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UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

Technical Memorandum No. 610

COMPARISON OF FIELD AND MODEL HYDRAULIC EXPERIMENTS
BOULDER OUTLET WORKS
BOULDER CANYON PROJECT, ARIZONA-CALIFORNIA-NEVADA

by

S. P. WING, ENGINEER

Denver, Colorado,
July 22, 1940

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MEMORANDUM TO CHIEF DESIGNING ENGINEER

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COMPARISON OF MODEL AND FIELD MEASUREMENTS OF THE HYDRAULIC LOSSES IN THE BOULDER OUTLET WORKS

S. P. Wing, Engineer Bureau of Reclamation, Denver, Colorado

SUMMARY

The theoretical studies for the hydraulic design of the Boulder Dam penstocks and outlet works were aided by elaborate tests on a 1/64-scale hydraulic model, the results of which were published in Boulder Canyon Project Final Reports, Part IV, Bulletin 2. The present memorandum covers the hydraulic tests made in the field during 1937, 1938, and 1939, and compares these prototype tests with those made on the model. The tests, made on 30.0-foot and 8-foot 6-inch bitumastic-coated steel conduits, with maximum velocities up to 90 feet per second (maximum Reynolds number of the order of 70,000,000) and with gross heads up to 337 feet, are believed to have a probable accuracy of about two percent for the discharges and 10 percent for the loss coefficient for the largest discharge measured. The discharge coefficients for the needle valves were found to decrease with increase in head, the values 0.77 - 0.73 based on the gross outlet area, and on the energy head as measured at the entrance to the valve being consistent with values found from the model. Likewise the equations giving the discharge in terms of the drop in water surface from the lake to the intake tower with one gate open were found in reasonable agreement with the model results the values being:

$Q = 6{,}500 \sqrt{H}$ for the model, $6{,}640 \sqrt{H}$ for the prototype 1939 tests, and $6{,}750 \sqrt{H}$ as determined from stream gagings below the dam.

Inherent difficulties in making field tests of such magnitude, the actual physical layout with fittings close together so that hydraulic disturbance from one fitting affects the flow at fittings downstream, and the lack of adequate data as the rate at which energy is dissipated following a fitting, make it impossible to separate reliably the gross measured loss into its individual components. However, within the limits of the tests, such information as has been obtained indicates the probability that in the prototype entrance, bend and branch losses are dissipated at a much slower rate than shown by the model. This may be due to the greater absolute roughness of the model compared to the

prototype, the friction coefficient being 0.02[illegible] in the one case and 0.0134 in the second (n = 0.015). The field tests show definitely the value of model testing in cases where special fittings make an estimate of losses from existing data uncertain, but they also seem to indicate that more theoretical and experimental data are required as to the mechanism of energy loss if the quantitative results are to be used to the best advantage.

INTRODUCTION

The outlet works of the Boulder Dam, of which the hydraulic testing of one unit forms the subject of this report, consist of four independent structures which are located in pairs on either side of the canyon, designated as the upper and lower Nevada and the upper and lower Arizona outlet works. (See plate No. 1.) Each comprises a 30-foot inside diameter vertical intake tower situated in the reservoir from the bottom of which a 30-foot pipe leads to the powerhouse manifold. Beyond the manifold the pipe is reduced to 25 feet inside diameter changing to another manifold from which six 8-foot 6-inch pipes discharge through balanced needle valves to the atmosphere. The original hydraulic design contemplated a total maximum discharge through the entire works of 121,600 c.f.s., 30,600 c.f.s. passing through the powerhouse units and 91,000 c.f.s. through 24 waste valves. The maximum head anticipated was 570 feet and maximum velocities approaching 100 feet per second were involved. To forecast the discharge from such an installation and to dimension economically its parts, requires the ability to estimate accurately the hydraulic loss coefficients of its intake, bends, straight pipe and valves. Such forecasting is not difficult in ordinary installations. Adequate data are available for estimating friction losses of straight pipe and although data dealing with fittings are not so satisfactory, losses caused by them are usually of minor importance compared with the total. At Boulder, on the contrary, it was estimated that entrance, bend, and branch-connection losses made up 35 percent to 45 percent of the entire loss. Few data were available to show how such losses might vary in large pipes at high velocities. Even estimating straight pipe losses involved extrapolation by the empirical formula of hydraulics in which the data are limited to tests on 12- and 16-foot diameter pipes with velocities of 10 feet per second, (Reynolds number of the order of 15,000,000) to values five times greater. To reduce the uncertainties to the greatest extent possible, as well as to determine loss coefficients for certain unusual fittings, which it was proposed to use, a number of hydraulic model tests were made[1].

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[1]Model Studies of Penstocks and Outlet Works, Bul. 2, Part VI, Hydraulic Investigations, Boulder Canyon Project Final Reports.

Model tests can only be wholly satisfactorily extrapolated to the prototype when the hydraulic laws which cause the losses are fully known. These have not yet been developed for entrances, bends, and branches. Thus even after model tests, estimates for these items are subject to considerable uncertainty.

For example, Thoma[2] in 1929 presented branch losses measured

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[2] Dr. Ing. Thoma, Hydraulic Losses in Pipes; Paper 26, Transactions World Power Conference, Tokyo 1929, vol. 2, pp. 446-72.

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in a 0.6-inch pipe leading from a 1.75-inch manifold (Reynolds number about 225,000). The results of the Bureau's tests with a model similar to his but six times the size (R = 600,000) showed only six-tenths the loss he found, though a decrease with increase in Reynolds number was neither indicated from his data nor from the Bureau's tests. What the coefficient of loss would be for a prototype 30 times still larger, clearly remained in doubt. It was resolved, therefore, that when the field performance tests of the needle valves were run, as much additional information bearing on the performance of the installation as a whole should be obtained as might be found convenient.

This paper reports the tests with information of the following hydraulic features:

(1) Needle-valve coefficients of discharge at various openings and heads; (2) relative discharge of different fully open needle valves with various numbers of valves discharging; (3) velocity distribution in pipe at entrance to needle valve; (4) coefficient of discharge of intake tower in terms of piezometric drop from the lake to the inside of the tower; (5) hydraulic friction factors; (6) coefficients of loss for a combined entrance bend and for a branch intake, and finally (7) an over-all check of the performance of the outlet works as compared to the results obtained from the model.

The Upper Arizona outlet works was chosen as the most suitable location for making the experiments. Here no main units yet were operating and by using their penstocks as piezometric connections to the main pipe and by wasting through various combinations of the canyon-wall needle valves farther down the line, with discharges up to 20,000 second-feet a satisfactory determination of the hydraulic grade line could be made from which fitting losses could be segregated. Three groups of tests were made: Those of 1937, 1938, and 1939, the first having already been provisionally reported.[3] The 1937 tests included complete pitot

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[3]"Memorandum to Chief Engineer, John Parmakian, May 1, 1937." Bureau of Reclamation.

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traverses for 20 different valve settings of a single 84-inch waste valve with a static head of 212 feet; the 1938 tests included a pitot traverse for the fully opened valve, plus numerous piezometric profiles, using various combinations of the valves discharging fully open: The static head was 281 feet; and the tests of 1939, most accurate of all, using improved equipment and technique, included a final pitot traverse at a gross head of 336 feet and supplied various data missing in the earlier experiments.

In this memorandum the three sets of tests are treated as a single series and are presented as a unit. First is given the experimental layout and the calibration of the instruments; next, the field procedure and summaries of the field measurements; then the discharge computations followed by the loss coefficients; next, a comparison of the results with the model tests, and, finally, the discussion and conclusions.

The field work of the 1937 and 1938 tests was directed by John Parmakian and that of 1939 by N. G. Noonan. Especial appreciation is due to R. E. Kennedy who made the lengthy computations and who was untiring in tracing down errors and devising means of consolidating the data into easily appreciated form. The model was tested under the direction of Jacob E. Warnock. All design and research work is carried on under the direction of J. L. Savage, Chief Designing Engineer of the Bureau of Reclamation. All engineering and construction work was under the direction of the late Chief Engineer, R. F. Walter, and all activities of the Bureau are under the general charge of John C. Page, Commissioner, with headquarters at Washington, D. C.

EXPERIMENTAL LAYOUT

The Upper Arizona outlet works as tested (see plate 1) is essentially as follows: A 30-foot inside diameter intake tower admits water horizontally by means of radial ports controlled by the upper cylindrical gate; the water than turns and flows vertically downward a distance of 5.2 D; enters a 90-degree vertical bend (R/D = 2-1/3), turns after two-thirds diameter of tangent through a 40-degree horizontal bend (R/D = 4), a total distance of 7.4 D, then passes through 10.9 D of straight 30-foot inside diameter steel welded pipe to the first of the four 13-foot, 75-degree penstock offtakes to the powerhouse, spaced about 5 D apart. One and one-half diameters from the last of these, the main penstock is reduced to 25 feet in diameter, the reducer leading through 7.6 D of straight pipe to a manifold from which, at an angle of 60°, six 8-foot 6-inch branches 22 D long (pipe intersection to valve face = 188 feet) discharge by means of a reducer through 84-inch needle valves. The main powerhouse units were not operating at the time of the tests so that their individual penstocks could be used as piezometric connections to the main pipe. In a similar manner, by discharging through waste valve V-A2 instead of V-A1, V-A1 penstock could be used to obtain the piezometric grade in the manifold. Plate 2 indicates the elaborate system of over a mile of piping and 50 valves as installed for the 1938 tests so that differential measurements by mercury manometers could be taken between the intake tower, powerhouse, and valve house, structures nearly one-quarter of a mile apart having a maximum difference of 500 feet in elevation. In addition to the differential manometers, numerous calibrated Bourdon gages were used for obtaining total pressures. The gaging station for the pitot traverses by which the discharge was measured was just upstream of V-A1 valve in the 1937 and 1938 tests and in a similar position on V-A4 for the 1939 test. Both V-A1 and V-A6 were equipped with a set of piezometer rings 125 feet apart for measuring the friction losses with individual connections to each separate piezometric opening so that points could be read separately. The drop of the water surface within the tower intake was read by direct measurement by a steel tape from a bench mark.

INSTRUMENTS AND THEIR CALIBRATION

Pitot tubes. The discharge measurements for determining the efficiency of the turbines made entirely apart from the present tests were obtained by the Gibson method (tests of October 12, 1937 on N-3 by Norman R. Gibson). It was desirable, in the case of the outlet works, to obtain in addition to the discharge, the velocity distribution in the 96-inch approach conduit to the needle valves. For this purpose a rugged, impact-static pitot tube was designed five-eighths-inch diameter and 14 inches long (plate 2). Velocities up to 85 feet per second, as well as considerable turbulence, were anticipated. To withstand the forces involved, a pair of streamline struts 2.5 inches thick at their midsection and set at right angles to one another were bolted into the conduit, just upstream from the needle valve with their centers supported on its point. In the leading edges of these struts a groove was cut in which the pitot tubes rode as they traversed the section.

A commercial replica of the pitot-static tubes used in the field was calibrated with the streamline strut attached in a 300-foot, straight, still-water rating channel 6 feet wide, 4 feet deep at the hydraulic laboratory of the Colorado Agricultural Experiment Station prior to the tests of 1937 and again in 1938. The calibration was made by mounting the tube on a car driven by an induction motor. The travel of the car, simultaneously with the time, was electrically recorded on a tape. A man riding on the car recorded one to six readings of a differential water manometer connected across the impact and static legs of the tube during the midhundred feet of the car travel. An investigation of the accuracy of this method of rating showed that while the time was accurate to one-tenth of one percent, the rate of travel of the car varied by about two and one-half percent on account of the varying pull of the power cable dragging behind the car. Since the exact time at which manometer readings were taken was not recorded, the possibility of the maximum error of about one and one-fourth percent exists in the tube coefficient as obtained from a single run. Analysis of the computed results from several runs indicates a probable precision of about 0.3 percent in the mean coefficients as determined from a single series of runs at

1 1/4% may error in pitot tubes

In this report unless otherwise specified, stated figures expressing the reliability of the result are the writer's estimate of the probable error. In most cases enough data are not available for this to be anything but a guess but, in the writer's opinion, in these days of prolific hydraulic experiment, much of it conflicting, there is an advantage to the reader in having the author give some indication of the reliability he places on his work. In using the probable error as an index of precision, it should be kept in mind that the true error may frequently be twice this amount.

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varying velocities, and of about 0.8 percent in the coefficients as obtained by different observers on different days.

The tube was rated first, for varying degrees of pitch and yaw with respect to the channel axis, since the gaging station at Boulder is just upstream from the needle valve and consequently the streamlines passing the section were slightly curved[4];

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[4] C. N. Zangar - "Stream Lines Through Boulder Dam Needle Valves," June 17, 1938.

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second, it was rated for wall effect, that is, to determine the change in coefficients as the tube neared the wall; and third, a determination was made of the effect of arranging the one-sixteenth-inch static openings in three rings with four openings per ring, as compared to three rings with two openings each in a plane normal to the pitot handle. The latter construction was that of the tube rated in 1937 and reported in the 1937 memorandum, although the tube as used in the field, both in 1937 and 1938, had the three sets of four holes.

The results of the rating tests are shown on plate 2. The tests indicated that the added static holes increased the coefficient from 0.918 to 0.933. This conclusion must be accepted with reserve and with consideration as to the accuracy of the tests. It is seen that up to 10 degrees there is small change in the coefficient with varying pitch and yaw. The effect on the coefficient as the tube neared the wall was, however, pronounced. In the prototype a 1- by 10- by 10-inch plate was required to fasten the streamline strut to the pipe wall, the upstream edge of the plate being about 3 inches downstream from the static openings of the tube. A reflected full-scale model of the wall plate, strut, and pitot tube was rated and it was found that when the pitot tube was located 2-7/16 and 7-13/16 inches from the wall, the pitot coefficients were 0.878 and 0.922, respectively, as compared to a value of 0.933 for a location free from disturbance. It is estimated the corresponding values for the prototype will be 0.893 and 0.925 on account of the high value of the velocity gradient as the wall is approached. Finally, all coefficients were increased by 0.01 as an allowance for the slightly curved stream lines.

Following the 1937 rating tests in a still-water channel, a test for the constancy of the tube coefficient was made by placing the tube without its supporting strut in the center of a pipe and subjecting it to velocities up to 70 feet per second. No variation of the coefficient with velocity was found, though, of course, the coefficient itself was altered.

An impact-static tube is an instrument designed to measure the difference between the total energy and the pressure at a given point in the cross section of a conduit. The impact leg, as is well known, measures with negligible error the total energy. On the other hand, the static opening, instead of recording the pressure as it exists in the stream prior to the introduction of the instrument, records the pressure in those particular filaments passing the openings as altered by the introduction of the instrument and its supporting struts. In making ratings in an open still-water channel, what is obtained is not a coefficient, which applied to the impact-static reading will give the velocity in the immediate vicinity of the tube, but a coefficient which represents the mean velocity of the section. The ratio which the velocity in the filament passing the impact opening bears to this mean velocity varies with the particular shape and arrangement of the pitot tube and its supports and on the percentage obstruction which they offer in respect to the total cross section.

The streamline struts used to support the tube in the 8-foot diameter pipe at Boulder actually occupied 6-2/3 percent of the gross cross section as compared with two percent which existed in the rating channel. A two-dimensional investigation by the electric analogy method[5] for the effect of varying percentages

[5] Memorandum J. E. Soehrens - Pitot tube calibration for Boulder Dam outlet works, April 5, 1940.

of obstruction on the coefficient indicated (plate 2) the need of increasing the coefficient by a mean amount of two percent for use in the field. This increase was applied as a variable correction at each gaging station (table 1 and plate 2) assuming the water to flow in annular rings in each of which the struts offered a different percentage obstruction.

The preliminary 1937 report used a constant coefficient of $C_p = 0.92$ to all readings. The comparable coefficient from the present study making all the connections and including the effect of the struts is $C_p = 0.95$. This applies to the formula

$$Q = \frac{A}{C_p} \sqrt{2gH}$$

where

A = gross area of section, C_p = velocity coefficient, and

H = mean of the impact-static readings (20 in all) taken at the centers of five equal concentric areas in two traverses at right angles to one another across the pipe.

It is believed, in spite of uncertainties involved in determining a pitot tube coefficient from a rating in a still-water channel at velocities limited to 15 feet per second for application in turbulent water at high velocity, at a location where nonparallel flow exists and with struts in the pipe to support the tube, the maximum range of error in the rating is not likely to exceed plus or minus one and one-half percent with a probable error of perhaps one-half of one percent. Certain data confirming this estimate will be given later.

Pressure-recording instruments, calibration, and corrections. The hydraulic measurements at Boulder involved differential heads up to 100 feet of water and total heads of the order of 350 feet. Although it would have been desirable to have had special experimental instruments for the measurements, it was decided to make use of the commercial Bourdon-type pressure gages bought in connection with the operating equipment. These had 6-inch dials either graduated in 5-foot units up to 500 feet of water or the equivalent in pounds with divisions about 3/16 of an inch apart. Readings were interpolated to the nearest foot.

The gages were calibrated prior to the tests in a dead-weight tester using water at 94° F. One-pound weights were added until the gages read at even divisions. Due to lost motion in the instruments the calibration with increasing loads as compared to decreasing loads differed by as much as three feet. The corrections were not constant over the range of the gage, varying up to 20 feet but were sensibly constant for the range of field measurements. Analysis of the errors of the corrected field readings indicates a probable precision of about 0.8 foot for a single reading, the range of error being ± 3.5 feet. The extent to which this error was reduced by repeated observation and by the plot of graphs is taken up in the detailed discussion of the results. The 1938 tests, in addition to the Bourdon gages, made use of differential mercury manometers capable of reading up to 20 feet of water. These could be interpolated to 0.1 inch of mercury (roughly 0.1 foot of water) with a probable error of about this amount.

The gages were connected to the lower ends of the individual penstocks which served the turbines and needle valves, and which took off at an average angle of 65° from the manifold. The turbines and valves were not operated at the times readings of the gages were made. Sixteen runs from the model* showed that the

*This model did not have the two vertical 12-inch diameter ties used for structural reasons in the prototype. It is believed these would not have affected the pressure registration of the gages.

pressures registered at the end of the penstocks acting as piezometers resulted in the gages reading $0.04h_v$ in excess of the mean pressure at the junction as determined from piezometers on each side. Corrections of this amount were applied to the penstock readings.

FIELD PROCEDURE AND SUMMARIZED FIELD MEASUREMENTS

General. The field measurements were made under handicaps of limited time, widely separated observation points, and non-technical observers, the latter consisting of nine men taken from the maintenance crew of the power station. Although detailed test programs were set up, valves were labeled, and telephone communications were installed, it was impossible for the engineer in charge to personally make sure instructions were exactly carried out and, as a result, in the field records there appear to be missing entries, reversed manometer readings, etc. The fact that, except for the 1937 tests, it was necessary to keep in operation a small turbo-generator diverting through penstock P-A8, up to 800 second-feet of water, introduced a variable, which while not important in the final accuracy of the results, made it difficult to check readings in the field and complicated computations in the office. This quantity of water equalled 25 to 2 percent of the water being measured, dependent upon the number of valves in operation.

Individual test runs were conducted in the following manner: First the required number of 84-inch needle valves was opened to a fixed reading on their indicating dials, usually the 95-percent reading. This opening could be controlled to within about one-half of one percent. After a wait of five minutes for the flow to become steady, telephoned instructions were given to the various observation points for readings to be taken. Next, piezometric lines were bled to eliminate air and to enliven the gages, then the mercury manometers were successively placed in circuit with the desired differential points of measurement by opening and closing the appropriate gate valves. For example, (see plate 1) by opening the powerhouse manometer between P-A1 and P-A6, the loss in head in 10.5 D of 30-foot pipe was obtained; by placing it between P-A6 and V-A1 with V-A2 discharging, the loss across the 30 by 25 reducer could be read. Fluctuations of 4 or 5 inches in the mercury columns were dampened by partially closing the line valves. After the manometer readings, the static Bourdon gages were likewise connected to the appropriate points, read, and the readings checked by another observer.

The 1937 Tests

(April 1937; reservoir elevation 1033; lower gate intake tower open; no units operating; water temperature 55° F. ±, gaging on V-A1.)

Purpose. The 1937 tests were arranged for the following purposes:

(1) To obtain by complete pitot traverses, for each 5-percent valve opening, the complete discharge performance of the 84-inch valve on V-A1 conduit; simultaneously with these readings were measured the piezometric drops between two piezometric rings placed 125 feet apart on the same conduit, to obtain a permanently calibrated measuring section.

(2) To obtain the piezometric drop between two piezometric rings 125 feet apart on the V-A6 conduit at varying valve openings; a comparison of this drop with that of V-A1 was expected to rate valve V-A6.

(3) To obtain the total discharge for the entire canyon-wall outlet works for any combination of the six valves by observing the simultaneous piezometric drops of V-A1 and V-A6, and from those interpolating the discharge of the intermediate valves.

The original data of the tests are given in "Report of Tests on Arizona Canyon Outlet Works by John Parmakian, May 1, 1937" and a summary, in which the original readings are converted into elevations in feet, is given in tables 3, 4, and 5. The observations are "dead-weight-corrected," the corrections having been applied at the time of taking the observations from a table of errors obtained from the dead-weight gage calibrations. It will be noted that the elevations for zero flow do not coincide with the lake elevations. From the graphs it is estimated that 0.5 foot and 1.0 foot must be added to the pitot and piezometric gage elevations, respectively, to obtain true values. Both field readings and dead-weight corrections were made to the nearest half a pound (1.2 ft.), which means that from this cause along a single reading may have an error of 0.6 foot. Larger errors doubtless occurred since, for example, under zero flow conditions, both the piezometer gage on the penstock and the gage on impact leg of the pitot tube were reported to read 84 pounds (194 ft.), whereas the zeros of the gages differed by 1.1 feet. Individual total pressure gages were connected to read the impact end pressure openings of the pitot tube and the same applies to the two piezometer rings placed for determining the drop in the conduit. Since for the fully opened valve the mean impact-static difference in the first case was 25 pounds, and 7.5 pounds for the second, a 0.5-pound error in a single reading would involve corresponding errors in the velocity of 1 and 3⅓ percent, respectively. At half-gate the errors would be approximately twice these values. Multiple readings ordinarily tend to increase the accuracy but in the case of the piezometer rings even though the four connections were read separately, with gages calibrated to read the same, errors less than half a pound are apt to remain. Measurements below half-gate, therefore, may be considerably in error. On the other hand, the pitot gagings, with two different traverses across the pipe, each involved 20 varying impact-static differences so that the error in the mean velocity should be only of the order of one-quarter of that obtained from a single reading.

The 1938 Tests

(April 1938; reservoir elevation 1102.3 - 1103.5; upper gate of intake tower open; station units using 300-800 second-feet; water temperature 54° F.; pitot gaging of V-A1.)

Purpose. Preliminary computation from the 1937 data indicated an unexpectedly large friction loss as well as an unexplainable distribution of velocity in the gaged section. It was decided, therefore, that the 1938 tests should be comprehensive and include the following:

(1) A complete gaging of penstock V-A1 with the valve fully open for comparison with the 1937 gagings.

(2) Complete determinations of the hydraulic grade line from the lake to the upstream face of the valves for various combinations of valves discharging fully open, the relative discharge of the various valves to be determined by comparison of the relative pressures existing at each valve.

The equipment and rating of the apparatus for these tests have already been described. With numerous direct readings of Bourdon gages to check the differential manometer readings, it was expected that the tests would be accurate and complete. The tests extended over a period of three days and fully occupied the time of the nine men engaged so that it was impossible to make computations in the field. It was only when computations were made in the Denver office that it became obvious that some of the equipment had failed to function and that, in many cases, the data were incomplete. By plotting and cross checking, it has proved possible to eliminate errors and to obtain surprisingly consistent results. Nevertheless, in presenting the data, it is felt appropriate to mention some of the difficulties both so that the probable accuracy of the work can be judged and, in the future, similar trouble may be avoided.

Reference to plate 1 will make clear the location of the various points for which field data are tabulated. In the summarized results the readings from various runs are averaged in accord with the number of valves operating regardless of the particular valves open; that is, the total flow is considered the same with V-A1 and V-A3 open as with V-A5 and V-A6 discharging. Model tests showed this assumption to be true within an error of about one-half of a percent which is beyond the accuracy of the present tests.

Piezometric drops from lake to intake tower. The water surface elevations within the intake tower were obtained by measuring down from a bench mark to the water surface with a steel tape and were recorded to the nearest hundredth of a foot. Zero flow readings could not be taken as the exciter was continuously on the line drawing a small amount of water. Fluctuations of the water surface within the tower increased with the flow, reaching a maximum of 0.1 foot for all gates open. For one valve discharging, the drop of the water surface through the screens was measured as less than 1/32 inch, which indicates that for six valves open the loss would be less than 0.07 foot or 0.008 h_v (velocity head is given in terms of the 30-foot pipe) which can be neglected in the computations.

It had been expected that the lake levels would be obtained from the continuous records of the automatic gage registering in the powerhouse and that they would be checked by the Bourdon gage on P-A2, which was connected to the lake through the unused lower Arizona manifold and intake tower. Unexplainably, both these gages went out of commission. As no zero flow readings of any of the other gages were taken, except for a single direct measurement within the intake tower at the beginning of the test, later elevations of the lake had to be obtained by interpolation between this reading and one made two days later, within which period the lake rose two feet. To obtain intermediate lake elevations, allowance had to be made for a change in the rate of lake rise due to the occurrence of an upstream flood which, according to a study, required about a day to reach the reservoir. These facts result in a range of uncertainty in the lake elevations estimated at about 0.05 foot for the first day and 0.10 foot for the second day of the tests, the rise in the lake being estimated at 1.2 feet.

Field measurements of the drop in the water surface between the lake and the interior of the tower have been plotted on plate 3 against the piezometric drops between P-A1 and P-A6 which were measured with a mercury manometer. The maximum loss measured for the latter, 3.98 feet, is estimated to have an accuracy of about one percent. While the straight line drawn in the graph gives a good approximation to the points, indicating but small errors in measurements, a curve slightly concave downwards fits equally well, and it is impossible to choose between them.

<u>Piezometric drop, lake to mean - P-A1 - P-A6.</u> The piezometric grade in the 30-foot penstock header at a point midway between the P-A1 and P-A6 penstocks was determined by using the mean readings of these respective gages. This procedure was used to reduce the errors inevitable in using single gage readings. Table 6 shows the individual gage readings as corrected from the initial calibration, the manometric readings between P-A1 and P-A6, and the computation giving the piezometric drops from the lake to P-A1 and P-A6. On plate 3, the drop from the lake to the mean of the P-A1 and P-A6 gages, is plotted against the measured drop within the tower, from which a constant correction to the mean gage readings of - 2.4 feet is indicated. Separate analysis shows P-A1 gage read 1.3 feet low and P-A6 gage, 6.2 feet high. In table 6, as a limiting check, the drop from the lake to P-A2 as measured with a mercury manometer, corrected by 0.04 h_v, is also tabulated, and a plot shows general agreement with the mean drop as measured by the gages, though individual readings have large discrepancies.

The piezometric drop from the lake to P-A1 for the maximum flow measured was 22.1 feet; the combined error due to all causes is estimated at about 1.0 foot. This gives a probable error in the piezometric drop of about $4\frac{1}{2}$ percent at this station. Downstream, while any absolute error remains in the over-all loss, the percentage of error due to this measurement becomes less.

<u>Piezometric drop - P-A6 - V-A1.</u> The piezometric drop from P-A6 to the valve manifold was obtained by using the V-A1 penstock as a piezometer connection and then discharging through the other five valves. A mercury differential manometer was used giving consistent readings, the maximum drop measured for five valves being the equivalent of 14.4 feet of water. The drop measured by this manometer required an addition of 0.04 Δh_v, the difference between the velocity heads at V-A1 and P-A6, to give a value equivalent to that obtained with ordinary piezometer connections.

Approximately four diameters below P-A6 water was diverted through P-A8 to supply the exciter unit in an amount from 1[illegible] percent of the lowest flow to $1\frac{1}{2}$ percent of that existing for 6 valves open. The quantity diverted was measured by the performance curve of the exciter. Heavy percentage diversions introduce material energy losses below a junction and a rise in the piezometer grade but for small diversions, the model tests[6] showed

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[6]Boulder Canyon Project Final Reports, Part VI, Hydraulic Investigations, Bul. 2, p. 40.

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an energy gain reaching a maximum of one percent. The appropriate correction is small and uncertain and is neglected in the present computations.

The readings only included values for one, three, four, and five valves open. The missing readings were interpolated by plotting a graph of the P-A6 to V-A1 drop against P-A1 to P-A6 measurements reduced to the values equivalent to the flow in the valve manifold. This was done by multiplying the actual measurements by the square of the ratio of the discharge below the diversion to that above. The computations are shown in table 7 and the plot on plate 3.

Table 9 gives a summary of the corrected section drops and their summation and against this is plotted the drop as directly measured by the V-A1 Bourdon gage. After allowing for a zero flow correction of - 2.0 feet, the individual measurements are shown to agree with discrepancies of the order of 1.5 feet. This would represent 3.1 percent for the maximum measured drop of 47 feet. It is believed appropriate to mention at this point that the average constant found necessary to add to the readings of the six different Bourdon gages after their original dead-weight calibration was - 2.5 feet. Whether this change in correction from the dead-weight ratings was due to a change in the temperature from the 94-degree water of the calibration to the 54 degrees of the test, or whether it was due to gage handling or to other causes is not known. Corrections of the same order were found in the gages used in the 1937 tests which, however, were checked for no-flow conditions before and after being used.

Table 10 gives a summary of the pitot gage readings in V-A1. Tables 11 and 12 give the impact readings at the center of the pipe for varying numbers of valves discharging, and various mercury manometer readings showing the increase in pressure in the valve manifold due to the diversion of water to the valve conduits.

The 1938 tests did not include gagings at other than 95-percent opening nor did the field data include readings at zero flow so there is no method of determining gage constants for the pitot gages other than constants found necessary in all the other gages. An indirect method, computing the anticipated drops in V-A1 as more and more valves were opened, from the 1937 data indicated the impact gage read 0.8 foot low. Small confidence can be placed in the computation and with no means to check the static leg of the pitot, it seems possible that material-residual errors may exist in these readings. These are discussed later in a comparison of the results of the various gagings.

The 1939 Tests

(March 27, 1939; reservoir elevation 1157.1; upper gate of intake tower open; station unit using 617 c.f.s.; water temperature 61° F., room 80° F., pitot gaging of V-A4.)

Purpose. The primary purpose of the 1939 tests was to obtain an accurate pitot gaging of the discharge for the purpose of explaining discrepancies in the 1937 and 1938 results, and to provide a check on the discharges at partial gates by measuring the relative heavy losses to the impact leg of the pitot tube. It was also desired to rate the drop from the lake to the interior of the intake tower so that by this simple measurement the entire flow through the outlet works could be obtained regardless of the combination of power units and outlet valves discharging on the line. The test was made with the exciter diverting a constant amount of 617 second-feet from P-A8 penstock, an amount equivalent to ~~160~~ 68 percent of the flow of the valve at 10-percent opening and 16 percent at 95-percent opening, the opening for which complete pitot traverses were made. Since with only one valve operating the lost head from the lake to the junction in the valve header is negligible, the diversion does not affect the accuracy of the lost head coefficients in the valve branch, but any errors in the reported quantity of water diverted through P-A8 would affect the tower rating at small gate openings.

The experience of previous gagings was utilized to obtain a greatly improved experimental set-up. Instead of measuring the impact and static pressures separately by different Bourdon gages, the two gages were arranged so that both were connected first to the impact opening and then immediately afterwards to the static opening. This minimized calibration errors and gave two independent measurements of pitot-static differences at each gaging point. The gages also were deliberately set to different zero gage readings to eliminate any tendency on the part of the observers to record the same readings. From the results it is estimated that the pitot difference obtained at a given point from the pair of gages had a probable error of about three percent. In addition to the readings at the pitot station, the connections were so arranged that the same two gages could be connected to a piezometer manifold 0.46 diameter upstream from the pitot station, which connected with numerous openings into the upper half of the pipe. Upstream an additional 0.25 diameter from this point was the center line of the paradox gate in the hood of which another gage was connected. The paradox gate was open throughout the tests and is so designed that a pipe section replaces the gate when the latter is fully open. The clearance spaces around the pipe section in effect form a continuous piezometer manifold. All gages were read before and after the test to provide an absolute check on their calibration. The drop in the intake tower was directly measured from a bench mark with a steel tape, the readings being given to 1/16 inch (0.005 ft.). For the maximum drop measured this is equivalent to one percent.

The gage readings for the pitot traverses transformed to elevations are shown in table 13. Computations of discharge are given in table 14. Table 15 gives the readings for valve manifold, piezometer manifold, pitot readings for variable valve openings, and the measured drop from the lake to the water surface within the intake tower. (Gage constants have not been added.)

RESULTS OF DISCHARGE MEASUREMENTS

The Boulder field measurements were designed so that, based on the gaging of a single valve, it would be possible to determine the flow from any one valve or all six on the assumption that their coefficients of discharge were identical. This was done in the following manner:

(1) Separate gagings were made of valve V-A1 in 1937 and 1938, and of V-A4 in 1939, at gross heads at the manifold of 212.0, 281.0, and 336.0 feet, respectively. The results obtained were compared at the 1938 head by means of appropriate factors.

(2) As more valves were put into operation the discharge in the valve gaged, that is V-A1, decreases due to the increased head consumed in the main penstock. The discharges for V-A1 under these conditions were determined by a comparison of its relative impact heads, these being measured in the 1937 and 1938 tests (plate 3).

(3) The discharge of V-A6 with reference to V-A1 was determined from the 1937 tests by a comparison of the losses between two pairs of piezometer rings on V-A1 and V-A6 for which simultaneous measurements were made for various combinations of discharging valves. An alternative method compared the pressure readings just ahead of the valves. The intermediate valves' discharges were interpolated between the two outside valves by reference to the tests on the model.

(4) The discharge in the intake tower was obtained by adding the discharge of the exciter to the sum of the valve discharges.

(5) The discharge of a single valve at variable valve settings was obtained by a complete set of 20 gagings in the 1937 tests and by measurement of the velocities at two points in the cross section for each of 10 openings in the 1939 tests.

It is estimated that the relative accuracy of the 1939, 1937, and 1938 gagings is somewhat in the order of 4 to 2 to 1. For this reason the 1939 gagings are described first.

<u>1939 pitot gagings.</u> Ten observations taken somewhat at random were made on each of two traverses located at 45 degrees to the horizontal just upstream of valve V-A4. After applying the variable pitot coefficients appropriate to their position in the cross section, profiles of the velocity distribution were made and the mean velocity was determined from the profile. Plate 4

shows the distribution of the energy, velocity, and pressure across the pipe. It will be noted that the profiles from the two traverses are in reasonable agreement with one another. Plate 5 is a contour distribution looking downstream and typical of flow below a bend. As has been found by other observers[7], the high velocity current in the tailpiece has been thrown away from the center of curvature, the manifold junction being in effect a reducing bend. It is of interest that the nonuniform distribution should persist in well defined form 22 diameters downstream and through an 8.5- by 8.0-foot reducer. The profile clearly shows the cone of pressure produced upstream by the valve needle and the less than average pressure at the pipe wall. A satisfying check on the general accuracy of the pitot coefficients is afforded in table 15 by the agreement of the mean pressures computed from the pitot readings with the pressures separately measured at the wall manifolds 0.2 and 0.5 diameters upstream. From the latter location under maximum flow, the computed friction drop is about one foot. In comparing these piezometric grades, it should be kept in mind that the wall manifolds give only the point pressures at the wall, whereas the pitot tabulations are the mean of the pressures for the entire section. It is believed this agreement of pressures, independently measured, justifies placing a probable error in the velocity of about 0.7 percent and a maximum error of not more than 2.0 percent. For reference, the ratio of the mean to maximum velocity was 0.902, the mean pitot coefficient 0.95 and the pipe diameter 7.98 feet, giving a discharge of 3,855 cubic feet per second at a velocity of 77.1 feet per second. The valve coefficient based on gross outlet area has a value of 0.732.

<u>1937 pitot gagings.</u> Mention has already been made that the 1937 gagings of valve V-A1 were made with the handicap of having to use a limited amount of commercial equipment. For example, under zero flow the piezometer gages and pitot gages all read 84 pounds which, added to the elevations of their different locations, gave lake levels of 1031.4 and 1032.5, although the lake was actually at elevation 1033. This type of error was partially eliminated by velocity-head piezometric-drop plots to obtain the gage constants, but the scatter of the plot was such that con-

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7 James Williams, Considerations on flow in large pipes, conduits, tunnels, bends, and siphons; Jour.-Inst. C.E.; ~~vol. 11,~~ April 1939, p. 451.

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stant errors of the order of 0.5 foot may remain.

The pitot gagings consisted of 20 observations located at the center of gravity of five divisions of equal area, the discharge being computed from the mean of the velocities thus determined. No center-line readings were taken. The mean pitot coefficient found applicable was 0.95 instead of the 0.918 used in the preliminary report, the more comprehensive pitot ratings made later being responsible for the change. Twenty complete gagings were made corresponding to five-percent increments in valve opening. The energy, velocity, and pressure distributions found in the test with the valve 95-percent open are plotted in light lines on plate 4. In spite of the fact that these distributions were consistent in all 20 of the gagings, they are not in agreement with the 1939 observations made with improved technique nor with theory. It seems impossible that the pressure at the wall was 0.40 velocity head head higher than at a location one foot distant. If it could be considered that in tabulation the readings became reversed with respect to the center line, there would be reasonable agreement with the 1939 results. That any such interchange occurred is considered impossible by the engineer in charge. Since the original notes no longer exist, the results can only be presented as found.

Whether or not the individual points and distributions are in error, it is believed some weight can be given to the discharge obtained from the mean of 20 readings. In the upper part of table 19 are tabulated for comparison the results of the 1937, 1938, and 1939 gagings. The 1937 coefficient of loss from the manifold to the impact leg of the pitot tube is given as K = 0.65, the value found for the 95-percent valve opening. However, a comparison of the coefficients for the other 19 gate openings plotted against Reynolds number, indicates that the coefficients for both the 95 and 100-percent openings were too high. This is indirectly shown by curve F of plate 7. The value which best agrees with the other data is 0.675 which compared with 0.61 found in the 1939 tests, is believed to indicate satisfactory agreement. However, the gaging results are considered definitely low by one or two percent. The following summarize the 1937 results for the valve 95-percent open: Velocity 64 feet per second; discharge 3,195 cubic feet per second; valve coefficient 0.77.

<u>1938 pitot gagings.</u> The 1938 gagings consisted of a pair of traverses at valve V-A1 with independent gages read to the nearest foot. As was true of the other gages of the tests of this series, the gages were calibrated against a dead-weight tester

prior to installation, but no observations were made afterwards of their readings under zero flow conditions. All the other gages used in the series were found to require the use of gage constants from one to five feet to bring them into agreement with the other data. No adequate means for determining the constants of the impact and static gages of the pitot tube are available so that there is considerable uncertainty in their results. The observations and computations of the discharge were similar to those made for the 1937 tests, and the distributions found are plotted on plate 4 for ready comparison with the other measurements. It is seen that, on the whole, they agree with neither the 1937 or 1939 measurements. In contrast to the 1937 tests with multiple gagings at various valve openings, and the 1939 tests made with improved equipment, the 1938 gagings lack supporting data as to their reliability. Moreover, a comparison of the loss and discharge coefficients obtained in each of the series of tests (table 9) seems to indicate that the 1938 gaging was from 3 to 5 percent low. This would correspond to 4 to 6 feet differences between the constants of the two legs of the pitot, values of the order to be expected from the analysis of the other gages of this series.

Since the 1938 tests were the only ones for which the piezometric profiles of the outlet works as a whole are available, it was decided to use in the computations of the loss coefficients, discharge values as obtained by weighing the various pertinent data rather than using the discharge as measured. This was done indirectly by interpolating between the 1937 and 1939 results, making use of the measured friction losses to the impact leg of the pitot tube and the computed valve coefficients, by a comparison of the measured friction losses between the ring piezometers 125 feet apart on V-Al calibrated in the 1937 tests and by a comparison of the relative drops from the lake to the tower intake in the 1939 and 1938 tests. It will be observed that different methods of interpolations shown in table 20 give results varying by about one percent from the value assumed as most probable, but this must be expected, corresponding as it does to the approximate probable error in the measured quantities. On the other hand, if the interpolation is based on the relative drops in the intake tower as measured in 1939 and 1938 when one and six valves were respectively discharging, a method of comparatively high precision, then the discharge is about 3.8 percent greater than the value chosen. Although this is a possible value, in the face of the balance of the data the lower value is used. For reference, the following are the results assumed applicable to the 1938 tests with V-Al alone discharging 95-percent open:

Q = 3,540 c.f.s.; coefficient of discharge, 0.735.

Discharge when more than one valve was open. The discharge of V-A1, when other valves were likewise discharging from the manifold, was obtained by comparison of its different impact heads measured on the center line. Only a single observation for each combination of valves operating was made. For this reason, the relative impact heads of V-A1 as found in the 1937 tests was compared with the 1938 observations. This required rather extensive computations to correct for upstream diversion, some valves 100-percent open, variable reservoir elevation, etc. The results are plotted on plate 3, from which it appears that the two series of tests were in good agreement, except for the combination with five valves open. An inspection of the 1938 data clearly indicates this observation to have been in error. From the curve approximating the results of the two series, the relative discharge of V-A1 as a function of the number of valves operating was obtained applicable to 1938 conditions.

Discharge of groups of valves with respect to V-A1. The friction drop between piezometer rings located 125 feet apart on V-A6 was compared with that measured in a similar location on V-A1 to obtain the relative flow in V-A6 with respect to V-A1. Such comparison lacks accuracy both because the drops compared, of the order of 16 feet, were small and measured by differences between Bourdon gages and because with different type entrances to the respective conduits, the entrance losses may affect the measured drops.

Plate 8 shows moreover, based on tests both of the model and prototype, that even with the same entrance, the entrance loss varies inversely with the rates of the quantity of water diverted through the branch to the quantity flowing in the manifold.

The question as to the magnitude of errors introduced by entrance and fitting losses in reported straight pipe losses will be discussed later, but the use of a calibrated section of pipe as a metering section in branches taking off from manifolds must be considered questionable. The relationship between the flows in V-A1 and V-A6 as measured by the tests of 1937, when all valves were 100-percent open, are considered applicable for all valves 95-percent open, which was the general condition of the 1938 tests. For certain of the valve combinations, however, some valves were 100-percent open for which an allowance of one-half percent per valve was made. The various coefficients and final discharges as used in the computations are shown in table 21.

It is recognized that the method used for obtaining the discharges of the intermediate valves lacks accuracy. However, differences between the discharges of various valves in a series taking off from a manifold are small and if the discharges from the outside two of a group are directly known, the average is unlikely to be greatly in error. Internal evidence of the tests indicate reasonable precision.

It is concluded, finally, that while the 1939 discharge of a single valve has a probable accuracy of about 0.7 percent, the error in the discharges for the various groupings of valves in the 1938 tests is between 1.0 and 2.0 percent with maximum errors unlikely to exceed ± 4 percent.

RESULTS OF ENERGY GRADE MEASUREMENTS

The Reynolds number - Loss chart. The results of the hydraulic grade measurements are shown in the form of loss coefficients "f" in the Weisbach-Darcy formula, plotted against Reynolds number (plate 7). The three measuring sections, lake to P-A1, lake to P-A6, and lake to V-A1, are 23, 34, and 57 equivalent diameters long, respectively, and the coefficients reflect the total loss to these points including friction, entrance, and fittings. It will be noted that at increasing distances from the intake, the mean loss per diameter decreases, showing the increasing dissipation of entrance and bend losses. The curves drawn to average the observations taken at five different velocities indicate reasonable precision, except for the section P-A6 to V-A1 in which two points at the lower velocities seem to be out of line; a possible explanation will be offered later.

One of the main purposes of these tests, of considerable theoretical as well as practical importance, was to determine whether at high Reynolds numbers the losses followed the quadratic law or varied at some other power of the velocity. Logarithmic friction formulas have been satisfactory within the range of their tests but for extrapolation they have given widely different results. Nikuradse showed, with systematic tests on pipes roughened with sand grains of uniform diameter that the exponent of the velocity in the loss equations starting with a value of about 1.7, with increasing velocities gradually reached a maximum of about 2.2 before decreasing to the square and becoming constant. A portion of his results are shown on plate 7.

Recently Colebrook[8], making use of modern theoretical and

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[8]Cyril Frank Colebrook - Turbulent flow in pipes, with particular reference to transition region between smooth and rough pipe laws; Jour. Inst. C. E., Feb. 1939.

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experimental knowledge of the nature of frictional resistance, bridged the gap between these rough pipe experiments and smooth pipe, and showed that where the roughness is nonuniform there is a gradual transition from one law to the other, the velocity exponent gradually increasing to its final value of two without passing through a maximum. His formula, which has a substantial theoretical background indicates that the quadratic law is reached at a roughness Reynolds number* of the order of 60 to 80.

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*The roughness Reynolds number is the shear force velocity (or wall velocity, V_*) $= \sqrt{\frac{\tau}{\rho}}$ multiplied by the average wall protuberance divided by the kinematic viscosity (ν). It is easily computed from its equivalent $\frac{V_* k}{\nu} = \left(\frac{f}{8}\right)^{\frac{1}{2}} \cdot \frac{k}{D} \cdot \frac{VD}{\nu}$. The value $\frac{k}{D}$, the relative roughness, can be obtained from the equation $\frac{1}{\sqrt{f}} = 2 \log 3.7 \frac{D}{k}$ after values of "f" have been found from experiment.

Prior to reducing the field measurements, it was impossible to forecast within which class of pipe the results would fall so every effort was made to eliminate bias in the reduction of the observations. This is not easy for to paraphrase Alexis Carrel in "Man, the Unknown," "when faced with many and imprecise data the temptation is great to choose among them those that please us and which conform to our feelings and belief." Hydraulic observations are variable at best and there is a temptation to assume that differences from an assumed law are to be charged to field errors. However, the group of curves summarizing the losses per diameter in the main penstock plotted on plate 7 do show a consistent trend downwards. The same trend is shown in the 1937 measurements of the losses in the V-A1 branch plotted on the same plate. These relative measurements here are more reliable than the 1938 measurements in the main penstock, the curve being based on 20 independent gagings. Friction tests made on four different 13-foot penstocks in connection with the efficiency tests of the turbines by the Gibson method are also shown. Since the velocity determinations for those tests are considered to have a high order of accuracy, the scatter of observations, about 20 percent from the means, must be considered as due to errors in the pressure observations made with a differential mercury gage. If a line is drawn approximating the observations for the four series made on different penstocks, it too, shows a decreasing trend with increase in velocity. However, considerations which will be discussed later, offer an alternative reason for the decrease. Here it is enough to say that the data of three independent series of tests show coefficients decreasing with increase of Reynolds number, the losses varying about as $V^{1.9}$. Since this is in conflict with Colebrook's formula, the roughness Reynolds numbers of the tests being between 200 and 300, the result should be used with caution.

Energy grade for model and prototype. A comparison of the energy grade, as found by the model, and as measured in the prototype is shown on plate 9. The energy grades are, in a sense, synthetic as conditions were not exactly the same in the two determinations. Two tie rods, 12 inches in diameter, required for structural reasons across each penstock opening in the prototype, were not included in the model. For the purpose of comparing the two results, an amount of 0.025 h_v per rod was assumed as the ultimate head loss per rod and 3/4 of this amount was added to the losses found in the model. This reduction was thought justified since, as will be shown later, with but a few diameters of pipe below the powerhouse manifold, the full losses probably would not have taken place in the model. Reference to the curve on plate 9 entitled "Cylinder Drag," from which the loss for the rods was estimated, is incidentally worth careful consideration by all those engaged in model tests of streamlined shapes and entrances. It can be seen, for example, that model tests on cylindrical shapes made within a range of Reynolds numbers from 2,000 to 25,000, would give totally erroneous results extrapolated for regions of R = 5,000,000. It is possible that the high coefficients of loss found at the low velocities in the prototype in the P-A6 to V-A1 reach previously mentioned, may have been caused by some abrupt change in the rod coefficients within the range of the tests. It has been stated elsewhere that the type of phenomena shown by the rod quite probably applies to shapes built of curves of many types, such as streamlined entrances or curved vanes.

The model test in addition to having been corrected for tie-rod losses has other features which need to be stated. The results shown are a combination of two different series of tests in which in the first, the losses were measured from the lake through the screens, through the upper gate to P-A1, and in the second, from the lake through the screens, through upper and lower gates to the canyon-wall needle valves. The screens for the model had been built to scale and indicated a hydraulic loss of the order of 0.10 h_v (30 ft. pipe). The prototype showed no measurable loss, certainly less than 0.01 h_{v30}. Investigation indicated that the velocity of water past the screens of the model was probably less than critical. The screen loss for the model was, therefore, subtracted from all model losses.

Two gates open at the intake tower give about one-fourth the entrance loss of one. To obtain an entire profile of the energy grade for the model comparable to that obtained in the prototype in 1938, in which the upper gate alone was open, the two tests of the model were combined, the first part of the profile coming from the first test and the lower part from the second. The two tests have a common profile between the bends and P-A1. This procedure has the defect that although a portion of the profile from each test was found similar, if the higher entrance losses in the model with one gate open would have affected the hydraulic grade at greater distances than 22D from the intake, then the lower part of the profile is less steep than it should be. It is believed any error is small.

Finally, before concluding this presentation of data, it is desired to re-emphasize certain things which must be kept in mind in reading the discussion which follows. Energy losses as spoken of in this report are purely differences in the energy grade line obtained by adding to the piezometric grade line the mean velocity head. The piezometric grade as measured by penstocks opening on only one side of the penstock, and with disturbed flow from fittings upstream, may not measure the mean pressure, the mean velocity head may be far from the true velocity head, and finally, the physical layout is such that errors tend to multiply. In most of the sections, the coefficient of loss due to fittings is obtained by subtracting from the measured piezometric drop a velocity head which is about one-half its size; from the remainder the friction is subtracted, which is again about half, and this quantity is divided by the velocity head. The result is that a one-percent error in the velocity produces a six-percent error in the loss coefficient. If the total piezometric drop is one percent in error in the same direction, there is an additional four-percent error in the coefficient and an additional one percent for each one-percent error in the friction coefficient.

The most accurate of the main penstock measurements is the loss from the lake to V-A1 which includes 57 effective diameters of 30-foot pipe. Assuming a 2-percent error for each of the items mentioned, the total coefficient as measured at this point might have a maximum error of 25 percent, with a probable error of perhaps 8 percent. At intermediate points, probable errors would be correspondingly greater.

It is regretted the probable error is so large but it is believed typical for layouts of this sort.

In spite of the possibility of large errors it is believed the tests have value, not only because they indicate provisions which must be made in the test facilities if better comparisons are to be made between model and prototype, but also because at least qualitatively certain phenomena are indicated which, if taken into account, should improve hydraulic design. This will be discussed in the next section.

DISCUSSION

General design for power plant and diversion tunnels differs from that of flow lines and municipal water supply, in that entrance and fitting losses make up a substantial portion of the total hydraulic losses, about 40 percent in the case of the Boulder penstocks. The penstock system also differs from a diversion tunnel in that in its dimensioning, no hydraulic factor of safety is possible. Its dimensions are based on making a minimum the cost of the installation plus the capitalized estimated value of lost head which, at Boulder was estimated at $110,000 per foot. Either an under estimate or over estimate of the hydraulic losses leads to an uneconomic size. While the curve of combined cost is comparatively flat, the increment in cost per foot of diameter, though small within a certain range, is enough to make desirable any changes in design which will lead to closer estimates. The present discussion has that in view. Before taking up in detail the experiments, it is proposed to briefly consider the nature of entrance losses.

Comparatively little experimental data seem to be available on the effect of size, velocity, and character of surface on loss coefficients for most forms of entrances and fittings. It has commonly been assumed that these vary with the square of the velocity and that the other factors are unimportant. Practically no data exist on the rate at which the loss takes place below the fitting but it has been stated that at 40 or 50 pipe diameters downstream, the disturbances have disappeared. On the other hand experimenters working with smooth pipes and well rounded entrances have found as much as 100 D were required to fully develop turbulence; a case is reported where due to a disturbance caused by a valve, irregularities in the velocity distribution were noted 200 D downstream in a 30-inch pipe line; and recent tests on bend losses have shown that the roughness of the pipe affects their magnitude.

Plate 10 is a diagram showing the percentage of total loss at varying diameters downstream for some tests made by the Bureau in which the total loss was measured some 30 diameters downstream. The losses are nominal, that is, they involve wall pressures and Q/A velocity heads. It is quite possible these functions are far from integrated pressure and velocity heads and, indeed, it is certain that they are in error at fractional diameters below an entrance. With the limited tests made and the exact determination of the curves of percentage losses a matter of much experimental difficulty, no explanation can be given of the differences. But they agree in indicating that substantial percentages of entrance loss are being dissipated up to 30 diameters downstream.

The Boulder installation, though laid out on a grand scale, if measured in diameters is typical of many power projects with fittings 10 to 15 D apart. Thus both from the point of view of design and experiment, something more than total loss coefficients must be available if the hydraulics are to be correctly designed and interpreted. The writer has not seen a complete theoretical discussion of fitting losses and their rate of dissipation. Such a paper, with a review of existing data and systematic experiments to fill in the gaps, would be valuable. Here all that can be done is to review briefly some of the well known phenomena on which losses must depend. To be specific, a circular flush intake is used as an example.

When water is freely discharged from a tank through a circular orifice striking against a flat plate, there results the following:

The jet contracts to 0.61 A where A equals the area of the orifice; it has a velocity corresponding to the total head and in being deflected 90° by the plate, it reacts against it with a force whose maximum intensity is h_v but whose total magnitude is equivalent to this intensity times twice the minimum area of the jet, $h_v \times 2$ [$0.61A$]. The fact that the relation between momentum and impulse requires a minimum area of at least twice that of the contracted section against which forces can act if the flowing water is to be completely decelerated is sometimes overlooked. Actually four to six times this area is required since the intensity of pressure on the area is not uniformly of maximum value. These same statements apply equally to acceleration at the intake. Since the orifice supplies only 1.6 the jet area there is a large area of wall surrounding the orifice which must be subjected to a reduced pressure. The maximum intensity of the reduction is at the edge of the orifice equal to the velocity head.

What has been said of the jet applies qualitatively to a practical intake connected with a pipe. Just below the throat of the intake there is a vena contracta whose shape depends upon the dimensions and forms of the intake. In contrast with the jet, if downstream conditions permit, less than atmospheric pressure can exist. Moreover, if (in the intake) departures from perfect streamlining occur in the form of sharp edges or short radius curves due to the requirements for gate seats, etc., the reduced pressure in a portion of the flowing stream theoretically will tend to approach absolute zero. For example, where the effective passage is equivalent to a curve with $R/D = 0.75$ the velocity head at the inside of bend is theoretically nine times the average, though actually it is reduced by the wall friction.

It is suggested that it is this tendency to vacuum around gate frames in front of those intakes over which the static head is only three or four times the velocity head, which tends to create and maintain vortices. These latter, to be stable, must extend from the water surface to a surface from which air is removed.

Below the throat of an intake the jet expands, thus involving curvature of the streamlines and a change in momentum. Since the latter can only be produced by an impulse, that is, by a force multiplied by time, and since in a straight pipe the side walls provide the only source of reaction, the shear at the pipe wall becomes a controlling factor. Analysis shows that the impulse received by each unit mass of water passing one diameter of pipe, which ultimately converts the high momentum of the jet into uniform momentum across the section, is solely dependent on the friction factor "f" and is independent of scale, from which the conclusion is drawn that in phenomena concerned with momentum, correspondence between model and prototype necessitates a common friction coefficient. This further leads to the supposition that if the friction coefficient of the model is twice that of the prototype, then twice as many diameters will be required for the prototype to produce the same change in momentum as observed in the model. But in both cases the ultimate total loss will be the same, being given by the formula $\left(\frac{V_1 - V_2}{2g}\right)^2$ integrated across the section, since it is believed this largely depends on the distribution of velocities at the contracted section.

Curvature of the streamlines means that velocities and pressures are not uniform in the pipe cross section and nodes of pressure can be expected at the wall of the pipe. These were noticed about three diameters apart in the model tests. The writer has not seen data showing the variation in pressure at varying diameters across the section below a contracted jet and it appears questionable as to what wall piezometers actually measure. Williamson (loc. cit.) suggests the wall pressure is higher than the mean; this appears reasonable before full turbulence is established, where the theory of streamlines, to a degree at least, is applicable.

What has been said about an intake also applies qualitatively to an elbow. Considering the dead end beyond a 90-degree angle branch through which all the water is being diverted, the cross section of the dead end has only one-half the area required to provide a reaction at a maximum intensity of pressure equal to the velocity head to decelerate the incoming water. The remainder of the decelerating force must act in the branch as excess pressure on the outside wall and reduced pressure at the inner wall. These forces combined with those in the dead end provide the total decelerating force but, since their line of action is not in common with that of the dynamic force of the water, there remain "moments" in the water which dissipate energy downstream.

The "moments" are largely eliminated, it is interesting to note, by the device of deflecting vanes. These at one and the same time provide excess area upon which decelerating forces can act, provide the area near the line of action of the forces, and deflect the water into the tailpiece with approximately normal distribution of momentum. It is suggested that a consideration of probable "moments" at entrances and branches might prove a useful tool in hydraulic design.

<u>Friction.</u> The general discussion of entrance losses has anteceded the presentation of the experimental data to emphasize the fact that in the usual power-plant installation all measure-

ments of hydraulic losses almost certainly contain a substantial percentage of loss due to fittings upstream. The friction coefficients for the Boulder experiments have to be derived from the Gibson tests which measured the friction in several sections of pipe 7.7 D long. In one of these A-8, the section was located 6.7 D below the entrance while for the others it was located 14.4 D distant. The question arises as to what percentage of the losses measured in these tests was due to the entrance loss at the junction upstream? These 13-foot penstocks joined the 30-foot manifold at an angle of 75° and were connected with it with one-quarter diameter fillets. Plate 8, curve 1-A, shows the entrance losses for this connection, known as the "streamline fitting;" curve 2-A shows the same for a cone entrance; both as determined by a Commercial Hydraulic Laboratory, the losses having been measured about 22 D downstream. The results are plotted against the $\frac{Q_s}{Q_a}$ ratio where Q_s equals the discharge in the branch and Q_a that in the pipe upstream from the diversion.

Above these curves are plotted losses for somewhat similar fittings, the work of other laboratories. Though all the tests were made in pipes classed as smooth, the differences in the results for only slight changes in the form of the branch connection are startling. For example, the filleted junction (R = 0.10 D) curve 1, tested by Thoma, is very much like the "streamlined" junction of curve 1-A but shows seven times the loss. Since Harris[9] has shown there is little reduction in entrance

[9]C. H. Harris, Elimination of Hydraulic Eddy Current Loss at Intake, Bul. No. 54, Engineering Experiment Station, University of Washington.

loss in increasing an entrance radius beyond 0.14 D, the difference in the radii of the connecting fillets in the two cases R = 0.10 D and R = 0.25 D can hardly be the explanation. Similarly, curve 2, an interpolated value from Thoma's tests on cone branches similar to the cone tested by the Commercial Laboratory curve 2-A, shows three times the loss.

The approximate formula for junction losses derived in bulletin 2 (loc. cit.) also shows about three times the loss. The formula approximately is

$$\text{Loss} = 0.50\, h_v \left(\frac{D_1}{D}\right)^4$$ where D_1 = diameter of branch and D equals the diameter at its intersection with the manifold.

These differences, far too large to be explained by changes in Reynolds numbers, angles, or pipe diameter ratios, can apparently only be explained as differences in laboratory technique. Mention has been made of the great experimental difficulty in obtaining precise measurements of fitting losses. Examination of the experimental procedure by which the loss for the streamline fitting was obtained, indicates it was of less precision than that used in the other tests. It is believed that the loss found by the Commercial Laboratory for the streamline junction used for the penstock connections in the prototype, 0.06 h_v is much too low and that in a repeat test a value of 0.15 to 0.20 h_v would be found. Although this particular junction was not tested in the prototype, a test was made of the manifold prototype and this showed good agreement with the Bureau model, rather than with the tests of the Commercial Laboratory.

The importance of knowing, within reasonable accuracy, the total junction loss lies in the fact that from plate 10 it can be estimated that with a total entrance loss of about 0.18 h_v, something like 0.03 h_v will be lost in a stretch 7 to 15 D from the entrance and 0.025 h_v in the portion located between 14 and 22 D downstream. Since the measured loss was about 0.16 h_v, these quantities represent 20 to 15 percent of the measured loss.

There is no object in trying to be exact in these matters. The test on the model branch is believed unsatisfactory; the model "rate of loss" curve for this particular fitting is not known and, as will be indicated, the model and prototype may not perform alike. It is believed, however, that if a value equal to 85 percent of the average friction coefficient determined in the Gibson tests on penstocks N-3, N-5, and A-7 be taken as the straight pipe coefficient, the probable error in the experimental value will be of the order of 4 percent.

While it is believed present knowledge of the rate of dissipation of energy losses does not justify a closer estimate of the friction coefficient from the field data, other factors possibly influencing the measurement should be noted. The measuring section in A-8 was seven diameters closer to the entrance than the others. Plate 10 shows that the effect of this would be to increase the measured loss. This is confirmed, the A-8 measurements showing a 5-percent greater friction coefficient than the balance of the tests. Since, however, due to the size of the unit, less water was being used, resulting in running the test at a lower Reynolds number, this factor may also have influenced the result.

Finally, the friction tests were made with variable $\frac{Q_s}{Q_a}$ ratios, that is, with a varying percentage of the incoming flow at the manifold diverted to the unit under test but with a constant amount passing downstream in the manifold. Plate 8 shows that for values of $\frac{Q_s}{Q_a}$ up to 0.20, there is an extremely rapid decrease in the coefficient of entrance loss with increase in the ratio. If entrance loss is dissipated in the piezometered section, the result will be an apparent decrease in the friction coefficient with increase in Reynolds number.

The mean friction factor given by the measurements on N-3, N-5, and A-7 was $f = 0.0207$. Eighty-five percent of this is 0.0177 which is the factor considered applicable for use in straight 13-foot pipe at Reynolds number $30 - 70 \times 10^6$. The friction coefficients applicable to 30, 25, and 8.5-foot pipe were obtained from this figure on the assumption that the coefficient varies inversely as $D^{1/3}$ this being the relationship of the Manning formula and more recently confirmed by Nikuradse's systematic tests on rough pipe(7) for application to losses where the quadratic law is applicable.

The following table contains in column 2 the values of the straight-pipe friction coefficients used in deriving fitting losses from the 1938 tests. The values as actually measured which include entrance losses are tabulated in column 3.

1938 Friction Coefficients

Pipe diameter 1	Friction factor used 2	Friction factor measured 3	Length of measuring section 4	Ratio column 2 to column 3 5	Notes
8.5'	0.0204	0.0211	14.7	0.97	Upper piez. in region of rising hyd. grade.
13.0'	0.0177	0.0207	7.7	0.85	14-D below entrance.
25.0	0.0142	-	-	-	
30.0	0.0134	0.029	10.5	0.46	About one-half excess loss due to tie rods; balance bends.

The differences between the estimated straight-pipe friction factor and the coefficients measured show how hard it is to find a section of pipe in which there is only pure friction, and the magnitude of errors likely to be introduced if no correction is made for upstream losses.

Concluding the subject of the field determination of friction losses in straight pipe, the value of $f = 0.0177$ found for the 13-foot pipe corresponds to Manning's n of 0.15 to a value ? High of Scobey's $K_s = 0.475$ and to Nihuradse's roughness factor $\frac{K}{R} = 0.00134$. The latter factor can be interpreted as meaning that the bitumastic-covered surface of the 13-foot pipe was as rough hydraulically as though it had been coated with uniform sand grains 0.1 inch in diameter. Those in charge of the tests stated that broom marks and joints were readily visible in the bitumen, a 20-foot straightedge would allow 1/8-inch "feelers" underneath and the rivet heads (really pinheads) of the girth joints 4 inches in diameter projected one-eighth of an inch. The writer anticipated a friction coefficient about one-half of these values and although the experimental results do not appear entirely out of line with the character of the surface just described, it is believed the value of $f = 0.0177$ is, if anything, on the high side. The value is greater than that found in any class of concrete pipe of equal diameter and approximates the value to be expected in a heavy, fully riveted pipe.

It would appear that there is an opportunity for hydraulic laboratories to devise means for pretesting, and specifying the degree of hydraulic roughness to which penstocks must conform to be acceptable under a contract. With one foot of head worth \$110,000, considerable money could be economically spent to reduce friction and even if a reduction were not possible, the advance knowledge of what the friction was to be would lead to material savings in a better economic balance in proportioning the installation.

<u>Entrance and fitting losses.</u> Plate 9 has plotted on it the energy grades as measured both in the model and the prototype. To the plate has been added the corresponding friction slopes. The difference between the energy and friction grades are the energy losses caused by the entrance and fittings as conventionally computed. The most noticeable feature of this plate is that whereas in the model the energy and friction grades are more or less parallel, in the prototype they converge and then diverge at P-A1. This appears to imply that the total fitting loss decreased as the water passed down the penstock, which could only be true of

the nominal losses, that is, those based on a $\frac{Q}{A}$ h_v.

The relationships between the fitting losses of the model and prototype at various points is best shown on plate 11 which plots the total fitting loss at each point as obtained by subtracting the straight-pipe friction from the measured energy grade. At the bottom has been plotted the entrance and fitting losses as assumed in the preliminary design. The sum of the total fitting losses at the end of a section 57 D long containing at least six distinct fittings; a tower entrance, one 90° bend, one 40° bend, four nonoperating branch offtakes, each with two 12-inch tie rods in the waterway, and one 30 by 25 reducer are found to be 1.75, 1.00, and 0.85 h_v for the design, the model, and the prototype, respectively. The great improvement brought about in the estimate of losses by the use of the model is evident. The 15-percent less loss shown by the prototype while thought to be significant, is less than the estimated maximum error possible in the tests. The writer cannot help continuing to marvel at the exactitude of nature's laws by which using experiments with 2 second-feet in a 6-inch pipe, it is possible to tell what will happen with 20,000 second-feet in a pipe 30 feet in diameter. If the losses at intermediate points in the penstock had shown equal correspondence, no further comment on the tests would be needed. That they do not, either reflects on the accuracy of the tests or requires explanation.

It so happens that the total fitting loss measured by the model at the end of the 57 D section equals 1.0 h_v so that the graphs on plate 11, in addition to showing the absolute losses measured in velocity heads, also give the percentages of loss in terms of the total which was measured in the model. Thus 23 D from the intake, where the model shows 70 percent of the loss having taken place, the prototype shows only 35 percent and the design estimate 150 percent. If this actually represents the facts, evidently much improvement in design procedure and model interpretation is necessary to bring the three values together.

The first measurement of the entrance loss in the model was made at a point 2.5 D below the intake, the piezometric grade being measured at the center of the tower with a siphon piezometer. To plate 11 has been transferred the detailed measurements of entrance losses for cone entrance, "A" of plate 10. These losses are plotted in percent of the total loss. It is to be noted that the entrance losses obtained by measurements made at various distances below the entrance up to 5 diameters show from 115 to 58 percent of the final loss. Since the tower entrance undoubtedly acts in a similar manner to the cone, the measurement of entrance loss at 2.5 D below the intake, which in the model gave a value of 0.57 h_v, yields no definite information as to the ultimate loss caused by the entrance, the pressures farther down the penstock being affected by the bend at the base of the tower.

An endeavor was made in the model to obtain more information on the probable ultimate loss caused by the intake by making a separate test in which a straight piece of pipe was substituted for the bend at the base of the tower. Measurements made at 10 D from the entrance showed an increased loss of about two percent. However, since this determination was made with wall piezometers whereas the pressures at $2\frac{1}{2}$ D were determined with the siphon piezometer placed near the center of the flowing stream in which possibly excess pressure existed due to the curvature of the jet, and since the curve of nominal losses tends to be flat between 5 - 10 D, no conclusions can be derived. A guess may be hazarded that the ultimate entrance loss would be of the order of 0.63 h_v, 15 percent greater than the measured.

It is desirable at this point to review the relationship which exists between the energy loss as measured 2.5 D below the entrance and the piezometric drop in water surface from the lake to that within the tower. Referring to plate 9, water enters the tower radially and has to be deflected through 90° to flow down the tower. As a result, water within the tower instead of standing at an elevation equal to the energy head minus the velocity head is raised enough to provide the accelerating or reaction head (reference 6, p. 92) required by the change in direction. Experimentally, the reaction head in the model was determined from the following equation $K_T + K_R = 1 + K_e$ where from the model

$K_T = 0.81$ = lake to tower drop per h_v,

$K_e = 0.57$ = entrance loss per h_v, from which

$K_R = 0.76$ = reaction head per h_v.

This reaction head, 0.76 h_v, checks closely the mean pressure obtained by integrating the pressures exerted by a jet of water striking against a plate, if the integration is only carried across an area equal to that of the orifice, though the entire pressure against the plate is $P = 2.0A\ h_v$. In the tower entrance the difference between these two values, $1.24A\ h_v$ must act as an upward force in the passages of the intake.

Making use of the equation giving the relationship between reaction, energy loss, and piezometric drop coefficients, using the reaction coefficient as determined in the model (0.76) and the piezometric tower drop measured for the prototype, (0.80; 1938 value) the field measurement of the entrance loss at 2.5 D below the intake is found to be 0.56 h_v, a value 2 percent less than that found in the model. The writer could wish that his comparison of data might stop here but additional data from both model and prototype throw doubt on the precision just indicated.

<u>Coefficient of discharge for tower.</u> The most certain indication of the entrance performance of model and prototype is given by the direct measurements of piezometric drops from lake to tower expressed in terms of the discharge equation $Q = C_1 \sqrt{D_1}$ where D_1 measured in feet is the piezometric drop from the lake to the tower water surface and Q is the discharge in cubic feet per second referred to the prototype. The following are the values of C_1 found in different tests:

Values of C_1

Model		Prototype	
Model tests No. 2 used in this report, mean of 5 runs at highest velocity, h_v = 0.014' ±	6,304	1938 interpolated measurements, h_v = 13' ±	6,340
Bul. No. 2, part VI, Boulder Final Reports, p. 99 - Mean of all measurements	6,500	1939 (most accurate of pitot measurements)..	6,640
		1939 current meter measurements in river	6,750

The equation $Q = 6{,}560 \sqrt{D_1} \pm 70$, best represents the weighted results of the field tests applicable for one tower gate open. The equation for two gates open derived from the same coefficient, and making use of the relative model discharges is $Q = 16{,}400 \sqrt{D_2}$. It is believed that high discharges through the Boulder outlet works can most easily be determined by the application of these equations with field measurements of the lake tower drop and with a probable accuracy of about two percent. This is perhaps better than could be done by stream gaging for the same conditions.

Referring again to plate 11 showing the fitting losses, the mechanism of energy dissipation seems to be as follows. At the throat the nominal losses as measured in model and prototype, tend to be the same, the piezometric grade being controlled by the shape of the intake. The slightly lower values of the loss coefficient found in the prototype is probably caused by the smaller internal friction of the prototype as compared to the model and could be estimated if relative friction factors were predetermined. Below the throat the lower straight-pipe friction coefficient of the prototype brings about a slower change in momentum than occurred in the model with the result that the prototype curve of losses is shifted downstream somewhat in proportion to the relative friction coefficients.

Referring to the upper part of plate 10, curves A and C, the first shows the measured loses found for a model with a cone entrance and the second, according to the proposed hypothesis, the losses which might be formed for a prototype, if it had one-half the friction coefficient. It will be seen that dependent on the distance (diameters) below the intake at which losses are measured, the prototype would show greater, the same, or less losses than its model and that it would have to have a greater total length if it were to finally reach the same total loss. It has been suggested that the formula $L = 0.7R^{\frac{1}{4}} D$ expresses the length required to fully develop turbulence in smooth pipes.[10]

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[10] S. Goldstein; Modern Developments in Fluid Dynamics.

A similar formula is required to predict the rate of loss from fittings.

Turning again to plate 11, it is seen that corresponding to this hypothesis, the intermediate losses measured in the prototype are relatively much less than found in the model. At P-A1, the first turbine penstock, the fitting loss was only half that registered in the model and about 60 percent of that at P-A6. On the other hand, these are nominal losses and whether energy is available for use in kinetic form or whether wall pressures are in excess of mean pressures at present is unknown.

Finally, at 57 equivalent diameters from the entrance the prototype showed 85 percent of the model's loss but with the indication from a comparison of the friction and energy grades that fitting losses were still taking place.

It is thought that in a long penstock the loss would equal that shown by the model in accord with the equation for shock loss $\frac{(v_1 - v_2)^2}{2g}$ providing the equation was applied to each filament of water starting at the section at which velocities are a maximum.

The measurements at 57 equivalent diameters from the intake provide the only possibility for approximating the losses due to the various fittings. The table below shows the design, model, and prototype losses segregated in accord with the judgment of the writer. The item of most general interest is perhaps the loss due to the 90° and 40° bend, which regardless of the distribution chosen, would apparently be less than the values ordinarily assumed for such a case.

Estimated Loss in Fittings from Boulder Tests

Item	Design	Model	Prototype
Entrance	1.10 h_v	0.68 h_v	0.63 h_v
Bend, combined 90° + 40°	0.40	0.11	0.08
Tie rods and penstock openings, 8 - 12" tie rods - 20 feet long across 4 openings	0.20	0.17	0.12
Reducer 30 x 25	0.05	0.04	0.02
Total	1.75 h_v	1.00 h_v	0.85 h_v

Branch losses. Plate 9 shows the energy grade for the loss through the 8.5-foot branch connection to V-A1 for 100-percent diversion plotted against equivalent diameters, that is, the last 2.2 D of 8-foot pipe including the reducer has been lengthened to 2.7 D to give equivalent 8.5 D losses. The energy grade was measured at three points, the upper and lower piezometer rings, and the pitot tube. The first two represent nominal energy grades whereas that at the pitot tube is the true grade being the mean of the impact readings. It is thought that the sharp break in the energy loss line in a distance of 2.7 equivalent diameters just in front of the pitot tube may indicate that the energy grades determined by adding mean velocity heads to wall pressures give unsatisfactory measures of energy head in large installations where unequal velocity distributions exist, many diameters below the disturbance. A comparison of the friction and energy grades on the drawing shows that here, as in the main penstock, the loss 2.6 D from the throat is very much less than the total and that the curve of losses is displaced downstream as compared to the model. On the other hand, the sharp break in the energy grade at the valve suggests the possibility that this apparent change in performance from the model, both in this case and in the main penstock, might be caused by wall pressures being higher than the mean across the section. The numerical loss was of the order of 10 feet, too great it is believed, for error to be entirely responsible for the break in grade. It is suggested that if the tests represent facts, then efficiencies determined for low head plants based on wall pressures may require significant corrections.

The branch losses for various ratios of $\frac{Q_s}{Q_a}$, the ratio of flow in the branch to flow in the main penstock, is compared with the model on plate 8, curves 4A and 4. The latter is extrapolated from a single series of tests made at $\frac{Q_s}{Q} = 0.17$ by means of curve 3, an almost identical fitting. The prototype shows about 60 percent of the model losses. Since the loss determination involves a friction factor determined by formula from the tests on the 13-foot penstocks, it may be much in error but again there is agreement with the other tests in showing less losses in the prototype than in the model.

<u>Valve coefficients.</u> The equation for discharge through the valve was determined by the equation $Q = CA\sqrt{2gH}$ for which;

C = coefficient of discharge $= \frac{1}{(K_1 + K_2)^{\frac{1}{2}}}$

A = gross area of outlet end of valve

H = mean energy head at inlet

K_1 = coefficient of total energy loss within valve per mean velocity head at outlet

K_2 = energy correction factor per mean velocity head at outlet

$$= \frac{\int v^3\, da}{\bar{v}^3 A}$$

Experimentally the coefficient "C" was determined for the model by measuring the discharge and the pressure in front of the valve and adding to the latter the mean velocity head to obtain "H." In the prototype "H" was determined directly from the mean of the impact readings. Normally, since H is large and enters the equation as the square root, errors from its determination are small. But it should be noted that for models with low heads and small pipe, neglect of the energy correction factor at the inlet may make the coefficient too high by an amount of the order of one percent.

Referring to the equation defining the coefficient in terms of fundamentals, $C = \frac{1}{(K_1 + K_2)^{\frac{1}{2}}}$ it can be seen that it depends upon the internal energy losses within the valve caused by friction and the stay vanes and upon the velocity distribution at the outlet whose value is practically a constant, depending upon the shaping of the valve body and needle. Since K_1 is small with respect to K_2, the equation can be put in the form of

$$C = \frac{K_2^{-\frac{1}{2}}}{1 + \frac{1}{2}\left(\frac{K_1}{K_2}\right)}$$

Taking a mean value from the model tests of $C = 0.77$ and estimating the friction loss through the valve in terms of the Q/A mean velocity at the outlet as equal to that occurring in two diameters of smooth pipe at a Reynolds number of 300,000 ($0.035\ h_v$) and assuming the loss caused by eight stay vanes as $0.045\ h_v$, the equation becomes $C = \frac{1.62^{-\frac{1}{2}}}{1.025}$.

The numerator of this equation corresponds to the energy correction factor for the outlet. It roughly checks that given by Williams (7) for a bend with $\frac{R}{D} = 0.80$, a ratio approximating that used in the design of the valve. The fractional part of the denominator, $2\frac{1}{2}$ percent, is a variable and depends on the friction losses within the valve. It can be seen that ~~for a decrease or increase of these by 100 percent~~ halving or doubling this factor, the valve coefficient will in turn increase or decrease $1\frac{1}{4}$ and $2\frac{1}{2}$ percent. These are the order of variations which were found in the model and prototype and which can be ascribed to changes in "f" with varying Reynolds number.

The highly nonuniform velocity distribution at the outlet indicated by the value of K_2, confirmed incidentally in the careful exploration of the pressures and velocities in a Pelton nozzle by Tenot[11] which all interested in valves should investigate. It

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[11] Tenot; Etude Experimentale d'un jet de Turbine Pelton etc.; Revue generale de l'Hydraulique, No. 27, May-June 1939, p. 119.

implies also a large reduction in pressure on the inside of the bend which Williams gives as 6 h_v for $\frac{R}{D}$ = 0.8. This theoretical value is, of course, greatly reduced by friction and can only fully develop in a bend of large central angle. However, at the outlet of the Boulder valves, a negative pressure of 0.232 h_v was measured, the zone of negative pressure extending as drawn on plate 1 in the region of the filleted outlet. Computation showed that this value will approach a perfect vacuum with a head on the prototype of about 200 feet.

It has been stated that the energy correction factor for any given shaped valve tends to remain constant. This only applies as long as no vacuum exists. Where it does, the effective waterway decreases and the energy correction factor increases, resulting in a decrease in the discharge coefficient. On plate 7 have been plotted the valve discharge coefficients, C_v. For the model, it is computed that full vacuum is reached at R = 4 x 10^6, and at a value of 6 x 10^7 for the prototype. (Reynolds numbers are based on the outlet diameter.) From plate 7 it is seen that the curves showing variation in the valve coefficients begin a trend downwards, somewhat before reaching these values, and that they are in good agreement with one another.

Plate 6 shows the valve coefficients for partial gate opening. Neither the model nor the field work was accurate enough to evaluate the causesof the differences as shown. It is believed the relative coefficient s found in the 1937 tests have the greater accuracy.

CONCLUSIONS

The following conclusions, subject to the limitations of tests, can be tentatively drawn:

(1) Pressure differences measured close to or within a fitting such as intake drops, valve coefficients, bends, etc., reproduce accurately from model to prototype providing small corrections for internal friction are made and providing vacuum does not exist in either system.

(2) Model tests provide the best means of estimating fitting losses in installations such as power plant penstocks where the influence of one fitting on another makes results derived from tests on single fittings inapplicable.

(3) Model tests showing rate of fitting loss at varying hydraulic radii below a fitting will not reproduce in the prototype unless the friction slope is the same. Though ultimately the total loss may be equal, intermediate losses will generally be less than found in the model tests, the percentage reduction varying with the point of measurement and the friction coefficient. The maximum difference was about 50 percent in the Boulder tests.

(4) Qualitative results in the prototype may differ from those in the model for streamline fittings and shapes and for cases in which velocity distribution upstream affects the fitting loss.[12]

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[12] Yarnell, David L.; Flow of water through 6-inch pipe bends; Tech. Bul. No. 577, United States Department of Agriculture.

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(5) Reliable comparisons between model and prototype cannot be made without additional knowledge as to the relationship between integrated velocities and pressures and measured mean velocities and wall pressures in regions below fittings where disturbed velocity distributions exist. Nor can fitting losses be accurately obtained from the prototype until a means is devised for obtaining the friction coefficient for straight pipe free from errors introduced by fitting losses.

(6) Reducers of 0.9 area reduction ratio were not found sufficient to produce a uniform velocity 23 D below a branch in an 8-foot 6-inch pipe. The mean velocity was four percent greater in one-half the pipe than in the other.

TABLES

TABLE 1

Pitot Coefficients C_p

See Plate No. 2

Distance from wall		Rating channel calibration	Corrections			
Inches	Radii		Velocity gradient	Turbulance and curved streamlines	Obstruction in channel	Total
2-7/16	0.051	0.878	0.015	0.010	0.008	0.911
7-13/16	0.163	0.922	0.003	0.010	0.009	0.944
14-1/16	0.293	0.933		0.010	0.014	0.957
21-11/16	0.452	0.933		0.010	0.019	0.962
32-13/16	0.684	0.933		0.010	0.031	0.974
Mean		0.920			Mean	0.950

TABLE 2

Effect of Pitch and Yaw on Coefficient C_p

See Plate No. 2

Angle in degrees	Four holes per ring			Two holes per ring
	Pitch		Yaw with strut	Pitch
	Without strut	With strut		
0	0.994	0.929	0.948	0.918
3			0.937	
5	0.998			
6			0.943	
7			0.954	
9			0.937	
10	1.022	0.937		
15			0.986	
20			1.041	
21	1.080			

TABLE 3

1937 Measurements at Piezometer Stations

Relation of average of recorded readings at ends of 125-foot section in V-A1 and V-A6 to number of valves open.

Number of valves open 100%	V-A1			V-A6		
	Upstream ft.	Downstream ft.	Diff. ft.	Upstream ft.	Downstream ft.	Diff. ft.
1	2	3	4	5	6	7
1	974.9	957.6	17.3	969.1	953.0	16.1
2	973.5	956.4	17.1	967.3	951.2	16.1
3	970.6	953.8	16.8	964.6	948.9	15.7
4	969.1	953.1	16.0	962.7	947.2	15.5
5	965.8	950.1	15.7	958.8	943.7	15.1
6	963.6	948.3	15.3	956.4	941.4	15.0

TABLE 4

1937 Pitot Traverse in V-A1 Valve 95-Percent Open

Traverse is taken looking downstream and reading from left to right. Lines A and B are on diameters at right angles to each other and at 45 degrees with the horizontal diameter. Line B is above line A on left side. No corrections have been made. Reservoir elev. = 1033.0.

Points on line A	Points on line B	Dist. from wall inches	Impact Line A ft.	Impact Line B ft.	Impact Mean ft.	Pressure head Line A ft.	Pressure head Line B ft.	Pressure head Mean ft.	Impact minus press. ft.	C_p from curve plate 2	Vel. ft. per sec.	h_v ft.
1	2	3	4	5	6	7	8	9	10	11	12	13
Wall		0										
1	10	2-7/16	1002.5	1002.5	1002.5	959.9	962.2	961.0	41.5	0.911	56.7	50.0
2	9	7-13/16	1000.2	1001.4	1000.8	947.2	949.5	948.4	52.4	0.944	61.5	58.8
3	8	14-1/16	997.9	997.9	997.9	931.0	932.2	931.6	66.3	0.957	68.2	72.4
4	7	21-11/16	997.9	997.9	997.9	932.2	932.2	932.2	65.7	0.962	67.6	71.1
5	6	32-13/16	999.1	1000.2	999.7	939.1	942.5	940.8	58.9	0.974	63.2	62.1
Center line		48										
6	5	63-3/16	999.1	1000.2	999.6	939.1	942.5	940.8	58.8	0.974	63.2	62.0
7	4	74-5/16	999.1	997.9	998.5	934.5	932.2	933.4	65.1	0.962	67.3	70.4
8	3	81-15/16	999.1	997.9	998.5	934.5	931.0	932.8	65.7	0.957	67.9	71.7
9	2	88-3/16	1000.2	1000.2	1000.2	944.8	946.0	945.4	54.8	0.944	62.9	61.5
10	1	93-9/16	1002.5	1002.5	1002.5	959.9	959.9	959.9	42.6	0.911	57.4	51.3
Wall		96										
Mean			999.76	999.86	999.81	942.22	943.02	942.63	57.18	0.954	63.59	63.13
										(1) 0.95	63.9	63.5

Area = 49.97 sq. ft.

Avg. velocity = 63.5 ft. per sec.

Q = 3,195 c.f.s.

Valve coef. disch. C_v = 77.0%

(1) Average from table 1

TABLE 5

1937 Data for Various Openings of V-A1 Valve

Opening of V-A1	Discharge (7)	Velocity head (4) h_v 8-1/2-foot pipe	Estimated energy grade at junction of V-A1 with main conduit (5)	Recorded piezometric elevation (1) at upper piezometer ring V-A1	Recorded piezometric elevation (1) at lower piezometer ring V-A1	Recorded mean elevation at pressure leg pitot tube (2) 8-foot pipe	Recorded mean elevation at impact leg pitot tube (2) 8-foot pipe	Recorded impact minus pressure	Energy grade at upper piezometer ring in V-A1	Energy grade at lower piezometer ring in V-A1	Energy loss junction to upper piezometer ring	Energy loss between piezometer rings	Energy loss lower piezometer ring to impact	Total energy loss in V-A1 from junction to impact leg	Energy loss coef. from junc. to upper piez. ring	Energy loss coef. from upper to lower piez. ring	Energy loss coef. from lower piez. ring to impact leg pitot tube	Energy loss coef. from junc. to impact leg	Valve coefficient C_v (6)	Reynolds number temp. 55°F. (x 10^7)
%	c.f.s.	ft.	ft.	ft.	ft.	ft.	ft.	ft.	ft.	ft.	ft.	ft.	ft.	ft.						
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
				add 1.0'	add 1.0'	add 0.5'	add 0.5'													
0		0.0	1033.0	1031.4	1031.4	1032.5	1032.5	0.0	1032.4	1032.4	0.6	0	-0.6	0	0		0			
5	250	0.3	1033.0	1031.4	1031.4	1032.1	1032.5	0.4	1032.7	1032.7	0.3	0	-0.3	0	1.00		-1.00	0	5.5	
10	475	1.1	1033.0	1030.3	1030.3	1030.1	1031.4	1.3	1032.4	1032.4	0.6	0	0.5	1.1	0.55	0	0.45	1.00	10.5	
15	685	2.3	1033.0	1029.1	1028.0	1027.5	1030.2	2.7	1032.4	1031.3	0.6	1.1	0.6	2.3	0.26	0.48	0.25	1.00	15.3	
20	905	4.0	1033.0	1026.8	1025.7	1024.0	1028.6	4.6	1031.8	1030.7	1.2	1.1	1.6	3.9	0.30	0.28	0.40	0.98	20.1	
25	1110	6.0	1032.9	1023.3	1021.0	1020.1	1027.0	6.9	1030.3	1028.0	2.6	2.3	0.5	5.4	0.43	0.38	0.08	0.90	24.8	1.28
30	1350	8.9	1032.9	1021.0	1017.6	1014.5	1024.8	10.3	1030.9	1027.5	2.0	3.4	2.2	7.6	0.22	0.38	0.25	0.85	30.4	1.56
35	1565	12.0	1032.9	1017.6	1014.1	1009.5	1023.3	13.8	1030.6	1027.1	2.3	3.5	3.3	9.1	0.19	0.25	0.27	0.76	35.5	1.79
40	1785	15.6	1032.8	1014.1	1009.5	1003.6	1021.5	17.9	1030.7	1026.1	2.1	4.6	4.1	10.8	0.13	0.29	0.26	0.69	40.7	2.06
45	1975	19.1	1032.8	1009.5	1003.7	997.2	1019.3	22.1	1029.6	1023.8	3.2	5.8	4.0	13.0	0.17	0.30	0.21	0.68	45.3	2.28
50	2160	22.8	1032.7	1004.9	998.0	991.1	1017.3	26.2	1028.7	1021.8	4.0	6.9	4.0	14.9	0.18	0.30	0.18	0.65	49.8	2.49
55	2330	26.4	1032.7	1000.3	992.2	985.0	1015.4	30.4	1027.7	1019.6	5.0	8.1	3.7	16.8	0.19	0.31	0.14	0.64	53.8	2.68
60	2490	30.2	1032.6	995.7	986.4	977.3	1012.1	34.8	1026.9	1017.6	5.7	9.3	5.0	20.0	0.19	0.31	0.17	0.66	58.1	2.87
65	2620	33.6	1032.6	991.0	980.7	971.6	1010.3	38.7	1025.6	1015.3	7.0	10.3	4.5	21.8	0.21	0.31	0.15	0.65	61.5	3.02
70	2765	37.4	1032.6	987.6	976.0	965.4	1008.4	43.0	1026.0	1014.4	6.6	11.6	5.5	23.7	0.18	0.31	0.15	0.63	65.3	3.19
75	2890	40.8	1032.5	984.1	971.4	959.3	1006.3	47.0	1025.9	1013.2	6.6	12.7	6.4	25.7	0.16	0.31	0.16	0.63	68.5	3.33
80	3010	44.2	1032.5	981.8	968.0	953.8	1004.7	50.9	1027.0	1013.2	5.5	13.8	8.0	27.3	0.12	0.31	0.18	0.62	71.6	3.47
85	3085	46.5	1032.5	978.3	963.3	949.3	1002.8	53.5	1025.8	1010.8	6.7	15.0	7.5	29.2	0.14	0.32	0.16	0.63	73.8	3.55
90	3165	48.7	1032.4	976.0	961.0	946.4	1001.4	56.0	1025.7	1010.7	6.7	15.0	8.8	30.5	0.14	0.31	0.18	0.63	76.0	3.65
95	3195	49.7	1032.4	974.9	(3)959.9	942.6	999.8	57.2	1025.6	1010.6	6.8	15.0	10.3	32.1	0.14	0.30	0.21	0.65	77.0	3.70
100	3205	50.1	1032.4	974.9	957.6	940.8	998.5	57.7	1026.0	1008.7	6.4	17.3	9.7	33.4	0.13	0.35	0.19	0.67	77.6	3.70

NOTES (Table 5)

(1) Add one foot to all readings at both ring piezometers in V-A1 to correct for gage zero setting as determined graphically.

(2) Add 0.5 foot to both impact and pressure-head pitot readings to correct for gage zero setting as determined graphically.

(3) Graph indicates this reading to be one foot too high.

(4) h_v for 8-1/2-foot pipe = (col. 9) x $\frac{1}{C_p^2}$ x $\left(\frac{8.0}{8.5}\right)^4$ = (col. 9) x $\frac{0.784}{(0.95)^2}$ = (col. 9) x 0.869.

(5) Estimated loss from lake to junction of V-A1 with main conduit =

$1.8\ h_v$ x $\left(\frac{8.5}{30.0}\right)^4$ = (col. 3) x 0.0116.

(6) Valve coefficient (C_v) is ratio of discharge through 8.5-foot pipe (A_1) to discharge through outlet end of the valve where the area is 38.5 sq. ft. (A_2) or

$$C_v = \frac{A_1}{A_2}\left(\frac{h_v}{H}\right)^{1/2} = 1.467 \times \left[\frac{\text{Col. 3}}{(\text{Col. 8} + 0.5 - 820)}\right]^{1/2}$$

(7) Net diameter for 8.5-foot pipe assumed to be 8.48 foot because of bitumastic coating. Area = 56.5 square feet.

TABLE 6

1938 Piezometric Drops from Lake to Mean of P-A1 and P-A6

Valves open	Gage reading		Manometric		Mean gage read-ing P-A1 and P-A6	Piez. elev. from gages col.6 plus 633.3	Lake elev.	Drop from lake to mean piez. elev.	$0.04h_V$	Corrected mean drop 2.4ft. plus cols. 9 plus 10	Drop from lake		
	P-A1 ft.	P-A6 ft.	Diff. P-A1-P-A6 ft.	Mean P-A1-P-A6 ft.	ft.	ft.	ft.	ft.	ft.	ft.	to P-A2 Hg.mano. ft.	to P-A1 col.11-$\frac{1}{2}$ col.5 ft.	to P-A6 col.11+$\frac{1}{2}$ col.5 ft.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	466 467	471.0 474.0	0.27 0.19	0.22	469.5	1102.8	1102.3	-0.5	0	1.9	4.4	1.8	2.0
2	465 466 462	470.1 472.1 472.1	0.82 0.63 0.52	0.66	467.9	1101.2	1102.4	+1.2	0.1	3.7	4.8	3.4	4.1
3	462	467.2	1.05 1.28 1.13	1.12	464.6	1097.9	1102.6	4.7	0.1	7.2		6.6	7.8
4	459	463.5	2.09 1.89	1.98	461.2	1094.5	1103.4	8.9	0.2	11.5	6.5	10.5	12.5
5	453	456.8	2.93	2.93	454.9	1088.2	1103.3	15.1	0.4	17.9	13.5	16.4	19.4
6	448	*450.0 447.0	3.98	3.98	*449.0 447.5	1082.3	1103.5	21.2	0.5	24.1	22.1	22.1	26.1

* Assumed values
recorded value
below seems in error.

TABLE 7

1938 Piezometric Drops from P-A6 to V-A1 Junction

1	2	3	4	5	6	7	8	9	10	11
Valves open	Tower Q c.f.s.	Diverted Q c.f.s.	Valve manifold Q c.f.s.	Percent diverted	$\left(\frac{\text{Col. 4}}{\text{Col. 2}}\right)^2$	Meas. P-A1-P-A6 drop ft.	Equiv. drop col. 3 in P-A1-6 for manifold flow ft.	Meas. P-A6-V-A1 drop ft.	$0.04\ (h_{v25} - h_{v30})$ ft.	Corrected P-A6 - V-A1 drop ft.
1	4,350	810	3,540	18.6	0.66	0.22	0.15	0.63	0.01	0.64
2	7,780	780	7,000	10.0	0.81	0.66	0.53	2.75*	0.05	2.8*
3	11,100	675	10,425	6.1	0.88	1.12	0.99	5.71	0.13	5.84
4	14,220	420	13,800	2.9	0.94	1.98	1.86	9.21	0.23	9.44
5	17,490	340	17,150	1.9	0.96	2.93	2.81	14.36	0.38	14.74
6	20,670	300	20,370	1.4	0.97	3.98	3.86	20.16*	0.54	20.7*

* Values interpolated. Columns 8 and 11 plotted on plate 3.

TABLE 8

1938 Drop between Piezometer Stations

Drop between piezometer rings at ends of 125-foot section in V-A1 and V-A6 by mercury manometer.

Valves open		Piezometric drop ft.
Name	Percent	
V-A1	94	19.47
V-A6	100	20.94

TABLE 9

Summary of 1938 Section and Total Drops from Lake to V-Al Junction

No valves open	Lake elev. ft.	Lake-tower drop ft.	Lake-P-Al drop ft. (corrected)	P-Al - P-A6 ft. (hg. man.)	P-A6 - V-Al junction ft. Hg. man. corrected	Lake - V-Al junction ft. (summation section drops)	Lake - V-Al junction ft. Bourdon gage + 0.04 hv	Lake - V-Al junction ft. Bourdon gage corrected + 2 ft.	Difference ft. columns 7 - 9
1	1102.3	0.47	1.8	0.22	0.64	2.7	-4.0	-2.0	4.7
2	1102.4	1.55	3.4	0.66	(2.3)	(6.9)	-		
3	1102.6	3.05	6.6	1.11	5.84	13.6	11.3	13.3	0.3
4	1103.4	5.12	10.5	1.98	9.44	21.9	21.5	23.5	-1.6
5	1103.3	7.59	16.4	2.93	14.74	34.1	32.5	34.5	-0.4
6	1103.5	10.40	22.1	3.98	(20.7)	(46.8)	-		

( ) Partly from interpolation.

TABLE 10

1938 Pitot Traverse in V-A1 Valve 94-Percent Open

Traverse is here shown looking downstream and reading from left to right about a horizontal axis. Lines A and B are on diameters at right angles to each other and at 45 degrees with the horizontal diameter. Line A is above line B on the left side.

Points on line A	Points on line B	Dist. from wall inches	Impact Line A ft.	Impact Line B ft.	Impact Mean ft.	Pressure head Line A ft.	Pressure head Line B ft.	Pressure head Mean ft.	Impact minus press. ft.	C_p from curve plate 2	Vel. ft. per sec.	h_v ft.
1	2	3	4	5	6	7	8	9	10	11	12	13
Wall		0										
10	1	2-7/16	1059.5	1062.5	1061.0	996.8	996.8	996.8	64.2	0.911	70.6	77.4
9	2	7-13/16	1063.5	1061.5	1062.5	995.8	996.8	996.3	66.2	0.944	69.1	74.3
8	3	14-1/16	1064.5	1063.5	1064.0	997.8	997.8	997.8	66.2	0.957	68.2	72.3
7	4	21-11/16	1064.5	1065.5	1065.0	998.8	999.8	999.3	65.7	0.964	67.4	70.7
6	5	32-13/16	1064.5	1063.5	1064.0	1002.8	1002.8	1002.8	61.2	0.972	64.6	64.8
Center line		48	1064.5	1064.5	1064.5	1003.8	1003.8	1003.8	60.7	0.978	63.9	63.4
5	6	63-3/16	1063.5	1064.5	1064.0	1001.8	1001.8	1001.8	62.2	0.972	65.1	65.8
4	7	74-5/16	1064.5	1063.5	1064.0	998.8	998.8	998.8	65.2	0.964	67.2	70.2
3	8	81-15/16	1064.5	1064.5	1064.5	995.8	996.8	996.3	68.2	0.957	69.2	74.5
2	9	88-3/16	1062.5	1063.5	1063.0	998.8	996.8	997.8	65.2	0.944	68.6	73.2
1	10	93-9/16	1061.5	1060.5	1061.0	999.8	997.8	993.8	62.2	0.911	69.4	75.0
Wall		96										
Means omitting center line			1063.3	1063.3	1063.3	998.7	998.6	998.65	64.65	0.951 (1)	67.94	71.82

(1) From 64.65

Area = 49.97 sq. ft.
Average velocity = 67.94 ft. per sec.
Q = 3,395 c.f.s.
Valve coef. of discharge = 70.6%

N.B. Q = 3,540 c.f.s. from interpolated values table 20
C_v = 73.5%

TABLE 11

1938 Piezometric Rise in Valve Manifold
Across Discharging Valve Branches
(Dead water in manifold downstream from branch)

Valves open		Mean discharge	Corrected impact pressure center line V-A1	Pressure rise			Diameters from
Number	Name	c.f.s.	ft. H_2O	From	To	Amount ft. H_2O	cols. 5 to 2
1	2	3	4	5	6	7	8
1	A-2		244.5	A-1	A-6	2.20	1.2
1	A-3	3,540	244.5	A-2	A-6	1.57	1.2
1	A-4		244.5	A-3	A-6	0.10	1.3
1	A-5		244.5	A-4	A-6	0.63	1.4
						(Mean 1.12)	(1.3)
2	A-1,2 1,4 1,5 1,3 1,6	7,000	243.5				
3	A-1,2,3		237.5				
3	A-2,3,4	10,425		A-1	A-5	5.66	
3	A-3,4,5	10,425		A-2	A-6	4.40	
						(Mean 5.05)	
4	A-2,3,4,5	13,800		A-1	A-6	7.75	
4	A-1,2,4,5	13,800	233.5				
5	A-1,2,3,4,5	17,150	233.5				
6	A-1,2,3,4,5,6	20,370	225.5				

TABLE 12

1938 Drop in Energy Grade in Dead End of Manifold after 100-Percent Diversion

Number valves open	Mean discharge c.f.s.	$0.04h_v$ ft.	Mean rise ft.	Diameters from upstream piez.	h_v ft.	Est. friction loss at 0.025 ft.	K loss = cols. $\frac{6-4-3-7}{6}$
1	2	3	4	5	6	7	8
1	3,500	0.03	1.12	1.3	0.81	0.03	$-0.46\ h_v$
3	10,425	0.13	5.05	-	7.01	0.32	$0.21\ h_v$
4	13,800	0.49	7.75	-	12.27	0.70	$0.27\ h_v$

TABLE 13

1939 Pitot Traverse Data in V-A4 Valve 95-Percent Open

Traverse is taken looking downstream and reading from left to right. Lines A and B are on diameters at right angles to each other and at 45 degrees with the horizontal diameter. Line B is above line A on left side.

Line A

Traverse		Impact	Pressure	Impact minus press. ft.	C_p from curve plate 2	Vel. ft. per sec.	h_v ft.	True pressure head ft.
Points from left to right	Distance from wall inches	Average elevation from gages 4 and 5 ft.	ft.					
Wall	0							
13	5-1/8	1104.5	1028.9	75.4	0.934	74.6	86.5	1017.8
12	13	1121.4	1025.4	96.0	0.955	82.3	105.3	1016.1
11	20	1129.3	1029.4	99.9	0.963	83.2	107.7	1021.6
10	27	1120.2	1032.3	87.9	0.968	77.7	94.0	1026.2
9	34	1128.2	1032.9	95.3	0.972	80.5	100.9	1027.3
8	41	1124.7	1041.4	83.3	0.975	75.0	87.5	1037.2
7 cen. line	48	1122.5	1051.6	70.9	0.978	69.0	74.0	1048.5
6	55	1121.4	1041.5	79.9	0.975	73.5	84.0	1037.4
5	62	1109.8	1032.3	77.5	0.972	72.6	82.0	1027.8
4	69	1109.8	1030.0	79.8	0.968	74.0	85.2	1024.6
3	76	1106.4	1024.2	82.2	0.963	75.5	88.7	1017.7
2	83	1103.2	1025.4	77.8	0.955	74.0	85.1	1018.1
1	90	1102.6	1022.5	80.1	0.938	76.5	91.0	1011.6
Wall	96							

Line B

Wall	0							
1	7	1105.5	1026.6	78.9	0.942	75.6	88.9	1016.6
2	13	1119.1	1026.6	92.5	0.955	80.7	101.2	1017.9
3	20	1127.0	1026.5	100.5	0.963	85.5	113.8	1013.2
4	27	1130.5	1031.7	98.8	0.968	82.3	105.3	1025.2
5	34	1125.9	1032.9	93.0	0.972	79.6	98.5	1027.4
6	41	1123.7	1040.7	83.0	0.975	74.9	87.3	1036.4
7 cen. line	48	1108.7	1053.8	54.9	0.978	60.7	57.3	1051.4
8	55	1114.5	1042.5	72.0	0.975	69.8	75.8	1038.7
9	62	1127.0	1034.7	92.3	0.972	79.2	97.5	1029.5
10	69	1108.7	1030.6	78.1	0.968	73.2	83.3	1025.4
11	76	1112.2	1025.4	86.8	0.963	77.6	93.7	1018.5
12	83	1109.8	1024.8	85.0	0.955	77.4	93.2	1016.6
13	89	1106.4	1023.8	82.6	0.942	77.4	93.2	1013.2
Wall	96							

N.B. velocities from this table are plotted on plate 5.

TABLE 14

1939 Pitot Traverse Computation for V-A4 Valve 95-Percent Open

From graphic determination of values at centroid radii of 1/5 areas.

Distance from wall inches	Impact ft.	Velocity ft. per second	h_v ft.	True pressure head ft.
2-7/16	1095.4	70.8	78.0	1017.4
7-13/16	1109.3	77.0	92.2	1017.1
14-1/16	1122.6	82.4	105.6	1017.0
21-11/16	1126.9	83.8	109.1	(1) 1018.0
32-13/16	1125.4	78.6	96.1	1029.3
48 C.L.				
63-3/16	1115.0	74.5	86.4	(1) 1028.8
74-5/16	1110.0	76.1	90.0	1020.0
81-15/16	1106.8	76.4	90.8	(1) 1016.2
88-3/16	1104.2	76.3	90.6	1013.6
93-9/16	1101.4	75.3	88.2	1013.2
Means	1111.7	77.12	92.70	1019.1

Area = 49.97 sq. ft.
Mean vel. = 77.12 ft. per sec.
Q = 3,855 c.f.s.

(1) Does not agree exactly with impact minus h_v

Valve coefficient of discharge C_v = 73.2%

TABLE 15

1939 Data for Partial Openings of V-A4 Valve (5)

Valve opening %	Drop lake to tower inches	Upstream wall pressure (1)			True mean pitot press. (4) elev. ft.	Pitot impact (1)				Pitot pressure head (1)			
		Valve bonnet	Piez. manifold			Point A-1		Point B-7		Point A-1		Point B-7	
		Gage No. 10 (2) ft.	Gage No. 4 (3) ft.	Gage No. 5 (3) ft.		Gage No. 4 ft.	Gage No. 5 ft.	Gage No. 4 ft.	Gage No. 5 ft.	Gage No. 4 ft.	Gage No. 5 ft.	Gage No. 4 ft.	Gage No. 5 ft.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
0	35-1/4	1156.9	1156.7	1157.2	1157.1	1156.9	1156.7	1156.9	1156.7	1156.9	1156.7	1156.9	1156.7
10	35-1/4	1152.5	1152.1	1152.5	1154.9	1154.6	1156.7	1152.5	1156.7	1152.5	1156.7	1152.5	1156.7
20	35-3/4	1142.4	1142.6	1143.1	1143.0	1150.2	1154.4	1151.4	1154.4	1143.5	1144.9	1144.5	1147.2
30	36-1/4	1126.7	1128.5	1125.8	1126.5	1143.5	1144.9	1146.8	1147.2	1126.7	1128.5	1133.4	1133.4
40	37-1/8	1105.5	1103.2	1107.1	1105.0	1139.2	1135.7	1143.5	1144.9	1107.8	1105.5	1120.0	1119.3
50	38-3/8	1082.8	1084.5	1083.8	1084.4	1125.5	1123.9	1134.5	1138.0	1087.5	1084.5	1105.5	1100.8
60	38-7/8	1063.7	1063.7	1065.1	1061.3	1123.2	1114.7	1139.2	1138.0	1067.2	1063.7	1090.7	1086.8
70	39-3/8	1045.7	1045.2	1047.8	1045.9	1105.5	1103.2	1132.2	1133.2	1050.3	1047.5	1076.2	1075.2
80	40	1031.2	1031.2	1036.0	1029.6	1109.8	1105.5	1125.5	1114.7	1036.7	1035.8	1063.7	1059.1
90	40-3/8	1019.8	1019.6	1024.2	1023.7	1103.2	1093.9	1121.2	1105.5	1026.5	1028.8	1057.0	1054.5
95	40-9/16	1016.4	1017.3	1022.8	1019.1	1104.3	1100.8	1114.2	1103.2	1023.1	1021.9	1055.8	1052.2

(1) All gage readings at 0-percent opening should record 1157.1. Any difference is considered a constant correction for that gage.

(2) Gage No. 10 is located 0.2 diameters upstream from gages 4 and 5.

(3) Gages 4 and 5 are located 0.5 diameters upstream from the tip of the pitot tube.

(4) Copied from column 11, table 16, for comparison with columns 4 and 5.

(5) Six hundred and seventeen c.f.s. diverted upstream in P-A8.

TABLE 16

1939 True Mean Pitot Pressure-Head Elevation for Partial Openings of V-A4 Valve

Valve opening % 1	Lost head to manifold (estimated)(1) ft. 2	Energy elev. at manifold(2) ft. 3	Energy elev. at impact log(3) ft. 4	Piez. elev. at press. log(4) ft. 5	Nominal energy loss(5) ft. 6	Nominal velocity head(6) ft. 7	True energy loss to impact(7) ft. 8	True velocity head(8) ft. 9	True energy elev. at impact(9) ft. 10	True mean pitot press.(10) elev. ft. 11
0	0	1157.1	1157.1	1157.1	0	0	0	0	1157.1	1157.1
10	0	1157.1	1155.4	1154.9	1.7	0.5	1.5	0.7	1155.6	1154.9
20	0.1	1157.0	1152.9	1145.3	4.1	7.6	3.6	10.4	1153.4	1143.0
30	0.3	1156.8	1145.9	1130.8	10.9	15.1	9.6	20.7	1147.2	1126.5
40	0.4	1156.7	1141.1	1113.5	15.6	27.6	13.8	37.9	1142.9	1105.0
50	0.6	1156.5	1130.8	1094.9	25.7	35.9	22.7	49.4	1133.8	1084.4
60	0.7	1156.4	1129.1	1077.4	27.3	51.7	24.1	71.0	1132.3	1061.3
70	0.9	1156.2	1118.8	1062.6	37.4	56.2	33.1	77.2	1123.1	1045.9
80	1.0	1156.1	1114.2	1049.1	41.9	65.1	37.1	89.4	1119.0	1029.6
90	1.0	1156.1	1106.3	1042.0	49.8	64.3	44.0	88.4	1112.1	1023.7
95	1.1	1156.0	1105.9	1038.5	50.1	67.4	44.3	92.6	1111.7	1019.1

(1) Manifold is junction of V-A4 with main conduit loss from lake taken as 1.8 h_v.
(2) Lake water surface 1157.1 minus column 2.
(3) Mean of impact elevation of A-1 and B-7, columns 7 to 10, table 15, with 0.3 foot added for gage zero error.
(4) Mean of pressure elevation at A-1 and B-7, columns 10 to 13, table 8, corrected as in (3) above.
(5) Loss from manifold to impact columns 3 minus 4.
(6) Impact minus pressure head columns 4 minus 5.
(7) Column 6 x 0.884 which is ratio [drop from manifold (1156.0) to mean elevation of impact across pipe at 95-percent opening (1111.7) or 44.3 feet] to [drop to corrected mean elevation at A-1 and B-7 (1105.9) or 50.1]; $\frac{44.3}{50.1} = 0.884$.
(8) Column 7 x 1.37 which is ratio of mean h_v across pipe at 95-percent opening (92.6 feet) to impact minus pressure for mean of A-1 and B-7 at 95 percent (67.7); $\frac{92.6}{67.4} = 1.374$.
(9) Col. 3 minus 8.
(10) Col. 10 minus 9.

TABLE 17

1939 Tower Discharge
for Partial Openings in V-A4 Valve

Valve V-A4		Discharge			Tower functions			
opening %	h_v ft.	V-A4 c.f.s.	P-A8 c.f.s.	tower c.f.s.	drop ft.	h_v ft.	K	R_n T = 16°C.
0	0	0	617	617	0.008*	0.01		
10	7.0	1,060	617	1,677	0.008	0.88		
20	10.4	1,290	617	1,907	0.050	11.32	0.44	6.8×10^6
30	20.7	1,825	617	2,442	0.091	18.60	0.49	8.7×10^6
40	37.9	2,465	617	3,082	0.163	28.60	0.57	1.09×10^7
50	49.4	2,815	617	3,432	0.268	36.70	0.73	1.22×10^7
60	71.0	3,375	617	3,992	0.310	49.60	0.625	1.42×10^7
70	77.2	3,520	617	4,137	0.350	53.20	0.66	1.47×10^7
80	89.4	3,790	617	4,407	0.403	60.40	0.67	1.57×10^7
90	88.4	3,765	617	4,382	0.435	59.70	0.73	1.56×10^7
95	92.6	3,855	617	4,472	0.453	62.20	0.73	1.59×10^7

* This amount estimated and added to readings to allow for drop due to exciter unit, P-A8.

TABLE 18

1939 Valve Coefficient of Discharge, C_v,
for Partial Openings in V-A4 Valve

Valve opening %	h_v $8\frac{1}{2}$" pipe ft.	Total head(1) ft.	C_v (Note 2)	R_n T = 61° F.
0		337.1		
10	5.5	335.6	0.186	1.34×10^7
20	8.1	333.4	0.229	1.63×10^7
30	16.2	327.2	0.327	2.30×10^7
40	29.7	322.9	0.444	3.11×10^7
50	38.7	313.8	0.516	3.55×10^7
60	55.6	312.3	0.619	4.26×10^7
70	60.5	303.1	0.656	4.44×10^7
80	70.0	299.0	0.710	4.78×10^7
90	69.2	292.1	0.714	4.75×10^7
95	72.6	291.7	0.732	4.86×10^7

(1) Col. 10 Table 16 minus 820 ft.

(2) See note 6 Table 5.

TABLE 19

Comparison of Pitot Gaging Data 1937-8-9

Item		1937	1938	1939
1. Valve 95% open		V-A1	V-A1	V-A1
2. Reservoir elevation	ft.	1033.0	1102.3	1157.1
3. Discharge measured at valve	c.f.s.	3195	3395	3855
4. Diverted upstream to exciter P-A8	c.f.s.	0	810	617
5. Energy loss to valve manifold (estimated)	ft.	0.6	0.9	1.1
6. Energy elevation at manifold	ft.	1032.4	1101.3	1156.0
7. Gross head at manifold above valve outlet (Elev. 820)	ft.	212.4	281.3	336.0
8. Measured mean energy elevation at impact of pitot tube	ft.	1000.3	1063.4	1111.7
9. Velocity head from measured discharge (8.5-ft. pipe)	ft.	49.7	56.2	72.6
10. Lost head from manifold to impact of pitot tube	ft.	32.1	37.9	44.3
11. Loss coefficient $K \frac{v^2}{2g}$ from manifold to impact		0.65	0.675	0.61
12. Energy head on valve outlet (El. 820)	ft.	180.3	243.4	291.7
13. Valve coefficient using outlet area		0.770	0.706	0.732
14. Relative discharge from gross head at manifold (square root law)		0.795	0.916	1.0
15. Relative discharge from pitot gaging		0.828	0.882	1.0
Comparison with Interpolated Values*				
16. Discharge	c.f.s.	3195	3540*	3855
17. Relative discharge from arrived value		0.795	0.916	1.0
18. Lost head from manifold to impace of pitot tube		31.3	37.9	44.3
19. Relative discharge for values in line 18 (square root law)		0.84	0.925	1.0
20. Coefficient of loss, manifold to impact		[illegible]	0.62	0.61
21. Valve coefficient of discharge from interpolated values		0.770	0.735	0.732
22. Water temperature	°F.	55±	54	61
23. Reynolds Number	$\times 10^7$.	3.7	3.9	4.9

*See Table 20.

TABLE 20

Interpolation for 1938 Discharge

Method of interpolation	Head (h) proportional to	Measured quantities in			Computed 1938 value of Q from data of		
		1937	1938	1939	1937	1939	Mean
1. Valve open 95%		V-A1	V-A1	V-A4			
2. Pitot gaging results c.f.s		3195	3395	3855			3395
3. Measured drop between 125-ft. spaced piezometer rings ft.	V^2	(1) 16.5	(2) 20.2		3540		
4. Measured drop between 125-ft. spaced piezometer rings ft.	$V^{1.8}$	16.5	20.2		3520		
5. Lost head manifold to impact leg of pitot tube, measured ft.	V^2	32.1	37.9	44.3	3470	3570	3520
6. Lost head manifold to impact leg of pitot tube, measured ft.	$V^{1.8}$	32.1	37.9	44.3	3510	3530	3520
7. Lost head manifold to impact leg of pitot tube from 1937 curve ft.	V^2	31.3	37.9	44.3	3510	3570	3540
8. Lost head manifold to impact leg of pitot tube from 1937 curve ft.	$V^{1.8}$	31.3	32.9	44.3	3550	3530	3540
9. Lost head in intake tower for 1939 Q = 3855 c.f.s. ft.	V^2		(3) 10.4	0.453		(4) 3670	

(1) Taken from curve of 1937 readings, Table 5.

(2) Mean of V-A1 and V-A6 mercury manometer readings, Table 8.

(3) Six valves open.

(4) See Table 21, second part.

Conclusion: Use Q = 3540 c.f.s. ± 35 c.f.s.

TABLE 21

1938 Relative Flows in V-A1 and Manifold
for Various Numbers of Valves Open

No. of valves open	1	2	3	4	5	6
1. Relative energy head at V-A1 (from curve)	1.0	0.992	0.974	0.958	0.943	0.923
2. Relative velocity in V-A1	1.00	0.996	0.987	0.979	0.972	0.961
3. Mean discharge per valve in terms of V-A1 flow c.f.s.	1.000	0.994	0.995	0.996	0.997	0.998
4. Mean discharge per valve in terms of gaged flow in V-A1	1.0	0.988	0.982	0.974	0.969	0.959
5. Mean discharge per valve c.f.s.	3,540	3,500	3,475	3,450	3,430	3,395
6. Total discharge in manifold c.f.s.	3,540	7,000	10,425	13,800	17,150	20,370
7. Diverted in P-A8 c.f.s.	810	780	675	420	340	300
8. Q in tower c.f.s.	4,350	7,780	11,100	14,220	17,490	20,670

<u>1939 Valve Discharge (1938 conditions)</u>

1938 - Measured tower drop = 10.4 ft. (6 valves open).
1939 - Measured tower drop = 0.453 (V-A4 valve open).

1939 - Tower Q = 3,855 c.f.s. measured + 617 c.f.s. in P-A8 = 4,472 c.f.s.

1938 - Tower Q (from 1939 gaging) $= \left(\frac{10.4}{0.453}\right)^{\frac{1}{2}} \times 4,472 = 21,450$ c.f.s. (6 valves open).

Diverted in P-A8 = $\underline{300}$ c.f.s.

1938 - Manifold Q (1939 gaging) = 21,150 c.f.s. (6 valves open).
1938 - Manifold Q (line 6 above) = 20,370 c.f.s. (6 valves open).
1938 - Q in V-A1 from 1939 gage $= \frac{21,150}{20,370} \times 3,540 = 3,670$ c.f.s.
= 1.038 % greater than value used.

TABLE 22

1938 Hydraulic Losses
Based on Interpolated 1937 - 39 Ratings

Valves open		1	2	3	4	5	6
1. Lake elevation	ft.	1102.3	1102.4	1102.6	1103.4	1103.3	1103.5
2. Q in 25-ft. manifold in terms of flow in V-A1 as unity		1.00	1.98	2.95	3.90	4.85	5.76
3. Q in 25-ft. manifold	c.f.s.	3,540	7,000	10,425	13,800	17,150	20,370
4. Q in P-A8, average	c.f.s.	810	780	675	420	340	300
5. Q in tower	c.f.s.	4,350	7,780	11,100	14,220	17,490	20,670
6. Q in tower in terms of one valve as unity		1.00	1.79	2.55	3.27	4.02	4.75
7. Valves open during tower to P-A6 readings		2,6	1,2 1,5 1,6	1,2,3	1,4,5,6	1,2,3, 4,5	1,2,3,4, 5,6
8. Ratio to (7) of valves open 94% (Nos.1,4,5)		0	0.50 1.00 0.50	0.33	0.75	0.60	0.50
9. Valves open during P-A6 to V-AI manometer readings		6		4,5,6 2,3,4	2,3,4,5; 3,4,5,6	2,3,4, 5,6	
10. Ratio of (9) open 94%		0		0.33	0.25	0.40	
11. Q in V-A1	c.f.s.	3,540	3,525	3,495	3,465	3,440	3,400
12. h_v tower to P-A8 Q in line 5, 30-ft. pipe area = 707 sq.ft.	ft.	0.59	1.88	3.84	6.30	9.53	13.30
13. h_v P-A8 to end 30-ft. pipe, Q in line 3	ft.	0.39	1.52	3.38	5.92	9.15	12.90
14. h_v (25-ft. pipe area = 491 sq.ft.) = line 13 $\times\left(\frac{707}{491}\right)^2$ Q in line 3	ft.	0.81	3.15	7.01	12.27	18.97	26.74
15. h_v (8.48-ft. pipe area = 56.5 sq.ft.) Q in V-A1, line 11	ft.	61.0	60.5	59.5	58.5	57.7	56.4
16. h_v (8.0-ft. pipe area = 50.0 sq. ft.) = line 15 $\times\left(\frac{56.5}{50.0}\right)^2$	ft.	78.1	77.4	76.1	74.8	73.9	72.2

TABLE 22
(Continued)

Valves open	1	2	3	4	5	6
17. Δh_v between tower and 25-ft. section (line 14 minus 12) ft.	0.22	1.27	3.17	5.97	9.44	13.44
18. Δh_v between P-A8 and 25-ft. section (line 14 minus 13) ft.	0.42	1.63	3.63	6.35	9.82	13.84
19. Δh_v between 25-ft. and 8.5-ft. pipe sections (line 15 minus 14) ft.	60.2	57.3	52.5	46.2	38.7	29.7
20. Δh_v between 8.5-ft. and 8.0-ft. pipe sections (line 16 minus 15) ft.	17.1	16.9	16.6	16.3	16.2	15.8
Measured Piezometric Drops						
23. Drop from lake to tower water surface by tape ft.	0.47	1.55	3.05	5.12	7.59	10.40
24. From lake to mean of P-A1 and P-A6, table 6, column 11 ft.	1.90	3.70	7.20	11.50	17.90	24.10*
25. From P-A1 to P-A6 by manometers ft.	0.22	0.66	1.12	1.98	2.93	3.98
26. From P-A6 to V-A1 manifold ft.	0.63	2.75*	5.71	9.21	14.36	20.16*
27. Total drop from lake to V-A1 manifold [Lines 24 + ½ (25) + 26] ft.	2.64	6.78	13.47	21.70	33.73	46.25
28. Total drop from lake to V-A1 manifold by Bourdon gages, Table 9, col. 8 ft.	-4.0		11.3	21.5	32.5	
29. Head lost from lake to mean impact in V-A1 (center impact 1.5% above mean on plate 4) = line 1 - 820 + col. 4 table 11 + 1.5% h_v line 16 ft. (See also line 42)	39.0	40.1	46.2	51.0	50.9	59.1

*Interpolated.

TABLE 22
(Continued)

Valves open		1	2	3	4	5	6
<u>Piezometric Drops by Sections</u>							
30. From lake to tower	ft.	0.47	1.55	3.05	5.12	7.59	10.40
31. From tower to P-A1 (lines $\frac{24-25-23}{2}$)	ft.	1.32	1.82	3.59	5.39	8.85	11.73
32. From P-A1 to P-A6 by manometer	ft.	0.22	0.66	1.12	1.98	2.93	3.98
33. From P-A6 to V-A1 manifold	ft.	0.63	2.75*	5.71	9.21	14.36	20.16*
34. From V-A1 manifold to pitot tube impact = line 29 minus 27	ft.	36.4	33.2	32.7	29.3	17.2	12.8
35. From pitot impact to atmosphere	ft.	243.3	242.4	236.4	232.4	232.4	224.5
36. Elev. center line of valve	ft.	820.0	820.0	820.0	820.0	820.0	820.0
37. Total equals lake elevation (sum of lines 30 to 36, inc.)	ft.	1102.3	1102.4	1102.4	1103.4	1103.3	1103.5
<u>Accumulated Piezometric Drops</u>							
38. From lake to tower	ft.	0.47	1.55	3.05	5.12	7.59	10.40
39. From lake to P-A1	ft.	1.79	3.37	6.64	10.51	16.44	22.11
40. From lake to P-A6	ft.	2.01	4.03	7.76	12.49	19.37	26.09
41. From lake to V-A1 manifold	ft.	2.64	6.78	13.47	21.70	33.73	46.25
42. From lake to pitot impact	ft.	39.04	39.98	46.17	51.00	50.93	59.05
<u>Energy Loss by Sections</u>							
44. Reaction head in intake tower $h_{(pu)}$. R_n is taken from Fig. 48 Bul.2, p.91 for a prototype Q equal to tower Q line 5 above. Then these R_n's are substituted in Fig.49 for $\frac{h_p}{h_v}$ for R_n 18.000 - 72,500 and $\frac{h_p}{h_v}$ = 0.80 for 72,500 + for upper gate. $h_{(pu)} = h_p$ for upper gate open	ft.	0.63	1.74	3.27	5.04	7.62	10.63

TABLE 22
(Continued)

Valves open	1	2	3	4	5	6
45. Loss from lake to tower = (line 23, ft.)	0.47	1.55	3.05	5.12	7.59	10.40
$h_{(pu)}$ + tower drop - h_v (line 12, ft.)	-0.59	-1.88	-3.84	-6.30	-9.53	-13.30
= lines 44 + 23 - 12 = line 45, ft.	0.51	1.41	2.48	3.86	5.68	7.73
46. From tower to P-A1 (line 39, ft.)	1.79	3.37	6.64	10.51	16.44	22.11
= lines 39 - (45 + 12) (line 12 + 45, ft.)	-1.10	-3.29	-6.32	-10.16	-15.21	-21.03
ft.	0.69	0.08	0.32	0.35	1.23	1.08
47. From P-A1 to P-A6 paralleling manometric drop line 25 ft.	0.22	0.66	1.12	1.98	2.93	3.96
48. From P-A1 to V-A1 manifold = lines 41 - 40 - 17 ft.	0.41	1.48	2.54	3.24	4.92	6.72
49. From V-A1 manifold to pitot impact = lines 42 - 41 + 44 ft.	37.21	36.35	39.71	41.57	36.17	39.54
50. Total: lines 45 to line 49 = line 42, ft.	39.04	39.98	46.17	51.00	50.93	59.05
Total Energy Loss						
52. From lake to P-A1 = lines 45 + 46 ft.	1.20	1.49	2.80	4.21	6.91	8.81
53. From lake to P-A6 = lines 52 + 47 ft.	1.42	2.15	3.92	6.19	9.84	12.79
54. From lake to V-A1 manifold = line 53 + 48 ft.	1.83	3.63	6.46	9.43	14.76	19.51
55. From lake to pitot impact line 54 + 49 = line 50 ft.	39.04	39.98	46.17	51.00	50.93	59.05
Energy Loss Coefficient K by Sections						
K = (Energy loss) ÷ h_v						
56. Piezometric drop from lake to tower water surface line $\frac{23}{12}$	0.797	0.824	0.794	0.813	0.797	0.782
57. Energy loss from lake to tower line $\frac{45}{12}$	0.864	0.750	0.646	0.613	0.596	0.581

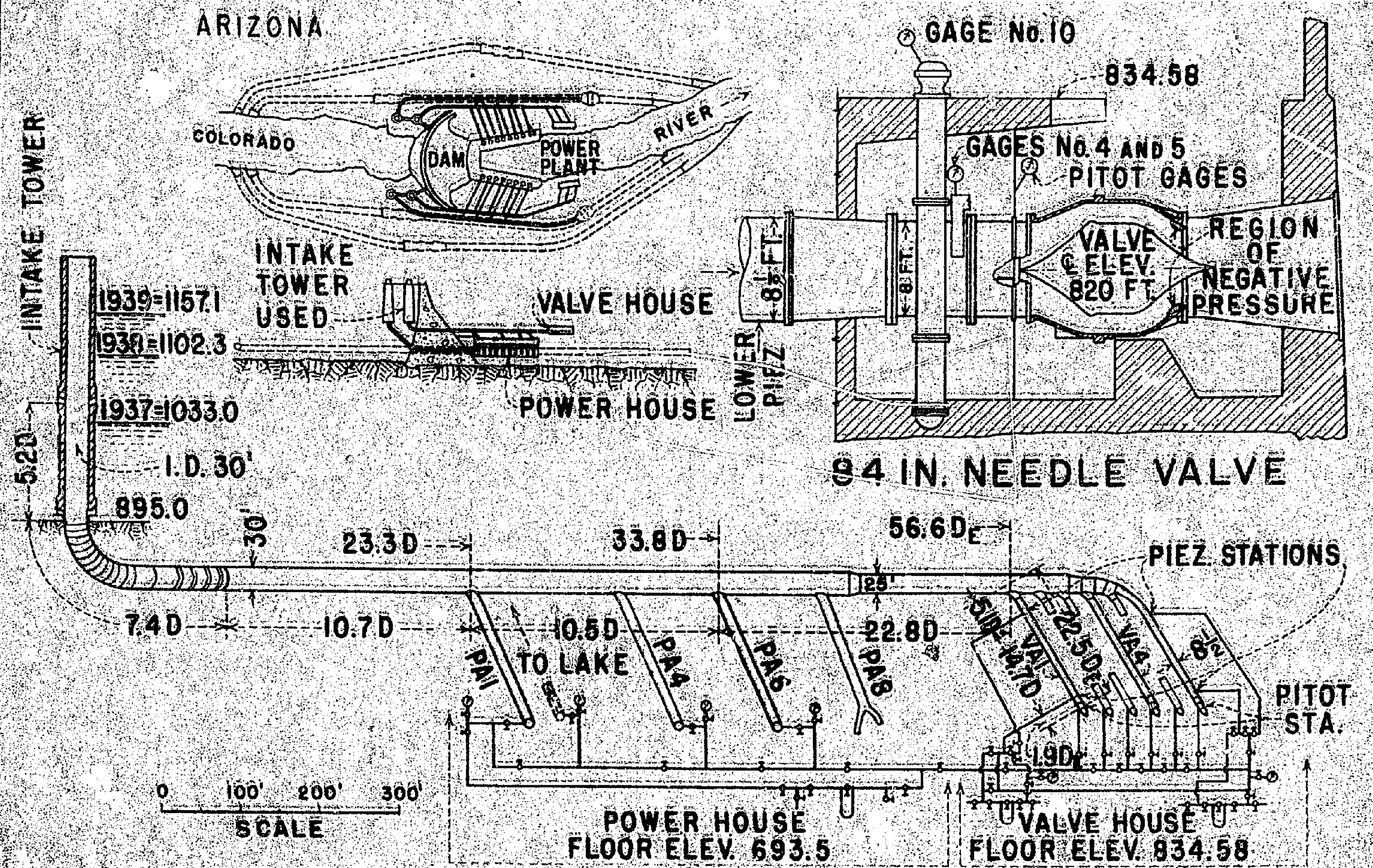
TABLE 22
(Continued)

Valves open	1	2	3	4	5	6
58. From tower to P-A1 = line $\frac{46}{12}$	1.170	0.043	0.083	0.056	0.128	0.081
59. From P-A1 to P-A6 = line $\frac{47}{12}$	0.373	0.351	0.291	0.314	0.309	0.299
60. From P-A6 to V-A1 manifold = line $\frac{48}{13}$	1.051	0.974	0.752	0.548	0.538	0.521
61. From V-A1 manifold to pitot impact = line $\frac{49}{15}$	0.610	0.601	0.668	0.694	0.627*	0.701
"f" by Sections						
"f" = K ÷ $\frac{L}{D}$						
62. From lake to P-A1, L = 23.3 diam. = line 74 below	0.087	0.034	0.031	0.029	0.031	0.0284
63. From P-A1 to P-A6 L = 10.5 diam. = $\frac{\text{line 59}}{10.5}$	0.036	0.033	0.028	0.030	0.029	0.0285
64. From P-A6 to V-A1 manifold, L = 22.8 diam. (30 ft.) = $\frac{\text{line 60}}{22.8}$	0.046	0.043	0.033	0.024	0.024	0.023
65. From P-A6 to V-A1 manifold, L = 10.46 diam. (25 ft.) = $\frac{\text{line 48}}{10.46 \times \text{line 14}}$	0.048	0.045	0.035	0.025	0.025	0.024
66. From V-A1 manifold to pitot impact, L = 22.5 diam. ($8\frac{1}{2}$ ft.) = $\frac{\text{line 61}}{22.5}$	0.027	0.027	0.030	0.031	0.028	0.031

*Interpolated.

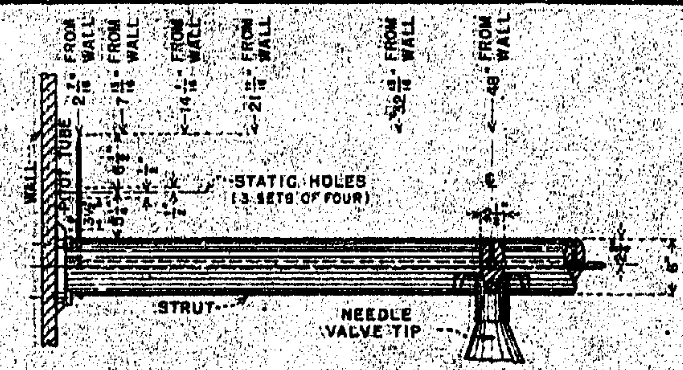
TABLE 22
(Continued)

Valves open	1	2	3	4	5	6
Over-all Energy Loss Coeff. K						
67. From lake to tower = line 57	0.864	0.750	0.646	0.613	0.596	0.581
68. From lake to P-A1 = line $\frac{52}{12}$	2.03	0.793	0.729	0.668	0.724	0.662
69. From lake to P-A6 = line $\frac{53}{12}$	2.41	1.14	1.02	0.982	1.03	0.962
70. From lake to V-A1 manifold Sum lines 57 to 60 inc.	3.46	2.12	1.77	1.53	1.57	1.48
Over-all "f" from Lake						
74. From lake to P-A1 (length = 23.3 diam.) = $\frac{\text{line 68}}{23.3}$	0.087	0.034	0.031	0.029	0.031	0.0284
75. From lake to P-A6 (L = 33.8 diam.) = $\frac{\text{line 69}}{33.8}$	0.071	0.034	0.030	0.029	0.030	0.0285
76. From lake to V-A1 manifold (L = 56.6 D.) = $\frac{\text{line 70}}{56.6}$	0.061	0.037	0.031	0.027	0.028	0.026
Reynolds Number (55 deg. F.)						
77. V from line 12 to lake to P-A6 - 30 ft. pipe $\times 10^7$	1.42	2.53	3.61	4.63	5.70	6.74
78. V from line 14 25-ft. pipe in manifold $\times 10^7$	1.38	2.73	4.08	5.39	6.71	7.98
79. V from line 13 V-A1 manifold to pitot impact $8\frac{1}{2}$-ft. diam. $\times 10^7$	4.08	4.03	4.00	3.97	3.95	3.90

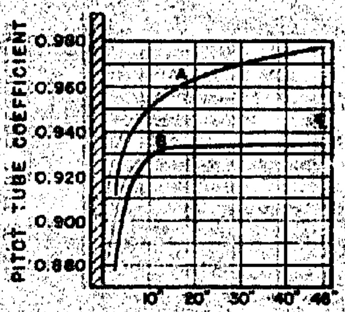


GENERAL LAYOUT

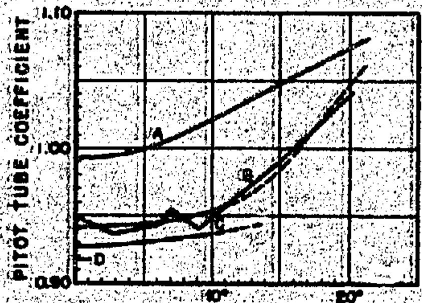
PLATE I



PITOT TUBE ASSEMBLY

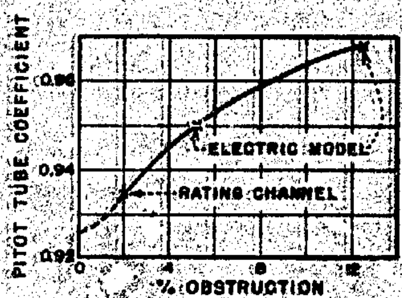


A. FINAL PITOT TUBE COEFFICIENT
B. RATING CHANNEL COEFFICIENT

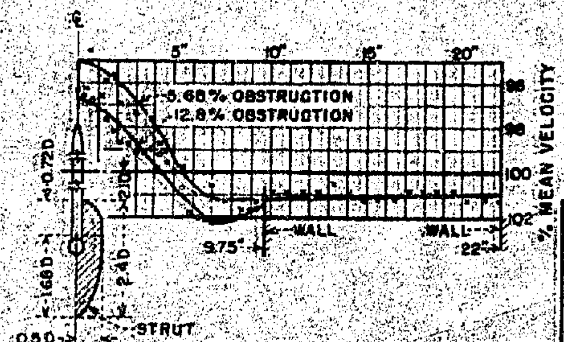


EFFECT OF PITCH AND YAW

A. PITCH WITHOUT STRUT, 4 HOLES PER RING.
B. YAW WITH STRUT, 4 HOLES PER RING.
C. PITCH WITH STRUT, 4 HOLES PER RING.
D. PITCH WITH 2 HOLES PER RING.



EFFECT OF OBSTRUCTION ON COEFFICIENT



VARIATION OF VELOCITY ACROSS CHANNEL IN PLANE OF STATIC OPENINGS, ELECTRIC MODEL

FORMULA

$$V = \frac{1}{C_p}\left(2gH\right)^{\frac{1}{2}}$$

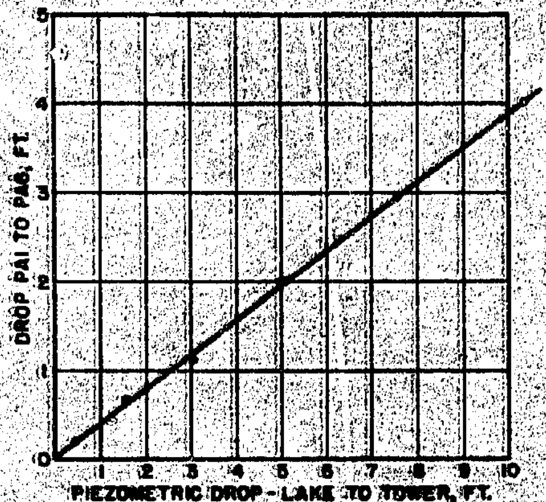
UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
BOULDER CANYON PROJECT

HYDRAULIC TESTS OF OUTLET WORKS
PITOT TUBE COEFFICIENT

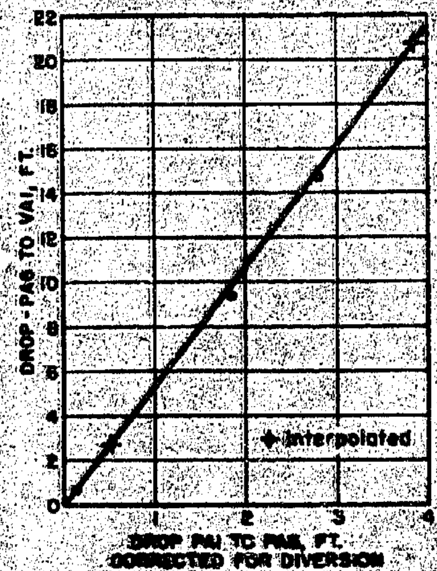
DRAWN... R.E.T.	SUBMITTED
TRACED... R.R.C. - M.E.K.	RECOMMENDED
CHECKED	APPROVED
	DENVER, COLORADO — JUN. 27, 1940

No. VALVES OPEN	DESIGNATION OF VALVES OPEN ARIZONA SIDE	PIEZOMETRIC DROP LAKE TO TOWER, FT.	MEAN PIEZ. DROP, FT. (2) PLUS 0.01 FT.	± PROBABLE ERROR %	DROP PA1 TO PA6, FT.	LAKE ELEVATION, FT.
1	6 1 5	0.45 0.47 0.44	(1) 0.47	2	0.22	1102.3
2	1,2 1,5 1,6	1.56 1.52 1.55	1.55	1/2	0.56	1102.4
3	1,2,3 4,5,6	3.03 3.04	3.05	1/2	1.11	1102.6
4	2,3,4,5 1,2,4,5 3,4,5,6	5.08 5.15 5.10	5.12	1/2	1.96	1103.4
5	1,2,3,4,5	7.50 7.67	7.59	1/2	2.93	1103.3
6	ALL	10.39	10.40	1/2	3.96	1103.6

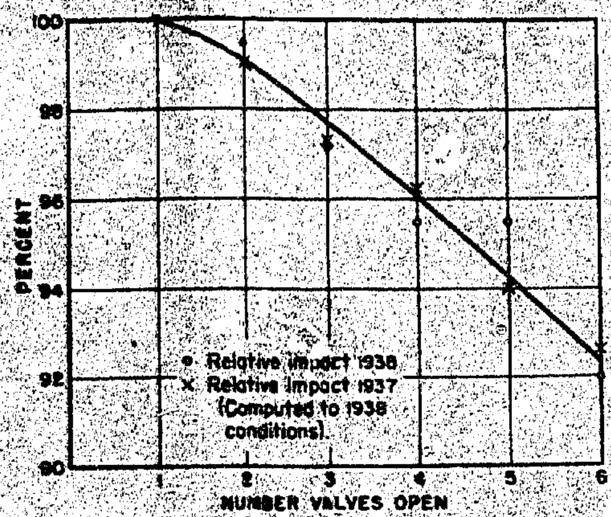
(1) Weighted.
(2) No measurement of tower drop without PA6 flowing. Estimated effect is 0.01 ft.



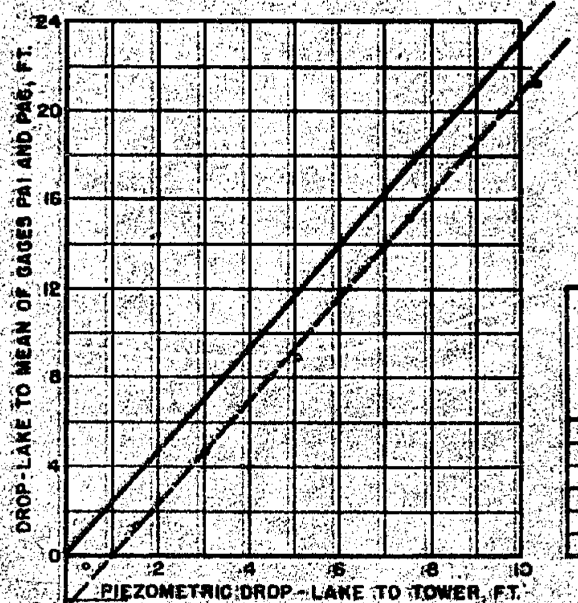
RELATION BETWEEN PIEZOMETRIC DROPS LAKE TO TOWER AND PA1 TO PA6



PA6 TO VA1 DROP VS. PA1 TO PA6 DROP CORRECTED

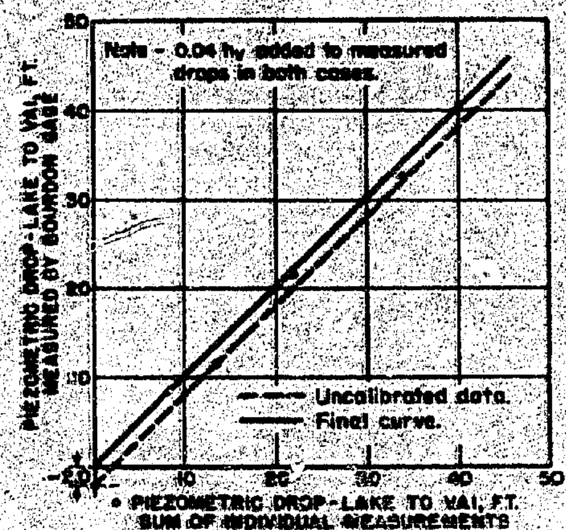


RELATIVE MEAN IMPACT HEAD AT VA1 VALVE

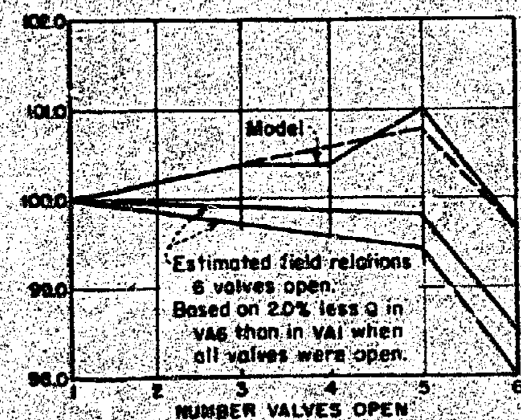


No. GATES OPEN	TOWER DROP, FT.	DROP, LAKE TO PA1-PA6, FT.
1	0.47	-0.5
2	1.55	1.3
3	3.05	4.7
4	5.12	8.9
5	7.59	15.1
6	10.40	21.2

MEAN PA1-PA6 DROP VS. TOWER DROP
FIELD DATA



COMPARISON OF PIEZOMETRIC DROPS - LAKE TO VA1 AS MEASURED BY TWO METHODS
SUM OF SEPERATE MEASUREMENTS ASSUMED CORRECT



RELATION OF VALVE DISCHARGE TO DISCHARGE OF VA1; MODEL TEST

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
BOULDER CANYON PROJECT

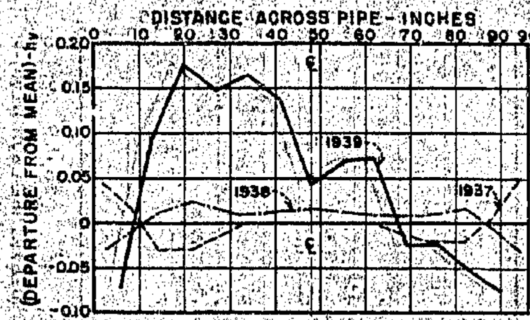
HYDRAULIC TESTS OF OUTLET WORKS

PIEZOMETER CALIBRATIONS
1938 TESTS

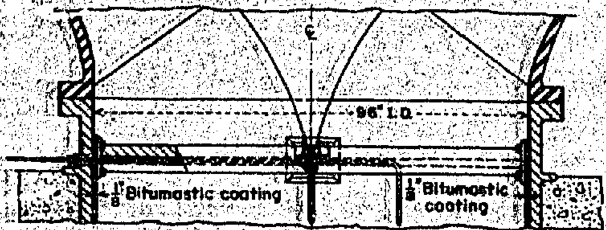
DRAWN	S.P.W.	SUBMITTED	
TRACED	D.R.D. R.L.A.	RECOMMENDED	
CHECKED	R.E.K.	APPROVED	
	DENVER, COLORADO JUNE 6, 1940		

PLATE 3

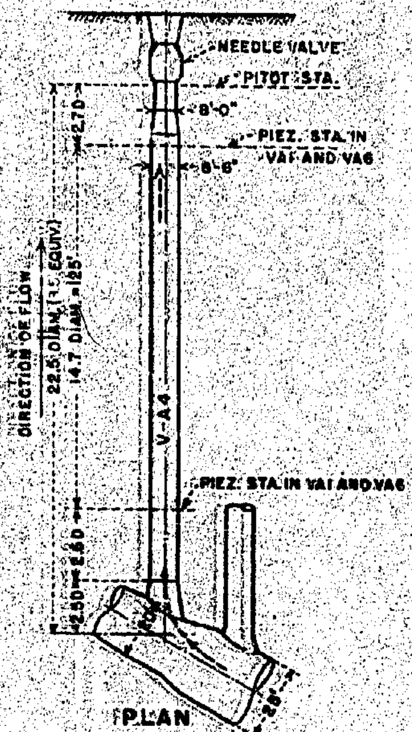
PLATE 3



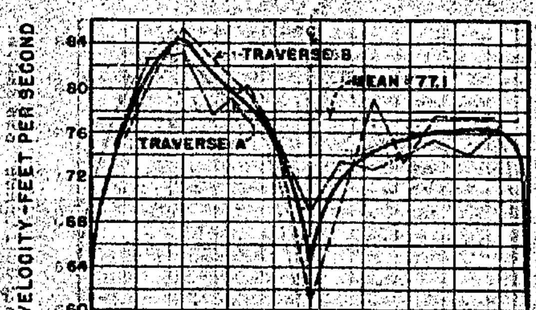
ENERGY HEAD



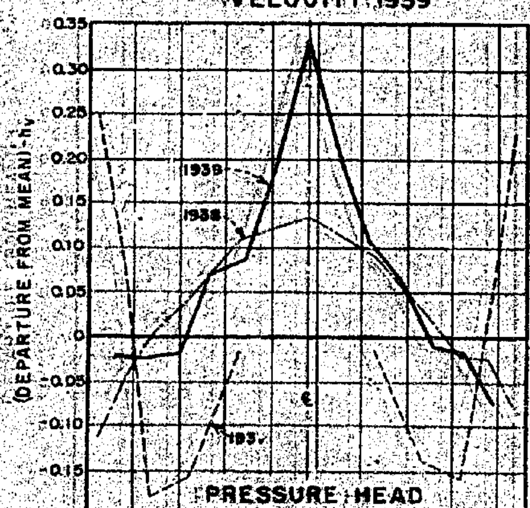
NEEDLE VALVE

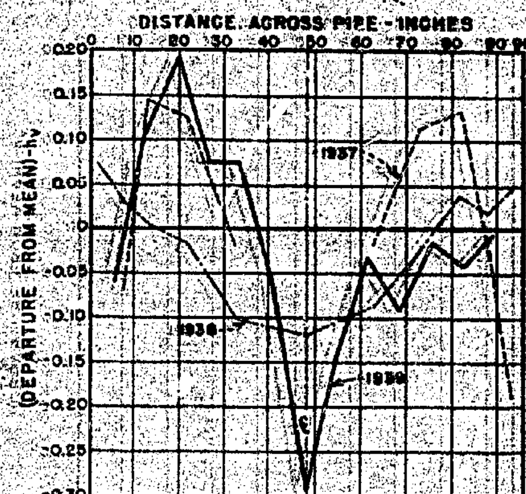


PLAN



VELOCITY 1939





VELOCITY HEAD

MEAN DATA (95% OPEN)

	1937	1938	1939
ENERGY HEAD-FT.	160.3	243.4	291.7
PRESSURE HEAD-FT.	116.8	171.6	199.0
VELOCITY HEAD-FT.	63.5	71.8	92.7
VELOCITY-FT/SEC.	53.9	67.9	77.1
DISCHARGE-C.F.S.	31[illegible] 5	339.5	385.5

UNITED STATES
DEPARTMENT OF THE INTERIOR
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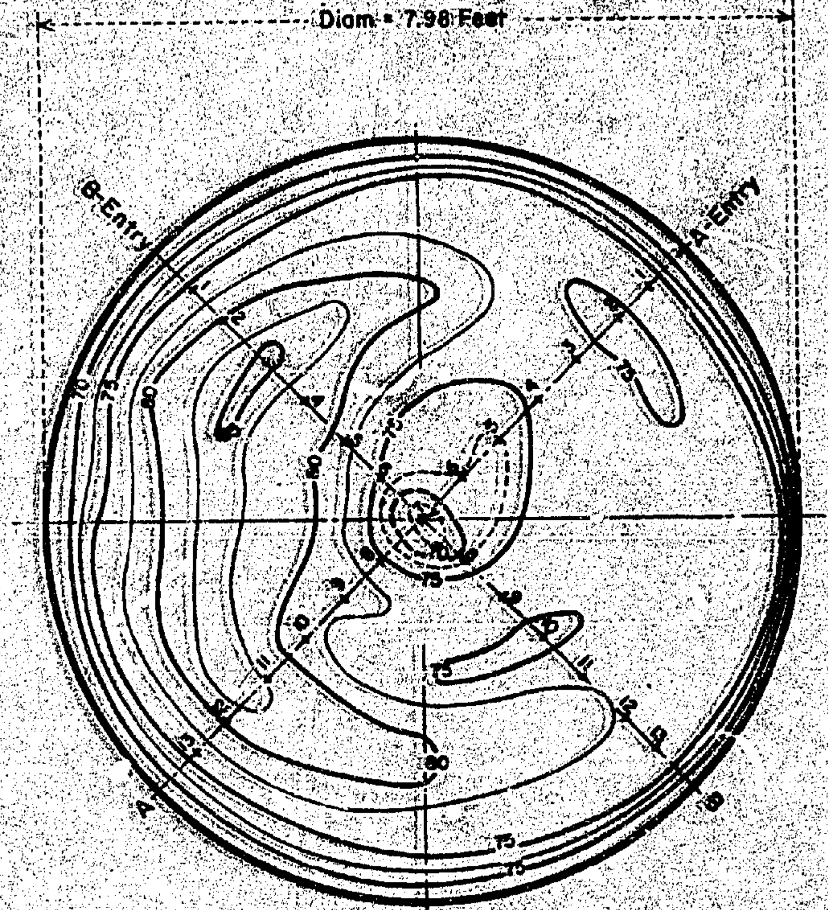
BOULDER CANYON PROJECT

HYDRAULIC TESTS OF OUTLET WORKS

IMPACT, PRESSURE AND VELOCITY HEADS
1937-1938-1939 MEAS. - 95% VALVE OPENING

DRAWN R.E.A. SUBMITTED
TRACED B.R.C.-E.R.S. RECOMMENDED
CHECKED APPROVED

DENVER, COLORADO JUNE 17, 1940



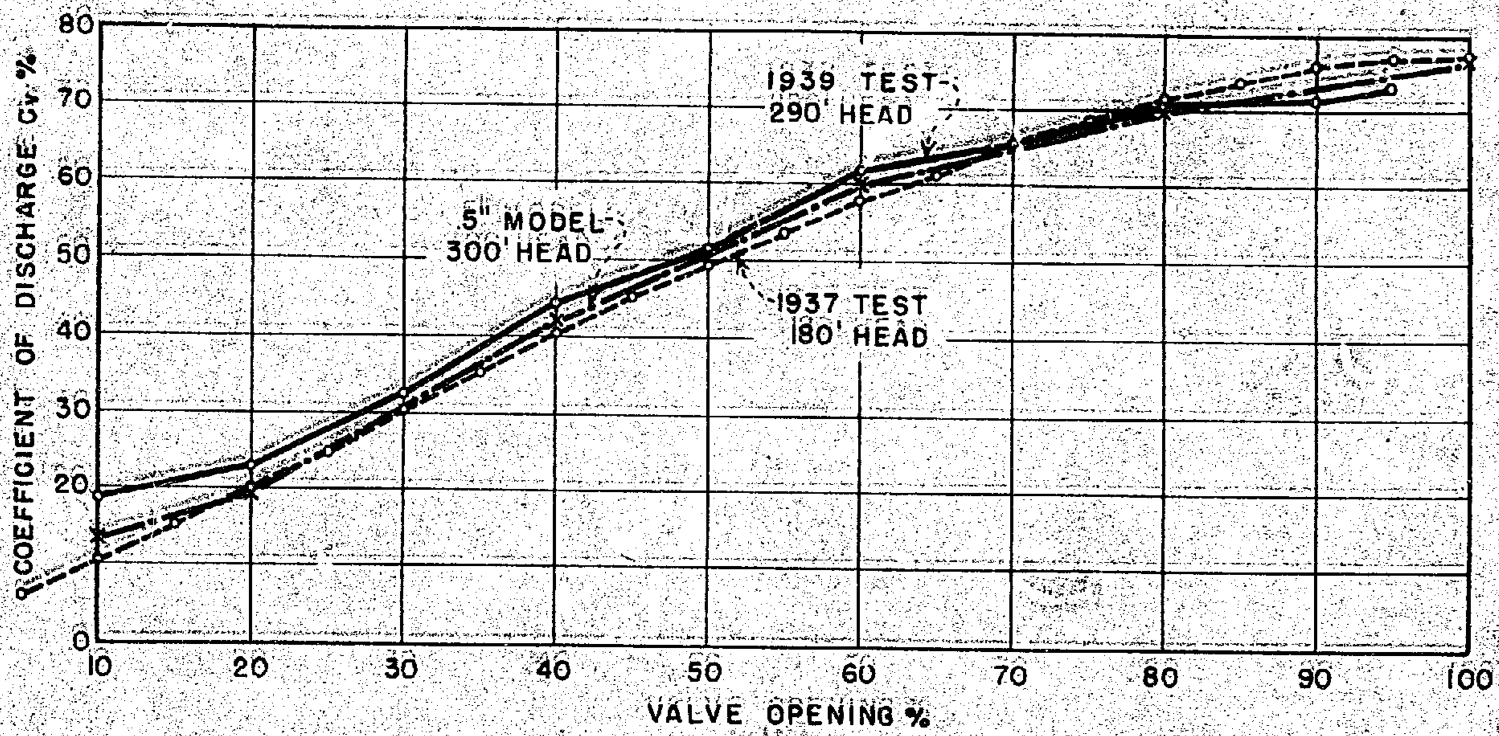
VIEW LOOKING DOWNSTREAM
1939 TEST

MEAN VELOCITY = 77.1 FEET PER SECOND
MAX VELOCITY = 85.5 " " "
RATIO; MEAN/MAX = 0.902 " " "

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
BOULDER CANYON PROJECT

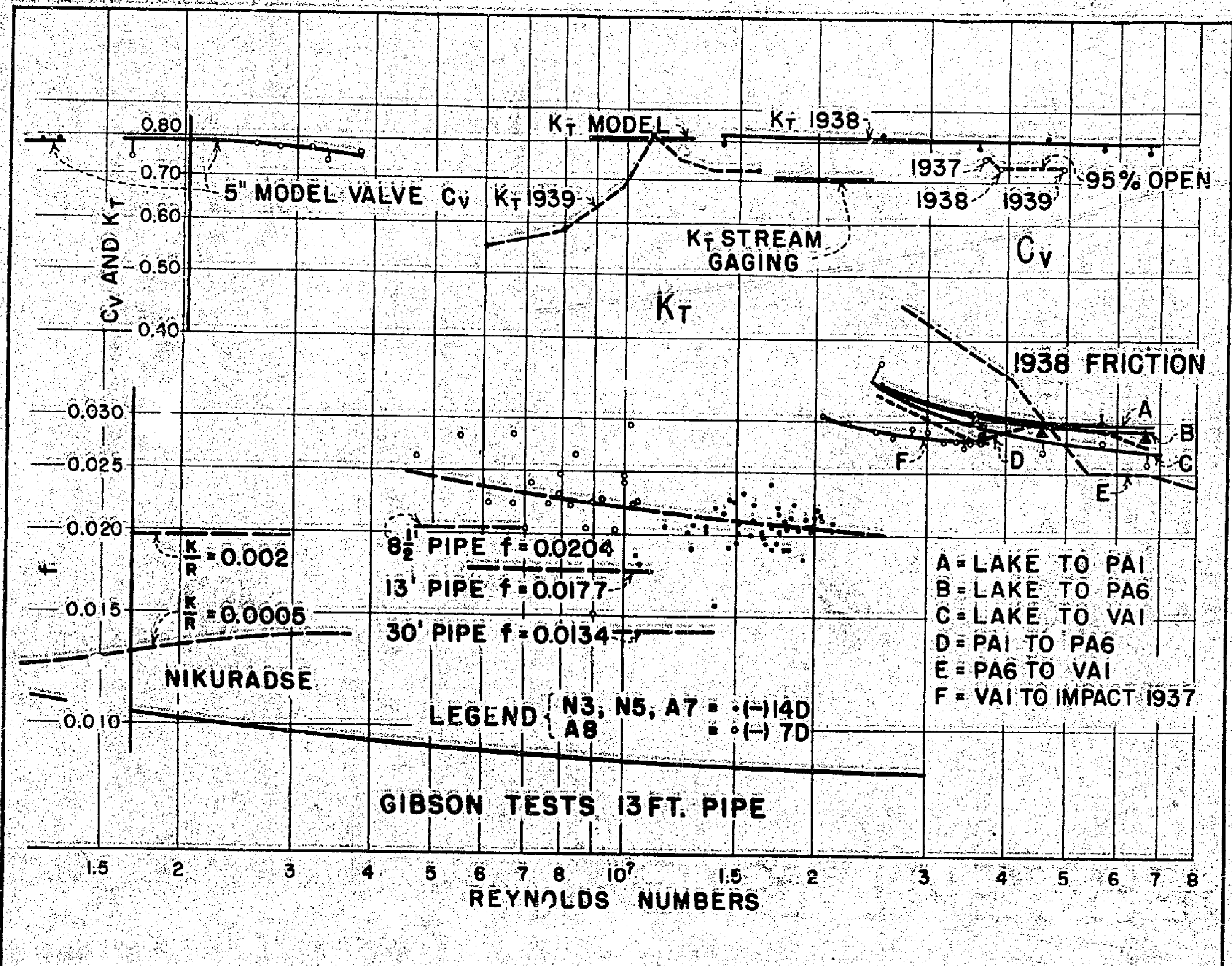
HYDRAULIC TESTS OF OUTLET WORKS
VELOCITY CONTOURS - VALVE ENTRANCE

DRAWN... R.E.K. .... SUBMITTED.............
TRACED. G.R.C.-H.S. RECOMMENDED.............
CHECKED............ APPROVED.............
DENVER, COLO. - JUNE 21, 1940



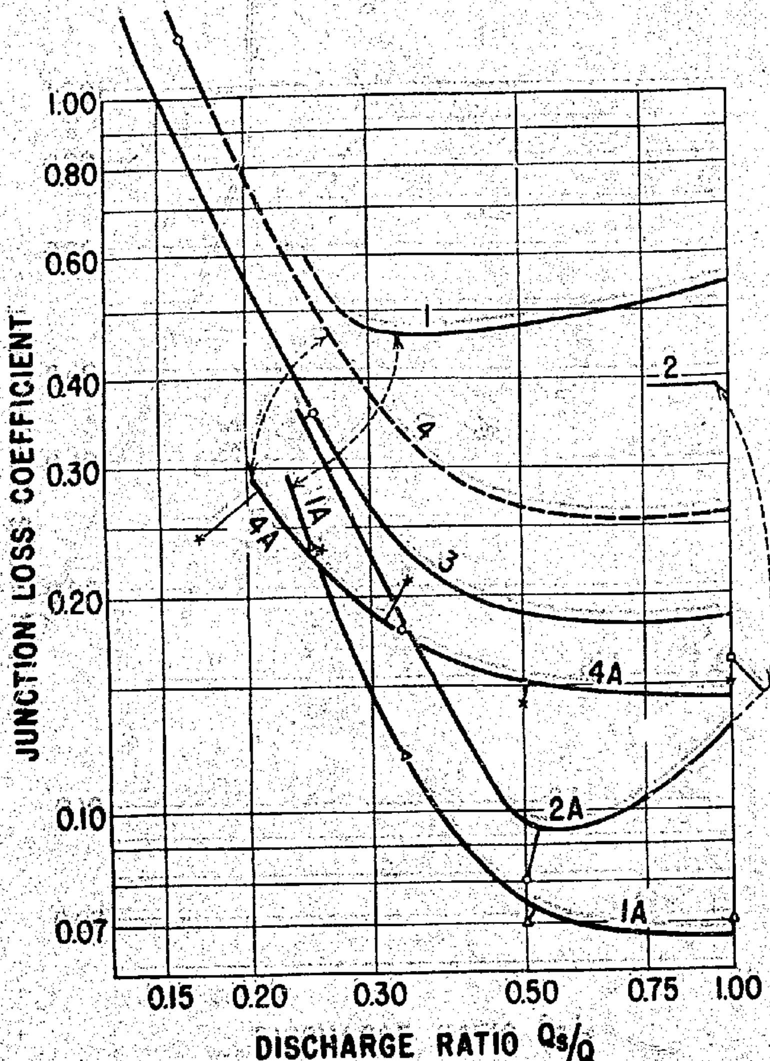
RELATION OF Cv TO % VALVE OPENING

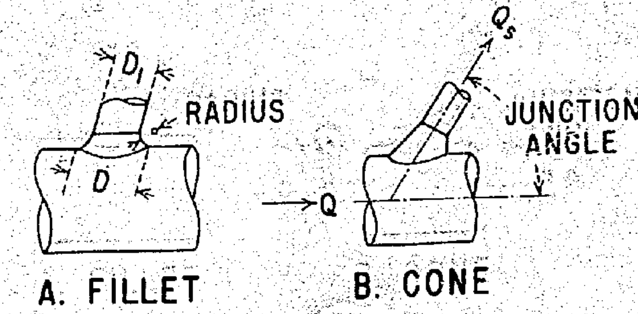
PLATE 6



CV AND KT
0.80
0.70
0.60
0.50
0.40
5" MODEL VALVE CV
KT MODEL
KT 1939
KT 1938
KT STREAM GAGING
1937
1938
1939
95% OPEN
CV
KT
1938 FRICTION
f
0.030
0.025
0.020
0.015
0.010
K/R = 0.002
K/R = 0.0005
NIKURADSE
8½" PIPE f = 0.0204
13' PIPE f = 0.0177
30' PIPE f = 0.0134
LEGEND { N3, N5, A7 = •(−) 14D; A8 = ∘(−) 7D }
A = LAKE TO PA1
B = LAKE TO PA6
C = LAKE TO VA1
D = PA1 TO PA6
E = PA6 TO VA1
F = VA1 TO IMPACT 1937
A
B
C
D
E
F
GIBSON TESTS 13 FT. PIPE
1.5
2
3
4
5
6
7
8
10^7
1.5
2
3
4
5
6
7
8
REYNOLDS NUMBERS

PLATE 7

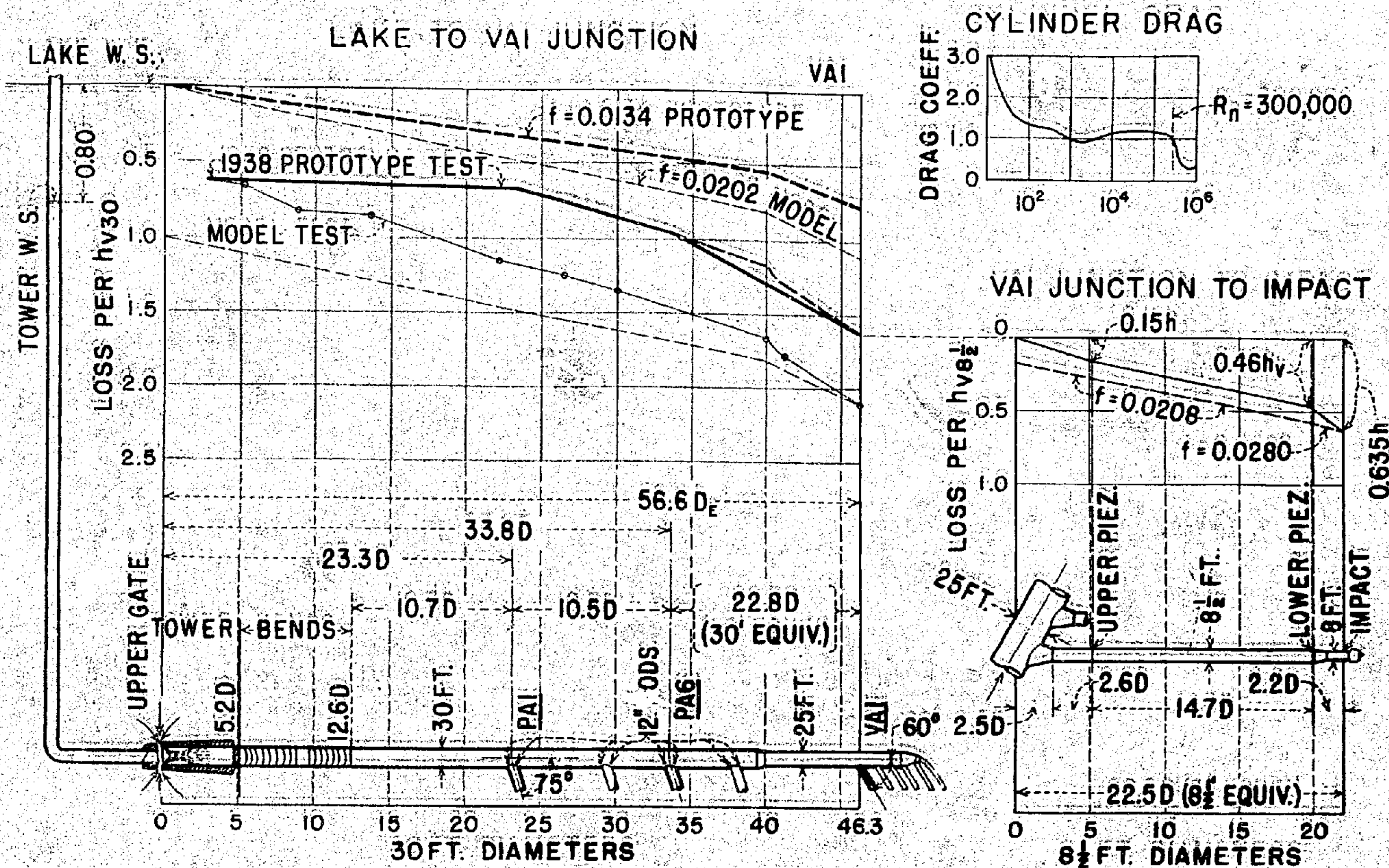




No.	TYPE	R_N	$\left(\frac{D_1}{D}\right)^4$	DESCRIPTION
1	A	10^5	0.61	THOMA; 45°; R = 0.10 D.
1A	A	3×10^5	0.65	COM. LAB; 75°; R = 0.24 D.
2	B	10^5	0.65	EST. FROM THOMA DATA.
2A	B	10^5	0.65	COM. LAB.; 75°.
3	B	10^5	0.56	U.S.B.R. 75° NOT USED.
4	B	5×10^4	0.61	U.S.B.R. VALVE-MAN. MODEL.
4A	B	4×10^7	0.61	U.S.B.R. 1938 PROTO.

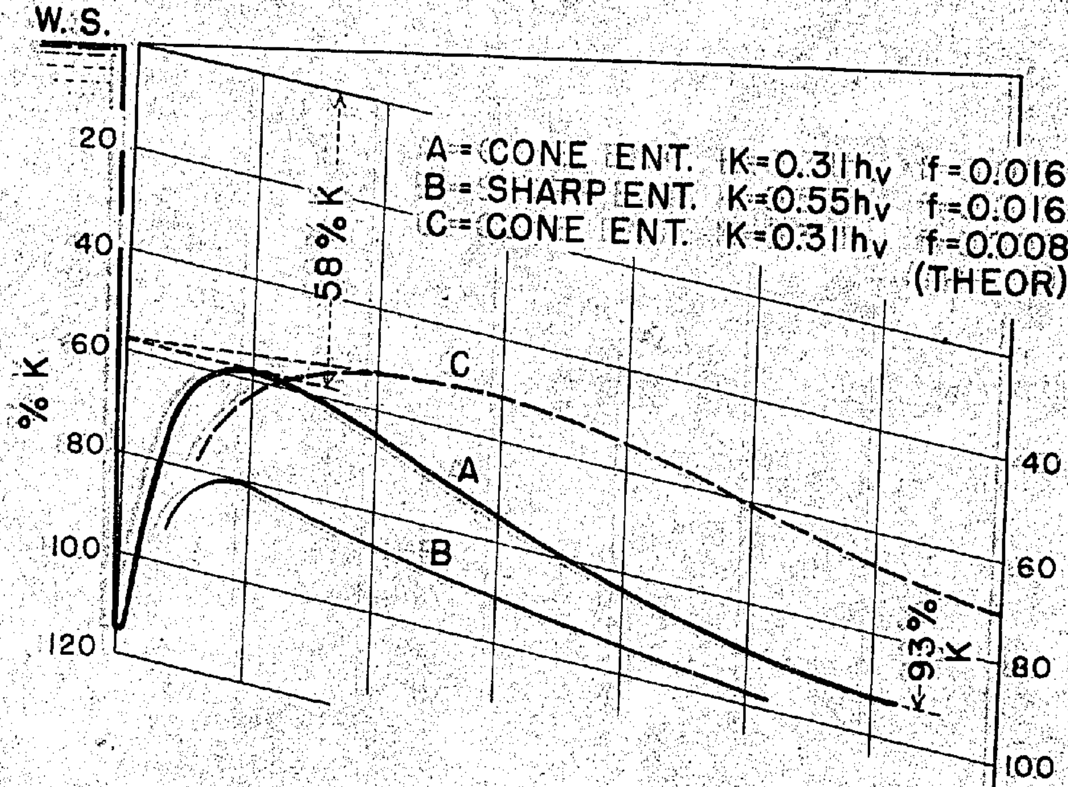
JUNCTION LOSSES

PLATE 8

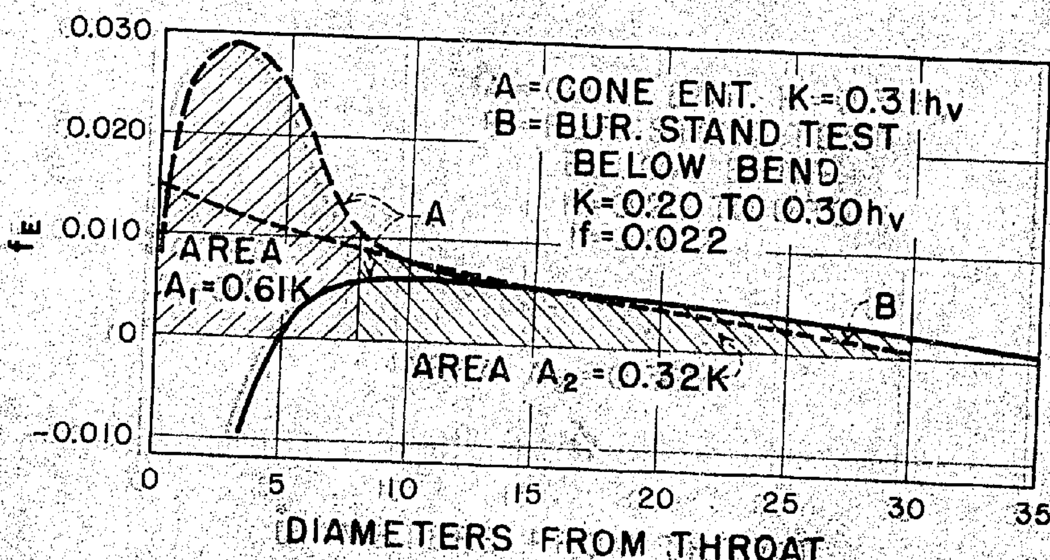


COMPARISON OF ENERGY GRADE LINES

PLATE 9

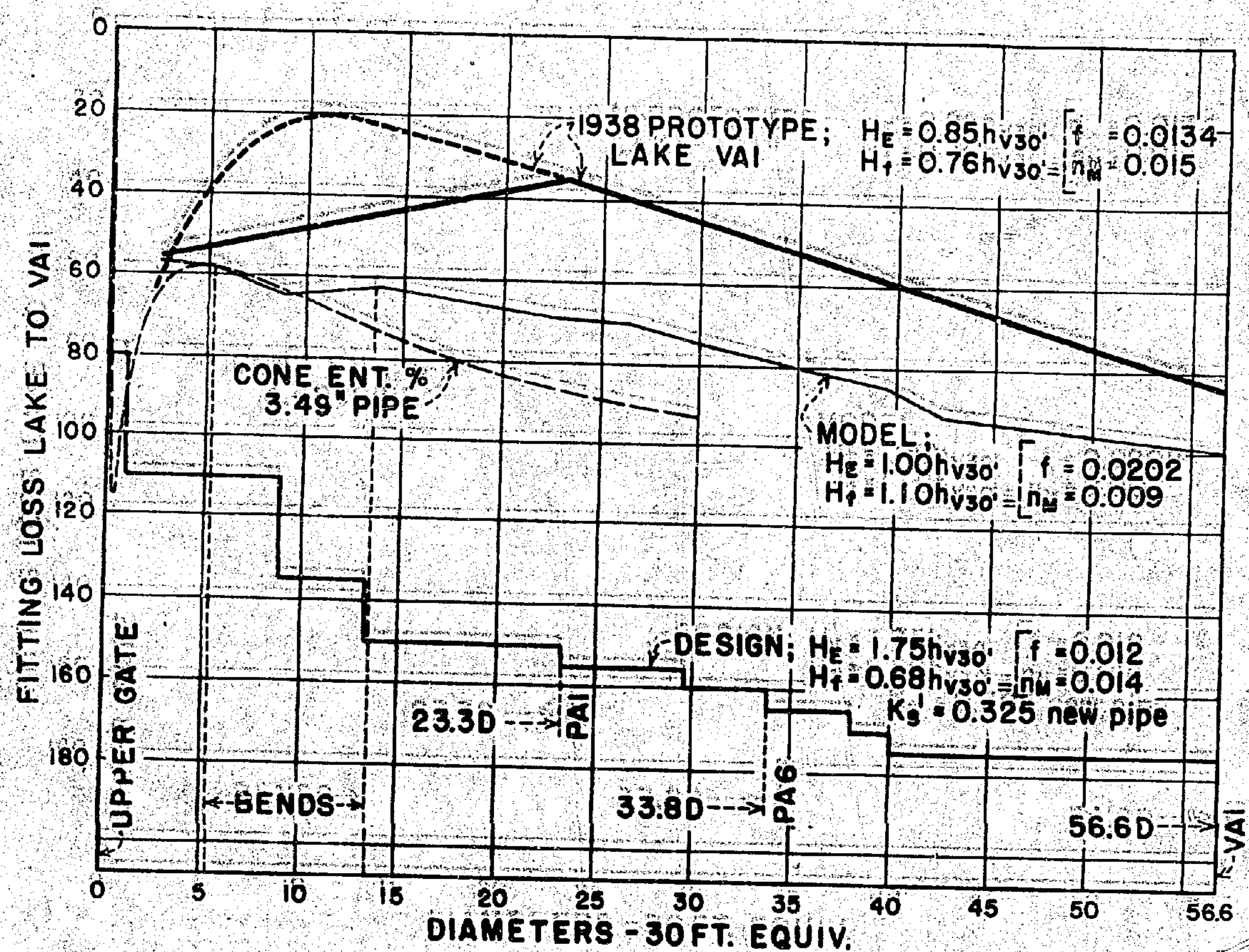


ENTRANCE LOSS IN % TOTAL



ENTRANCE LOSS PER DIAMETER

PLATE 10



COMPARISON OF FITTING LOSSES

PLATE 11