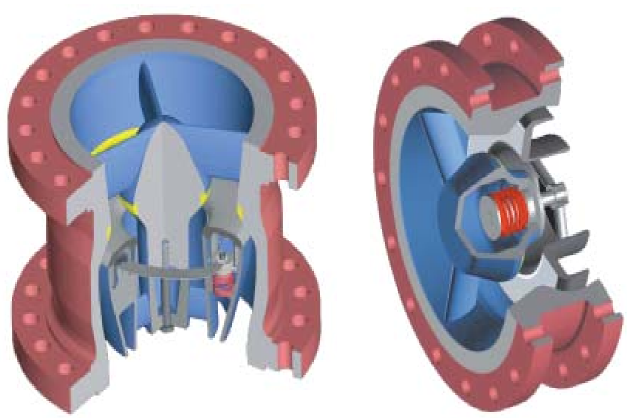
Technical Data Package

for

Enertech NozzleCheck Valve

Solution for

Pressure Surge Problems



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Introduction

The NozzleCheck valve was developed in 1935 to eliminate damaging pressure surge transients cause by the rapid closure of pump discharge check valves in response to pump trips. The first NozzleCheck was installed in a commercial nuclear plant in 1972, as a root-cause solution to water hammer transients that fractured feed system piping. There are now more than 1000 NozzleCheck valves installed in nuclear plants worldwide that have solved the most difficult application challenges. Enertech, as the exclusive worldwide manufacturer of safety related and ASME Section III NozzleChecks, continues to support the nuclear industry with this unique valve technology.

This document collection provides technical information on the dynamic performance of NozzleCheck valves, in addition to case studies that document the increases in system reliability realized by replacing swing check valves with NozzleCheck designs. We look forward to discussing this information with you and offering a cost effective solution to your pressure surge problems. Please contact your Enertech Regional Manager or our main office at 714-528-2301 to discuss your water hammer/pressure surge related problems.

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White Paper

Axial Flow Check Dynamic Response Testing -- Using Test Results to Select the Optimum Valve Design

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Axial Flow Check Valve Dynamic Response Testing – Using Test Results to Select the Optimum Valve Design

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ABSTRACT

A comprehensive set of dynamic response curves is essential to selecting the optimum check valve design for applications susceptible to pressure surge. This paper illustrates the accuracy of these curves by calculating expected performance and comparing this to actual system response measured during transients. By using the dynamic test results and the analytical sizing methodology illustrated in this paper, design engineers can accurately predict the performance of various axial flow check valves before they are purchased and installed. Variables such as disc geometry, bearing design, spring selection, and internal component configuration contribute greatly to the dynamic performance of axial flow check valves. This paper analyses how dynamic performance can be controlled by proper selection of these variables.

INTRODUCTION

Steady State and Dynamic flow loop testing (Ref. 1,2,3,4,5) has been completed on a number of, axial flow check valves designed for use in Nuclear plants. This testing offers the design engineer the ability to evaluate these designs as solutions for applications experiencing damage due to pressure surge. This paper will identify nuclear plant applications susceptible to pressure surge, describe the need for dynamic testing, summarize the test methodology and provide a step-by-step approach to using the curves, including examples of actual applications.

Check valves are utilized in numerous applications throughout Nuclear Power plants to prevent reverse flow of fluids. One of the most common uses for check valves is at the discharge of pumps to prevent reverse flow while a pump is in standby. This ensures that pumps do not rotate backwards and system inventory is not depleted when pumps are placed in standby with the discharge isolation valve left open.

In most cases, a swing check was the original valve design specified for Balance of Plant (BOP) and Safety Related pump discharge applications. During the design phase, consideration of a pump-discharge check valve's dynamic performance characteristic was rarely factored into the design specifications. Swing checks, and to a lesser extent, duo-disk check valves, were chosen for pump discharge applications based mainly on their price, flow capacity, seat leakage capability and conformance to ANSI B31.1 and ASME Section III design criteria. Steady state operating pressures are greatly exceeded during severe pressure spikes resulting from rapid check valve closure during pump trips and gas expansion transients. Design specifications were typically written around steady state performance criteria, not the more limiting transient condition.

Transient surge pressures will evolve in a piping system when fluid velocity is changed rapidly. This can occur by rapid valve closure or opening or vapor pocket collapse. Resultant forces exerted by pressure surge have caused pipe movement damaging supports and anchors, fractured pipe, over-ranged gages and forced pumps/motors out of alignment. High transient pressures have also resulted in rupture of upstream and downstream piping.

All Nuclear Power plants have applications susceptible to pressure spikes, of varying degrees of severity, caused by the rapid closing of check valves. Some plants are fortunate to have system configurations that do not generate pressure surges high enough to cause damage or exceed design pressures. In many cases, pump start-up and shutdown procedures have been modified requiring an "Operator Work Around." Operations personnel are dispatched to the pump to close the manually operated discharge isolation valve on a pump being placed in standby. After the pump is secured, the discharge valve is typically reopened. Another option is to install accumulators at the discharge of pumps to absorb the pressure surge after a pump trip.

There are some sites that have replaced originally supplied check valves with similar or alternate designs in an effort to eliminate problems associated with pressure surge. For the nuclear plants in the process of implementing design changes to eliminate the potential for damage due to pressure surge or to remove the need for an Operator Work Around, having the data needed to accurately calculate the dynamic performance of a replacement check valve design is essential. As an example contrasting the performance of a swing check vs. an axial flow check valve, see Figure 1. Check valve "A" is an axial flow design, valve "B" is a swing check. The resultant pressure surge with an axial flow design is roughly ten times less due to the low mass disc, short stroke and fast closure time characteristic of axial flow check valve designs. How does an engineer determine the maximum pressure surge as a function of check valve performance?

The only way to accurately predict the dynamic performance of a check valve is to conduct testing that determines accurate reverse velocity (V_R) vs. system deceleration (dV/dT) curves. Although a check valve design may look similar, seemingly irrelevant differences can provide vastly different dynamic test results. Swing check valves with lever arms will react differently than standard swing checks. Axial flow check valves that are center guided with multiple bearing surfaces will react differently to rapid flow reversal than ring shaped, floating disc designs.

The design engineer or system engineer tasked with eliminating problems related to pressure surge must have access to dynamic test results in addition to the experience to know how to use them. They must also be familiar with the various check valve designs available and recognize the differences that affect dynamic performance. One important lesson learned in the Nuclear Industry is that valves of similar design may have exceedingly different performance characteristics. As an example, when the operating torque required for symmetric butterfly valves is used to size double or triple offset butterfly valve actuators, the result is an undersized actuator. Although symmetric and high performance butterfly valves both contain very similar components, disc, shaft, bearings, the torque required for seating and unseating is vastly different. With all valve designs, the engineer must be careful not to assume valve performance characteristics based on generic valve types. As illustrated in this paper, seemingly similar axial flow designs will have different dynamic performance characteristics.

SYSTEMS SUSCEPTIBLE TO CHECK VALVE INDUCED PRESSURE SURGE IN NUCLEAR POWER PLANTS

Systems at risk are those with multiple pumps discharging to a common discharge header and those with one or two pumps that discharge against a high head. There are also applications, such as Residual Heat Removal, that have experienced high pressure spikes generated as the result of gas expansion after the restart of pumps following a Loss of Offsite Power (LOOP).

Parallel pump configurations - Service Water, Heater Drain, Cooling Tower Makeup, Screen Wash & Component Cooling Water Systems

These systems are typically configured with multiple pumps in parallel, See Figure 2, with one pump in standby and either one or two pumps running, depending on the cooling load required. A loss of power to one running pump, or an intentional pump trip, can cause a damaging pressure transient. Once the tripped pump begins to coast down the standby pump starts automatically. The pressure between the tripped pump and the associated check valve drops below the header pressure. Provided other pumps continue to feed the common header or a high discharge head exists, flow will quickly reverse and run from the header to the pump discharge closing the check valve. The instant the check valve closes, the fluid momentum is transferred into a pressure wave that travels downstream causing a spike in downstream pressure. This scenario takes place in typically less than one second.

Generally, the worst case configuration is the trip of one pump while the other pumps continue to run discharging to the common header. Observed in some applications is a high transient pressure both upstream and downstream of the check valve. Upstream pressures are caused by column separation and may exceed the peaks measured on the downstream side.

High Head - Main Feed System

High head applications such as Main Feed, Main Feed Booster and Condensate pumps are at risk of severe pressure surge transients if one or all of the running pumps were to trip. One Nuclear Plant designed with two 100% duty Main Feed pumps experienced severe pipe movement causing fractures and serious damage, but no pipe rupture (Ref. 11). When alternating pumps, this site started the shutdown pump and secured the previously running pump after the system became stable. This evolution was done at reduced power levels. When the pump was secured, a large pressure transient was initiated. The damage was due to the delayed closure of the installed swing check valves. When the check valves closed during high reverse velocity a large downstream pressure spike occurred. This was followed by the formation of a vapor cavity on the upstream side, which quickly collapsed causing additional pressure spikes. This incident was one of the precursors to further in-depth studies on the affect of various check valve designs and their ability to mitigate pressure surge.

Vapor Bubble Recoil - RHR System

Residual Heat Removal Systems typically incorporate a vertical, U-tube heat exchanger where non-condensable gasses may accumulate. After a Loss of Offsite Power (LOOP) event, the RHR pumps lose power causing this gas bubble to expand. Once the Emergency Diesel Generators start and load the Essential Bus, the RHR pumps receive a start signal. During the pump start transient, there is a pressure surge due to the fluid momentum that compresses the gas bubble in the heat exchangers which subsequently expands once the pressure drops to steady state levels causing the check valves to slam closed.

In an effort to isolate the cause of high pressure transients associated with a specific RHR application, Enertech instrumented an RHR system as illustrated in Figure 3. The RHR pump was started after instrumenting the loop using instrumentation of a sampling rate of 2,500 to 50,000 samples per second (SPS). This was electronically reduced to either 250 SPS or 5,000 SPS depending on the resolution needed to analyze the specific event. This instrumentation included position indicators (pos.) and accelerometers (accl.) on pump suction, discharge and header check valves and pressure (press) transducers in various locations as indicated in Figure 3. During this transient, caused by the recoil of a vapor pocket trapped in the high point of the RHR heat exchanger, a pressure spike of over 500 psig was recorded in the pump suction line. This was due to column separation and subsequent collapse after the discharge swing check valve slammed closed.

DYNAMIC TESTING

What tools will dynamic testing offer the Design Engineer?

When a check valve is being evaluated for purchase as a replacement for an existing design, a modification package and the issuance of design specifications control the process. Valve data sheets and a request for quotation to various vendors are issued that ideally, illustrate all service, testing and design conditions that the valve must meet. Critical check valve characteristics vary based on the application but generally contain some minimum requirements such as:

- Size and Pressure class
- Pressure Retaining Material Spec's
- Trim Material Specifications
- Minimum Cv
- Vmin (the velocity required to maintain a check valve disc in full open position without oscillation).
- Hydro and Seat Leakage Test Criteria

Rarely, if ever, do nuclear plants request that a check valve manufacturer provide dynamic performance curves. In applications susceptible to water hammer, this information is as critical as minimum Cv, weight, or pressure/temperature limitations. Having access to dynamic response curves will allow the design engineer to accurately calculate the resulting pressure surge magnitude affected by installing the replacement valve. To use these curves will require either:

- The dynamic response curves of both the original and the replacement check valve and the magnitude of pressure surge experienced with the original valve installed, or
- The maximum fluid deceleration through the check valve determined by hand calculations for a simple system or via computer generated hydraulic model for more complex systems. In addition, the engineer must have dynamic response curves for the check valve being evaluated for purchase and installation.

By taking advantage of these available curves, the design engineer will have the data necessary to calculate the new, resultant pressure surge under various transient conditions. This information can be used to:

- evaluate the cost-effectiveness of the modification prior to approval
- provide data necessary to compare other options such as air vessels or other system modifications
- select the most economical check valve design that limits pressure below the level where damage may occur to system components
- improve the accuracy of hydraulic system models

By eliminating the uncertainty of how a new check valve design will mitigate transient pressures, the cost justification process is made much easier. It allows the engineer to determine which valve designs will meet the transient design criteria. Once the acceptable options are selected, regarding dynamic performance, a more detailed comparison can be made using other criteria important in selecting the best option. Cost, delivery, V_{min}, C_V, will likely play an important role in making the optimum selection of check valve design. This design information also allows plant management to assign the appropriate weight/value to a pressure surge reduction project if they know the final result prior to making an investment in a plant modification. These projects will be competing for limited budgets alongside other projects that can also increase plant efficiency and/or safety.

How is check valve dynamic performance modeled in a flow loop?

Valve manufacturers providing check valves to Nuclear plants for critical applications, both safety related and Balance of Plant (BOP) should be responsible for conducting flow tests to determine steady state and dynamic performance of their product. Steady state testing will identify characteristics specific to the valve design such as flow capacity (Cv), the velocity required to fully open the check valve (V_0), V_{min} and the liquid pressure recovery factor (F_L). This information is useful in calculating pressure drops, cavitation, choke points and predicting degradation of the check valve internals. Dynamic testing provides information characterizing a check valve's ability to react to rapidly changing system conditions that generate a rapid deceleration of fluid.

Dynamic test loops are designed to generate variable rates of fluid deceleration through a check valve. This rapid deceleration closely resembles the conditions associated with a pump trip or the recoil of a gas bubble in downstream piping. The test loops are instrumented to record the ability of the specific check valve to react to the change in forces acting on the disc. A typical test will measure time, flow/velocity, valve position and upstream and downstream pressure. Figure 4 illustrates velocity, disc position, upstream pressure, downstream pressure and differential pressure as a function of time. The elapsed time is 1 second.

This data is used to calculate:

- average dv/dt at the start of the transient
- maximum reverse velocity at the instant of initial disc closure

The test is repeated at different decelerations until sufficient data points have been collected to allow a plot to be generated comparing V_R (The reverse velocity at the instant of check valve closure) to dv/dt.

V_{Rmax} = f (dv/dt)

This curve is referred to as the check valve's dynamic performance curve and can be used to determine the maximum reverse velocity based on a given system deceleration. For each one foot per second change of velocity a pressure change of approximately 50-60 psi occurs for metal pipes. Using the Joukowski equation, the maximum reverse velocity can be used to calculate the resulting pressure surge.

$$\Delta p = \pm \rho a v_r$$

Δp = Pressure Surge at the instant of check valve closure

ρ = The density of the fluid

a = The wave speed

v_r = The reverse velocity at the instant of check valve closure

Axial flow and duo-disc check valves, among others, utilize a spring to apply a force to the obturator in the closed direction. With the wide range of potential spring rates available for use in any one check valve design, it is impractical to perform dynamic testing for each potential spring selection. It is also not economically justified to test every size of a given check valve design. Fortunately, a method exists to incorporate these variables, valve size and spring force, into a dimensionless form of a dynamic response curve. These curves allow check valves that are geometrically similar, but of different sizes and spring forces, to be characterized by one performance dimensionless curve. A 32" valve of a specific design will have different dynamic characteristics than a 12" version of the same design. Most likely, little difference will be noticed between a 30", 32" or 36" valve of the same design. A discussion of the theory and derivation of dimensionless performance curves is beyond the scope of this paper.

How were the check valves tested?

Dynamic testing of various axial flow check valves was conducted at the Delft Hydraulics Lab in The Netherlands. The purpose of this testing was to validate both steady state and dynamic performance characteristics of different check valve designs. The steady state check valve characteristics collected during the testing include flow capacity, pressure recovery factor and critical velocity for check valves with different spring configurations. One unique aspect of the steady state testing was the measurement of disc position as a function of flow rate in both the stroke open direction and stroking closed evolution. Although the steady state testing results are important, this paper will focus on the results related to dynamic tests. The dynamic tests were conducted to generate dimensional and non-dimensional performance curves that plot reverse velocity as a function of system deceleration.

Two test loops were used, one for 12" valves and smaller, the other for valves between 12" and 32" NPS. The test loops are designed to generate variable deceleration rates of water in the reverse direction through the installed check valves after a constant flow rate is established in the forward flow direction. This deceleration is effectuated by means of rapidly increasing the pressure downstream of the check valve. For small bore check valves the upstream pressure was rapidly decreased. Either method closely duplicates power plant conditions causing check valve slam during transients. This paper compares only the tests conducted using the large bore test loop on 32" valves.

Large Diameter Check Valve Dynamic Testing

Tests conducted on 32" and 24" axial flow check valves have been conducted using the large test loop as illustrated in Figure 5. This is an open loop with constant upstream pressure maintained by a head tank. The head tank level is maintained constant by an overflow line and eight centrifugal pumps taking a suction from a reservoir. Downstream of the check valve, mounted in the test section, is the High Pressure Tank, followed by a 24" throttle valve used to adjust flow to a point sufficient to fully open the check valve. Flow in the reverse direction is caused by rapidly pressuring the high-pressure tank via a fast acting (<0.5 seconds) air valve that is connected to an Air Reservoir. The rate of deceleration is controlled by varying the pressure in the Air Reservoir (2472 ft^3). During steady state testing, the Air Reservoir is isolated from the water filled High Pressure Tank (212 ft^3) by the fast acting air valve. From an initial steady state flowing condition, the fast-acting air valve is opened, rapidly reversing flow through the check valve. Fluid velocity and pressure are measured as a function of time. The capacity of this test rig is limited to a deceleration of 65 ft/sec^2. This dynamic test measures the following parameters as a function of time:

- flow rate
- disc position using a strain gauge
- upstream pressure (P1)
- downstream pressure (P2)

These parameters are used to calculate

- The fluid velocity gradient dv/dt (ft/sec^2)
- The maximum reverse velocity $V_{R\ max}$ (ft/sec)

The test is repeated for a variety of decelerations. The complete test is typically conducted once with the check valve fitted with relatively weak spring; then repeated using a stronger spring.

Dynamic Test Equipment

- Flow rate was measured with an electromagnetic flowmeter, accuracy +/- 5% of measured value.
- The check valve position was measured using a strain gauge mounted on one of the three radial guide assemblies. On axial flow designs that utilize a center guiding stem, instead of a radial guide assembly, valve position was not measured.
- Dynamic upstream and downstream pressures were measured using piezo-electric pressure transducers, 100 bar range, 40 kHz frequency and charge amplifiers with frequency range of 0-180 kHz, accuracy +/- 1%.

Presentation of Dynamic Test Results

When evaluating the performance of various automobiles, criteria such as 0-60 time, top speed and braking distance are normally provided corresponding to the specific model of car along with any performance enhancing options such as: turbo, larger engine displacement, body style etc. How valuable is performance data associated with only a generic car type such as sports car, sedan or pickup truck instead of the specific model? Would you assume that a Chevrolet Cavalier offers the same performance as the Chevrolet Z06 Corvette? Of course not, any more than you should assume that all axial flow check valves offer the same dynamic performance characteristics. When using dynamic performance curves, the design engineer should ensure they are also provided with the exact configuration of valve tested and how the test was conducted. For each of the axial flow check valves tested, we have provided a standardized method to present the relevant data. Type, Manufacturer/Model, Size, V_o, Orientation, Test Medium, Test Method, Drawing of Valve, Description of Valve and Description of Test Specifics and provided for each test conducted.

Since different axial flow designs have different performance characteristics, we have tested multiple designs of the same 32” size, Model DRV-G, DRV-B and KRV-B. We have also tested 12” and 24” DRV-B’s, see Figure 6. In addition to NozzleCheck valve test results we have included dynamic performance curves for Mokveld 32” Axial Flow designs (Ref. 7) in addition to swing check and Duo-Disc designs (Ref.6) . Although all of the relevant information was not available, the Mokveld check valve curves provide a comparison of different results for valves of the same type. The test results of swing checks and Duo-Discs are offered for use as comparison to axial flow check valve performance characteristics.

When testing spring loaded check valve designs it is critical to identify the specific spring design used during testing. The strength of the spring is one variable, within one valve’s design, that significantly affects dynamic performance. One simple way to differentiate between spring designs is to categorize them by the associated V_{min} or V_0. The stronger the spring force, the higher velocity required to fully open the valve. In addition, the closing times are faster with stronger springs resulting in lower reverse velocities. In the testing conducted on the NozzleChecks at Delft Hydraulics, we identify the specific spring by the minimum flow required to fully open the valve, V_0. It is also important to compare the steady state velocity prior to the start of the dynamic test and compare it to the valve's V_0 value. The velocity must be higher than valve's V_0 to fully open the valve. Running the test at less than the fully open position results in lower reverse velocity values. Note that the 32" DRV-G tests were conducted with the valve slightly less than fully open. As a general description, you will sometimes see various springs identified as weak, medium or strong. This general categorization will vary between different valve manufactures and can introduce large errors in calculating pressure surge magnitudes; the actual V_0 values should be used when available.

Type: NozzleCheck
Manufacturer/Model:Entech/NozzleCheck Model DRV-B
Size: 32" NPS
V_o -Strong Spring- 8.81 fps
V_o -Weak Spring- 6.1 fps
Orientation: Horizontal
Test Medium: 66 ˚F Water
Test Loop: Large Bore Loop at Delft Hydraulics
Dynamic Response Curves and Drawing:

Figure 7

Description of Valve

This test was conducted using the original style of DRV-B. The DRV-B has a single-piece body with a ring style disc that fits in a recessed area of the diffuser. The disc is acted on by a set of helical springs evenly spaced around the circumference of the disc. The DRV-B inlet geometry consists of inlet vanes that straighten the flow stream equalizing uneven velocity gradients. A cone shaped section in the center of the inlet directs the flow into the vanes and gradually away from the center.

The ring shaped disc provides two fluid paths past the inner and outer seats. Relative to a circular disc of the same size valve, typical of swing checks or axial flow designs such as the DRV-Z, the DRV-B disc is smaller and lighter. The disc face remains perpendicular to the inlet flow direction throughout the full stroke. The DRV-B has a stroke length longer than both the DRV-G and KRV-B. There is no stem in this valve which minimizes the force needed to overcome the high friction typical of center-guided designs. With flow in the reverse direction and the valve fully open, the diffuser shields the disc from drag forces tending to close the valve.

Description of Test

A graphical representation of the Delft Hydraulics test results is provided in Figure . This test was conducted twice, once with the strong spring and again with the weak spring. The initial steady-state velocity prior to introduction of the transient was 6.23 ft/sec with the weak spring and 8.86 ft/sec with the strong spring. Both tests began with the valve fully open. The dynamic characteristic with weak spring was measured up to a maximum deceleration of 27.13 ft//sec^2 corresponding to a backflow of 2.36 ft/sec. A maximum deceleration of 54.23 ft/sec^2 was generated with the strong springs installed, resulting in a maximum backflow of 3.12 ft/sec.

Type: Axial Flow
Manufacturer/Model:Entech/NozzleCheck Model KRV-B
Size: 32" NPS
V_o: 6.56 fps
Orientation: Horizontal
Test Medium: 66 ˚F Water
Test Loop: Large Bore Loop at Delft Hydraulics
Dynamic Response Curves and Drawing:

Figure 8

Description of Valve

A ring shaped disc, short face-to-face, and center disc guide characterizes this axial flow design. The disc has two seating surfaces with flow being divided between inner and outer cavities. The ring shaped disc is connected to the concentric shaft via multiple vanes. In the fully open position, a single shaft/bearing interface carries the weight of the disc. The disc face remains perpendicular to inlet flow at all positions from fully closed to fully open applying maximum fluid force to the valve obturator. In the fully open position, the KRV-B disc area is exposed to maximum fluid drag forces when flow reverses. The stroke length is longer than the 32" DRV-G but shorter than the DRV-B.

A single helical spring is used to increase closure speed in addition to providing added seat load in the closed position. The minimum velocity required to fully open the valve is dependent on the spring design and will typically vary from a minimum of 3 ft/sec.

Description of Test

This testing was commissioned to determine both the resistance coefficient (C_V) and dynamic performance characteristic for a 32", ANSI 150 Model KRV-B. The valve was dynamically tested using a spring configuration related to a Vmin of 6.56 ft/sec with an initial velocity of 7.87 ft/sec. Testing was repeated 11 times varying deceleration from 9.3 ft/sec^2 to 56.13 ft/sec^2. The maximum backflow recorded was 2.0 ft/sec. Valve position was not measured.

Type: Axial Flow
Manufacturer/Model:Entech NozzleCheck Model DRV-G
Size: 32"
V_o -Strong Spring-8.92 fps
V_o -Weak Spring- 6.2 fps
Orientation: Horizontal
Test Medium: 66 ˚F Water
Test Loop: Large Bore Loop at Delft Hydraulics
Dynamic Response Curves and Drawing:

Figure 9

Description of Valve

The model DRV-G is similar to the DRV-B. The main differences are the two piece body and larger flow area of the DRV-G, which results in a higher Cv. There are differences in the shape of inlet and outlet flow passages. As in the DRV-B design, downstream of the disc is a diffuser that shields the disc from drag force in the reverse flow direction when fully open. The DRV-G has the shortest stroke length compared to the DRV-B or KRV-B.

Description of Test

This test was conducted twice, once with the strong spring and again with the weak spring. The initial steady-state velocity prior to introduction of the transient was 8.27 ft/sec with the weak spring and 11.81 ft/sec with the strong spring. Both tests began with the valve nearly full open. The dynamic characteristic with weak spring was measured up to a maximum deceleration of 30.74 ft/sec^2 corresponding to a backflow of 2.59 ft/sec. A maximum deceleration of 51.1 ft/sec^2 was generated with the strong springs installed, resulting in a maximum backflow of 2.52 ft/sec.

Type: Axial Flow
Model: Mokveld Circular Disk- TKZ-E or similar.
Size: 32"
V_o - 6.9 fps
Orientation: Horizontal
Test Medium: Unknown
Test Loop: Unknown, Possibly Large Bore Loop at Delft Hydraulics
Figure 10

Description of Valve

This model uses a center guided, circular disc similar to the model DRV-Z design. The valve has a diffuser that shields the disc from drag forces in the reverse flow direction. The flow testing information comes from Koetzier, Kruisbrink & Lavooij (Ref. 7), which does not provide specific information regarding the exact configuration of valve tested. We have included these curves to illustrate the wide range of results obtainable when comparing axial flow check valves of the same size, but different types.

Type: Swing Check and Duo-Disc
Model: Unknown
Size: Various
V_o -Various
Orientation: Horizontal
Test Medium: Water
Test Loop: Analytically determined based on flow test data.
Figures 11 & 12

Dynamic response curves, Figures 11 and 12, are provided using data from Ellis & Mualla (Ref. 8) and are based on numerical modeling used to extend the value of actual test data. The values of v_r as a function of dv/dt are optimistic and should provide the user with pressure surge magnitudes lower than actual. Specific information regarding the configuration of the tested swing check or duo-disc is not provided.

DISCUSSION OF TEST RESULTS

Test results are different for each type of axial flow check valve tested. An interesting point is the difference between the performance of the DRV-B compared to the DRV-G. Even though their designs seem nearly identical except for the center flange, the resulting reverse velocity is markedly less with the DRV-G resulting in a lower pressure surge . This is likely due to a number of factors related the test method and the valve design. The DRV-G dynamic test was conducted with the valve slightly less than fully open which, provides added drag force on the disc in the reverse direction in addition to a shorter stroke length. The DRV-G has a larger flow and a shorter stroke length than the DRV-B. This provides for faster closure. Both the DRV-G and DRV-B have similar disc geometry and weight.

The most interesting result of the testing is the performance of the model KRV-B. It has the longest stroke and heaviest disc but closes quicker under reverse flow than the DRV-B or DRV-G. The KRV-B also has to overcome the higher friction forces attributed to a heavier disc sliding along a center bushing. The KRV-B spring is slightly stronger than the weak springs used for the DRV-B and DRV-G but not enough to explain the difference in performance. The reason for the exemplary dynamic performance is likely due to the disc design. Since the disc protrudes from the back end of the valve and is fully exposed to the force of the reverse flow the drag force is much higher under reverse flow conditions. As a result the disc closes faster, and results in lower pressure surges than the other models.

Comparing the NozzleCheck flow testing to test results conveyed from (Ref. 7) illustrate a drastic difference in performance. The main difference between the DRV-B, DRV-G and KRV-B compared to the Mokveld design is the disc shape. The Mokveld design uses a disc and diffuser shape similar to the NozzleCheck Model DRV-Z. A circular, center-guided disc is attached to the stem and recessed in a diffuser as illustrated in Figure 10. The weight of this disc is higher than that of a ring style of the same size, there is also a large bearing surface that adds a frictional force component that is much higher than the DRV-B or DRV-G designs. During a reverse flow condition, flow is diverted away from the backside of the disc by the diffuser, similar to the DRV-B and DRV-G. There is effectively no drag force on the disc until it leaves the fully open position.

HOW TO CALCULATE PRESSURE SURGE USING DYNAMIC RESPOSNE CURVES

Nomenclature:

Δp = Transient pressure surge (psig)
a = Wave propagation speed (ft/s)
ρ = Fluid density (lb/ft^3)
p_1 = Operating line pressure (psig)
$p_{allowable}$ = Maximum allowable pressure in the line (psig)
v_r = Reverse velocity through the header at the instant of disc closure (ft/s)
g = Gravity (ft/s^2)
$p_{transient}$ = Downstream line pressure at valve closure (psig)
K = Bulk modulus (lbf/ft^2)
D = Pipe diameter (ft)
E = Young's modulus of elasticity (lbf/ft^2)
e = Pipe wall thickness (ft)
ϕ = Restraint factor (for simplicity, this parameter is assumed to be equal to 1)

When selecting a check valve for systems susceptible to water hammer the resulting pressure surge due to rapid valve closure should be predicted. Once predicted, it is combined with the line pressure to determine the maximum pressure that will occur during a transient. This resultant maximum pressure should be less that the maximum allowable system pressure to ensure safe operation.

There are two parameters that are required to determine the maximum transient pressure:

a) The deceleration of the flow dv/dt
b) The line pressure p_1

Deceleration, related to check valve dynamic performance, is typically defined as rate of change in velocity with time of the fluid at the check valve outlet. This is measured or calculate from the time velocity begins to decay after a pump trip until the initial check valve closure. This deceleration is nearly constant over the entire period. We calculated an average dv/dt value for each of the testing in (Ref 1,2,3,4.)

In simple systems, deceleration can be readily determined. For complex systems, a computer analysis is needed to determine the fluid deceleration.

There are two methods that can be used for check valve selection or evaluation. Method 1 is used if transient deceleration is known. Method two is used in lieu of having deceleration but requires the engineer to know maximum transient pressure as well as access to dynamic performance curves for the installed valve. The first would be used to select a valve based on the known parameters of the system prior to installing the valve.

METHOD 1

The resultant pressure surge due to valve closure can be calculated using the Joukowski Formula:

$$\Delta p = \pm \rho a v_r \qquad (1)$$

However, gravity and a unit conversion must be incorporated into the equation to make the units work. For application, the equation becomes:

$$\Delta p = \frac{\pm \rho a v_r}{144g} \qquad (2)$$

Where:

Δp = Pressure change, surge (psig)
a = Wave propagation speed (ft/s)
ρ = Fluid density (lb/ft^3)
p_1 = Operating line pressure (psig)
$p_{allowable}$ = Maximum allowable pressure in the line (psig)
v_r = Reverse velocity of the system (ft/s)
g = Gravity (ft/s^2)
$p_{transient}$ = Downstream line pressure at valve closure (psig)

The following steps can be used to assist in selecting a check valve for a particular system.

1) Determine either analytically or through computer simulation, the deceleration dv/dt, through the subject check valve as a result of a system transient causing the closure of the check valve. This can be calculated assuming no check valve in the line.
2) Determine what the wave propagation speed and fluid density are for the systems particular media, a and p_f.

   To assist with determining wave propagation speed, a, refer to the following basic formula for thin walled pipes:

$$a = \left(\rho \left(\frac{1}{K} + \frac{D}{Ee}\phi \right) \right)^{-1/2} \qquad (3)$$

   Refer to Figure 13 for reference:

3) Establish the maximum allowable pressure surge for the system based on the piping and component design limitations. Maximum allowable line pressure. $(p_1 + \Delta p) = p_{allowable}$
4) Calculate, using the Joukowski Formula, the maximum allowable reverse velocity of the system (v_r).

$$v_r = \frac{\Delta p}{a\rho}$$

5) Evaluate check valve performance curves, such as those provided at the end of this report, to determine which meet the systems reverse velocity requirements. Locating the v_r on the y-axis and tracing a horizontal line does this. Then, locate the dv/dt value and trace a vertical line. Any check valve performance curves that intercept the vertical line between the x-axis and the horizontal v_r line will meet the requirements.

EXAMPLE 1

In designing a new system, consider there are three pumps operating in parallel, discharging into a common header. A check valve is required immediately after each pump discharge. Assume an ANSI 150# 12" valve is required. The media being pumped is ambient temperature demineralized water through stainless steel piping. The operating pressure is approximately 100 psig, with maximum allowable system pressure of 200 psig. The calculated deceleration of the system is 40 ft/s^2 during a single pump trip with the remaining two running. The System Engineer is in the process of evaluating the system for possible check valve options. Refer to Figure 13 for the wave propagation value.

Known:

a = 4150 ft/s
ρ = 62.4 lb/ft^3
g = 32.2 ft/s^2
Δp = 100 psig

(1) Determine maximum allowable v_r using the Joukowski Formula:

$$v_r = \frac{(100)(144)(32.2)}{(62.4)(4150)}$$

$$v_r = 1.79\, ft/s$$

(2) Evaluate which check valves would meet this reverse velocity requirement based on a deceleration of 40 ft/s^2 during a pump trip. Compare the available dynamic performance curves of potential replacement check valves. Assume that the only check valve models evaluated as potential replacements are 12" DRV-B axial flow valves, standard swing checks and duo-disc valves.

a) Refer to Figure 14 in the reference section of this report. At 40 ft/s^2 the 12" DRV-B has a reverse velocity of approximately 0.9 ft/s with a weak spring and 0.24 ft/s with a strong spring. Both will meet the reverse velocity requirement of this system.

b) Refer to Figure 14 in the reference section of this report. At 40 ft/s^2, the 12" Swing Check Valve has a reverse velocity of approximately 8.8 ft/s. This valve will not meet the reverse velocity requirement.

c) Refer to Figure 14 in the Reference section of his report. At 40 ft/s^2, the 12" Duo-Check Valve has a reverse velocity of approximately 5.2 ft/s. This valve will not meet the reverse velocity requirement.

(3) Based on the system specifications, the calculations performed and the graphical evaluations, the DRV-B Check Valve is the only design suitable for this application.

METHOD 2

The second method is used when the deceleration of the system associated with a particular transient is not known. The dynamic performance curve of the installed valve and the maximum downstream pressure level during check valve closure must be known. This method allows the user to estimate the system deceleration based on how the installed check valve reacts to rapidly changing conditions compared to results from past dynamic flow testing. Although not as accurate as method 1, this will provide the engineer with a means to evaluate various check valve designs without having to build a hydraulic model of a complicated system. The following steps would be used.

(1) Once again, use the Joukowski Formula to calculate the reverse velocity associated with the installed valve at the subject transient.

(2) Using the performance curve for the installed check valve, the system's dv/dt associated with a specific check valve closure time can be determined. Draw a horizontal line from the v_r value on the y-axis to the point where it intersects the curve. Then, from the intersecting point on the curve, draw a vertical line to the x-axis. This will determine the dv/dt of the system.

(3) Continue with Step 2 of Method 1 described above.

EXAMPLE 2

A pump discharge, swing check valve in the Low Pressure Safety Injection (LPSI) system (Ref. 10) is causing damage to system components when the pump is tripped. After pump trip, resulting pressure surge is exceeding the design limitation of system components upstream and downstream of the check valve. The Design Engineer is faced with the task of identifying a replacement valve that would prevent these surges from occurring. The existing Swing Check Valve has a diameter of 12".

Based on measurements taken, the pressure surge experienced when the pump trips is 174 psig. The maximum allowable pressure surge based on system design is 100 psig

Known:
a = 4150 ft/s
ρ = 62.4 lb/ft^3
g = 32.2 ft/s^2
Δp = 174 psig
Desired Δp = 100 psi
D_i = 12"

(1) First determine what the reverse velocity is for the existing valve. Use the Joukowski Formula:

$$v_r = \frac{(174)(144)(32.2)}{(62.4)(4150)}$$
$$v_r = 3.12\,ft/s$$

(2) The next step is to determine what the reverse velocity should be to achieve the desired pressure surge. Again, use the Joukowski Formula:

$$v_r = \frac{(20)(144)(32.2)}{(62.4)(4150)}$$
$$v_r = 0.36\,ft/s$$

(3) Now that the required v_r is known, the next step is to review the performance curve for the installed check valve and determine what is the system's dv/dt .

(4) Refer again to Figure 14 provided in the Reference section for the performance curve for a Swing Check Valve. By using $v_r = 3.12\,ft/s$ and the curve, it can be determined that the system's dv/dt is approximately 21 ft/s^2.

(5) The next step is to determine which check valve will produce the desired reverse velocity of $v_r = 0.36\,ft/s$ at the system's

dv/dt of 21 ft/s^2. Knowing the dv/dt of the system, and the size of the desired valve, it is only a matter of reviewing the different check valve curves.

(6) Refer again to Figure 14 for the performance curve for the Duo-Check valve. For a dv/dt of 21 ft/s^2, a 12" Duo-Check Valve will produce a reverse velocity of 2.5 ft/s.

(7) Refer again to Figure 14 provided in the Reference section for the performance curve for the DRV-B Check Valve. For a dv/dt of 21 ft/s^2, a 12" DRV-B Check Valve will produce a reverse velocity of 0.32 ft/s with a weak spring and 0.26 ft/s with a strong spring.

(8) Based on this evaluation, a 12" DRV-B Check Valve with a weak or strong spring would reduce the pressure surge to acceptable levels.

This extensive testing completed on the DRV-B, KRV-B and DRV-G axial flow designs should offer the Design Engineer a valuable tool when exploring methods to mitigate the adverse affects of pressure surge. It is apparent that dynamic performance curves as stand-alone documents, without a detailed description of both the valve and test methodology, tell only half the story. Within the family of axial flow check valves there are an assortment of designs available that offer various advantages to the end user. With regard to dynamic performance, each design reacts differently to rapid deceleration of fluid and produces pressure waves of varying magnitudes. Hopefully the information within this paper can be used as a tool to select the most economical design to mitigate pressure surge and improve the accuracy of system hydraulic analysis.

CONCLUSION

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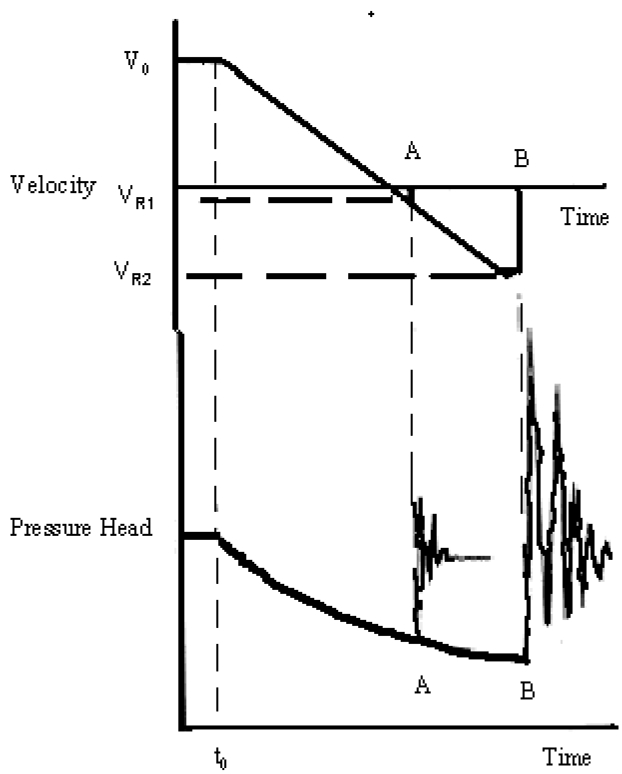


Velocity and Pressure History after a pump trip, downstream of the discharge check valve. The "A" check valve closes faster, with a lower V_{R1}, and much smaller pressure rise. The "B" check valve closer slower, with a higher V_{R2}, resulting in a higher pressure rise.

Figure 1.

Typical Parallel Pump Cooling System

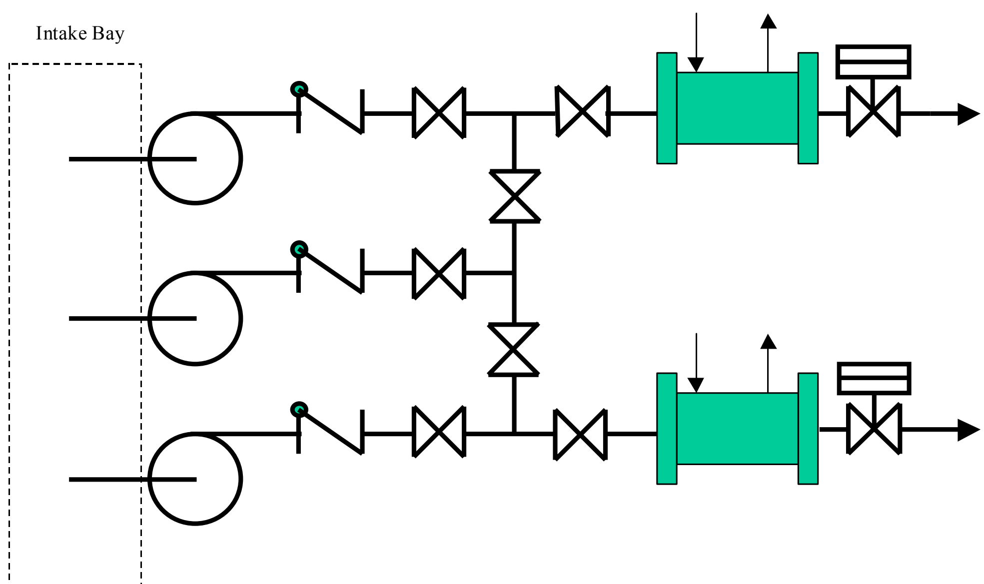


Figure 2.

RHR Waterhammer Test Instumentaion

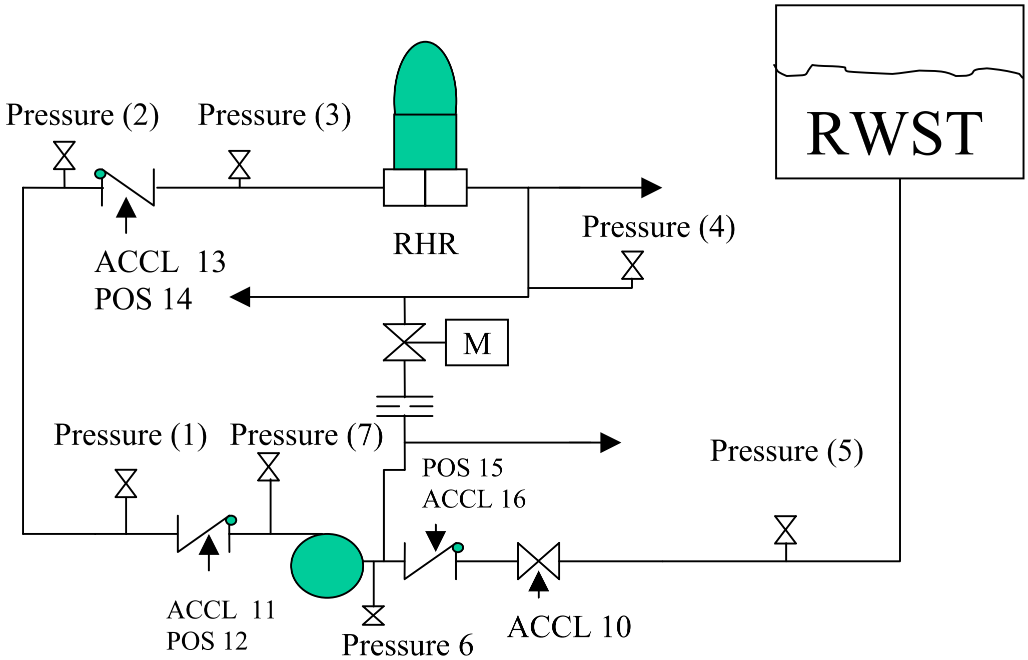
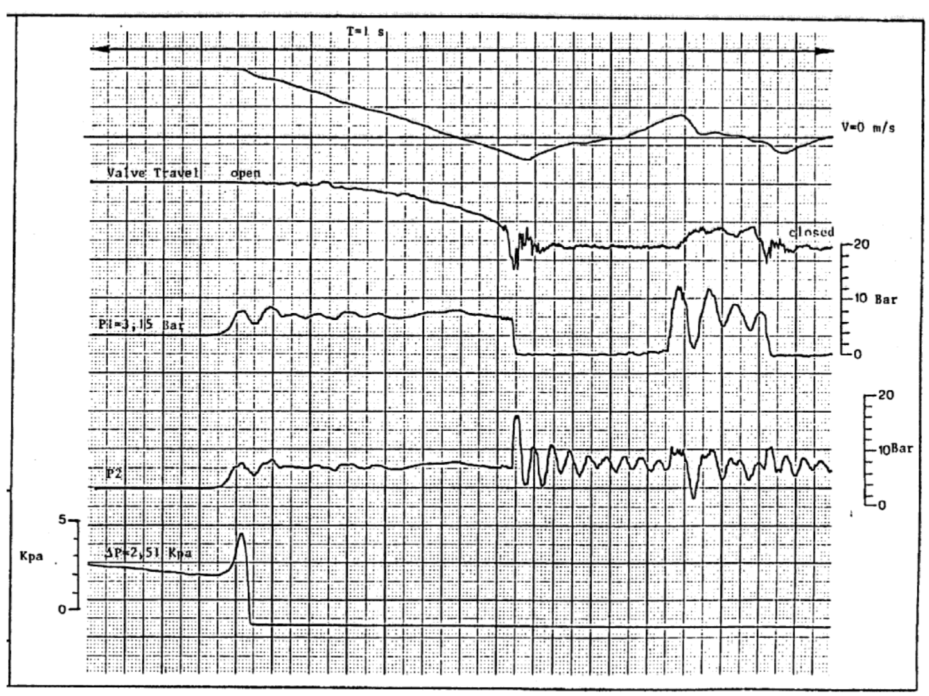


Figure 3.



Recordings of Dynamic Test Data for Large Bore Axial Flow Check Valves

Figure 4.

Delft Large Bore Test Loop

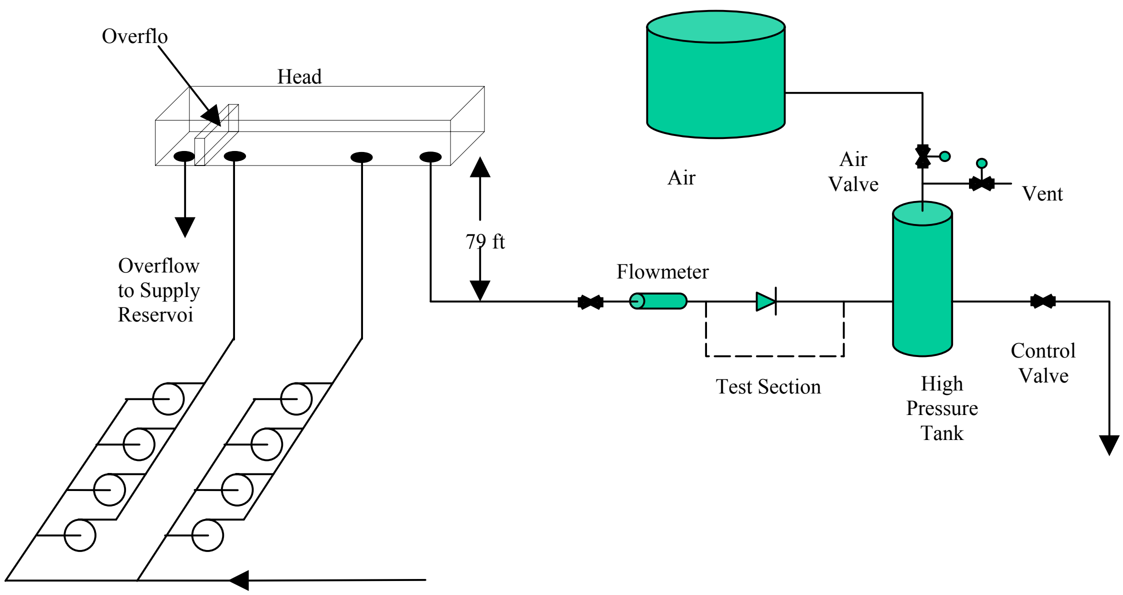


Figure 5.

DRV-B Check Valve Performance Graphs

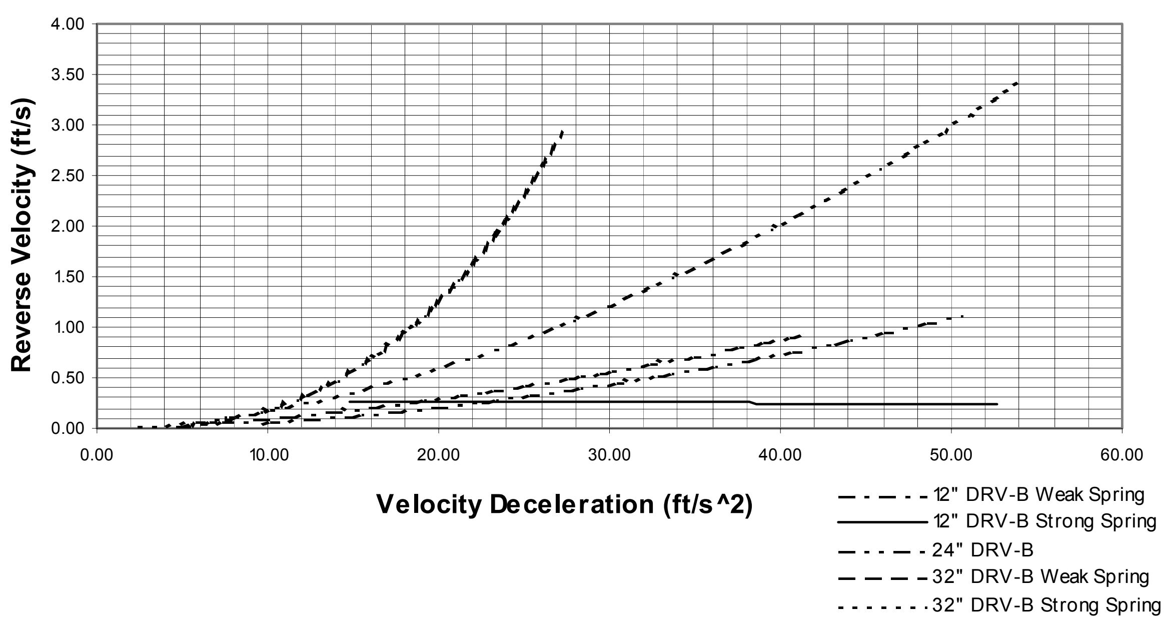
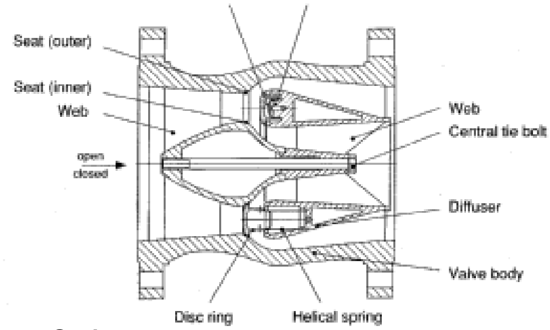


Figure 6.

Type: NozzleCheck
Manufacturer/Model: Entech/NozzleCheck Model DRV-B
Size: 32” NPS
V_o -Strong Spring- 8.81 fps
V_o -Weak Spring- 6.1 fps
Orientation: Horizontal
Test Medium: 66 °F Water
Test Loop: Large Bore Loop at Delft Hydraulics



DRV-B Performance Graph for Weak and Strong Spring

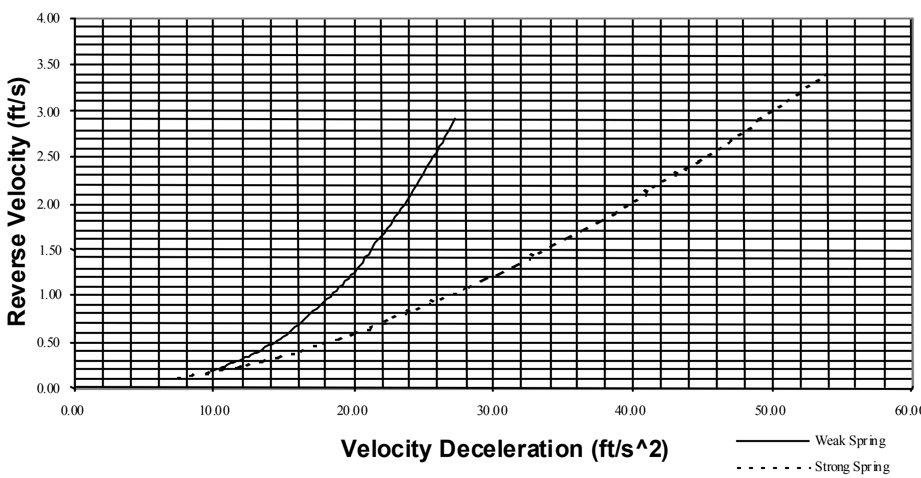
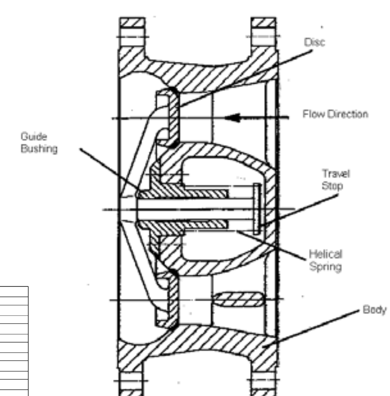


Figure 7.

Type: Axial Flow
Manufacturer/Model: Entech/NozzleCheck Model KRV-B
Size: 32” NPS
V_o: 6.56 fps
Orientation: Horizontal
Test Medium: 66 °F Water
Test Loop: Large Bore Loop at Delft Hydraulics



32" KRV-B Performance Graph

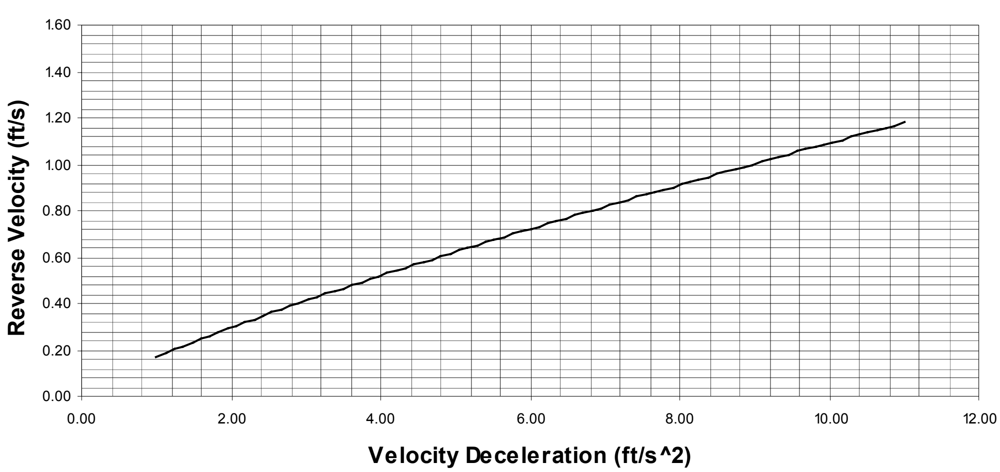
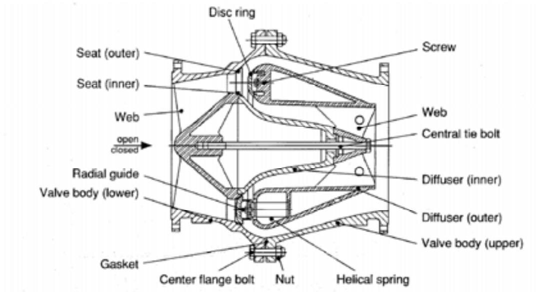


Figure 8.

Type: Axial Flow
Manufacturer/Model: Entech NozzleCheck Model DRV-G
Size: 32" NPS
V_o **-**Strong Spring-8.92 fps
V_o **-**Weak Spring- 6.2 fps
Orientation: Horizontal
Test Medium: 66 °F Water
Test Loop: Large Bore Loop at Delft Hydraulics



32" DRV-G Performance Graph for Weak and Strong Springs

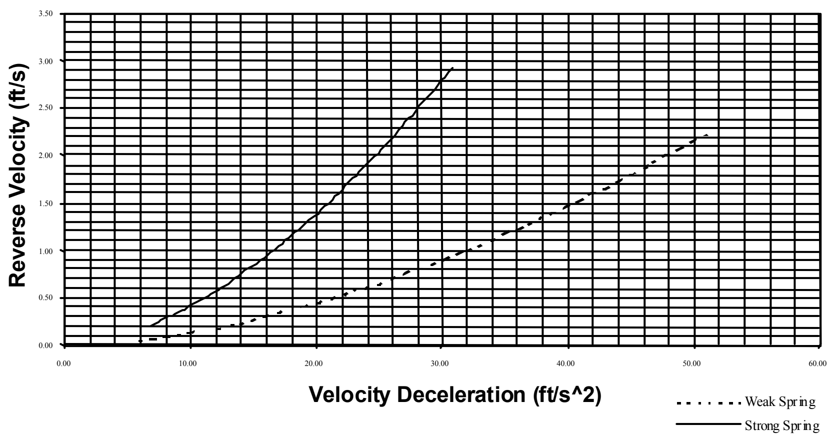
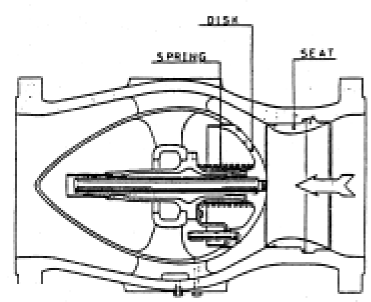


Figure 9.



Type: Axial Flow
Model: Mokveld Circular Disk- TKZ-E or similar.
Size: 32" NPS
V_o - 6.9 fps
Orientation: Horizontal
Test Medium: Unknown
Test Loop: Unknown, Possibly Large Bore Loop at Delft Hydraulics

32" Mokveld Performance Graph

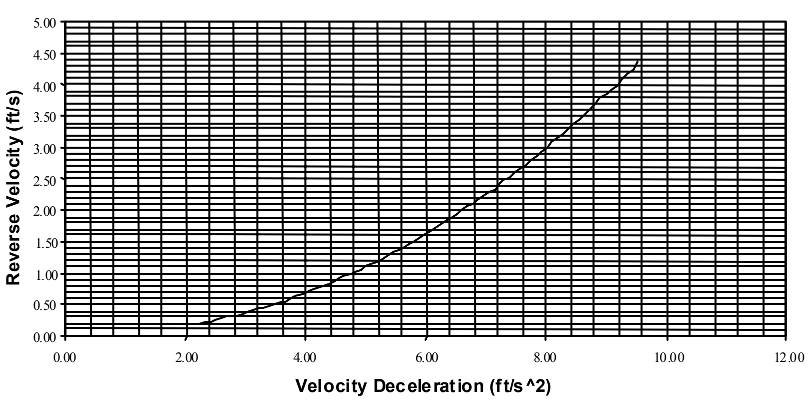


Figure 10.

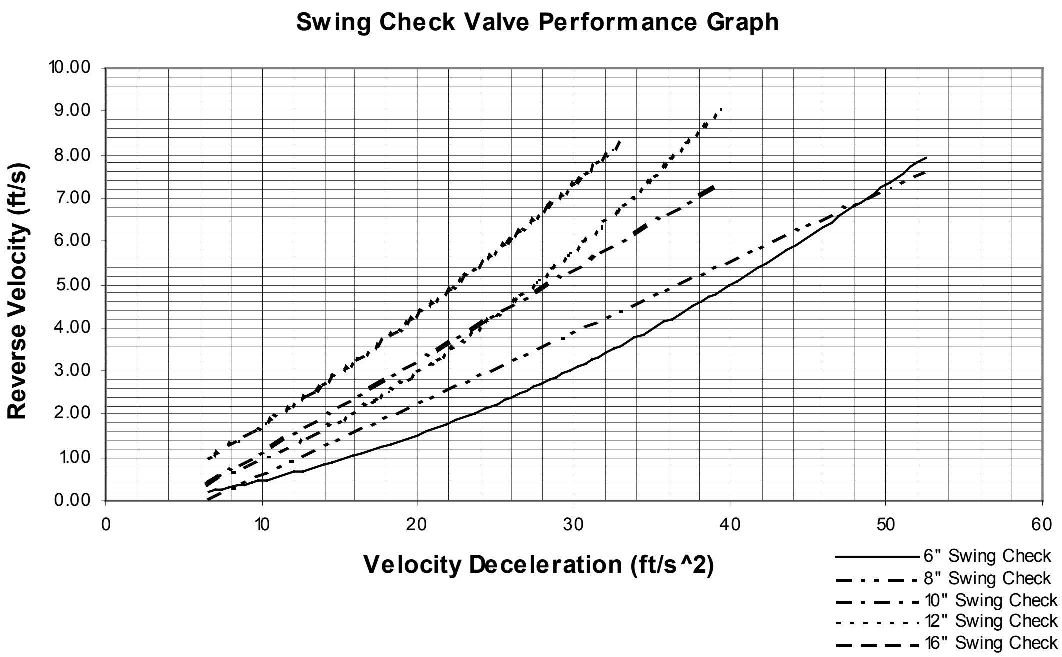


Figure 11.

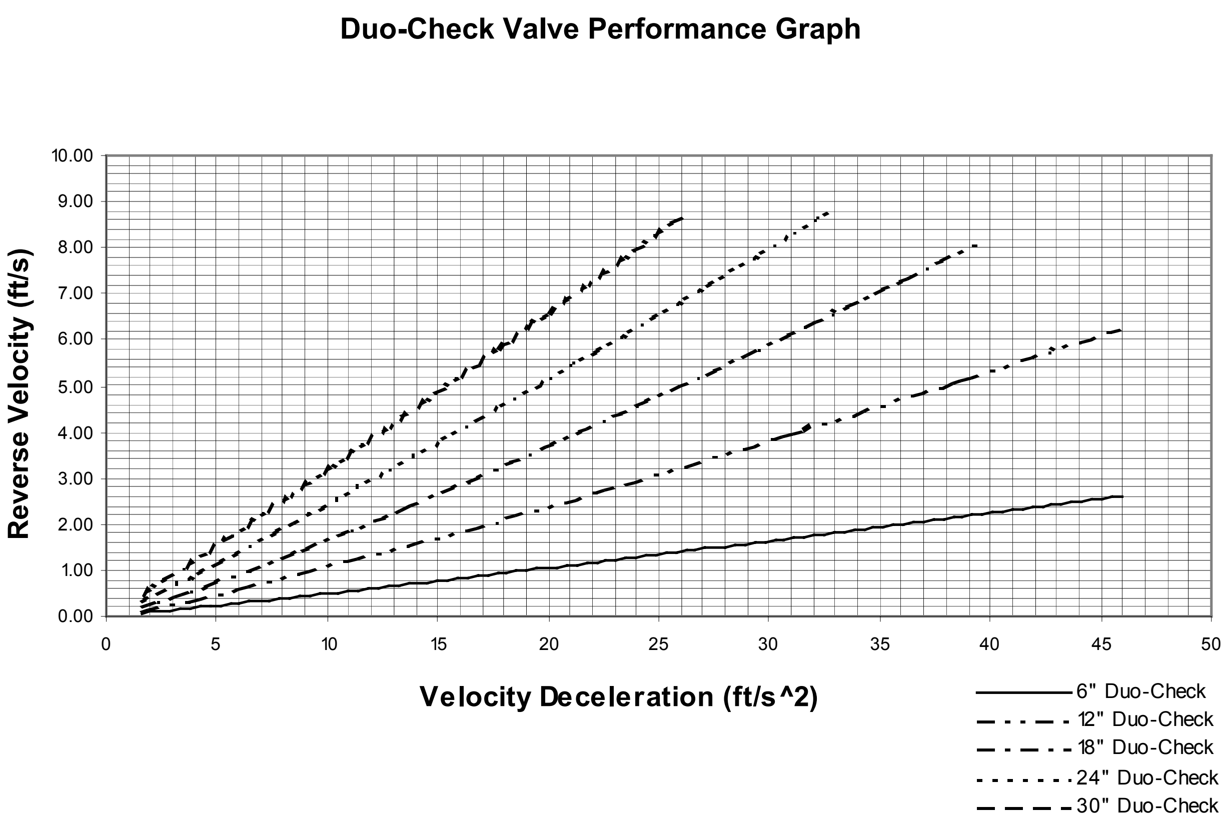


Figure 12.

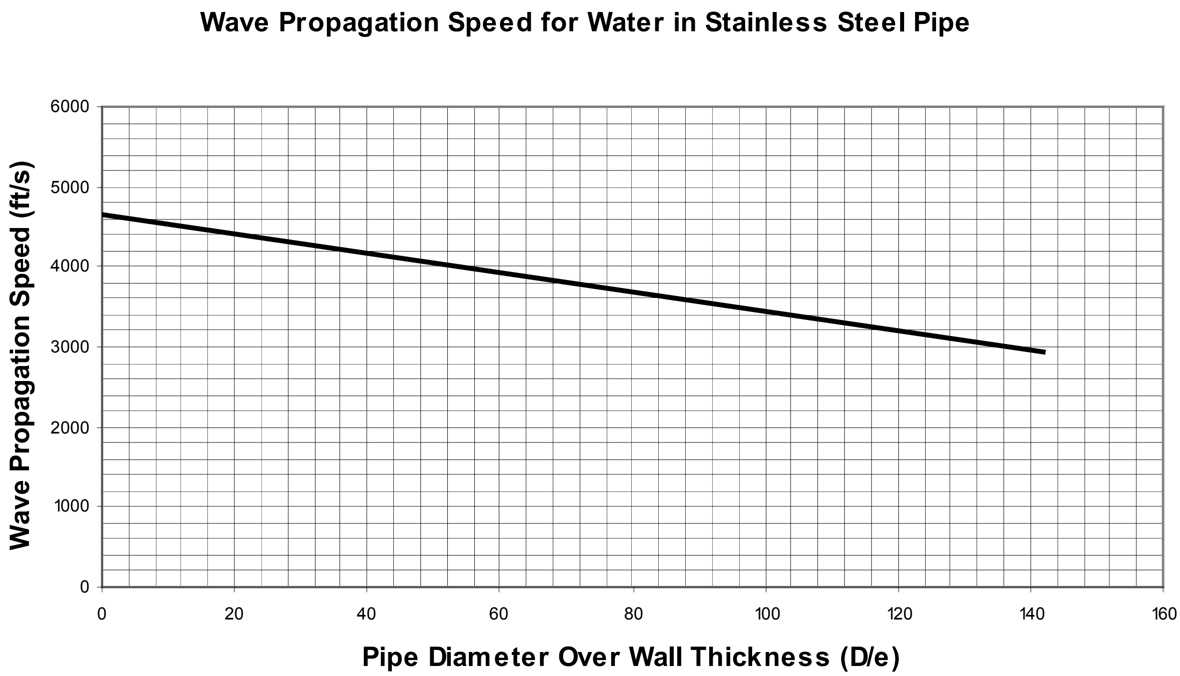


Figure 13.

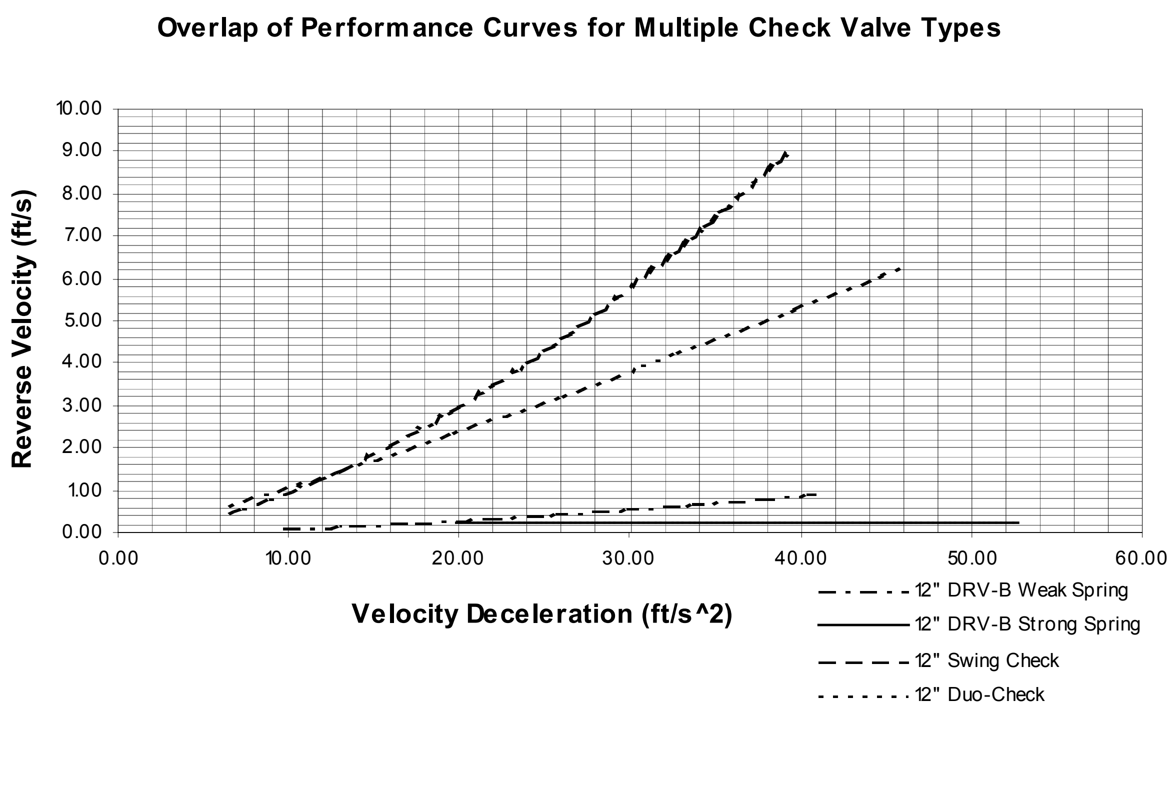


Figure 14.

Presentation

Dynamic Test Results and Methodologyfor large Bore, Axial Flow Check Valves

Presented 2002
Nuclear Industry Check Valve (NIC)
Summer Meeting

ENERTECH

Nuclear Industry Check Valve (NIC) Summer 2002

Dynamic Test Results and Methodology for Large Bore, Axial Flow Check Valves

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CURTISS WRIGHT FLOW CONTROL CORPORATION

2002 Summer NIC

ENERTECH

Objectives of Paper

The technical paper entitled

"Axial Flow Check Valve Dynamic Testing-Using Test Results to Select the Optimum Valve Design"

will be published in the Proceedings of the

"Seventh NRC/ASME Symposium on Valve and Pump Testing."

To obtain a copy, please forward a request to:

enertech@et.curtisswright.com

Enertech

Axial Flow Check Valve Dynamic Testing

Objectives of Paper

- Introduce check valve dynamic testing methods
- Present results of dynamic testing
- Illustrate multiple methods to calculate pressure surge
- Compare dynamic performance of different style axial flow designs

2002 Summer NIC

3

Enertech

Axial Flow Check Valve Dynamic Testing

Presentation Outline

- Purpose of Dynamic Testing
- Test Methodology
- Presentation of Data
- How to use the curves
- Conclusion

2002 Summer NIC

4

Enertech

Axial Flow Check Valve Dynamic Testing

Purpose of Dynamic Testing

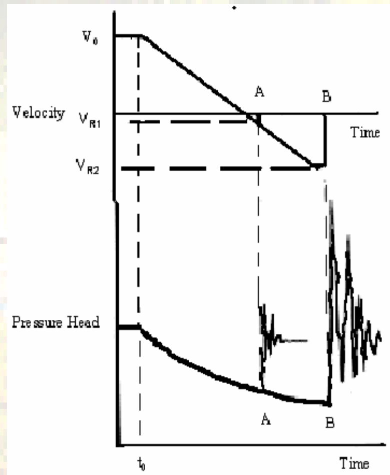
$$\Delta p = \pm \rho a v_r$$

Δp = Pressure change (psig)

a = Wave propagation speed (ft/s)

ρ = Fluid density (lb/ft^3)

v_r = Reverse velocity of the system (ft/s)



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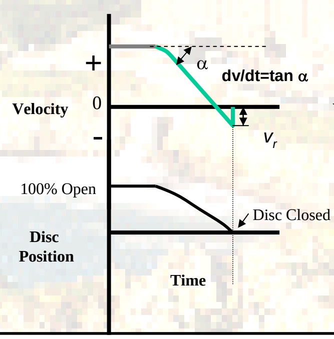
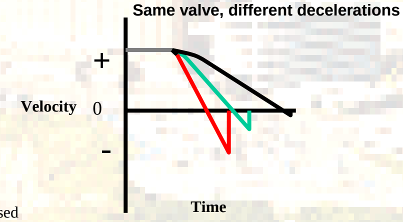
Axial Flow Check Valve Dynamic Testing

Purpose of Dynamic Testing

Dynamic Testing is required to establish a relationship between v_r and system deceleration.

- Deceleration is defined as the average change in velocity from the time a pump is tripped until the check valve closes.
- Deceleration is a function of system configuration.
- Deceleration of fluid is caused by increasing downstream pressure or decreasing upstream pressure.





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Axial Flow Check Valve Dynamic Testing

Testing Methodology

- **Steady State Testing**
  - **Resistance Coefficient(K_{factor})**
  - **Stroke vs. K_{Factor}**
  - **Pressure Recovery Factor**
  - **Critical Velocity (V_o)**
  - **Friction vs. stroke direction**
- **Dynamic Testing**
  - **Multiple Decelerations**
  - **Multiple Spring Strengths**
  - **Rapid Upstream Pressurization or Downstream Venting**

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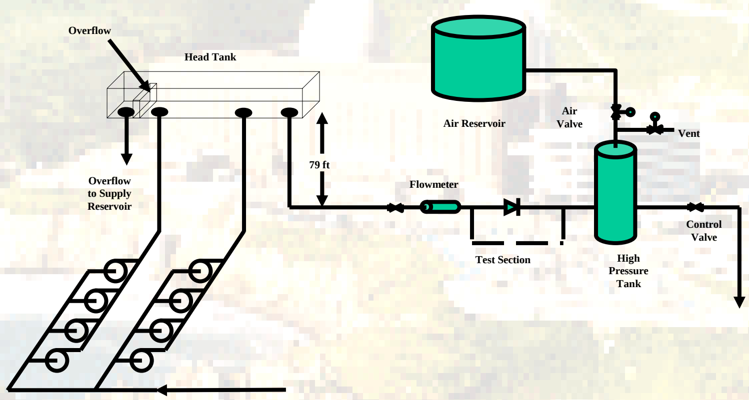
Axial Flow Check Valve Dynamic Testing

Testing Methodology

Large Bore Test Loop-Downstream Pressurization



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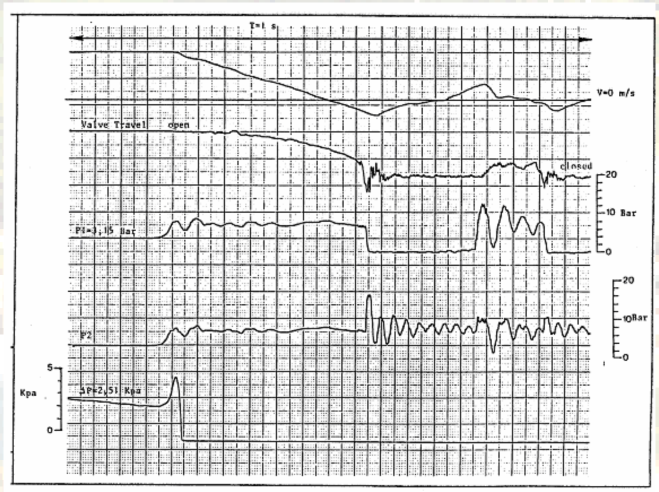
Axial Flow Check Valve Dynamic Testing

Example of one Large Bore Test Run



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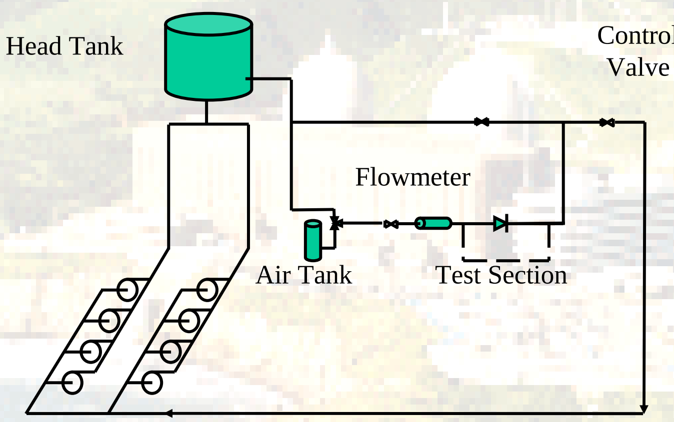
Small Bore Test Loop-Upstream Venting



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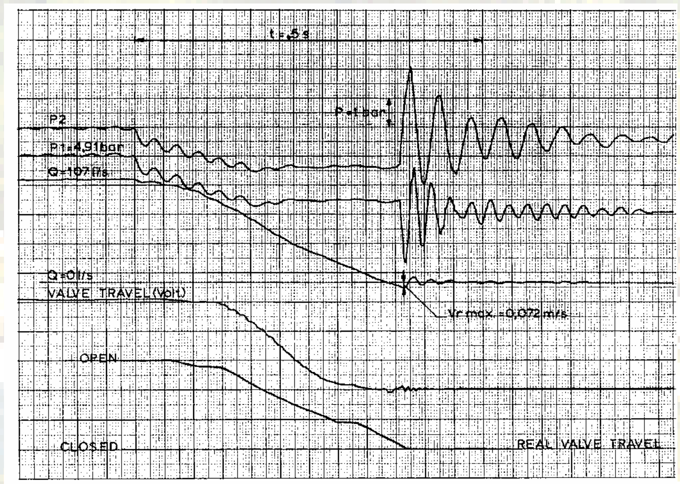
Axial Flow Check Valve Dynamic Testing

Example of one Small Bore Test Run



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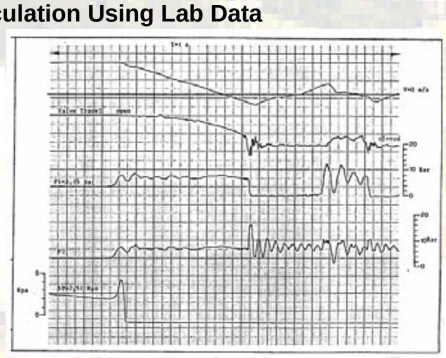
Axial Flow Check Valve Dynamic Testing

Example Pressure Surge Calculation Using Lab Data



Actual Result:

V_r=0.71m/s,

Pressure Surge= 9 bar

Conversion: 2.32 ft/s, 130 psi,

Calculated Result:

$\Delta p = a * \rho * v_r$

Δp=4250ft/s * 62.3lb/ft^3 * 2.32ft/s * s^2/32ft * ft^2/144in^2

Δp=125.5 psi = 9.1 bar

Difference: Actual vs. Calculated- 9 bar vs. 9.1 bar = ~1% difference

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Axial Flow Check Valve Dynamic Testing

Presentation of Data

Five Different Axial Flow Valve Types Tested

- 32” Split Body, Ring Style Disc
- 32” One-Piece Body, Ring Style Disc
- 32” Wafer Body Style, Ring Style Disc
- 24” High Flow, One-Piece Body, Ring Style Disc
- 12” One-Piece Body, Ring Style Disc Test

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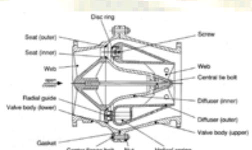
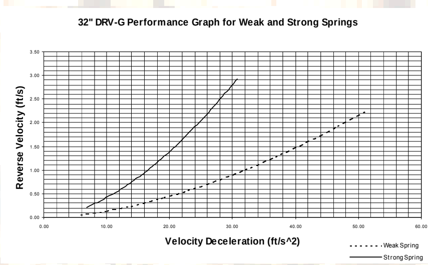
Axial Flow Check Valve Dynamic Testing

Presentation of Data

32” Split Body, Ring Style Disc

Type: Axial Flow
Manufacturer/Model: Entech NozzleCheck Model DRV-G
Size: 32” NPS
V_o **-**Strong Spring-8.92 fps
V_o **-**Weak Spring- 6.2 fps
Orientation: Horizontal
Test Medium: 66 °F Water
Test Loop: Large Bore Loop at Delft Hydraulics





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Axial Flow Check Valve Dynamic Testing
Presentation of Data

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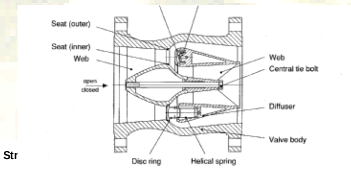
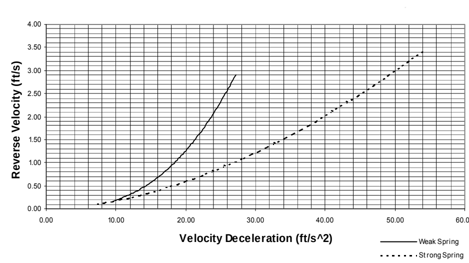
32" One-Piece Body, Ring Style Disc

Type: NozzleCheck
Manufacturer/Model: Entech/NozzleCheck Model DRV-B
Size: 32" NPS
V_o -Strong Spring- 8.81 fps
V_o -Weak Spring- 6.1 fps
Orientation: Horizontal
Test Medium: 66 °F Water
Test Loop: Large Bore Loop at Delft Hydraulics



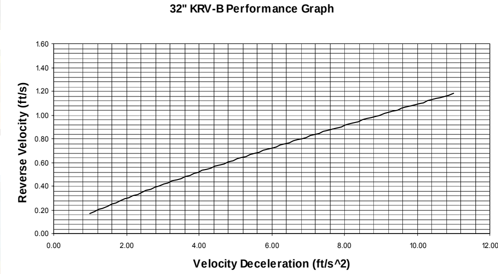


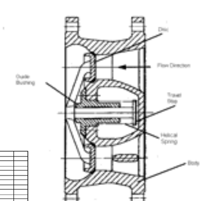
Enertech
Axial Flow Check Valve Dynamic Testing
Presentation of Data

2002 Summer NIC

32" Wafer Body Style, Ring Style Disc

Type: Axial Flow
Manufacturer/Model: Entech/NozzleCheck Model KRV-B
Size: 32" NPS
V_o: 6.56 fps
Orientation: Horizontal
Test Medium: 66 °F Water
Test Loop: Large Bore Loop at Delft Hydraulics





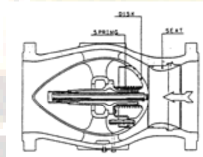
Enertech

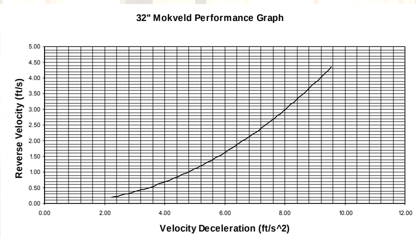
Axial Flow Check Valve Dynamic Testing

Presentation of Data

Test Data for 32” Solid Disc, Axial Flow Design

Type: Axial Flow
Model: Mokveld Circular Disk- TKZ-E or similar.
Size: 32” NPS
V_o - 6.9 fps
Orientation: Horizontal
Test Medium: Unknown
Test Loop: Unknown, Possibly Large Bore Loop at Delft Hydraulics

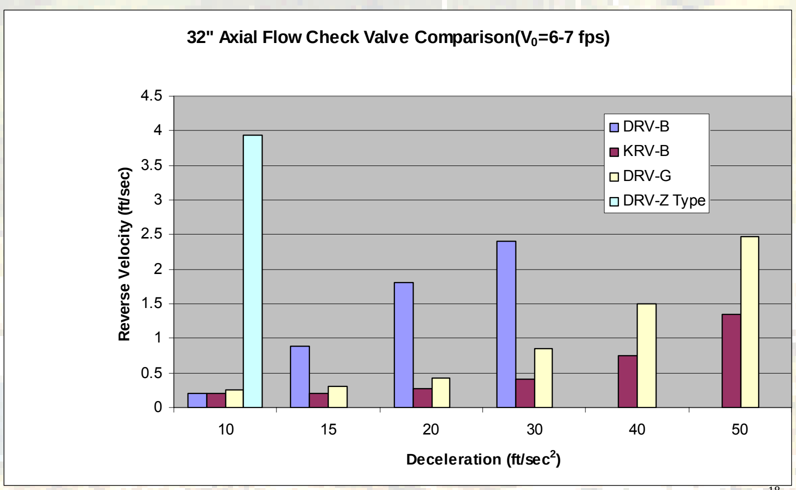




Enertech

Axial Flow Check Valve Dynamic Testing

Presentation of Data



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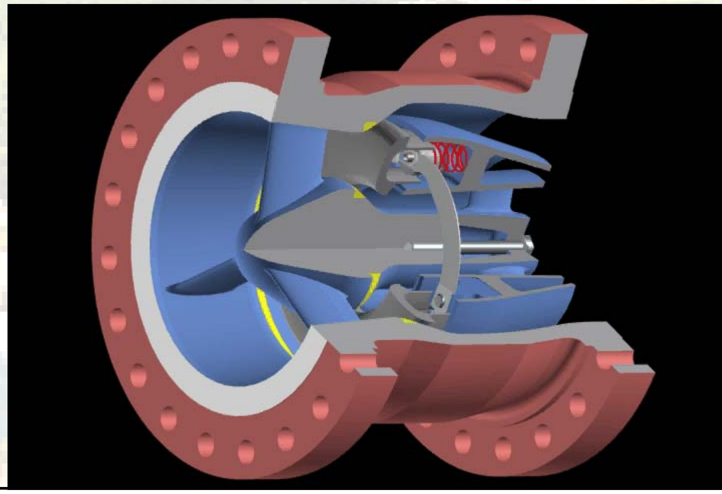
Enertech

Axial Flow Check Valve Dynamic Testing

Presentation of Data

DRV-B Type



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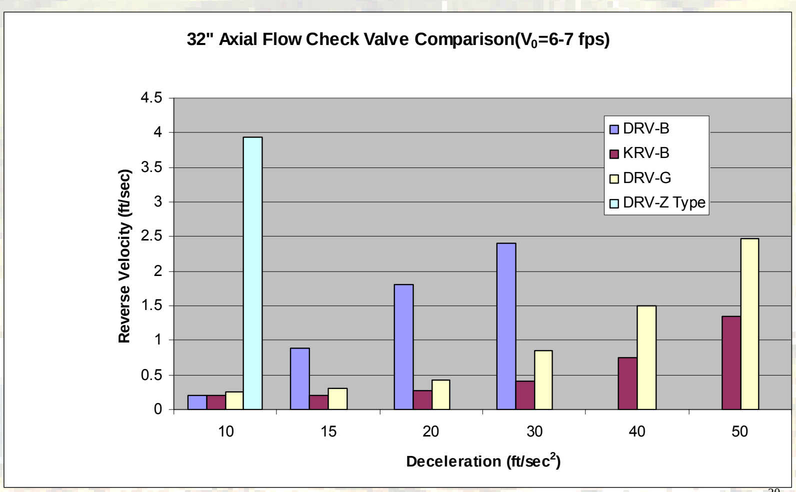
Axial Flow Check Valve Dynamic Testing

Presentation of Data



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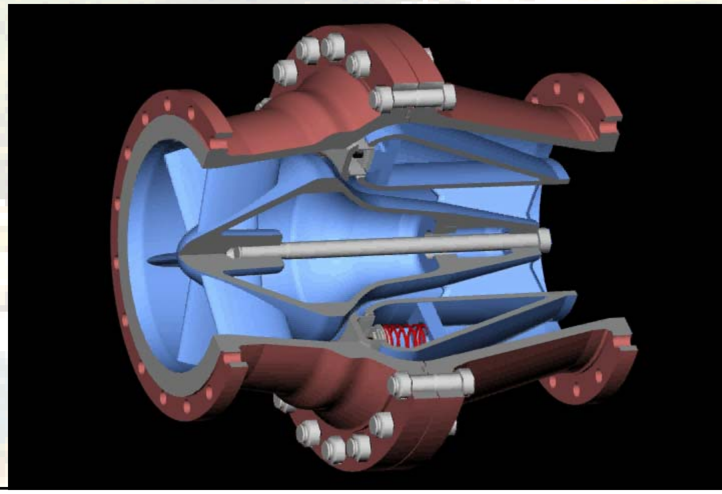
Axial Flow Check Valve Dynamic Testing

Presentation of Data

DRV-G Type



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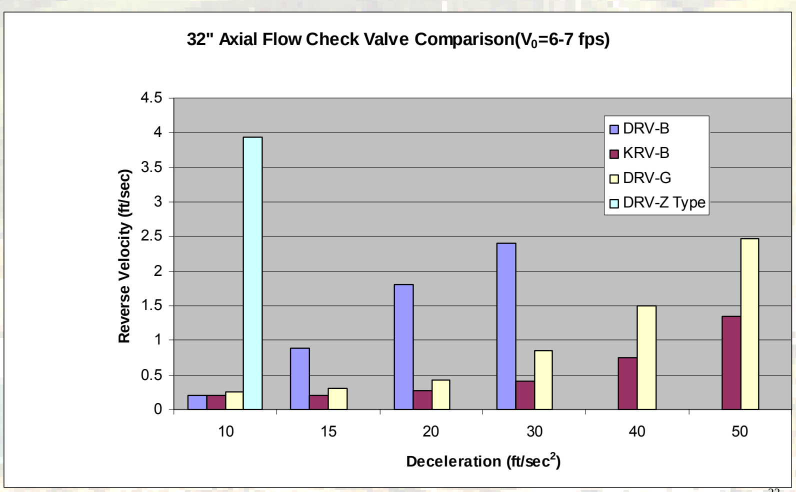
Axial Flow Check Valve Dynamic Testing

Presentation of Data



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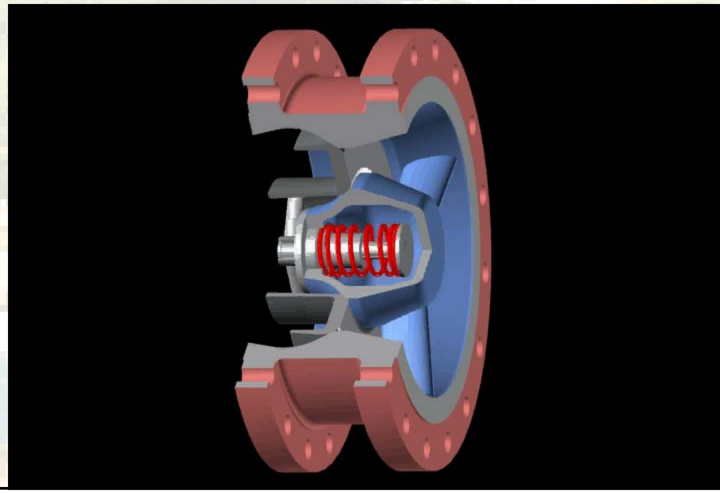
Enertech

Axial Flow Check Valve Dynamic Testing

Presentation of Data

KRV-B Type



ENERTECH

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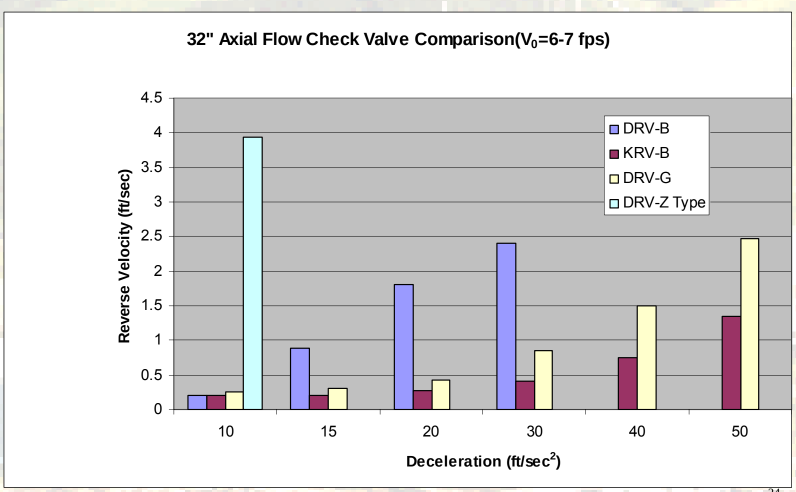
Enertech

Axial Flow Check Valve Dynamic Testing

Presentation of Data



ENERTECH

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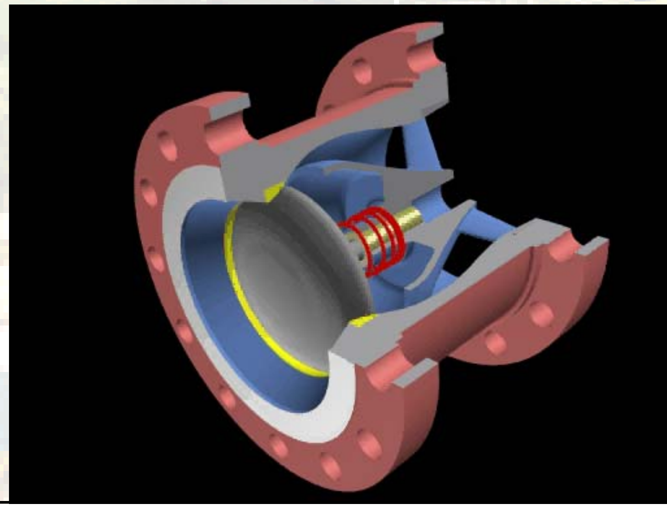
Axial Flow Check Valve Dynamic Testing

Presentation of Data

DRV-Z Type



Enertech

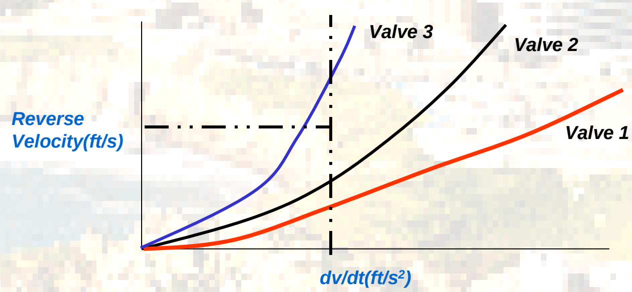
Axial Flow Check Valve Dynamic Testing

Using the Test Results

If you know the deceleration and want to evaluate the resultant pressure surge with different check valves installed...

Dynamic Characteristic Curve

Establish maximum allowable V_r using $\Delta p = \pm \rho a v_r$



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Axial Flow Check Valve Dynamic Testing

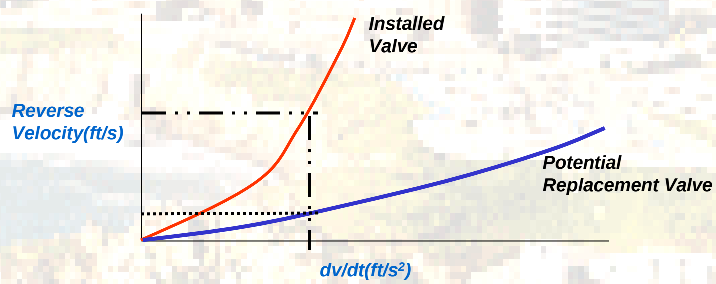
Using the Test Results

If you know the magnitude of pressure surge associated with a specific event but don't know the deceleration and want to calculate the effect of potential valve replacement...

Dynamic Characteristic Curve

1. *Estimate Deceleration*
2. *Calculate V_r and pressure surge with replacement valve*

$$\Delta p = \pm \rho a v_r$$



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Axial Flow Check Valve Dynamic Testing

Conclusion

- ***Dynamic Testing Data is Essential to Accurately Predict Pressure Surge***
- ***Test Date Exists for Numerous Check Valve Designs***
- ***Valves of the Same General Type Have Vastly Different Dynamic Performance***

28

White Paper

Detailed Performance Comparisions Between Swing Type and In-Line Nozzle Check Valves

Presented 2001
EPRI Valve Technology Symposium

EPRI Valve Technology Symposium
August 14-16, 2001

Detailed Performance Comparisons Between Swing Type and In-Line Nozzle Check Valves

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Abstract

Older style swing and split butterfly check valves have been used for many years to provide check valve capability on the outlets of power plant pumps. Detailed high-speed measurements were taken of the water hammer effects due to a swing type check valve closing due to a pump trip on a multi-pump flow manifold. Detailed measurements were later made with a nozzle check valve installed on the same system. Comparisons were made between the two valve types in terms of time to close, peak discharge and suction pressures and pump speed at closure.

The Problem

Older style swing check valves were installed in the outlets of two low-pressure safety injection (LPSI) pumps, feeding a common outlet header. Normally when shifting pumps, the pump to be stopped was decreased in flow to minimize the check valve slam effects. However, on rare occasion a pump would trip under normal flow conditions. The resulting water hammer effects were significant, causing permanent deflections in piping support structure. As a result, a comprehensive test program was initiated to provide data supporting increased piping support structure. The measured results also prompted the investigation of alternate valve types, one of which was later installed. After installation of the new valves the same series of tests were run, with lessor instrumentation due to the expectation of significantly reduced water hammer effects.

An ideal check valve should have three principle properties:

- valve closure at zero flow
- low moving mass
- short moving distance

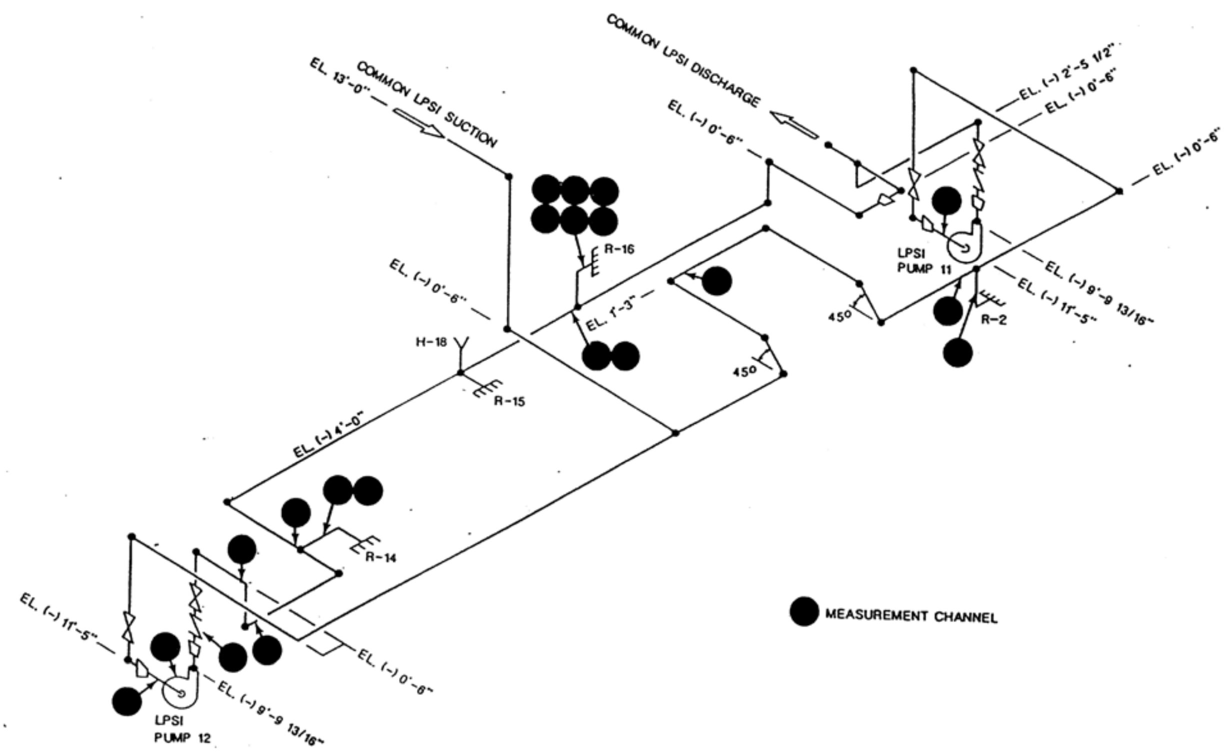
The first minimizes water hammer by closing at the time the flow has transitioned from outlet flow to reverse flow. The second and third properties contribute to rapid closure times, minimizing the time for reverse flow to build.

The classic swing valve does not start to close until reverse flow is established. The moving mass is large and moves through a significant arc to closure. All contribute to large water hammer effects by closing well after the reverse flow has been established, the water hammer being caused by the pressure wave necessary to negate the fluid motion.

System Configuration

Figure (1) shows the general layout of the pumping and instrumentation system.

Figure (1): General Test Layout

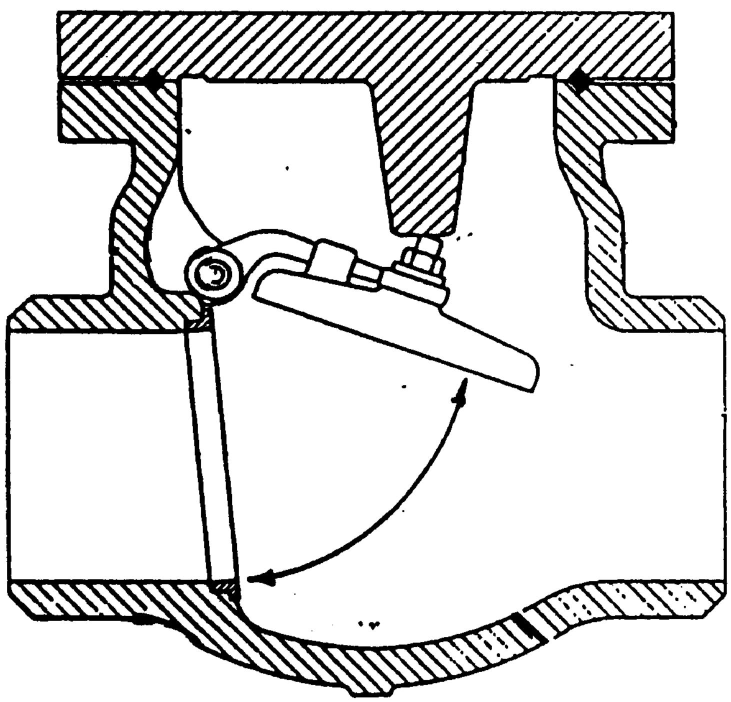


The LPSI pumps shown in Figure (1) have the following properties:

pump:	model 8x21AL single stage horizontal centrifugal pump
design flow:	3000 GPM
maximum flow rate:	4500 GPM
design head:	300 ft.
nominal pump speed:	1780 RPM
suction piping:	14 inch OD, 0.25 inch wall
discharge piping:	10.75 inch OD, 0.25 inch wall

The check valve was a 10 inch ANSI 300 lb. single element swing type as shown in Figure (2).

Figure (2): Swing Check Valve



Instrumentation System

The keys to successful instrumentation for check valve testing is two-fold:

- time response of the transducers
- bandwidth of the recorders

For the initial swing valve testing, there was a concern about the loading on the piping support by the water hammer shocks. As a result, an extensive suit of sensors was employed to measure the force on piping supports, motion of piping elements, and system transient pressures. The types and numbers of transducers are shown in Table (1).

Table (1): Test Sensors

Sensor	Type	Number Used	Locations
key phasor	laser	1	pump shaft
accelerometer	piezoelectric	1	check valve body
pressure	piezoresistive strain gage	5	suction, discharge and header piping
velocity	integrating piezoelectric	6	piping
displacement	eddy current	3	piping and support structure
displacement	optical tracker	1	piping support structure
strain	film strain gage	3	piping support structure

The accelerometer had a natural frequency of 35 kHz with a range, at +/- 5%, of 3 to 10 kHz, The acceleration range was +/- 300 g.

The pressure transducers had a natural frequency of 200 kHz with a pressure range of 0 to 500 psia. Combined non-linearity, hysteresis and repeatability were +/- 5 psi with overpressure capability to 750 psia and burst pressure of 1500 psi. The pressure transducers were statically calibrated before and after valve testing to ensure stability.

The velocity sensors had a maximum range of +/- 50 in/sec. The eddy current displacement sensors had a range of 0-1 inch and a frequency range of 0 to 50 kHz.

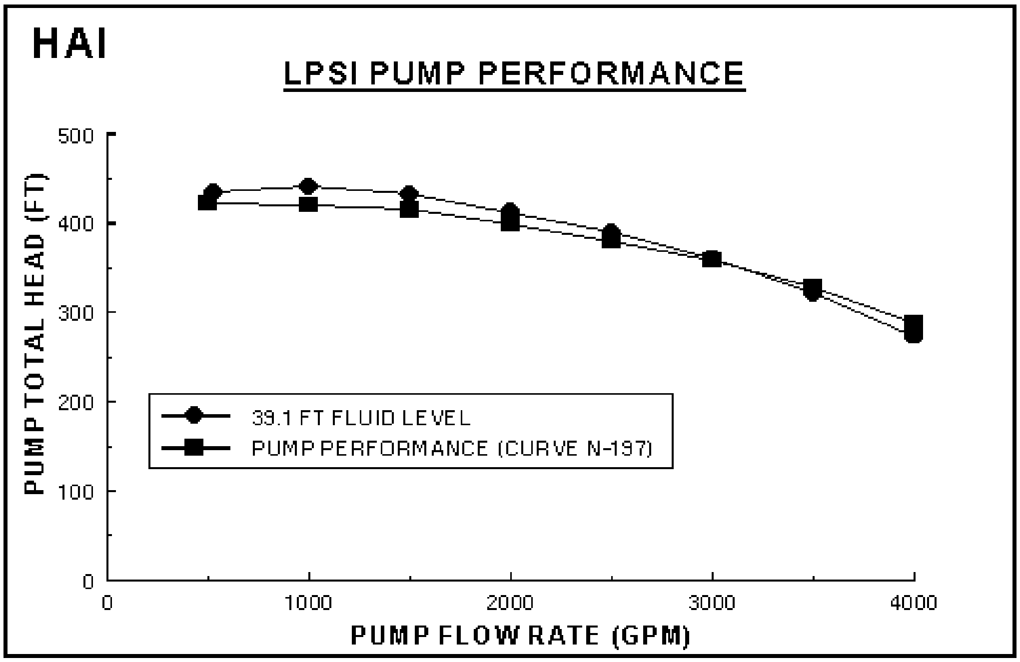
All signals were recorded on analog tape recorders with bandwidths of 0 to 50 kHz. Data was transcribed from the tape to digital form by a multi-channel analog to digital converter operating at a scan frequency of 50 kHz (scan interval of 20 micro seconds). As an example of the accuracy effect of scan rate, the pulses from the laser key phasor occur at intervals of 0.337 seconds (1780 RPM). The result of the scan frequency would therefore represent less then an error of 1 RPM in the measurement.

In addition to the described data recording system, a four-channel digital storage oscilloscope with a built in plotter was used to provide realtime data. This allowed immediate evaluation of the data to avoid risk to the plant piping system.

Flow Model Correction

The pressure transducer for the pump outlet pressure was not mounted directly on the pump outlet but on a vent opening on the pump outlet header. The header was 9.4 feet above the pump outlet with 14.6 feet of 10 inch schedule 20 stainless steel pipe, one gate valve and one check valve between the pump and pressure transducer. A commercial flow code (AFT Fathom) was used to model the steady state pressure difference as a function of configuration and flow rate. Figure (3) shows a comparison of measured pump performance versus flow rate as a check on the accuracy of the measurements and flow model. The computed pressure differential (between the pressure transducer location and the pump outlet) was used in later data reduction to correct the pressure readings to the pump outlet location.

Figure (3): LPSI Pump Performance



Swing Valve Test Data

The most severe pump trip water hammer occurred at a flow rate of 1500 GPM (vice 550 GPM or 850 GPM tests run earlier). Testing at 2000 GPM was not run because of the severity of the pressure surges measured at 1500 GPM. The first channel reduced was the key phasor to determine relative pump trip time. The measured pump speed prior to pump trip was 1789 RPM (vs. the spec value of 1780 RPM). The measured (from the key phasor data) pump trip time was 0.3280 seconds after the start of the digitized window of data. It should be noted that the “zero” time was arbitrary and represented the beginning of the eight second “digitizer window” used to digitize the data. The “digitizer window” was set to capture the pump trip within the first second of digitization, providing at least seven seconds of detailed high speed information on the flow system reaction to the pump trip.

Figure (4) shows the discharge pressure as a function of time. Pump trip time is shown as 0.3280 seconds.

Figure (4): Pump Discharge Pressure

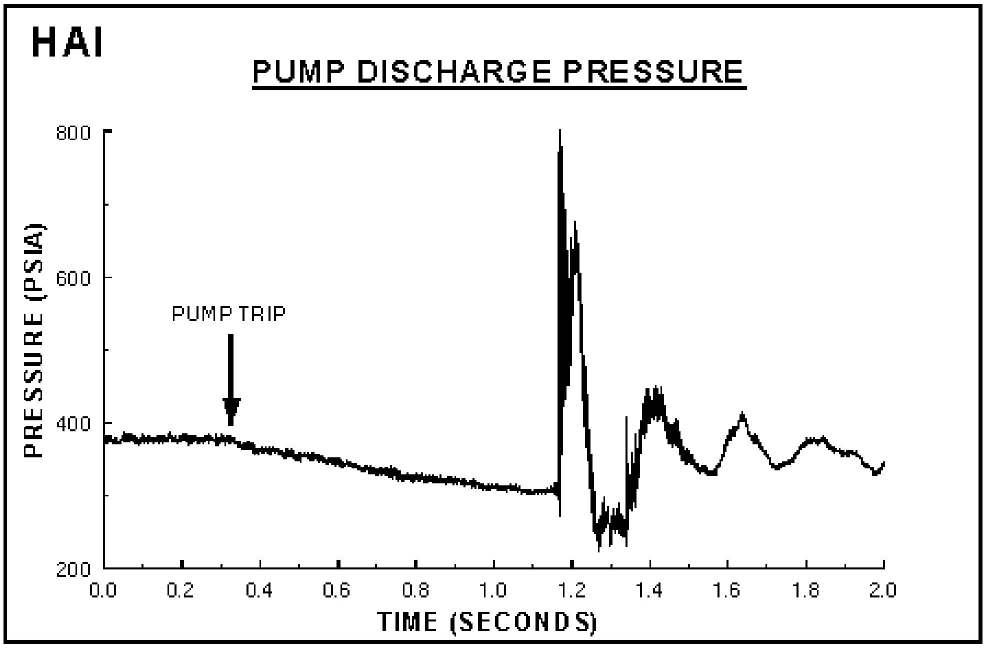


Figure (5) expands the time scale after the first reaction at 1.1650 seconds with a second reaction at 1.3380 seconds. The initial reaction shows high frequency high amplitude waves consistent with water hammer phenomena resulting from a rapid blockage of reverse flow by the check valve

Figure (5): Pump Discharge Pressure

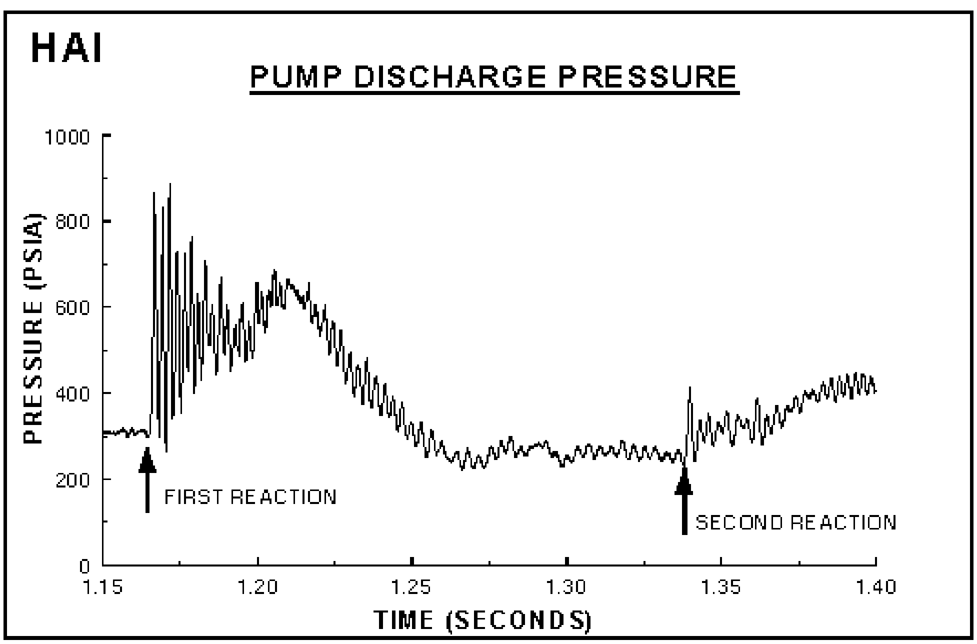


Figure (6) shows the trace from the accelerometer mounted on the swing check valve body. The major shocks at 1.1630 and 1.3360 seconds are 0.002 seconds earlier then the pressure events downstream. The time differential and distance would indicate a wave speed of 4,220 ft/sec, not inconsistent with the computed wave speed in this pipe size [4180 ft/sec, Reference (1)].

Figure (6): Pump Accelerometer Output

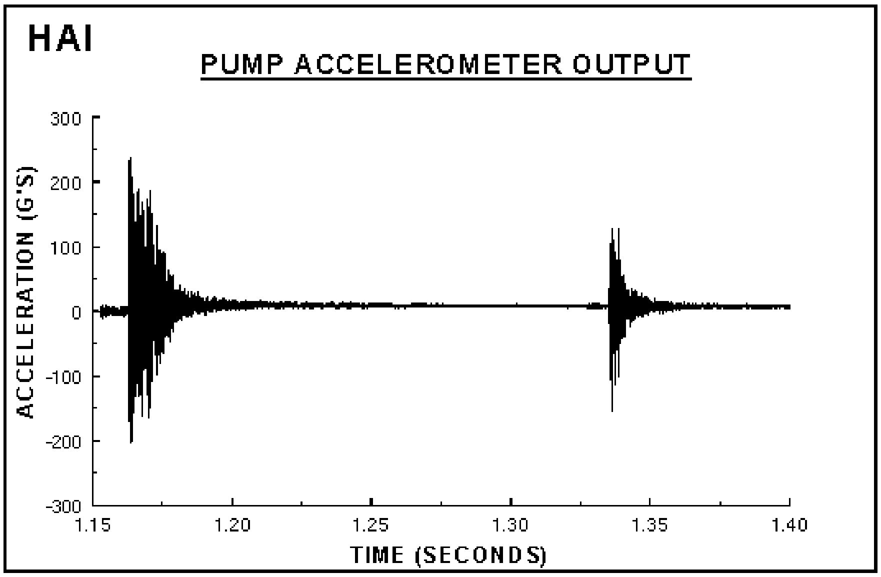


Figure (7) shows an expansion of the time around the first shock. Note the increase in vibration at 1.1530 seconds prior to the major shock at 1.1630 seconds. If this is interpreted as motion of the valve it indicates an extremely short period of valve motion (0.010 seconds).

Figure (7): Pump Accelerometer Output

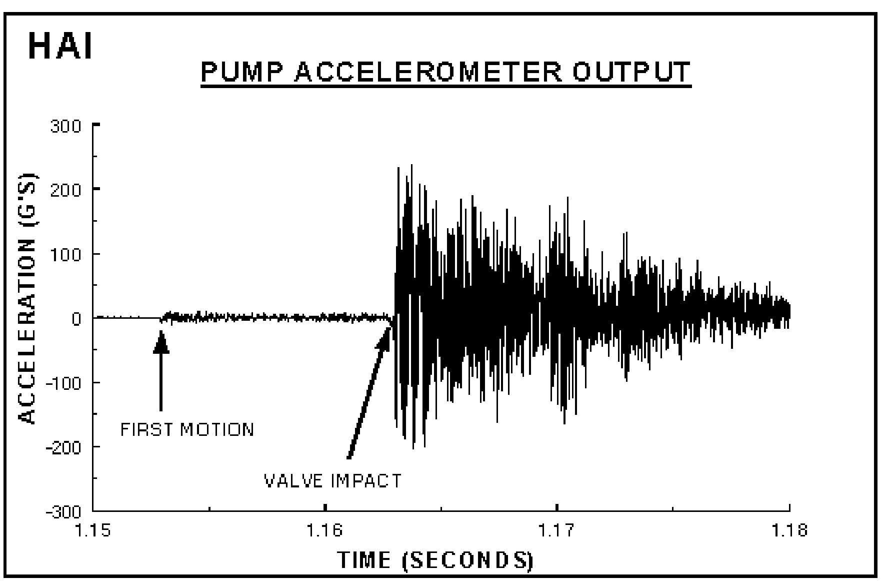
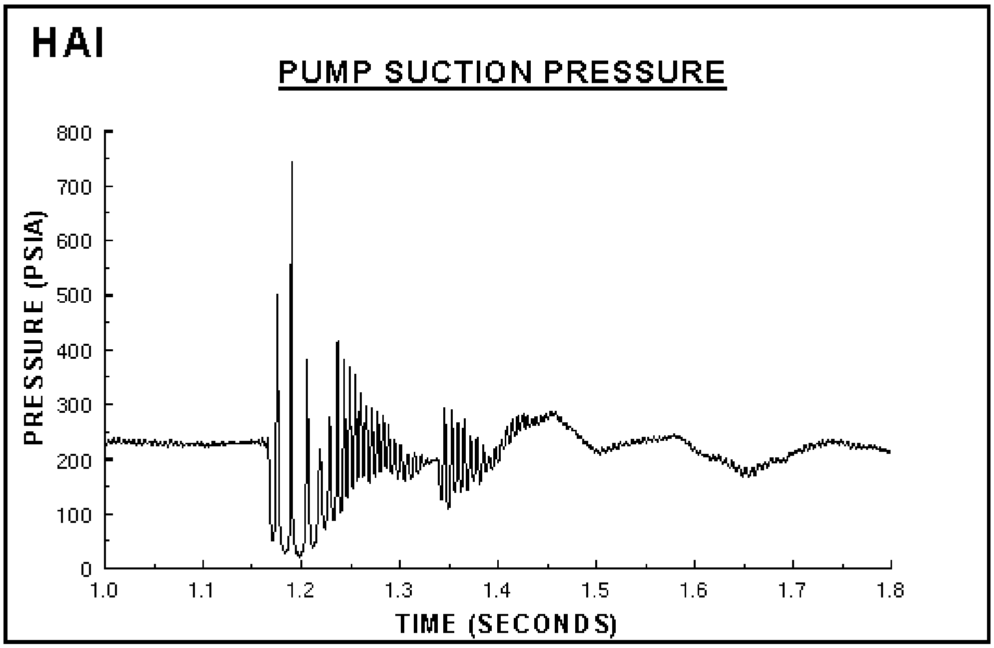


Figure (8) shows the pump suction pressure. The time of first change coincides with the first pressure surge in the discharge line and the second event coincides closely with the second event in accelerometer and discharge pressure. The pressure shown in Figure (8) is typical of traces for a separated flow column with high pressure spikes occurring at column recombination.

Figure (8): Pump Accelerometer Output



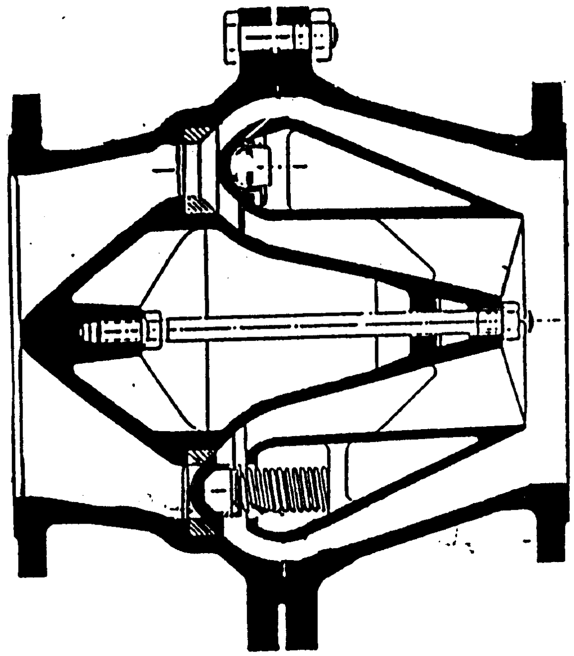
The remainder of the data set showed large pipe motion and high structural loading of the support system. As a result, the decision was made to install in-line nozzle check valves to reduce transient loading on the piping system.

Replacement Valve Characteristics

The replacement check valve chosen was a Mannesmann 10 inch model DRV-G in-line nozzle check valve, ANSI B16.5, class 300 pound with a C_V of 2900 GPM. Figure (9) shows a cutaway of the type of valve installed. The upper portion shows the valve in the open position, the bottom shows the valve in the closed position. Flow would move from left to right.

It was anticipated that this type of valve would minimize the water hammer effects due to its impact on the three “ideal” check valve properties. The flow pressure acting against a valve spring, to open the valve, addresses the first consideration, valve closure at zero flow. Just prior to flow reversal the spring force should overcome the flow force on the face of the moving portion of the valve. This should provide closure very close to the zero flow condition. In addition the moving mass of the valve is greatly reduced, as is the range of motion needed to close the valve.

Figure (9): In-Line Nozzle Valve



In-Line Nozzle Valve Test Data

The pump trip testing was re-run after the installation of the new valves. The test set-up was identical to the original set-up used for the swing valve testing valve testing with the following exceptions.

- substitution of the in-line nozzle check valve in place of the swing check valve
- the system pressure was lower in the second series due to plant line-ups and system status

The instrumentation suit was considerably smaller than previously used because of the anticipation that the piping system reaction would be minimal. The laser key phasor was again used as the timing fiducial for the pump trip time. Pre-trip pump speed was measured as 1790 RPM with pump trip time of 0.4090 seconds into the 8 second digitized data window. Figure (10) shows the discharge pressure for the 8 second window.

Figure (10): Pump Discharge Pressure

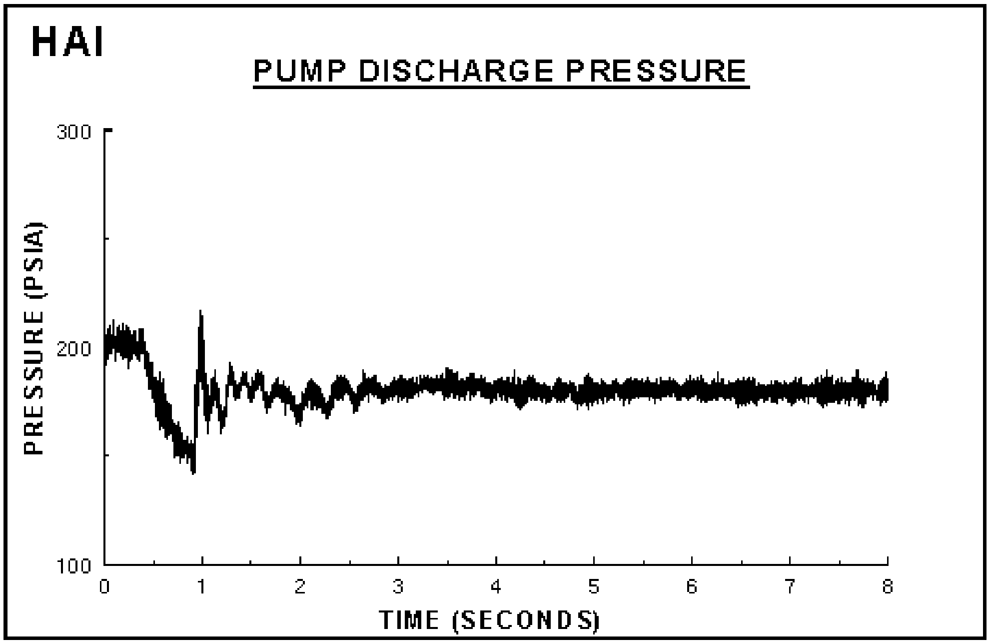


Figure (11) shows a time expansion over the first 1.3 seconds of the window. Note that the pressure peak at valve closure (0.9200 second) is barely back up to the pre-trip level.

Figure (11): Pump Discharge Pressure

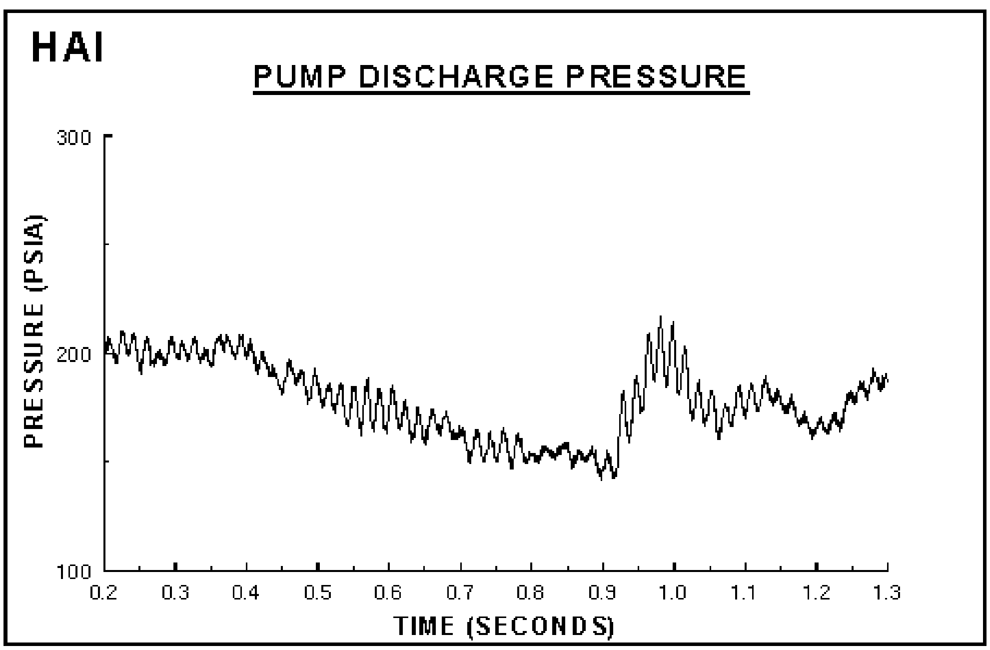


Figure (12) is the suction pressure of the pump over the 8 second data window.

Figure (12): Pump Suction Pressure

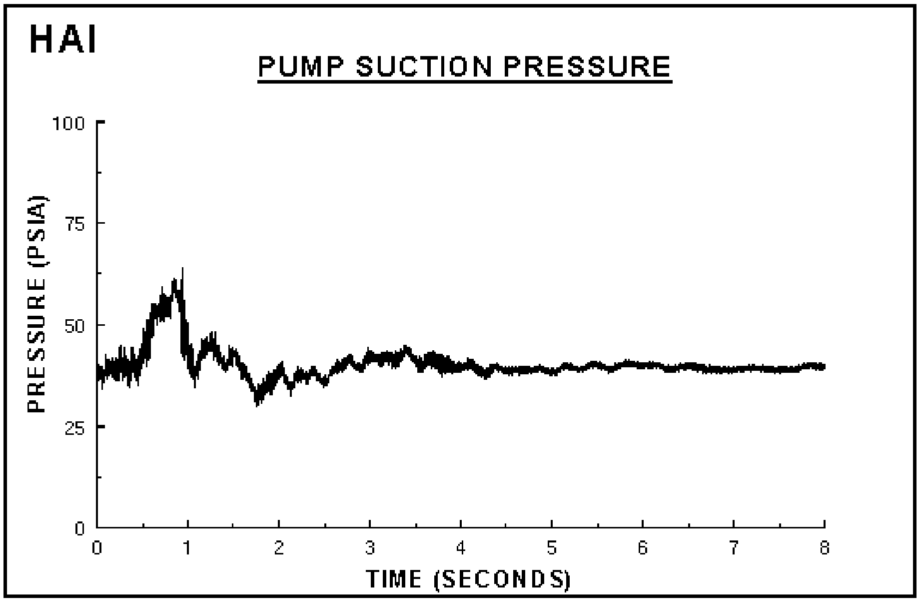


Figure (13) is a time expansion of the first 1.4 seconds of the data window. The most severe pressure oscillations occur at valve closure (0.9200 seconds).

Figure (13): Pump Suction Pressure

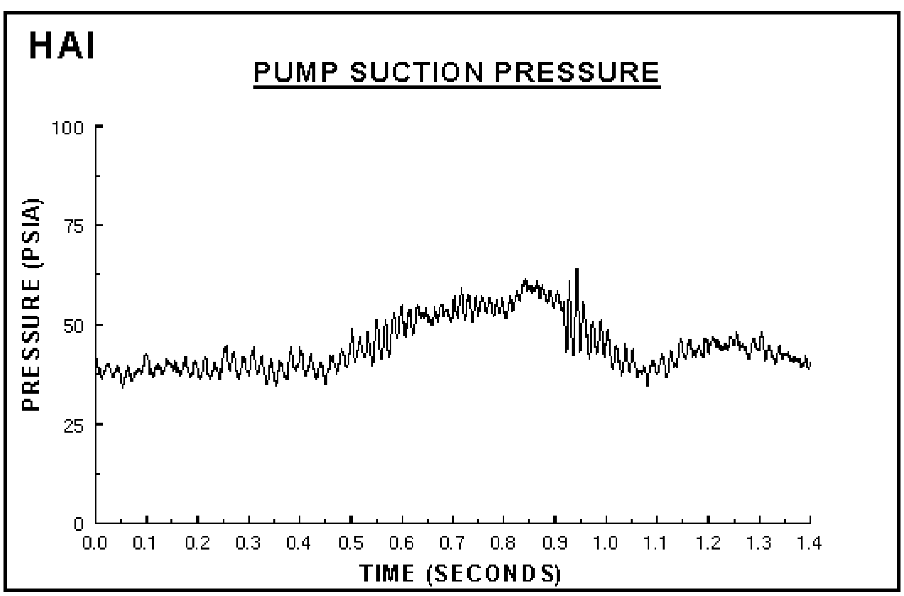


Figure (14) is the accelerometer on the valve case. In the case of the inline nozzle check valves, it was anticipated that the valve slam levels would be very low so no scale factor on the accelerometer was carried. However, it should be noted that the signature of the accelerometer at valve closure (0.9200 seconds) is barely above the pre-trip noise level of the pump vibration.

Figure (14): Check Valve Accelerometer

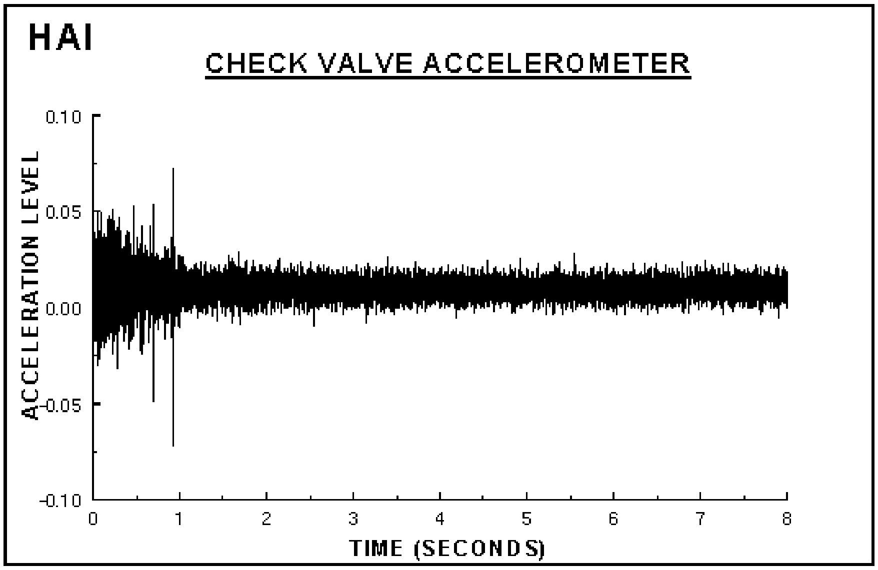
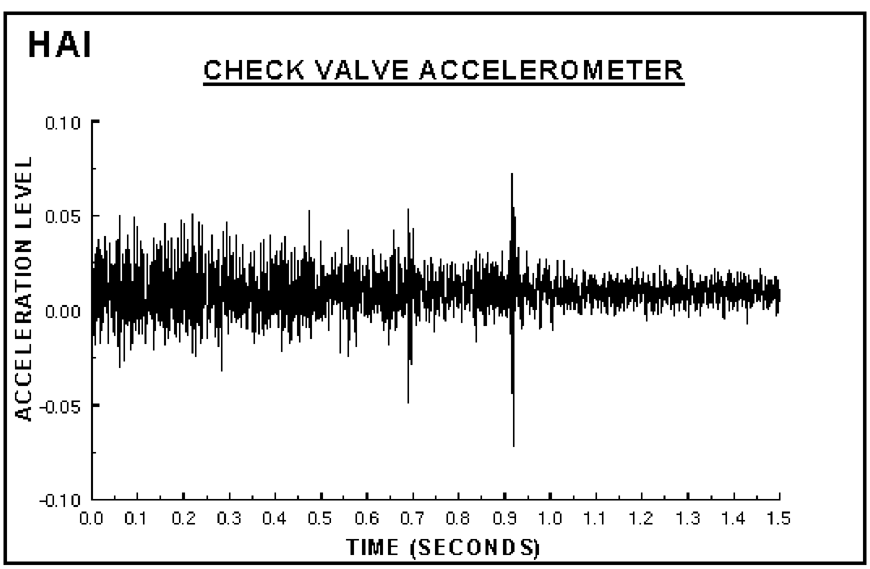


Figure (15) is the time expansion of the accelerometer trace for the first 1.5 seconds of the data window and shows the valve closure at 0.9200 seconds.

Figure (15): Check Valve Accelerometer



Data Comparisons and Conclusions

Table (2) shows a comparison of pertinent parameters, comparing the performance of the swing type check valve to the in-line nozzle valve.

Table (2): Comparison Values

Parameter	Swing Valve	In-Line Nozzle Valve
pre-trip RPM	1789	1790
pre-trip discharge pressure (psia) at transducer	381.13	201.80
pre-trip suction pressure (psia)	219.59	38.91
pre-trip pressure[1] differential across pump (psi)	168.47	169.82
time from trip to valve closure (sec)	0.8350	0.5110
peak discharge pressure[2] after closure (psia)	542.23	202.00
peak suction pressure after closure (psia)	758.93	63.85
pump speed at valve closure (RPM)	1157	1333
maximum pressure rise above pre-trip in discharge (psi)	161.10	0.20
maximum pressure rise above pre-trip in suction (psi)	539.34	24.94
pressure pulse from valve closure (psi)	249.82	49.56

Note: (1) 6.93 psi has been added to the discharge pressure to correct for the transducer location relative to the pump discharge.
(2) The measured peak values for the swing valve discharge has been corrected for transducer pipe resonance (see explanation below).

It was postulated that the high frequency large amplitude pulses in the discharge header measured during the swing valve testing were due to ringing in the transducer connection piping. Per Reference (1), the cycle time for an impulse in a pipe was given as:

$$4\,L/a \;=\; \textit{cycle time} \qquad (1)$$

where:

L = pipe length (ft)
a = speed of wave propagation (ft/sec)

The transducer pipe length was 2.57 feet with a wave speed computed, for the transducer pipe, of 4424 ft/sec. This would result in a predicted cycle time of 0.00232 seconds. The average measured cycle time for the first twenty cycles of discharge pressure oscillations for the swing valve data was 0.00227 seconds. With this confirmation of the postulated transducer pipe effect, the average of the first twenty cycles, computed at 542.23 psia, was used as the actual discharge pressure shown in Table (2).

Table (2) clearly demonstrates that the in-line nozzle check valve closes more quickly after pump trip (0.5110 seconds versus 0.8350 seconds), and as a result generates considerably lower water hammer pressure pulses (49.56 psi versus 249.82 psi) in the pump discharge header. It is postulated that this faster closure time is much nearer to the zero velocity flow condition in the valve and therefore generates considerably low pressure waves in the piping system. On the suction side, the pressure trace [Figure (8)] exhibits the classic form of a separated flow column, caused by the rapid valve closure against a fluid stream already moving upstream from the check valve because of the decreasing pump head. The resulting high pressure surge of 539.34 psi is the most extreme encountered in this series of testing.

The measured discharge pressure rises were compared to those provided by the simple scaling laws provided in Reference (1).

$$\Delta H = -\frac{a\Delta V}{g} \tag{2}$$

where:

ΔH = pressure surge in feet of head
a = wave propagation speed (ft/sec)
V = flow velocity through valve (ft/sec)
g = gravitation constant (ft/sec)

The original fluid velocity in the discharge header is 5.83 ft/sec at 1500 GPM. Table (3) gives the computed values of reverse velocity predicted by the above relationship.

Table (3): Reverse Velocity Scaling

Parameter	Swing Valve	In-Line Nozzle Valve
ΔH (ft)	536.01	114.40
a (ft/sec)	4180	4180
ΔV (ft/sec)	-4.13	-0.881

If the transition from a plus velocity of 5.83 ft/sec to a negative velocity of –4.13 ft/sec were linear in time, then the time to zero velocity (and, therefore, zero pressure pulse) would be:

$$t_o = \left(\frac{5.83}{5.83+4.13}\right)0.8350 = 0.489\,\text{sec} \tag{3}$$

The swing check valve shows a difference of 0.346 seconds where the in-line nozzle check valve shows a difference of 0.022 seconds. The ideal valve would close at exactly 0.489 seconds to preclude a pressure pulse.

The final conclusion is that a check valve which has a short stroke, low moving mass, and closes close to the point of zero flow velocity, will produce the lowest water hammer pressure surge. Per the data previously presented, the in-line nozzle check valve tested is close to the desired properties and produces very low values of pressure pulses, a definite advantage to the related piping system.

ACKNOWLEDGEMENTS

We would like to acknowledge the assistance provided by Applied Flow Technology, Woodland Park, Colorado for their assistance in the fluid system modeling described in this paper. We would also like to acknowledge the cooperation of Enertech Corporation, Brea, California in providing detailed information and graphics for the in-line nozzle check valves.

REFERENCES

(1) Chaudhry, M. H., 1987, "*Applied Hydraulic Transients*," 2nd Edition, Van Nostrand Reinhold Co.

(2) Hannah, B. W., Sponsel, J. and Kostelnik, M.; "*Detailed Performance Measurements of In-Line and Swing Type Check Valves*"; ASME 2000 Fluids Engineering Division Summer Meeting; June 11-15, 2000.

White Paper

Solution to Ginna's Component Cooling Water System and Service Water System Check Valve Problems

Presented 1996
Nuclear Industry Check Valve (NIC)
Summer Meeting

Nuclear Industry Check Valve
Summer Meeting
San Diego, California
June 18, 1996

Presentation

Solution to Ginna's
Component Cooling Water System
Service Water System
Check Valve Problems

Rochester Gas and Electric, Ginna Station
John C. Grzybek
Gregg E. Joss
Joseph P. Mallia

Solutions to Ginna's Component Cooling Water System (CCW) and Service Water System (SW) Check Valve Problems

ABSTRACT

Conventional swing type check valves installed in CCW and SW had persistently caused system problems when valve disk "slamming" occurred during valve closure. The check valves are located on the system's pump discharge and violently close upon flow reversal occurring at pump starts and stops. This presentation will describe the evaluation, resolution and results of the installation of nozzle type check valves.

PROBLEM DESCRIPTION

Swing type check valves were originally installed in the CCW and SW systems. Signs of excessive force from the swing check "slamming" were manifested as:

<u>CCW System</u>

... Grout cracking beneath the CCW pump/motor pads and base.
... Damage to electrical cable to the motor.
... Greater than normal vibrations caused by induced misalignment between the CCW pump and motor.
... Pressure gauges and switches becoming abnormally out of calibration.
... Delays in valve prompt closure caused inadvertent auto-starts of standby CCW pump and difficulty in passing quarterly ASME XI check valve closure tests.
... Initial "jerry rigging" backstop extension (early in plant, 1969) to force faster valve closure.
... Cracked valve seat surfaces.
... CCW heat exchanger tube fretting.

<u>SW System</u>

... SW heat exchanger tube failures corresponding to SW pump cycling activations.
... Excessive SW header displacement, >5/8".
... Worsening parts availability from manufacture (Crane), forced to customize parts.

INVESTIGATION

Resolving the "slamming" problem meant finding a means of controlling reverse flow and dissipating the energy emitted from the rapid closure.

... Explore various replacement type check valves (swing, lift, nozzle, piston, tilting disk).

... Visit to Calvert Cliffs, similar check valve installation in Brine Water (SW) system and Safety Injection (SI) system (only plant at the time that used nozzle check valves in safety related systems). Experienced problems with Hilte bolts tearing out of supports.

... Discussions with industry valve experts (NIC, ASME, USNRC, various Engineering Consultants and various Valve manufacturers).

DESIGN AND STATION TECHNICAL ENGINEERING

Design and Station Technical Engineering prepared for a new replacement style by:

... Test the existing swing check valve with instrumentation sensitive enough to detect pipe deflection, pressure surge spikes and pipe strain. Video tape of pump activation. Results were quite revealing.

CCW System performance parameters :

	Design	Operating
Flow rate (gpm)	2980	600 - 2750
Pressure (psig)	150	78 - 90
Temperature (F)	200	70 - 120

Swing check valve test results:

Lanyard, pipe displacement (in)	0.003 - 0.005	
Pressure Transducer (psig)	1,600 - 1,800	*(>10 times design, very brief)*
Strain Gage (in/in)	not detectable	

(Testing was repeated three times for statistical validation)

... Matrix evaluation weighing various objectives and relationships. Optimize relationships between cost/benefit, risk and payback period.

... Decision to install ENERTECH nozzle check valves, DRV-Z in the CCW system and DRV-B in the SW system.

... Review of the "as found" system isometrics, preparing for differences in valve characteristics.

... Review of hydraulic flow models, preparing for differences in valve Cv.

... Review of pipe stress models, preparing for differences in valve weight and Cg.

... Evaluate vertical orientation of the CCW swing check valve.

Characteristics	Existing Swing Check	Replacement Nozzle Check	
Length, end-to-end (in)	19 ½	17 ½	
End Configuration	BW	Flanged	
Weight (lb)	235	176	
Flow Coefficient (Cv)	1755	1694	
Seismic, Center of Gravity (Cg)	6"	0"	(from pipe center line)

PERFORMANCE

Installation of flanges and nozzle check valves went as planned. Performance testing of the nozzle checks using the same sensitive instrumentation was not warranted based on the valves smooth, silent and "slam-free" operation.

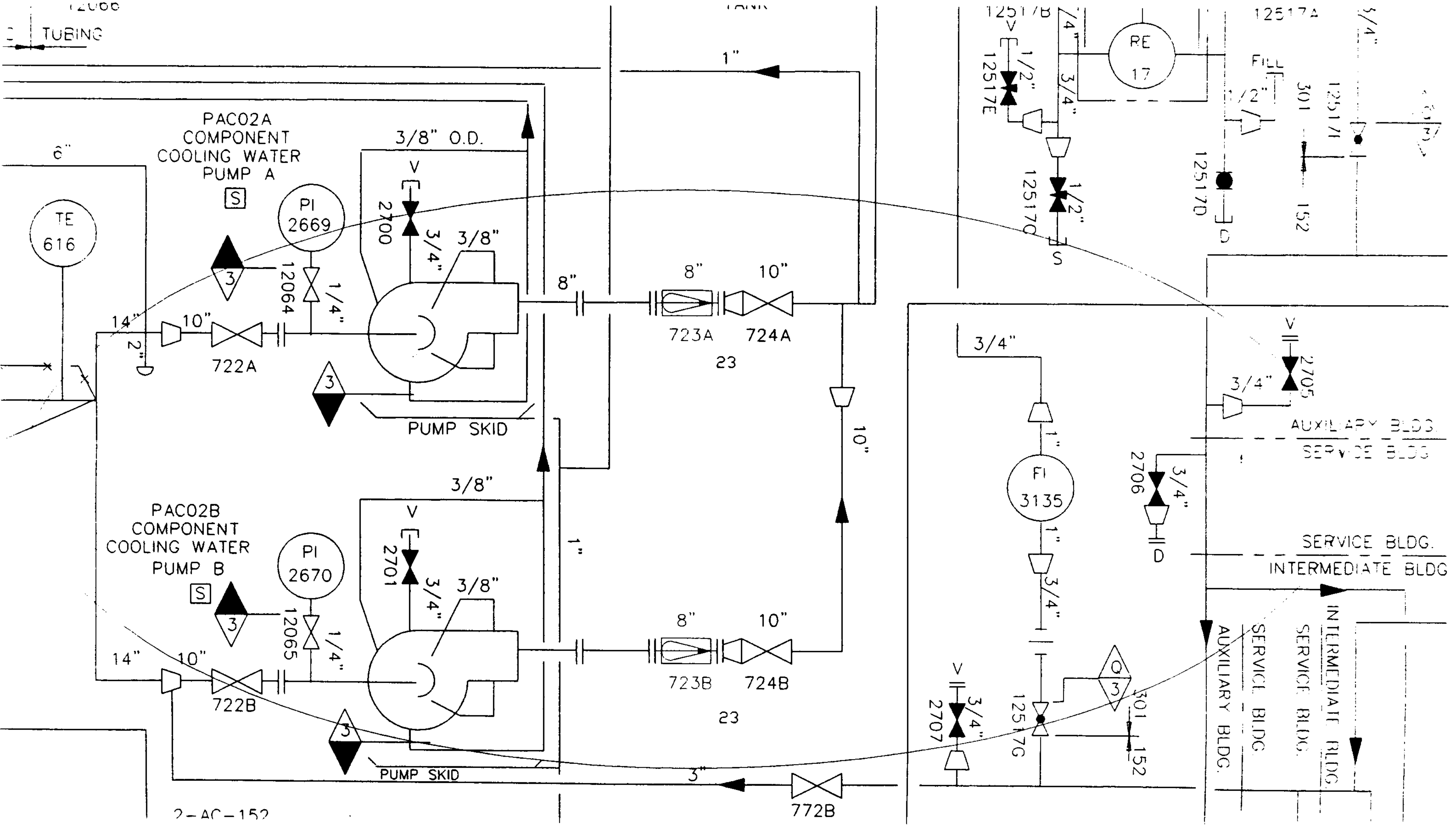
... Video tape of pump activation, for before and after comparisons.
... Subsequent operational and maintenance problems have not reappeared.

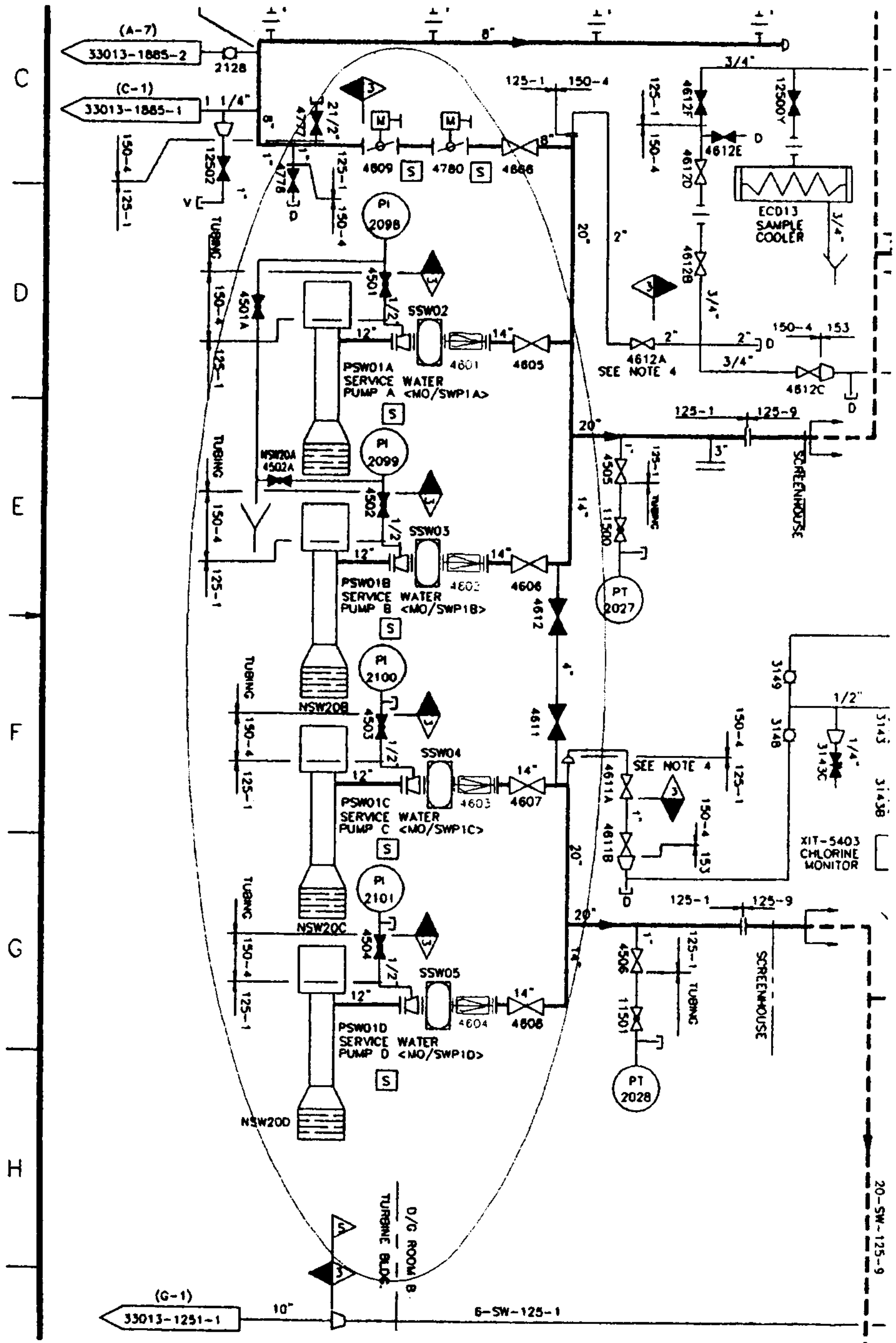
PROJECT RESULTS

The installation of the nozzle checks has solved the CCW and SW reverse flow and subsequent "slamming" problems.

... Nozzle check valves have two years of service in CCW and SW, delivering excellent performance.
... NRC recognition, written in the 1994 Outage follow up letter, of the "initiative and long term resolution of long standing problem".
... These check valves are capable of being fully tested during operation, which allowed elimination of a Ginna IST Program, NRC Cold Shutdown Relief Request.
... Elimination of Operator "work around" on CCW system where pump control switch had to be held open.
... Elimination of Operator "work around" on SW system for visual confirmation of no pump reverse rotation.
... Emerging problem of SW nozzle checks, the check valves close so tightly that a vacuum forms as the valves close and water attempts to flow back down to lake level (about 16 feet). Pump packing is beginning to wear a groove in the shaft.

N.I.C. Presentation
RGE-Ginna
Check Valve Solution





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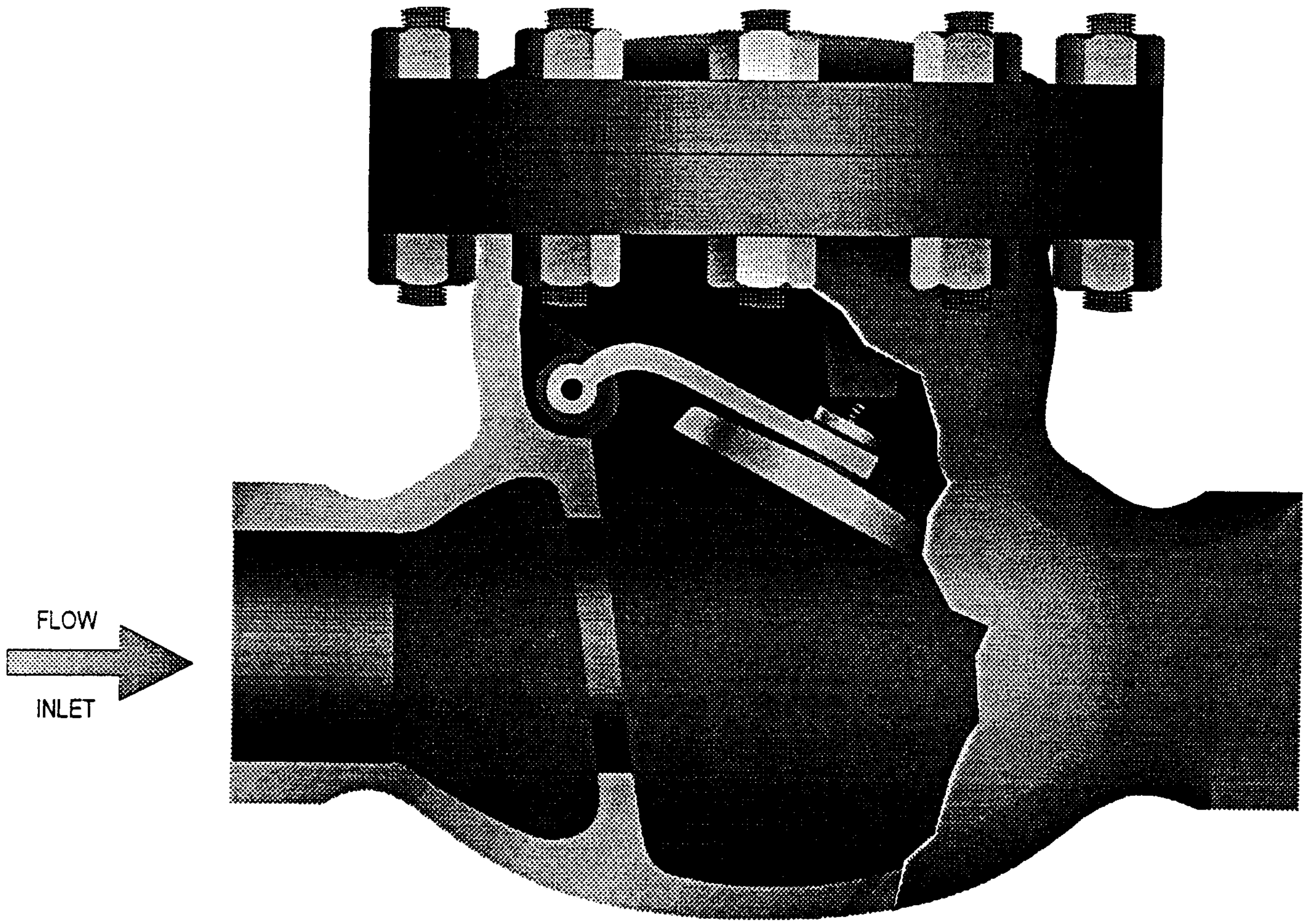
Swing Check



Forged Bolted Cover
8" Swing Check

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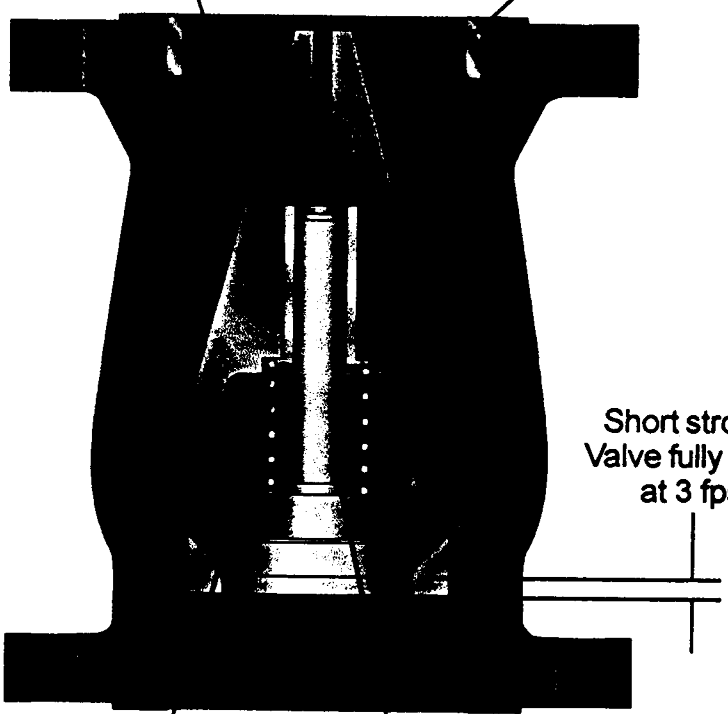
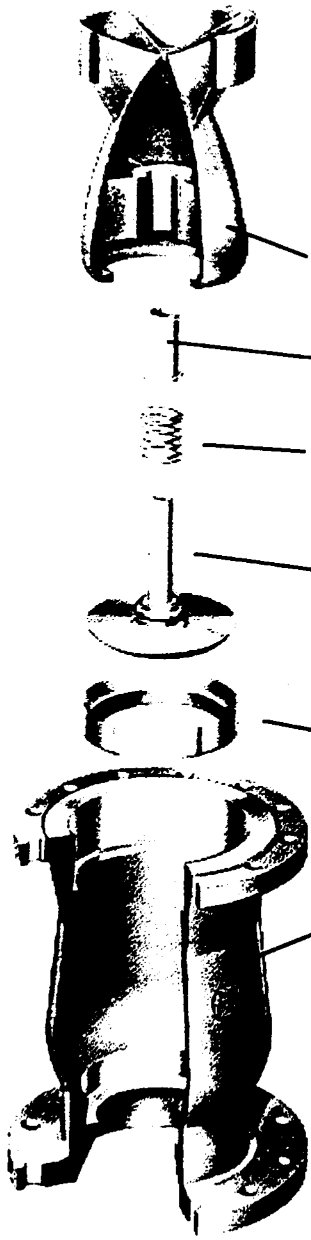
NozzleCheck DRV-Z

The SOLUTION to Ginna CCW Check Valve Problems





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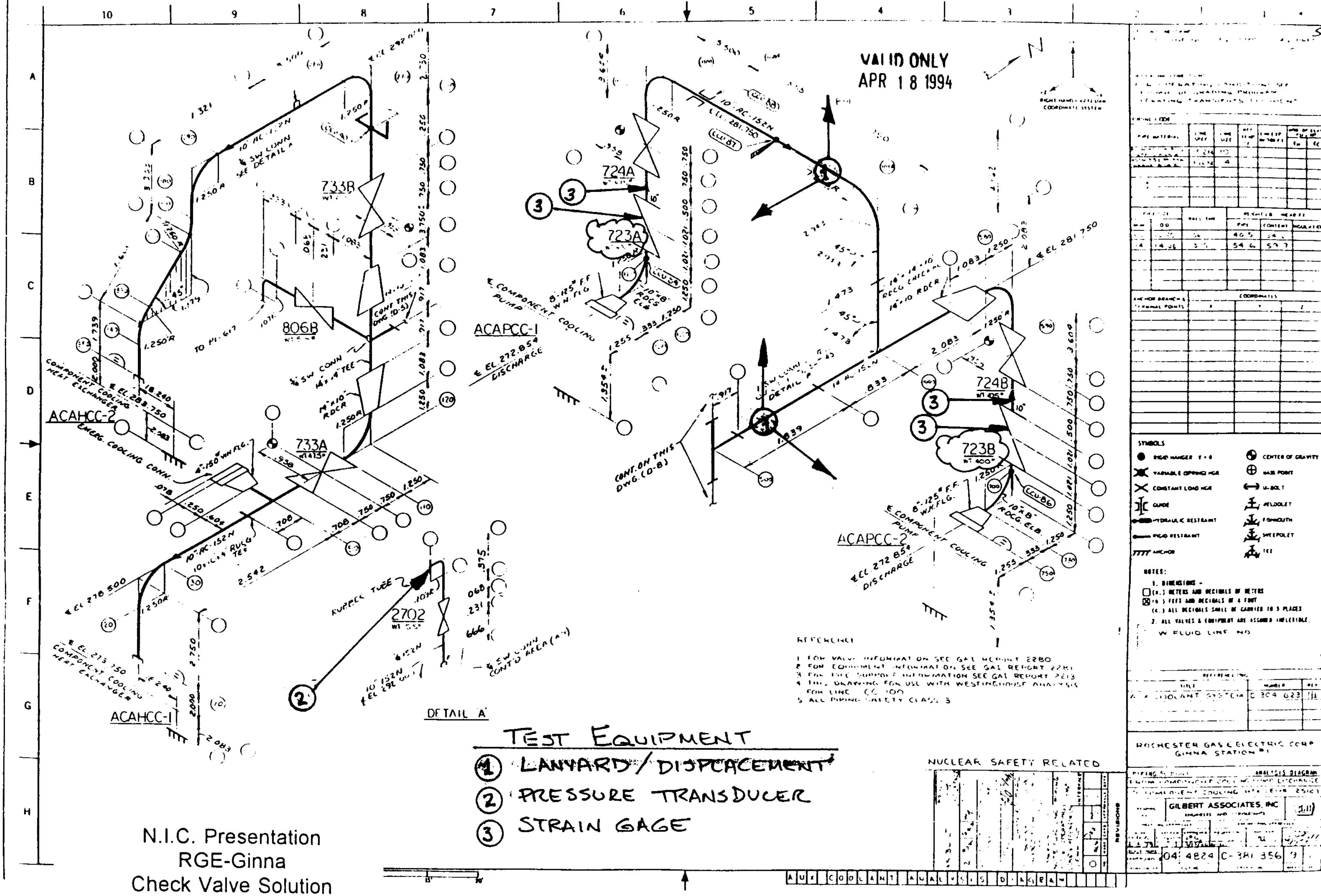

TEST EQUIPMENT

1. LANYARD / DISPLACEMENT
2. PRESSURE TRANSDUCER
3. STRAIN GAGE

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RGE-Ginna
Check Valve Solution

Swing Check Valve Replacement

(EWR5284)

Description: Check valves (swing type) are "slamming" upon closure at the CCW and SW pump discharges. This "slamming" has caused damage to the systems and surrounding equipment. Evaluate replacement of the swing checks with nozzle check.

Scope:

- o Evaluate need for change of check valve type.
- o Determine options for types of check valves.
- o Develop options and recommend the most cost effective option with acceptable risk.

Approach:

- o Evaluate most realistic actions for resolving the concerns.
- o Deliver options to management and make decision on which option to proceed.

Options: Four (4) viable options for resolving the CCW and SW check valve problems :

1) Leave the swing check valves as-is, maintain only..

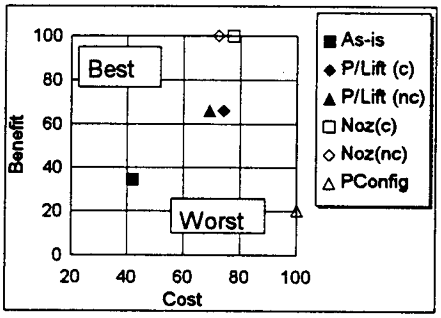
2) Replace with piston/lift type check valves.

3) Replace with nozzle type check vavles.

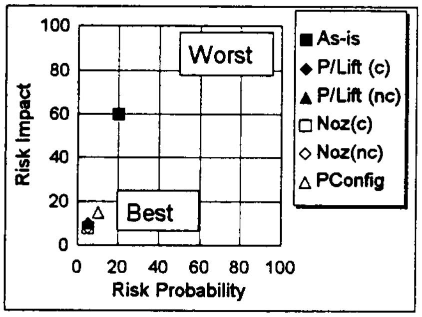
4) Change Plant piping configuration, correcting and "tuning" slamming potential.

Relationships: Evaluate the options

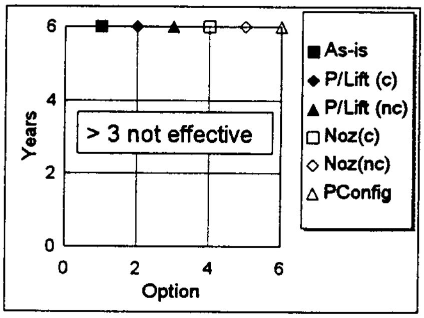
Cost / Benefit



Risk



Pay Back



Recommendation: The optimum proposed action is Option 3, replace swing check with nozzle check valves.

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N.I.C. Presentation
RGE-Ginna
Check Valve Solution

Swing Check Valve Replacement

Goal: Determine the optimized overall recommendation for check valve resolution.

Technical Objectives	note	Weight	As-is	Piston/Lift (c)	Piston/Lift (nc)	Nozzle (c)	Nozzle (nc)	Plant Config
			Achievement: low (1) ... high (10)					
Tangable								
Minimize closure time/flow reversal (WHammer)		10	3	5	5	10	10	9
Maximize Disc position (full open) at min flow		10	3	7	7	10	10	5
Maximize Cv at max/min flows		9	7	7	7	10	10	7
Minimize pressure drop and unrecoverable pressure		9	10	9	9	10	10	10
Maximize shielding of moving internal parts		9	7	8	8	10	10	7
Maximize Flow recovery/correction		9	8	8	8	10	10	8
Maximize leak tightness		8	8	9	9	10	10	8
Minimize maintenance requirements		8	1	8	8	10	10	1
Minimize Construction piping modifications		7	10	9	9	9	9	1
Minimize Maintenance repairs		7	2	6	6	10	10	2
Minimize Engineering reanalyzes		6	10	7	7	7	7	2
Minimize spare parts requirements		5	7	7	7	10	10	7
Minimize potential gasket leak paths		4	10	10	10	8	8	10
Non-Tangable								
Possess proven Design/Operation		10	10	10	10	10	10	9
Maximize Ginna first-time-use confidence		10	10	9	9	8	8	9
Maximize Valve Quality (perception)		9	9	10	10	10	10	10
Maximize confidence in Safety Related Systems		8	4	9	9	10	10	9
Maximize "Ruggedness" (perception)		8	8	9	9	10	10	8
Maximize Warrantee		7	9	9	9	10	10	9
Made in the USA		5	10	10	10	10	10	10
Technical Evaluation			1999	3818	3818	5784	5784	1178
Percent of Highest Value			34.6%	66.0%	66.0%	100.0%	100.0%	20.4%
Cost $K (through Plant end-of-life)		n/a						
Procurement (ASME Code Valves)			0	30		35		0
(Commercially Dedicated)			0		22		27	0
Engineering			15	10	10	10	10	20
Construction and Maintenance			50	75	75	75	75	135
Cost Evaluation	1		65	115	107	120	112	155
Percent above Lowest Cost			0.0%	76.9%	64.6%	84.6%	72.3%	138.5%
Percent below Highest Cost			-58.1%	-25.8%	-31.0%	-22.6%	-27.7%	0.0%
Evaluation		n/a						
70% Tech/30% Cost	2		[illegible]	[illegible]	[illegible]	76.8%	[illegible]	[illegible]

1. Not included: AFUDC, CAC, ESC or replacement power cost from forced outage.
2. The largest value indicates the most attractive recommendation.

Relationship(s)	note	n/a	As-is	Piston/Lift (c)	Piston/Lift (nc)	Nozzle (c)	Nozzle (nc)	Plant Config
			Evaluate: low (1) ... high (100)					
Cost	1		41.9	74.2	69.0	77.4	72.3	100.0
Benefit	2		34.6	66.0	66.0	100.0	100.0	20.4
Risk Probability	3		20.0	5.0	5.0	5.0	5.0	10.0
Risk Impact	4		60	10	10	8	8	15
Pay Back Period			>6	>6	>6	>6	>6	>6
>3 years is not considered cost effective								

1. Reflects the Cost.
2. Reflects the technical objectives.
3. Probability
4. Impact

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Check Valve Solution

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J Wayland

March 3, 1995

Dr. Robert C. Mecredy
Vice President, Ginna Nuclear Production
Rochester Gas and Electric Corporation
89 East Avenue
Rochester, New York 14649

Dear Dr. Mecredy:

SUBJECT: GINNA ENGINEERING INSPECTION 95-02

This letter refers to the inspection of your engineering activities, conducted by Mr. L. Prividy and Ms. L. Harrison from January 9-13, 1995, at the Ginna Nuclear Power Plant in Ontario, New York. The inspection was focused on issues important to public health and safety. It consisted of an independent evaluation of activities in the on-site engineering groups, including their involvement in plant modifications and the identification and technical resolution of problems. The preliminary inspection findings were discussed with Mr. T. Marlow and other members of your staff on January 13, 1995.

The modifications, which our inspectors reviewed, were found to be well engineered. Temporary modifications were observed to be appropriately controlled. We acknowledge your initial efforts to pilot a new plant change review process to improve design phase inputs and interfaces. Your service water system self assessment was found to be thorough, providing constructive comments for improving system performance.

No reply to this letter is required, and your cooperation with us in this matter is appreciated. We would be pleased to discuss the conclusions of this inspection with you.

Sincerely,

Eugene M Kelly

Eugene M. Kelly, Chief
Systems Section
Division of Reactor Safety

Docket No. 50-244

Enclosure: Inspection Report No. 50-244/95-02

N.I.C. Presentation
RGE-Ginna
Check Valve Solution

DETAILS

1.0 INSPECTION SCOPE

The objective of this inspection was to evaluate the engineering organization's performance of various activities, including the development of plant modifications and the identification and technical resolution of problems. NRC Inspection Procedure 37550, Engineering, was used for inspection guidance. The inspection was conducted at the Ginna plant to include reviews of the onsite engineering activities since an earlier inspection had emphasized design engineering work at the corporate office. Areas inspected included reviews of the recent corporate reorganization, plant major and minor modifications, temporary modifications, self and independent assessments of engineering activities, the system engineer program, engineering work backlog, technical staff training, and the operational experience review program. Also, the inspectors reviewed the status of unresolved item 94-26-01, regarding potential erosion in a throttle valve.

2.0 INSPECTION FINDINGS

2.1 Engineering Reorganization and Work Control

A corporate reorganization established a new onsite engineering support group effective on November 1, 1994, to administer the details of plant modifications. This group reports to the corporate engineering organization. The licensee expects that this reorganization item will enable the system engineering group to develop more quickly, since the system engineers will not be responsible for the details of plant modification work.

A good effort by the licensee in prioritizing engineering work items was observed by the inspector. The licensee appeared to have reasonable goals established for controlling major engineering work items, including engineering work requests (EWRs), technical staff requests (TSRs), and vendor technical manual change requests.

2.2 Major Modifications

The inspectors reviewed selected EWRs to verify that changes to the station's safety systems were supported by appropriate design criteria, design input requirements, and design analyses. The inspectors performed this review to confirm that the design changes did not adversely affect the function of safety-related systems.

2.2.1 EWR #5284B - Service Water (SW) and Component Cooling Water (CCW) Pump Discharge Check Valves

The engineering work reviewed by the inspector, in support of this modification, was performed well with some examples as follows. A thorough search for a better check valve demonstrated design engineering's initiative to achieve a good long-term resolution to an old problem. Also, engineering properly analyzed the impact on system performance upon recognizing the higher pressure drop characteristics of the new check valves.

This modification involved the replacement of the original swing check valves located at the discharge of each SW and CCW pump with nozzle type check valves. This different design check valve was selected to provide smoother operation during pump startup and tighter shutoff for any idle pumps. Initial operating experience in both systems has shown that these check valves have performed well in satisfying both of these requirements.

The inspector noted that the nozzle check valves had a larger hydraulic resistance than the original check valves. The impact of this change on SW system performance had been evaluated as documented in Design Analysis DA-ME-94-101. The licensee concluded in this analysis that no significant adverse changes were expected on overall SW flow to safety related components during safety injection or recirculation operation. The inspector also reviewed the post modification test result that demonstrated satisfactory operation prior to return to service.

N.I.C. Presentation
RGE-Ginna
Check Valve Solution

ENERTECH
Curtiss-Wright Flow Control

White Paper

Success Revisited
Solving Performance Problems
Using NozzleCheck Valves

Presented 2000
ASME Pump & Valve Symposium

Success Revisited
Solving Performance Problems
Using NozzleCheck Valves

ASME Pump & Valve Symposium - Summer 2000

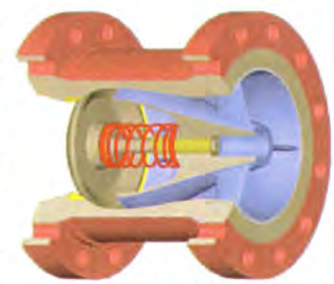
Presented By:

Gregg Joss
Inservice Test Coordinator
Rochester Gas & Electric
Ginna Station

Jim Zulawski
Performance Monitoring Supervisor
Rochester Gas & Electric
Ginna Station

Abstract

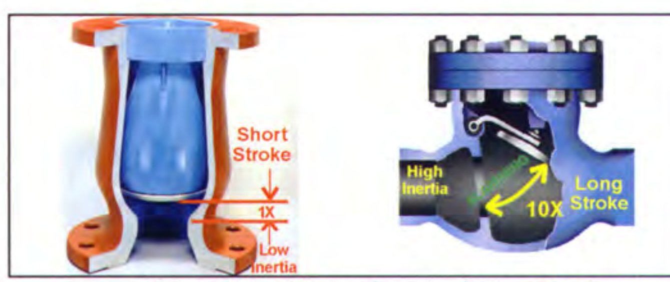
For 25 years, Rochester Gas & Electric's Ginna Nuclear Power Plant had conventional swing check valves installed in its Component Cooling Water (CCW) and Service Water (SW) systems which resulted in persistent system problems when valve disk "slamming" occurred during valve cycling. These check valves are located at the discharge of the CCW and SW pumps and are in parallel configurations. During routine pump swaps, the valves closed violently when flow reversal occurred. This presentation describes the role the nozzlecheck replacement valves played in achieving a permanent and highly successful problem resolution.



NozzleCheck Valve

Introduction

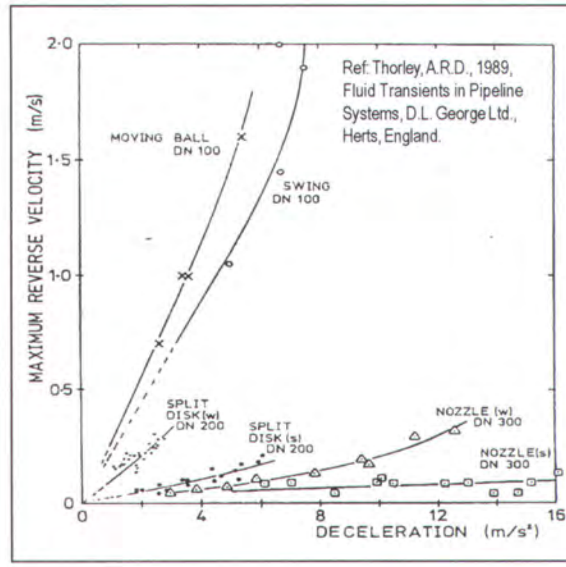
Swing type check valves were originally installed in Ginna's CCW and SW systems at the discharge of the parallel configured pumps. Signs of excessive force from swing check valve "slamming" and inappropriate valve application were manifested as:



Short Stroke: NozzleCheck vs Swing Check

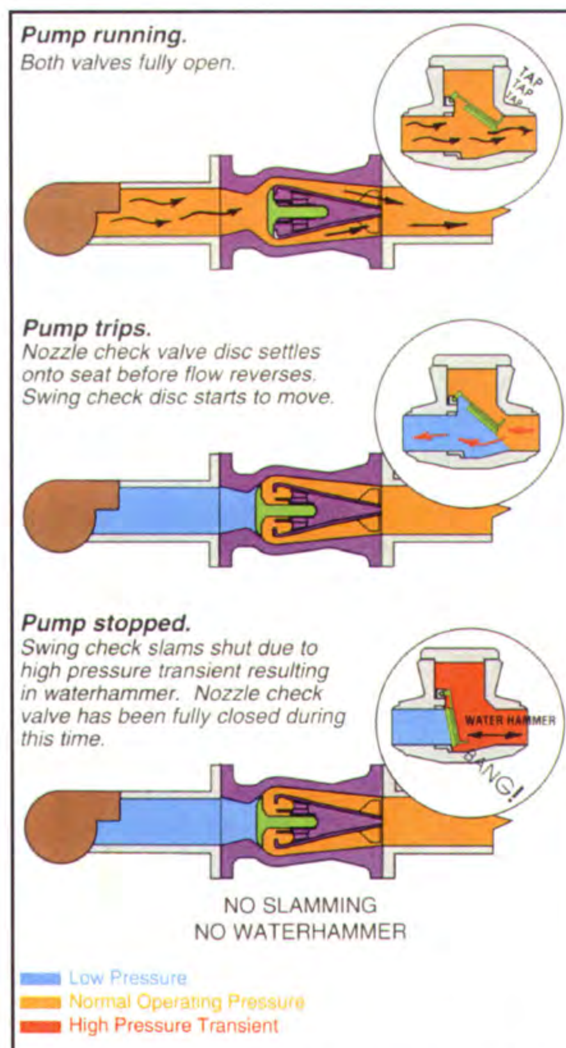
1) CCW System
   - Grout cracking beneath the CCW pump/motor concrete pads and base
   - Damage to motor electrical power cable
   - Abnormally high pump and motor vibrations caused by induced misalignment between the CCW pump and motor
   - Support system pressure gauges and switches frequently found "out of tolerance" during periodic calibrations
   - Delays in valve "prompt closure" resulted in inadvertent auto-starts of the standby CCW pump
   - Delays in valve "prompt closure" often resulted in difficulty passing quarterly ASME Section XI check valve exercise/closure tests
   - Cracked valve seat surfaces
   - CCW heat exchanger tube fretting

2) SW System
   - SW heat exchanger tube failures corresponding to SW pump cycling activities (Containment Recirculation Fan and Motor Coolers, Standby Auxiliary Feedwater Pump Room Coolers)
   - Excessive SW header piping displacement (>_ ")
   - Poor parts availability from the valve manu facturer forced use of customized parts



Investigation

1) To resolve the "slamming" problem, a means of controlling reverse flow, ensuring prompt closure and dissipating the energy emitted from the rapid closure was necessary. The following investigation activities took place in an attempt to identify potential corrective action(s).
   - Explored various replacement type check valve designs (swing, lift, nozzle, piston, tilting disk)
   - Visited Calvert Cliffs which already had nozzle check valves installed in the Brine Water (a.k.a. Service Water) and Safety Injection (SI) systems.
     - At that time, Calvert Cliffs was the only plant in the United States to have nozzle checksinstalled in safety related systems.
     - The SI application solved very severe Hilte bolts/support degradation problems induced by check valve slamming events.
   - Conducted in-depth discussions with various industry valve experts such as:
     - Nuclear Industry Check Valve (NIC) group
     - ASME
     - USNRC
     - Multiple Engineering consultants
     - Numerous valve manufacturers



Operational Comparison of NOZZLECHECK and Swing Check Valves.

Technical Approach

1) Ginna Station Design and Technical Engineering personnel prepared for a new style replacement valve by:

- Testing the existing swing check valve and CCW piping by monitoring test parameters with instrumentation sensitive enough to detect pipe deflection, pressure surge spikes and pipe strain.

2) Comparing the CCW system design and operating performance parameters:

	Design	Operating
Flow Rate (gpm)	2980	600-2750
Pressure (psig)	150	78-90
Temperature (F)	200	70-120

3) Obtaining swing check valve test data during CCW pump activations:
   Note 1: The actual CCW test was videotaped to provide audio and visual evidence as well.
   Note 2: Testing was repeated three times for statistical validation.
   - Lanyard (pipe displacement (in)) = 0.003-0.005
   - Transducer (pressure surge spikes (psig)) = 1600 - 1800 (> 10 times expected design magnitude, very brief spikes)
   - Strain Gauge, (pipe strain (in/in)) = not detectable

4) Using a matrix evaluation model to evaluate and assign weighting factors to various objectives and potential solutions
   - The intent of the matrix is to optimize relationships between cost/benefit, risk and payback period.

5) Reviewing the "as-found" system isometrics, preparing for differences in replacement valve characteristics.

	Existing Swing Check	Replacement NozzleCheck
Length, end-to-end (inches)	19 1/2"	17 1/2"
End configuration	Butt Weld	Flanged
Weight (pounds)	235	176
Flow Coefficient (Cv)	1755	1694
Seismic, Center of Gravity (Cg, inches from pipe centerline)	6	0

6) Reviewing hydraulic flow models, preparing for differences in replacement valve Cv.

7) Reviewing pipe stress models, preparing for differences in replacement valve weight and Cg.

8) Evaluating acceptability of vertical orientation of the CCW swing check valve.

Final Decision

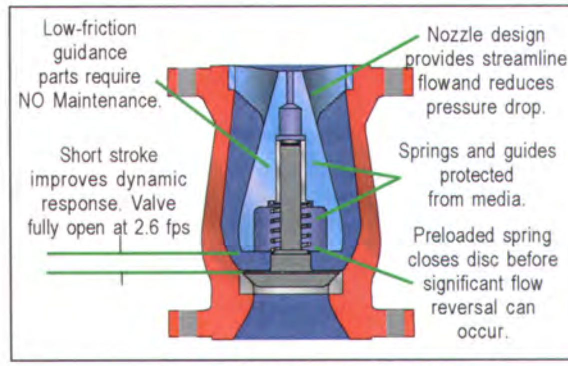
1) A decision was made to install nozzlecheck valves in the CCW and SW systems.

Valve Installation/Testing

1) Installation of flanges and nozzlecheck valves went as planned
2) Post-installation performance testing of the new valves using the same sensitive instrumentation was not warranted based on their smooth, silent and virtually "slam-free" operation.
3) CCW and SW pump activations were videotaped to provide before and after comparisons.

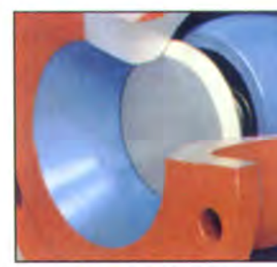
Conclusions

1) Installation of the nozzle check valves first in the CCW system and later that same year (1994) in the SW system, has totally solved all reverse flow and "slamming" problems.
2) Other notable post-installation points of interest include:
   - In the NRC's 1994 RFO on-site inspection report, Ginna was recognized for "good initiative and long term resolution of a long standing problem".



NozzleCheck Valve Features

   - All six nozzlecheck valves are fully capable of being tested during plant operation which allowed elimination of a Ginna Station IST Program, NRC Cold Shutdown Relief Request.
   - An Operator "work around" was eliminated since the CCW pump control switch no longer needed to be held to the "off" position for 5 seconds after securing the pump (afforded sufficient time for the pressure spike to dissipate without auto-starting the just secured pump).



Metal-to-Metal Tight Shutoff

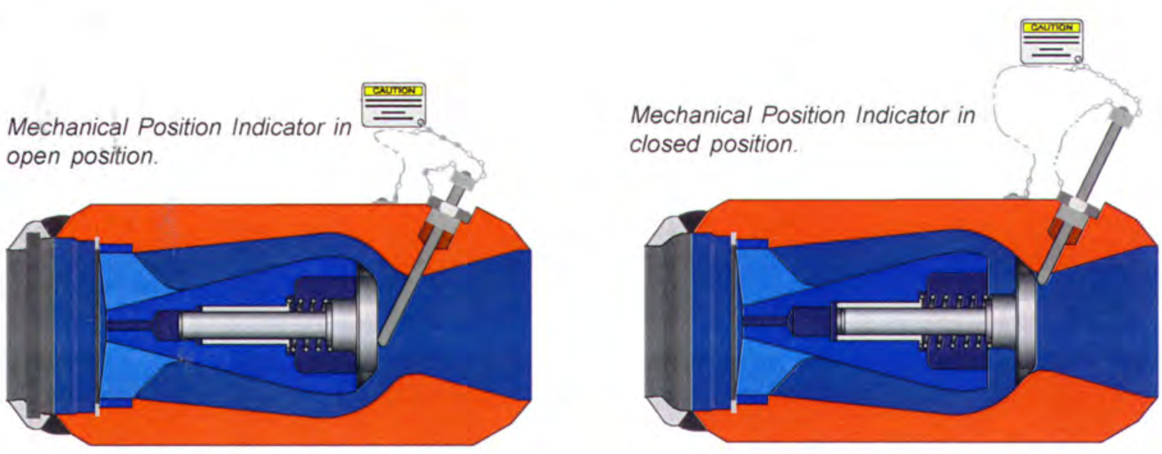
   - Emergent problems with SW nozzlecheck valves:
     - Valves close so tightly that a vacuum forms as the valves close and water attempts to flow back down into the screen bay (~16 feet) to existing lake level. Resultant pump packing consolidation actually caused shaft wear (grooves) and required installation of individual vacuum breakers on pump side of the valve.
     - There have been no recurrences of the shaft wear since the vacuum breakers were installed.
3) Following five years of continuous operating service, one CCW and three out of four SW nozzlecheck valves have been removed and inspected to assess valve wear and overall condition, Inspection results:
   - V-723B-CCW Pump "B" Discharge (March 1999)
     - 100% freedom of movement
     - No evidence of seat or internal valve wear, valve plug and seat condition described as "like new"
     - No evidence of degradation internal or external to the valve
     - Preventative Maintenance (PM) frequency extended from 5 years to 7 years
   - V-732A-CCW Pump "A" Discharge (N/A)
     Based on results of V-723B PM inspection, V-732A PM inspection deferred to 2001 and frequency extended from 5 years to 7 years.

- V-4601-SW Pump "A" Discharge (February 1999)
  - 100% freedom of movement
  - No evidence of seat or internal valve wear, valve plug and seat condition descr ibed as "like new"
  - No evidence of degradation internal or external to the valve
  - Cleaned inner body walls
- V-4603-SW Pump "C" Discharge (February 1999)
  - 100% freedom of movement
  - No evidence of seat or internal valve wear, valve plug and seat condition described as "like new"
  - No evidence of degradation internal or external to the valve
  - Cleaned inner body walls
- V-4604-SW Pump "D" Discharge (January 2000)
  - 100% freedom of movement
  - Lapped seat and disc to obtain 360 positive contact (valve is not a Category A IST component)
  - Replaced all 3 disc springs as a preventative measure
  - Cleaned inner body walls
- Preventative Maintenance frequency (5 years) not changed for V-4601, V-4603 and V-4604 due to evidence of sludge buildup on inner valve body walls caused by "raw water" (Lake Ontario) service environment. This condition did not affect valve performance.
- V-4602-SW Pump "B" Discharge slated for inspection September 2000.

Future NozzleCheck Endeavors at Ginna Station

1) Based on current problems with swing check valves located in the steam supply lines to the Turbine Auxiliary Feedwater Pump (sluggish and often incomplete closure, large backstop slam and excessive corrective maintenance), a modification to replace these valves with normally closed nozzlecheck valves will be performed during the September 2000 Refueling Outage.

2) Efforts to improve check valve performance (i.e. back leakage) in the Safety Injection system include consideration of nozzlecheck valves as potential problem solvers.

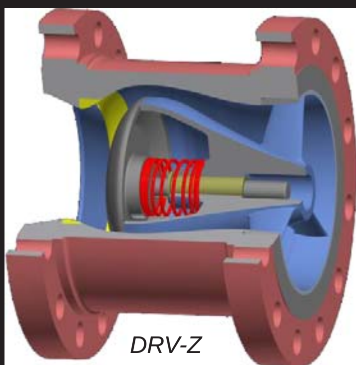


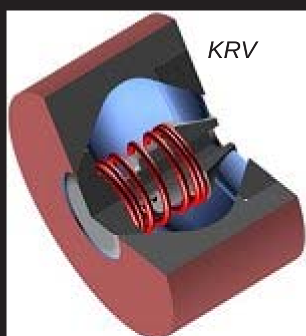
Normally Closed NOZZLECHECK Valves with Position Indicator for operational verification in-place. Meets 89-04 IST requirements (reduced outage time by requiring the removal of only one pipe plug.).

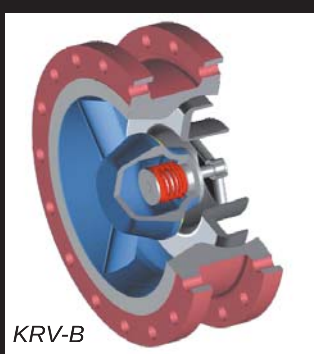
NozzleCheck Valves

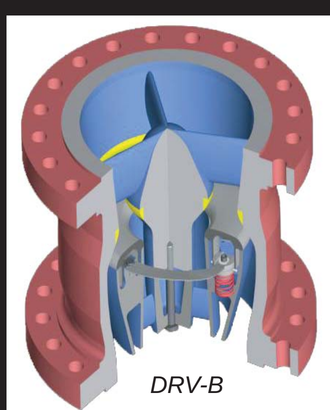
for use in Nuclear Power Applications
ASME Section III, Safety-Related, Commercial

ENERTECH
Curtiss-Wright Flow Control Corporation


DRV-Z


KRV


KRV-B


DRV-B

Model DRV-Z and ERV-Z

- **High capacity valve for small bore applications**
  - Size: 3/4” to 10”
  - ANSI Pressure Class: 150 -2500
  - End Connections: socket weld, buttweld, flange
  - Drop-in replacement for standard swing check valves
  - Available in standard short F-F dimensions

Model KRV

- **Wafer style valve with short face-to-face dimensions**
  - Size: 1” to 10”
  - ANSI Pressure Class: 150 -2500
  - End Connections: wafer
  - Short face-to-face makes it an excellent drop-in replacement for duo check valves
  - Lightweight, economical, tight shutoff design

Model KRV-B

- **Short face-to-face for large bore applications**
  - Size: 12” to 72”
  - ANSI Pressure Class: 150 -2500
  - End Connections: flange, lug wafer, wafer, buttweld
  - Drop-in replacement for obsolete large bore duo check valves
  - Excellent dynamic performance

Model DRV-B and ERV-B

- **High flow for large bore applications**
  - Size: 10” to 88”
  - ANSI Pressure Class: 150 -2500
  - End Connections: flange, buttweld
  - High Cv and excellent dynamic performance
  - Ring-style disc for low friction cycling

Axial Flow Control Valves

- **Designed for dirty, high capacity, severe service applications**
  - Size: 6” to 48”
  - ANSI Pressure Class: 150 -2500
  - End Connections: Flange
  - Balanced design to reduce torque requirements

Support Services

- Outage Services
  - Field inspection, maintenance and repair
  - Site-specific O&M training
- Enertech Applications Engineering
- Spare Parts

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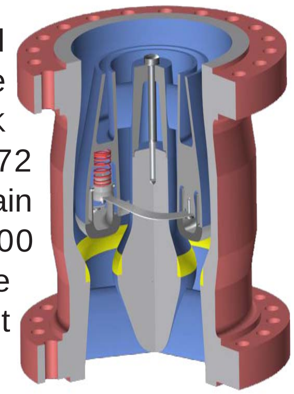

The NozzleCheck valve was first developed and patented in 1935 to mitigate system damage caused by water hammer. The first NozzleCheck valve was installed in a nuclear plant in 1972 to eliminate damaging transients caused by Main Feed pump trips. Since that time, over 800 NozzleChecks have been selected to replace conventional check valve designs in the most challenging nuclear plant applications.

The NozzleCheck valve has been manufactured under Enertech's N-Stamp and Appendix B Program in support of the Nuclear Industry since 1992. ASME Section III Class 1, 2, and 3, safety-related and commercial valves are available as a cost-effective alternative to resolving issues such as:

- Accelerated wear of swing checks and dual plate check valves caused by disc oscillation at low flow
- Frequent maintenance required due to soft seat degradation by check valves using elastomers to obtain tight shutoff
- Minimizing weight and face-to-face dimensions on new check valve applications
- Obsolescence of first-generation check valve designs
- Water hammer as a result of pump trips
- Water hammer caused by gas recoil after pump start evolutions
- Packing and bonnet leakage
- LLRT and seat leakage test failures due to slam-induced seat degradation and misalignment
- The need to function in horizontal, vertical-flow-up or vertical-flow-down applications

Special designs available:

- Normally open NozzleCheck valves for Auxiliary Feed Pump Turbine Steam Supply applications to eliminate wear induced in swing checks due to fluctuating steam generator pressures
- External position indication to support ISI/IST programs
- Soft-seat options for low pressure seat leakage requirements
- Vacuum breakers for Service Water and HPCI/RCIC Turbine Exhaust

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NozzleCheck valve vertically oriented and located above elbow.

NozzleCheck valve in Service Water System at River Bend NPP.

NozzleCheck valves installed in cooling system at nuclear power plant in France.

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