PAP 502

Hydraulic Model Study of fixed-Cone Valve Energy Dissipator Central Arizona Project Bypass Structure, Waddell Pumping-Generating Plant

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U.S. Bureau of Reclamation

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PAP-502

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INFORMATIONAL ROUTING

D-1531

Dodge

Memorandum
Chief, Concrete Dams Branch

Denver, Colorado
August 15, 1986

Chief, Hydraulics Branch

Hydraulic Model Study of Fixed-Cone Valve Energy Dissipator - Central Arizona Project Bypass Structure, Waddell Pumping-Generating Plant

As requested in your memorandum dated December 11, 1985, we have completed the subject model study.

A memorandum report is attached covering the work done on the model.

GPO 852320

Philip H. Burgi

Attachment

Copy to: D-1530
D-1530A
D-1531
D-1531 (Dodge)
D-1531 (PAP file)
D-1532
D-1532 (Peck)
D-223 (Hepler)
D-252 (Simmonds)
D-230 (Duster)
(with attachment to each)

HYDRAULIC MODEL STUDY
FOR
WADDELL FIXED-CONE VALVE ENERGY DISSIPATOR

by

Hilaire Peck

INTRODUCTION

A hydraulic model study was performed for the proposed 11-foot-diameter fixed-cone valve energy dissipator on the CAP (Central Arizona Project) bypass structure, Waddell Pumping-Generating Plant. The fixed-cone valve was sized to pass 1,800 ft^3/s at a low head during reservoir evacuation. The purpose of the study was to obtain optimum structural dimensions for the dynamic loads acting on the energy dissipation structure due to the annular jet, dimensions for the air vent, and an estimate of spray, noise, and possible vibration during operation.

The model scale (1:22) was determined by the 6-in fixed-cone valve available in the laboratory. The model was operated according to Froude similarity since gravity and inertia forces are dominant. The model included the valve and energy dissipator on the right abutment (figure 1) and 55 feet of the downstream canal. Maximum head on the centerline of the prototype valve will be 190 feet, and the maximum allowable discharge will be 1,800 ft^3/s. Head loss in feet from the reservoir to the valve is given by the equation:

$$H_L = 3.03 \times 10^{-6} Q^2 \qquad (1)$$

where Q = discharge in ft^3/s.

GATE CHAMBER FLOW

The energy dissipator easily contained the energy for the design flow and double design flow (figures 2-5). The side deflectors inside the dissipator were moved upstream into the same vertical plane as the roof deflector. This caused no discernible change in the condition of the flow leaving the dissipator for either the design flow or double design flow (figures 6-9). At design flow rate there was no problem with spray or the condition of the flow leaving the dissipator for either deflector design. Therefore, the choice of deflector design should be based on structural strength and/or cost considerations.

The energy dissipator studied was 90 ft long, 36 ft wide and 38 ft high (see figure 1). It is felt the dimensions of the structure should not be reduced because the distance available for air entrainment in the jet is already short and the energy in the flow leaving the dissipator is appreciable. Any reduction in structure dimensions will decrease air entrainment and therefore increase the possibility of cavitation. Reduction in structural dimensions could also considerably increase the energy in the flow leaving the energy dissipator.

AIR VENT DESIGN

Falvey [1] states that relative airflow rate in fixed-cone valves is proportional to the total upstream head. Thus,

$$\frac{Q_{air}}{Q_{water}} = f\left(G, \frac{\Delta p/\gamma}{H_t}\right) \tag{2}$$

where G = gate opening in percent
H_t = total potential and kinetic energy upstream in ft
$\Delta p/\gamma$ = differential between atmospheric pressure and air pressure at the end of the vent in ft
γ = specific weight of water in lb/ft^3

Using the relationship shown in equation (2), plots of relative air pressure at the valve versus relative airflow rate were drawn (figures 10 through 14). These plots were drawn using model data converted to Waddell prototype values. Each figure represents data collected at one gate opening. These figures show percent of maximum valve flow area rather than percent gate opening because the model valve does not represent the prototype valve for discharge calibration purposes.

Maximum valve flow area was established to maintain control at the downstream end of the valve. This was accomplished by restricting the sleeve travel to 0.45 D or less. This prevents opening the valve to a greater flow area than is available within the flow passage, so that back pressure is maintained on the vanes and control does not shift to the leading edge of the vanes.

This opening was determined in the following manner:

1. Cross-sectional area of flow versus sleeve travel from the downstream face of the valve was plotted on a graph. The area was calculated by taking the full area of the upstream pipe minus the cross-sectional area of the vanes and the cone at several specific sleeve travel positions.

2. Area of the flow leaving the valve versus sleeve travel was plotted on the same graph. This area is different than the area calculated in 1 above because the flow was assumed to be leaving

the valve at a 45° angle from the centerline of the valve due to deflection by the cone and is controlled by the distance from the cone (normal to the cone surface) to the outer sleeve.

The point where the two lines plotted in 1 and 2 cross gives equal area for two methods.

The relationship between prototype sleeve travel and percent of maximum flow area is shown in figure 15.

According to Falvey [1], relative airflow rate (Q_{air}/Q_{water}) can be expressed as:

$$\frac{Q_{air}}{Q_{water}} = \frac{A_v}{A_p C_D} \left(\frac{\rho_{water}/\rho_{air}}{\Sigma K_s + fL/4R}\right)^{1/2} \left(\frac{H_{air}}{H_{water}}\right)^{1/2} \tag{3}$$

where
- A_v = cross-sectional area of the air vent
- A_p = area of pipe immediately upstream of valve
- C_D = discharge coefficient of the valve
- ρ = density (slugs/ft^3)
- K_s = singular (form) loss in the air vent
- f = Darcy-Weisbach friction factor
- L = air vent length (ft)
- R = hydraulic radius of air vent (ft)
- H_{air} = difference between atmospheric and energy dissipator pressure head (ft)
- H_{water} = pressure head on centerline of valve (ft)

To determine airflow rate and air pressure for a given air vent design, equation (3) should be plotted on the appropriate graph (figures 10 through 14). Equation (3) is solved by assuming values of H_{air}/H_{water} and calculating the corresponding value of Q_{air}/Q_{water}. The intersection of the empirical data plot and the computed curve gives the relative airflow rate and the relative air pressure for a given flow of water and vent design (as shown in figure 12).

Example Vent Design Calculation

Figure 16 shows three different air vent designs used to calculate relative air flow. The curves computed using equation (3) were plotted on figure 12. The losses (K_s) shown on figure 16 include entrance, bend, and exit losses. The following values were used in equation (3) for all three air vent designs.

A_v = 10 ft^2
A_p = 95 ft^2
C_D = 0.63
ρ_{water} = 1.94 slugs/ft^3
ρ_{air} = 0.00187 slugs/ft^3
$fL/4R$ = assumed negligible

The computed airflow curve for figure 16A intersects the empirical data curve for a water flow rate of 4,925 ft^3/s at a relative airflow rate of 0.38 and a relative air pressure of 0.58. Therefore,

Q_{air} = (0.38) (4925) = 1872 ft^3/s and
H_{air} = (0.58) (104)/100 = 0.60 ft.
The pressure head inside the energy dissipator is 0.60 feet of water below atmospheric pressure.

The allowable air demand is often limited by factors such as noise due to high air velocities or accessibility of the air vent opening to people. Air duct velocities near buildings are normally limited to 50 ft/s. For inaccessible air vents on a dam, velocities as high as 300 ft/s would be allowable without incurring compressibility effects. In the above example the air velocity was 187 ft/s through the vent. If it is felt that the velocity or the air pressure in the dissipator is not acceptable, a new air vent design could be selected and the above process repeated until acceptable values are obtained.

DYNAMIC LOADS

Nineteen piezometer taps were installed on the model structure. Pace 5-lb/in^2 transducers were installed on the four piezometer taps that had the highest pressures (2, 6, 13, 14). These four taps also had the greatest fluctuations in pressure. Figure 17 shows the locations of the piezometer taps. Taps No. 1 through 4 are 11 inches above the energy dissipator floor. Tap No. 20 is 60 inches below the dissipator roof on the top deflector (see figure 1). Tap No. 19 is 11 inches below the dissipator roof. The remainder of the taps are flush with the bottom of the dissipator roof. Table 1 shows the manometer readings obtained from the piezometer taps. Table 2 shows the data obtained from the Pace transducers. The standard deviations in table 2 are presented to indicate the amount of fluctuations that can be expected.

Table 1. - Manometer readings.

Percent open (percent maximum flow area)	Head on centerline of valve (ft)	Flow rate (ft^3/s)	Average pressure head (ft of water)																		
			Tap No.																		
			1	2	3	4	5	6	7	9	10	11	12	13	14	15	16	17	18	19	20
6.4	198	800	6.7	9.5	9.0	9.7	0.0	0.7	1.2	0.0	0.0	0.2	0.2	2.3	1.2	0.5	0.2	13.2	1.2	-0.5	-0.5
19.7	183	1,825	5.3	12.0	9.2	12.2	1.2	4.4	1.8	0.0	0.5	9.7	10.2	2.1	0.2	0.2	1.8	2.3	1.2	-0.9	-0.2
47.1	170	4,120	3.2	16.6	9.7	17.8	1.8	26.1	2.3	0.0	1.4	34.4	9.2	6.7	1.8	0.7	2.3	21.9	2.3	8.3	0.5
67.1	110	4,310	5.5	17.1	10.9	17.8	0.0	10.9	6.0	0.2	12.2	27.0	14.1	6.5	2.5	0.7	0.0	12.2	6.0	6.0	0.7
84.8	100	5,120	6.2	18.9	11.1	18.7	0.0	3.9	7.2	6.5	19.6	21.3	17.3	10.4	4.2	1.2	0.0	4.6	7.4	-3.2	0.0
100	75	8,230	1.8	28.9	7.2	25.2	1.6	19.9	14.8	32.3	38.1	37.2	45.5	43.7	21.3	6.0	2.3	21.0	12.2	-4.6	0.0

Table 2. - Transducer results.

Percent of maximum flow area	Head on center-line of valve (ft)	Flow rate (ft^3/s)	Tap No.	High pressure (ft of water)	Low pressure (ft of water)	Average pressure (ft of water)	Standard deviation (ft of water)
6.4	196	800	2	8.3	1.4	5.8	0.5
6.4	196	800	6	114.3	-21.3	28.6	33.0
6.4	196	800	13	13.6	-9.2	0.0	0.9
6.4	196	800	14	9.7	-4.9	-0.5	0.2
19.7	184	1830	2	12.7	1.6	8.8	1.4
19.7	184	1830	6	27.0	-5.5	3.0	0.7
19.7	184	1830	13	58.2	-6.5	7.2	5.1
19.7	184	1830	14	40.2	-5.3	3.9	3.7
47.1	186	3610	2	18.7	6.5	12.7	1.6
47.1	186	3610	6	58.2	-6.5	15.5	9.7
47.1	186	3610	13	97.5	-8.5	22.6	9.5
47.1	186	3610	14	46.7	-2.1	8.1	4.9

CONCLUSIONS

The model indicated there was very little air demand at the design flow rate of 1,800 ft^3/s. Using equation 3 and figures 10-14, different types and sizes of air vents can be checked for air demand, air velocity, and negative pressure inside the energy dissipator as discussed previously.

The present design easily dissipates the energy resulting from the design flow. There is little spray at the design flow rate (figures 2 and 3). It is recommended that the dimensions of the structure not be reduced due to the short distance available for air entrainment by the jet leaving the valve and because a decrease in structure dimensions will reduce its energy dissipation capability.

Noise levels could be a problem if there are high air intake velocities at the mouth of the air vent. High air intake velocities could also be a safety problem if the intakes are accessible to people. There was minimal vibration in the model for all flows studied.

Dynamic and static loads and negative pressures inside the energy dissipator were generally low as indicated in tables 1 and 2.

REFERENCES

1. Falvey, H. T., "Air-Water Flow in Hydraulic Structures," Engineering Monograph No. 41, Bureau of Reclamation, December 1980.

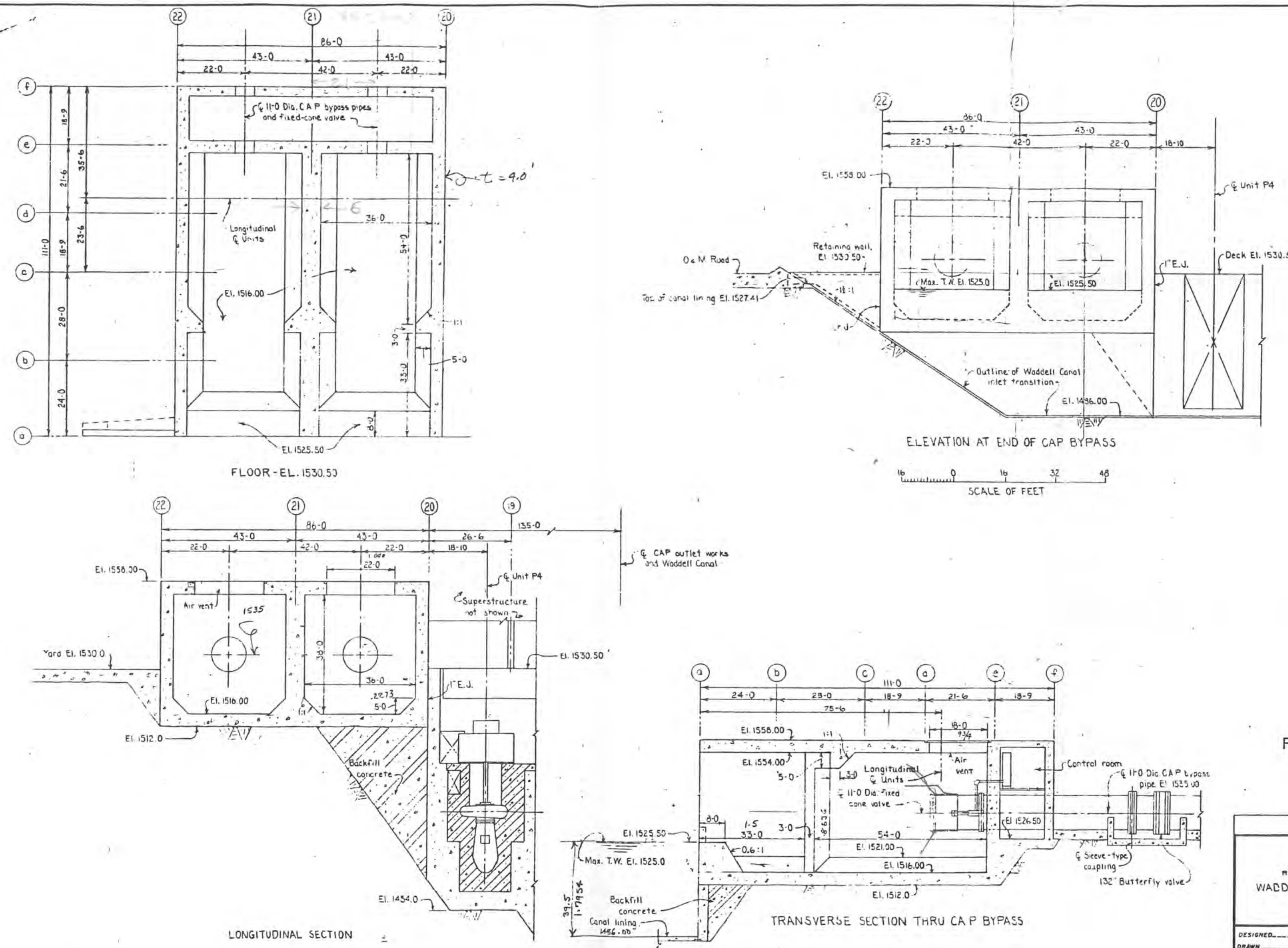

Figure 1

STUDY, DEC 20 1985

ALWAYS THINK SAFETY

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL ARIZONA PROJECT
REGULATORY STORAGE DIVISION-ARIZONA

WADDELL PUMPING-GENERATING PLANT
PRELIMINARY ARRANGEMENT
CAP BYPASS STRUCTURE

DESIGNED	TECHNICAL APPROVAL
DRAWN	SUBMITTED
CHECKED	APPROVED
DENVER, COLORADO DEC. 6, 1985	344-D223-3

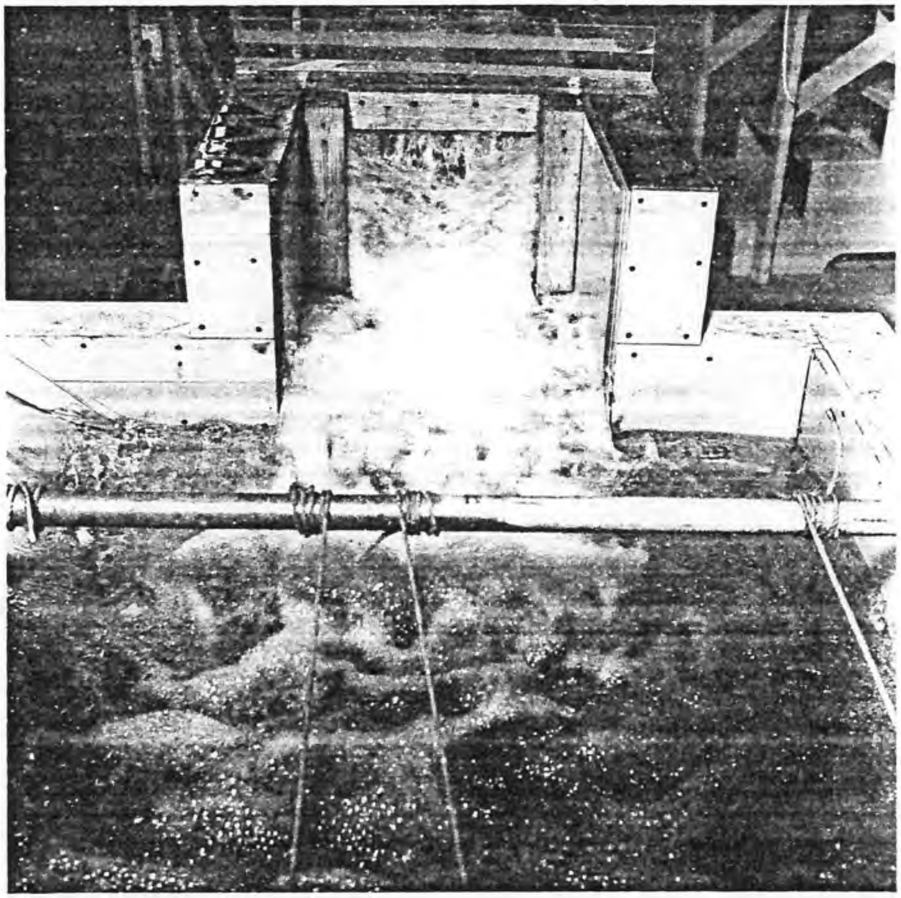

Figure 2. - Q = 1,800 ft^3/s.

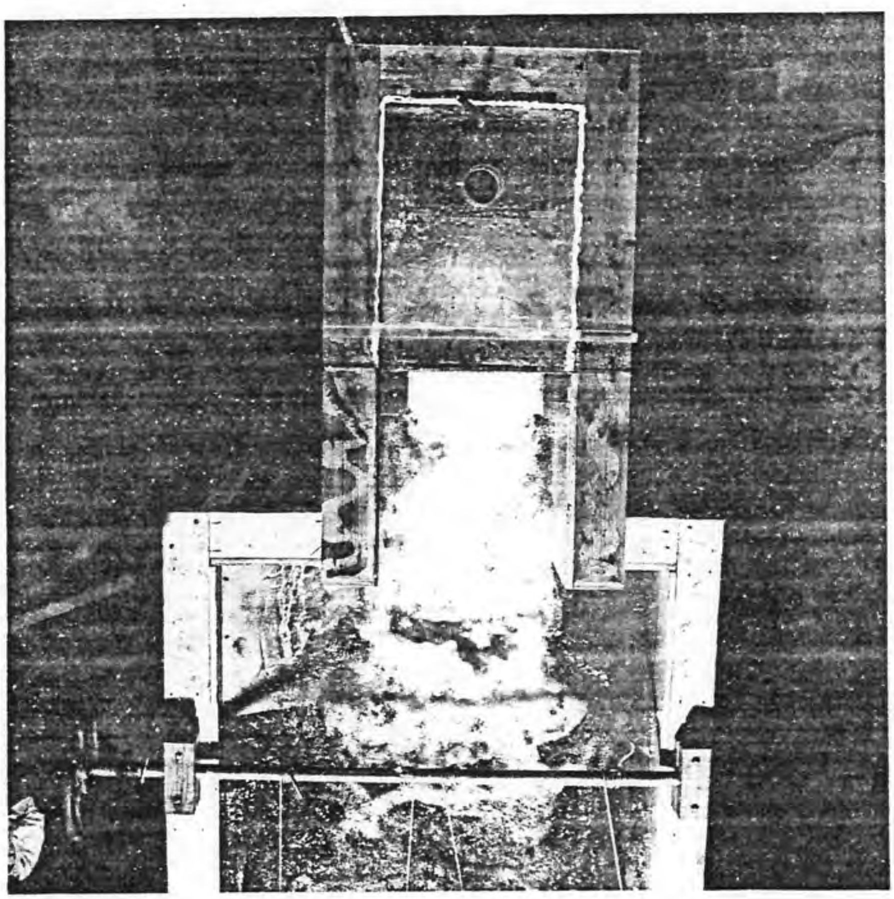

Figure 3. - Q = 1,800 ft^3/s (top view).

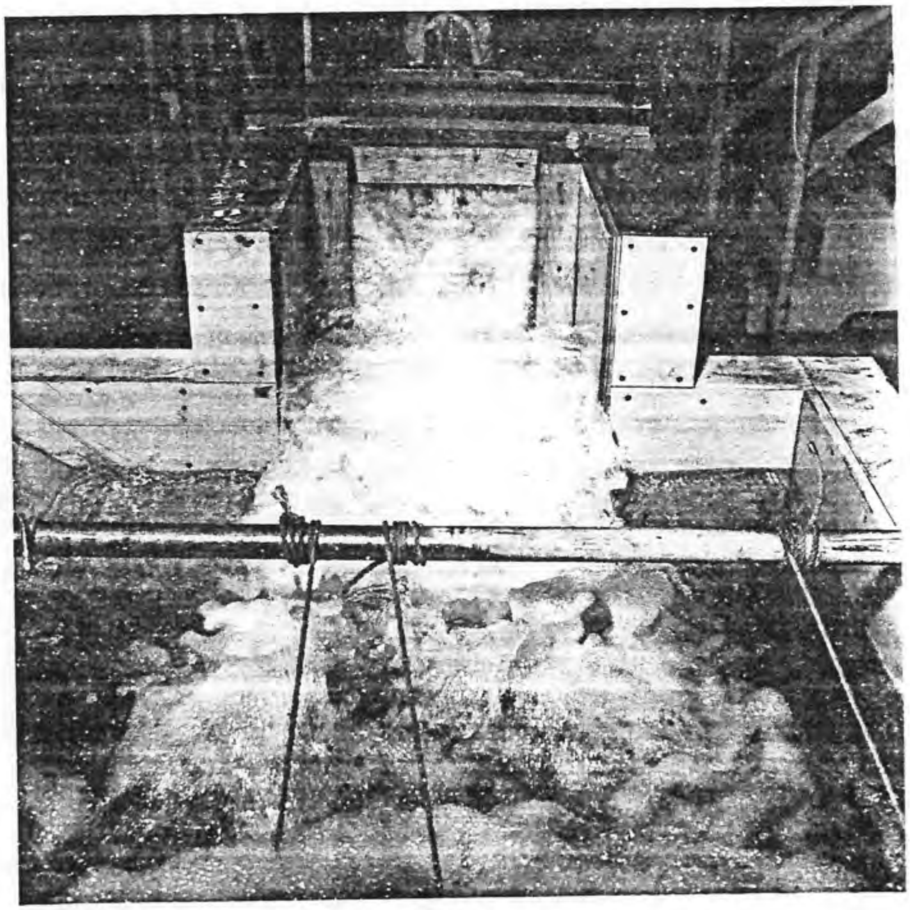

Figure 4. - $Q = 3,600 \ ft^3/s$.

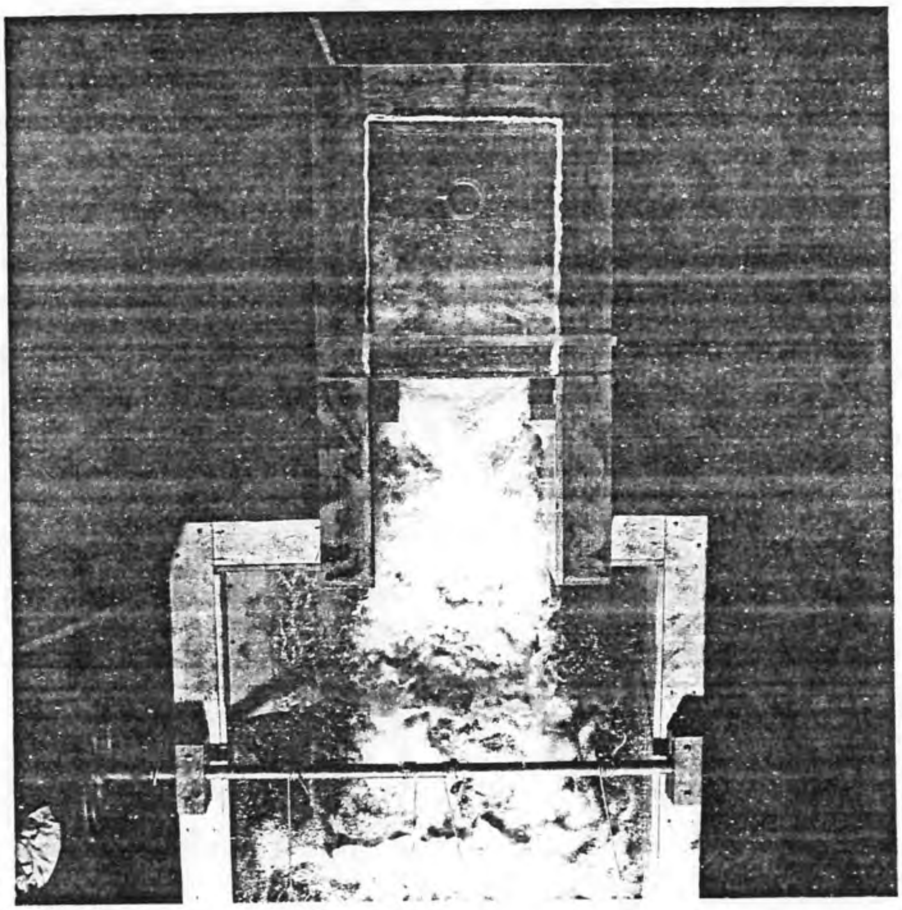

Figure 5. - Q = 3,600 ft^3/s (top view).

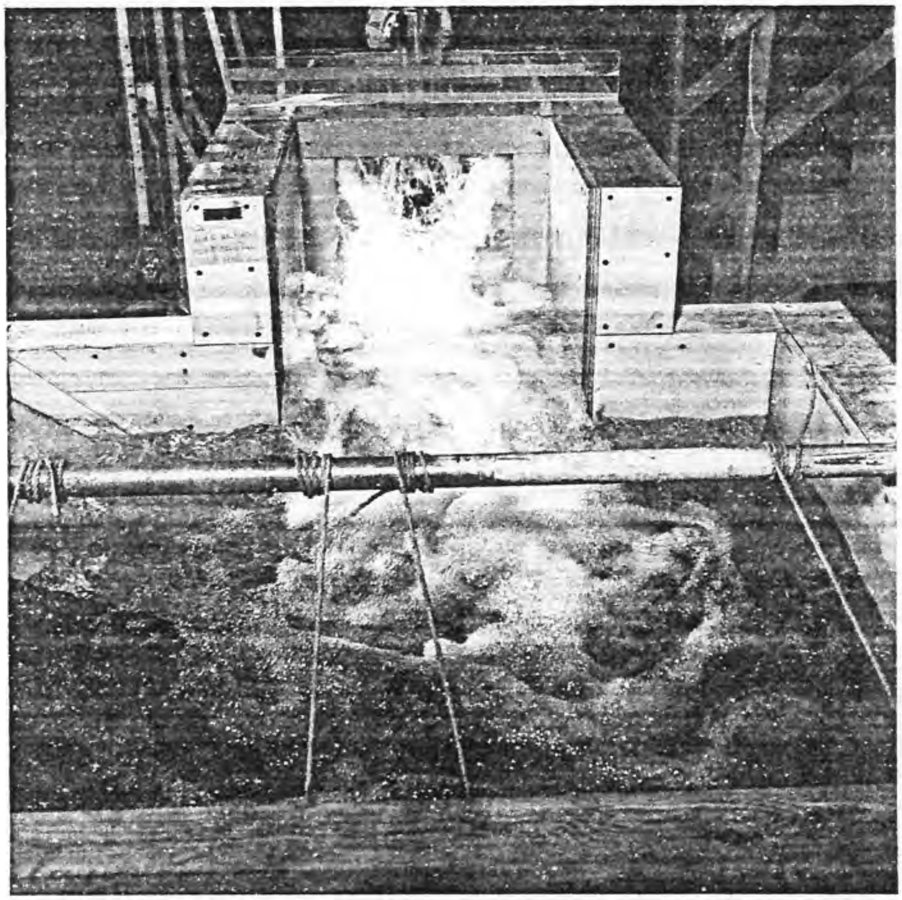

Figure 6. - 1,800 ft^3/s, deflector modified.

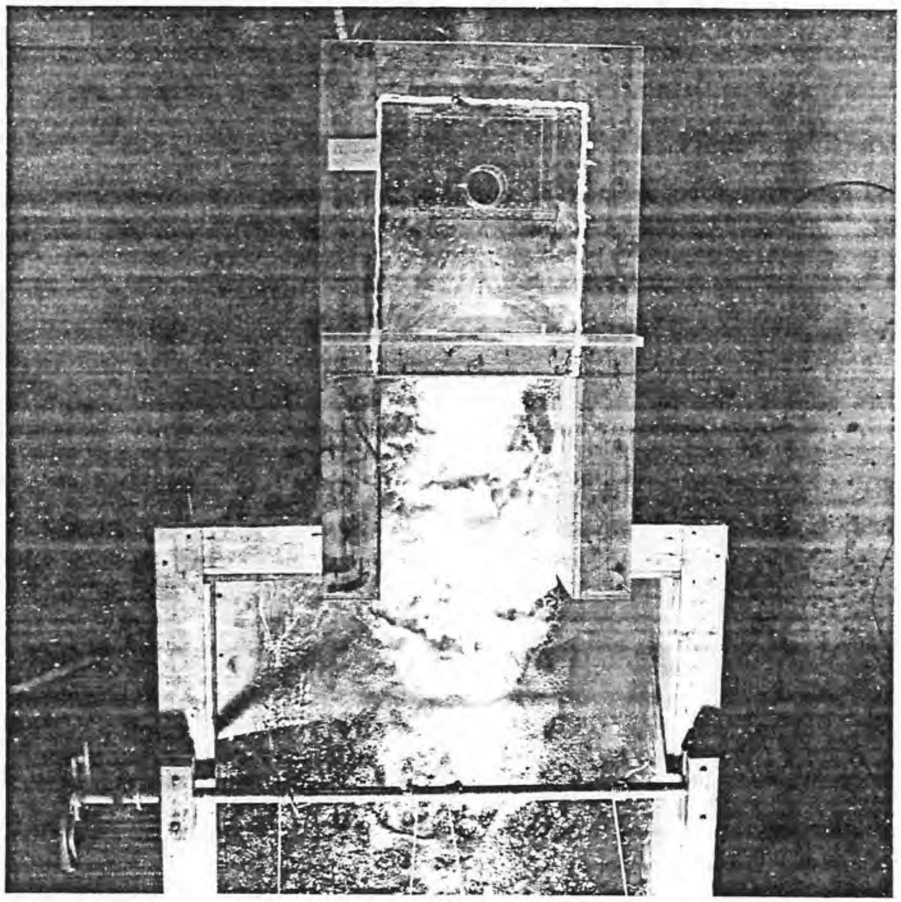

Figure 7. - 1,800 ft^3/s, deflector modified (top view).

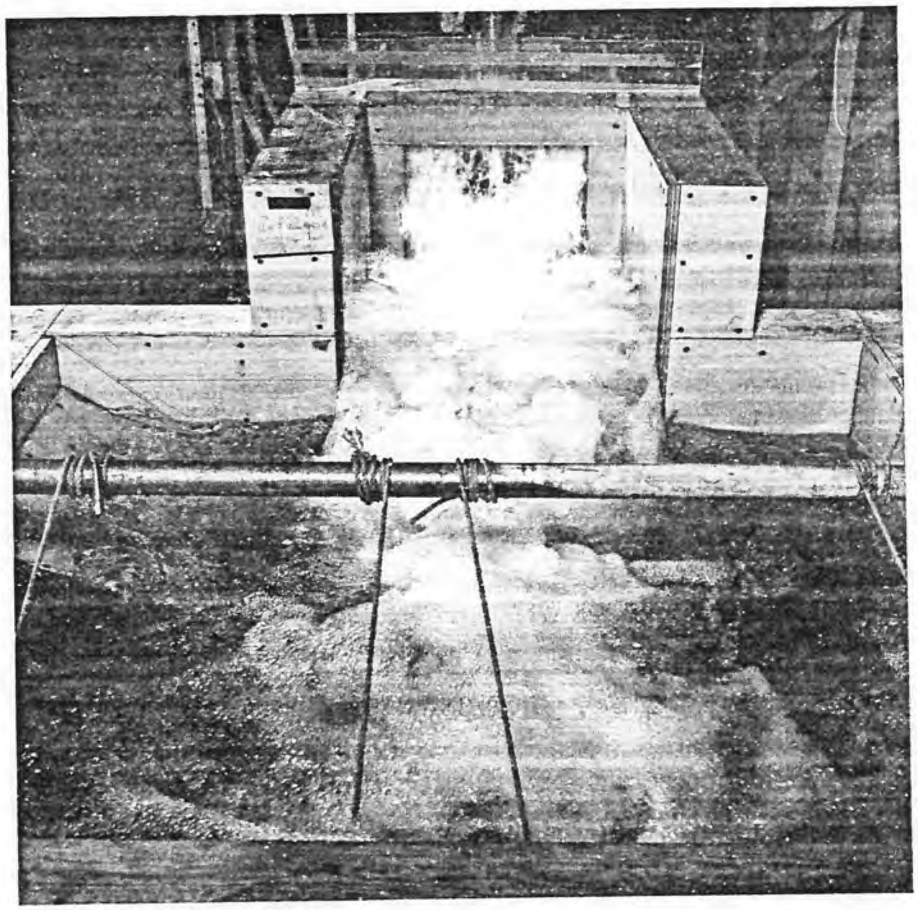

Figure 8. - 3,600 ft^3/s, deflector modified.

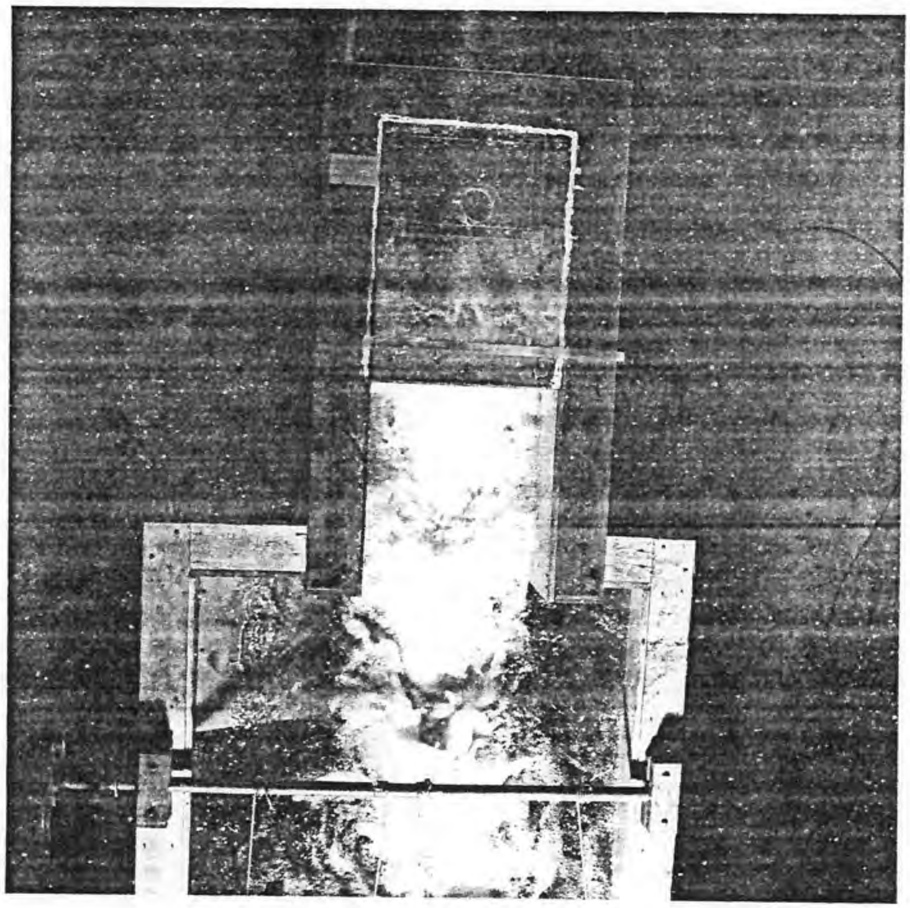

Figure 9. - 3,600 ft^3/s, deflector modified (top view).

Figure 10

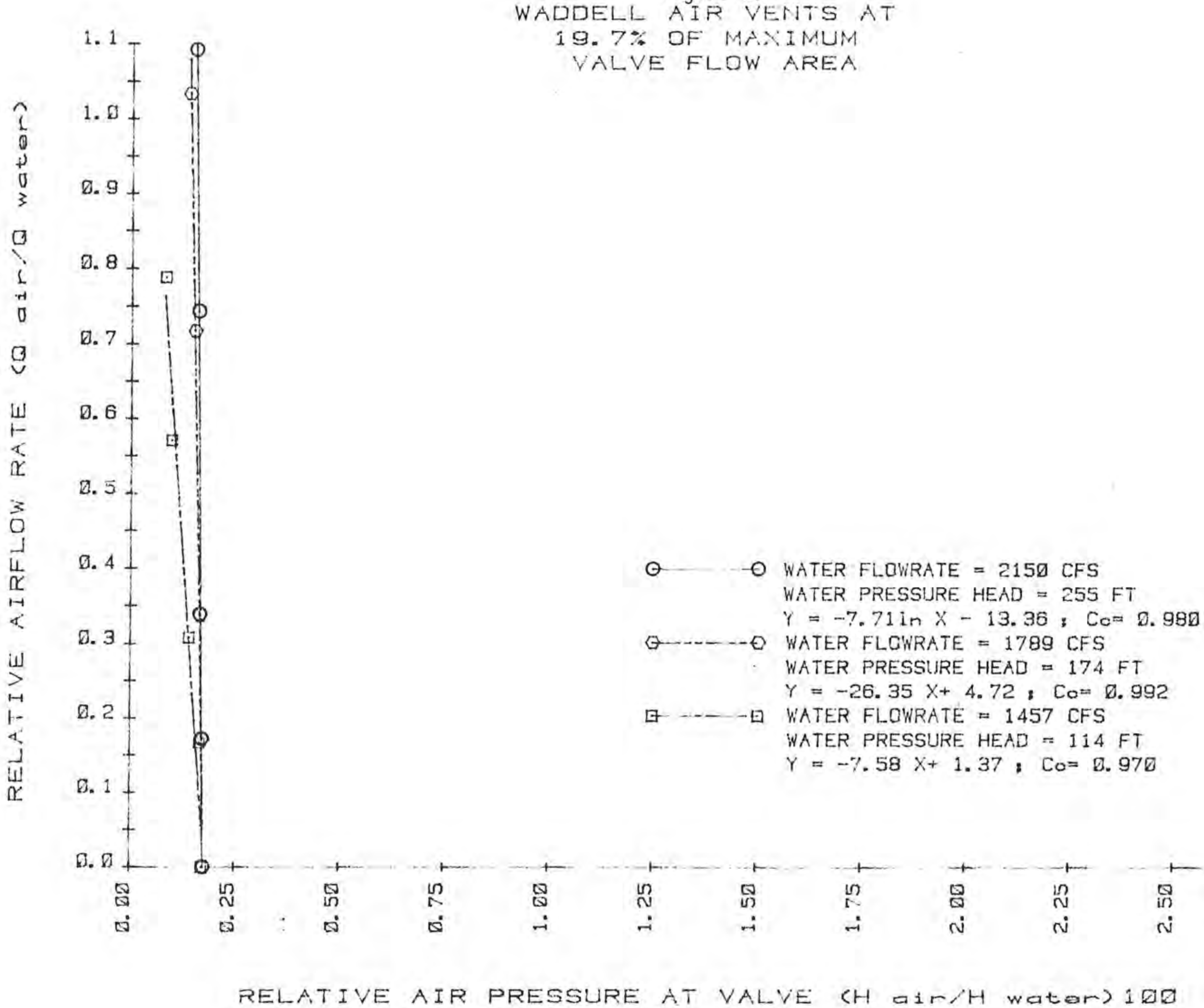
WADDELL AIR VENTS AT
19.7% OF MAXIMUM
VALVE FLOW AREA
RELATIVE AIRFLOW RATE (Q air/Q water)
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
2.25
2.50
2.75
WATER FLOWRATE = 2150 CFS
WATER PRESSURE HEAD = 255 FT
Y = -7.71ln X - 13.36 ; Co= 0.980
WATER FLOWRATE = 1789 CFS
WATER PRESSURE HEAD = 174 FT
Y = -26.35 X+ 4.72 ; Co= 0.992
WATER FLOWRATE = 1457 CFS
WATER PRESSURE HEAD = 114 FT
Y = -7.58 X+ 1.37 ; Co= 0.970
RELATIVE AIR PRESSURE AT VALVE (H air/H water) 100

Figure 11

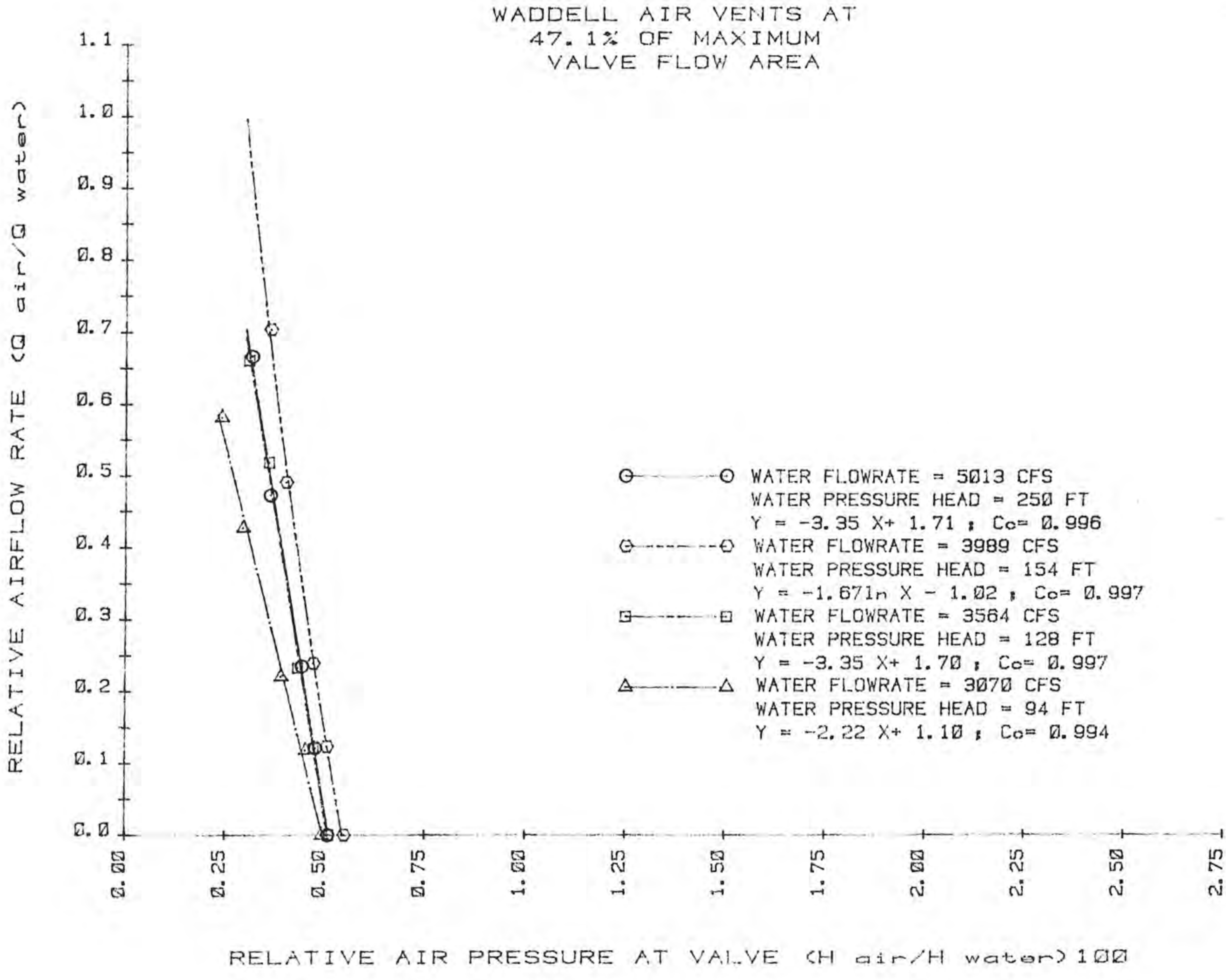
WADDELL AIR VENTS AT
47.1% OF MAXIMUM
VALVE FLOW AREA
RELATIVE AIRFLOW RATE (Q air/Q water)
RELATIVE AIR PRESSURE AT VALVE (H air/H water) 100
WATER FLOWRATE = 5013 CFS
WATER PRESSURE HEAD = 250 FT
Y = -3.35 X+ 1.71 ; Co= 0.996
WATER FLOWRATE = 3989 CFS
WATER PRESSURE HEAD = 154 FT
Y = -1.671n X - 1.02 ; Co= 0.997
WATER FLOWRATE = 3564 CFS
WATER PRESSURE HEAD = 128 FT
Y = -3.35 X+ 1.70 ; Co= 0.997
WATER FLOWRATE = 3070 CFS
WATER PRESSURE HEAD = 94 FT
Y = -2.22 X+ 1.10 ; Co= 0.994
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
2.25
2.50
2.75

Figure 12

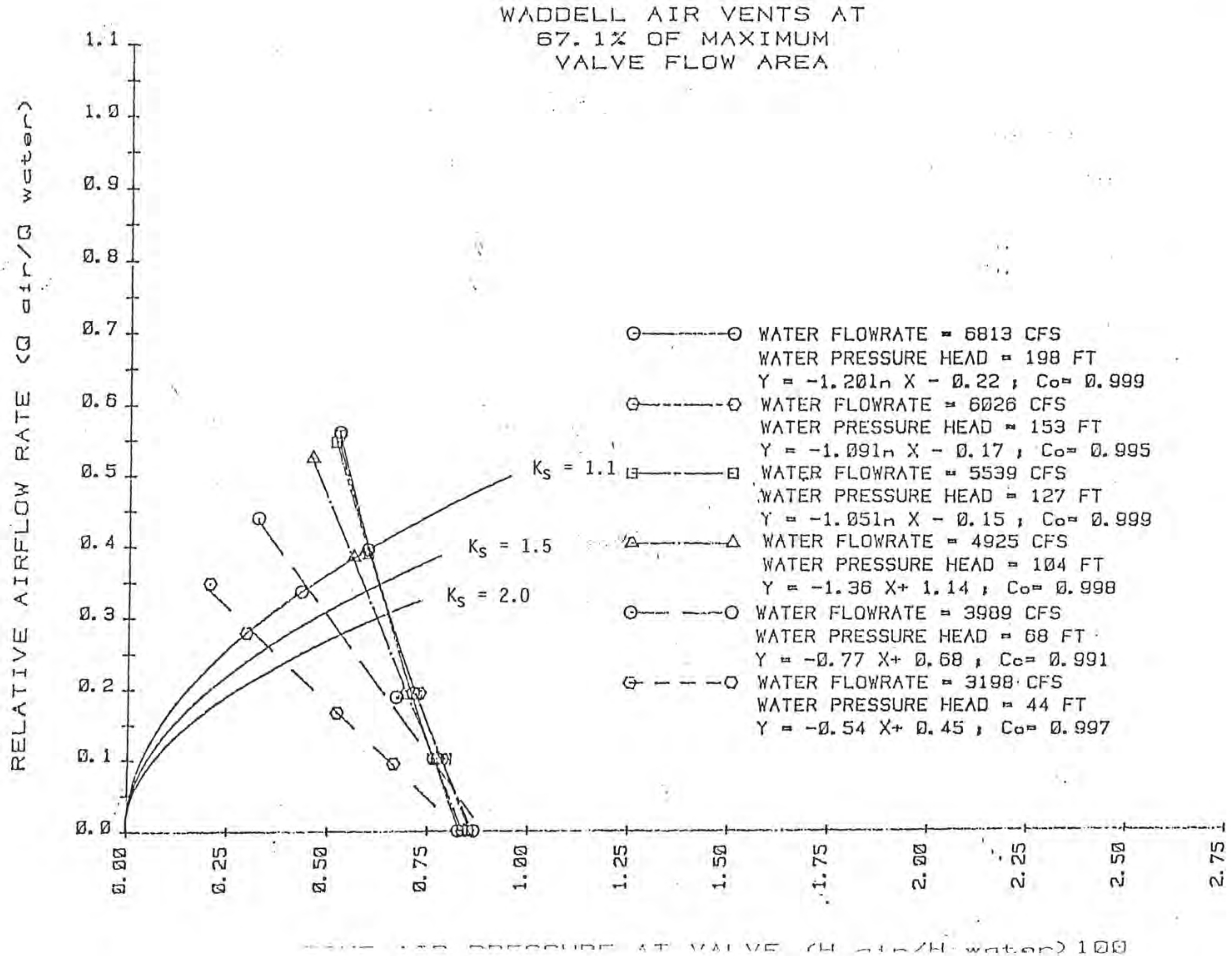
WADDELL AIR VENTS AT
67.1% OF MAXIMUM
VALVE FLOW AREA
RELATIVE AIRFLOW RATE (Q air/Q water)
K_S = 1.1
K_S = 1.5
K_S = 2.0
WATER FLOWRATE = 6813 CFS
WATER PRESSURE HEAD = 198 FT
Y = -1.20ln X - 0.22 ; Co= 0.999
WATER FLOWRATE = 6026 CFS
WATER PRESSURE HEAD = 153 FT
Y = -1.09ln X - 0.17 ; Co= 0.995
WATER FLOWRATE = 5539 CFS
WATER PRESSURE HEAD = 127 FT
Y = -1.05ln X - 0.15 ; Co= 0.999
WATER FLOWRATE = 4925 CFS
WATER PRESSURE HEAD = 104 FT
Y = -1.36 X+ 1.14 ; Co= 0.998
WATER FLOWRATE = 3989 CFS
WATER PRESSURE HEAD = 68 FT
Y = -0.77 X+ 0.68 ; Co= 0.991
WATER FLOWRATE = 3198 CFS
WATER PRESSURE HEAD = 44 FT
Y = -0.54 X+ 0.45 ; Co= 0.997

Figure 13

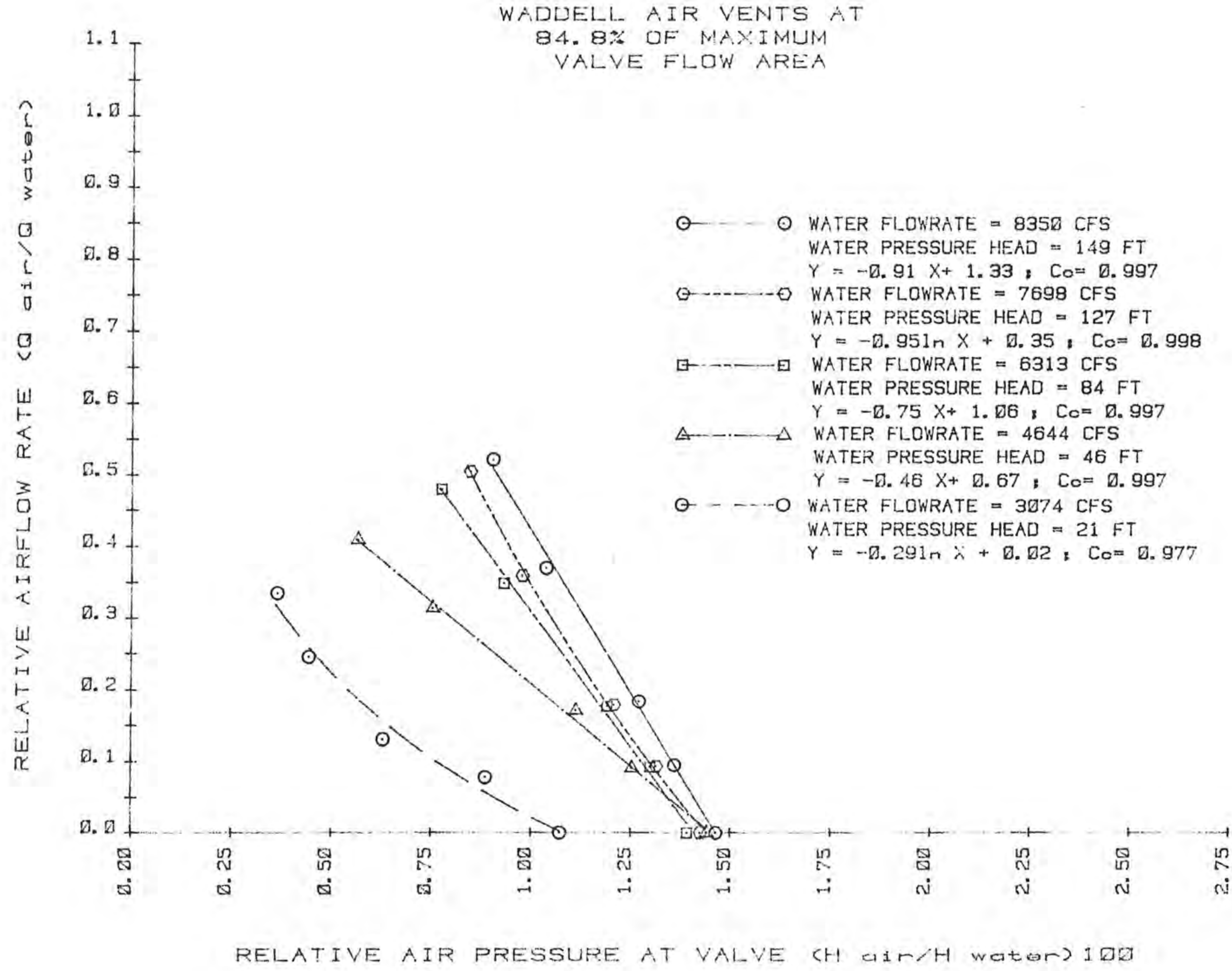
WADDELL AIR VENTS AT
84.8% OF MAXIMUM
VALVE FLOW AREA
WATER FLOWRATE = 8350 CFS
WATER PRESSURE HEAD = 149 FT
Y = -0.91 X+ 1.33 ; Co= 0.997
WATER FLOWRATE = 7698 CFS
WATER PRESSURE HEAD = 127 FT
Y = -0.951n X + 0.35 ; Co= 0.998
WATER FLOWRATE = 6313 CFS
WATER PRESSURE HEAD = 84 FT
Y = -0.75 X+ 1.06 ; Co= 0.997
WATER FLOWRATE = 4644 CFS
WATER PRESSURE HEAD = 46 FT
Y = -0.46 X+ 0.67 ; Co= 0.997
WATER FLOWRATE = 3074 CFS
WATER PRESSURE HEAD = 21 FT
Y = -0.291n X + 0.02 ; Co= 0.977
RELATIVE AIRFLOW RATE (Q air/Q water)
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
2.25
2.50
2.75
RELATIVE AIR PRESSURE AT VALVE (H air/H water) 100

Figure 14

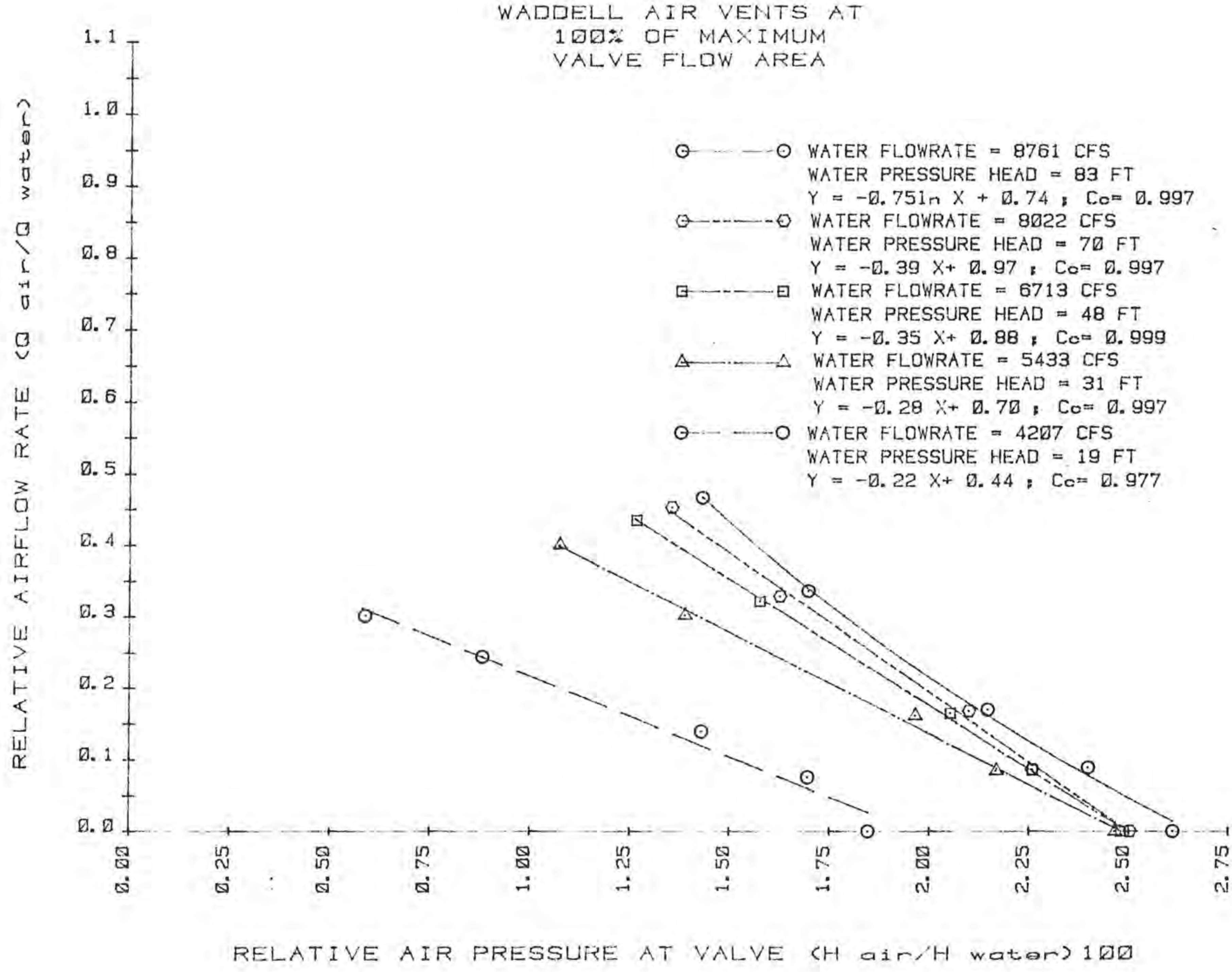
WADDELL AIR VENTS AT
100% OF MAXIMUM
VALVE FLOW AREA
WATER FLOWRATE = 8761 CFS
WATER PRESSURE HEAD = 83 FT
Y = -0.75ln X + 0.74 ; Co= 0.997
WATER FLOWRATE = 8022 CFS
WATER PRESSURE HEAD = 70 FT
Y = -0.39 X+ 0.97 ; Co= 0.997
WATER FLOWRATE = 6713 CFS
WATER PRESSURE HEAD = 48 FT
Y = -0.35 X+ 0.88 ; Co= 0.999
WATER FLOWRATE = 5433 CFS
WATER PRESSURE HEAD = 31 FT
Y = -0.28 X+ 0.70 ; Co= 0.997
WATER FLOWRATE = 4207 CFS
WATER PRESSURE HEAD = 19 FT
Y = -0.22 X+ 0.44 ; Co= 0.977
RELATIVE AIRFLOW RATE (Q air/Q water)
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
2.25
2.50
2.75
RELATIVE AIR PRESSURE AT VALVE (H air/H water) 100

Figure 15

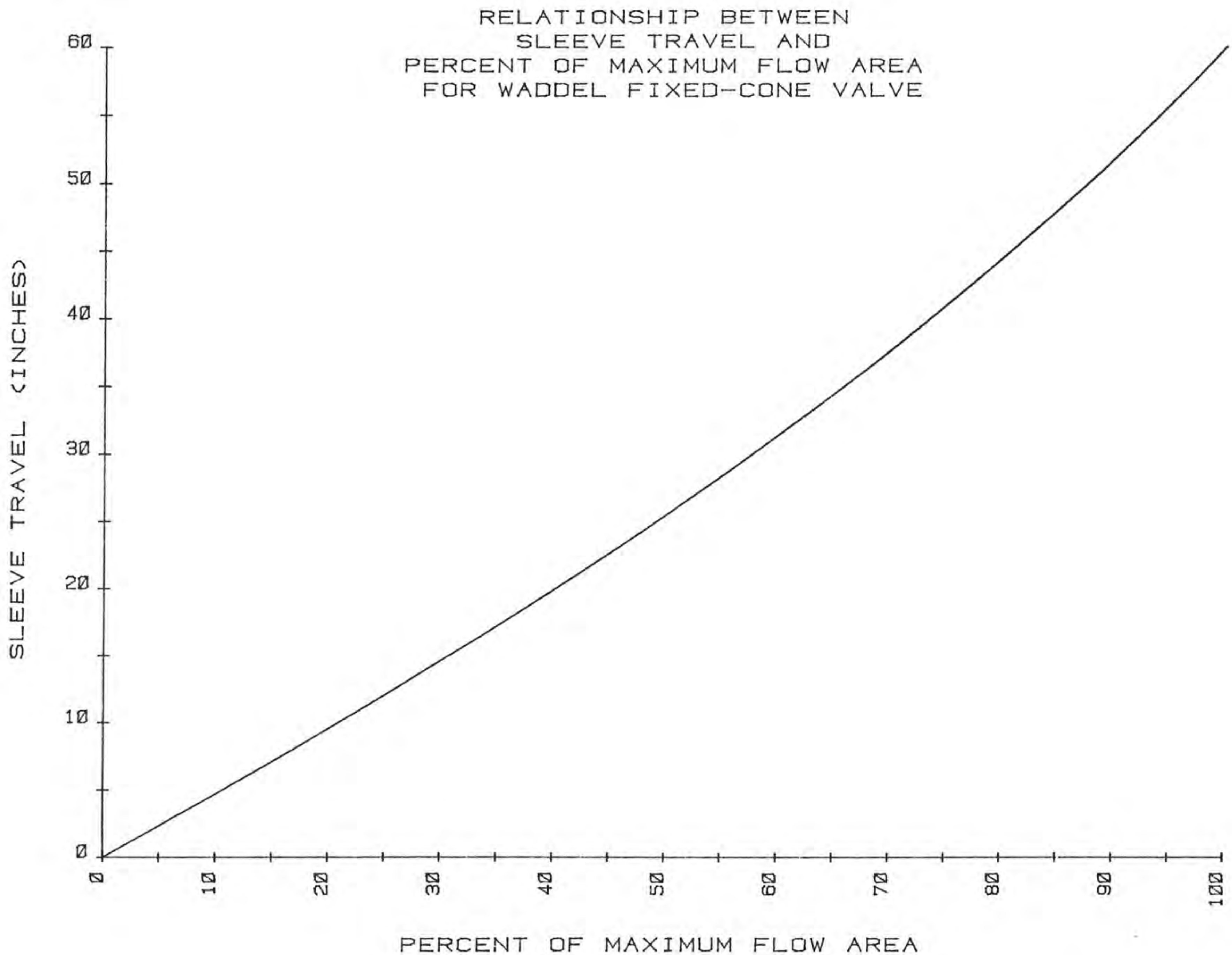
RELATIONSHIP BETWEEN
SLEEVE TRAVEL AND
PERCENT OF MAXIMUM FLOW AREA
FOR WADDEL FIXED-CONE VALVE
SLEEVE TRAVEL (INCHES)
0
10
20
30
40
50
60
0
10
20
30
40
50
60
70
80
90
100
PERCENT OF MAXIMUM FLOW AREA

Figure 16A. - Streamlined bell mouth entrance.

$K_S = 1.1$

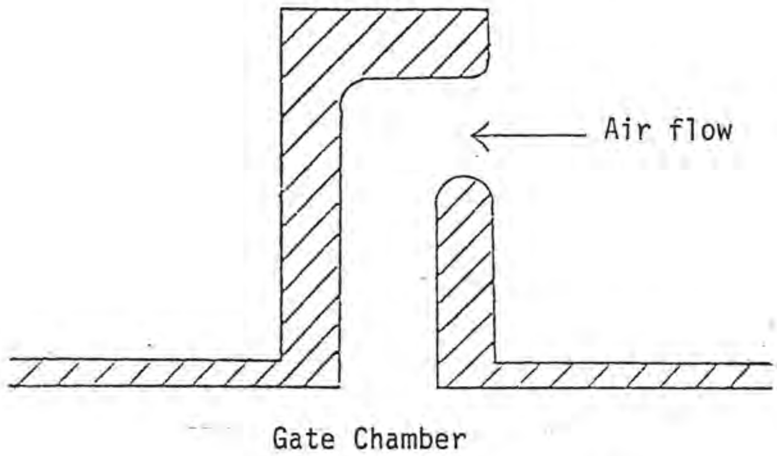

Figure 16B. - Square-edged entrance.

$K_S = 1.5$

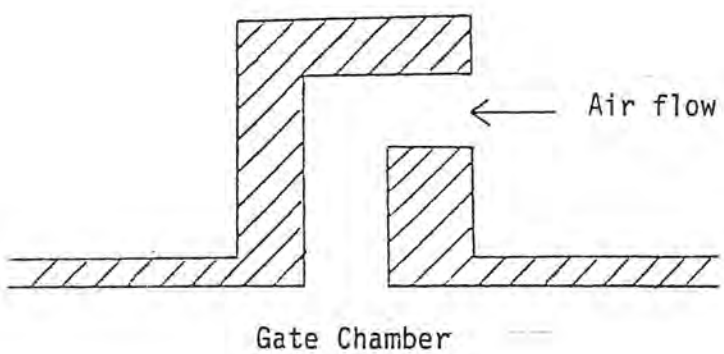

Figure 16C. - Pipe entrance.