AMERICAN SOCIETY OF CIVIL ENGINEERS

Founded November 5, 1852

TRANSACTIONS

Paper No. 1875

IMPROVED TYPE OF FLOW METER FOR HYDRAULIC TURBINES

By Ireal A. Winter[1], Assoc. M. Am. Soc. C. E.

With Discussion by Messrs. C. Maxwell Stanley, R. C. Johnson, E. A. Dow, E. B. Strowger, and Ireal A. Winter.

Synopsis

The operation of the modern hydro-electric power plant for high efficiency necessitates the determination of the actual turbine and generator performance curves. Such curves can now be determined to a high degree of accuracy by rating the turbine water input by either the Allen or the Gibson method. These ratings are usually in the nature of a turbine efficiency test made to determine whether the manufacturer's guaranties have been met, or to check the efficiency of old installations. As the performance curves thus established apply to only one head and one plant-loading condition, its utility as a means of presenting operating data has been considered secondary. This attitude has been justifiable, to some extent, as no generally applicable means have been available for rating permanently, a device that would indicate the turbine discharge independent of the head and load on the plant.

The necessity of using, to the fullest extent, the rate of flow data obtained during the turbine efficiency test, led to the development of the Winter-Kennedy principle for obtaining a permanent plant discharge record. The principle utilized is based on fundamental laws of flow, enabling the prediction of performance before installation. Pressure differences existing in the conventional turbine scroll case serve as the prime movers and, therefore, no loss of pressure head, change in direction of flow, or turbulence, is set up in the water passages of the turbine. The means utilized are flush piezometers located on the inner and outer surfaces of the scroll case.

Note.—Published in April, 1933, *Proceedings*.

[1] Denver, Colo.

The writer describes the fundamentals of this new method of determining the turbine discharge, and discusses, at some length, the results obtained on models tested in the laboratory and actual plant installations.

Introduction

The increased efficiency of present-day hydraulic turbines and the added refinements to power plants representing large financial investments, suggests the need of accurate knowledge of the water input and power output.

When the turbine is driving an electrical generator, the determination of the power output is relatively a simple matter. Electrical metering instruments, capable of a high degree of accuracy, are available for every type of service. The measurement of the rate of flow of large quantities of water in the field, however, is still recognized by hydraulic engineers as one of the most difficult of engineering problems.

Methods of rating water passages have been developed by American engineers, to a point where reliable and consistent results are obtained, covering a wide range of conditions. The methods developed by C. M. Allen, M. Am. Soc. C. E. (salt-velocity method[2]), and N. R. Gibson, M. Am. Soc. C. E. (time-pressure method[3]), have met with wide approval. The momentary rate of flow may be determined by both these methods, leaving the problem of permanent plant recording to be solved successfully.

Early attempts to solve the combined problem of rate of flow and plant recording were made, using the Venturi tube. This type of meter has proved practical for high-head impulse installations where ideal pipe-line conditions are to be had.

When attempts were made to adapt Venturi tubes to plants of relatively low head and short penstock, the expense of the tube and loss in pressure head made them economically unsound. It was discovered that Venturi tube characteristics were subject to considerable variations when departures from ideal conditions were encountered. The meter coefficients were found to vary as much as 10% from the ideal, and square law ratios did not exist.[4]

Another attempt to combine rate of flow and plant recording was made with the pitometer. This instrument proved unsatisfactory because it was subject to plugging by trash or because it was knocked out of position by solids carried in the water stream.

Many successful installations of equipment for obtaining permanent plant records have been made by placing piezometers in converging sections of water passages or on elbows in the penstock. As these devices are dependent upon plant conditions, they do not have general application.

It is significant to note that all recording devices for use with plants of large turbine discharge and low head must be rated in the field by some independent method of water measurement. This is also true of the method

[2] *Transactions*, A. S. M. E., Vol. 45, No. 1902, p. 285.

[3] *Loc. cit.*, No. 1903, p. 343.

[4] Serial Report, Publication 278-34, National Electric Light Assoc., pp. 6-7.

developed by Mr. A. M. Kennedy and the writer. The advantages of the improved type of flow meter are that it has general application, and its performance can be predicted with sufficient accuracy to enable the preparation of designs before the plant is constructed.

Scroll-Case Design

An understanding of the basis of designing scroll cases is necessary in considering the operation of the differential pressure taps. There are three general methods of arriving at the progressive areas of the scroll, namely, the accelerating velocity method, in which the area progressively decreases relative to the increment of water discharged into the turbine; the constant velocity method, in which the area decreases directly as the increment of flow into the turbine; and the decreasing velocity method, in which the net area of the scroll case is progressively greater than the increment of flow into the turbine.

The three methods involved are based on the fundamental law of conservation of angular momentum, or the law of the free spiral and circular vortices. Only in the accelerating scroll case of the free spiral vortex design does the center of the turbine and vortex coincide.

For structural considerations, a scroll case is not the path following the stream lines of a vortex. Investigation of the flow within the scroll shows a wide range of stream lines into the turbine speed-ring.[5] The discharge varies from a radial flow on the upper and lower surfaces of the scroll adjacent to the speed-ring, to an angular flow greater than the angle of the speed-ring stay-vane at the center line of the distributor.

Concrete scrolls of rectangular section and plate-steel scrolls of circular section are of similar design. The difference in shape is due to structural rather than to hydraulic requirements. The baffles in plate-steel scrolls are located substantially at a point 90° to the transverse counter line of the turbine. In the case of the concrete scroll, considerable latitude is taken in locating the baffle. Efficient designs have been executed with placement at points ranging from 90° to the transverse center line of the turbine, to a point substantially on the down-stream center line, covering an arc of 270 degrees. In all cases the remaining part of the true scroll passage is designed according to one of the three methods outlined.

Notation

The following notation is used throughout the paper:

A = area of scroll case at piezometer section.

C = coefficient of deflection.

H = head: H_e = effective pressure head on turbine; H_{V1} = velocity head corresponding to tangential velocity component, V_1; H_{V2} = velocity head corresponding to V_2; H_t = effective pressure head on turbine at time of test; and H_c = a common pressure head selected for convenience of comparing test data.

P = differential pressure between piezometers: P_t = pressure at test head; P_c = pressure at a common head.

[5] *Transactions*, A. S. M. E., Vol. 58, No. 13, HYD-53-4, p.32.

Q = discharge: Q_t = discharge through the turbine at time of test; Q_o = discharge at a common pressure head; and Q_n = net expected maximum discharge through the piezometer section.
R = a distance measured from the center of the turbine: R_1, R_2, etc. = distances to piezometers; and R_x = distance to center of gravity of piezometer section.
V = velocity: V_1, V_2, etc. = tangential velocity components at piezometers; and V_x = mean velocity at center of gravity of piezometer section.
g = acceleration due to gravity.
h = head: h_v = any velocity head; and h_f = any friction head.
k = experimental constant.
m = mass of water flowing at any point in the piezometer section.
n = logarithmic exponent.

Flow Meter Design

As previously pointed out, the flow lines within the scroll case do not follow definite directions, but are subject to considerable variation. This combination of flows invites an investigation as to their possible effect upon pressures at various points across the section. The following laws of flow are significant:

(a) For radiating currents, the pressure head at any point distant from the center is a function of,

$$R_1 \; V_1 = R_2 \; V_2 \qquad (1)$$

(b) Likewise, for a revolving mass of water in which the stream lines are concentric circles and the total pressure head for each stream line is the same, the pressure head at any distance is a function of Equation (1).

(c) Furthermore, for a revolving mass of water having a radiating flow combined with a circular flow, the pressure head at any distance from the center line of the turbine is a function of Equation (1), for both components of flow, and with positive or negative acceleration.

(d) If no change in direction or relative magnitude of flow takes place within the scroll case, the differential pressure, P, existing between any two points is found to be,

$$P = \frac{V^2_1}{2g} - \frac{V^2_2}{2g} \qquad (2)$$

Therefore, the flow, Q_n, through the scroll at the piezometer section is a function of the square root of the differential pressure, P. Laboratory and plant tests covering a wide range of conditions confirm these conclusions.

In designing an installation of scroll-case differential-pressure taps, the relation, $R_1 \; V_1 = R_2 \; V_2$, is used as a basis. The velocities, V_1 and V_2, are considered as tangential components of the absolute velocity, and the radii, R_1 and R_2, are their distances from the center of the turbine. This assumption is necessary as the absolute direction of the flow is unknown. Therefore, Equation (2) must take the form of,

$$P = C\left(\frac{V^2_1}{2g} - \frac{V^2_2}{2g}\right) \qquad (3)$$

The quantity of water flowing at the section selected for trial, is determined by assuming the entrance edge of the speed-ring stay-vanes as the orifice of discharge from the scroll case (see Fig. 1). The total arc of the speed-ring orifice, through which the total discharge, Q, flows, is determined by deducting the thickness of the scroll baffle on a line with the cir-

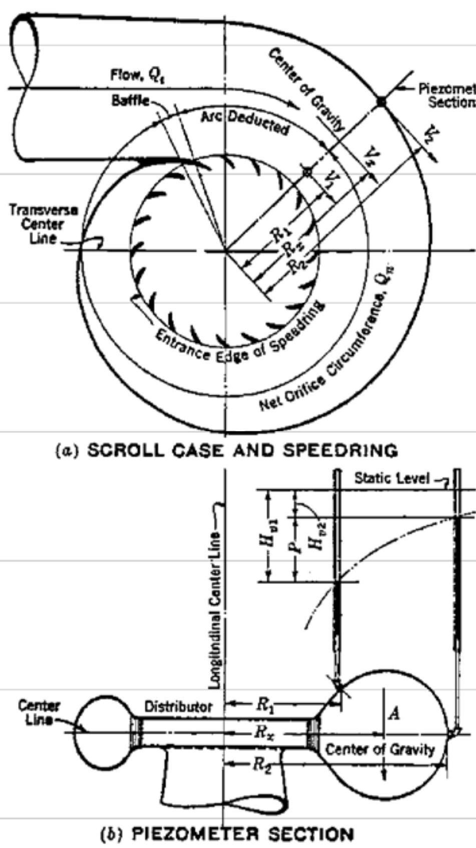


FIG. 1.—BASIS FOR DESIGN OF DIFFERENTIAL PRESSURE TAPS

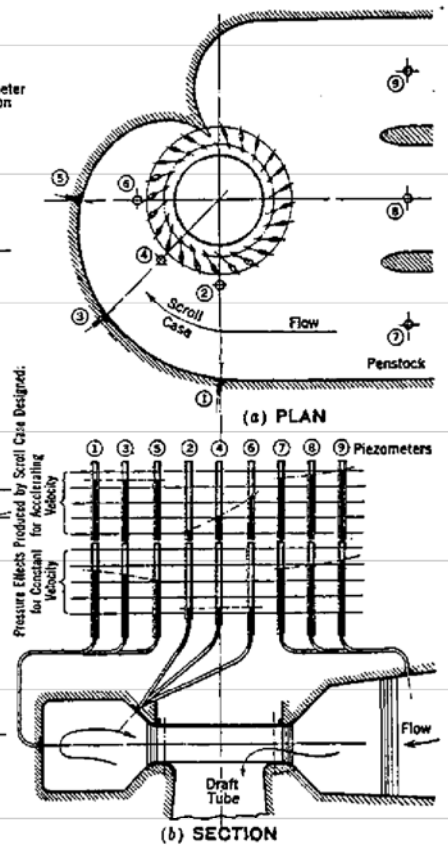


FIG. 2.—INFLUENCE OF SCROLL-CASE DESIGN ON PRESSURE EFFECTS OBTAINED BY DIFFERENTIAL PRESSURE TAPS

cumference of the stay-vane tip of the speed-ring. The net quantity, Q_n, flowing past the piezometer section is taken as the percentage of orifice circumference past that section. This establishes the net flow to be used in the calculations. The quantity, Q, when used as a basis of design, is always taken as the maximum expected discharge through the turbine. With the net flow and the area of cross-section of the scroll case known, the mean velocity is,

$$V_x = \frac{Q_n}{A} \quad \ldots\ldots (4)$$

As shown by the relation, $R_1 V_1 = R_2 V_2$, the increment velocities vary inversely as the radii. Hence, the velocity curve from the entrance of the speed-ring to the outer wall of the scroll case takes the form, $y = cx^n$, with the exponent, n, at unity.

For stability of flow, the revolving mass of water must have a common center of gravity. This common center of gravity is assumed to be the center of gravity of the cross-sectional area of the scroll-case section. Then, by the law of constant moment of angular momentum,

$$R_1 \; m \; V_1 = R_2 \; m \; V_2 \dots\dots\dots\dots (5)$$

the point of mean velocity, R_x, is about the center of gravity of the cross-sectional area. With the point of mean velocity, R_x, established, the velocity at any point of the section is determined by the relation of its distance (R_1, R_2, etc.) to the center of the turbine.

In order that the differential pressure existing between any two points on the section may be clearly understood, the static and flowing water levels are indicated in Fig. 1. To determine the flowing level at R_1, the velocity head corresponding to V_1, is computed as,

$$H_{V_1} = \frac{V^2_1}{2g} \dots\dots\dots\dots (6)$$

Likewise, the velocity head at R_2 is computed as,

$$H_{V_2} = \frac{V^2_2}{2g} \dots\dots\dots\dots (7)$$

Therefore, the net differential pressure is,

$$P = H_{V_1} - H_{V_2} \dots\dots\dots\dots (8)$$

Laboratory Investigations

As the assumption of performance of the proposed differential pressure taps was based on tangential velocity components, it was evident that a coefficient would have to be introduced to obtain the true relation of flow to deflection. The determination of this coefficient was made by use of models in the laboratory.

The first experiments were conducted using sheet-copper models of plate-steel scroll cases. The models were complete with penstock, turbine speed-ring, and fixed guide-vanes. Three types of scrolls were investigated: One with areas corresponding to full acceleration of a free spiral vortex; a second, with acceleration comprising the spiral and circular vortex; and the third, with areas corresponding to negative acceleration. Piezometer sections were established at quarter-points in the cases.

The total rate of flow as determined by pressure readings for the various sections of all three scroll cases was found to be, approximately,

$$Q = k \; P^{0.500} \dots\dots\dots\dots (9)$$

The coefficient, C, in Equation (3), was found to vary with the ratio of height of scroll case at R_x, to height of turbine guide-vanes—the smaller the ratio, the larger the coefficient. The absolute value of C ranged from 0.75 at the entrance to 1.25 at the last quarter of the scroll.

Subsequent tests were made on models of a concrete scroll case in connection with a 16-in. runner, operating under a head of 10 ft. These results proved more interesting as greater deflections were obtained and the influence of the change of angle of turbine gates could be observed.

The characteristics of pressure differences observed in the two types of scrolls are shown in Fig. 2. It is significant to note that for the case with constant velocity areas, the outer, or high-pressure, piezometers (Nos. 1, 3, and 5) record substantially the same elevation. The inner, or low-pressure, piezometers (Nos. 2, 4, and 6), vary as a function of R_x.

For the accelerating velocity case, the characteristics are reversed. The outer piezometers (Nos. 1, 3, and 5), varied substantially as the change in radius of the center of gravity of the scroll section, while the inner piezometers (Nos. 2, 4, and 6), registered equal elevations at various points around the scroll.

These interesting facts suggest that a flow is obtained more nearly approaching the vortex principle than is indicated by the flow lines on the surface of the scroll.

Similar coefficient characteristics as observed in the experiments on the 6-in., plate-steel models were found to exist for the rectangular sections of concrete cases. The absolute value of C is higher for the rectangular section, when based on the ratio of height of the scroll case at the center of gravity, to height of the turbine guide-vanes.

Piezometers Nos. 7, 8, and 9, shown in Fig. 2, are included as interesting information as to the conversion of centrifugal force into pressure head within the scroll proper. Pressures registered by these piezometers are influenced by many outside factors, and a definite relation of flow to deflection does not hold true when compared with each other, or with the pressure units within the scroll.

These differences in pressures over the scroll sections should be more clearly understood by the structural engineer. The eccentric loadings indicated by these tests, introduce bending moments not taken into consideration in the ordinary procedure of designing power houses.

The influence of a change in head on the turbine on the performance of the pressure taps is shown in Fig. 3. The plotting is for readings made on Piezometers Nos. 1 and 2, as shown in Fig. 2. The test points indicated are for five gate-openings, covering a turbine speed ranging from 75 to 115% of the normal designed head (4 to 7 speeds each, over a range from 84 to 110 rpm.). This curve clearly illustrates that no change in characteristics takes place due to a change in angle of the turbine guide-vane, or relative speed of the runner. This is important as there are three factors tending to produce unstable flow conditions at the speed-ring entrance where the low-pressure tap is usually located.

The distribution of velocities into the speed-ring is not symmetrical or uniform about the center line of the distributor. This is due to the lower section of the speed-ring supplying water to a larger area, and the shorter path to the turbine runner. As the guide-vanes are closed, the distribution is more symmetrical because the gate-opening becomes the controlling orifice.

A second influence in the distribution of velocity adjacent to the speed-ring is the relative position of the entrance edge of the guide-vane to the discharge tip of the speed-ring stay-vane. Cases have been known in which the change of flow from one side of the speed-ring vane to the other caused a drop of $2\frac{1}{2}\%$ in over-all turbine efficiency. Any pressure tap within the influence of this velocity change would register unstable characteristics.

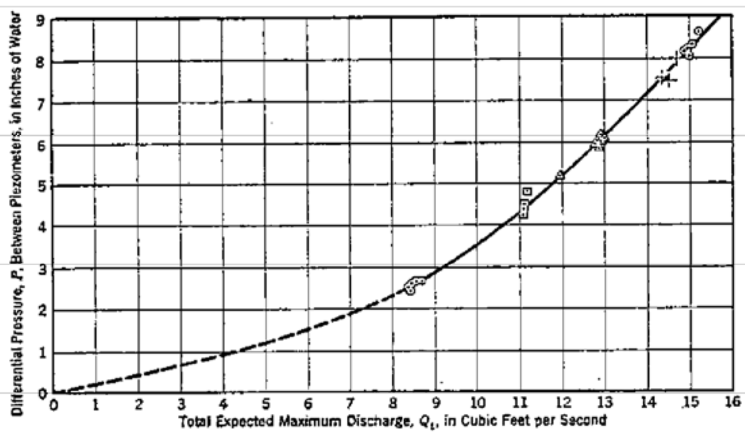


FIG. 3.—DIFFERENTIAL PRESSURES FOR 16-INCH TURBINE MODEL, SHOWING EFFECT OF CHANGE IN GATE-OPENING AND SPEED OF RUNNER

The third factor producing undesirable pressure effects is the constant speed of the runner for all heads. This item is seldom of importance because the head on the turbine rarely ever approaches a value such that a short circuit of flow is likely to take place on the entrance side of the runner.

In all cases tested in the laboratory, the inner piezometers were outside the speed-ring casting. The influence of gate changes and speed of runner could not be found for any combination of conditions.

From the results of laboratory investigations, the following conclusions may be deducted:

First.—For all scroll cases designed according to one of the three methods outlined, the following relation was determined:

$$Q = k\ P^n \text{.............................} (10)$$

Second.—The coefficient, C, was determined as varying from 0.75 to 1.25.

Third.—The coefficient, C, is substantially constant when the laws of similitude are applied. This makes possible the utilization of experimental coefficients for locating pressure taps in any scroll case.

Fourth.—The exponent, n, was found to be substantially 0.500.

Fifth.—The low-pressure taps, when properly located with reference to the speed-ring, are not subject to influence by a change in the angle of the turbine guide-vanes, or to a change in head on the turbine.

FLOW METER APPLICATION

Plant Installation.—A typical section through an hydro-electric power plant is shown in Fig. 4. An installation of differential pressure taps is shown, diagrammatically, on the down-stream side of the power house. The differential pressures are related to the static forebay pressure. The notation, $h_f + h_v$, indicates the total friction-head and velocity-head losses incidental to the penstock and other intervening water passages leading to the forebay. This loss in head is shown by the difference in elevation of the water in the high-pressure piezometer line and the static forebay level.

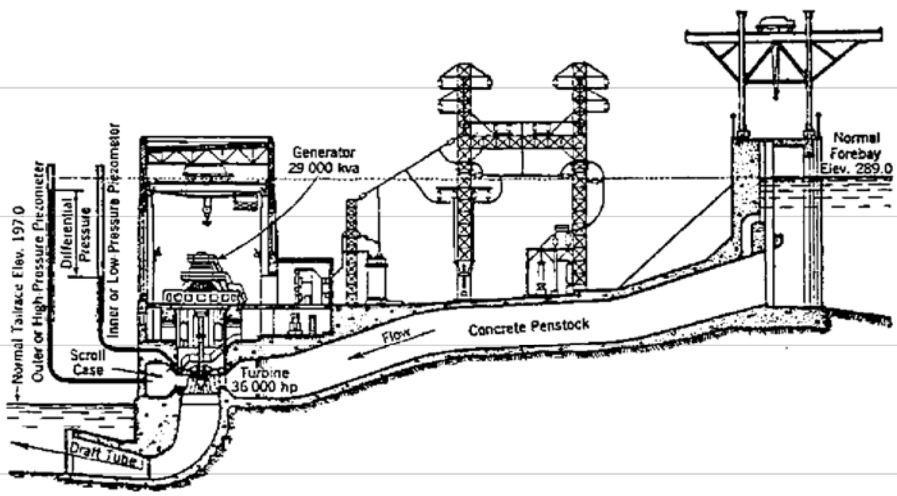


FIG. 4.—TRANSVERSE SECTION THROUGH HYDRO-ELECTRIC PLANT, SHOWING EFFECT PRODUCED BY SCROLL-CASE DIFFERENTIAL PRESSURE TAPS

A typical installation of metering and testing equipment connected to scroll-case differential pressure taps, is shown in Fig. 5. Such a combination gives complete water information for the unit. The mechanical register indicates the rate of flow, in cubic feet per second, draws a chart record showing the change of flow through the turbine, and totalizes the quantity of water passing through the machine, in cubic feet. With these data before him, the operating engineer can determine the efficiency of the unit covering any particular period of time.

The throttling manometer, shown in the illustration, is especially designed for determining the quantity-deflection relation at the time of test. Its several features consist of an engraved scale, graduated in inches and tenths, with the zero of both scales at the bottom. The scales are placed on the center line of the glass tubes, enabling accurate observations on the meniscus of the mercury column. An equalizing valve and suitable air vents provide means of obtaining a clear column of water connecting each leg of the manometer. The throttling valve at the bottom of the U-tube enables the successful dampening of the mercury column without introducing the usual errors inherent with throttling at points in the water line. By use of this manometer, check tests can be made on the performance of the register, and readings on higher or lower pressure taps can be compared with those on the standard taps in case of an emergency.

In Fig. 5, there are three low-pressure taps designated as Piezometers Nos. 1, 2, and 3. Tap No. 1 is a special connection to obtain a large differential pressure. The performance of this tap is influenced by a secondary factor due to the centrifugal force, in a vertical plane, of the water flowing around the curvature of the speed-ring casting. This secondary force is of prime importance as it can reach a value at high flows for which the quantity-deflection ratio departs from the square law. This is due to the tendency of the water to leap clear of the casting as the velocity increases and the restraining pressure decreases.

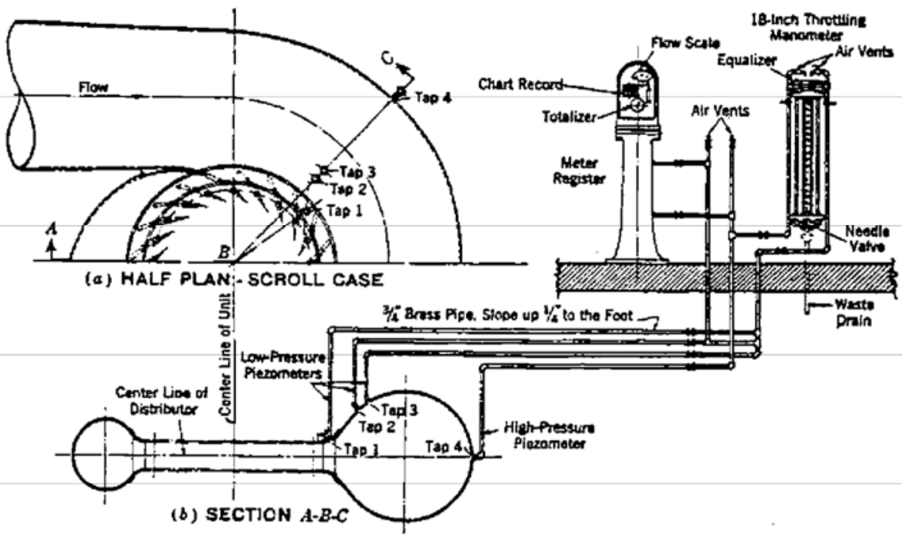


Fig. 5.—Typical Installation of Metering and Testing Equipment on Plate-Steel, Scroll-Case, Differential Pressure Taps

Pressure Taps Nos. 2 and 3 are designed as standards to meet the range of deflection of commercial meter registers. In all cases there should be two inner taps so designed that a variation of 20% in flow will produce like pressures. This is necessary, due to a possible increase in expected turbine discharge. If only one tap is installed, the capacity of the register might be exceeded, making necessary the purchase of a special instrument.

The type and method of installing the piezometers within the scroll case are of first importance. Two of the most typical designs are shown in Fig. 6. Fig. 6 (*a*) and Fig. 6 (*b*) illustrate the correct method of fastening piezometers to wooden forms and of extending pipes through the concrete. Details of two types of piezometers are shown in Fig. 6 (*c*) and Fig. 6 (*d*). The Type *A* piezometer is for use with the plate-steel scroll case and speed-ring castings. The Type *C* piezometer (for use with a concrete scroll case) is anchored to the forms with three No. 10, flat-head, brass screws, $1\frac{1}{2}$ in. long. After the forms are stripped, the screws are cut off flush with the face of the brass plate. Thus, after the walls have been cleaned, the drilling of the $\frac{3}{8}$-in. hole is completed so as to make the pressure connection to the $\frac{3}{4}$-in. line.

The reason for drilling the piezometer opening only part way in Type *C* is that cement mortar is likely to run into the piezometer line and plug the passage during the process of pouring the concrete. The completion of the hole after the forms have been removed is a simple matter. By providing the anchorage holes for the brass screws, damage to the piezometers when removing the forms is eliminated.

The Type *A* piezometer is made of monel metal for rust resistance and metallic strength. The threaded shank should never be tapered. Invariably, if it is, the nose of the piezometer will not extend to the inside of the scroll, thus creating a most embarrassing situation. ??

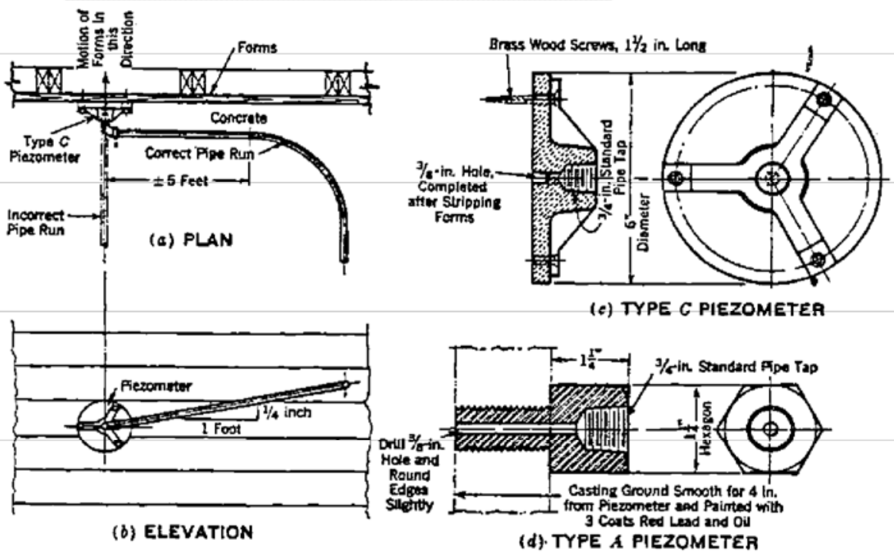


FIG. 6.—STANDARD PIEZOMETERS AND THEIR INSTALLATION

The method of making the pipe run adjacent to the scroll case is of importance. The reason for providing a 5-ft run, parallel to the surface of the scroll (Fig. 6 (*a*)), is due to the "giving way" of the forms as the concrete is being poured. When the pipe is run straight out, sufficient embedding is often found to hold the piezometer in place, while the forms move out. This leaves the piezometer recessed in the concrete, and a poor installation results. The same care should be exercised in connection with plate-steel cases as the pipe threads might be stripped in the piezometer nut, permitting leakage in the pressure line.

As the differential pressure taps in the scroll case are capable of indicating flow to a high degree of accuracy, the installation of piezometers and pressure lines should be given extreme care. Close attention should be given to tightness against leaks, proper location of valves, and careful alignment of pipe to obtain a positive upward slope to convenient valves for the elimination of air. The piezometers should have ample clearance from any projection or depression within the walls of the scroll case. A smooth surface, capable of being maintained in that condition, should surround the piezometer.

Plant Calibration.—Pressure taps in the scroll case are of considerable value during the turbine efficiency test. They provide a convenient and accurate means of comparing the measured flow through the machine with the power output, the distance traveled by the piston of the servo-motor, and net head on the turbine. Inconsistent points are weighed as to their probable value, or are discarded as being in error.

The instruments used in measuring the differential pressure should be of the highest type available. The observer should be skilled in the reading of the true pressures and in the manipulation of the pressure lines. Static readings, before and after every test, should be included as routine procedure.

In the case of a plant with two or more units of identical design, it is customary to make a measured test on only one unit. The remaining units are then calibrated for discharge, by using the electrical output as a measure of flow, assuming the efficiency to be the same for all units. This is done to best advantage by reducing kilowatts and cubic feet per second to a common head, H_c, the kilowatts being reduced or increased as $H^{1.5}$, and the cubic feet per second as $H^{0.5}$.

To obtain the discharge value for the machine being calibrated, the kilowatt load is reduced to the common head on which the test curve is based, as $\left(\frac{H_c}{H_t}\right)^{\frac{3}{2}}$. The discharge corresponding to that load is then reduced to the calibration head as $\left(\frac{H_c}{H_t}\right)^{\frac{1}{2}}$. This determines the quantity of water flowing during the calibration run, and establishes the quantity-deflection curve for the pressure taps.

If a manometer is used to obtain the deflection reading, the process may be simplified by reducing the observed values on the manometer to the common head, directly as the ratio of the common to the calibration head (H_c to H_t). This can be seen by the following relation: Since Q varies as $\sqrt{H}$ and P varies as Q^2,

$$Q_c = Q_t \left(\frac{H_c}{H_t}\right)^{\frac{1}{2}} \qquad (11)$$

and,

$$P_c = \frac{H_c}{H_t} P_t \qquad (12)$$

It has not been proved that machines of like design and installation have like efficiency. The kilowatt output of a unit is the result (not the cause), of a quantity of water flowing through the turbine. It is important, therefore, that the factors directly influencing the discharge through the runner be taken into consideration.

The principal factors are: (1) The net pressure head on the turbine; (2) the turbine gate-opening, in inches; (3) the diameter of the entrance tips of the runner blades; (4) the discharge area of the runner blades; (5) the speed of the runner; and (6) the degree of air venting into the draft-tube.

The net pressure head on the turbine is determined by net-head piezometers in the lower reaches of the penstock, and by suitable water-level gauges in the tail-race.

The gate-opening is calibrated for every tenth graduation on the governor dial. Three points between each pair of guide-vanes—one at the top, one in the center, and one in the bottom—are calipered for all gates. These values are averaged for the gate-opening curve.

The gate-opening readings are transferred to the outside of the machine by making simultaneous readings on a graduated scale clamped to the piston trunk of the servo-motor that operates the gates. Thus, the gate-opening is translated into the distance that the piston of the servo-motor travels. Consistent readings are obtained by setting the scale at zero on the servo-motor piston with a known oil pressure in the governor system. This pressure should be used at all times and for all machines when setting the scale at zero. The motion of the gates should be in the same direction for making all tests. These precautions are necessary to eliminate the effect of "gate-squeeze" and lost motion in the operating mechanism.

A typical calibration curve for the gate-opening, piston-travel relation is shown in Fig. 7. The governor-dial curve in this diagram is for operating reference and has no significance in connection with testing or calibration.

Factors (3) and (4) relate to the physical dimension of the runner and should be investigated to determine the percentage variation from the tested unit. The discharge area of the runner should be calibrated by triangulation at close points, and the developed sections measured by planimeter so as to determine the outflow areas accurately. The usual method of calibrating the area by taking three or four widths at optional points is not sufficient.

Factors (3) and (5) produce whirls in the water that fill the annular space between the guide-vanes and the runner. These are usually of no importance because the calibration head should not vary greatly from the common test head. The output and discharge relation tends to depart too far from the three-halves and square-root laws when the runner is more than 5% "off speed."

As most hydraulic units are now provided with automatic air vents in the turbine cover-plates, care should be taken that these vents are closed and the automatic mechanism is removed when calibrating the flow meter.

The advantages of making accurate readings of the piston travel to be used as a parameter for plotting the turbine discharge is shown in the flow-travel curve in Fig. 7. The original calibration of the differential pressure taps was made by using the time-pressure method of water measurement.

The subsequent test points were made by manometer readings, and these readings were converted to cubic feet per scond by the equation,

$$Q = 1\,448\ (Hg)^{0.495} \qquad \ldots\ldots(13)$$

in which, (Hg) denotes inches of mercury. Table 1 shows the agreement of discharge measurement as determined by the manometer and as read on the rate-of-flow scale of a standard Venturi tube register after three years of service.

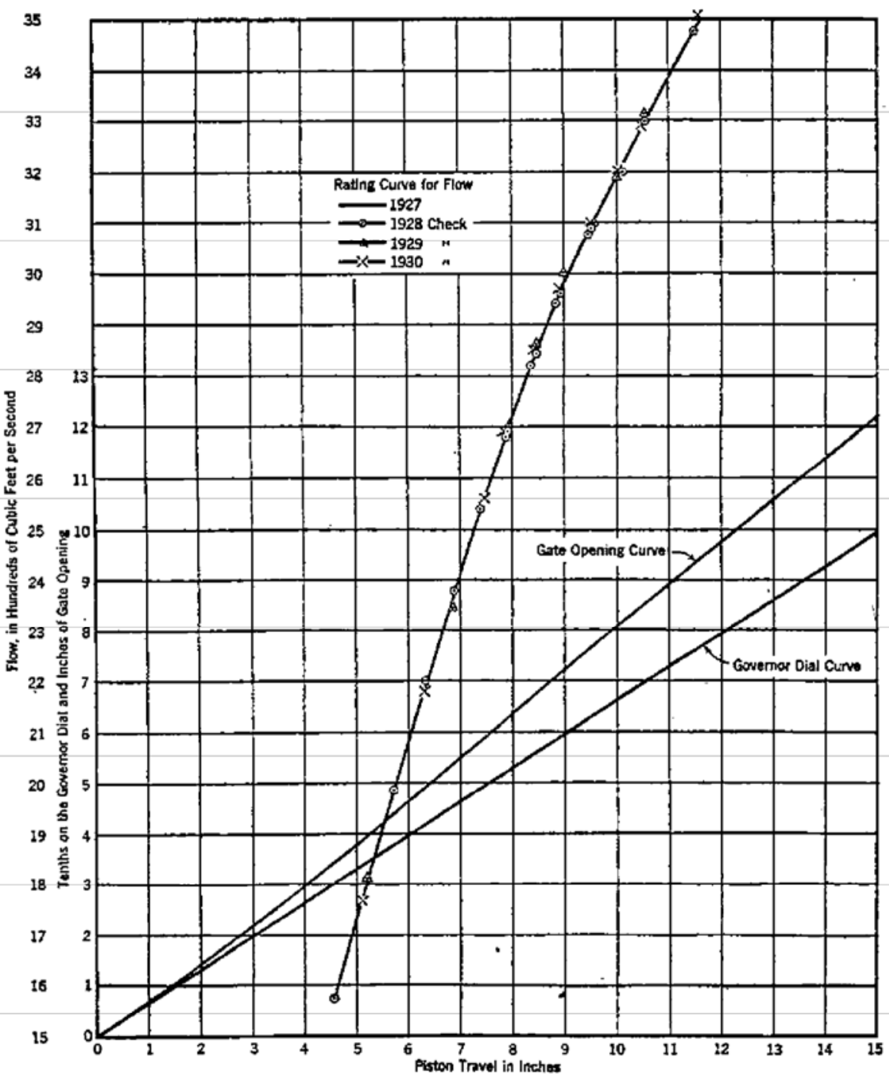


FIG. 7.—RELATION BETWEEN PISTON TRAVEL, AND THE VARIOUS FACTORS SHOWN

TABLE 1.—Comparison of Reading by Manometer and Flow Register, in Cubic Feet per Second

Run	By manometer	By register	Run	By manometer	By register
1	880	878	7	3 100	3 085
2	1 340	1 330	8	3 300	3 300
3	1 770	1 764	9	3 525	3 523
4	2 180	2 172	10	3 210	3 210
5	2 545	2 528	11	2 980	2 968
6	2 850	2 840	12	2 690	2 681

Table 1 lists the actual test values from which the points for the 1930 check test, indicated in Fig. 7, were computed. The installation of equipment is identical to that shown in Fig. 5.

Plant Performance.—The performance of a typical installation of pressure taps in a plate-steel scroll case, is shown in Fig. 8. The piezometer numbers, 1 and 4, 2 and 4, and 3 and 4, refer to the corresponding numbers in Fig. 5. The curves represent test values as determined by using the salt-velocity method of water measurement.

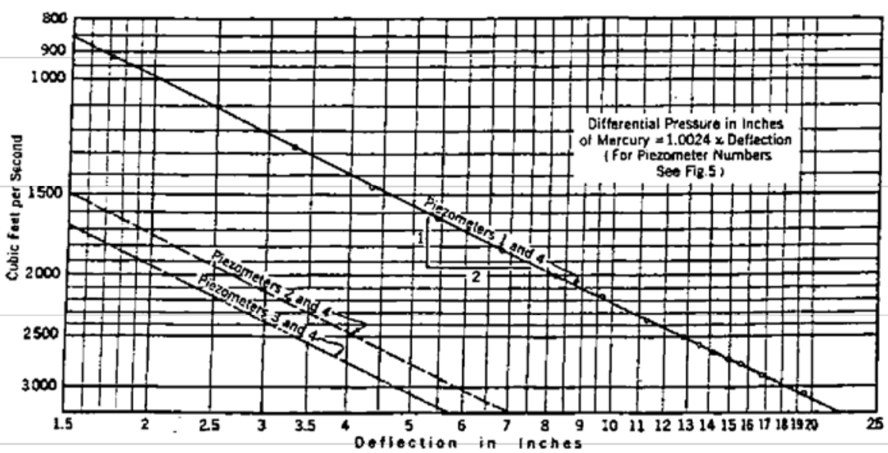


Fig. 8.—Performance of Differential Pressure Taps in Scroll Case

By inspection, the exponent, n, in Equation (10) is 0.500 for all three combinations of taps. There is no evidence of a change of flow around the speed-ring stay-vane; nor is there a change of the quantity-deflection relation due to the centrifugal force of the water flowing around the short radius of the speed-ring casting. Piezometer Nos. 2 and 4 give deflections in the ideal range for standard Venturi tube registers.

Three similar machines in the same plant were calibrated for turbine discharge by reference to the kilowatt-discharge values obtained during the test, as shown in Fig. 8. The following deflections, in inches of mercury, for Taps Nos. 1 and 4 (see Fig. 5) were found, corresponding to a discharge of 3 000 cu ft per sec:

Turbine Unit No.	Deflection, in Inches of mercury.	Percentage variation from the average.
1	21.3	+ 10
2	18.2	— 6
3	19.3	— 1
4	19.0	— 2

The percentages of variation of the four units was found by averaging the deflections. These values are greater than those found for piezometers corresponding to Taps Nos. 2 and 4, and 3 and 4. The predicted and observed deflection for the more conservative designs seldom differ more than 5 per cent.

The reasons for a difference in the pressures found in similar installations are as follows, in order of their importance:

First.—The pressure taps are designed assuming that the water is distributed equally around the unit. This condition does not exist.

Second.—The efficiency is assumed to be the same for like units. Actual data on plant tests, confirming this point, are lacking.

Third.—The cross-sectional area at the piezometer section is assumed to be accurately known. No field measurements are made to determine the point.

Fourth.—The location of the piezometers is not of a precision nature. As long as they are not subject to alteration, no special attention as to exact location is necessary.

Fifth.—When the inner or low-pressure tap is within the influence of the absolute gate-opening, identical quantity-deflection relations are not to be expected. The irregularity of the guide-vane castings and the eccentricity of the guide-vane stem to the face of the casting produce flows of varying

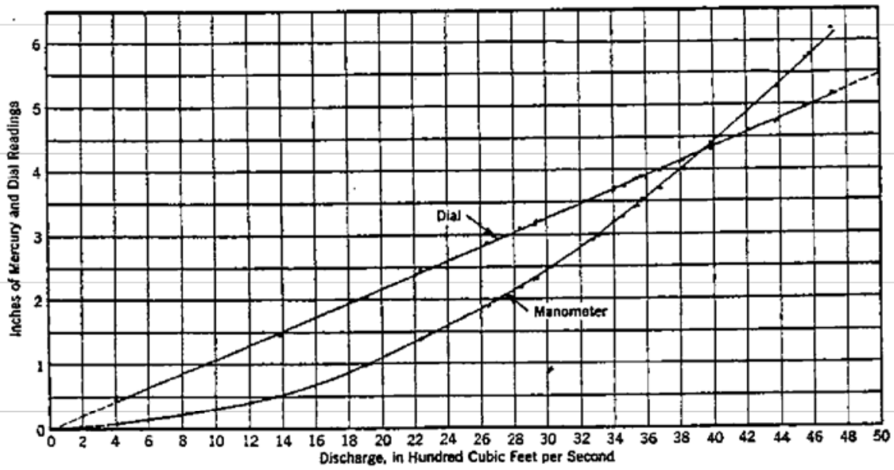


FIG. 9.—RELATION OF FLOW, AND REGISTER INDICATION, AND INCHES OF MERCURY DEFLECTION FOR DIFFERENTIAL PRESSURE TAPS

quantities. This fact becomes of first importance in installations using Piezometers Nos. 1 and 4 (Fig. 5).

An interesting exhibit of plant results is shown in Fig. 9. These curves are from plant test data made by the salt-velocity method of water measurement, and are for the plant shown in Fig. 4. The piezometer installation is as shown in Fig. 2, Taps Nos. 1 and 2. The manometer readings, in inches of mercury, compare with the laboratory results, in inches of water, as shown in Fig. 3. The comparative agreement between the laboratory and plant curves is within 2%, the laboratory values being consistently lower.

By plotting the actual test points as given in Table 2, the fine agreement between the measured discharge, manometer, and dummy dial reading of the meter register, can be studied. The readings were obtained on instruments installed as shown in Fig. 5. The meter used is a standard register, designed to operate with a Venturi tube. The flow scale, or dummy dial, is graduated, in inches of the arc traveled by the pointer, and is an arbitrary value used for reference purposes only. By installing the meter register before making the plant test, the values for final calibration are read directly on the scale during the efficiency test. The engraving of a new flow scale is a simple matter, necessitating only the shifting of the instrument zero and re-spacing of the graduations to read in cubic feet per second.

TABLE 2.—PLANT EFFICIENCY TEST DATA

Run	Head, H_t, in feet	Measured discharge, in cubic feet per second	Differential pressure, P, = inches of mercury	Readings, register dummy dial	Horse-power	Inches of piston travel	Inches of gate-opening	Readings on governor dial, in tenths
1....	91.99	1 381	0.51	1.46	10 194	4.18	3.15	3.0
2....	91.97	1 907	0.97	2.04	15 526	5.78	4.45	4.0
3....	91.84	2 639	1.81	2.87	23 933	7.56	5.95	5.0
4....	91.81	2 930	2.33	3.17	26 515	8.28	6.60	5.5
5....	91.79	3 269	2.92	3.53	30 649	9.25	7.45	6.0
6....	91.81	3 527	3.42	3.84	33 233	9.93	8.05	6.5
7....	92.01	3 806	4.00	4.15	35 430	10.93	8.90	7.0
8....	91.82	3 984	4.36	4.42	37 004	11.65	9.50	7.5
9....	91.65	4 207	4.87	4.59	38 312	12.50	10.20	8.0
10....	91.56	4 377	5.26	4.74	39 000	13.25	10.80	8.5
11....	91.33	4 555	5.70	4.96	39 586	14.12	11.50	9.0
12....	91.10	4 719	6.15	5.16	39 902	15.06	12.25	10.0
13....	92.14	2 236	1.36	2.43	19 457	6.62	5.15	4.5
14....	92.09	2 935	2.35	3.20	26 906	8.37	6.65	5.5
15....	92.01	3 291	2.96	3.58	30 986	9.28	7.45	6.0
16....	91.97	3 402	3.21	3.71	32 647	9.62	7.75	6.2
17....	91.98	3 465	3.28	3.76	32 843	9.72	7.85	6.4
18....	91.92	3 544	3.43	3.86	33 773	9.97	8.06	6.5
19....	91.92	3 534	3.44	3 86	33 611	9.97	8.06	6.5
20....	91.88	3 567	3.55	3.89	34 196	10.09	8.15	6.6
21....	92.18	3 681	3.71	3.99	34 657	10.40	8.40	6.7
22....	92.13	3 701	3.77	4.01	34 854	10.53	8.55	6.8
23....	92.04	3 825	4.00	4.15	35 622	10.97	8.91	7.0
24....	92.09	402	0.05	0.46	1 676	1.25	0.90	1.0
25....	92.09	....	0.00	0.00	—228	0.00	0.00	0.0

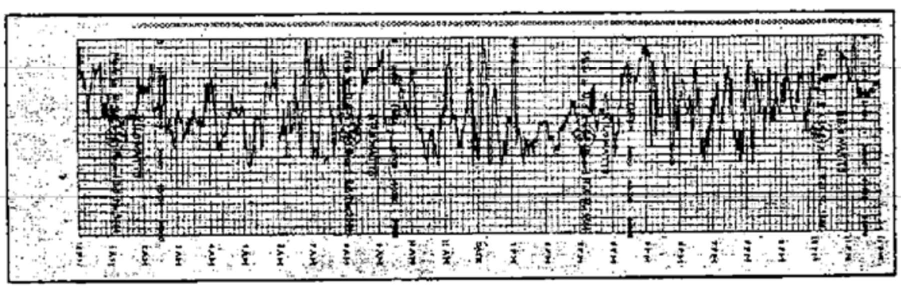
It is significant to note that the quantity-deflection relation for the differential pressure taps, follows those of the Venturi tube sufficiently close to enable the use of standard Venturi tube equipment.

The plotting of the curves in Fig. 9 clearly indicates that a deflection of 6 to 8 in. of mercury is sufficient for all practical purposes. A greater deflection must carry the same error in percentage, and, therefore, can have no advantage over the lower deflection for absolute values. Furthermore, as the rating of the pressure taps is dependent upon the measured flow, a degree of accuracy greater than that of the measuring means is of no value.

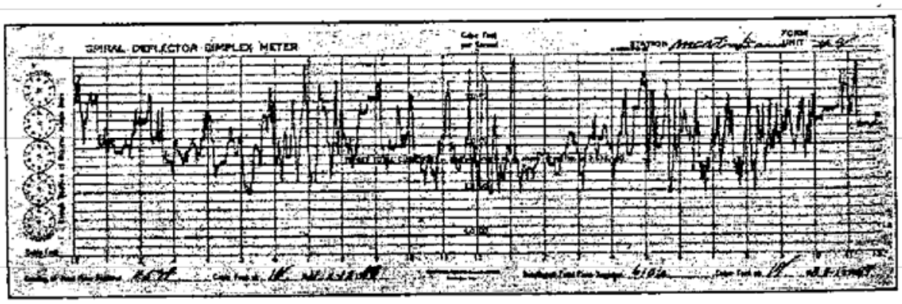
The height of the diagrams obtained in the time-pressure and salt-velocity methods of water measurement do not ordinarily exceed 3 or 4 in.; and a second dimension, time, is introduced in the computation for flow. In the case of the manometer and dummy dial reading, only one dimension is involved. With a deflection of 6 to 8 in. of mercury for the differential taps, the advantages are obvious.

In the case of a calibration test for similar machines where the electrical output is used to determine the turbine discharge, the same relation exists. A high-grade portable watt-meter will ordinarily show a scale deflection of 2 to 3 in. The water-meter register will have a corresponding scale deflection of from 5 to 8 in.

The water-meter register is capable of continuous integration with a prime moving force in the same proportion as the deflection. This method of rating can be used in connection with the measurement of water by current meter and by rotating standards for electrical output, if the engineer believes greater accuracy is obtained by continuous integration.



(a) Kilowatt Load Chart



(b) Water-Input Rate Chart for Hydraulic Turbine

Fig. 10.—Typical Kilowatt-Load and Water-Input Charts

Fig. 10 (a) is a sample record chart showing the flow characteristic obtained for a turbine equipped with a water register connected to differential pressure taps. The corresponding record chart (Fig. 10 (b)) of electrical output is included for purposes of comparison.

Plant Operation.—The operating advantages to be had by reference to water-measuring means are not fully realized. The general method in use at present to arrive at the daily operating efficiency is by reference to the kilowatt-hours per hour reading. This value is converted to cubic feet per second by assuming the unit operating at one load, and at constant head. Unless the plant has been on base load for a long time, such methods give results that are not in agreement with the facts.

The following example illustrates the errors possible:

Total kilowatt-hour per hour reading....	=	24 500
Over-all efficiency at 24 500 kw..........	=	89.5%
Apparent operating efficiency...........	=	89.5%

The load was actually carried as follows:

45 min at 29 000 kw..................	=	82.0%
15 min at 11 200 kw..................	=	76.0%

Then, the weighted average efficiency is:

82 × 29 000..........................	=	2 380 000
76 × 11 200..........................	=	850 000
Σ kw......40 200		3 230 000 = Σ products

$$\text{Efficiency} = \frac{3\,230\,000}{40\,200} = 80 \text{ per cent.}$$

Therefore, the error in the efficiency value by using the kilowatt-hour per hour reading is $89.5 - 80.0 = 9.5$ per cent. The absolute error in quantity of water flowing through the unit is $\frac{89.5}{80.0} = 1.12$, or 12 per cent. The change in tail-race elevation between hourly readings can frequently account for errors of 2 or 3% in efficiency and flow values.

Concrete proof of the inability of operating engineers to secure the best results by the use of an efficiency curve only is shown by actual plant records. Increases in over-all plant efficiency amounting to 4% in generation have been effected by the use of water registers.

The water power generated in the United States accounts for about 6% of the total generation. As water power can be brought on the line within 1 min to 3 min, with maximum efficiency, the importance of hydro-electric plants as peak-load power can be seen. As the ratio of hydro-electric to steam generation decreases, the necessity of water-recording will increase.

On rivers being developed for power throughout their length, the formation of slack-water pools drowns out the stream-gauging stations. Unless plant-recording devices are provided, the very necessary stream-flow records will cease.

In the case of joint use of a stream by two or more companies, the quantity of water flowing must be known because charges are made for regulated flows. Any method, subject to possible errors as shown to exist when positive recording devices are not provided, might lead to expensive controversies over water rights.

Conclusions

The following summary of conclusions is offered for discussion, based on the evidence presented in the paper.

(1) Laboratory tests on models of scroll cases indicate that the pressure differences on opposite walls of the scroll case can be used to indicate the rate of flow through the turbine.

(2) These pressures were found to follow definite laws of motion, and a flow relation could be established involving fundamentally sound principles.

(3) Laboratory investigations show that a change in head on the plant will not alter the quantity-deflection relation.

(4) The differential pressures in the scroll case are related to the flow through the turbine with characteristics of the Venturi tube. These characteristics are sufficiently close to enable the use of standard Venturi-tube instruments.

(5) The exponent, n, in Equation (10) ($Q = k\,P^n$) is substantially 0.500. The information being compiled on various plant tests indicates that a true square law relation will agree more uniformly with plant calibrations than the Venturi-tube exponent.

(6) There is no definite information indicating a change in quantity-deflection relation at low flows, as is manifested in the Venturi tube with throat velocities less than 5 ft per sec.

(7) Plant records indicate that the differential pressure taps in the scroll case will give consistent values over a period of years.

(8) The coefficient, C, in Equation (3) was found to cover a wide range, but in agreement with the fundamental laws used as a basis for design.

(9) By application of the laws of similitude, the experimental coefficient, C, can be used to predict the quantity-deflection relation within 5% of the observed value.

(10) Plant recording and operation can be brought to a high degree of efficiency by the use of water-recording devices.

(11) Experience with the scroll-case taps to date, indicate that they are not satisfactory for determining the rate of flow through a turbine without first being calibrated by other means of water measurement.

Acknowledgment

The writer wishes to express his appreciation to the Alabama Power Company, W. S. Barstow and Company, and the Simplex Valve and Meter Company for their co-operation in supplying a large part of the data presented in the paper; and to Mr. Kennedy for his valuable aid in the conception and development of the flow meter.

DISCUSSION

C. MAXWELL STANLEY,[9] JUN. AM. SOC. C. E. (by letter).—The flow meter for hydraulic turbines, which has been developed by Mr. Winter and his associates, and which is described in his paper, fills a decided need in the field of hydro-electric power development. Hydraulic engineers have long desired and searched for a suitable method for obtaining continuous records of discharge of water through turbines for use as a check on the efficiency of operation. Such a method must be simple, positive, accurate, and economical, if it is to have general application.

Numerous methods are available for the measurement of water. These methods include the weir, Venturi meter, and current meter. They also include the salt-velocity method and the Gibson pressure method, which have been widely used for efficiency tests. The list would not be complete without adding the stream-gauging method, which is in general use on rivers. Some of these methods have definite applications, together with definite limitations, as devices for obtaining a continuous record of discharge through a hydraulic turbine or power plant. Obviously, the salt-velocity and Gibson pressure methods, and the use of the current meter, are too complicated for continuous service and must be limited to testing purposes.

The weir is an ideal device for obtaining continuous records of discharge, but it is limited to small quantities, and, even then, it requires a loss of head, which is uneconomical. The Venturi meter may also serve for continuous records where its installation is feasible and economical, but it will be limited generally to relatively small quantities. The stream-gauging method is suitable where a satisfactory control is available and will give good results if used intelligently. Such a station, however, can only measure the total flow and cannot segregate the discharge to various turbines, or to flow over the spillway.

Unfortunately, none of the aforementioned devices is fitted for general application to individual turbines for the purpose of obtaining continuous records of discharge. For such purposes, a device must give a single, unvarying indication for any given discharge and must be susceptible of calibration in terms of true discharge. Such requirements lead into the field of the so-called "index" methods, in which the pressure differential between two points, or the velocity at some one point, is related definitely to the total discharge. Within recent years, considerable attention has been given to the development of such "index" methods for use in obtaining continuous records and as a check against more elaborate test methods.

The use of pressure differentials appears to offer greater possibilities than the use of velocities because pressures are more stable than velocities, and the instrument for measuring pressures—the piezometer—is a more simple and more reliable device than the instruments for measuring velocity—the Pitot tube, or the current meter.

[9] Engr. (Young & Stanley, Inc.), Muscatine, Iowa.

The Winter-Kennedy method is an ingenious application of the pressure differential method to scroll-case installations. This paper outlines the design and application of such equipment in an interesting manner. The writer believes that the principles of this method may be extended to a wider application to fit cases other than those having closed scroll cases. Whenever there is a flowing water, there will be pressure differentials between various points which will result from friction losses, changes in velocity, or centrifugal action. In all these cases, the pressure differential will probably be a function of the square of the discharge. It is probable that differentials resulting from changes in velocity, or centrifugal action, will be more reliable and consistent than those resulting from friction losses, as the friction losses tend to vary with time

Wherever pressure differentials are of sufficient magnitude to allow satisfactory measurement, and wherever they are a definite function of the discharge, they may be utilized as an index of the discharge; with proper calibration, they may be used to express the total discharge. No doubt, in many instances, these differentials will have to be measured in inches of water rather than in inches of mercury, and equipment will have to be arranged to fit the particular application.

As an example, assume an intake structure to a tunnel with a water velocity of 2 ft per sec immediately behind the trash racks, and a velocity in the tunnel of 8 ft per sec. This change in velocity involves a pressure differential of nearly 1 ft of water which will occur in a relatively short distance. Such a differential is capable of measurement. Other applications might be feasible at bends of penstocks or pipe lines, and at various points on the turbine speed-ring, guide-vanes, or gates of units having an open flume setting. Of course, such installations will require careful investigation and experimentation to obtain reliable results.

R. C. Johnson,[7] Assoc. M. Am. Soc. C. E. (by letter).—The author has presented clearly the mathematical reasoning and experimental evidence leading to very definite conclusions. An understanding by the profession of the operation and installation of this improved type of flow meter should certainly lead to its more general adoption.

In addition to the operating advantages mentioned, such flow meters should offer particular advantages when used in connection with adjustable-blade propeller turbines, where the operating conditions sometimes vary through much wider limits than for ordinary installations. Continuous flow records for the purpose of checking efficiencies would be especially useful in connection with hand-operated adjustable-blade runners. The use of an efficiency curve as the sole guide in the operation of a unit is indeed inadequate. The character of an efficiency curve is by no means constant even under the same apparent conditions of operation. An easy method of checking the efficiency quickly, under any combination of operating conditions, would often lead to the location of sources of waste which would otherwise go unnoticed.

[7] Assoc. Prof., Civ. Eng., Univ. of South Carolina, Columbia, S. C.

The author points out the eccentric loading condition due to differences in pressures over the scroll-section and the resulting bending moments produced by it. Expressing the pressure difference by Equation (3), with C having a range of values from about 0.75 to 1.25, it can be seen that the pressure difference between any two points in the scroll case could scarcely be sufficient to cause undue annoyance in the structural design, yet stresses produced by such pressure should certainly be taken into consideration. Fig. 8 shows the differential pressure between Piezometers Nos. 1 and 4 (Fig. 5) to be about 22 in. of mercury (approximately 25 ft of water), for maximum discharge conditions. While moments produced by such eccentric loads could not be enormous, a knowledge of their magnitude and distribution throughout the entire scroll-section, based on experimental investigations, would remove considerable guesswork now necessarily used by the structural designer, resulting in a more economical use of materials.

E. A. Dow,[8] M. Am. Soc. C. E. (by letter).—An interesting presentation of a most useful method of metering the discharge of large hydraulic turbines is presented in this paper. Undoubtedly, it will be applied with increasing frequency in future hydro-electric practice.

The turbines of the Comerford Plant of the New England Power Association System (15-Mile Falls development, near Littleton, N. H.), are similar to that from which the data shown in Fig. 8 were obtained, and the results of the calibration test at this plant may be of interest for comparative purposes.

The Comerford turbines are equipped with a number of piezometers to provide a wide variety of differentials for use as mutual checking systems in connection with the measurement of turbine discharge by the salt-velocity method. Among these were four piezometers installed in accordance with designs furnished by Mr. Winter and corresponding to Piezometers Nos. 1, 2, 3, and 4, shown in Fig. 5. In Fig. 11, the performance of these piezometers is presented in logarithmic form similar to Fig. 8, except that the differentials are expressed in feet of water. The data are for the piezometers corresponding to Piezometers Nos. 2 and 4 of Figs. 5 and 8, these being the ones observed during the salt-velocity test.

The solid line ($Q = 1\ 355 H^{0.490}$), represents the calibration formula derived from twenty test runs. This was arrived at by least squares to eliminate personal bias, and assumes equal probability of errors in determining the differentials and the discharges. The dotted line ($Q = 1\ 343 H^{0.500}$), represents a square law calibration, with the coefficient (0.500), assumed to make the average of the ratios of computed discharge to salt-velocity discharge equal to unity.

It will be noted that the two calibrations intersect at about 2 100 cu ft per sec, or 65% of capacity; and that the square law discharge is 101% of the logarithmic discharge at 100% of capacity, and 96% at 10% of capacity. It is obvious that the square law coefficient may be slightly modified to cause the line to pass through some preferred point, sacrificing the average agree-

[8] Designing Engr., New England Power Eng. & Service Corporation, Boston, Mass.

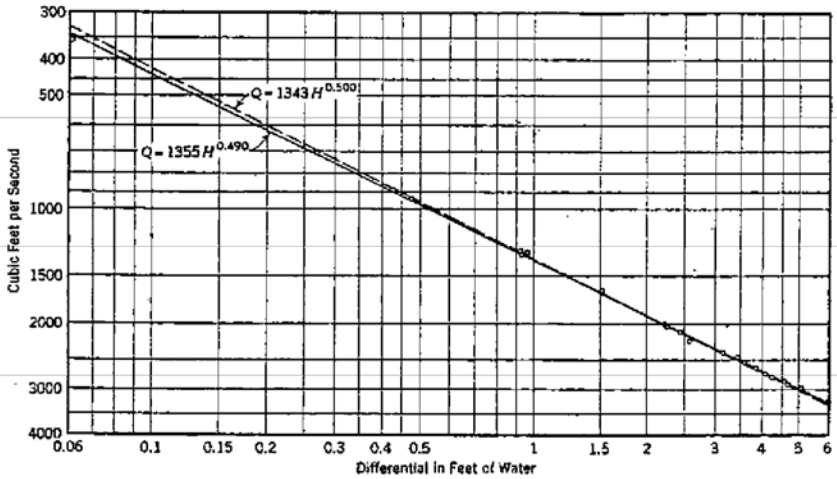


FIG. 11.—PERFORMANCE OF PIEZOMETERS IN TURBINES OF COMERFORD PLANT.

ment with the salt velocity to some extent. The calibration thus reduces to a choice of assumptions, either that the square law is inviolate, or that, on the average, the measured discharges are correct.

E. B. STROWGER,[9] ESQ. (by letter).—The flow meter described by Mr. Winter seems to be in fairly wide use in the case of relatively recent hydraulic turbine installations. This has been brought to the writer's attention by the increasing number of turbine tests made by the Gibson method where a rating of casing piezometers is specified as a part of the work. Of the turbines tested under the direction of Mr. Gibson during the four years (1929–1933), nearly one-half have had casing piezometers installed. The inference that may be made from this trend is that some, or perhaps all, of the advantages pointed out by Mr. Winter for the installation of these piezometers, are becoming generally recognized.

In thirty-three recent tests by the Gibson method, the exponent, n, in the author's Equation (10), was found to vary from 0.484 to 0.544, the average value being 0.517. These data seem to confirm the author's Conclusion (5) which states that the value of the exponent, n, is substantially 0.50 (see, also, Equation (9)). The relation between the computed value of P and the test value is expressed by the coefficient, C, of the author's Equation (3). In the tests cited this coefficient varied from 0.65 to 1.27, the average value being 0.99. These values roughly check Mr. Winter's experiments involving the range in C from 0.75 to 1.25. This relatively wide spread in the value of C is probably accounted for mainly by (1) the assumption that the velocity at each of the taps can be calculated by use of the vortex law; and by (2) neglecting the effect of the centrifugal force of the water as it flows around the case. This wide variation in C makes it impossible to use the casing piezometers as a primary method of water measurement but, as pointed out

[9] With Buffalo, Niagara & Eastern Power Corp., Buffalo, N. Y.

by the author, the piezometers should be rated by an approved method, such as the Allen or Gibson method, and thereafter may be used for measuring purposes.

In designing the piezometer and metering equipment, it is important, of course, to know the relative amount of mercury deflection to expect for the full-gate discharge. Some allowance must be made for the possible error involved in the calculation and, as a practical matter, this may be taken care of by adding two more low-pressure taps, one on either side of the main one, as indicated by the author.

IREAL A. WINTER,[10] ASSOC. M. AM. SOC. C. E. (by letter).—The importance of using pressure differences as a means of determining the flow of water in hydraulic channels has been pointed out by Mr. Stanley. He thus states a fundamental rule that makes possible the correct approach to the problem of measuring, continuously, water in motion.

The structural significance of varying pressures across the scroll case, as noted by Professor Johnson, are not important in connection with rectangular concrete sections, where the structure is usually designed as a cantilever with no external loads carried by the turbine speed-ring. The water pressure inside the scroll tends to relieve the stress and increase the equilibrium of the structure. With plate-steel scroll cases, under relatively high heads, the eccentric loadings do introduce secondary stresses of appreciable magnitude.[11]

This loading may best be illustrated by considering the scroll case empty and superimposing the unbalanced water pressure, due to the difference in velocities across the scroll, external to the casing. Under this loading the casing will assume a shape that will introduce considerable stress in the extreme fiber of the plate at the junction of the plate and speed-ring. The secondary stress, added to the ring stress due to internal loading, is likely to reach a value several times that used in the assumptions for design purposes. As pointed out by Professor Johnson, an understanding of these loadings will result in a better design of structure with perhaps an actual, instead of an imaginary, factor of safety.

The exponent, n, in Equation (10) is of interest in connection with all prime movers that operate registers. As a matter of practical application, the indication of a register must follow within certain limits of deflection. Fig. 9 illustrates the agreement of a Venturi tube register with the exponential characteristic of the scroll-case meter. Figs. 8 and 11 differ in curve slope from each other and from the Venturi tube. It is always important that the register be read at the time of calibration as the lower values of differential pressure are more easily read on the register dial than can be determined by the manometer. With manometer readings being converted into flow from calibration equations or curves, the slope of the quantity-deflection curve is of no moment.

The interesting comparison made by Mr. Dow in Fig. 11, between the basic and actual exponent determined by test, does not indicate that either

[10] Denver, Colo.

[11] *Transactions*, Am. Soc. C. E., Vol. 98 (1933), p. 101.

the test or the differential pressure is in error. It is more likely that both are correct. As a matter of further interest regarding the exponent, n, of the quantity-deflection curve, Mr. Strowger reports values determined by tests ranging from 0.484 to 0.544. These values are significant when considered with an understanding of the relation of the low-pressure and high-pressure taps to the mean velocity at the measuring section.

Investigation as to the slopes of the quantity-deflection curves for various prime movers where the velocities of the threads of flow vary considerably, shows the same trend of exponential variation. A Pitot tube, when placed in one thread of flow, and calibrated to indicate the quantity flowing in a relatively larger body of water, may have a slope character of wide variance from the square law. In the case of large Venturi tubes or converging sections in penstocks, as referred to in the paper, the slope exponent is not at all consistent with the square law ratio. This apparent inconsistency is a matter for further research.

It may be stated that any pressure or impact orifice, subject to the action of a single thread of a relatively large stream of flowing water will tend to function according to Bernoulli's theorem of the relation of velocity and pressure heads for that particular thread of flow. This statement is at odds with experience in connection with small conduits in which a piezometer ring is said to give the net mean pressure head for the piezometer section. A velocity traverse of the section will show a wide range in velocities without a corresponding difference in pressure.

This apparent phenomenon has been the source of amazement to hydraulic engineers since the relation was determined. It is not inconceivable that Mr. Dow and Mr. Strowger should consider inviolate the theory that a single pressure tap should integrate the entire stream of flow.

As a matter of exact science, according to the fundamental laws of flow, the pressures and velocities should be integrated for the entire flow stream. This process is approached by traversing a measuring section with a Pitot tube of both impact and pressure orifices. The exponential function of the Pitot tube will be found to approximate the square law without measurable differences.

The pressure taps in a scroll-case flow meter intersects threads of flow of widely different velocities in the same plane of pressure. One thread may be leading or lagging in relation to the other, and hence the exponent, n, in Equation (10) may vary between the limits as found without departing from a true relation of velocity to pressure heads.

In presenting the fundamentals of the flow-meter design, the complexity of the relation of flow to vortex pressures has been purposely omitted. When the inherent limitations are considered, of a turbine installation for determining (independently of other means) absolute discharge, it is obvious that a direct approach with easily determined functions, should be used as a matter of interest and utility.