UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

HYDRAULIC STUDIES OF A 3-INCH QUICK-OPENING SLOW-CLOSING AIR VALVE

Hydraulics Branch Report No. Hyd-488

DIVISION OF ENGINEERING LABORATORIES

OFFICE OF ASSISTANT COMMISSIONER AND CHIEF ENGINEER
DENVER, COLORADO

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CONTENTS

FIGURES

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Office of Assistant Commissioner and Chief Engineer
Division of Engineering Laboratories
Hydraulics Branch
Structures and Equipment Section
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Report No. Hyd-488
Compiled by: H. T. Falvey
Checked by: W. P. Simmons
Reviewed by: J. W. Ball
Submitted by: H. M. Martin

Subject: Hydraulic studies of a 3-inch quick-opening, slow-closing air valve--Friant-Kern Canal Distribution Systems, Central Valley Project, California.

PURPOSE

The purpose of this study was to investigate the hydraulic characteristics of a 3-inch quick-opening, slow-closing air valve and the effect of modifications to increase the speed of closure under low heads and to decrease the water spillage.

CONCLUSIONS

1. The discharge coefficient, $C_d = \frac{Q}{A\sqrt{2g\Delta H}}$, was 1.04 for the valve 100 percent open (Figure 3A). The area, A, is based on the nominal size of the valve.

2. The total closing forces were determined for the whole range of air valve openings (Figure 3B). For use with a riser pipe the ordinates of Figure 3B must be increased by 0.8h, where h = height of riser, except for the fully closed position for which the ordinate remains unchanged.

3. The force fluctuations are negligible for openings less than 50 percent. For openings greater than 50 percent the fluctuations become very large (Figure 4).

4. Closing times for normal conditions of a pipeline filling and for conditions of a surge in the pipeline were determined for various differential heads. Adding a riser pipe reduced the closing time (Figure 5).

5. The dashpot oil must have a Saybolt viscosity in the range 80-90 at 100° F in order for the quick-opening, slow-closing air valve to close in 30 seconds under a 100-foot head differential at normal operating temperatures.

6. Neither the heavy counterweight supplied by the manufacturer, nor the light one being used in the field allowed much latitude in adjustment. An intermediate weight would be required for greater latitude in adjustment.

7. The valve would open 100 percent by vibrating the valve body or lightly tapping the spindle when the distance between the heavy counterweight and the lever hub was 0.10 foot. The valve would open freely to 100 percent without the riser when the distance was 0.05 foot, but would only open 55 percent with the riser. The light counterweight had to be used before the valve with the riser would open freely to the 100-percent opening.

8. The closing time increased approximately 1 minute under a 7-foot head differential when using the light counterweight set so the valve, with the riser, opened freely to 100 percent.

9. A piezometric head differential of 2.5 feet was required to close the valve with a 15-foot riser pipe. Without the riser, a differential of 5.0 feet was required.

10. When operating under low heads, a slight reduction in closing time was obtained by sealing the bottom of the valve float (Figure 5).

11. Time was not available for sufficient tests to determine the effect of limiting the valve opening as suggested by the manufacturer. Therefore, no definite conclusions were reached.

12. Operation under low heads can be improved by reducing the friction in the moving parts of the air valve, such as providing lubrication for the counterweight lever hub bearings.

ACKNOWLEDGMENT

This study and report is a result of cooperative efforts of the Technical Engineering Analysis, Mechanical, and Hydraulics Branches of the Assistant Commissioner and Chief Engineer's Office in Denver, Colorado. The tests were facilitated by the willing cooperation of the Tea Pot Dome Water District, who supplied the 3-inch quick-opening, slow-closing air valve to the laboratory.

INTRODUCTION

A quick-opening, slow-closing air valve is a mechanical device that protects a pipeline from collapsing under a vacuum or from bursting due to high-pressure surges. These valves are normally placed at the high points in pipelines. Then, when the water or pressure level drops below the high points, the valves relieve the negative pressures by opening quickly to admit air into the system. Extreme positive pressures can occur as the pumps restart. Air from the pipelines is expelled through the valves and the water columns rejoin. A quick closure of the valves at this time would result in excessive pressures due to the rapid change in momentum of the water columns. Therefore, the valves close slowly to allow a gradual change in momentum of the water columns and thereby reduce the high-pressure surges. In the field, the closing rate is set to be slow enough to prevent excessive pressures from being generated and yet rapid enough to prevent excessive quantities of water from being wasted. Generally, a 30-second closure under the maximum head is considered satisfactory.

The principal features of this type of valve are (Figure 1A):

(1) An inverted bucket or float which provides buoyancy to help close the valve.

(2) A shaft connecting the float to a dashpot.

(3) An oil-filled dashpot which controls the closing and opening of the valve. The closing rate is varied by restricting the amount of oil that passes through an adjustable needle valve. The opening rate is controlled by a spring-loaded flap valve which allows free flow of oil from the fixed piston assembly into the traveling cylinder so the downward movement of the float is relatively unrestricted.

(4) A lever and counterbalance assembly to balance the dead weight of the moving parts.

Quick-opening, slow-closing air valves have been used successfully in many high-pressure systems and perform satisfactorily under the operating conditions for which they were designed. However, in the Tea Pot Dome Water District, on Lateral 99.4, between Pumping Stations T2 and T3, several quick-opening, slow-closing air valves failed to close in reasonable lengths of time. This failure occurred while the valves were operating with piezometric head differentials of less than 15 feet as the pipeline filled. During this time, excessive amounts of water were wasted, and the storage wells provided for the waste water were flooded. However, after the lines were filled and the full operating head developed, the air valves functioned satisfactorily.

As a result of the valves failing to close under low heads, and because of the need to investigate the hydraulic characteristics of the valve in order to develop remedial measures, one of the field valves was delivered to the Denver Office for testing.

LABORATORY APPARATUS

The air valve was mounted on a 3-inch riser pipe that was welded to the 8-inch supply main (Figure 1B). Eight-inch-diameter regulating valves were placed upstream and downstream from the air valve so the pressure and discharge through the air valve could be varied. Two separate waste pipes were provided for connection to the valve discharge port. The original hydraulic characteristics of the valve were determined with a horizontal pipe which was connected to the valve discharge port and led to the laboratory channel. The effect of the riser was studied by replacing the horizontal pipe with a 5-inch vertical pipe, 14.85 feet high.

The closing force was measured by means of a brass proving bar whose overall dimensions were 1/4 by 1-1/2 by 9-1/2 inches (Figure 2). Enlarged sections at the ends of the bar acted as fixed supports. A boss in the center of the bar provided for transfer of load from a threaded rod to the proving bar. This bar was mounted on top of the valve after first removing the oil and fixed piston assembly from the dashpot. The threaded rod, passing through the center of the proving bar, was set on an indentation in the center of the traveling cylinder. This rod was used to adjust the valve opening and to prevent the air valve from closing. The counterweight assembly and the closing forces prevented the valve from opening. The closing force was transmitted from the float through the shaft, the traveling cylinder, and the threaded rod to produce deflections in the proving bar. These deflections were measured with four SR-4, Type A-5-1, Baldwin strain gages which were connected electrically in a bridge circuit. By calibrating the bar with dead weights, the relationship between strain gage reading and closing force was determined. The strain gage readings, recorded on a 150-Sanborn Recorder, gave a continuous trace of the instantaneous closing forces. The maximum force that could be measured with the bar, without exceeding the proportional limit of the brass, was 500 pounds.

All pressures in this report are given in feet of water and are referenced to the centerline of the 5-inch discharge port. Pressures upstream from the air valve were measured with a direct reading, high-head, mercury-filled manometer. The downstream pressures were measured with a U-Tube mercury manometer (Figure 1). Discharge measurements were made with calibrated venturi meters mounted permanently in the laboratory. All tests were conducted without the small vacuum and air release valves which were supplied to the Denver Office with the air valve. A 1/2-inch plug was placed in the hole where the appended valves had been connected. The 1/2-inch drain plug was left open.

THE INVESTIGATION

Discharge coefficients and closing forces for various openings of the valve and closing times for various piezometric head differentials were obtained. The effects of the counterweight adjustment, the addition of a riser pipe, and sealing the bottom of the float were also determined. In addition, analytical consideration is given to the effect of limiting the valve opening.

Discharge Coefficients

Discharge coefficients were obtained at approximately 10-percent increments of the valve float's vertical travel. These coefficients are based on the equation:

$$Q = C_d A \sqrt{2g\Delta H}$$

where Q = discharge in cfs

C_d = discharge coefficient

A = nominal area of the 3-inch valve = 0.0492 feet^2

ΔH = piezometric head differential across the valve, measured at the centerline of the discharge port, in feet.

The discharge coefficient with the air valve 100-percent open was found to be 1.04. Discharge coefficients for the entire range of openings are given in Figure 3A.

Closing Forces

The total force tending to close the air valve consists of the following components: a buoyant force, an impulse force, a drag force, an uplift force, and a hydrostatic force. The buoyant force was found to be essentially constant. Impulse, drag, and uplift forces were dynamic and functions of the valve opening and differential head across the valve. The hydrostatic force was a function of the piezometric head and dependent upon whether or not the air valve was open or fully closed.

The buoyant force is equal in magnitude to the weight of water displaced by the float and that part of the shaft which is below the main guide bushing. If the float were completely filled with air, 0.23 cubic foot of water would be displaced. This amounts to a buoyant force of 14.4 pounds. Of this 14.4 pounds, 11.1 pounds are due to the weight of water displaced by air in the float. However, the jetting action of the 3-inch stream entering through the bottom of the valve, compression of the air in the float, and possibly absorption of the air during prolonged periods of operation, all tend to reduce the volume of air in the float. Therefore, the buoyant force will be less than 14.4 pounds unless the open bottom of the float is sealed. The weight of the float and shaft should not be subtracted from the buoyant force because the counterweight tends to balance their dead weight.

An impulse force is created by the 3-inch-diameter jet of water which enters through the bottom of the valve and impinges on the float. The change in the direction of flow results in a great deal of turbulence at large valve openings. Obtaining an accurate value for the magnitude of the impulse force by analytical means is presently impossible because the flow paths in and around the float are not known. The flow pattern obviously changed as the valve opening became smaller. For the 50-percent opening, closing force fluctuations were negligible.

The drag force is composed of friction drag around the outside of the cylinder and of form or body drag due to the shape of the float. The friction forces can be predicted with a reasonable degree of accuracy. However, further tests would be necessary to accurately determine the body drag coefficients. Both of the drag forces will vary with the differential head, but their order of magnitude is small when compared with the total closing force.

The uplift force in the air valve is analogous to the downpull force in a gate. The water, as it passes through the restricted area around the valve seat, increases in velocity. This reduces the pressure near the seat creating a differential that results in an uplift force on the float. In order to estimate the magnitude of the uplift force, pressure coefficients would have to be known on the top and bottom surfaces of float for various valve openings. Time did not permit a determination of these coefficients.

The hydrostatic force, when the air valve is open, is a result of the piezometric pressure acting over a circular area whose diameter equals the diameter of the valve stem. When the valve is closed the hydrostatic force is a result of the piezometric pressure acting over a circular area whose diameter equals the inside diameter of the valve seat. Thus, the hydrostatic force due to the piezometric pressure acting over an area equals 0.8h, in pounds, with the air valve open and $10.0\,\Delta H$ with the valve closed. ΔH is the piezometric head differential across the valve and h is the height of the riser, both measured in feet.

Because of the difficulties in measuring or estimating the magnitude of the dynamic force components, it was decided to measure the total closing force by means of a proving bar attached to the valve body. Runs were made without a riser pipe at approximately 20-percent increments of the valve travel for various differential heads. For each valve opening, a curve of closing force versus piezometric head differential was plotted. A compilation of these plots to give the total closing force measurements for the whole range of air valve openings is given in Figure 3B.

For use with a riser pipe, the ordinates of the constant head differential curves (Figure 3B) must be increased by 0.8h for each valve opening except for the fully closed position. The ordinate for the fully closed position should not be changed because the water in the riser drains out through the main guide bushing and the 1/2-inch drain plug.

The dynamic force fluctuations were so large at the 100-percent opening that an accurate determination of the mean closing force was impossible. However, at the 85-percent opening, the mean closing force was well defined; and with the air valve 50 percent open, the fluctuations were negligible (Figure 4). These fluctuations explain an apparent paradox that occurs during a valve closure under certain low head differentials. The valve begins closure but stops with the valve still 60- to 75-percent open. The apparent mean closing force is greater at the 60- to 75-percent openings than with the valve 100 percent open (Figure 3B). However, the instantaneous forces at the 100-percent opening are greater than at the partial openings. Thus, these large fluctuations start the closure even though an equivalent mean closing force without the violent fluctuations will not move the valve.

Closing Time

The closing time was found to be a function of the following variables: the differential head across the valve, the type of oil used in the dashpot, the needle valve adjustment, and the counterweight adjustment.

Two operating conditions were simulated in the laboratory: a high-pressure surge, and a line-filling condition. To simulate a surge, water was first allowed to discharge freely through the 8-inch main. Then the downstream regulating valve was closed as quickly as possible. As the air valve closed, the piezometric head at the valve inlet gradually increased until the shutoff head was reached. After the air valve had closed, the maximum piezometric head (shutoff pressure) and closing time were recorded (Figure 5).

A line-filling condition was simulated by maintaining the pressure in the 3-inch pipe at a constant value. This was achieved by manipulation of the downstream regulating valve. When the air valve had closed, the constant pressure and closing time were recorded (Figure 5).

An SAE 10 motor oil, with a Saybolt viscosity of 170 at 100° F, was used in the dashpot for preliminary tests. With this oil, the air valve could not be closed in less than 40 to 45 seconds. However, the manufacturer of the valve stipulated that the closing time should be approximately 30 seconds under the maximum head for proper water hammer control. Therefore, a less viscous oil was obtained which allowed the air valve to close in 30 to 35 seconds. This oil was designed for lubrication of refrigeration units and had a Saybolt viscosity of 82 at 100° F. All results presented in this report were obtained with the low viscosity oil.

The principal means of controlling the closing time is a screw-type needle valve located in the dashpot. Precise adjustment of the closing time was impossible with the valve tested because the needle was in very poor condition (Figure 6). Both the needle and its seat seemed to be machined from the same kind of brass. This led to scoring of the needle with some damage to the seat. With the needle in its present condition, one-fourth turn gave a total closing time of 45 seconds; and one-half turn gave a time of 30 seconds under 100 feet of shutoff head. However, these values are for comparison only, and should not be used for other air valves. The locking device to hold the needle valve in a constant position was found to be ineffectual.

The influence of the counterweight adjustment on the closing time is discussed in the Riser Pipe section of this report.

Counterweight Adjustment

A heavy (original) and a light (presently used) counterweight were supplied to the Denver Office with the air valve. Either one would counterbalance the dead weight of the moving parts in the valve. However, very little latitude was available for adjustment with either counterweight because the light one had to be placed on the outer end of the spindle and the heavy one on the inner end. A weight, whose mass was between the two weights supplied, would have allowed a greater latitude in adjustment. The heavy weight was used in the tests because the greatest change in valve performance could be realized from a given change in the weight position.

The manufacturer's instructions stated that the counterweights should be adjusted so the valve just opens with atmospheric pressure in the main line. Making this adjustment was very difficult due to friction in the counterweight bearings and the piston rings. When finally adjusted, the valve remained at any opening to which it was placed. However, bumping the spindle lightly or vibrating the valve body caused the valve to open 100 percent. At this adjustment, the distance between the heavy counterweight and the shoulder of the lever hub was 0.10 foot.

The following table gives the opening characteristics of the valve for various heavy counterweight adjustments and discharge pipe conditions.

Distance between the heavy counterweight and the shoulder of the lever hub	Valve opening* (Percent of valve travel)	Discharge pipe conditions
0.10 foot	55%	With 15-ft riser
0.10 foot	0	No riser
0.05 foot	55%	With 15-ft riser
0.05 foot	100%	No riser

*Spindle was not bumped nor the valve body vibrated. Tests begun with valve fully closed.

The table illustrates that, for opening the valve, minor changes in the counterweight adjustment are more significant without the riser than with the riser. Without the riser, a slight negative pressure in the pipeline was not enough to overcome the friction in the valve with the counterweight 0.1 foot from the spindle hub. However, moving the counterweight 0.05 foot toward the hub allowed the friction to be overcome and the valve opened 100 percent whenever atmospheric pressures existed in the pipeline. With a riser, an adjustment of 0.05 foot did not have a measurable effect on the percent of valve travel.

Riser Pipe

A riser pipe on the air valve outlet has two beneficial effects. It increases the hydrostatic head and, hence, the hydrostatic force on the stem and reduces the spillage. The increase in hydrostatic force allows the valve to close under smaller head differentials. In the laboratory, the 14.85-foot riser enabled the valve to close under a 2.5-foot head differential, whereas, without the riser a differential of 5 feet was required. The increased closing force due to the riser is noticeable up to a head differential of about 40 feet (Figure 5). The riser pipe greatly reduces spillage because no water will be wasted until the pipeline fills to elevation 530 on the suction side of Pumping Station T3. After the line fills, due to the alinement of the piping, the piezometric head will rise rather rapidly until a differential of 10 feet exists across the valves. Thus, spilling will occur only for the short time it takes to achieve the differentials that close the valves promptly.

Although there are beneficial effects, a riser pipe on the air valve outlet has two detrimental effects. First, the riser pipe prevents the valve from opening 100 percent with the counterweight adjusted to the manufacturer's recommendations. When the pressure level drops below the valve, the water in the riser pipe flows down through the valve. However, the buoyant force due to the float prevents the valve from opening fully. After the water has drained, the friction within the valve prevents the valve from opening further. With the recommended counterweight adjustment, the valve will open only 55 percent. However, the light counterweight can be adjusted so the valve opens 100 percent when the water in the riser pipe drains.

The second detrimental effect is caused by the need to use the light counterweight. Since a smaller amount of the dead weight of the valve is counterbalanced with the light counterweight, a greater force is needed to close the valve. This means a longer closing time under low head differentials. The tests revealed approximately a 1 minute increase in the closing time under a 7-foot head differential when using the light counterweight. No measurable change in the closing time was noted with either counterweight when operating under head differentials greater than 40 feet. The dynamic closing forces at these head differentials are so great they mask any effect of the counterweight adjustment.

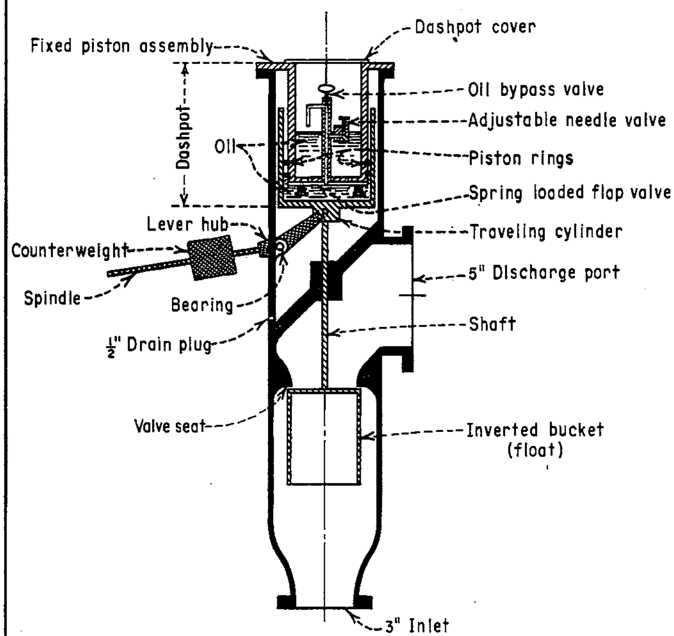
Sealing the Float

Sealing the float with a bottom plate performs two useful functions. First, the plate will prevent air in the float from being carried away by the water jet entering through the base of the valve. Second, sealing the float will prevent air from being dissolved by the water over long operating periods. Both cases serve to retain the full buoyancy of the float. However, laboratory tests with the sealed float revealed no major reduction in the closing time because the increased buoyant force is negligible when compared with the total closing force (Figure 5).

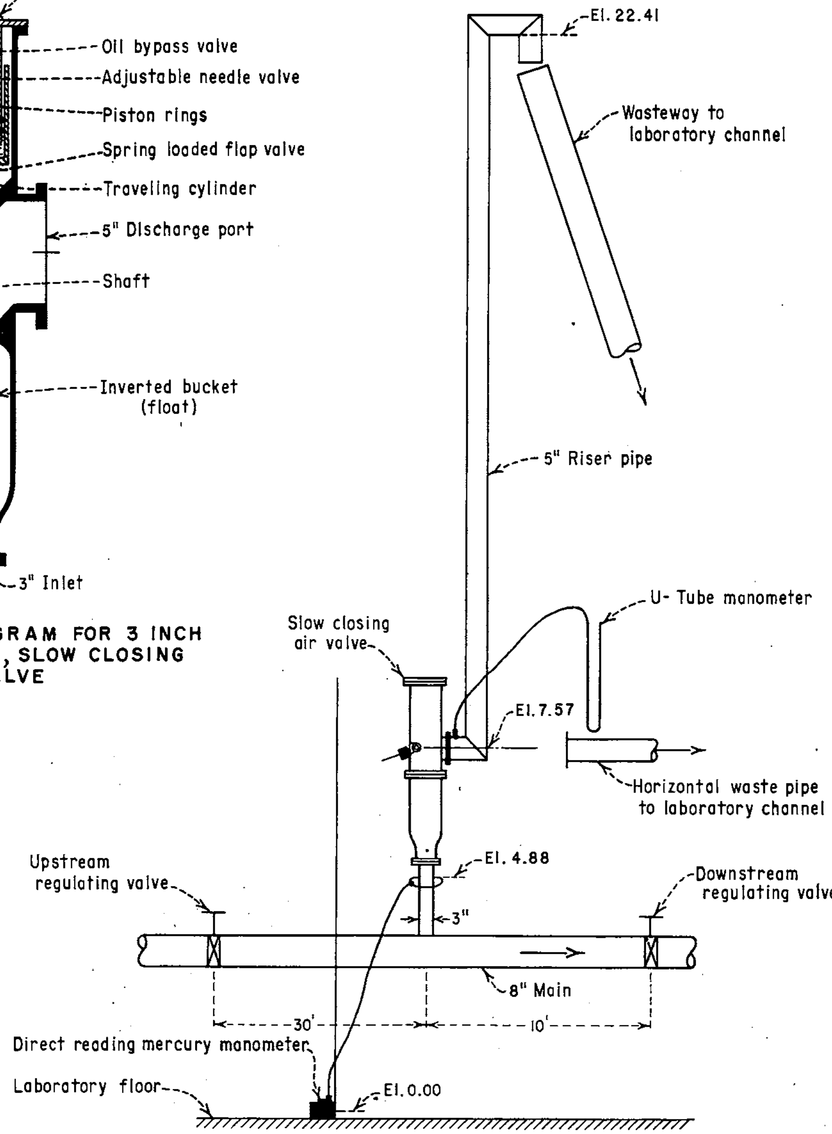
Limiting the Valve Opening

The manufacturer suggested that shorter closing times could be achieved under low heads by preventing the valve from opening 100 percent. With a maximum opening of 85 percent, the discharge is approximately 88 percent of the discharge at the 100-percent opening (Figure 3A). This reduced discharge capacity is sufficient to control water hammer surges in the Tea Pot Dome Distribution System.

A greater mean closing force is available to start closing the valve from the 85-percent opening than is available from the fully open position. This increased force makes the start of closure positive and eliminates any delay which might occur at the fully open position. However, if the valve opening is limited, the valve must still be adjusted to close in 30 seconds under a 100-foot head differential. This adjustment will, to some extent, counteract the decrease in closing time that is obtained by the greater initial closing force and the shorter stroke. Time was not available for the tests needed to determine the effect limiting the valve opening and no definite conclusions were reached.



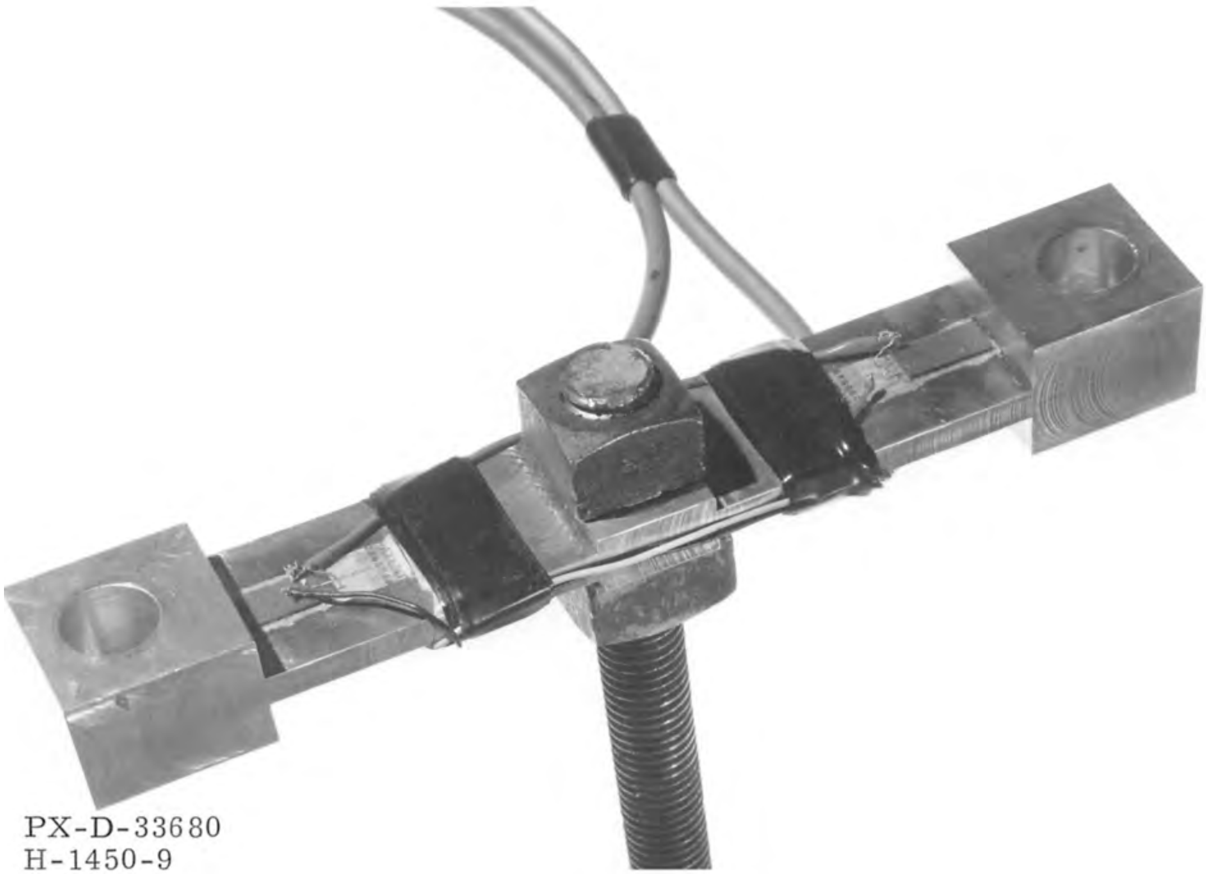
A. SCHEMATIC DIAGRAM FOR 3 INCH QUICK OPENING, SLOW CLOSING AIR VALVE



B. SCHEMATIC DIAGRAM OF LABORATORY FACILITY

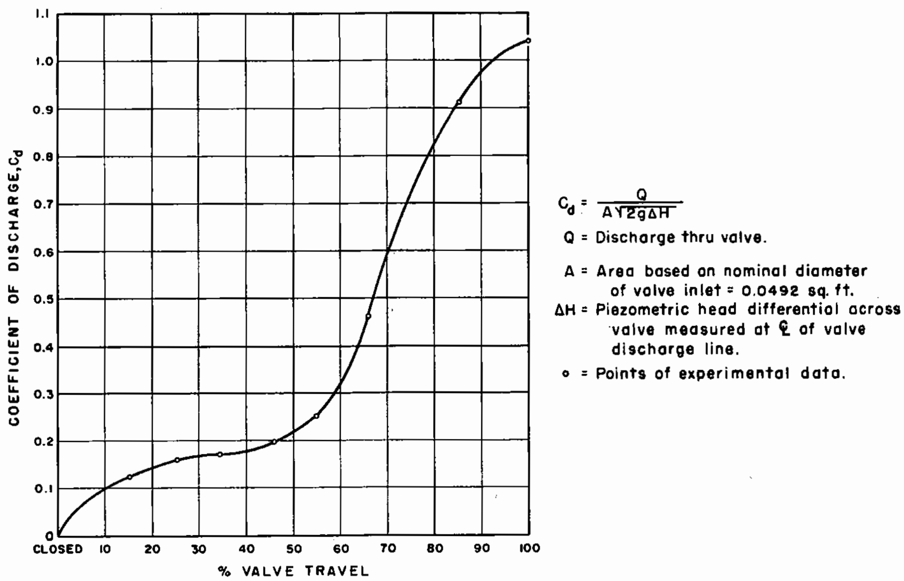
3-INCH QUICK-OPENING, SLOW-CLOSING AIR VALVE

SCHEMATIC DIAGRAM OF VALVE AND TEST FACILITY

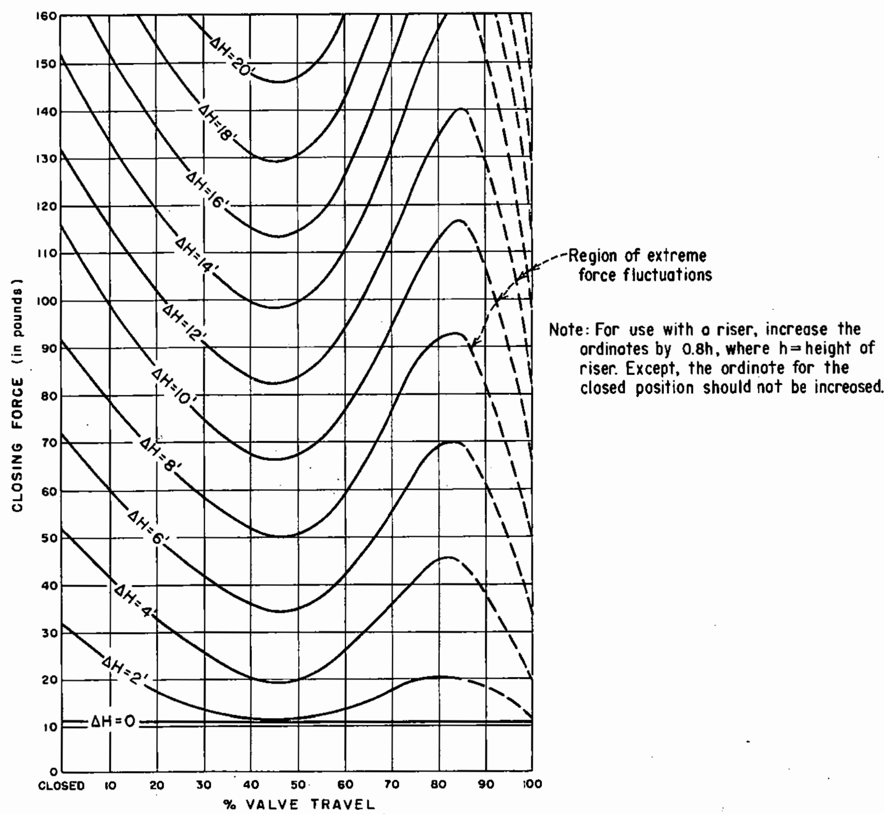


PX-D-33680
H-1450-9

3-INCH QUICK-OPENING, SLOW-CLOSING AIR VALVE
Proving Bar for Determination of Closing Force



A. DISCHARGE COEFFICIENT FOR VARIOUS VALVE OPENINGS



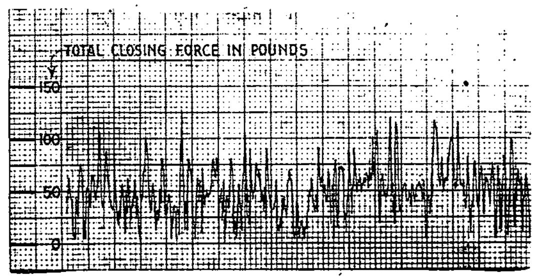
B. TOTAL CLOSING FORCE FOR VARIOUS VALVE OPENINGS, WITHOUT A RISER

3-INCH QUICK-OPENING, SLOW-CLOSING AIR VALVE

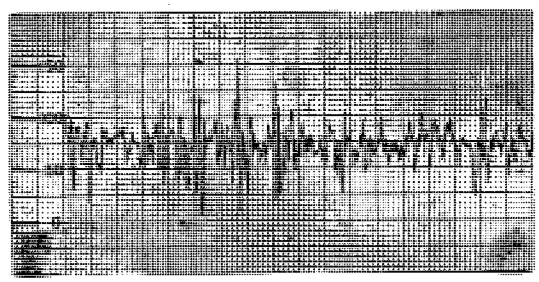
DISCHARGE COEFFICIENTS AND CLOSING FORCES

GPO 857157

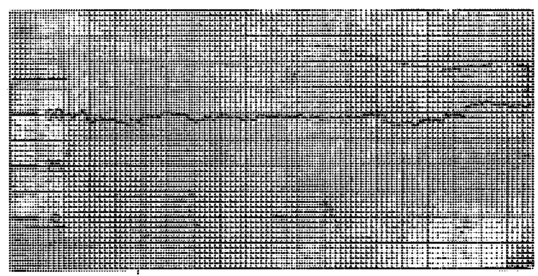
100 % OPEN
Q = 1.00 cfs



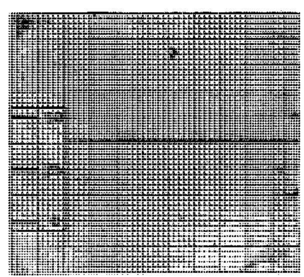
85.6 % OPEN
Q = 0.89 cfs



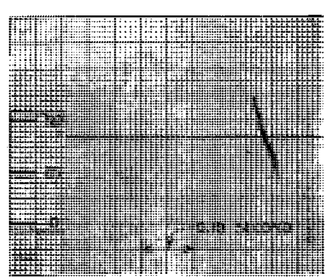
66.2 % OPEN
Q = 0.45 cfs



45.9 % OPEN
Q = 0.20 cfs

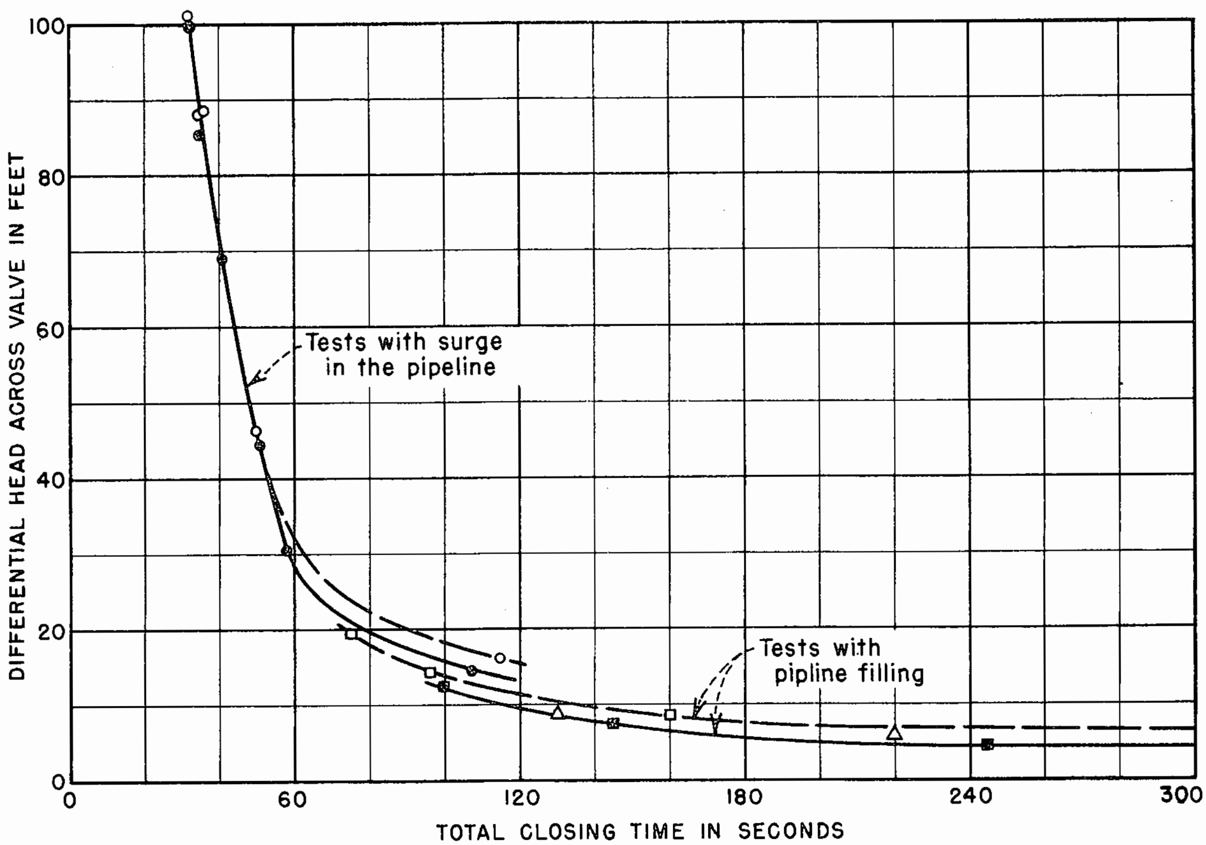


25.7 % OPEN
Q = 0.16 cfs



3-INCH QUICK OPENING SLOW CLOSING AIR VALVE

DYNAMIC CLOSING FORCE WITH A 6.2-FOOT HEAD DIFFERENTIAL



——— Denotes operation with a 15' riser
— — — Denotes operation without a 15' riser
△ Denotes operation without a 15' riser and with the float sealed
Counterweight set so valve opens 100% when valve is lightly tapped.

3-INCH QUICK OPENING SLOW CLOSING AIR VALVE

TOTAL CLOSING TIMES

Approximately 2.5x enlargement

3-INCH QUICK-OPENING, SLOW-CLOSING AIR VALVE
Needle Valve Condition