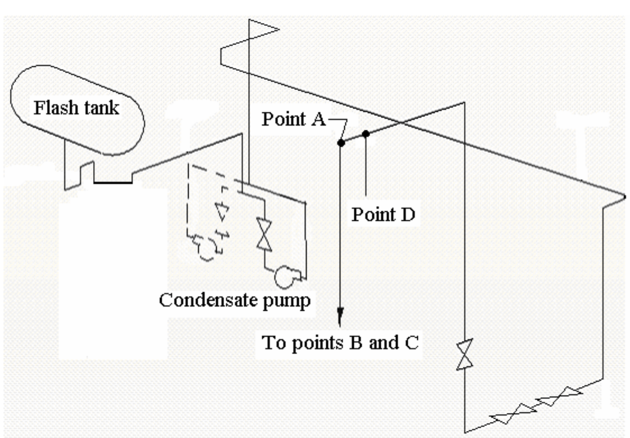

Figure 3: Piping Configuration Inside of the Building

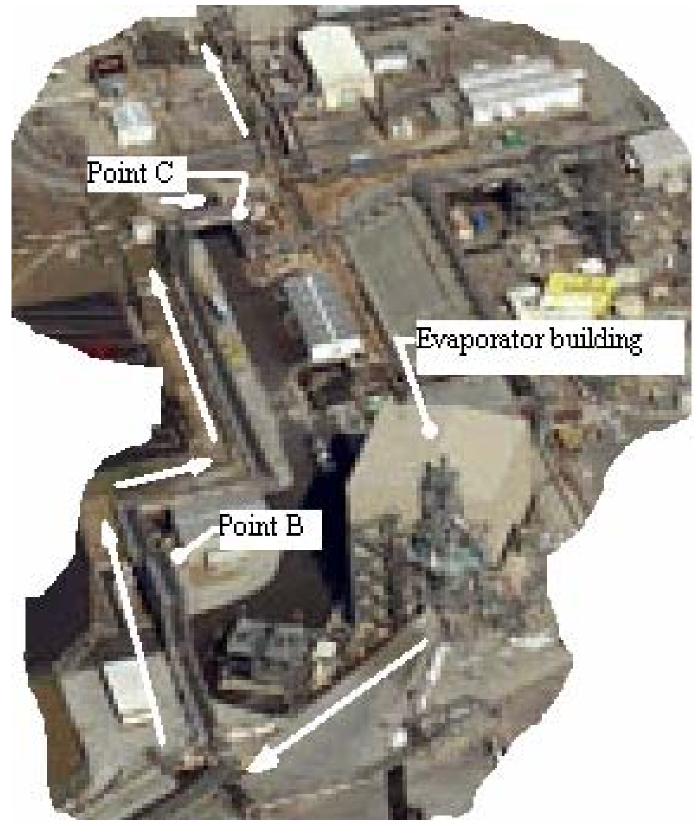

Figure 2: Aerial View of the Condensate System

INITIATIONOF THE PRESSURE TRANSIENT

Each time the pump started, three distinct pressure transients were observed. The transients were initiated at points A, B, and C, and some discussion of the testing performed to reach this conclusion is required.

The transients were observed in process as engineering personnel were stationed inside the evaporator building near the pump and outside the evaporator building with radio communications in place. When the pump started, the first water hammer was heard at point A. Eight seconds later the water hammer at point C was heard. To determine the cause of the hammer, water levels in the horizontal pipe between points A and D were measured.

Measurement of the Vapor Cavity

The fluid transients were caused by vapor cavity collapse, which was verified using ultrasonic transducers. Transducers were attached to the 2 NPS pipe at point D, shown in Figs. 1and 3. Transducer measurements were used to evaluate changes in the water level in the horizontal pipe at point D while the pipe filled at point D when the pump started.

The transducer emitted an ultrasonic signal through the pipe wall and water in the pipe. The time for the wave to be reflected back to the transducer was recorded by an attached processor, which then calculated the wave speeds in the pipe wall and water and determined the water level in the pipe. The water level was thus measured to be ≈1 inch, or half full, before the pump started. When the pump was turned on, the pipe filled nearly instantaneously at point D. This change in water level indicated that the vapor in the pipe collapsed immediately.

A typical processor display of the water level in the pipe is shown in Figs. 4 and 5. Note that the level changes from a half full pipe to a full pipe in one frame, which occurs in about 1/60 of a second. A summary of the data during the complete 10 minute cycle from pump shut down to pump shut down is shown in Fig. 6. Measurements were not taken near points B and C. Although ultrasonic measurements were only used at one location, the vapor collapse mechanism creates the pressure surges at all three locations: points A, B, and C.

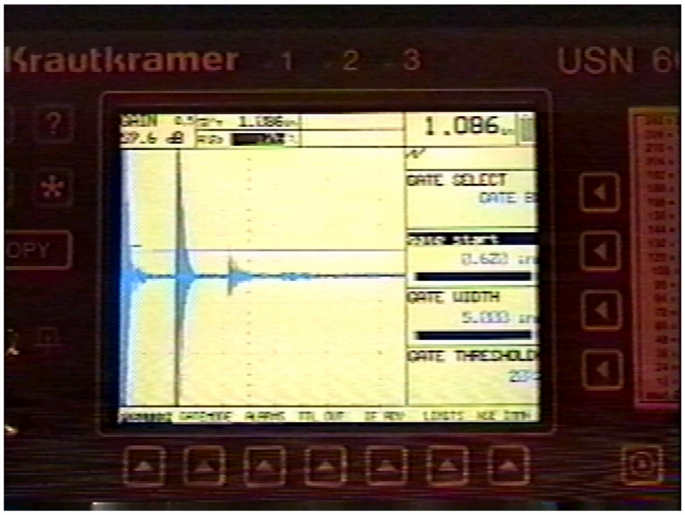

Figure 4: Water Level in the Pipe Prior to Pump Start

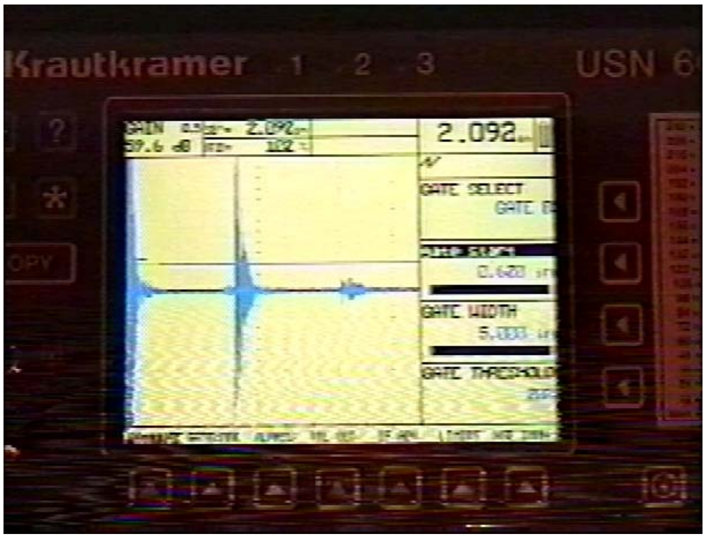

Figure 5: Water Level in the Pipe after Pump Start

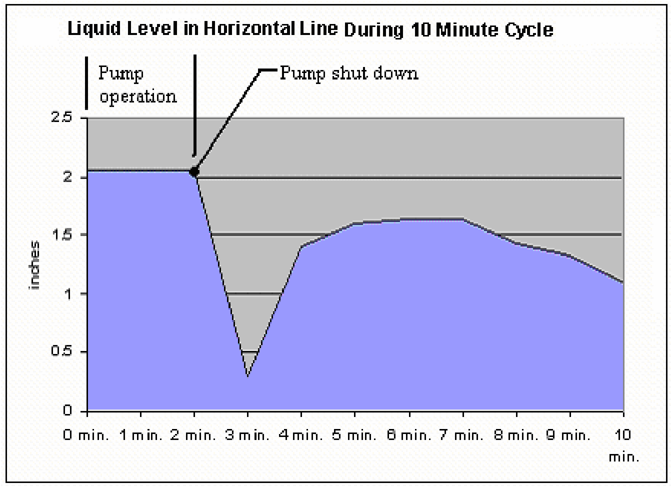

Figure 6: Change of Condensate Level at a High Point in the Pipe

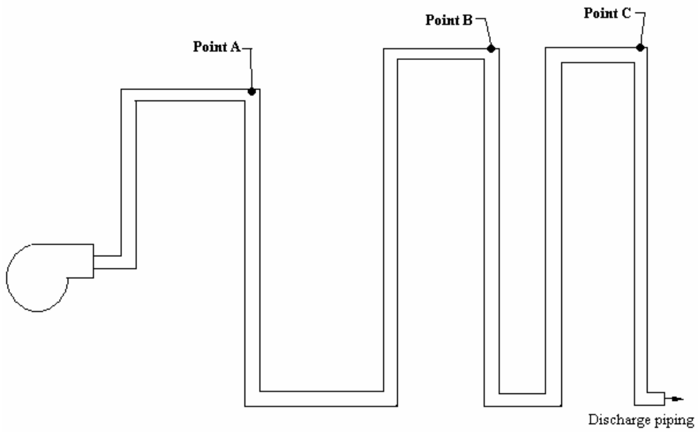

Figure 7: Pipe Schematic

Vapor collapse mechanism

Why do the voids remain in the piping? A description of a typical vapor collapse provides some explanation of this phenomenon (Streeter [1]). To facilitate the discussion Fig. 1 is redrawn as Fig. 7. When the pump stops all of the fluid in the pipe begins to slow down. As it slows down, the pressure in the pipe decreases. If the pressure at point A decreases to the vapor pressure, the water vaporizes and the flow separates. The water to the left of point A continues to slow until it comes momentarily to rest, while the flow to the right of point A continues to flow toward points B and C where the process repeats, and three separate voids are thus created. At each void a vacuum will be created which will force the fluid in the pipe to reverse direction and collapse each of the voids, thus creating the observed water hammers at three separate locations at different times. The hammer at point C occurred 8 seconds after the hammer at point A since the slug flow took longer to reach point C. Numerous high frequency pressure spikes were observed as the voids continued to open and collapse. If the end of the discharge piping were submerged, the voids in the piping would be eliminated. However, any fluid exiting the discharge piping does not return to the system. Consequently, the system volume is less when the transient completes than before the transient begins and the voids remain in the system. More water is expelled from the system as the temperature of the condensate increases, since the vapor pressure is lowered as the temperature increase toward the boiling point. The system usually operates near 180° F. Consequently, larger voids remain following successive water hammers at the higher operating temperature.

The larger voids created even larger pressure surges when the pump restarted as slugs of fluid were propelled through the void into a stationary fluid. That is, the magnitude of the hammer increased as the temperature increased. A fluid transient analysis capable of performing a concise description of the pressure transients associated with this collapse mechanism is unavailable to date. Short duration, high frequency, pressure spikes were observed, which are assumed here to be due to repetitive vapor collapse. Current solution techniques, such as the method of characteristics, assume that high frequency pressure excursions do not occur.

Even so, the approximate magnitude of a pressure surge due to slug flow into a vapor void can be calculated as follows and can be used to obtain a course approximation of the pressure.

Approximated Maximum Pressure Surge

Having proven the existence of the collapse mechanism at point A shown in Fig. 1, the magnitude of the pressure transient was approximated. Water hammer created in this fashion is typically referred to as vapor bubble formation and collapse, and the maximum pressures were calculated using the method of characteristics approach to fluid transients prescribed by Streeter [1].

A slug of fluid was accelerated by the pump toward point A, which then collapsed the vapor void in the pipe. The slug is assumed to be accelerated by a constant pressure supplied by the pump. The maximum pressure, p, thus created is calculated using the parameters provided in Table 1 as follows.

OD = 2.375 inches	Outside diameter
ID = 2.067 inches	Inside diameter
t = 0.154 inches	Pipe wall thickness
R = 1.118 inches	Median pipe radius
E = 29,500,000 psi	Modulus of elasticity for steel
k =320,800 psi	Bulk modulus of water
ν = 0.29	Poisson's ratio
g = 32.174 ft / second^2	Gravitational constant
g = 386.1 in / second^2	Gravitational constant
ρ = 60.13 pounds / ft^3	Density of water at 200° F

Table 1: Material properties and pipe dimensions

The wave speed, a, for a thick walled pipe equals,

$$a := \sqrt{\frac{\frac{k \cdot g}{\rho water}}{1 + \left(\frac{k}{E} \cdot \frac{2 \cdot R}{t}\right) \cdot \left[\frac{t}{R} \cdot (1 + \nu) + \frac{2 \cdot R \cdot \left(1 - \nu^2\right)}{2 \cdot R + t}\right]}} \quad \text{Streeter [1] (1)}$$

Then the wave speed for a 2 NPS pipe is

$$a = 4568 \text{ feet/ second} \tag{2}$$

To find the maximum pressure at point A, the velocity of the slug equals

$$Vs = \sqrt{\frac{2 \cdot Lv \cdot p_{pump}}{\rho \cdot Ls}} \quad \text{Green [2]} \tag{3}$$

where the slug length, Ls, at the moment of vapor collapse equals

$$Ls = 38 + 14 = 52 \text{ ft} \tag{4}$$

Since the pipe is half full, the void length will be assumed to be one half of the horizontal pipe length at point A. This assumption will yield a slug velocity and consequent pressure lower than expected. The void length, Lv, is

$$Lv \approx 14/2 = 7 \text{ ft} \tag{5}$$

A 50 gpm rate was determined from flash tank level changes using Pi, and the pressure was obtained from the pump curve in Fig. 2. The pressure, p_{pump}, supplied by the pump during normal operation at 50 gpm is

$$p_{pump} = 21.2 \text{ psi} \tag{6}$$

Then, the slug velocity at point A equals

$$Vs := \sqrt{2 \cdot \frac{(7) \cdot 21.2 \cdot 144 \cdot 32.174}{60.13 \cdot (38 + 14)}} \tag{7}$$

$$Vs = 20.971 \text{ ft/second} \tag{8}$$

The maximum pressure at point A then equals one half of the velocity obtained from a rapidly closing valve, such that the over-pressurization, p, due to water - water impact at point A is

$$p = \rho \cdot a \cdot \frac{Vs}{2} \tag{9}$$

$$p = 623 \text{ psi} \tag{10}$$

Similar calculations yield a pressure of p = 399 psi at point C, shown in Fig. 1. The void sizes at points B and C required to approximate the pressure surge at those points were not measured. For lack of measured data, an assumption of the void size at points B and C is required to approximate the pressure surge. When the pump shut down, the ultrasonic measurements depicted in Fig. 6 indicated that the level change in the system varied rather slowly. Consequently, the system can be assumed to be near an equilibrium condition, which would require that water maximum levels throughout the system are equal. In this case, the height of the void would simply be the difference in elevation between the three high points. The elevations are shown in Fig. 1 and the void heights between the water level and the top of the pipe at points B and C are thus 1.5 feet. The length of the void was then found from the dimensions in Fig. 1, and the pressure surge was calculated using the equations provided for the pressure at point A. In other words, the basic mechanism creating the water hammer has been identified, but a method to concisely calculate the magnitude of this transient has not been identified. The only valid conclusion with respect to the maximum pressure surge is that the pressure in the evaporator building is ≥ the measured pressures at point C.

PRESSURE SURGE MAGNITUDES

Empirical measurements of the pressure surges were the basis of corrective actions. Measurements of actual transients are discussed prior to an evaluation of the solution to the problem. The pressures were measured using a piezoelectric transducer, data collection was initiated with respect to specified trigger level. That is the data was collected with respect to an arbitrary time equal to zero when a preset pressure level was obtained. A minimum trigger of 10 psi was initially used and raised as required during the test.

Pressure Measurements

The magnitude of the pressure transients increased as the temperature increased from a pipe temperature of 90° F to a typical operating temperature of 208° F at the flash tank and 176° F at the transducer. The variations of pressure surges with respect to temperature are shown in Figs. 8 – 11. The measured pressure transients increased from 50 to 925 psi as the system temperature changed. When the temperature stabilized, the magnitude of the pressure transient stabilized near 925 psi for successive pump starts. In short, the surges in Figs. 8 – 10 only occur when the evaporator starts up as the condensate approaches the normal operating temperature, and the pressure surges shown in Fig. 11 were observed following several pump starts at operating temperature.

Before considering the normal operating pressure displayed in Fig. 11, Figs. 8 – 10 bear discussion. When the system initially started up the pipes were measured to be 90° F, and at this temperature water hammer was not observed when the pump started. As the condensate system began heating up, water hammers were heard in the evaporator building, and were audibly noted to be of less magnitude than the hammers which occurred during normal operation. The pressure surges increased to a constant maximum value of 925 psi after a 4 – 1/2 hour time period following evaporator startup, and the pressure increases can be observed in the figures. Water hammer magnitude in the evaporator building was consistent once the system reached operating temperature.

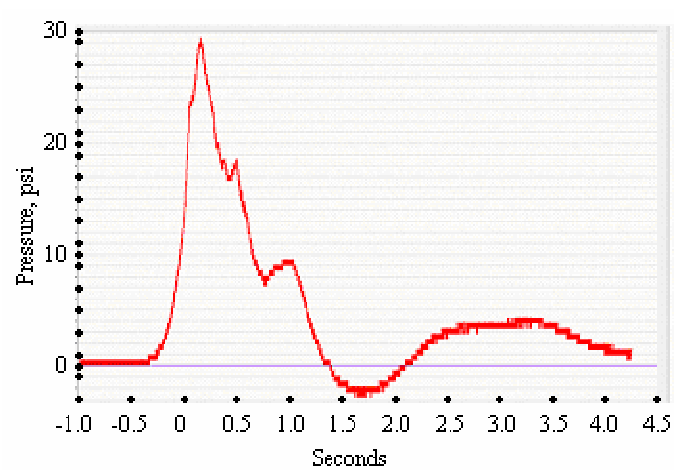

Figure 8: Pressure Data Measured 1 Hour after Startup

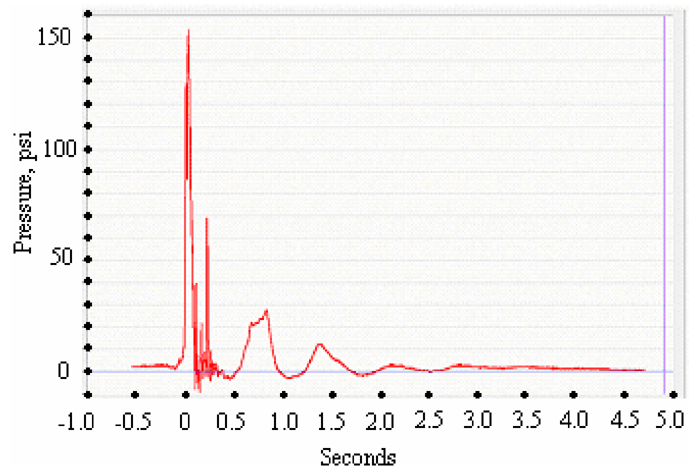

Figure 9: Pressure Data Measured 3 Hours after Startup

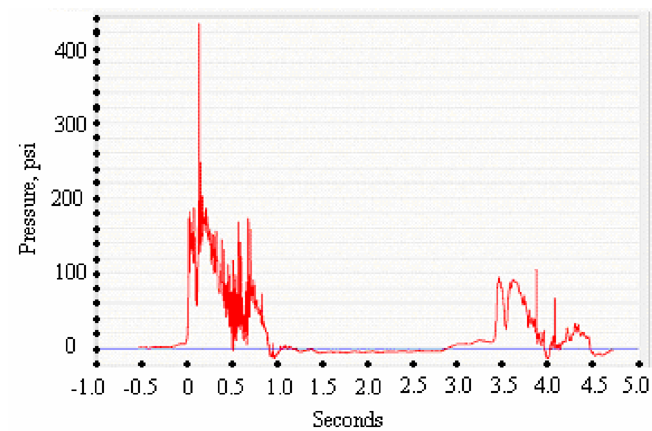

Figure 10: Pressure Data Measured 4 Hours after Startup

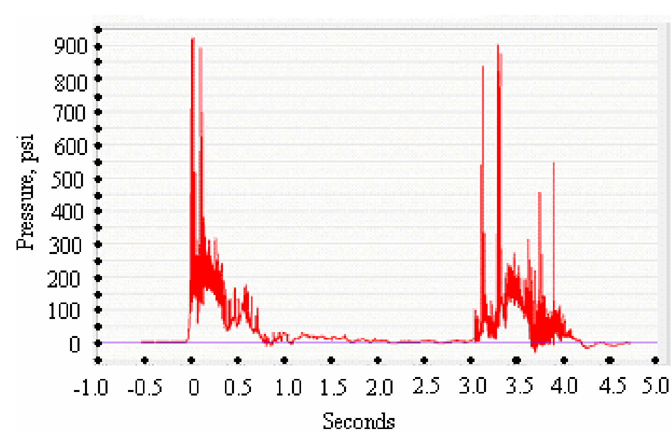

Figure 11: Typical Pressure Data (Measured 4 – 1/2 Hours after Startup)

VFD REQUIREMENTS

The corrective action consisted of an installed VFD, which required a specification of the motor speed with respect to time. The specifications for the installed VFD are shown in Fig. 12 and the pump curve is shown in Fig. 13. These two figures completely define the pump operation on startup.

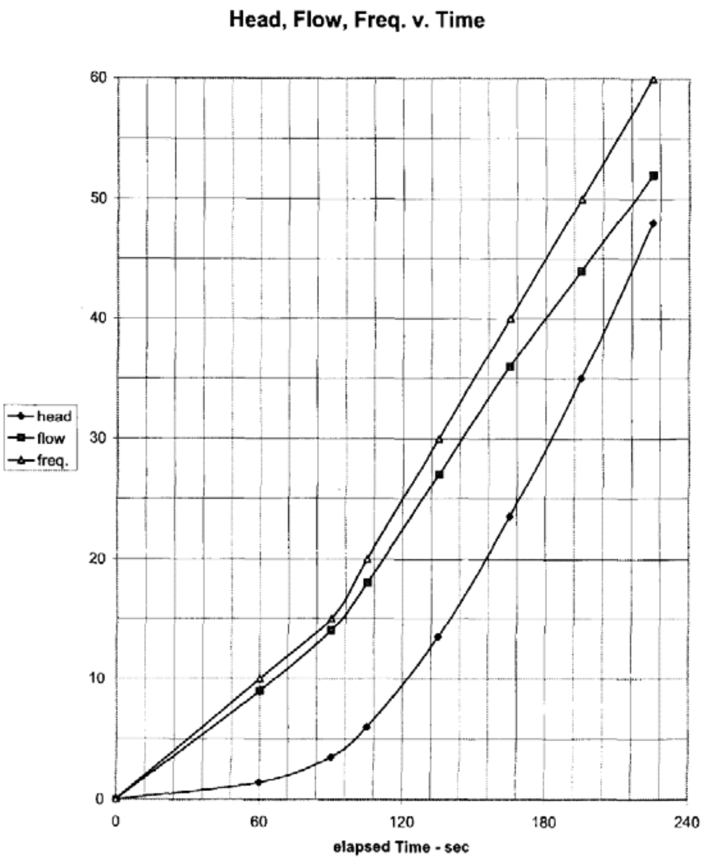

Figure 12: VFD Operating Parameters

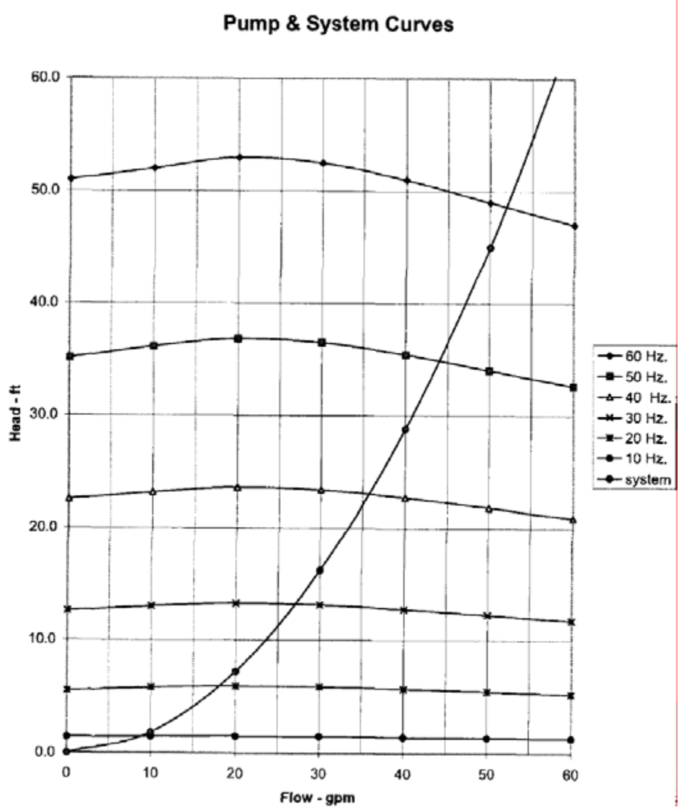

Figure 13: Pump Curve

RESULTS OF THE VFD INSTALLATION

The installed VFD significantly reduced the pressure surges. The pump speed ramps up to the required operating speed in accordance with Fig. 12 any time the pump starts. Following modifications test data was taken as the pump typically operated near 900 rpm, which corresponds to a pressure increase on pump start of 14 psi, as seen in Fig. 13.

Numerous measurements were taken and the system pressure typically had a gradual increase to the expected 14 psi pressure, as shown in Fig. 14. This pressure is expected throughout the system. On one occasion the pressure increased to 80 psi as shown in Fig. 15 and on another the pressure increased to 37 psi (not shown). The occasional higher pressure is attributed to the opening of the vacuum breaker which breaks the siphon between the flash tank and the discharge pipe, shown in Fig. 1.

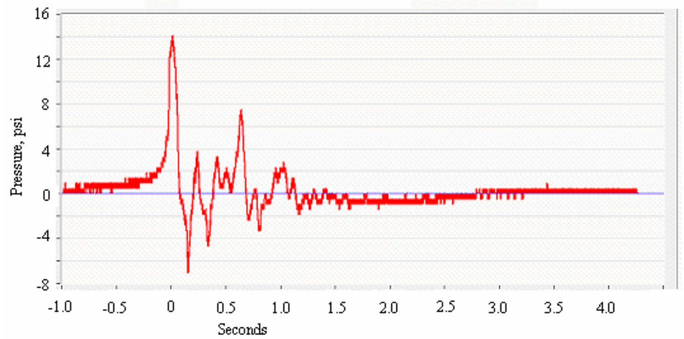

Figure 14: Typical Pressure Surge after VFD Installation

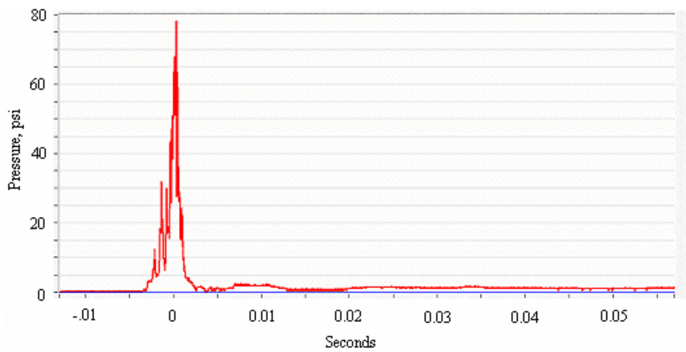

Figure 15: Occasional Pressure Surge after VFD Installation

CONCLUSION

A VFD is an effective means of reducing water hammer effects induced by vapor collapse. Empirical measurements are presently the best means for determining transient pressures, since installed gauges indicated 55 psi, and hand calculations also estimated low pressures. Perhaps the most significant finding of this study is that large, high frequency, pressures are improperly indicated by pressure gauges typically in use. In this case, the measured 925 psi pressure was well in excess of the gauge pressure.

ACKNOWLEDGEMENTS

Numerous SRS personnel contributed to this research. Among then are A. Soosaipillai, M. Wood, G. Reeves, C. Fulghum, K. Carden, E. Lee, and D. Rochelle.

APPENDIX A: TRANSDUCER CALIBRATION

The pressure transducers were calibrated at SRS using the test setup shown in Fig. 16. Basically, a cylinder pressurized to different pressures up to 1000 psi was connected to the transducer through a ball valve, which was quickly opened. The measured pressures are recorded in Fig. 17. There are 2 sets of data. One set of data was recorded before testing was performed. Pressures between 0 and 500 psi were used and the tolerance between input and measured pressures was less than 2 %. Other data recorded after testing was performed at 0 - 1000 psi. This data varied by as much as 8 %, as shown in Fig. 17. The error in the data indicated that all empirical data was less than or equal to the actual data. That is, the measured 925 psi pressure spike was between 925 and 1000 psi.

Figure 16: Calibration Equipment

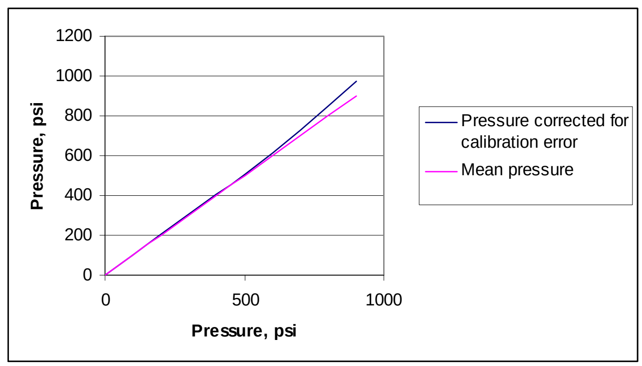

Figure 17: Calibration Tolerances

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[1] Streeter,V. L. , and Wylie, E. B., "Fluid Transients in Systems", 1993, Prentice hall, N. J.

[2] Green, D. J., "Technical Evaluation: 300 Area Steam Line Valve Accident", 1993, Westinghouse, Hanford, Washington.