HYDRAULIC CHARACTERISTICS OF DISCHARGE FROM ORIFICES IN ALUMINUM IRRIGATION PIPE

by

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INTRODUCTION

The use of irrigation to provide adequate soil moisture for growing crops is a long established practice in Kansas. The United States Census of 1890 (11) tabulated twenty-one thousand acres of Kansas farmland under irrigation. By 1954 the total acreage of irrigated land had increased to four hundred twenty one thousand acres. During this period the United States Bureau of Reclamation was developing and investigating 20 project areas for irrigation development which would add approximately three hundred fifty thousand acres of irrigated land in the state of Kansas. One of these experimental projects, the Bostwick Irrigation District, consisting of forty thousand acres, was being irrigated from the Courtland Canal near Belleville, Kansas, at the time of this writing.

The fact-finding committee of the Kansas Water Resources Board (11) in a report to the governor of Kansas predicted that by 1975 the irrigated acreage in Kansas will reach one million acres. This committee reported that fourteen percent of the land under irrigation in 1954, about sixty thousand acres, was irrigated by overhead sprinkler systems and the remaining by ditch or gravity systems.

Irrigated acreage in Kansas in 1960 was estimated by the Engineering Extension Department at Kansas State University to be nearly one million acres which indicated that irrigation was increasing more rapidly than had been predicted by the fact-finding committee for the Kansas Water Resources Board in 1954.

Kohler (13) in discussing water requirements and uses in the United States stated that irrigation accounted for the largest single use of fresh water and was estimated to require seventy-five to one hundred billion gallons of water per day in 1955. This was approximately one half of the amount of fresh water used annually for all purposes. Irrigation efficiency or percent of water available to crop roots which farmers achieve in their operation varies from 15 to 90 percent. Losses accounting for the low efficiencies were evaporation, deep percolation and run-off.

Kohler stated that the rapid population increase has been causing an increasing demand for domestic water. Individual requirements for water vary from approximately 60 gallons a day in communities of 500 or fewer people to 180 gallons per day in cities over 10,000 population or more.

Increasing industry demands additional supplies of fresh water and with industrial expansion moving west the problem of stream and river pollution has been accelerated according to Kohler.

Conservation of fresh water in the United States has long been a problem of significance and since irrigation constitutes the greatest single consumption of fresh water improved irrigation practices and research in achieving greater irrigation efficiency should contribute significantly to any program of water conservation.

The use of gated irrigation pipe is one method which offers increased irrigation efficiency by eliminating losses which occur in transporting water from the point of supply to the crop in the field. Improvement in the design of gated pipe would aid in reducing run-off losses and permit the farmer to set all gates uniformly.

STATEMENT OF THE PROBLEM

Equal flow in each furrow would aid in reducing losses that occur in run-off when water reaches the end of some rows ahead of other rows in the area being irrigated. Uniform distribution of water throughout the field would also increase production by preventing over irrigation of some areas causing deep percolation losses while leaving other areas without sufficient soil moisture for maximum crop yields.

Uniform flow from all gates in a gated pipe system would be desirable in preventing waste of irrigation water and increasing crop production. Equal flow at each gate has usually required a different setting of each gate. However, if uniform flow could be obtained with equal gate settings irrigation efficiency could be improved and the operation and management of gated pipe could be simplified.

The purpose of this study was to determine whether it is possible to achieve equal flow from orifices in a level pipe by eliminating gates and resulting friction losses inherent in present gated irrigation pipe design.

More specifically the purpose of this study was to determine whether an equal discharge of water for distribution in an irrigation system can be obtained from uniformly spaced orifices of constant area in a level pipe with a uniform cross-sectional area by varying the pressure head at the pipe inlet within the limits of practical values. If equal flow can be obtained from orifices, then it may be possible to manufacture gated pipe from which equal flow can be obtained with uniform gate settings and level pipe. This will provide system efficiency previously not intentionally included. Availability of water was the irrigator's first consideration and conservation of water a factor to be included if and only if the system adequacy was first established. This principle could be incorporated in the design for land leveling making it possible to provide zero side slope when gated pipe is to be used.

REVIEW OF LITERATURE

Gated irrigation pipe was developed after World War II and is a system which provides control and distribution of irrigation water under low heads ranging from 0.5 to 8.0 feet of water. The use of a low pressure pipe system is an efficient method of conveying irrigation water and provides positive control of the stream of water delivered to each furrow (24).

Hansen (7) stated that there was an urgent need for controlling and measuring the flow of water from gated irrigation pipe and he further explained that a method was needed to assure uniform distribution of water from each gate with uniform gate settings. Hansen experimented with two 20-foot lengths of four-inch gated pipe with gate spacings of 22 inches and two 20-foot lengths with 36-inch gate spacing. By placing the pipe on a slope of 1 in 300, equal flow was achieved from all of the gates. Fig. 1 shows the equipment design of the irrigation system used in this experiment.

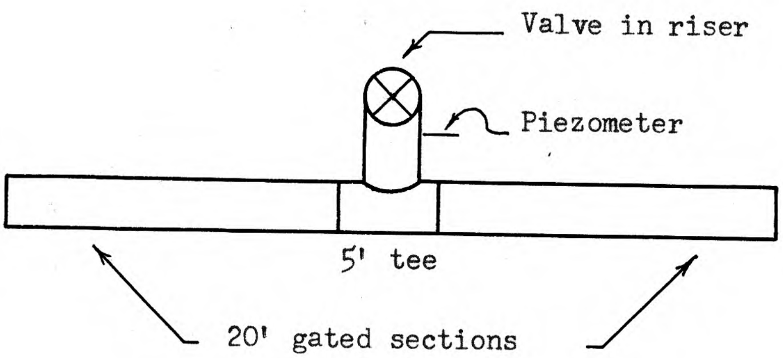

Fig. 1 Irrigation system design

The discharge for the gates at a given head and gate opening was determined and these data were plotted on two-cycle log log graph paper. Hansen proposed that by measuring the head above the center line at the inlet end of the pipe, and with a known gate setting, the discharge could be determined from the graph or for a desired discharge the gate opening and head could be determined from the graph.

Tovey in evaluating hydraulic losses of four-inch pipe used 1 in 300 slope recommended by Hansen (7). The water flowing from all the gates was measured and from these measurements the average discharge per gate was calculated. The average head loss per gate was determined by measuring the piezometric head at the inlet and at the last gate and dividing the difference in head by the total number of gates.

$$\text{Average head loss per gate} = \frac{H-h_o}{\text{Number of gates}} \qquad (1)$$

Tovey (24) pointed out that when the average head loss per gate exceeded 0.1 foot for a gate opening of 0.55 inches, 0.15 foot for gate opening of 0.95 inches and 0.17 foot for gate opening of 1.7 inches, it was not possible to obtain equal flow. He did not state the head at the pipe inlet corresponded to these losses per gate or the manufacturer of the gated pipe used. The erosive nature of the jet at heads higher than these values was too great to be practical according to Tovey. In evaluating flow from gated pipe he concluded that gated pipe offers a refined method for controlling small streams of irrigation water.

Somerhalder (22), experimenting with eight-inch gated pipe, used one manufacturer's pipe to calibrate the discharge for various gate openings at heads ranging from 3½ inches to 39½ inches of water. Leaking at the gates was also determined for heads ranging from 3½ to 27½ inches of water. He found that the average leakage per gate was 1.49 pounds per minute at 3½ inches, 0.122 pounds per minute at 15½ inches and 0.065 pounds at 27½ inches of water. Bursting pressure required to blow the gates out ranged from 165 to 200 pounds per square inch.

Fischbach (5), using eight-inch gated pipe from the same manufacturer and non-gated eight-inch pipe, found that the presence of gates did not affect friction loss until flow reached about 850 gallons per minute. Increasing the flow to 1100 gallons per minute indicated an increasing loss of head due to the presence of gates. Fischbach stated that the orifice formula,

$$Q = C_d A \quad 2gH \tag{2}$$

had been used to determine flow from gates in gated pipe and that the value used for C_d was 0.6.

The review of literature revealed no additional experimental work with gated pipe. Similar studies have been undertaken involving the flow of fluid in a manifold system. These experiments dealt with the flow of liquids and gases and were concerned with small manifold diameters.

One of the early investigations was undertaken in 1927 by Ellms (3) who published results of pressure distribution in a perforated pipe filter lateral system.

Enger and Levy (4) proposed that the flow of water from a manifold or perforated pipe could be analyzed by considering the variation of pressure in a long narrow slot. In their analysis pipe friction was neglected and a straight pipe with uniform cross-sectional area was assumed. These experimenters wrote a momentum equation for an elemental length of pipe:

$$\left[w\ A(h+dh) - wAh\right] dt = w\ \frac{VA}{g}\ dt \left[(V-dV) - V\right]$$

$$dh = -\frac{V}{g}\, dV$$

$$\int dh = -\frac{1}{g}\int V dV$$

$$h = -\frac{V^2}{2g} + h_o \tag{3}$$

Where:

w = specific weight of fluid in pounds per cubic foot

A = cross-sectional area of pipe in square feet

h = static pressure head in feet

t = time in seconds or minutes

V = velocity in feet per second

g = acceleration of gravity in feet per second squared

h_o= static pressure at dead end in feet

When pipe friction is neglected, the pressure at any point along the slot is equal to the static head at the dead end and minus the velocity head at the given point.

Enger and Levy assumed that the coefficient of discharge in the flow equation, $q = C_d b dx \sqrt{2gh}$, remains constant and derived the following equation for pressure at any point along the slot:

$$h = \frac{h_o}{2} \text{ Vers} \left[\pi \frac{-2C_d b\ x}{A} \right] \tag{4}$$

Where:

C_d = coefficient of discharge

b = slot width

x = distance from dead end of main line in feet

Enger and Levy found that the coefficient of discharge decreased with an increase in velocity in the pipe and proposed the following empirical formula as a result of an experiment with ten 3/8-inch openings in a two-inch water pipe.

$$C_d = \frac{h - V^2/2g}{h} \quad C_{do} \tag{5}$$

$V^2/2g$ is the velocity head of water in the pipe approaching the opening and C_{do} the coefficient of discharge for the last opening.

A tabulation by Enger and Levy for one set of experiments showed close agreement between observed values of pressure head and values calculated with equation 3, with a pressure range of 3.22 feet of water at the inlet to 3.38 feet of water at the dead end.

Kunz (14) used an analytical method to derive Enger's and Levy's equation for the variation of the discharge coefficient. In his derivation of an equation similar to equation 4, he assumed that the coefficient of discharge was proportional to the static head at the point under consideration.

In considering the loss of head in a uniformly tapped pipe, Gladding (6) assumed uniform outlets evenly spaced along a main line would discharge an equal amount of fluid. Edwards (2) reported that Gladdings assumption was not possible and did not represent a close approximation of actual conditions except for a pipe having a large cross-sectional area compared to the total outlet area. Edwards explained that the velocity in the direction of flow decreases after passing each orifice and is accompanied by a corresponding increase in pressure. The decrease in transfacial velocity is responsible for the increase in efficiency of each succeeding orifice resulting in a greater discharge from each successive opening downstream.

Oakey (18) studied the hydraulic losses peculiar to the flow of a fluid from a short tube in the side of a pipe and represented the losses by multiplying the upstream velocity by appropriate coefficients. In his analysis he began with the energy equation:

$$Q_u\, w \left[\frac{Vu^2}{2g} + h_u\right] = Q_d\, w \left[\frac{V_d^{\,2}}{2g} + h_d\right] + qw \left[\frac{q}{Q_u}\;\frac{V_u^{\,2}}{2g} + h_u\right]$$

This was reduced to:

$$h_d - h_u = \left[3\ \frac{q}{Q_u} - \left(\frac{q}{Q_u}\right)\right]^2 \cdot \frac{V_u^2}{2g} \qquad (6)$$

where:

Q_u = flow rate in upstream cross-section of main line in cubic feet per second

w = specific weight of fluid

V_u = velocity in upstream cross-section of main line in feet per second

h_u = pressure head in upstream cross-section of main line in feet

Q_d = flow rate in downstream cross-section of main line in cubic feet per second

V_d = velocity in downstream cross-section of main line in feet per second

h_d = pressure head in downstream cross-section of main line in feet

q = rate of flow in branch pipe in cubic feet per second

From the energy equation he assumed the head on the discharging tube to be:

$$\frac{q}{Q_u}\ \frac{V_u^2}{2g} + h_u$$

Oakey called the change in pressure given by equation 6, the theoretical rise in the hydraulic gradient between a point upstream from the opening and a point downstream from the opening. In his experiment the actual rise in pressure was defined by the relationship:

$$k_2\ \frac{V_u^2}{2g}$$

Where: k is the fraction of momentum change of the entering or leaving fluid produced by a pressure change in the main duct.

The lost head in the pipe was defined as the difference between the theoretical rise in the hydraulic gradient and the observed rise. Oakey expressed this relationship by:

$$k_1 \frac{V_u^2}{2g}$$

Values of k determined by experiment were tabulated for four diameter ratios, D_b/D 1 to 4.24, 1 to 2.82, 1 to 1.82 and 1 to 1.21. As the flow ratio q/Q_u varied from 0.1 to 1.0, k_1 varied from approximately 0.1 to 1.4 while k_2 varied from approximately 0.2 to a high of 0.85 when the flow ratio was 0.8. As the flow ratio increased to 1 the value of k_2 decreased to approximately 0.7.

Oakey stated the values for the coefficients are independent of the actual discharge or pressure and are affected only by flow ratio q/Q_u. If this ratio and the diameter ratio are constant, the k values will be correct regardless of the actual value of D, d, Q_u or q. The following equation was proposed by Oakey for determination of the coefficient of discharge:

$$q = C_d \, a \sqrt{2g \left[\frac{q}{Q_u} \frac{V_u^2}{2g} + h_u \right]} \qquad (7)$$

Keller (12) in discussing the problems encountered in manifold flow, stated that only two forces (1) inertia and (2) friction determine the distribution of flow from manifolds. Inertia corresponds to a change in velocity head and in most cases as the fluid moves along the manifold its longitudinal velocity decreases because a part of the fluid volume is being discharged through the ports. Thus the fluid in the manifold is being decelerated and causes an increase in pressure as predicted by Bernoulli's equation. Friction causes a reduction in pressure and the relative magnitudes of these two forces will determine whether there will be an increase or decrease of static pressure at the dead end of the manifold.

In analyzing manifold flow Keller wrote the following basic equation for pressure rise in the direction of flow:

$$\frac{dP}{w} = -d\left(\frac{V^2}{2g}\right) + f\,\frac{dx}{D}\,\frac{V^2}{2g} \tag{8}$$

Where:

P = pressure of fluid in pounds per square foot

w = specific weight of fluid in pounds per cubic foot

V = velocity

f = friction factor defined by $h = f\,\frac{LV^2}{D2g}$

D = diameter of pipe

g = acceleration of gravity in feet per second squared

x = distance from dead end of main line in feet

The deceleration term, $-d\,\frac{V^2}{2g}$ is negative since an increase in pressure results from a decrease in velocity. The second term is the friction factor and although friction causes a reduction in pressure in the direction of flow it is positive because x is measured from the dead end, in the direction opposing flow as shown in Fig. 2.

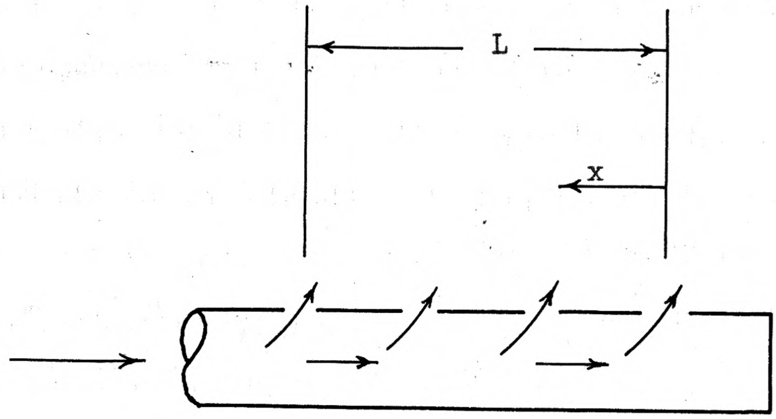

Fig. 2 Inlet manifold

To determine the pressure P, at any distance from the dead end for an inlet manifold, Keller wrote the following expression:

$$P = P_L + w\left[\frac{V_L^2 - V^2}{2g}\right] + \int_{L}^{(L-x)} f\,\frac{w}{D}\,\frac{V^2}{2g}\,dx \qquad (9)$$

Where: subscript L represents inlet end of main line

For uniform discharge the manifold velocity decreases linearly from its initial velocity to zero at the dead end. Keller substituted $V = V_L \frac{x}{L}$ in equation and integrated to obtain:

$$P = P_L + w\,\frac{V_L^2}{2g}\left[1 - \left(\frac{x}{L}\right)\right]^2 - f\,\frac{wL}{3D}\,\frac{V^2}{2g}\left[1 - \left(\frac{x}{L}\right)^3\right] \qquad (10)$$

This equation is similar to the one proposed by Enger and Levy (4) although more complicated, since it includes a friction term

$$-\frac{wL}{3D}\,\frac{V_L^2}{2g}\left[1 - \left(\frac{x}{L}\right)^3\right]$$

A second order differential equation for the velocity in a manifold of uniform cross section and constant slot width was derived by Keller (12) who stated no solution had been found. He used a numerical point by point method to find the velocity distribution.

Keller used the following two dimensionless ratios to define manifold flow:

$$\text{L/D ratio} = \frac{\text{active length of manifold}}{\text{diameter of manifold}}$$

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

Manufacturers of gas burners, according to Keller, stated that if the area ratio does not exceed one, the height of gas flames along the length of the pipe will be practically uniform. This statement did not take into consideration the length/diameter ratio and when L/D = 70 Keller stated that friction practically nulifies the deceleration regain. Keller found that for an area ratio of unity and L/D ratios greater or smaller than 70 discharge will not be uniform and uniformity of discharge for area ratios of two or greater could not be obtained regardless of the L/D ratio.
Dow (1) found that uniform distribution of flow was not dependent on the L/D and area ratios but also affected by the rate of flow through the manifold and pointed out that Keller had neglected the variation of the Reynolds number and a corresponding variation of the friction factor along the manifold. Dow succeeded in showing that the variation in discharge could be altered for a constant diameter, constant discharging pipe by changing the total flow rate.

Using the same fundamental equations that Keller (12) had proposed for uniform discharge, that static pressure along the entire length of the manifold must remain constant, Dow (1) proceeded to develop expressions for the variation in the diameter or hydraulic radius for an inlet manifold. He verified his results by placing tapered plugs in pipe burners and observing the resultant flame heights.

Howland (9), in discussing the gain in total head that may be observed in a straight flowing stream when a side stream separates from it, stated:

> . . . . .if the Bernoulli equation is written. . . .the so-called head loss term that must be introduced on the right side of the equation in order to balance it will, in general, be found to be negative for ratios of side flow to main flow of less than 1 to 2. In other words a gain in head is observed.

This apparent gain in head was explained by Howland:

> . . . . .the branch scoops off the relatively slow moving edge layers of water, leaving the fast moving and therefore high-energy containing central core of water continuing past the take off has a higher average unit energy content, or head, than the complete stream approaching the take off.

In designing a perforated pipe for uniform discharge, Howland used Enger's and Levy's (4) simple Bernoulli equation:

$$h = \frac{P}{w} = \frac{Po}{w} - \frac{V_L^2}{2g} \left[\left(\frac{x}{L}\right)^2 - \frac{fL}{3D} \left(\frac{x}{L}\right)^3 \right] \qquad (11)$$

Howland stated that the coefficient of discharge for the pipe and submerged discharge produced more variation than free discharge. Howland used a copper pipe 16 feet long and 1.606 inches in diameter in his experiment. Values for the coefficient of discharge C_d were determined experimentally by Howland and used in his calculation. He compared the observed discharge from each orifice with the discharge predicted by the equation,

$$q = C_d \; a \sqrt{2gH} \qquad (12)$$

Shove (20) reported that the effect of friction in fluid flow had been studied in detail, but less attention has been given to the pressure change accompanying the acceleration or deceleration which occurs as a portion of the fluid enters or leaves through a duct. Figure (3) illustrates the conditions for combining flow.

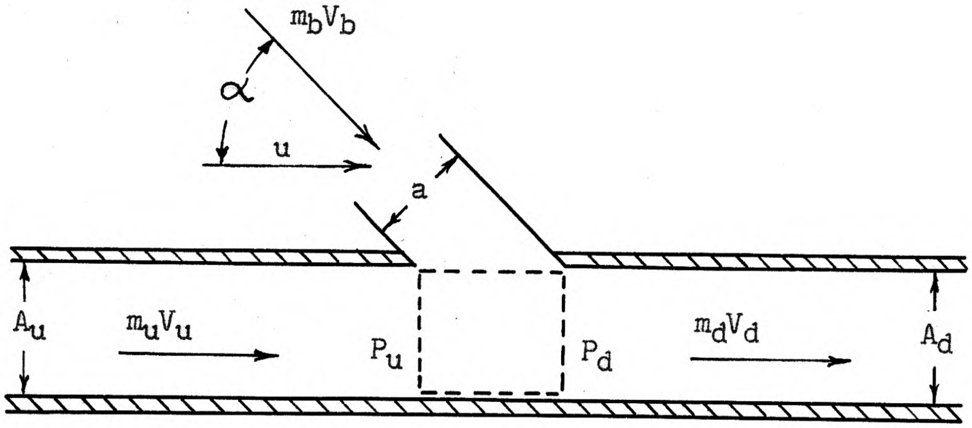

Fig. 3 Frictionless combining flow

Shove began his analysis of pressure change associated with velocity change by writing the impulse-momentum equation for combining flow:

$$m_dV_d - m_uV_u - m_bV_b \ \cos\alpha = Ft \tag{13}$$

Where:

m_d = mass downstream from branch in pounds per feet per second squared

V_d = velocity in main line downstream from branch in feet per second

m_u = mass upstream from branch in pounds per feet per second squared

V_u = velocity in main line upstream from branch in feet per second

m_b = mass in branch in pounds per foot per second squared

V_b = velocity in branch in feet per second

F = force in pounds

t = time in seconds

The force F in equation 13 is the resultant horizontal force on the fluid.

Shove substituted $P_uA_u - P_dA_d$ for F and divided by t to obtain:

$$\frac{m_dV_d}{t} - \frac{M_uV_u}{t} - \frac{m_bV_b}{t} \cos\alpha = P_uA_u - P_dA_d \tag{14}$$

Where:

P_u = pressure of fluid in the main line upstream from branch in pounds per square foot

A_u = area of main line upstream from branch in square feet

P_d = pressure of fluid in the main line downstream from the branch in pounds per square feet

A_d = area of main line downstream from the branch in square feet

Since $\frac{m}{t}$ = mass per unit, $\frac{m}{t}$ was replaced by ρVA and simplified for a duct of uniform cross section which resulted in the following equation:

$$\rho V^2{}_d - \rho V^2{}_u - \rho \frac{a}{A} V_b{}^2 \cos\alpha = P_u - P_d \tag{15}$$

Where: ρ is the density of the fluid in pounds seconds squared divided by feet to the fourth power

Substituting $\frac{W}{g} = \rho$ and writing in terms of velocity heads equation 15 was written in the following form:

$$2\left[\frac{V_d{}^2}{2g} - \frac{V_u{}^2}{2g}\right] - \frac{2a}{A}\frac{V_b{}^2}{2g} \cos\alpha = -\Delta h \tag{16}$$

Where:

w = the specific weight of the fluid in pounds per cubic foot

g = the acceleration due to gravity in feet per second squared

a = the area of the branch in square feet

h = static pressure head in the main line in feet

A friction loss term was introduced to the equation and was justified on the basis that friction loss in a duct can be considered equal to the normal friction loss observed when there is no flow through the wall provided that the area of the intake or discharge openings is a small percentage of the total duct wall area. The resulting equation for head loss for combining flow could now be written in differential form:

$$-\frac{dh}{dx} = 2\frac{d}{dx}\left[\frac{V^2}{2g}\right] - 2\frac{a}{A}\frac{V_b^2}{2g}\cos\alpha + \frac{dh_f}{dx} \tag{17}$$

Where:

h_f = the head differential in the Darcy Weisback equation

x = distance from the dead end of the main line in feet

Making the following substitutions:

$$aV_b = q,\ \frac{q}{A} = \frac{dV}{dx},\ V_b \cos\alpha = u$$

Where:

q = the rate of flow in the branch in cubic feet per second

u = velocity component of V_b parallel to the main line in feet per second

Equation 17 was reduced to:

$$-\frac{dh}{dx} = \left[\frac{2V-u}{g}\right]\frac{dV}{dx} + \frac{dh_f}{dx} \tag{18}$$

An identical analysis can be made for dividing flow and since in this case the decreasing velocity head tends to off set the friction loss, equation 18 can be written to cover this case as follows:

$$\frac{dh}{dx} = -\left[\frac{2V - u}{g}\right] \frac{dV}{dx} + \frac{dh_f}{dx} \tag{19}$$

If in the general case, u is a constant proportion of V, equations 18 and 19 can be written:

$$\frac{dh}{dx} = - K \frac{V}{g} \frac{dV}{dx} \pm \frac{dh_f}{dx} \tag{20}$$

Where:

k = constant applied to velocity head change

The plus sign on the friction term $^{dh_f}/dx$, applies to dividing flow when the static pressure change associated with velocity change tends to off set the static pressure change associated with friction. The minus sign on the friction term applies to combining flow when the pressure change due to increasing velocity adds to friction loss.

Since:

$$\frac{V_u}{V_d} = \frac{Q_u}{Q_d} = \frac{Q_d - q}{Q_d} = 1 - \frac{q}{Q_d}$$

Equation 20 can be written in terms of the flow ratio and a given increment of pipe length as:

$$\Delta h = - K \frac{V_d^2}{2g} \left[\frac{q}{Qd} \left(\frac{2-q}{Qd}\right)\right] \pm \Delta h_f \tag{21}$$

Where:

Q_u = flow rate in main line upstream from branch in cubic feet per second

Q_d = flow rate in main line downstream from branch in cubic feet per second

Shove (20) wrote the following relationship for the rate of static pressure change in an air duct with air outlets along its length as a result of his experiment.

$$\text{Rate of static head change} = -K \left[\text{Rate of velocity head change} \right] + \text{Rate of friction head loss} \qquad (22)$$

Shove used the Darcy Weisback formula to represent the friction loss and equation 22 was then written as:

$$\frac{dh}{dx} = -K \quad \frac{V}{g}\frac{dV}{dx} + \frac{fV^2}{D\ 2_g} \qquad (23)$$

$\frac{Q}{A} = V$ was then substituted in equation 22 and was written:

$$\frac{dh}{dx} = -\ K\frac{Q}{A^2g} \quad \frac{dQ}{dx} + f\ \frac{Q^2}{A^2D2g} \qquad (24)$$

and simplified to:

$$\frac{dh}{dx} = -MqQ + NQ^2 \qquad (25)$$

Where:

$$M = \frac{K}{A^2\ g}$$

$$N = \frac{f}{A^2\ D2g}$$

A point-by-point evaluation can be made of equation (25) to predict the static pressure gradient in the pipe. The solution can be achieved by starting at the dead end of the pipe if the static head, h_o, at the dead end, the q entering or leaving the opening nearest the dead end and K for the system is known or can be determined.

Shove (20) assumed a constant friction factor throughout the test section and justified this assumption on the basis that an air flow of 164 c.f.m. in a five-inch duct corresponds to a Reynolds number of 50,000 and for a friction factor of 0.25 any marked change in the friction factor would occur below a Reynolds number of 50,000. Shove experimented with air flows equal to or greater than 164 c.f.m. which is in the region of turbulent flow and constant friction factor as shown by a Moody or Stanton diagram found in most textbooks on fluid mechanics.

EQUIPMENT AND PROCEDURE

For the purpose of this experiment it was essential that some positive method for controlling and determining the pressure head at the pipe inlet be designed and constructed. It was also necessary to provide a method for reducing the velocity of approach of the water to a negligible value. These conditions were achieved by means of a reservoir equipped with a system of baffles and a vertical opening in one side which could be closed or opened by inserting or removing one or more one by six-inch by two feet long redwood boards. The water level in the reservoir was controlled by the height of the boards placed in the opening.

The value of approach velocity becomes negligible when the ratio of the area of the contracted section to the area of the channel is small. In this study the calculated ratio of the area of discharge to the area of the reservoir varied from 0.0356 for a head of 1.03 feet to .00825 for a head of 3.025 feet.

The pressure head and approach velocity controls were considered in the design of the head control reservoir, four feet square and eight feet high as shown in Fig. 4. The tank was constructed of 18 gauge galvanized steel with the seams locked together in a sheet metal break. The seams were soldered to make them watertight. The bottom of the tank was made from 18 gauge sheet metal and riveted to the sides. A stress analysis of the tank under maximum load conditions revealed that it would withstand the load when re-enforced with four angle iron frames. At the base a $2\frac{1}{2}$ by $\frac{1}{4}$-inch angle iron frame was used. Frames $2\frac{1}{2}$ by 3 1/8-inches were used two feet and four feet above the base of the tank. A $1\frac{1}{2}$ by 3/16-inch frame of angle iron was found to provide sufficient strength around the top. The angle iron frames were cut, welded together and secured in place around the reservoir with copper rivets. The sides for the opening in the tank were formed with $1\frac{1}{4}$-inch channel iron. This channel was riveted in the corner of the tank to the sheet metal side. The channel for the inside edge was welded to a 2 by 1/8-inch angle iron which in turn was riveted to the tank. This construction is shown in Fig. 6.

The discharge opening, six inches in diameter for the pipe inlet in the tank, was cut in the sheet metal side near the bottom as shown in Fig. 4. A collar with an outside diameter of eight inches was cut and welded to a one-foot length of six-inch aluminum irrigation pipe. A second eight-inch collar was constructed and placed inside the tank to provide a means of distributing the pressure of the eight bolts which were used to attach the collar to the tank, thus preventing leakage around the rubber gasket between the tank and the outside collar. A Philadelphia rod was clamped to the side of the tank. A glass tube was attached to the face of the rod to provide a manometer for the tank. The manometer was attached to 1/8-inch pipe by means of a rubber hose at the points where head measurements were desired.

EXPLANATION OF PLATE I

Fig. 4. Complete instrumentation for calibration of triangular weir.

PLATE I

Fig. 4

Baffles were used to provide nearly uniform velocity across the approach channel. Construction of the head control tank included installation of baffles to provide nearly uniform distribution of velocity in the reservoir. The first baffle was made from 1/8-inch perforated well casing which was installed at approximately a 45-degree angle with the vertical supply pipe discharging into the reservoir. The second set of baffles was made from 7/8-inch by 3 3/4-inch lumber mounted perpendicular to the flow of the water across the reservoir in two vertical planes. The baffles were arranged as shown in Figs. 5 and 6.

A triangular weir was constructed and calibrated for accurate water measurement. The decision to use a triangular weir was based on the principle that with increases or decreases of flows of water of less than 100 gallons per minute there is a correspondingly larger change in head on the weir. Measurement of small flows in a triangular weir is considered more accurate than in other standard weirs.

A 90-degree triangular notch forming a 45-degree angle with a vertical line at the vertex of the notch was cut in a $\frac{1}{4}$-inch aluminum sheet measuring four feet wide and two feet high. The base of the triangle opposite the 90-degree angle was two feet wide. The edge of the notch was then beveled in a milling machine to provide a sharp edge over the length of the notch. The plate of aluminum with the triangular notch was then mounted in the reservoir in order to calibrate the rate of discharge. A watertight bulkhead made from a sheet of $\frac{1}{4}$-inch aluminum four feet long and two feet high was installed in a vertical plane perpendicular to the flow across the center of the tank. On top of the bulkhead the triangular weir was mounted. Figure 7 shows the weir installed and ready for

EXPLANATION OF PLATE II

Fig. 5. Inside view of reservoir showing baffle made from 1/8-inch perforated well casing.

PLATE II

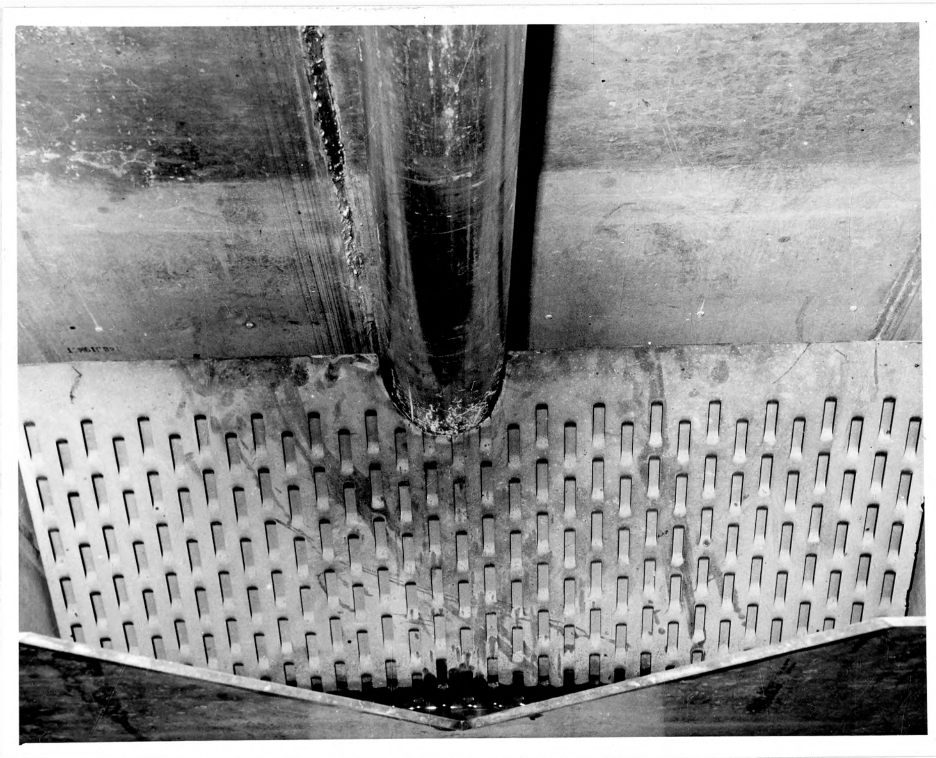

Fig. 5

EXPLANATION OF PLATE III

Fig. 6. View of opening of the tank used for regulating the head of water showing channel with sliding boards and baffle.

PLATE III

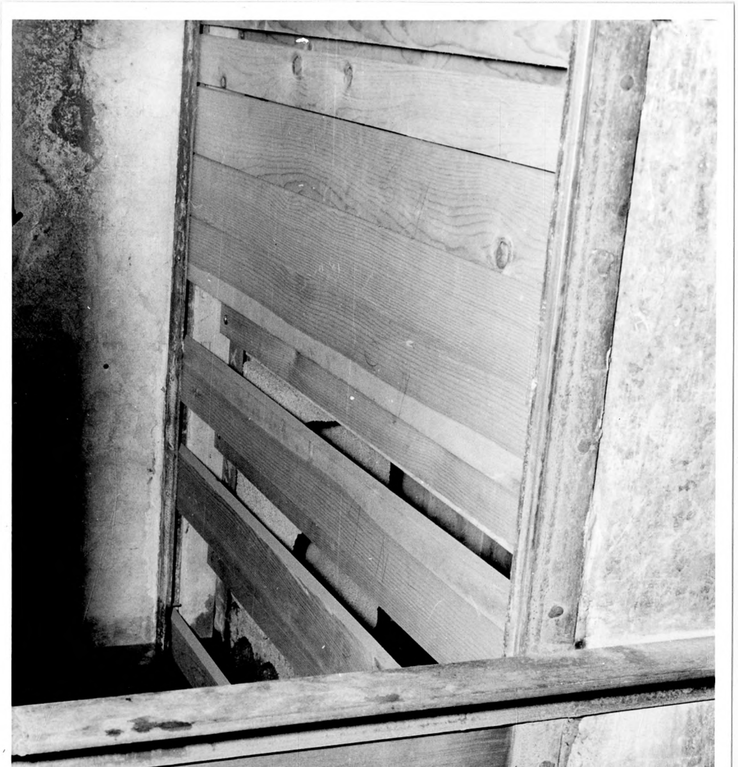

Fig. 6

calibration. Figure 5 shows the weir with the baffle made of well screen in place and the incoming supply pipe for the reservoir. The incoming water supply was discharged below the surface of the pool in front of the weir. The supply pipe was perforated near the discharge end so that a nearly uniform velocity could be achieved throughout the reservoir. In Fig. 8, the baffle has been set-up vertically to show the perforated supply pipe and the use of a concrete block to prevent a large discharge from the open end of the supply pipe. This figure also shows the baffle arrangement and a 3/4-inch pipe which was connected to an outside stilling basin by means of a ½-inch garden hose where a hook gage was used to measure the head on the weir. The stilling basin consisted of a galvanized tube eight inches in diameter and six feet eight inches high. The outside connection from the reservoir to the stilling basin and a section of the stilling basin is shown in Fig. 9. In Fig. 10, the installation of the hook gage and top of the stilling basin is shown. The head on the weir inside the reservoir was read to 1/1000 of a foot with this gage.

A weighing tank was necessary for accurate determination of discharge over the weir. A tank two feet wide, three feet ten inches long and two feet deep was modified for this purpose. A six-inch diameter opening was cut in the bottom of the tank and a quick acting valve was installed as shown in Fig. 11. The valve was made from a circular steel plate ¼-inch thick with a rubber seal glued to the face of the valve. This valve was attached to a right angle lever which was hinged at the bottom of the tank. When the lever was raised the valve moved down over the opening in the tank. A dog, constructed from a short piece of steel, was welded to the lever which locked over the edge of the tank holding the valve

EXPLANATION OF PLATE IV

Fig. 7. Triangular weir mounted in head control reservoir for calibration.

Fig. 8. Inside view of tank showing perforated supply pipe and tap for the outside stilling basin.

PLATE IV

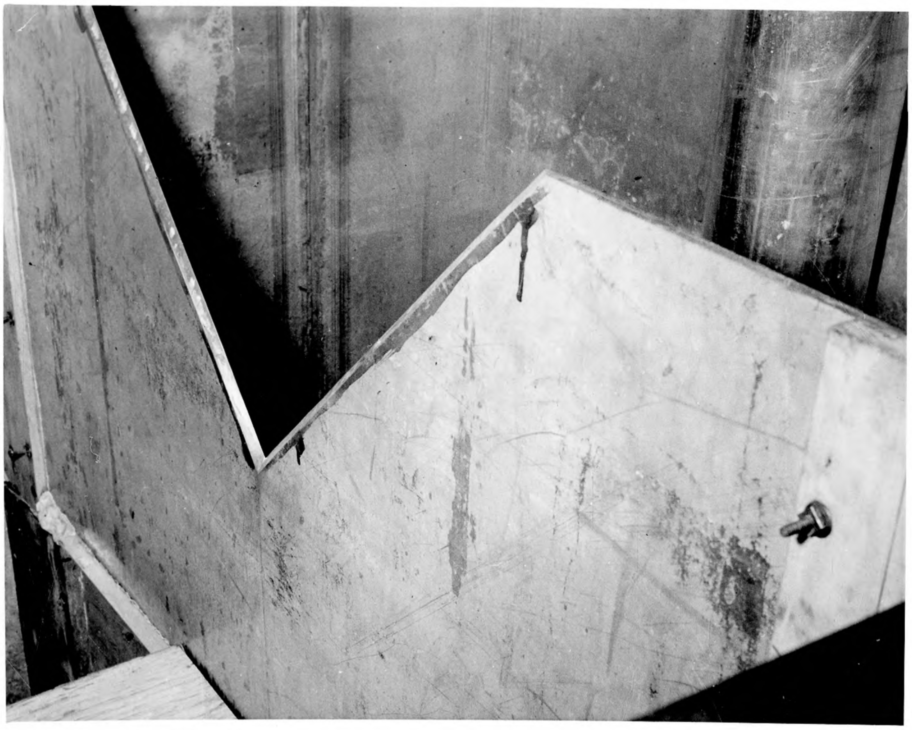

Fig. 7

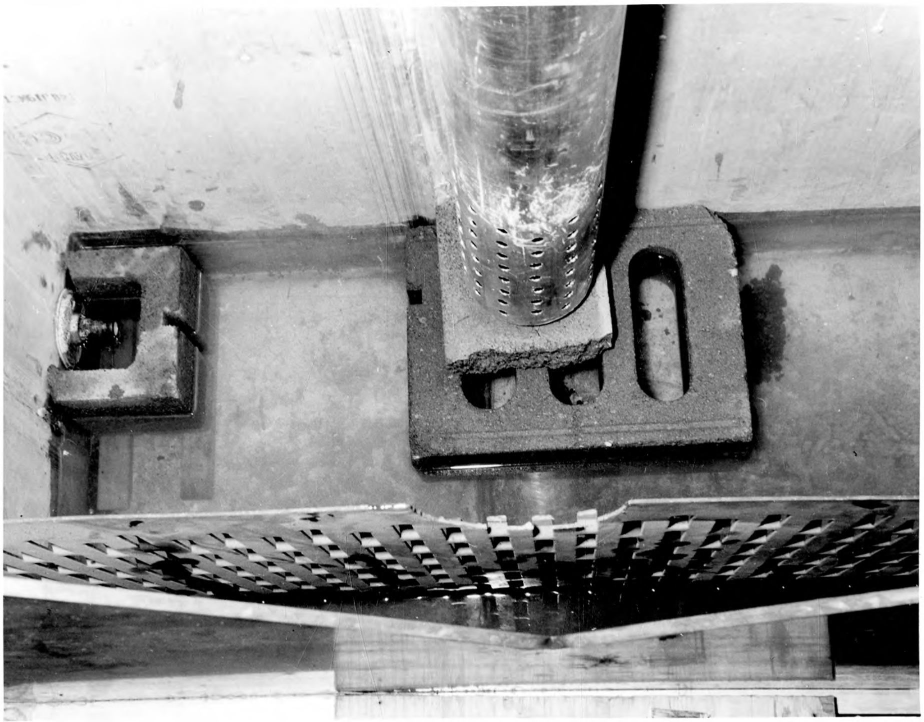

Fig. 8

EXPLANATION OF PLATE V

Fig. 9. Outside valve and hose connection for stilling basin.

PLATE V

Fig. 9

EXPLANATION OF PLATE VI

Fig. 10. A close up view of hook gage and top of the stilling basin.

Fig. 11. Inside view of weighing tank with quick acting valve in open position.

PLATE VI

Fig. 10

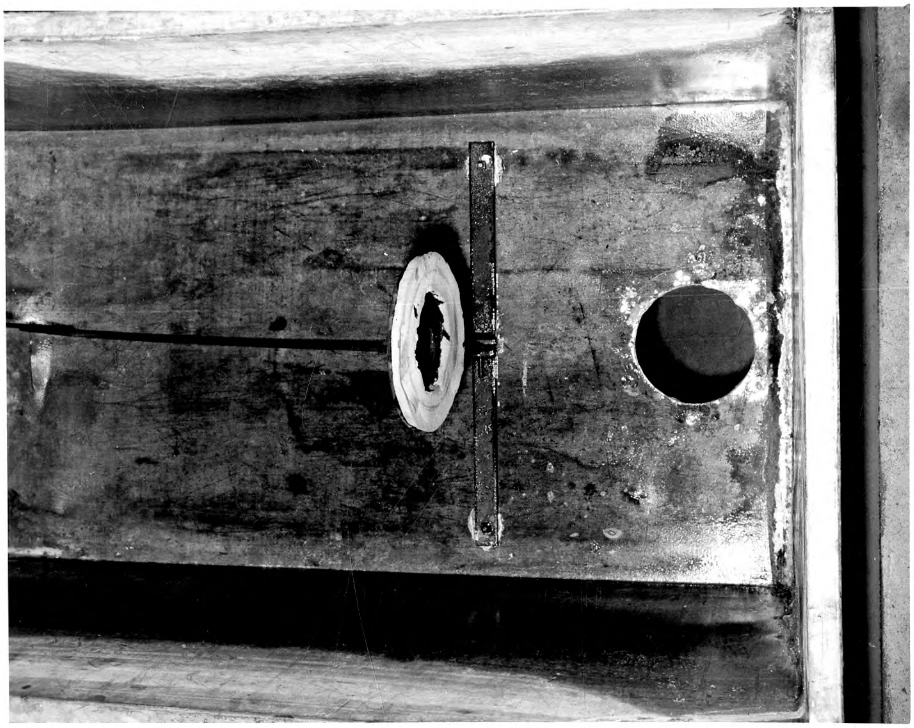

Fig. 11

closed under pressure and maintaining an adequate seal around the opening as shown in Fig. 12. Leakage at this valve was negligible. The weighing tank and scales used for calibration of the weir are shown in Fig. 12.

A Measure-Rite flow meter was used to determine the discharge of the weir when flows were 100 gallons per minute or more. A six-inch Venturi meter was also used to measure the flow of water over the weir when the flow was equal to or greater than 0.2 c.f.s. (89.76 g.p.m.). Readings from these instruments provided comparative information for a check of validity of the data obtained by the previously mentioned weighing method.

A Cornell four-inch centrifugal pump was mounted on a variable speed power unit driven by a $7\frac{1}{2}$ h.p. electric motor. This unit as shown in Fig. 13 was installed to pump the water for this experiment. Flows were regulated by a four-inch gate valve on the discharge side of the pump and also by increasing or decreasing pump speed with the variable speed drive. The equipment for calibration of the weir is shown in Fig. 4. It was necessary to place the reservoir on concrete blocks four rows high in order to catch and weigh the discharging flow. This structure is shown in Fig. 4.

Procedure For Weir Calibration

It was necessary to run a series of tests starting with a small flow and increasing the flow to the limit of the weighing tank in order to plot a curve showing discharge for the weir against the head on the weir. Thirty such tests were conducted for this phase of the study ranging from 6.78 to 407.78 g.p.m. These data were treated by curvilinear regression techniques to obtain a prediction equation for discharge over the weir. The data recorded for each test were: (1) head on the weir in feet of

EXPLANATION OF PLATE VII

Fig. 12. Weighing tank with valve closed and scales used for determining flow of water over the triangular weir.

Fig. 13. Four-inch centrifugal pump, discharge valve and variable speed power unit used for pumping the water for this experiment.

PLATE VII

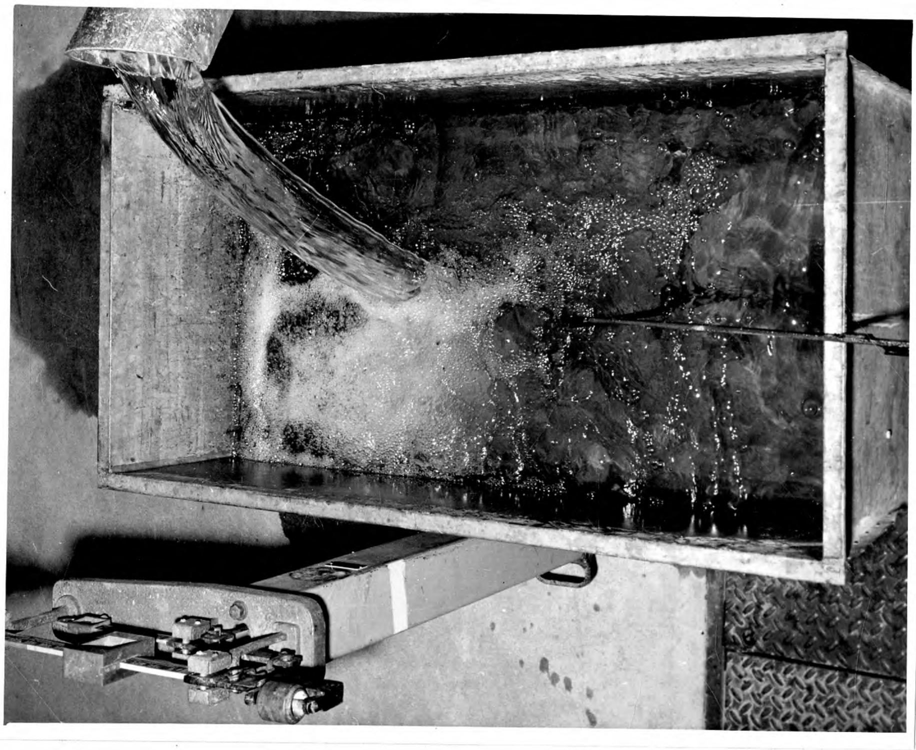

Fig. 12

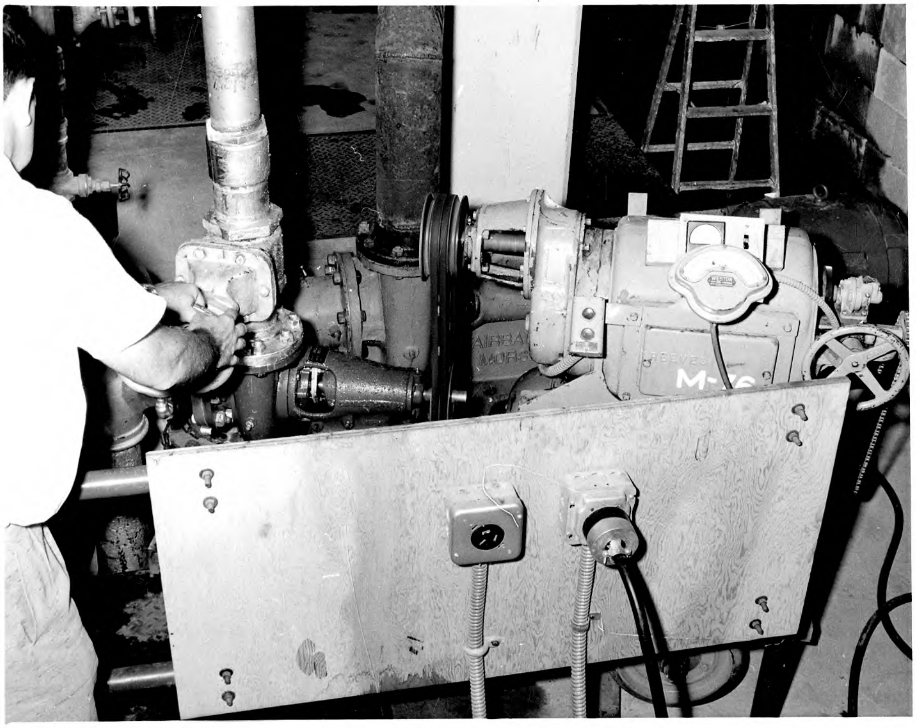

Fig. 13

water, (2) time in seconds to accumulate a predetermined number of pounds of water in the weighing tank. When the flow was increased to values that could be measured by the Venturi meter and the Measure-Rite flow meter, data were recorded for the flow as indicated by these instruments. Five replications were made for each flow and the mean values of these replications were calculated and used in deriving a prediction equation for the discharge of the weir. Head measurements on the weir were recorded from the hook gage as shown in Fig. 14. Figure 15 shows the method used for weighing the water discharged. Several flow rates over the weir are shown in Figs. 16 and 17. These figures indicated that the weir installation provided: (1) free admission of air under the falling sheet of water or nappe, (2) head sufficient to prevent nappe from clinging to back of the weir, (3) sharp edged crest, normal to flow and straight, (4) crest not affected by tail water.

A minimum depth of three feet in front of the weir in conjunction with the baffle helped reduce turbulence in the approach channel.

Equipment For Determining Flow From Orifices In A Manifold

Following the calibration of the triangular weir, the equipment necessary for determining flow from orifices in a manifold was assembled as shown in Fig. 18. The triangular weir was removed from the reservoir and mounted in the channel six inches above the point where the channel makes a 90-degree turn for returning water to the sump.

The weir was installed with $\frac{1}{4}$-inch flat head bolts counter sunk in the face of the weir. These bolts secured the weir to 3/4-inch aluminum angle bolted to the side walls and bottom of the channel with anchor bolts. A watertight seal between the aluminum angle and channel was achieved by

EXPLANATION OF PLATE VIII

Fig. 14. Reading hook gage to determine the head in feet of water over the vertex of the triangular weir.

Fig. 15. Weighing the discharge of water over the triangular weir and recording time with a stop watch.

PLATE VIII

Fig. 14

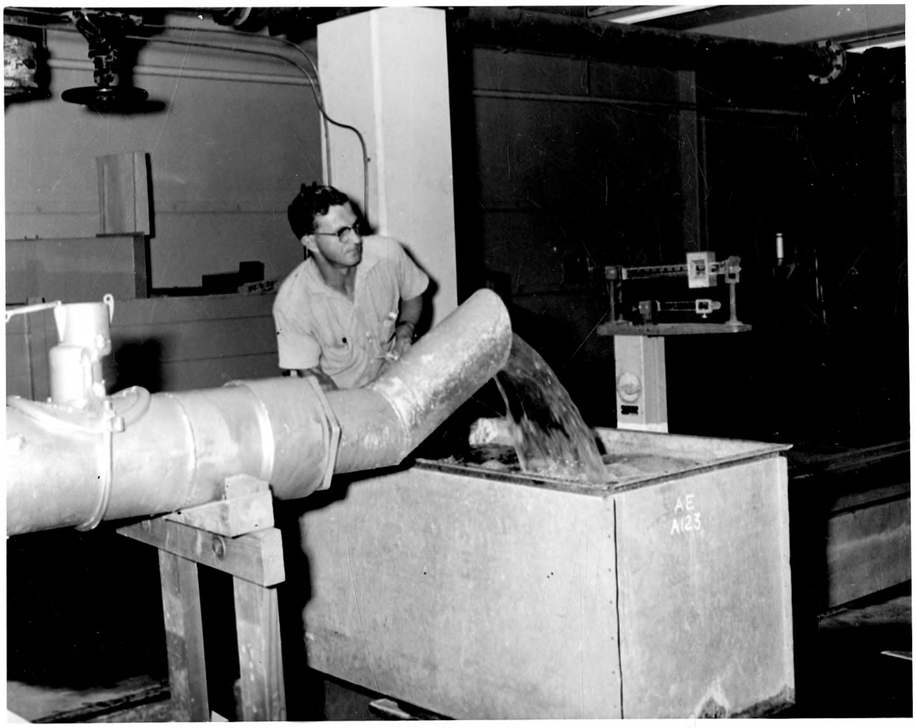

Fig. 15

EXPLANATION OF PLATE IX

Fig. 16. A small discharge of water flowing over the weir.

Fig. 17. A large discharge of water flowing over the weir.

PLATE IX

Fig. 16

Fig. 17

using a welding compound of epoxy resin. A watertight seal between the weir and aluminum angle was accomplished by the use of calking compound so that the weir could be removed.

The hook gage was installed in the channel 42 inches above the weir, a distance sufficient to eliminate the effect of downward curvature of the water surface at the weir. A stilling basin for the hook gage, shown in Plate X, was made by placing a four-inch perforated well casing inside an eight-inch well casing and back filling the area between the two casings with Lincoln sandstone ranging in size from 3/8 to 1-inch diameter.

Turbulent flow in the channel caused by jet streams issuing from the manifold was controlled by means of a baffle installed upstream from the hook gage. Two rectangular frames made from 1 by 1/8-inch angle iron were welded together with a height and length equal to the inside dimensions of the channel. These two separate frames were welded together with angle irons providing an inside space of 8½ inches. The frame was then lined with ¼-inch hardware screen, placed in the channel and filled with crushed rock. This baffle is shown in Plate X.

It was necessary to place a dam in the channel as shown in Fig. 18 in order to force all the flow in the channel over the weir. Aluminum angles were bolted to the wall with anchor bolts and a sheet of 3/4-inch water resistant plywood secured to these angles completed this dam. A watertight seal was effected with the use of caulking compound.

The locating and drilling of the orifices in six-inch aluminum pipe was accomplished by the following methods. Two 2 by 4-inch boards were bolted together at right angles to form an L-shape frame. Two such frames were made and the pipe then placed on them. A small block was nailed to

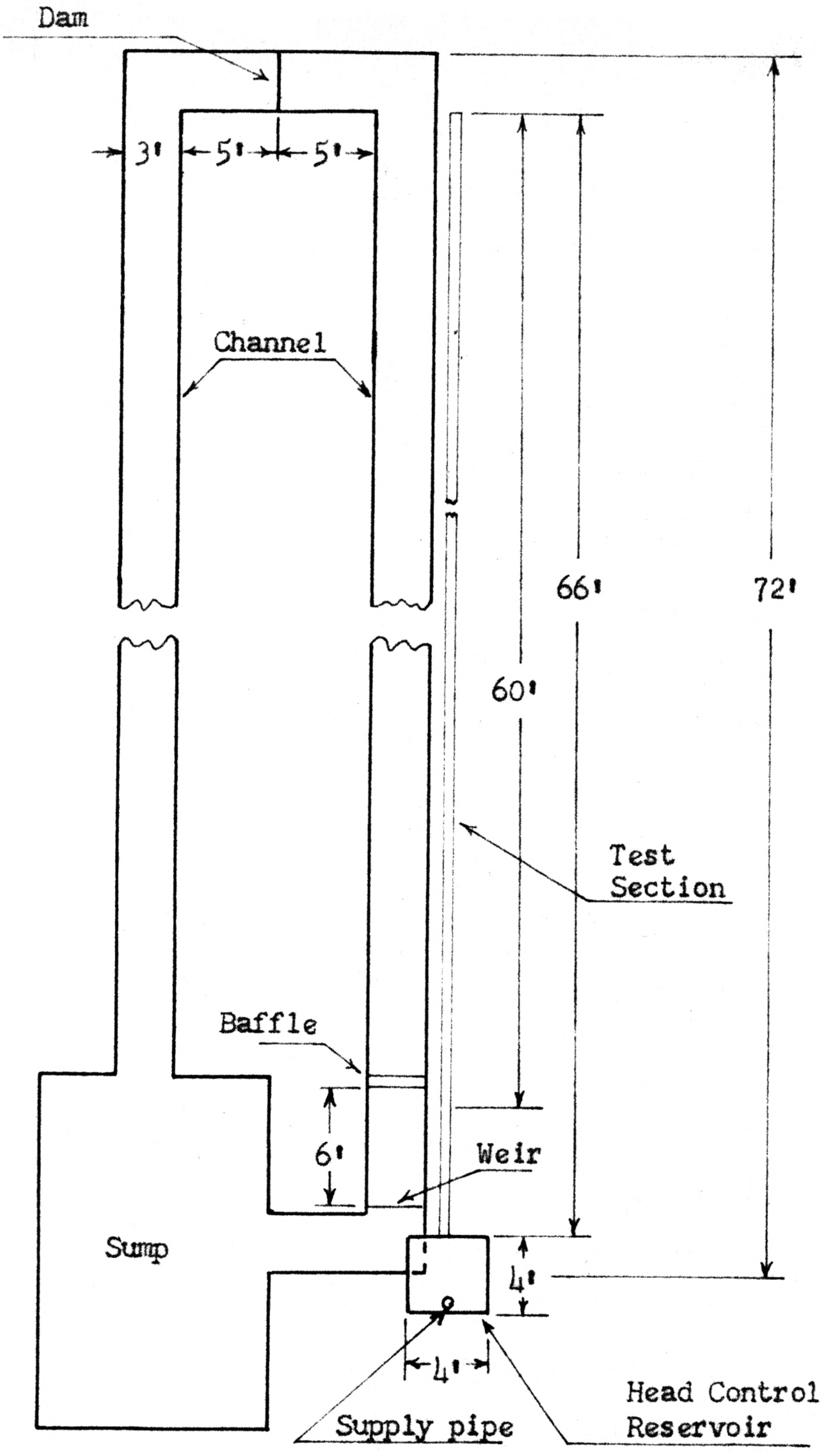

Fig. 18. Schematic diagram of equipment used for this study.

the horizontal 2 by 4-inch piece to hold the pipe firmly against the vertical 2 by 4-inch board. The horizontal location of each orifice was determined by measuring from one end of the pipe. The center of the first orifice was 20 inches from the end of the pipe. The center-to-center distance of 40 inches for the remaining openings were marked leaving a 20-inch spacing from the center of the last hole to the end of the pipe. The second right angle frame was then placed at the point of the first orifice and the center of the orifice located at the mid-point of the pipe by measuring normal to the horizontal board a vertical distance equal to the radius of the pipe as shown in Fig. 19. The center for each orifice as measured was then located and center punched. This procedure was followed for locating orifices in both the 40-foot and 20-foot lengths of pipe. The right angle clamped to the one end of the pipe was left in place while the orifices were drilled. The pipe was placed in a drill press and both ends supported by floor stands to hold the pipe level. A five foot spirit level was used to determine when the pipe was level. Another level was placed against the frame at the end of the pipe to adjust the pipe so the twist drill would enter the pipe normal to a horizontal line tangent to the center of the orifice. The pipe was then held with a drill vise and the hole drilled in the pipe after the drill had been centered in the depression left by the center punch. All holes were drilled by the above procedure. To obtain the highest possible uniformity each hole was drilled undersize 1/64 of an inch and then reamed with a machine reamer to a diameter of 3/4-inch. The same procedure for alignment was followed during the reaming process as outlined above.

EXPLANATION OF PLATE X

Shown at the left is the hook gage and stilling basin. Beyond the hook gage, a baffle to reduce the effect of turbulence in the channel above the weir can be seen. Overflow of head control reservoir may be noted at the right.

PLATE X

Burrs of aluminum were found around the inside edges of the pipe after the orifices had been drilled and reamed. These burrs were removed by a 5/8-inch diameter disk on a shank $\frac{1}{4}$-inch diameter and $1\frac{1}{2}$-inches long. A mixture of epoxy resin and silicon carbide was applied to the disk on the shank side to provide a fine grind stone.

A cap made from $\frac{1}{4}$-inch aluminum was welded over the dead end of the test section of pipe to provide a watertight end plug which could also be tapped for a manometer connection. It was essential that the end plug prevent leakage so that all the water flowing in the pipe would be discharged through the orifices in the pipe assuring that flow conditions in the manifold would be comparable to the final lengths of gated pipe under field conditions. The end plug was tapped below a horizontal line in the same plane as the bottom edge of the orifices so that it would be possible to obtain a zero reading of the manometer with reference to the lower edge of all orifices when the pipe was level.

A manometer tap was also provided at the beginning of the 60-foot test section by building up the wall thickness of the pipe by welding a flat circular nodule on the pipe. This was drilled and tapped for 1/8-inch pipe. All irregularities caused by welding, drilling and tapping were removed with a portable power tool equipped with a resin bonded wheel.

When the 3/4-inch orifices were drilled out to 1-inch diameter a third manometer tap was installed at the end of the 40-foot section of the test pipe.

Seven wooden frames were made to support the pipe at an elevation sufficient to allow the weighing tank mounted on scales to be moved under each orifice. Each frame consisted of two 2 by 4-inch uprights with a

2 by 4-inch horizontal piece bolted to each upright. At the base of each upright a 2 by 4-inch board one foot long was bolted to each side providing better stability. The pipe was then placed on the horizontal cross piece of each frame as shown in Plate XIII.

A five-foot section of plain pipe was located between the combined 40-foot and 20-foot test sections and the reservoir. The five-foot section of pipe was used so that the first two orifices would not discharge in the stilling basin area ahead of the triangular weir in the channel. The jets issuing from the orifices would have caused a turbulence in the stilling basin and made it impossible to achieve accurate readings on the hook gage.

This study was designed to deal primarily with the flow characteristics of orifices in six-inch aluminum pipe. It was therefore essential that leakage and friction losses resulting from pipe couplers be eliminated. A butt joint sealed with plastic tape and re-enforced with a split metal sleeve was used to accomplish these requirements. A butt joint was used to fasten the five-foot section to the discharge pipe installed in the head control tank and the joint taped with two-inch plastic tape. A metal split sleeve which fit securely over the joint was fastened in place by two bolts clamping the two pipes together. A flexible connection was desired between the 60-foot test section of pipe and the five-foot section which completed the connection to the tank. This flexible connection was made by slipping a section of inner-tube over the butt joint and clamping the tube on both sides of the joint with pipe strap. A flexible connection was provided so that the slope of the test section could be varied when desired. Water-proof cement was used to obtain a watertight seal on both sides of the joint. A 20-foot section and a 40-foot section of six-inch aluminum pipe were joined to form a 60-foot test section. These were connected with a butt joint which was taped and clamped with a sleeve. Successful joining of the test section at this point was complicated by the problem of alignment of orifices in both pipes so that the center of all orifices would lie on a straight line. To facilitate alignment a semi-circle was cut out of a sheet of ½-inch plywood at an angle of 45 degrees as shown in Fig. 20. The distance around the circumference of the pipe from top center to the top of the orifice was calculated. This distance varied with the orifice size and angle of discharge for the orifices. For one orifice size and discharge angle the length of the arc was constant and could be measured from a point determined by a line drawn parallel to the top of plywood and tangent to the semi-circle. The distance was measured from this point along the arc and a steel point set in the curved edge of the plywood at this point. The steel point was then hooked against the top of the orifice and a spirit level placed on the top edge of the plywood. The pipe was rotated until the bubble was centered and this section of pipe was clamped to one of the frames supporting the pipe. This process was repeated for the other section of pipe.

The test section was leveled by adjusting the supporting frames. The horizontal supporting 2 by 4-inch board could be raised and lowered by means of a slot provided for this purpose in the vertical uprights as shown in Plate XIII. The adjustments necessary were determined and checked with a surveyor's level.

A three-foot manometer was installed on a wooden frame at the dead end of the pipe and connected with rubber tubing to the 1/8-inch pipe

EXPLANATION OF PLATE XI

Fig. 19. Method for locating center of orifices is illustrated.

PLATE XI

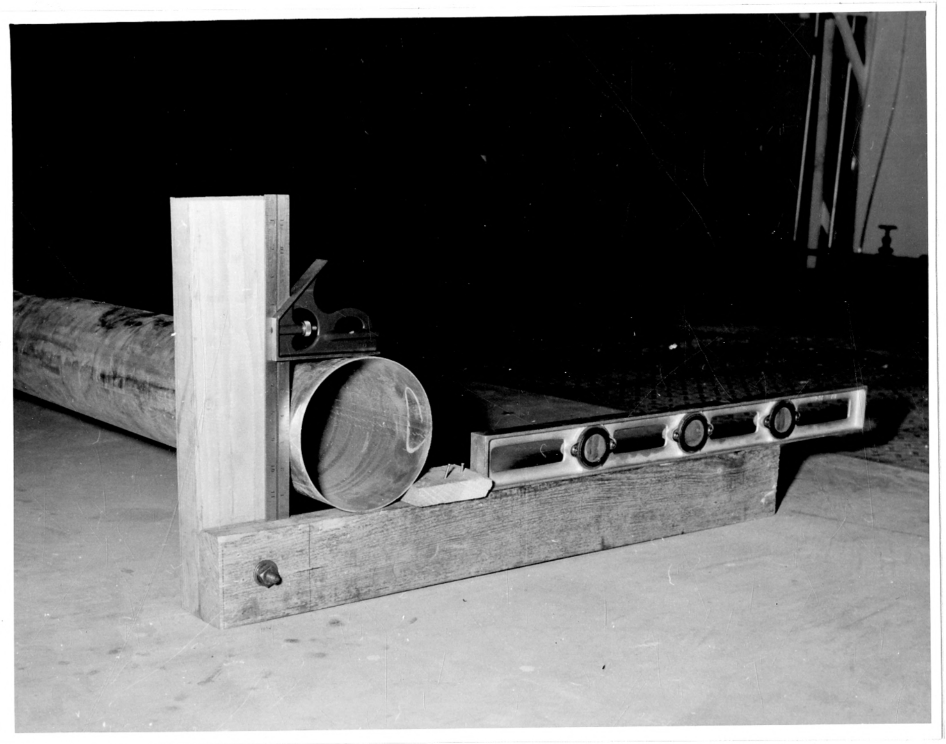

Fig. 19

EXPLANATION OF PLATE XII

Fig. 20. Alignment of the two sections of pipe used to provide a 60 foot test section.

PLATE XII

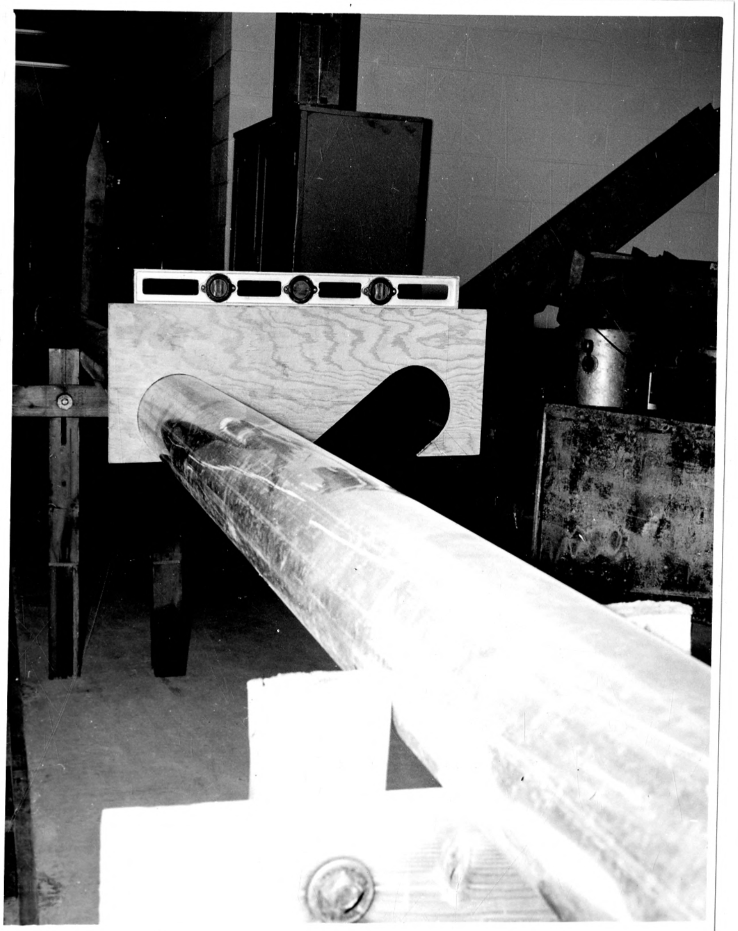

Fig. 20

EXPLANATION OF PLATE XIII

Equipment assembled for determining the flow of water from orifices in a manifold.

PLATE XIII

which was threaded in the end cap. This completed the construction of equipment and calibration necessary for this experiment.

Procedure For Measuring Flow From Orifices

The water level in the head control reservoir was raised until full flow in the discharge pipe existed. The water level was held at this level long enough to allow the air in the pipe to escape. Increasing the head slowly until full flow in the pipe was attained permitted the remaining entrapped air in the pipe to escape.

The water level in the head control tank was then adjusted by adding or removing boards in the opening and by regulating the pump discharge. The head in the tank was determined by reading the manometer mounted on the outside of the tank. The pump discharge was regulated to provide a small flow over the last board in the opening of the tank, which was returned directly to the sump. This small overflow made it possible to visually detect any serious variation in pump discharge. The system was then allowed to run until steady flow conditions were achieved in the channel.

Data recorded during each head determination were: (1) hook gage reading, (2) manometer reading for head above the bottom of orifices in the head control tank, (3) manometer reading at a point 10 inches upstream from the first orifice, (4) manometer reading at the dead end, (5) discharge of the orifices measured by weight. Three observations of the discharge were taken at each orifice. After half of the orifices had been checked, manometer and hook gage readings were recorded. The discharge of the remaining orifices was checked and the test was concluded by making a final reading of the manometers and hook gage. This same procedure was followed for all tests with the head on the orifices the only variable.

Initially the orifices were set to discharge normal to a vertical plane running through the center of the pipe parallel to the length of pipe; but when a head of two feet was reached the discharge from the orifices overshot the channel. To prevent this overshot the center of the orifices was rotated 45 degrees downward from normal position as described above in order to allow the discharge at all heads to fall into the channel. Plate XIII shows the orifices discharging in this position and Fig. 21 shows orifices discharging normal to the pipe.

The angle of the discharging water was measured with a sliding T bevel at all orifices for heads ranging from 0.503 to 4.688 feet. Fig. 22 shows the angle of discharge of the jet at orifice number 18. This angle was measured at all orifices and these data were recorded in Table 8.

RESULTS

Results Of Calibration Of Triangular Weir

The results for calibration of the triangular weir are shown in Table 1 which includes the data for all three methods used to determine the discharge over the weir. At the lower flows it was not possible to obtain readings from the Venturi and Measure-Rite flow meters; consequently the values for these rates of flow had to be determined by weighing a specified amount of water and the time required to accumulate this weight was recorded. From this information the rate of flow was calculated in g.p.m. The rate of flow was recorded from the Venturi and the Measure-Rite flow meters when

EXPLANATION OF PLATE XIV

Fig. 21. Discharge of orifices normal to a longitudinal vertical plane at the center of the pipe.

PLATE XIV

Fig. 21

EXPLANATION OF PLATE XV

Fig. 22. Angle of discharge of the jet from the orifice next to the inlet end.

PLATE XV

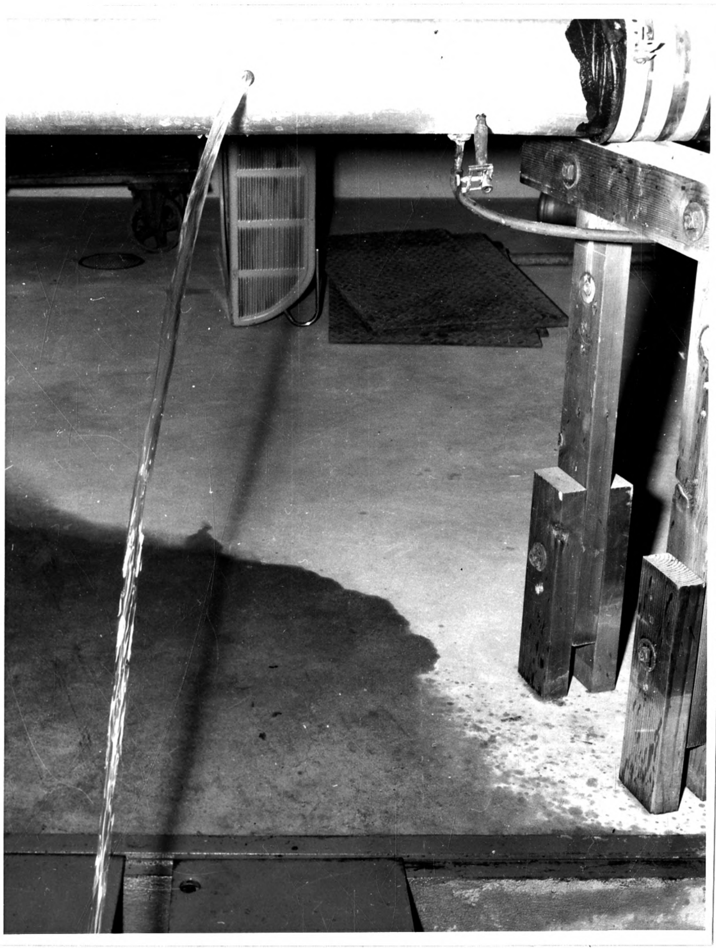

Fig. 22

flows of sufficient magnitude were reached.

Examination of the data in Table 1 revealed that there was close agreement in the rate of flow as determined by weighing and with the Venturi meter. The discrepancy in the results by these two methods ranged from 0.4 to 4.16 percent based on the assumption that the weighing method was accurate. The Venturi meter usually recorded greater flows than were found by weighing. When the rate of discharge reached 400 g.p.m. the capacity of the weighing tank became a limiting factor; therefore its use was discontinued for flows over 407.78 g.p.m. because it became more and more difficult to achieve dependable results. At these larger flows it was necessary to take short time periods and reduce the weight of water retained in the weighing tank to prevent overflow of the tank during the test. A shorter time period tended to introduce an additional variable of time measurement which could be eliminated by the use of other measurement systems. The weir was calibrated for flows greater than 407.78 g.p.m. by means of the Venturi meter and these results are recorded in Table 1.

The results for the Measure-Rite flow meter indicated that it could not be used as an accurate measuring device for calibration; however for flows of 200 g.p.m. or more the results of this flow meter are compatible with the results obtained by the two afore-mentioned methods. Generally, the Measure-Rite flow meter indicated larger flows than the other methods and may have been the result of entrained air in the water flowing through the meter, a problem which could not be eliminated in this experiment.

Since the flows to be measured in this study of discharge from uniformly spaced orifices in a six-inch aluminum pipe were to be of

Table 1 Table of means of observation for calibration of the triangular weir

Test* No.	Discharge lbs. H_2O	Discharge Time sec.	Discharge** g.p.m.	Discharge*** Venturi g.p.m.	Discharge*** Measure-Rite g.p.m.	Hook Gage Head on Weir feet
1	100	106.00	6.78	---	---	0.126
2	100	40.60	17.75	---	---	0.190
3	100	23.60	30.40	---	---	0.234
4	200	31.90	45.09	---	---	0.274
5	500	54.90	65.50	---	---	0.315
6	500	49.10	73.24	---	---	0.332
7	500	40.50	88.70	89.76	---	0.354
8	700	44.40	113.46	116.69	---	0.394
9	600	32.50	132.66	134.64	---	0.416
10	700	32.70	154.65	157.08	100.00	0.446
11	700	28.30	177.73	179.52	150.00	0.470
12	700	24.40	206.09	201.96	210.00	0.505
13	700	23.00	219.37	224.40	220.00	0.513
14	600	18.70	230.14	224.40	240.00	0.520
15	600	17.00	254.17	251.33	260.00	0.548
16	600	16.20	267.13	269.28	275.00	0.556
17	600	15.90	272.51	269.28	285.00	0.558
18	600	15.20	284.63	291.72	305.00	0.577
19	600	13.70	315.51	314.16	330.00	0.589
20	550	11.90	332.47	336.60	360.00	0.610
21	500	10.10	357.43	359.04	365.00	0.616
22	500	9.50	380.13	385.97	390.00	0.639
23	450	8.20	393.78	403.92	405.00	0.640
24	350	7.10	407.78	426.36	430.00	0.664

* Each test mean computed from five observations
** Measurements not applicable at large flows
*** Measurements not applicable at small flows

Table 1 (cont) Table of means of observations for calibration of the triangular weir

Test* No.	Discharge lbs. H_2O	Discharge Time sec.	Discharge** g.p.m.	Discharge*** Venturi g.p.m.	Discharge*** Measure-Rite g.p.m.	Hook Gage Head on Weir feet
25	——	——	——	471.24	480.00	0.676
26	——	——	——	493.68	500.00	0.688
27	——	——	——	516.12	520.00	0.701
28	——	——	——	538.56	540.00	0.712
29	——	——	——	561.00	565.00	0.735
30	——	——	——	590.00	590.00	0.751

* Each test mean computed from five observations

** Measurements not applicable at large flows

*** Measurements not applicable at small flows magnitudes of 400 g.p.m. or less, the results for flow over the weir determined by weighing were used to calculate the constants for the discharge equation for a 90 degree triangular weir. The discharge for the weir was plotted as the ordinate and corresponding values of head were plotted as the abscissa. The resulting curve is shown in Fig. 23. The exact values for the equation for discharge over a triangular weir were determined by the use of curvilinear regression (21), a standard statistical technique. The values and procedure for these computations are shown in the Appendix and resulted in the following equation for the triangular weir:

$$Q = 1154.4 \quad H^{2.49} \qquad (26)$$

Where:

Q = discharge in g.p.m.

H = head in feet of water above the vertex of the weir

The weir was then used in subsequent tests to measure the total discharge of all orifices and equation 26 was used to compute the flow in g.p.m.

Results Of Flow From Orifices

Flow from the uniformly spaced openings fulfilled the requirements for orifice flow with the discharging flow touching only the inside edge of the opening. The results of the first seven tests are tabulated in Table 2. It is evident that the flow from all the orifices is not identical. The maximum difference in flow for test one was 0.27 of a gallon and represented 6.9 percent of the mean discharge of all the orifices at an inlet head

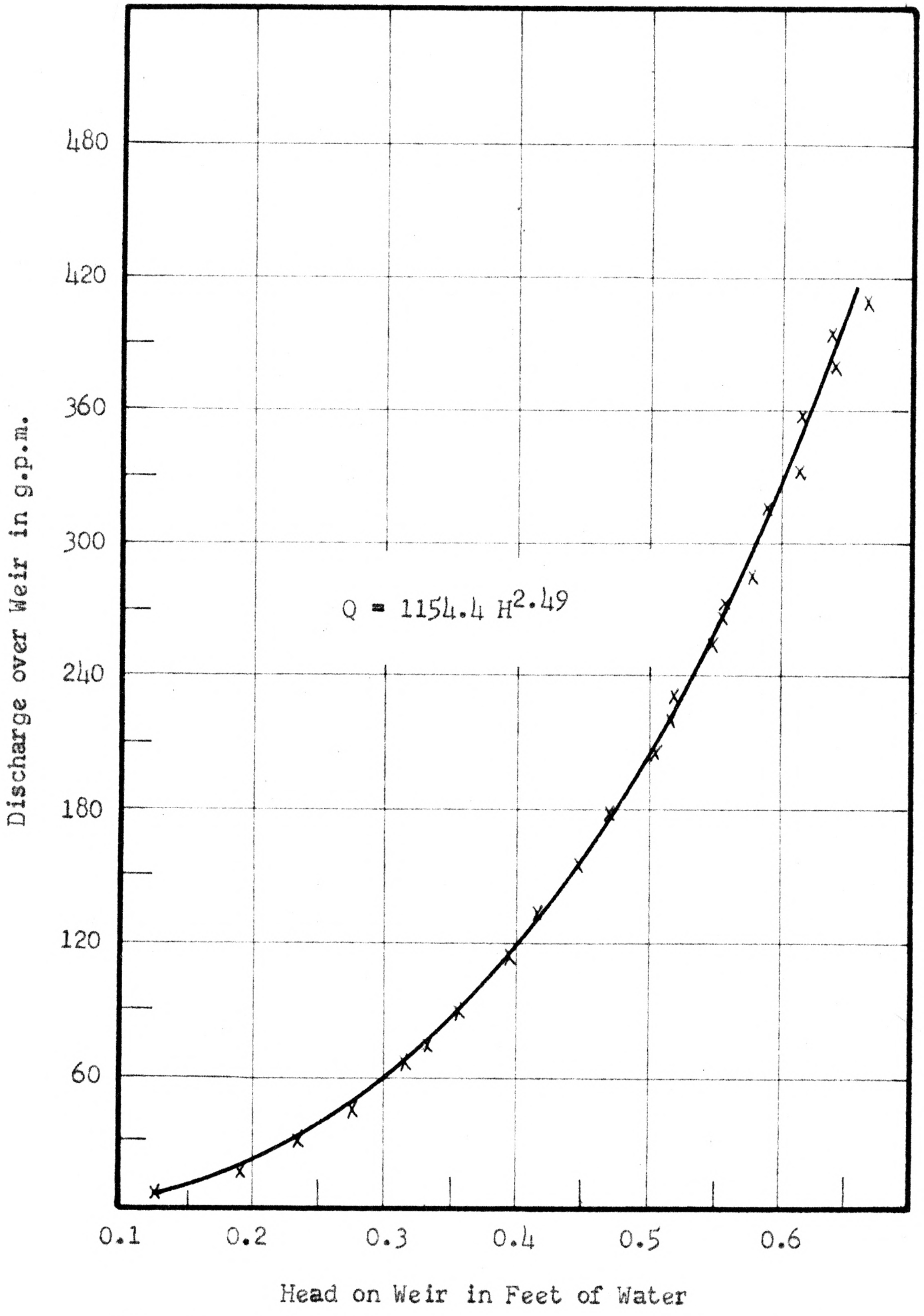

Fig. 23. Calibration curve for the discharge of the triangular weir.

of 0.264 feet. The maximum variation of flow from the orifices varied with varying head and for test two the variation represented 3.2 percent of the mean of all orifices; test three, 6.0 percent; test four, 3.3 percent; test five, 5.5 percent; test six, 6.6 percent and test seven, 5.4 percent. In Fig. 24 the discharge for each orifice was plotted against orifice position with the head held constant for each test. These curves indicate the differences of flow for each orifice and also show that for these seven tests the same pattern exists with a trend to increased flow at the dead end and maximum discharge occurred at the third orifice from the dead end. The minimum discharge occurred near the inlet end and at orifices 16 and 17 numbered from the dead end.

Errors of five percent or less have generally been accepted as the practical accuracy obtainable with water measuring equipment of the type used in this experiment. However, a difference of 0.27 g.p.m. between the maximum and minimum discharge of the gates would represent a difference in discharge of 16 gallons per hour. For a commonly used irrigation set of 11 hours the difference in flow would amount to 178 gallons and in 23 hours it would increase to 373 gallons. Therefore, it cannot be assumed that equal flow existed for test number 1 in Table 2. The difference between the maximum and minumum flows for the remaining six tests in Table 2 were: 0.32 g.p.m. for test 2, 0.37 g.p.m. for test 3, 0.23 g.p.m. for test 4, 0.53 g.p.m. for test 5, 0.71 g.p.m. for test 6, and 0.82 g.p.m. for test 7. Following the above reasoning it is apparent that the difference in flow would be a real value not to be ignored. The data for the remaining tests has been presented in Tables 4,5 and 6. Although some of the variation in discharge may be due to unequal

Table 2 Table of means of observations for three-quarter-inch diameter orifices placed on a 45-degree angle below a longitudinal horizontal plane at the center of a level pipe

Head at reservoir	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7
	0.283	0.563	0.713	1.008	1.998	3.548	4.945
Head H_L*	0.264	0.549	0.695	0.978	1.934	3.467	4.832
Head h_u**	0.253	0.528	0.668	0.943	1.866	3.350	4.670
Head h_o***	0.257	0.528	0.674	0.949	1.893	3.391	4.734
V****	0.83	1.15	1.33	1.51	2.09	2.75	3.23
Total Q*****	73.22	101.06	117.15	131.55	183.71	242.06	284.75
Orifice	Means of discharge for each orifice in g.p.m. (Three observations)						
1	3.89	5.32	5.91	6.88	9.49	12.72	14.94
2	3.89	5.30	5.86	6.86	9.77	12.62	14.82
3	4.06	5.58	6.21	7.19	9.99	13.27	15.64
4	4.00	5.53	6.15	7.11	9.80	12.95	15.61
5	3.93	5.41	6.03	7.02	9.80	12.95	15.40
6	3.89	5.34	5.98	6.87	9.62	12.86	15.18
7	3.87	5.34	5.95	6.85	9.62	12.80	15.06
8	3.88	5.34	5.97	6.86	9.62	12.76	14.98
9	3.89	5.33	5.90	6.84	9.46	12.70	14.98
10	3.92	5.38	5.99	6.90	9.65	12.86	15.06
11	3.91	5.33	5.97	6.76	9.56	12.94	14.96
12	3.89	5.33	5.94	6.85	9.65	12.76	15.01
13	3.90	5.37	5.97	6.85	9.69	12.80	15.06
14	3.91	5.35	5.94	6.83	9.61	12.68	15.08
15	3.85	5.33	5.92	6.81	9.59	12.70	14.91
16	3.82	5.29	5.90	6.76	9.49	12.76	14.86
17	3.79	5.26	5.84	6.80	9.46	12.56	14.86
18	3.87	5.34	5.94	6.84	9.56	12.68	14.94
Total	70.16	95.47	107.37	123.88	173.43	230.37	271.35
Mean	3.90	5.30	5.97	6.88	9.64	12.80	15.08

* H_L is the total energy head in feet ten inches upstream from orifice No. 18

** h_u is the static head ten inches upstream from orifice No. 18

*** h_o is the static head at the dead end in feet

**** V is the velocity in feet per second upstream from orifice No. 18

***** Q is the total flow measured with the triangular weir in g.p.m.

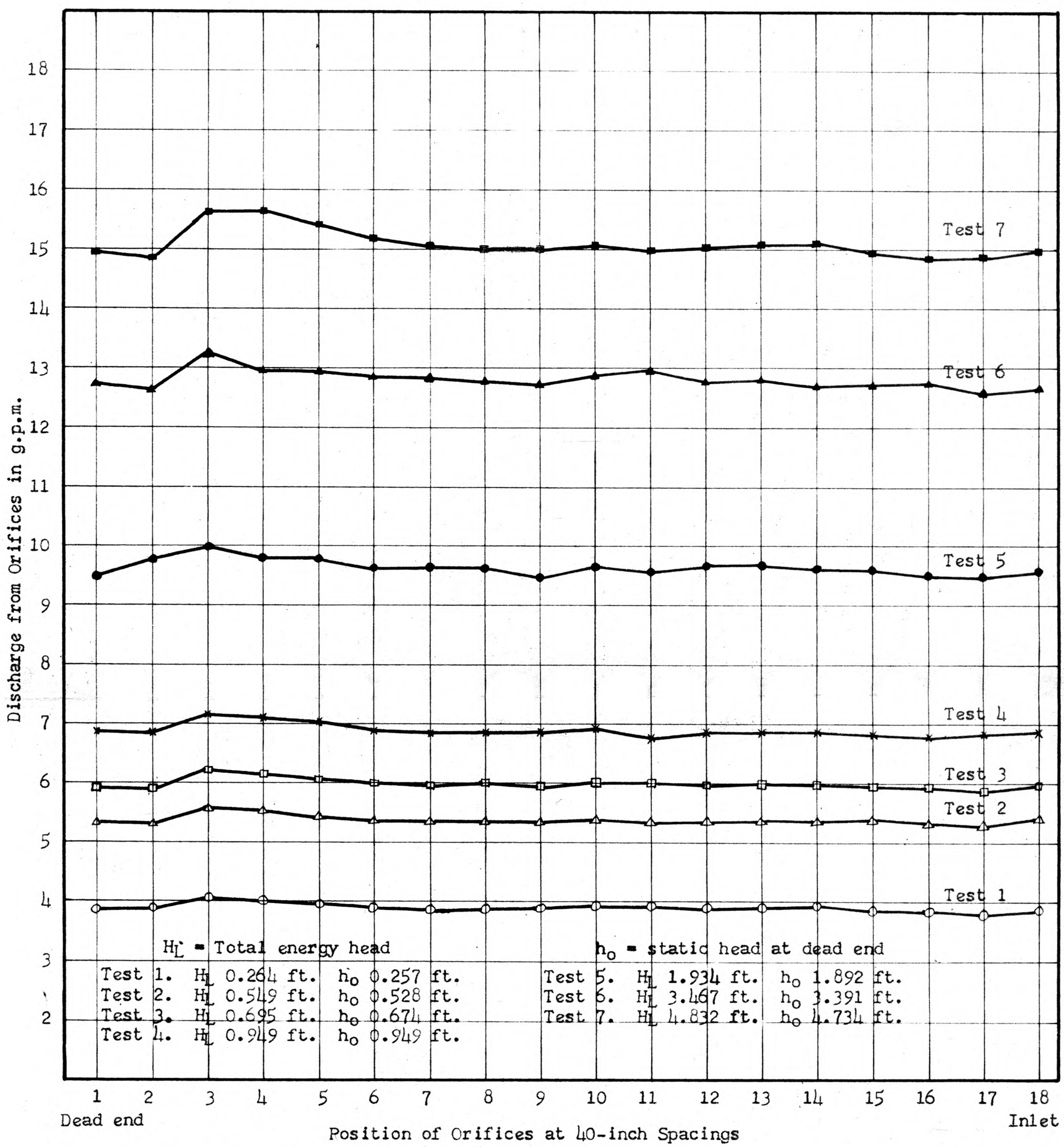

Fig. 24. Discharge of 3/4-inch diameter orifices placed on a 45° angle below a horizontal longitudinal plane at the center of a level pipe.

cross-sectional area of the orifices it is doubtful that it accounts for all the variance in discharge. In order to determine whether this was a factor in this experiment the horizontal and vertical diameters of each orifice were measured with an inside micrometer to determine whether the differences in discharge from the orifices were a result of variation in orifice area. The results of these measurements are shown in Table 3.

The trend toward increased discharge at the dead end was the result of increased static pressure. The conversion of velocity head to pressure head exceeded the losses due to friction in this experiment, resulting in a greater static pressure at the dead end than was observed at a point ten inches upstream from orifice number 18. Although the total energy head H_L at a point ten inches upstream from orifice number 18 is greater than the static head h_o at the dead end it may be noted that the static head h_u, ten inches upstream from orifice number 18 is less than the static head h_o in Table 2. This condition existed at all heads and for all tests for a level pipe.

A two-way analysis of variance (21) revealed a significant difference at the one percent level between the discharges from all the orifices. The error term in these calculations was 0.0685 which resulted in a significant difference between observations at the one percent level for all tests analyzed except tests number one in Table 2. All tests for this study followed the same trend toward greater flows at the dead end; all the tests retained the same general pattern for both orifice sizes; therefore it was not considered necessary to repeat an analysis of variance for each test.

An orthogonal comparison (21) was made for the means for test four

Table 3. Table of horizontal and vertical measurements in inches of the diameter of orifices in a 6-inch aluminum irrigation pipe numbered from the dead end.

Orifice	Horizontal*	Vertical*	Horizontal**	Vertical**
1	.751	.752	1.000	1.001
2	.752	.753	1.002	1.000
3	.750	.754	1.002	1.000
4	.750	.750	1.000	1.003
5	.751	.751	1.001	1.004
6	.754	.754	0.999	1.002
7	.754	.754	1.000	1.001
8	.748	.748	0.999	1.001
9	.751	.751	1.004	1.001
10	.753	.753	1.001	1.001
11	.753	.753	1.002	1.003
12	.753	.753	1.001	1.001
13	.757	.757	0.998	1.001
14	.753	.753	1.000	1.002
15	.754	.754	1.001	1.002
16	.752	.752	1.000	1.001
17	.751	.751	1.002	0.997
18	.754	.754	1.002	1.004

* Diameter measurements made with an inside micrometer for 3/4-inch orifices
** Diameter measurements made with an inside micrometer for 1-inch orifices

in Table 2 in order to fit a curve that would best describe the data. These comparisons were carried out to a fifth order polynomial and show that a linear relationship fit 39 percent of the data, a quadratic curve described 0.5 percent of the data, a cubic curve would fit 9.5 percent of the data, a quartic relationship fit 11.4 percent of the data and a Quintic curve would fit 12.8 percent of the data. This left 26.8 percent of the data to be described by higher order polynomials. The F test (21) indicated that all the relationships were significant at the one percent level except the quadratic curve and it was significant at the five percent level. Further statistical analysis appeared to be unnecessary since there was a marked consistancy in the data and values for the F test exceeded tabular values at the one and five percent levels by large margins.

The effect of orifices discharging at a 45-degree angle below a longitudinal horizontal plane at the center of the pipe was compared to orifices discharging normal to a longitudinal vertical plane at the center of the pipe. As previously discussed it was possible to run these tests only for heads less than one foot of water because of the overshooting of the channel by the jet from each orifice. The data for two such tests are shown in Table 4 and are represented graphically in Fig. 25. Comparison of tests one and two in Fig. 25 with tests two and three in Fig. 24 indicate little difference in trends or pattern. The small difference in discharge per orifice was accounted for by the difference in head between the tests. Tests for discharge in a normal position could not be considered conclusive but do indicate that discharge is not significantly influenced by the two angles at which the orifices were placed for these tests except as the head on the orifices was increased or decreased by the rotation of the pipe to change the orifice discharge angle.

It was decided to use the slope Hansen (7) recommended of 1 in 300 to achieve equal flow from gates in gated irrigation pipe. For the first trial the pipe was placed on a rising slope of 1 in 300 with the dead end high and the 3/4-inch orifices discharging normal to a longitudinal vertical plane at the center of the pipe. This slope was achieved by raising each supporting frame the desired amount with small blocks. This was an approximate setting of slope and the purpose of these tests was to ascertain whether it would be possible to reverse the trend toward higher flows at the dead end. The results of these tests is shown in Table 5

Table 4 Table of means for three-quarter-inch orifices placed normal to a longitudinal vertical plane at the center of a level six-inch aluminum pipe

	Test 1	Test 2
Head at reservoir	0.503	0.733
Head H_L*	0.486	0.713
Head h_u**	0.468	0.688
Head h_o***	0.482	0.695
V****	1.08	1.28
Q*****	95.15	112.82

Orifice	Means of discharge for each orifice in g.p.m. (Three observations)	
1	5.02	5.95
2	4.99	5.96
3	5.24	6.25
4	5.20	6.19
5	5.09	6.09
6	5.01	5.97
7	5.01	5.97
8	5.04	6.05
9	5.00	5.97
10	5.06	6.03
11	5.02	5.96
12	5.01	5.94
13	5.05	5.99
14	5.04	6.00
15	5.06	5.97
16	5.02	5.97
17	4.91	5.88
18	5.00	5.94
Total	90.77	108.08
Mean	5.04	6.00

* H_L is the total energy head in feet ten inches upstream from orifice No. 18

** h_u is the static head ten inches upstream from orifice No. 18

*** h_o is the static head at the dead end in feet

**** V is the velocity in feet per second upstream from orifice No. 18

***** Q is the total flow measured with the triangular weir in g.p.m.

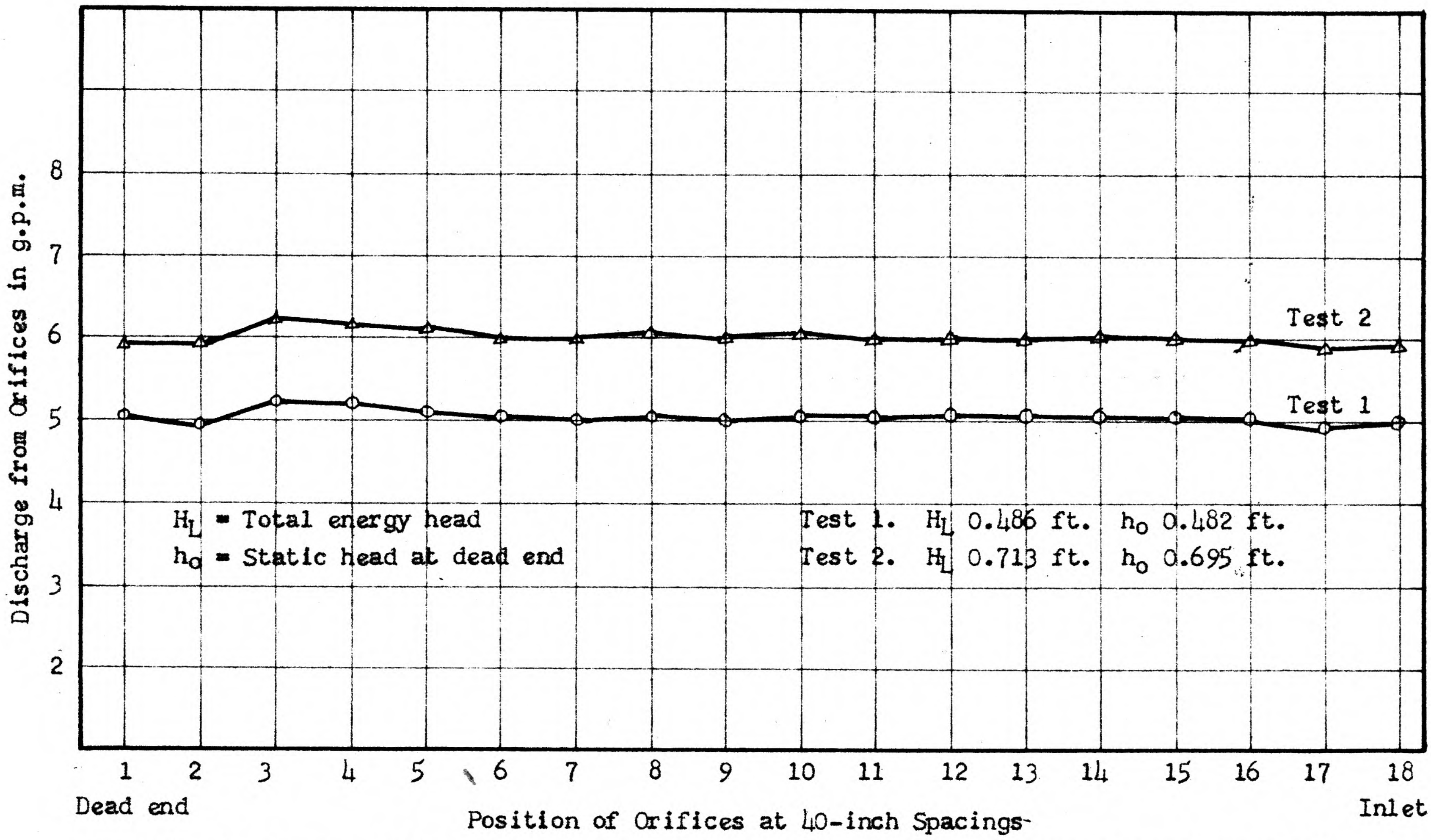

Fig. 25. Discharge of 3/4-inch diameter orifices placed normal to a longitudinal vertical plane at the center of a pipe.

as tests one and two and are graphically represented in Fig. 26. This change in slope did reverse the trend with a greater flow per orifice at the inlet than at the dead end; however the basic pattern still persisted when these curves were compared to earlier test results.

The pipe was then placed on a falling slope of 1 in 300 as Hansen (7) had used to achieve equal flow. These results are shown as test three and four in Table 5 and Fig. 26. This slope resulted in an increase in the static head at the dead end and a corresponding increase in discharge per orifice at the dead end. The same pattern was present in these tests.

The effect of increasing the orifice diameter on the discharge from the orifices was next considered. Increasing the orifice diameter to one inch increased the discharge per orifice resulting in larger flows in the pipe when compared to 3/4-inch diameter orifices. With larger flows, it was desired to study the conversion of velocity head to pressure head and whether this conversion would increase the discharge from orifices near the dead end. The results of tests for one-inch orifices are shown in Table 6 and in Fig. 27. Tests one, two and three indicated a fairly uniform increase in discharge per orifice toward the dead end with maximum discharge for an individual orifice occurring at orifice number two. When the total energy head, H_L, was increased to 1.800 and later to 2.798 feet, the discharge from the orifices became erratic even though the same general trends were present. There was still an increase in flow as the dead end was approached. Orifice number one constantly discharged less flow than orifices number two and three, while orifice two produced greater flows than any of the other orifices for the first four tests. The largest

Table 5 Table of single observations for three-quarter-inch orifices placed normal to a longitudinal vertical plane at the center of a six-inch aluminum pipe at slopes indicated

	Test 1	Test 2	Test 3	Test 4
Head at reservoir	0.508	0.733	0.488	0.728
Head H_L*	0.499	0.716	0.473	0.801
Head h_u**	0.483	0.693	0.453	0.673
Head h_o***	0.338	0.558	0.611	0.841
V****	1.00	1.22	1.13	1.34
Q*****	88.20	107.19	99.73	117.89

	Discharge for each orifice in g.p.m. (one observation)			
Orifice	Slope 1 in 300 dead end high		Slope 1 in 300 dead end low	
1	4.24	4.56	5.53	6.47
2	4.27	4.56	5.50	6.40
3	4.56	4.87	5.72	6.70
4	4.57	4.84	5.64	6.59
5	4.53	4.76	5.51	6.45
6	4.51	4.99	5.40	6.33
7	4.67	4.76	5.38	6.25
8	4.60	4.81	5.30	6.22
9	4.60	4.78	5.27	6.19
10	4.67	4.89	5.24	6.22
11	4.62	4.86	5.21	6.25
12	4.76	4.91	5.11	6.07
13	4.87	4.99	5.08	6.07
14	4.90	5.05	5.08	6.10
15	4.94	5.01	5.07	6.12
16	4.99	5.05	4.99	5.97
17	4.93	5.02	4.90	5.90
18	5.02	5.15	4.90	5.92
Total	84.25	87.86	94.83	112.22
Mean	4.68	4.88	5.27	6.23

* H_L is the total energy head in feet ten inches upstream from orifice No. 18

** h_u is the static head ten inches upstream from orifice No. 18

*** h_o is the static head at the dead end in feet

**** V is the velocity in feet per second upstream from orifice No. 18

***** Q is the total flow measured with the triangular weir in g.p.m.

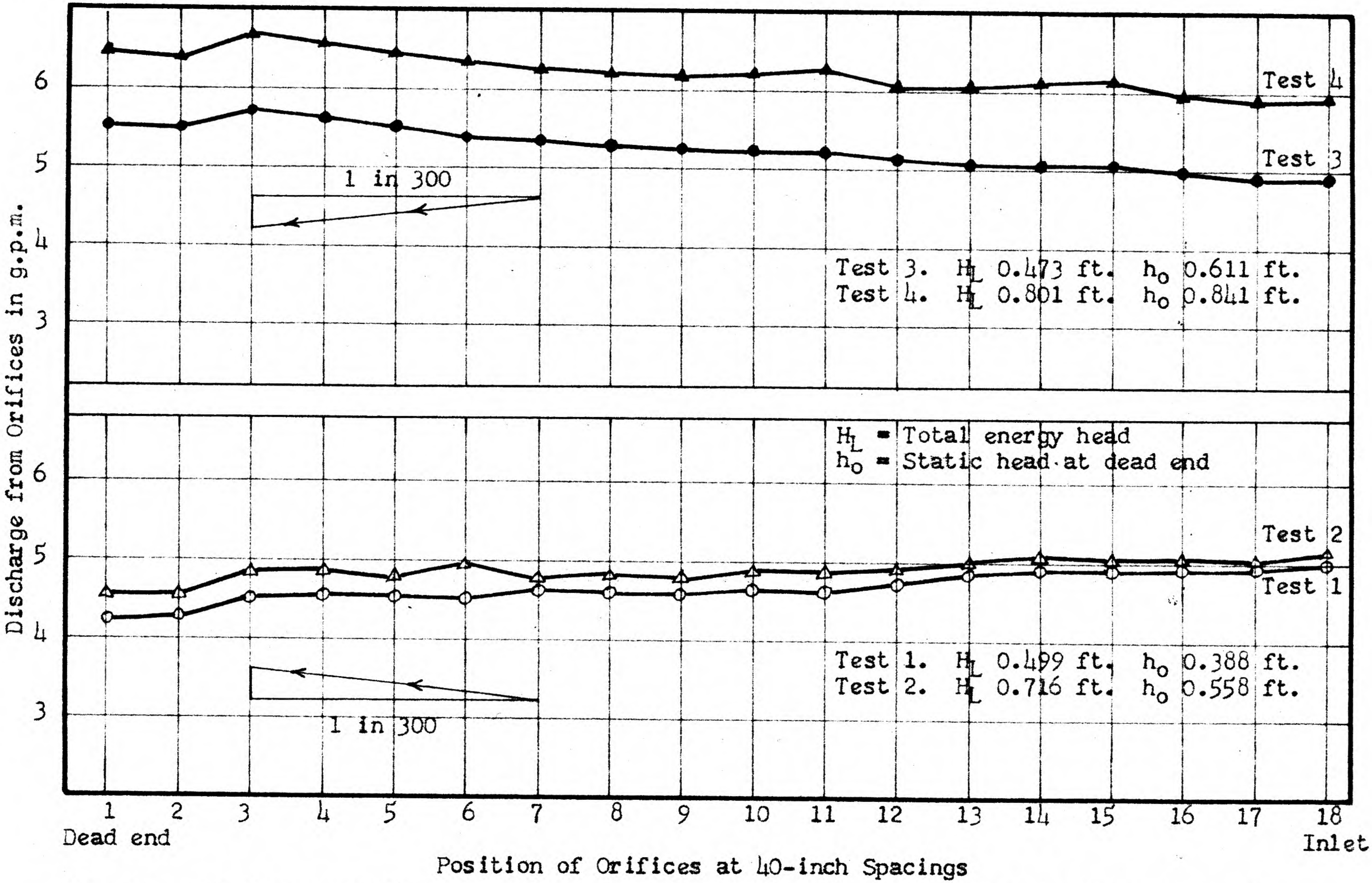

Fig. 26. Discharge of 3/4-inch diameter orifices placed normal to a longitudinal vertical plane at the center of the pipe with pipe slope as indicated.

Table 6 Table of means for one-inch diameter orifices placed on a 45-degree angle with a longitudinal horizontal plane at the center of a level pipe

	Test 1	Test 2	Test 3	Test 4	Test 5
Head at reservoir	0.481	0.726	0.976	1.976	2.996
Head H_L*	0.417	0.618	0.820	1.800	2.798
Head h_u**	0.371	0.551	0.731	1.676	2.528
Head h_o***	0.413	0.613	0.829	1.684	2.584
V****	1.73	2.08	2.39	3.37	4.17
Q*****	152.88	183.71	210.64	297.37	367.80
Orifice	Means of discharge for each orifice in g.p.m. (Three observations)				
1	8.26	9.97	11.52	16.04	19.67
2	8.46	10.13	11.71	16.33	19.98
3	8.38	10.08	11.65	16.31	19.98
4	8.27	9.91	11.53	16.11	19.76
5	8.29	10.01	11.60	16.25	19.93
6	8.18	9.88	11.39	16.05	19.60
7	8.17	9.90	11.47	16.02	19.33
8	8.20	9.89	11.40	15.84	18.82
9	8.34	10.09	11.59	15.89	19.24
10	8.22	9.91	11.36	15.19	19.19
11	8.04	9.74	11.13	14.98	18.99
12	8.08	9.75	11.20	15.25	18.99
13	8.26	9.92	11.38	15.72	20.12
14	8.05	9.63	11.07	15.64	19.76
15	8.07	9.64	10.97	16.11	19.76
16	7.76	9.25	10.49	15.72	19.23
17	7.78	9.33	10.47	15.78	19.35
18	7.74	9.31	10.49	15.89	19.39
Total	146.55	176.34	202.42	285.12	351.09
Mean	8.14	9.80	11.25	15.84	19.50

* H_L is the total energy head in feet ten inches upstream from orifice No. 18

** h_u is the static head ten inches upstream from orifice No. 18

*** h_o is the static head at the dead end in feet

**** V is the velocity in feet per second upstream from orifice No. 18

***** Q is the total flow measured with the triangular weir in g.p.m.

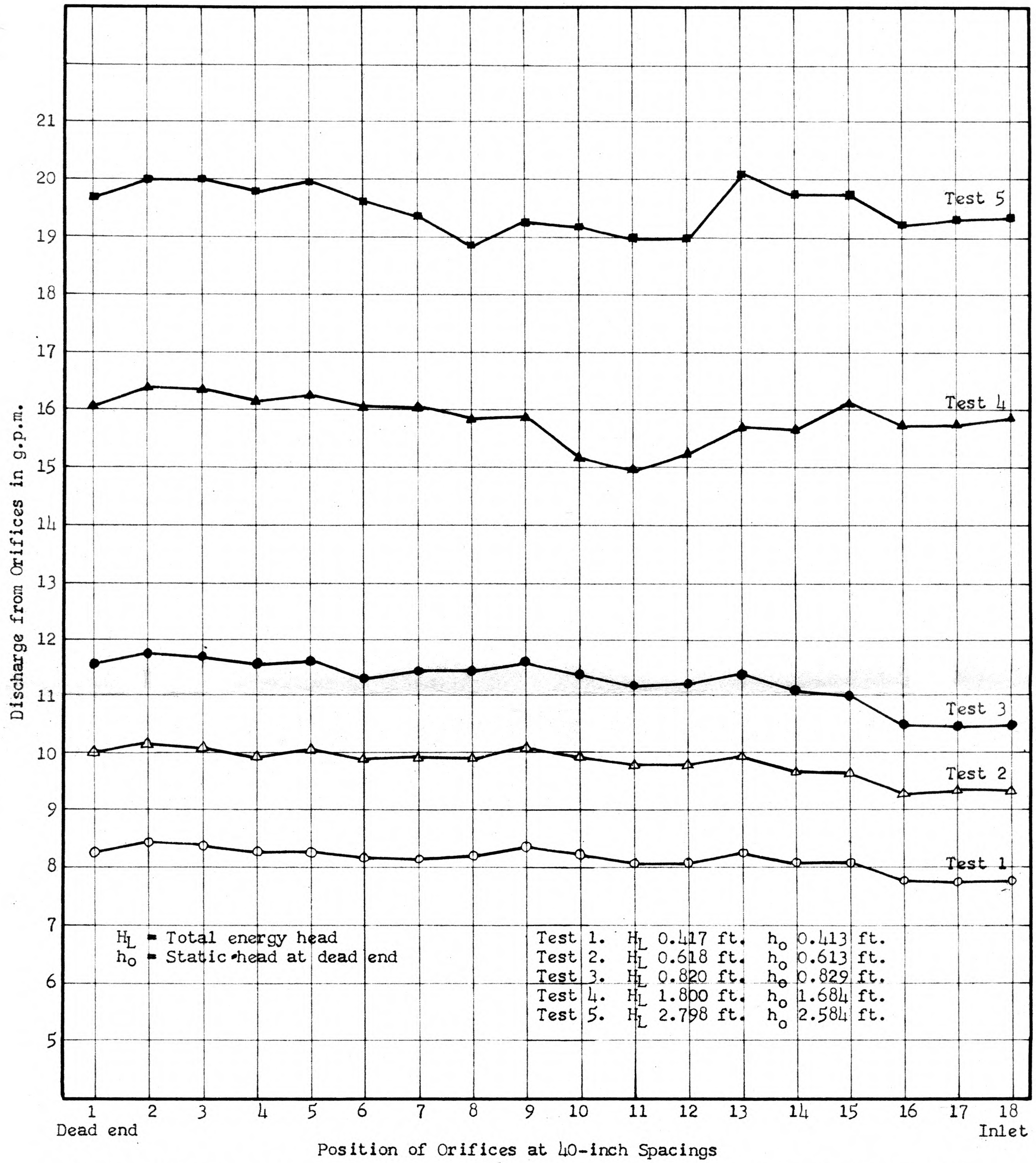

Fig. 27. Discharge of one-inch diameter orifices placed on a 45° angle below a horizontal longitudinal plane at the center of a level pipe.

flow occurred at orifice number 13 in test five and seems to be an unexpected event based on previous results.

At the conclusion of these tests the horizontal and vertical diameters of the orifices were measured with an inside micrometer and these measurements are tabulated in Table 3. The maximum variation of these diameters is 0.007 of an inch and little if any relationship to the rate of discharge is evident.

The obtuse angle of discharge of the jet for each orifice was measured with respect to a plane parallel to the pipe for heads ranging from 0.494 to 4.679 feet for the 3/4-inch orifice diameter. The results of these measurements are given in Table 7 and show that the angle of discharge approaches a perpendicular angle with respect to the pipe as the dead end is approached. The changing angle of discharge from the inlet to the dead end indicates that a constant value cannot be used to multiply the velocity head in order to calculate the conversion to pressure head. In other words an absolute value for such a constant would change as the fluid passes each successive orifice.

Comparing the angle of discharge for each orifice at the five different heads revealed that the angle remains practically constant for any one orifice. The discrepancies which exist may be a result of human error in adjusting the sliding T bevel used for these determinations. The uniformity of the angles for each orifice at all heads indicated that the conversion of velocity head to pressure head as the fluid passes each successive orifice can be expressed as a proportion and remains constant for the range of heads used in this experiment.

Table 7 Angle of jet discharging from all orifices for five values of head with orifice discharge normal to a longitudinal vertical plane at the center of a level pipe.

	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
Head h*	0.494	0.724	1.099	2.079	3.639	4.679
Orifice						
1	91°00'	91°00'	90°30'	91°30'	91°00'	90°30'
2	92°00'	91°00'	92°00'	91°30'	91°00'	92°15'
3	92°35'	93°30'	92°00'	92°00'	91°45'	92°00'
4	93°10'	93°30'	92°30'	93°00'	92°15'	93°00'
5	94°05'	94°30'	93°05'	93°05'	92°30'	93°45'
6	94°45'	94°30'	93°00'	93°00'	92°40'	93°45'
7	95°00'	95°00'	93°40'	93°00'	93°55'	94°30'
8	95°30'	96°00'	94°00'	95°00'	95°00'	94°10'
9	95°40'	95°30'	95°30'	95°30'	95°00'	94°15'
10	96°35'	96°30'	95°15'	96°00'	96°20'	95°30'
11	97°30'	97°00'	96°30'	96°40'	96°30'	96°20'
12	97°30'	97°10'	98°30'	97°00'	96°45'	96°10'
13	97°30'	98°00'	98°00'	97°15'	97°15'	97°45'
14	98°00'	98°20'	97°35'	98°30'	98°15'	99°00'
15	99°00'	98°05'	98°00'	99°00'	99°15'	97°45'
16	99°25'	98°05'	97°55'	98°45'	101°00'	98°45'
17	100°30'	100°00'	99°05'	100°40'	99°30'	98°00'
18	100°15'	99°30'	98°00'	99°50'	99°15'	98°30'

* h is the static head measured at the reservoir with respect to the center of the orifices

SUMMARY AND CONCLUSIONS

Gated irrigation pipe is an irrigation system which provides increased irrigation efficiency. This increased efficiency results from the elimination of losses that occur in an open ditch. Improvement in the design of gated irrigation pipe would permit uniform gate settings for equal flow and aid in reducing run-off losses that occur as a result of unequal flow.

The purpose of this study was to determine whether it is possible to achieve an equal flow of water from orifices in a level pipe by eliminating gates and resulting friction losses inherent in present gated irrigation pipe design.

To study this problem it was necessary to build a head control reservoir which would maintain a constant head for the duration of each test. Baffles were included in the design of the reservoir to provide negligible approach velocity. The discharge of each orifice was determined by weighing a pre-determined weight of water and the time required to catch this weight. Static pressure heads were measured at the reservoir, at the dead end and at a point ten inches upstream from orifice number 18, the orifice next to the pipe inlet. Total flow of all orifices was recorded from the triangular weir.

The first series of seven tests show that a difference exists in the discharge from 3/4-inch orifices placed on a 45-degree angle below a longitudinal horizontal plane at the center of a level pipe. The maximum discharge for one orifice consistently occurred at orifice number 3 from the dead end for these tests. The minimum discharge was at orifices number 16 and 17 near the pipe inlet. A trend toward greater discharge from orifices as the dead end was approached was noted. Measurement of the vertical and horizontal diameters of each orifice did not reveal sufficient variation to account for all the variance in discharge from the orifices.

Discharge from the orifices was not found to be influenced by the two angles used in this study. Discharging normal to a longitudinal vertical plane at the center of the pipe and also at a 45-degree angle below a horizontal plane at the center of the pipe affected discharge of the orifices only as the head was increased or decreased on the center of the orifice.

Experimenting with a rising 1 in 300 slope a decrease in discharge per orifice was noted as the flow approached the dead end. For a falling slope of 1 in 300 an increase in discharge per orifice existed as the dead end was approached.

Increasing the orifice size to one-inch diameter resulted in increased discharge from the orifices as the flow approached the dead end. For heads up to 0.820 feet, this increase was relatively uniform from the pipe inlet to the dead end but a greater variation between individual orifices was present than was observed for 3/4-inch diameter orifices. Higher heads, from 0.820 to 2.798 feet resulted in greater variations between orifices than was noted at the lower heads. It was assumed that an increase in head tends to magnify the variation in discharge between orifices. Highest discharge rate for one orifice occurred at orifice number 2, second orifice from the dead end, in these tests except for the test with a head of 2.798 when orifice number 13 recorded the highest flow.

Measurement of the discharge angle of the fluid with respect to the pipe indicated that the discharge angle for each orifice remained constant within the range of 0.494 to 4.679 feet of head. This indicated that the conversion of velocity head to pressure head remained constant for each section of pipe between adjoining orifices for this range of head.

These tests indicated that it may be possible to design a gate which would provide nearly uniform flow from each gate in a level pipe within a discharge rate of 4 to 15 g.p.m. The major design problem appears to be the friction losses caused by the gates. If these losses can be prevented from exceeding the pressure head regain resulting from the conversion of velocity head to pressure head as the fluid passes each successive gate, then uniform flow throughout the system is possible.

SUGGESTIONS FOR FUTURE RESEARCH

Comparison of the results of the several methods used for the calibration of the triangular weir were in close agreement and a satisfactory accuracy was obtained for these studies. It is the writer's opinion however, that a greater degree of accuracy in measuring flows can be achieved with the weir if it is re-calibrated in place. Flow conditions over the weir were not the same in the channel as they were in the reservoir. Space limitations in the reservoir made it impossible to have a long straight channel above the weir to insure normal velocity distribution and smooth flow whereas these conditions did exist when the weir was installed in the channel. Entrained air and turbulence in the pool above the weir was not completely eliminated in the reservoir during the calibration of the weir.

The problems observed as a part of this study might well be considered as bases for future research applicable to hydraulic characteristics peculiar to the flow of a fluid in a manifold system of the nature considered in this study.

1. Why does the maximum flow occur at orifice two and three?
2. What is the value of the friction factor between each successive orifice?
3. Is there a change from turbulent to laminar flow in the main line and if such a change exists is it present at all practical heads and discharge rates for irrigation?
4. Can the conversion of velocity head to pressure head be expressed as a proportion and does it remain constant for each section of pipe between orifices regardless of head?
5. What is the value of pressure head and velocity in each section of the pipe between orifices?
6. What is the coefficient of discharge for each orifice and does it remain constant for each orifice when the head is varied?
7. What is the effect of other uniform orifice spacings on the discharge?
8. Do the same flow conditions exist for smaller and larger orifice diameters than those used for this experiment?

In any experimental design the experimentor observes some limitations to his instrumentation which, if considered could enable future researchers to improve upon the basic system. The following suggestions are made in critique of these efforts:

1. Install manometer taps at the midpoint of each section between orifices to determine the static head at each of these points.

These data would also enable the experimenter to calculate the value for the friction factor for each section of pipe.

2. Use clear plastic pipe to study flow characteristics in the main line. Turbulence and any change from turbulent to laminar flow in the pipe could readily be detected. Flow lines of the fluid as it passes each orifice could also be observed.

3. Improved method for providing uniform cross-sectional area of orifices. One suggestion would be to secure machined orifices with one-thousandth of an inch tolerance and use an adaptor which would hold different size orifices and could be easily inserted into a large opening in the pipe. Care to prevent excessive friction losses would be necessary in the design of such an adaptor. The use of such an adaptor would make it possible to use the same pipe for testing different orifice sizes and spacings.

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REFERENCES

(1.) Dow, Willard M.
The uniform distribution of a fluid flowing through a perforated pipe. Journal of Applied Mechanics. 17: 431-438. Dec. 1950. In American Society of Mechanical Engineers Transactions. Vol. 72. 1950.

(2.) Edwards, F. W.
Uniformly tapped pipes. Engineering News-Record. 126:78. 1941.

(3.) Ellms, J. W.
Design of perforated pipe strainer system. Journal of American Water Works Association. 18: 664-674. 1927.

(4.) Enger, M. L. and M. I. Levy
Pressures in manifold pipes. Journal of the American Water Works Association. 21: 659-667. 1929.

(5.) Fischbach, Paul E.
Personal Correspondence.

(6.) Gladding, R. D.
Loss of head determinations in uniformly tapped pipes. Engineering News-Record. 115: 697. 1940.

(7.) Hansen, Vaughn E.
Determination of water flow from gated pipe. Agricultural Engineering. 35: 496-7. July 1954.

(8.) Howland, W. E.
Design of perforated pipe for uniformity of discharge. Third Midwestern Conference on Fluid Mechanics. 1953: 687-701. 1953.

(9.) ________.
Gain in head at take-offs. Journal of New England Water Works Association. 49, No. 1: 14. 1935.

(10.) ________.
Uniformly tapped pipes. Engineering News-Record. 126: 79. 1941.

(11.) Kansas State Water Resources Board
Water in Kansas. 53-60. 1955.

(12.) Keller, J. D., Jr.
The manifold problem. Journal of Applied Mechanics. 16: 77-85. March 1949. In American Society of Mechanical Engineers Transactions. Vol. 71. 1949.

(13.) Kohler, Karl D., Jr.
Trends in the utilization of water. U.S.D.A. Yearbook of Agriculture. 35-40. 1955.

(14.) Kunz, Jacob
Jets from manifold tubes. American Society of Mechanical Engineers Transactions. 53: 181-186. 1931.

(15.) Malishevsky, N.
Pressure head in perforated pipes. Journal of the American Water Works Association. 27: 413-418. 1935.

(16.) Markland, E.
Analysis of flow from pipe manifolds. Engineering, January 30, 1959. Vol. 187. No. 4847: 150-151.

(17.) McNown, John S.
Mechanics of manifold flow. American Society of Civil Engineers Proceedings, August 1953. Vol. 79. No. 258: 1-22.

(18.) Oakey, John A.
Hydraulic losses in short tubes determined by experiments. Engineering News-Record. 110: 717-718. 1933.

(19.) Russell, George E.
Hydraulics. 5th ed. New York: Henry Holt and Co. 468. 1948.

(20.) Shove, Gene
Air flow analysis of grain ventilation ducts. Unpublished Ph. D. Thesis, Iowa State College, 1959.

(21.) Snedecor, George W.
Statistical Methods. 5th ed. Ames, Iowa, The Iowa State College Press. 1959.

(22.) Somerhalder, B. R.
Research for hastings gated pipe company. University of Nebraska, Dept. of Agricultural Engineering. 1949.

(23.) Stevens, J. C.
Theoretical energy losses in intersecting pipes. Engineering News-Record. 98: 140-141. 1926.

(24.) Tovey, Rhys and Victor I. Myers
Evaluation of some water control devices. Idaho Agricultural Experiment Station Bulletin. 319: 6-9. Dec. 1959.

(25.) U.S.D.A. Soil Conservation Service
Irrigation on western farms. Information bulletin 199. Washington, D. C. U. S. Government Printing Office. 1959.

APPENDIX

Table 8 Table of means for head on weir in feet and discharge determined by weighing in g.p.m. and corresponding logarithm to the base ten for statistical analysis by curvilinear regression technique

Run No.	Head on weir in feet	Logarithm of head base 10	Algebraic value of logarithm X	Discharge of weir by weighing	Logarithm of Discharge g.p.m. Y
1	0.126	9.10037-10	-0.8996	6.78	0.83123
2	0.190	9.27875-10	-0.7312	17.75	1.24920
3	0.234	9.36922-10	-0.6308	30.40	1.48287
4	0.274	9.43775-10	-0.5622	45.09	1.65408
5	0.315	9.49831-10	-0.5017	65.50	1.81624
6	0.332	9.52114-10	-0.4789	73.24	1.86475
7	0.354	9.54900-10	-0.4510	88.70	1.94792
8	0.394	9.59550-10	-0.4045	113.46	2.05484
9	0.416	9.61909-10	-0.3809	132.66	2.12274
10	0.446	9.64933-10	-0.3507	154.65	2.18935
11	0.470	9.67210-10	-0.3279	177.73	2.24973
12	0.505	9.70329-10	-0.2967	206.09	2.31406
13	0.513	9.71012-10	-0.2899	219.37	2.34118
14	0.520	9.71600-10	-0.2840	230.14	2.36200
15	0.548	9.73878-10	-0.2612	254.14	2.40513
16	0.556	9.74507-10	-0.2549	267.13	2.42672
17	0.558	9.74663-10	-0.2534	272.51	2.43539
18	0.577	9.76118-10	-0.2388	284.63	2.45428
19	0.589	9.77012-10	-0.2299	315.51	2.49901
20	0.610	9.78533-10	-0.2147	332.47	2.52175
21	0.616	9.78958-10	-0.2104	357.43	2.55319
22	0.639	9.80550-10	-0.1945	380.13	2.57993
23	0.640	9.80618-10	-0.1938	393.78	2.59526
24	0.664	9.82217-10	-0.1778	407.78	2.61043
Total			-8.8194		51.56128
Mean			-0.3675		2.14839

SAMPLE CALCULATIONS

Calculations based on the values from Table 8 to determine the constants for the equation of discharge over the triangular weir using flow measurements determined by the weighing method as a standard.

$X = (-)8.8194$	$Y = 51.56128$	$xy = (-)16.96261$
$\bar{x} = (-)0.36748$	$\bar{y} = 2.14839$	$C = (-)18.94748$
$X^2 = 4.03896$	$Y^2 = 115.71176$	$xy = 1.98487$
$x = 0.79805$	$C = 110.77356$	$b = \frac{xy}{x^2} = \frac{1.98487}{0.79805} = 2.48715$
	$y^2 = 4.93820$	

Where:

X = sum of logarithm values for head

$\bar{x}$ = mean of logarithm values for head

X^2 = square of the logarithm values for head

C = correction factor, square of the sum of logarithm values of head divided by the number of observations, $\frac{(X)^2}{n}$

n = number of observations

x^2 = square of the deviations from the mean

Y = sum of the logarithm values for discharge

$\bar{y}$ = mean of the logarithm values for discharge

Y^2 = square of the logarithm values for discharge

C = correction factor $\frac{(Y)^2}{n}$

y^2 = square of the deviations from the mean

XY = product of logarithmic values for head and discharge

C' = correction factor $\frac{(X)(Y)}{n}$

xy = product of the deviations from respective means

b = slope of the curve or the rate of change of Y with respect to X

The standard equation for flow over a triangular weir is:

$$Q = C\ ^{8}/_{15} \tan \sqrt{2g}\ \ H^{5/2}$$

Where:

Q = discharge in g.p.m.

C = coefficient of contraction

= angle between a vertical line at vertex of weir and one side of weir in degrees

H = head above vertex of weir in feet

Let:

$c\ ^{8}/_{15} \tan\ = (a)$ a constant for a 90 degree weir

b = an exponent

Then:

$$Q = aH^{b}$$

Let:

$$\log \bar{y} = \log Q$$

$$\log \bar{x} = \log H$$

Then:

$$\log \bar{y} = \log a + b \log \bar{x}$$

From Table

$$\log \bar{y} = 2.14839$$

$$\log \bar{x} = -0.36748$$

$$b = 2.48715$$

Then:

$$2.14839 = \log a + 2.48715\ (-0.36748)$$

$$\log a = 3.06237$$

$$a = \text{anti} \log 3.06237$$

$$= 1154.4$$

substituting the values for a and b in equation $Q = aH^b$, it becomes:

$$Q = 1154.4\ H^{2.49} \tag{26}$$

HYDRAULIC CHARACTERISTICS OF DISCHARGE FROM ORIFICES IN ALUMINUM IRRIGATION PIPE

by

RALPH GEORGE SPOMER

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This study was conducted for the purpose of determining whether an equal discharge of water for distribution in an irrigation system can be obtained from level pipe with uniformly spaced orifices. The use of orifices made it possible to eliminate friction losses inherent in present gated irrigation pipe design.

All the factors affecting the flow from orifices were held constant with the slope of the pipe and head of water on the orifices as the only variables.

A triangular weir was constructed and calibrated to provide an accurate means for measuring the total flow of water from the orifices. The flow as measured by the weir was used as the actual total flow for calculation of results.

A head control reservoir was designed and constructed to maintain a constant head throughout each test. Baffles were included in the design of the reservoir to provide a negligible approach velocity as the water entered the irrigation pipe.

Manometer taps were installed in the head control reservoir, ten inches upstream from orifice number 18, the orifice next to the pipe inlet, and at the dead end. These taps were connected to manometers by the use of rubber tubing. Static pressure head was recorded at these points for each test.

The discharge from each orifice was determined for each test by weighing a specific amount of water and recording the time required to accumulate this amount. Three observations were taken at each orifice for each test, and the mean discharge per orifice was determined from these observations.

The results of this study indicated that there was an increase in discharge from the orifices as the dead end of the pipe was approached. The difference between the maximum and minimum flow for the orifices ranged from

0.27 to 0.82 g.p.m. for tests with 3/4 inch diameter orifices and a level pipe. For one inch diameter orifices the difference between maximum and minimum flow from the orifices ranged from 0.72 to 1.35 g.p.m. The orifices were numbered consecutively from the dead end. The minimum discharge for 3/4 inch diameter orifices occurred at orifices 16, 17 and 18. The minimum discharge for one inch diameter orifices occurred at 16, 17 and 18 with the exception of one of the five tests when orifice 13 recorded minimum discharge. The variation in discharge between orifices was significant at the one percent confidence level. The differences were also considered significant when irrigation periods of 11 or 23 hours were considered.

Experimenting with a rising 1 in 300 slope a decrease in discharge per orifice was noted as the flow approached the dead end. For a falling slope of 1 in 300 an increase in discharge per orifice existed as the dead end was approached.

Discharge from the orifices was not found to be influenced by the two angles used in this study. Discharging normal to a longitudinal vertical plane at the center of the pipe and also at a 45 degree angle below a horizontal plane at the center of the pipe affected discharge of the orifices only as the head was increased or decreased on the center of the orifice.

Some of the variation in the discharge between orifices was assumed to be a result of unequal cross-sectional orifice area. The regular pattern present in most of the curves; however must be a result of flow conditions peculiar to this type of manifold system.

It was concluded that it may be possible to design a gate which would provide nearly uniform flow from each gate in a level pipe within a discharge rate of 4 to 15 g.p.m. per gate.