FACTORS AFFECTING ORIFICE DISCHARGE IN A MULTI-OUTLET IRRIGATION PIPE

by

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B. S., Kansas State University, 1977

A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Agricultural Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1979

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ACKNOWLEDGEMENT

In a study like this, carried out under the constraints of both time and facilities, the guidance of an experienced professor is almost indispensable. I, therefore, shall like to acknowledge with profound gratitude the consistent guidance provided to me during the course of this undertaking by Dr. H. L. Manges, my major advisor and a professor in the Agricultural Engineering Department.

I shall also like to express my appreciation to Dr. J. Zovne of the Civil Engineering Department with whom I had consulted at several points during this study. My gratitude also goes to Dr. J. E. Kipp, a professor in Mechanical Engineering Department who gracefully agreed to discuss my problems at different times during the investigation.

Lastly, though by no means the least, I shall like to again express my appreciation this time to Mr. Darrell Oard who assisted me throughout the setting up of the experiment.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

INTRODUCTION

Irrigation of agricultural lands is one of the greatest advancements that man has achieved in his ever increasing battle against the unfavorable conditions that stand in the way of good and enormous agricultural production.

In irrigation, like many other fields of human endeavor, the struggle to move the system toward perfection has been the concern of many. Engineers working in the area have been forced, by the constraints that most irrigation systems provide, to redouble their efforts toward a higher efficiency in the use of the single most important resource required for the survival of the system, namely water.

Water losses due to evaporation and seepage in open-ditch surface irrigation systems have, for a long time, been matters of concern. The advent of pipe irrigation systems have proved a remedy to those problems but not without presenting some of their own. In furrow irrigation, for instance, gated pipes replace the ditches and siphon tubes for supplying water to furrows. However, the gated pipes present a problem of non-uniform flow through the various openings of the pipe. The ramifications of the problem are tantamount to an inefficient use of the water. If we take, for example, the condition in which more water comes out of the inlet end gates than from the closed end gates, the tendency would be either to irrigate adequately at the beginning and under-irrigate towards the end furrows along the pipe or to over-irrigate at the beginning furrows and irrigate adequately at the end furrows. The phenomenon reverses when there is more water coming out of the end furrows than at the beginning. In either case the consequences are undesirable. At present, farmers manually adjust the gates in an attempt to obtain uniform flow; an exercise that requires skill and good judgement and also one that shall always produce a result which is, at best, only approximate. Moreover, in places where labor-saving is an element of major concern to the economic well being of the system, there is a desire to automate the system. This, at present, is practical except for the problem of attaining uniform flow out of the gated pipes. Since with an automated system the aim is to minimize labor input, it shall be necessary to find a means of obtaining a uniform flow through the gates without the need for making adjustments.

The solution to this problem requires a careful examination of the factors that influence flow out of the various orifices of the pipe. The two factors that are known to have a direct effect upon the flow are the static head at the orifice and the discharge coefficient of the orifice. The possibility exists that this latter factor, the discharge coefficient, can in turn be influenced by several other factors. In fact, investigations and analysis of the problem made to date have indicated the presence of such factors. Two of them that have received an almost unanimous recognition are the velocity of water inside the pipe as it approaches the orifice and the static head at the orifice. However, the nature of the relationship between these factors and the coefficient has not been clearly understood or established.

The purpose of this study was to carry out an investigation to determine the relationship between the coefficient of discharge for orifices in a gated pipeline and the two factors, velocity of water approaching the orifice and static head at the orifice.

REVIEW OF LITERATURE

The problem of obtaining a uniform flow through several openings on the side of a pipe is not unique to gated irrigation pipes. It has been of concern to engineers dealing with manifolds in different systems well before the development of gated irrigation pipes. These, among others, include the pipe burner for gaseous fuels, manifold of the radiant fire headers for air heaters and distribution pipes in water filtering systems.

Some of the work was more or less of a trial and error nature in which the goal was to obtain uniform discharge through orifices of the manifold. Attention in earlier works had not been paid specifically to the factors on which the uniformity of discharge depends.

As early as the middle of the nineteenth century, Francis (1865) gave rules for the relation of orifice to pipe size in order to accomplish uniform discharge in the specific case of perforated pipes in a fire protection system. However, he did not analyze the various factors involved in the variation of the discharge from one orifice to another. Later, Jenks (1922) published more elaborate rules of a similar kind. These seemed to be confirmed by the results of studies on pressure distribution conducted by Elms (1927).

Enger and Levy (1929) published results of experimental studies which indicated that the Bernoulli equation could be used to establish the pressure distribution at various points in a pipe. The results further revealed the effect of water velocity inside the pipe on the coefficient of discharge through orifices. In their analysis, they negelected the pipe friction and hence, determined that pressure at any point on a slot is equal to the head at the end of the slot minus the velocity head at the given point. This they concluded from the following equation:

$$v = \sqrt{2g\ (H-y)} \tag{1}$$

where

v = velocity in the pipe at any section in question

H = head at the end of the slot

y = pressure at any point in question

g = acceleration due to gravity

They derived the above equation from the assumption of zero friction, the impulse momentum principles and the boundary condition that at the end of the slot, velocity inside the pipe was zero.

Assuming a constant discharge coefficient, Enger and Levy derived another expression, this time for pressure:

$$y = \frac{H}{2}\ \text{vers}\ \left(\pi - \frac{2Ce}{A}(L-X)\right) \tag{2}$$

where

C = coefficient of discharge of the slot

e = slot width

L = slot length

X = a distance from the beginning of the slot to a point in question

However, they later conceded to the fact that the discharge coefficient was not constant but rather decreased with an increase of the velocity in the pipe in accordance with the empirical formula:

$$C = \left(\frac{y - \frac{v^2}{2g}}{y}\right) Ce \tag{3}$$

where

C = coefficient of discharge on any opening

Ce = coefficient of discharge of the last opening

Computed values for the discharge coefficient using Equation 3 were found to be in close agreement with experimental values obtained in other studies. It is worth noting here that Enger and Levy did address themselves to the relation of orifice coefficient with both the static and velocity heads inside the pipe. Nevertheless, it should be pointed out that in their entire analysis they did not take into account the rate of recovery of pressure due to deceleration of flow in the axial direction which is due to the redirection in fluid momentum as a result of the discharge through the various orifices. Further, they didn't consider the effect of friction on the discharge rate.

Wills (1931) had performed some investigations on gas flow through burner manifolds. He also neglected friction losses and regarded the manifold as an infinite reservoir. Thus, the cross-sectional area of the manifold is a lot larger than the cross-sectional areas of all the orifices combined. Hence from these assumptions, uniformity of flow was eminent.

Keller (1949), in a comprehensive paper, dealt with the case of a manifold supplying fluid to a set of parallel pipes or ducts, or discharging through numerous orifices distributed along a pipe. He did not assume any particular state of flow, neither laminar nor turbulent. He took the friction factor to be constant along the entire length of the manifold irrespective of the flow rate. He also regarded as perfect the pressure recovery due to the axial deceleration of the flow. In addition, he neglected pressure losses due to branching. In his calculations, Keller determined the variation in width of a longitudinal slot in a pipe which would provide uniform outflow. He also determined the required change in cross-sectional area of a pipe essential for the provision of uniform outflow through slots of constant width. This latter aspect was treated by W. M. Dow as will be mentioned later.

The equation obtained by Keller, for the variation of discharge along a pipe length of constant cross-sectional area, is:

$$V_1 = \frac{V_o}{\mathrm{Sin}\ (KR)} \cdot \mathrm{Cos}\ (KR\ S/L) \qquad (4)$$

where

S = distance from the dead end of the manifold

V_o = inlet velocity

V_1 = discharge velocity at distance S from the dead end of the manifold

K = coefficient of discharge of the holes or slot

L = length of the manifold

R = area ratio = $\dfrac{\text{sum of areas of all discharge openings}}{\text{cross-sectional area of manifold}}$

It can be noted that the above equation is a form of the combination of Equations 1 and 2 given earlier as determined by Enger and Levy.

Dow (1950), in his theoretical analysis of flow through a perforated pipe with a closed end, assumed that it was possible to obtain a uniform discharge throughout the length of the pipe when the spacing and diameter of the orifices in the pipe wall were uniform. He also assumed a perfect rate of pressure recovery. His analysis, however, was restricted to finding the cross-sectional area of the pipe as a function of the distance along its axis. An implicit assumption that can be gathered from Dow's analysis is that if uniform discharge is to be attained with constant orifice area and spacing, then of necessity, the cross-sectional area of the manifold has to vary. Hence, he determined the area as a function of the axial length. The results of his analysis were verified by the photographs of flame heights from gas burners both before and after modification. This was, however, restricted to relatively short pipes.

The manifold problem was considered in a relatively more detailed form by Van Der Hegge Zijnen (1951). One of the typical cases of the problem he discussed was that of a turbulent flow inside the manifold and turbulent discharge through the orifices. The assumptions he made were those of isothermal conditions, incompressibility of the fluid, continuity of discharge over the manifold length and a continuous cross-sectional area of the manifold. He further assumed a constant coefficient of pressure recovery and constant coefficient of discharge.

Van Der Hegge Zijnen, in his analysis, started with the requirements that to obtain uniform discharge per unit length of the manifold, the orifice spacings, $^1/n$, could be determined if the diameters of both the pipe and the orifices were kept uniform, or the orifice diameter could be determined with the pipe diameter and orifice spacings made uniform or else the pipe diameter at any point could be determined while the orifice diameters and their spacings were kept constant. He assumed that the initial discharge into the pipe and pressure head at inlet were always known or could be prescribed. Among the conditions he considered were pressure recovery due to deceleration of flow as water exits through the orifices which he assumed counteracts the fluid friction and the bending of flow from the axial direction of the manifold to the lateral direction of the discharge. The latter condition, he believed, would make fluid resistance accompany the discharge through the orifices. He further suggested that the magnitude of the fluid resistance was determined by the velocity of flow just upstream from the orifice in question. He started his analysis with the equation of motion for the flow in a manifold, which was given as:

$$\frac{d}{dx}\left(p + K\cdot\tfrac{1}{2}PV^2\right) = -\lambda\cdot\tfrac{1}{2}PV^2/D \qquad (5)$$

where

p = pressure just upstream from an orifice

K = rate of pressure recovery

P = water density

V = velocity upstream from the same orifice

λ = coefficient of fluid friction

D = diameter of the pipe

For laminar flow in a pipe discharging through orifices, Van Der Hegge Zijnen set out to determine either the spacing or the diameter of the orifice per unit length, given that one of them was uniform and the pipe diameter was the same everywhere. In order to obtain uniform flow, the following condition had to be satisfied:

$$q = \frac{\pi}{4} d^2 \alpha[P_o - \tfrac{1}{2}PV^2 (\frac{64}{Re_o} - 2K \frac{D}{L}) \frac{x}{D} (1 - \frac{x}{2L})]^{\frac{1}{2}}(2/P)^{\frac{1}{2}} \qquad (6)$$

where

q = discharge through each of the orifices

P_o = the pressure at the inlet

Re_o = the Reynold's number at the inlet

α = a proportionality constant

x = distance from the inlet end

L = the total length of the pipe

The discharge q through the various orifices must sum up to the discharge Q_o into the pipe at the inlet. Hence $q_o = Q_o/nL$ in which n is the number of orifices per unit length. Equation 6 can then be used to determine the spacing, $\frac{1}{n}$, when both the pipe and the orifice diameters are uniform. On the other hand, the same equation can be used to determine the various sizes of the orifices if the spacing and pipe diameter are made uniform. In the case of turbulent flow, the following equation had to be satisfied for uniform flow:

$$P_o + \tfrac{1}{2}PV_o^2 \left[2 \frac{Kx}{L} \left(1 - \frac{x}{2L}\right) + 0.158 \, Re_o^{-\frac{1}{4}} \frac{L}{D} \frac{8}{11} \left[\left(1-\frac{x}{L}\right) \frac{5}{4} - 1\right] \right] = Q^2 \Big/ \left[\left(\frac{\pi}{4} d^2 nL\alpha\right)^2 2/p\right] \qquad (7)$$

The other alternative that Van Der Hegge Zijnen considered was that of maintaining a uniform spacing and equal orifice sizes while the pipe diameter changes per unit length. He developed the following equation:

$$\frac{A}{A_o} = 1 + 0.158 \frac{Re_o^{-\frac{1}{4}}}{K} \left[1-(1-x/L)^{3/8}\right]^{8/3} (1-x/L) \qquad (8)$$

where

A = cross-sectional area at any point

A_o = cross-sectional area at the inlet

Gladding (1940) discussed the loss of head in a pipe of uniform diameter. Using a simple analysis made possible by the assumption that it was possible to obtain an equal discharge through evenly spaced outlets on the side of a pipe, he showed the loss of head can be given by the expression:

$$h = FKLQ^m \qquad (9)$$

where

h = loss of head

K = a constant

m = a constant

L = length of the pipe

Q = total discharge into the pipe

F = a quantity which itself is given by

$F = \Sigma N^m / N^{m+1}$ in which N is the number of outlets

Howland (1953) presented a paper to the 3rd Midwestern Conference on Fluid Mechanics in which he discussed the design of perforated pipe for uniformity of discharge. He established from Bernoulli's equation the variation of pressure head along a manifold pipe with closed end when uniformity of discharge was attained. He gave the expression as:

$$h = h_o - \frac{V_1^2}{2g} (X/X_1)^2 + h_L \qquad (10)$$

where

h = pressure head at any point along the pipe

h_o = pressure head at the closed end of the pipe

V_1 = velocity at entrance to the pipe

h_L = head loss due to pipe friction in the portion of the pipe between the point under consideration and the closed end

X = distance from the point to the closed end

X_1 = length of the pipe

Howland further showed that the friction loss, h_L, could be expressed as:

$$h_L = \int_o^X \frac{dh}{dx} dx = \int_o^X S dx \qquad (11)$$

where

$S = \frac{dh}{dx} = S_1 (V/V_1)^n$

S_1 = head loss per unit length at the entrance

S = head loss per unit length at point under consideration

V = velocity at point under construction

n = a constant varying between 1.75 and 2.00

The final form of the total head loss equation was:

$$h - h_o = \frac{V_1^{\,2}}{2g} \quad [\ \frac{f_1 X_1}{(n+1)\ D}\ (X/X_1)^3 - (X/X_1)^2\ ] \qquad (12)$$

where

f_1 = friction factor

D = pipe diameter

Howland gave consideration to the change of orifice coefficients with a change in velocity. He developed an empirical curve that established the orifice coefficient as a function of pipe velocity.

Hansen (1954) wrote a paper on the determination of water flow from gated pipe in which he indicated that uniformity of flow was obtained when he used two 20-foot, 4-inch gated pipes connected at the middle by a 5-foot tee section and placed on a 1 in 300 slope. He used 22-inch and 36-inch gate spacings to obtain data to establish discharge-opening-head relationships.

Tovey (1959) followed the recommendation of Hansen using similar 4-inch gated pipe and attempted to duplicate his results but found that uniform flow could not be obtained for 0.55-inch gate openings where the average head loss per gate exceeded 0.1 foot. In a similar manner he also found that with 0.95-inch and 1.7-inch gate openings, uniform flow could not be obtained when the average head loss per gate exceeded 0.15 foot and 0.17 foot, respectively.

Spomer (1961) also used the Hansen slope recommendation for a 6-inch pipe. He found that instead of uniform flow, a rising slope of 1 in 300 gave a decrease in discharge from the gates as one approaches the dead end. The reverse occurred when the pipe was on a falling slope.

Ramirez-Guzman and Manges (1971), in a paper presented to the annual meeting of the American Society of Agricultural Engineers, discussed uniform flow from orifices in irrigation pipe. They used a method of analysis similar to that of Howland mentioned earlier. They also obtained an equation similar to Equation 12 derived by Howland. The major difference between the two analyses was the difference in the initial conditions of the pipe. Howland considered the pipe to be horizontal while Ramirez-Guzman and Manges took the pipe to be sloping. Hence, the elevation term appeared in the latter's derivation. Further, while Ramirez-Guzman and Manges used the Hazen-Williams equation to express the energy slope as a function of the velocity, the friction coefficient and the hydraulic radius of the pipe, Howland expressed the same quantity (i.e. the slope) as a function of pipe friction, the pipe diameter and the flow velocity inside the pipe. The following two equations express slope as used by Ramirez-Guzman and Manges, and Howland, respectively:

$$h_1 = \left[\frac{V_1}{1.318C_H}\right]^{1.85} \frac{1}{R^{1.17}} \tag{13}$$

where

h_1 = energy slope at pipe entrance

V_1 = velocity at the entrance to the pipe

C_H = Hazen Williams friction coefficient

R = hydraulic radius

$$S_1 = \frac{f_1 V_1^2}{2g\ D} \tag{14}$$

where

f_1 = pipe friction

V_1 = velocity at entrance

D = pipe diameter

The final form of the equation derived by Ramirez-Guzman and Manges was:

$$h = h_o - \frac{V_1^2}{2g} (X/X_1)^2 + \left[\frac{V_1}{1.318C_H}\right]^{1.85} \frac{X_1}{2.85R^{1.17}} (X/X_1)^{2.85} + \frac{SX}{X_1} X_1 \quad (15)$$

where

h = pressure head at any point in the pipe

h_o = pressure head at dead end

X_1 = distance of the inlet end from the dead end

X = distance from the dead end to the point in question

Using the above equation, Ramirez-Guzman and Manges calculated the head at various points along the pipe and thence, determined the discharge out of each orifice assuming a unity coefficient of discharge. They then went back to correct for the assumption of unity coefficient of discharge. This time around, they took the coefficient to be the ratio of the combined discharges out of the orifices to the total discharge that entered the pipe. Based on this, they adjusted their calculations for the discharge through each of the orifices. Upon comparison with experimentally determined data, as observed by Spomer, some agreements were achieved with variation of up to 6.5%. They, therefore, concluded that the equation they derived was accurate enough for field use.

Even though in their calculations Ramirez-Guzman and Manges had assumed that the orifice coefficients were the same, they nonetheless conceded that, due to the consistent overestimation of the orifice discharge at the inlet end and an underestimation near the dead end, there was an inverse variation in the magnitude of the discharge coefficient with distance from the dead end.

Chu and Moe (1971) followed up the analysis made by Ramirez-Guzman and Manges. They introduced a new element by taking into consideration the variation in the coefficient of discharge coefficient with velocity. It should be mentioned, however, that their initial approach differs from that of Ramirez-Guzman and Manges even though it was obviously based on the latter's analysis.

Chu and Moe started with Bernoulli's energy equation expressed between two points at the centers of two adjacent orifices. Thus they wrote:

$$h_{i+1} + \frac{V_{i+1}^2}{2g} + Z_{i+1} = h_i + \frac{V_i^2}{2g} + Z_i + h_L \tag{16}$$

where

Z = elevation at a point

i = 1, 2......., N = the subscript identifying the outlet

N = the total number of outlets

h_L = head loss due to friction and branching of flow

Flow through individual orifices can be expressed as:

$$q = ca\sqrt{2gh} \tag{17}$$

where

q = discharge out of an orifice

c = orifice coefficient of discharge

a = area of the orifice

h = head at each orifice

g = acceleration due to gravity

Chu and Moe pointed out that, in order to have uniform flow from the various orifices at constant spacings and equal sizes, the head at each outlet should be the same, if the orifice coefficient remains constant. Thus:

$$h_{i+1} = h_i \tag{18}$$

where

$i = 0$ to N

The energy equation is thus reduced to:

$$\frac{V_{i+1}^2}{2g} + Z_{i+1} = \frac{V_i^2}{2g} + Z_i + h_L \tag{19}$$

They further suggested:

$$h_L = h_f + \frac{K}{2}\left[\frac{V_{i+1}^2}{2g} + \frac{V_i^2}{2g}\right] \tag{20}$$

where

h_f = the frictional loss of head of pipe

K = coefficient of energy loss of head

It is worth noting here that Chu and Moe took into consideration the loss that resulted from the branching of flow. In the last part of their analysis, they removed the assumption that the orifice coefficients are the same for all of the orifices.

In their analysis, Chu and Moe developed a factor which is a function of the number of outlets on a lateral. They expressed the factor, F(N; CF, K) as:

$$F\,(N;\ CF,\ K) = \frac{CF}{N^{m+1}} \quad \frac{K}{3}\ N^3 - (1+\frac{K}{2})\ N^2 + (1 + \frac{SK}{12})\ \ N - \frac{K}{4} \quad + (F - \frac{1}{N}) \tag{21}$$

where

$CF = \frac{1}{2}\, F_1^{\,2}/S_{f_1}$

F_1 = Froude number with respect to 1st section of the lateral counting from the dead end

S_{f_1} = the slope of the energy grade line for the same section

F = Christianson's factor

Also:

$$F\ (N;\ CF,\ K) = (Z_N - Z_1)/S_{fN}\ (NL) \tag{22}$$

where

S_{fN} = head loss due to friction in the corresponding main pipeline

$= S_{fi}\ L(N)^{m+1}$

$$F = \frac{1}{m+1} + \frac{1}{2N} + \frac{(m-1)^2}{6N^2} \tag{22a}$$

m = constant = 1.9

A table of values for F has been generated by Christianson (1942) for different values of N and K.

Substituting Equation 22a into Equation 21 and simplifying yields the following:

$$F\ (N;\ CF,\ K) = \frac{CFK}{3}\ N^{2-m} - CF\ N^{1-m} + \frac{1}{m+1} - \frac{1}{2N} \tag{23}$$

Chu and Moe finally concluded with an expression for the slope of a pipe, S_p', where

$$S_p' = \frac{1}{I-1}\ [F(jI;\ CF,\ K)\ S_{f\ jI} - F(j-1)I+1;\ CF,\ K)\ S_{f(j-1)I+1}] \tag{24}$$

where

S_p' = slope of the j^{th} pipe counting from the dead end

j = the number that identifies the gated pipe

$S_p = S_p'/(1-S_p'^2)^{\frac{1}{2}}$

S_p = slope of the j^{th} pipe counting from the dead end, with respect to ground surface

The slope, S_p', was also regarded as the average slope on which the pipe should be installed.

In a paper published four years later by the Journal of Mechanical Engineering Science, Bailey (1975) discussed and analyzed the nature of "fluid flow in perforated pipes." Bailey pointed out two factors as being the major causes of variation of flow along a perforated pipe or a manifold, namely friction and decrease in fluid momentum. While a lot of work had been accomplished in the field of friction losses in pipes, he stated, there was little, if anything done or known about the nature in which the decrease in fluid momentum affected the fluid pressure downstream of an outlet. Elaborating on the point, he maintained that the decrease in fluid momentum resulted in an inevitable decrease in the fluid velocity inside the pipe after passing each discharging orifice. This, he said, resulted in a corresponding increase of the fluid pressure, a phenomenon variously described as the "diffusion," "inertia" or the "static regain" effect.

In his analysis, Bailey suggested that the component of the velocity of the emerging fluid, which was perpendicular to the plane of the orifice, was derived from the excess static pressure in the pipe. The fluid was assumed to be coming out at an angle to a perpendicular line which, in turn, is at right angles to the plane of the orifice. The velocity component, Vy, was, therefore, expressed accordingly as:

$$V_y = \left[\frac{2(P_1 + P_2)}{2\rho}\right]^{\frac{1}{2}} \qquad (25)$$

where

V_y = y-component of the emerging fluid velocity

P_1 = pressure inside the pipe in the pipe section preceding the orifice in question

P_2 = pressure in the pipe after passing the same orifice

ρ = fluid density which is constant

It can be noted that Bailey had assumed, by implication, that the pressure, P, at the orifice is the average of the pressure upstream and the one downstream of the orifice (i.e. $P = (P_1 + P_2)/2$). Equation 25 above can hence be reduced to the one usually obtained by the Terricelli's principle (i.e. $Vy = 2gh$, where $h = P/\rho g$). Bailey then described the flow out of each orifice as:

$$q = \frac{\pi d^2}{4} \; C_d \left[\frac{P_1 + P_2}{\rho} \right]^{\frac{1}{2}} \tag{26}$$

where

q = the discharge out of the orifice

C_d = discharge coefficient

d = the orifice diameter

As a result of this discharge, he explained that the momentum balance parallel to the pipe axis can be established as:

$$P_2 - P_1 = \rho \; (V_1^2 - V_2^2) \tag{27}$$

where

V_1 = velocity in the pipe prior to the orifice of interest

V_2 = velocity in the pipe after passing the same orifice

The relation, however, was established upon the assumption that the emerging fluid had lost all of its axial momentum. That is, it was emerging at 90^o to the axis of the pipe. However, since this was not so in reality, a correction factor was introduced into the above expression. Thus:

$$P_2 - P_1 = \frac{C_r}{2} \rho \;\; (V_1^2 - V_2^2) \tag{28}$$

$P_2 - P_1$ is the pressure increase across the orifice which has been referred to as the 'static regain'. The coefficient C_r introduced in the above equation is, therefore, defined as the 'coefficient of static regain'. Rearranging the equation, he showed the coefficient can be given by:

$$C_r = \frac{P_2 - P_1}{\frac{1}{2}\rho(V_1^2 - V_2^2)} \qquad (29)$$

Further to these analyses, Bailey had carried out an experiment using two lengths of 0.2m diameter rigid PVC tubing, each of length 1.2m. The two were connected with flanges. At 0.25m downstream from the flange a 25mm orifice, punched in a 0.064mm thick polyethelene sheet was mounted beneath a 100mm diameter hole made on the wall of the PVC tubing. In a plane perpendicular to the axis of the orifice, static pressure tappings were made on the surface along one side of the test section. Air was used as the system's fluid. A pitot probe was used to determine the dynamic head of the emerging air with the aid of a collecting chamber. A pitot traverse was used to determine the dynamic head inside the pipe.

As a result of his experiment, Bailey came up with an empirical equation expressing the orifice discharge coefficient as:

$$C_d = 0.62 + 0.070\beta + 0.088\beta^2 \qquad (30)$$

where

$$\beta = \log\ [\log\ [(1 + \frac{P_1 + P_2)}{V_1^2}]\ ]$$

Since the distributor he considered in his paper consisted of a large number of diametrically opposed pairs of perforations, Bailey found it impractical to measure the changes in velocity after passing each hole, because of the small amounts of air coming out of each. Hence, he determined the static pressure profile experimentally and developed a model with the aid of computer that gave the coefficient of static regain as:

$$C_r = 0.78 + \phi \log \left[\frac{V_1}{V_1 - V_2}\right] \tag{31}$$

ϕ can be expressed as:

$$\phi = 0.284 + 0.098 \log \frac{d}{D} \tag{32}$$

where

d = diameter of an orifice

D = diameter of the perforated pipe

In addition, Bailey was able to measure, by means of an anemometer, the angle of discharge for each orifice. He subsequently established the following empirical relationship:

$$\theta = \partial \arctan \left(\frac{V_1^2 \rho}{P_1 + P_2}\right)^{\frac{1}{2}} \tag{33}$$

where

θ = 0.71 + 0.0043d, the angle that the fluid made with the axis of the orifice

d = diameter of the orifice

V_1 = approach velocity to the orifice

P_1 = pressure in the section just before the orifice

P_2 = pressure just after the orifice at the section immediately downstream from it

In conclusion, Bailey suggested that the use of these coefficients will enable predictions to be made of both the static pressure and fluid discharge variations along uniformly perforated distributors. He maintained that such predictions had agreed with experimental observations.

INVESTIGATION

Objectives

Analyses presented earlier and studies by other researchers have suggested the presence of some effect of an approach velocity towards an orifice and of pressure head at the orifice upon its discharge coefficient. The review of literature revealed the existence of only one empirical formula relating these quantities. The formula, given earlier as Equation 3, cannot be used without assuming some value for the discharge coefficient at the dead end. However, since the value at the dead end is not constant, the criterion for choosing it shall have to be arbitrary. Hence, the accuracy of any prediction of the discharge coefficient at points other than the dead end shall have to depend upon the accuracy of the assumed value at the dead end. Therefore this investigation was undertaken with the following objectives.

1. To determine the relationship between orifice discharge coefficient and two parameters; approach velocity and static head.
2. To determine if the results of studies by Enger and Levy on 2-inch diameter pipes can be applied to larger pipe sizes, such as those used for gated pipe.
3. To determine if equations obtained as a positive achievement of objective 1 fit other published data of flow from orifices of a manifold pipe.
4. To determine a procedure for designing a system that shall provide uniform flow out of the various outlets of a gated pipe system.

Theory

Whenever there is an opening in the wall of a conduit or any container such that a fluid issues through the entire area of the opening under pressure, the opening is referred to as an orifice.

The conservation of energy equation, better known as Bernoulli's equation, requires that for any two points 1 and 2 ir a stream tube:

$$\frac{V_1^2}{2g} + h_1 + Z_1 = \frac{V_2^2}{2g} + h_2 + Z_2 + h_L \tag{34}$$

where

V_1 and V_2 = velocity at points 1 and 2, respectively

h_1 and h_2 = static head at points 1 and 2, respectively

Z_1 and Z_2 = elevation at points 1 and 2, respectively

h_L = the total specific energy lost in the journey of the fluid from point 1 and 2

Figure 1 shows one such stream tube. The velocity at point 1 is very low and can be taken to be zero. The elevation is the same for both points 1 and 2 and, thus, cancel. Loss of energy as water moves from point 1 to 2 was neglected. Also since the static heads were derived from the gage pressures, the head outside the tank is necessarily zero. With these boundary conditions, Equation 34 becomes:

$$V_2 = \sqrt{2gh} \tag{35}$$

V_2 is the ideal velocity of water issuing from the orifice and shall henceforth be referred to as simply V.

Losses due to various factors such as the fluid viscosity make the introduction of a correction factor, C_V, essential. Hence; $V = C_V \sqrt{2gh}$.

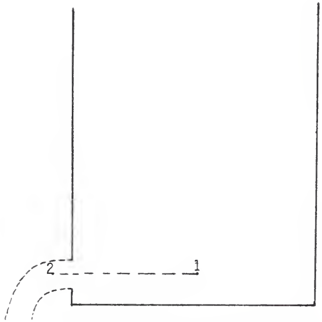

Figure 1. Discharge Through an Orifice in a Tank.

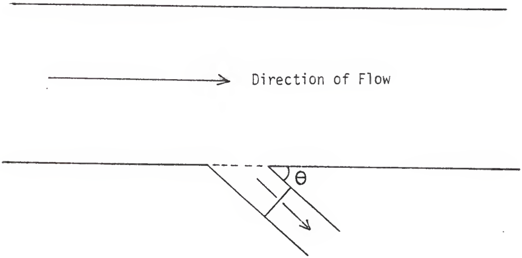

Figure 2. Discharge Through an Orifice in a Pipe.

Continuity requires that:

$$q' = aV \tag{36}$$

where

q' = orifice discharge

a = area of the orifice

Anywhere outside the orifice, the true area of discharge is always less than the area of the orifice. Another correction factor, Ca, is introduced to take care of the reduced area. Thus, the true discharge becomes:

$$q = C_a C_V A \sqrt{2gh} \tag{37}$$

The coefficient of discharge, C_d, is then defined as the product of the two correction factors (i.e. $C_d = C_a C_V$).

Hence:

$$q = C_d A \sqrt{2gh} \tag{38}$$

Equation 38 has been used widely in determining orifice flows even in the case of manifold pipes. Nevertheless, since the premises upon which the theory was built is not the same as what is found in the case of a manifold pipe, reassessment of the various factors involved is in order.

One of the approximations made in the above analysis was that the velocity at point number 1 was zero due to the fact that the amount of water that exits through the orifice is relatively very small as compared to the total amount of water in the tank. In the case of a manifold pipe, however, since most of the water is moving past the orifice, the velocity inside the pipe cannot be assumed to be zero. This provides a basis for another new approach to the analysis.

If Bernoulli's equation is written for two points, one inside a pipe directly opposite the center of an orifice and the other outside on an axis of discharge through the orifice, the two points can be referred to as point 1 and point 2, respectively. Bernoulli's equation, given as Equation 34, can again be used.

The orifice in this pipe shall in this analysis be taken to be on the side of the pipe such that the axis of discharge through it is at right angles to the axis of flow inside the pipe and the discharge exits horizontally. The elevation of the two points 1 and 2 are, therefore, the same. Hence the terms Z_1 and Z_2 in Equation 34 are eliminated. In addition, the static head at point 2 (i.e. h_2), shall also be zero since the gage pressure outside the pipe is zero.

Equation 34, then, reduces to:

$$V_2 = \sqrt{2g\left(\frac{V_1^2}{2g} + h_1 - h_L\right)} \qquad (39)$$

If the head loss due to branching and exit is taken to be a fraction of total head inside the pipe, that is both dynamic and static, the head loss can then be expressed as;

$$h_L = K\left(\frac{V_1^2}{2g} + h_1\right) \qquad (40)$$

Equation 39 then becomes:

$$V_2 = \sqrt{2g\left[\left(\frac{V_1^2}{2g} + h_1\right) - K\left(\frac{V_1^2}{2g} + h_1\right)\right]} \qquad (41)$$

Therefore:

$$V_2 = (1 - K)^{\frac{1}{2}} \sqrt{2g \left(\frac{V_1^2}{2g} + h_1\right)} \tag{42}$$

If we let $C_V = (1 - K)^{\frac{1}{2}}$, then:

$$V_2 = C_V \sqrt{2g \left(\frac{V_1^2}{2g} + h_1\right)} \tag{43}$$

If the total area, occupied by the discharging flow while inside the pipe, is A', then continuity requires that:

$$A'V_1 = (C_c A)V_2 \tag{44}$$

where

C_c = correction factor for the cross-sectional area of the discharge that comes out of the orifice since a contraction always occurs; the factor is also known as the coefficient of contraction

a = area of the orifice

Thus:

$$V_1 = \frac{C_c A}{A'} V_2 \tag{45}$$

Equation 43 becomes:

$$V_2 = C_V \sqrt{2g \left[\left(\frac{C_c A}{A'} \frac{V_2^2}{2g} + h_1\right]} \tag{46}$$

Simplifying, we obtain:

$$V_2 = \frac{C_V \sqrt{2gh_1}}{\sqrt{1 - (C_V C_c \frac{A}{A'})^2}} \tag{47}$$

The factor, $(C_V C_c \frac{A}{A'})^2$, in the demoninator of the equation is so small compared to 1 that it can be assumed to be zero. In which case:

$$V_2 = C_V \sqrt{2gh_1} \tag{48}$$

The discharge out of the orifice then becomes:

$$q = (C_c A) \quad C_V \sqrt{2gh_1} \tag{49}$$

Thus:

$$q = C_d A \sqrt{2gh} \tag{50}$$

where

$h = h_1$ in the preceding equation

A completely different approach to the analysis of orifice discharge is worth considering. In the interest of clarity, the following conditions for both the pipe and the flow through it shall be stated as:

1. The pipe is horizontal
2. The orifices are in the vertical plane
3. The end of the pipe is plugged

With these conditions we can then move on to say that at any point of exit, the force that will push the water through the hole causing it to change directions must be the gage pressure inside the pipe. The angle that the discharge out of the orifice makes with the axis of flow at the point of exit depends upon how much of its axial momentum it has lost. If it loses all, then the angle has to be 90^o. However, since this is seldom the case, we shall assume a general case in which the water retains a significant amount of its momentum after exit.

Figure 2 shows the diagram of water flowing out of one of the orifices of a pipe. Since the water is coming out at an angle θ, to the axis of flow inside the pipe, the actual cross-sectional area of the stream shall be:

$$a_s = a \sin \theta \tag{51}$$

where

a = the area of the orifice

It can further be stated that the effective force acting on the issuing stream of water must be one that is applied in the same direction as the flow. Since the water pressure, P, at the orifice acts in all directions, the effective force shall be the pressure applied on the cross-sectional area of the water stream. Thus, the force, F, can be given by:

$$F = P\ a \sin\theta \tag{52}$$

The momentum principle requires that:

$$\Sigma F = \frac{D}{DT}\int_{\forall \text{ material volume}} \rho V d\forall = \frac{\partial}{\partial t}\int_{\text{control volume}} \rho V D\forall + \int_{\text{control volume}} \rho V(V\cdot ds) \tag{53}$$

where the term ΣF on the left represents all the forces acting on a fluid while the terms on the right represent the total rate of change of momentum. These can be rewritten as:

$$\Sigma F = \frac{\partial M}{\partial t} + \iint_S \rho V(V\cdot ds) \tag{54}$$

where

$M = \iiint_{\forall} V\rho d\forall$, the momentum of fluid within the control volume for the fluid initially within a control surface

$\frac{\partial M}{\partial t}$ = the rate of change of momentum within the control volume

$\iint_S \rho V(V\cdot ds)$ = the momentum flow rate outward through the control surface

For steady flow, that is when tne velocity at any one point does not change with time, the $\frac{\partial M}{\partial t}$ goes to zero if the density of the fluid is also assumed to be constant. Hence

$$\Sigma F = \iint_S \rho V(V\cdot ds)$$

This can be simplified into the form:

$$\Sigma F = (\dot{M}V)\text{ leaving} - (\dot{M}V)\text{ entering} \tag{55}$$

Breaking the forces into their various components,

$$\Sigma F_x = (\dot{M}V_x) \text{ leaving} - (\dot{M}V_x) \text{ entering} \tag{56}$$

$$\Sigma F_y = (\dot{M}V_y) \text{ leaving} - (\dot{M}V_y) \text{ entering} \tag{57}$$

$$\Sigma F_z = (\dot{M}V_z) \text{ leaving} - (\dot{M}V_z) \text{ entering} \tag{58}$$

where

F_x, F_y, F_z = the x, y, and z components of the sum of the forces

V_x, V_y, V_z = the x, y, and z components of the exit and inside velocities of the issuing water.

In Figure 2, a flow was shown issuing out of an orifice at an angle θ to the axis of flow. If the effective force acting on it is P a sinθ as earlier stated, then, using the convention indicated in the figure, the y-component of the force, F_y, d_L is given by:

$$F_y = Pa\sin^2\theta \tag{59}$$

Since F_y is the only force acting in the y direction and the initial velocity of the water in the same direction was zero, the principle of momentum requires that:

$$F_y = \rho q\ (V_e \sin\theta - 0) \tag{60}$$

where

V_e = the exit velocity

q = the discharge out of the orifice

ρ = water density

Thus:

$$Pa\sin^2\theta = qV_e\sin\theta$$

But, since asinθ is the actual cross-sectional area of the issuing stream:

$$V_e = \frac{q}{a\sin\theta} \tag{61}$$

$$\therefore\ \rho q \cdot \frac{q}{a\sin\theta} \cdot \sin\theta = Pa\sin^2\theta$$

$$\text{or } \rho q^2 = Pa^2\sin^2\theta$$

But $P = \rho gh$

where

h = head in feet of the pressure at the orifice

$$\therefore\ \rho q^2 = gha^2\sin^2\theta$$

$$\text{or } q^2 = gha^2\sin^2\theta$$

$$\therefore\ q = \frac{\sin\theta}{\sqrt{2}} \cdot a\sqrt{2gh} \tag{62}$$

This equation is of the same form as Equation 38. So, if we are to use the latter, then we shall know that $C_d = \dfrac{\sin\theta}{\sqrt{2}}$.

The above analysis indicates that the discharge coefficient is a function of the angle of discharge for the orifice. It can be recalled that the angle of discharge itself has been stated to be a function of both the velocity of flow inside the pipe and the pressure at the orifice. It can therefore be inferred that the discharge coefficient, C_d, is a function of both the velocity of the fluid as it passes the orifice and the pressure head at the same point.

If there is no analytical way of relating the angle of discharge to the two factors affecting it, then an empirical relationship can be sought through experiments to determine a direct relationship between the discharge coefficient and the two factors.

Materials and Procedure

The experiment was conducted in the hydraulics laboratory of the College of Engineering at Kansas State University. Some of the basic equipment used was already available. An underground tank in the laboratory was employed as the water source.

Among the available equipment was a horizontal centrifugal pump driven by a 20 h.p. variable speed motor shown in Figure 3. This was used for drawing the water from the tank. The pump was primed by means of a vacuum pump which can be seen in Figure 4.

The equipment was initially arranged such that water was pumped into a six-inch pipe which led to a six-inch valve. The valve, shown in Figure 5, controlled the rate of flow. Flow then passed through another short section of six-inch pipe into a six-inch to eight-inch pipe adapter. A 90^{o} elbow was connected to the adapter and to this elbow was, in turn, connected a tee. The middle outlet of the tee was plugged so that the flow that came out of the elbow could pass straight through it. It was, thus, used simply as a short section of pipe. From the tee, the flow passed into a second 90^{o} elbow. Another connection of 20-foot length of an eight-inch plain pipe was made at the other end of the elbow. To this pipe was connected the test pipe as shown in Figure 6.

Later, when surging was found to be a problem, the set up was modified by making the flow pass through a stand pipe. The tee used earlier as a pipe section was connected directly to the six-to-eight-inch adapter mentioned earlier. The other end of it was plugged while a plain pipe about 10 feet in length was connected to its middle outlet and the flow was made to go vertically upwards through the pipe. The flow then discharged into the stand pipe after passing through two 90^{o} elbows which

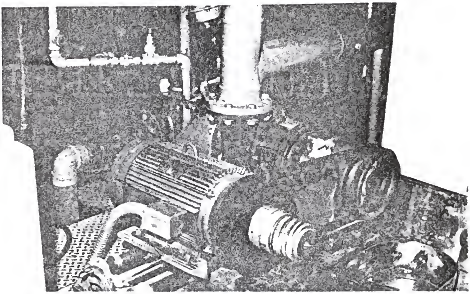

Figure 3. Pump in Kansas State University Hydraulics Laboratory.

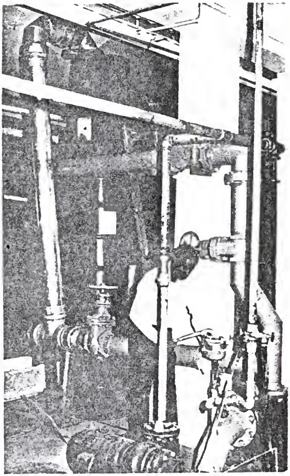

Figure 4. Vacuum Pump for Priming Pump and Associated Valves and Piping.

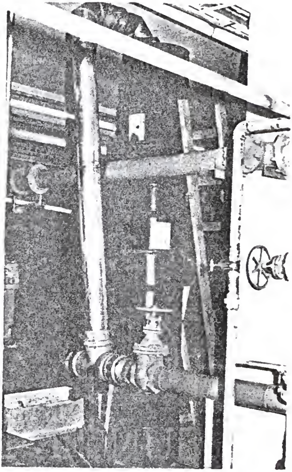

Figure 5. The Flow Control Valve.

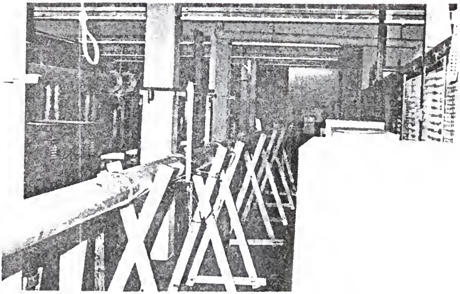

Figure 6. Adjustable Wooden Supports that Held the Pipe.

were connected together. The 20-foot length of pipe mentioned earlier was then directly connected to the stand pipe.

The test pipe used was thirty feet long and eight-inches in diameter. The eight-inch, 20-foot long pipe, was used for stabilizing the flow as it came out of the last elbow before it entered the test section. Measurements of the total discharge passing through the pipe were made with a Jaccuzi flow meter. Figure 7 shows the flow meter. A differential manometer, shown in Figure 8, was used whenever the determination of head difference between two points was desired. Copper tubings, about 1 1/2 inch in length and 0.125-inch internal diameter, were glued to the test pipe at the various orifices and through them holes were drilled into the pipe to provide pressure tappings as shown in Figure 9. Head readings were taken from a set of manometers mounted in a protective steel frame against a set of scales marked from zero to seventy-two inches. These are shown in Figure 10. Connections between the manometers and the copper tubings were made with 0.25-inch inside diameter flexible polyethelene tubings.

An initial test was made to determine the friction coefficient of the test pipe. Using two pressure tappings, 27 1/2 feet apart, with one being 15-inches from the inlet end and the other being the same distance from the outlet end, the head loss between the two points was determined with the aid of the differential manometer. The velocity of flow was measured with the flow meter. Readings were taken at ten different points across a section of the pipe, each representing one of ten equal areas bounded by ten concentric circles. The average of the ten was then taken to be the average velocity.

Figure 7. The Jaccuzzi Flow Meter.

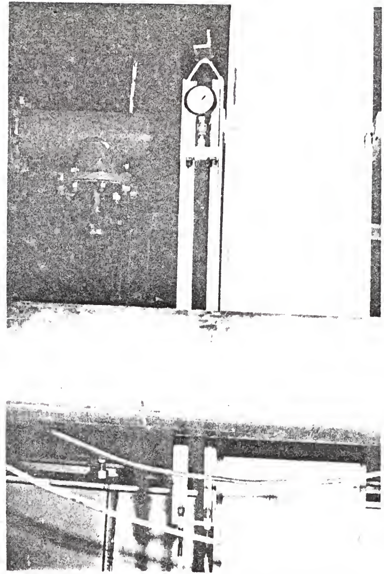

Figure 8. The Differential Manometer.

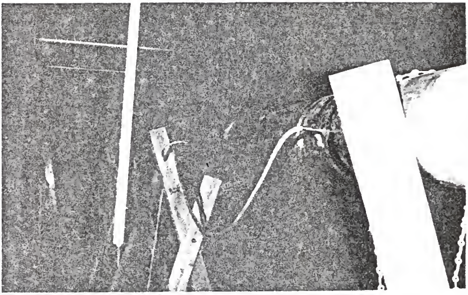

Figure 9. Pressure Tappings on the Test Pipe.

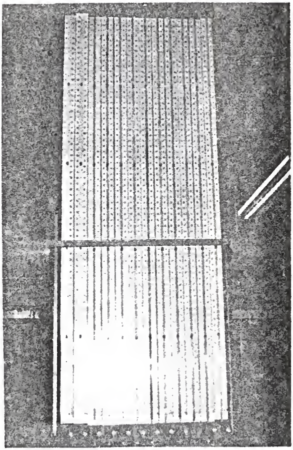

Figure 10. Set of Manometers.

The test pipe was then disconnected from the 20-foot pipe. Holes were drilled in the pipe with a drill press. During the drilling process, the pipe was placed in a wooden saddle such that about half of its perimeter was in the saddle. A metal band was strapped around the upper part of the pipe and screwed onto the saddle for added stability. The drill bit was lowered gently onto the pipe, each time, and was run at its slowest speed to avoid any wobbling that would easily jeopardize the preciseness of the holes. The arrangement can be seen in Figure 11. Only six holes were drilled on the test pipe, each 13/16 inch in diameter and all equally spaced at 5 feet from center to center. A manually operated crane was used for lifting the barrels which were used for collecting the discharge out of the various orifices of the pipe. The crane used is shown in Figure 12.

The pipe was then connected back onto the 20-foot pipe and flow diverting devices were mounted on it with one unit at each hole. The open end was plugged, thus becoming a dead end.

Water was run through the system and discharged through the six orifices. The static head at each outlet with reference to the center line of the orifice was recorded from the set of manometers previously mentioned. The discharged water was collected in barrels for some recorded time interval. The barrels were always weighed while empty, just before the water collection started. They were reweighed again while containing the water.

The flow diverting device mounted on the pipe initially was one that allowed the discharge out of all the orifices to be diverted simultaneously. The device consisted of six identical units, all connected together and activated from the same point at one time. Each unit was made up of three pieces of metal bars as illustrated in Figure 13. One

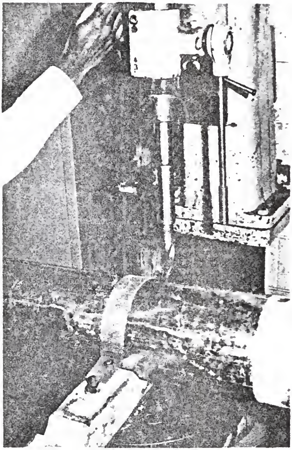

Figure 11. Arrangement for Drilling the Orifices.

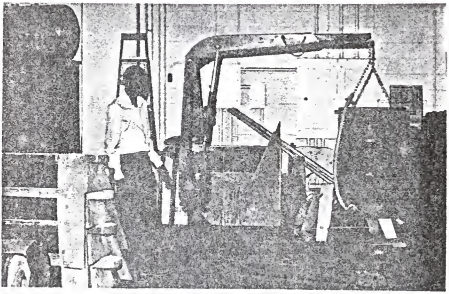

Figure 12. Manually Operated Crane for Lifting a Can-full of Water.

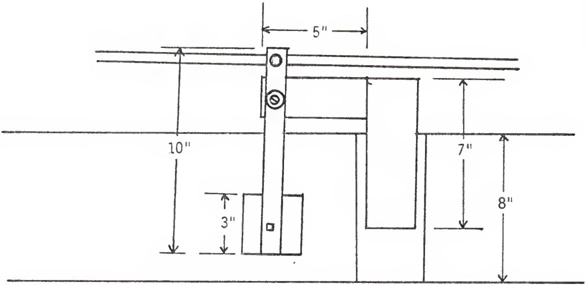

Figure 13. Components of the Flow Diverting Device.

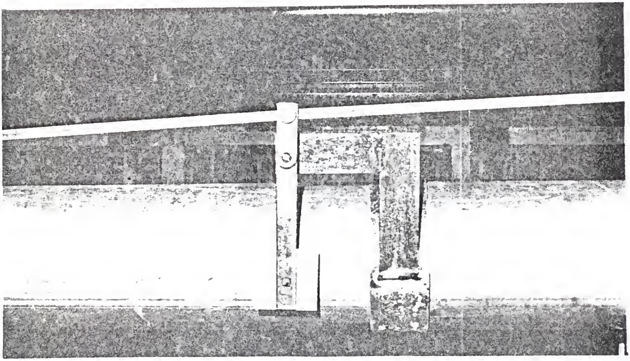

Figure 14. The Flow Diverting Device in Place.

about 10 inches in length while the other two were 7 inches and 5 inches. In the interest of clarity, the three pieces will be referred to as A, B, and C. The pieces B and C were welded together forming a right angle at their joint. On the piece C, a hole was drilled about 4.5 inches from the joint. Another hole was drilled on piece A, 1/2 inch from one end. Through these holes the two pieces, A and C, were connected by means of a bolt which acted as a pivot point around which piece A could swing freely. Two more holes were drilled in piece A, one at each end. A metal plate was bolted onto it at the end furthest from the pivot point while two handles were connected to the other end such that a pull on one of them could cause the piece to swing one way or the other. Thus, when the device was put in place around an orifice, the plate could be made to either cut across the flow or be swung away from it. The device was welded to a pipe clamp and then mounted on the pipe around an orifice. Figure 14 shows the flow diverting device in place. All the handles were connected together such that any action at one end could cause a simultaneous identical response in all units.

This was attained by turning the pipe so that the discharge exited horizontally. When the barrels were ready to be put in place, the diverter was activated so that the plates at each unit cut across the flow and remained there, thus obstructing flow. The barrels were then placed in position and the diverter was again activated, this time to remove the plate obstruction. At the same time, a stop watch was started. When the water in the barrel collected up to a selected level, the diverter was activated for a third time to cut the flow into the barrels. Weighing then followed as mentioned earlier. The process was repeated for four different flow-rates. Thus four runs were made.

The test pipe was then removed for a second time. The holes were enlarged to 1 1/4 inches in diameter by using a drill bit of proper size. The procedure used for the drilling was very much the same as that described earlier in the case of the smaller holes. Additional care was, however, taken to make sure that the bit was well centered.

The pipe was then put back in place. Water was run through as before only this time, the pipe was oriented such that the orifices discharged directly downwards to facilitate collection of the flow. A different diverter had to be used because the first diverter could not control flow through the enlarged orifices. The second diverter consisted of a single unit and was made up of two pieces of rigid plastic pipes connected together with a 75^{0} elbow. A chain tied around the shorter of the two pieces was made to go around the test pipe at the required position and the loose end was hung on to a hook fastened to the part of the chain that was tied to the plastic pipe. The arrangement, which is shown in Figures 15 and 16, was used to divert water into a barrel for a measured time interval. This interval was recorded with an electronic stopwatch. So, at each orifice the diverter was swung into place and, simultaneously, the start button of the stop watch was pressed. The water flowed through the plastic tubing, into the elbow and through the second tubing into the barrel which was sitting on the scale. As in the case of the first set of orifices, in this also, four runs were made by varying the flow rate into the system.

The diverter was swung away after some recorded time interval and readings were taken on the scale. The same procedure was followed for determining the discharge out of the rest of the orifices taking one outlet at a time. Four runs were made with this arrangement.

Figure 15. The Plastic Flow Diverting Device.

Figure 16. Operation of the Plastic Flow Diverter.

The scale used was a regular platform scale that read to one-tenth of a pound. However, an estimate to the nearest hundredth was acceptably possible. Thus weight, instead of volume, was measured for each sampled discharge. Both volume and the average velocity at various sections could be determined.

Results

The test for the determination of the friction coefficient for the eight-inch diameter test pipe showed that, when the average velocity inside the pipe was 4.255 feet per second, the total head loss between the two tappings, 27 1/2 feet apart, was 2.8 inches. The friction coefficient, C_H, was determined using the Hazen-Williams' equation of velocity. The equation is:

$$V = 1.318 C_H R^{0.63} S^{0.54} \tag{63}$$

where

V = velocity in the pipe, cfs

C_H = pipe friction coefficient

R = the pipe hydraulic radius, ft

S = the hydraulic slope

The head loss due to fricition can be expressed as:

$$h_f = SL \tag{64}$$

where

h_f = the head loss due to friction, ft

L = length of segment of the pipe under consideration; ft

After solving Equation 64 for S, substituting in Equation 63 and rearranging, the following expression can be established:

$$C_H = \left[\frac{V}{1.318 R^{0.63}}\right] \left[\frac{L}{h_f}\right]^{\frac{1}{1.85}} \tag{65}$$

Using Equation 65 the friction coefficient was calculated to be 131.48.

Following the determination of the friction coefficient, two sets of tests were carried out using the same test pipe. These tests were conducted after drilling holes in the pipe. The aim as stated in the objectives was to determine the flow rate from the orifices and the static head at each. The approach velocity toward the orifices and the discharge coefficient at each were then calculated. Holes of 13/16-inch diameter were used in the first set of tests while 1.25-inch diameter holes were used in the second set. The results are shown in Tables 1 and 2, respectively.

Suggestions, from earlier analyses and experiments, indicated that the coefficient of discharge is affected by both the velocity approach and the head at an orifice.

Regression analyses using a computer were, therefore, carried out to determine the unknown effects and the full dimension of the nature of any relationship that may exist between them.

The computer program used, known as the "Statistical Analysis System," or simply SAS performs various statistical analyses depending upon the procedure required and the model indicated.

Statistical theory suggested that the relationship between any two or more quantities could be expressed as a polynomial equation. So if a set of experimental values of two or more quantities were available, in order to determine if some relationship existed between them, attempts could be made to express one of the quantities as a polynomial function of the rest. However, a polynomial could be of the first, second, third, or any other order. But in choosing an order for a model, it is always logical to choose the first order model as the starting one.

For the analysis of the data obtained in this study, the regression procedure of the SAS program was used. The first order model was tried first, then the second, then the third and so on. For every order of model considered an F-test was conducted to determine if the order of the model in question had shown a significant reduction of the residual sum of the squares, at 5% level, as compared to the model order immediately preceding it. For instance, an F-test was made to find out if the

Table 1. Pressure Head, Approach Velocity and Discharge Coefficient for a 13/16-inch Diameter Orifice.

Orifice Number	Initial* Weight (lbs)	Final** Weight (lbs)	Net*** Weight (lbs)	Time Interval (secs)	Pressure Head Δ(ft)	Orifice+ Discharge (gpm)	Approach+ Velocity (fps)	Discharge+ Coefficient
Run Number 1, Total Inflow = 50.37 gpm								
1	41.50	243.25	201.75	179.9	1.2683	8.07	0.051479	0.5522
2	41.00	245.25	204.25	179.9	1.2667	8.17	0.103597	0.5594
3	40.75	252.00	211.25	179.9	1.25	8.45	0.157500	0.5824
4	42.00	259.00	217.00	179.9	1.2583	8.68	0.212871	0.5963
5	42.50	253.25	211.75	179.9	1.2583	8.47	0.266902	0.5819
6	43.00	256.25	213.25	179.9	1.2883	8.53	0.321316	0.5860
Run Number 2, Total Inflow = 72.77 gpm								
1	41.00	238.75	197.75	120.0	1.8500	11.85	0.075646	0.6719
2	41.25	239.00	197.75	120.0	1.8500	11.85	0.151292	0.6719
3	40.50	240.25	199.75	120.0	1.8400	11.97	0.227704	0.6805
4	41.75	252.75	211.00	120.0	1.8300	12.65	0.308418	0.7208
5	41.25	244.00	202.75	120.0	1.7800	12.15	0.385977	0.7823
6	42.75	248.00	205.25	120.0	1.8400	12.30	0.464492	0.6943

Table 1. Continued

Orifice Number	Initial* Weight (lbs)	Final** Weight (lbs)	Net*** Weight (lbs)	Time Interval (sec)	Pressure Head Δ(ft)	Orifice+ Discharge (gpm)	Approach+ Velocity (fps)	Discharge+ Coefficient
Run Number 3, Total Inflow = 86.62 gpm								
1	43.00	279.00	235.00	120.0	3.2100	14.15	0.090278	0.6087
2	42.00	284.75	242.75	120.0	3.0050	14.55	0.183138	0.6471
3	41.75	282.75	241.00	120.0	2.8780	14.45	0.275329	0.6565
4	42.00	290.25	248.25	120.0	2.6167	14.88	0.370293	0.7092
5	41.75	277.00	235.25	120.0	2.5033	14.10	0.460284	0.6871
6	43.25	285.00	241.75	120.0	2.5033	14.49	0.552762	0.7061
Run Number 4, Total Inflow = 128.26 gpm								
1	41.00	394.00	353.00	120.0	6.4983	21.16	0.135035	0.6399
2	41.00	396.75	355.75	120.0	6.6317	21.32	0.271121	0.6384
3	40.00	396.75	356.75	120.0	6.7317	21.38	0.407590	0.6354
4	41.75	405.00	363.25	120.0	6.2433	21.77	0.546546	0.6718
5	41.50	392.50	351.00	120.0	6.2433	21.04	0.680815	0.6492
6	43.00	403.25	360.25	120.0	6.2433	21.59	0.818623	0.6663

* The initial weight was one of the barrel before sampling
** The final weight was of barrel and the water it contained after sampling
*** The net weight was then the weight of the water sampled
\+ These values were computed -- See sample calculations in the appendix

Note: In these runs samplings were made simultaneously. The orifice number 1 was always the one at the dead end.

Table 2. Pressure Head, Approach Velocity and Discharge Coefficient for a 1.25-inch Diameter Orifice.

Orifice Number	Initial* Weight (lbs)	Final** Weight (lbs)	Net*** Weight (lbs)	Time Interval (sec)	Pressure Head Δ(ft)	Orifice Discharge (gpm)[+]	Approach Velocity (fps)[+]	Discharge Coefficient[+]
				Run Number 5, Total Inflow = 169.11 gpm				
1	50.25	282.00	231.75	60.0	1.7083	27.78	0.177305	0.6924
2	53.25	292.75	239.50	60.0	1.7083	28.71	0.360538	0.7156
3	52.50	286.25	233.75	60.0	1.7083	28.02	0.539373	0.6984
4	53.25	291.25	238.00	60.0	1.7000	28.53	0.721459	0.7128
5	54.75	288.50	233.75	60.0	1.7000	28.02	0.900294	0.7001
5	57.25	291.25	214.00	60.0	1.6917	28.05	1.079320	0.7025
				Run Number 6, Total Inflow = 222.18 gpm				
1	41.25	348.25	307.00	60.0	3.2833	36.80	0.234876	0.6616
2	41.00	355.75	314.75	60.0	3.2750	37.73	0.475681	0.6792
3	41.00	350.75	309.75	60.0	3.2833	37.13	0.712661	0.6675
4	41.25	351.50	310.25	60.0	3.2500	37.19	0.950024	0.6720
5	51.50	358.25	306.75	60.0	3.2333	36.77	1.184709	0.6662
6	40.50	345.50	305.00	60.0	3.2083	36.56	1.418054	0.6649

Table 2. Continued

Orifice Number	Initial* Weight (lbs)	Final** Weight (lbs)	Net*** Weight (lbs)	Time Interval (secs)	Pressure Head Δ(ft)	Orifice+ Discharge (gpm)	Approach+ Velocity (fps)	Discharge+ Coefficient
Run Number 7, Total Inflow = 270.24 gpm								
1	41.50	394.25	352.75	59.7	4.8125	42.50	0.271234	0.6311
2	42.00	395.25	353.25	59.8	4.7917	42.49	0.542399	0.6323
3	43.00	407.25	364.25	59.8	4.7875	43.81	0.822007	0.6522
4	41.25	418.75	377.50	62.3	4.7583	43.58	1.100157	0.6508
5	41.25	414.75	373.50	60.3	4.7583	44.55	1.384489	0.6653
6	40.75	402.00	361.25	60.0	4.6333	43.31	1.660870	0.6554
Run Number 8, Total Inflow = 296.28 gpm								
1	41.50	449.25	407.25	60.0	6.0708	48.88	0.311957	0.6462
2	41.50	460.75	419.25	60.0	6.0708	50.26	0.632712	0.6645
3	41.50	456.50	415.00	60.5	6.0708	49.34	0.947591	0.6523
4	41.50	456.50	415.00	60.0	6.0583	49.75	1.265095	0.6584
5	42.50	451.25	408.75	59.8	6.0458	49.16	1.578862	0.6513
6	40.75	449.25	408.75	60.1	6.0208	48.89	1.890873	0.6490

* The initial weight was one of the barrel before sampling
** The final weight was of barrel and the water it contained after sampling
*** The net weight was then the weight of the water sampled
\+ These values were computed -- See sample calculations in the appendix

Note: In these runs samplings were made independently. The orifice number 1 was always the one at the dead end.

second order model had shown a significant reduction of the residual sum of the squares over the first model. When two consecutive models show insignificant reductions of sum of squares over their preceding ones, then the model immediately preceding the two was taken as the model of best fit. For example, if the second order model showed no significant reduction of the residual sum of the squares over the first, and nor did the third over the second, then according to this methodology, the first order model was taken as the model with the best fit.

Using the above methodology in the analysis in this study, a fourth order model was obtained having 15 terms. Since, some of the terms were of no significance in the relationship between the quantities, another procedure in the SAS program was used to determine the significant terms in the relationship. The procedure, known as the "stepwise procedure," took only one term initially and added the other terms in the model one at a time. An F-test was made in each case to determine the significance of the term to the model. Thus, all insignificant terms were rejected.

Results showed that the discharge coefficient at any orifice could be expressed as a third order model and a function of both the velocity of approach and the pressure head at the orifice. The prediction equation obtained was:

$$C_d = 0.5836 + 0.3723V - 0.01098hV - 0.346V^2 + 0.1084V^3 \tag{66}$$

where

C_d = discharge coefficient of an orifice

V = approach velocity toward the orifice

h = pressure head at the orifice

The appearance of the pressure head in only one term and the predominance of the approach velocity term in the equation raised some questions about the significance of the pressure head effect upon the coefficient.

A similar analysis was carried out, taking the approach velocity as the only factor of concern. Again another prediction equation of the same third order was obtained. It was established as:

$$C_d = 0.5883 + 0.3106V - 0.3141V^2 + 0.0898V^3 \qquad (67)$$

A comparison of discharge coefficients predicted by each of these two equations with the observed values showed a good agreement. The comparisons are given in Tables 3 and 4. The differences between the observed and the predicted values were found to be statistically insignificant, after performing an F-test. It can be recalled that part of the objective of this study was to determine if the Enger and Levy equation, given earlier as Equation 3, can be applied to the case of eight-inch irrigation pipes. It was found that the accuracy of its prediction depended very much on how good an assumption was made of the discharge coefficient for the dead end. Values predicted by this equation can also be seen in the comparison Tables 3 and 4.

Table 3. Comparison Between Observed and Predicted Discharge and Discharge Coefficients for 13/16-inch Orifices.

Orifice Number	Coefficient of Discharge				Orifice Discharge			
	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***
				Run Number 1				
1	0.5860	0.6667	0.6587	0.5515	8.53	9.6983	9.5818	8.0229
2	0.5819	0.6567	0.6505	0.5517	8.47	9.5533	9.4637	8.0261
3	0.5963	0.6453	0.6411	0.5519	8.68	9.3872	9.3257	8.0287
4	0.5824	0.6319	0.6298	0.5520	8.45	9.1625	9.1315	8.0042
5	0.5594	0.6171	0.6172	0.5521	8.17	9.0077	9.0088	8.0589
6	0.5522	0.6011	0.6035	0.5522	8.07	8.7799	8.8138	8.0648
				Run Number 2				
1	0.6993	0.6834	0.6738	0.6707	12.30	12.0214	11.8533	11.7983
2	0.7023	0.6744	0.6666	0.6710	12.15	11.6695	11.5330	11.6104
3	0.7208	0.6625	0.6569	0.6714	12.65	11.6227	11.5237	11.7782
4	0.6805	0.6471	0.6438	0.6716	11.97	11.3838	11.3255	11.8147
5	0.6719	0.6293	0.6284	0.6718	11.85	11.1006	11.0848	11.8497
6	0.6719	0.6083	0.6100	0.6719	11.85	10.7299	10.7607	11.8514

Table 3. Continued

Orifice Number	Coefficient of Discharge				Orifice Discharge			
	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***
Run Number 3								
1	0.7061	0.6868	0.6792	0.6075	14.49	14.0922	13.9361	12.4662
2	0.6871	0.6796	0.6735	0.6079	14.10	13.9442	13.8190	12.4735
3	0.7092	0.6689	0.6648	0.6082	14.88	14.0322	13.9466	12.7593
4	0.6565	0.6534	0.6519	0.6085	14.45	14.3763	14.3421	13.3866
5	0.6471	0.6348	0.6352	0.6086	14.55	14.2711	14.2801	13.6820
6	0.6087	0.6113	0.6138	0.6087	14.15	14.2036	14.2630	14.1429
Run Number 4								
1	0.6663	0.6599	0.6813	0.6388	21.59	21.3822	22.0784	20.7011
2	0.6492	0.6642	0.6825	0.6392	21.04	21.5241	22.1165	20.7118
3	0.6718	0.6640	0.6789	0.6394	21.77	21.5152	21.9992	20.7203
4	0.6354	0.6551	0.6688	0.6397	21.38	22.0421	22.5037	21.5231
5	0.6384	0.6415	0.6512	0.6398	21.32	21.4251	21.7486	21.3672
6	0.6399	0.6182	0.6247	0.6399	21.16	20.4374	20.6536	21.1540

* The predicted value I is the value obtained using Equation (66)
** The predicted value II is the value obtained using Equation (67)
*** The predicted value III is the value obtained using Equation (3)
The observed value for the coefficient was actually calculated from the observed discharge out of and the pressure head at the orifices

Note: The Orifice number 1 was always the one at the dead end.

Table 4. Comparison Between Observed and Predicted Discharge and Discharge Coefficients for 1.25-inch Orifices.

Orifice Number	Coefficient of Discharge				Orifice Discharge			
	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***
				Run Number 5				
1	0.7025	0.6986	0.6705	0.6850	28.05	27.8913	26.7706	27.3477
2	0.7001	0.7006	0.6789	0.6873	28.02	28.0405	27.1696	27.5059
3	0.7128	0.6993	0.6826	0.6891	28.53	27.9890	27.3195	27.5793
4	0.6984	0.6906	0.6785	0.6906	28.02	27.7081	27.2226	27.7051
5	0.7156	0.6712	0.6637	0.6916	28.71	26.9269	26.6257	27.7458
6	0.6924	0.6360	0.6340	0.6922	27.78	25.5163	25.4355	27.7707
				Run Number 6				
1	0.6649	0.6749	0.6532	0.6552	36.56	37.1079	35.9132	36.0211
2	0.6662	0.6772	0.6647	0.6571	36.77	37.3792	36.6898	36.2704
3	0.6720	0.6841	0.6769	0.6587	37.19	37.8535	37.4566	36.4529
4	0.6675	0.6867	0.6826	0.6600	37.13	38.1960	37.9674	36.7094
5	0.6792	0.6770	0.6746	0.6609	37.73	37.6049	37.4756	36.7119
6	0.6616	0.6449	0.6451	0.6614	36.80	35.8686	35.8795	36.7882

Table 4. Continued

Orifice Number	Coefficient of Discharge				Orifice Discharge			
	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***	Observed Value	Predicted Value I*	Predicted Value II**	Predicted Value III***
				Run Number 7				
1	0.6554	0.6596	0.6491	0.6253	43.31	43.5838	42.8900	41.3124
2	0.6653	0.6512	0.6546	0.6272	44.55	43.6002	43.8277	41.9923
3	0.6508	0.6613	0.6694	0.6286	43.58	44.2769	44.8221	42.0897
4	0.6522	0.6728	0.6813	0.6297	43.81	45.1895	45.7548	42.2932
5	0.6323	0.6725	0.6787	0.6305	42.49	45.1867	45.6025	42.3643
6	0.6311	0.6470	0.6512	0.6310	42.50	43.5644	43.8521	42.4866
				Run Number 8				
1	0.6490	0.6583	0.6597	0.6402	48.89	49.5842	49.6855	48.2216
2	0.6513	0.6307	0.6491	0.6421	49.16	47.6038	48.9932	48.4591
3	0.6584	0.6362	0.6604	0.6435	49.75	48.0633	49.8911	48.6215
4	0.6523	0.6548	0.6770	0.6447	49.34	49.5206	51.2007	48.7598
5	0.6645	0.6659	0.6818	0.6455	50.26	50.3641	51.5663	48.8220
6	0.6462	0.6486	0.6574	0.6460	48.88	49.0511	49.7156	48.8599

* The predicted value I is the value obtained using Equation (66)
** The predicted value II is the value obtained using Equation (67)
*** The predicted value III is the value obtained using Equation (3)
The observed value for the coefficient was actually calculated from the observed discharge out of and the pressure head at the orifices

Note: The Orifice number 1 was always the one at the dead end.

Discussion

The results of the theoretical discussion and analysis in this study suggested that the discharge coefficient for any orifice is a function of the angle at which the discharge exits from the orifice and was given in Equation 62. It was further suggested that the angle of discharge itself is a function of both the velocity of approach towards and the pressure at the orifice. The obvious inference was therefore that the discharge coefficient is a function of the approach velocity and the static head at the orifice. The analysis, however, failed to establish any relationship similar to those obtained as a result of this experiment (Equations 66 and 67). These equations, though subject to some limitations, had nonetheless strengthened the claim that was based on the theoretical analysis. The limitation alluded to here is one of scope of validity.

It must be conceded that this study was not designed to include many other factors that are likely to play a role in the influence of the discharge coefficient. These factors were dropped from consideration on the grounds that some earlier studies had found their effect insignificant as compared to the effects of approach velocity and static head. Nevertheless, since a generalization cannot be made without some risk of oversimplification of the problem, the decision to consider those factors ineffective is, to some extent, tantamount to an assumption. This exposes one of the sources of limitations of the obtained empirical relations.

Moreover, the prediction Equations 66 and 67 may be valid only within the range of the data collected. Even though this can best be ascertained by a comparison between values predicted by the equation, and those calculated from some observed data of flow well beyond the range recorded in this experiment.

This notwithstanding, the empirical formula can be used to determine the rate of flow out of the various orifices of a pipe when the total discharge and static head at the point of entry are known. The accuracy of the values will of course depend on how closely the conditions of the system in question resemble those under which this experiment was conducted. Enger and Levy's Equation showed good agreements with the observed data in some of the runs. However, since the value of any coefficient can be predicted only after assuming a value for the coefficient of the orifice nearest the dead end, the observed value for it was used. Hence the value of the observed and that obtained by the prediction equation are always the same.

In order to find out how good a prediction for orifice discharge can be made employing those empirical relations, the data of a different experiment was used for comparison. The experiment was performed by Spomer (1969) and the data gave the total discharge into a manifold pipe along with the static head at the entry point. Also included were the observed discharge out of the various orifices.

The following methodology was used for calculating the orifice discharges using the empirical equations 66 and 67.

First, the inflow and the head at entrance to the gated pipe was known. The velocity of approach towards the first orifice from the inlet end was then determined using the continuity equation:

$$Q = AV \tag{68}$$

where

Q = the total flow at entry into the pipe

A = the cross-sectional area of the pipe

V = the velocity inside the pipe

Then, using the empirical equation 66, the discharge coefficient for the first orifice was obtained under the assumption that the pressure head just before the orifice is the same as that at the orifice. This pressure head was determined by subtracting the head loss due to friction between the point of measurement of head at entry and the orifice. Hazen-Williams' equation of velocity was used in this connection.

Since the head at the orifice was known along with the discharge coefficient, the flow out of the first orifice was calculated using Equation 38. Bernoulli's equation was, therefore, written between the point just prior to the orifice and one immediately following it. Thus:

$$h_1 + \frac{V_1^2}{2g} = h_2 + \frac{V_2^2}{2g} \tag{69}$$

where

h_1 = the static head immediately before the orifice

V_1 = approach velocity toward the orifice

h_2 = the static head immediately after the orifice

V_2 = velocity of water leaving the first orifice

The velocity of water inside the pipe leaving the orifice, V_2, can be expressed as:

$$V_2 = V_1 - \frac{q_1}{A} \tag{70}$$

This equation is true, since the total flow inside the pipe after passing the first orifice, Q_2, is equal to $Q_1 - q_1$. Another form of this statement from which the above equation was directly derived can be expressed as, $AV_2 = AV_1 - q_1$. The energy equation, therefore, became:

$$h_2 = h_1 + \frac{V_1^2}{2g} - \frac{(V_1 - \frac{q_1}{A})^2}{2g} \tag{71}$$

From this equation the static head immediately after the orifice, h_2, was calculated and the velocity V_2 in the section downstream from the first orifice was calculated from Equation 70 . This velocity became the approach velocity toward the second orifice. The static head immediately before the second orifice, h_2', was the static head just after the first orifice less the friction loss in between. Again Hazen-Williams' equation was used for determining the head loss due to friction. Therefore:

$$h_2' = h_1 - h_{f1-2} \tag{72}$$

where

h_2' = static head just before the second orifice

h_{f1-2} = the head loss due to friction between orifice 1 and 2

Then, onward, the process was repeated until the dead end was reached.

Approximations made for the static head at an orifice, in this procedure, contain the assumption that the static head just before the orifice was the same as the head at the orifice. Since this is anything but the actual case, adjustments were made to correct for the assumption. With all heads both before and after each orifice calculated, the average of the two was taken to be the actual head at the orifices. However, this adjustment was started with the first orifice from the inlet end while the rest were left as they were. The discharge out of the first orifice was recalculated using the new value for the head. The process of recalculating other heads both upstream and downstream of the orifices followed in accordance with the methodology described earlier. The new value for the head was examined to determine the percentage difference between it and the previous value of the head. When the percentage was found to be less than 1, the adjustment process proceeded to the next orifice. The procedure was repeated until the last orifice was reached. Thus, the head at every orifice and the discharge from each were established. Table 5 compared the value calculated and those actually observed by Spomer. Even though satisfactory values of head everywhere along the pipe were obtained, the computed discharge out of the orifices were not equal. In order to attain uniform flow, an amount of discharge equal to the quotient of the total inflow into the pipe divided by the total number of the orifices had to issue from each orifice. This, then, was regarded as the required flow. A head, other than the one computed, was needed to produce this required flow out of each of the orifices. When the computed head was greater than the required one, the difference was regarded as positive and when it was the other way around, the difference was considered negative. All the differences were determined associating the appropriate signs with each.

The pipe under discussion was level and all the orifices in it were at the same elevation. When the head differences were subtracted from the pipe elevation at every orifice, a different elevation resulted in each case. However, since the pipe could not be bent all over such as to obtain the ideal elevation at each orifice, it was found most appropriate to determine an optimum pipe slope that would approximate most closely the ideal elevations. The slope of the line of least squares was calculated and was taken to be the optimum slope. A computer program was written and used for all the calculations described in this procedure.

Table 5. Comparison of Calculated and Observed Discharge Using Spomer's Data.

Calculated Discharge	Observed Discharge	Difference of the Two	Percentage* Deviation
5.09	5.00	-0.09	-0.02
5.10	4.91	-0.19	-0.04
5.13	5.02	-0.11	-0.02
5.16	5.06	-0.10	-0.02
5.18	5.04	-0.14	-0.03
5.19	5.05	-0.14	-0.03
5.20	5.01	-0.19	-0.04
5.20	5.02	-0.18	-0.04
5.20	5.06	-0.14	-0.03
5.18	5.00	-0.18	-0.04
5.15	5.04	-0.11	-0.02
5.11	5.01	-0.10	-0.02
5.06	5.01	-0.05	-0.01
5.00	5.09	0.09	0.02
4.93	5.20	0.27	0.05
4.84	5.24	0.40	0.08
4.74	4.99	0.25	0.05
4.64	5.02	0.38	0.08

Mean = -0.02 Square of Standard Deviation = 0.04 T = -0.40

A printout obtained as a result is shown in the Appendix. It should be pointed out here that the program was written such as to allow the use of it by anybody, in any design for optimum pipe slope that would provide uniform flow, so long as the total inflow and the head just before the first orifice were known.

The Enger and Levy's equation was found unworkable under the methodology just described.

CONCLUSIONS

The following were concluded as a result of this study:

1. The coefficient of discharge of an orifice in the side of a multiple outlet pipe can be expressed as a function of either the approach velocity towards the orifice or as a function of both the approach velocity and the static head at the orifice. Further, the value of the coefficient can be predicted by either of the following two equations:

$$C_d = 0.5836 + 0.3723V - 0.01098hV - 0.346V^2 + 0.1084V^3$$

and

$$C_d = 0.5883 + 0.3106V - 0.3141V^2 + 0.0898V^3$$

2. The Enger and Levy Equation can be applied to the case of a multi-outlet eight-inch irrigation pipe with the accuracy of prediction depending to a large extent on the accuracy of the assumption for the value of the discharge coefficient of the orifice at the dead end.

3. The prediction equations, obtained as a result of the investigation, predicted satisfactory values for orifice discharge in the case of different pipe and orifice sizes. The predicted values, also, compared favorably with observed values obtained from a different experiment.

4. A methodology was established for determining an optimum slope needed to provide uniform flow out of various orifices of a gated pipe. A computer program was developed for easy application. In reality exact uniformity may not be achieved but deviations therefore should be practically insignificant. Lastly, the Enger and Levy Equation cannot work with this methodology.

SUMMARY

Non-uniformity of flow through the various orifices of a gated pipe irrigation system has presented a problem that is becoming of increasing concern. At present farmers try to overcome it by adjusting the individual gates until equal flow is approximated as nearly as possible. Increasing costs of operation due to high labor inputs in many irrigation systems call for automation of the system. If, however, gates have to be adjusted manually employing both the skill and judgement of the irrigator, any automation shall be far from complete.

This study, therefore, addressed itself to the problem. A review of the literature carried out showed that other people have looked into similar problems in studies with gas and air systems. Yet others have worked with multiple outlet pipes, some with perforated pipes and some even looked directly at the problem of uniformity of flow from gated irrigation pipes. There appeared, generally, to be an agreement between the people that considered the problem that the discharge coefficient at each orifice is affected by the approach velocity and the head at each orifice. The coefficient was accepted as a factor having a direct influence on the discharge out of the orifices. Some have alluded to the possibility of the effects of such factors as the orifice to pipe diameter ratios and the curvature of the pipe just to mention a few. Others have dismissed them as insignificant compared to the effects of the two factors previously mentioned. Despite all these suggestions, none but one relationship between the discharge coefficient and the approach velocity and, or the static head was found to have been established. Enger and Levy gave that relationship expressing the discharge coefficient at an

orifice as a function of the dynamic and static head upstream of an orifice along with discharge coefficient at the dead end. This latter quantity has to be assumed every time the equation is used. The equation was obtained in a study with a 2-inch pipe having 3/8-inch orifices and was expressed as:

$$C_d = (1 - \frac{\frac{V^2}{2g}}{h})\, C_{d_o}$$

An investigation was carried out with the following objectives:

1. To determine a direct relationship between the orifice discharge coefficient and the parameters; approach velocity and static head.
2. To determine if the Enger and Levy Equation can be applied to eight-inch irrigation pipes.
3. To determine how any empirical relations obtained compare with other independently observed data.
4. To determine a design procedure for ensuring uniform flow.

Theoretical analysis made as part of the investigation led to the establishment of the relation

$$C_d = \sin\theta / \sqrt{2}$$

The angle of discharge was believed to be directly influenced by both the velocity of water approaching the orifice and the static pressure at the orifice. However, no theoretical relationship could be established. An experiment in pursuit of the above mentioned objectives was conducted.

For the purpose of the experiment, an irrigation pipe, 30-feet long with six circular holes drilled in it was used as a test pipe. It was connected to another pipe as part of a simulated pipe irrigation system. Water was run through the system with the aid of a centrifugal pump. The discharge out of each orifice, within a certain recorded time, was collected in a can and weighed. The head, again at each outlet, was read from a set of manometers connected to pressure tappings at the various orifices. Four runs were made with each of the two orifice sizes used in the experiment. The flow rate was different for every run.

The data collected in the experiment was used to determine the discharge coefficients and the approach velocities towards each orifice. The data, also, helped confirm the theoretical suggestion that the discharge coefficient of an orifice is influenced by both the head at and the approach velocity towards it. It further revealed the fact that the relation between the coefficient and these influencing factors is not a linear one. Statistical analysis showed that a cubic model is the best fit for the data in establishing a relation between the coefficient and the two quantities, pressure head and approach velocity. When it was carried further, the analysis precipitated two prediction equations. One gave the discharge coefficient as a function of both the pressure head and the approach velocity towards the orifice. The other expressed the coefficient as a function of only the velocity of approach. The two equations were expressed as:

$$C_d = 0.5836 + 0.3723V - 0.01098hV - 0.346V^2 + 0.1084V^3$$

and

$$C_d = 0.5883 + 0.3106V - 0.3141V^2 + 0.0898V^3$$

Both were found to predict equally well within the range of values of the static heads and the pipe velocities observed in the experiment.

The heads at each outlet were established by series of calculations. Hence, the difference between the required and the calculated heads were computed. The orifices were first taken to be at the same elevation with respect to an arbitrary datum. The differences in head then was translated into an elevation difference. The slope of the line of least square through the points corresponding to the elevation at the various orifices was taken to be the optimum slope. It was found that Enger and Levy's Equation could not be used with this methodology. A computer program was written for easy application of it.

It was, therefore, concluded that:

1. The coefficient of discharge could be expressed as a function of either the approach velocity only or both the approach velocity and the static head.

2. The Enger and Levy Equation can be applicable to the case of eight-inch pipe if good assumptions can be made for the coefficient at the dead end.

3. The prediction equations obtained as a result of this study predicted satisfactory values of orifice discharge for different conditions of flow and different pipe and orifice sizes.

4. A methodology has been established for determining the optimum slope needed to provide a uniform flow, for all practical purposes, out of various orifices of a gated pipe.

SUGGESTIONS FOR FUTURE RESEARCH

This study did not include a wide range of static pressures at the different orifices. Therefore, the full extent of the interplay between a static regain due to momentum decrease and the possible loss of pressure head due to branching of the flow could not be determined. Hence, a further investigation to determine this shall help eliminate some of the assumptions being made in calculating the static head at various points on the pipe, thus making the process of determining the optimum slope more accurate.

The pipe used in this experiment had plane circular orifices cut with a drill press while the conventional gated irrigation pipes have adjustable gates, of varying shape, with some rubber fittings on the inside. How much this difference in the nature and shape of the orifices can affect the applicability of the results of this study calls for a further investigation.

The ratio of orifice to pipe diameters has been suggested as a possible contributor to the variation in the discharge coefficient of orifices. Experimental study can provide evidence for either the justification or the dismissal of any assumptions that might have been made with respect to this ratio. Thus, it may contribute to a better understanding of the problem.

Theoretical analysis made, as shown earlier herein, has indicated a direct relation between the discharge coefficient and the angle of exit of the flow out of the orifice. A further research to determine the factors affecting it are essential.

Finally, the range of the velocities observed in this experiment, due to the constraints of available facilities, was relatively narrow as compared to those obtained in the field. So, a research project, perhaps in the field, that can include the range of flows for irrigation systems shall help determine whether or not the prediction equation obtained in this study has to be confined to the range of values of velocity observed.

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APPENDIX A

Computer Program Developed

for

The Determination of the Optimum Slope

that Would Provide Uniform Flow

Out of the Orifices of a Multi-Outlet

Irrigation Pipe.

```
$JOB
C     THIS COMPUTER PROGRAMME WAS WRITTEN BY SALIHU S. ABUBAKAR
C     TO DEMONSTRATE HOW FOR A MANIFOLD IRRIGATION PIPE,AN
C     OPTIMUM SLOPE THAT WOULD MOST CLOSELY APPROXIMATE EQUAL
C     DISCHARGE THROUGH ITS VARIOUS ORIFICES,COULD BE DETERMINED.
C
C
C     DEFINITION OF VARIABLES USED IN THIS PROGRAMME:
C
C     TOTFLO=THE RATE OF FLOW OF WATER INSIDE A PIPE WITHIN A SEGMENT.
C
C     TOTALQ=THE TOTAL FLOW ENTERING THE PIPE.
C
C     INIHED=THE INITIAL STATIC HEAD JUST BEFORE THE FIRST ORIFICE.
C
C     TOTHED=THE TOTAL HEAD BOTH DYNAMIC AND STATIC AT ANY POINT.
C
C     PHEDBE(I)=THE STATIC HEAD JUST BEFORE AN ORIFICE (I).
C
C     PHEDAF(I)=THE STATIC HEAD JUST AFTER AN ORIFICE (I)
C
C     PHEAD=THE STATIC HEAD AT AN ORIFICE.
C
C     CH=HAZEN-WILLIAMS' FRICTION FACTOR.
C
C     DPIPE=THE DIAMETER OF THE PIPE.
C
C     ORFD=THE DIAMETER OF THE ORIFICE.
C
C     A=THE CROSS-SECTIONAL AREA OF THE PIPE.
C
C     P=THE PERIMETER OF THE PIPE.
C
C     R=THE HYDRAULIC RADIUS OF THE PIPE.
C
C     AO=THE AREA OF THE ORIFICE.
C
C     DCOEFF=THE DISCHARGE COEFFICIENT OF THE ORIFICE.
C
C     Q(I)=THE DISCHARGE OUT OF AN ORIFICE.
C
C     V=THE VELOCITY OF APPROACH TOWARDS AN ORIFICE.
C
C     VHEAD=VELOCITY HEAD WITHIN A SECTION.
C
C     L=LENGTH OF SPACING BETWEEN ORIFICES.
C
C     FRCLOS=THE FRICTION LOSS BETWEEN TWO CONSECUTIVE ORIFICES.
C
C     AVEHED=AVERAGE OF HEADS BEFORE AND AFTER AN ORIFICE
C
C     PERC=PERCENTAGE DEVIATION FROM THE AVERAGE HEAD
C
C     SUMSQ=SUM OF SQUARES
C
C     ELDIFF=DIFFERENCE BETWEEN REQUIRED AND ACTUAL HEAD AT AN ORIFICE
C
C     B=THE SLOPE OF THE LINE OF LEAST SQUARES
C
C
```

```
C
      DIMENSION PHEDAF(18),PHEDBE(18),AVEHED(18),Q(18),PHEAD(18),H(18)
      REAL INIHED,MEAN,L,NEWHED,NEWQ,NTACEP
      READ,TOTFLO,INIHED,CH,DPIPE,DRFD,N,L
      TOTALQ=TOTFLO
      A=(DPIPE**2)*3.14159/4
      AD=(DRFD**2)*3.14159/4
      P=DPIPE*3.14159
      R=A/P
      V=TOTFLO/(448.83*A)
      VHEAD=V*V/64.4
      TOTHED=VHEAD+INIHED
      PHEAD(1)=INIHED
      PHEDBE(1)=INIHED
      DO 22 I=1,N
      CALL SALIFU(PHEAD,V,Q,TOTFLO,A,VHEAD,CH,R,PHEDBE,PHEDAF,FRCLOS,AD,
     XI,TOTHED,N,L,DCOEFF)
   22 CONTINUE
   25 TOTFLO=TOTALQ
      DO 44 K=1,N
      I=K
      AVEHED(I)=(PHEDBE(I)+PHEDAF(I))/2
      PERC=(AVEHED(I)-PHEDBE(I))/AVEHED(I)
      IF(PERC.LT.(0.01))GO TO 33
      PHEAD(I)=AVEHED(I)
      DO 30 I=K,N
      CALL SALIFU(PHEAD,V,Q,TOTFLO,A,VHEAD,CH,R,PHEDBE,PHEDAF,FRCLOS,AD,
     XI,TOTHED,N,L,DCOEFF)
   30 CONTINUE
      GO TO 25
   33 TOTFLO=TOTFLO-Q(I)
      V=TOTFLO/(448.83*A)
   44 CONTINUE
      XYSUM=0
      YSUM=0
      XSUM=0
      XSSQ=0
      EQUALQ=TOTALQ/(N*448.83)
      ELEV=10.0
      DO 70 I=1,N
      TOTFLO=TOTALQ-(I-1)*EQUALQ*448.83
      V=TOTFLO/(448.83*A)
      DCOEFF=0.5883+0.3106*V-0.3141*V*V+0.0898*V*V*V
      H(I)=((EQUALQ/(DCOEFF*AD))**2)/64.4
      ELDIFF=PHEAD(I)-H(I)
      RECELV=ELEV-ELDIFF
      XSUM=XSUM+(I-1)*L
      XSSQ=XSSQ+((I-1)*L)**2
      YSUM=YSUM+RECELV
      XYPROD=(I-1)*L*RECELV
      XYSUM=XYSUM+XYPROD
   70 CONTINUE
      SXY=XYSUM-(YSUM*XSUM)/N
      SXX=XSSQ-((XSUM)**2)/N
      B=SXY/SXX
      PRINT 90
      NTACEP=YSUM/N-B*XSUM/N
      TOTFLO=TOTALQ
      DO 80 I=1,N
      V=TOTFLO/(448.83*A)
```

```
      DCOEFF=0.5883+0.3106*V-0.3141*V*V+0.0858*V*V*V
      NEWHED=PHEAD(I)-(IO-(NTACEF+8*(I-1))*L))
      NEWQ=DCOEFF*AO*(SQRT(64.4*NEWHED))*448.83
      TOTFLO=TOTFLO-NEWQ
   80 PRINT 100,I,PHEAD(I),H(I),NEWHED,Q(I),EQUALQ*448.83,NEWQ
      PRINT 200,B
   90 FORMAT('1',T32,'OFF     CALCULATED   REQUIRED   ADJUSTED   CALCULATED
     X REQUIRED   ADJUSTED'/
     X' ',T32,'NO.          HEAD         HEAD        HEAD     DISCHARGE    DISCHA
     XRGE  DISCHARGE'/
     X' ',T42,'(FT)         (FT)        (FT)        (GPM)         (GPM)       (GP
     XM)'/)
  100 FORMAT('0',T32,I2,6X,F7.4,4X,F7.4,3X,F7.4,5X,F5.2,7X,F5.2,6X,F5.2)
  200 FORMAT('0',T36,'THE OPTIMUM SLOPE FOR THE PIPE IS',2X,F8.5/'1')
      STOP
      END

      SUBROUTINE SALTHU(PHEAD,V,Q,TOTFLO,A,VHEAD,CH,R,PHEDBE,PHEDAF,
     XFRCLOS,AO,I,TOTHED,N,L,DCOEFF)
      DIMENSION PHEDAF(18),PHEDBE(18),AVEHED(18),Q(18),PHEAD(18)
      REAL INIHED,L
      DCOEFF=0.5883+0.3106*V-0.3141*V*V+0.0858*V*V*V
      Q(I)=DCOEFF*AO*(SQRT(64.4*PHEAD(I)))*448.83
      TOTFLO=TOTFLO-Q(I)
      V=TOTFLO/(448.83*A)
      VHEAD=V*V/64.4
      PHEDAF(I)=TOTHED-VHEAD
      IF(I.EQ.N)GO TO 11
      FRCLOS=((V/(1.318*CH*(R**0.63)))**1.85)*L
      PHEDBE(I+1)=PHEDAF(I)-FRCLOS
      PHEAD(I+1)=PHEDBE(I+1)
   11 RETURN
      END

$ENTRY
```

Table 6. Comparison of Calculated, Required and Adjusted Heads and Discharges.

ORF NO.	CALCULATED HEAD (FT)	REQUIRED HEAD (FT)	ADJUSTED HEAD (FT)	CALCULATED DISCHARGE (GPM)	REQUIRED DISCHARGE (GPM)	ADJUSTED DISCHARGE (GPM)
1	0.4680	0.4596	0.4273	5.09	5.04	4.86
2	0.4672	0.4559	0.4308	5.10	5.04	4.90
3	0.4691	0.4526	0.4372	5.13	5.04	4.95
4	0.4710	0.4500	0.4434	5.16	5.04	5.00
5	0.4728	0.4481	0.4496	5.18	5.04	5.05
6	0.4744	0.4471	0.4556	5.19	5.04	5.09
7	0.4759	0.4470	0.4615	5.20	5.04	5.12
8	0.4773	0.4481	0.4673	5.20	5.04	5.15
9	0.4786	0.4504	0.4729	5.20	5.04	5.17
10	0.4798	0.4541	0.4784	5.19	5.04	5.18
11	0.4808	0.4594	0.4838	5.15	5.04	5.17
12	0.4817	0.4666	0.4891	5.11	5.04	5.16
13	0.4825	0.4758	0.4942	5.06	5.04	5.14
14	0.4831	0.4876	0.4993	5.00	5.04	5.10
15	0.4836	0.5021	0.5041	4.93	5.04	5.05
16	0.4840	0.5200	0.5089	4.84	5.04	4.98
17	0.4843	0.5420	0.5136	4.74	5.04	4.90
18	0.4844	0.5688	0.5181	4.64	5.04	4.81

THE OPTIMUM SLOPE FOR THE PIPE IS 0.00131

APPENDIX B

Sample Calculations for Tables 1 and 2

Sample Calculations for Tables 1 and 2

In Table 1 run number 1, the weight of the sample discharge out of the orifice no. 1 at the dead end was measured as 201.75 pounds and was collected in 179.9 secs. Therefore:

$$q = \frac{201.75 \text{ lbs}}{179.9 \text{ secs}} \times \frac{1}{62.4} \text{ lbs/ft}^3 \times 448.83 \text{ gpm/cfs}$$

or

$$q = 8.07 \text{ gpm}$$

where:

q = the discharge out of the orifice

The approach velocity towards the orifice no. 1 is:

$$V = \frac{Q}{A}$$

where:

V = approach velocity toward the orifice

Q = the total discharge inside the pipe approaching the orifice

A = the cross-sectional area of the pipe

Therefore:

$$V = \frac{q}{A} = \frac{8.07 \text{ gpm}/(448.83 \text{ gpm/cfs})}{\pi \, (8/12 \text{ ft})^2/4}$$

or

$$V = 0.051479 \text{ cfs}$$

The discharge coefficient is given as

$$C_d = q/(a\sqrt{2gh})$$

where:

C_d = coefficient of discharge

a = cross-sectional area of the orifice

g = acceleration due to gravity

h = the head at the orifice

Therefore:

$$C_d = \frac{8.07 \text{ gpm}/(448.83 \text{ gpm/cfs})}{\pi(13/(16 \times 12))^2/4 \cdot \sqrt{2 \times 32{\cdot}2 \times 1{\cdot}2683}}$$

or

$$C_d = 0.5522$$

FACTORS AFFECTING ORIFICE DISCHARGE IN A MULTI-OUTLET IRRIGATION PIPE

by

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B. S., Kansas State University, 1977

AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Agricultural Engineering

KANSAS STATE UNIVERSITY

Manhattan, Kansas

1978

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ABSTRACT

Adjustments have to be made manually in order to approximate equal flow out of the various gates in gated pipe. Complete automation of the pipe system can not be attained until a way can be found to solve this problem.

This study was undertaken to solve the problem of non-uniform flow through gates. Studies made by many people in a similar or related area and well established theoretical analyses have indicated the dependence of the discharge out of any orifice upon the static head at the orifice and its discharge coefficient. Earlier studies, also,suggested the effect of approach velocity toward an orifice and the head at the orifice upon the discharge coefficient. There was only one empirical formula given in the literature which related the coefficient with the dynamic and the static heads. The shortcoming of this relationship lies in the fact that the coefficient of discharge for the orifice at the dead end must be assumed before the coefficient for other orifices can be determined.

Investigation was, therefore, made with the objectives of determining the relationship between the orifice discharge coefficient and the approach velocity towards the orifice and between the coefficient and both the approach velocity and the static head at the orifice. Further, the applicability of the only empirical relationship available from the literature on the conditions of flow with an eight-inch irrigation pipe was investigated. The accuracy of predictions by the empirical relationships obtained as a result of this study were also to be determined. Finally, a procedure for using the relationships obtained in designing optimum slopes of pipes to provide uniform flow was sought.

A simulation of field conditions was carried out in the laboratory. An eight-inch pipe, thirty feet long was used as the test pipe. The friction factor of the pipe was determined by running water through and determining the head loss between two points, one near the inlet end and the other near the outlet end. Circular holes were then drilled in the pipe to represent the gates that are found in gated irrigation pipes. The pipe was connected to a water source at one end while the other was plugged. Water was then run through such that it was discharged through the orifices. Samples were caught and weighed within some recorded time intervals at each orifice. Thus, the rate of discharge out of each was determined. Manometer taps positioned opposite each orifice were used in determining the static head at each outlet. From the data collected, the approach velocity and the discharge coefficient at every orifice was calculated.

Statistical methods employing a "SAS" (Statistical Analysis System) computer program was used in analyzing the data. Using several statistical procedures, two prediction equations were obtained; one expressing the orifice discharge coefficient as a function of approach velocity while the other expressed the coefficient as a function of both the approach velocity and the static head at the orifice.

The two equations were used to establish a methodology for designing an optimum slope that would give flows with minimum deviation from equality of flow.