UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

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HYDRAULIC LABORATORY REPORT NO. 180

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HYDRAULIC STUDIES FOR THE DESIGN OF THE TUBE VALVES IN THE OUTLETS IN THE SHASTA DAM CENTRAL VALLEY PROJECT, CALIFORNIA

By

D. J. HEBERT

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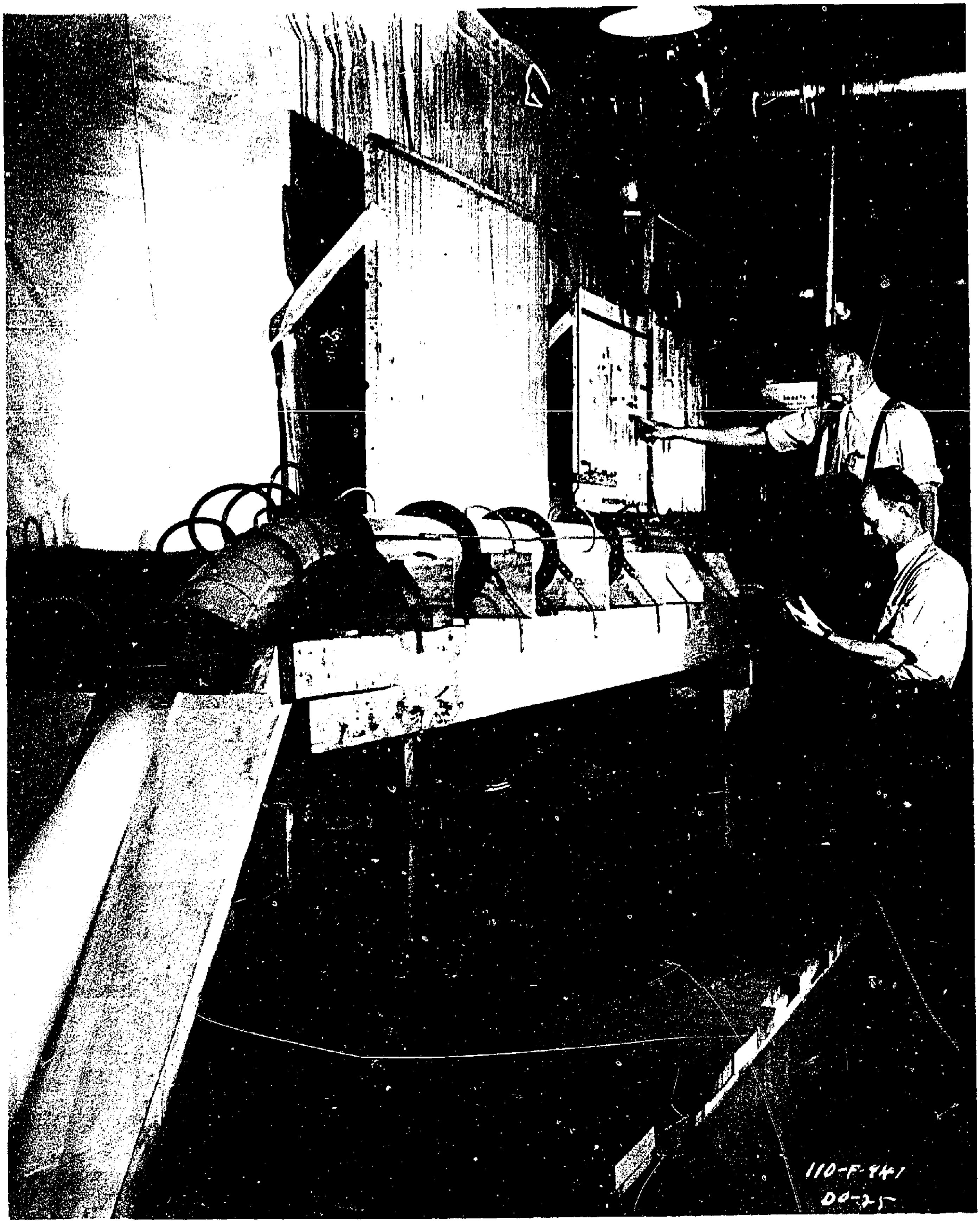
Denver, Colorado
August 7, 1945



6-INCH MODEL OF SHASTA TUBE VALVE AND CONDUIT

CONTENTS

V - DETAILED STUDIES LEADING TO FINAL DESIGN

VI - AIR STUDIES

VII - STUDIES OF FRIANT NEEDLE AND TUBE VALVES

VIII - COMPARISON OF 6- AND 20-INCH MODEL TESTS

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FOREWORD

In August 1939 the problem of determining the efficiency of a new type of control, known as the tube valve, to be used on the 102-inch outlets in Shasta Dam was assigned to the hydraulic laboratory. Subsequent testing proved the inadequacy of the design which led to a comprehensive program of studies on a model in the Denver hydraulic laboratory to a scale of 1:17 and a model in the Arizona canyon wall outlet house at Boulder Dam to a scale of 1 to 5.1 from which a design was evolved which met the original requirements without including the adverse pressure conditions of the original design.

The valves discussed in this report were designed under the direction of W. C. Beatty and P. A. Kinzie. The laboratory investigations, both in Denver and at Boulder Dam, from August 1939 to August 1940, were conducted by C. W. Thomas, with the particular assistance of D. M. Lancaster, R. A. Goodpasture, and A. N. Smith, and others. In August 1940 Mr. Thomas was called to active military duty and Mr. Lancaster continued the studies until August 1941, when he reported for active military duty. The writer, who had followed the progress of the studies throughout, completed them and prepared this report with the assistance and under the direction of J. E. Warnock, engineer in charge of the hydraulic laboratory. Particular credit is due to B. H. Staats, who prepared the design of the various models, and to J. N. Bradley, who analyzed the data on air requirements which developed in the final design.

The electric analogy studies described in Chapter III were made by J. E. Soehrens, under the supervision of J. H. A. Brahtz, and K. B. Keener, engineer in charge of design. His report dated January 11, 1940, has been incorporated intact except for modification of the introductory paragraph and section headings.

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

Branch of Design and Construction
Engineering and Geological Control
and Research Division
Denver, Colorado
August 7, 1945

Laboratory Report No. 180
Hydraulic Laboratory
Compiled by: D. J. Hebert
Reviewed by: J. E. Warnock

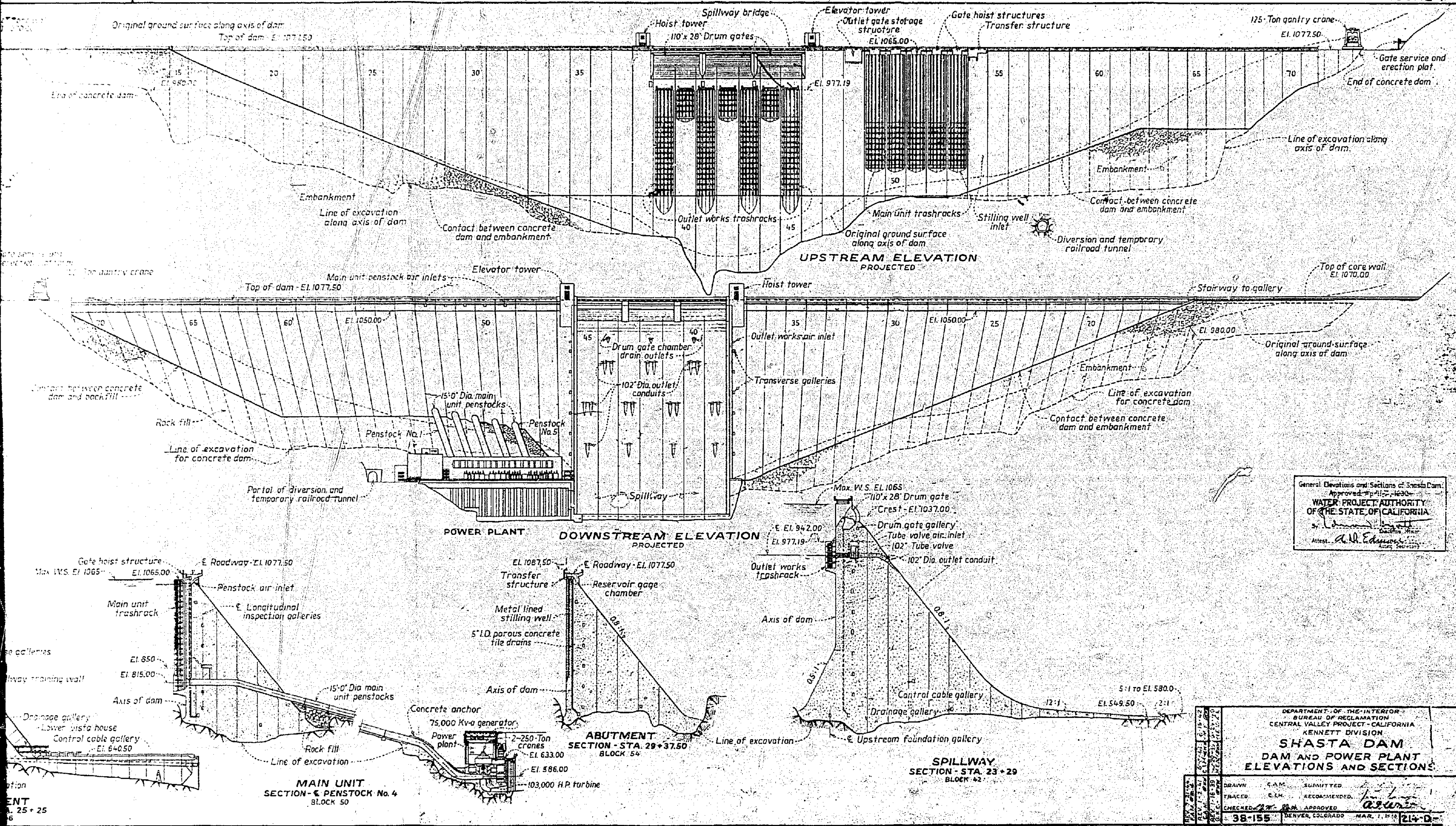
Subject: Hydraulic studies for the design of the tube valves in the outlets in Shasta Dam - Central Valley Project, California.

INTRODUCTION AND SUMMARY

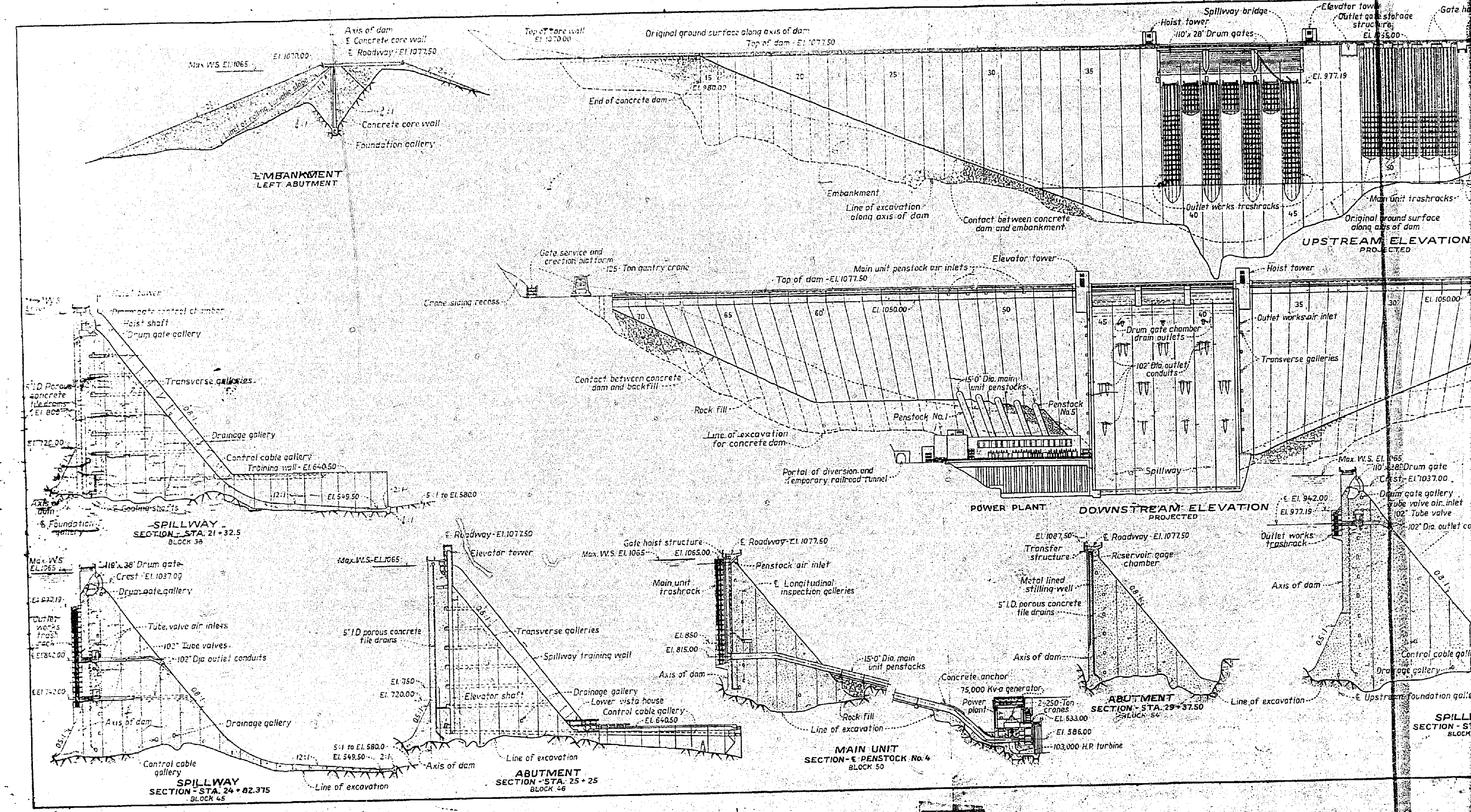
1. Purpose of investigation. The Shasta Dam being built on the Sacramento River, nine miles above Redding, California, is designed as a multipurpose dam to regulate the flow of the river for flood control, irrigation storage, and power generation. The release of stored water for flood storage evacuation and consumptive demand downstream will be primarily through the powerhouse turbines. Releases in excess of the capacity of the turbines will be made through eighteen 102-inch outlets placed at three elevations; four outlets in the lower tier at elevation 742, eight in the intermediate tier at elevation 842, and six in the upper tier at elevation 942, as shown in section and elevation on figure 1. The general design of these outlets follows quite closely that installed in Grand Coulee Dam except for length and method of control. Because of the limitations of the ring-seal ring-follower gates as installed in the Grand Coulee Dam, particularly the necessity of operating at only the full open or full closed position due to adverse pressure conditions downstream from the gate, a regulating valve in the conduit near the face of the dam was considered. The particular regulating valve is known as a tube valve. Fundamentally, it is a needle valve with the downstream tip removed.

Since no previous studies of valves discharging in a closed conduit were available, the hydraulic laboratory was assigned the

FIGURE 1



UPSTREAM ELEVATION
PROJECTED
Original ground surface along axis of dam
Top of dam - El. 1077.50
Hoist tower
Spillway bridge
110' x 28' Drum gates
Elevator tower
Outlet gate storage structure
El. 1065.00
Gate hoist structures
Transfer structure
125-Ton gantry crane
El. 1077.50
Gate service and erection plat.
End of concrete dam
El. 977.19
Line of excavation along axis of dam.
Embankment
Contact between concrete dam and embankment
Line of excavation along axis of dam
Outlet works trashracks
Main unit trashracks
Stilling well inlet
Original ground surface along axis of dam
Diversion and temporary railroad tunnel
DOWNSTREAM ELEVATION
PROJECTED
Elevator tower
Main unit penstock air inlets
Top of dam - El. 1077.50
Hoist tower
Top of core wall El. 1070.00
Stairway to gallery
El. 1050.00
Drum gate chamber drain outlets
Outlet works air inlet
102" Dia. outlet conduits
Transverse galleries
15'-0" Dia. main unit penstocks
Penstock No. 1
Penstock No. 5
El. 980.00
Original ground surface along axis of dam
Embankment
Line of excavation for concrete dam
Contact between concrete dam and embankment
Contact between concrete dam and backfill
Rock fill
Line of excavation for concrete dam
Portal of diversion and temporary railroad tunnel
Spillway
POWER PLANT
MAIN UNIT
SECTION - ℄ PENSTOCK No. 4
BLOCK 50
Gate hoist structure
℄ Roadway - El. 1077.50
Max. W.S. El. 1065
El. 1065.00
Penstock air inlet
Main unit trashrack
℄ Longitudinal inspection galleries
El. 850
El. 815.00
Axis of dam
15'-0" Dia. main unit penstocks
Drainage gallery
Lower vista house
Control cable gallery
El. 640.50
Rock fill
Line of excavation
Concrete anchor
75,000 Kv-a generator
Power plant
2-250-Ton cranes
El. 633.00
El. 586.00
103,000 H.P. turbine
ABUTMENT
SECTION - STA. 29+37.50
BLOCK 54
El. 1087.50
℄ Roadway - El. 1077.50
Transfer structure
Reservoir gage chamber
Metal lined stilling well
5" I.D. porous concrete tile drains
Axis of dam
Line of excavation
SPILLWAY
SECTION - STA. 23+29
BLOCK 42
Max. W.S. El. 1065
110' x 28' Drum gate
Crest - El. 1037.00
Drum gate gallery
Tube valve air inlet
102" Tube valve
102" Dia. outlet conduit
℄ El. 942.00
El. 977.19
Outlet works trashrack
Axis of dam
Control cable gallery
Drainage gallery
℄ Upstream foundation gallery
5:1 to El. 580.00
El. 549.50
General Elevations and Sections of Shasta Dam.
WATER PROJECT AUTHORITY OF THE STATE OF CALIFORNIA
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL VALLEY PROJECT - CALIFORNIA
KENNETT DIVISION
SHASTA DAM
DAM AND POWER PLANT
ELEVATIONS AND SECTIONS
DENVER, COLORADO
38-155
214-D-
214-D-711



EMBANKMENT
LEFT ABUTMENT
Axis of dam
℄ Concrete core wall
℄ Roadway - El. 1077.50
Max W.S. El. 1065
El. 1070.00
Concrete core wall
Foundation gallery
Top of core wall El. 1070.00
Original ground surface along axis of dam
Top of dam - El. 1077.50
End of concrete dam
El. 980.00
Embankment
Line of excavation along axis of dam
Contact between concrete dam and embankment
Spillway bridge
Hoist tower
110' x 28' Drum gates
Elevator tower
Outlet gate storage structure
El. 977.19
Outlet works trashracks
Main unit trashracks
Original ground surface along axis of dam
UPSTREAM ELEVATION
PROJECTED
Gate service and erection platform
125-Ton gantry crane
Crane siding recess
Main unit penstock air inlets
Top of dam - El. 1077.50
El. 1050.00
Elevator tower
Hoist tower
Outlet works air inlet
Drum gate chamber drain outlets
102" Dia. outlet conduits
Transverse galleries
Contact between concrete dam and backfill
Rock fill
15'-0" Dia. main unit penstocks
Penstock No. 1
Penstock No. 5
Line of excavation for concrete dam
Portal of diversion and temporary railroad tunnel
Spillway
POWER PLANT
DOWNSTREAM ELEVATION
PROJECTED
Hoist tower
Hoist shaft
Drum gate gallery
Transverse galleries
5' I.D. Porous concrete tile drains
Drainage gallery
Control cable gallery
Training wall - El. 640.50
El. 549.50
5:1 to El. 580.0
Axis of dam
℄ Foundation gallery
SPILLWAY
SECTION - STA. 21 + 32.5
BLOCK 38
Max. W.S. El. 1065
110' x 28' Drum gate
Crest - El. 1037.00
Drum gate gallery
Tube valve air inlets
102" Tube valves
102" Dia. outlet conduits
Outlet works trash rack
Axis of dam
Drainage gallery
Control cable gallery
El. 580.0
El. 549.50
Line of excavation
SPILLWAY
SECTION - STA. 24 + 82.375
BLOCK 45
℄ Roadway - El. 1077.50
Elevator tower
Max. W.S. - El. 1065
5' I.D. porous concrete tile drains
Transverse galleries
Spillway training wall
El. 750
El. 720.00
Elevator shaft
Drainage gallery
Lower vista house
Control cable gallery
El. 640.50
Line of excavation
Axis of dam
ABUTMENT
SECTION - STA. 25 + 25
BLOCK 46
Gate hoist structure
℄ Roadway - El. 1077.50
Max. W.S. El. 1065
El. 1065.00
Penstock air inlet
Main unit trashrack
℄ Longitudinal inspection galleries
El. 850
El. 815.00
Axis of dam
15'-0" Dia. main unit penstocks
Rock fill
Line of excavation
Concrete anchor
75,000 Kv-a generator
Power plant
2-250-Ton cranes
El. 533.00
El. 586.00
103,000 H.P. turbine
MAIN UNIT
SECTION - ℄ PENSTOCK No. 4
BLOCK 50
El. 1087.50
℄ Roadway - El. 1077.50
Transfer structure
Reservoir gage chamber
Metal lined stilling well
5' I.D. porous concrete tile drains
Axis of dam
ABUTMENT
SECTION - STA. 29 + 37.50
BLOCK 54
Line of excavation
Max. W.S. El. 1065
110' x 28' Drum gate
Crest - El. 1037.00
℄ El. 942.00
El. 977.19
Drum gate gallery
Tube valve air inlet
102" Tube valve
102" Dia. outlet conduit
Outlet works trashrack
Axis of dam
Control cable gallery
Drainage gallery
℄ Upstream foundation gallery
SPILLWAY

problem of the investigation of the hydraulic characteristics. The scope of the laboratory studies was circumscribed by the request that they be made to determine a design of tube valve which will operate satisfactorily at all openings with aeration, if necessary, and with a sufficiently high coefficient of discharge that the downstream portion of the conduit would flow full at full valve opening. To insure against damage by pitting due to cavitation, it was assumed that at no point in the prototype valve and conduit should the pressure be less than 25 feet below atmospheric.

2. Summary and conclusions. Preliminary tests on the 1 to 17 model of the original tube valve design proved conclusively that a valve discharging into a closed conduit must be provided with adequate air relief for operation at partial openings and that the original design could not be revised to perform satisfactorily at any opening.

A new design of valve, characterized by its long slim shape and referred to loosely as the "Shasta Tube Valve", was developed specifically for operation in a closed conduit. This valve when fully opened had a discharge coefficient high enough to fill the conduit under pressure. Despite extensive development of air relief measures, a valve of this type located in the lower tier under maximum head would be inoperative over nearly 40 percent of its range of opening because of the presence of subatmospheric pressures conducive to cavitation erosion. For heads less than the maximum of 322 feet the inoperative range is reduced to some 7 percent for a head of 222 feet which corresponds to the head on the intermediate tier of valves for maximum reservoir surface elevation.

Model tests on a 20-inch diameter valve (1 to 5.1 scale) of the Shasta Tube Valve type were conducted in the Arizona valve house at Boulder Dam and the results confirmed those obtained in the tests made with the 6-inch diameter valve (1 to 17 scale) in the Denver laboratory. The quantity of air required to relieve the negative pressures created by the condition of a valve discharging into a conduit was measured in both models and the sizes of air piping required to supply each prototype valve were determined by using the criterion that air velocities should not exceed 300 feet per second in the interest of quiet operation.

Another series of tests was conducted using two valves, a tube valve and a needle valve, which had been developed by a separate model study for free discharge conditions at Friant Dam. The tests proved that, with air relief provided at a point immediately downstream from either of the valves, they were as satisfactory at all openings, from the standpoint of pressures, for conduit operation, as for free discharge operation. The only change due to operation of these valves in a conduit was a minor decrease in pressure at the end of the valves for supplying the required flow of air through the air relief piping. The maximum amount of air relief required for either valve was approximately equal to the amount required by the Shasta tube valve so the size of the air supply piping would be the same for all three valve designs.

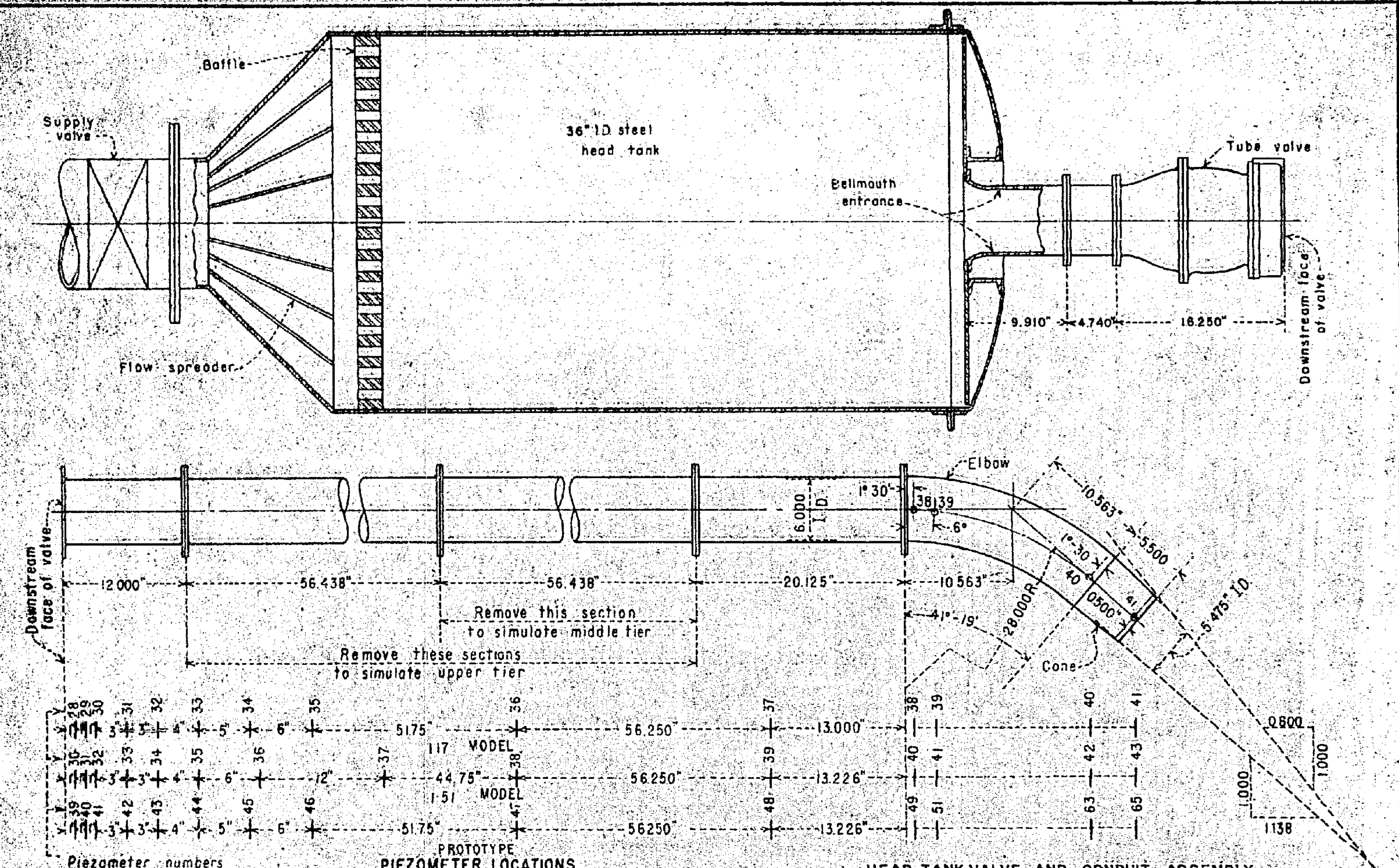
The possibilities of damage, to the portion of the metal-lined conduit downstream from the valve, by corrosion due to the large amount of air mixed with the water at partial valve openings, which at times may reach as high as 215 percent more air than water, were not investigated in the studies described in this report. The accelerated corrosion which may occur is, however, believed to be a definite factor in any consideration of valves for operation in closed metal-lined conduits and a check of field conditions from time to time is recommended to establish its importance.

3. Laboratory apparatus. During the course of model studies for the detailed design of the 102-inch outlets for Grand Coulee and Shasta Dams, equipment and models had been designed and installed in the Denver laboratory to an extent that when the problem of the

Shasta tube valve was assigned, it was necessary to construct only the model of the valve to complete a model of one outlet in the lower tier at Shasta Dam. Since all previous experiments had been made using a diameter of 6 inches for the model conduit, the scale of the new model was established by the ratio of 6 to 102 or 1 to 17. The model was then complete from the bell-mouth entrance at the reservoir to the deflecting elbow at the downstream face of the spillway (figure 2). The prototype reservoir was assimilated by a cylindrical pressure tank, 36 inches in diameter, equipped with baffles to give uniform steady flow. This tank was supplied through a venturi meter and regulating gate valve by a 12-inch centrifugal pump direct-connected to a variable speed, 100-horsepower motor. The desired head was obtained by adjustments of the variable speed motor and the regulating valve.

The pressure conditions within the model valve were measured by piezometers installed at intervals along the boundary surfaces so that complete pressure profiles could be constructed from the test data. Piezometer holes 1/16-inch in diameter were drilled normal to the surface of the flow boundaries and brought through the valve body by copper tubes. Rubber tubing was used to connect the copper leads in the valve to a bank of open manometers. Pressures too high for the open manometers were measured on a mercury differential gauge, and subatmospheric pressures were measured with either a water or a mercury differential gauge depending on the magnitude of the vacuum. Discharges were measured by a venturi meter which had been calibrated volumetrically in the laboratory. Measurements of the air demand were made with a 4-inch anemometer supplemented by a pitot tube.

As the tests progressed on the 1:17 model in the Denver laboratory, it became increasingly apparent that confirmation tests were desirable. To make these confirmation tests, equipment was designed and installed in the Arizona canyon wall valve house at Boulder Dam. The 102-inch outlet conduit at Shasta Dam was represented by a model conduit with a diameter of 20 inches so that the scale ratio was 20 to 102 or 1 to



PIEZOMETER LOCATIONS

Dimensions for 1:51 model and prototype converted for comparison with 1:17 model.

HEAD TANK, VALVE AND CONDUIT ASSEMBLY FOR 6-INCH MODEL OF SHASTA DAM OUTLET TUBE VALVE

SHOWING LOCATIONS OF PIEZOMETERS IN DOWNSTREAM CONDUIT IN 6- AND 20-INCH MODELS AND IN PROTOTYPE.

FIGURE 2

5.1. The model was installed on a platform in the space over the needle valve discharge guides (figure 3). A heavy cast steel reducer was attached to the A-1 valve in place of the seating ring. The larger end of this reducer was shaped so the needle seated as usual while the smaller end was designed to fit a 24-inch outside diameter pipe. From the reducer the 24-inch line was brought up to the platform through a 60-degree bend, two 45-degree bends and a riser pipe at a 45-degree slope. At this point, the line was expanded to 30 inches outside diameter with a flow straightener immediately following the expander. After a run of 48½ feet, the 30-inch pipe was reduced to a 20-inch inside diameter pipe in a length of 4 feet. The drop in head across this reducer was calibrated against discharge by a pitometer traverse, and then used to measure discharge during the tests. The reducer was followed by approximately 34 feet of 20-inch inside diameter pipe which connected to the model proper. The model of the river outlet downstream from the valve was designed with three lengths of pipe of the proper lengths so that the valve could be shifted one or two lengths downstream to assimilate the intermediate or the upper tier. The deflecting elbow of the model protruded through the end of the valve house and discharged against the canyon wall. Photographs of the 1 to 5.1 model are shown in figure 4.

Several changes were made in this arrangement as the testing program progressed to obtain the required range of heads and discharges. Most of the difficulties arose from the necessity of dissipating the energy from the available head of 370 feet to the required model head of 63.3 feet. The 84-inch needle valve proved to be unsatisfactory as a regulating valve because of the great difference in size between it and the 20-inch model tube valve. To obtain any appreciable reduction in head, the 84-inch valve had to be operated at very small openings, and with the low pressures created by discharging into a closed conduit, the needle could not be held steady and tended

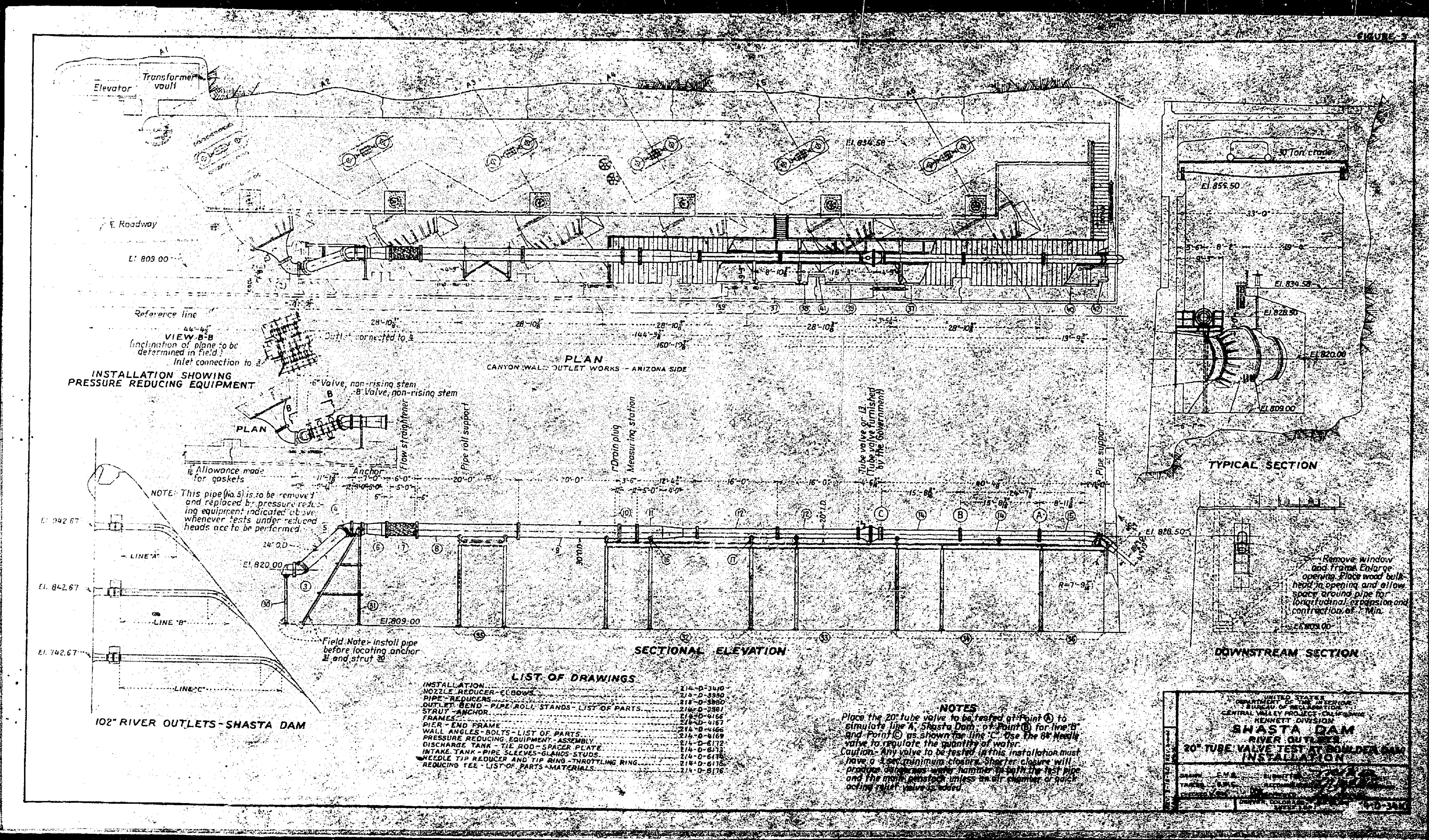
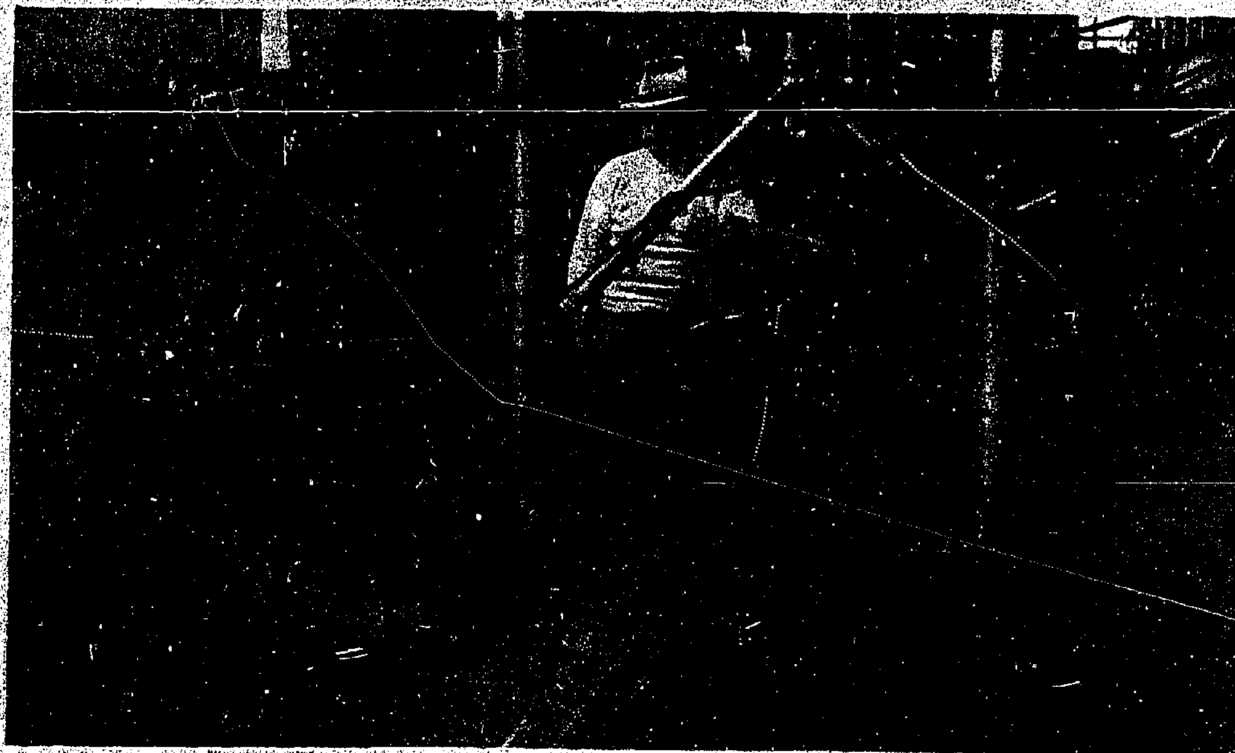
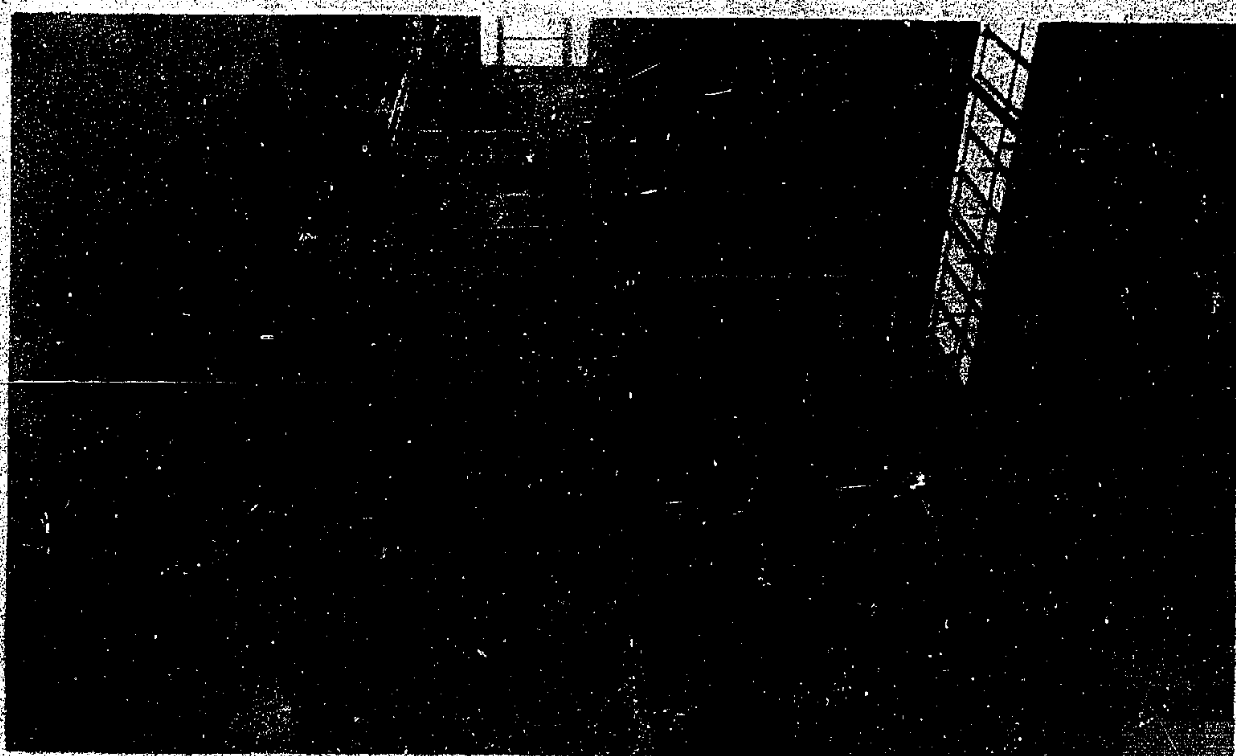


FIGURE 3
Elevator
Transformer vault
℄ Roadway
El. 809.00
El. 834.50
Reference line
VIEW B-B
(Inclination of plane to be determined in field.)
Inlet connection to 2
INSTALLATION SHOWING
PRESSURE REDUCING EQUIPMENT
Outlet connected to 4
PLAN
CANYON WALL OUTLET WORKS - ARIZONA SIDE
6" Valve, non-rising stem
8" Valve, non-rising stem
PLAN
1/16" Allowance made for gaskets
Anchor
Flow straightener
Pipe roll support
1" Drain plug
Measuring station
Tube valve or 13 (Tube valve furnished by the Government)
Pipe support
NOTE: This pipe (No. 5) is to be removed and replaced by pressure reducing equipment indicated above whenever tests under reduced heads are to be performed.
El. 942.67
LINE "A"
El. 842.67
LINE "B"
El. 742.67
LINE "C"
102" RIVER OUTLETS - SHASTA DAM
El. 820.00
El. 809.00
Field Note - Install pipe before locating anchor 31 and strut 30
SECTIONAL ELEVATION
El. 828.50
30 Ton crane
El. 855.50
33'-0"
El. 834.50
El. 828.50
El. 820.00
El. 809.00
TYPICAL SECTION
Remove window and frame. Enlarge opening. Place wood bulkhead in opening and allow space around pipe for longitudinal expansion and contraction of 1" Min.
El. 809.00
DOWNSTREAM SECTION
LIST OF DRAWINGS
INSTALLATION ... 214-D-3410
NOZZLE-REDUCER-ELBOWS ... 214-D-3950
PIPE-REDUCERS ... 214-D-3980
OUTLET BEND - PIPE ROLL STANDS - LIST OF PARTS ... 214-D-3981
STRUT-ANCHOR ... 214-D-6166
FRAMES ... 214-D-4167
PIER-END FRAME ... 214-D-4166
WALL ANGLES-BOLTS-LIST OF PARTS ... 214-D-4167
PRESSURE REDUCING EQUIPMENT-ASSEMBLY ... 214-D-6169
DISCHARGE TANK-TIE ROD-SPACER PLATE ... 214-D-6172
INTAKE TANK-PIPE SLEEVES-GLANDS-STUDS ... 214-D-6173
NEEDLE TIP REDUCER AND TIP RING-THROTTLING RING ... 214-D-6174
REDUCING TEE-LIST OF PARTS-MATERIALS ... 214-D-6176
NOTES
Place the 20" tube valve to be tested at Point A to simulate line "A", Shasta Dam, at Point B for line "B" and Point C as shown for line "C". Use the 84" Needle valve to regulate the quantity of water.
Caution:- Any valve to be tested in this installation must have a 4 sec. minimum closure. Shorter closure will produce dangerous water hammer in both the test pipe and the main penstock unless an air chamber or quick acting relief valve is added.
UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL VALLEY PROJECT - CALIFORNIA
KENNETT DIVISION
SHASTA DAM
RIVER OUTLETS
20" TUBE VALVE TEST AT BOULDER DAM
INSTALLATION
DENVER, COLORADO

FIGURE 4



Arrangement for test B8EA5



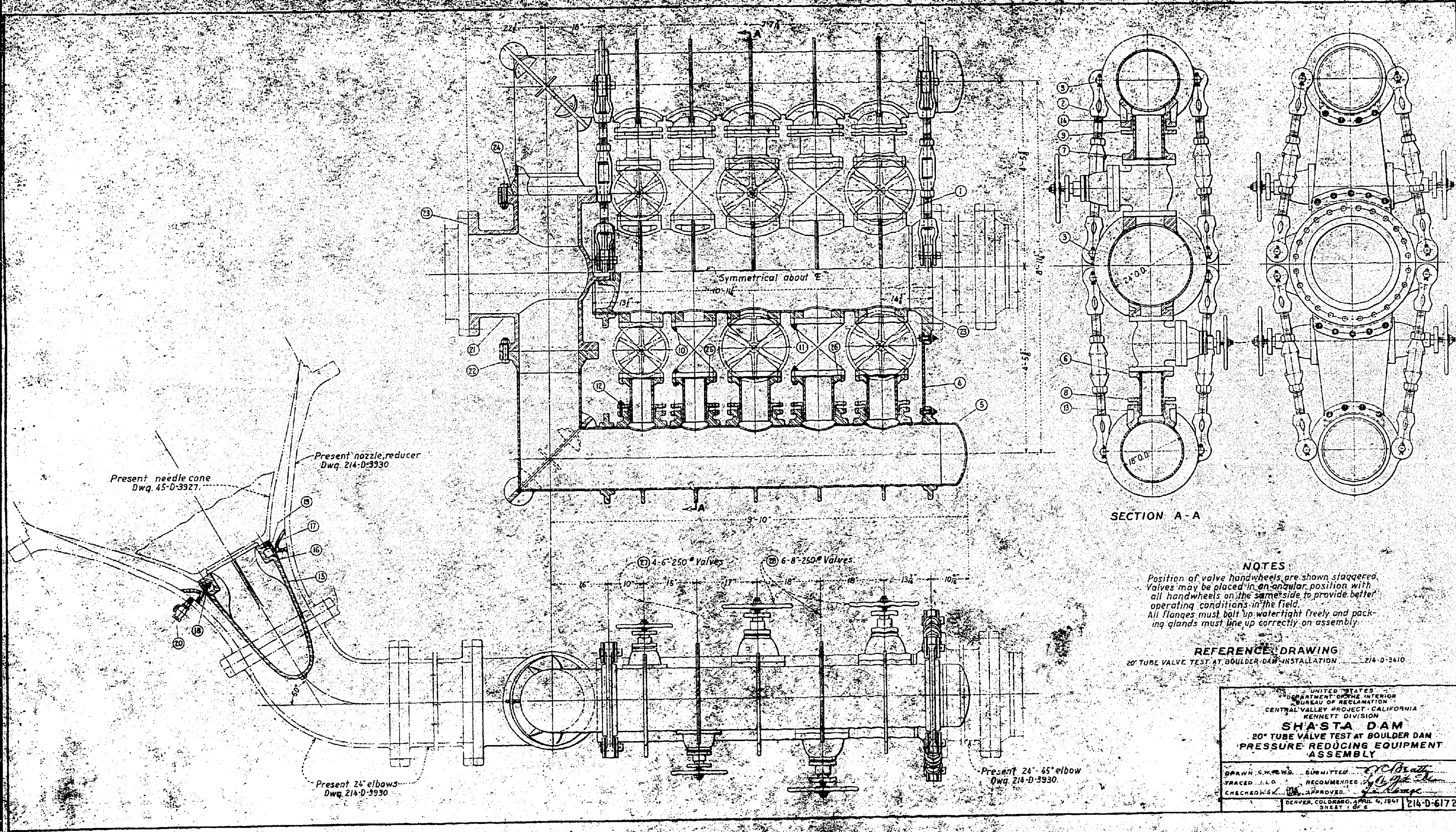
Arrangement for test B8EA512

INSTALLATION OF 20-INCH MODEL IN
ARIZONA CANYON WALL HOUSE AT BOULDER DAM

to slam shut. Three bypass lines with valves were welded into the 24-inch riser in an attempt to lower the head by diverting part of the discharge from the 84-inch valve, but the head was still too high. Some tests at heads below the required model head were obtained by supplying the model through an angle valve from the air manifolds of paradox gates A-2 and A-3. These manifolds were under full reservoir pressure when the gates were opened with the corresponding needle valves closed. A further change was made by installing a new bulbous tip on the needle, a constricting ring in the large reducer, and an energy dissipator in place of the 24-inch riser pipe. The new tip and constricting ring formed in effect a new smaller valve at the end of the 84-inch needle valve and maintained pressure on the needle so that it held any setting. The dissipator consisted of three pairs of 8-inch and two pairs of 6-inch gate valves discharging normally into a 24-inch central collecting pipe. The installation of the dissipator is shown on figure 3. Details of this arrangement are shown in figure 5. This arrangement gave satisfactory control of the head for the model testing. To assure the success of this energy dissipator, a model study was made of it before completion of design.

Measurements of the air demand were made with an intake orifice and a pitot tube in the first tests and later with individual flow nozzles in each of the three air supply lines.

4. Nomenclature and model scales. A system of test numbers was adopted in identifying the numerous tests with their respective valve shapes and data. The number corresponding to each test contains both numerals and letters and has anywhere from 2 to 7 places. The first place is a number indicating the valve contour, the second is a letter identifying the shape of the tube lip. The presence or absence of air relief is indicated by the presence or absence of the letter (A) in the third place. The fourth place indicates, by letter or by the absence of a letter, the manner in which air is admitted, with the exception of the letter (J) which appears in this place to


Symmetrical about ℄
24" O.D.
18" O.D.
SECTION A-A
Present needle cone
Dwg. 45-D-3927
Present nozzle, reducer
Dwg. 214-D-3930
(27) 4-6"-250# Valves
(28) 6-8"-250# Valves
Present 24" elbows
Dwg. 214-D-3930
Present 24"-45° elbow
Dwg. 214-D-3930
NOTES
Position of valve handwheels are shown staggered.
Valves may be placed in an angular position with all handwheels on the same side to provide better operating conditions in the field.
All flanges must bolt up watertight freely and packing glands must line up correctly on assembly.
REFERENCE DRAWING
20" TUBE VALVE TEST AT BOULDER DAM INSTALLATION 214-D-3410
UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL VALLEY PROJECT - CALIFORNIA
KENNETT DIVISION
SHASTA DAM
20" TUBE VALVE TEST AT BOULDER DAM
PRESSURE REDUCING EQUIPMENT
ASSEMBLY
DRAWN ..... SUBMITTED
TRACED ..... RECOMMENDED
CHECKED ..... APPROVED
DENVER, COLORADO, APRIL 4, 1941
SHEET 1 OF 5
214-D-6172

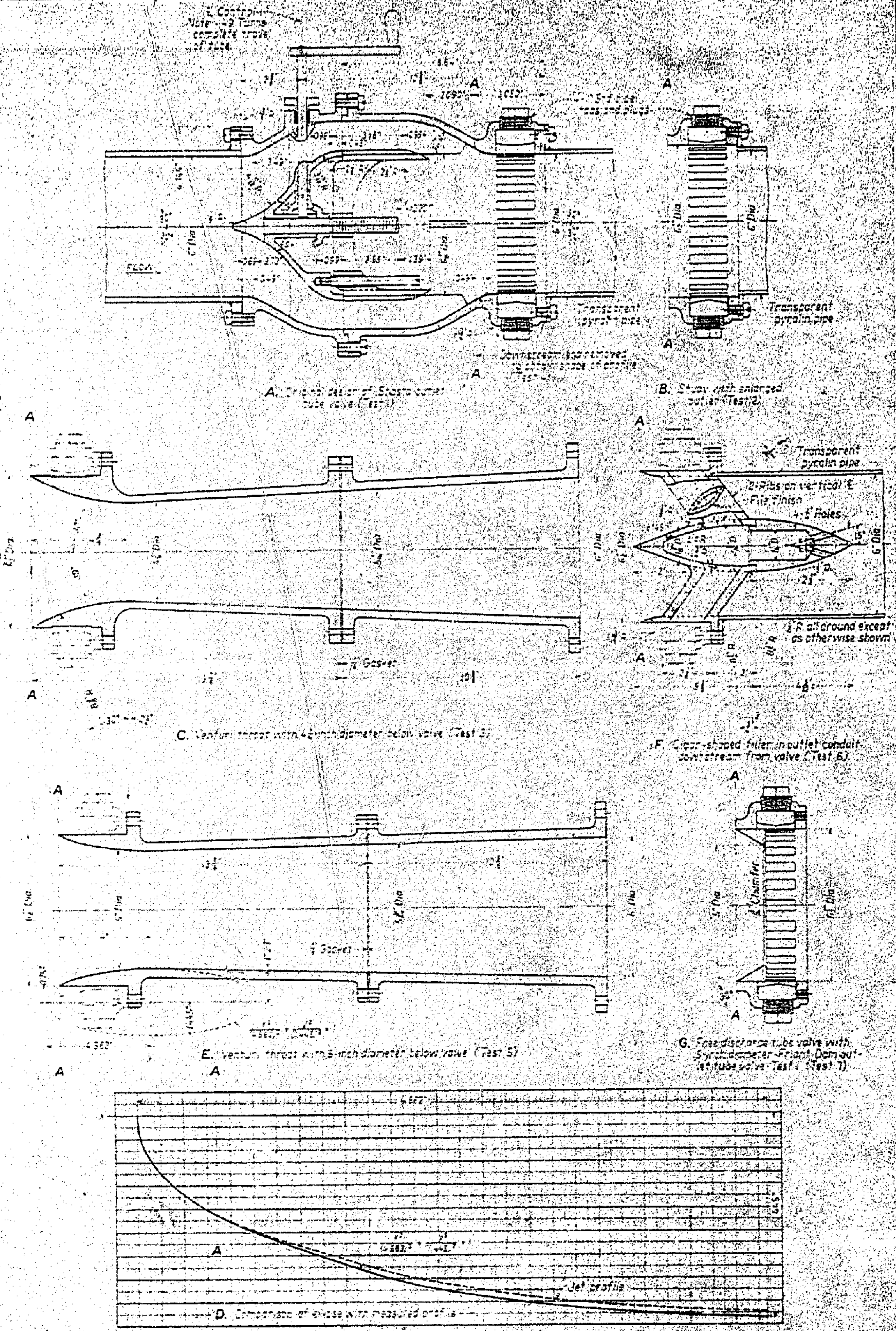
indicate the presence of the jet-centering guide. The absence of a letter in the fourth place indicates the use of holes; the letter (G) indicates a groove, and the letter (S) indicates a slot. The number terminates in a numeral which indicates the number in a series for which the conditions are identified by the preceding places. For the relatively few confirmation tests made with the 20-inch model at Boulder Dam, a (B) is placed in front of the number of the corresponding test with the 6-inch model.

II - STUDIES OF VARIATIONS OF ORIGINAL DESIGN

5. Poor hydraulic conditions of original design (Test 1). In the initial test, a 1:17 model of the original design (figure 6A) of the conduit tube valve was installed in the Denver laboratory testing apparatus with a length of conduit to represent one of the lower tier of outlets at Shasta Dam. In this original design, the valve was made comparatively short to minimize the necessary size of the operating chamber and to prevent disruption of the stress distribution within the concrete of the spillway section.

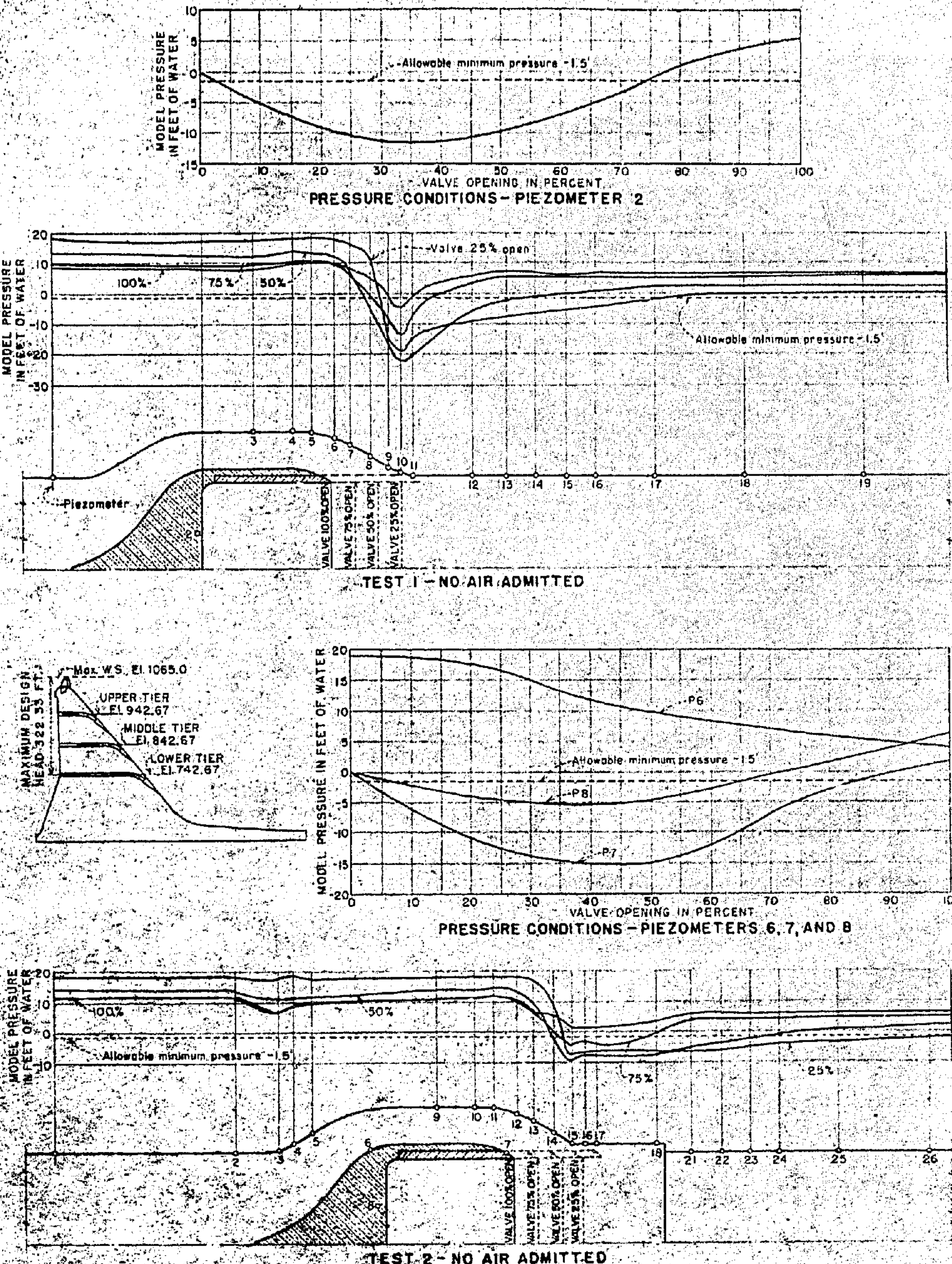
The results from this initial test were unsatisfactory to an extreme. Figure 7 shows the pressure gradients for various valve openings. At no opening were the valve and conduit free of subatmospheric pressures of a magnitude such that if transferred to prototype, would not only exceed the assumed allowable limit of -25 feet of water, but would have reached the vapor pressure of water where cavitation would have occurred with its accompanying destructive erosion by pitting. With a scale ratio of 1 to 17 and an assumed tolerable pressure of -25 feet of water, the allowable minimum on the 1 to 17 model was -1.5 feet of water. At 100 percent of the designed head, or the maximum head on the lower tier, figure 7 shows that the limit was exceeded at all valve openings on the curve tangent to the 6-inch conduit at the downstream end of the valve. In fact, during the tests in which the model valve was operated at the maximum available test head, the extent and magnitude of the low pressure was so severe at 50 percent valve opening that the transparent plastic conduit downstream from the valve collapsed.

Although this test provided strong evidence that a valve discharging into a conduit will set up severely low negative pressures unless air relief is provided, the assignment of the problem to the laboratory stipulated that air relief be provided only if necessary. In order to determine the degree of necessity for air relief, several



STUDIES OF VARIATIONS OF ORIGINAL DESIGN OF SHASTA OUTLET TUBE VALVE

FIGURE 7



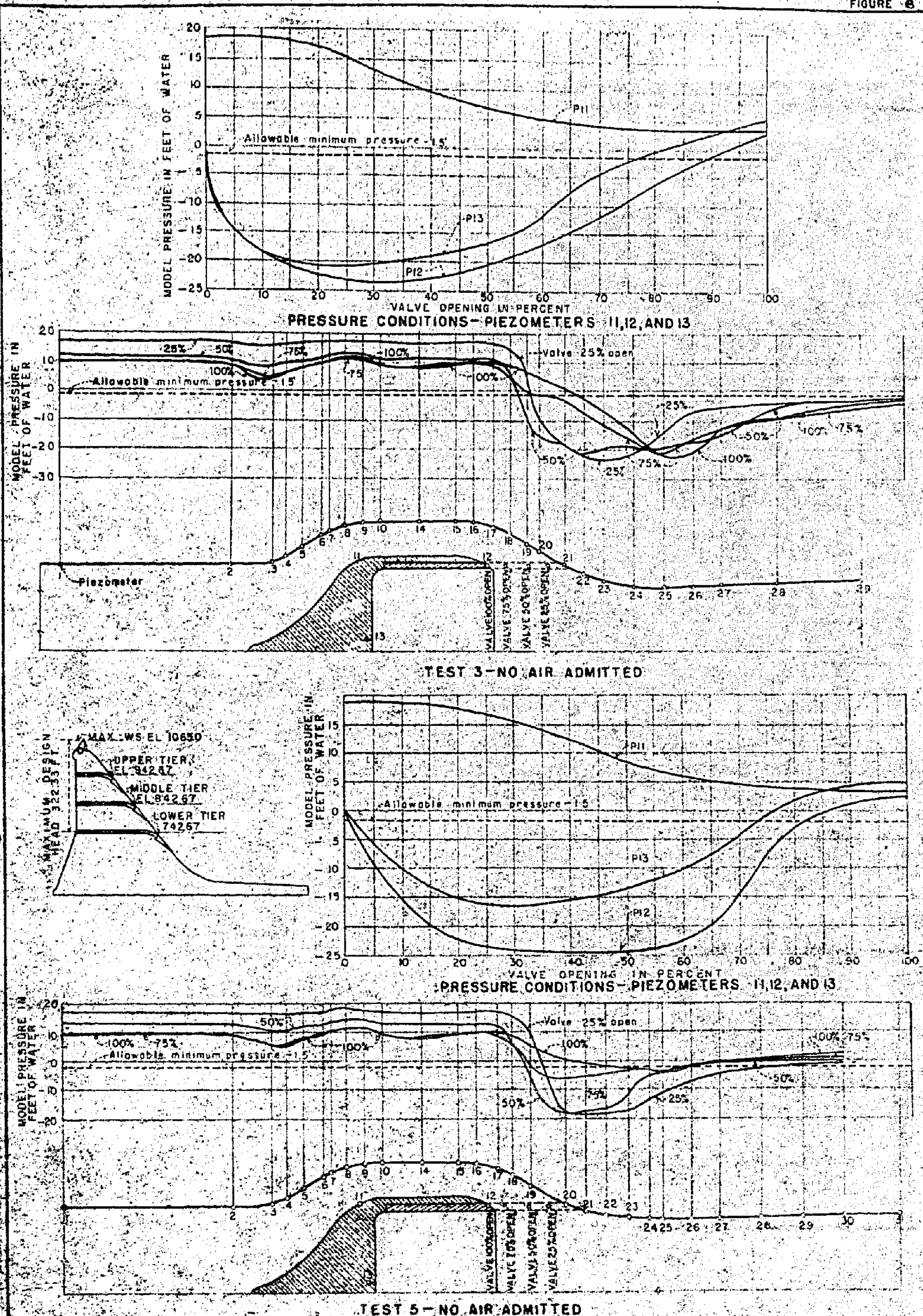
SHASTA DAM OUTLETS
PRESSURE CONDITIONS IN TUBE VALVES
FOR MAXIMUM HEAD ON LOWER TIER
MODEL SCALE 1:17

additional tests were made in an effort to devise a valve design which would operate without air relief. Each new test provided additional evidence that the need for air relief at partial valve openings could not be circumvented. Some of the methods tried will be described briefly to serve as a matter of record.

6. Study with enlarged outlet (Test 2). At the time, the apparent solution appeared to be a closer coincidence between the shape of the free jet from the valve and the shape of the conduit below the valve. The use of a venturi-shape appeared to be an approach, so construction was started on the shape later used in test 3. To install this new throat, the original valve model was bored to a diameter of $6\frac{1}{8}$ inches, which removed the original short-radius curve back to the point of curvature. While the new throat was being constructed, the downstream conduit was replaced and pressures observed on the model, as shown in figure 6B. This was not a practical arrangement because the valve seating surface was removed and complete closure could not be made, but the data obtained served to indicate the trend of future tests. The observed pressures (figure 7) were materially raised at 75 and 100 percent valve openings as the positive pressure immediately upstream from the critical zone was sufficient to allow the jet to fill the nozzle.

7. Venturi throat with $4\frac{1}{4}$-inch diameter below valve (Test 3). The installation of the $4\frac{1}{4}$-inch venturi throat in an attempt to obtain a coincidence between the jet and the boundary was a complete failure, as shown by the pressure gradients in figure 8, except for the information it furnished. The subatmospheric pressures extended over a longer length of conduit, and their intensity was not decreased. This was to be expected inasmuch as, according to Bernoulli's Theorem for a given discharge in a horizontal conduit, the area of cross-section is decreased, the velocity is increased, and the pressure decreased.

FIGURE 6



SHASTA DAM OUTLETS

PRESSURE CONDITIONS IN TUBE VALVES FOR MAXIMUM HEAD ON LOWER TIER

MODEL SCALE 1:17

8. Removal of downstream end of valve to determine shape of jet (Test 4). For any given orifice, the jet profile, through the vena contracta, is independent of the head. This fact, applied to the tube valve, means that the contracted section would remain constant for a particular valve opening. A change of the valve opening would change the orifice diameter and produce an entirely different jet diameter.

Since the $4\frac{1}{4}$-inch throat in test 3 had been the result of a pure guess, and test 3 showed the low pressures in the conduit to be due to the high velocities, in test 4 the downstream end of the test valve was removed at the point of curvature, section A-A, figure 6A, to permit measurement of the actual shape of the discharging jet. From those measurements, a new throat was designed, having a 5-inch diameter and a bell-mouth, as described by rotation of an ellipse $\frac{X^2}{(4.562)^2} + \frac{Y^2}{(1.445)^2} = 1$ about the center line of the conduit, as shown in figure 6E. A comparison of the profile described by the ellipse with the measured profile is shown in figure 6D. The profile of the jet was measured only at full opening of the valve since the first step in the design was to obtain a valve which would function properly at that opening. The profile was measured with a model head of 263.33 percent of the scaled head as the higher head produced a more stable jet.

9. Venturi throat with 5-inch diameter below valve (Test 5). The pressure gradient for the new nozzle was materially higher than previously obtained (figure 8). The reason the gradient drops below atmospheric pressure at an opening of 100 percent is due to the error in determining the actual contour of the jet. However, this test did indicate that progress was being made and, furthermore, that all possibilities of using the original valve had been exhausted. The conclusion was to design a longer and more streamlined valve.

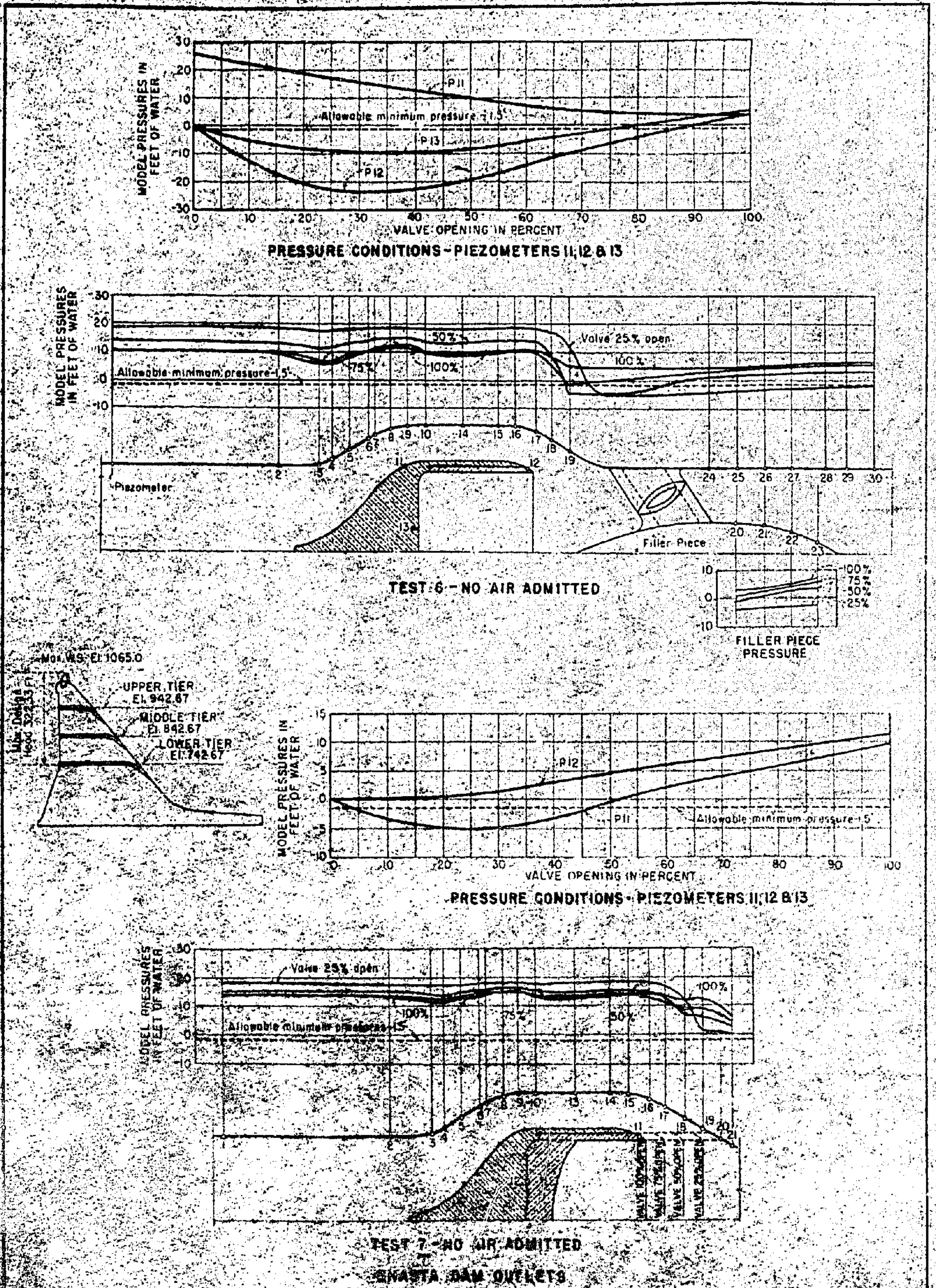
Time being an important factor, the problem of the new design was given to the photoelastic laboratory for study in the electric-analogy apparatus where changes could be made more easily than on the hydraulic model. Since this investigation necessarily considered only a

100 percent valve opening and the results apply in detail only to flow of an ideal fluid, a hydraulic model would be required to check the results and make additional studies. The electric-analogy studies are described in chapter III.

10. <u>Cigar-shaped filler in outlet conduit downstream from valve (Test 6)</u>. To force the stream against the wall of the conduit, a filler piece was proposed, as shown in figure 6F. Air was admitted at the downstream end of the filler through passages in the supporting vanes. The pressure intensity (figure 9) along the wall of the conduit at 75 and 100 percent openings was definitely improved, but the pressures on the lip of the tube and within the tube were adverse to an extreme. These undesirable pressures and the structural limitations led to the abandonment of further consideration of this idea.

11. <u>Free discharge tube valve with 5-inch diameter - Friant Dam outlet tube valve - Test 1 (Test 7)</u>. In the design of Friant Dam, ten large and two small outlets were provided to release water; four 110-inch valves in the Friant-Kern canal outlets, four 110-inch and two 18-inch valves in the river outlets, and two 91-inch valves in the Friant-Madera canal outlets. To reduce the costs of the controls on these outlets, it was proposed to design a valve requiring less material and less complicated control mechanism than required for a needle valve. The tube valve being studied for the conduit controls at Shasta Dam was suggested, only in this case the valves would be at the downstream end of the conduits and would discharge into the atmosphere instead of into the conduit; hence they are referred to as free-discharge tube valves. It was proposed that two of the river outlets and two of the Friant-Kern canal outlets be tube valves, and the other two in each case be needle valves. The objection to using the tube valve throughout was the objectionable spray at openings of less than 20 percent. It will probably be necessary to operate in that range to secure adequate regulation.

FIGURE 9



SHASTA DAM OUTLETS
PRESSURE CONDITIONS IN TUBE VALVES
FOR MINIMUM HEAD ON LOWER TIER

Since there were no standard designs available, model studies were made to determine whether any adverse pressures would occur which would cause cavitation and pitting, and to determine the discharge capacity in terms of coefficients. To expedite the testing, the original Shasta tube valve model was revised, as shown in figure 6G, by casting a solder insert in the $6\frac{1}{2}$-inch bore.

This preliminary test demonstrated that a tube valve may not be suitable for regulation at small openings because of the severe spray caused by the disintegration of the jet due to cross-flow. The pressures on the lip of the tube for valve openings of less than 50 percent opening were extremely low, as shown for piezometer 11 on figure 9, making the destruction of the lip practically an assured occurrence. The curvature of the lip caused a divergence of the passage between the lip of the tube and the body of the valve. It was this divergence which produced the severe subatmospheric pressures at piezometer 11 on the lip.

At large openings, a solid jet was formed; but as the valve was closed, the diameter of the jet became smaller until at approximately 15 percent, a single jet no longer formed. Instead, the cone of water from the ring between the tube and body met at a point below the valve and disintegrated into spray. Such spray would be undesirable in many prototype structures, especially in the vicinity of electrical equipment or in outlets where the spray may be carried over the walls and saturate the confining ground, as in the case of the Friant-Madera and Friant-Kern canals.

The coefficient of discharge in the equation $Q = CA\sqrt{2gH}$ was $C = 0.51$ in this test. This coefficient is based on the area of the conduit one diameter upstream from the upper flange of the valve, and $H = h_p + h_v$.

Subsequent tests on the free discharge tube valves, which are described in another report,* led to an improved design which was

* Hydraulic Laboratory Report No. Hyd. 163

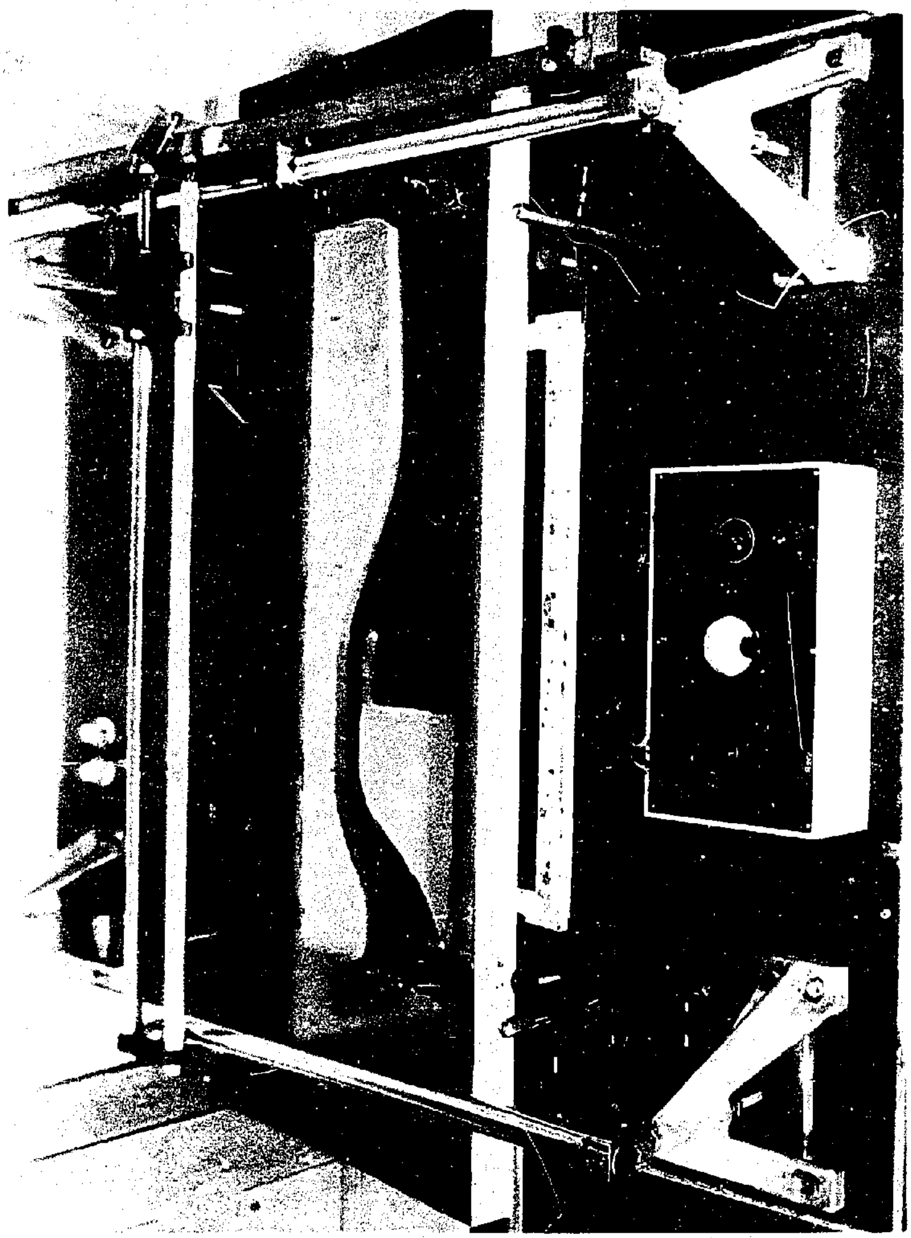
later tested as a valve in a conduit with the results described in chapter VII of this report.

III - ELECTRIC-ANALOGY STUDIES

12. Purpose of analogy studies. The purpose of these studies was to analyze, by the electric-analogy method, several proposed tube valve designs and to select in the shortest possible time the best of several proposed modifications.

A characteristic feature of valves, in general, is their tendency to cavitate. It has been found that this occurs wherever velocities are high and the pressure drops below atmospheric and approaches absolute zero. In the electric analogy, the study of the flow of electricity through a salt solution furnishes a relatively rapid means of predetermining velocities and pressures and arriving at a shape that should prove to be relatively free of low pressure areas when tested by a hydraulic model.

13. Description of apparatus. In the structure, because of the symmetry of the valve about the center line, all stream lines lie in planes containing the center line of the valve. The state of flow is, for this reason, the same in all sectors bounded by imaginary planes containing the center line. In the model shown in figure 10, a sector shape was effected by building the model on a sloping glass sheet. The valve shape was constructed of modeling clay, and made to conform to the proper curvature of the inside of the valve shell. A salt solution was brought to a featheredge to correspond to the center line of the valve. Electrodes were installed at the ends of the model, and an applied electric potential difference caused current to flow through the sector-shaped salt solution. The electric potential diminished uniformly between the electrodes. The relative potential at any point was conveniently determined by a Wheatstone bridge. In the Wheatstone Bridge used in this analysis, the conventional slide-wire arrangement was replaced by twenty series-connected fixed resistors brought to a multiple-pole selector switch. By means of this bridge, settings for equipotential lines of five percent potential increments were readily



SHASTA DAM
ELECTRIC ANALOGY STUDY
OF TUBE VALVE

obtained.

The scanning mechanism shown in figure 10 provides a means of moving the probing point to all points on the surface of the electrolyte. The two graduated scales make possible the rapid reading and plotting of the coordinates of points along any equipotential line.

14. Analogy between electric and hydraulic flow. In the hydraulic structure, the total potential at any point is by Bernoulli's Theorem equal to the sum of the elevation head, pressure head, and the velocity head. In this study, since the axis of the valve was horizontal, and the vertical distance between top and bottom of the valve was small in comparison with the depth of the reservoir above the valve, the elevation component was neglected and considered equal to zero. Although the lines of equal potential in the electric analogy are only a measure of voltage, their relative positions, spacing, and shape are identical with those that would occur in the hydraulic structure. In the hydraulic structure, if the pressures at each end of the valve and the total flow are known or can be assumed, the total potential at each end can be computed. In the electric analogy, the electrodes are placed at the positions at the ends of the valve where the total potential is known, and the equipotential lines determine the potential at all points between electrodes.

The pressure head may be obtained at any point by subtracting the velocity head from the total potential as given by the lines of equal potential. The velocity head is readily obtained from the spacing of the equipotential lines, since the velocity is directly proportional to the potential gradient, or inversely proportional to the distance between equipotential lines. The constant of proportionality is obtained by measuring the spacing of the equipotential lines in the straight pipe portion of the model at each end of the valve proper.

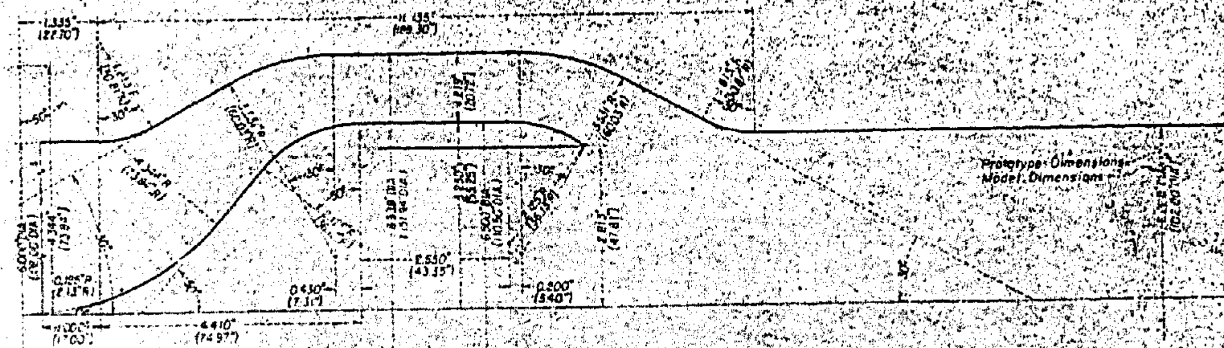
15. Hydraulic conditions assumed in electric analogy. Figures 10 through 14 show eight shapes that were tested by the electric

analogy in the layouts shown on the top of the drawings. The dimensions in parentheses refer to the prototype, whereas the other refer to the hydraulic models, which, in those cases where they have been constructed, have been to a 1:17 scale. In the lower figures both the equipotential lines and lines of equal pressure are shown. From the equipotential lines as given, any pressure system may be constructed depending upon the boundary conditions of discharge and pressures at the end of the valves. The pressure nets shown on the figures are those that would probably exist in 1:17 hydraulic models when tested in a complete assembly of pipe, valve, bellmouth, etc. All pressures are those that would accompany a reservoir head of 18.96 feet on the hydraulic model. With this head, it has been assumed that tests 1, 3, 4, 6, 7, and 8, would discharge approximately 4.78 cubic feet per second. This is the discharge that was actually measured in the hydraulic model of test 1. In the narrow-throated valve, test 2, a discharge of 3.91 cubic feet per second has been used, and in test 5, in which the upstream pipe diameter has been increased in the prototype from 102 inches to 120 inches, the discharge has been estimated to be 4.83 cubic feet per second. In the prototype, the di[illegible]arges would be seventeen to the five halves power times the above, and the pressures would be seventeen times as great as those shown by the solid contours.

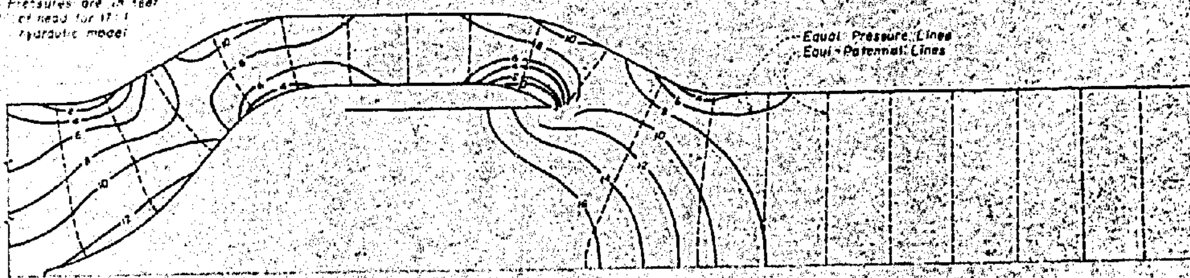
16. Analysis of electric-analogy results. In figures 10 through 14, the minimum pressures in feet of head in the hydraulic model are as follows:

Test	Minimum upstream pressure	Minimum downstream pressure
1	0	2
2	3	-9
3	6	6
4	5	8
5	9.5	8
6	8	5
7	5	6
8	6	6

FIGURE 1[illegible]

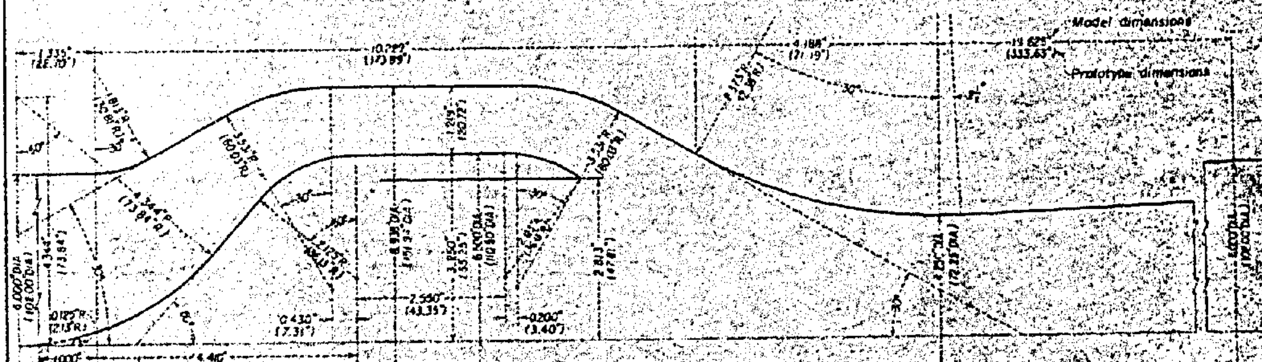


SHAPE 1

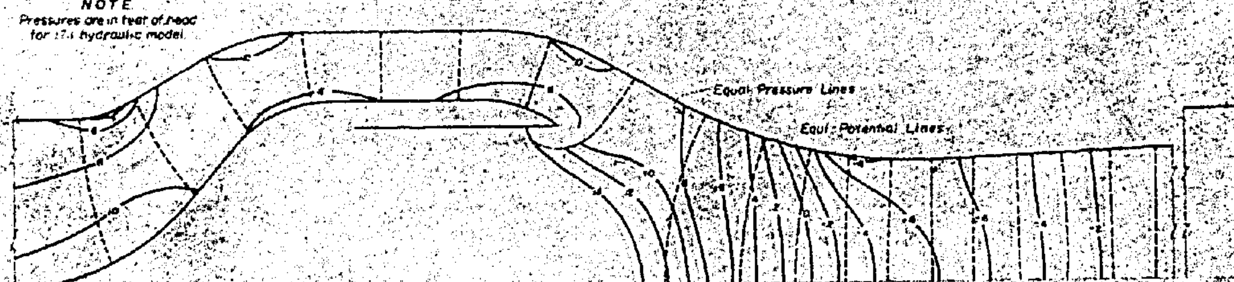


CONTOURS OF EQUAL PRESSURE AND POTENTIAL

TEST 1



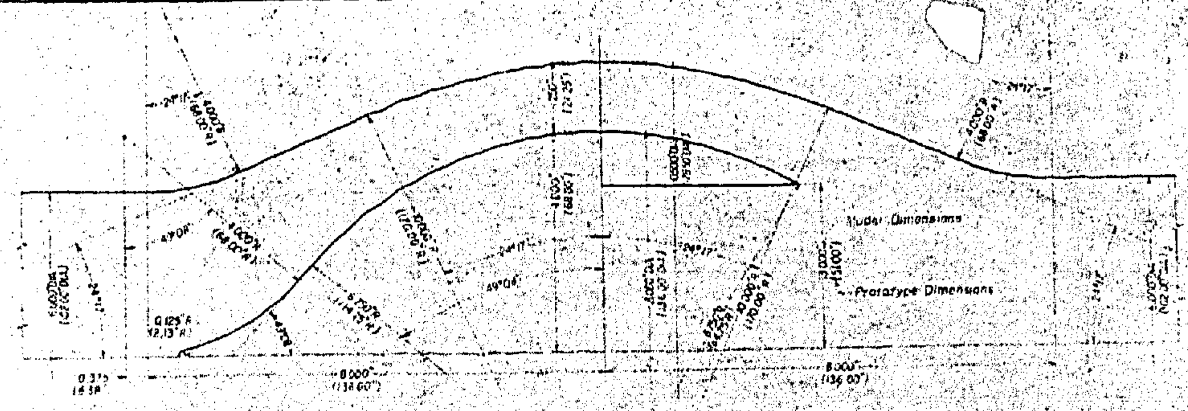
SHAPE 2



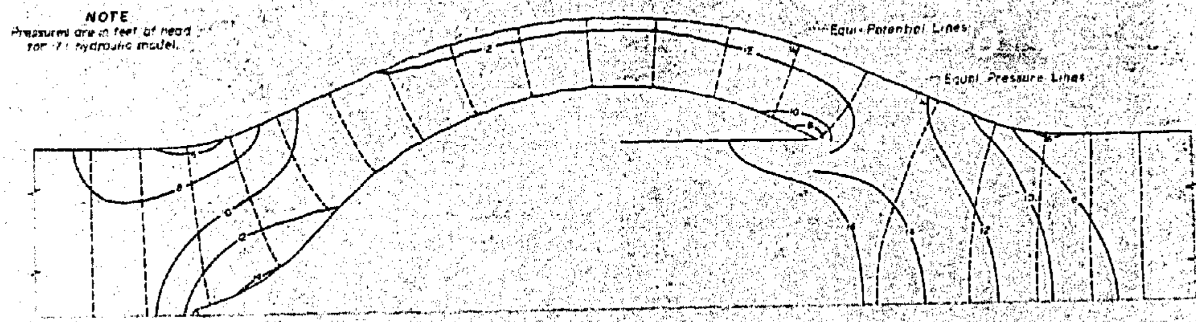
CONTOURS OF EQUAL PRESSURE AND POTENTIAL

TEST 2

SHASTA DAM TUBE VALVE
ELECTRIC ANALOGY TESTS

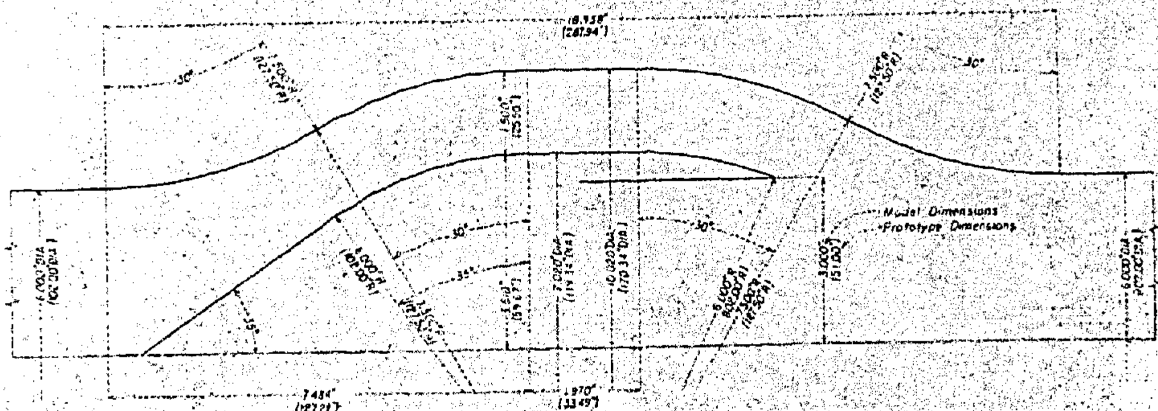


SHAPE 3



CONTOURS OF EQUAL PRESSURE AND POTENTIAL

TEST 3



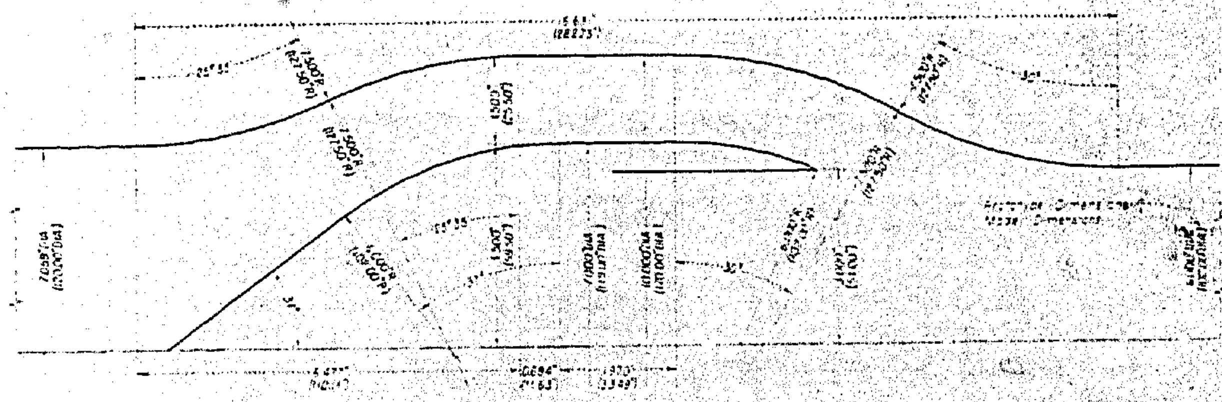
SHAPE 4

CONTOURS OF EQUAL PRESSURE AND POTENTIAL

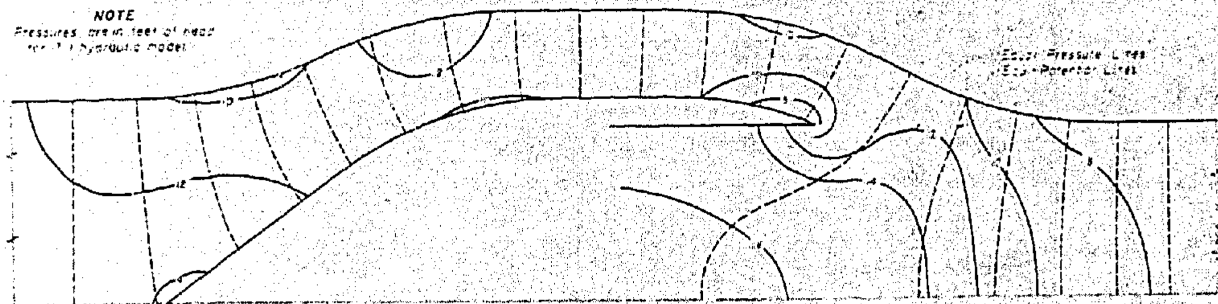
TEST 4

SHASTA DAM TUBE VALVE

ELECTRIC ANALOGY TESTS

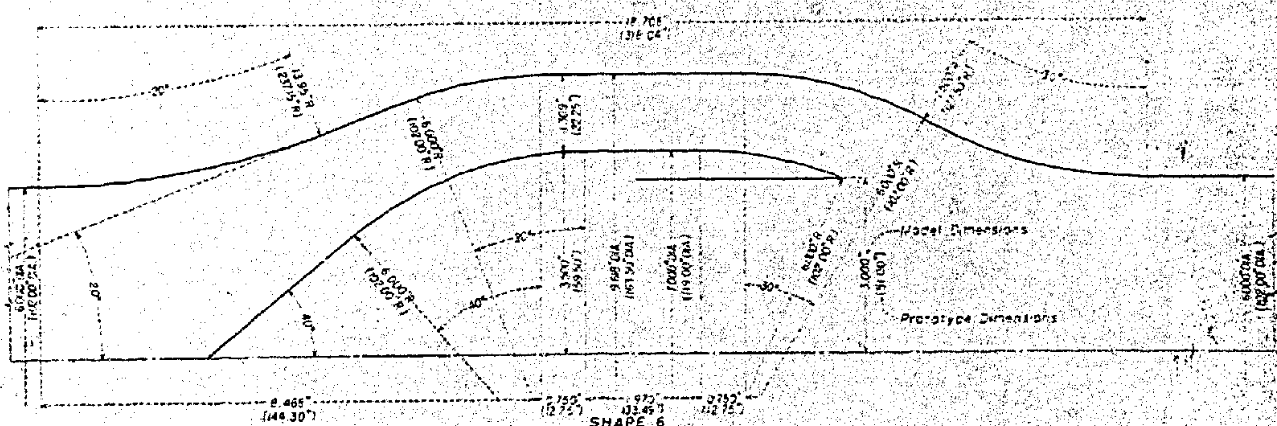


SHAPE 5

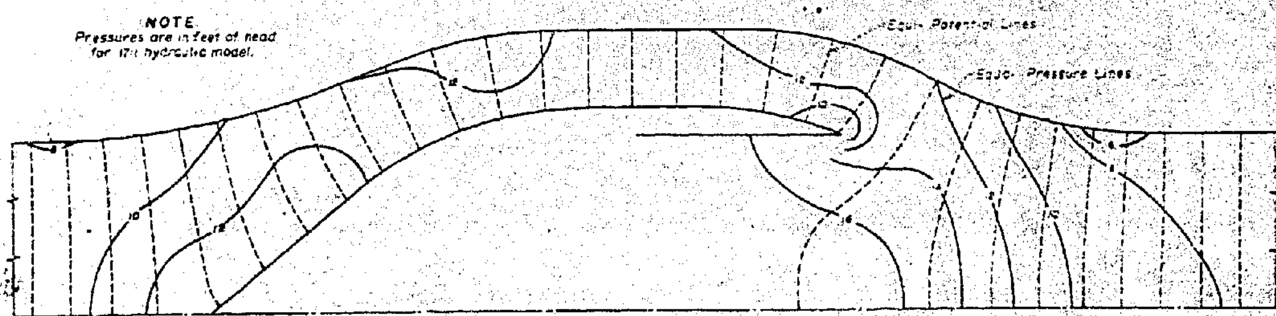


CONTOURS OF EQUAL PRESSURE AND POTENTIAL

TEST 5



SHAPE 6

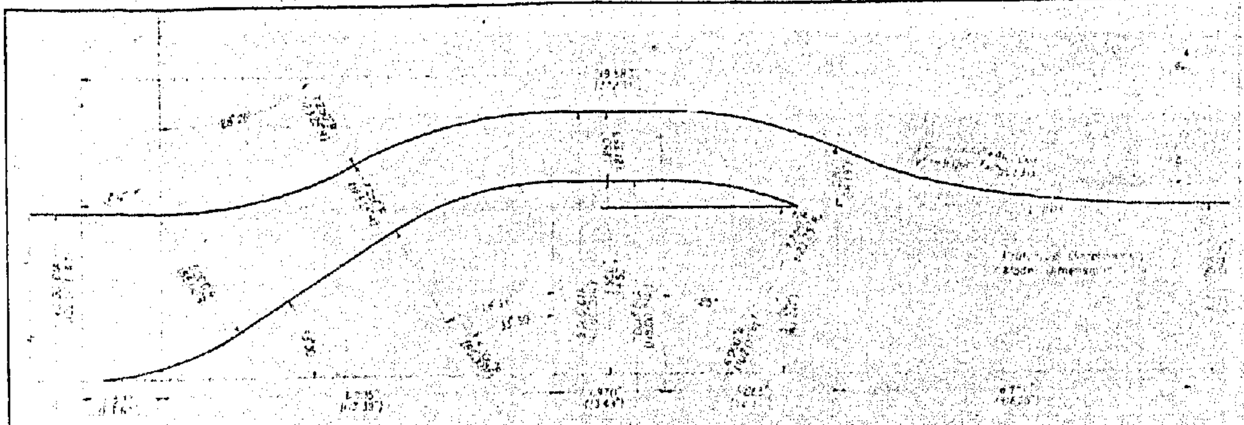


CONTOURS OF EQUAL PRESSURE AND POTENTIAL

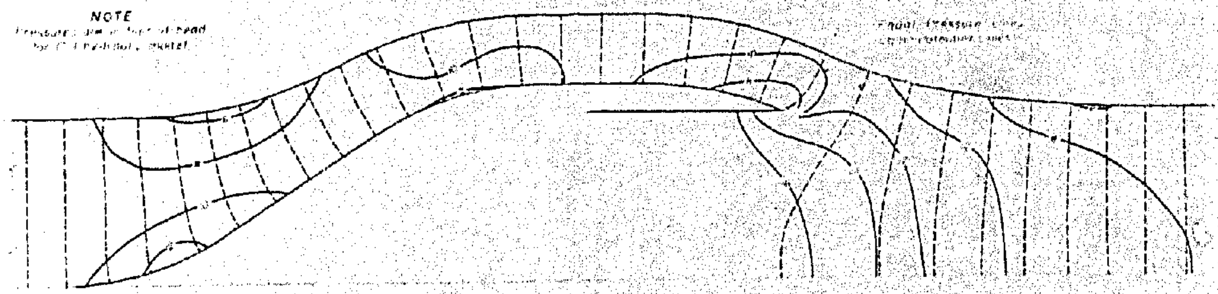
TEST 6

SHASTA DAM TUBE VALVE

ELECTRIC ANALOGY TESTS

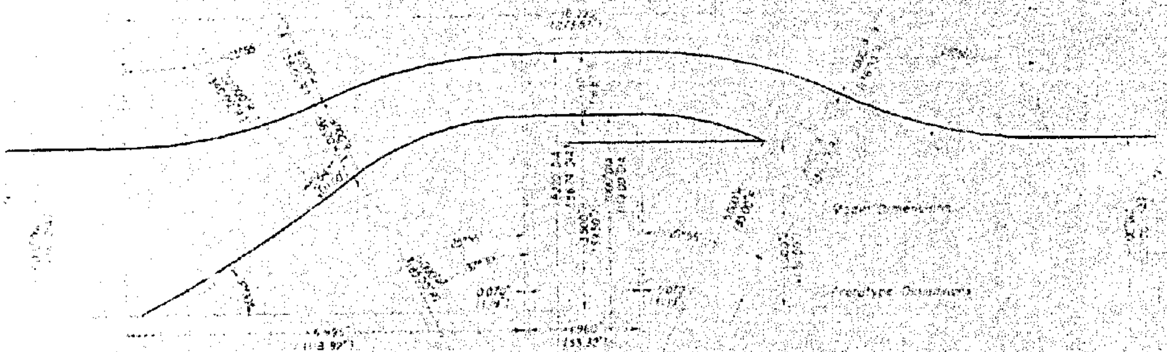


SHAPE 7

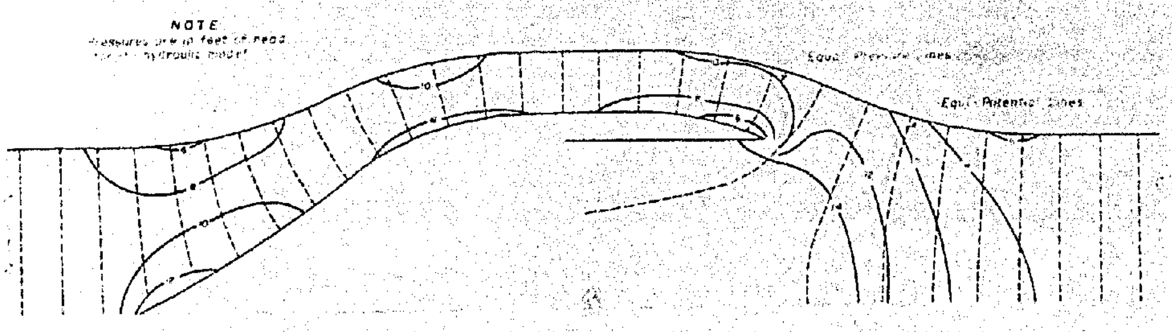


CONTOURS OF EQUAL PRESSURE AND POTENTIAL

TEST 7



SHAPE 8



CONTOURS OF EQUAL PRESSURE AND POTENTIAL

TEST 8

SHASTA DAM TUBE VALVE

ELECTRIC ANALOGY TESTS

In test 1, the relatively low pressures can be attributed to the short radii in the valve shell. In test 2, the negative pressure is a direct result of the narrow throat. The cross-sectional area at the throat is one-half that of the rest of the system. This doubles the average velocity across this section and produces an average velocity head four times as great as would otherwise exist. In this instance, the negative pressure of nine feet of head cannot be multiplied by seventeen to obtain the prototype pressure since it would give a negative pressure of one hundred fifty-three feet, or about one hundred and twenty feet below absolute zero, an absurdity. In the remainder of the tests, there are no regions of severely low pressures, and for this reason they are all relatively free of tendencies to cavitate. It must be borne in mind that all of these tests are for the condition of valves wide open, and are not indicative of low pressures which might exist for partial openings. With the exception of shapes one and two, all the valves tested appear to be hydraulically good designs. Structurally they are quite different. Shape three, for example, would be the heaviest and most expensive. It was tested to show that a shape could be found that would operate efficiently under wide-open conditions. The wide variation in the design of the upstream point of the needle has little effect upon the pressures. Unless the space within the needle point is needed for structural purposes, it could be efficiently replaced by a blunt-nose needle, similar to the blunt-nosed leading edges of airfoils.

The analogy of electric to hydraulic flow is quite close but not exact. For example, there can be no tendency for vortex action in electric flow. In the hydraulic structure there will exist a stable vortex or "water needle" within the tube. Its angular velocity will probably be such as to lower the pressure within the tube and to prevent flow around the sharp edge of the tube. This eliminates the low-pressure zone indicated for all tests in this region by the electric analogy. With this exception, the pressures obtained by the electric analogy should agree closely with those obtained in hydraulic models.

IV – PRELIMINARY STUDIES OF SELECTED DESIGN

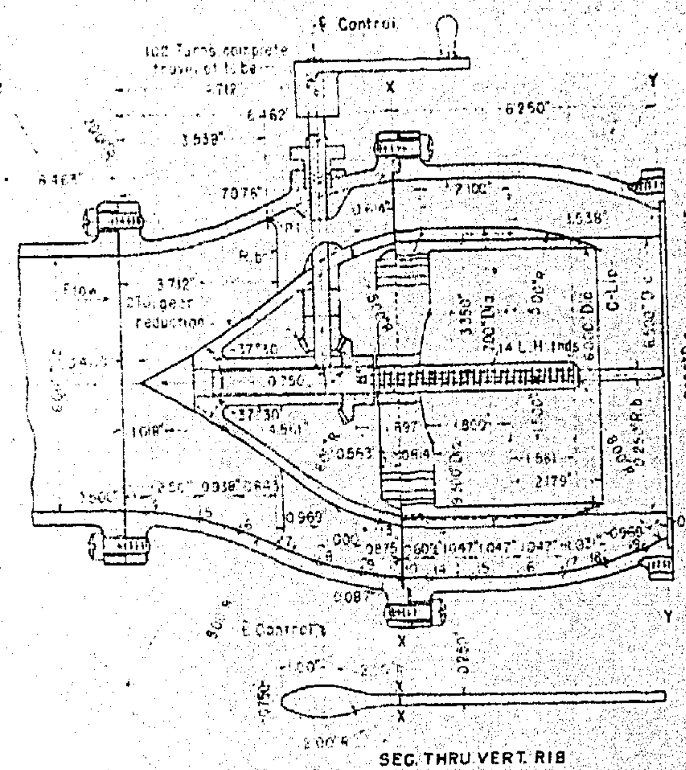
17. Measurement of jet profile (Test 8A). The electric-analogy investigation included eight proposed shapes of tube valves. The first was the same as used in hydraulic test 1 and was unsatisfactory also in the electric-analogy study. Electric analogy 1 and 2 were discarded because of subatmospheric pressure in the nozzle. Test 5 was not satisfactory because of the increase of diameter from 102 to 120 inches at the upstream end which would have increased materially the size of the bellmouth outlet entrance and the emergency coaster gate. Tests 3, 4, and 6 were discarded because of the large diameter of the valve body necessitating an increase in the size of the operating chamber. Test 7 was undesirable due to the excessive length of the valve. So, by elimination, the shape used in test 8 was chosen for further development in the hydraulic model.

The upstream section, body, tube, and control mechanism for the valve model, based on the electric-analogy study, was machined and assembled for testing, as shown in figure 15A. The nozzle was omitted and the valve operated as a free-discharge valve to obtain the shape of the jet to determine the profile of the nozzle.

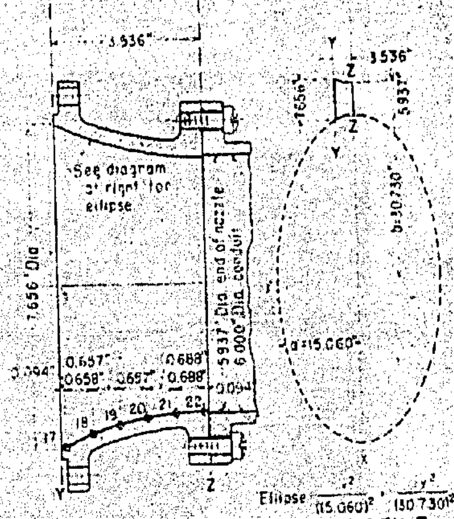
18. Elliptical nozzle outlet with diameter of 5.937 inches (Test 8B). The actual profile of the jet obtained in test 8A is shown in figure 15E. For test 8B, a nozzle was machined, as shown in figure 15B, using a sector of the ellipse $\frac{X^2}{(15.060)^2} + \frac{Y^2}{(30.730)^2} = 1$ and a downstream diameter of 5.937 inches instead of 6 inches. The reduced diameter was used to avoid removing material from the nozzle which would be needed for subsequent studies. A comparison of the sector of the ellipse and the actual profile is shown in figure 15E.

With the nozzle discharging freely into the atmosphere, the nozzle did not flow full so no attempt was made to measure the pressures. Although the ellipse selected was slightly inside the measured profile, its curvature differed sufficiently to explain the

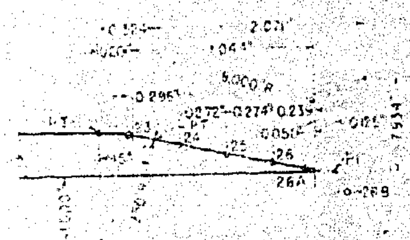
NOTE: Piezometer numbers 14 to 19 incl. shown for test 8A change to no's. 11 to 16 incl. for tests 8B, 8C and 8D.



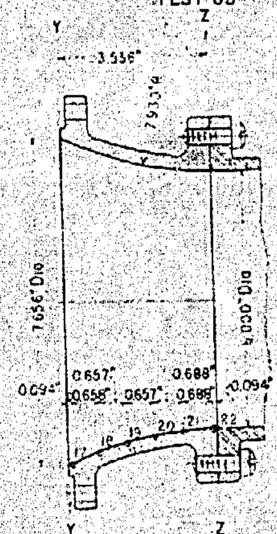
A. ASSEMBLY OF MODEL VALVE BASED ON ELECTRIC ANALOGY STUDY B
TEST 8A



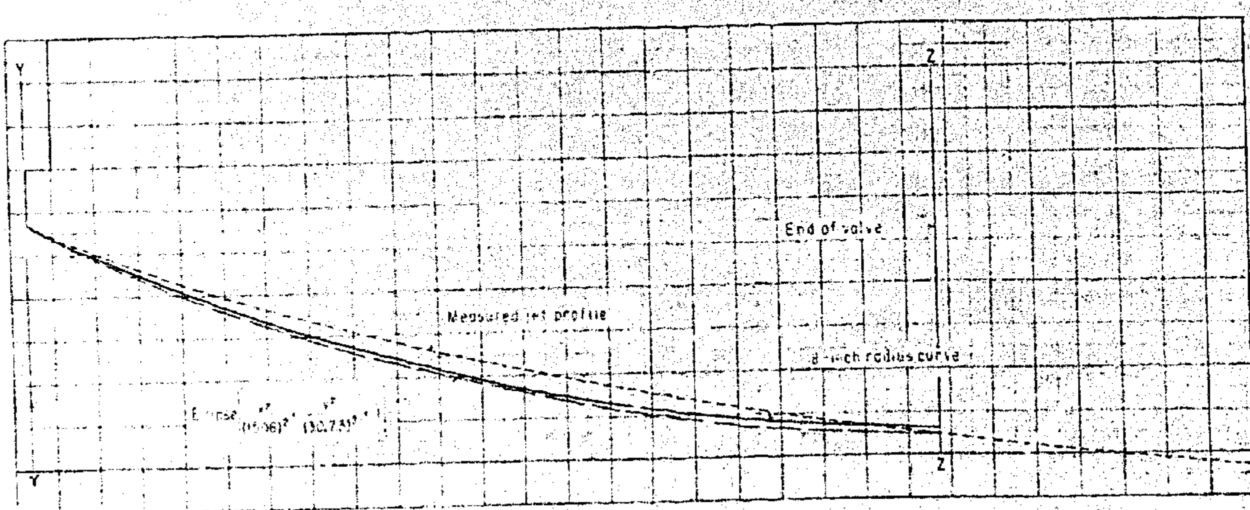
B. NOZZLE WITH ELLIPTICAL PROFILE
TEST 8B



C. REVERSE CURVED LIP ON TUBE
LIP E, TESTS 8D AND 8E



D. NOZZLE WITH RADIUS OF 7.93 INCHES
TESTS 8C AND 8D



E. COMPARISON OF JET WITH ELLIPTICAL AND CIRCULAR CURVES ON NOZZLE PROFILE

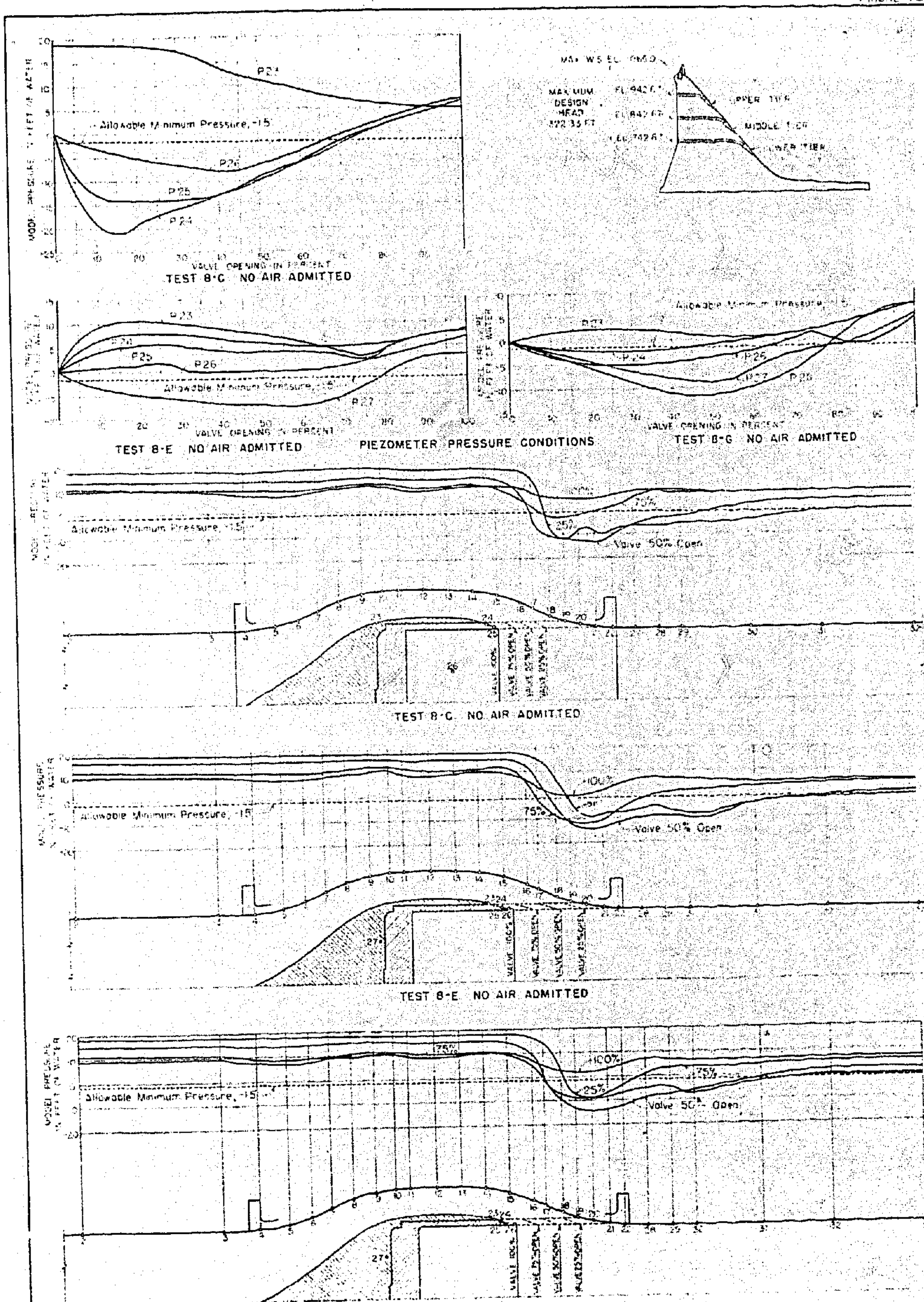
MODEL DETAILS FOR DESIGNS 8A THROUGH 8D

failure of the nozzle to flow full. If the valve could have been forced to flow full, there would have been negative pressures on the nozzle. This possibility would be offset by the back pressure which would be obtained if the conduit were attached. The use of the ellipse in the nozzle would have increased the difficulty of the manufacture of the prototype nozzle, so to simplify the construction details, an arc of a circular curve with a radius of 7.93 inches was used in test 8C instead of the ellipse (figure 15B and D). Measurements of pressure (figure 16) showed that although the pressures were positive at 85 and 100 percent opening, extremely severe subatmospheric pressures existed at smaller openings. The tube lip was also subject to severely low pressures at the smaller openings. This was the first test in which conditions were satisfactory at a valve opening of 100 percent.

20. Alteration of tube lip and increase of nozzle radius. With a valve in which conditions were acceptable at the full-open position, in fact, from 85 to 100 percent opening, attention was directed to revision of the tube lip to eliminate the severe low pressures on it. In tests 8A, 8B, and 8C, the tube lip was a simple circular curve with a radius of 5 inches. Inspection of a section of the valve will show that as the valve is closed, the water passage between the lip and the body is diverging which would create low pressures with high-velocity flow. The low pressures on the nozzle at part openings were not considered serious inasmuch as they could be relieved by aeration, a factor which had not been introduced into the program since other considerations predominated, whereas no such improvement could be expected in the condition of low pressures on the lip because the area could not be directly relieved.

Applying the same logic as used on the nozzle, the tube lip was changed to conform to the contour of the jet by a reverse curve. In test 8D, the lip was revised by introducing a reverse curve, as shown in figure 15C. Test 8D consisted of observations of the pressures on the lip only, and the results are not included. Similar results were

FIGURE 16



SHASTA DAM OUTLETS

PRESSURE CONDITIONS IN TUBE VALVES

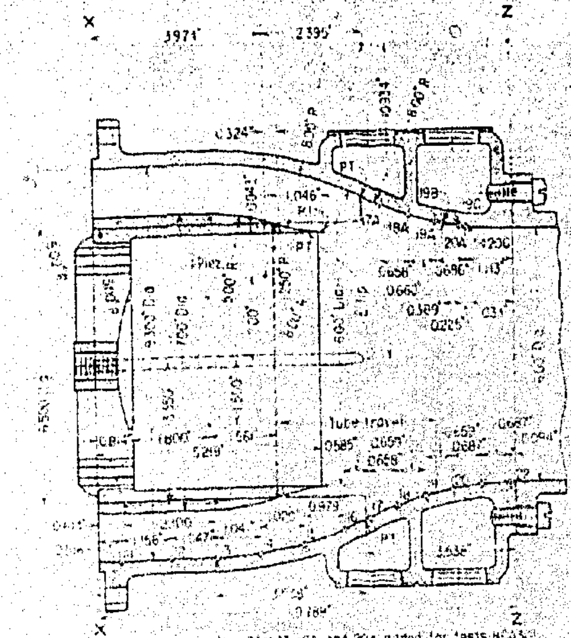
FOR MAXIMUM HEAD ON LOWER TIER

MODEL SCALE 1:17

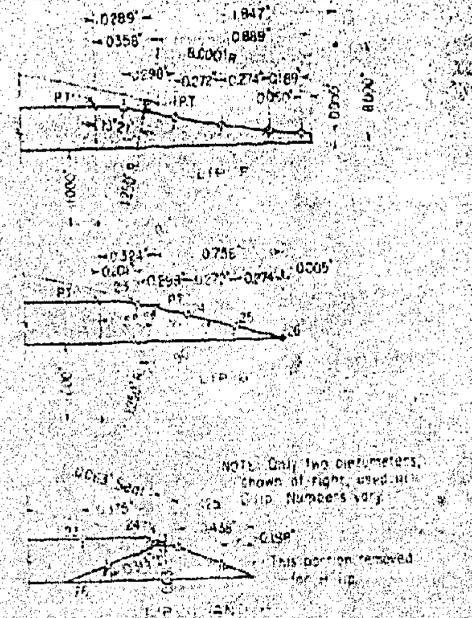
obtained in test 8E. Briefly, the pressures on the lip were practically all positive, which indicated that the change was an improvement. The pressures in the nozzle at part openings were still too low.

At this point in the program, it appeared desirable to combine the improvements attained and introduce air to relieve the low pressures at partial openings. A new casting (figure 17A) was made in which the downstream portion of the body and the nozzle were combined, and on which two air boxes were provided through which to introduce air into the nozzle. In the new model, first used in test 8E, (1) the radius in the nozzle was changed from 7.930 to 8.000 inches; (2) the reverse curve was used on the tube lip, henceforth referred to as lip E from its use in test 8E; (3) the thickness of the supporting vanes in the body of the valve was increased to permit installation of air conduits into the tube interior; and (4) the intermediate section and nozzle were combined as one casting. The change in radius from 7.930 to 8.000 inches was made for psychological reasons; a radius of 7.930 x 17 or 134.81 inches would look very peculiar on the prototype drawings, whereas 8.000 x 17 or 136.00 inches looked logical. The larger radius formed a slightly more gradual transition from the valve body to the 6-inch diameter outlet conduit, and also made the nozzle a duplicate of that used in electric-analogy study No. 8.

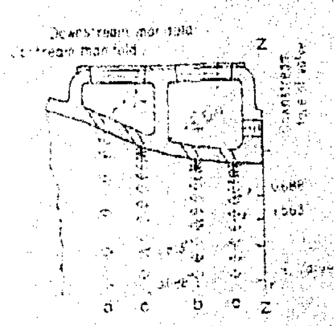
The detailed pressure measurements of test 8E (figure 16) showed that the pressures on the nozzle were negative for 85 and 100 percent openings. As the pressures for 85 and 100 percent openings had been positive for test 8C, it was necessary to determine whether the change to negative pressures was due to the reverse curve on the tube lip, the 8-inch radius nozzle, or the increased thickness of side vanes. The C lip was installed in the test valve (test 8F) to determine if the pressure gradient would coincide with the one measured prior to the change of the nozzle radius and supporting vanes. The pressure gradient agreed with that measured in test 8C, so the adverse conditions on the



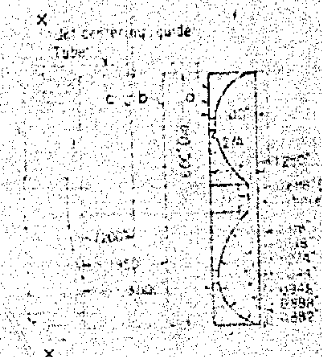
A. NOZZLE CASTING WITH TWO AIR BOXES



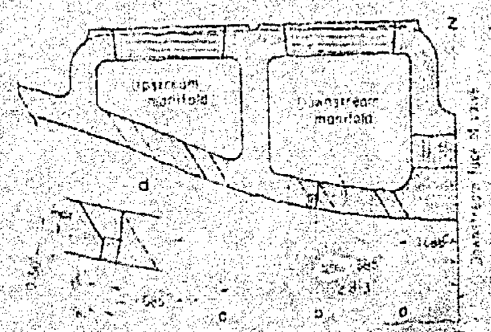
B. DETAILS OF VARIOUS LIP DESIGNS



C. LOCATION OF $\frac{3}{16}$-INCH DIA. AIR RELIEF HOLES IN NOZZLE

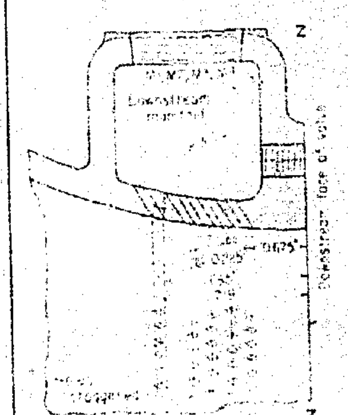


D. JET CENTERING DEVICE

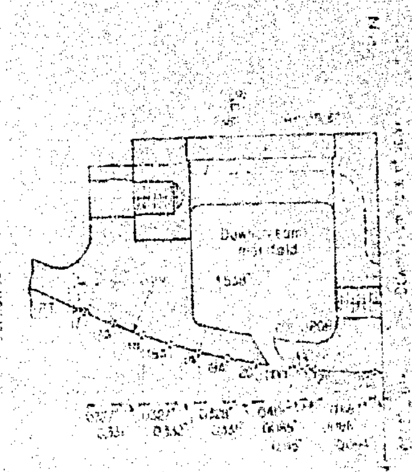


E. ENLARGED DETAIL OF NOZZLE SHOWING EQUALIZING GROOVE

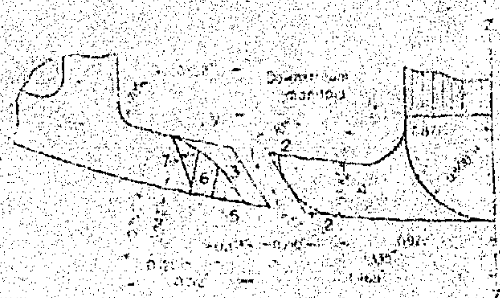
TESTS BEA5? AND BEA302



F. SINGLE AIR BOX WITH SMALLER HOLES



G. REVISION OF AIR BOX WITH AIR RELIEF SLOT



H. DEVELOPMENT OF SLOT DESIGN

SHASTA TUBE VALVE

DETAILS OF REVISIONS FOR TESTS 8E TO BEAS16 nozzle could be charged to lip E.

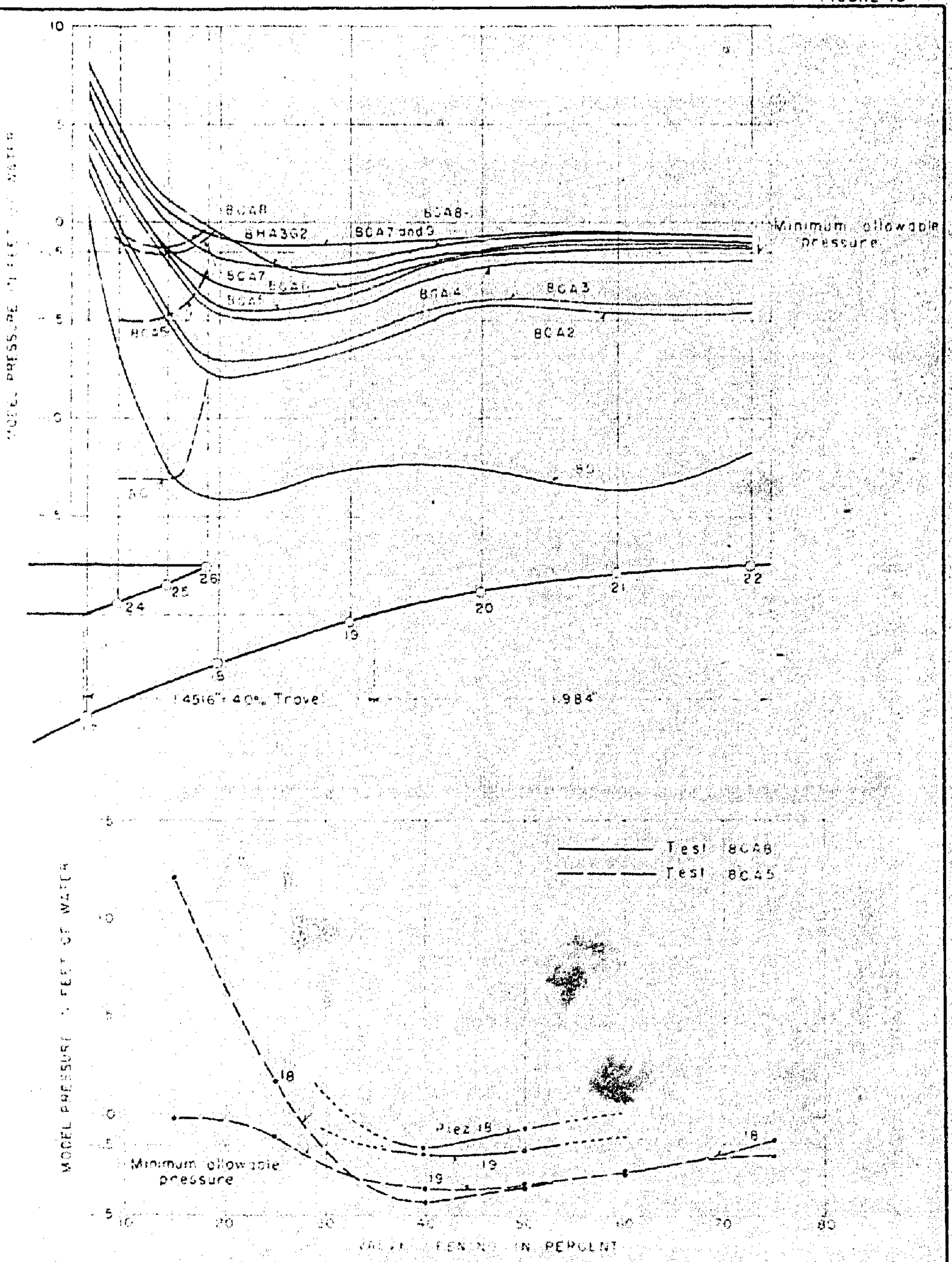
21. Lip G installed and lip pressures decreased. A second lip (figure 17B), known as the G lip from its use in test 8G, was installed in which the lip was a circular arc with a radius of 1.25 inches and a tangent. It was a compromise between the simple curve of test 8C and the reverse curve of test 8E. The pressures were again critical, as shown in figure 16, but the pressures on the lip were higher at 100 percent opening than for lip E in test 8E. The conditions to maintain positive pressures on the tube lip caused subatmospheric pressures on the valve nozzle, and vice versa.

V - DETAILED STUDIES LEADING TO FINAL DESIGN

22. C lip reinstalled, effect of holes checked (Test 8CA1). Conditions at full opening were assumed to be satisfactory and attention was directed toward improving conditions at partial opening by admitting air to relieve the negative pressures which prevailed. Since the pressure varied from positive to negative over the nozzle and the exact location for admitting air was difficult to determine, it was decided that air relief be provided by holes drilled through the nozzle into the double, annular air chamber, called the air box, which surrounds the nozzle casting as shown in figure 17C. The air box was divided into two separate chambers, each of which had four supply lines, scheme A, figure 20, to prevent circulation when one chamber is under pressure and the other under a vacuum. Before any air was admitted, the effect of the presence of the holes was determined by drilling 24 equally spaced holes, 3/16-inch in diameter, in row (d) as shown in figure 17C and testing with the air-supply lines shut off. The tests indicated no change from test 8C. The holes produced water throughout the range of valve opening, so were plugged as useless.

23. Air admitted through holes in nozzle. Air was admitted through 24 holes in row b test 8CA2) with the location and size shown in figure 17C and the pressures in the critical range for piezometers 17 to 22 were materially raised at 40 percent valve opening, as shown in figure 18. The pressures shown in figure 18 and subsequent figures were determined with a model head of 19 feet which corresponds to the maximum head on the lower tier or 322 feet, prototype. This opening was selected as the basis of comparison for several of the following revisions because it had low, although perhaps not the lowest pressures, before air was admitted.

In an effort to enlarge on the improvement in the preceding test, 24 more holes were added in row (b) (test 8CA3). The slight increase in pressure as shown in figure 18 indicated that no significant



COMPARISON OF PRESSURES IN CRITICAL ZONE
FOR DESIGNS 8C THROUGH 8CA9

improvement could be expected from an increase in the number of holes in this row.

A second row of holes, (test 8CA4) row (a) in figure 17C was drilled and air was admitted through this new row with the further small increase in pressure, shown in figure 18.

To exhaust the possibilities of this row (a), 24 holes were added (test 8CA5) but no significant increase in pressure occurred, see figure 18.

Although it has been stated previously that the low pressures on lip C could not be directly relieved by aeration, since it is impractical to admit air to the tube lip, an improvement in pressure was expected as a result of any increase in pressure on the nozzle or inside the tube. With the amount of aeration provided in test 8CA5, the lip pressures increased considerably as shown in figure 18.

A third row of 48 holes, row (c) in figure 17C, was drilled in the upstream chamber of the air box which made it possible to again check for any effects due to the presence of the holes by keeping the upstream chamber closed for test 8CA6. The test results, shown in figure 18, indicate no appreciable effects due to presence of air holes. When air was admitted to row (c) in test 8CA7 the pressures were materially increased and the increase was reflected in the pressures on the tube lip, as shown in figure 18. From the standpoint of pressures on the nozzle this design was very promising but since row (c) of holes was located upstream from the valve seat and would, therefore, be under full reservoir pressure at closure, it would be necessary to install an automatic relief valve which would shut off the upstream air chamber when the pressures became positive at row (c) or extend the air-supply piping from this portion of the air box to maximum reservoir elevation.

24. <u>Air admitted to tube interior</u>. When the valve was dismantled to drill holes in the nozzle in one of the previous tests,

holes were drilled through the tube supporting ribs into the tube interior in the manner shown in figure 20, scheme A, because it was anticipated that in due course air would have to be admitted to the tube interior. Although the previous test (8CA7) had satisfactory pressures, row (c) of holes was located above the seat which was objectionable as already explained. In an attempt to avoid the use of row (c) air was admitted to the tube interior through holes in the ribs, test 8CA8. The nozzle pressures were increased over those of 8CA5 as shown in figure 18 which shows the comparison of nozzle pressures at 40 percent valve opening. It is evident from figure 18 that although aerating the tube interior did raise the pressures it was not as effective as admitting air to row (c) in test 8CA7. The situation is reversed in the case of the lip pressures which were higher for test 8CA8 than for test 8CA7.

Although it was not appreciated at the time, this valve, 8CA8, was a reasonably satisfactory design and a few additional remarks are pertinent. The data taken on this valve were very meager because the test was intended to show only the effect of aerating the tube interior for use later in the testing program. For this reason it is impossible to describe definitely the performance of the design at all openings. However, an estimate can be made by using test 8CA5 as an index of pressure variation with valve opening since the only difference between tests 8CA5 and 8CA8 was that in the latter the tube supply was open. A plot of the pressures at the two piezometers 18 and 19 against valve opening for 8CA5 and 8CA8 is shown in figure 18. It is evident from the curves that the data taken in test 8CA8 which consisted of tests at 40 and 50 percent openings represent the minimum pressures and if the shape of the remainder of the curve is assumed to follow test 8CA5, the arbitrary limit of negative pressures in model terms of -1.5 feet of water is exceeded for only some 15 percent of the total travel as shown in figure 22. This design is, therefore, recommended for further study at some future date.

To complete this particular phase of the study the upstream air chamber was opened (test 8CA9) and air was thereby admitted through row c. All air inlets were now open and the pressures were increased back up to those of test 8CA7 as shown in figure 18.

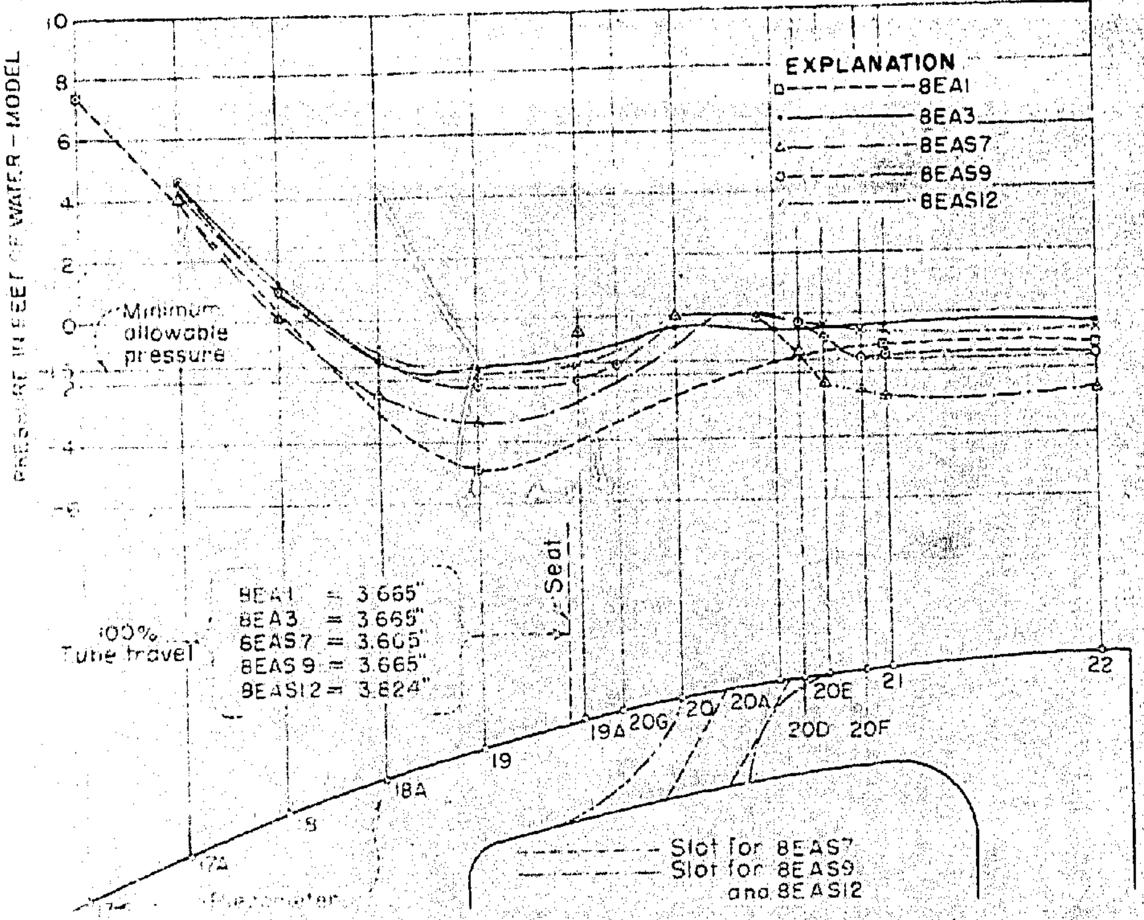
25. Lip changed to E, air relief through nozzle and tube. It was fixed definitely at this point that further development be concentrated principally upon the valve with the E lip installed because the necessity of using row c of holes to relieve the negative pressures on the tube lip made the C lip unacceptable. With only the downstream air chamber open, the pressures at 60 percent opening (test 8EA1) were too low as shown in figure 19A, and more air relief was indicated. It will be noted that the basis of comparison has been changed from 40 percent to 60 percent opening because of the greater amount of test data at the larger opening. The pressure comparisons shown in figure 19 are divided into groups but one test has been made common to each pair of groups to provide continuity.

When the upstream air chamber was opened (test 8EA2) it was evident that this portion of the valve was under pressure so no test was made.

When air was admitted to the tube interior (test 8EA3) the pressures were raised considerably as shown in figure 19A.

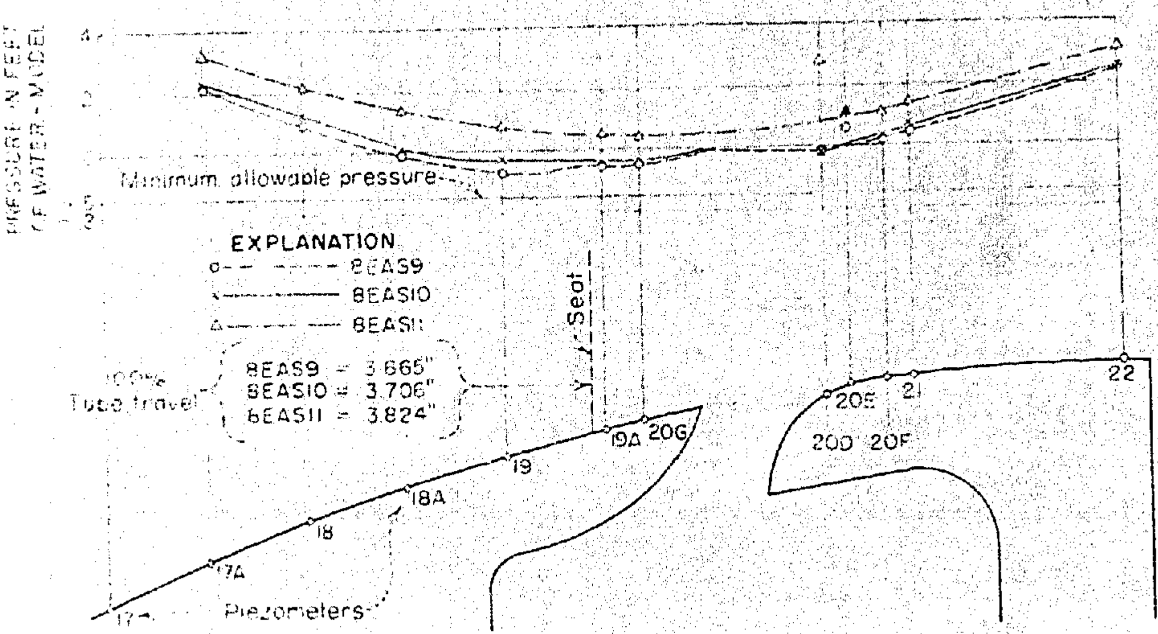
(Test 8EAJ). A digression was made at this point to test a so-called jet-centering device which was designed to stabilize the eddy inside the tube and possibly produce higher pressures. The nature of the device is shown in the various positions tested in figure 17D. Since no significant change in pressures could be detected, test data are not shown for this design.

26. Equalizing groove installed at row b. A groove, 1/16-inch in width and located 1/16-inch downstream from the seat, (test 8EA3G1) was machined around the nozzle in the manner shown in figure 17E. This groove was supplied through the existing row of holes, row (b).



A. 60 PERCENT OPENING

DESIGNS 8EA1, 8EA3, 8EAS7, 8EAS9 AND 8EAS12



B. 100 PERCENT OPENING

DESIGNS 8EAS9, 8EAS10 AND 8EAS11

COMPARISON OF PRESSURES IN CRITICAL ZONE

10-12-45

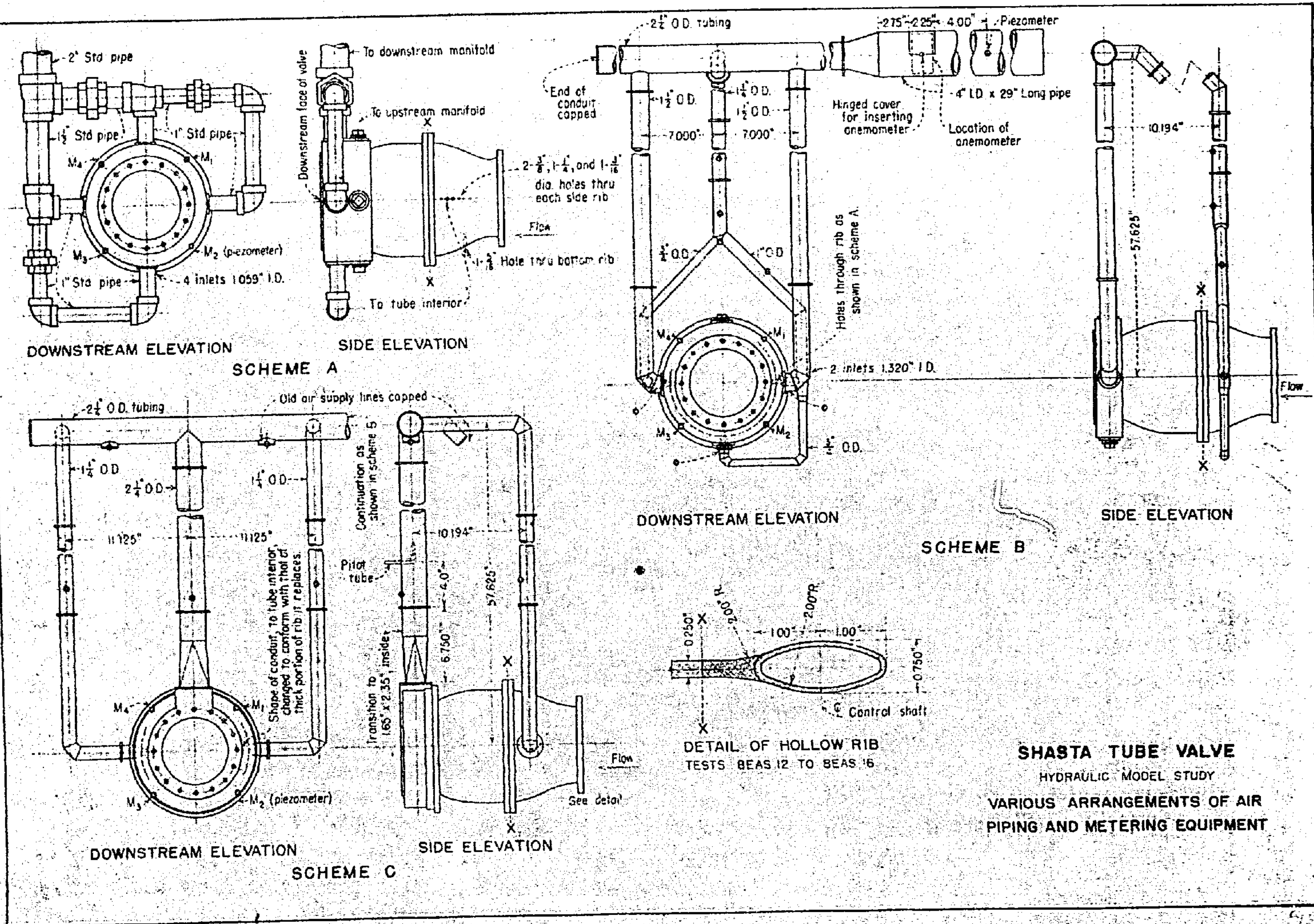
It was reasoned that the groove would act as an equalizer and would supply air in a continuous line rather than just at the positions of the holes. The change in pressure due to this groove was very slight.

When the groove was widened to 3/32-inch, (test 8EA3G2) as shown in figure 17E, there was no increase in pressure which indicated that the capacity of the holes supplying the groove had been reached. To verify that the low pressures were a function of the curvature of the valve boundaries rather than the siphonic action of the deflecting elbow at the end of the conduit, the previous test was repeated with the conduit detached from the valve. The pressures agreed with those in the previous test indicating that the low pressures were definitely a function of the boundary curvature.

The lip C was reinstalled (test 8CA7G2) on the supposition that with this lip the pressure at the groove would be lowered and more air would be pumped but the test results indicate that there was no improvement.

(Test 8HA2G2 and G3). While attention was again focused on the C lip an attempt was made to improve the condition of low lip pressures by removing the portion subject to the low pressures. The lip was removed as shown in figure 17B and designated as lip H. The pressures for this lip are compared with those for the C lip in figure 18 and it is apparent that there had been no improvement at 40 percent. When this design was operated at low openings the jet disintegrated badly so the design was abandoned.

27. <u>Further development of E lip design</u>. The E lip was reinstalled and the 3/16-inch diameter holes were replaced by double their number of 3/32-inch diameter holes because the latter size would be more practical in the prototype, see figure 17F. The four supply lines to the air box scheme A were replaced by two risers of equivalent area - scheme B, figure 20. The test results (test 8EA4) indicated that the pressures were not changed by any significant amount



SHASTA TUBE VALVE
HYDRAULIC MODEL STUDY
VARIOUS ARRANGEMENTS OF AIR PIPING AND METERING EQUIPMENT

FIGURE 20

at a valve opening of 60 percent. The design of the nozzle was simplified by eliminating the upstream air box as shown in figure 17F. At openings below 60 percent the groove had a detrimental effect. Visual observations revealed that water flowed through the groove into the air chamber from whence it returned to the nozzle through the air supply holes thus effectively reducing the amount of air relief. For this reason the groove was abandoned. Although in all previous tests with air relief the quantity of air involved was treated implicitly from a consideration of pressure conditions, the design had reached a point where direct measurements of the amount of air flowing were necessary to provide data for the design of the prototype air piping. The arrangement of piping and the apparatus for metering air are shown in figure 20. Measurements of the quantities of air flowing were made for comparison with later tests.

28. Groove replaced by holes, all holes enlarged (Test 8EA5). The groove of the previous test was filled and replaced by a row of 96 holes of 0.104-inch diameter. All other holes were enlarged from 3/32-inch diameter to 0.104-inch diameter. These changes had very little effect on the pressures. The similarity in the performance of tests 8EA4 and 8EA5 is shown by a comparison of air discharges in figure 27. It was concluded that the possibilities of air relief by means of holes had been thoroughly exhausted.

29. Further simplification of air-box supply (Test 8EA6). The two supply lines feeding the air box were replaced by one line of greater cross-sectional area and the attachment to the air box was accomplished by a transition as shown in scheme C figure 20. The adequacy of the single supply line was demonstrated by the uniformity of the pressure around the periphery of the air box. Another feature of the problem was high-lighted when a more or less routine change was made in the model set-up. The transparent conduit, including the deflecting elbow and terminal cone, had become warped so it was replaced by a brass conduit with the elbow and cone made of sheet metal.

This change resulted in a decrease in pressure at the end of the valve of approximately one foot model at 100 percent opening. Due to this decrease at the end of the valve some of the pressures in the valve were negative at 100 percent opening. An investigation revealed that the small end of the sheet metal cone, which was made to the proper size, was 0.01-inch larger in diameter than the transparent cone which it replaced. The transparent cone which was fabricated of a cellulose plastic had obviously shrunk by this amount. The purpose of of the cone was to create back pressure at 100 percent opening and its exit diameter is very critical as indicated by this test. It is apparent that in the prototype the dimensions of this cone will have to be held to fairly close tolerances.

30. Development of slot to replace holes (Test 8EAS7). A different method for supplying air to the nozzle was next undertaken since the use of holes had been thoroughly explcited. The downstream air box or manifold was revised as shown in figure 17G. A slot was machined with an area equivalent to the former holes at an angle of 30 degrees from the vertical as shown in figure 17H. The nozzle pressures were lowered by an appreciable amount as shown in figure 19A. Also the discharge of air was decreased as shown in figure 27. The reason for the decrease in pressure is apparent from a consideration of the location of the slot and the holes which it replaced, and of the shape of the pressure gradient. Comparing the slot and the five rows of holes in test 8EA5, it will be noted that one of the rows was upstream from the slot and two of them were downstream. Referring to figure 19A it can be seen that the elimination of the air relief provided by the row upstream from the slot lowered the pressure at this point and the effect was reflected upstream. Also the absence of the two downstream rows lowered the pressure considerably at the end of the valve. Apparently this last effect was due to the sharpness of the downstream corner of the slot which created a contraction of the air stream.

In an effort to relieve the pressures at the end of the valve the downstream corner of the slot was rounded for test 8EAS8 by removing the portion marked 2 as shown in figure 17H. The pressures were relieved downstream from the slot but the pressures upstream were not materially improved.

The slot was next extended upstream by removing portion marked 3 in figure 17H in an attempt to regain some of the advantages which were lost when the upstream row of holes were plugged, (test 8EAS9). The pressures were raised by a significant amount, as shown in figure 19A, and were now just as satisfactory as those of test 8EA5. A test at 100 percent valve opening revealed that there were negative pressures because the conduit was not flowing full and under pressure.

31. Increase in tube travel. The valve discharge at 100 percent opening was increased by lengthening the tube travel from 3.665 inches to 3.706 inches, (test 8EAS10), and the pressures were raised slightly as shown in figure 19F but were still negative.

The travel was further increased from 3.706 inches to 3.824 inches (test 8EAS11). The latter figure was derived by plotting the pressures at piezometers 19A and 20C against opening and extending the curves until they indicated positive pressures. The opening at which this occurred was the desired opening. With a travel of 3.824 inches all pressures were positive at 100 percent opening as shown in figure 19B. Because of the change in total travel of the tube the percentage openings cannot be directly compared but comparisons can be made on the basis of equal distances from the closed position which does not change with tube travel.

32. Tube supply enlarged. In the previous series of tests the tube air supply, which was always open, had a constant area. Now this area was increased (test 8EAS12) as shown in figure 20, scheme C. The pressures were raised somewhat as shown in figure 19C but the only significant increase occurred at the downstream end of the valve. The change was due to the fact that the increase in pressure inside the

tube allowed the jet to contract more and this allowed more air to flow between the outer surface of the jet and the valve boundary. This contention is borne out by the increase in air discharge shown in figure 27.

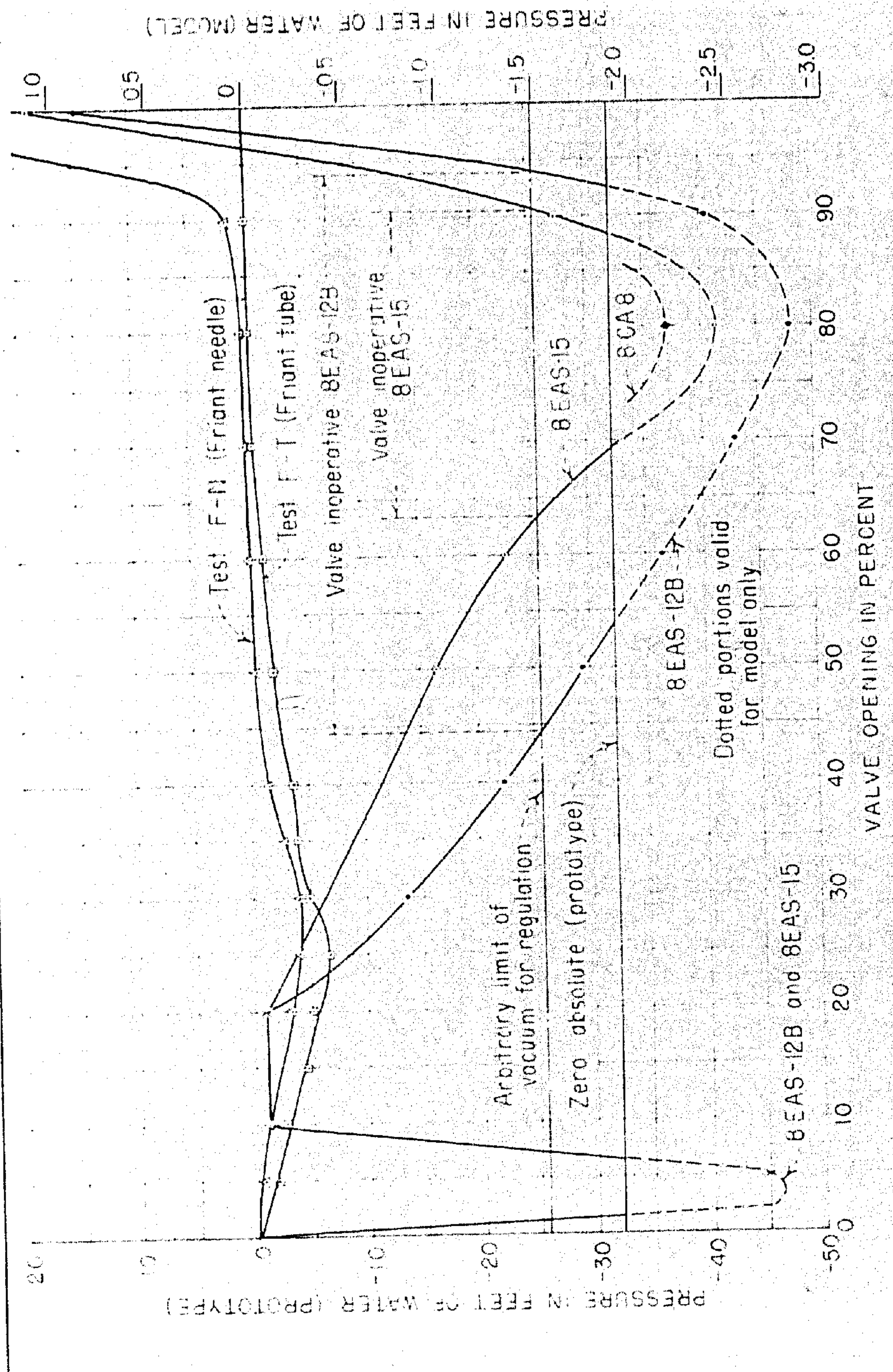
33. Characteristics of tentative final design. Design 8EAS12 was selected as a tentative final design and rather complete data were taken. The significance of the pressure measurements will be discussed in connection with the comparison of the results of the 20-inch and 6-inch models but the variation of the minimum pressure with valve opening is shown in figure 22. The permissible pressure was arbitrarily set at -25.5 feet of water prototype which corresponds to -1.5 feet in the 1 to 17 model. The range of openings from 44 to 93 percent for which the permissible negative pressure was exceeded is indicated in figure 21.

For purposes of comparison with later designs the coefficient of discharge of this design was determined in two ways. For a valve in the lower tier operating under full head the conduit downstream from the valve flowed full and the complete outlet could be treated as a unit. Under these conditions two different coefficients of discharge can be derived for the same pipe area and discharge by using two different heads. First, there is the coefficient based on the head from the reservoir level to the elevation of the discharge elbow. This coefficient will be designated as C_T to denote that it is based on the total head including the siphon head of the discharge elbow. The value of C_T for design 8EAS12 was 0.694 where

$$C_T = \frac{\text{measured discharge}}{\text{conduit area } \sqrt{2gH} \text{ (total)}}$$

This coefficient applies only when the conduit downstream from the valve flows full and the siphon head of the discharge elbow is effective. A second coefficient which is probably more versatile for comparative purposes is one based on the head drop across the valve only. Such a coefficient is an index of the performance of the valve

FIGURE 21



MINIMUM PRESSURES FOR LOWER TIER

PROTOTYPE HEAD = 323 FT.

since the effect of the downstream conduit is eliminated by referring the head to the pressure downstream from the valve. This coefficient will be designated as C_v to indicate that it expresses the performance of the valve only. The value of C_v for design 5EAS12 was 0.745. This coefficient may be regarded also as the free discharge coefficient.

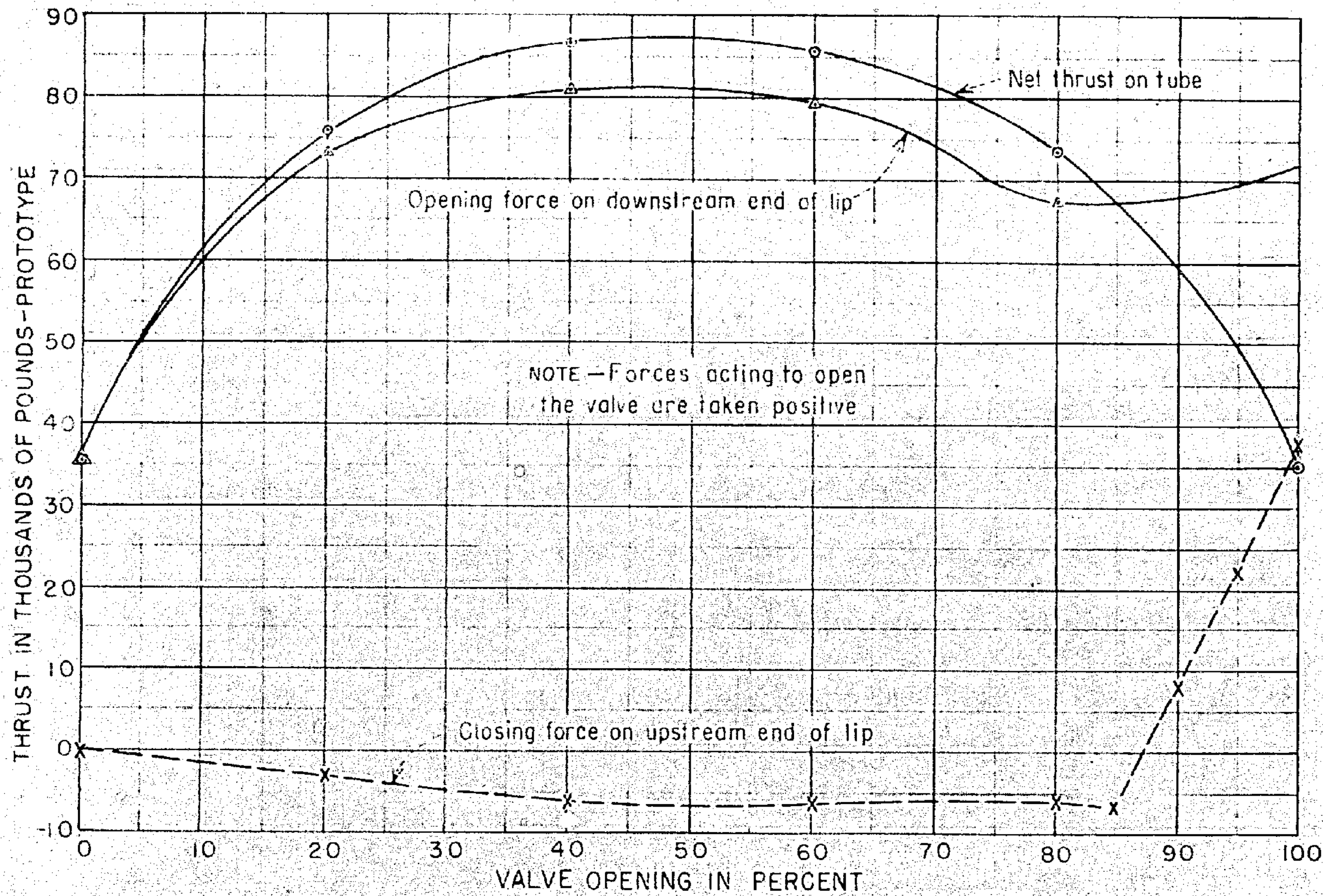
The data required for designing the valve operating mechanism were obtained in two ways, first, by measuring the torque required to operate the 20-inch model under prototype head and secondly, by integrating over the area of the tube the pressures measured in the 6-inch model. The results of the first method are questionable and are, therefore, not included in this report. The torque required to operate the valve was measured for both the opening and closing cycles on the premise that the difference in the two measurements would be equal to the hydrodynamic thrust. It was known from the pressure measurements that the hydrodynamic thrust acted to open the valve and since the torque measurements indicated a consistently higher torque for the opening cycle it was obvious that the frictional forces were the controlling or predominating ones and that their variation with direction was enough to completely mask the hydrodynamic forces. Since the frictional forces in the model bear no definite ratio to those of the prototype which will be operated hydraulically, no estimate of the operating forces required for the prototype could be made from the torque measurements.

The second method of evaluating the forces encountered during operation of the valve consisted of integrating the pressures measured in the 6-inch model. To the forces thus obtained would be added an estimate of the friction forces which oppose motion of the tube. The estimate of friction forces is not treated in this report which deals only with the hydraulic forces. The unbalanced force on the tube is that caused by the difference in pressures on the lip of the tube and its projected area which is subject to the pressure prevailing in the interior of the tube. Each force was treated separately and the net force of the unbalanced load which must be overcome by the operating mechanism was obtained by adding the two forces vectorially. The measured piezometric pressures on the tube lip were scaled up to prototype terms for a head of 322 feet or the maximum head on a valve in the lower tier. The pressure at each piezometer location for each of several valve openings was multiplied by the corresponding prototype radius and the product was plotted against the radius. The area under this curve was integrated graphically and multiplied by 2πto give the total thrust for each opening. This force acted to close the valve since all the measured pressures were positive. The force acting over the back of the tube on the projection of the lip area was obtained by assuming that the pressure measured at piezometer No. 27 which was located inside the tube prevailed over the entire projected area. The variation with valve opening of the two forces and the net thrust on the valve which acts to open the valve are shown in figure 22 for a valve in the lower tier. The force on the back of the tube is negative over most of the range of openings because the pressure inside the tube was negative for all openings where the conduit did not fill. The unbalanced thrust reached a maximum value of 8.7×10^4 pounds at a valve opening of 50 percent as shown in figure 22.

Measurements of air discharge were also made for this design but they will be discussed subsequently in connection with the confirmation tests when the performance of the 20-inch model is compared with that of the 6-inch model.

34. New lip F substituted for E. It was indicated in 8EAS12 that for valve openings less than five percent there were very low pressures at piezometer 19A as shown in figure 21. When the tube is in this position the passage formed by the tube lip and the nozzle is diverging and critically low pressures are set up in its minimum section as in

FIGURE 22



SHASTA TUBE VALVE

UNBALANCED THRUST ON TUBE

DATA FROM 8EAS-12

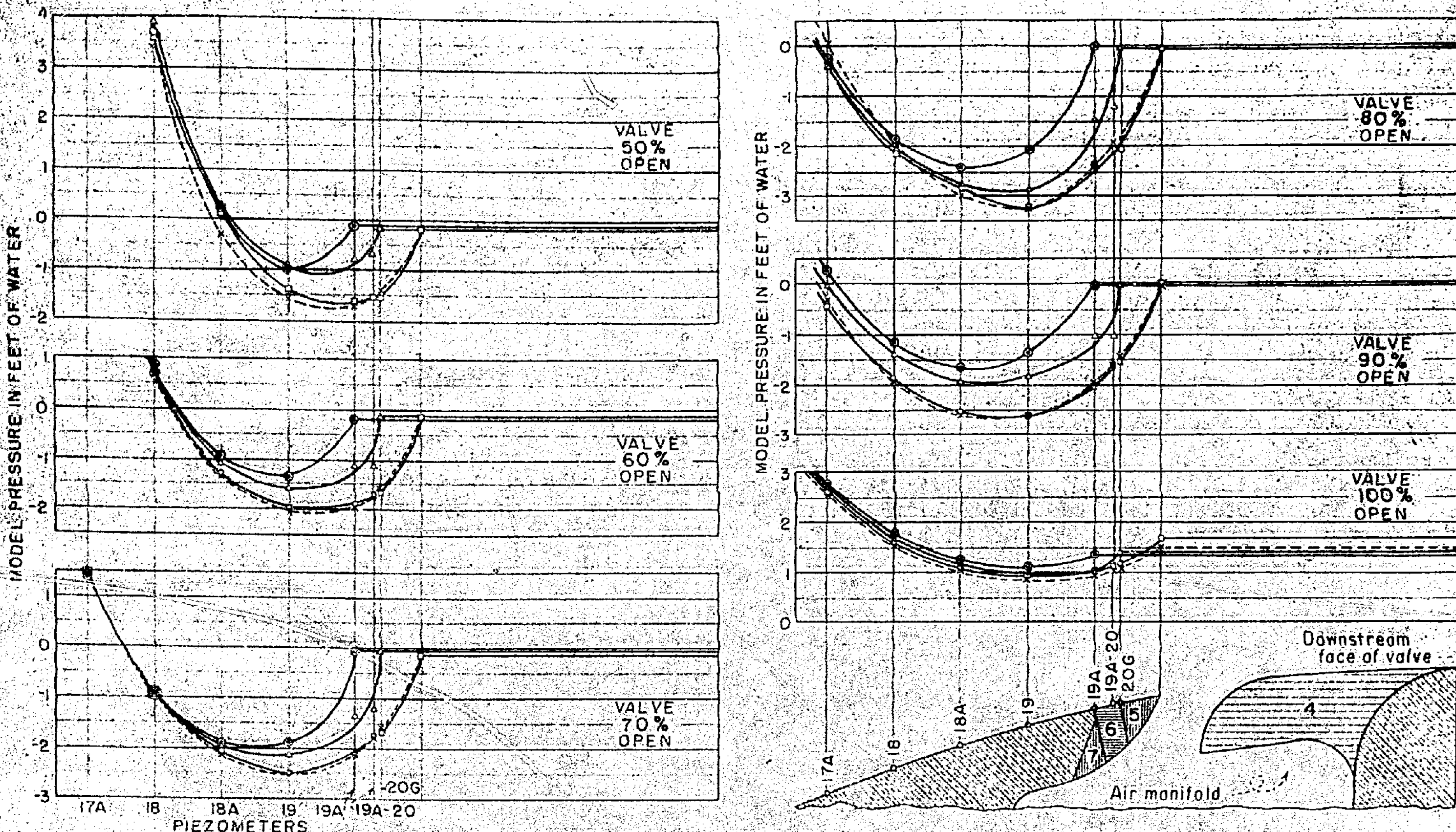
the throat of a venturi tube. This divergence was eliminated by shifting the center of the 8-inch lip radius as shown in figure 17F so that the lip and nozzle would coincide at closure and the new lip was designated "F" test 8FAS12. This change eliminated the low pressures at five percent opening, but the design was rejected because of the negative pressures at piezometers 19, 19A, and 20E for the 100 percent opening. Another factor in its rejection was the objectionable wedging action which would occur when the valve was closed.

35. Lip "E" reinstalled, new line of piezometers (Test 8EAS12B). A rather considerable time interval between these last tests and further developments of the valve design was spent in making confirmation tests with the 20- and the 6-inch models. As a part of these tests a new row of piezometer holes, row "B", was drilled to replace the former row "A" because the latter row showed indications of being too close to one of the supporting ribs. The piezometer openings were located similarly in each row but the rows were located in different longitudinal planes. A more complete discussion of this change and its effects will be taken up under the heading of confirmation tests and it is mentioned at this point only because both rows will be referred to in the following data and discussions.

36. Downstream lip of air box removed. The valve of test 8EAS12 was changed by removing the lip downstream from the air slot, portion marked 4 and shown in figure 17H, test 8EAS13. This lip was removed because it was preventing the free passage of air from the air box to the air space surrounding the jet immediately downstream from the valve. A choking effect was indicated by the difference in pressure between these two points in test 8EAS12. Referring to figure 27, it can be seen that the situation was materially relieved as indicated by the substantial increase in the quantity of air passing through the air box. The amount passing to the tube interior decreased because the additional relief provided through the air box allowed the jet to

close at a point closer to the valve and this decreased the air requirement of the jet interior. The change in the jet shape lowered the coefficient of discharge, C_T, as defined previously, from 69.4 to 68.5 percent but it imporved the stability of the jet at the small openings. In test 8EAS12, the shape of the jet changed from conical to cylindrical at openings below 10 percent whereas in 8EAS13 the jet retained its conical shape at all openings. The removal of the lip had practically no effect on the pressures through the critical zone as shown in figure 23 which are based on the pressures in row "B", of piezometers.

37. Nozzle cut-back. (Tests 8EAS14 and 8EAS15). The contour of the valve nozzle was derived initially from a calipered jet as explained previously. Therefore, it should be possible to terminate the nozzle contour at any point without materially affecting the coefficient of discharge for the valve 100 percent open. At the same time any cut-off of the nozzle would result in an increase in pressure at partial openings because the pressure at the point of cut-off would be forced to that of the air box with a reflected increase in pressure at points upstream. The nozzle was, therefore, cut back in two stages, marked 5 and 6, as shown in figure 17H, in order to have some record of the amount of cut-off versus increase in pressure. The first cut-off was designated as 8EAS14 and the second which was made just short of the seating ring was designated as 8EAS15. The changes in pressure due to each change are shown in figure 23. A significant improvement in pressure conditions had been accomplished but this had been done at the sacrifice of some discharge capacity. The discharge coefficient C_T which was substantially the same for both cut-offs was lowered to 67.4 percent, but the permissible range of operation for regulation was extended appreciably. The valve coefficient or C_v was not determined experimentally but it was estimated to have a value of 0.72 by using the head across the valve. In obtaining the head



EXPLANATION

SYMBOL	TEST
x- - - - -x	8EAS-12
□———□	8EAS-13
△———△	8EAS-14
•———•	8EAS-15
○———○	8EAS-16

NOTES

No sections removed for test 8EAS-12
Section #4 removed for test 8EAS-13
Sections #4 and 5 removed for test 8EAS-14
Sections #4,5 and 6 removed for test 8EAS-15
Sections #4,5,6 and 7 removed for test 8EAS-16

SHASTA TUBE VALVE

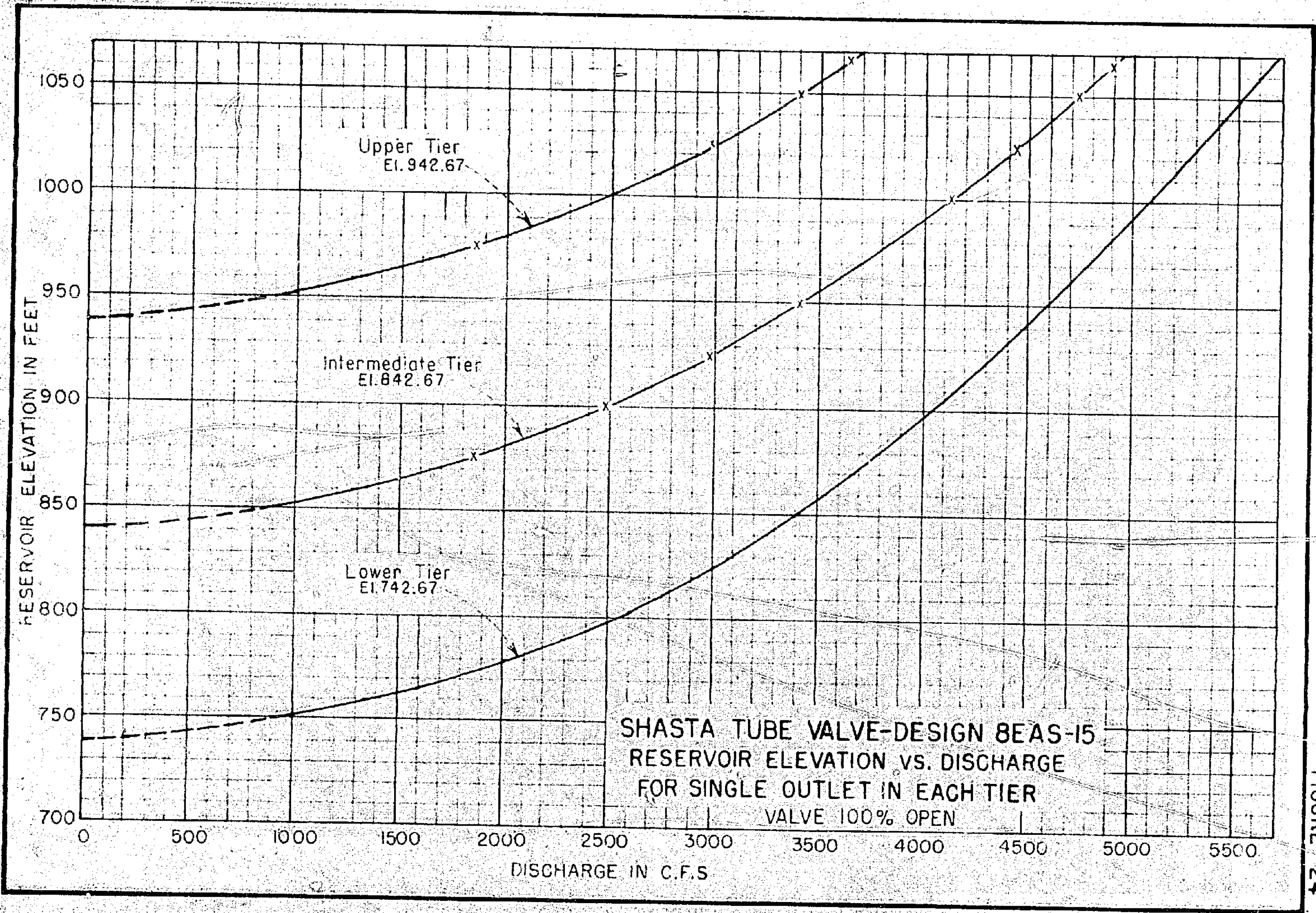
PRESSURE GRADIENT THROUGH CRITICAL ZONE

FOR TESTS 8EAS12 TO 8EAS16

across the valve the pressure head at the end of the valve was assumed to be equal to the pressure head measured in the air box. For any consideration of this design for the free discharge condition the coefficient of 0.72 should be used. The extension of the operating range for maximum head on the lower tier compared to the range of 8EAS12 is shown in figure 21 where the minimum pressure for each valve opening is shown. With the limit arbitrarily set at -1.5 feet model as explained previously, the range of permissible operation is indicated. The inoperative range was narrowed to the region from 63 to 91 percent opening. The portion of the curves below zero absolute apply only to the model and have been plotted only to aid in drawing the curves.

This design was not tested in the intermediate and lower position so the operating ranges cannot be determined from the model tests directly although they can be approximated from tests 8EAS112 and U12. It is proposed that the safe operating ranges for the prototype valves be determined in the prototype by means of piezometers provided in one valve of each tier. For purposes of comparison only, the negative ranges for the intermediate and upper tiers for design 8EAS15 are estimated to be from 75 to 80 percent for the former and no inoperative range for the latter.

The variation of discharge with reservoir level is shown for a valve, 100 percent open in each tier in figure 24. These curves were obtained from the model data in the following manner. In test 8EAS6 it was determined that for a valve in the lower tier the pressure in the conduit downstream from the valve dropped to zero when the reservoir level was reduced to 20 percent of its maximum level or when the reservoir level reached a level corresponding to elevation 806.2 feet, prototype. For reservoir levels lower than elevation 806.2 which will be designated as condition 1 the discharge was computed by using the coefficient C_v with the head measured from the center line of the valve to the reservoir level. For reservoir levels above 806.2,



Upper Tier
El. 942.67
Intermediate Tier
El. 842.67
Lower Tier
El. 742.67
SHASTA TUBE VALVE-DESIGN 8EAS-15
RESERVOIR ELEVATION VS. DISCHARGE
FOR SINGLE OUTLET IN EACH TIER
VALVE 100% OPEN
RESERVOIR ELEVATION IN FEET
DISCHARGE IN C.F.S
700
750
800
850
900
950
1000
1050
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
5500

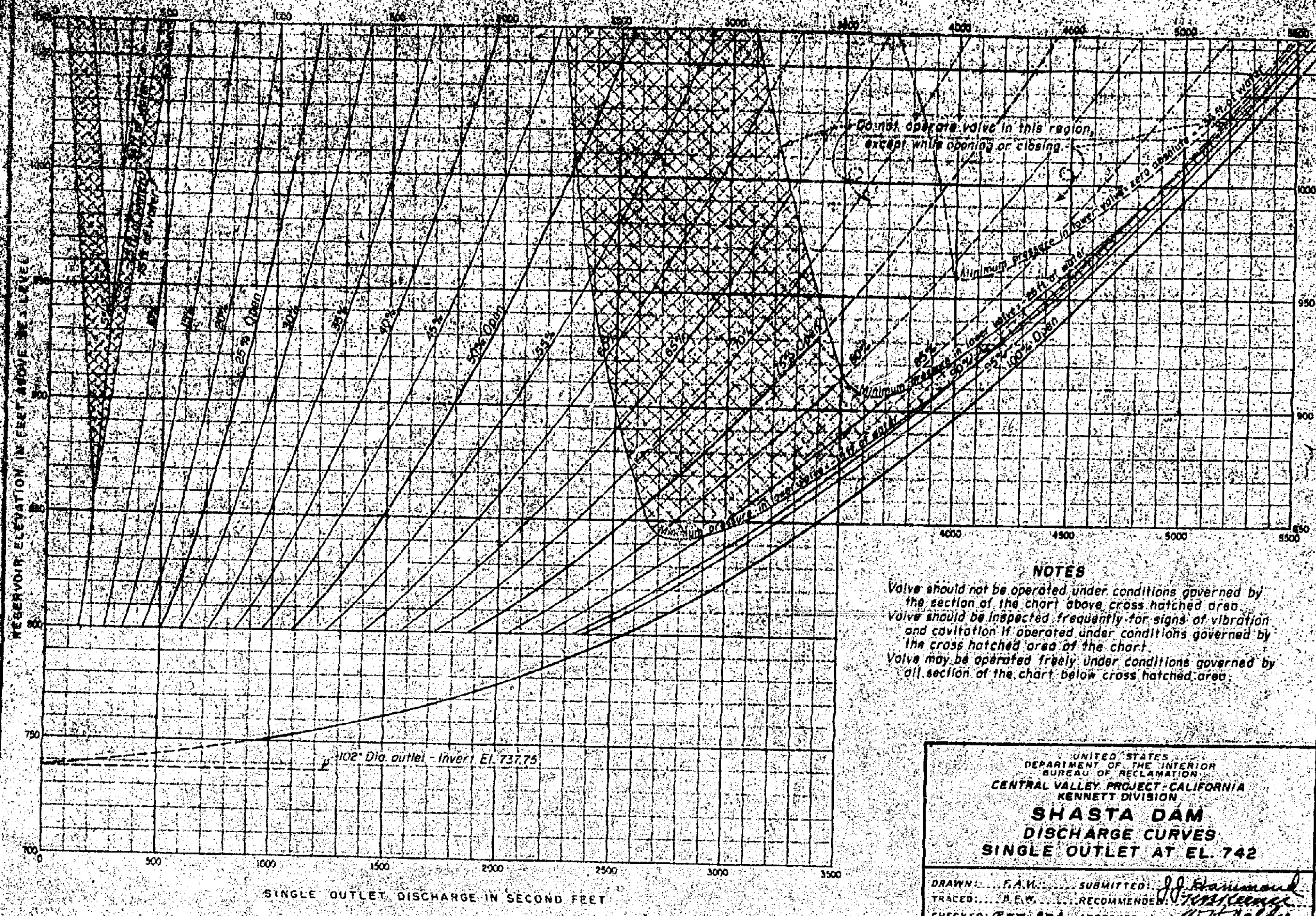
FIGURE 24

called condition 2, the coefficient C_T was used with the head measured from the reservoir level to the elevation of the discharge elbow. For valves in the upper and intermediate tiers condition 1 will prevail at all reservoir levels due to the shorter conduit downstream from the valve so the computation of discharge was based on the coefficient C_V.

The valves in the lower tier will be installed first and they will be required to operate under a gradually increasing head as the reservoir fills. The variation in permissible operating range for different reservoir surface elevations was compiled in graphical form for guidance of the field operators, as shown in figure 25.

The predicted values of air and water discharge for a prototype valve in the lower tier under a head of 323 feet are shown in figure 26. The portion of the total air which passes through the air box is also shown and the portion passing to the tube interior can be obtained by subtraction from the total. The prototype values of water discharge for the intermediate and lower tiers are also shown for the maximum water surface elevation of 1065.

The values of maximum air demand for the intermediate and upper tiers were obtained by applying the ratio of maximum air discharge to water discharge at the same opening as determined for the lower tier. This method yields the values of $(2200 \div 2000) \times 1600 = 1{,}760$ c.f.s. of air for the intermediate tier and $(2200 \div 2000) \times 1200 = 1{,}320$ c.f.s. of air for the upper tier. The size of the air piping could be reduced for the intermediate and upper tiers since their air requirements are smaller. The amount of reduction can be computed by selecting the size pipe which will have equal velocity considering the reduced air discharge. For equal velocities the diameters must have the ratio of the square root of the ratio of air discharges - $\sqrt{QA/QA}$ which will be for the intermediate tier, $D = \sqrt{1{,}760/2{,}200} \times 36 = 32$ inches; and for the upper tier, $D = \sqrt{1{,}320/2{,}200} \times 36 = 28$ inches.



NOTES

Valve should not be operated under conditions governed by the section of the chart above cross hatched area.

Valve should be inspected frequently for signs of vibration and cavitation if operated under conditions governed by the cross hatched area of the chart.

Valve may be operated freely under conditions governed by all section of the chart below cross hatched area.

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL VALLEY PROJECT-CALIFORNIA
KENNETT DIVISION

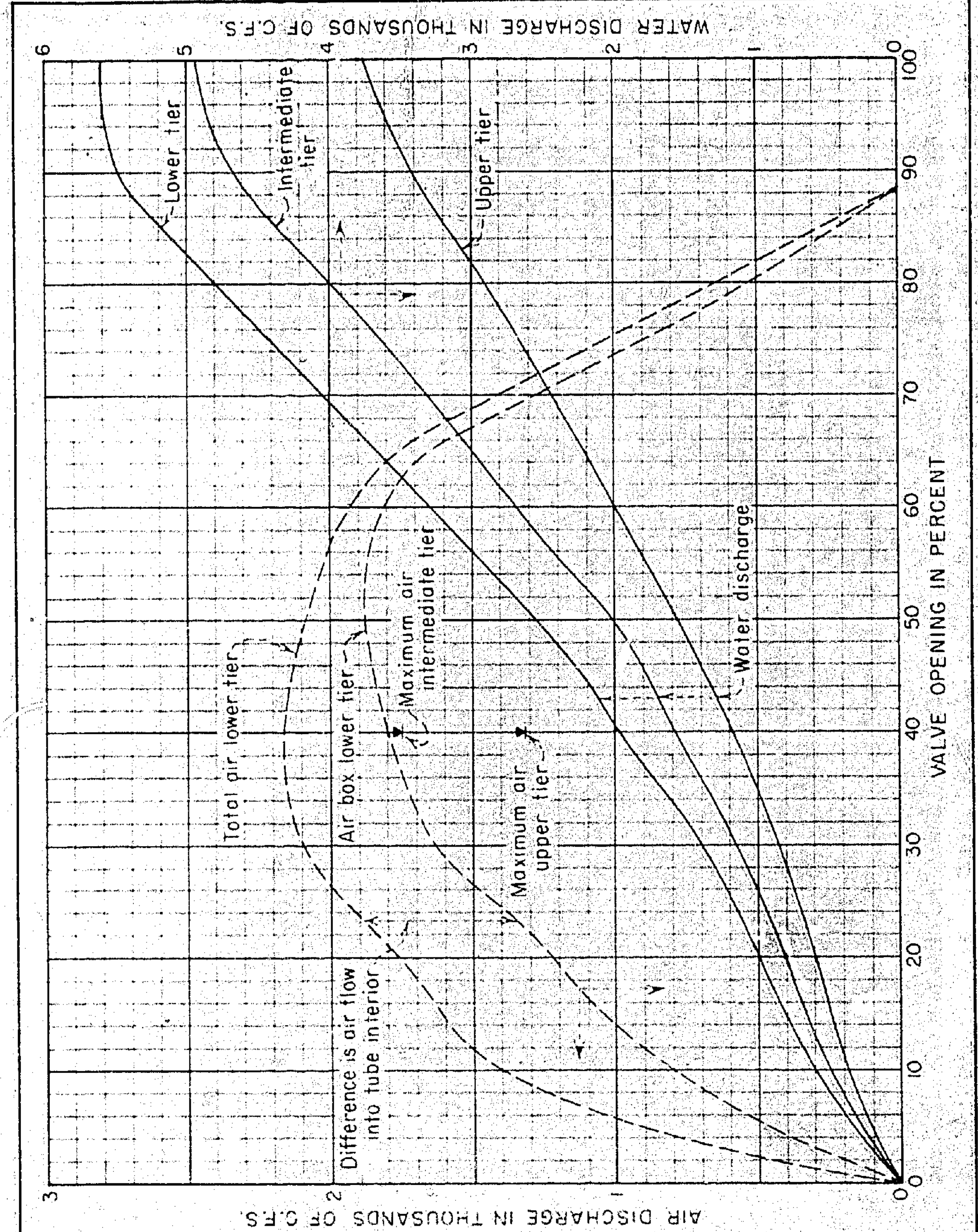
SHASTA DAM
DISCHARGE CURVES
SINGLE OUTLET AT EL. 742

DRAWN: F.A.W. SUBMITTED: J.J. Hammond
TRACED: R.E.W. RECOMMENDED: [signature]
CHECKED: [illegible] APPROVED: W.H. Nalder

DENVER, COLORADO - JUNE 20, 1942 | 214-D-10951

FIGURE 25

FIGURE 26



SHASTA TUBE VALVE-DESIGN 8EAS-15

PROTOTYPE VALUES OF AIR AND WATER DISCHARGES

FOR MAXIMUM RESERVOIR ELEVATION 1065

In designing the prototype air piping the maximum pressure head in the air box must be known and these values are given herewith as 23.0 feet of water for the lower tiers, zero for the intermediate tier, and slight vacuum of approximately 0.2 feet of water for the upper tier.

38. Angle of cut-off changed (Test 8EAS16). This test differed from the previous test in the angle at which the cut-off was made. The portion marked 7 as in figure 17H was removed. The original angle of 8EAS15 was maintained for a distance of 3/64 inch and the remainder was machined to an angle of 70 degrees with the axis of the valve. Tests were run at openings of 100 to 50 percent, inclusive, and a comparison made with 8EAS15. No difference in pressure could be detected as shown in figure 23.

VI - AIR STUDIES

42. General discussion. After it had been established in the earlier tests that a considerable quantity of air must be provided to relieve low pressures in the valve, the question arose as to how much air was required to accomplish this purpose. No attempt was made to answer this question until a design was evolved which operated at full opening with no negative pressures. Therefore, no measurements of air were made previous to test SEA4. For this test and all subsequent tests the quantity of air was measured and this quantity proviede not only a basis for designing the air piping for the prototype but also a basis for evaluating the effectiveness of changes in design when it was considered in conjunction with the measurements of subatmospheric pressures. The use of the measured air discharge in the latter sense was possible because there was a direct connection between the magnitude of the negative pressures and the discharge of air pumped by the valve.

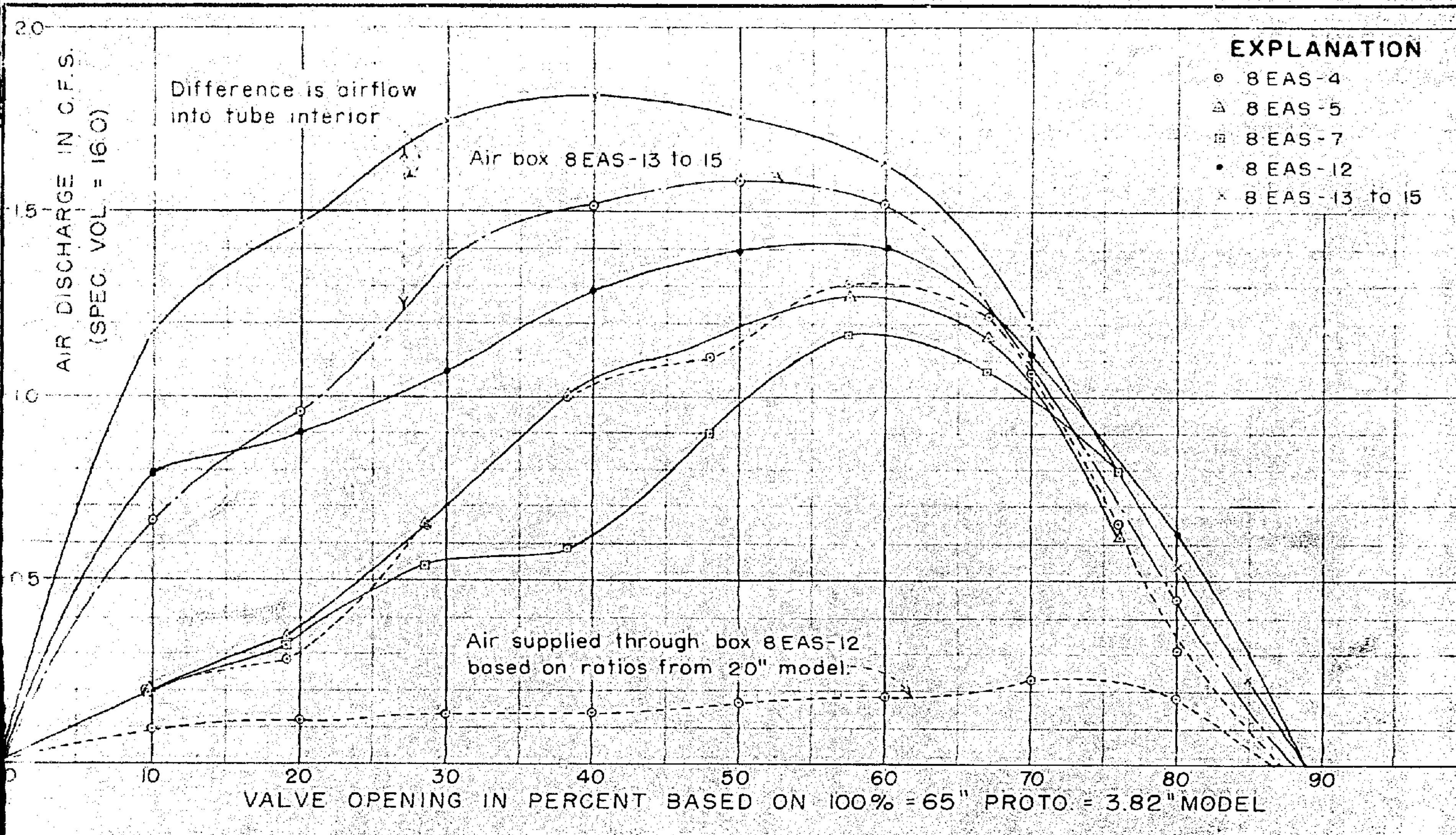
The purpose in admitting air was to relieve negative pressures so if any change in design resulted in an increase in air discharge there was always an improvement in pressure conditions at some point in the valve. This statement is perhaps oversimplified and further discussion is necessary because the behavior and effects of the air supplied from the air box differed from those of the air supplied to the tube interior.

The magnitude of the vacuum in the space surrounding the jet is a function of the quantity of air being discharged through the air box which in turn is equal to the quantity being pumped by the jet of water as it expands to fill the conduit at some point downstream. In the case of the tube interior the quantity of air itself is a function of the pressure within the tube because the air being pumped depends on the shape of the hollow cone which comprises the interior of the jet and this shape in turn depends on the difference in pressure between the interior and the exterior of the jet. This interrelationship was demonstrated by arbitrarily constricting the supply to the tube interior which resulted in a decrease in air demand for the tube. On the other hand a constriction in the supply of the air box should not affect its air demand to any great extent.

In the case of SEAS12 the two separate air supply lines were more interdependent than this rather idealized description would indicate. This was due to the fact that the passage from the air box to the conduit downstream from the valve was constricted by the exterior surface of the jet. There was insufficient clearance between the valve contour and the jet downstream from the air slot. Any change in the shape of the jet, such as that produced by constricting either the tube or the air box supply, produced a change in the relation between pressure and air discharge through the air box because of a change in the area of the passage through which this air must be supplied.

In the preceding discussion the amount of air being pumped per unit of time has been referred to as the air demand. The word demand is somewhat of a misnomer as applied to the problem of a valve discharging into a conduit because the air supplied to the valve represents only a partial fulfillment of the actual demand. The magnitudes of the negative pressures are a good index of the discrepancy between supply and demand. There must always be some negative pressure to supply the potential for moving the air but when the negative pressures exceed this minimum value some barrier must be present.

43. Air discharges for progressive valve changes (SEA4 to SEAS12). The changes in air discharge with progressive changes in valve design are shown in figure 27. All values of air discharge are given in model terms in order to avoid the complications introduced by the phenomenon of critical pressure. This phenomenon stated succinctly is that when air is flowing under a differential pressure, measured from atmospheric conditions, a maximum weight discharge is attained when the differential reaches a value equal to approximately one-half atmospheric pressure.



NOTE

All values of air discharge represent total quantities unless otherwise noted.

COMPARISON OF AIR DEMANDS FOR DIFFERENT DESIGNS AT 19.0 FT. HEAD IN 1:17 MODEL

FIGURE 27

The possibility of critical pressure existing at some point in the air relief system was one of the major reasons for building and testing the 20-inch model at Boulder Dam. Critical pressures were encountered in some of the high head tests on the 20-inch valve.

44. Remedy of choking action of air box lip. As the area of the air inlets was increased the amount of air taken by the tube interior increased but the amount taken through the air box did not increase in the same proportion. The reason for this condition was the insufficient clearance between the valve boundary and the jet as discussed previously. When this condition was remedied by removing the lip which formed the downstream protion of the air slot, tests 8EAS13 to 8EAS15, the total quantity of air was materially increased as shown in figure 27. Also, the pressure below the valve was increased substantially and was only a slight vacuum at all valve openings. Further effects of this change were the decrease in the quantity of air required by the tube and the increase in pressure inside the tube.

The negative pressures at all points where air relief would be effective were so small for this test that any further benefit which could be obtained by increasing the size of the air piping would be quite small and would not be economically justified. The relatively small portion of the total air supply which passes through the tube interior, as shown in figure 27, indicates that the supply lines to the tube interior can be materially reduced without appreciably changing the pressure inside the tube.

45. Recommendations for prototype air piping. In making a recommendation for the sizes of the air supply lines for the prototype by scaling up the model results it is necessary to consider the effects of differences in air density. These effects will be discussed as a part of the comparison between the 6- and 20-inch models and it is sufficient at this point to merely state that they were neglected in scaling up model results.

The sizes of the air supply pipes were established from the air discharge data of tests SEA6 and BSEA6 by limiting the velocity of the air to a maximum of 300 feet per second. For the air duct design intended for valve design SEAS12 the air box is supplied by one pipe and the tube interior by two pipes of equal diameter. The arrangement of air supply piping is shown in figure 28. The sizes were determined to be a 36-inch diameter pipe for the former and two 20-inch diameter pipes for the latter. In obtaining the air velocities in the tube supply pipes the air discharge measured in the model and scaled to prototype terms was used. In the case of the air box a computed air discharge was used. It was obtained by assuming a coefficient for the air holes into the nozzle of SEA6 and by assuming a drop in head of 16 feet, water, which amounts to assuming critical pressure to exist. The discharge computed in this manner was considerably higher than the discharge indicated by the model so the pipe size, 36-inch diameter, resulting from its use was very generously proportioned particularly for design SEAS12 in which the proportion of the total air demand passing through the air box was less than for design SEAS6 and even at its maximum value it amounted to only 20 percent of the total air discharge. The prototype air velocities for SEAS12 computed by dividing air discharges, scaled up from model measurements, by the area of the pipes would be 38 feet per second maximum in the 36-inch pipe and 320 feet per second maximum in the 20-inch pipes. It is evident that for this design (SEAS12) the air box supply should be materially decreased in size whereas the tube supply should be increased in size to reduce the velocities. The latter change could be made only by increasing the number of hollow ribs which convey air to the tube interior. For designs SEAS13 to 15 the situation was changed completely and the prototype velocities computed in the same manner would be 266 feet per second, maximum, in the 36-inch pipe and 55.0 feet per second in the two 20-inch pipes. The entire capacity of the 36-inch pipe could be utilized but the tube

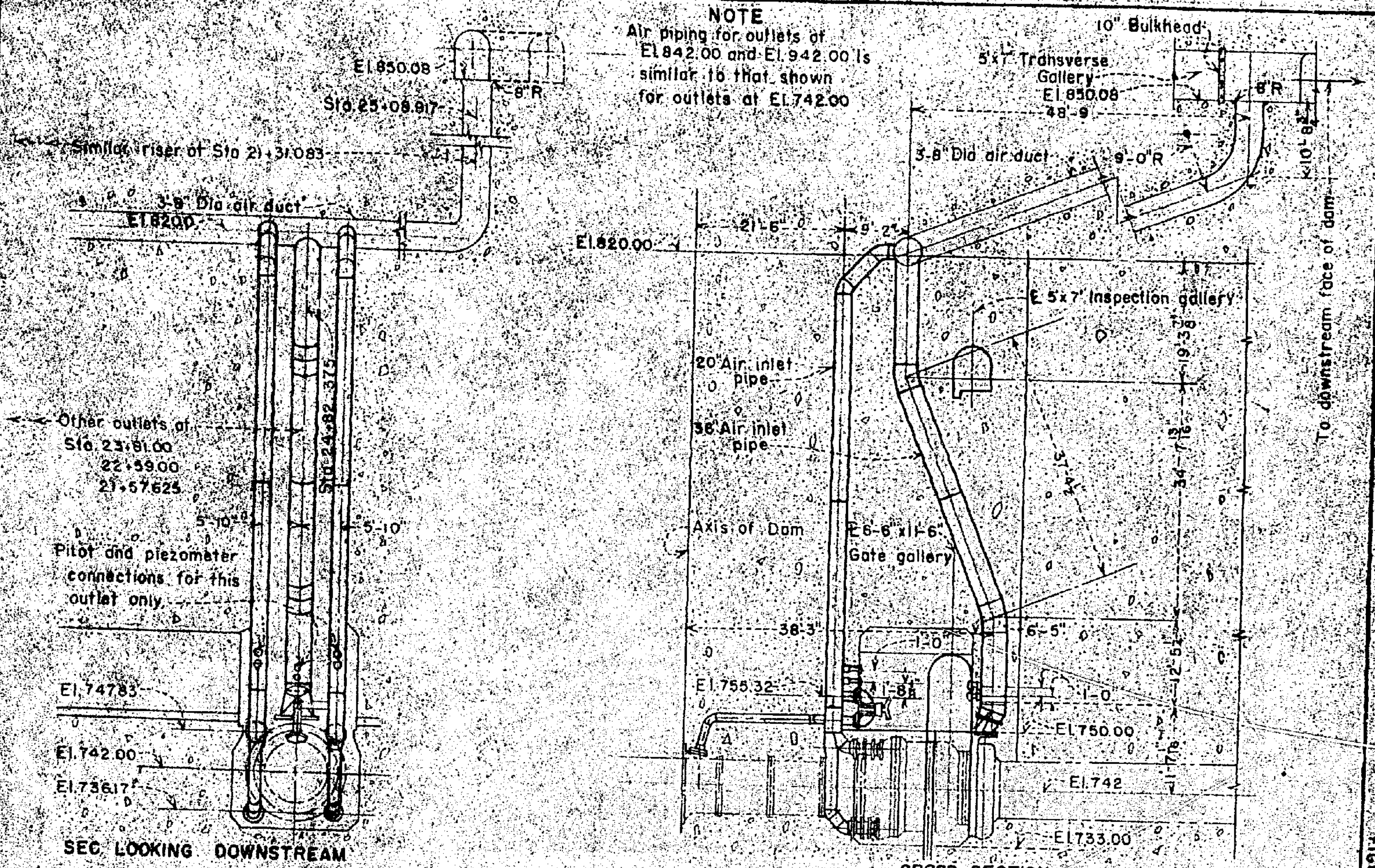


FIGURE 28

SHASTA TUBE VALVE
ARRANGEMENT OF AIR SUPPLY PIPING FOR VALVE AT ELEV. 742

supply could be reduced. If one 20-inch diameter pipe were used the velocity in it would be twice the value for two pipes or 110 feet per second. Therefore, one 20-inch pipe would be adequate for the tube supply and by the same token only two of the present four hollow ribs would be necessary.

All estimates for prototype air piping sizes were based on valves in the lower tier for maximum head of 323 feet. Since this is the worst condition the sizes of air piping will be more than adequate for the intermediate and upper tiers.

The drop in pressure, from that of the atmosphere, which would be required to supply the necessary air relief through the prototype piping was computed for the condition of the two inside valves in the lower tier operating under maximum head at the opening which requires maximum air discharge. Assuming an atmosphere of specific weight equal to 0.075 pounds per cubic foot the various losses of head through the supply piping add up to a drop in head of 1.90 foot of water to the air box and 0.8 foot of water to the tube interior. The pressure in the air box may, therefore, be expected to reach a vacuum of nearly 2.0 feet of water. The value of this vacuum obtained by scaling up the negative pressure measured in the 6-inch model was 3.4 feet of water. The difference in the two values is due to the relatively smaller frictional loss in the prototype, because of the higher Reynolds number. The operating characteristics of the prototype valves were based upon an air box pressure of -3.4 feet of water as obtained from the model measurements. The fact that the prototype pressure will be higher by approximately one foot of water will be on the safe side and will operate as a small factor of safety.

VII. - STUDIES OF FRIANT NEEDLE AND TUBE VALVES

46. Purpose of studies. The development of the Shasta tube valve to this point has been directed by the criterion of a high coefficient of discharge which was maintained at the unusually high value of 0.70 or better. It was decided that it would be profitable to study two valve designs which had been developed by separate model studies for operation as free discharge valves at Friant Dam. The first few tests of these studies were made with one of the preliminary Shasta tube valve designs and have been described previously, see section II. There were two types developed, one a needle valve, and the other a tube valve, with discharge coefficients of 0.59 and 0.51, respectively.*

47. Model scale and apparatus. Both valve models are shown in figure 29 and it will be noted that although both valves have a 6-inch diameter inlet, the outlet diameters differ and in both cases are less than 6 inches. The smaller outlet diameters of these valves necessitated a smaller conduit downstream than the 6-inch diameter conduit used in the previous tests. The new conduit size was obtained by equating the diameter at the end of the discharge cone to the jet diameter of the needle valve at 100 percent opening so that the conduit would flow full at this opening. This resulted in a scale ratio of 1:20 which makes the diameter of the model conduit 5.10 inches. Through some error the diameter of the conduit was made only 5.01 inches which gives a scale ratio for the conduit of 1:20.36. Using the latter scale ratio based upon the actual diameter of the model conduit, its length and the diameter at the end of the discharge elbow which were sized according to the ratio 1:20 exceeded their proper values by 1.8 percent. The excess in length of conduit was unimportant as the additional friction loss introduced by the extra length was very small. The excess in the diameter at the end of the conduit cannot be

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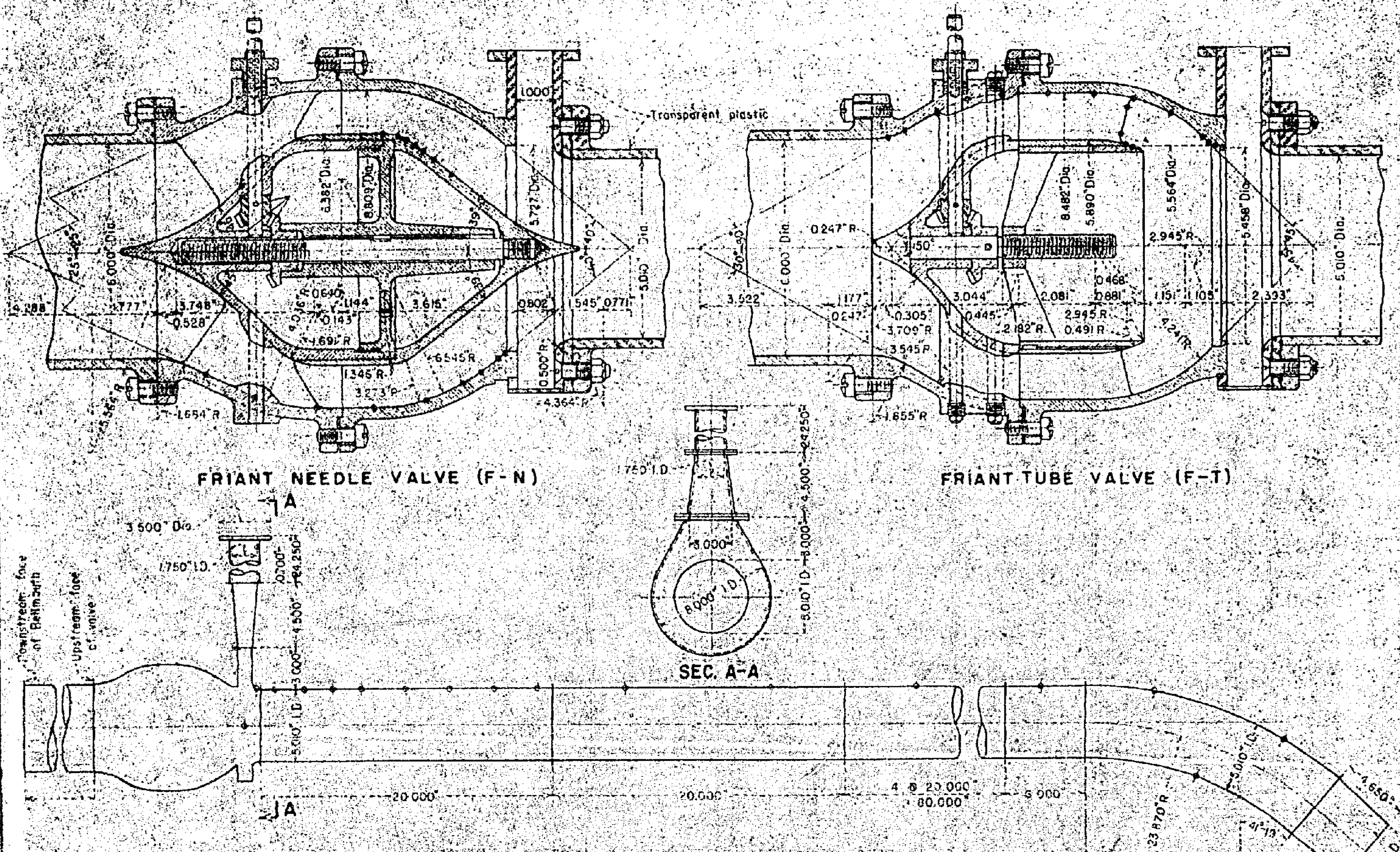
* Hydraulic Laboratory Report No. Hyd. 163.



INSTALLATION OF FRIANT NEEDLE AND
TUBE VALUE MODELS IN A CONDUIT

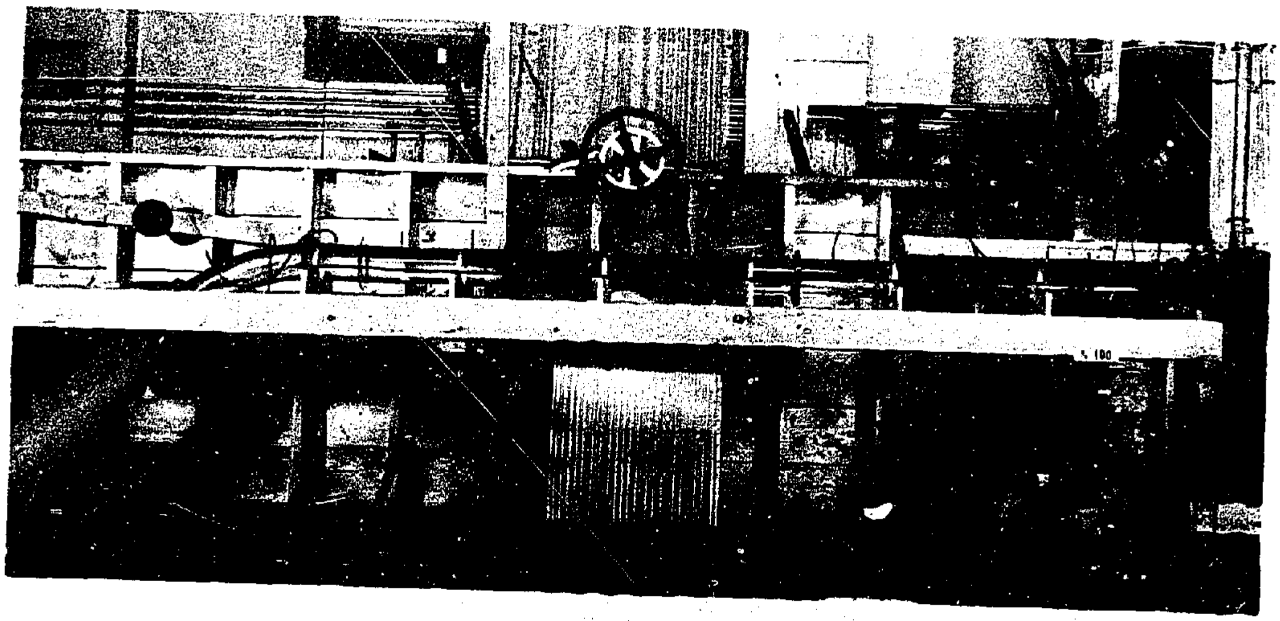
FIGURE 29

neglected because the back pressure created by the terminal cone is very sensitive to changes in diameter. The back pressure on the conduit varies inversely as the fourth power of the ratio of the diameters which gives $(1/1.018)^4 = 0.9312$ for the ratio of back pressures. With the model as constructed, the back pressure is then 7.9 percent lower than it should be and any comparisons must take this discrepancy into account. The conduit upstream from the valve was also out of scale but this was ignored since it has very little influence on the hydraulic action in and downstream from the valve. The valve was connected to the conduit by an air box which slipped over the end of the valve and was faired into the conduit to make an easy passageway for the relief air as shown in figure 29. With the exception of the discrepancies already noted the conduit downstream from the valve was similar to that of the previous tests. It was made of transparent plastic, as shown in figure 30, to permit observation of the downstream conditions. The Friant tube valve was tested with the same conduit although it was realized that the conduit would not flow full at 100 percent valve opening because of the lower discharge coefficient of the valve.

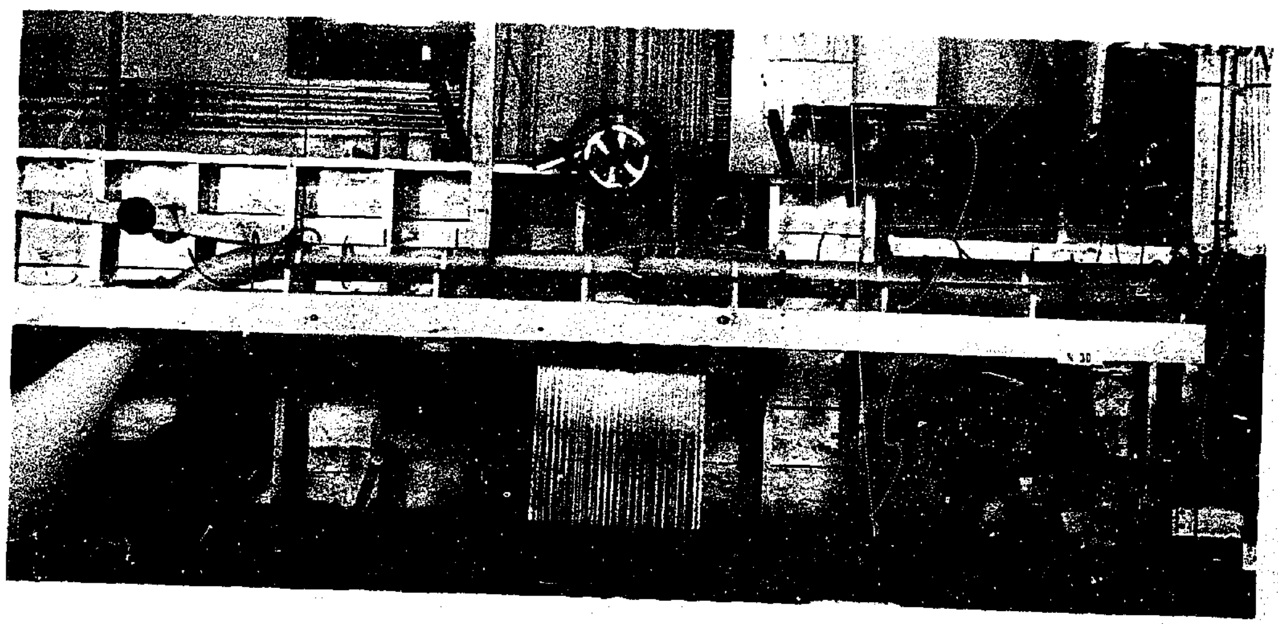
48. Test of Friant Needle Valve in a conduit. Since a complete set of tests had been made of both valves for free discharge conditions*, only a limited number of tests were made to establish the changes due to placing the valves in a conduit test F-N. In the first series, tests were made for valve openings of 100, 80, 60, 40, and 20 percent at four different heads which were not set to any scale but resulted from different settings of the variable-speed motor driving the pump. Also in this series, tests were run for valve openings of 90, 70, 50, 30, 10, and 5 percent but only at one head. A second series of tests was run in which only the head on the valve, discharges of air and water and air box pressures were measured. In this series the needle valve was

*Loc. Cit.

FIGURE 30



100-Percent open



30-Percent open, maximum air demand

FRIANT NEEDLE VALVE INSTALLED IN CONDUIT tested at six settings of the pump motor for openings of 10, 20, 25, 30, 35, 40, 50, 60, 70, 80, and 90 percent. The tube valve was tested at the same settings and valve openings with additional tests at openings of 15 and 100 percent.

To simplify comparisons with the previous tests the Friant needle valve will be referred to as test F-N and the Friant tube valve will be referred to as F-T.

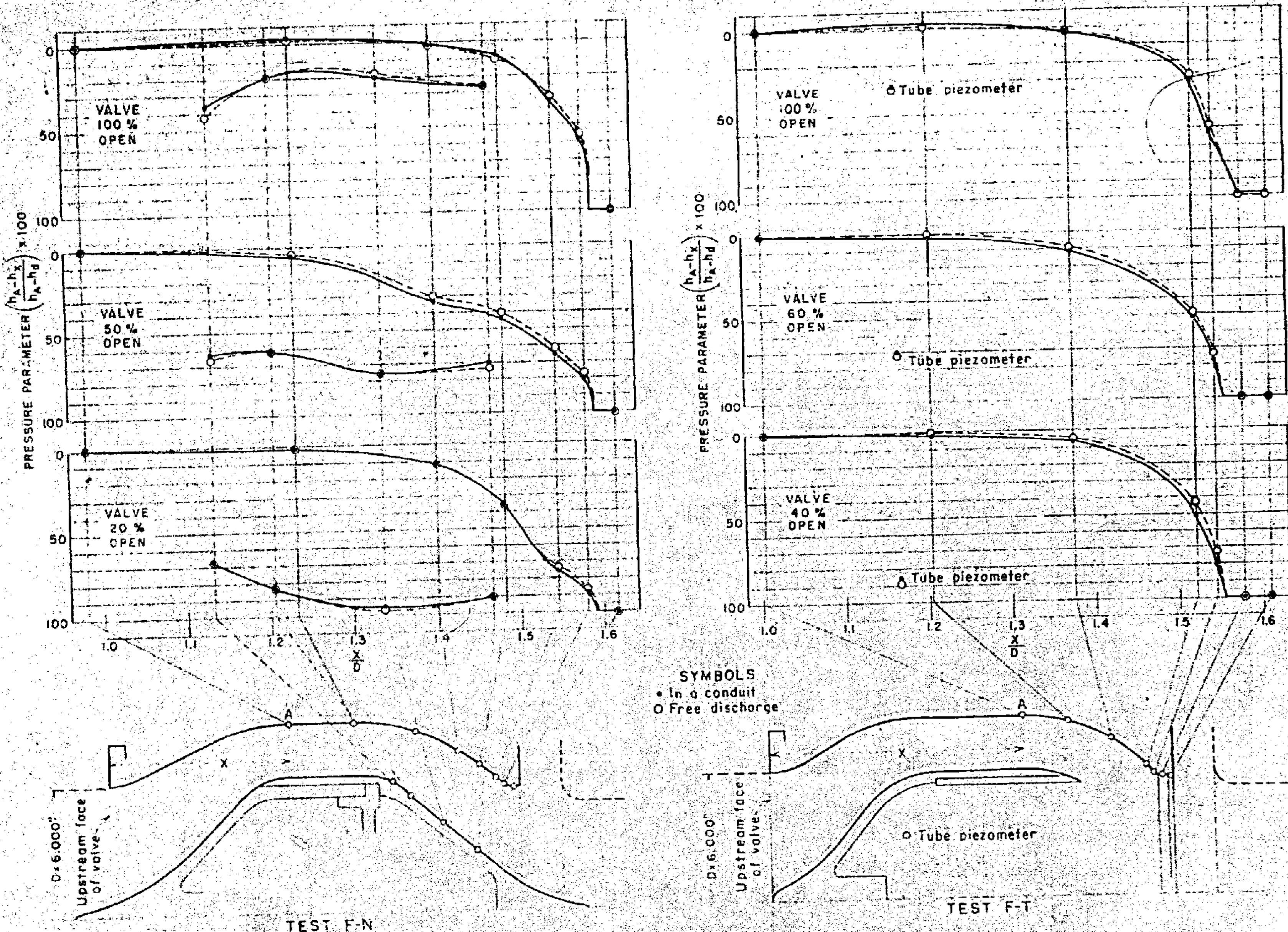
The needle valve operated satisfactorily at all openings and at 100 and 90 percent the conduit filled with a positive pressure in the air box. The pressure in the air box became negative at 70 percent and for all smaller openings the pressure was below atmospheric and the water jet pumped air into the conduit. The pressures along the valve boundaries were positive at all openings although somewhat less than those for free discharge conditions for openings less than 70 percent because the valve was discharging into a partial vacuum.

With the exception of the reduced pressure at the discharge end of the valve there was no difference in the performance of the valve attributable to operation in a conduit. Because of the distortion of the model conduit with its influence upon pressure conditions, it would be rather difficult to compare pressures directly for the two conditions, free discharge and discharge into a conduit. To avoid this difficulty the comparison was made on the basis of a dimensionless number which is now developed. Let it be assumed that the flow through the valve, that is the pattern of streamlines by which the flow and the resultant pressure distribution can be represented, is the same for free discharge and for discharge into a conduit. It can be demonstrated that the drop in pressure between any two piezometers will be a constant percentage of the total pressure drop through the valve. Expressed mathematically this would be for points A, X, and C:

$$\frac{h_A - h_X}{h_A - h_C} = \text{constant}$$

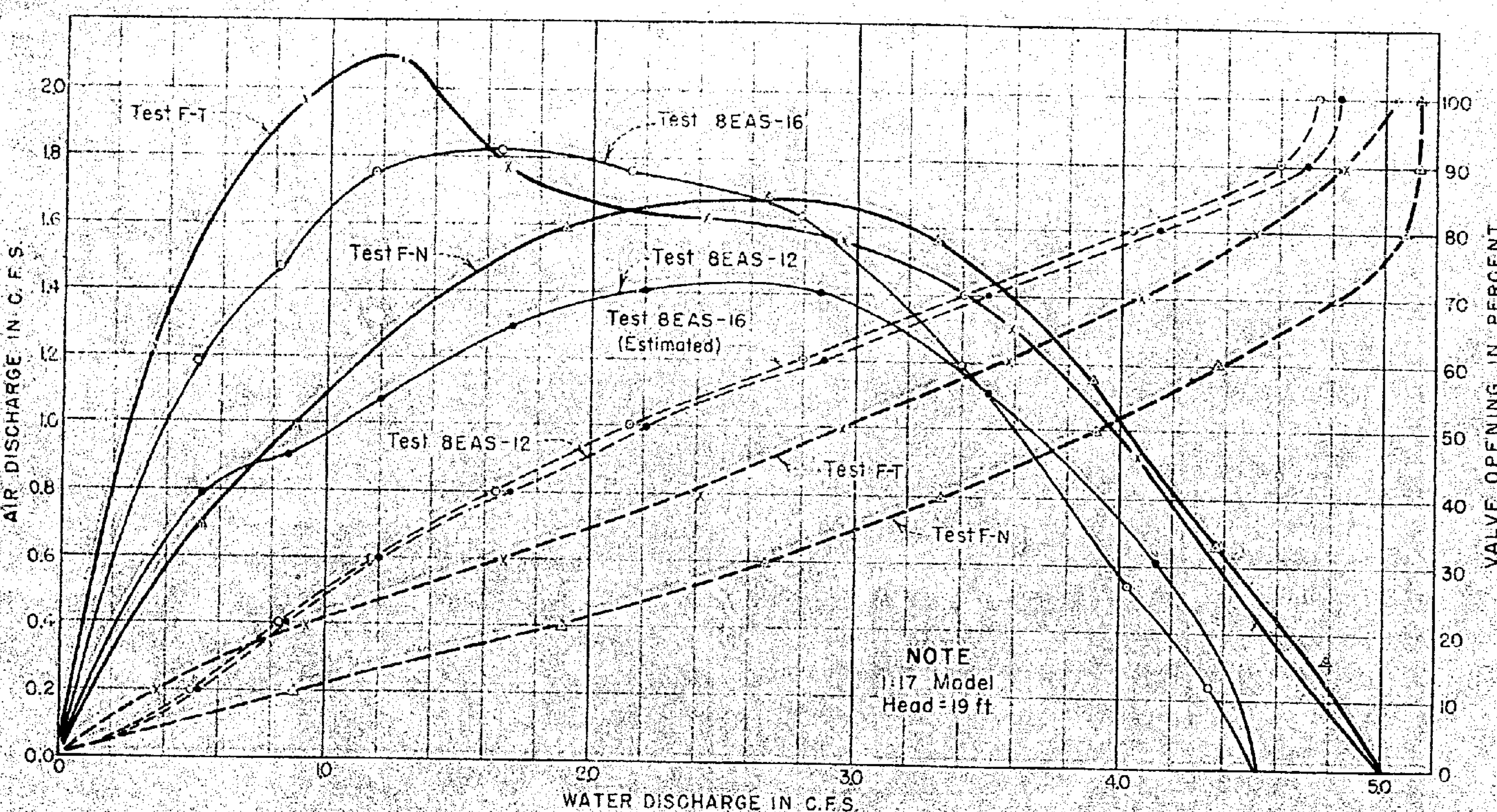
If point (C) is taken just downstream from the end of the valve, the partial vacuum in the conduit as indicated by the air box pressure will be included in the dimensionless ratio hereafter referred to as the pressure parameter. Point (A) would serve best if taken at the piezometer one diameter upstream from the valve. This location is generally used for determinations of discharge coefficient. Since pressures at this piezometer were not observed in either F-N or F-T, point (A) was taken at a piezometer located at approximately the midpoint of the valve where the boundary is straight and the effect of curvilinear flow is a minimum. Point (X) is any other piezometer and the locations of those used in the comparisons are shown in figure 31. Values of the non-dimensional pressure paramenter for piezometers in the critical zone are compared for free discharge and discharge into a conduit in figure 31 for 100, 50, and 20 percent openings. The agreement at 50 and 20 percent is very good and somewhat better than at 100 percent. This may be explained by the fact that when the conduit filled at 100 percent a jump occurred at the end of the valve and the pressure surrounding the jet was not uniform as in the case of free discharge or discharge into the conduit at partial valve opening when the conduit is not filled at the end of the valve. The agreement proves definitely that placing a valve of this type in a conduit has no effect on the pressure gradient other than lowering the end point by the amount of vacuum required to furnish a potential for supplying the relief air. It follows then that, for a valve which is designed to have no negative pressures under free discharge conditions, the lowest pressure when the valve is placed in a conduit will be the vacuum at the beginning of the conduit into which the valve is discharging.

The magnitude of the vacuum below the valve is some complex function of resistance of the conduit, the manner of admitting air, the quality of the water jet, and the valve opening. In test F-N the only variables were the valve opening and the quality of the jet



COMPARISON OF PRESSURES FOR FRIANT VALVES WITH AND WITHOUT CONDUIT and the latter was practically a constant since the jet was smooth and stable at all openings. The measurements of air discharge for the various valve openings under the different heads were converted for comparison with the previous tests by the following procedure. Air discharge was plotted against head for constant valve openings and the value of air discharge at 19-foot head was picked off. This value was then multiplied by the factor $(6.00/5.01)^2$ to give the results which would have been obtained if the model for test F-N had been built to the same scale (1:17) as the previous models. This scaling-up of the test results does not account for the fact that the back pressure is too low because the end cone is oversize. A comparison of the scaled-up results with tests SEAS12 and SEAS16 is shown in figure 32. The comparison is made on the basis of discharge (water) rather than valve opening because the latter loses its significance when different types of valves are being compared. The variation in discharge of water with valve opening for the different types of valves is also shown in figure 32.

The air discharge for F-N is higher than SEAS12 which had a restricted air relief and lower than SEAS16 because the jet was smoother at the lower openings, see figure 32. It is apparent from figure 32 that the resistance of the conduit was too low for test F-N since the water discharge for zero air discharge was higher than in test SEAS16. The curves have the characteristic that the air discharge increases with opening until a maximum is reached at some discharge of water which depends on the quality of the jet. Then the air discharge decreases until it finally reaches zero when the water discharge is sufficient to fill the conduit. If the conduits had been similar this point of zero air demand would be the same for both tests so the curve for test F-N can be shifted down to agree with SEAS16 at this point. The amount of downward shift in the rest of the curve would be difficult to determine but it would be expected to decrease with increasing air demand and to practially disappear at a water discharge



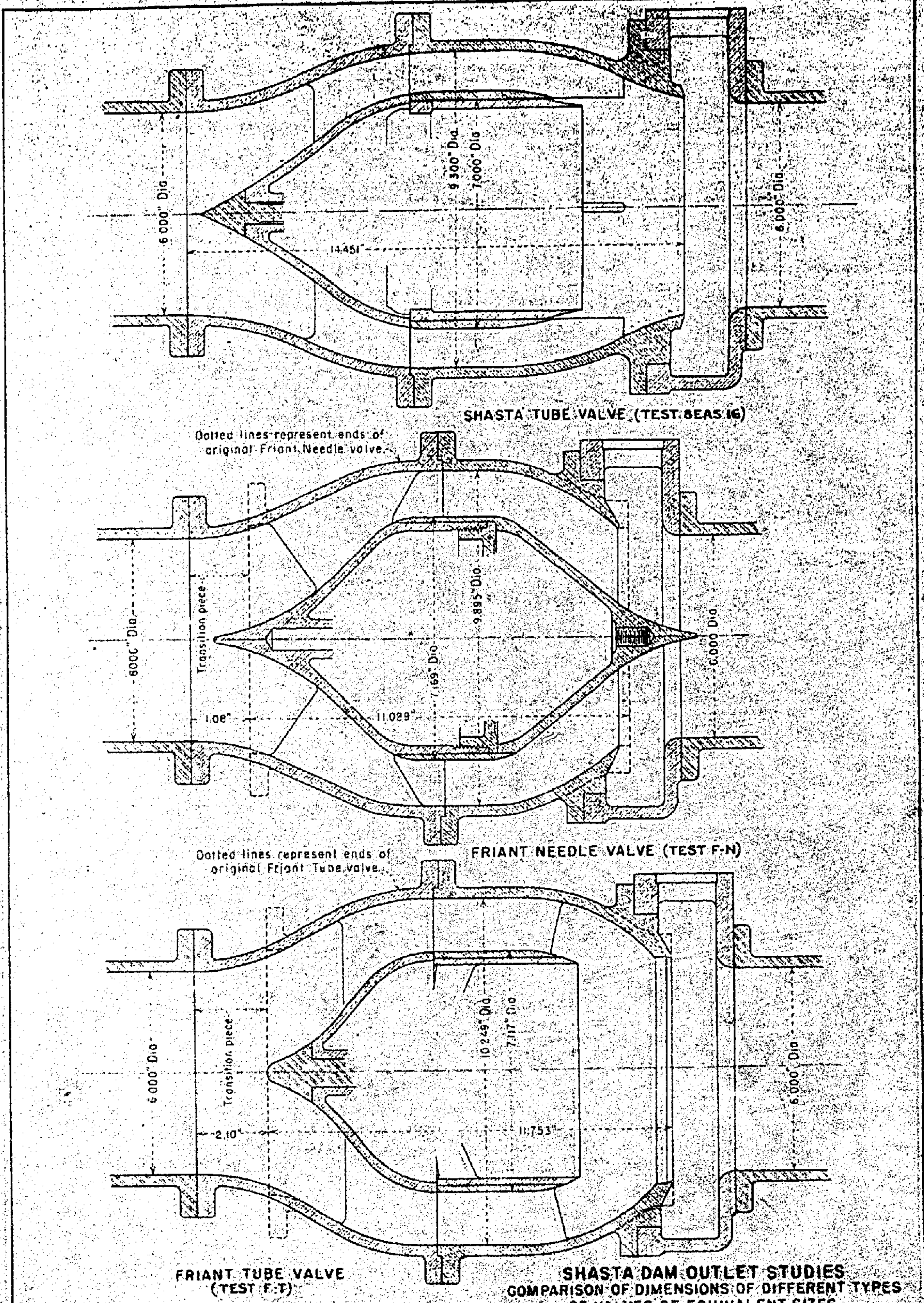
EXPLANATION

——— Air discharge vs. water discharge
– – – Valve opening vs. water discharge

AIR AND WATER DISCHARGE FOR FRIANT NEEDLE AND TUBE VALVES of 2.5 c.f.s. because in both curves it appears that for discharges less than 2.5 c.f.s. the quality of the jet is the controlling factor whereas for discharges greater than 2.5 c.f.s. the conduit assumes control. It was, therefore, assumed that the maximum air discharge was substantially correct as measured for test F-N. The maximum occurred at an opening of 30 percent and the vacuum reached a value of 0.24 feet for a head of 19 feet which corresponds to 0.24 x 17 or a vacuum of 4.1 feet in the prototype for a head of 323 feet. This vacuum will have to be increased in the prototype by the amount of loss in the air piping but this increase will be small and the magnitude of the vacuum in the prototype would be approximately the same as that for SEAS16 since the air demand is approximately the same.

To complete the comparison of F-N with SEAS16 it is necessary to discuss the relative sizes of the two designs. As already explained the size of the F-N model was overdesigned with the result that even though the back pressure was low the conduit filled at an opening of less than 80 percent. For a true comparison, valve F-N must be scaled to a size such that it would discharge the same quantity of water as SEAS16 under the same head. This adjustment can be made by means of the discharge coefficients, based on inlet area, of the two designs for free discharge conditions. The coefficient for the F-N valve was determined in previous studies as 0.59. The free discharge coefficient (C_v) for SEAS15 was not measured but its value has been estimated previously as 0.72. To pass equal quantities of water under the same head the inlet diameters must be inversely proportional to the square root of the ratio of free discharge coeffici[illegible] and since the inlet diameter of SEAS15 is 6 inches; the diameter of the inlet for F-N would be $\sqrt{0.72/0.59} \times 6.00 = 6.63$ inches. All other dimensions of the valve must be increased by the ratio 6.63/6.00 where 6.00 inches is the diameter of the model for test F-N. The needle valve with these new dimensions is compared for size with SEAS15 in figure 33. The reduction in diameter from 6.63 to 6.00 inches to connect with

FIGURE 33



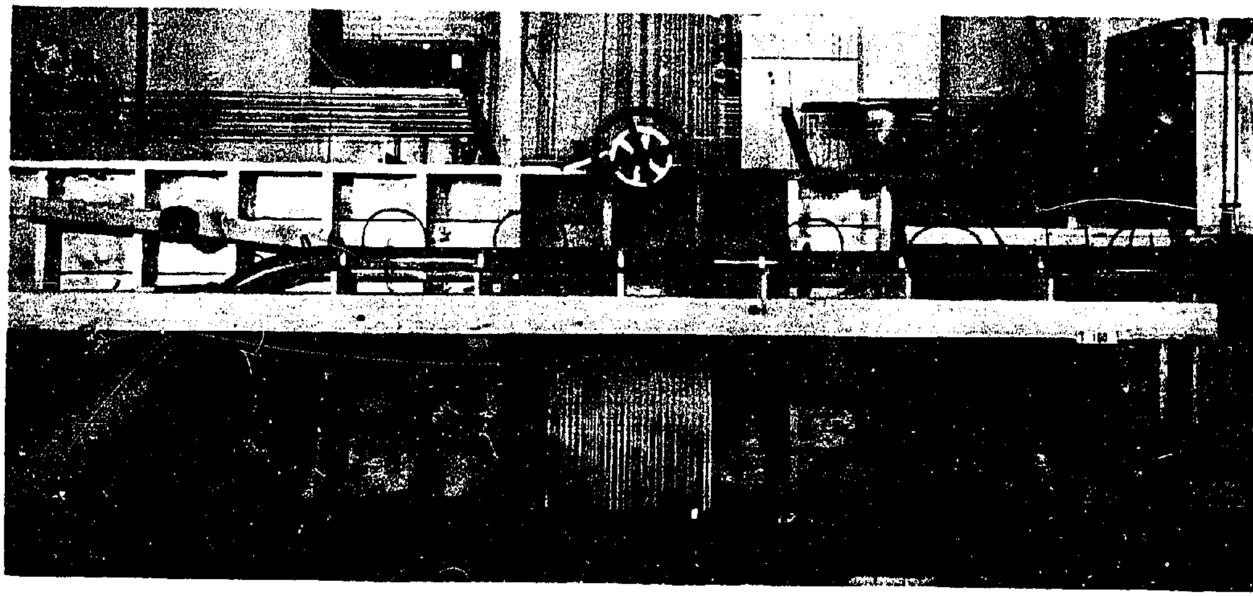
SHASTA DAM OUTLET STUDIES
COMPARISON OF DIMENSIONS OF DIFFERENT TYPES OF VALVES OF EQUIVALENT SIZES

the size of the conduit upstream from the valve can be accomplished as shown in figure 33.

49. <u>Test of Friant Tube Valve in a conduit</u>. Although the Friant tube valve, test F-T, failed to fill the conduit at 100 percent opening because of its lower coefficient it otherwise performed like the F-N design except at openings below 30 percent when its jet became rough, see figure 34. A comparison of the pressure distribution in this valve for free discharge and discharge into a conduit was made on the basis of the parameter developed for this purpose in test F-N. The agreement shown in figure 31, for valve openings of 100, 60, and 40 percent, proves that the pressure gradient is effected only by lowering the end point by the amount of vacuum downstream from the valve.

The air discharge at a head of 19 feet was obtained from a plot of air discharge versus head and scaled up by the ratio $(6.00/5.01)^2$ as in test F-N. The results are shown in figure 32. It will be noted that air discharges for F-N and F-T are practically the same for water discharges above 2.00 c.f.s. but for smaller discharges valve F-T pumped considerably more air. The divergence in air discharge started at an opening of approximately 35 percent on F-T and it was observed that for openings less than 30 percent the jet became rougher and the pressure inside the tube became negative. The greater amount of air for F-T was undoubtedly due to the rougher jet which became very unstable at openings below 25 percent. The variation of water discharge with valve opening is shown for comparison with the other designs in figure 32. The values of air discharge are subject to the same qualifications that have been discussed for test F-N. The maximum negative pressure at 25 percent opening for a head of 19 feet was 0.40 feet which corresponds to 0.40 x 17 or 6.8 feet in the prototype for a head of 323 feet.

The dimensions of this valve for comparison with 8EAS15 were obtained in the same manner as those for the needle valve. Since the



100-Percent open

20-Percent open, nearly maximum air demand

FRIANT TUBE VALVE INSTALLED IN CONDUIT

discharge coefficient is 0.51 the new inlet diameter will be $\sqrt{0.72/0.51} \times 6.00 = 7.13$ inches and all the other dimensions will be increased by the ratio 7.13/6.00. The valve with the new dimensions is shown in figure 33. The reduction from 7.13 to 6.00 inches diameter to connect with the conduit upstream from the valve may be accomplished as shown in figure 33.

VIII - COMPARISON OF 6- AND 20-INCH MODEL TESTS

50. General discussion. The primary purpose of the 20-inch model tests was that of establishing the validity of the 6-inch model tests for predicting the performance of the prototype. For this purpose the 20-inch valve was regarded as the prototype of which the 6-inch valve was a model to a scale of 1 to 3.33 and the actual performance of the 20-inch valve was compared with the performance as predicted from the 6-inch model tests. This confirmation was required because the complexities introduced into the problem by the presence of the two fluids, air and water, are as yet imperfectly understood. Some recent experimental work has yielded promising results but the test conditions were too limited to permit generalization in terms of the subject problem.

In establishing a program for operation of the 20-inch model in a setting where high heads were available the possibilities of operating the model at exaggerated heads were considered. Whereas for flow in closed conduits where no free water surfaces exist it has been well established that the proper manner of obtaining complete dynamic similarity is to duplicate Reynolds number, it has also been shown that when the Reynolds number reaches values of the order of 1×10^5 dynamic similarity in a practical sense becomes more or less independent of the Reynolds number. This fact permits the operation of a model at any head and it is customary for the sake of convenience to use a head scale equal to the geometric scale ratio. In the case of the Shasta tube valve the condition of no free water surface was definitely not satisfied. The necessary conditions for similitude in such a case include constancy of both Reynolds and Froude numbers where Reynolds number represents the relative importance between inertia and viscous forces and Froude number represents the relative importance between inertia and gravity forces. When the same fluid is used in both model and prototype the two numbers or force parameters cannot be satisfied simultaneously and it is necessary to adopt the usual expedient of satisfying the Froude relationship and, if possible, of establishing the relative independence of the Reynolds relationship. The major portion of the tests were made at model heads based on the Froude relationship which resulted in a head scale ratio equal to the geometric scale ratio and only a relatively few tests were made at higher heads to check the influence of a variation in Reynolds number. Unfortunately, the losses due to the presence of the pressure regulator even when it was wide open limited the head available at the valve to values only 100 percent or so greater than the scaled head as previously defined. Some data on air requirements were taken at the higher heads especially for the earlier designs where the restriction of the air relief might be expected to cause critical pressure and thus limit the weight flow of air. Since these data were taken more to supply general information regarding the general problem of air-water flow than to serve a particular purpose in the final design of the valve, only a small portion of the data are presented in this report.

51. Comparison of hydraulic measurements. The testing arrangements for the two models have been fully discussed so it is not considered necessary to add any further description except to point out the difference in air density. The 6-inch model was tested in Denver, Colorado, which is situated some 5000 feet above sea level and has a moderately dry atmosphere. The 20-inch model was tested at Boulder Dam in the Arizona valve house which is situated at an elevation of 800 feet above sea level. Due mostly to the difference in elevation the specific weight of air at Denver was 0.062 as compared with 0.075 at Boulder Dam. This range of specific weight is large enough to include any likely difference between model and prototype.

Some of the difficulties encountered in the tests of the 20-inch model were discussed along with the description of the testing arrangements for this model. Additional difficulties will be briefly sketched in order to serve as a record for future high head tests. In the initial tests all pressures were measured by means of pressure gauges of the Bourdon element type. Inconsistencies in the test results led to a check of the gauges by means of manometers and it was found that the gauges were not reliable enough for the tests. They were replaced by a differential mercury manometer which served the purpose admirably with the only exception that for the higher heads the mercury column became too long for easy reading. This drawback was finally eliminated by using a Crosby Fluid Pressure scale for the heads which exceeded a mercury column of 48 inches.

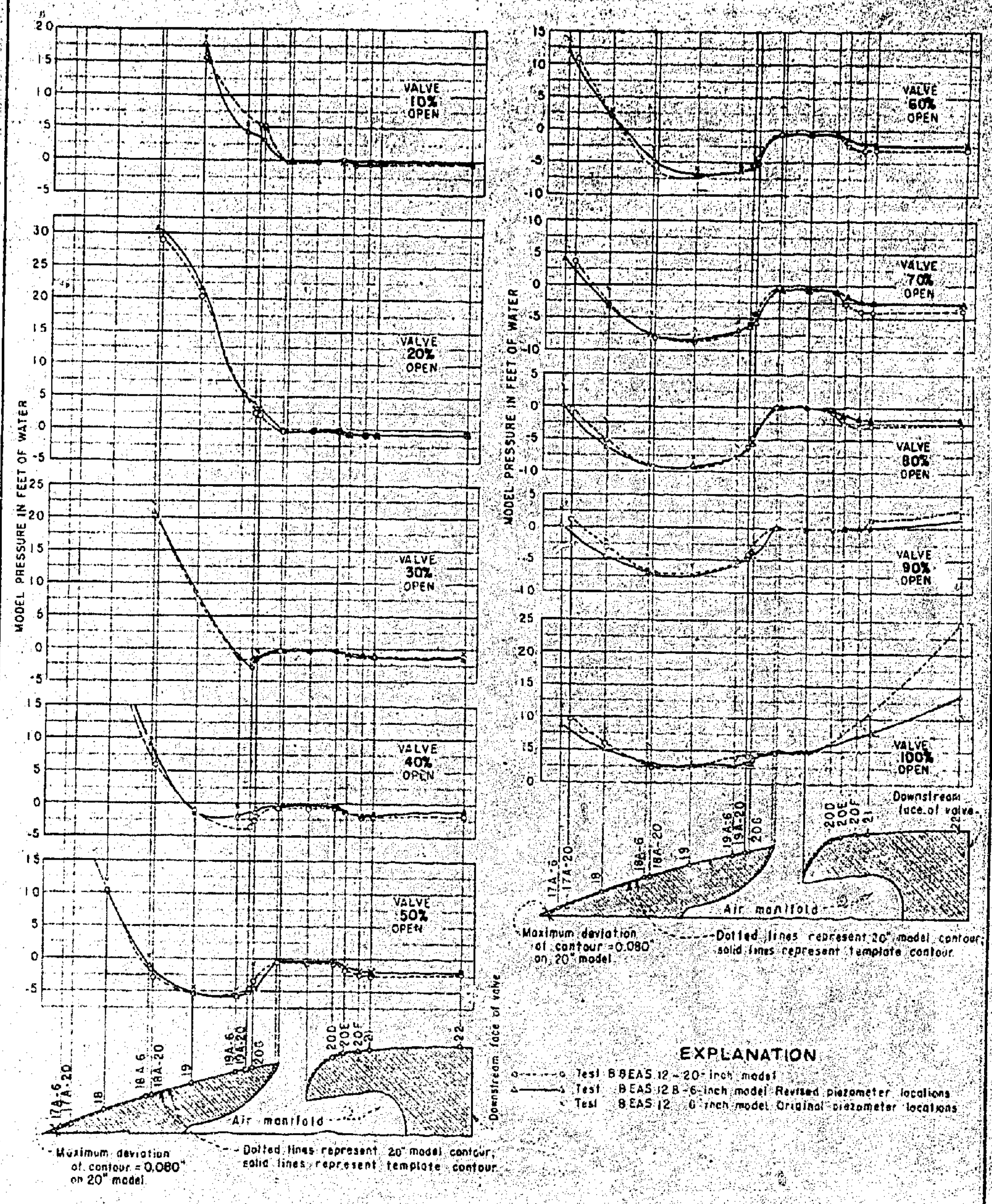
A considerable amount of testing was done before an arrangement was devised which was effective in reducing the available head of 350 feet to the desired model head of 62 feet so none of these tests could be used in checking the Froude similitude law directly. Only the later tests using the improved head control were used for this purpose. However, the earlier tests were used to good advantage as an extension of the tests with the 6-inch model. These high head tests indicated very definitely that critical pressures would occur under prototype conditions in all the designs which provided air relief by means of holes drilled through the nozzle. Critical pressure has been defined previously as the pressure at which the weight flow of air reaches a maximum. Whenever critical pressure occurs a definite change in regimen takes place and similitude between the two models is invalidated. Other undesirable features of designs with air relief by means of holes as brought out in these tests were (1) the presence of very low pressures between the air relief holes and (2) the occurence at some valve openings of a condition where air entered some holes while others were under pressure and discharged water into the air chamber. These undesirable features, which were brought out by the high head tests, led to the development of the air slot type of air relief. Further development of this type in the 6-inch model led to design 8EAS12 which was adopted as a basis for the detailed confirmation tests between the two models.

The tests which will be used for confirming the results of the 6-inch model study were made with the 20-inch model at Boulder Dam with the 8EAS12 shape for reasons explained previously and with the model placed to represent the upper and lower outlets. These tests were designated as B-8EAS12 and B-8EASU12, respectively. It had been determined in previous tests that it was necessary to read only the piezometers in the critical zone. Measurements of pressure distribution and air demand were made at valve openings of 10 percent intervals from 0 to 100 percent. Each opening was operated as closely as possible to the scaled model head of 63.2 feet. Attempt to operate the model under the prototype head were unsuccessful, except at the smaller openings, due to the loss in the energy dissipator. The larger openings were operated under the maximum heads attainable which in the case of openings above 80 percent were considerably less than prototype head.

It became apparent after a few preliminary tests that the air supply lines had too much resistance as indicated by the fact that the pressure inside the tube at piezometer 27 was lower than for the corresponding condition in the 6-inch model. The source of this excess resistance was found to be the design of the transitions between the circular air piping and the hollow rib leading to the tube interior. These transitions, which, incidentally, did not assimilate the prototype, were too sharp and created an obstruction which raised the resistance of the supply line. Since the pressures in the critical portion of the valve were affected by any change in the pressure inside the tube it was necessary to determine the relationship between the pressure at each piezometer and the pressure inside the tube. This was accomplished by dismantling the air piping progressively and measuring pressures after each change with approximately the same head. Eventually the air piping was stripped down to the transitions and a flow nozzle followed be a 3-inch gate valve was attached to each transition to facilitate changes in the resistance.

The test results were corrected to the proper head when necessary and the results were compared with the corresponding values scaled up from the 6-inch model. In other words the 6-inch model was treated as a scaled model of the 20-inch valve with a scale ratio of 1 to 3.33. For making this comparison between the predicted and measured pressure distribution through the critical zone, the pressure at piezometer 27 was used as a basis and the air demand of the valve was neglected. From the several tests at each opening on the 20-inch model one or more were selected which had a pressure inside the tube at piezometer 27 which was similar to, that is 3.33 times, the pressure measured in the 6-inch model. This approach was used to simplify the comparison and to eliminate the discrepancy between the specific weight of the air for the two different models. When the predicted and measured pressures were compared as shown on figure 35, the agreement was satisfactory for openings up to 60 percent but from this opening to 100 percent there was an appreciable discrepancy particularly for piezometers 17A through 19A.

The possible sources of this discrepancy were investigated thoroughly in the 20-inch model and it was found first that the contour of the valve did not correspond exactly to that of the 6-inch valve. The nature and magnitude of the difference, as shown in figure 35, were not considered sufficiently serious to account for the discreancy in test results. The 6-inch valve was examined carefully and it was found that the piezometers were located quite close to one of the ribs whereas in the 20-inch model the piezometers were located midway between ribs. This location in the 6-inch model derived from an attempt to keep the piezometers as close to the center line as possible and since one of the ribs was on the center line the piezometers were quite close to it. To eliminate any effect of the rib on the piezometer openings new piezometers were drilled midway between the ribs to correspond with those in the 20-inch valve. This



EXPLANATION

- Test 8 BEAS 12 - 20-inch model
- Test 8 EAS 12 B - 6-inch model Revised piezometer locations
- Test 8 EAS 12 - 6-inch model Original piezometer locations

SHASTA TUBE VALVE

HYDRAULIC MODEL STUDY

COMPARISON OF PRESSURES FOR 6-INCH AND 20-INCH MODEL change placed the openings comparatively farther away from the rib in the 6-inch model because there were only four ribs as compared with eight in the 20-inch model. The test results for the new line of piezometers, labeled SEAS12B, when scaled up to compare with the results for the 20-inch valve showed remarkably good agreement and proved conclusively that the lack of agreement in previous tests was due to the proximity of the rib to the piezometer openings. A comparison of BSEAS12, SEAS12, and SEAS12B is given in figure 35. It is evident that the main discrepancy between the two models had been eliminated but there was still a certain lack of agreement at the higher openings for piezometers 17A and 18, which was probably due to the difference in contour of the two valves. The effect of this difference would be more pronounced at the higher openings, as indicated in the plots, due to the higher velocities but at no opening was it considered serious enough to justify remachining the 6-inch valve for a more positive check.

It will be noted in figure 35 that there was some difference for piezometers 20D to 22, inclusive, which reached a maximum at 100 percent opening. While there was a wide divergence at number 22 the agreement at 28, only ½-pipe diameter downstream, was very close. In addition there was a sharp rise in pressure in the region from 20D to 28 which indicates that an hydraulic jump was occuring. The difference in pressures at any given point within the region occupied by the jump can be readily accounted for by a slight shift in the location of the jump. This explanation is valid only for 100 percent and 90 percent openings where the jump, as indicated by the sharp rise in pressure, occurred at the downstream end of the valve. For openings smaller than 90 percent, the jump moved out of the valve into the conduit.

When the 20-inch model was checked over for possible sources of the disagreement between the results from the two models it was found that the discharge cone at the end of the conduit was somewhat undersize.

This was due to the method of bracing it by jacking against the sides of the opening in the concrete wall of the valve house through which the model discharged. The jacks were tightened so much that the cross-section of the end of the cone was forced to take an elliptical shape which decreased its area. This distortion was detected by measuring 4 diameters of the end section. A computation of the approximate area showed that there was a decrease in area large enough to create an excess back pressure of approximately 1.5 feet at the scaled head of 63 feet. Again as in the case of the 6-inch model when the sheet metal cone was substituted for the shrunken plastic cone the influence of the sizing of this cone was clearly demonstrated. For purposes of comparison the conditions, as measured in the 20-inch model, were duplicated in the 6-inch model by constructing the end of the 6-inch model outlet until the proper pressure at piezometer 27 was obtained. This was done at only two openings, 100 percent and 90 percent, for which the conduit flowed full because at all lesser openings the effect of back pressure from the terminal cone did not extend to the valve. The pressure distributions for 100 percent and 90 percent shown in figure 35 were measured with the constriction.

Another factor involved in the comparison of pressures in the region from 20D to 22 was demonstrated when this portion of the valve contour was removed for test 8EAS13 as shown in figure 37. From the effects of this change on the flow of air it was evident that in test 8EAS12 the jet was flowing more or less in contact with the boundary in this region. Any air relief for the space surrounding the jet of water downstream from the valve must either pass between the jet exterior and the valve contour or be picked up by the jet proper by reason of the roughness of its surface due to turbulence. The pressure in the region from 22D to 22 was a function of turbulence which was not similar in the two models. Therefore, the disagreement of pressures in this region was understandable and since the differences were minor no further investigation was made.

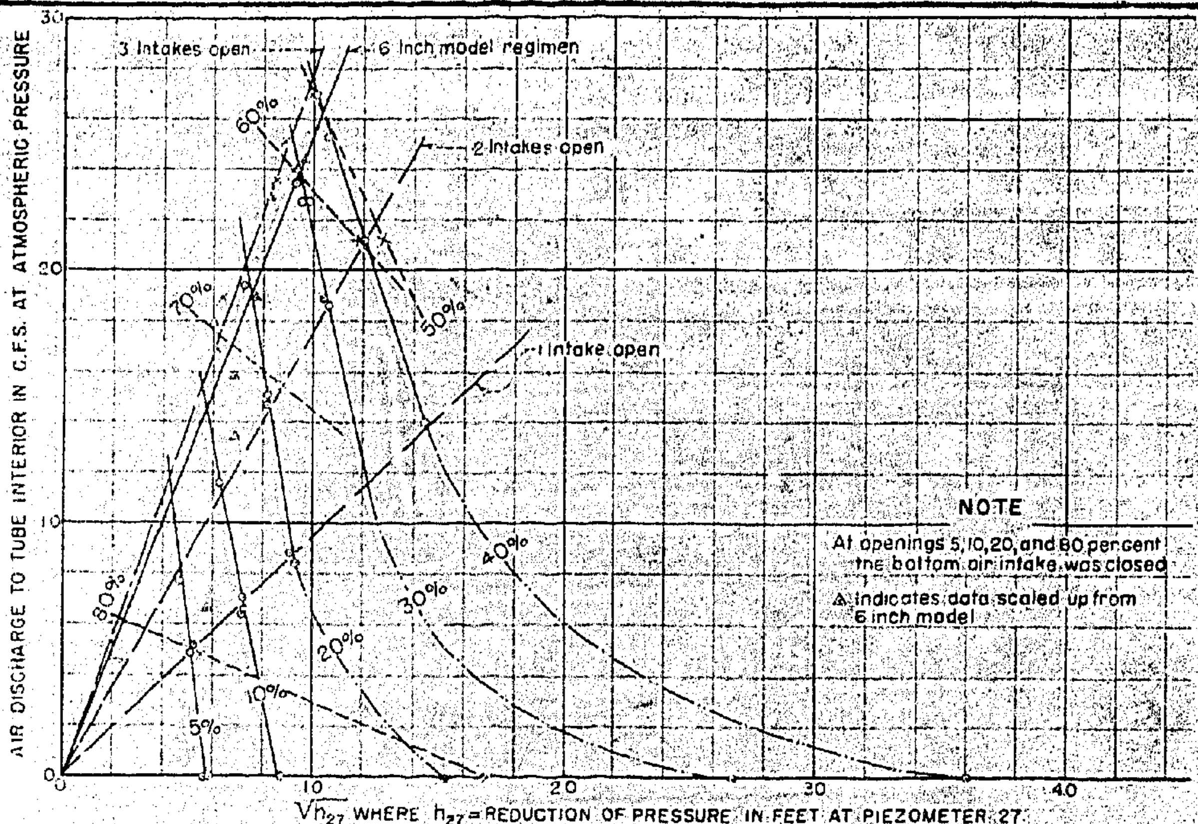
In order to complete the comparison of the two models a discussion is necessary concerning the tendency of the jet to break away from the boundary in the region between piezometers 19 and 20G. In the 6-inch model it was necessary to suck on the piezometer tube and thus force the jet against the boundary before a true reading could be obtained. If this was not done the piezometers from 19 through 20G indicated only the pressure of the air box. The flow in this region was decelerating and would not fill the end of the valve unless forced to do so. In the case of the 20-inch model it was not necessary to force the jet to remain in contact with the boundary. The higher degree of turbulence in the approach piping of the larger model together with the higher Reynolds number of the flow through the valve prevented the separation action as experienced in the smaller model. The phenomena of separation and its prevention by setting up a turbulent boundary layer are treated in modern textbooks of fluid mechanics. At a few openings in the 6-inch model it was impossible to remedy the separation so the pressures are not comparable. There is no doubt that conditions in the prototype with its higher Reynolds number, which is a qualitative index of turbulence, will assimilate the conditions of the 20-inch model and no separation will occur.

52. Comparison of air measurements. One of the factors involved in the comparative air studies is the difference in the specific weight of the air for the two models. A prediction of the effects of this difference would seem to offer no insurmountable difficulties if it is assumed that the relief air can find its way unobstructed to the regions of low pressure. The jet of water at any given valve opening together with the action in the conduit into which it is discharging has the capacity for pumping a definite quantity of air expressed in terms of the pressure in the conduit. If the atmospheric conditions were somehow changed so that for example the specific weight of the air was increased the following effects would be

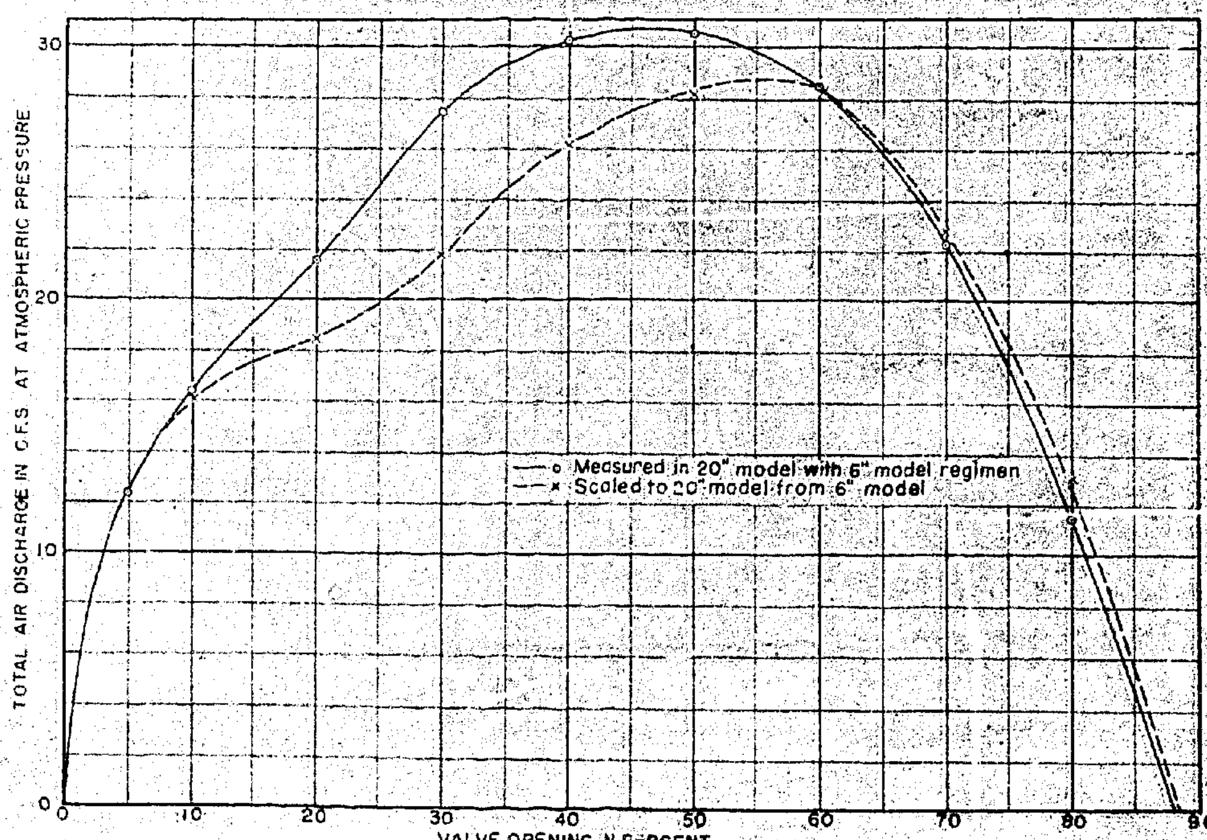
expected. The jet of water would attempt to pump approximately the same quantity of air as before but because of its higher density a greater drop in pressure through the air supply piping would be required. The volume of air would expand due to the pressure drop and eventually a point would be reached at which the higher differential pressure plus the effect of expansion would supply the required quantity of air. Since the gage pressure in the conduit would be lower for the atmosphere of higher specific weight, the gage pressures on the valve boundaries would also be lower but the absolute pressure would tend to remain the same.

The experimental verification of these predictions was complicated because in the case of the valve design selected for the confirmation studies the condition of unobstructed access for the air was not present. For this reason any attempt to evaluate the effect of change in air density from the confirmation tests would be difficult and of questionable value. The confirmation tests will be used only to show that the model results can be converted to prototype terms by applying similitude relationships with some measure of confidence.

As described previously the measurements of air flow to the tube interior in the 20-inch model were made with various amounts of resistance in the air supply lines. The air requirement in model terms for different resistances is shown in figure 36, plotted against the square root of the drop in pressure to piezometer No. 27 (inside the tube). The slope of the line is an index of the resistance of the air supply lines. The air discharge used in this plot was intended to be the quantity at the pressure of piezometer No. 27 but the difference in pressure from atmospheric was so small in all cases except where the tube supply was entirely closed that atmospheric pressure was substituted with a very small error. The air discharge shown in figure 36 does not include the quantity of air taken by the air box which was found to be very small. The air discharge through the air box was so small in fact that a 3-inch diameter intake orifice



A AIR DEMAND FOR VARIOUS RESISTANCES IN TUBE SUPPLY 20 INCH MODEL



B COMPARISON OF AIR DEMAND IN 6-INCH AND 20-INCH MODELS
(DESIGN 8EAS12)

in the 7.0-inch diameter supply line was required to create a pressure drop large enough for a reliable measurement. Since the flow through the air box was only a small part of the total flow for valve design 8EAS12, it is accorded only a minor place in the following discussion.

The air demand for any given condition or size of supply lines may be obtained from the curves in figure 36. For the purpose of comparing the pressures as measured in the two models the pressure at piezometer No. 27 was adjusted in the 20-inch model until it conformed to the pressure scaled up from the 6-inch model. This comparison, which has been discussed previously, eliminated all minor effects due to difference in air densities and frictional losses which would occur as a result of the difference in Reynolds number of the two models. A curve has been drawn in figure 36 to indicate the regimen of the flow of air into the tube in the 6-inch model scaled up to the 20-inch model. This curve was obtained from the 6-inch model tests by the following process. In measuring the flow of air in this model the total flow was metered by means of a 4-inch anemometer and the portion passing through the air box by means of a pitot tube placed at the center line of the air box supply line. Because only a small part of the total flow of air passed through the air box the head differential across this pitot tube was too small to be considered reliable. The portion of the total flow passing into the tube interior of the 6-inch model was, therefore, obtained by computing the ratio of the corresponding flows as measured in the 20-inch model and using this ratio to obtain the desired flow in the 6-inch model from the measurement of the total flow with the anemometer. This ratio was approximately 85 percent for most of the range of gate openings except for openings approaching 80 percent where the ratio dropped to 50 percent. The values of air flow to the tube interior thus obtained for the 6-inch model were scaled up to the 20-inch model according to the scale ratios based on a constant value of Froude's number. The results are shown in figure 36A. The data scattered considerably but a straight line was drawn through the points representing the higher discharges because they were assumed to be more reliable. The intersections of this line with the lines representing constant valve opening indicate the air discharge to the tube which would occur if the 6-inch model regimen were duplicated in the 20-inch model. The total flow for any condition was obtained by applying the ratios of air box flows to total flows as derived previously from the 20-inch model tests. The total air discharges which are shown in figure 36B represent the values which would have been obtained if the air piping in the 20-inch model had been similar to that in the 6-inch model. For comparison with these values the measurements of total flow in the 6-inch model, scaled up by the ratio $\left(\frac{20}{6}\right)^{\frac{5}{2}}$ are also shown in figure 36. With the exception of the range between 20 and 50 percent the agreement is quite satisfactory. It will be recalled that in a previous section of this report a consideration of the complexity of the flow involving two separate phases and of the differences in the experimental methods led to the conclusion that a close agreement between the results of the two models could not be expected. The comparison does indicate, however, that the results of a model can be scaled up on the basis of Froudes similitude relations with a reasonable degree of certainty. Some tests were run to furnish comparative data for the upper outlet but they were not completed because it was decided to make the comparison on the basis of the lower outlet since it operated under a much higher head.

The results of the tests with the revised design, 8EAS15, were scaled to prototype dimensions and the results are shown in figure 26. The distribution of the total flow between the air box and the tube interior was measured in the same manner as test 8EAS12 but the results are more reliable because the increased flow through the air box made the differential of the pitot tube much greater.

The conditions set up at the beginning of this section for a proper evaluation of the effects of a difference in the density of the air were present in design 8EAS15 but unfortunately no opportunity presented itself for duplicating this test in the 20-inch model at Boulder Dam. The effects of a change in density could be studied in the 6-inch model by enclosing the entire model so that it could be operated at reduced and elevated pressures but the considerable expense which would be involved was not considered justified. The magnitude of the effects of a change in density can be estimated as follows. Assuming two different atmospheric pressures, $P_1 = 28$ inches of mercury and $P_2 = 25$ inches of mercury. The differential pressure required to supply the necessary quantity of air, assuming this quantity to be constant for a given valve opening and head, will be proportional to the specific weight which in turn will be proportional to the atmospheric pressure. The ratio of pressures would be $28/25 = 1.14$ which means that the negative pressure in the air box would be 14 percent higher for the higher specific weight corresponding to the higher atmospheric pressure. The computed magnitude of this negative pressure has been estimated previously as 1.90 feet of water for a valve operating under 323 feet head. The effect of the difference in density will then be $0.14 \times 1.9 = 0.27$ feet in the prototype. However, the higher atmospheric pressure which gives rise to the higher specific weight more than offsets the increased differential required for driving the higher density fluid. The absolute pressure will be the sum of the two effects. The effect of a change in density of the air where considered with the change in atmospheric pressure can be safely neglected for all practical purposes.