UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

HYDRAULIC MODEL STUDIES FOR TURBINES AT GRAND COULEE POWER PLANT

COLUMBIA BASIN PROJECT, WASHINGTON

Hydraulic Laboratory Report No. Hyd.-198

BRANCH OF DESIGN AND CONSTRUCTION
DENVER, COLORADO

FEBRUARY 20, 1946

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UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

Branch of Design and Construction
Engineering and Geological Control
and Research Division

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Laboratory Report No. 198
Hydraulic Laboratory
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Subject: Hydraulic model studies for the turbines at Grand Coulee Power Plant, Columbia Basin Project.

SYNOPSIS.

The unprecedented capacities of the proposed turbine units for the Grand Coulee Power Plant made it necessary to insure that every possible improvement be incorporated in their design. The more important features of the turbine setting such as draft tubes and penstock conditions near the entrance to the scroll case were studied by means of a complete homologous model turbine built to a scale of 1 to 24. Three draft-tube designs selected by pilot tests in a smaller model were tested extensively. Several other features of turbine design, including scroll case, speed ring, and fairwater cone were also studied to determine the relative effect of certain design modifications on turbine efficiency.

INTRODUCTION

The Grand Coulee Dam, a feature of the Columbia Basin Project, is located on the Columbia River about 80 miles northwest of Spokane, Washington, at a point adjacent to the head of the Grand Coulee, a wide valley in the plateau extending southward from the left bank of the river. The dam will be about 550 feet high over-all, and will raise the water level about 353 feet above normal river level at this point. A part of the potential energy stored in the reservoir will be used to pump water from the reservoir into the Grand Coulee from which it will be distributed by gravity for irrigation and the remainder will be transmitted as electrical energy to various load points. The two power plants will be located at the toe of the dam, one on each side of the spillway.

Each power plant is designed to accommodate nine of the largest hydroelectric power units ever built in point of capacity. The units will be of the vertical direct-connected type. The turbines are expected to deliver approximately 153,000 horsepower to the generators when operating at best efficiency and discharging 4,400 second-feet of water under the rated head of 335 feet.

The highest possible efficiency consistent with simplicity and sturdiness of design in these turbine units was especially desirable because of the unprecedented capacity of the units and the enormous investment involved. For this reason certain component features of the turbine unit which might affect its efficiency in converting hydraulic energy into mechanical energy were given consideration. In hydraulic turbine units the performance of each of the elements, or component parts, may be materially influenced by some other element. For instance, the performance of the draft tube may be influenced by runner characteristics, runner performance may be influenced by gate and speed-ring characteristics, and so on to the penstock intake.

Modern turbine-runner design leaves little to be expected in the way of improving runner efficiency but it appears that some features, such as the draft tube, speed ring, and scroll case have been somewhat neglected. Also there has been some question as to the effect of bends in the penstock especially when located near the scroll-case entrance. Lack of reliable design data on draft tubes led the Alabama Power Company to authorize a series of tests on model draft tubes at the Alden Hydraulic Laboratory in 1921.* Later studies** made by the Alabama Power Company at its own laboratory, without the use of a turbine runner, culminated in the Lower Tallassee draft-tube design.

* "Comparative Tests on Experimental Draft Tubes," by C. M. Allen and I. A. Winter, Transactions, A.S.C.E., volume 87, 1924, p. 893.

** Studies conducted by I. A. Winter, G. J. Hornsby and R. R. Randolph, Jr. - unpublished.

The Lower Tallassee turbines (36,000-horsepower at 88-foot head), with this draft tube, yielded an efficiency of 93 percent under test in 1930. This draft tube has been used as a basis for all Bureau of Reclamation draft-tube designs since 1933. Studies made on speed rings about the same time as the Lower Tallassee draft-tube studies were being made indicated that turbine efficiencies might be improved by certain departures from usual design practice in the shaping and positioning of the vanes.

These studies, as well as the work of other experimenters, yielded valuable information with respect to the design of individual elements but they did not supply the data necessary for evaluating the relative effects of any one element on the performance of any other element. In order to study these relative effects it was considered expedient to build a complete homologous model of one of the proposed turbine units.

At the time these model studies were authorized, the turbines for the Grand Coulee Power Plant, which were without precedent from the standpoint of horsepower requirements, had not been designed. It was therefore necessary to design a hypothetical turbine unit for normal operating conditions of 4,400 second-feet at 335-foot head to develop at least 150,000 horsepower depending upon efficiency. Under maximum conditions this unit could develop approximately 187,000 horsepower. The speed of this unit was to be 128.57 r.p.m. In designing the turbine unit, the design theories of various manufacturers capable of building turbines of this size were considered and every effort was made to anticipate, as nearly as possible, the design which might be expected from the manufacturer.

A model of the completed design was built to a geometric scale of 1 to 24. This scale ratio was adopted after careful consideration of the accuracy of the test results, laboratory space available, and cost. The special methods and techniques developed for constructing the model, are given in the appendix of this report.

The turbine features to be studied in this series of model tests were the draft tube, the speed ring, the scroll case, and the bend in the penstock near the entrance to the scroll case. Other features, such as the proportions of the fairwater cone, the use of fins and vanes for the control of power swings, etc., were incidental and not included in the original plans. Among the features to be studied, the draft tube was considered the most important because it may affect the turbine efficiency to a greater extent than any one of the other features listed. Experience has shown that the proportioning and shaping of the draft tube can be quite critical, especially in the elbow or quarter-turn type.

In laying out the plans for the powerhouse it was found that by placing the axis of the turbine unit four feet off the centerline of the bay the unit spacing could be reduced by that amount. The width of the horizontal leg of the draft tube did not permit it to be moved off the centerline of the bay by this amount so it seemed necessary to make the vertical leg and elbow of the draft tube eccentric. This was believed to be an undesirable condition as the major portion of the work done by the draft tube is accomplished in the vertical leg and the elbow. In order to make the draft tube symmetrical, it was suggested during the course of the studies that the width of the outflow end of the elbow be reduced, and the plane of symmetry be rotated about the vertical axis of the turbine so that it would intersect the central plane of the bay at the downstream face of the powerhouse substructure. This suggestion was favorably received and the draft-tube studies proceeded on this basis.

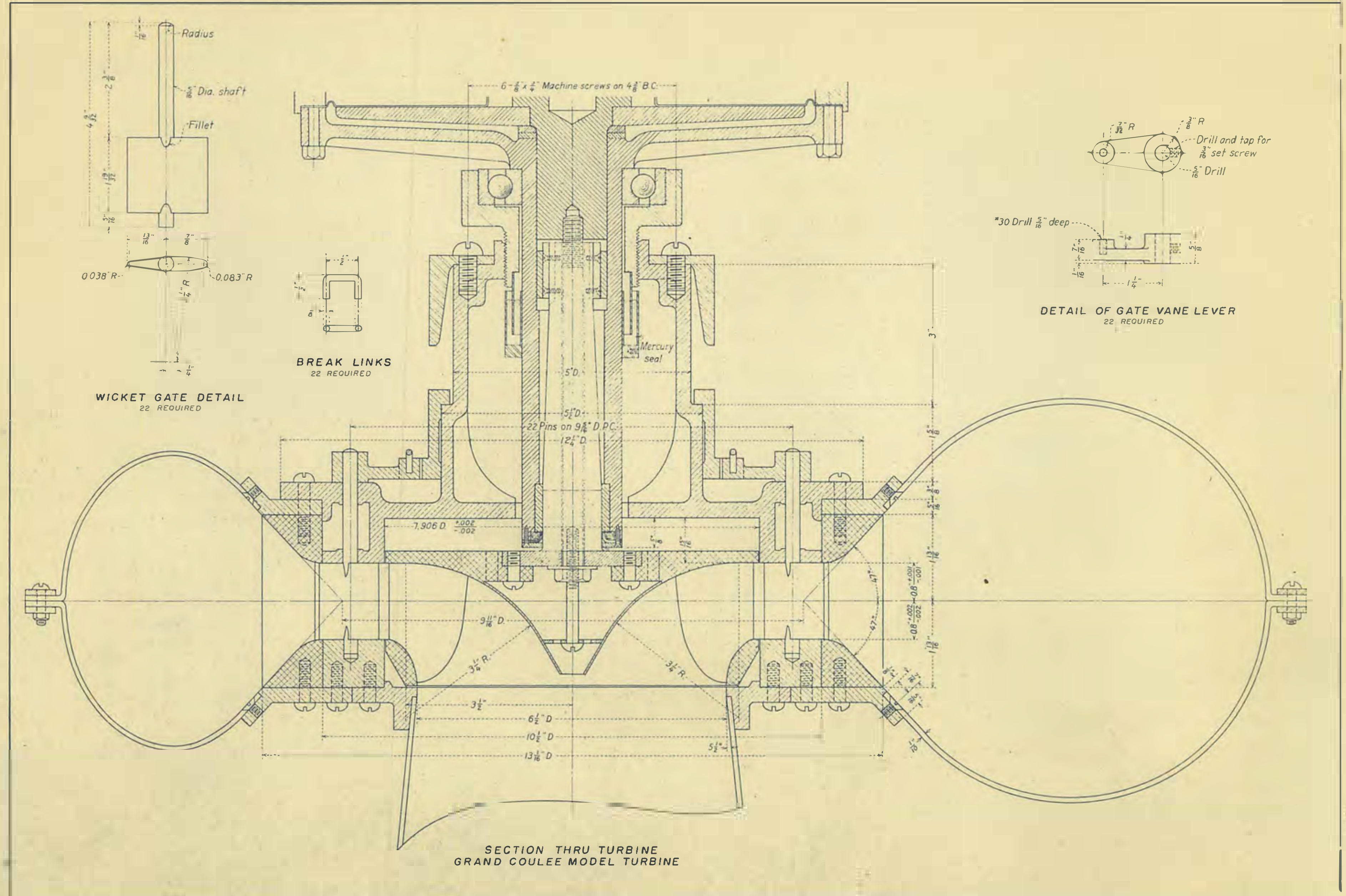
A series of preliminary tests were made without a runner, using draft-tube models having a throat diameter of three inches, for the purpose of selecting two or three of the more promising types for final comparative study in the complete model. For these tests, the flow conditions associated with a turbine runner were reproduced by using various sets of vanes which introduced the required spiral flow. Three draft-tube designs were selected for further study with the complete model turbine unit on the basis of ability to convert kinetic energy into useful work, flow characteristics, and suitability for the Grand Coulee installation. These three designs were referred to as the Norris, Lower Tallassee, and the proposed design for Grand Coulee.

DESCRIPTION OF MODEL TURBINE

The turbine model used in connection with these tests was a complete homologous model of the anticipated prototype turbine, including the draft tube and a short section of penstock. The penstock, scroll case, and draft tube were molded from a transparent, thermoplastic material to permit visual inspection and photographic recording of flow conditions. The speed ring, wicket gates, and runner were cast in a bearing metal known as "Government Genuine" babbit and were held in their respective positions by a bronze support casting. The draft tube and the scroll case were attached to this casting by bronze connecting rings as shown in figure 1.

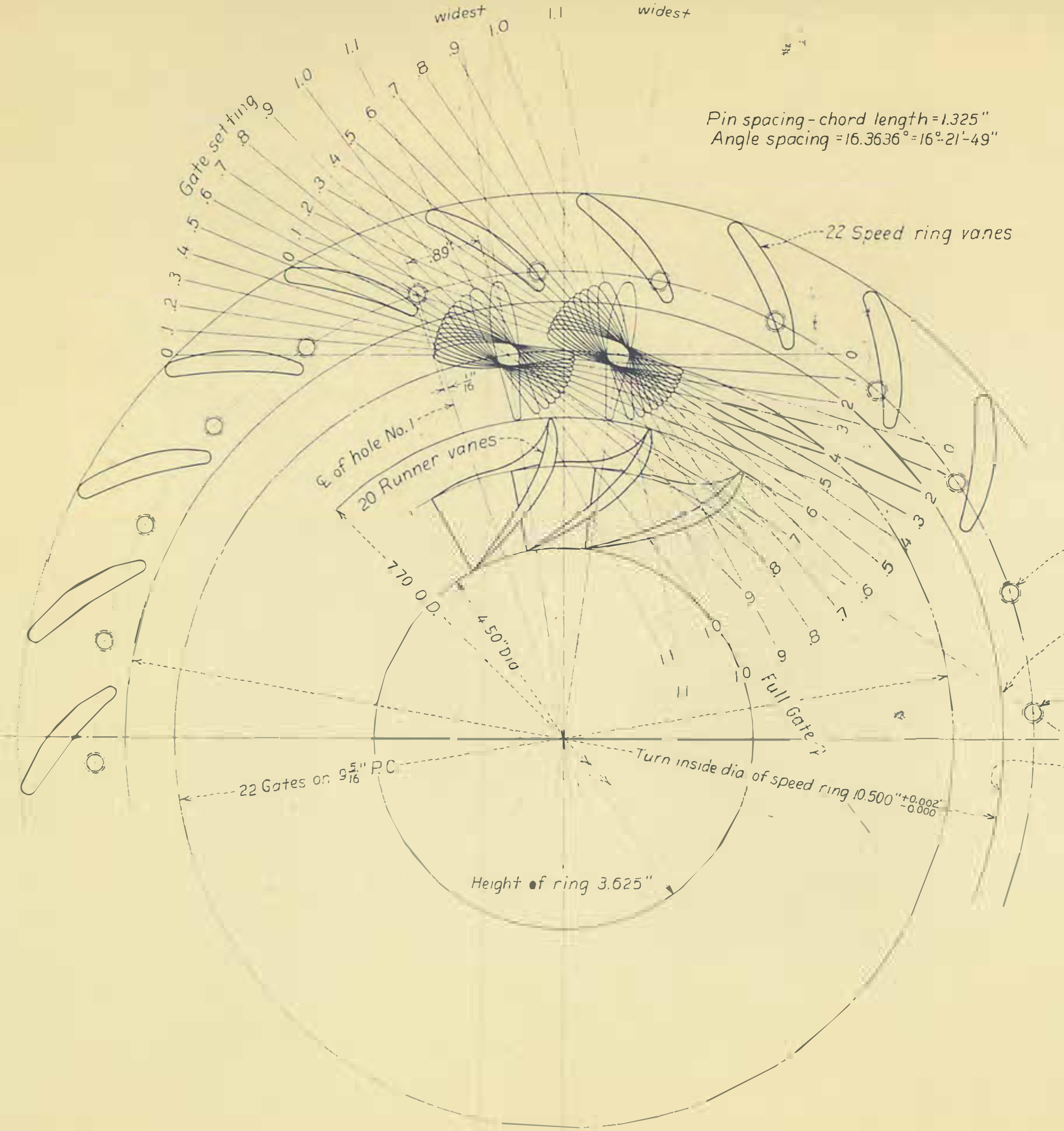
The proposed design conditions for the Grand Coulee turbines at the time the model was designed and built were as follows: head 335 feet; discharge 4,400 second-feet; speed 128.57 r.p.m. Assuming an efficiency of 90 percent, the output of the turbine would be 150,000 horsepower. The specific speed for these conditions would be 34.4. The value of ϕ, or the ratio of the peripheral velocity of the runner at the inlet diameter to the spouting velocity of the water was set at 0.70 for maximum efficiency.

The selection of the model scale of 1 to 24 was dictated principally by the fact that a velocity of not less than four feet per second in the throat of the draft tube within the test range was necessary for satisfactory model draft-tube performance. Previous studies in connection with decelerated flow, such as in draft tubes and venturi tubes, have demonstrated that there is a sudden decrease in efficiency



SECTION THRU TURBINE
GRAND COULEE MODEL TURBINE

FIGURE 1



GATE SETTING	INCHES OPENING
0	0.000
.1	0.089
.2	0.178
.3	0.267
.4	0.356
.5	0.445
.6	0.534
.7	0.623
.8	0.713
.9	0.802
1.0	0.891
1.1	0.980
widest	1.045

GRAND COULEE MODEL TURBINE

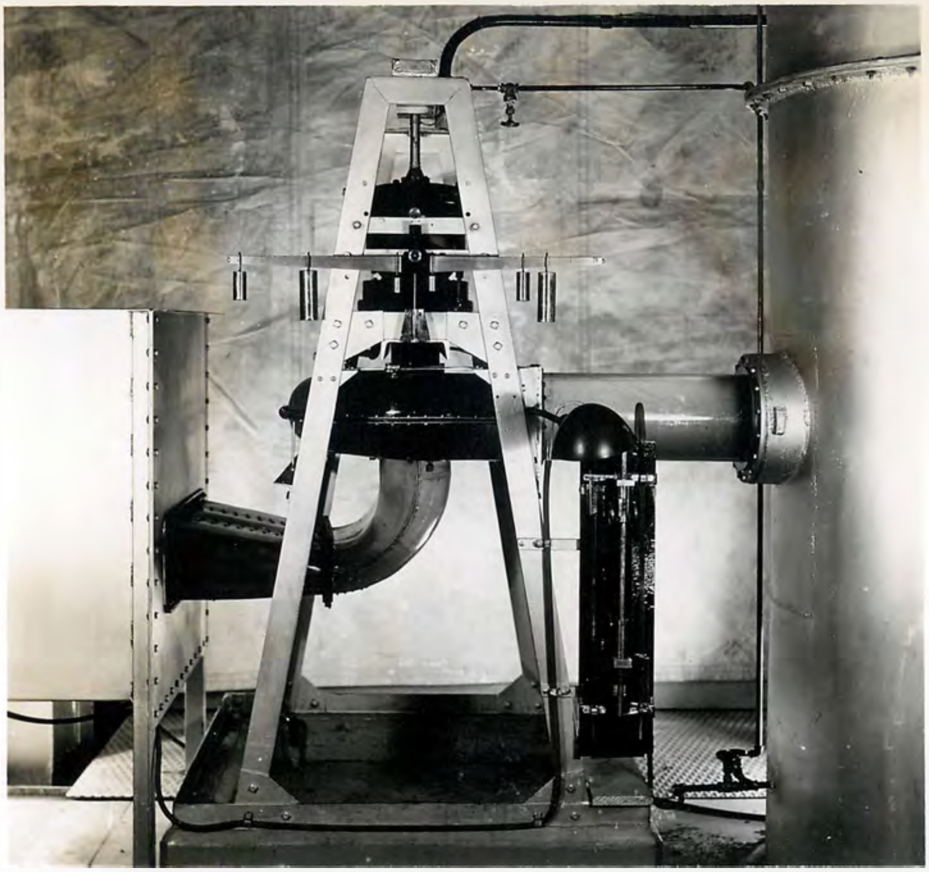
PLAN SECTION THROUGH TURBINE

AT CENTRAL PLANE OF ROTATION

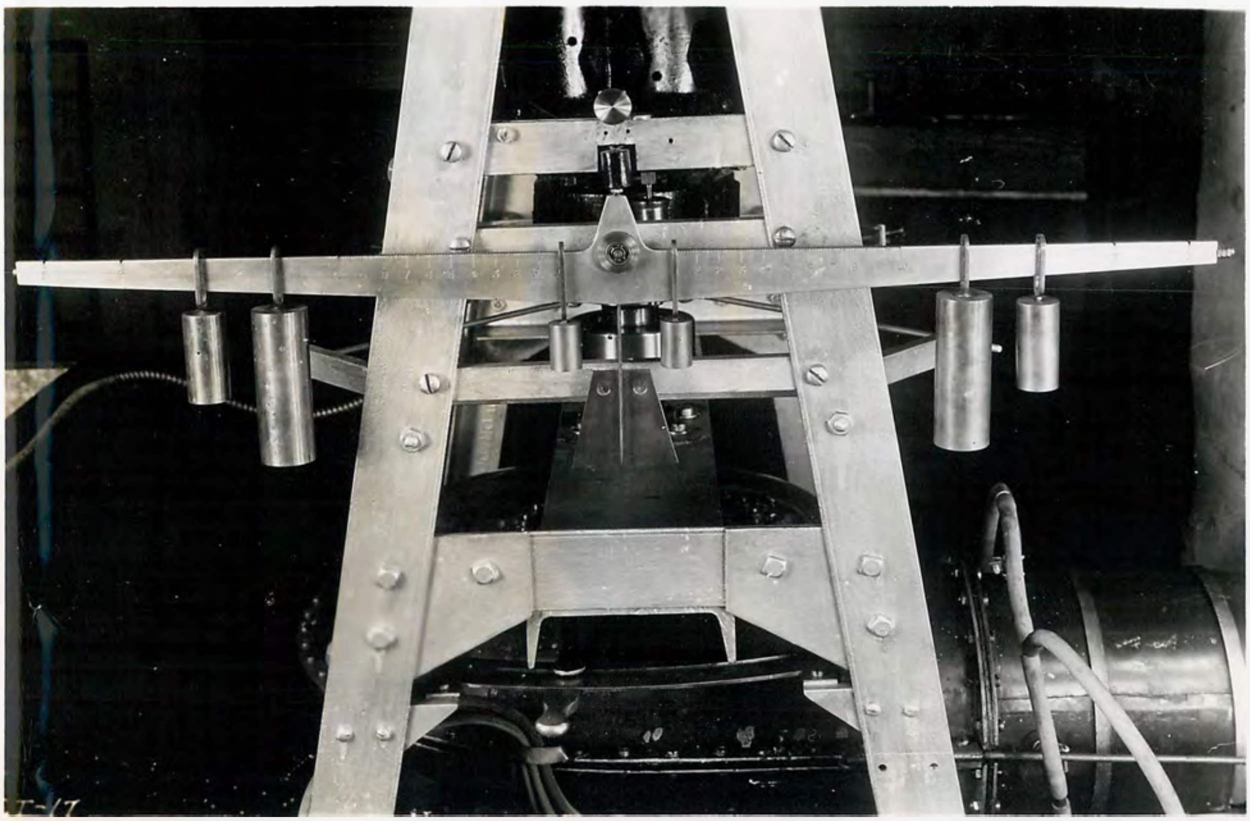
FIGURE 2

of velocity-head recovery when the velocity at the throat is reduced to a critical value of approximately four feet per second. By making the model to a scale of 1 to 24, the critical velocity would occur at a model gate opening of approximately 0.2, which represents the lower limit of the usual operating range. At the gate opening for best efficiency the velocity in the model draft-tube throat was 6.77 feet per second. A larger model would have resulted in somewhat greater accuracy but the additional cost was not considered justified.

The design conditions reduced to model dimensions were: head 13.96 feet; discharge 1.56 second-feet; speed 630 r.p.m. The specific speed and ϕ, which are both dimensionless, had the same values in the model as in the prototype, or 34.4 and 0.70, respectively. A sectional drawing of the model turbine giving details and principal dimensions is shown in figure 1. A photograph of the complete operating model is shown in figure 3A. The arrangement for controlling the gates and for measuring the torque is shown in figure 5, and in the photograph, figure 3B. The gates were positioned by means of a gate-operating lever attached to the top of the gate-shift ring. The gate-vane levers and the break links formed linkages between each wicket-gate vane and the shift ring. The gate position was indicated by a graduated quadrant along which the gate-control lever moved. The quadrant was graduated in tenths of full gate so that the gates could be accurately adjusted to any desired opening. The lever could be securely clamped to the quadrant to prevent creeping of the gates during a test. Full gate opening was arbitrarily taken as the position where the vertical plane of symmetry of the gate vanes was tangent to the circular trace of the discharge edge of a runner vane in the central plane of the turbine runner as shown in figure 2. With the gate-operating lever set at the full open position the clear openings between the gate vanes were carefully measured. A series of gage blocks of thickness varying by increments of one-tenth from zero to ten-tenths of this measurement was made for use in graduating the quadrant. The gate vanes were closed down



A - Grand Coulee Model Turbine

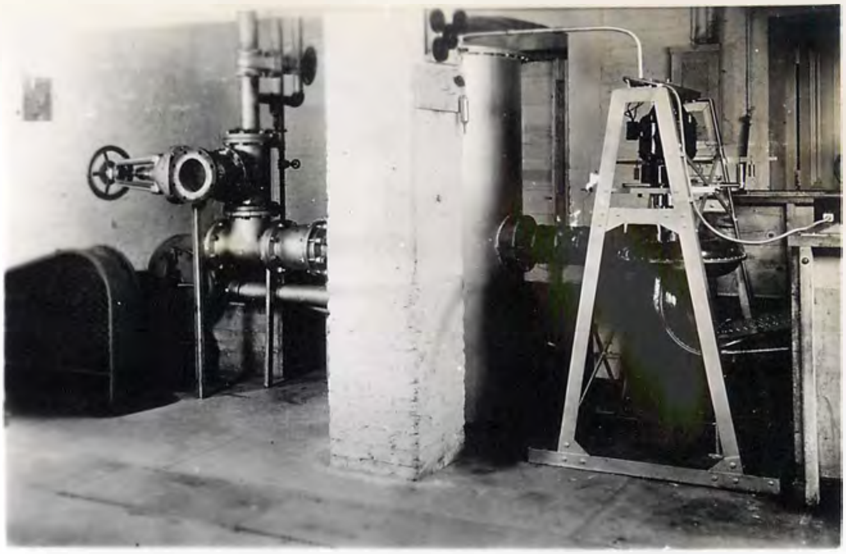


B - Close-up of Torque Beam and Gate Lever on each gage block successively and the corresponding positions of the shift lever were marked on the quadrant. These graduations, although not necessarily the same as those on the prototype turbine gates, were used as a basis for coordinating the model performance data such as power, discharge, and efficiency.

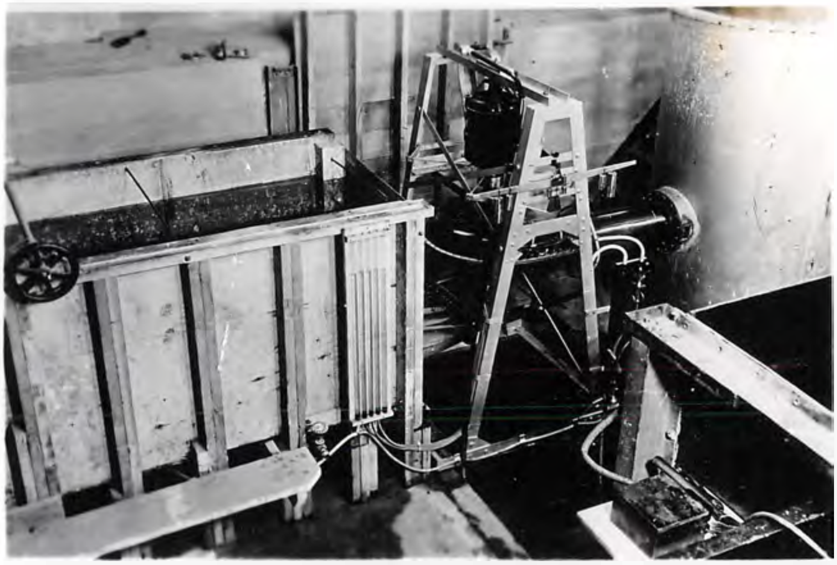
The turbine assembly was mounted on a steel "A" frame, as shown in figure 3.

TESTING ARRANGEMENT

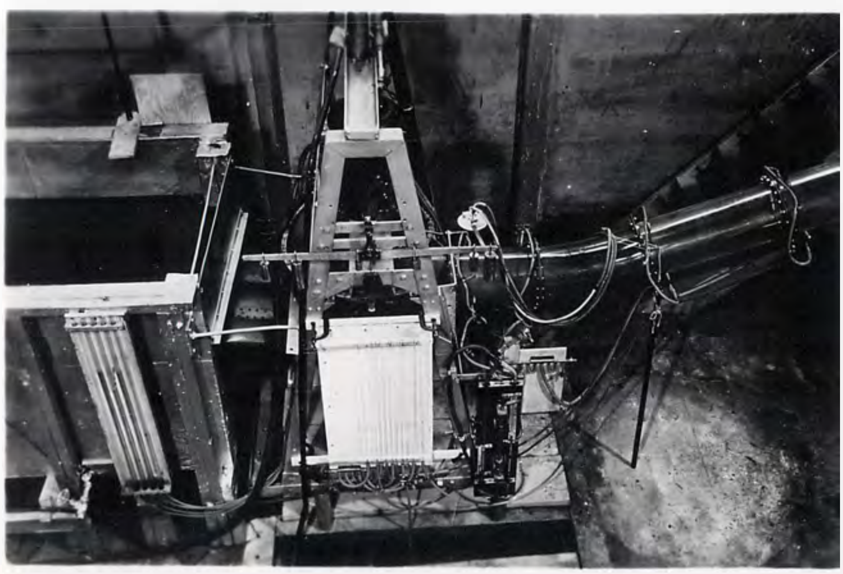
The arrangement by which water was circulated through the model under the desired head consisted of an 8-inch certrifugal pump, head tank, tailwater box, and weir flume as shown in figure 4. The pump delivered water from a sump to the head tank through an 8-inch-diameter pipe and a gate valve which served as a means for adjusting the model head and discharge. Two different designs of head tanks were used during the course of the testing program. In the first design the available headroom dictated the use of a short steel tank which was operated under pressure to build up the required model head. The tank was fitted with baffles to direct and still the flow and was operated with air under pressure above the water surface to act as a cushion. The difficulties encountered in maintaining a constant head under this air cushion led to a redesign of the tank when the location of the laboratory was changed to another building where the headroom was greater. Two additional sections were added to the tank to make it high enough for operation with a free water surface. The head tank and scroll case were connected by a short length of transparent pipe, 8.25 inches in diameter which matched the diameter at the model scroll-case entrance. The inlet end of the transparent pipe was equipped with a bellmouthed-entrance piece designed to give smooth flow and a uniform velocity distribution. This short length of pipe was later replaced by a complete scale model of the Grand Coulee penstock including the bend adjacent to the scroll-case entrance, see arrangement No. 3, figure 4. Later studies showed



Arrangement No. 1



Arrangement No. 2



Arrangement No. 3

that the penstock bend had no measurable effect on the turbine efficiency so the substitution of a short conduit for the actual penstock was admissible for studies of turbine elements located downstream from the scroll-case entrance such as the draft tube, speed ring, and scroll case. The draft tube was connected to the tailwater box by means of a bolted flange so that it could be readily removed. The tailwater box was made approximately two feet wider than the draft-tube exit to minimize its effect upon discharge conditions of the draft tube. An adjustable gate at the end of the box served as a means for regulating the level of the tailwater surface. From the tailwater box, the water flowed into a weir flume, two feet in width, and then over a rectangular, sharp-edged weir, two feet in length, and back to the sump for recirculation.

TESTING APPARATUS

The efficiency of the model turbine, which was to serve as the index of performance of the various turbine elements, is defined as the ratio of brake horsepower output to water horsepower input.

The determination of water horsepower input involved measurement of both head and discharge. The discharge was determined by measurement of head over a rectangular, fully suppressed, sharp-crested weir two feet in length. The weir box which was fabricated from plate steel was constructed to the same dimensions and with the same system of baffling as a volumetrically calibrated weir used previously except that the width was increased from one foot to two feet. The head over the weir was determined by means of a hook gage set in a well connected to piezometers located on both sides of the weir box at a distance upstream from the weir equal to approximately four times the depth of water over the weir at normal model discharge. The head over the weir was taken as the average of five readings of the hook gage made at intervals of 30 seconds. The discharge was computed by the Francis formula,

$$Q = 3.33\, Lh_w^{3/2} \tag{1}$$

in which Q = discharge in cubic feet per second

h_w = head over the weir in feet, and

L = length of the weir in feet.

The determinations of discharge made in this manner were quite consistant and, it is believed, very nearly accurate.

The net or effective head on the turbine was determined by means of a mercury differential gage of the U-tube type. One leg of the U-tube was connected to the tailwater box at a suitable distance away from the draft-tube outlet and the other was connected to a piezometer ring consisting of four connections equally spaced around the penstock near the entrance to the scroll case. The mercury-column differential in the U-tube indicated the net head on the turbine minus the velocity head in the penstock at the entrance to the scroll case. The net head, H, on the turbine was determined as follows:

$$H = 12.6\, h_z + h_v \tag{2}$$

in which h_z = mercury column differential in feet, and

h_v = velocity head in the penstock.

The water horsepower input to the turbine was then computed from these measurements by the following relations:

$$WHP = \frac{62.4\, QH}{550} \tag{3}$$

in which Q = discharge in c.f.s. as measured by the weir,

H = net head on the turbine, and

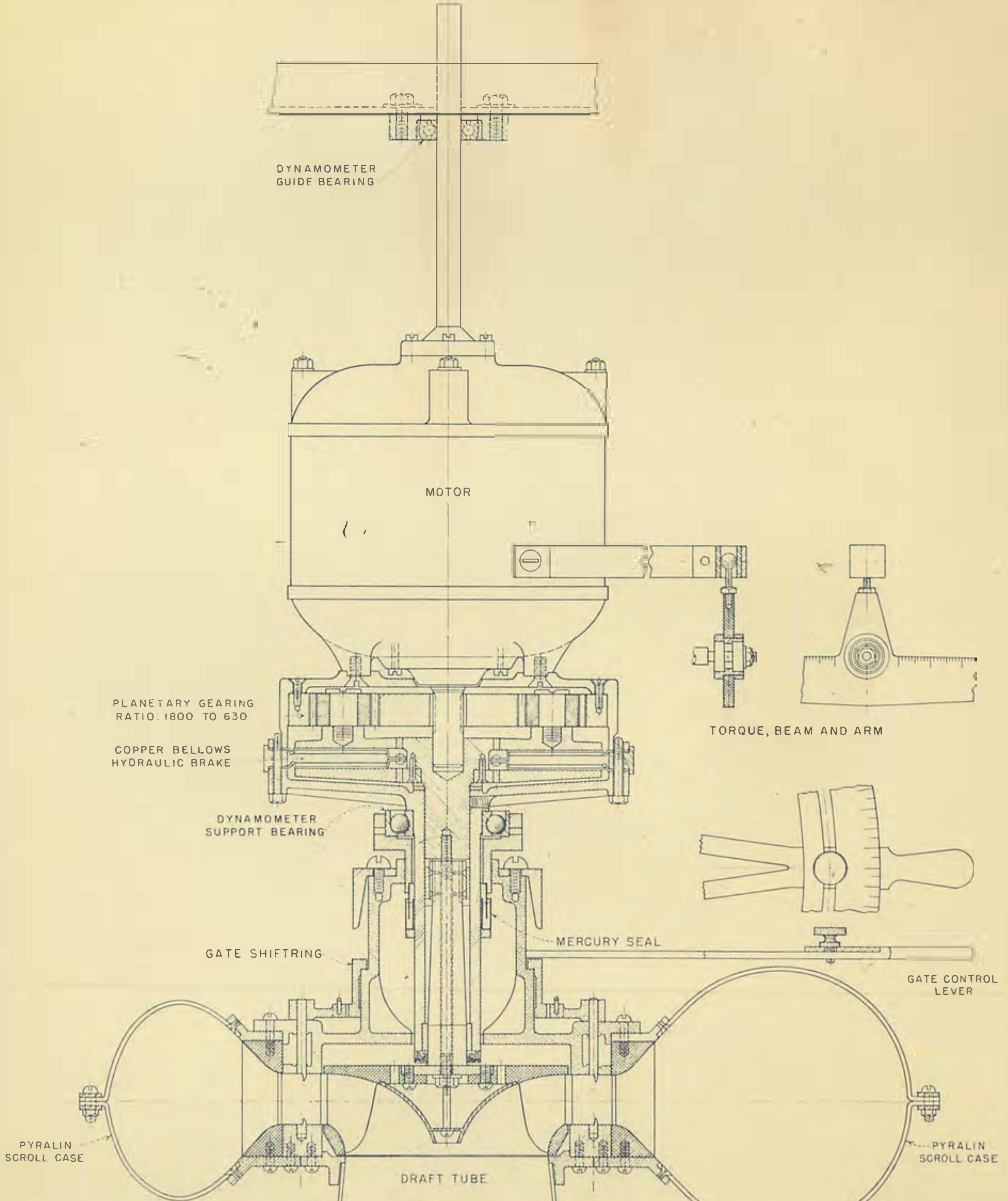
62.4 = weight of one cubic foot of water.

The determination of brake horsepower involved measurement of speed and torque. The arrangement for measuring the torque and controlling the speed was somewhat involved. A variable-speed electric dynamometer would have been ideal for this purpose but it was not available. Instead, a 3-horsepower, 220-volt, 3-phase, 60-cycle, 1,760 r.p.m. induction motor was used. By cutting four slots in the rotor, this motor was made to synchronize at 1,800 r.p.m. for loads up to about one horsepower which was not quite enough to hold the turbine down to synchronous speed at the larger gate openings. The turbine would drive the motor beyond 1,800 r.p.m. by an amount which could be determined by the number of pole slips per minute, each pole slip being equal to one-quarter revolution of the motor. The pole slips were indicated by an ammeter connected in the motor leads.

In the first testing arrangement the motor was connected to the turbine shaft through a set of speed reduction gears such that the ratio was normally 1,800 to 630 but could be changed to various other ratios by changing one of the gears. Seven gears were provided such that the speed of the turbine runner could be changed in seven steps from 560 to 690 r.p.m. for establishing the best speed, or value of ϕ, for best turbine performance at designed head. This could have been established by changing the head on the turbine except that it was questionable whether the plastic model scroll case would stand the pressure. After the proper model speed of 630 r.p.m. ($\phi = 0.70$) had been established, the original reduction-gear system was replaced by one of constant speed ratio into which an absorption braking system was built to relieve the load on the motor and eliminate the pole slippage previously mentioned. The arrangement of the braking system is shown in figure 5.

In measuring the torque developed by the turbine runner it was desirable to eliminate all possible running friction which might introduce error in the data. This was accomplished by including in

FIGURE 5



SECTION THRU TURBINE MODEL AND ABSORPTION DYNAMOMETER

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
COLUMBIA BASIN PROJECT-WASHINGTON

GRAND COULEE DAM
HYDRAULIC MODEL STUDIES
TURBINE ASSEMBLY

TRACED
CHECKED
RECOMMENDED
APPROVED
DENVER, COLORADO - FEB 19, 1940
222-D-6099

the power-absorption unit, or dynamometer, all guide and thrust bearings and all running friction. The dynamometer, consisting of motor, gear box, and the extension tube which supported the runner bearing, was supported on a large radial and axial thrust ball bearing set in the upper part of the turbine support casting and was held in a vertical position by a small radial ball bearing mounted in the top cross member of the model supporting frame figure 5. This arrangement permitted the dynamometer to float on its bearings uninfluenced by any major external force other than the torque exerted by the turbine runner. The slight movement of the dynamometer was resisted only by the friction in the two ball bearings which was negligible. A mercury seal was provided between the turbine-support casting and the turbine-shaft housing.

The torque, all of which was manifested in the dynamometer unit, was transmitted to the scale beam through a torque arm attached to the motor housing as shown in figure 5. The aluminum scale beam, which was supported on a small, double ball bearing, had a lever extending upward and terminating in a polished steel ball. This ball worked in a hardened-steel bushing fixed in the end of the torque arm which was attached to the motor frame. The torque was evaluated by balancing the scale beam with movable weights. Every precaution was taken to make the weighing beam as accurate and sensitive as possible. The graduations on the beam were marked off by means of micrometer measurements and the weights were carefully checked against appropriate standards. Friction in the fulcrum ball bearing supporting the beam was so small that it was neglected. The least count of the beam was 0.01 foot-pound, or approximately 0.10 percent of the torque developed at normal turbine output. A photograph of scale beam is shown in figure 3B.

With the speed held constant and the torque determined by measurement, the brake horsepower was computed by the following formula:

$$\text{Brake horsepower (B.H.P.)} = \frac{2\ TN}{33{,}000} \qquad (4)$$

where T = torque in foot-pounds and

N = turbine speed in r.p.m.

The efficiency of the turbine was then obtained by taking the ratio of B.H.P. to W.H.P. from equations (3) and (4).

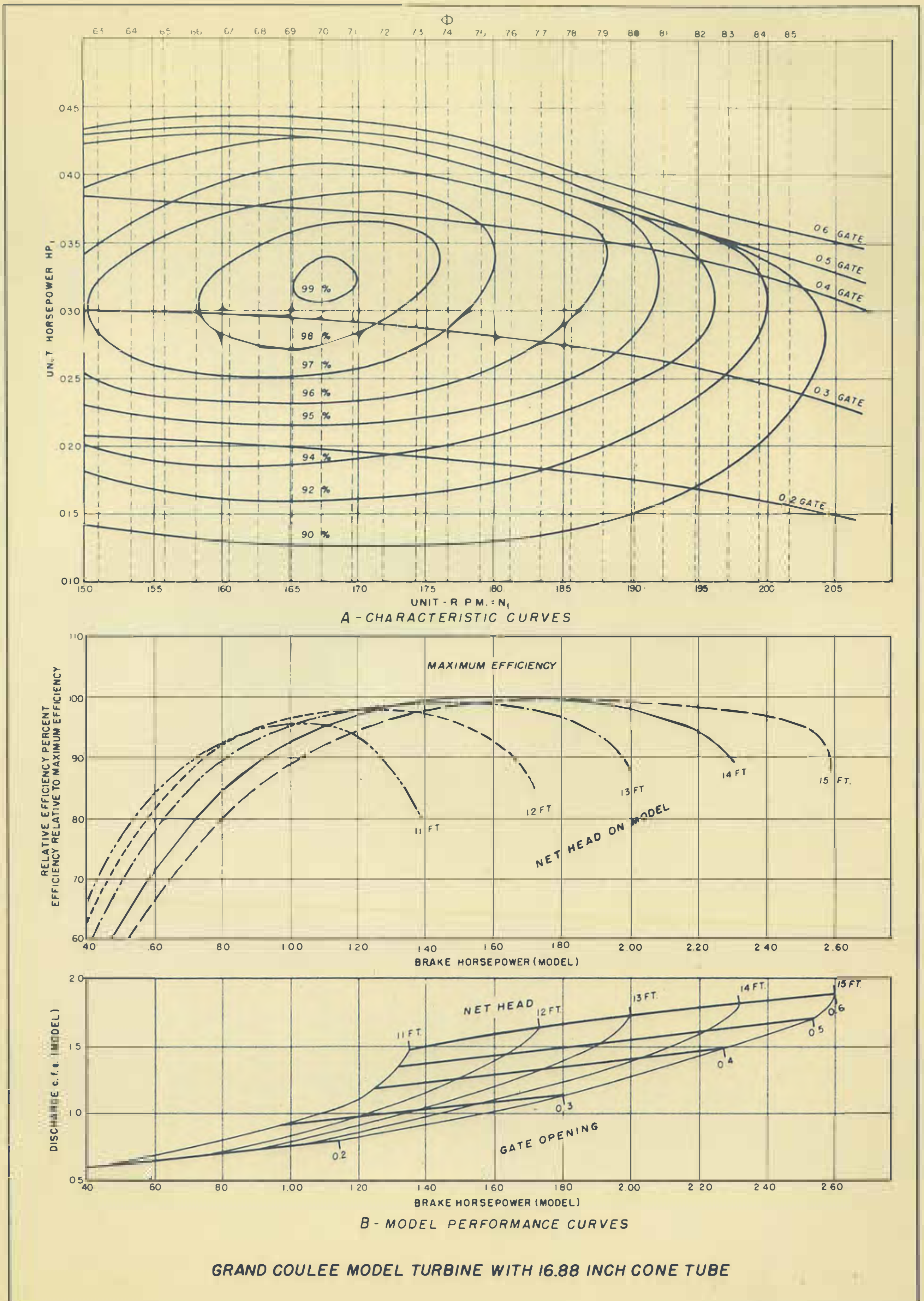
To facilitate comparisons of performance for the various designs of each element the data were put in graphical form by plotting efficiency against brake horsepower. For these plots it was desirable that all values be referred to a common head which for convenience was taken as the scaled head of 13.96 feet. Since it was difficult to hold the model head exactly at this value, much of the data was taken with model heads which sometime varied slightly from the desired value. In order to adjust the data to the values which would have prevailed if the model head had been exactly 13.96, the following procedure was used.

It is evident in the curves of figure 6A that the efficiency is affected only very slightly by changes in unit speed such as would occur with slight changes in head for a constant speed. The efficiency may therefore be assumed as constant for small changes in head and the measured value used as if it had been measured at the required head. The horsepower curves in figure 6B show that, on the other hand, it cannot be assumed constant but must be adjusted to the required head. Let the subscript (m) indicate the measured value and the subscript (a) indicate the value adjusted to a head of 13.96 feet, then the following equation may be written:

$$\frac{(B.H.P.)_a}{(B.H.P.)_m} = \frac{E_a\,(W.H.P.)_a}{E_m\,(W.H.P.)_m} = \frac{E_a\,Q_a\,H_a}{E_m\,Q_m\,H_m} \qquad (5)$$

It was assumed that at a given gate opening the turbine acts as an orifice for which the relationship between discharge and head may be expressed as $Q = k\sqrt{H}$. Inserting this relationship and the fact that efficiency was constant into equation 5 gives the following expression:

$$(B.H.P.)_a = (B.H.P.)_m \left(\frac{13.96}{H_m}\right)^{3/2} \qquad (6)$$



GRAND COULEE MODEL TURBINE WITH 16.88 INCH CONE TUBE

ADJUSTMENT TESTS

The model turbine was designed as carefully as possible for a ϕ of 0.70 but to ascertain that best efficiency did occur at this value of ϕ it was necessary to make pilot tests with various turbine speeds. This procedure is customary practice in turbine design where a small-scale model is built and tested to check the selection of best speed for a given head. These tests were made with the first reduction-gear design which was arranged so that change gears could be substituted to give various turbine speeds. The seven change gears used gave a range of speeds from approximately 560 to 690 r.p.m. as compared to the designed best speed of 630 r.p.m. for normal head.

The characteristics of the turbine were determined by running tests with 5 different heads between 11 and 15 feet for each of the seven change gears. The data were plotted as characteristic and performance curves shown in figure 6A and B. The nondimensional units used in these plots other than those previously defined are as follows:

$$\text{Unit horsepower } (HP_1) = \frac{HP(\text{measured})}{H^{3/2}}$$

$$\text{Unit speed } (N_1) = \frac{N(\text{measured})}{H^{1/2}}$$

Relative efficiency = efficiency (measured) in percent of maximum value.

Both plots indicate that best gate, or the gate opening at which maximum efficiency was obtained, occurred at a value of 0.35. The curves also show that the efficiency was nearly constant near its maximum at a value of N_1 close to 166. For $N_1 = 166$ the corresponding value of ϕ is 0.699 which is very close to the design value of 0.700 which, based on the design head of 13.96 feet, corresponds to a model speed of 630 r.p.m. The second reduction-gear design in which the turbine operated at a fixed speed was, accordingly, designed to hold the turbine runner at a speed of 630 r.p.m.

DRAFT-TUBE STUDIES

While the model of the complete turbine unit for Grand Coulee Power Plant was under construction a series of tests* on small wooden models of draft tubes was carried on in an open flume for the purpose of selecting one or more designs which would be efficient and at the same time be adaptable to the Grand Coulee turbine setting. These models, which had a throat diameter of three inches, were tested in connection with an orifice made in the form of a turbine with gate vanes and runner vanes removed. The required flow conditions at the throat of the draft-tube models were accomplished by means of sets of removable speed-ring vanes designed to produce whirls of different angles from 0° to 45°. Paint tests were made on these models to study the flow characteristics. These models were made in two pieces so that they could be taken apart for paint-test studies and photographing.

Three designs were selected for further study in the complete model on the basis of simplicity of structural design and effectiveness of the draft tube as a head regainer. Certain of the models showed a higher discharge coefficient than those selected but were rejected because of structural limitations.

The problem, as related to draft-tube studies, was not one of research and development, but rather one of comparing the performance of a draft tube of new and slightly different design with that of designs which have been tried and proven. Draft-tube development during recent years has shown considerable progress as exemplified in the "hydraucone" developed by W. M. White of the Allis-Chalmers Company, the Moody spreading draft tube developed by L. F. Moody of the Baldwin-Southwork Corporation, and the elbow type developed by I. A. Winter, G. J. Hornsby, and R. R. Randolph, Jr., of the Alabama Power Company for the Lower Tallassee turbines.

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*Hydraulic Laboratory Report No. 18 - "Progress Report on Hydraulic Model Experiments for the Design of Turbine Draft Tubes for the Grand Coulee Power Plant" by L. E. Winkelhaus and G. J. Hornsby.

The selection of the elbow type for Grand Coulee Power Plants was influenced by slightly better performance, adaptability to physical limitations, certain structural advantages, and cost of construction.

After a selection of type of draft tube had been made, the problem resolved itself into that of determining what effect slight variations in proportioning to meet specific conditions would have on its performance. The draft tube installed at Norris Power Plant and the one proposed for Grand Coulee Power Plants were both similar to the Lower Tallassee design. The principal differences were that the Norris tube was somewhat longer and the Grand Coulee tube was slightly shorter and considerably smaller in cross-sectional areas, especially at the termination of the elbow section. The Norris tube had given a good account of itself under field tests but there was some question as to how the Grand Coulee tube, with its reduced areas, would perform.

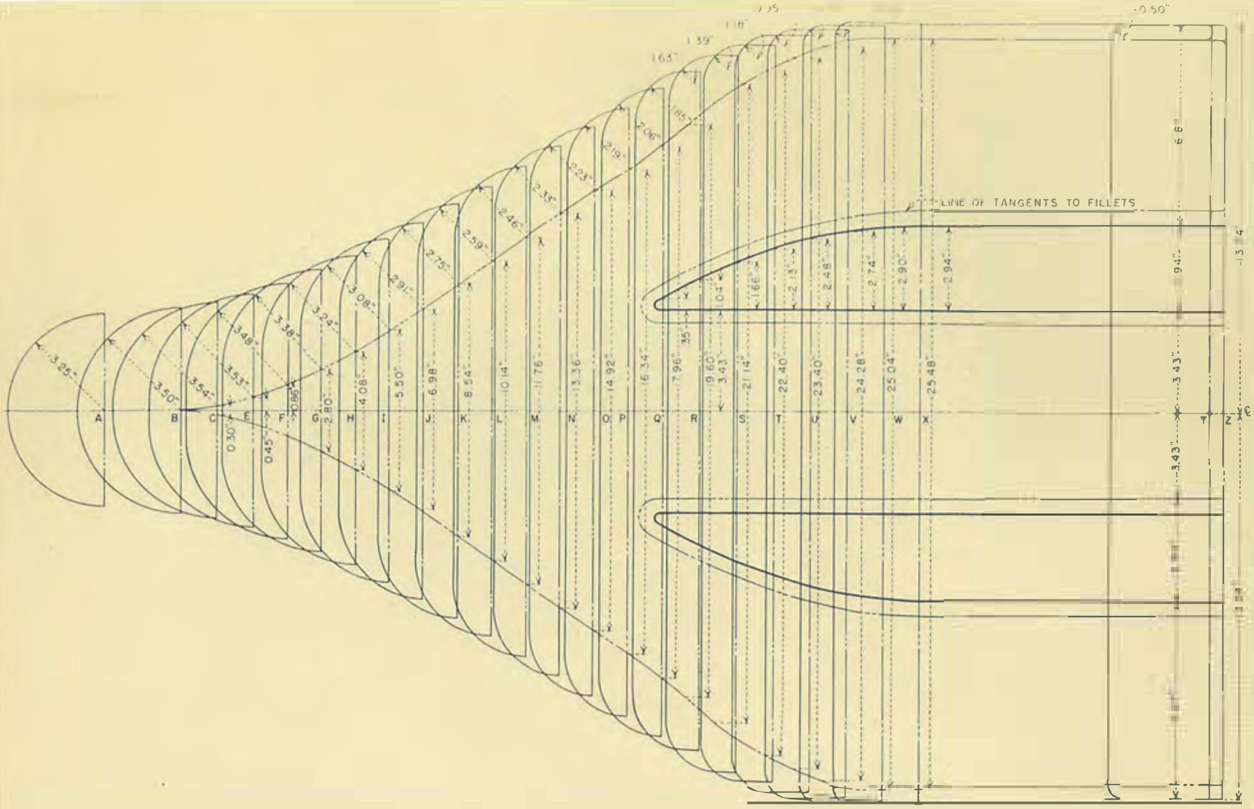
A draft tube is judged by its ability to convert the velocity head rejected by the turbine runner into useful head on the turbine with a consequent increase in efficiency of the turbine unit. The efficiency of the turbine unit was used therefore as a measure of the performance of the draft tube as well as the performance of all other features considered in these tests.

Since the function of the draft tube is to convert as much as possible of the velocity head rejected by the turbine runner into useful head on the turbine, the ideal draft tube would be one which would convert all of the velocity head into useful head. This is impossible because of certain inherent limiting conditions which may not be entirely overcome. These conditions are friction along the wetted surface of the water passages, viscosity of the fluid, eddy losses, and maldistribution of flow at the inlet and outlet ends of the draft-tube. Assuming no internal losses and uniform distribution of velocities at the inlet and outlet of the draft tube, the amount of velocity head converted in the draft tube will be the inlet velocity head minus the outlet velocity head. The straight conical draft tube is believed to meet most nearly the conditions of these assumptions. Friction can be minimized by making the surfaces as smooth as practicable, eddy losses can be reduced by streamlining the passages and controlling sectional areas for gradual changes in velocity. The design of the turbine runner controls the flow conditions at the inlet and the outflow conditions are controlled by the general shape of the draft tube. Viscosity is a characteristic property of the fluid and its relative importance in the draft-tube action is an inverse function of Reynolds number. Due to the latter relationship it may be expected that the prototype will be more efficient than the model in an amount which is a function of the model scale.

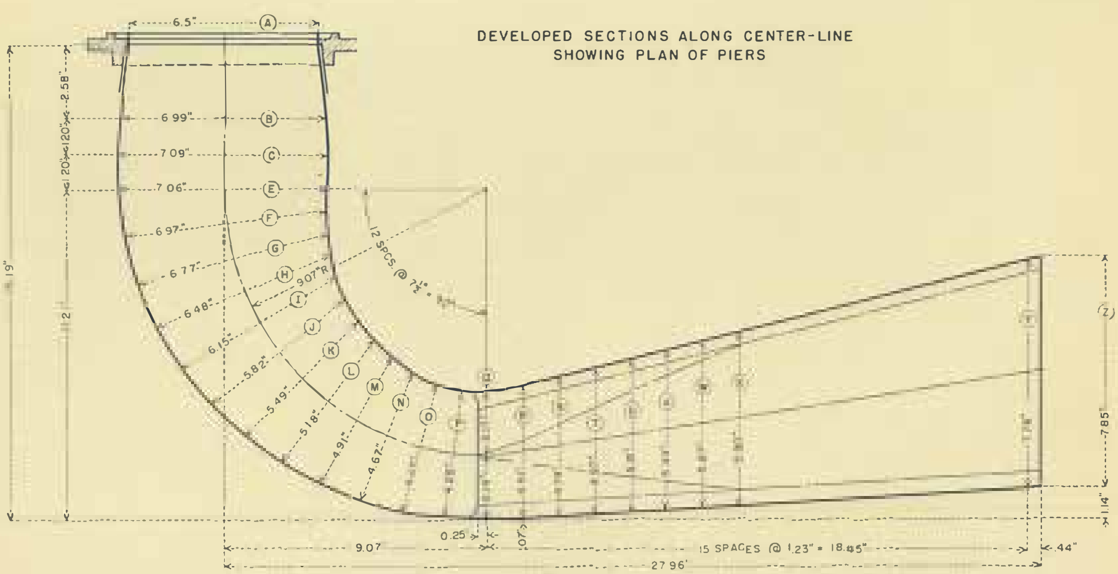
The length of the draft tube is usually a matter of economics. The rate of change of area against distances, or the rate of velocity deceleration, must be less than a certain experimentally determined maximum or eddies will form in the water passages and lessen the effectiveness of the draft tube. In practice, a discharge velocity of less than six feet per second, or more than eight feet per second, at rated turbine discharge, is seldom encountered.

A model of each of the three draft-tube designs selected for further study was constructed on a scale ratio of 1 to 24 to fit the model turbine. The models were fabricated from "Pyralin," a transparent plastic, to permit visual observation of flow at all points within the draft-tube models. A description of the materials and methods used in making these models is given in the appendix. Drawings of these models are shown in figures 7, 8, and 9 and photographs of the completed models are shown in figures 10, 11, and 12.

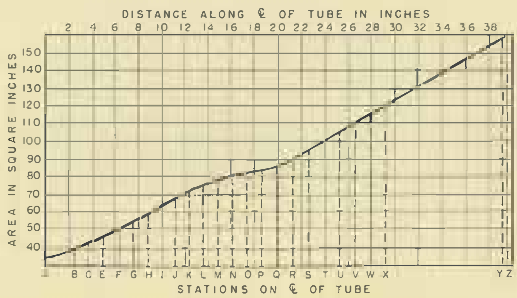
Several test runs were made on each of the three draft-tube models over the range of gate openings ordinarily used in power-plant operation. The tests were made with constant turbine speed and with the head held as nearly constant as practicable. The data were then adjusted to a common head corresponding to the design head on the turbine and used to compute the model efficiencies. Relative efficiency was plotted



DEVELOPED SECTIONS ALONG CENTER-LINE
SHOWING PLAN OF PIERS



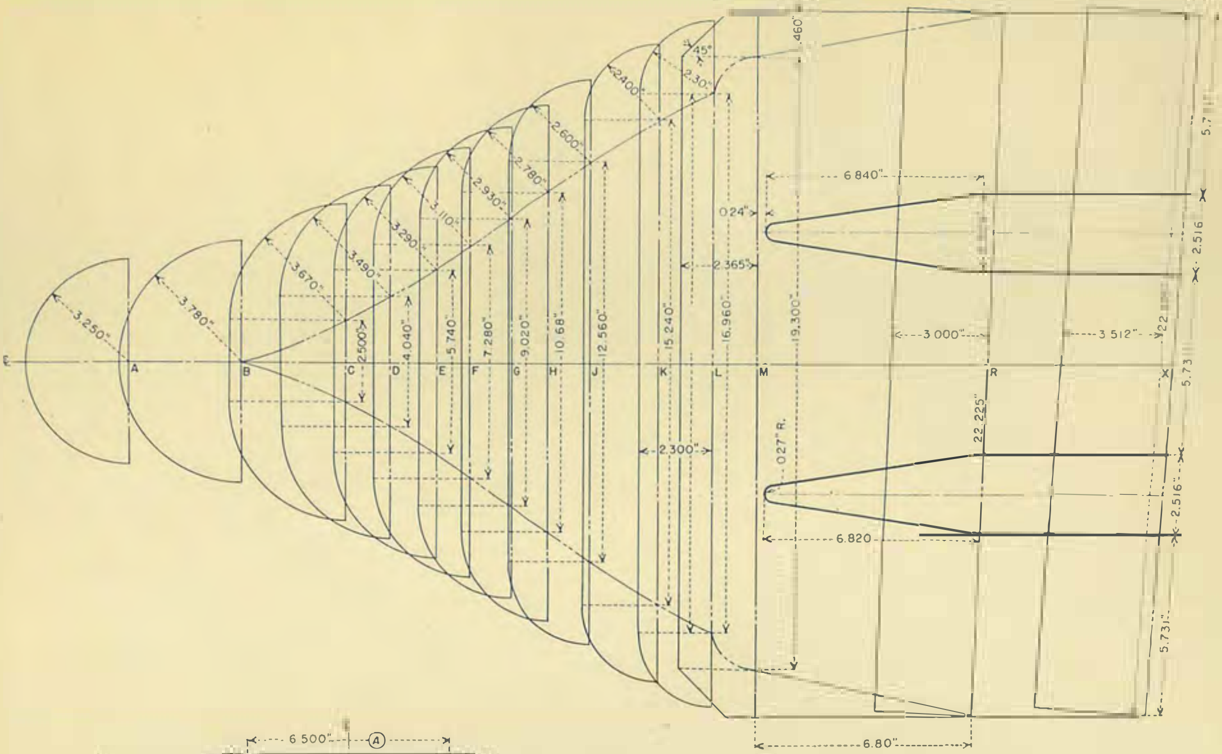
SECTION ON CENTER-LINE OF TUBE



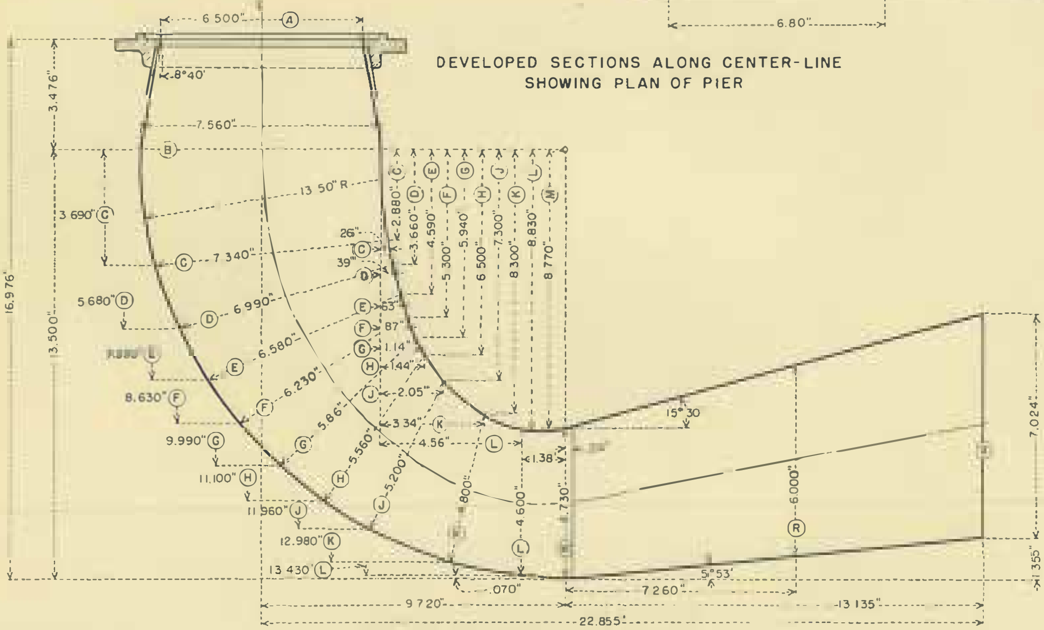
CURVE SHOWING AREAS OF SECTIONS
ALONG CENTER-LINE OF TUBE

GRAND COULEE DRAFT TUBE MODEL TESTS
NORRIS DRAFT TUBE

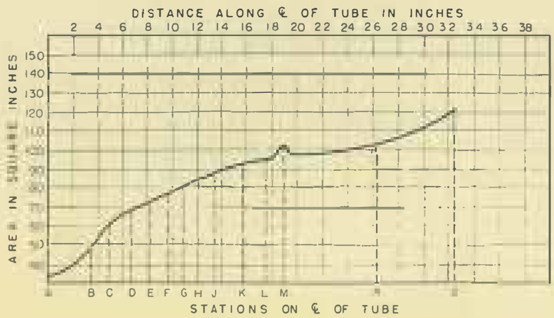
FIGURE 8



DEVELOPED SECTIONS ALONG CENTER-LINE
SHOWING PLAN OF PIER



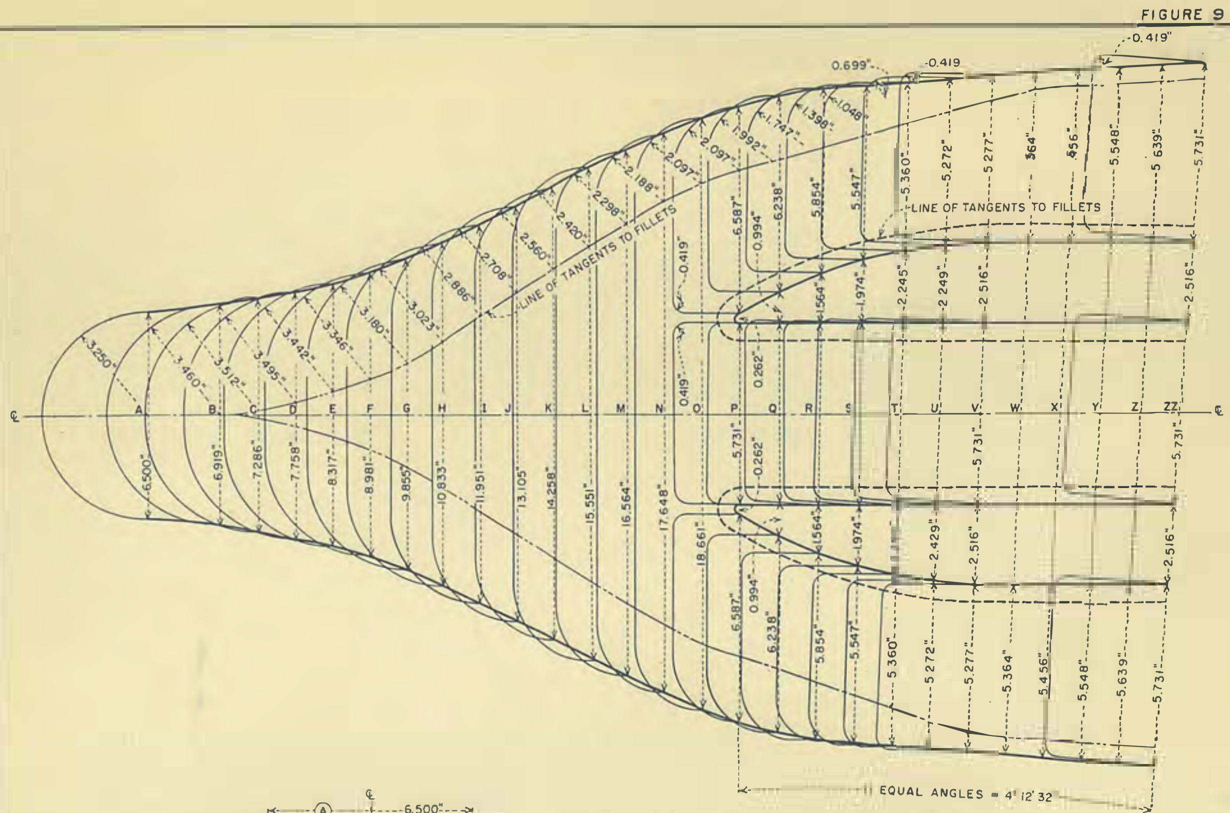
SECTION ON CENTER-LINE OF TUBE



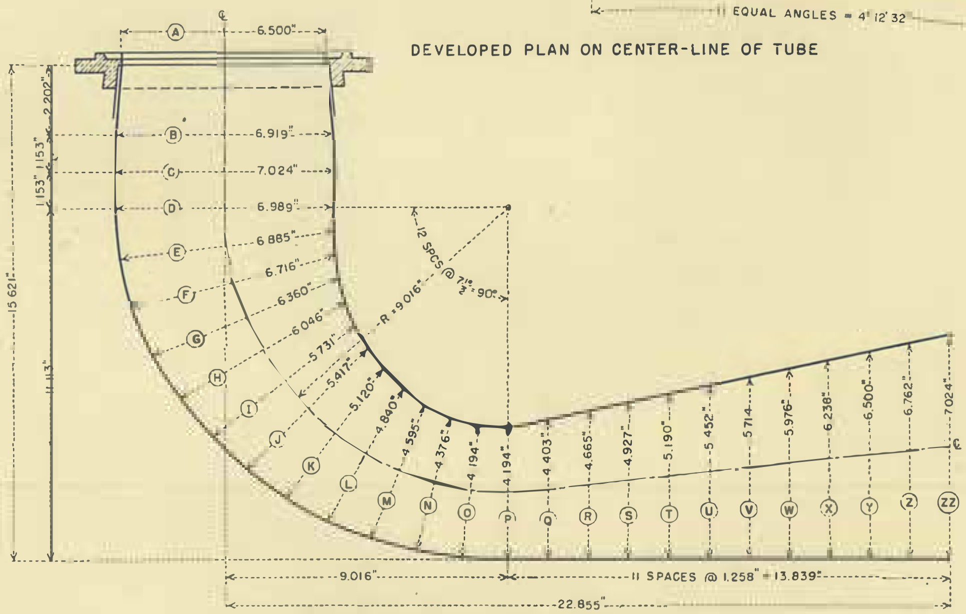
CURVE SHOWING AREAS OF SECTIONS
ALONG CENTER-LINE OF TUBE

GRAND COULEE DRAFT TUBE MODEL TESTS
LOWER TALLASSEE TUBE

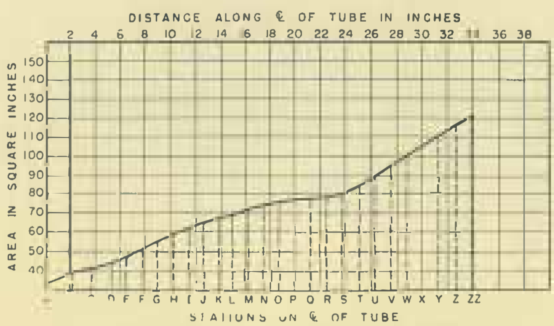
FIGURE 9



DEVELOPED PLAN ON CENTER-LINE OF TUBE



SECTION ON CENTER-LINE OF TUBE



CURVE SHOWING AREAS OF SECTIONS ALONG CENTER-LINE OF TUBE

GRAND COULEE DRAFT TUBE MODEL TESTS

GRAND COULEE TUBE

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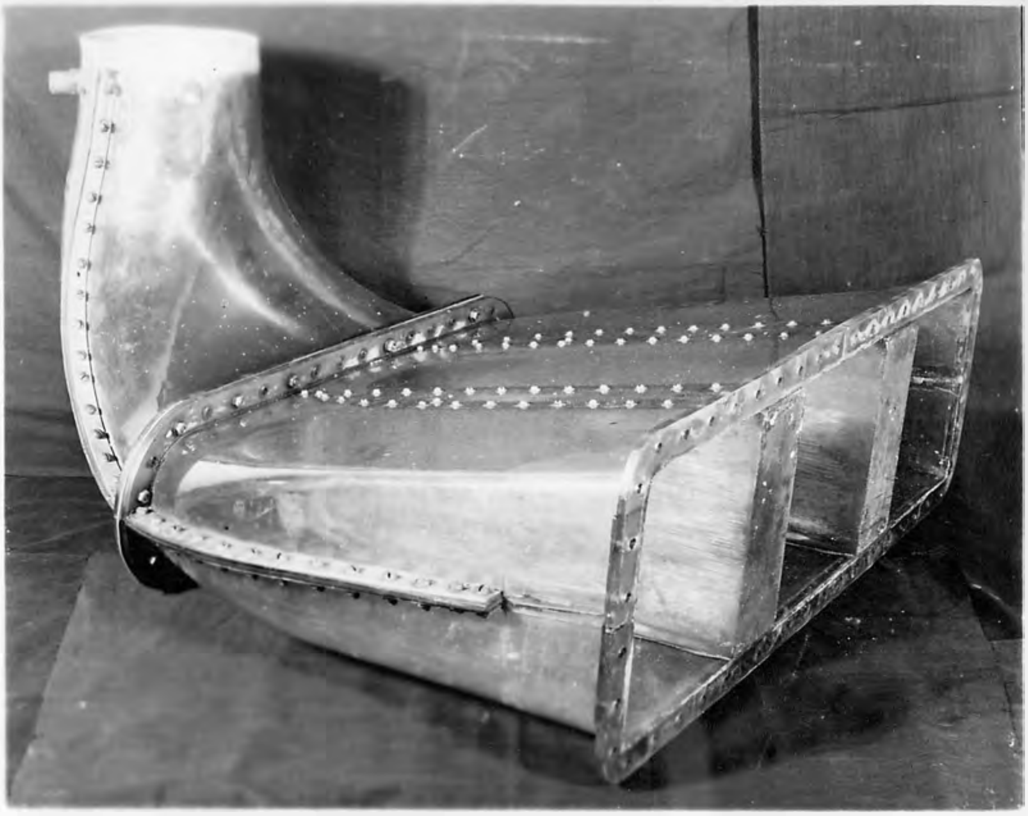
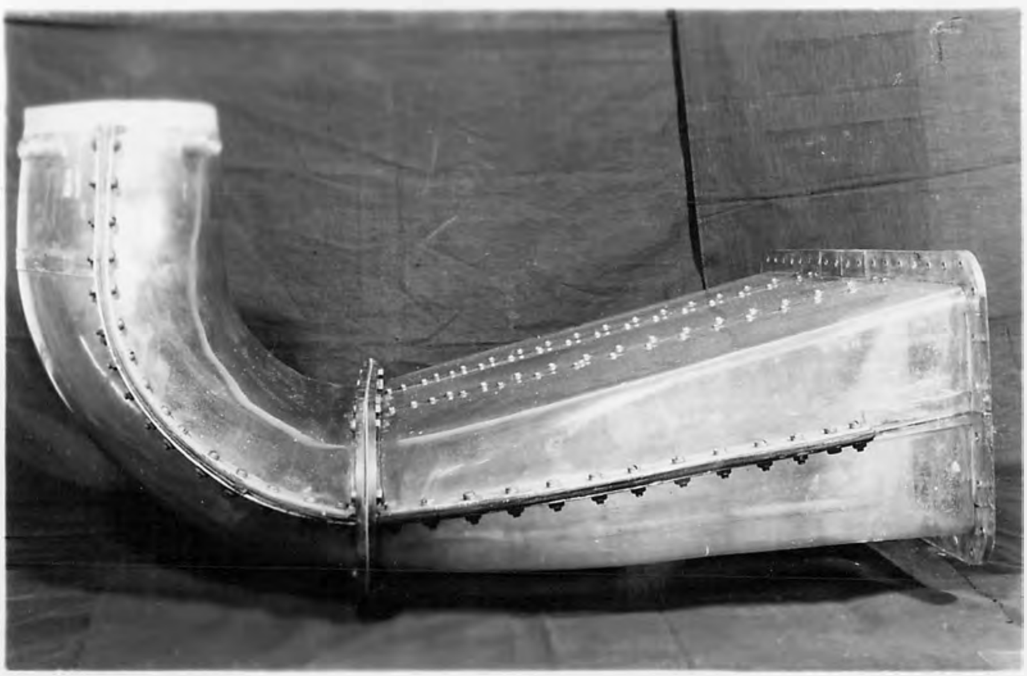
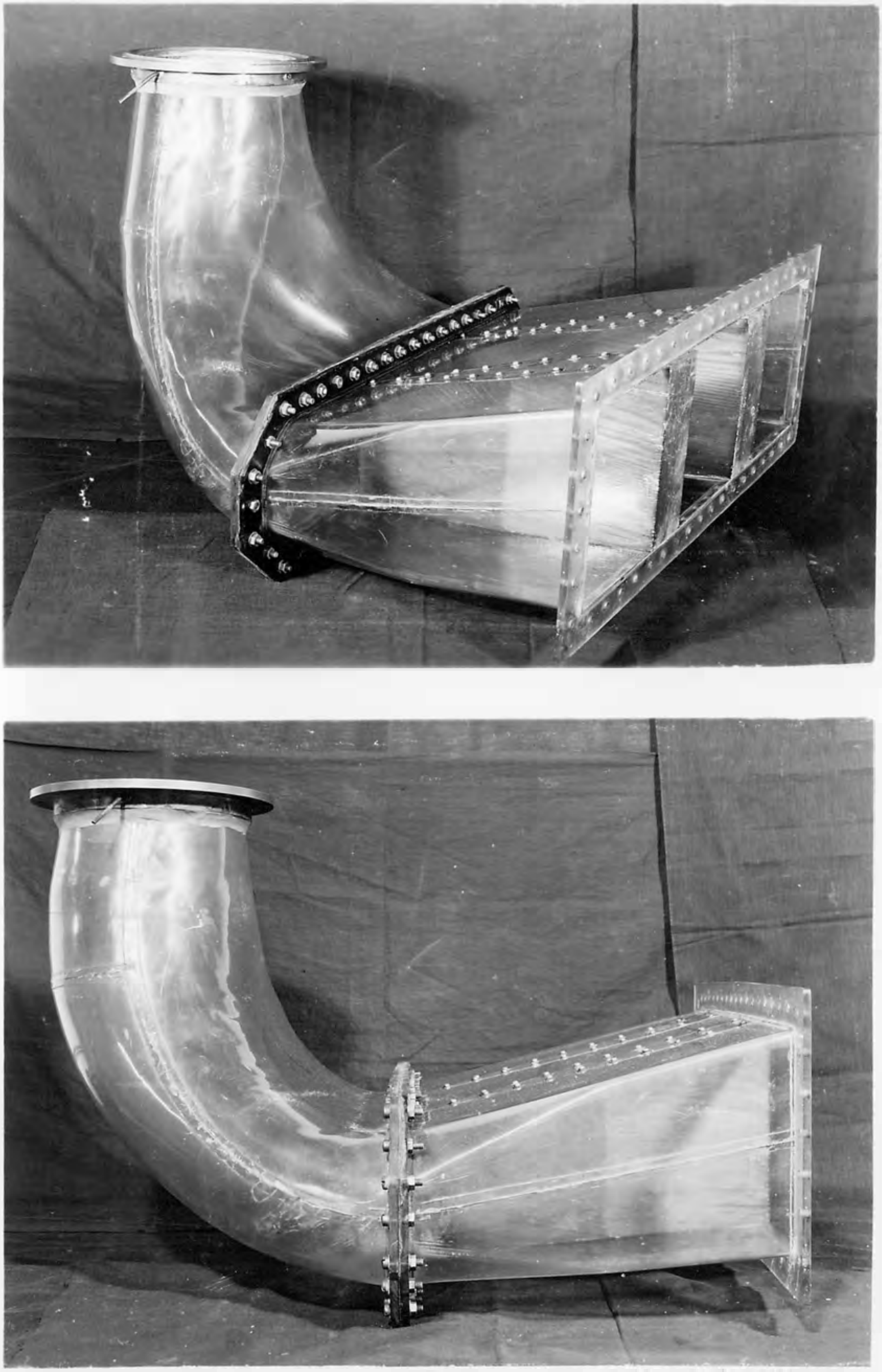
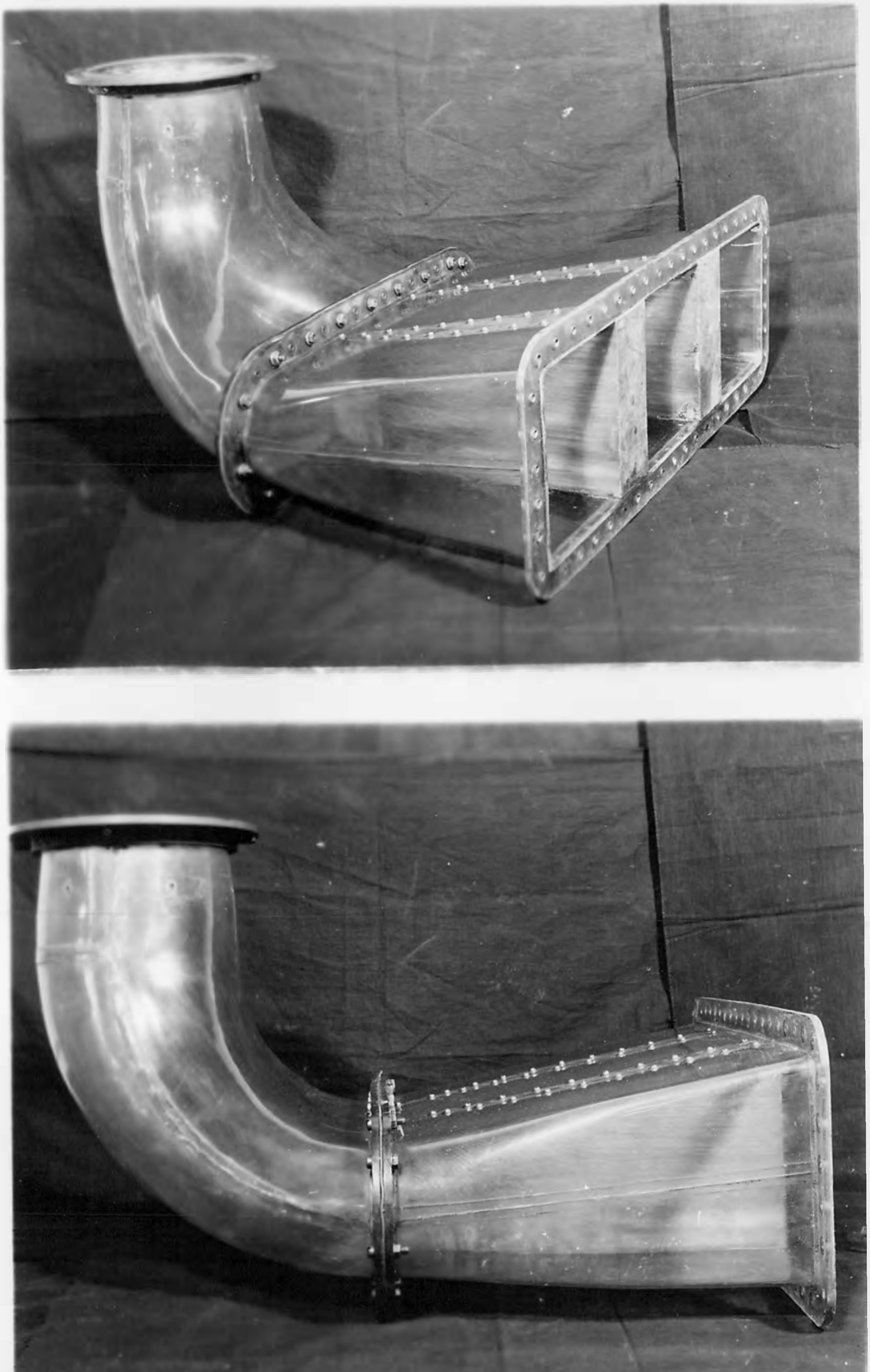




Model of Norris Draft Tube



Model of Lower Tallassee draft tubes



Model of Grand Coulee draft tube

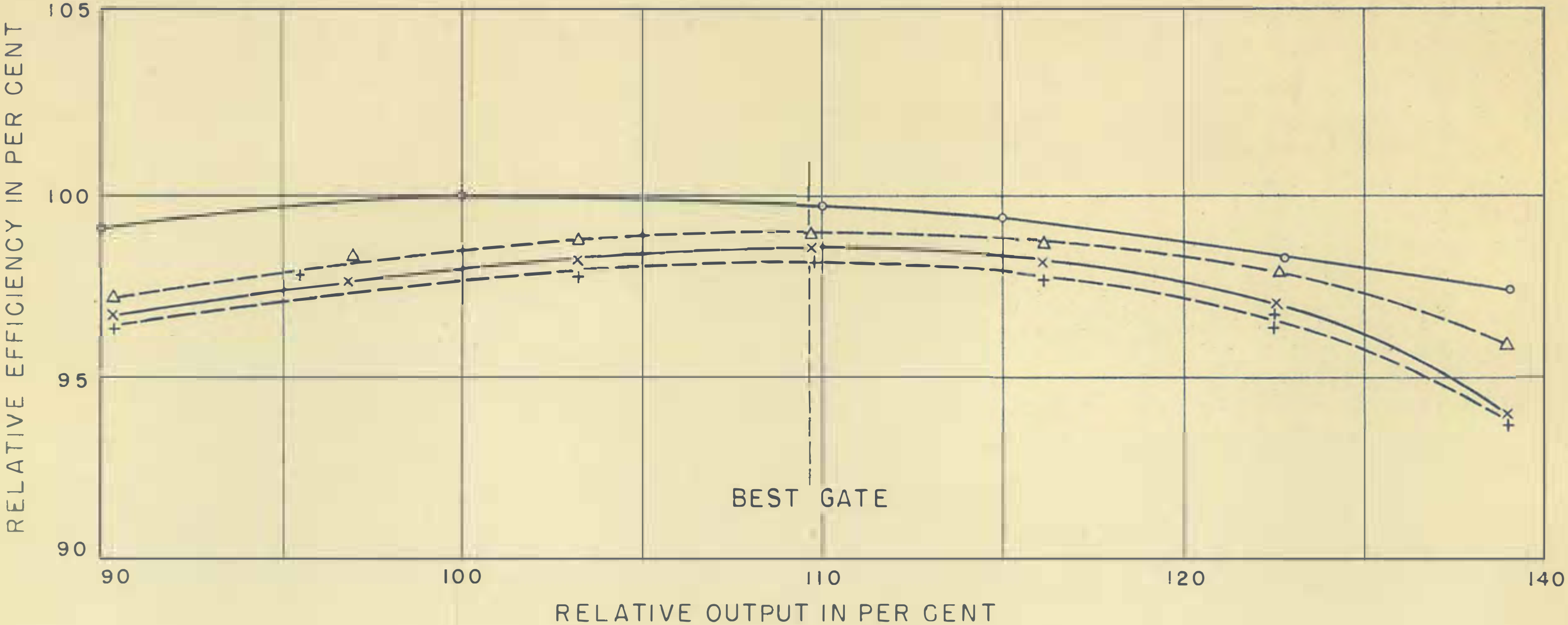
against relative output as shown in figure 13. The curves are of the same general shape and they show that peak efficiency occurred at approximately 0.35 gate or best gate for each of three models. The flatness of the curves indicates that the draft tubes all performed well over a wide range of gate openings. The test and check points were quite consistent, being on the order of 0.2 percent plus or minus.

Although the primary purpose of this series of tests was to select the best one of three elbow-type draft tubes, it was deemed expedient to set up some standard by which future investigations could be easily correlated with the present studies. For this purpose a conical draft tube designated as master, whose dimensions are shown in figure 14, was included in the tests.

The performance of each of the three elbow-type draft tubes was compared with that of the conical draft tube as shown in figure 13. The turbine efficiency with each draft tube is expressed in percent of the maximum efficiency with the conical draft tube. The turbine output with each draft tube is expressed in percent of the output at peak efficiency of the conical draft tube. By plotting the performance curves in nondimensional terms the data can be correlated more readily with data obtained in future investigations.

The characteristic curve of the three elbow draft-tube models are quite similar, all three curves indicating maximum efficiency at the same output. The model tests indicate that the Grand Coulee draft tube is slightly better than either of the other two elbow draft tubes. Under acceptance tests the Lower Tallassee turbines reached a maximum efficiency of 93.1 percent and the Norris turbines reached 93.0 percent. By comparison, the Grand Coulee turbines should give excellent performance so far as the draft tube is concerned.

Another criterion by which the performance of similar draft tubes may be judged is their effectiveness in distributing the flow uniformly over their exit sections. For draft tubes having comparable exit areas any parameter which expresses the uniformity of exit velocity distribution can be used to express the comparative recovery efficiency of the



EXPLANATION

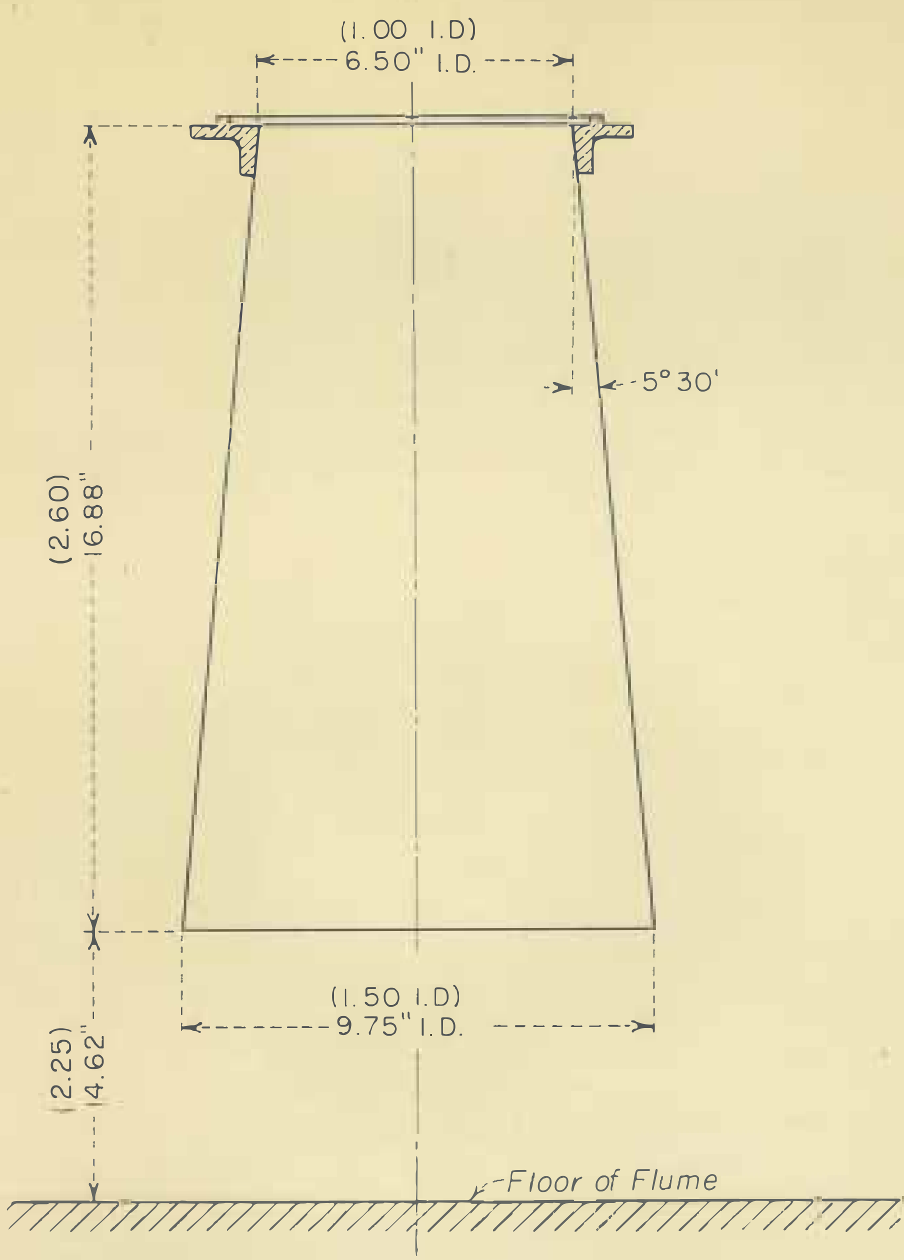
○ 16.88" Cone Tube　　△ Grand Coulee

+ Norris　　× Lower Tallassee

GRAND COULEE DRAFT TUBE STUDIES

PERFORMANCE OF TURBINE WITH CHANGE IN DRAFT TUBE RELATIVE TO EFFICIENCY AND OUTPUT AT PEAK EFFICIENCY OF 16.88 CONE TUBE

FIGURE 13



MASTER DRAFT TUBE DETAILS

NOTES

Make cone of No. 18 gage galvanized iron with seam on outside. Fill joint on inside with solder and grind smooth. Unit dimensions shown in parentheses.

GRAND COULEE MODEL TURBINE

CONICAL DRAFT TUBE

draft tubes. Velocity measurements were made at the exit section of the three draft-tube models with a direction-finding pitot sphere with the turbine operating at best gate. The results are plotted in figure 15. The velocity traverses indicate that the Grand Coulee and Lower Tallassee models most nearly utilize the area at the exit section. As the extent to which each draft tube utilizes the area of its exit is difficult to determine from the velocity traverses, the Coriolis coefficient, "α", was evaluated at the exit section for each model draft tube. This coefficient is defined as

$$\alpha = \frac{\int_A V^3 \, dA}{\bar{V}^3 \, A}$$

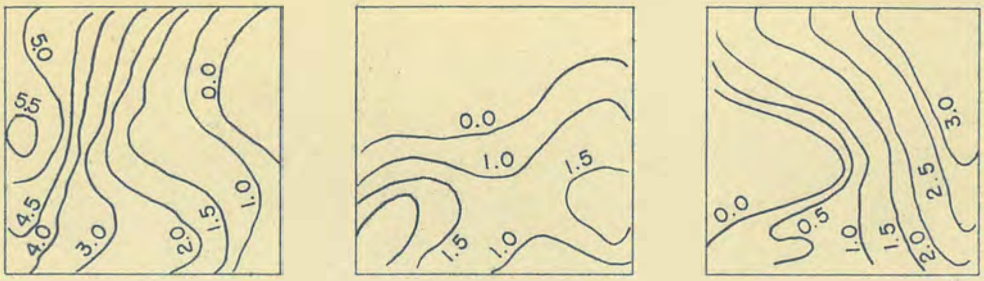
where V = measured velocity,

$\bar{V} = \frac{Q}{A}$, or average velocity,

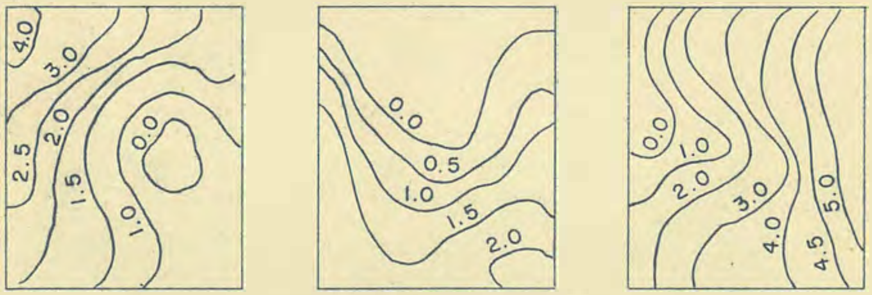
A = area at exit,

Q = discharge in c.f.s.

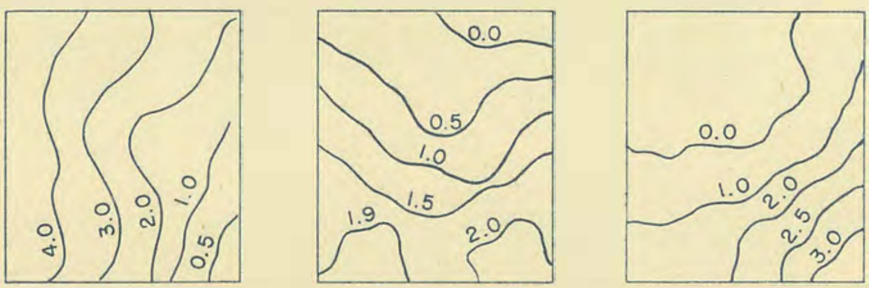
In other terms, "α" is the coefficient which, when applied to the head corresponding to the mean velocity, expresses the true kinetic energy per pound of water passing the section. The values of this coefficient were obtained by graphical integration and are as follows: Lower Tallassee, 3.13; Grand Coulee, 3.28. The coefficients indicate that the Lower Tallassee and Grand Coulee models were practically equal from the standpoint of ability to distribute the flow uniformly over the exit section. The coefficient of the Norris tube was not evaluated because the exit area differed too much from that of the other two tubes for a valid comparison by this method.



NORRIS



LOWER TALLASSEE



GRAND COULEE

GRAND COULEE MODEL TURBINE

VELOCITY TRAVERSE AT EXIT OF DRAFT TUBE

HEAD 13.96 FEET GATE .35

INSTRUMENT- PITOT SPHERE

(VELOCITY EXPRESSED IN FEET PER SECOND)

Additional measurements of velocity were made on the Norris and Grand Coulee models only, at a section 2.5 inches below the discharge edge of the runner band to determine the flow characteristics at the inlet of the draft tube. The results of these tests are shown in figure 16. The similarity of these curves indicates that the flow pattern at the draft-tube entrance is determined by the turbine runner, as previously stated, rather than by the draft tube.

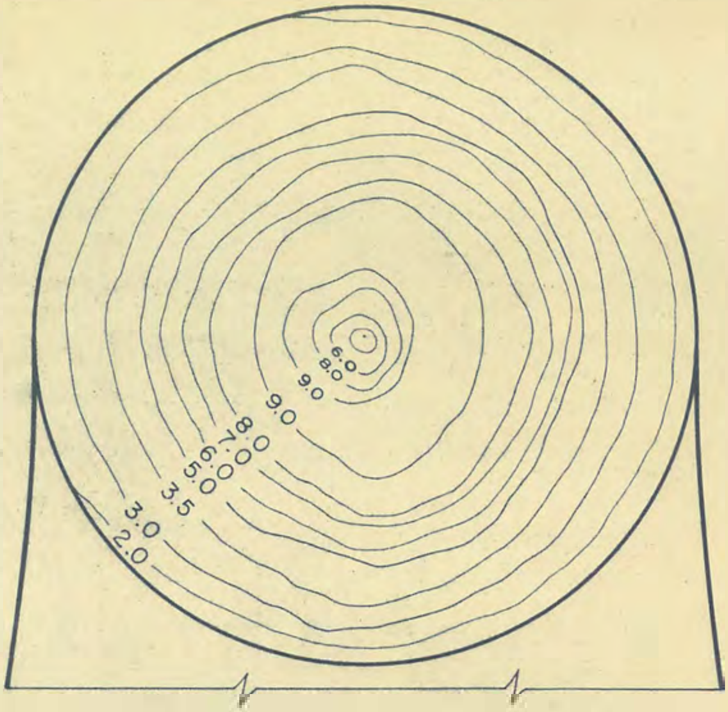
PENSTOCK STUDIES

The initial test program included investigations of the effect on the turbine performance of such shut-off devices as ring-follower gates, needle valves, butterfly valves, etc., in the penstock near the scroll-case entrance, but before these tests were started it was decided that no shut-off device was to be incorporated in the final design and these investigations were not made. The penstock, however, was to approach the turbine at a slope of 20° 32' from the horizontal necessitating a bend of this amount near the entrance to the turbine scroll case as shown in figure 17. The possibility that such a bend in the penstock at this point would affect the performance of the turbine was considered of sufficient importance to warrant an investigation.

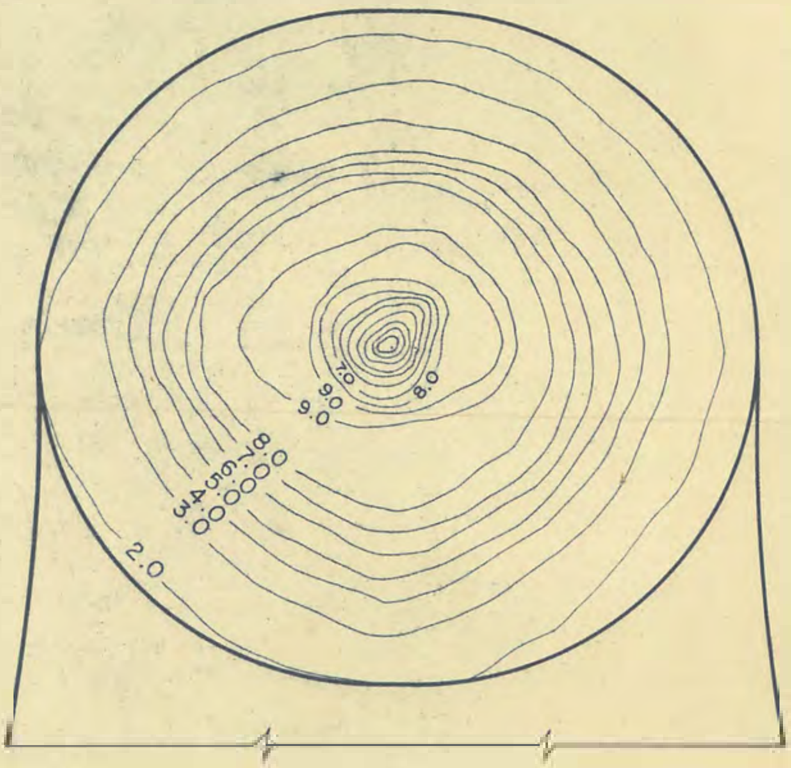
It is well known that a bend in a pipe tends to set up a double spiral in the flow and that this disturbance will persist for some distance beyond the bend. Tests were made to determine whether or not this condition would continue into the scroll case and affect the performance of the turbine.

The head tank was moved to accommodate the penstock design as proposed for the prototype. The revised arrangement is shown in figure 4, (arrangement No. 3) and in figure 17. Tests were made to determine the model performance with the revised penstock. A comparison of the performance curves with those of the short penstock showed that there was no significant difference attributable to the presence of the penstock bend.

Velocity measurements in both penstocks near the scroll-case entrance failed to disclose any effect of the bend.



NORRIS DRAFT TUBE



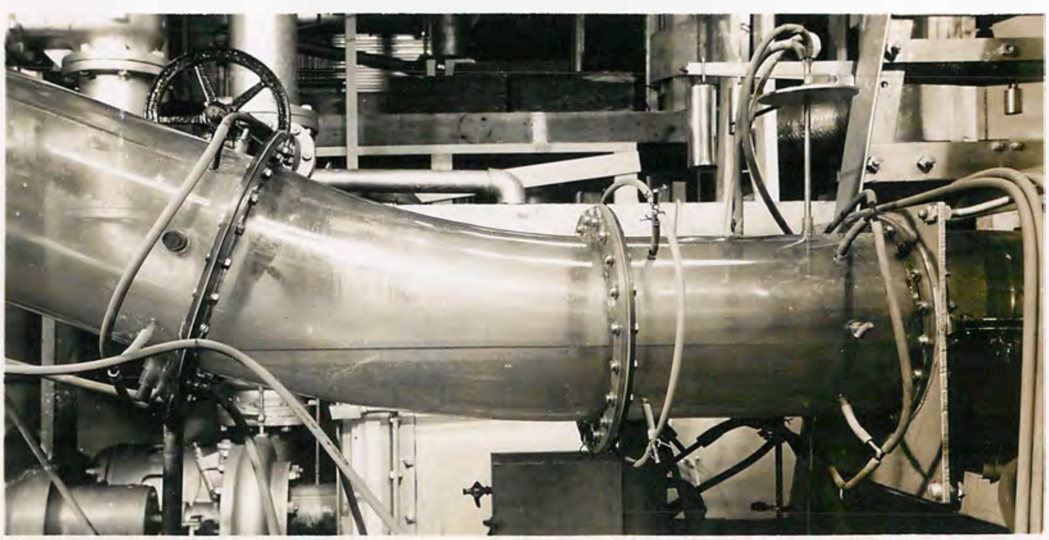
GRAND COULEE DRAFT TUBE

VELOCITY TRAVERSES AT THROATS OF DRAFT TUBES AS MEASURED WITH A PITOT CYLINDER. VELOCITIES IN FEET PER SECOND. HEAD 13.96 FEET. GATE .35

GRAND COULEE MODEL TURBINE

FIGURE 17

A. View of complete penstock and head tank



B - Close-up of penstock elbow

In the light of experimental data concerning flow conditions in pipe bends at the time of these tests the fact that the bend made no discernible difference in the flow conditions at the entrance to the scroll case is somewhat surprising. The explanation may lie in the fact that the penstock was 18 feet in diameter and the scroll-case entrance was 15 feet in diameter necessitating a converging bend instead of one of constant cross-sectional area. On the other hand, the disturbance of flow approaches a minimum as the angle of the bend approaches zero, and it may be that the small angle of 20° 32' did not cause enough disturbance to affect the performance of the model to an appreciable extent. No effort was made to measure the actual head loss in the bend.

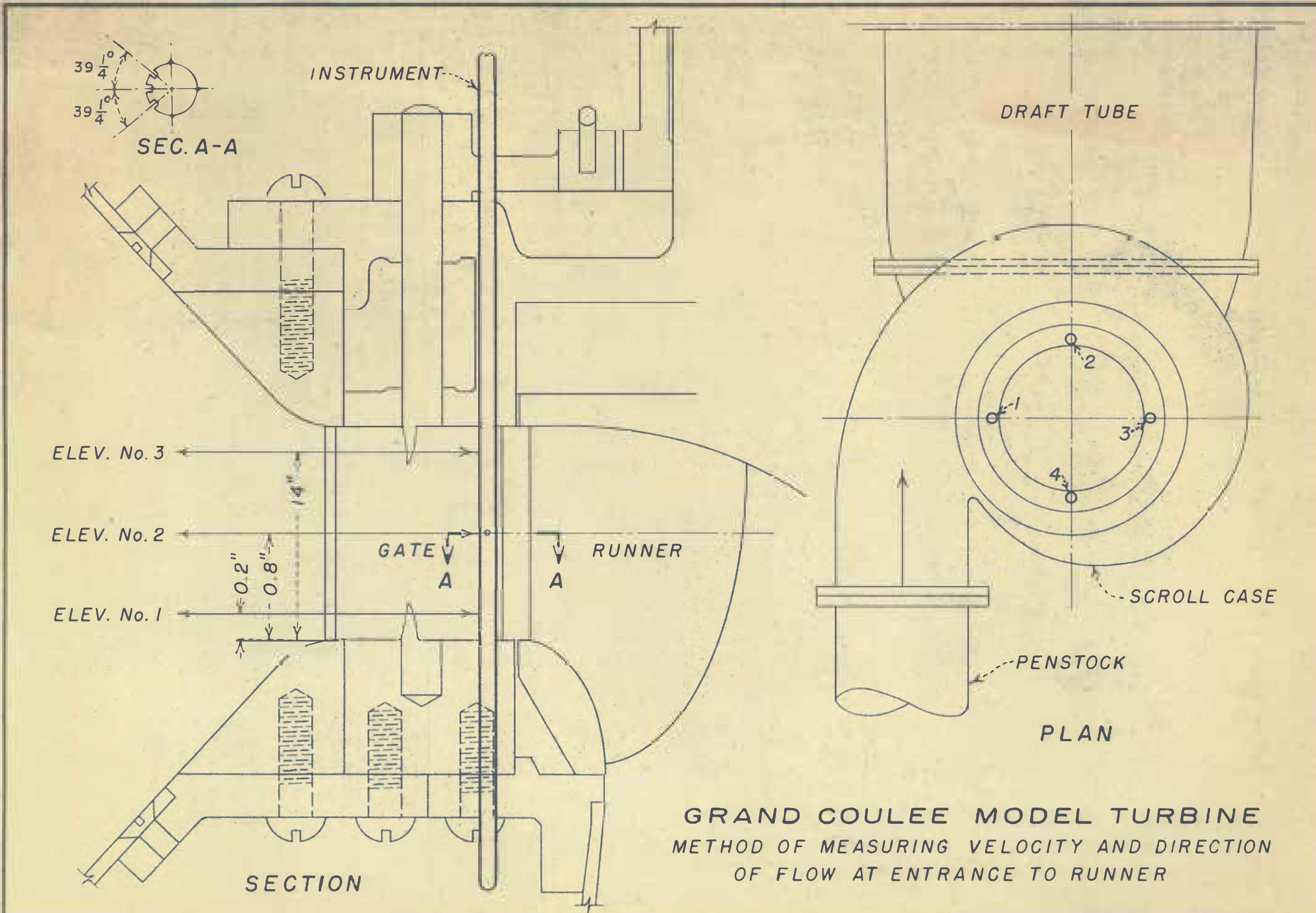
SCROLL-CASE AND SPEED-RING STUDIES

The function of the scroll case in connection with a water turbine is to distribute the water equally around the periphery of the turbine runner. The water flowing through each of the turbine gates should have the same relative direction and have the same energy content in order that each turbine vane may do its full and equal share of the work.

Scroll-case designers have different theories for accomplishing this objective and although these theories are somewhat divergent the results appear to coincide remarkably well with respect to performance. As the model-testing program did not contemplate an investigation of the various design theories, one of them was selected on the basis that it gave dimensions which more or less averaged those given by the other theories. The theory adopted for the design of the model scroll case is based on the law of the free vortex according to which the velocity varies inversely with the radius, or $rV = c$, where c is a constant. In applying this relationship to scroll-case design, r is the radius from the axis of the runner to the center of gravity of any radial section of the scroll case, and V is the mean velocity normal to that section.

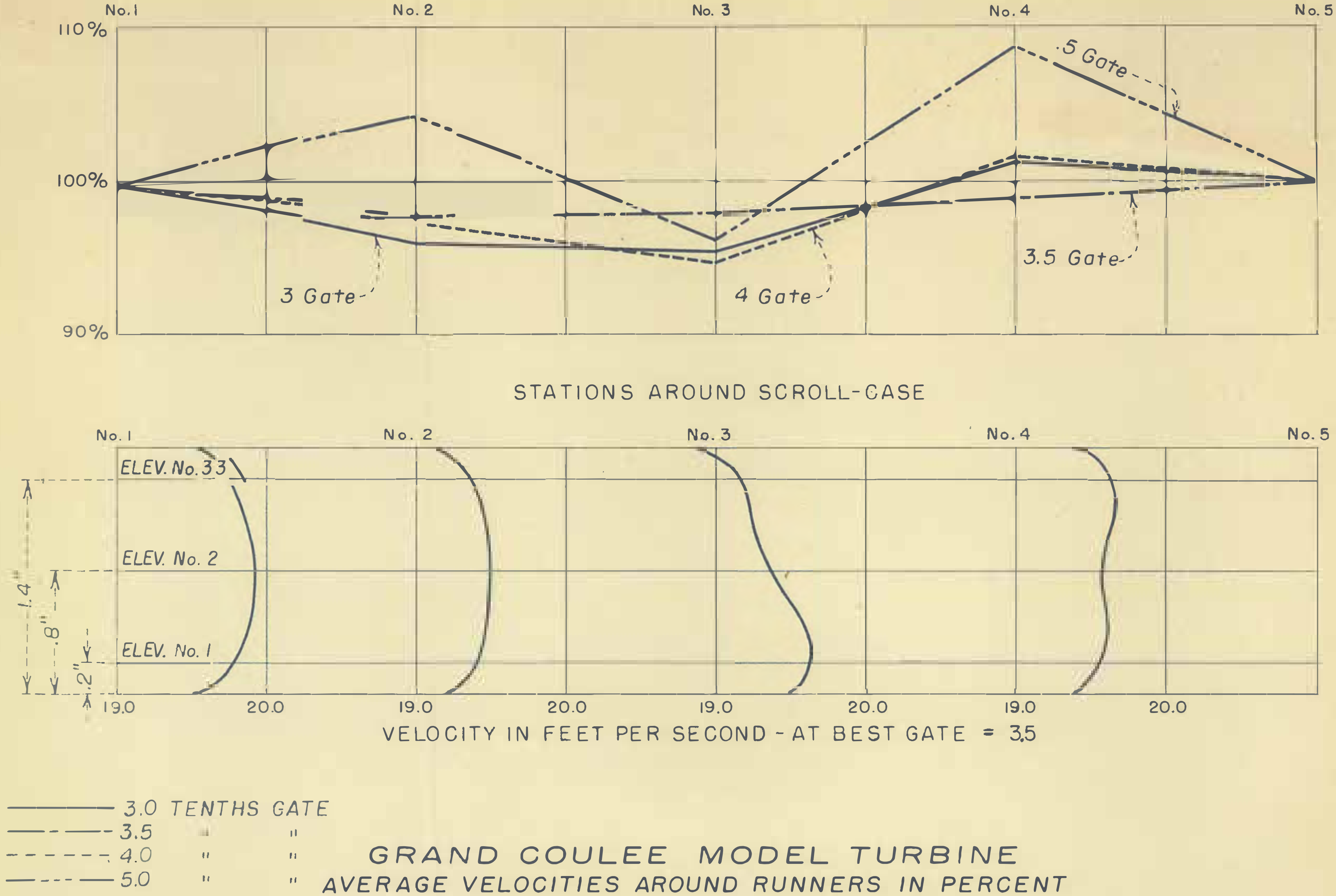
As it was not known how effectively this scroll-case would distribute the flow to the runner and as the flow distribution would certainly be reflected in the performance of the draft tube, it was considered advisable to make some tests in this connection. The arrangement used in the velocity-distribution tests is shown in figure 18 and the results are shown plotted in figure 19. Measurements were made at four points around the circumference of the runner and in three rotational planes as indicated in figure 18. The instrument used was a pitot cylinder having three openings 39.25 degrees apart. The three openings were each connected to a mercury U-tube through small brass tubes and rubber-tube connections. In making measurements, the pitot cylinder was rotated until the two outside U-tubes indicated equal pressures, establishing the direction of flow. The differential between either of the outside holes and the middle hole together with a calibration coefficient indicated the velocity. The velocities at each station and in each plane, with the turbine model operating at best gate (0.35), are plotted in figure 19. The velocity distributions at stations one and two are excellent though the average velocity at station one is slightly higher than that at station two. Station four shows very good velocity distribution but station three shows some slight distortion. The average velocities at each station for various gate openings plotted nondimensionally as percentages of the average velocity at station one, are also shown in figure 19. At best gate, the velocities at the four stations show a variation of less than two percent but for other gate openings, the variation is much greater. No attempt was made to determine the cause of this rather unexpected phenomena as the variation was not sufficient to affect the results of the draft-tube tests appreciably. These tests were made using the conventional speed ring.

The speed ring in a hydraulic turbine serves primarily as a structural member to hold the scroll case together and, in a vertical



GRAND COULEE MODEL TURBINE
METHOD OF MEASURING VELOCITY AND DIRECTION
OF FLOW AT ENTRANCE TO RUNNER

FIG. 18



GRAND COULEE MODEL TURBINE
AVERAGE VELOCITIES AROUND RUNNERS IN PERCENT
AT 3-3.5-4-5 TENTHS GATE —BEST GATE = 3.5

FIG. 19

turbine, to support the weight of the generator. Since it is necessarily in the path of flow, it must be designed to offer the least possible resistance if high turbine efficiencies are to be realized. Previous studies* concerning the effect of speed-ring design on turbine performance have indicated that the conventional design may be improved by so placing the vanes that they will conform more nearly to the lines of flow. Paint tests on the speed ring of the Grand Coulee test model, figure 20, showed that the vanes did not conform to the flow lines. A second speed-ring model was made as shown in figure 21, with vanes shaped to conform more nearly to the flow lines as deduced from the paint tests. Figures 20 and 21 show that there was very little improvement and figure 22 shows that no appreciable change in efficiency occurred.

The flow lines of water in scroll cases, especially those of circular cross section, are very complex. Paint tests on the model scroll case indicate that the direction of flow along the boundary was nearly radial as the water entered the speed ring. While this was true adjacent to the speed-ring shrouds, it is quite likely that the direction of flow was different at other points between the shrouds, because there were several forces acting in various degrees and directions at different points within the fluid. These forces were friction, pressure, and centrifugal force acting on the fluid while in motion. Take any section of the scroll case in a radial plane. Centrifugal force, f, acting upon a unit volume of the fluid the mass times the velocity squared divided by 2g times the radius of curvature of the path. Friction acted to reduce the angular velocity of the fluid in contact with the wetted surface of the scroll, with a consequent reduction in its centrifugal force. The velocity at the center of the section for any given radius, being least affected by friction, reached its maximum value. Therefore, the centrifugal force in this region was greater than that near the top and bottom surfaces of the scroll. The water near the center of the section tended to flow

* "Applications of Hydraulic Laboratory Researches," by I. A. Winter, A. S. M. E. Transactions 1931, vol. 53, HYD-53-4, pp. 27-40.

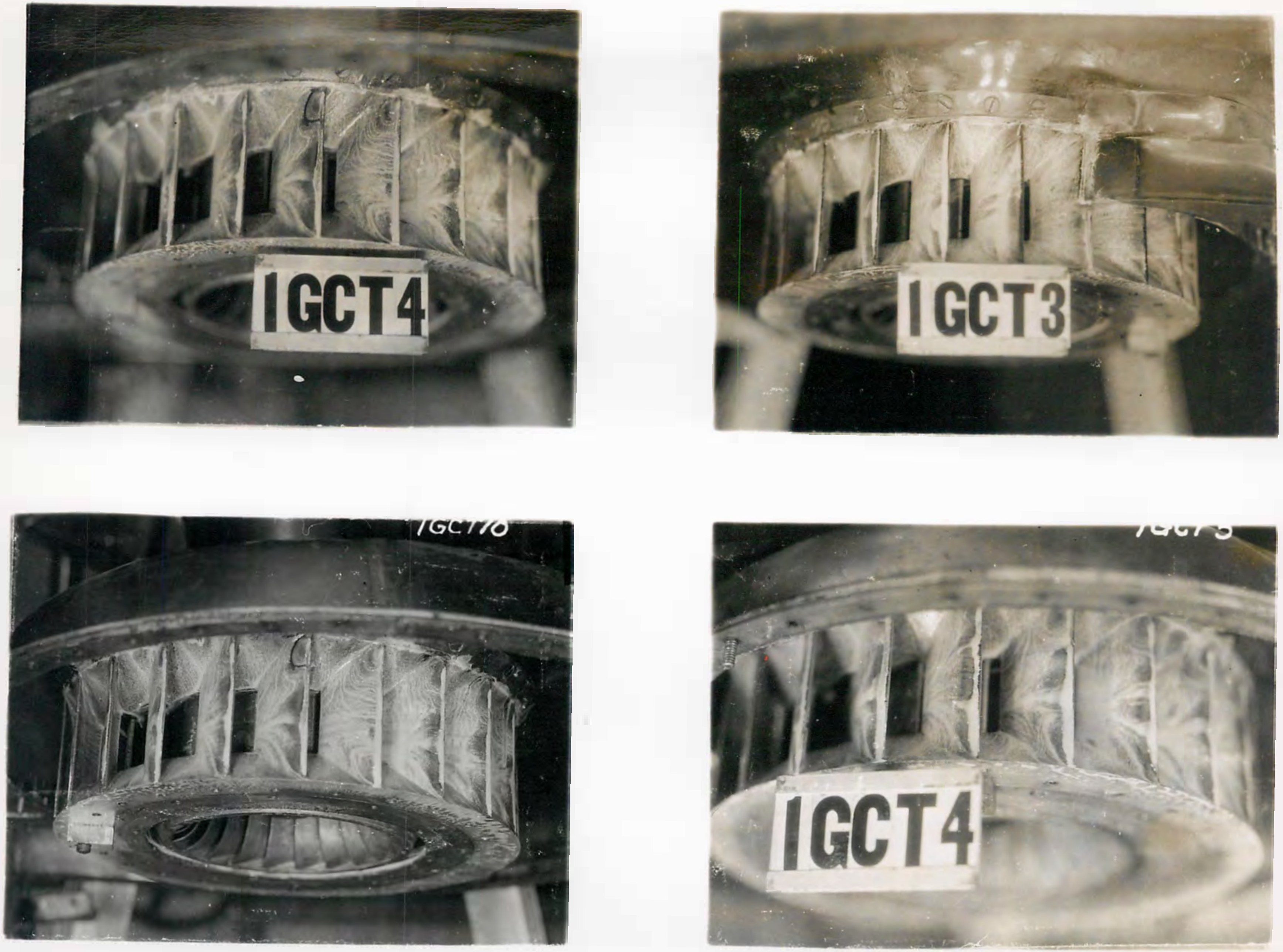
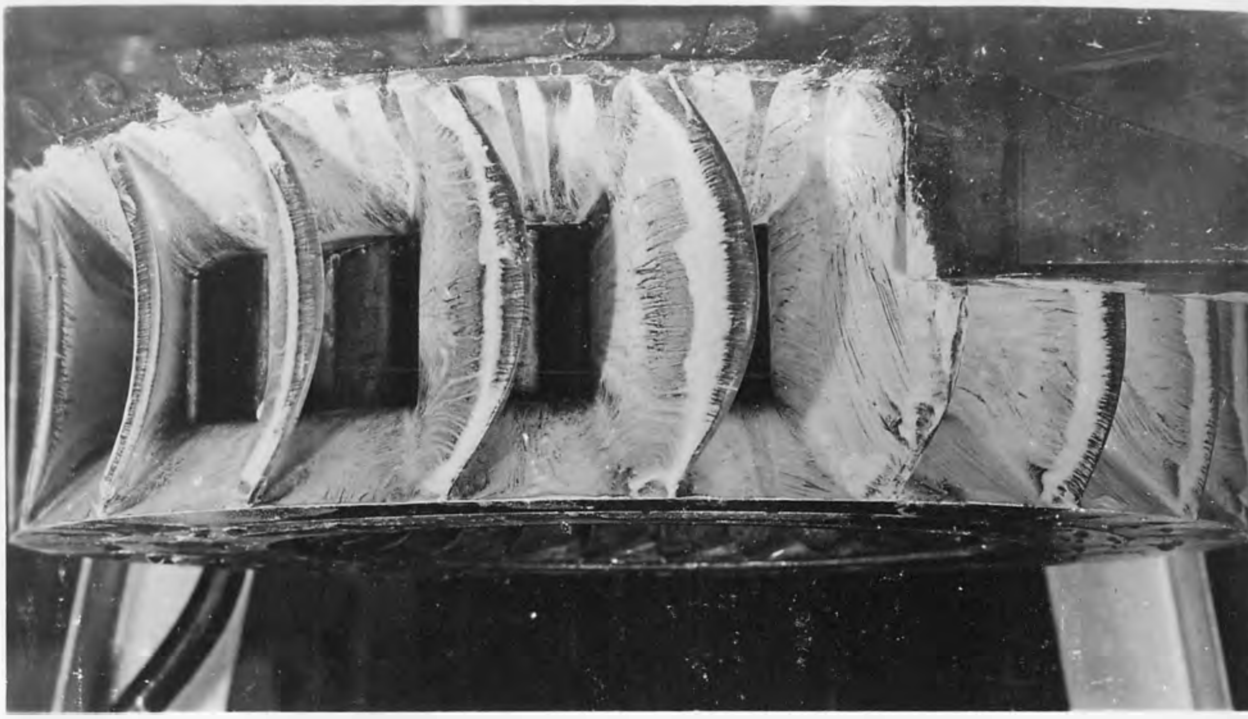
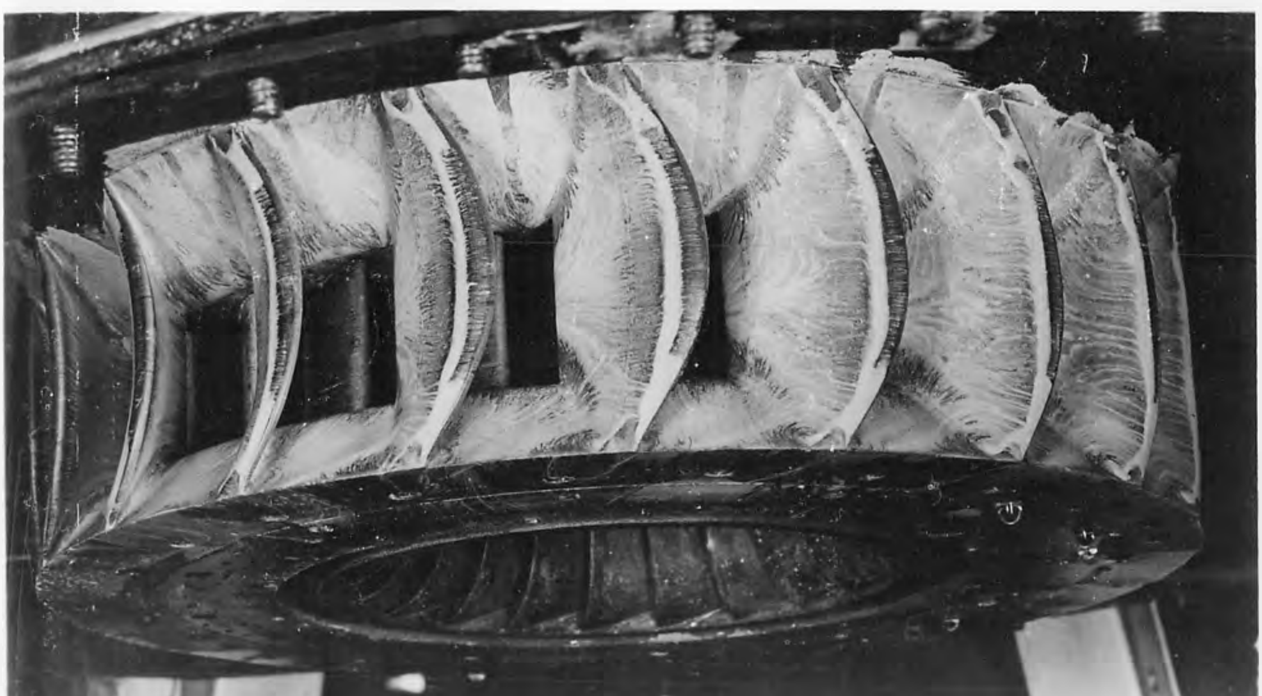


FIGURE 20

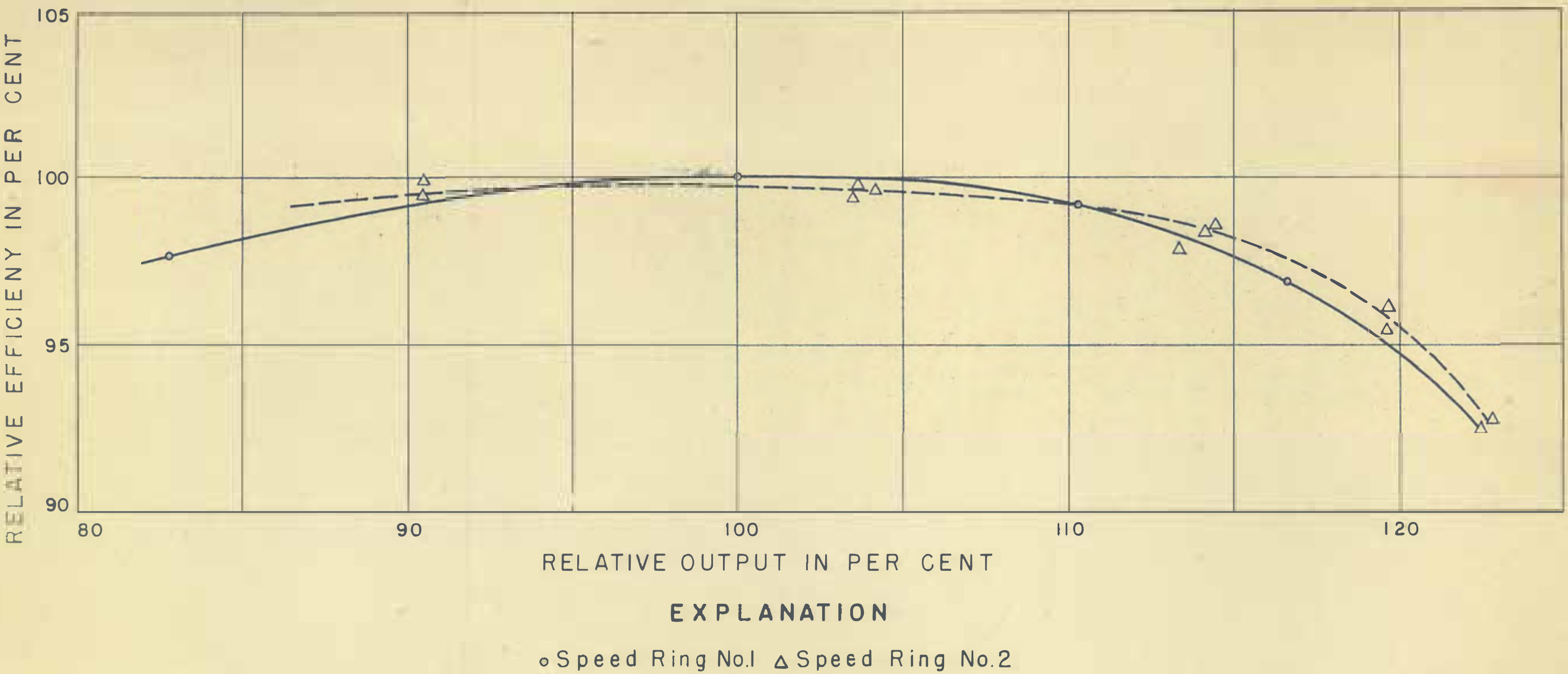
Paint tests of Speed Ring No. 1

FIGURE 21





Paint Tests of Speed Ring No. 2



GRAND COULEE SPEED RING STUDIES

PERFORMANCE OF TURBINE WITH CHANGE IN SPEED RING RELATIVE TO EFFICIENCY AND OUTPUT AT PEAK EFFICIENCY WITH GRAND COULEE TUBE AND SPEED RING NO. 1

FIGURE 22

outward and that near the top and bottom tended to flow inward toward the speed-ring entrance. This resulted in a double spiral flow in the scroll case; one spiral being clockwise and the other being counterclockwise. The convergence of the scroll-case in the direction of flow and flow of part of the water through the speed ring, acted to retard the outward component of velocity and the resultant velocity was nearly tangential in the central region of the scroll case. For this reason the vanes in No. 2 speed ring were designed to approach the tangent at the central section and to approach the radial at the upper and lower shrouds, as shown in figure 21.

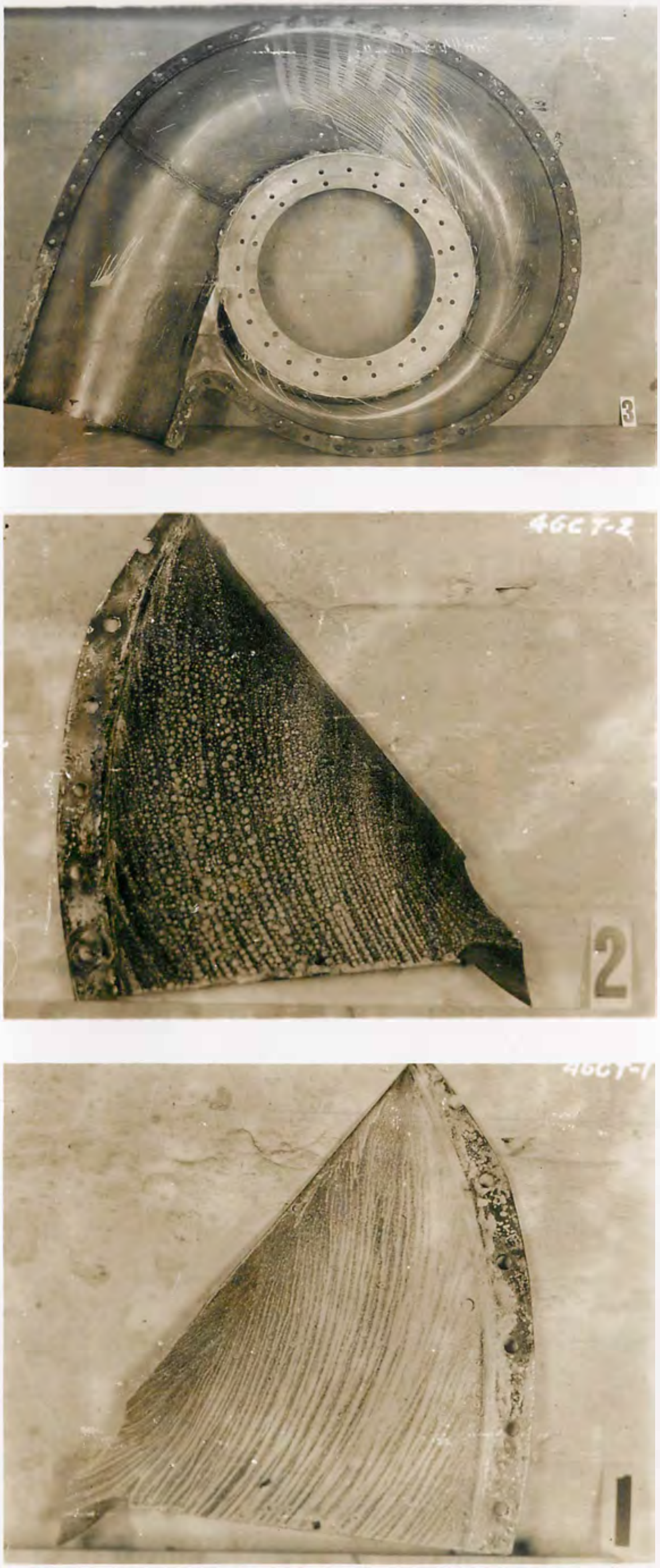
To verify the assumed direction of flow lines used in designing speed ring No. 2, a paint test was made by inserting in the plane of symmetry of the scroll case the metal plate shown in figure 23. The results were disappointing because the presence of the metal plate altered the velocity distribution. The scroll case being divided into two parts, the flow in each part behaved as it would in two separate scrolls. Each part developed a pair of spirals rotating in opposite directions. It should be kept in mind that paint tests show the direction of flow lines only along the boundary surface and that the direction of flow lines in the interior of the water passage must be deduced from these.

Turbines operating in connection with scroll cases having a rectangular section nearly always give higher efficiencies than those using scroll cases with circular sections. It is believed that this is due to the fact that the double spiral flow as described above is discouraged by the square corners in the rectangular-section type of scroll case.

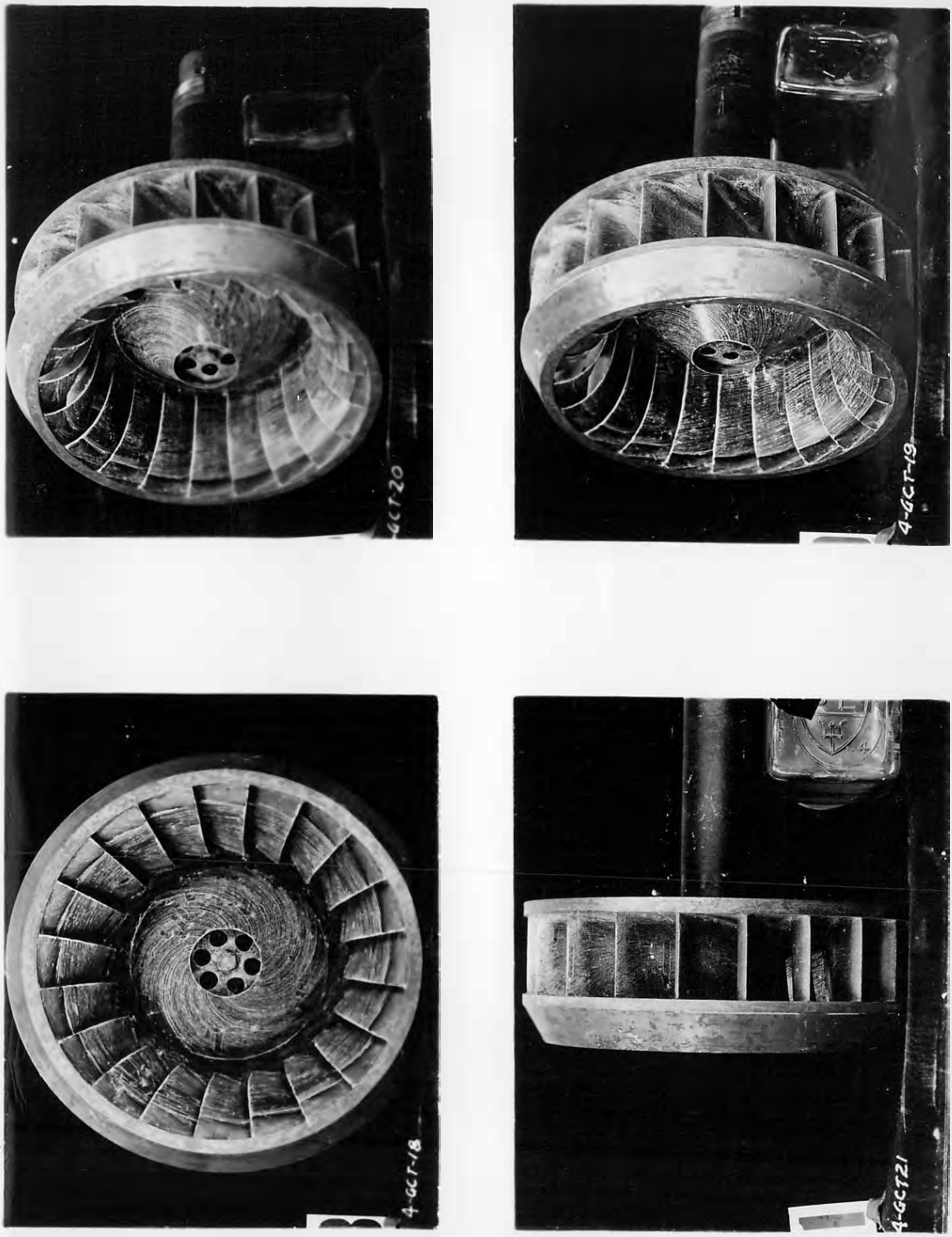
TURBINE-RUNNER AND FAIRWATER-CONE STUDIES

A paint test was made on the runner and fairwater cone to determine the flow lines through the buckets and on the cone. The results are shown on the photographs, figure 24. The general pattern of the flow

FIGURE 23



Paint Studies of Scroll Case



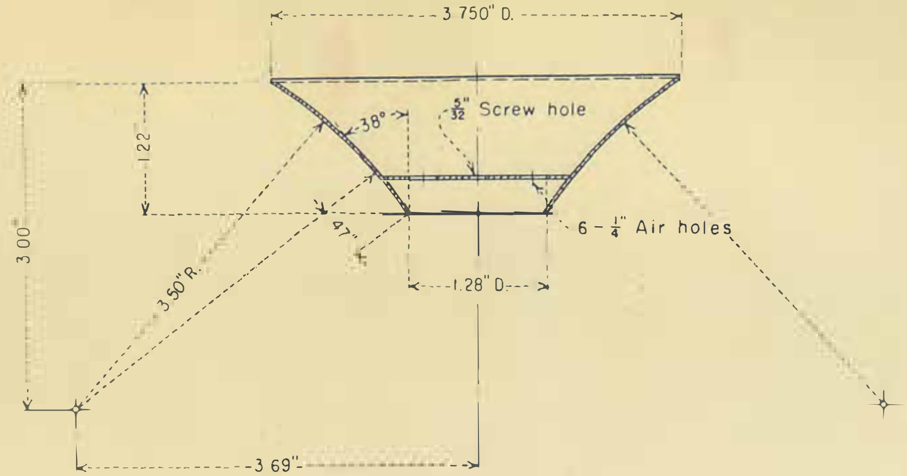
Paint Studies of Runner

lines indicates that the flow through the runner and on the runner cone occurred as anticipated in the design except near the runner band. It is believed that the markings in this region were obliterated by the gate leakage after the gates were closed and before the water could be drained away from the runner.

Tests were made to determine the relative effect of various fairwater cones on the performance of the turbine. Fairwater cones of various proportions as shown in figure 25 were tested, No. 3 being the one originally designed for the model runner. The results are shown plotted nondimensionally in figure 26. Cone No. 3 resulted in highest efficiency but the turbine reached its highest-peak horsepower output with no cone at all. Cone No. 2 gave a reduction in both the efficiency and horsepower output. The reduction was doubtlessly due to the restriction in area at the outflow of the turbine runner due to this cone.

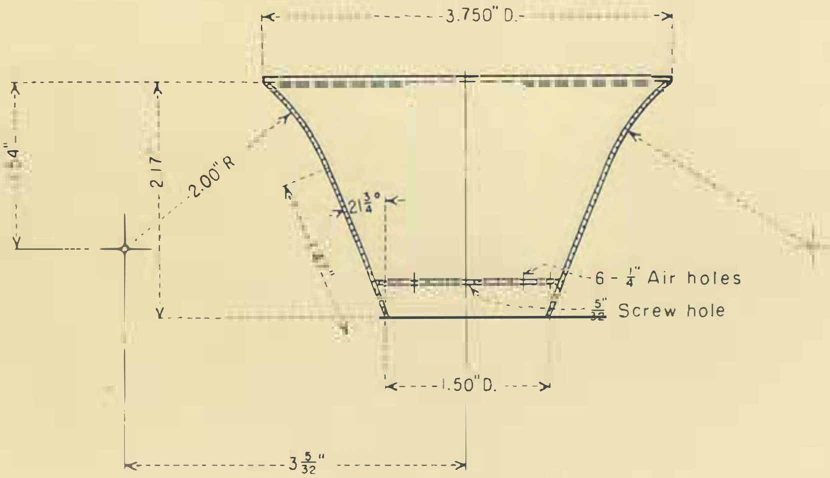
INVESTIGATION OF DRAFT-TUBE DISTURBANCES

The transparent draft-tube models used in the tests made it possible to observe the behavior of the water as it discharged from the runner and passed through the draft tube. When a small amount of air was admitted into the scroll case it was noticed that the air collected below the runner fairwater cone in the form of a rotating spiral vortex extending some distance down into the throat of the draft tube. The speed of rotation of the spiral was relatively slow at small gate openings and increased with the gate opening. Also the pitch of the spiral decreased as the gate opening increased. The action was most severe between 0.1 gate and 0.3 gate. As the tail of the spiral passed any given point there seemed to be a pressure surge and a listener could detect an audible swish. At gate openings greater than 0.4, the frequency of the surges was so high that they were not readily detected. So far as could be determined on the model this phenomenon did not affect the torque developed by the runner.



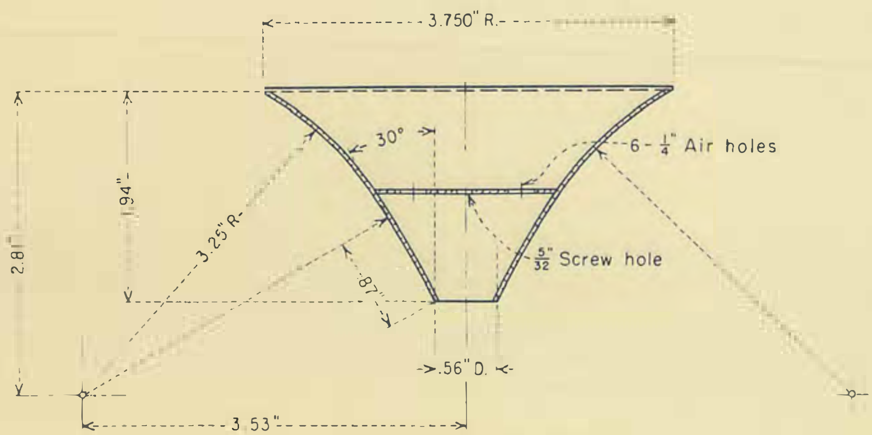
RUNNER CONE No. 1

1/32" COPPER



RUNNER CONE No. 2

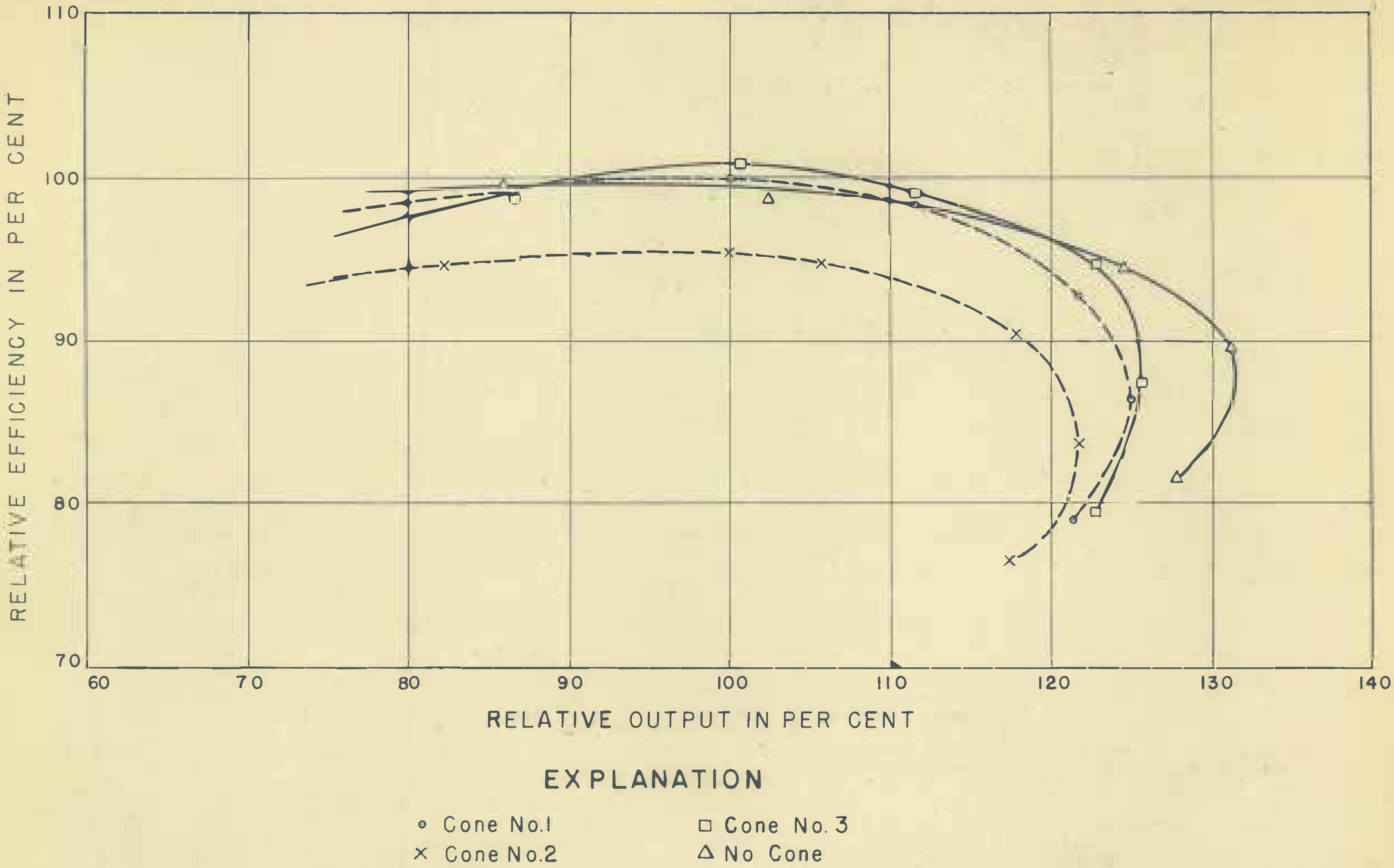
1/32" COPPER



RUNNER CONE No. 3

1/32" COPPER

GRAND COULEE MODEL TURBINE
RUNNER CONES



GRAND COULEE TURBINE RUNNER STUDIES

PERFORMANCE OF TURBINE WITH CHANGE IN RUNNER CONES RELATIVE TO THE EFFICIENCY AND OUTPUT AT PEAK EFFICIENCY OF THE GRAND COULEE TUBE AND CONE NO.1

FIGURE 26

The spiral vortex assumed various shapes, depending upon gate opening. At small gate openings, the shape was something like a carrot twisted into a corkscrew spiral loosely attached to the tip of the fairwater-cone and swinging rather lazily around in the conical section of the draft tube. As the gate opening increased the vortex whipped wildly around until the gate opening approached that of best turbine efficiency (0.35). Then it straightened out and assumed a shape similar to a loosely twisted wisp of strings hanging vertically downward. At about 0.4 gate the twist reversed to the opposite from the direction of rotation of the runner. At 0.5 to 0.6 gate, the stringlike vortex again assumed the spiral shape and whipped violently around, still keeping its slender proportions. At approximately 0.70 gate, which is beyond the gate for maximum horsepower, the vortex virtually disappeared.

Motion pictures of this phenomenon were taken on 16 mm. film and placed in the files of the hydraulic laboratory for future reference.

Power swings, or pulsations, have been a source of concern to operators and engineers ever since about 1912, and reports received from the field indicated that some of the Bureau operators in plants such as Seminoe, Elephant Butte, and Boulder were experiencing trouble of this kind. There is little literature on the subject but the consensus of opinion seems to be that there is a relationship between power swings and draft-tube disturbances. A paper by Mr. W. J. Rheingans*, together with discussions, goes rather deeply into the subject but does not offer a definite solution of the problem.

The laboratory was not equipped to make a complete investigation of the problem but attempts were made to devise some means of breaking up the vortex in the throat of the draft tube in the hope that this would either eliminate the power swings or at least reduce them to tolerable dimensions without materially affecting the horsepower and efficiency of the turbine.

* "Power Swings in Hydroelectric Power Plants," by W. J. Rheingans, Transactions A.S.M.E., vol. 62, 1940, pp. 171-184.

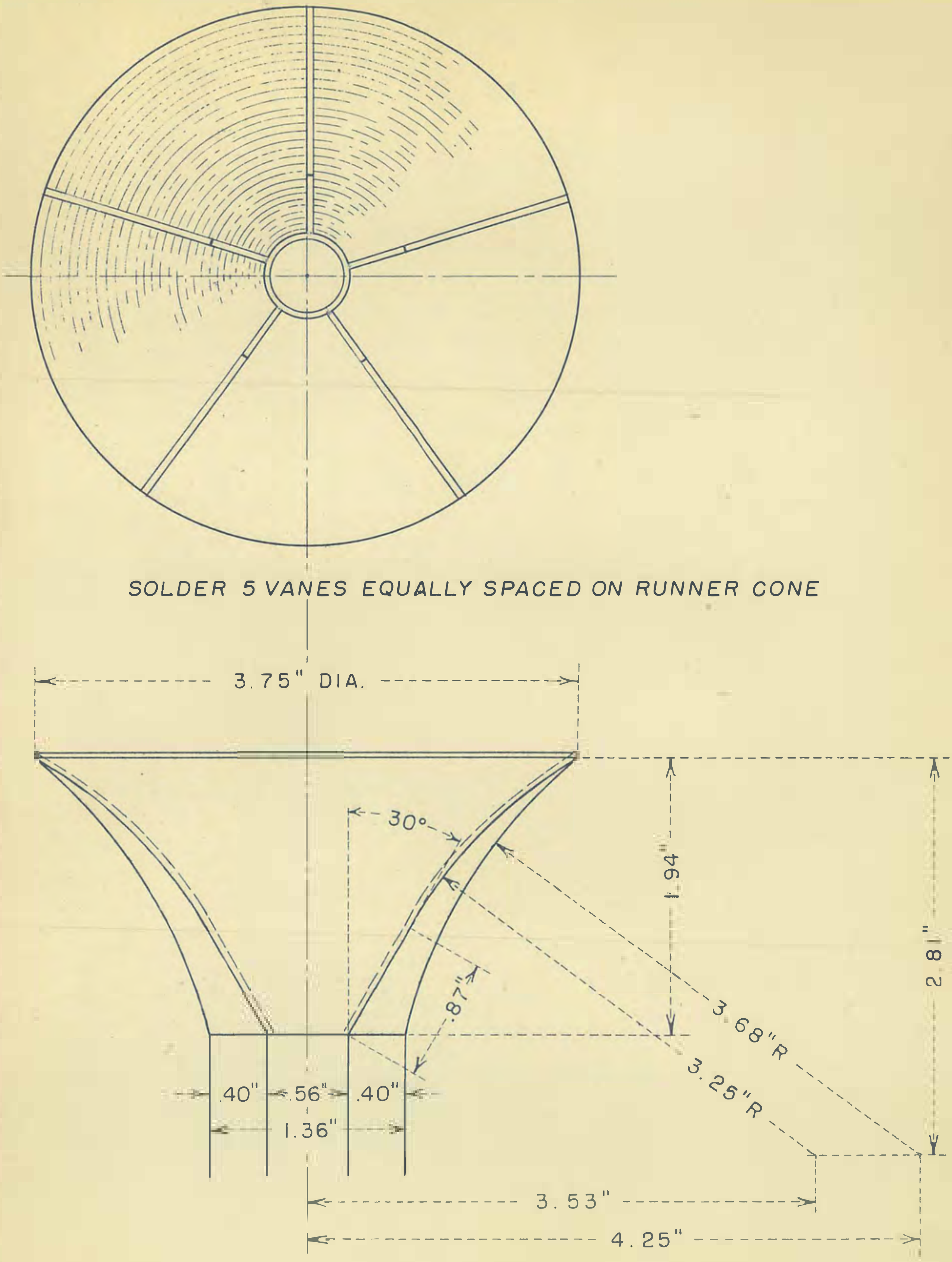
Three types of corrective devices were studied. First, fins were placed inside the throat of the draft tube; second, fins were attached to the fairwater cone; and, third, vanes radiating from a central hub under the runner were anchored to the wall of the draft tube.

The dimensions of the draft-tube fins were 0.10 inch by 0.25 inch by 2.00 inches. Five of the fins were positioned in radial planes equally spaced around the inside of the draft-tube throat, beginning in the plane of symmetry of the draft tube on the downstream side. The 2.00-inch dimension extended from the throat flange downward in a direction parallel with the turbine axis and the 0.25-inch dimension extended radially inward toward the turbine axis. There appeared to be a tendency for these fins to confine the spiral vortex into slightly closer limits about the axis of the turbine. It was expected that the fins would straighten out the vortex filaments and cause the discharge to flow in a more nearly vertical direction and thus disrupt the spiral. There were no visual indications that this was accomplished. The efficiency and horsepower of the turbine were not affected.

The second type of corrective device consisted of fins placed on the runner fairwater cone. A fairwater cone with five radial vanes as shown in figure 27 was tried. These vanes not only failed to break up the vortex but resulted in an appreciable decrease in the efficiency of the turbine. A fairwater cone with three vanes placed spirally as shown in figure 28A was then tried. This also resulted in a loss in efficiency and peak horsepower. Neither of these devices effectively eliminated the vortex.

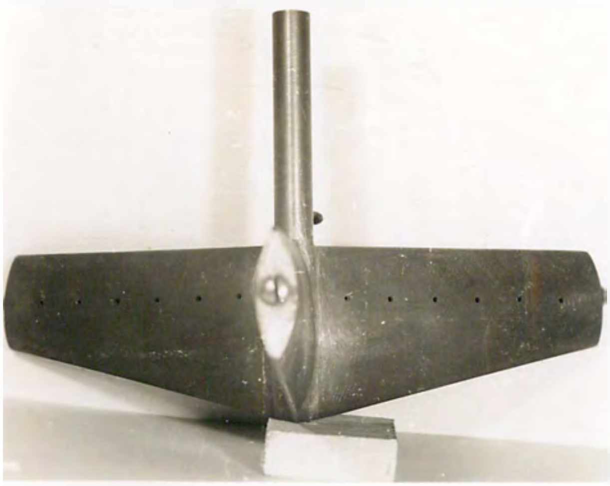
The third type consisted of streamlined splitter vanes extending radially from a central hub and anchored to the wall of the draft-tube throat. Two sets of these splitter vanes of somewhat similar design, and referred to as No. 1 and No. 2, were tested. The hub, which formed an unattached continuation of the fairwater cone, was hollow. The three streamlined vanes radiating from the hub were also hollow and

FIGURE 27



GRAND COULEE MODEL TURBINE
FAIRWATER CONE STUDIES
VANES ON CONE

A - Fairwater cone



B - Splitter No. 1

C - Splitter No. 2

were provided with holes along the sides for bleeding air into the water to prevent pitting on the vanes. Photographs of these splitters are shown in figure 28B and C.

The splitters were tested in two positions to determine which would be most effective in breaking up the vortex and in rectifying the flow in the draft tube. The first position was with one of the legs to the front, or pointing downstream in the direction of flow out of the draft tube. The second position was with one leg back, or upstream from the direction of flow from the draft tube.

The splitters not only confined the vortex to the center of the draft-tube throat but almost entirely disrupted it. In addition, the splitters appeared to function as a flow straightener to distribute the flow more uniformily over the exit section of the draft tube at all gate openings. In other words, the tendency of the flow to switch from one side of the draft-tube to the other as the gate opening passed through that of best efficiency was considerably reduced, making the draft tube more effective at over-gate and under-gate positions.

The tests indicated that splitter No. 2 was more efficient than No. 1 and that the position with one leg to the front, or downstream, was more effective than with the leg to the back, or upstream. Figure 29 shows in nondimensional form the relative effect of the splitters on the over-all efficiency of the turbine. The relative values of efficiency and power output are based on the output of the turbine without the splitter. It is apparent that the splitter will increase the efficiency of the turbine over the operating range of gate openings, but with a sacrifice of peak power output. This is probably due to a reduction in discharge capacity resulting from constriction of the effective area of the water passage.

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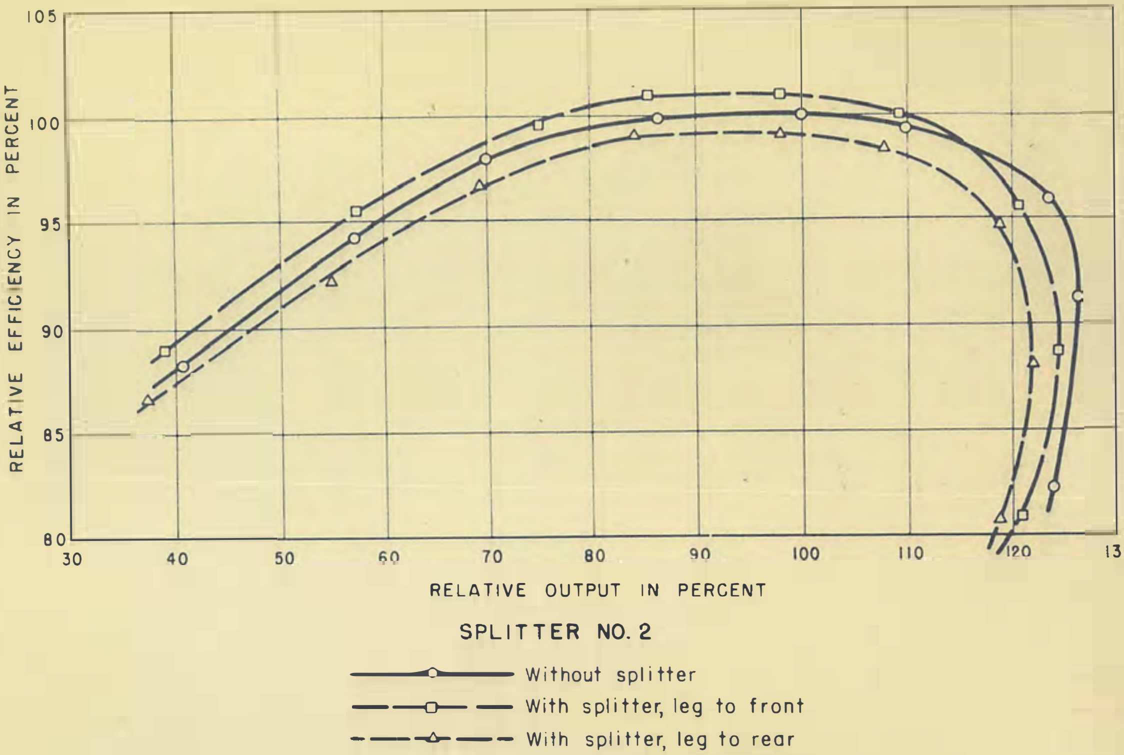
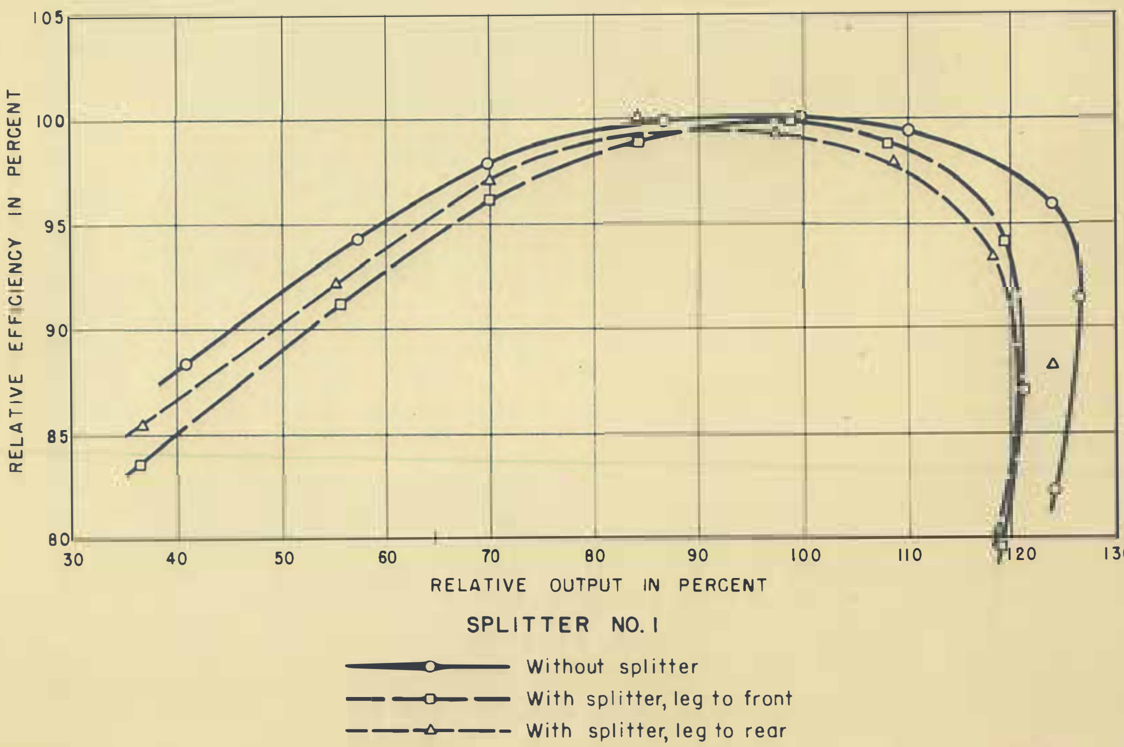




MODEL TURBINE PERFORMANCE AS AFFECTED
BY SPLITTERS IN THE THROAT OF THE DRAFT TUBE

GRAND COULEE DRAFT TUBE STUDIES

APPENDIX

APPENDIX

CONSTRUCTION DETAILS OF THE GRAND COULEE TURBINE MODEL

1. Introduction. The design and construction of the model for the studies described in the foregoing report presented many problems the like of which had not been encountered by the laboratory personnel at the time the model was constructed. Considerable training was therefore necessary in order to accomplish the desired results.

To facilitate the study of flow conditions in the penstock, scroll case and draft tube, a complete homologous model of the turbine unit was built of transparent plastic material as far as practicable. The penstock, the scroll case, and draft tube were made of transparent material, but the turbine case, speed ring, gates and runner, being difficult to make of transparent material, were made of suitable metals. Considerable process-development work was necessary in the making of these parts. This appendix is being added to this report in order to record the details of the methods, processes, and techniques used in building the models for the Grand Coulee turbine and draft-tube investigations.

2. The model design. It was necessary to create a complete turbine design on the basis of the requirement at Grand Coulee such as head, horsepower, speed, discharge, etc., as the turbine contract had not been let and there was no turbine design available to meet the specified conditions. When the design was completed the physical dimensions affecting the hydraulics of the model were reduced in the scale ratio of 1:24.

3. Constructing the model. Some of the features of the model such as the circulating pump, the model supporting structure, the weir box and water channels presented no unusual construction problems, but the construction of the turbine model with appurtenances called for skills, techniques, and methods outside the field of usual hydraulic laboratory practice. The features requiring special skills, techniques and methods were as follows:

a. The turbine runner and wicket gates.

b. The speed ring.

c. The scroll case and draft tube.

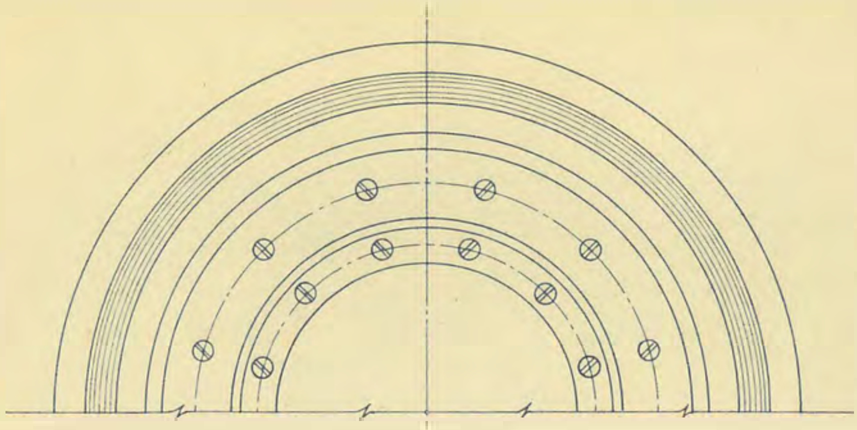
4. <u>Casting the turbine wicket gates and runner.</u> After considering various methods of constructing these features of component parts, soldered, brazed, or welded together, it was decided to cast the gates in one piece, and if successful, to cast the runner complete in one homogeneous casting. This appeared to be an extremely difficult undertaking as the runner was only 7.90 inches in diameter and contained 20 vanes of a rather complicated form.

The metal selected for casting the wicket gates, the speed ring, and the turbine runner was "Government Genuine" babbit metal made by the Syracuse Smelting Works. This metal was selected because of its low melting point, high tensile and compressive strength and machinability.

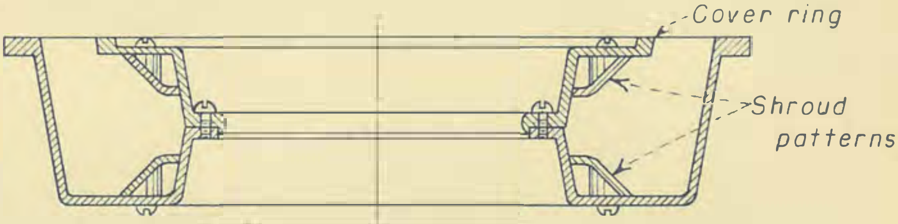
The mold for casting the wicket gate and gate stem was made of ordinary cold-rolled steel in two parts guided together by dowels. No trouble was expected in making these simple castings. However, it was found that there was an indenture about the size and volume of half of a small peanut in each vane casting near the base of the gate stem. No amount of preheating or venting of the mold was sufficient to eliminate the defect which was always alike and in exactly the same location. Finally it was suspected that the defect was due to shrinkage. The metal in the gate stem had solidified before that in the vane and would not flow down into the vane to supply the deficiency due to shrinkage when the vane solidified. The stem portion of the mold was then heated up to about the melting point of the metal and the vane portion of the mold was kept comparatively cool. The result was a vane casting entirely free from defects. This information proved exceedingly useful in casting the turbine runner and speed rings.

The mold for casting the turbine runner shown in figure 1A was far more involved. First, the mold had to be designed and castings had to be made in a local foundry. The main casting was of iron and shaped somewhat like a cake bake pan with a cone in the center. The cone served as a mold for the under side of the runner and also as a support for the runner-crown pattern and the runner-crown top mold, or cover plate. The material selected for the cores forming the mold for the runner vanes was "dental investment," such as is used for casting gold bridge work and crowns for teeth. With the runner-band pattern in place and the crown pattern and cover plate bolted down, two solid segments of the plaster were cast. The end of one of the segments was then carved, using thin metal templates, exactly to the form of the back of a turbine vane. Then the adjacent end of the other segment was carved in exactly the same manner to form the front of a turbine vane. The carved ends of the segments were given a thin coat of shellac. The segments were then returned to the mold and spaced by means of a spacer block at the proper positions to form a segment of 18°, or 1/20th of a complete circle. This formed the core box for molding the runner cores. The required twenty runner-vane cores were then molded from the dental plaster in this core box, using a very thin coat of grease to prevent sticking. The cores were then dried for 24 hours in an electric oven at about 260° F. The core box, cores, and forming blocks are shown in figure 2A.

The mold was then set up for casting the runner. First the runner-band pattern and the runner-crown pattern were removed from the mold. The mold was thoroughly cleaned and the dental plaster cores were stacked in place in the mold as shown in figure 2A. Note that a small notch was cut in one of the cores to provide a place to pour the molten metal. The joints between the cores provided sufficient venting for the gases to escape. The cover plate was then bolted down and the mold was ready for pouring.

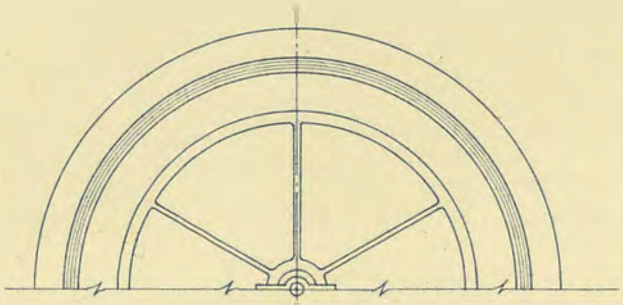


HALF PLAN

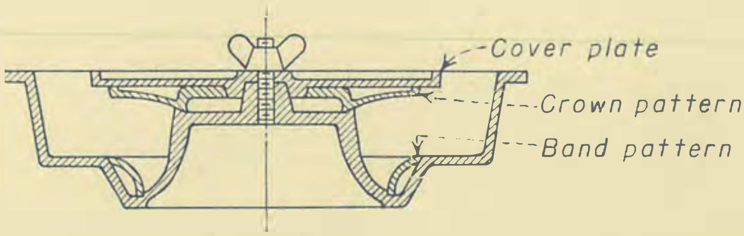


SECTION

SPEED RING MOLD



HALF PLAN

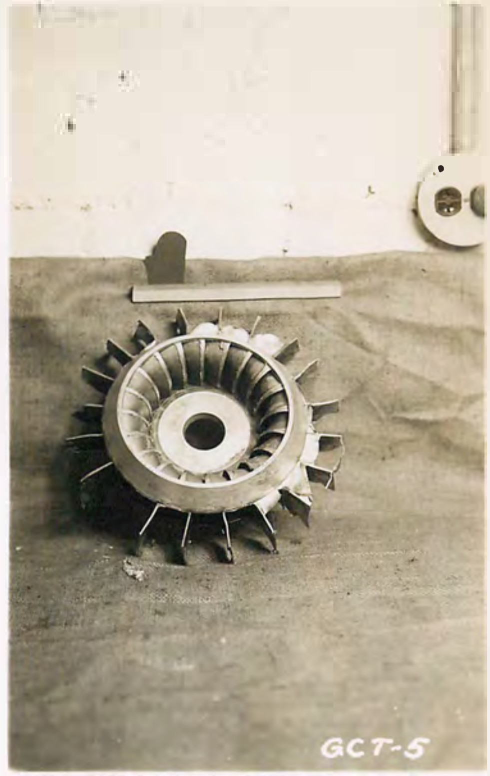


SECTION

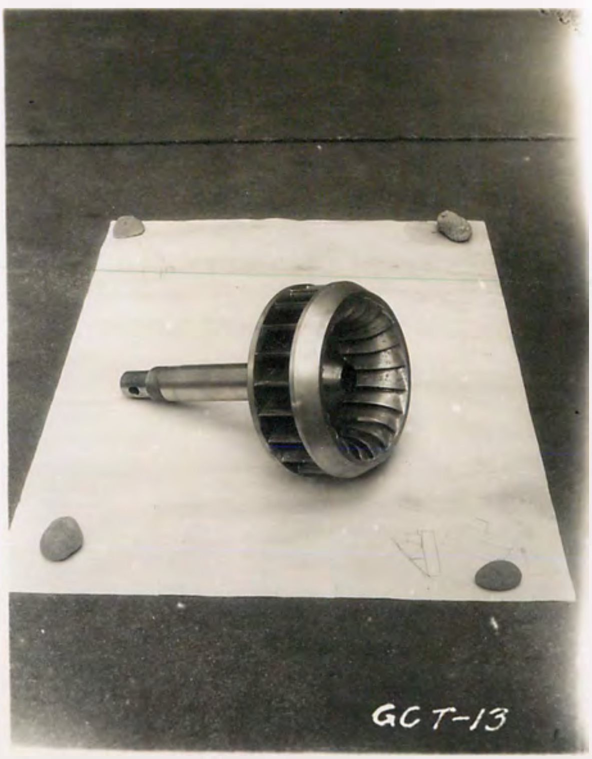
TURBINE RUNNER MOLD

GRAND COULEE DRAFT TUBE MODEL TESTS

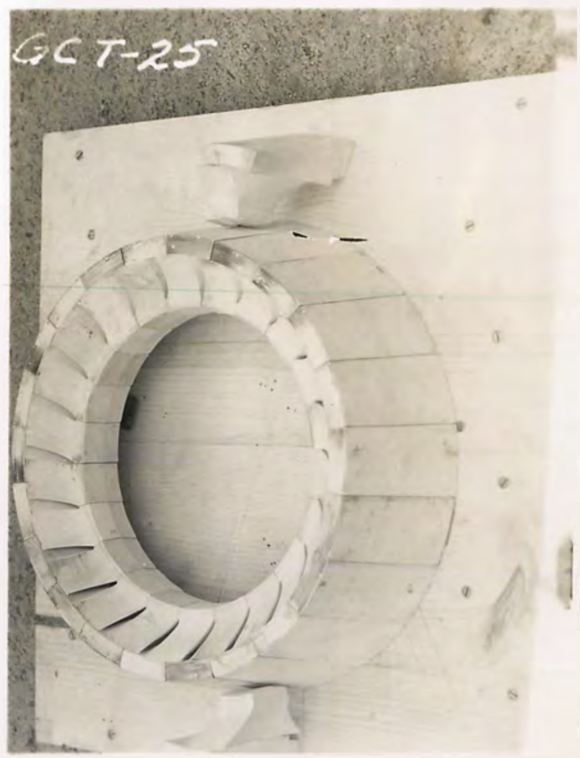
Runner mold



Runner just out of mold



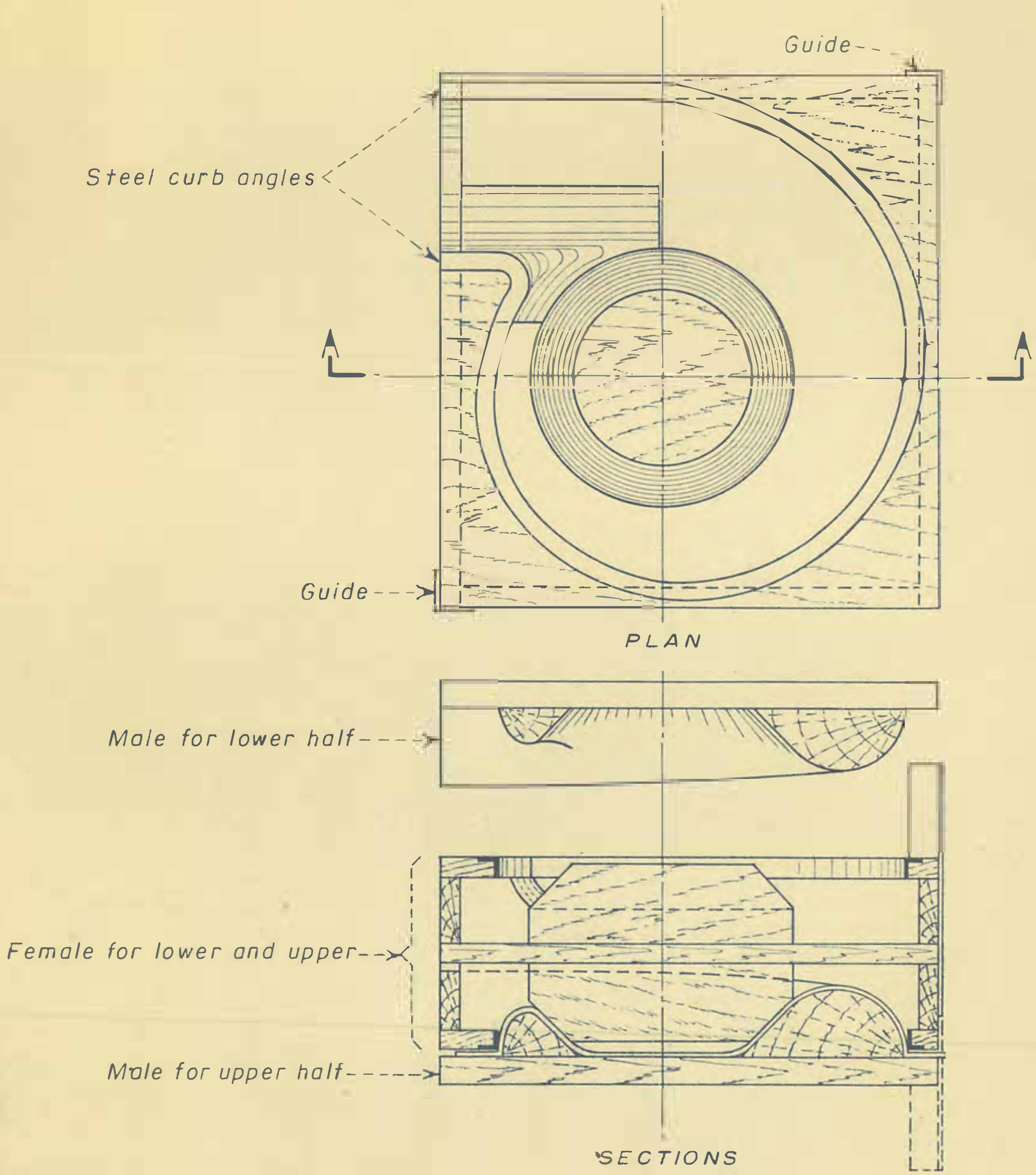
Finished Runner



Speed Ring Mold

Before pouring the metal the mold was heated up to about 300° F. to prevent chilling the metal. The cover plate was then heated up to slightly above the melting point of the metal. The metal was poured slowly and continuously until the mold was full. More heat was then applied to the cover plate by means of a blow torch and a fan was turned on the bottom of the mold for cooling. The cooling caused the runner band to solidify while the metal in the vanes and the runner crown remained molten. As the cooling progressed upward through the mold the runner vanes solidified. Shrinkage during the solidifying action made it necessary to add more molten metal. Finally the cover plate was allowed to cool along with the runner crown. The resulting casting is shown in figure 2A. The fins were sheared off with a pocket knife and the casting machined. Some trimming and scraping at the discharge edge of the runner vanes was necessary. Otherwise the vanes and water passages were so smooth that no polishing was necessary. The finished turbine runner with shaft attached is shown in figure 2A.

5. Casting the speed ring. The method and technique used in casting the speed ring were the same as used in casting the turbine runner. The mold was of cast iron machined inside only. The patterns for the upper and lower shrouds were also of cast iron. The technique used to cast the speed ring was a duplication of that used for the runner. In spite of all the precautions taken there were some slight defects in some of the vanes which had to be patched up by soldering. This serves to emphasize the importance of temperature control in pouring such castings; figure 20 of the report shows the finished casting of the No. 1 speed ring. The cores for this casting, with two extras, are shown stacked, bottom side up in figure 2A. The complete mold is shown in half plan and section in the upper drawing, figure 1A.



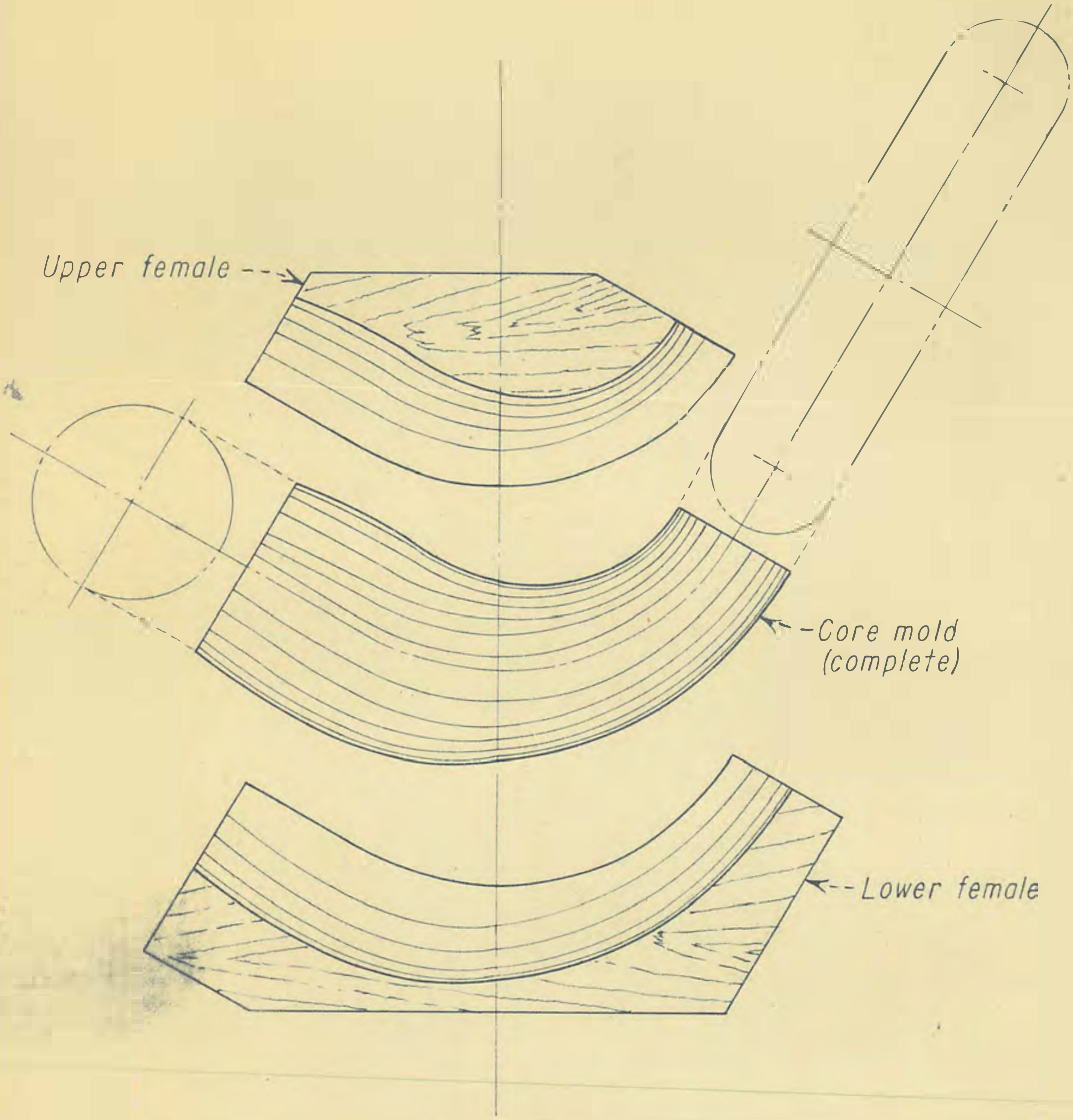
DRAWING AND FORMING MOLD FOR
PLASTIC MODEL SCROLL CASE

GRAND COULEE DRAFT TUBE MODEL TESTS

6. <u>Scroll-case and draft-tube models.</u> The scroll-case and draft-tube models, being made of transparent plastics, necessitated methods, skills, and techniques from an entirely different field. Instead of casting methods, forming and drawing methods had to be used. The material selected for the scroll case and draft tubes was a plastic known as "Pryalin". This material was selected because of its transparency, toughness, and workability at reasonably low temperatures. This material, however, had some objectionable characteristics. When overheated it turned an amber color and bubbles formed in its interior. Also it would shrink and lose much of its transparency, within a few months after forming. Later, it was found that "Plexiglas," though somewhat more brittle, has better optical qualities, less shrinkage, and does not discolor with age.

Transparent plastics had been used for visual flow studies in the laboratory before this time (1935) but this was the first time an attempt had been made to form the material into complicated shapes by means of forming tools. The scroll case was made in two sections, the separation being in the plane of symmetry. The molds, therefore, had to be for right- and left-hand spirals. The female halves were made in one piece with the left-hand matrix on one side and the right-hand matrix on the other, as shown in figure 3A. This saved material and storage space. The male halves were made separately. Provisions were made on the female for guiding the male half into position. The tool was so made that it would form bolting flanges on the parts.

A sheet of pyralin 1/10-inch thick was placed between the forms and the assembly placed in an oven and heated to about 260° F. It was soon found that this was the wrong approach as an edge of the pyralin near one of the electric heater elements in the oven ignited before the forming mold reached the proper temperature for forming.



DRAWING AND FORMING MOLD FOR
PLASTIC MODEL DRAFT TUBE ELBOW

GRAND COULEE DRAFT TUBE MODEL TESTS

It was then decided to place the forming mold in the oven without the pyralin until it reached the proper temperature for forming. The pyralin was then inserted in the mold and left until it, too, reached the temperature of the forming mold. The two halves of the forming mold were then pressed together, clamped, and removed from the oven to cool. As the pyralin sheets were not wide enough to form the entire section at one pressing it was necessary to form the parts in two pieces and join the pieces together, using acetone to "weld" the joint. The joint was reinforced by means of thin pyralin strip. The method of joining pieces together was later improved by one of the craftsmen. The pieces to be joined were beveled and a triangular strip of pyralin, which had been soaked in acetone until soft, was pressed into the "V" between the two beveled edges and allowed to harden. The joint made in this manner had the appearance of a butt weld between two sheets of steel and was remarkably strong. Fillets to reinforce the flanges were also made in this manner.

The draft-tube models were made in two sections, flanged and bolted together. The drawing and forming mold for the elbow section is shown in figure 4A. This section was very troublesome to make as there was considerable drawing of the material in forming. This resulted in wrinkling and the mold had to be very accurately made in order to press out the wrinkles. A study of the photographs on figures 10, 11, and 12 of the report show the evidence of this wrinkling in the elbows of the models. The mold for making the horizontal section of the draft-tube model is not shown as it was comparatively simple, being mostly a matter of bending the pyralin sheets with very little drawing in the forming process.

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