DISCUSSION ON PAPER "CAVITATION EFFECT ON DISCHARGE COEFFICIENT OF THE SHARP-EDGED ORIFICE PLATE"

By

James W. Ball[(1)]

The material contained in the paper is very interesting and establishes definitely that the basic laws for orifices are true. Work of this nature signifies progress and increases the confidence of those who use the results. The authors are to be commended for their contribution.

Actually, an examination of basic relationships for orifices confirms the results. The differential head between two pressure taps, one upstream and the other downstream of an orifice in a pipeline, is an indication of the amount of flow through the orifice irrespective of the location of the taps. The change in hydraulic grade between the two taps (differential head) results from three sources: (1) head to cause the flow through the orifice, (2) head loss due to boundary friction, and (3) head loss due to turbulent eddies. The amount contributed by each of the three sources will vary with the geometry of the jet issuing from the orifice and the distance of the taps from the orifice. For fully developed turbulence and a given orifice, the flow pattern geometry in the pipe does not change noticeably, provided the downstream

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pipe is kept under back pressures which assures that the jet is surrounded by water. The coefficient of discharge for a given orifice to pipe diameter ratio will be constant for any tap location as long as this condition exists. The location of the upstream tap is not too important. However, this is not true of the downstream tap. A fixed location for the upstream tap will be considered for the following discussion. As the pressure gradient downstream from an orifice is lowered and the cavitation envelope forms and extends sufficiently to alter the flow stream geometry, the coefficient, based on taps affected by this alteration will deviate from the constant value. The smallest deviation will occur for taps least affected by the change in the stream geometry. The stream geometry upstream from the vena contracta remains essentially the same regardless of the downstream pressure; thus the discharge coefficients based on measurements at downstream taps located at or upstream from the vena contracta will not be changed by the presence of cavitation downstream from the orifice. This explains why the coefficient of discharge is essentially constant for the flange taps used in the tests discussed in the paper and why similar results shown on Figure 1 of this discussion were obtained in tests at cavitation numbers as low as 0.001 and heads up to 160 feet made in a test facility (Figure 2) in the Bureau of Reclamation Hydraulic Laboratory. This also explains why the coefficient of discharge decreases abruptly when the downstream tap is located any appreciable distance downstream from the vena contracta as is the case for part of the data shown on Figure 3. However, the coefficients in these cases remain substantially constant until the hydraulic grade in the "recovery" region moves past the tap location.

As the cavitation envelope below an orifice lengthens due to decreased back pressure and the "recovery" region moves past the downstream tap, the pressure at this tap is lowered, increasing the apparent differential head between the upstream and downstream taps while neither changing the pressure immediately downstream from the orifice nor the discharge through the orifice. This results in a decrease in coefficient as shown on the curves of Figure 3. 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.

There is a difference in numerical value of the coefficient depending on how far the downstream tap is located from the orifice. This is due mainly to the difference in friction and eddy losses to the various locations. The differences for taps located 0, 2.1, 15, and 36 inches downstream from 2-3/8-, 1-3/4- and 1-1/4-inch orifices in a 3-inch standard pipe are shown in the plots on Figure 3. The differences for the two locations farthest downstream become insignificant for the smallest orifice where the velocities in the pipe are very low and the friction and eddy loss differences become an insignificant part of the total head differential. The shape of the curves, cavitation number (K) versus coefficient of discharge (C), in any case will be essentially the same for both locations in the range where the "recovery" region moves past the tap. Where the tap is located just downstream from the vena contracta as for the 2.1-inch location, Sta.(3), on the 2-3/8-inch orifice, the coefficient will be slightly larger than for flange taps until vapor pressure reaches the tap (Figure 3).

A flow nozzle 1.333 inches in diameter (Figure 4) was tested in the same setting as the three orifices. The data for the nozzle are plotted on both Figures 1 and 4. The K_d versus C curve on Figure 1 is quite different from that for orifices, indicating that cavitation does affect the efficiency of this particular shape. Also for this particular nozzle, there seemed to be a slight change in the value of C with head, the cause of which has not been definitely determined but may be an influence introduced by the shape of the nozzle. The discharge coefficient of this nozzle is affected by cavitation irrespective of the downstream tap location.

The cavitation data given in the paper are very interesting, and their publication have brought out some interesting facts. The cavitation number for incipient cavitation obtained by ear for the 1-1/4-, 1-3/4- and 2-3/8-inch orifices tested in the Bureau of Reclamation Hydraulic Laboratory and based on the relationships given in the paper were 1.3, 1.1 and 1.0, respectively. These values appear to agree reasonably well with the paper; however, the trend is for K_{di} determined by ear to increase with decrease in β rather than increase with β as does line "L" which is defined as "where noise and vibration from cavitation became quite apparent" (Figure 6 of the paper.)

In one series of tests made in the Bureau of Reclamation Hydraulic Laboratory on the 1-3/4-inch orifice, the value of K for incipient cavitation, using the relationships given in the paper, did not agree with that given in the paper. Also, an examination of the test results made on two 2.083-inch orifices, placed in tandem in a 3-inch pipeline to serve as a pressure reducing system,[2] indicated poor agreement. In these instances it appeared that cavitation occurred at much smaller values of K than shown in the paper. Check tests on the 1-3/4-inch orifice showed the initial tests to be in error. The check tests on this orifice agreed reasonably well with the results given in the paper and it was theorized that a small air leak into the system occurred downstream from the orifice, thus preventing cavitation noise (crepitation) until a lower value of K was reached.

In the tests in which the two 2.083-inch orifices were used in tandem in a 3-inch pipeline (Figure 5), the criterion was that the noise level was acceptable. This point was therefore somewhat above that for

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(2) "Hydraulic Studies of a Pressure Reducing System for the Transformer Cooling Water--Grand Coulee Power Plant," Bureau of Reclamation Hydraulic Laboratory Report Hyd-308, March 1951, by L. V. Wilson

incipient cavitation and agreement with the data in the paper could not be expected.

The cavitation number given in the paper is not in a form which could be readily used by designers to determine whether or not there would be objectionable cavitation and vibration in an orifice installation. In studies on valves in the Bureau of Reclamation Hydraulic Laboratory[3] another form of cavitation number was found most useful for this purpose. The same relationship can be applied to orifices in a pipeline. The data for three orifices using this relationship

$$K = \frac{H_x - H_v}{H_T - H_x} \quad \text{and} \quad C = \frac{Q}{A_o \sqrt{2g}\sqrt{H_o - H_x}}$$

are plotted on Figure 3 where the symbols in both equations are defined.

Knowing the critical cavitation number K_i at which cavitation is incipient for an orifice, it is possible to determine the back pressure required to prevent cavitation for a given upstream head, or to compute the allowable upstream head for a given back pressure. It is believed that the variation of K with β', where β' is the ratio of the area of the jet at the vena contracta to that of the pipe, should be somewhat like that shown in Figure 6. This relationship was obtained by assuming general cavitation just beginning in the pipe immediately

(3) "Cavitation Characteristics of Gate Valves and Globe Valves Used as Flow Regulators Under Heads Up to About 125 Feet." Paper 56-F-10, Volume 6, November 1957, A.S.M.E. Transactions, by James W. Ball.

downstream and a loss to the "recovery" pressure downstream equal to the Borda Loss expressed by $H_L = (V_o - V_p)^2/2g$. V_o = velocity of the contracted jet[(4)] and V_p = velocity in the pipe downstream.

Values of K_i determined by ear for the three orifices and based on the relationship in Figure 3 for taps located at Sta. 4 and Sta. 5 (4.9 and 11.7 pipe diameters downstream of the orifices) are also plotted on Figure 6. The curve formed by these values is much higher than that based on the occurrence of vapor pressure immediately downstream from the orifices and the Borda Loss. This signifies that cavitation occurs before vapor pressure is registered at flange or vena contracta taps.

Several tests on the three orifices were made for conditions where vapor pressure at the downstream flange tap was believed to be incipient. The values of K for this condition, based on downstream taps at Stations 4 and 5, were plotted for comparison with the values obtained using the Borda Loss relationship. The K values for the orifices, based on heads at Stations 4 and 5 and obtained from plots of the pressure data, were in good agreement, Figure 6. The agreement is excellent considering that the information was taken from tests not made for that specific purpose, that it was questionable that vapor pressure was incipient at the downstream flange taps, and that it was

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(4) Contraction coefficient obtained from page 35, "Engineering Hydraulics" by Hunter Rouse.

questionable that the taps were located or could be located to indicate the conditions assumed for the Borda Loss relationship.

The values of K_1 for the three different orifices were obtained by ear. The audibility of cavitation was noted for each test run. There were three conditions observed, (1) cavitation definitely audible, (2) cavitation definitely not audible, and (3) cavitation audibility questionable. The third condition gave the most difficulty but was very useful in selecting K_1 for each orifice.

It is questionable that cavitation actually exists in all the "manner of cavitation occurrence" definitions given in Table 2 of the paper. It is believed that the bubbles described for a cavitation number of 2.5 are those formed by air being released from solution as the flow passes into the low pressure zone immediately downstream from the orifice rather than vapor cavities. Bubbles formed by air coming out of solution and then moving in the eddy zone downstream, as described in Table 2 for a cavitation number of 1.0, are not necessarily vapor cavities and cannot be considered definite proof that cavitation is present. Air coming out of solution in sufficient quantity may influence the point of incipient cavitation and muffle the crepitation. Similar tests in a water tunnel, using de-aerated water, would be of particular interest.

It is of interest at this time to point out that because of relatively high head losses, orifices can be used in tandem for energy dissipation, Figure 5. However, because of the intense noise and vibration accompanying cavitation, it is important to prevent cavitation by providing sufficient back pressure. Also, it is imperative to consider the fact that wide fluctuations in pressure occur in the eddy zone immediately downstream from the orifice at all times when high-velocity flows are being handled. Because of the critical factors involved it is expedient and wise to make hydraulic investigations before designs employing this principle are used. More interesting facts about orifices will no doubt be forthcoming when studies are made of the transient pressures downstream.

A point of interest concerns measurement of head at pressure taps located downstream from orifices. Wide fluctuations in pressure make it imperative to obtain a sufficient number of readings to obtain average values or use ample damping in the pressure recording units to obtain average values. The consistency of data will depend on the care exercised in recording the head. This factor no doubt is the reason for the variable spread in points on the curves of the paper and this discussion.

The arrangement shown schematically in Figure 5 has been used to reduce the head from 163 feet to 38 feet for a transformer cooling water system.(5) An installation, using a needle valve for a variable orifice, has been employed successfully in reducing a head of 630 feet

(5) Same as footnote (2)

to 250 feet to supply and make use of an existing power plant designed for the lower head.(6) In both of these cases it appears that the major portion of the energy in a high-velocity jet discharging into a sudden enlargement is dissipated in the first 5 to 6 diameters length of the enlarged section. This condition of course is not true if a general zone of cavitation is permitted to occur immediately downstream from the point of entry. Extremely high heads may also alter this observed characteristic.

Some work not yet published has been performed by the Bureau of Reclamation Hydraulic Laboratory on the back pressure necessary to prevent destructive cavitation for valves discharging into sudden enlargements of various sizes. Briefly, the tests have shown that cavitation numbers, based on the relationship given in Figure 3, may be as low as 0.20 without causing pitting of the walls of the sudden enlargement when a diameter ratio, valve to pipe, of 1.75 or more is used. Also, the admission of a very small amount of air near the downstream end of the valve will quiet crepitation immensely.

The standard hydraulic symbols C and K, for discharge coefficient and cavitation number, are used in this discussion. The use of K for both, as in the paper, is somewhat confusing.

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(6) "Pressure Reducing Plant for a Water Power Station," by K. Stierlin. Escher Wyss News, Volume 30, 1957, No. 1.

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.

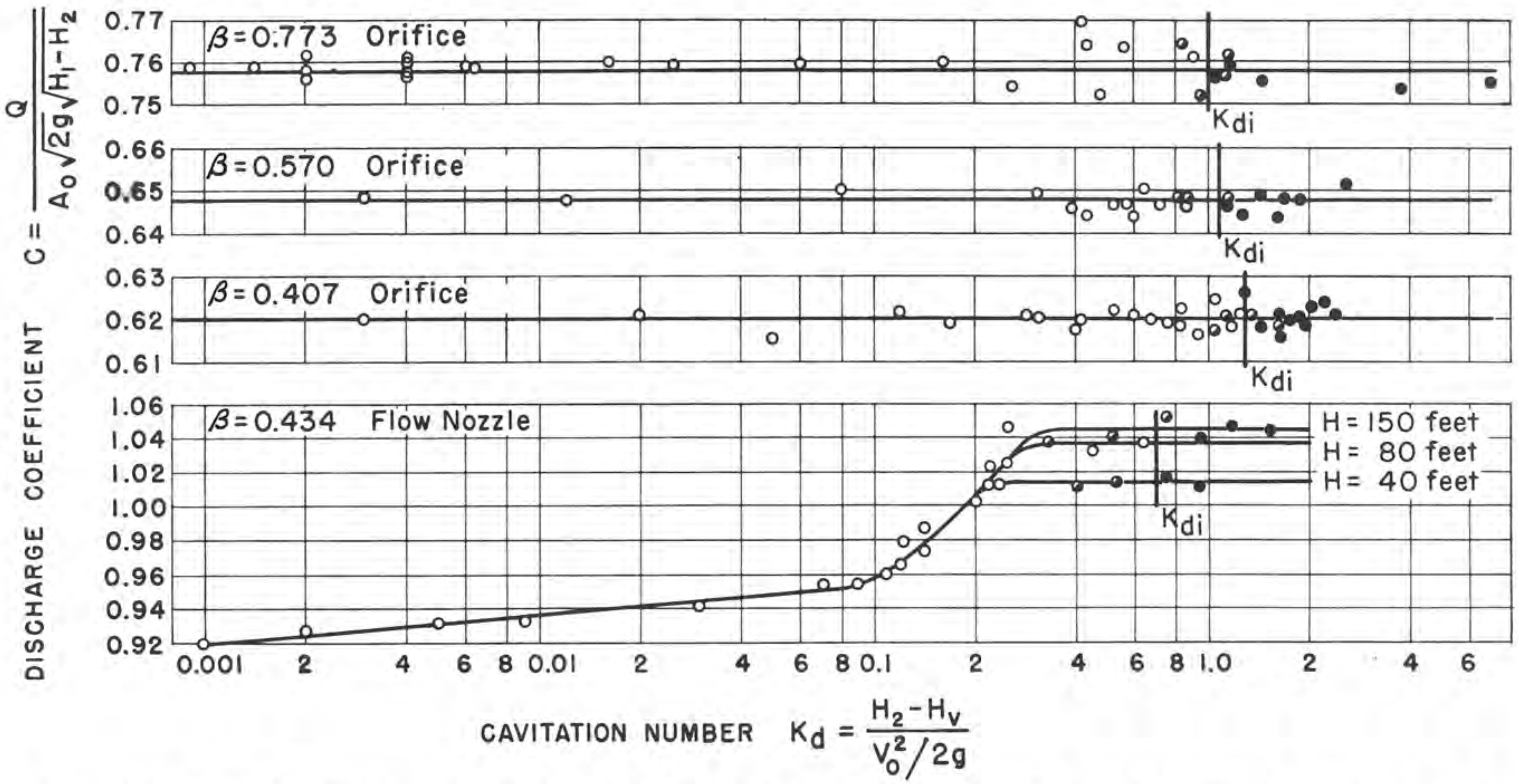

FIGURE 1. VARIATION OF DISCHARGE COEFFICIENT C WITH CAVITATION NUMBER K_d

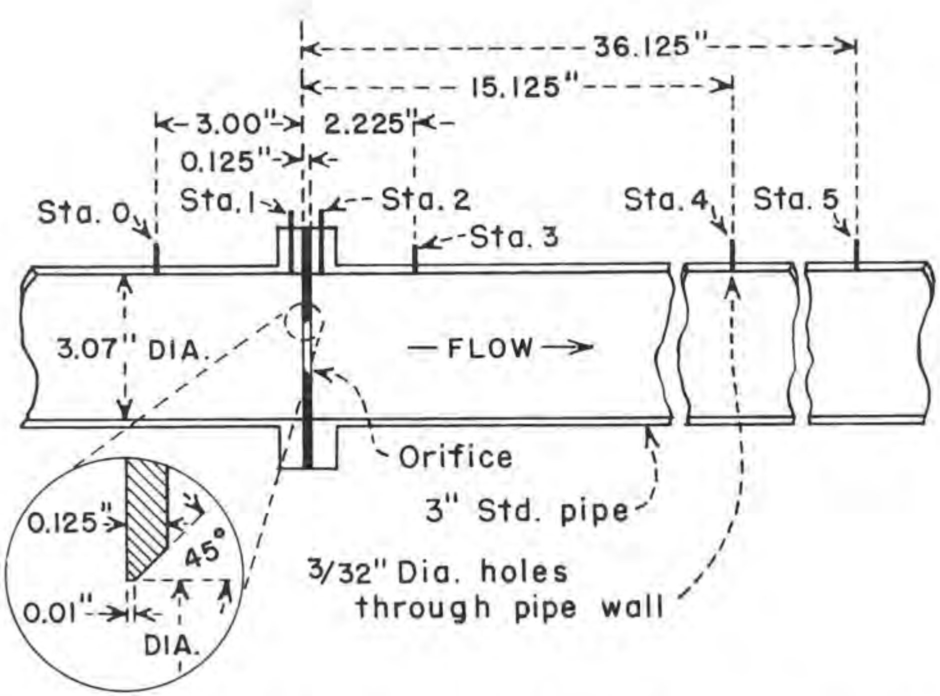

FIGURE 2. TEST FACILITIES FOR ORIFICES AND FLOW NOZZLE.

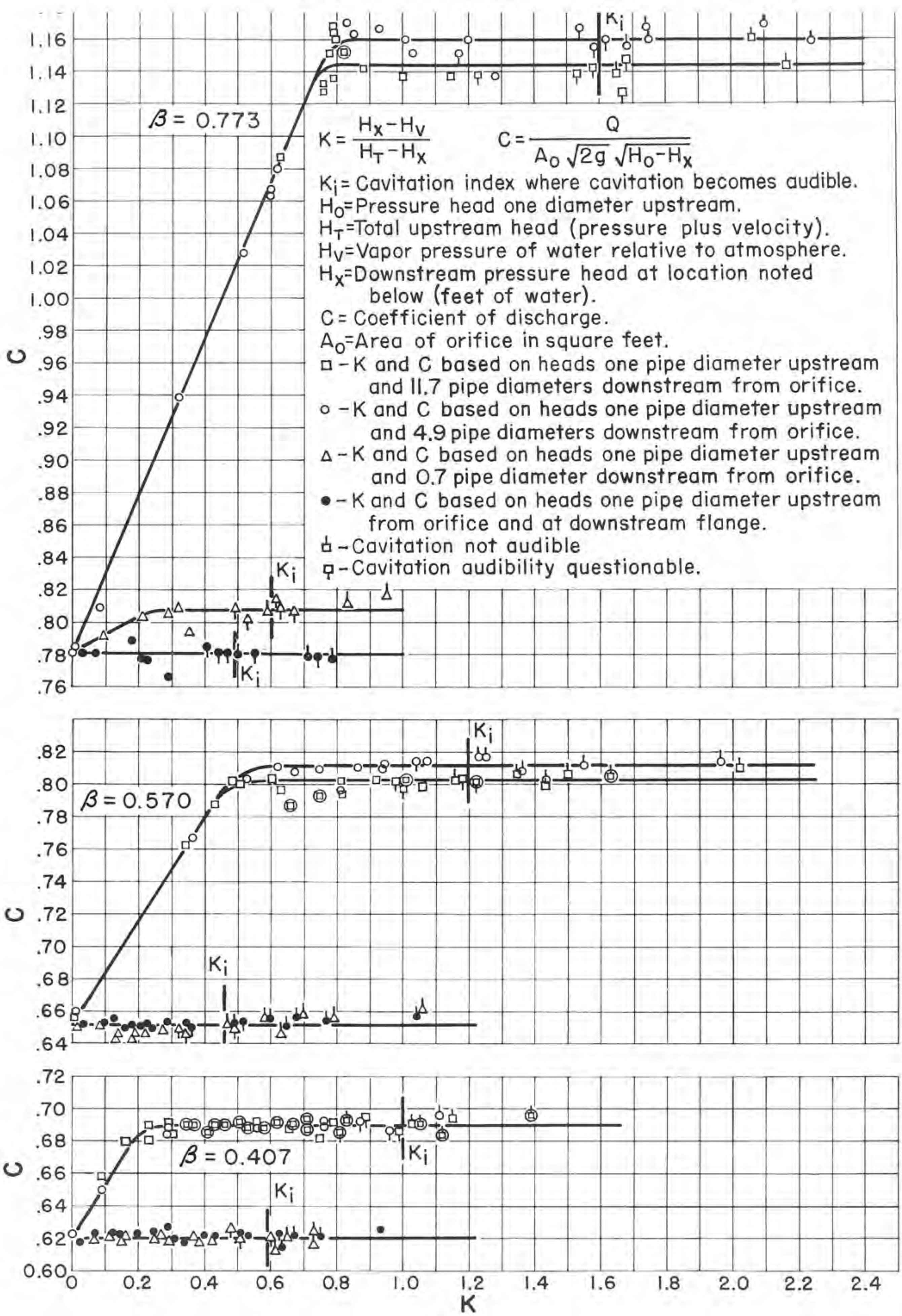

FIGURE 3. VARIATION OF DISCHARGE COEFFICIENT C WITH CAVITATION NUMBER K FOR ORIFICES 1¼-, 1¾- AND 2⅜- INCHES IN DIAMETER

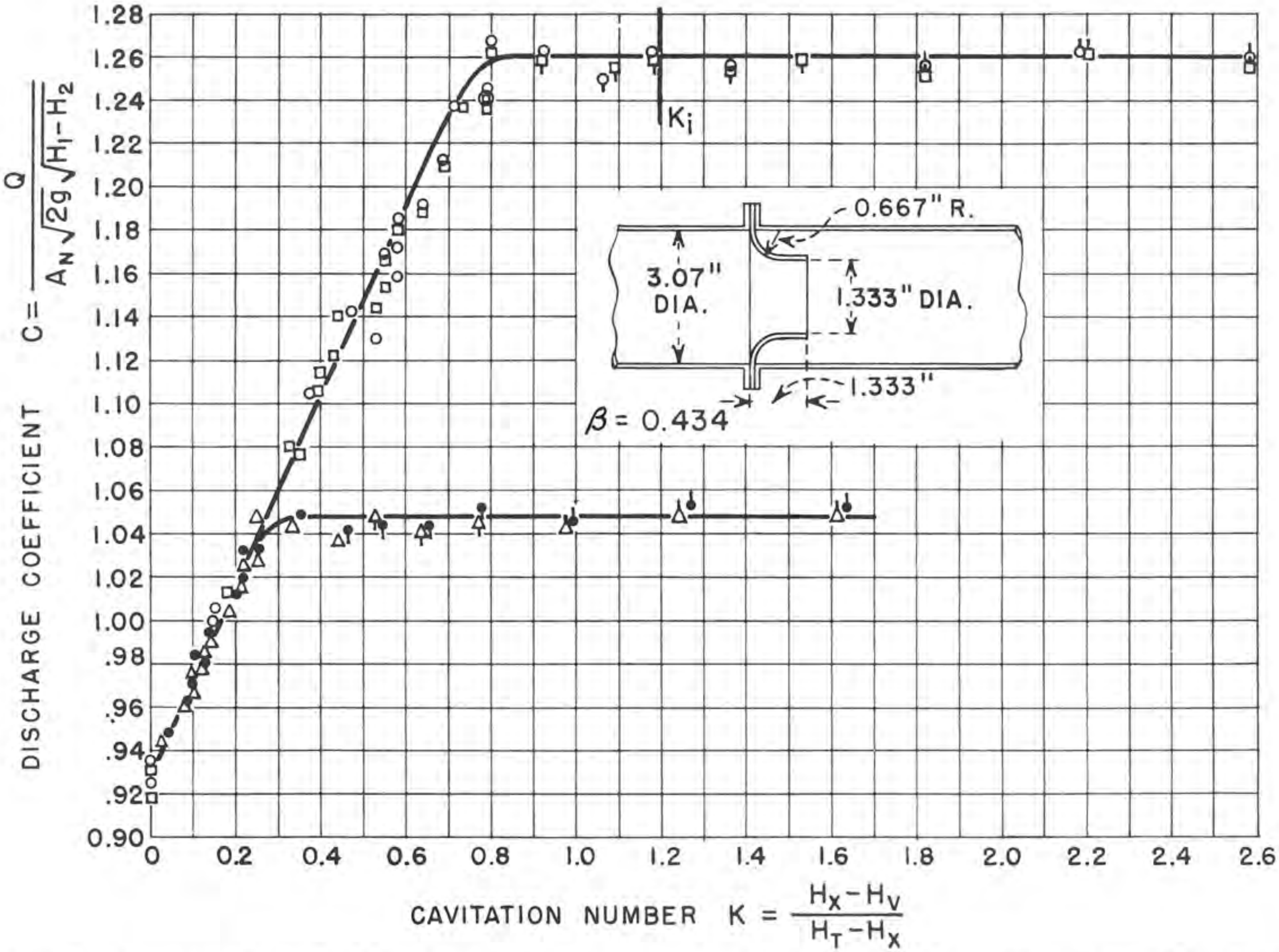

FIGURE 4. VARIATION OF DISCHARGE COEFFICIENT C WITH CAVITATION NUMBER K FOR FLOW NOZZLE 1.333 INCHES IN DIAMETER.

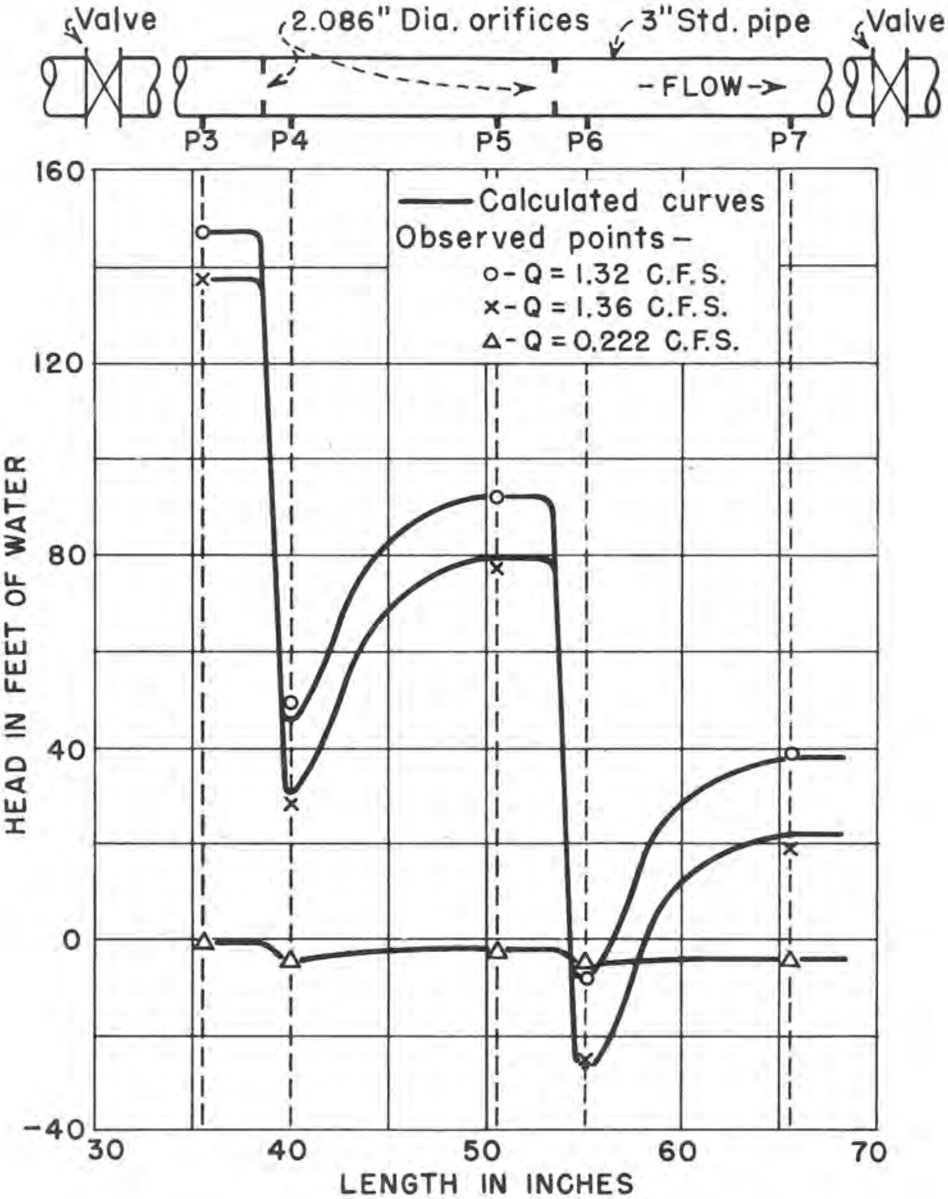

FIGURE 5. HYDRAULIC GRADIENT FOR ORIFICES IN TANDEM IN PIPELINES.

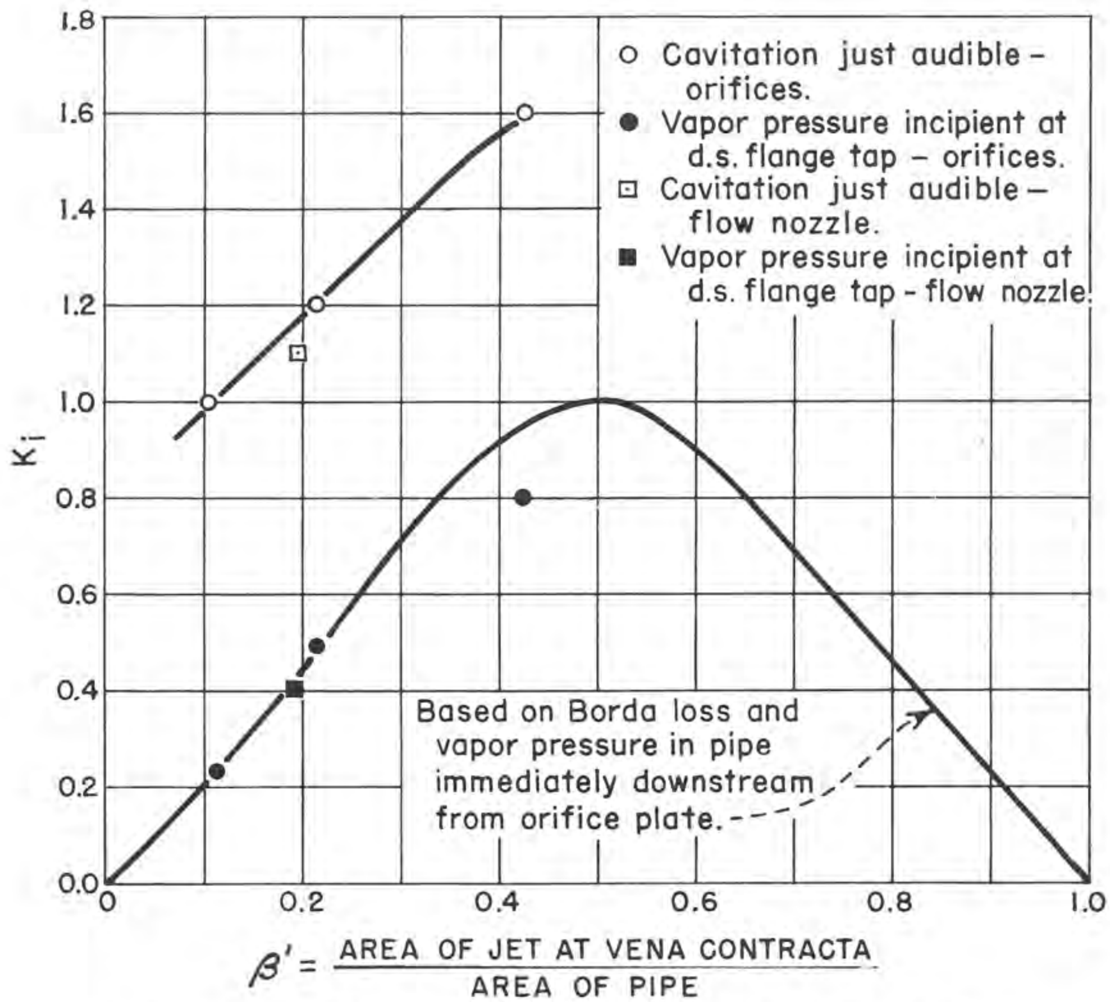

FIGURE 6. CRITICAL CAVITATION NUMBER K_i FOR ORIFICES AND FLOW NOZZLE.

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Cavitation Effect on the Discharge Coefficient of the Sharp-Edged Orifice Plate[1]

The object of this paper is to investigate the effects of cavitation on the discharge coefficient of sharp-edged orifice plates with reference to various degrees of cavitation as defined by a cavitation number. The experimental data described in this paper substantiate the fact that cavitation can exist to a minimum cavitation number of 0.2 without introducing errors in the orifice discharge coefficient in excess of the normal expected accuracy. In addition to this, it was found that the use of air-inhalation to suppress the vibration and noise from the cavitation had no effect on the discharge coefficient.

Introduction

While it is desirable to use sharp-edged orifice plates for water flow measurement, there has been some concern in the use of this device where cavitation is present, particularly in regard to small size pipes with relatively high velocities. While J. F. Bailey [1][2] and G. Ruppel [2] have made investigations regarding cavitation, the former experimenting with high water temperatures of 196.9 F and 209.1 F and the latter experimenting in large Reynolds number ranges, neither of these papers dealt directly with the cavitation effects on the discharge coefficient of the sharp-edged orifice plate. To this date, no literature nor papers dealing directly with the stated problem have been known to us.

It is the intent of this paper, therefore, to investigate and clarify the effect of cavitation occurrence on the discharge coefficient of sharp-edged orifice plates and to set allowable limits for cavitation in order to remain within acceptable flow measurement standards. The experiment described in this paper covered the range of cavitation numbers K_d down to 0.2, showing that no significant errors in flow measurement exist over this range of cavitation occurrence in the orifice sizes tested. It was also shown that air inhalation, used to suppress the vibration and noise due to cavitation, does not show a significant effect on the orifice discharge coefficient.

Orifice and Orifice Piping

The orifice flange used in the experiment employed corner taps conforming to the Japanese Engineering Standard [3] shown in Fig. 1. The dimensions of the JES flange and orifice assembly are quite similar to those used by the ASME [4] and VDI [5]. Five different-diameter orifice plates were used in this experiment, as tabulated in Table 1.

The flange and orifice were installed in the piping arrangement, as shown in Fig. 2. Straightening vanes of 12 mm square lattice, 0.8 mm thick, were employed 25 diameters upstream from the orifice to produce desirable metering conditions. The distance PS was varied to produce the required total straight run of pipe for the different-diam-ratio orifice plates, in accordance with the following tabulation:

$$PO = 30\ D \text{ for } \beta \leqq 0.499$$

$$PO = 39\ D \text{ for } \beta \geqq 0.593$$

The orifice plates were made of bronze with a 15-cm length of clear plastic immediately following the orifice flange for purposes of observation. The water was pumped by means of centrifugal pump C through the flow section to a tank containing a right-angled triangular notch weir T suitable for flow measurement.

Table 1 Dimensions of specimen pipe and orifice plates

Pipe diameter D, mm	Orifice diameter d, mm	Diameter ratio $\beta = d/D$
104.10	23.33	0.224[a]
	46.69	0.448
	51.94	0.499
	61.73	0.593
105.26	23.33	0.222
	46.69	0.444
	66.60	0.633[a]

[a] The orifice diameters were selected for even values of m, $(d/D)^2$.

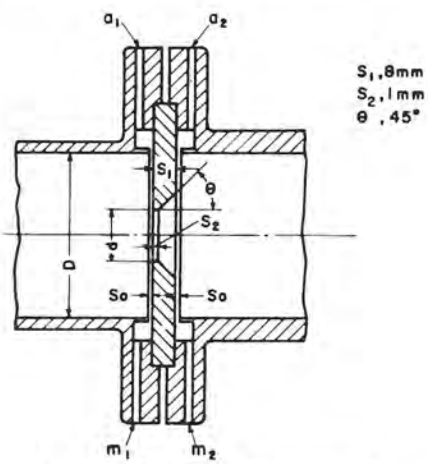

Fig. 1 Orifice plate and pressure taps

[1] English translation edited by D. M. Stough, Senior Meter Engineer, Hagan Chemicals & Controls, Inc., Pittsburgh, Pa.

[2] Numbers in brackets designate References at end of paper.

Contributed by the Research Committee on Fluid Meters for presentation at the Annual Meeting, New York, N. Y., November 30–December 5, 1958, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, March 8, 1957. Paper No. 58—A-93.

Discussion on this paper will be accepted at ASME Headquarters until January 10, 1959

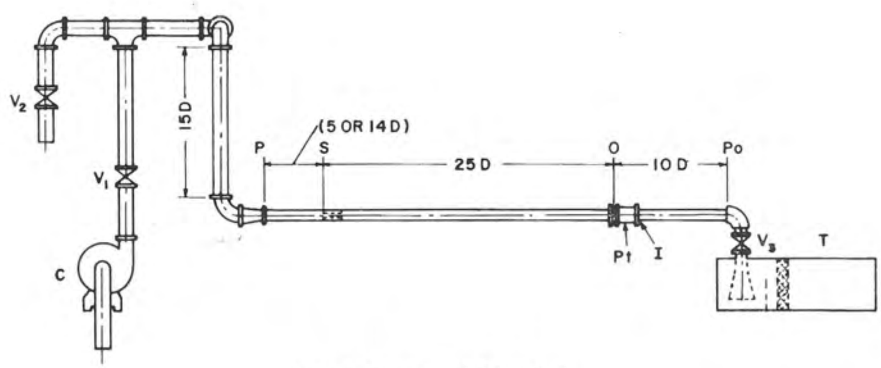

Fig. 2 Experimental installation

Method of Test

A U-tube mercury manometer was used to measure the pressure difference across the orifice plates at taps m_1 and m_2, with a measurement of static pressure at tap m_2. It was found necessary to frequently vent the air accumulated in the piping by means of vent cocks located at taps a_1 and a_2. The rate of flow through the orifice plates was measured by means of the right-angled triangular notch T, which had been previously calibrated by means of a volume tank at the Institute of High Speed Mechanics of Tōhoku University. The rate of flow and static pressure were regulated by means of valves v_1, v_2, and v_3.

The formation and size of the cavitation were observed and photographed throughout the experiment by means of the plastic section immediately following the orifice flange. These photographs are shown in Fig. 3 for various values of K_d [6], taken at a flash duration of 1/100,000 sec. The photographs show the cavitation occurrence much clearer than the naked eye.

The discharge coefficient K, the cavitation number K_d, and the pipe Reynolds number R_D were calculated from the following formulas:

$$K = \frac{Q}{\dfrac{\pi d^2}{4}\left[\dfrac{2g}{\gamma}(P_1 - P_2)\right]^{1/2}}$$

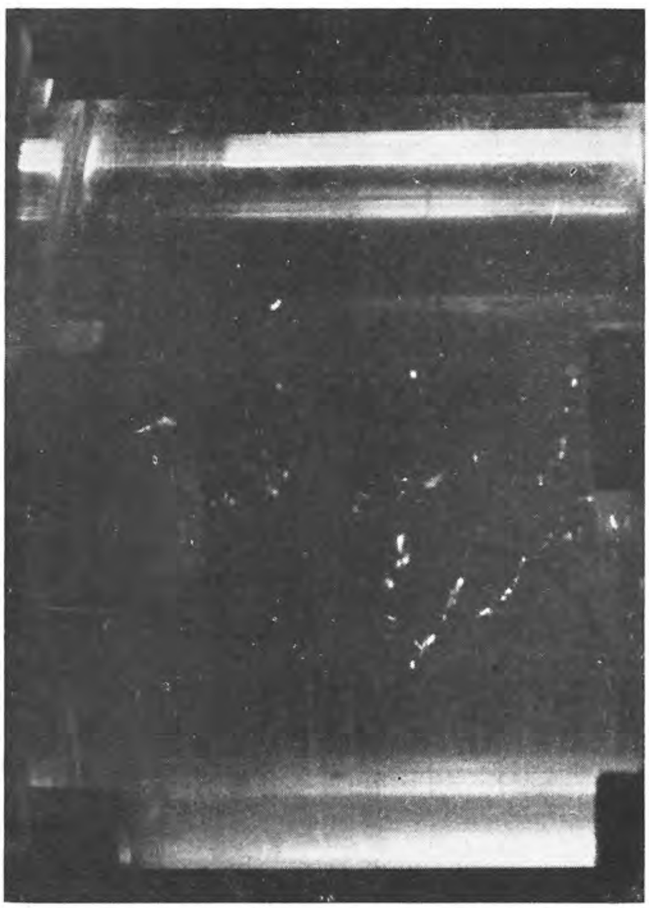

Fig. 3(a) Flash duration 1/100,000 sec; $t_w = 21.6$ C; $\beta = 0.444$; $K_d = 1.50$

Fig. 3(b) Flash duration 1/100,000 sec; $t_w = 21.6$ C; $\beta = 0.444$; $K_d = 1.10$

Fig. 3 Moment photographs of appearance of cavitation occurrence

where

Q = flow rate through pipe, ft^3 per sec
d = orifice diameter, ft
g = acceleration due to gravity, ft per sec^2
P_1 = static pressure at upstream tap, ft
P_2 = static pressure at downstream tap, ft
γ = specific gravity of water, dimensionless

$$K_d = \frac{2g(P_2 - P_d)}{V_o^2}$$

where

P_d = vapor pressure of water, ft
V_o = average velocity through orifice, ft per sec

$$R_D = \frac{VD}{\nu}$$

where

V = average velocity through pipe, ft per sec
D = inside pipe diameter, ft
ν = kinematic viscosity of water, ft^2 per sec

The point I shown in Fig. 2, located 14 cm downstream from the orifice plate, is a 6-mm-diam hole drilled in the pipe to permit air inhalation. This hole was used during certain portions of the experiment to suppress vibration and noise due to excessive cavitation, and will be discussed later.

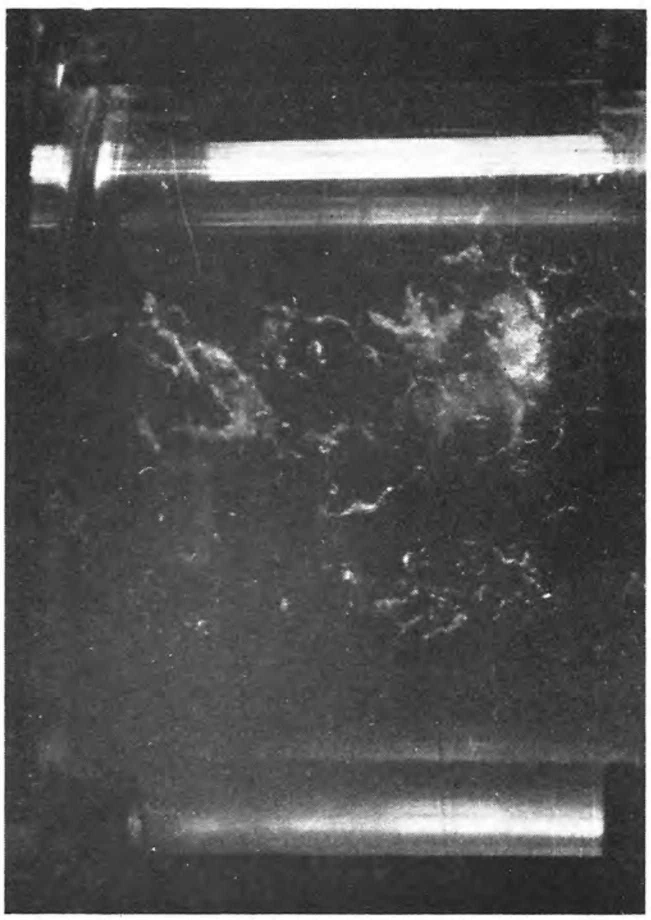

Fig. 3(c) Flash duration 1/100,000 sec; t_w = 21.6 C; β = 0.444; K_d = 0.90

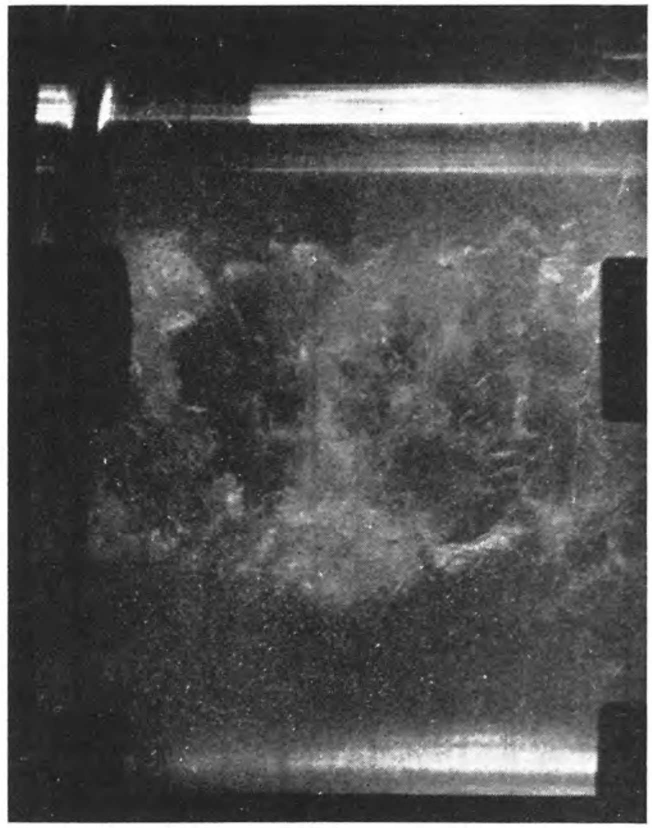

Fig. 3(d) Flash duration 1/100,000 sec; t_w = 21.6 C; β = 0.444; K_d = 0.70

Fig. 3(e) Flash duration 1/100,000 sec; t_w = 21.6 C; β = 0.444; K_d = 0.30

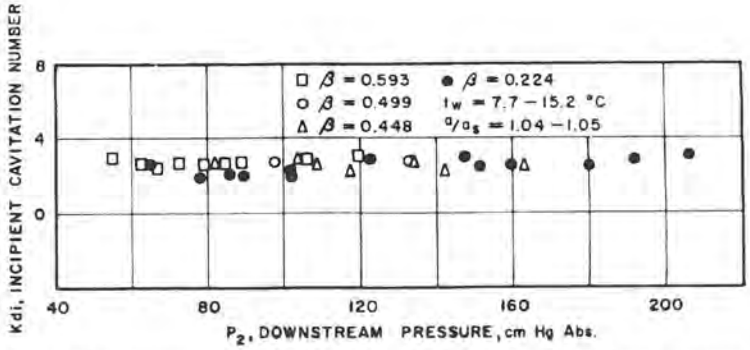

Fig. 4 Relation between incipient cavitation number K_{di} and downstream pressure P_2

Manner of Cavitation Occurrence

The relation between the incipient cavitation number K_{di}, defined as the state of flow at which small bubbles of cavitation are seen only intermittently, and the absolute static pressure P_2 at the downstream pressure tap m_2 is illustrated in Fig. 4. As shown in Fig. 4, the incipient cavitation number is approximately 2.5 irrespective of the orifice diameter ratio, β.

The symbol t_w is the water temperature, a is the absolute air content of the water, and a_s is the saturation air content of the water. It should be noted that the ratio of a/a_s is normally very close to one, and that changes in this ratio affect only slightly the incipient cavitation number and have no particular effect on cavitation numbers under the incipient cavitation number.

As a matter of convenience the observed conditions of cavitation with respect to cavitation number have been tabulated in Table 2.

Table 2 Manner of cavitation occurrence

Cavitation number K_d	Manner of cavitation occurrence
2.5	Small cavitation bubbles are seen by the unaided eyes intermittently
1.0	Small bubbles float in the dead space
0.6	Cavitation occurs from the orifice edge; somewhat unstable flow; large noise
0.25	The pipe is almost filled with cavitation bubbles; very unstable flow; large noise

Variation of Discharge Coefficient With Changing Values of Cavitation Number

A study of the orifice discharge coefficient K was made with respect to different values of cavitation number for four different-diameter-ratio orifice plates. This study is shown in Fig. 5. It can be seen clearly in Fig. 5 that no change in the discharge coefficient was apparent between the noncavitating and cavitating flow condition down to a cavitation number of 0.2. Below line L the noise and vibration from the cavitation became quite apparent.

Variation of Discharge Coefficient With Changing Values of Reynolds Number

The orifice discharge coefficients K are plotted against pipe Reynolds number R_D for various orifice-diameter ratios as shown in Fig. 6.

It is clear from this figure that the variations of the discharge coefficients with changing values of Reynolds number are smaller than the allowable tolerance of the standards in this experiment.

Table 3 Comparison between standard and experimental values of discharge coefficient K

Diameter ratio $\beta = d/D$	Discharge coefficient K: Standard value	Experimental value
0.224	0.598	0.598 ± 0.002
0.448	0.615	0.615 ± 0.003
0.499	0.623	0.623 ± 0.003
0.593	0.647	0.650 ± 0.003
0.222		
0.444		
0.633		

When the cavitation occurs, even violently ($K_d = 0.2$), the dispersion of the experimental values obtained is less than ±0.5 per cent. Therefore the variation of the discharge coefficient is smaller than the allowable tolerances stipulated by the following standards:

Standard	Diameter ratio	Allowable tolerance	Type
JIS	0.224	±0.5 per cent	Corner taps
	0.835	±1.0 per cent	
ASME	$0.15 \leq \beta \leq 0.70$	±0.5 per cent	Corner taps
DIN	$0.224 \leq \beta \leq 0.59$	±0.5 per cent	Corner taps
	0.835	±1.0 per cent	

Table 3 shows a tabulation of the discharge coefficients obtained experimentally for the various diameter-ratio orifice plates tested. This table also lists the DIN standard coefficients for the same diameter-ratio orifice plates. It will be noted that the discharge coefficient for the 0.593-diameter-ratio orifice plate is approximately 0.4 per cent higher than the standard value, which is attributed to a formation of rust in the upstream orifice piping, appearing before this orifice was tested. Table 3 substantiates that the test data for the various orifice plates agree with the standard DIN data within the allowable tolerance of ±0.5 per cent.

Counter Measure for Vibration and Noise

The objectionable noise and vibration noted in the lower cavitation number regions in Fig. 5 and tabulated in Table 4 can be reduced or eliminated by the use of air-inhalation. In this experiment a quantity of air was allowed to enter the piping at a point 14 cm downstream from the orifice plate through a 6-mm-diam hole.

Fig. 7 shows the effect of different air-inhalation coefficients on the discharge coefficients of various diameter-ratio orifice plates. The air-inhalation coefficient, q/Q, is the ratio of inhaled air volume at the downstream tap conditions to the flowing volume. Fig. 7 shows that the discharge coefficients for the lower diameter

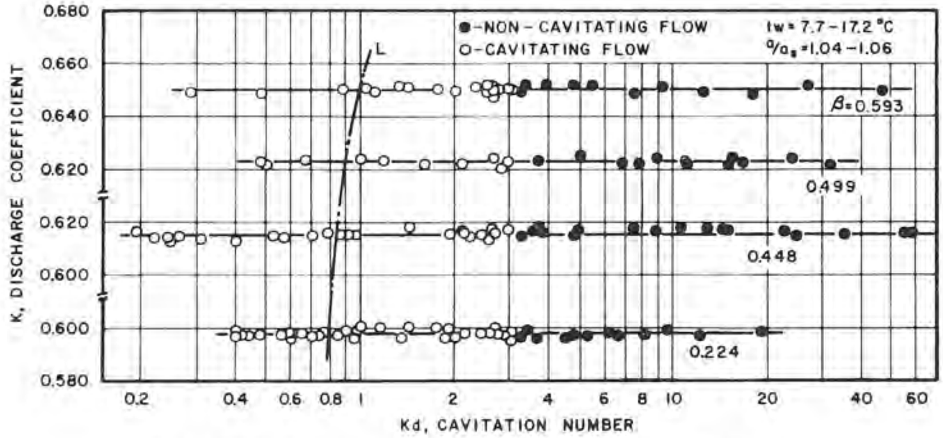

Fig. 5 Relation between cavitation number K_d and discharge coefficient K

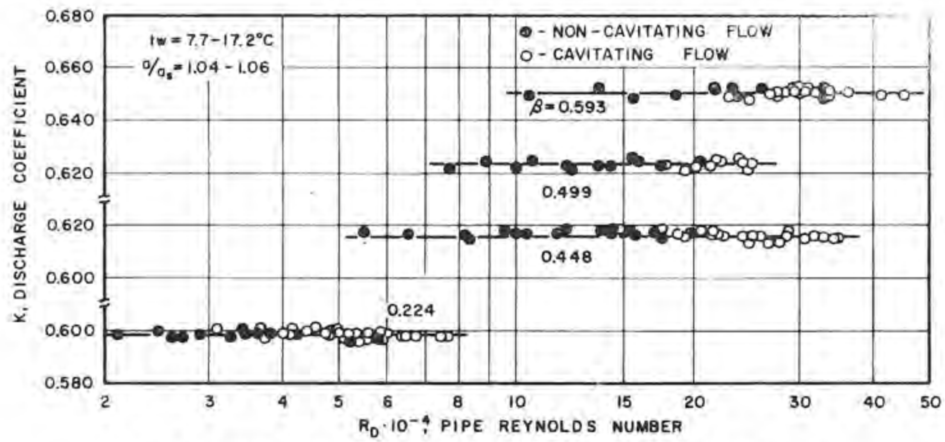

Fig. 6 Relation between Reynolds number R_D and discharge coefficient K

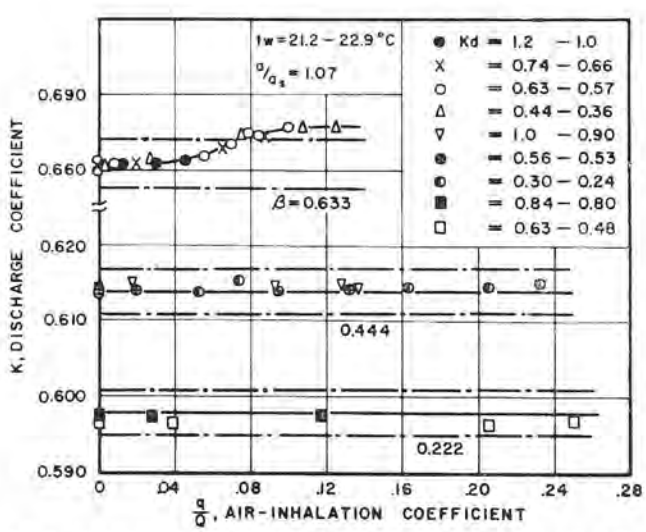

Fig. 7 Variation of discharge coefficient K with changing air-inhalation coefficient q/Q

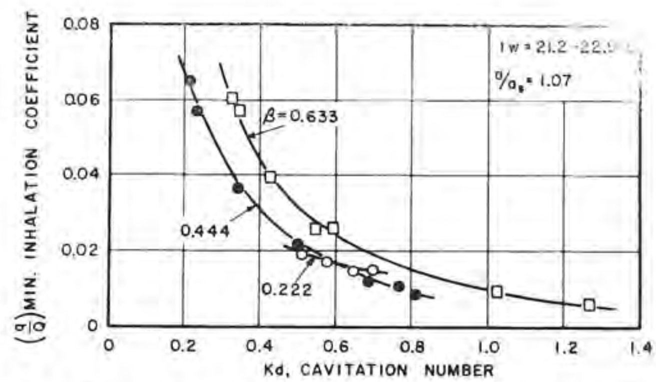

Fig. 8 Variation of cavitation number K_d with minimum air-inhalation coefficient $(q/Q)_{min}$ required to suppress vibration and noise

Table 4 Minimum cavitation number K_d producing unstable flow for each diameter ratio

β	0.22	0.45	0.50	0.60	0.63
K_d	0.80	0.85	0.90	1.00	1.25

ratio orifice plates, namely, for β equals 0.222 and β equals 0.444, remain within the allowable tolerance of ± 0.5 per cent with an air-inhalation coefficient as great as 0.25, while the air-inhalation coefficient for the higher diameter-ratio orifice plate, namely, 0.633, must not be greater than 0.08 to maintain the allowable tolerance for the discharge coefficient.

Fig. 8 shows the relationship between the minimum air-inhalation coefficient and cavitation number for various diameter ratio orifice plates to produce optimum noise and vibration suppression. It may be noted that the minimum air-inhalation coefficient required to suppress the vibration and noise increases with a desumcrease in cavitation number.

Conclusion

The results obtained concerning the effects of cavitation on the discharge coefficients of standard sharp-edged orifice plates are summarized as follows:

1 The discharge coefficient K is not influenced beyond the standard allowable tolerances to a minimum cavitation number K_d of 0.2.

2 The incipient cavitation number is approximately 2.5 and is independent of the orifice-diameter ratio in the range of the test orifices, the experimental velocity, and the static pressure.

3 The method of air-inhalation may be employed to suppress the vibration and noise due to cavitation without influencing the orifice-discharge coefficient beyond the allowable tolerance of ± 0.5 per cent.

References

1 J. F. Bailey, "Metastable Flow of Saturated Water," Trans. ASME, vol. 73, 1951, p. 1109.

2 G. Ruppel, "Die Durchflusszahlen von Normblenden und ihre Abhängigkeit von der Kantenlänge," *Zeitschrift VDI*, vol. 80, 1936, p. 1381.

3 "Standard on Measurement of Water Quantity With Pipe Orifice," *Journal of the JSME*, vol. 43, 1940, p. 295.

4 "Report of the Joint AGA-ASME Committee on Orifice Coefficients," 1935.

5 DIN 1952, VDI-Durchflussmessregeln, VDI-Verlage, 1948.

6 S. F. Crump, "Determination of Critical Pressures for the Inception of Cavitation in Fresh and Sea Water as Influenced by Air Content of the Water," David W. Taylor Model Basin, Report 575, 1949, p. 4.

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