UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

PROGRESS REPORT ON HYDRAULIC CHARACTERISTICS OF PIPELINE ORIFICES AND SUDDEN ENLARGEMENTS USED FOR ENERGY DISSIPATION

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ABSTRACT

Tests were made at heads up to 500 feet with circular concentric orifices and a gate valve in a 3-inch diameter pipeline to determine cavitation characteristics and effects of cavitation on head loss, discharge coefficients, pressure distribution, velocity distribution, and pressure fluctuations. Effects of admission of air downstream from the control station were also determined. The sizes of enlarged sections needed to prevent conduit wall damage downstream from gate valves operating under cavitating conditions were established. Conclusions were--1/ head losses across orifice stations were not significantly changed by cavitation, 2/ discharge coefficients were affected if the coefficients are based on pressures measured at a distance downstream from the orifice, 3/ most of the pressure and velocity redistribution to a normal profile occurred within five pipeline diameters of the orifice or valve station, 4/ easily-used cavitation index values were established for a range of orifice sizes to determine when cavitation would begin, and the pressure and velocity conditions needed to prevent it, 5/ pressure fluctuations significantly increased when cavitation occurred, particularly at large orifice-to-pipe diameter ratios, 6/ under steady-flow operation a simple system of orifices in series could be used safely for energy dissipation, 7/ for variable-flow systems a more complex system with variable-size orifices appears necessary.

DESCRIPTORS--*hydraulics/submerged orifices/gate valves/ *cavitation/head losses/*discharge coefficients/velocity distribution/pressure distribution/*energy dissipation/instrumentation/ pitot-tube/laboratory tests/closed conduits/air/

IDENTIFIERS--Cavitation erosion/cavitation index/cavitation suppression/air admission/cylindrical pitot tube.

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PURPOSE

Studies of pipelines with circular concentric orifices and control valves were made to determine the effects of cavitation on downstream velocity distribution, pressure fluctuation, and energy losses, and to evaluate orifice stations and sudden enlargements as energy dissipators.

CONCLUSIONS

1. The head loss for an orifice is comparatively high and varies with the orifice-pipe diameter ratio, approaching 1.0 $\left(\frac{H_o - H_2}{H_o - H_1}\right)$ for small ratios and decreasing to zero for a ratio of 1.0 (Figure 2A).

2. Most of the recovery of velocity head downstream from an orifice or valve station occurs within a distance of five pipe diameters downstream of the station (Figures 1, 8, and 12A).

3. Most of the flow redistribution is also completed within 5 diameters of the downstream piping (Figures 8, 9, 10, and 11).

4. Cavitation will occur downstream from orifice stations when the cavitation index, K, is below certain values (Figures 5, 6, and 7A). The values depend upon the orifice-to-pipe diameter ratio, or in the case of valve stations, the upstream and downstream pipe diameter ratios and upon where the pressures are measured.

5. The measured head loss across orifice stations was not significantly affected by the presence or absence of cavitation (Figure 5).

6. By use of the general cavitation index parameter, $K = \frac{H_2 - H_v}{H_T - H_2}$ and information contained in Figures 7A and 7E, systems can be analyzed for possible cavitation and any necessary remedial measures can be taken.

7. Increases in back pressure, the admission of air, or decreases in upstream head will reduce or eliminate cavitation.

8. The coefficient of discharge of orifices is not affected by cavitation if the downstream head measurement is made at or near the downstream face of the orifice (Figure 6). If the coefficient is based on downstream pressures measured farther from the orifice, cavitation has a pronounced effect on the values.

9. Cavitation significantly increases the pressure fluctuations downstream from orifices, particularly at orifice-to-pipe diameter ratios, β, of about 0.9 (Figure 7D). Considerable pressure fluctuation is present even without cavitation.

10. Cavitation damage in pipelines downstream from control valves can be reduced or eliminated by using enlarged pipe sections downstream from the valves (Figure 13).

11. Air may be used in certain installations to reduce or prevent cavitation and its resultant damage, vibration, and noise.

12. Limited tests show that several properly sized orifice or valve stations can be used in series to safely dissipate very high heads. Under continuous operation at design discharge, no difficulty would be encountered in a simple, fixed, properly sized orifice system. During starting, stopping, or partial flow conditions, a simple system is not believed to be satisfactory and adjustable stations with suitable back pressure provisions appear necessary.

13. Multiholed orifice stations, particularly with several plates in series and with one or more of them adjustable to place the holes "in line" or "out of line," can be very effective energy dissipators. Additional investigation is needed to design for large prototype releases under high heads.

INTRODUCTION

In the early part of the 17th century, Galileo discovered that the velocity of flow from an orifice was proportional to the square root of the head on the orifice. Then, about 100 years later, Bernoulli assembled the equation $V = \sqrt{2gh}$ which provides a means of computing the velocity of flow from an orifice.

Further developments established the basic discharge relationship

$$Q = CA_o \sqrt{2gh}$$

for incompressible fluids, where

Q = discharge in cfs

C = numerical coefficient which varies with orifice setting

A_o = area of orifice in square feet

g = acceleration of gravity (32.2 ft/sec^2)

h = effective head across orifice

After the discharge relationship was established, the use of orifices to measure rates of flow became widespread. Standards were developed so that consistency and accuracy could be obtained, and the standards are mainly concerned with locations of pressure taps with respect to the orifice plate, orifice size with respect to pipe size, and shape of the orifice edge. The shape and position of the orifice opening within the pipeline were also considerations.

Numerous and extensive investigations have been made throughout the world to establish the relationships between these variables. Most of the relationships are well known and are published in engineering literature and in textbooks on fluid mechanics (Figures 1 and 2). It has been only recently, however, that investigations have been made to determine the effects of cavitation downstream from orifices upon their flow-measuring characteristics.1/

In recent years another use of the orifice has developed wherein orifice stations are used as energy dissipators. This use requires that the orifice be in a pipeline which remains completely filled downstream from the orifice. The turbulent eddy zone created around the jet after it leaves the orifice and expands to fill the downstream pipe is the source of most of the energy dissipation.

Several factors become important when an orifice is used as an energy dissipator for very high heads. Consideration must first be given to the allowable pressure drop, or head loss, across the orifice stage.

1/Refers to bibliography

Other considerations are the effects of the orifice-to-pipe diameter ratio on capacity, pressure fluctuations, vibration, cavitation potential; effects of cavitation on other hydraulic characteristics; possibilities of cavitation damage by pitting on flow boundaries; and the overall performance of multiple orifices used in tandem.

Investigations of some of these characteristics of orifices, when used as energy dissipators, have been conducted by the Hydraulics Branch, Division of Research, in Denver, Colorado. This report describes tests made with circular, concentric orifices, or with a gate valve, placed in settings suitable for energy dissipators with and without cavitation occuring in the downstream pipeline.

TEST FACILITIES

Tests were made on three separate facilities (Figure 3). The first tests were made with an orifice station located in a 3-inch standard, wrought iron pipe with a 3.068-inch inside diameter. The orifice plates were machined from 1/8-inch-thick brass plate with the downstream part of the bore relieved on a 45° cone (Figure 3, Detail C). The orifice diameters were 1.250, 1.750, and 2.375 inches, and the orifice-to-pipe diameter ratios, β, were 0.407, 0.570, and 0.773, respectively. Piezometers were drilled through the pipe flanges upstream and downstream from the plate to form corner taps approximating those defined by ASME Standards. 2/ Additional taps were provided in the pipelines one conduit diameter upstream from the orifice and at several stations downstream. Water was supplied by the central laboratory pumping system and flows were measured by permanently installed, calibrated Venturi meters. The maximum head available in this test facility was 150 feet. Control valves in the test line permitted control of the pressures and discharges for the wide range of tests conducted.

The second test facility provided much higher heads (Figure 3B). A single-stage pump and a high-head, two-stage pump were connected in series to produce heads up to 700 feet. A new orifice test section was constructed of 3.00-inch-inside-diameter brass pipe and the orifice plates were machined from 1-inch-thick brass plates in order that true corner taps could be used. The thickness of the orifice plate within the pipe was made one-fourth inch to provide adequate strength for high differential pressures. The downstream surface of the plate was relieved on a 45° chamfer to provide a typical thin-edged orifice plate. Orifice diameters of 1.000, 1.625, and 2.625 inches were used. The diameter ratios, β, were 0.333, 0.542, and 0.875, respectively. Piezometer taps were provided in the pipe walls upstream and downstream from the orifice station, and flows were measured by a calibrated laboratory orifice Venturi meter.

Velocity and pressure distribution measurements were made upstream and downstream from orifices in the second test facility by means of a 1/8-inch-diameter, three-holed, stainless steel, cylindrical pitot tube. Pairs of machined bosses with O-ring seals were placed diametrically opposite one another at appropriate stations along the pipeline and the cylindrical pitot tube was inserted through them. Thus, the instrument extended through both sides of the pipe and received adequate support to withstand the hydraulic and dynamic loads for most of the tests. The instrument was positioned and held in place by a support that encircled the pipe (Figure 4). A scribe mark on the instrument body was alined with graduations on a machinists' scale on the support to position the pitot openings at the desired point across the passage. The tube was rotated to obtain equal pressures in the two static pressure openings and then clamped in position. Any angularity in flow about the axis of the tube cylinder was shown by the calibrated pointer at the top of the tube.

The third test facility used a gate valve as a variable-opening orifice (Figure 3C). Water at heads up to 500 feet was supplied to the test valve; upstream and downstream control valves permitted control of the pressures and discharges during tests. Piezometers were provided in the 3-inch, 300-pounds-per-square-inch valve body downstream from the seat rings and in the upstream and downstream pipes. Flows were measured by a standard laboratory orifice-Venturi meter.

The likelihood of cavitation erosion in enlarged pipe sections downstream from the control valve also was tested by placing concrete-lined pipes in the regions of cavitation collapse (Figure 3C). A mix of 1 part cement, by weight, to 6 parts sharp angular sand, with only enough water to produce a moderately stiff mortar, was used. The mortar was tamped into place inside an 8-inch lightweight pipe and screeded smooth. Curing time was usually about 48 hours, but difficulties of scheduling tests sometimes made the cure time as much as 96 hours. The test duration was 6 hours.

Pressure measurements were first made with high-quality, calibrated Bourdon gages. However, results were not sufficiently accurate and final measurements were made with a fluid pressure scale and with mercury-filled U-tubes. In all cases, the head differential across the orifice was measured with a mercury U-tube. This required a tube about 10 feet high in the high-head, high-velocity tests. Corrections were made for the net height of the water columns on the mercury columns to obtain the true differential pressure.

Photographs were taken through sections of transparent plactic pipe to record the cavitation clouds within the flow. Photographs were obtained by means of strobe lights with duration times of about 1/10,000 second. Motion picture records at rates up to 3,000 frames per second were also taken.

INVESTIGATION

General Characteristics--No Cavitation

Sudden enlargements--The flow in a pipeline just downstream from an orifice is an example of flow into a sudden enlargement. In such cases, the flow phenomena related to a submerged jet are applicable, and substantial dissipation of energy takes place. The head loss is adequately expressed by the Borda formula,

$$h = \frac{(V_j - V_2)^2}{2g}$$

where V_j is the jet velocity and V_2 is the velocity in the downstream pipeline.3/ This dissipation of energy is accompanied by substantial and rapid pressure fluctuations, particularly if high-velocity flows occur.

Pressure distribution--The hydraulic gradeline for orifices within or at the end of pipelines has been well established (Figure 1A). Upstream from the orifices, the pressure gradient follows the friction slope except near the orifice. There the pressure rises until it reaches essentially the stagnation value at the upstream face of the orifice plate. The pressure then drops as the flow accelerates to pass through the orifice and with free discharge flow conditions, reaches atmospheric pressure. In closed pipeline flow it drops to a minimum at the vena contracta a short distance downstream from the orifice (Figure 1B). The pressure subsequently rises rather rapidly to the maximum, or recovery, value several pipe diameters downstream from the orifice. The hydraulic gradient then coincides with the friction slope of the downstream pipe.

Coefficient of discharge--The relationship of coefficient of discharge to the orifice-to-pipe diameter ratio has long been established for orifices where cavitation does not occur (Figure 2B). Reynolds number affects the coefficients, tending to lower the coefficient as Reynolds number increases. The coefficient of discharge, C_d, is based on the equation

$$C_d = \frac{Q}{A_o \sqrt{2g} \sqrt{H_o - H_x}}$$

where A_o is the orifice area and H_o and H_x are the pressure heads at the upstream and downstream measuring taps, respectively. There is a difference in the numerical value of the coefficient depending on how far the downstream tap is located from the orifice. This is due to the extent of pressure redistribution that has occurred and the friction and eddy losses encountered (Figure 6).

Head loss--The change in static pressure from a point a short distance upstream from an orifice (minimum pressure station) to a point downstream where the pressure grade reaches maximum is considered the loss attributable to the orifice. This loss is often expressed as a percent of the difference between the minimum piezometric pressure measured a short distance upstream from the orifice to the minimum piezometric pressure downstream (Figures 1A and 2A). Test data for the work covered in this report and based on measurements at stations one conduit diameter upstream and 9 diameters downstream, for six orifice-to-pipe diameter ratios, agree well with the presented curve (Figures 2A and 5B).

The loss can also be expressed in terms of velocity head, such as $H_L = K_L\left(\frac{V_o^2}{2g}\right)$ where H_L is loss in feet, K_L is loss coefficient applicable to the orifice, and V_o is the average velocity based on orifice area. The value of K_L will vary depending on the orifice pipe diameter ratio and the Reynolds number.

Cavitation Potential of Orifices in Pipelines

The jets flowing from orifices into filled pipelines (sudden enlargements) represent cases of extreme separation where, in addition to the expansion and diffusion of the main jet, there is the generation of secondary flow and countless small eddies and vortices. The pressures within the eddies will be appreciably below that of the surrounding fluid, particularly when the velocity of orifice efflux is high. These low pressures can quite easily reach the vapor pressure of the water and vapor pockets or cavities will form. When this occurs, cavitation has started. An increase in flow velocity and/or a decrease in ambient line pressure will intensify the formation and the subsequent violent collapse of the vapor pockets and produce more severe cavitation. Conversely, a decrease in velocity and/or an increase in the downstream line pressure will reduce or eliminate the cavitation.

Measurements made on a variety of orifice-to-pipe diameter ratio systems at operating heads up to 600 feet and differential heads up to 300 feet have helped to determine the operating conditions for incipient cavitation (Figures 5 and 6). To make these determinations, particular attention was given to the conditions of

(1) Cavitation definitely audible, (2) cavitation audibility questionable, and (3) cavitation definitely not audible. The second condition, although difficult to delineate, was very useful in determining, K_i, the value for incipient cavitation. The general index value, K, is expressed by the formula

$$K = \frac{H_2 - H_v}{H_T - H_2} ,$$

where H_2 is downstream pipeline pressure at selected points, H_v is the vapor pressure of water relative to the atmosphere, and H_T is the total head upstream from the orifice. The cavitation data were also expressed in the form

$$Kd = \frac{H_2 - H_v}{V_o^2/2g} ,$$

where V_o is the average velocity through the orifice, for comparison with certain data of other organizations.

The index value where cavitation just starts, K_i or K_{di}, for a given orifice will vary depending on the station at which the reference pressure is measured. The values appeared comparatively constant (between 0.45 and 0.6) when based on pressures upstream from the orifices and at the downstream flange taps (Figure 6). The values varied up to about 2.0 when based on downstream pressures measured at taps located five or more pipe diameters downstream where the velocity distribution becomes more uniform (Figures 5 and 6).

Average values of K_i for various orifices-to-pipe diameter ratios are shown by the upper curve or dashed line in Figure 7A. With this relationship established, it is possible to determine the downstream pressure required to prevent cavitation and its inherent noise and vibration for given discharges and upstream conditions.

Incipient values, K_{di}, for the parameter using velocity head instead of pressure difference in the denominator were also determined by listening for the first sounds of cavitation (Figure 7B). The curve formed by these values is higher than the curve based on the occurrence of vapor pressure immediately downstream from the orifices, and on the Borda loss. This higher curve shows that cavitation occurs before vapor pressure is registered at flange or vena contracta taps. Such an occurrence is in accord with the concept of vapor formation in the much lower pressure regions within eddies and vortices created in the diffusion process.

General Characteristics--Cavitation Present

Pressure and velocity distribution--With fully developed turbulence and a given orifice and pipeline arrangement, the flow pattern geometry in the pipe does not change appreciably with changes in discharge or pressure, provided the downstream pipe is kept under sufficient back pressure to assure that the jet is surrounded by water (Figures 7C and 8A). Thus the coefficient of discharge for a given orifice-to-pipe diameter ratio will be constant for given tap locations as long as this operating condition exists. The numerical value of the coefficient will, however, be determined by the position of the chosen tap along the pipeline, and hence along the changing hydraulic gradeline (Figure 1B). The position of the upstream tap is the least critical because the pressure gradient immediately upstream from an orifice changes relatively little. However, due to the rapid change in pressure gradient downstream from the orifice, the location of the downstream tap has considerable effect on the discharge coefficient.

When the downstream pressure is not high enough to maintain liquid water around the jet, cavitation begins and the flow pattern and pressure distribution change (Figure 8). As the pressure is lowered past the point of incipient cavitation, an envelope of cavities forms around the jet (Figures 9, 10, and 11). Further lowering of the pressure causes the envelope to grow and to extend downstream to alter the flow geometry and pressure distribution. In the extreme case, not shown in Figure 8, the envelope occupies most of the pipe cross section for a distance of many pipe diameters downstream from the orifice. In such cases, the "zone of recovery," where the jet dissipates and uniform pipe flow is re-established, is moved well downstream (Figure 7E).

Detailed velocity profile measurements were made at Stations 1, 2, 3, 4, 5, and 9-1/3 pipe diameters downstream from the 1.625-inch orifice ($\beta = 0.542$) in the 3-inch test facility (Figure 8). The velocity profiles show the turbulent recirculation downstream from the orifice plates and the rapid change from high velocity at the orifice to the lesser velocity in the pipeline. Most of this change in velocity occurs within four to five pipe diameters. There was little change in the velocity distribution when the cavitation conditions varied from light to heavy, i.e., between K values of 0.573 and 0.067 (Figures 8B and 8C).

Although not delineated in these tests, the length of flow passage in which the velocity reduction takes place may increase materially as K decreases below the 0.067 value and vapor pressure surrounds the jet downstream from the orifice (Figure 7E).

The characteristic of rapid flow redistribution and rapid reduction in velocity was noted in tests made in recent years on a pressure reducing plant for a water power station.4/ In this case the enlargement, which consisted of a needle valve 0.9 meters in diameter discharging into a section of pipe 3 meters in diameter and 18 meters long, performed very effectively. The fact that the energy dissipation and velocity redistribution take place in such a short distance makes the sudden enlargement principle a useful and economical tool.

An observation made during velocity measurements in the pipe downstream from the orifices is of particular interest. A cylindrical pitot tube one-eighth inch in diameter was used for making velocity traverses across the pipe at the several stations noted previously. When inserted at distances of 1 and 2 diameters downstream from the orifice and with back pressure to prevent cavitation, the pitot tube was snapped in two within a few seconds. At first it was thought that high velocities were the primary cause of failure. However, when the back pressure was reduced to induce slight or severe cavitation, the measurements were made without difficulty. It was later concluded that at back pressures which precluded cavitation, the natural frequency of the tube and the periodicity of the Von Karman vortex street forming downstream from the tube were in resonance and caused extreme vibration and rapid failure of the tube.

Pressure fluctuations--The energy dissipation in the pipe, where the jet from the orifice expands and is surrounded by a zone of highly turbulent flow, is accompanied by highly fluctuating pressures. These pressure fluctuations can induce objectionable vibrations in lightweight structures, particularly if the natural frequency of the structure corresponds to that of the pressure fluctuations. Intense noise and vibration will occur when cavitation occurs in the turbulent zone.

To determine the magnitude and nature of the fluctuations, pressure cells were connected to piezometer taps in the pipe walls downstream from different size orifices in a 3-inch pipeline. Tests were made at heads up to 600 feet with varying degrees of cavitation.

Pressure cell records showed that pressure fluctuations existed in the flow upstream from the orifices. Part of this fluctuation was attributed to the pumping system and the remainder was attributed to disturbances caused by the flow in the sudden enlargement downstream from the orifice. Although it was not possible to isolate the pump-induced vibrations from the data, interesting conclusions concerning pressure fluctuations in the recovery zone beyond the orifice were nevertheless possible.

In general, for values of β below 0.6, the pressure fluctuation coefficients, H_F, attributed to the orifices were about 0.5 (Figure 7D). This coefficient is based on the equation

$$H_F = \frac{\Delta H}{h_{vo} - h_{vp}}$$

where ΔH = pressure fluctuation (feet of water)

h_{vo} = velocity head at orifice (feet of water)

h_{vp} = velocity head in downstream pipeline (feet of water)

For values of β higher than 0.6, the pressure fluctuation coefficients increase rapidly to a value of about 3.0 at $\beta = 0.8$. It appears that for high β values where large pressure fluctuations occur, there is greater danger of cavitation erosion. Thus, enlargements with these β values may not be suitable as energy dissipators. From Figure 7, it appears that a β value of about 0.5 would be optimum for good energy dissipation without damage to the conduit. This conclusion is supported by the fact that flow conditions with a cavitation index of less than 0.08 are required to produce damage on the downstream conduit walls of a valve installation having an upstream-to-downstream pipe diameter ratio of 0.50 (Figure 13B).

Head loss--The head loss, expressed by the coefficient K_L, and measured from a station one pipe diameter upstream from the orifice to stations 9 and 12 pipe diameters downstream, was not significantly affected by the presence or absence of cavitation (Figure 5B). Even violent, jarring cavitation did not cause a noticeable change in losses, and the losses based on the above parameters closely followed the curve in Figure 2A. This is in agreement with results reported in studies of a European energy absorber design.4/

Coefficient of discharge--Coefficients of discharge based on taps that are affected by gradient changes due to cavitation will deviate from standard values for those taps (Figure 6). As the cavitation envelope below an orifice lengthens due to decreased back pressure and the recovery zone moves past the downstream tap, the pressure at the tap is lowered. An apparent increase in differential head between the upstream and downstream taps occur, even though there is no change in the pressure immediately downstream from the orifice and no change in discharge. The result is a decrease in coefficient (Figure 6). As the cavitation envelope extends and vapor pressure reaches the downstream tap, the coefficient again becomes constant and has a value equal to that for flange taps.

The flow geometry from the orifice to the vena contracta remains essentially the same regardless of downstream pressure. Thus, discharge coefficients based on taps located at or upstream from the vena contracta will not be significantly changed by the presence of cavitation downstream from the orifice. This explains why the coefficient of discharge is essentially constant for flange taps regardless of the degree of cavitation.1/

Multiple orifices in series--The relatively high losses associated with sudden enlargements such as orifices in pipelines, 5/6/ and the short length of enlarged section required to redistribute the flow, suggested that head reduction and energy dissipation could be effectively accomplished by using orifices in series.

One use of multiple enlargements in series to safely dissipate high-velocity flow involved placing two orifices of the same size in a pipeline leading to the cooling system of an electrical transformer unit. 6/ The problem consisted of determining the optimum positions and spacing of the orifice and valve components to give satisfactory reduction in head from 185 to 70 feet without severe cavitation or vibration for 0.3 to 1.3 cubic feet per second of cooling water flowing continuously. The multiple enlargements were obtained by placing two 2.086-inch orifices in a 3-inch pipeline, together with a plug valve for flow regulation. The orifice-to-pipe diameter ratio was 0.695.

Tests showed that the plug valve should be placed upstream from the orifices where considerable back pressure existed, and that the space between the components should be about 15 inches, or 5 diameters of the 3-inch pipe. The pressure gradient in the system showed that head recovery was completed within the 5 diameters, and that the loss for one component was not influenced by the other components (Figure 12A). These conclusions were verified by visual observations and still and high-speed motion pictures of conditions in transparent pipe sections downstream from orifices (Figures 9, 10, and 11).

Cavitation at the orifices can be avoided by supplying sufficient back pressure to prevent pressures downstream from the orifices from reaching vapor pressure. Thus, a series of orifices, or other sudden enlargements, could be used in stages to dissipate energy from high heads, and cavitation and vibration could be eliminated when sufficient back pressure was provided at each stage to prevent the occurrence of vapor pressure. If sufficient back pressure is not available or is not desired, cavitation can be permitted without fear of erosive damage to the pipe walls if the pipe is made sufficiently large, or if air is admitted. The downstream pipe size required to prevent damage, and the effect of air admission, are discussed subsequently.

The principle of successive enlargements can be applied to the dissipation of energy in turbine bypasses. The number of stages could be determined so that there would be no cavitation in the last stage. A series of simultaneously operated needle valves spaced about five pipe diameters apart in a pipeline would be an effective, but perhaps costly, means of bypassing water around a turbine that was out of service (Figure 12B). Other types of controls, including variable-sized orifices, might be used depending on the nature of the installation and whether or not some satisfactory method, perhaps based on the type of variable orifice used, could be developed for starting and stopping flow in the system. This factor of safely starting and stopping the flow is sometimes overlooked in designing systems, and is obviously of the utmost importance with respect to cavitation at the primary control point. An ideal bypass facility might be developed using the variable orifice and the principle depicted in Figure 12B.

Multihole orifice plates in series--The Bureau of Reclamation has not yet performed research in the field of multihole orifice plates or eccentric orifices. The possible use of such plates in series for dissipating energy and bypassing water under high heads is to be considered as a part of the research on sudden enlargements now in progress. Some excellent work on this subject, using circular holes and slots, has already been accomplished by others. 7/ Multihole-multiplate orifices can be used very effectively for dissipating energy in a compact facility. Rotation of multiholed orifice plates to the "misalined" position increases losses and permits compactness by closer spacing of the plates. To date, no large-scale, high-head structures have been built using this principle.

Oversized pipe downstream from valves--A control valve with an oversized downstream pipe constitutes a special orifice where the size and shape of the opening vary. Tests were made to determine the size of the larger pipe needed downstream from the valve to prevent cavitation damage on the pipe. General investigations were first made with a 3-inch gate valve at heads up to 500 feet (Figure 3C). It was determined that the cavitation index for incipient cavitation damage for conventional gate valve installations with no enlargement downstream (β = 1.0) was about 1.0 (Figure 7F). Use of an enlarged section 1.5 times the diameter of the valve (β = 0.67) lowered the cavitation index for incipient damage to 0.15 (Figure 13B). Further increase in downstream pipe sizes (reductions in β) caused further lowering of the incipient damage cavitation index, and better protection from damage.

The above high-head tests were restricted to gate openings of 16 percent or less because of pump limitations. New facilities of greater capacity are being prepared so the studies can be extended.

In lower-head studies of a design for specific project use, tests were made with an 8-inch gate valve followed by a 14-inch-diameter pipe (Figure 13A). 8/ A riser pipe extended upward from the 14-inch section and maintained a 6-foot back pressure on it. The 14-inch-diameter test section was lined with an easily erodible portland cement mortar and the valve was operated at large and small openings at a 150-foot head. The diameter ratio, β , was 0.58. Tests showed that the lining was not damaged under any test condition, including sustained operation at a cavitation index value of 0.25. Extensive use of this design has subsequently been made for releasing irrigation water at heads up to 125 feet.

Effect of air--The continuous admission of small amounts of air to the pipeline immediately downstream from an orifice or valve was extremely effective in eliminating cavitation damage and reducing vibration and noise. In the case of the concentric orifices, the air could be admitted through any one of the circumferentially placed 1/16-inch-diameter piezometers near the orifice plates with equal and dramatic effectiveness. With the valve, best results were obtained when air was admitted into the bottom of the gate body just downstream from the seat rings. Good results were also obtained when air was admitted in the bottom of the pipeline immediately downstream from the valve. Admission of air to the sides of the pipeline was less effective, and air admission at the top was least effective. In all cases, however, the reduction in damage, vibration, and noise was pronounced. The quantities of air were not measured, but appeared to be a small percent of the waterflow.

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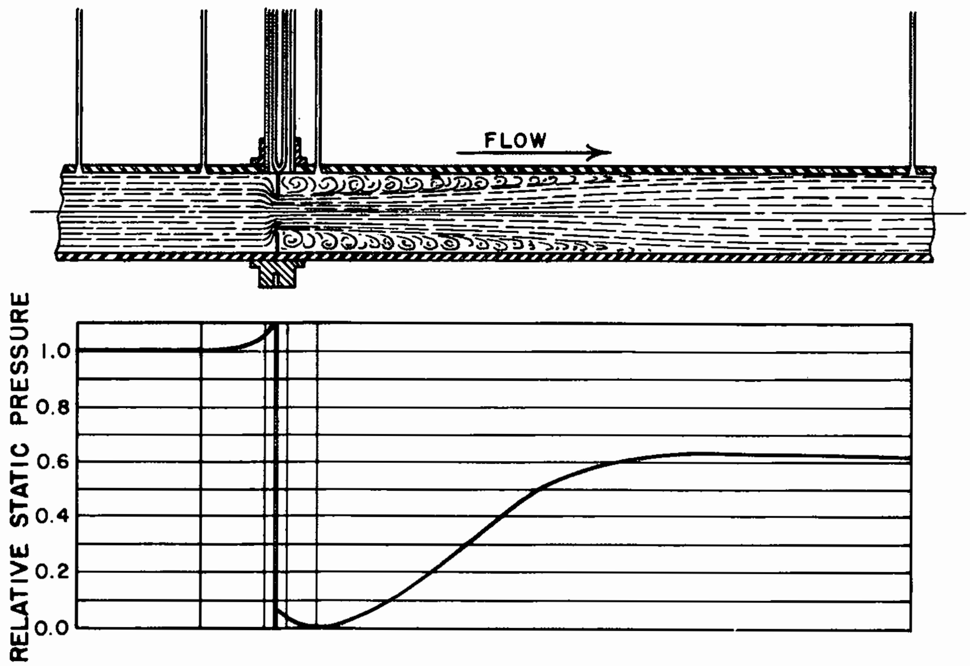
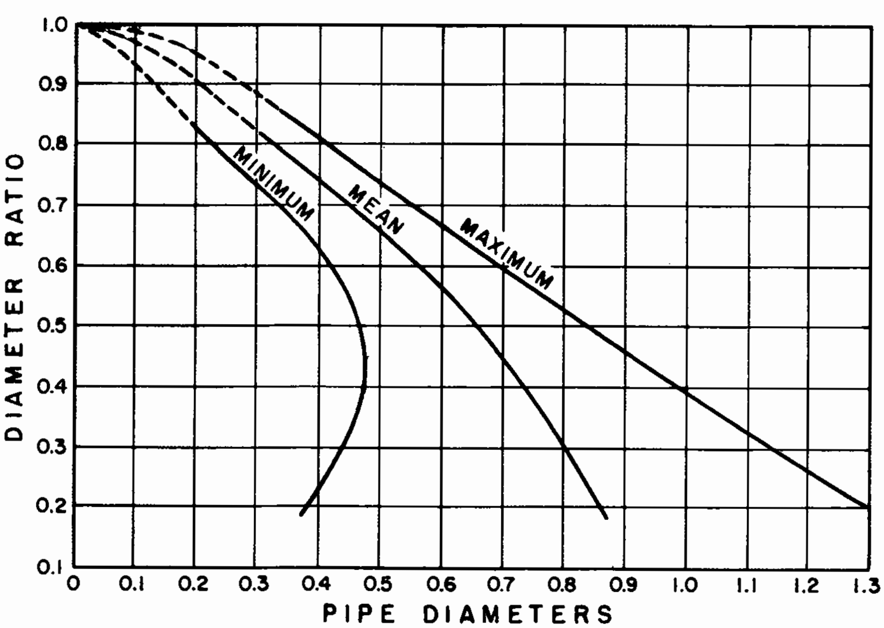
6. "Hydraulic Studies of a Pressure Reducing System for Transformer Cooling Water--Grand Coulee Powerplant," by L. V. Wilson, Bureau of Reclamation, Report Hyd-308, March 1957

7. "Variable Flow Resistances with Adjustable Multihole Orifice Plates in Series," by J. Silverman, F. A. Grochowski, and J. E. Sharbaugh, Journal of Basic Engineering, ASME, Volume 82, Series D, No. 3, September 1960

8. "Cavitation Characteristics of Gate Valves and Globe Valves Used as Flow Regulators under Heads up to 125 Feet," by J. W. Ball, Transactions, ASME, Volume 79, No. 6, August 1957



A. DIAGRAMMATIC REPRESENTATION OF FLUID FLOW THROUGH A THIN-PLATE ORIFICE SHOWING POSITIONS OF PRESSURE TAPS IN COMMON USE AND THE RELATIVE STATIC PRESSURE

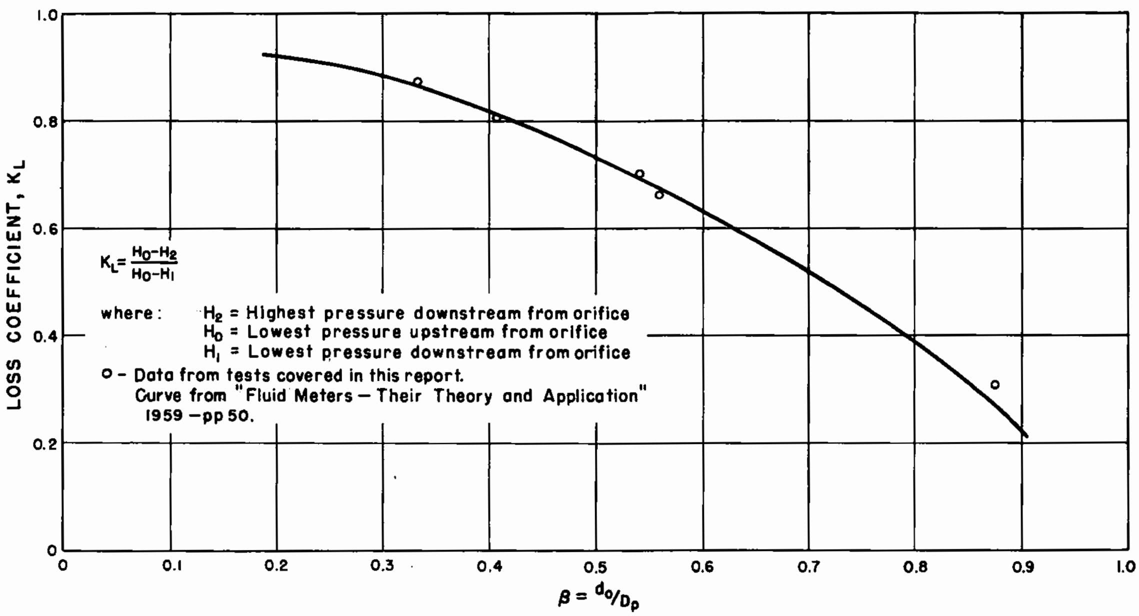


B. POSITION OF MINIMUM STATIC PRESSURE

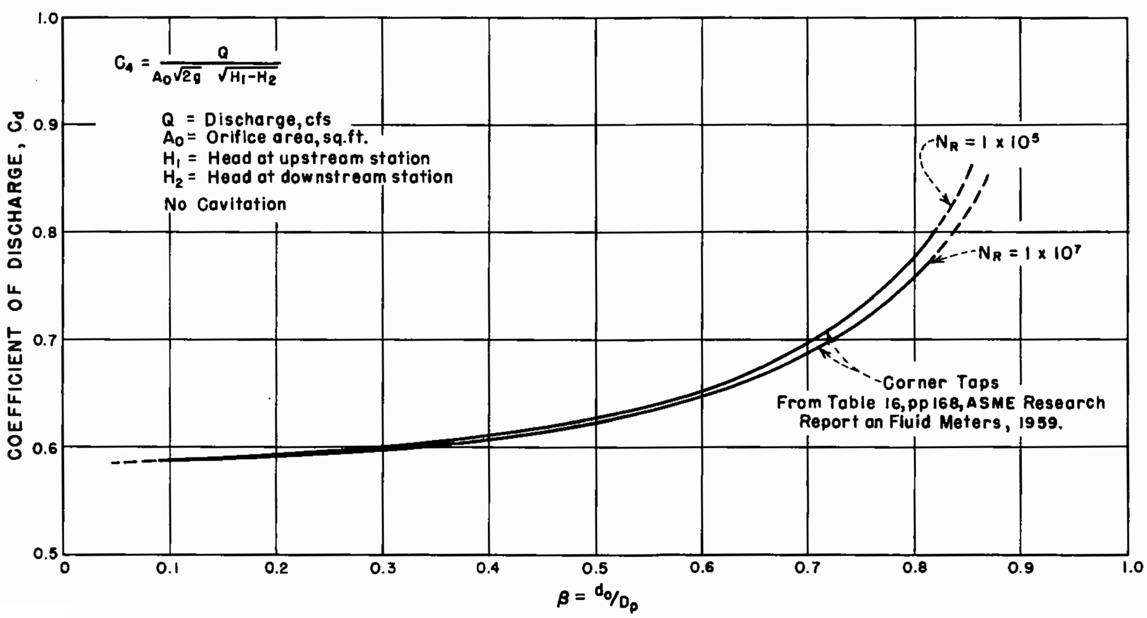
From: ASME Research Report on Fluid Meters, 1959

ORIFICES AND VALVES IN PIPELINES

PRESSURE DISTRIBUTION AT TYPICAL ORIFICE STATIONS
NO CAVITATION



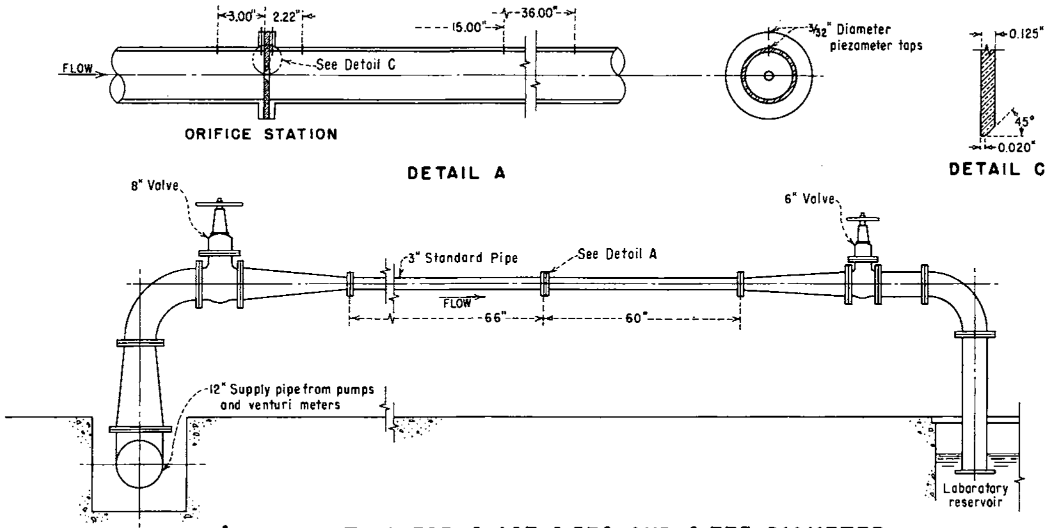
A. LOSS COEFFICIENT VS DIAMETER RATIO



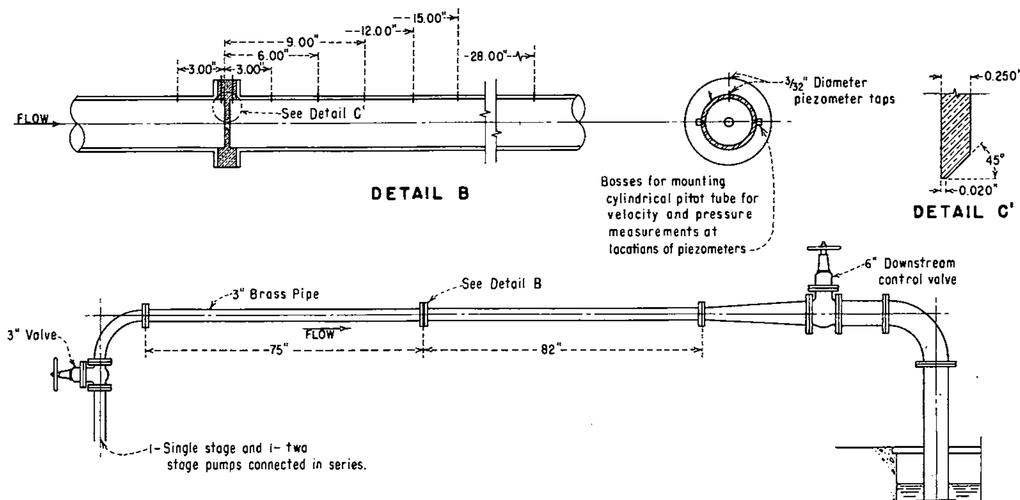
B. COEFFICIENT OF DISCHARGE VS DIAMETER RATIO

ORIFICES AND VALVES IN PIPELINES

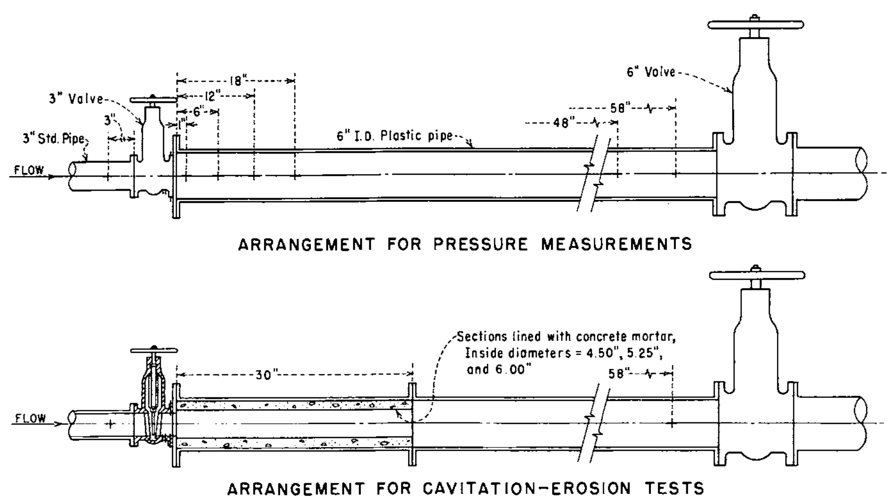
LOSS AND DISCHARGE COEFFICIENTS FOR TYPICAL ORIFICES NO CAVITATION



A. FACILITIES FOR 0.407, 0.570 AND 0.773 DIAMETER RATIO ORIFICE TESTS



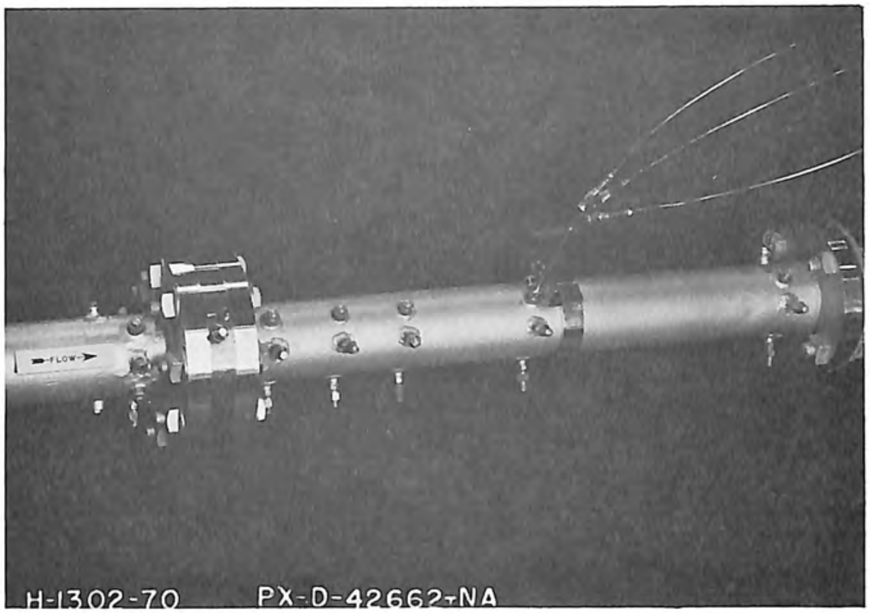
B. FACILITIES FOR 0.333, 0.542 AND 0.875 DIAMETER RATIO ORIFICE TESTS



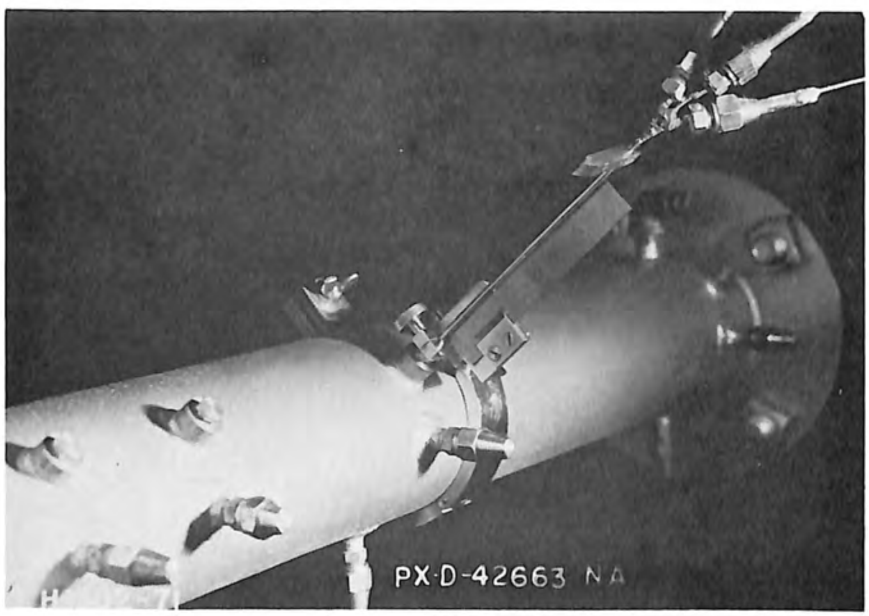
C. FACILITIES FOR GATE VALVE TESTS

ORIFICES AND VALVES IN PIPELINES
SCHEMATIC DIAGRAMS OF TEST FACILITIES

553



A. Orifice test facility with 1/8-inch-diameter cylindrical pitot tube installed for measuring velocity and pressure distribution.



B. Detailed view of instrument holder and bosses through which the tube is inserted.

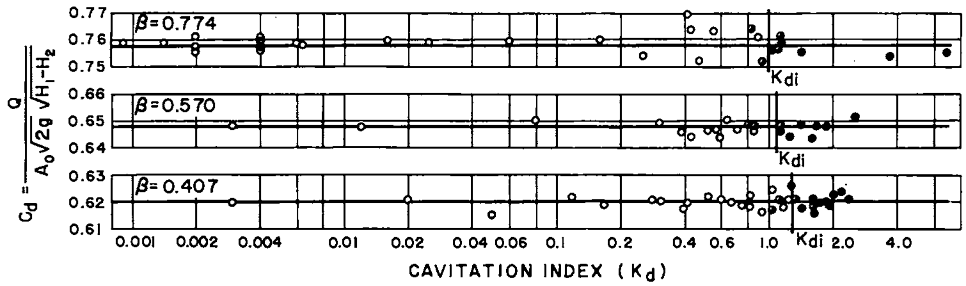
ORIFICES AND VALVES IN PIPELINES

Facilities for Orifice Tests with $\beta = 0.333$, 0.542 and 0.875

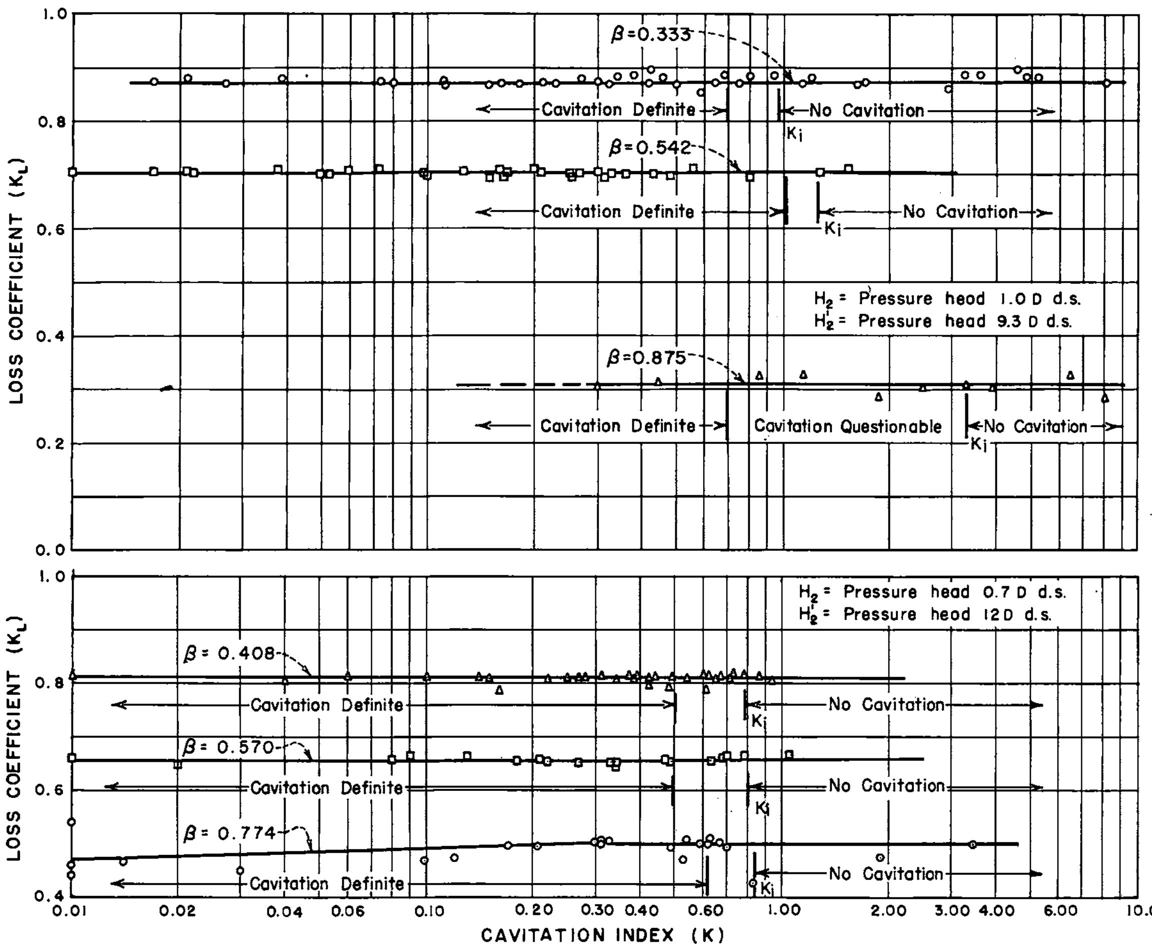
H_1 = Pressure head at upstream flange tap, in feet.
H_2 = Pressure head at downstream flange tap, in feet.
H_v = Vapor pressure of water, in feet.

V_0 = Average velocity through orifice, in feet per second.
A_0 = Area of orifice, in square feet.
Q = Rate of flow, in cubic feet per second.

$$K_d = \frac{H_2 - H_v}{V_0^2/2g}$$



A. VARIATION OF DISCHARGE COEFFICIENT, C_d, WITH CAVITATION INDEX, K_d



$$K_L = \frac{H_0 - H_2'}{H_0 - H_1} \quad \text{and} \quad K = \frac{H_2 - H_v}{H_T - H_2}$$

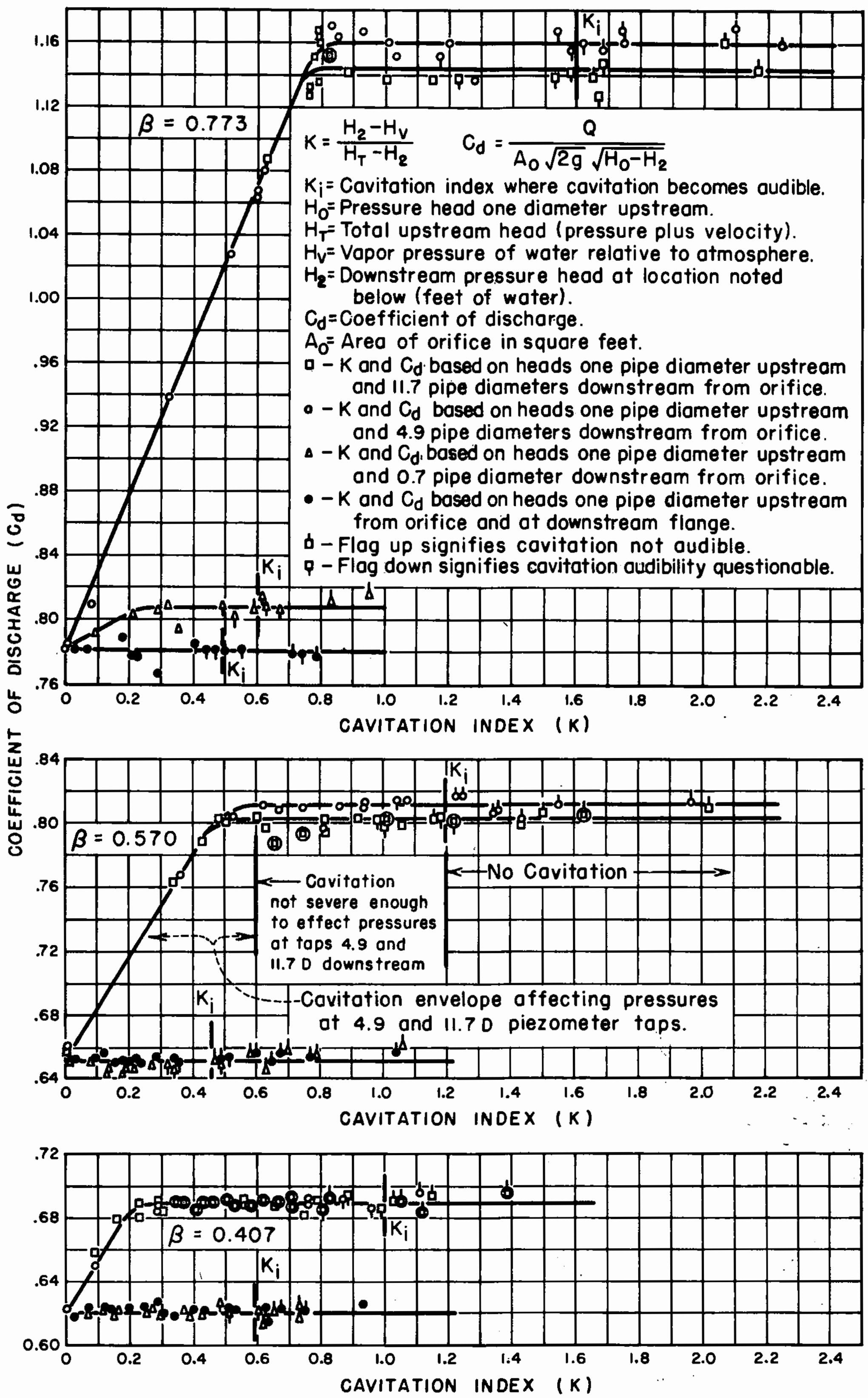
K_i and K_{di} cavitation index values where cavitation just becomes audible.

Where:
H_0 = Pressure head 1 dia. upstream
H_1 = Pressure head 1 dia. downstream
H_2' = See notes on graphs
H_2 = See notes on graphs
H_v = Vapor press. of water relative to atmosphere
H_T = Total upstream head

$$\beta = \frac{d_0}{D_P}$$

B. VARIATION OF LOSS COEFFICIENT, K_L, WITH CAVITATION INDEX, K

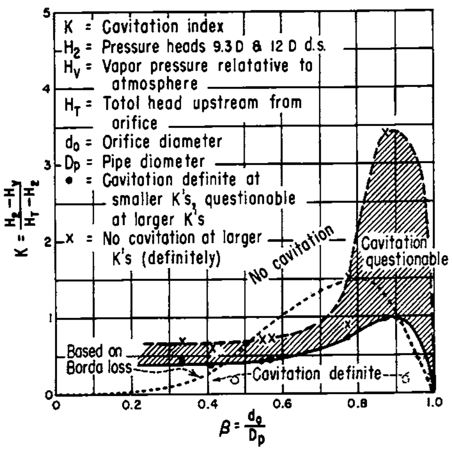
ORIFICES AND VALVES IN PIPELINES
EFFECT OF CAVITATION ON ORIFICE LOSSES



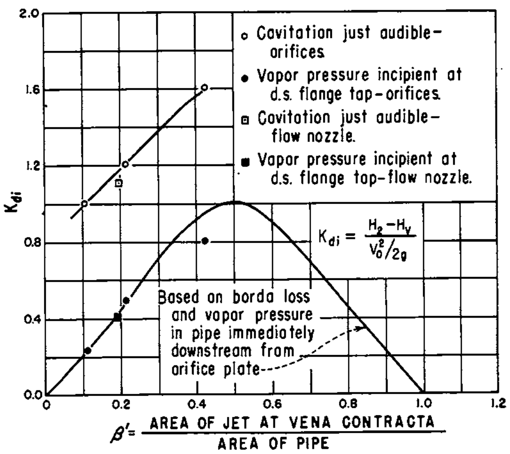
VARIATION OF COEFFICIENT OF DISCHARGE, C_d, WITH CAVITATION INDEX, K

ORIFICES AND VALVES IN PIPELINES

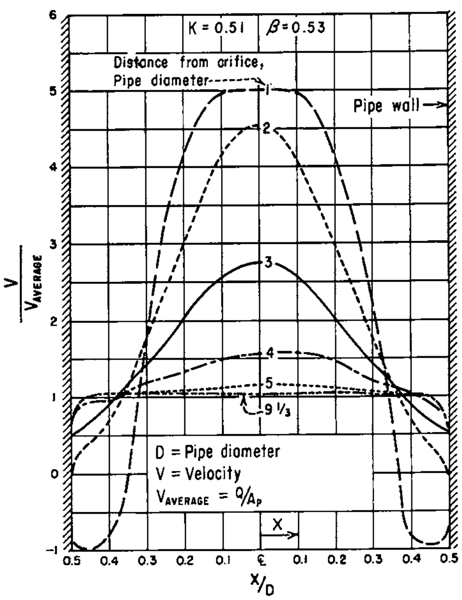
EFFECT OF CAVITATION ON COEFFICIENT OF DISCHARGE WITH VARIOUS DOWNSTREAM PRESSURE TAPS



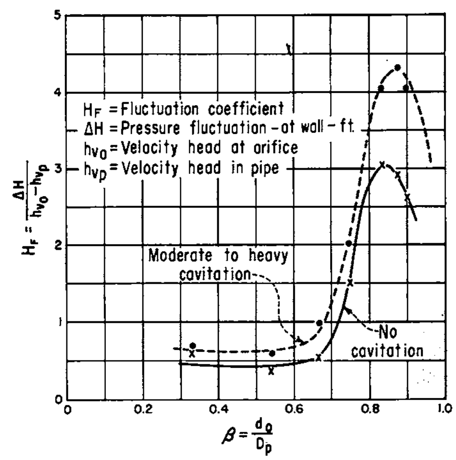
A. CAVITATION POTENTIAL OF ORIFICES IN A PIPELINE



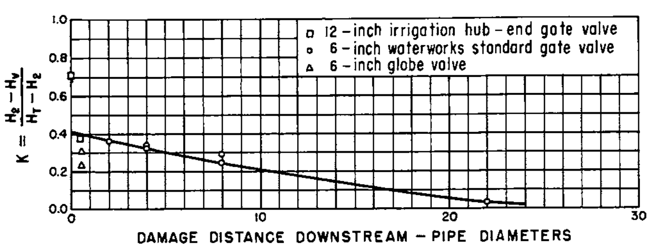
B. CRITICAL CAVITATION NUMBER K_{di} FOR ORIFICES AND FLOW NOZZLE



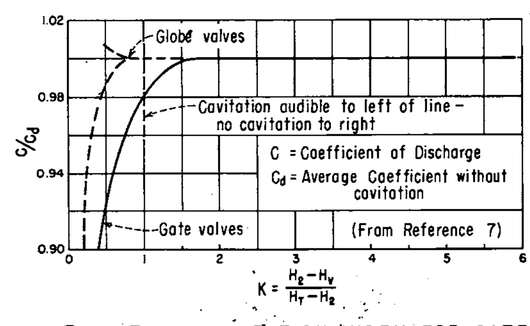
C. VELOCITY REDISTRIBUTION IN PIPE DOWNSTREAM FROM ORIFICE—GENERAL



D. PRESSURE FLUCTUATIONS DOWNSTREAM FROM ORIFICES IN PIPELINES



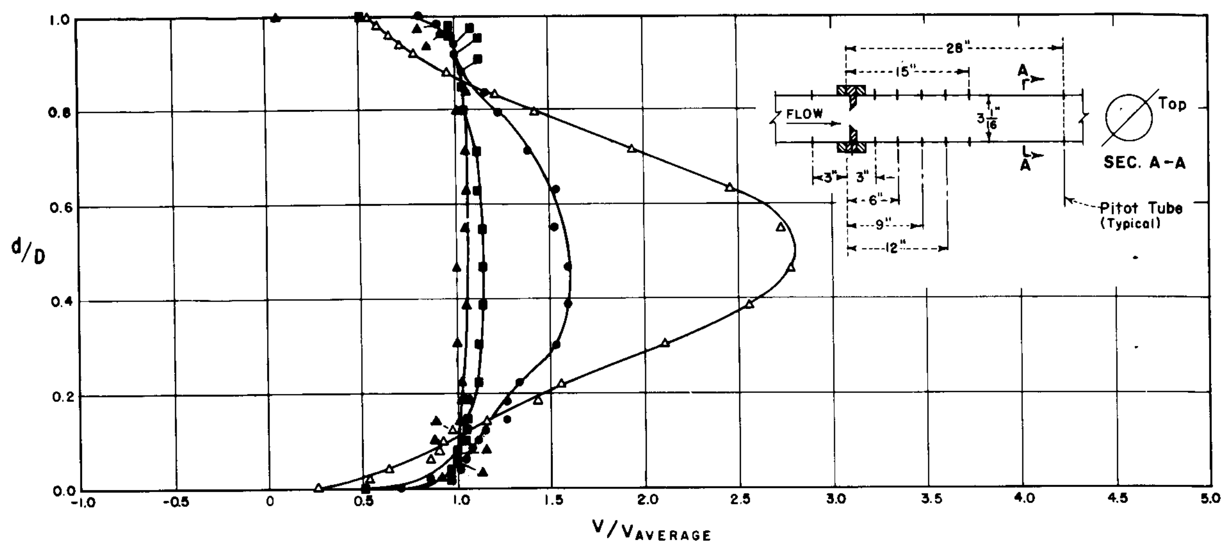
E. DISTANCE OF CAVITATION DAMAGE DOWNSTREAM FROM VALVE



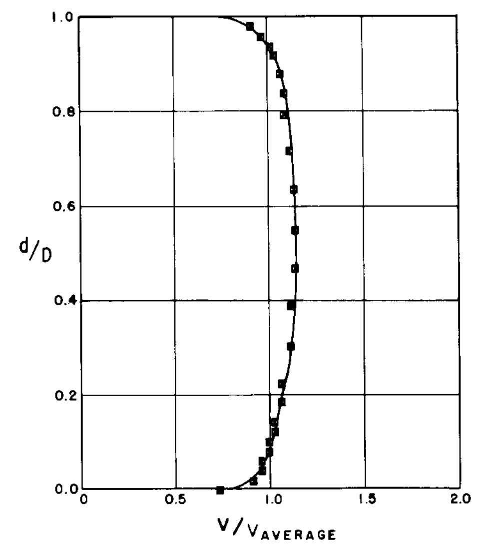
F. CRITICAL CAVITATION INDEX FOR GATE AND GLOBE VALVES

ORIFICES AND VALVES IN PIPELINES

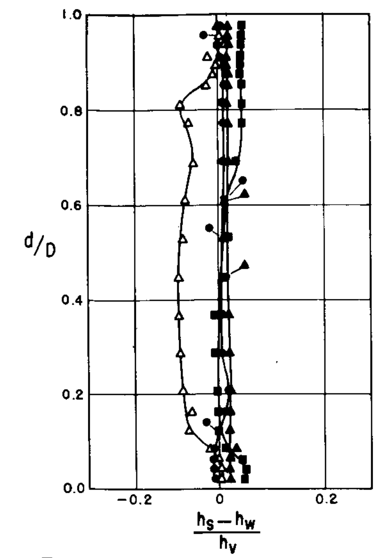
CAVITATION POTENTIAL, GENERAL VELOCITY DISTRIBUTION, AND PRESSURE DISTRIBUTION AT ORIFICE AND VALVE STATIONS



A. VELOCITY DISTRIBUTION - NO CAVITATION, K = 1.820



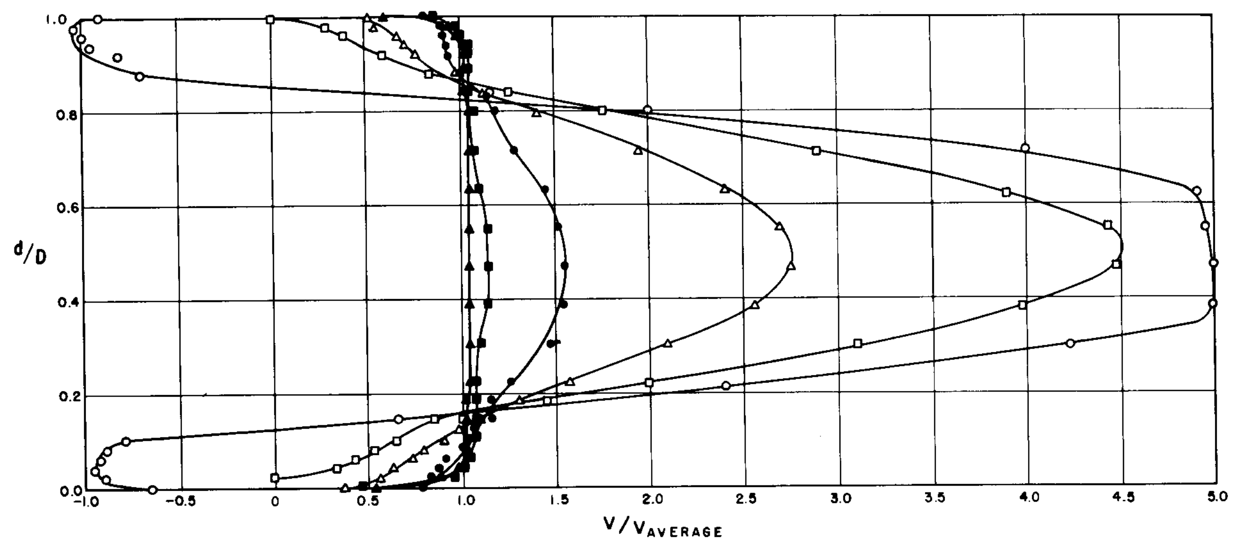
D. VELOCITY DISTRIBUTION AHEAD OF ORIFICE



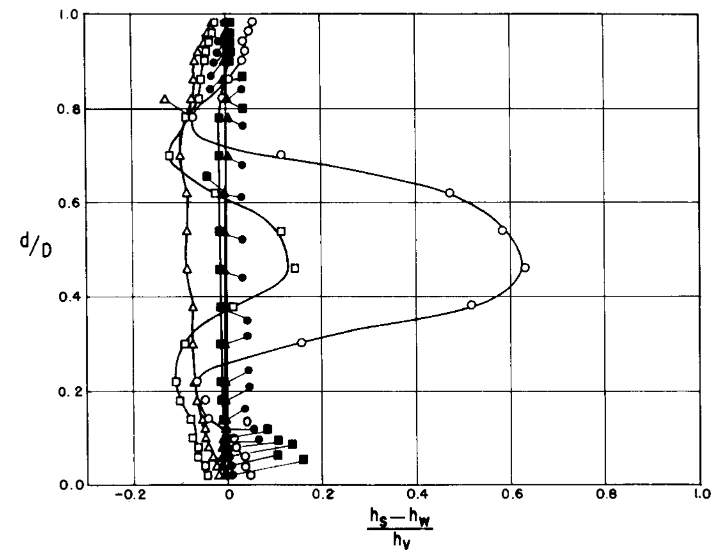
E. PRESSURE DISTRIBUTION - NO CAVITATION, K = 1.820

SYMBOLS

- ○ ....3" Downstream from Orifice
- □ ....6" Downstream from Orifice
- △ ....9" Downstream from Orifice
- ● ....12" Downstream from Orifice
- ■ ....15" Downstream from Orifice
- ▲ ....28" Downstream from Orifice
- ■ ....3" Upstream from Orifice.



B. VELOCITY DISTRIBUTION - LIGHT CAVITATION, K = 0.513



F. PRESSURE DISTRIBUTION - LIGHT CAVITATION, K = 0.513

$$K = \frac{H_2 - H_V}{H_T - H_2}$$

h_s = Static pressure at any point in traverse.

h_w = Static pressure at wall on traverse.

h_v = $V_0{}^2/2g$.

V_0 = Q/ORIFICE AREA.

d = Distance from wall to traverse point.

D = Diameter of pipeline (3.0")

V = Velocity at traverse point.

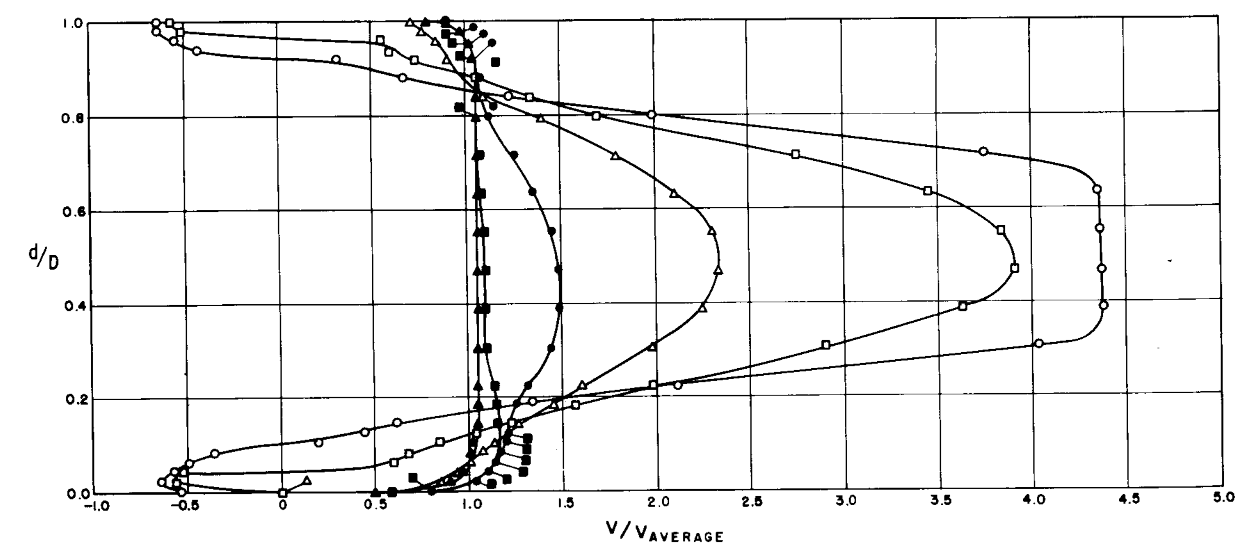
$V_{AVERAGE}$ = Q/PIPE AREA

H_2 = Pressure head at downstream flange top.

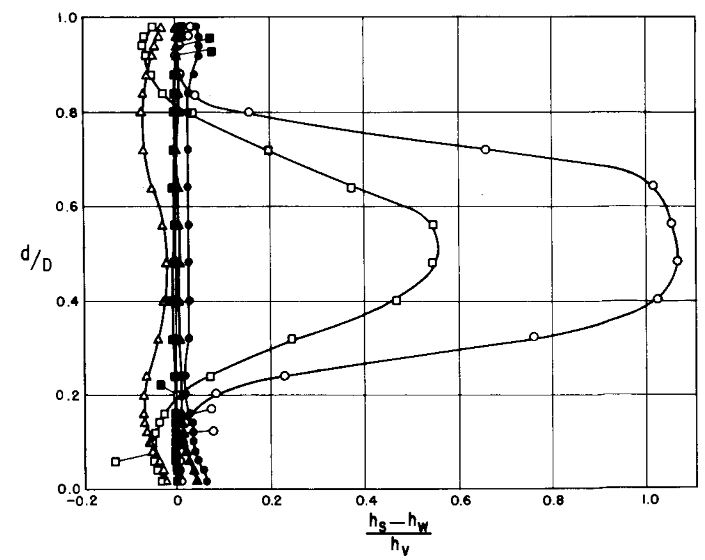
H_V = Vapor pressure relative to atmosphere.

H_T = Total head upstream from orifice.

$\beta = {}^{d_0}/_{D_p} = 0.542$



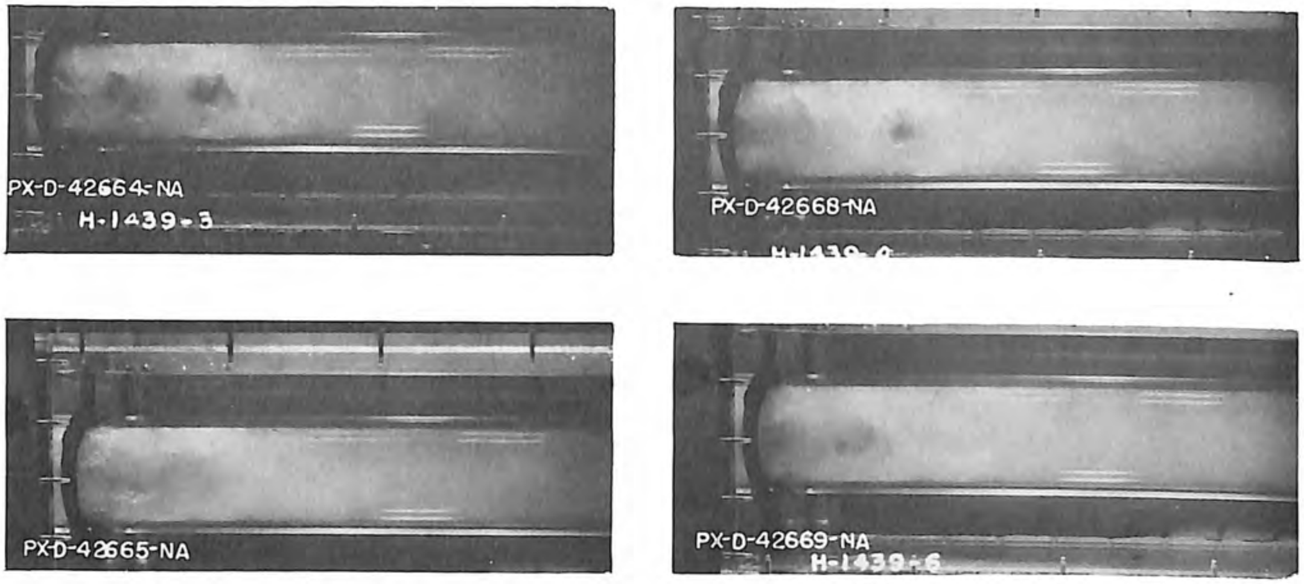
C. VELOCITY DISTRIBUTION - HEAVY CAVITATION, K = 0.067



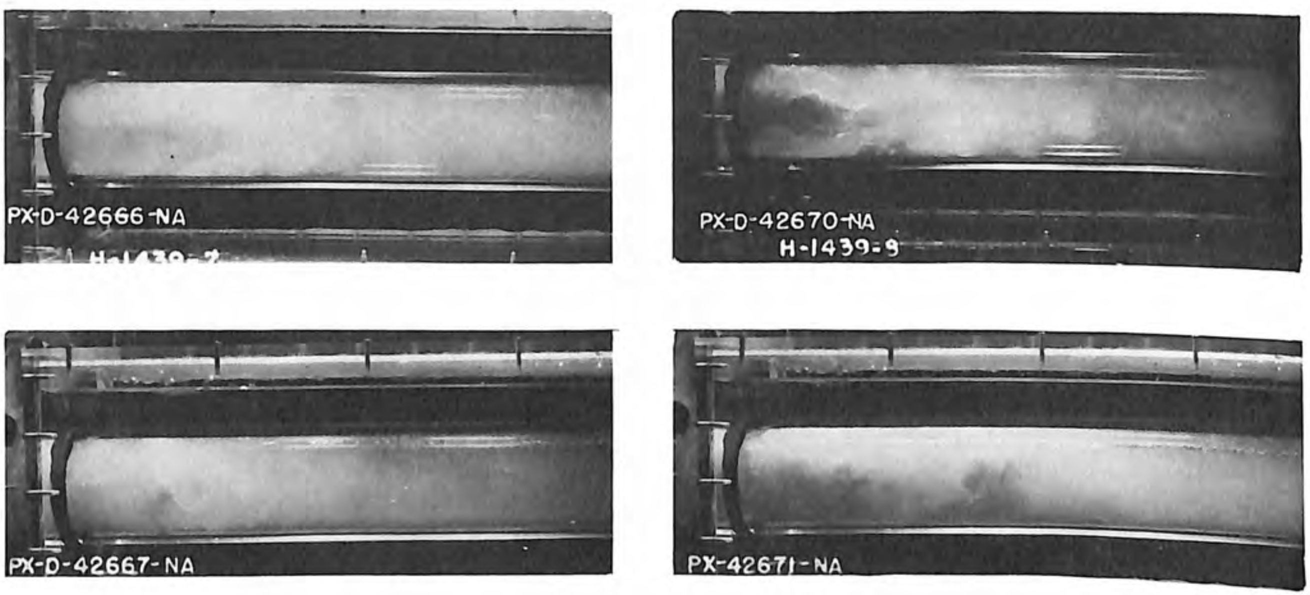
G. PRESSURE DISTRIBUTION - HEAVY CAVITATION, K = 0.067

ORIFICES AND VALVES IN PIPLINES

VELOCITY AND PRESSURE PROFILES AT ORIFICE STATIONS



A. Moderately heavy cavitation - K = 0.54
H_T = 145 feet, H_2 = 33 feet

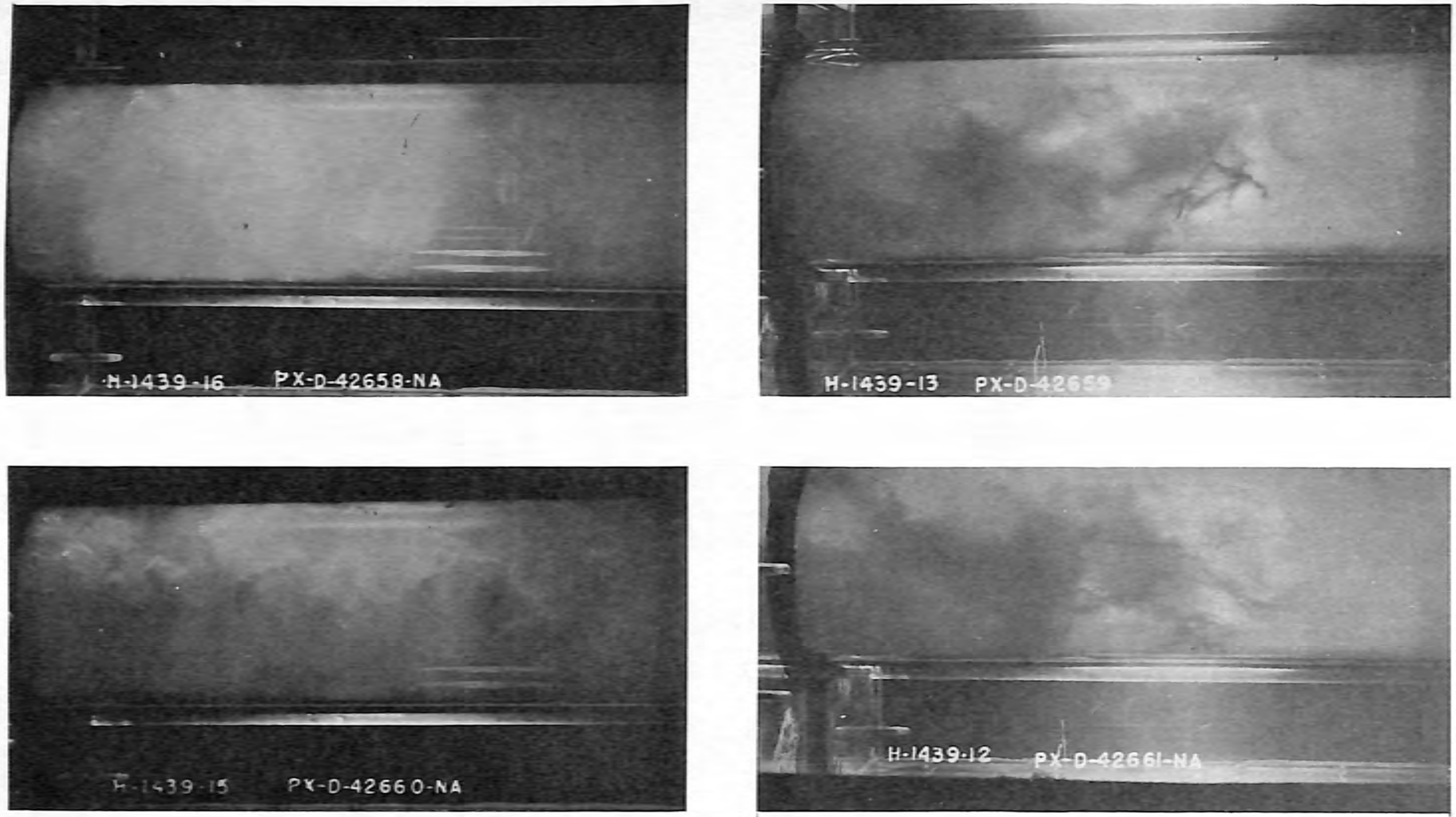


B. Moderately heavy cavitation - K = 0.42
H_T = 154 feet, H_2 = 27 feet

Exposure time 1/10,000 second
Lighting from top and bottom
H_2 measured 16D downstream from orifice

ORIFICES AND VALVES IN PIPELINES

Photographs of Cavitation Downstream from an Orifice
1.625-inch Orifice in a 3.000-inch Pipe



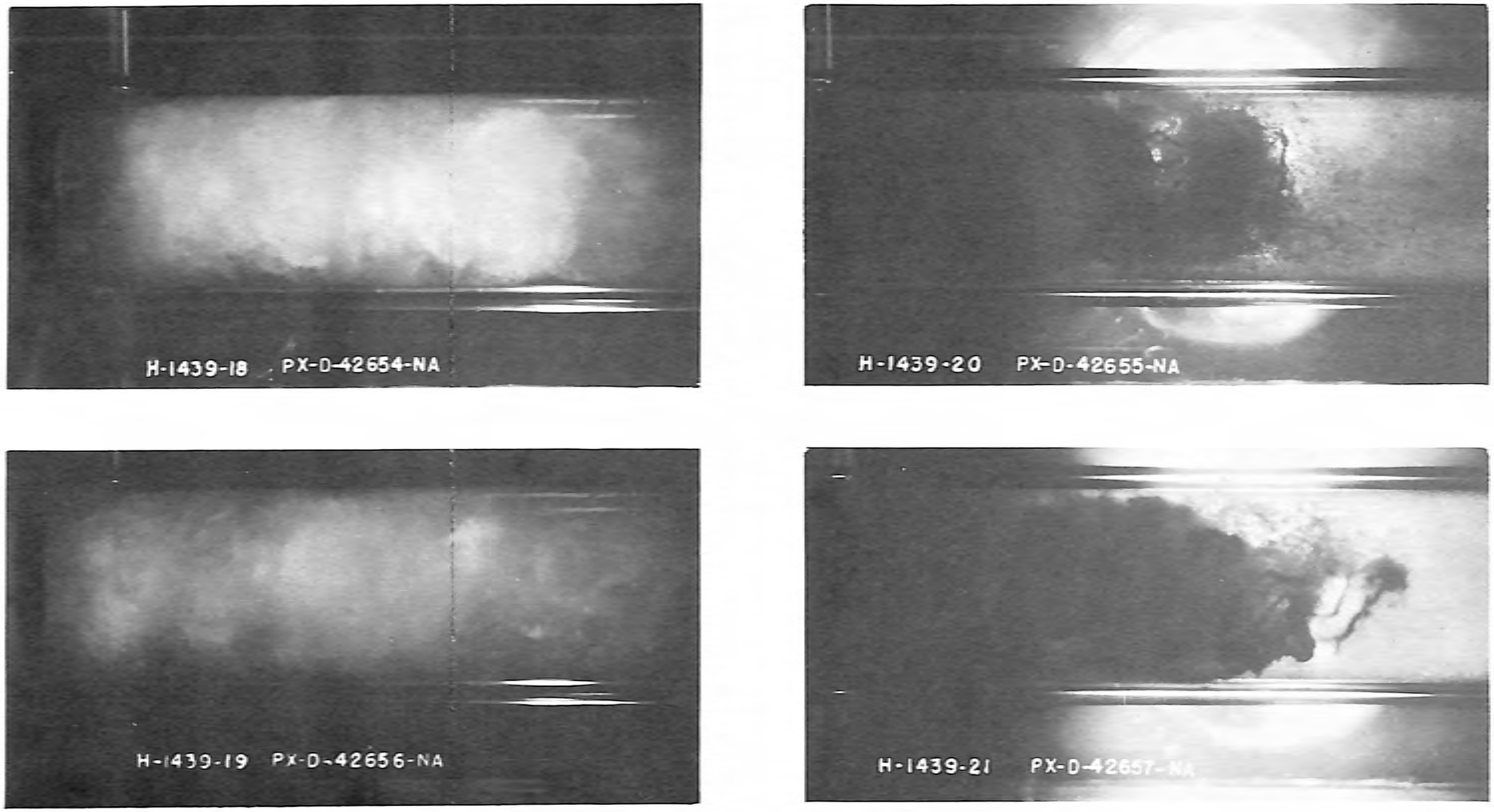
A. Front lighting only - Strobolux @ 2,525 cps
Exposure 1/100 second

B. Lighted from top, bottom and rear
1/10,000 second

H_T = 208 feet H_2 = 44 feet Q = 1.10 cubic feet per second
H_2 measured 16D downstream from orifice
Lighting from behind

ORIFICES AND VALVES IN PIPELINES

Photographs of Cavitation Downstream from an Orifice - K = 0.45
1.625-inch Orifice in 3.000-inch Pipe



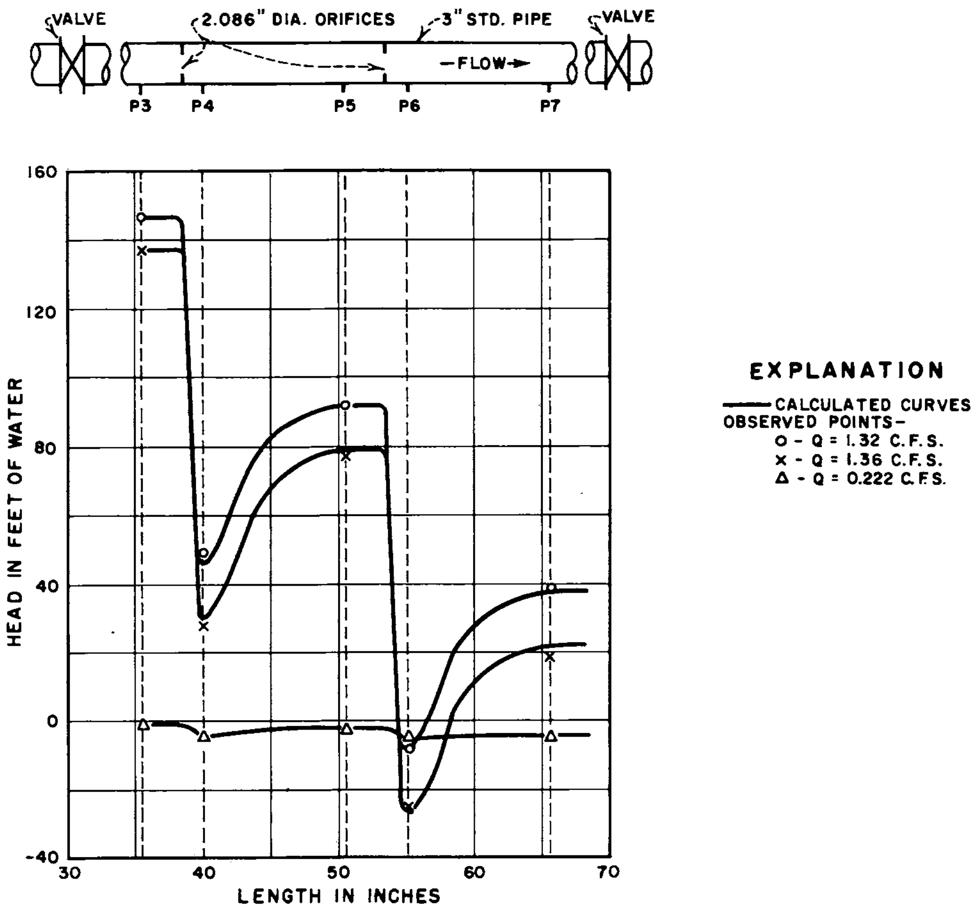
A. Front lighting only - Strobolux @ 2,525 cps 1/100 second

B. Lighted from top, bottom, and rear Exposure 1/10,000 second

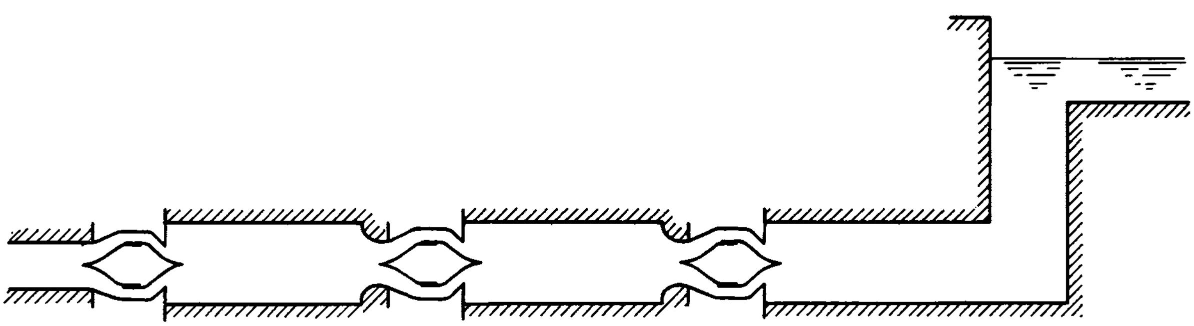
H_T = 165 feet H_2 = -18 feet Q = 0.984 cubic feet per second
H_2 measured 16D downstream from orifice
Lighting from behind

ORIFICES AND VALVES IN PIPELINES

Photographs of Cavitation Downstream from an Orifice - K = 0.05
1.625-inch Orifice in 3.000-inch Pipe



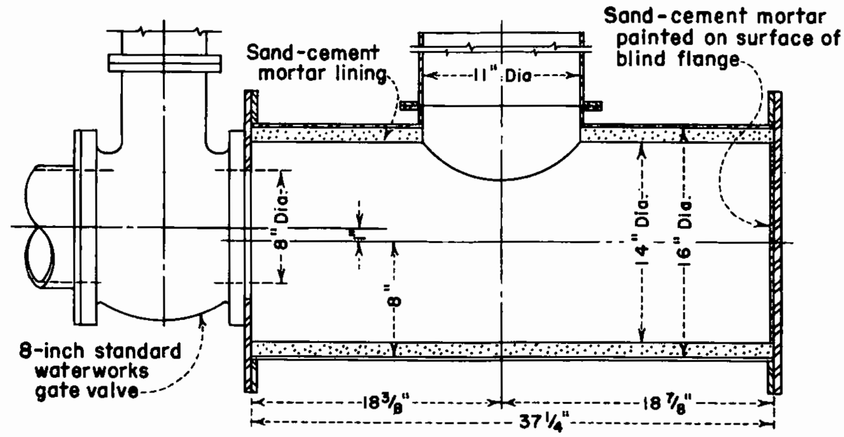
A. HYDRAULIC GRADIENT FOR ORIFICES IN TANDEM IN PIPELINES



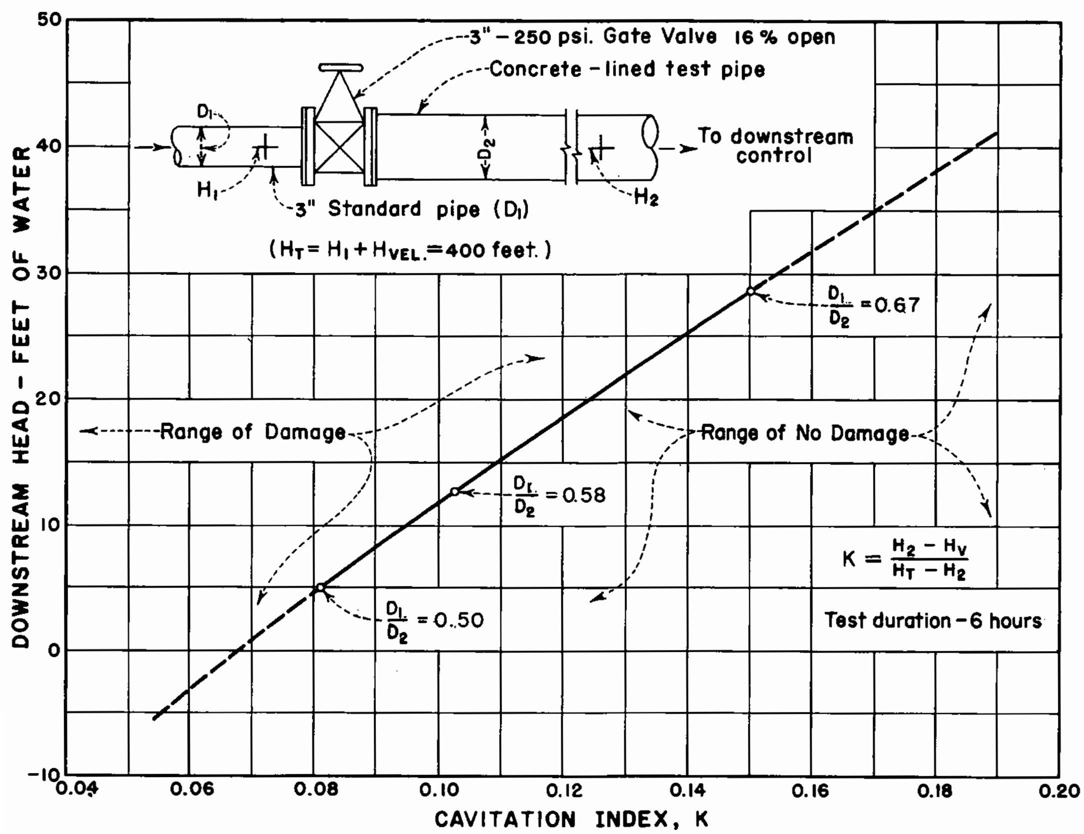
B. PRESSURE REDUCER FOR TURBINE BY-PASS, NEEDLE VALVES IN SERIES

ORIFICES AND VALVES IN PIPELINES

MULTIPLE ORIFICES IN SERIES



A. CONCRETE LINED T-SECTION AS SUDDEN ENLARGEMENT BELOW GATE VALVE



B. EFFECT OF ENLARGEMENT SIZE ON CAVITATION DAMAGE INDEX

ORIFICES AND VALVES IN PIPELINES

ENLARGEMENTS DOWNSTREAM FROM GATE VALVES

GPO 840-015

CONVERSION FACTORS--BRITISH TO METRIC UNITS OF MEASUREMENT

The following conversion factors adopted by the Bureau of Reclamation are those published by the American Society for Testing and Materials (ASTM Metric Practice Guide, January 1964) except that additional factors (*) commonly used in the Bureau have been added. Further discussion of definitions of quantities and units is given on pages 10-11 of the ASTM Metric Practice Guide.

The metric units and conversion factors adopted by the ASTM are based on the "International System of Units" (designated SI for Systeme International d'Unites), fixed by the International Committee for Weights and Measures; this system is also known as the Giorgi or MKSA (meter-kilogram (mass)-second-ampere) system. This system has been adopted by the International Organization for Standardization in ISO Recommendation R-31.

The metric technical unit of force is the kilogram-force; this is the force which, when applied to a body having a mass of 1 kg, gives it an acceleration of 9.80665 m/sec/sec, the standard acceleration of free fall toward the earth's center for sea level at 45 deg latitude. The metric unit of force in SI units is the newton (N), which is defined as that force which, when applied to a body having a mass of 1 kg, gives it an acceleration of 1 m/sec/sec. These units must be distinguished from the (inconstant) local weight of a body having a mass of 1 kg; that is, the weight of a body is that force with which a body is attracted to the earth and is equal to the mass of a body multiplied by the acceleration due to gravity. However, because it is general practice to use "pound" rather than the technically correct term "pound-force," the term "kilogram" (or derived mass unit) has been used in this guide instead of "kilogram-force" in expressing the conversion factors for forces. The newton unit of force will find increasing use, and is essential in SI units.

Table 1

QUANTITIES AND UNITS OF SPACE

Multiply	By	To obtain
	LENGTH	
Mil	25.4 (exactly)	Micron
Inches	25.4 (exactly)	Millimeters
	2.54 (exactly)*	Centimeters
Feet	30.48 (exactly)	Centimeters
	0.3048 (exactly)*	Meters
	0.0003048 (exactly)*	Kilometers
Yards	0.9144 (exactly)	Meters
Miles (statute)	1,609.344 (exactly)*	Meters
	1.609344 (exactly)	Kilometers
	AREA	
Square inches	6.4516 (exactly)	Square centimeters
Square feet	929.03 (exactly)*	Square centimeters
	0.092903 (exactly)	Square meters
Square yards	0.836127	Square meters
Acres	0.40469*	Hectares
	4,046.9*	Square meters
	0.0040469*	Square kilometers
Square miles	2.58999	Square kilometers
	VOLUME	
Cubic inches	16.3871	Cubic centimeters
Cubic feet	0.0283168	Cubic meters
Cubic yards	0.764555	Cubic meters
	CAPACITY	
Fluid ounces (U.S.)	29.5737	Cubic centimeters
	29.5729	Milliliters
Liquid pints (U.S.)	0.473179	Cubic decimeters
	0.473166	Liters
Quarts (U.S.)	9,463.58	Cubic centimeters
	0.946358	Liters
Gallons (U.S.)	3,785.43*	Cubic centimeters
	3.78543	Cubic decimeters
	3.78533	Liters
	0.00378543*	Cubic meters
Gallons (U.K.)	4.54609	Cubic decimeters
	4.54596	Liters
Cubic feet	28.3160	Liters
Cubic yards	764.55*	Liters
Acre-feet	1,233.5*	Cubic meters
	1,233,500*	Liters

Table II

QUANTITIES AND UNITS OF MECHANICS

MASS

Multiply	By	To obtain
Grains (1/7,000 lb)	64.79891 (exactly)	Milligrams
Troy ounces (480 grains)	31.1035	Grams
Ounces (avdp)	28.3495	Grams
Pounds (avdp)	0.45359237 (exactly)	Kilograms
Short tons (2,000 lb)	907.185	Kilograms
	0.907185	Metric tons
Long tons (2,240 lb)	1,016.05	Kilograms

FORCE/AREA

Multiply	By	To obtain
Pounds per square inch	0.070307	Kilograms per square centimeter
	0.689476	Newtons per square centimeter
Pounds per square foot	4.88243	Kilograms per square meter
	47.8803	Newtons per square meter

MASS/VOLUME (DENSITY)

Multiply	By	To obtain
Ounces per cubic inch	1.72999	Grams per cubic centimeter
Pounds per cubic foot	16.0185	Kilograms per cubic meter
	0.0160185	Grams per cubic centimeter
Tons (long) per cubic yard	1.32894	Grams per cubic centimeter

MASS/CAPACITY

Multiply	By	To obtain
Ounces per gallon (U.S.)	7.4893	Grams per liter
Ounces per gallon (U.K.)	6.2362	Grams per liter
Pounds per gallon (U.S.)	119.829	Grams per liter
Pounds per gallon (U.K.)	99.779	Grams per liter

BENDING MOMENT OR TORQUE

Multiply	By	To obtain
Inch-pounds	0.011521	Meter-kilograms
	1.12985×10^{6}	Centimeter-dynes
Foot-pounds	0.138255	Meter-kilograms
	1.35582×10^{7}	Centimeter-dynes
Foot-pounds per inch	5.4431	Centimeter-kilograms per centimeter
Ounce-inches	72.008	Gram-centimeters

VELOCITY

Multiply	By	To obtain
Feet per second	30.48 (exactly)	Centimeters per second
	0.3048 (exactly)*	Meters per second
Feet per year	0.965873×10^{-6}*	Centimeters per second
Miles per hour	1.609344 (exactly)	Kilometers per hour
	0.44704 (exactly)	Meters per second

ACCELERATION*

Multiply	By	To obtain
Feet per second^2	0.3048*	Meters per second^2

FLOW

Multiply	By	To obtain
Cubic feet per second (second-feet)	0.028317*	Cubic meters per second
Cubic feet per minute	0.4719	Liters per second
Gallons (U.S.) per minute	0.06309	Liters per second

FORCE*

Multiply	By	To obtain
Pounds	0.453592*	Kilograms
	4.4482*	Newtons
	4.4482×10^{-5}*	Dynes

WORK AND ENERGY*

Multiply	By	To obtain
British thermal units (Btu)	0.252*	Kilogram calories
	1,055.06	Joules
Btu per pound	2.326 (exactly)	Joules per gram
Foot-pounds	1.35582*	Joules

POWER

Multiply	By	To obtain
Horsepower	745.700	Watts
Btu per hour	0.293071	Watts
Foot-pounds per second	1.35582	Watts

HEAT TRANSFER

Multiply	By	To obtain
Btu in./hr ft^2 deg F (k, thermal conductivity)	1.442	Milliwatts/cm deg C
	0.1240	Kg cal/hr m deg C
Btu ft/hr ft^2 deg F	1.4880*	Kg cal m/hr m^2 deg C
Btu/hr ft^2 deg F (C, thermal conductance)	0.568	Milliwatts/cm^2 deg C
	4.882	Kg cal/hr m^2 deg C
Deg F hr ft^2/Btu (R, thermal resistance)	1.761	Deg C cm^2/milliwatt
Btu/lb deg F (c, heat capacity)	4.1868	J/g deg C
Btu/lb deg F	1.000*	Cal/gram deg C
Ft^2/hr (thermal diffusivity)	0.2581	Cm^2/sec
	0.09290*	M^2/hr

WATER VAPOR TRANSMISSION

Multiply	By	To obtain
Grains/hr ft^2 (water vapor transmission)	16.7	Grams/24 hr m^2
Perms (permeance)	0.659	Metric perms
Perm-inches (permeability)	1.67	Metric perm-centimeters

Table III

OTHER QUANTITIES AND UNITS

Multiply	By	To obtain
Cubic feet per square foot per day (seepage)	304.8*	Liters per square meter per day
Pound-seconds per square foot (viscosity)	4.8824*	Kilogram second per square meter
Square feet per second (viscosity)	0.02903* (exactly)	Square meters per second
Fahrenheit degrees (change)*	5/9 exactly	Celsius or Kelvin degrees (change)*
Volts per mil.	0.03937	Kilovolts per millimeter
Lumens per square foot (foot-candles)	10.764	Lumens per square meter
Ohm-circular mils per foot	0.001662	Ohm-square millimeters per meter
Millicuries per cubic foot	35.3147*	Millicuries per cubic meter
Milliamps per square foot	10.7639*	Milliamps per square meter
Gallons per square yard	4.527219*	Liters per square meter
Pounds per inch	0.17858*	Kilograms per centimeter

GPO 845-237

ABSTRACT

Tests were made at heads up to 500 feet with circular concentric orifices and a gate valve in a 3-inch diameter pipeline to determine cavitation characteristics and effects of cavitation on head loss, discharge coefficients, pressure distribution, velocity distribution, and pressure fluctuations. Effects of admission of air downstream from the control station were also determined. The sizes of enlarged sections needed to prevent conduit wall damage downstream from gate valves operating under cavitating conditions were established. Conclusions were--1/ head losses across orifice stations were not significantly changed by cavitation, 2/ discharge coefficients were affected if the coefficients are based on pressures measured at a distance downstream from the orifice, 3/ most of the pressure and velocity redistribution to a normal profile occurred within five pipeline diameters of the orifice or valve station, 4/ easily-used cavitation index values were established for a range of orifice sizes to determine when cavitation would begin, and the pressure and velocity conditions needed to prevent it, 5/ pressure fluctuations significantly increased when cavitation occurred, particularly at large orifice-to-pipe diameter ratios, 6/ under steady-flow operation a simple system of orifices in series could be used safely for energy dissipation, 7/ for variable-flow systems a more complex system with variable-size orifices appears necessary.

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Office of Chief Engineer, Denver, 15 pp., 13 figs., 1963.

DESCRIPTORS--*hydraulics/submerged orifices/gate valves/*cavitation/head losses/*discharge coefficients/velocity distribution/pressure distribution/*energy dissipation/instrumentation/pitot-tube/laboratory tests/closed conduits/air/

IDENTIFIERS--Cavitation erosion/cavitation index/cavitation suppression/air admission/cylindrical pitot tube

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