Energy Dissipation in Low Pressure Irrigation Pipelines: II Orifices

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ABSTRACT

EROSION caused by water discharged from gated pipe openings can be reduced by dissipating excess energy with orifices placed in the gated pipe couplings. Laboratory tests were conducted to determine graphical relationships and coefficients for estimating the head loss for orifices made from galvanized sheet metal. The loss coefficient, K_o, is a function of the orifice-to-pipe diameter ratio, β_o, and can be expressed by an equation of the form $K_o = a\beta_o^b$ where **a** and **b** are empirical constants determined from the tests. Comparisons made between machined, square edge orifices commonly used for flow measurement, and those made in sheet metal shops for irrigation showed that the irrigation orifices have a higher discharge coefficient and a lower head loss coefficient than do the square edge orifices. Square edge orifices placed in irrigation pipe couplings behaved similarly to those for flow measurement, particularly in the mid and lower ranges of the diameter ratio, β_o.

The head loss ratio, R, as defined by the ASME (1959) is the same for, (a) square edge orifices used for flow measurement, (b) square edge orifices installed in aluminum irrigation pipeline joint couplings, and (c) sheet metal orifices made for irrigation installed in pipe couplings. The ratio can be represented by the equation $R = 1 - 0.9\,\beta_o^{1.7}$.

INTRODUCTION

Gated irrigation pipe is often used on nonuniform and relatively steep slopes. When used on slopes that exceed the friction on hydraulic gradeline slope of the flowing water, pipeline pressures increase in downstream sections of pipe. The resulting high pressures can give nonuniform flow, make outlet gates difficult to adjust, and cause high velocity streams to be emitted from the pipe. These high velocity streams often cause excessive soil erosion, especially on erosive soils. Another problem encountered on steep slopes is that the pipe may not flow full and it is difficult to get sufficient flow from upstream outlets.

Orifice plates with concentric orifices, placed at intervals in the pipeline, can be used to dissipate excess energy. They can also "check" the water so that the pipe flows full. Pipe orifices are widely used for flow measurement, and discharge coefficients for these are readily available. However, when they are used for energy dissipation, velocity head recovery downstream from the orifice must be considered and information for this use is limited. Head loss information presented by the American Society of Mechanical Engineers (ASME, 1959) pertains to square edge orifices clamped between flanges at a pipeline joint with stringent installation requirements. An orifice for irrigation pipe is loosely installed inside the bell end of a pipe coupling and is held in place by the male end of a companion pipe. With this type coupling, there is a discontinuity in the pipeline at the joint. This is in contrast to ASME flow measurement conditions, where orifices are installed in a rigid joint having a uniform diameter. ASME orifices are also machined and honed to achieve a very exacting square edge. This degree of precision is not required for energy dissipating orifices and the cost of such orifices would be prohibitive. Orifices for irrigation pipelines were made by conventional tools normally used in sheet metal shops and their inside edges were not completely square.

Because of the different conditions noted, pressure loss data presented by the ASME would not be expected to apply exactly to sheet metal orifices used for irrigation. Therefore, laboratory tests were conducted to obtain energy or head loss coefficients for orifices used in gated irrigation pipe. The results of these tests are presented in this paper which also includes a comparison of the test results with the ASME data and presents coefficients with which the ASME data can be used to estimate energy dissipation in gated pipe systems.

PROCEDURE

Laboratory tests were conducted using 150 mm (6 in.), 200 mm (8 in.), and 250 mm (10 in.) aluminum conveyance pipe without gates or outlets to determine the head loss for different orifice sizes and discharge rates. Pipe lengths upstream from the orifice were 5 m (15 ft) and represented 30, 23, and 18 pipe diameters respectively for the three pipe sizes, while downstream lengths varied from 5 m to 9 m (15 to 30 ft). All pipe lengths were adequate for full downstream velocity head recovery. The orifice to be tested was installed inside the coupling which joined the two lengths of pipe. Pipe coupling losses were measured without an orifice in place and were found to be small (Humpherys, 1986). Since head loss for the couplings was small, all of the measured loss with an orifice installed was attributed to the orifice.

Orifices for the tests were made from 1.5 mm (16 gauge) galvanized sheet metal and were sized to fit inside the bell end of a gated pipe coupling as shown in Fig. 1. Guide pins were fastened to some of the orifice plates to position and hold them in place on the male end of the pipe while the pipe was inserted into the coupling.

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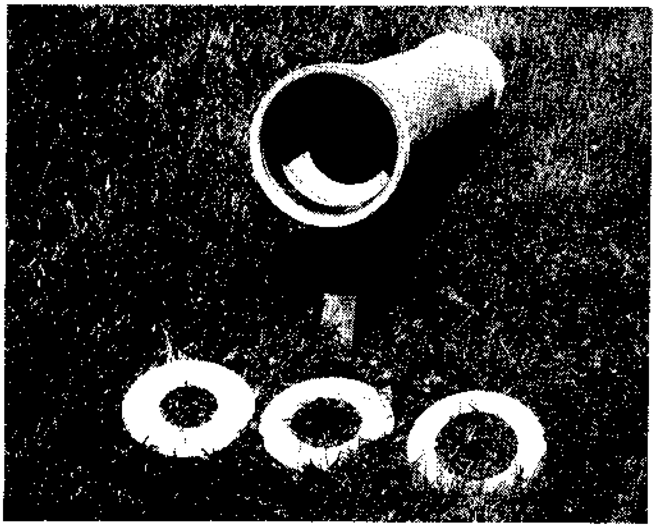

Fig. 1—Orifice plates used for energy dissipation in gated irrigation pipe.

However, subsequent tests indicated that this was not necessary. Plates with nominal orifice diameters ranging from 75 mm (3 in.) to 115 mm (4.5 in.) for the 150 mm pipe, 75 mm (3 in.) to 165 mm (6.5 in.) for the 200 mm pipe, and 125 mm (5 in.) to 190 mm (7.5 in.) for the 250 mm pipe were tested. These sizes represent orifice-to-pipe diameter ratios, β_o, ranging from 0.38 to 0.82 where $\beta_o = d_o/D$; d_o is the orifice diameter and D the inside pipe diameter. Most orifices were made using commercial shop procedures and circle cutters. The actual orifice diameters, which sometimes varied slightly from nominal, were used to determine β_o.

One series of tests was conducted using orifices made from the same sheet metal material but machined to provide a perfectly round orifice with a square edge. These were more nearly like the ASME orifices except that they fit loosely inside the pipe coupling.

Water for the tests was pumped from a laboratory sump and the flow measured with a 150 mm venturi-type flow meter. The test pipe was placed at zero slope in a flume and connected to a stilling head box at the inlet, so that the flow was free from swirls and eddies. Flow rates ranged from approximately 14 L/s (225 gpm) to 56 L/s (900 gpm). These flow rates represented a range of orifice Reynolds Numbers, N_R, from about 1.2 to 4.0 x 10^5 for the three pipe sizes with N_R based on d_o and the average orifice velocity. Piezometer taps were spaced 50 cm (20 in.) along the length of the test pipe with closer spacings down to 5 cm (2 in.) immediately downstream from the orifice. Piezometric head measurements were made with a water-column manometer. Head loss measurements were made for each orifice at different flow rates.

RESULTS AND DISCUSSION

The head loss, H_o, illustrated schematically in Fig. 2, is the elevation difference between the hydraulic gradelines extended upstream and downstream from the orifice. The downstream hydraulic gradeline was extrapolated upstream from the downstream section of pipe below the region where velocity head recovery was achieved. The head loss past an orifice can be expressed in the normal manner as a function of the velocity head $V_o^2/2g$ and a head loss coefficient as

$$H_o = K_o V_o^2/2g \quad \ldots\ldots [1]$$

where

- H_o = head loss representing the energy dissipated through an orifice, L
- K_o = dimensionless coefficient of head loss or energy dissipation through an orifice, a function of β_o
- V_o = orifice flow velocity = Q/A_o, L/T
- Q = flow discharge, L^3/T
- A_o = orifice area, L^2
- g = acceleration of gravity, L/T^2

Head loss coefficient

The head loss coefficient, K_o, was determined from the test data with a rearranged form of equation [1] where

$$K_o = 2g\,H_o/V_o^2 \quad \ldots\ldots [2]$$

The coefficient was found to be primarily a function of the diameter ratio β_o. It is nearly independent of flow rate and N_R in the higher ranges of N_R, where most irrigation flow rates fall, and in the mid and lower ranges of β_o as shown in Fig. 3. Published values of the discharge coefficient, C_d, for square edge orifices can be related to K_o, as noted later, and were used to compute K_o for square edge orifices for three diameter ratios. These are also shown in Fig. 3 to illustrate the variation of K_o with N_R.

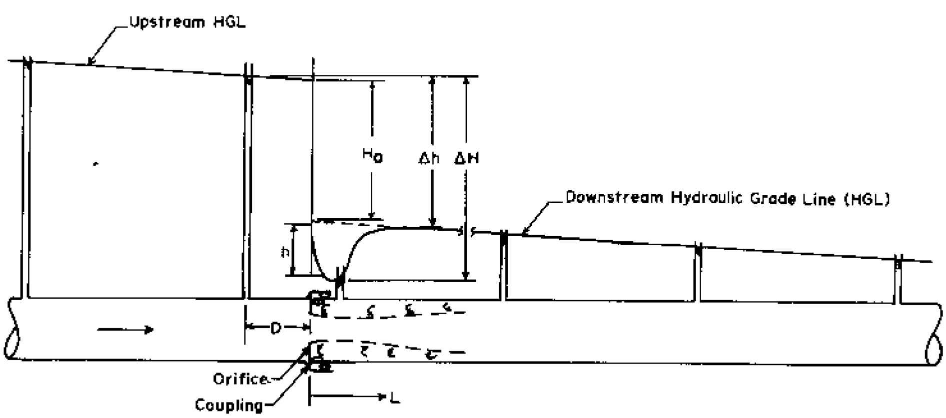

Fig. 2—Schematic diagram of the hydraulic gradeline for a pipeline with an energy dissipating orifice.

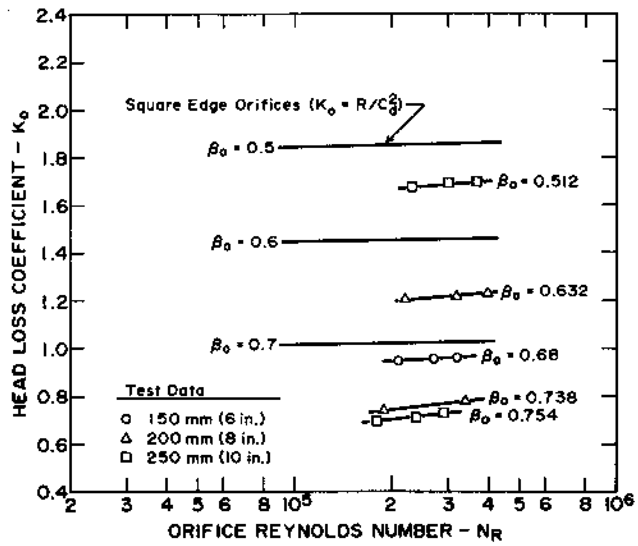

Fig. 3—Head loss coefficient K_o as a function of orifice Reynolds number for square edge orifices of three diameter ratios and representative data for irrigation sheet metal orifices from laboratory tests.

Values of K_o at different flow rates for a given orifice varied less than two percent from their average which, for practical purposes, is not significant. Therefore, average values of K_o for two or more test runs at different flow rates for a given orifice (Fig. 3) were plotted logarithmically as a function of $1-\beta_o$ as shown in Fig. 4. The factor $1-\beta_o$ was used rather than β_o so the data would plot as a straight line on a log-log plot. The head loss coefficient K_o approaches the coupling loss coefficient, K_c, as β_o approaches 1.0. Approximate values of K_c in the flow range of the tests were 0.15 for the 150 mm diameter pipe, 0.084 for the 200 mm pipe and 0.065 for the 250 mm pipe. As shown in Fig. 4, the head loss coefficient, K_o for 200 and 250 mm pipe can be represented by one curve while that for the 150 mm size is best represented by a separate curve. The function representing orifices for the 150 mm pipe ($r^2 = 0.995$) is

$$K_o = 3.5\,(1-\beta_o)^{1.2} \quad\ldots\ldots\ldots\ldots[3]$$

and for the 200 and 250 mm pipes ($r^2=0.997$) is

$$K_o = 4.85\,(1-\beta_o)^{1.38} \quad\ldots\ldots\ldots\ldots[4]$$

Combining each equation [3] and [4] with equation [1] gives the head loss for flow through orifices for 150 mm pipes as

$$H_o = 1.75\,(1-\beta_o)^{1.2}\,V_o^2/g \quad\ldots\ldots\ldots\ldots[5]$$

and for orifices in 200 and 250 mm pipe as

$$H_o = 2.42\,(1-\beta_o)^{1.38}\,V_o^2/g \quad\ldots\ldots\ldots\ldots[6]$$

Head loss curves such as those for 200 mm pipe shown in Fig. 5, can be constructed for different size orifices, flow rates, and pipe sizes from these equations for use in irrigation.

Comparisons with square edge orifices

As previously noted, a limited amount of information on head losses was presented by the ASME (1959) for flow measurement orifices which uses the basic orifice

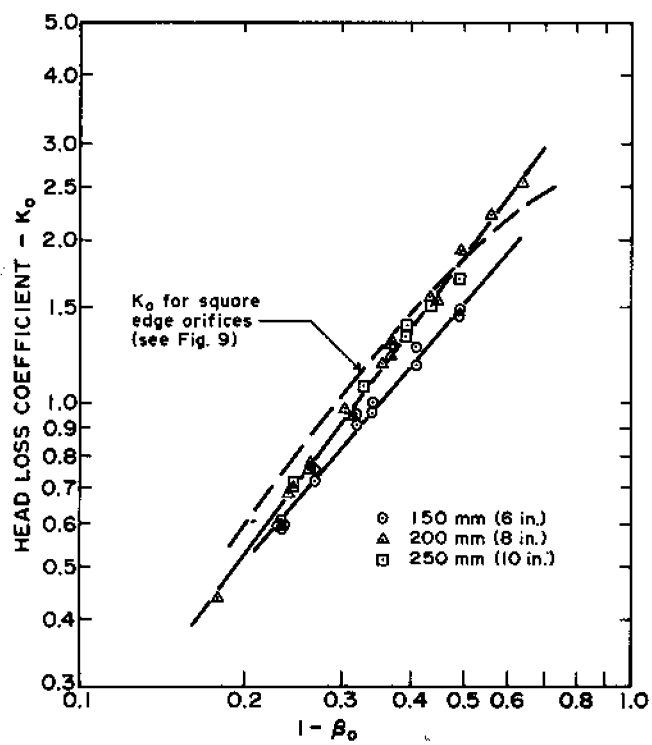

Fig. 4—Head loss coefficient K_o for energy dissipating sheet metal orifices used for irrigation as a function of the diameter ratio factor $1-\beta_o$.

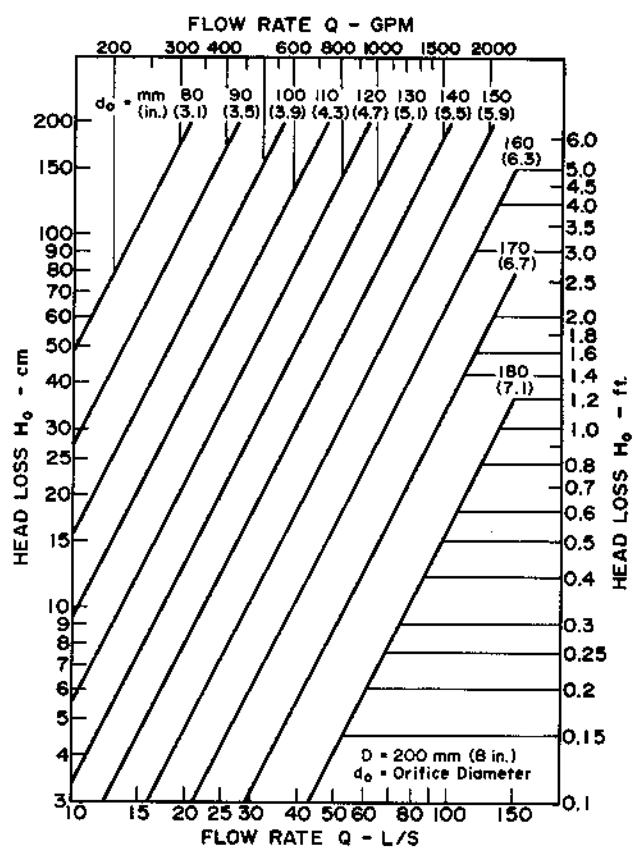

Fig. 5—Representative diagram of the head loss for energy dissipating orifices in a 200 mm (8 in.) irrigation pipe for different size orifices and flow rates.

flow equation

$$V_o = C_d(2g\,\Delta H)^{0.5} \quad [7]$$

where

C_d = coefficient of discharge for orifices,
ΔH = differential pressure head measured by pressure taps located upstream and downstream from the orifice.

Several types of pressure taps are used for measuring the differential head ΔH. This discussion and the ASME data are based upon vena contracta taps. As illustrated in Fig. 2, the high-pressure tap is located one pipe diameter upstream from the face of the orifice plate and the low pressure tap at the vena contracta which is the point of minimum downstream pressure (Brater and King, 1976).

Head loss ratio: The ASME (1959) defines a pressure head loss ratio as

$$R = \Delta h/\Delta H \quad [8]$$

where

Δh = difference between the minimum pressure head upstream from the orifice and the maximum head downstream from the orifice.
ΔH = difference between the minimum pressure head upstream and the minimum head downstream from the orifice.

The differential pressure heads as defined above are shown in Fig. 2. As shown in Fig. 2, Δh is nearly the same as the head loss, H_o. The difference between Δh and H_o is small compared to the loss and represents the pipe friction loss between the upstream tap and the point of maximum downstream pressure. This difference was generally less than 12 mm (0.5 in.) and for energy dissipating purposes can be neglected. Thus, if H_o is substituted for **Δh**, from equation [8], the head loss is

$$H_o = R\Delta H \quad [9]$$

From equation [7],

$$\Delta H = 1/C_d^{\,2}\,(V_o^{\,2}/2g) \quad [10]$$

and combining equations [9] and [10] gives

$$H_o = R/C_d^{\,2}\,(V_o^{\,2}/2g) \quad [11]$$

from which

$$K_o = R/C_d^{\,2} \quad [12]$$

As noted previously, a series of laboratory tests was conducted using machined square edge orifices installed in the irrigation pipe couplings. These tests were made to compare results between machined and non-machined orifices made from sheet metal and the precision-made ASME orifices. Some runs were made during the laboratory tests with both groups of sheet metal orifices to obtain Δh and ΔH data from which the pressure head loss ratio could be determined. The head loss ratio for these tests, expressed as (1-R), is plotted logarithmically

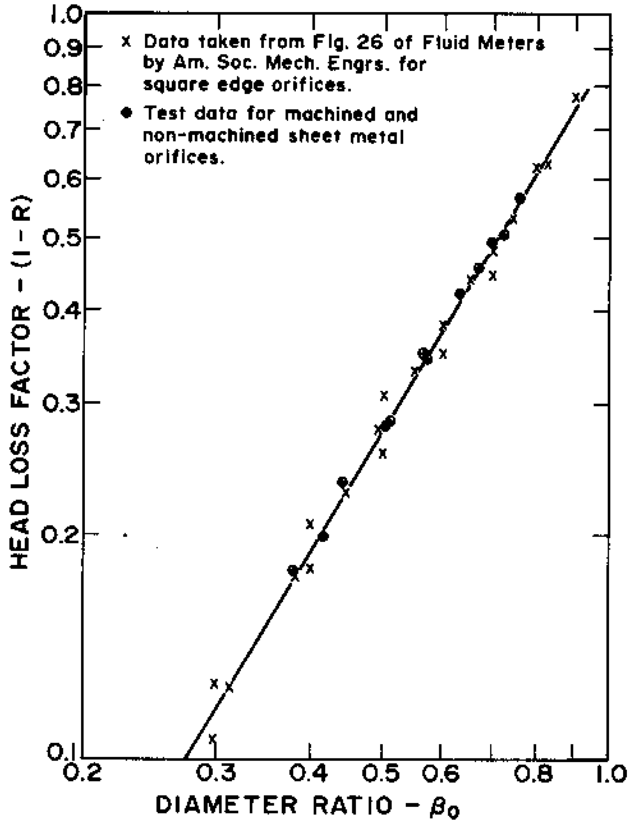

Fig. 6—Head loss factor (1-R) as a function of the diameter ratio β_o for common and square edge sheet metal orifices in irrigation pipe couplings and for ASME square edge flow measurement orifices.

as a function of β_o in Fig. 6. The data for both groups of sheet metal orifices fit the same curve. Individual data points taken from Fig. 26 of the ASME publication for square edge flow measuring orifices are also shown in Fig. 6 and fit the same curve. The head loss ratio R can be expressed by an equation of the form

$$(1-R) = a\,\beta_o^{\,b} \quad [13]$$

where **a** and **b** are constants and are shown in Table 1 for regressions on each set of data individually and combined.

As shown in Fig. 6 and Table 1, the head loss ratio is the same for all orifices and can be expressed as

$$(1-R) = 0.9\,\beta_o^{\,1.7} \quad [14]$$

from which

$$R = 1-0.9\,\beta_o^{\,1.7} \quad [15]$$

Since K_o is a function of R and C_d as shown by

TABLE 1. CONSTANTS FOR THE GENERAL EQUATION EXPRESSING R AS A FUNCTION OF β_o FOR TWO SETS OF DATA

Data set	a	b	r^2
Laboratory tests w/square edge and common orifices	0.899	1.68	0.999
Data from ASME Figure 26	0.887	1.69	0.997
Both data sets combined	0.893	1.69	0.997

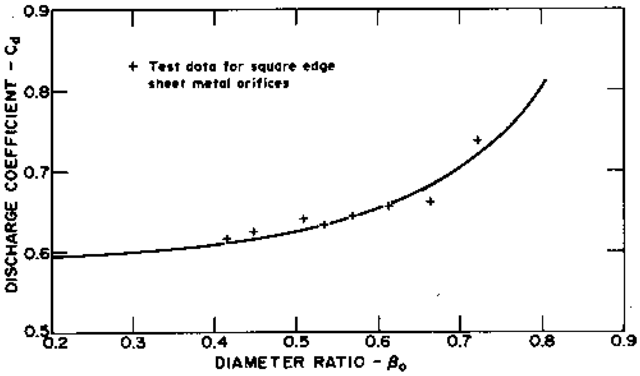

Fig. 7—Published values of the discharge coefficient C_d as a function of β_o for square edge orifices in the orifice N_R range from 2 to 2.5 x 10^5 with data from the laboratory tests for square edge sheet metal orifices superimposed.

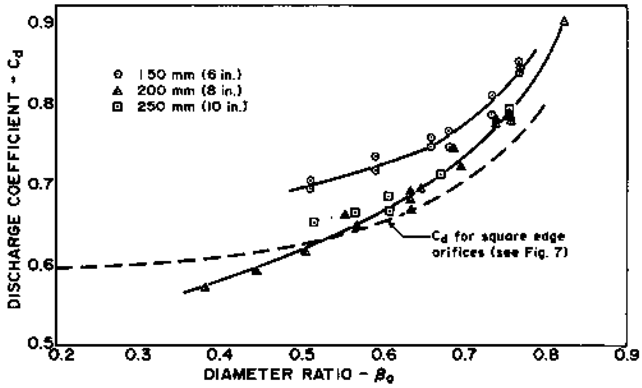

Fig. 8—Discharge coefficient C_d for irrigation sheet metal orifices tested in the laboratory as a function of β_o; the curve for square edge orifices is shown for comparison.

equation [12], and since R is apparently the same for all orifices, any differences between K_o for square edge orifices and those made for irrigation energy dissipation must result from differences in the discharge coefficient C_d.

Discharge coefficient: Published values of the coefficient of discharge C_d for orifice Reynolds numbers N_R in the range from 2 to 2.5 x 10^5 (Baumeister, 1967; Brater and King, 1976 and Rouse, 1950) are shown by the curve in Fig. 7 as a function of β_o. These coefficients all include the velocity of approach factor $1/[1-\beta_o^4]^{0.5}$. The coefficient C_d for the square edge orifices tested in the laboratory was determined from a rearranged form of equation [7] where

$$C_d = V_o/(2g\Delta H)^{0.5} \quad \ldots\ldots [16]$$

These data superimposed upon the curve in Fig. 7 show that the discharge coefficient for square edge orifices placed in irrigation couplings is similar to that for square edge orifices installed under more exacting conditions. The effects of coupling geometry and pipe discontinuity at the couplings are apparently relatively small, particularly, at low values of β_o. They would be expected to increase as β_o increases such that the flow streamlines are closer to the coupling boundary in the vicinity of the orifice plate.

A discharge coefficient, C_d, for the irrigation orifice plates was determined from equation (12) where

$$C_d = (R/K_o)^{0.5} \quad \ldots\ldots [17]$$

which combined with equation [15] gives

$$C_d = [(1-0.9\beta_o^{1.7})/K_o]^{0.5} \quad \ldots\ldots [18]$$

Equation [18] was used rather than equation [16] for the irrigation orifices because ΔH was not available from the test data for many of the runs. The discharge coefficient for the irrigation orifices is shown in Fig. 8 along with the curve for square edge orifices from Fig. 7 for comparison. As noted in the figure, the coefficient for 150 mm pipes was different from that for the 200 and 250 mm pipes. Except for three data points for small orifices, the coefficient is generally higher than for square edge orifices. This is as expected because the slight rounding of the edge of the orifice opening resulting from shop construction compared to an exact square edge, decreases the flow contraction through the orifice, and results in a higher discharge coefficient. For irrigation applications, orifices with β_o values less than about 0.5, would seldom be used because of their severe flow restriction.

The head loss coefficient K_o for square edge orifices calculated from published values of C_d with a rearrangement of equation [18] is shown in Fig. 9 where

$$K_o = (1-0.9\beta_o^{1.7})/C_d^2 \quad \ldots\ldots [19]$$

Correspondingly, coefficients from the laboratory tests for square edge orifices are also shown. This shows the close agreement between the loss for square edge orifices installed in irrigation couplings and that for flow measurement orifices. The curve from Fig. 9 is also shown in Fig. 4 for comparison. Fig. 4 shows that the

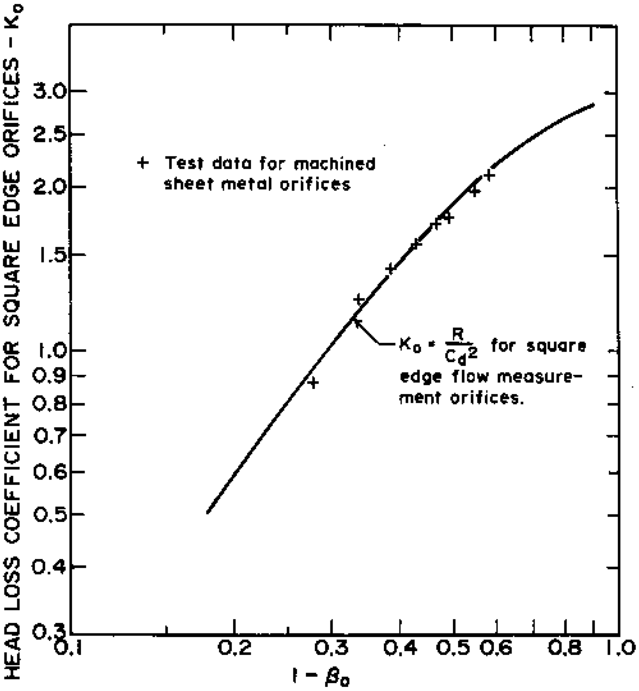

Fig. 9—Head loss coefficient K_o for square edge orifices determined from published values of C_d in the orifice N_R range from 2 to 2.5 x 10^5 with data from the laboratory tests for square edge orifices superimposed.

head loss for orifices that are not square edged, such as the irrigation orifices, can vary significantly from that for square edge orifices. Values of C_d from Fig. 8 can be used with the diameter ratio as in equation [19] to estimate the head loss coefficient for commonly-made sheet metal orifices.

Hydraulic Gradeline Depression

Orifice plates convert pressure head to velocity head as water flows through the orifice openings. Consequently, the pipe pressure immediately downstream from an orifice, near the vena contracta, is reduced beyond that represented by the head loss and is lower than that further downstream, as shown in Fig. 2. This reduces the flow from outlets or gates that may be located in this section of pipe. To minimize this effect, it is best to use orifices with either relatively large or relatively small diameter ratios. A greater number of orifices with large β_o values will be required, but they are relatively easy to install and their cost is nominal.

The piezometric head depression, h, is the elevation difference between the actual depressed hydraulic gradeline at a given point downstream from the orifice and its projected elevation at that point when extrapolated upstream from the downstream section of pipe where full velocity head recovery is achieved. By minimizing h when designing energy dissipating orifices, near uniform flow rates from the openings can be achieved with a small adjustment of the pipe gates.

The piezometric head depression is affected by the diameter ratio β_o, velocity head $V_o^2/2g$, distance downstream from the orifice, and to a lesser extent, pipe size. The depression, expressed as the ratio h/H_o is shown in Fig. 10 for different diameter ratios, β_o. The curves represent average values for all pipe sizes developed for even values of β_o from cross plottings of the laboratory test data. Distance downstream from the orifice is expressed in pipe diameters. As seen from the figure, the depression can be significant in relation to the head loss up to a distance of about 2½ to 3 pipe diameters downstream where it exceeds about one tenth of the head loss. The depression was related to H_o for ease of estimation. For practical purposes, a high degree of accuracy is not necessary when estimating h. The depression at the first downstream outlet or gate, which is the primary point of interest, can be estimated by using the curves in Fig. 10 to determine the ratio h/H_o. The depression is calculated from this ratio for a given orifice ratio and distance with H_o estimated from head loss curves such as those shown in Fig. 5.

Design Procedure

An elevation profile along the pipeline is needed to determine the number, size, and location of orifices to be used in the field. The desirable pressure head along the length of a gated pipe is between 0.3 and 0.7 m (1 to 2 ft). Orifices are selected to maintain this pressure head range. The pressure should be reduced to the required level in the first gated pipe section by using orifices in the conveyance pipe preceding the distribution section. This can be done, if necessary, with orifices having relatively small β_o values. Using small orifices here minimizes the number needed to maintain the optimum operating pressure at the upstream end of the distribution pipe and also helps to minimize h in the downstream gated pipe sections.

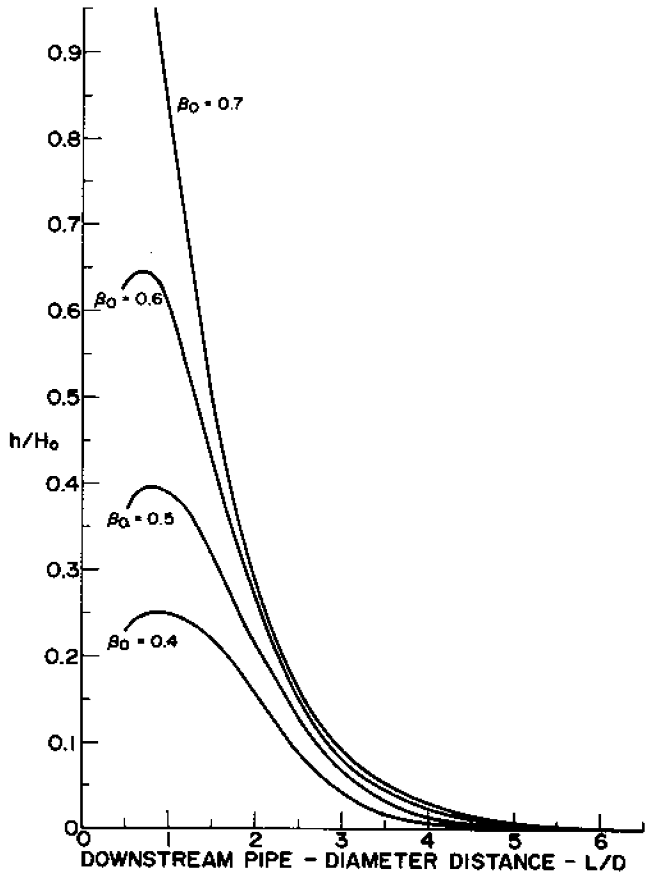

Fig. 10—Piezometric head depression ratio h/H_o for different diameter ratios β_o as a function of pipe-diameter distance L/D downstream from an orifice in irrigtion pipelines.

A design example is illustrated in Fig. 11. This is for a design flow of 34 L/s (540 gpm) in 200 mm gated pipe. The hydraulic gradient or slope of the pipe at the design flow is shown with the elevation profile of the pipeline. The design can be accomplished graphically by using a cardboard or plastic template with the upper side sloped to match the hydraulic gradient. The left side is vertical, to correspond to elevation, with marks representing the head loss for several orifices in the size range needed. The range of possible orifice sizes for a particular site or condition can be determined by estimating the head loss desired for one orifice (Fig. 5). Since the orifices are installed in the pipe joints, the loss represented by one orifice must be for a distance represented by a whole number of pipe lengths usually some multiple of 9 m. A trial set of orifice sizes is selected and the hydraulic gradeline depression, h, at the first downstream outlet from the orifice is estimated using Fig. 5 and 10 for the design flow. The larger the orifice sizes, the smaller h will be and the greater the number that will be required. In the example, if h is arbitrarily limited to about 10 cm (0.33 ft) at the first opening, which is 2.5 diameters downstream, then all orifices 120 mm and larger will be satisfactory. When a set of orifice sizes has been selected, their head loss for the design flow is marked on the template. The template is moved downstream from the inlet and parallel to the pipe hydraulic slope line until the accumulated excess elevation head at a pipe joint matches the loss for one of the selected orifices. The hydraulic gradeline is then reduced by that amount and drawn on the chart. This process is repeated such that

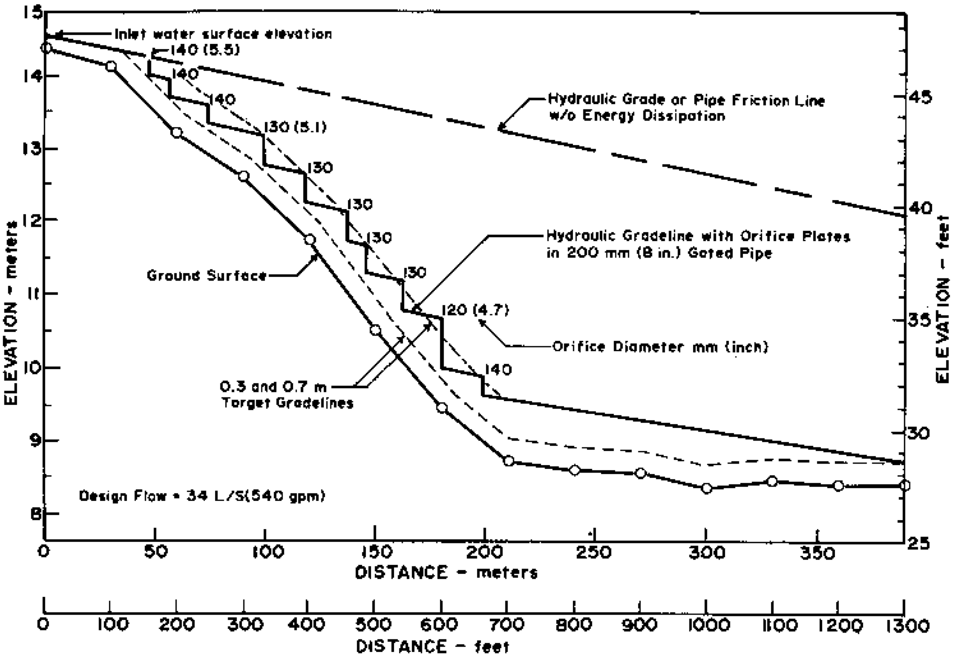

Fig. 11—Design example illustrating hydraulic gradeline reduction in steps or increments with energy dissipating orifices.

the hydraulic gradeline is reduced in steps to within the desired range after each orifice as illustrated in Fig. 11. The graphic design can be quickly checked by numerically summing the loss increments and comparing the elevation of the resulting gradeline to the pipeline elevation at its downstream end.

Maintenance

To minimize rusting of the orifice edge, pipelines should be permitted to drain after each irrigation (this will usually occur naturally with gated pipe) and the orifices stored in a dry place during the off-season.

SUMMARY AND CONCLUSIONS

Orifices placed at intervals in gated pipe couplings can be used to dissipate excess energy and thus minimize erosion caused by high velocity streams discharged from the pipe. The orifice can be made from galvanized sheet metal at most sheet metal shops. Head loss relationships for orifices used in irrigation pipelines for energy dissipation were obtained from laboratory tests. Comparisons were made between the head loss and discharge coefficients for machined square edge orifices and those made for irrigation in sheet metal shops.

General conclusions are:

1. Head loss for sheet metal orifices made for irrigation can be predicted using coefficients determined from this study.

2. The head loss coefficient, K_o, can be expressed by an equation of the form $K_o = a\beta_o^b$ where **a** and **b** are empirical constants determined from laboratory tests. K_o is nearly independent of orifice Reynolds number in the range normally encountered in the field (1.2 to 4.0×10^5) and in the mid and lower ranges of the diameter ratio β_o. The coefficient varied less than two percent from the average in the range of Reynolds numbers for the tests.

3. Sheet metal orifices made in sheet metal shops have a slightly rounded edge which results in a higher coefficient of discharge, and a lower head loss coefficient than for machined square edge orifices.

4. The head loss ratio, R, as defined by the ASME (1959) is the same for, (a) square edge orifices used for flow measurement, (b) square edge orifices installed in aluminum irrigation pipeline joint couplings and (c) sheet metal orifices made for irrigation installed in pipe couplings. The ratio can be represented by the equation $R = 1\text{-}0.9\,\beta_o^{1.7}$.

5. Square edge orifices in aluminum irrigation pipe couplings behave similarly to those for flow measurement, particularly in the mid and lower ranges of the diameter ratio, β_o. Both the discharge and the head loss coefficients for square edge sheet metal orifices installed in pipe couplings fit the calculated curves for these parameters determined from published values of the discharge coefficient for square edge orifices.

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Energy Dissipation in Low Pressure Irrigation Pipelines
I. Butterfly Valves and Discs

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Butterfly valves and discs inserted in low pressure irrigation pipelines, such as gated pipe, can be used to dissipate excess energy for erosion control and to "check" the water so that the pipes flow full. Butterfly discs installed in short energy dissipator pipe sections or in full lengths of gated pipe are described. The head loss for both valves and discs is expressed as a function of a head loss coefficient and the velocity head. The loss coefficient for butterfly valves is expressed exponentially as function of the angular position of the butterfly plate, while that for discs is also a function of the disc-to-pipe diameter ratio. Graphical relationships for the loss coefficient were developed for both valves and discs. These can be used to estimate head loss when designing and using the butterfly valves and discs in low pressure irrigation pipelines.

INTRODUCTION

One advantage of gated pipe for irrigation compared to open channels is that the pipe need not be placed on a uniform slope. It is often used on nonuniform and steep slopes. When used on relatively steep slopes that exceed the friction or hydraulic gradeline slope of the flowing water (>1% depending on length of pipeline), the pipeline pressure increases in downstream sections of the pipe, particularly in long pipelines. Pressure heads greater than about 0.7 m (2 ft) make the gates difficult to adjust, give nonuniform flow, and cause high velocity streams to be emitted from the pipe gates which often cause excessive soil erosion. Many field situations require that variable amounts of energy be dissipated in gated or surface irrigation pipelines. A given pipeline may require energy dissipation while serving one field or portion of a farm and not require it when serving a different area. Elevation differences within the same field also may require variable amounts of dissipation. Surface pipelines may not always flow full under low head and steep slope conditions. Thus, it is sometimes difficult to get sufficient and uniform flow from the pipe outlets unless the flow is "checked" in the pipe. The amount of checking needed often varies with flow rates and other conditions.

Commercial, low-pressure, irrigation butterfly valves are sometimes used to help satisfy these needs. However, because they completely shut off the flow in their closed position, the amount of energy dissipated is very sensitive to the angular position of the butterfly plate, particularly in the upper range of closing angles. The valves are sometimes difficult to adjust for a specific head loss requirement and flow rate variations can produce a relatively wide range of head losses at a given angular setting. Because of the need for a wider range of angular adjustability for a given range of head losses than that provided by butterfly valves, energy-dissipating butterfly discs were developed and rated for this purpose. The discs are similar to butterfly valves except that their diameter is smaller than that of the pipe and they do not fully close the pipe.

Inquiries to several irrigation valve manufacturers indicated that energy dissipation data are not available for "low pressure" irrigation butterfly valves. Therefore, laboratory tests were conducted to obtain head loss coefficients for both valves and discs. The results of these tests are presented in this paper which also describes the discs and presents head loss relationships for energy dissipation in gated pipe systems. Orifices for this purpose were also studied but are reported separately (Humpherys, 1987).

BUTTERFLY VALVES

Low pressure irrigation butterfly valves are commonly made in short pipe sections which can be inserted into a pipeline at any joint. The butterfly plate is usually coated or encased in rubber to form a tight pipe closure. In their full open (zero degree) position, the flow restriction is that caused by the shaft and projected edge of the plate. The flow restriction increases as the plate is rotated from its full open toward its closed position such that the accompanying head loss increases exponentially. The lower range of angular positions below about 30 to 40 deg is the most practical for energy dissipation because the flow is so severely restricted in the upper range. These valves usually have a pressure rating of about 170 kpa (25 psi).

Laboratory Tests

Laboratory tests were conducted with 150 mm (6 in.), 200 mm (8 in.), 225 mm (9 in.), and 250 mm (10 in.) valves from two different manufacturers. Test pipes, without gates, corresponding to each valve size were placed in a laboratory flume with 4.6 m (15 ft) of straight pipe upstream and 9 m (30 ft) of straight pipe downstream from the valve section. These lengths represented a range of 18 to 60 pipe diameters for the pipes tested and were more than adequate for full downstream velocity head recovery. Tests were also made to estimate pipe coupling losses.

Water was pumped from a laboratory sump and the flow measured with a 150 mm (6 in.) venturi type flow meter. The test pipe was placed at zero slope in the flume and connected to a stilling head box at the inlet end.

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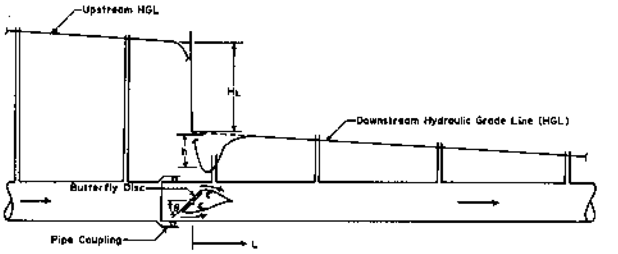

Fig. 1—Schematic diagram of the hydraulic gradeline and head loss for a pipe with a butterfly valve or disc energy disipator.

Flow rates ranged from approximately 14 L/s (0.5 cfs) to 56 L/s (2 cfs). Piezometer taps were spaced 50 cm (20 in.) along the length of the test pipe with closer spacings down to 5 cm (2 in.) downstream from the valve. Piezometric head measurements were made with a water-column manometer. The head loss was determined at various valve closing angles and flow rates for each valve. The valve angles began with zero degrees at the full open position and approached 90 deg at the fully closed position.

The head loss, illustrated schematically in Fig. 1, is the elevation difference between the hydraulic gradelines extended upstream and downstream from the butterfly valve. The downstream hydraulic gradeline was extrapolated upstream from the downstream section of pipe below the region where velocity head recovery was achieved. The head loss can be expressed in the normal manner as a function of the velocity head $V_p^2/2g$ and a head loss coefficient as

$$H_b = K_b \, V_p^2/2g \quad \ldots\ldots\ldots\ldots [1]$$

where

H_b = head loss representing the energy disspiated through a butterfly valve, L

K_b = dimensionless head loss coefficient for commercial butterfly valves, a function of θ

θ = valve closing angle in degrees

V_p = mean pipe flow velocity = Q/A_p, L/T

Q = flow discharge, L^3/T

A_p = inside pipe area, L^2

g = acceleration of gravity, LT^2

Results and Discussion

The head loss as illustrated in Fig. 1 also includes pipe coupling losses at the joint(s). With a valve inserted into the test pipe at the joint, the loss includes that for two couplings, whereas, only one additional coupling is added to the total pipeline. Therefore, the loss for one coupling was subtracted from the measured laboratory head loss so that the loss includes that for one coupling and the valve. Coupling losses were determined from a rearranged form of Equation [1] with a different loss coefficient such that

$$K_c = 2g \, H_c/V_p^2 \quad \ldots\ldots\ldots\ldots [2]$$

where

K_c = coupling loss coefficient determined from laboratory tests.

H_c = coupling head loss.

Coupling loss coefficients in the flow range used for the tests vary slightly with velocity and are shown in Fig. 2 for three pipe sizes. The coefficients are for pipes with a rolled end approximately 9.5 mm (3/8 in.) thick.

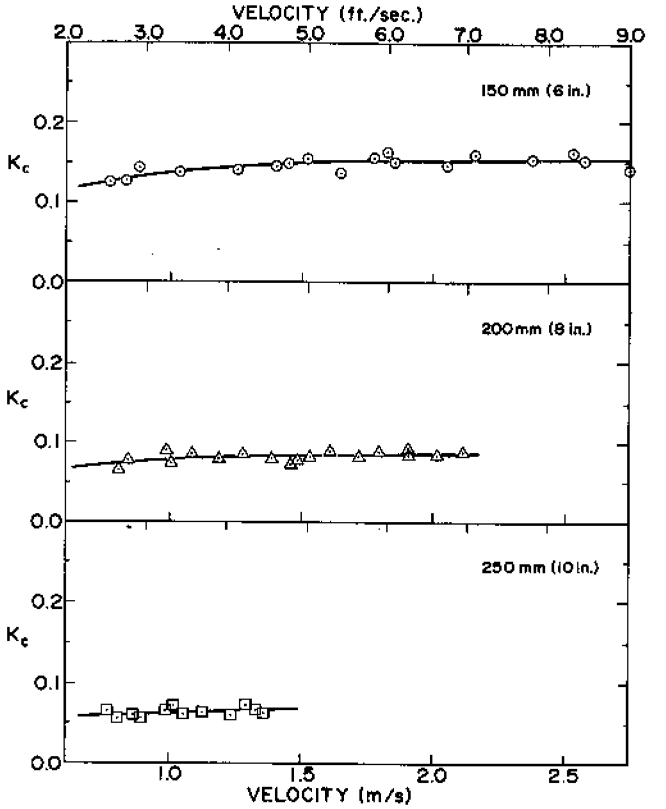

Fig. 2—Coupling loss coefficient K_c for three sizes of pipe used in the laboratory tests.

The head loss coefficient K_b was determined from the test data with an equation of the same form as Equation [2], with the subscript c changed to b for butterfly valves. The mean pipe velocity was used because the actual flow velocity past the butterfly plate could not be readily determined. The loss coefficient is practically independent of flow rate and Reynolds number, N_R, for fully developed turbulent flow. The tests were conducted in the Reynolds number range from about 1 X 10^5 to 3.5 X 10^5; N_R for most irrigation flow rates falls within this range. The coefficient K_b was found to be a function of the valve closing angle, θ, and can be related to θ by an equation of the form

$$K_b = a \, e^{b\,\theta} \quad \ldots\ldots\ldots\ldots [3]$$

where **a** and **b** are dimensionless constants and are shown in Table 1 for the curves plotted in Fig. 3.

As can be seen from Fig. 3, Equation [3] represents K_b for valve angles between approximately 15 and 60 deg. This includes the usable range for energy dissipation. With the valve open, the edge of the valve plate faces the flow and valve rotation through several degrees does not significantly affect the loss. At θ = zero, K_b represents the head loss caused by the edge of the valve plate, the shaft and one coupling. The projected thickness of the valve plate with the rubber-covered shaft in its center varied from about 34 mm (1.34 in.) for the valves of manufacturer B to 38 mm (1.5 in.) for the largest valves of manufacturer A. As shown in Fig. 3, K_b for the 150 mm valves is slightly different for the two makes of valves, but for practical purposes both can be represented by one curve. Curves for the largest valves

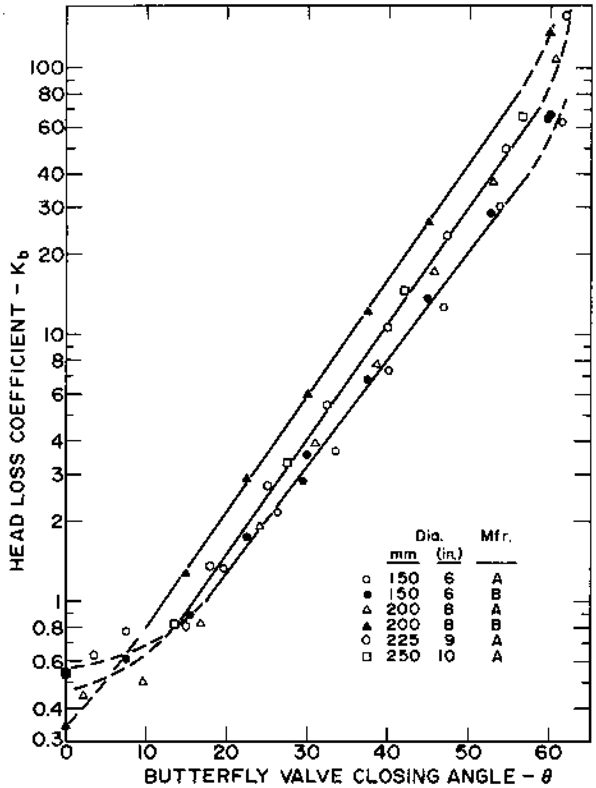

Fig. 3—Head loss coefficient K_b for different sizes of two commercial irrigation butterfly valves as a function of valve closing angle θ when the valves are used for energy dissipation.

TABLE 1. CONSTANTS FOR DETERMINING THE HEAD LOSS COEFFICIENT K_b FOR DIFFERENT SIZE IRRIGATION BUTTERFLY VALVES FROM TWO DIFFERENT MANUFACTURERS.

Valve size	Manufacturer	a	b	r^2
150 mm	A* and B†	0.202	0.092	0.996
200, 225, 250 mm	A	0.203	0.10	0.996
200 mm	B	0.292	0.10	0.999
200-250 mm (4 valves combined)	A and B	0.226	0.10	0.990

*Hastings Irrigation Pipe Co., Hastings, Neb.
†Midwest Irrigation Co., Henderson, Neb. Company names are shown for the benefit of the reader and do not imply endorsement or preferential treatment of the company or products noted.

indicate that K_b for valves of different sizes made by the same manufacturer can likely be represented by one curve, while those of a different make may vary depending upon valve plate thickness and configuration. The constants shown in Table 1 for Equation [3] can provide an estimate of the range of values of K_b for valves of this type.

Combining Equations [1] and [3] gives an expression for the relationship between valve closing angle, head loss, and velocity for irrigation butterfly valves:

$$\theta = \frac{1}{b} \log_e \left(\frac{2g\,(H_b)}{a\,V_p^2} \right) \qquad [4]$$

BUTTERFLY DISCS

Butterfly discs, as described in this paper, are similar to butterfly valves except that the disc diameter is smaller than that of the pipe as shown in Fig. 4. The amount of energy dissipated can be varied from minimum to maximum by changing the disc's angular position. Maximum dissipation occurs when the face of the disc is normal to the flow. The discs were installed in the pipe as shown in Fig. 5. Short, pipe dissipators with couplings as shown in Fig. 4, made from either aluminum or plastic pipe, can be inserted into a pipeline at any joint. Alternatively, a disc can be installed near the end of a full-length pipe. A linkage device for adjusting the angular position of the disc is shown in Fig. 6. This device will hold the disc in any position.

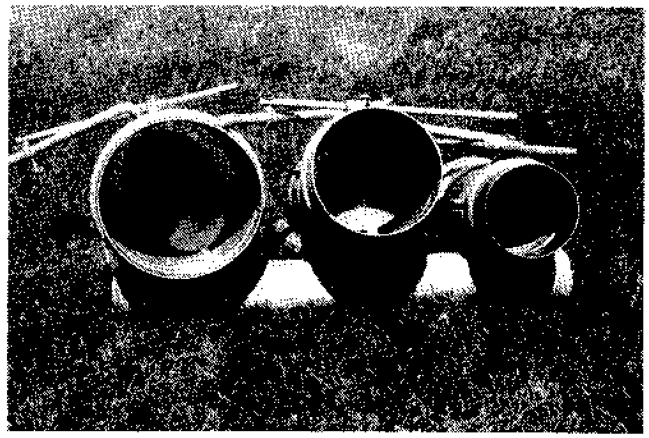

Fig. 4—Butterfly disc energy dissipators in short pipe sections for irrigation pipelines.

Laboratory Tests

Two series of laboratory tests were conducted in the same manner and with the same laboratory setup as previously described for the butterfly valves. The first series of tests was conducted for maximum dissipation with discs of different diameters placed in the 90 deg or closed position. The discs were held fixed with their full face area exposed to the flow for maximum head loss. Discs for this test series ranged in size from 70 mm (2 3/4 in.) to 108 mm (4 1/4 in.) for the 150 mm pipe, 108 mm (4 1/4 in.) to 162 mm (6 3/8 in.) for the 200 mm pipe and 100 mm (4 in.) to 200 mm (8 in.) for the 250 mm pipe.

The second series of tests was conducted with variable position discs of three sizes representing disc-to-pipe diameter ratios, β_d, of 0.6, 0.8 and 0.9 for each pipe. These ratios were chosen to provide sizes that would satisfy most conditions encountered. There is

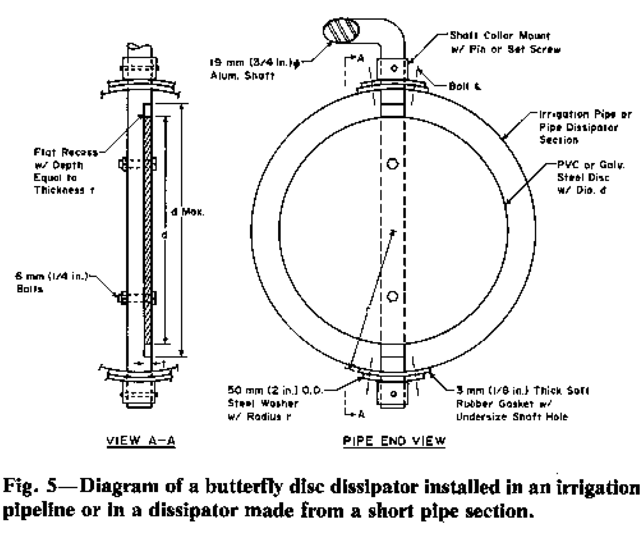

Fig. 5—Diagram of a butterfly disc dissipator installed in an irrigation pipeline or in a dissipator made from a short pipe section.

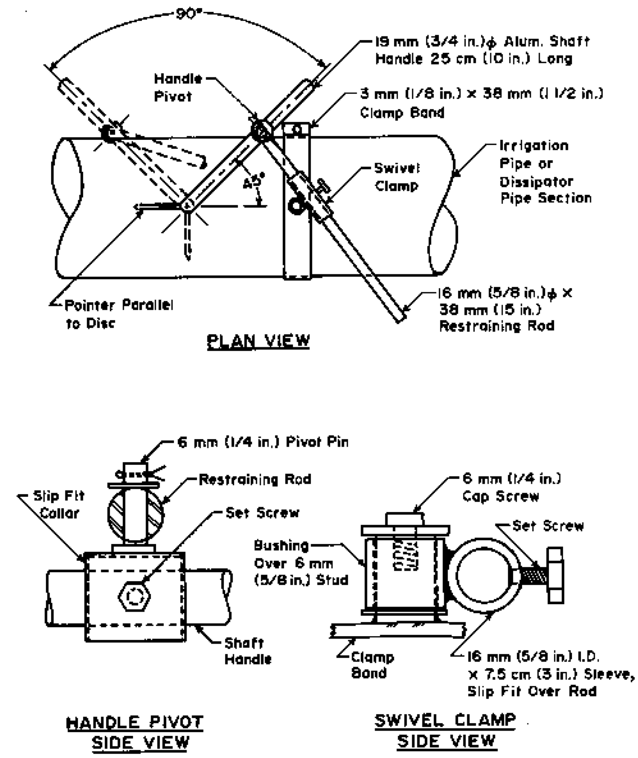

Fig. 6—Drawing of a linkage device to adjust and maintain the angular position of a dissipator disc.

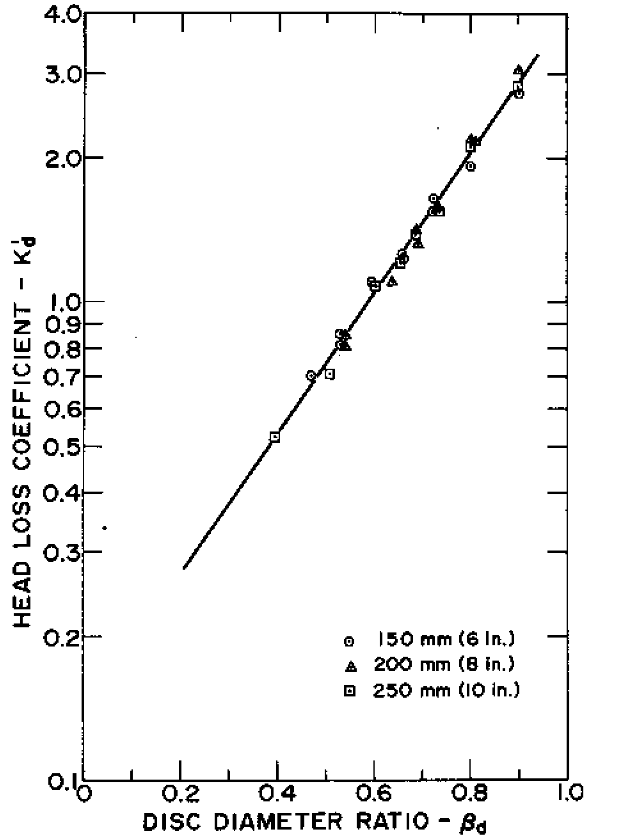

Fig. 7—Maximum head loss coefficient, plotted as K_d' vs. disc diameter ratio β_d for butterfly discs in a fixed (90 deg) position normal to the flow.

considerable overlap between sizes, and other sizes should not normally be needed. Test runs were made at various flow rates and disc closing angles θ ranging from 0 deg (open) to 90 deg (closed) for each disc for the three pipe sizes.

Discs for both tests were cut from 6.4 mm (1/4 in.) thick PVC plate.

Results and Discussion.

Fixed position discs: The head loss for this test series was determined for various disc sizes and flow rates, for three pipe sizes, and in the same range of N_R as for the butterfly valves. The head loss for discs is also expressed in the form of Equation [1] with the terms for butterfly valves (subscript b) changed to those for discs (subscript d). The coefficient for maximum head loss, K_d', for discs positioned at 90 deg was calculated from the test data such that

$$K_d' = 2g\,H_d'/V_a^2 \quad \ldots\ldots [5]$$

where

K_d' = head loss coefficient for discs positioned at θ = 90 deg for maximum energy dissipation

H_d' = maximum head loss corresponding to discs at θ = 90 deg, L

V_a = actual mean flow velocity past the disc = $Q/A_a = V_p/(1-\beta_d^2)$, L/T

A_a = actual flow area = $A_p - A_d = (1-\beta_d^2)\,A_p$, L^2

A_d = disc face area = $\beta_d^2\,A_p$, L^2

β_d = disc-to-pipe diameter ratio, d_d/D

d_d = disc diameter, L

D = inside pipe diameter, L

It was necessary to use the actual flow velocity to obtain a straight line on either a semilogarithmic or a logarithmic data plot for K_d'. With a disc fixed in its closed position, the actual mean velocity past the disc can be readily determined. The loss coefficient K_d' is a function of the diameter ratio β_d, and is shown on a semilogarithmic data plot in Fig. 7. The function representing this curve (r^2 = 0.994) is

$$K_d' = 0.14\,e^{3.38\,\beta_d} \quad \ldots\ldots [6]$$

Each data point in Fig. 7 represents the average value for two or three test runs at different flow rates with the same disc. Deviations from the average for different flow rates (and N_R) were usually less than about 2 %. The maximum head loss H_d' for discs of a given size positioned at θ = 90 deg can be determined by combining Equation [6] with a form of Equation [1] which uses the actual velocity such that

$$H_d' = [0.14e^{3.38\,\beta_d}]\;[1/(1-\beta_d^2)]^2\;V_p^2/2g \quad \ldots\ldots [7]$$

Variable position discs: The head loss for this test series was determined for various flow rates, disc angular positions, and three disc diameter ratios and pipe sizes. The head loss coefficient, K_d, was calculated from the test data using the form of Equation [2] with the term notation changed to that for butterfly discs. The loss coefficient is a function of disc closing angle θ and diameter ratio β_d and is shown in Fig. 8 on a semilogarithmic data plot for the three pipe sizes. The straight line portion of each curve shown in the figure for the three diameter ratios can be represented by Equation [3] with the appropriate constants. The constants **a** and **b** for the curves plotted are shown in Table 2.

As shown in Fig. 8, Equation [3] can represent K_d for

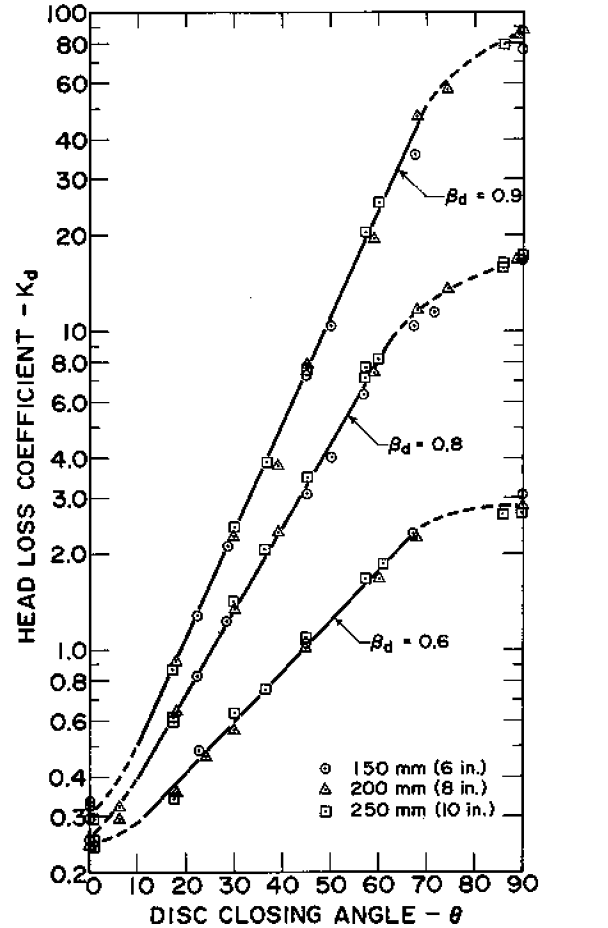

Fig. 8—Head loss coefficient, K_d, for variable position energy dissipating discs with three diameter ratios, β_d, as a function of disc angle, θ, for three pipe sizes.

values of θ between approximately 10 and 70 deg for all pipe sizes tested. In the full open position ($\theta = 0$), K_d represents the loss caused by the edge of the disc, the shaft and one coupling.

Combining Equation [3] with Equation [1] for discs and using the constants presented in Table 2 gives expressions for head loss, H_d, for variable position butterfly discs which have β_d ratios of 0.6, 0.8, and 0.9:

$$H_d = ae^{b\theta}\ V_p^2/2g \quad \text{[8]}$$

The resulting equations, when solved for θ, are the same form as Equation [4] with H_b changed to H_d. Using these equations, one for each β_d, diagrams such as that shown in Fig. 9 for $\beta_d = 0.8$ for 200 mm diameter pipe

TABLE 2. CONSTANTS FOR DETERMINING THE LOSS COEFFICIENT K_d FOR BUTTERFLY DISCS WITH THREE DIAMETER RATIOS, β_d

β_d	a	b	r^2
0.6	0.193 (0.204)*	0.037 (0.037)	0.997 (0.988)
0.8	0.215 (0.228)	0.060 (0.061)	0.998 (0.995)
0.9	0.233 (0.235)	0.077 (0.078)	0.999 (0.996)

*Numbers in parenthesis were determined with K_d estimated from equation [12].

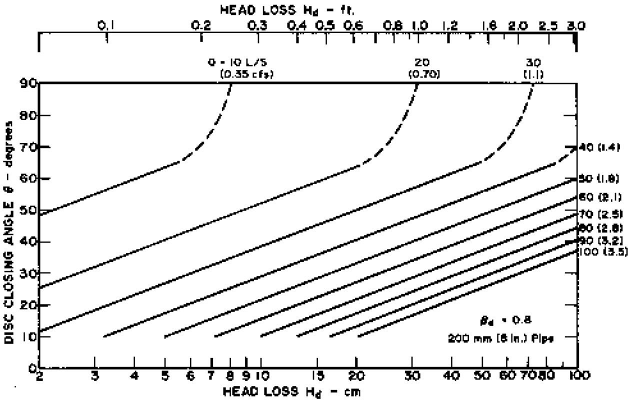

Fig. 9—Disc closing angle θ for a butterfly disc dissipator with a diameter ratio β_d of 0.8 as a function of head loss and flow rate for 200 mm (8 in.) diameter pipe.

which relate disc angular position to head loss and flow rate, can be constructed for the other two β_d values. Values for $\theta = 90$ deg in Fig. 9 were developed by using Equation [7].

The head loss increases exponentially with disc closing angle θ and, as can be seen by the slopes of the curves in Fig. 8, head loss becomes more sensitive to θ as β_d increases. By comparing the slopes of the curves in Figs. 3 and 8, it can be seen that disc dissipators can have a greater range of angular adjustment for a given range of energy dissipation than do butterfly valves. The curve slopes also show that the head loss for discs is not as sensitive to angular position as valves, particularly in the lower β_d range. Because of the wide range of adjustability, dissipators with discs having $\beta_d = 0.6$ or 0.8 will satisfy the general range of dissipation needed for most conditions as determined from diagrams such as Fig. 9.

Another method of expressing the loss coefficient for discs is to relate K_d to the projected area of the disc. The projected area at various angular positions is an ellipse with d_d as the major axis and $d_d \sin\theta$ the minor axis. Thus, the projected area, A'_{pr}, of the disc is

$$A'_{pr} = \pi/4\ d_d^2 \sin\theta \quad \text{[9]}$$

The total projected area also includes the projected area of the shaft, A_s, on each side of the disc where

$$A_s = T\,(D - d_d) \quad \text{[10]}$$

T = shaft diameter, L

Thus, the total projected area, A_{pr}, is

$$A_{pr} = \pi/4\ d_d^2 \sin\theta + T\,(D - d_d) \quad \text{[11]}$$

The straight line portion of the semilogarithmic relationship between K_d and the ratio of the projected area to the pipe area, A_{pr}/A_p, is shown in Fig. 10 ($r^2 = 0.996$) where

$$K_d = 0.12\ e^{7.31(A_{pr}/A_p)} \quad \text{[12]}$$

Equation [12] combines all of the variables into one expression for pipe size, angular positon and disc diameter ratio for area ratios less than about 0.75. The straight line portion of the curve can be used to construct

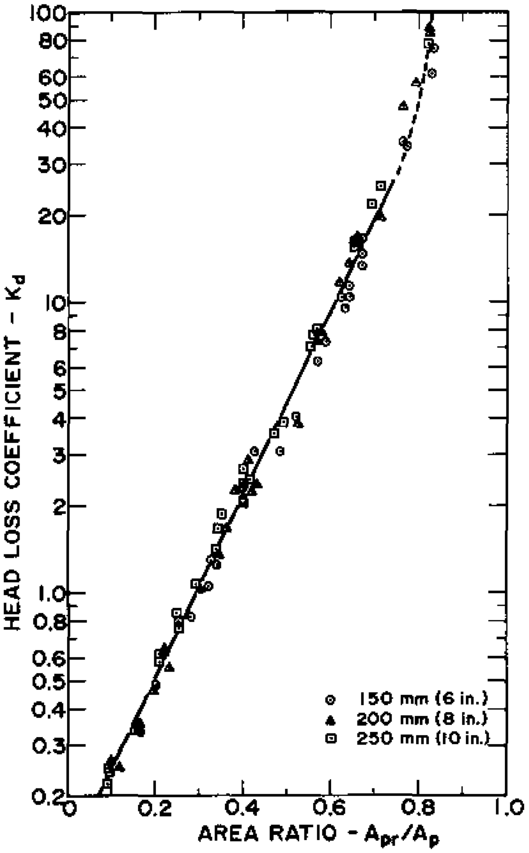

Fig. 10—Head loss coefficient, K_d, for variable position energy dissipating discs as a function of the disc projected area-to-pipe area ratio, A_{pr}/A_p.

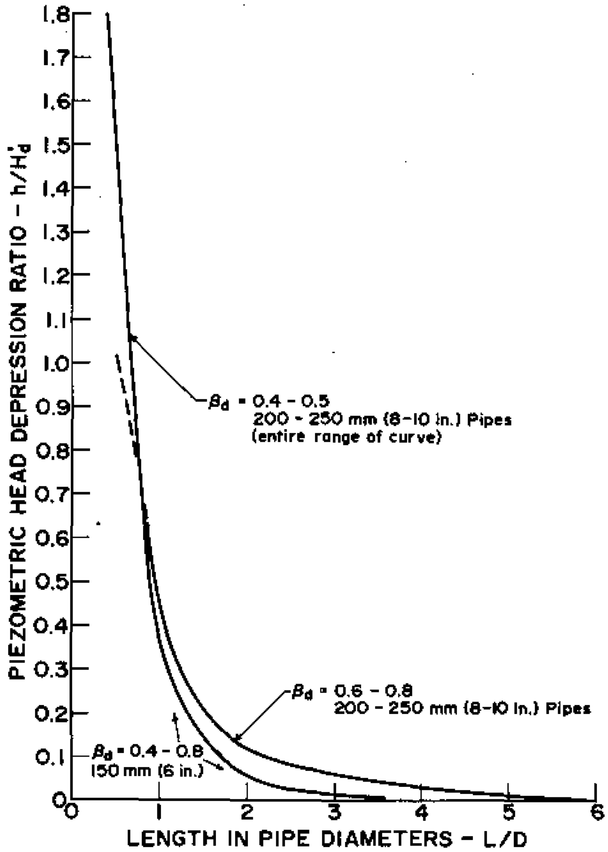

Fig. 11—Average piezometric head ratio, h/H_d', as a function of the pipe-diameter distance, L/D, downstream from butterfly discs positioned at 90 deg (normal) to the flow in an irrigation pipeline.

head loss diagrams such as Fig. 9 for other β_d values than those shown in Fig. 8. It will usually be more convenient to relate head loss to the disc closing angle θ and diameter ratio β_d directly as shown in Figs. 8 and 9.

As a cross verification of Equation [12], values of K_d were calculated for $\beta_d = 0.6$, 0.8 and 0.9 for three pipe sizes from which curves similar to those shown in Fig. 8 could be constructed. A regression on these estimates produced the constants shown in parentheses in Table 2. The values correspond to the test data within about 6 %, and thus verify the validity of Equation [12].

Disc dissipators do not appear to be very sensitive to errors in construction. One 250 mm dissipator was inadvertently assembled with one end of the shaft offset from the center approximately 12 mm (1/2 in.). Test data for this condition followed the $\beta_d = 0.8$ curve of Fig. 8 with the same deviation as subsequent data obtained after the error was corrected and the shaft realigned. Thus, if reasonable care is exercised in constructing the dissipators, the head loss can be determined from the relationships presented here with sufficient accuracy for practical purposes.

The dissipator angular setting and gate openings for a particular field layout may require a trial procedure during the first irrigation. However, after the settings are once determined for a given condition, only minor adjustments will generally be needed for subsequent irrigations. The excess energy head to be dissipated may be determined from pipeline or ground surface elevation profile measurements or estimated. Using a graph such as that shown in Fig. 9, the angular position of the dissipator disc is set for the desired head loss.

Hydraulic gradeline depression: Butterfly discs convert pressure head to velocity head as water flows through the restricted area around the disc. Consequently, the pipe pressure immediately downstream from the disc is reduced beyond that represented by the head loss, and is lower than that further downstream, as shown in Fig. 1. This reduces the flow from outlets or gates that may be located in this section of pipe. The piezometric head depression, h, is the elevation difference between the actual depressed hydraulic gradeline at a given point downstream from the disc and its projected elevation at that point when extrapolated upstream from the downstream section of pipe where full velocity head recovery is achieved. The annular shape of the flow past the disc contributes to strong turbulent mixing of the fluid elements; this causes rapid energy dissipation and velocity head recovery which occur over a short distance. The depression is not usually significant when short-pipe dissipator sections are inserted into a pipline because the distance between the disc and the first downstream outlet is sufficient for velocity head recovery. About the only time that depression could be a problem is when a butterfly disc is located in the upstream end of a full length of gated pipe. In this case, an outlet could be located close enough to the disc to be affected. To avoid this condition, the disc should be installed in the downstream end of the pipe below the last outlet. This would provide sufficient length for velocity head recovery before the first outlet in the next downstream length of pipe is reached.

The piezometric head depression is affected by the diameter ratio β_d, velocity head, pipe size, θ, and distance downstream from the disc. The depression, expressed as the ratio h/H_d', is shown in Fig. 11 for different values of β_d and pipe size. The curves in Fig. 11 were developed by cross plotting h/H_d' ratios obtained from the test data vs. pipe diameter length and β_d to obtain the ratio for even values of β_d vs. length. As seen in the figure, the depression is only significant within a distance of less than two pipe diameters downstream where it exceeds about one-tenth of the head loss. The depression at the first downstream outlet, which is the point of interest, can be estimated from the curves shown in Fig. 11 by knowing the head loss. The depression was related to H_d' for ease of estimation. For practical purposes, only a rough quantitative determination is needed when estimating h. The curves in Fig. 11 were developed from data obtained from the tests conducted with discs fixed in the 90 deg position. Since this is the most severe condition, depression data were not obtained for discs at smaller angular positions. The variable position discs were all tested in short-pipe dissipator sections, and the depression was not sufficient to be measured in the pipe downstream from the dissipator where the piezometer taps were located. For most practical purposes, H_d for values of θ less than 90 deg can be substituted for H_d' in the piezometric head depression ratio from which to estimate h.

SUMMARY

Butterfly discs, which are similar to butterfly valves, were developed and rated for energy dissipation. An energy dissipator consisting of a disc installed in a short length of pipe with couplings can be inserted into a pipeline at any joint. Laboratory tests were conducted to obtain head loss relationships for both butterfly valves and discs used for energy dissipation in low pressure pipelines such as gated pipe.

Head loss for butterfly valves and disc is a function of the velocity head and the angular position of the butterfly plate. The loss for discs is also a function of the disc-to-pipe diameter ratio. Coefficients determined for estimating the head loss can be described by an exponential equation of the form

$$K = a\,e^{b\,\theta} \qquad [13]$$

The constants **a** and **b** were determined for both valves and discs. The discs were also tested in a fixed position with their face normal to the flow for maximum restriction and head loss. The coefficient for this condition can be represented by an exponential function of the diameter ratio.

Low pipeline pressures immediately downstream from a disc can reduce the flow from pipe outlets if they should be located within a distance of about two pipe diameters downstream from the disc. This is not a problem if the disc is installed near the downstream end of a length of gated pipe, or in a short-pipe dissipator inserted into the pipeline at a joint.

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