The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX A

Geotechnical

The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX B

Hydraulic

CORPS OF ENGINEERS, U.S. ARMY WALLA WALLA DISTRICT

ESC DIVISION H BRANCH HYDRAULICS SECTION

PROJECT DALLES EFL EMERGENCY AWS

SUBJECT 8'X8' AWS BOX CULVERT & HEADER/INTAKE

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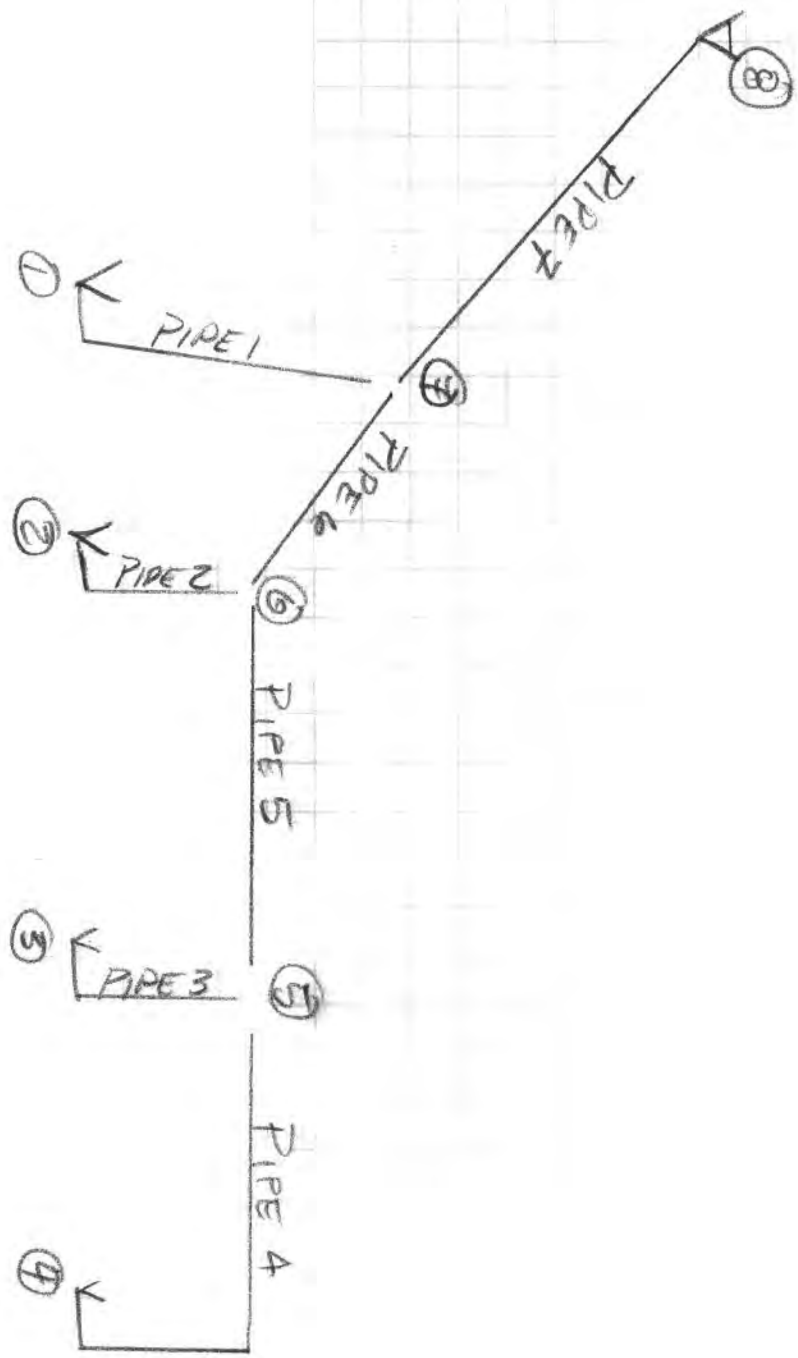

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PROJECT: DALLES EFL EMERGENCY AWS

SUBJECT: 8'X8' AWS BOX CULVERT & INTAKE

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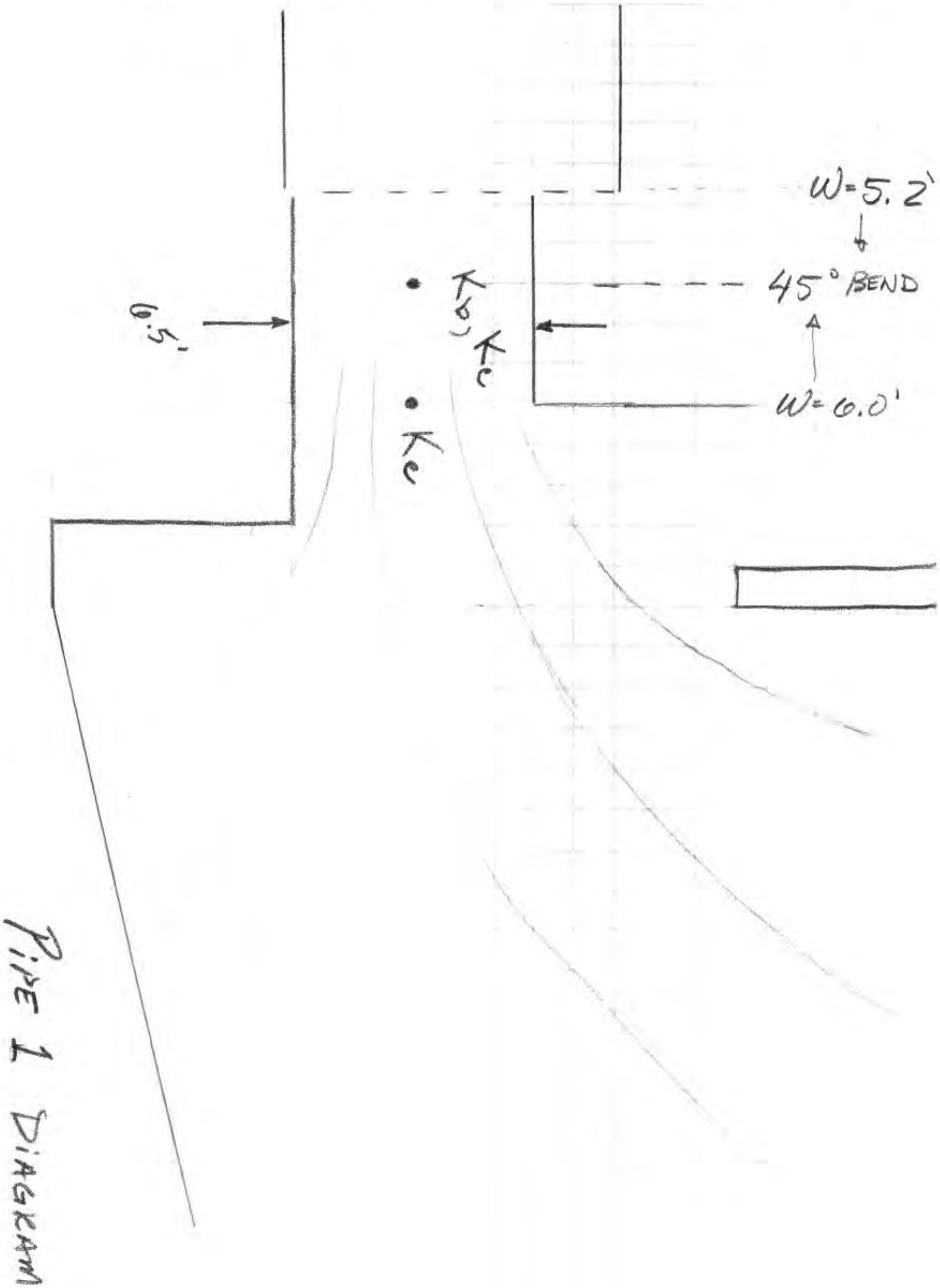

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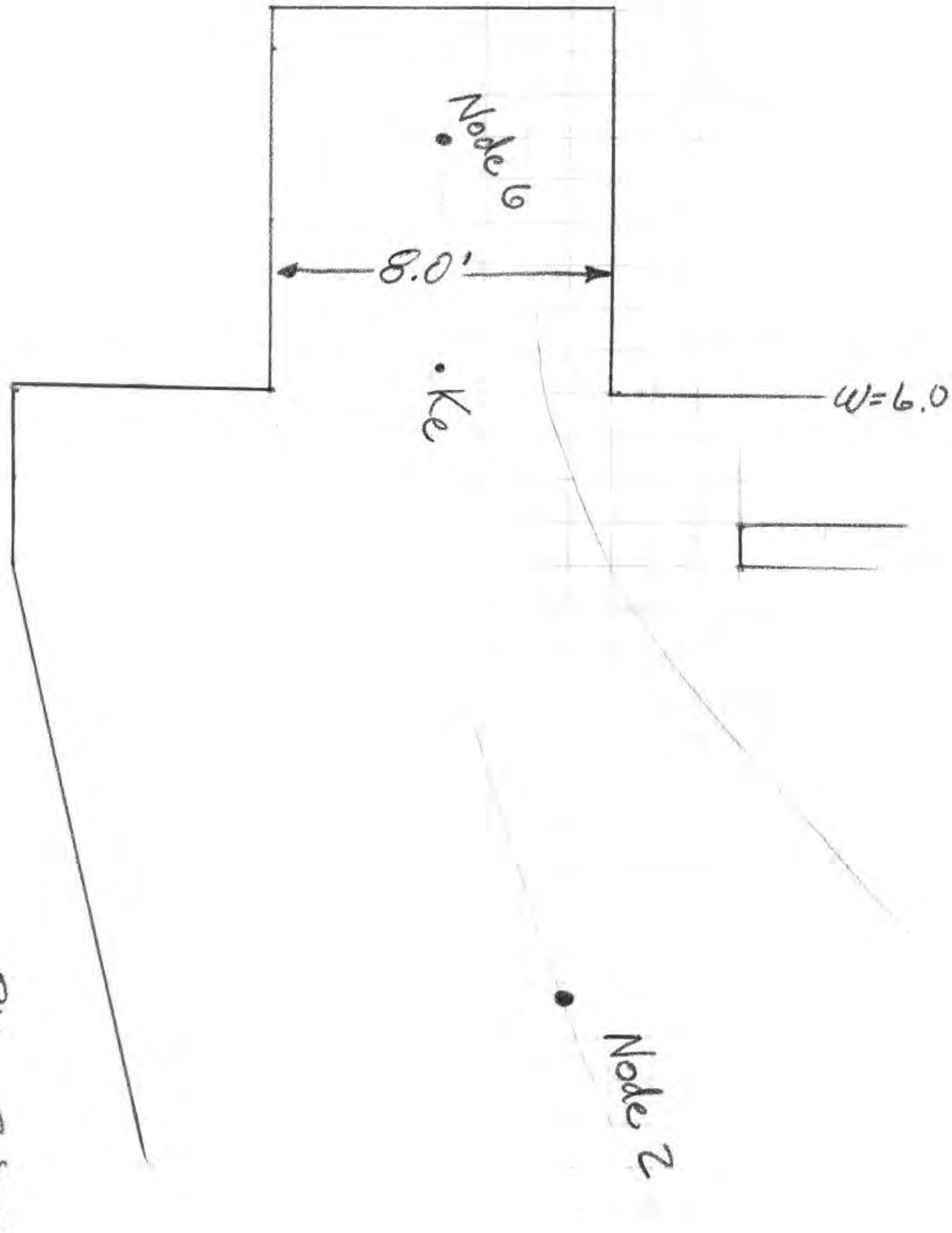

PIPE 2 DIAGRAM

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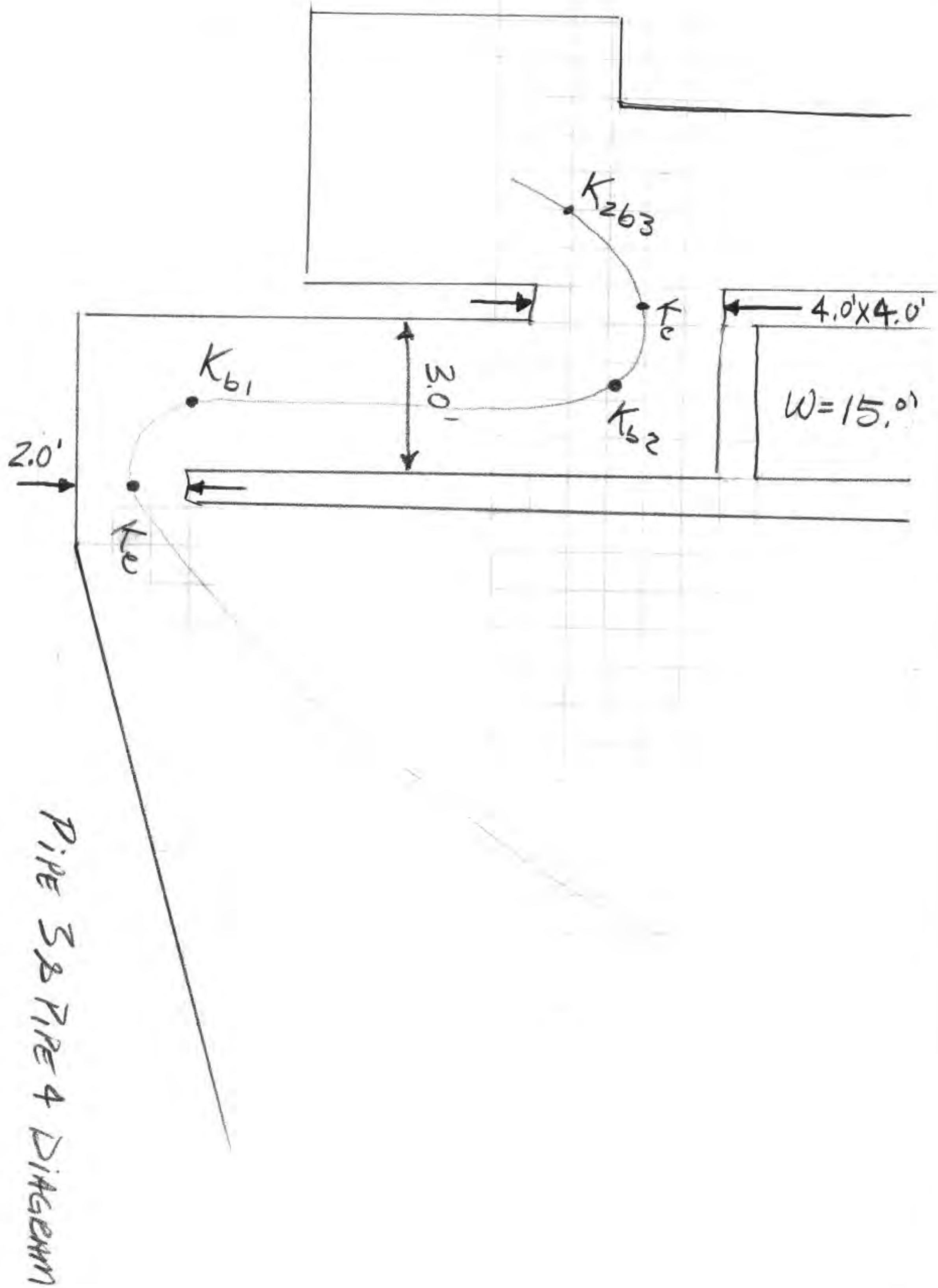

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WATER SURFACE ELEVATION

NODE	WSE
1	102.5'
2	102.5'
3	102.5'
4	102.5'
8	90.5' MAX

MAX HEAD AV = 12.0'

HEAD LOSS EQUALITIES

$$H_{L1} = H_{L2} + H_{L6} \quad (1)$$

$$= H_{L3} + H_{L5} + H_{L6} \quad (2)$$

$$= H_{L4} + H_{L5} + H_{L6} \quad (3)$$

$$H_{L2} = H_{L3} + H_{L5} \quad (4)$$

$$= H_{L4} + H_{L5} \quad (5)$$

$$H_{L3} = H_{L4} \quad (6)$$

MAX HEAD AV. = 12.0 ft

$$= H_{L7} + H_{L1} \quad (7)$$

$$= H_{L7} + H_{L6} + H_{L2} \quad (8)$$

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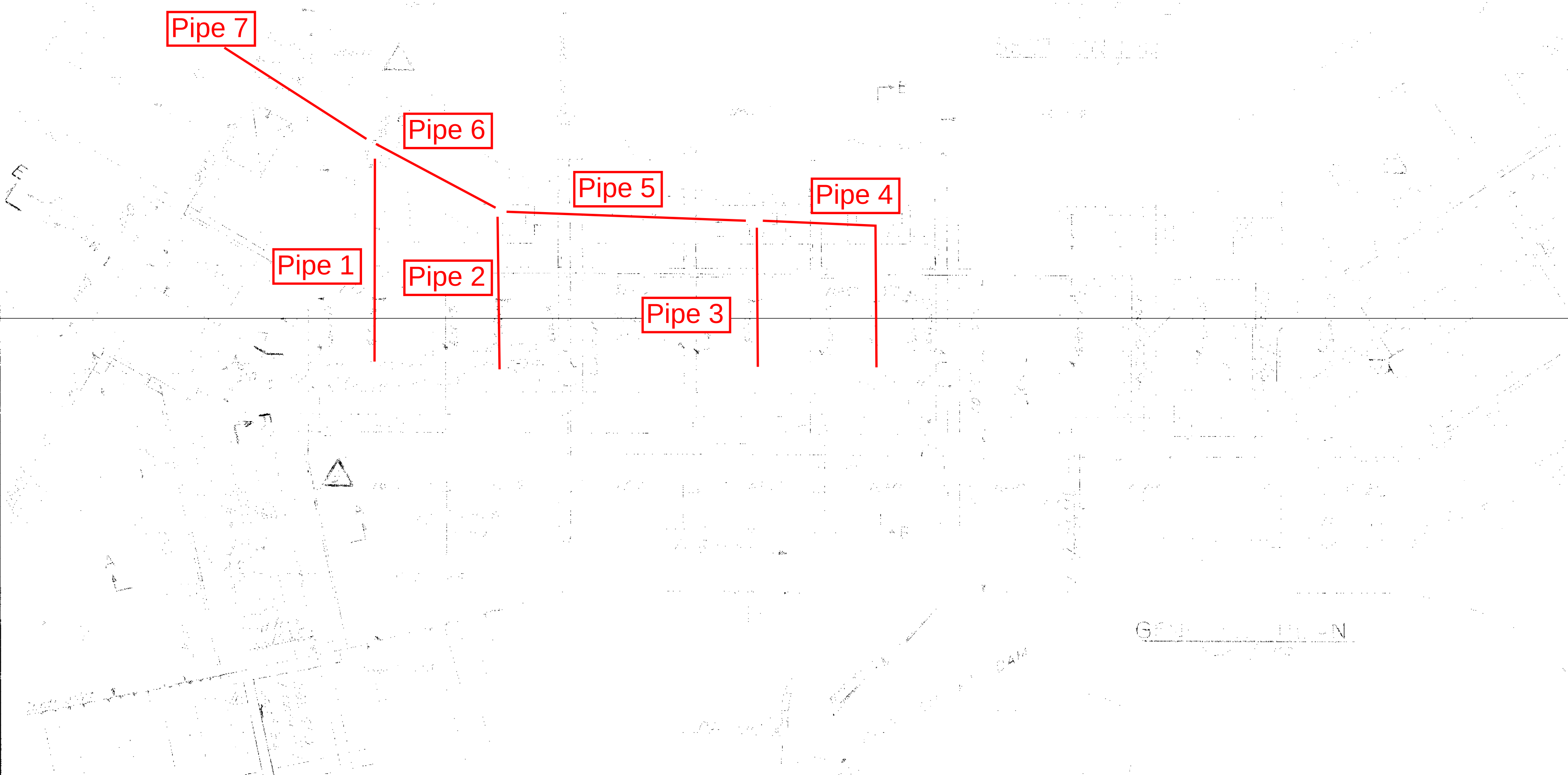
$$= H_{L7} + H_{L6} + H_{L5} + H_{L3} \quad (9)$$

$$= H_{L7} + H_{L6} + H_{L5} + H_{L4} \quad (10)$$

Network overlay for FLAC to AWS Chamber via 8 by 8 AWS Box Culvert



8 ft by 8 ft Culvert and Header Computations

Custom Units Definition

$fps := ft \cdot s^{-1}$ feet per second

$cfs := ft^2 \cdot fps$ cubic feet per second

Fluid Properties

$\rho := 1000 \frac{kg}{m^3}$ fluid density

Assumed temperature deg. F

$T_f := 50$ $T_c := (T_f - 32) \cdot \frac{5}{9}$ $T_c = 10$ Temp. deg. C

$\nu := \frac{1.792 \cdot 10^{-6}}{1.0 + \left(0.0337 \cdot T_c + 0.000221 \cdot T_c^2\right)} \cdot \frac{m^2}{s}$ $\nu = 1.319 \times 10^{-6} \cdot \frac{m^2}{s}$ Kinematic viscosity of water from temp. relationship

Global Functions

Area function: $A(h,w) := h \cdot w$

Equivalent diameter for rectangular conduit: $D'(h,w) := \frac{4 \cdot A(h,w)}{2 \cdot h + 2 \cdot w}$

Reynolds number: $Re(Q,h,w) := \frac{Q \cdot D'(h,w)}{A(h,w) \cdot \nu}$

Average velocity: $V(Q,h,w) := \frac{Q}{A(h,w)}$

Jain's equation for friction factor

$$f(Q,h,w,k_s) := \frac{0.25}{\log\left(\frac{k_s}{3.7 \cdot D'(h,w)} + \frac{5.74}{Re(Q,h,w)^{0.9}}\right)^2}$$

Ref: Swamee and Jain, 1976, "Explicit equations for pipe-flow problems," Journal of Hydr. Div. ASCE, Vol. 102, No. HY5, pp. 657-664

Assumed concrete equivalent sand grain roughness

$k_s := 0.002 ft$ assumed roughness

$k_{sr} := 0.01 ft$ extremely rough

$k_{ss} := 0.00015 ft$ fully smooth

Driving head characteristics

WSE in FLAC WSE in AWS chamber

$HW := 102.5 ft$ $TW := 90.5 ft$ $Available := HW - TW$ $Available = 12\,ft$

For the header losses through modified diffusers

Pipe 1 Floor diffuser 1 (node 1 to node 7)

$Q_{1'} := 300\text{cfs}$ Trial flow rate for loss coefficient estimation

$D_{h1} := 6.5\text{ft}$ Conduit height

$D_{w1.1} := 6.0\text{ft}$ Initial conduit width

$D_{w1.2} := 5.2\text{ft}$ Final conduit width

$L_1 := 5\text{ft}$ Estimation of conduit length

Entrance loss

$K_{1e} := 0.5$ Miller Fig. 14.11

Bend loss 45 deg

$k'_b := 0.3$ From Miller Fig. 9.9

$\text{Re}\left(Q_{1'}, D_{h1}, D_{w1.1}\right) = 3.382 \times 10^6$ Reynolds number at entrance

$C_{Re} := 1.0$ From Miller Fig. 9.3

$C_o := 1$ Miller Fig. 9.4

$C_f := \dfrac{f\left(Q_{1'}, D_{h1}, D_{w1.1}, k_{sr}\right)}{f\left(Q_{1'}, D_{h1}, D_{w1.1}, k_{ss}\right)}$ $C_f = 2.089$ From Miller Eq. 9.3

$K_{1b45} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{1b45} = 0.627$ From Miller Eq. 9.4

Contraction loss

$$\frac{A\left(D_{h1}, D_{w1.2}\right)}{A\left(D_{h1}, D_{w1.1}\right)} = 0.867$$

$K_{1c} := 0.08$ Miller Fig. 14.14

Combining tee loss

$K_{t17} := 0.4$ Miller Fig 13.10

Friction loss

Contracted section assumed negligible for friction due to short section

$$f_1(Q) := f\left(Q, D_{h1}, D_{w1.1}, k_s\right) \cdot \frac{L_1}{D'\left(D_{h1}, D_{w1.1}\right)}$$

Pipe 1 summation of losses

$$H_1(Q) := \left(K_{1e} + K_{1b45} + K_{1c} + f_1(Q) + K_{t17}\right) \cdot \frac{V\left(Q, D_{h1}, D_{w1.1}\right)^2}{2 \cdot g}$$

$H_1\left(Q_{1'}\right) = 1.489\text{ ft}$ $V\left(Q_{1'}, D_{h1}, D_{w1.1}\right) = 7.692\,\dfrac{\text{ft}}{\text{s}}$

Pipe 2 Floor diffuser 2 (node 2 to node 6)

Entrance loss

$Q_{2'} := 130\text{cfs}$ Trial flow rate for loss coefficient estimation

$D_{h2} := 6.5\text{ft}$ Conduit height

$D_{w2} := 6.0\text{ft}$ Conduit width

$L_2 := 0\text{ft}$ Estimation of conduit length

Entrance loss

$K_{2e} := 0.5$ Miller Fig. 14.11

Assumed friction loss negligible

Combining tee loss

$K_{t26} := 8$ Miller Fig 13.10

Pipe 2 summation of losses

$$H_2(Q) := \left(K_{2e} + K_{t26}\right)\cdot\frac{V\left(Q, D_{h2}, D_{w2}\right)^2}{2\cdot g}$$

$$H_2\left(Q_{2'}\right) = 1.468\text{ ft}$$

$$V\left(Q_{2'}, D_{h2}, D_{w2}\right) = 3.333\ \frac{\text{ft}}{\text{s}}$$

Pipe 3 Floor diffuser 3 (node 3 to node 5)

Entrance loss

$Q_{3'} := 100\text{cfs}$ Trial flow rate for loss coefficient estimation

$D_{h3} := 3\text{ft}$ Conduit height

$D_{w3} := 15\text{ft}$ Conduit width

$L_3 := 15\text{ft}$ Estimation of conduit length

Entrance loss

$K_{3e} := 0.5$ Miller Fig. 14.11

Bend loss

$k'_b := 1.2$ From Miller Fig. 9.9

$Re\left(Q_{3'}, D_{h3}, D_{w3}\right) = 7.829 \times 10^5$ Reynolds number at entrance

$C_{Re} := 1.0$ From Miller Fig. 9.3

$C_o := 1$ Miller Fig. 9.4

$C_f := \dfrac{f\left(Q_{3'}, D_{h3}, D_{w3}, k_{sr}\right)}{f\left(Q_{3'}, D_{h3}, D_{w3}, k_{ss}\right)}$ $C_f = 1.869$ From Miller Eq. 9.3

$K_{3b90.1} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{3b90.1} = 2.243$ From Miller Eq. 9.4

$K_{3b90.2} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{3b90.2} = 2.243$ From Miller Eq. 9.4

Orifice loss

$$K_{3o} := \left(1 - \frac{16\text{ft}^2}{D_{h3} \cdot D_{w3}}\right)^2 \cdot \left(\frac{D_{h3} \cdot D_{w3}}{16\text{ft}^2}\right)^2$$ $K_{3o} = 3.285$ Miller Eq. 14.2

Combining tee loss

$K_{t35} := 1.5$ Miller Fig 13.10

Friction loss Contracted section assumed negligible for friction due to short section

$$f_3(Q) := f\left(Q, D_{h3}, D_{w3}, k_s\right) \cdot \frac{L_3}{D'\left(D_{h3}, D_{w3}\right)}$$

Pipe 3 summation of losses

$$H_3(Q) := \left(K_{3e} + K_{3b90.1} + K_{3b90.2} + K_{3o} + f_3(Q) + K_{t35}\right) \cdot \frac{V\left(Q, D_{h3}, D_{w3}\right)^2}{2 \cdot g}$$

$H_3\left(Q_{3'}\right) = 0.754\,\text{ft}$ $V\left(Q_{3'}, D_{h3}, D_{w3}\right) = 2.222\,\dfrac{\text{ft}}{\text{s}}$

Pipe 4 Floor diffuser 4 (node 4 to node 5)

Entrance loss

$Q_{4'} := 90\text{cfs}$ Trial flow rate for loss coefficient estimation

$D_{h4.1} := 3\text{ft}$ Conduit height, segment 1

$D_{w4.1} := 15\text{ft}$ Conduit width, segment 1

$D_{h4.2} := 8\text{ft}$ Conduit height, segment 2

$D_{w4.2} := 4\text{ft}$ Conduit width, segment 2

$L_{4.1} := 15\text{ft}$ Estimation of conduit length

$L_{4.2} := 15\text{ft}$ Estimation of conduit length

Entrance loss

$K_{4e} := 0.5$ Miller Fig. 14.11

Bend loss, 1 & 2

$k'_b := 1.2$ From Miller Fig. 9.9

$\text{Re}\left(Q_{4'}, D_{h4.1}, D_{w4.1}\right) = 7.046 \times 10^5$ Reynolds number at entrance

$C_{Re} := 1.1$ From Miller Fig. 9.3

$C_o := 1$ Miller Fig. 9.4

$$C_f := \frac{f\left(Q_{4'}, D_{h4.1}, D_{w4.1}, k_{sr}\right)}{f\left(Q_{4'}, D_{h4.1}, D_{w4.1}, k_{ss}\right)} \qquad C_f = 1.845$$ From Miller Eq. 9.3

$K_{4b90.1} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{4b90.1} = 2.435$ From Miller Eq. 9.4

$K_{4b90.2} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{4b90.2} = 2.435$ From Miller Eq. 9.4

Orifice loss

$$K_{4o} := \left(1 - \frac{16\text{ft}^2}{D_{h4.1} \cdot D_{w4.1}}\right)^2 \cdot \left(\frac{D_{h4.1} \cdot D_{w4.1}}{16\text{ft}^2}\right)^2 \qquad K_{4o} = 3.285$$ Miller Eq. 14.2

Bend loss, 3

$k'_b := 1.2$ From Miller Fig. 9.9

$\text{Re}\left(Q_{4'}, D_{h4.2}, D_{w4.2}\right) = 1.057 \times 10^6$ Reynolds number at entrance

$C_{Re} := 1$ From Miller Fig. 9.3

$C_o := 1$ Miller Fig. 9.4

$$C_f := \frac{f\left(Q_{4'}, D_{h4.2}, D_{w4.2}, k_{sr}\right)}{f\left(Q_{4'}, D_{h4.2}, D_{w4.2}, k_{ss}\right)} \qquad C_f = 1.913$$ From Miller Eq. 9.3

$$K_{4b90.3} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f \qquad K_{4b90.3} = 2.296$$ From Miller Eq. 9.4

Combining Tee loss

$K_{t45} := 0.65$ Miller Fig 13.11

Friction loss segment 1

$$f_{4.1}(Q) := f\left(Q, D_{h4.1}, D_{w4.1}, k_s\right) \cdot \frac{L_{4.1}}{D'\left(D_{h4.1}, D_{w4.1}\right)}$$

Friction loss segment 2

$$f_{4.2}(Q) := f\left(Q, D_{h4.2}, D_{w4.2}, k_s\right) \cdot \frac{L_{4.2}}{D'\left(D_{h4.2}, D_{w4.2}\right)}$$

Pipe 4 summation of losses segment 1

$$H_{4.1}(Q) := \left(K_{4e} + K_{4b90.1} + K_{4b90.2} + K_{4o} + f_{4.1}(Q)\right) \cdot \frac{V\left(Q, D_{h4.1}, D_{w4.1}\right)^2}{2 \cdot g}$$

$$V\left(Q_{4'}, D_{h4.1}, D_{w4.1}\right) = 2\ \frac{ft}{s}$$

Pipe 4 summation of losses segment 2

$$H_{4.2}(Q) := \left(K_{4b90.3} + f_{4.2}(Q) + K_{t35}\right) \cdot \frac{V\left(Q, D_{h4.2}, D_{w4.2}\right)^2}{2 \cdot g}$$

$$V\left(Q_{4'}, D_{h4.2}, D_{w4.2}\right) = 2.813\ \frac{ft}{s}$$

Pipe 4 summation of losses

$$H_4(Q) := H_{4.1}(Q) + H_{4.2}(Q) \qquad H_4\left(Q_{4'}\right) = 1.013\ ft$$

Pipe 5 Node 5 to node 6 $Q_5 = Q_3 + Q_4$

Friction loss

$Q_{5'} := Q_{3'} + Q_{4'}$ Trial flow rate for loss coefficient estimation

$D_{h5} := 8\text{ft}$ Conduit height

$D_{w5.1} := 5\text{ft}$ Initial conduit width

$D_{w5.2} := 6\text{ft}$ Final conduit width

$L_5 := 32\text{ft}$ Estimation of conduit length

Combining loss Q3 and Q4 - Used for K_{t45} and K_{t35} above

$$\frac{Q_{3'}}{Q_{5'}} = 0.526$$

Adjust losses above

$$\frac{16\text{ft}^2}{D_{h5} \cdot D_{w5.1}} = 0.4$$

Gate area over entering conduit area

Friction loss

$$f_5(Q) := f\left(Q, D_{h5}, D_{w5.1}, k_s\right) \cdot \frac{L_5}{D'\left(D_{h5}, D_{w5.1}\right)}$$

Combining tee loss

$K_{t56} := 1.0$ Miller Fig. 13.11

Expansion loss

$$\frac{D_{h5} \cdot D_{w5.1}}{64\text{ft}^2} = 0.625$$

64 ft^2 is the main conduit area expansion

$K_{ex} := 0.2$ Miller Fig. 14.15

Pipe 5 summation of losses

$$H_5(Q) := \left(f_5(Q) + K_{t56} + K_{ex}\right) \cdot \frac{V\left(Q, D_{h5}, D_{w5.1}\right)^2}{2 \cdot g}$$

$H_5\left(Q_{5'}\right) = 0.449\text{ ft}$ $V\left(Q_{5'}, D_{h5}, D_{w5.1}\right) = 4.75 \frac{\text{ft}}{\text{s}}$

Pipe 6 Node 6 to node 7 $Q_6 = Q_5 + Q_2$

$Q_{6'} := Q_{5'} + Q_{2'}$ Trial flow rate for loss coefficient estimation

$D_{h6} := 8ft$ Conduit height

$D_{w6} := 8ft$ Initial conduit width

$L_6 := 16ft$ Estimation of conduit length

Combining loss Q5 and Q2 - Used for K_{t56} and K_{t26} above

$\frac{Q_{2'}}{Q_{5'}} = 0.684$

Adjust losses above

$\frac{16ft^2}{D_{h6} \cdot D_{w6}} = 0.25$

Friction loss

$f_6(Q) := f(Q, D_{h6}, D_{w6}, k_s) \cdot \frac{L_6}{D'(D_{h6}, D_{w6})}$

Combining tee loss

$Q_{7'} := Q_{6'} + Q_{1'}$ $Q_{7'} = 620 \cdot cfs$

$\frac{Q_{1'}}{Q_{7'}} = 0.484$ $\frac{D_{h1} \cdot D_{w1.2}}{64ft^2} = 0.528$

$K_{t67} := 0.45$ Miller Fig 13.11

$H_6(Q) := (f_6(Q) + K_{t67}) \cdot \frac{V(Q, D_{h6}, D_{w6})^2}{2 \cdot g}$

$H_6(Q_{6'}) = 0.186\, ft$ $V(Q_{6'}, D_{h6}, D_{w6}) = 5\, \frac{ft}{s}$

Pipe 7 Node 7 to AWS chamber

$D_{h7} := 8ft$ Conduit height

$D_{w7} := 8ft$ Initial conduit width

$L_7 := 85ft$ Estimation of conduit length

Friction

$Re\left(Q_{7'}, D_{h7}, D_{w7}\right) = 5.461 \times 10^{6}$

$f\left(Q_{7'}, D_{h7}, D_{w7}, k_s\right) = 0.015$

$$h_{f7}(Q) := f\left(Q, D_{h7}, D_{w7}, k_s\right) \cdot \frac{L_7}{D'\left(D_{h7}, D_{w7}\right)} \cdot \frac{V\left(Q, D_{h7}, D_{w7}\right)^2}{2 \cdot g}$$

$h_{f7}\left(Q_{7'}\right) = 0.226 \cdot ft$

15 deg bend loss, 1 & 2

$\frac{r}{d} = 6$

$k'_b := 0.05$ From Miller Fig. 9.7

$C_{Re} := 1.0$ From Miller Fig. 9.3

$C_o := 0.9$ 6 diameters away, Miller Fig. 9.4

$C_f := \frac{f\left(Q_{7'}, D_{h7}, D_{w7}, k_{sr}\right)}{f\left(Q_{7'}, D_{h7}, D_{w7}, k_{ss}\right)}$ $C_f = 2.081$ From Miller Eq. 9.3

$K_{b15.1} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{b15.1} = 2.248$ From Miller Eq. 9.4

$K_{b15.2} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{b15.2} = 2.248$ From Miller Eq. 9.4

90 deg bend loss
90 deg bend smooth

$\frac{r}{d} = 1$

$k'_b := 0.27$ From Miller Fig. 9.7

$C_{Re} := 1.0$ From Miller Fig. 9.3

$C_o := 2.7$ Immediate outlet, Miller Fig. 9.4

$$C_f := \frac{f(Q_{7'}, D_{h7}, D_{w7}, k_{sr})}{f(Q_{7'}, D_{h7}, D_{w7}, k_{ss})} \qquad C_f = 2.081$$ From Miller Eq. 9.3

$$K_{b90} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f \qquad K_{b90} = 1.517$$ From Miller Eq. 9.4

Exit Loss

$$K_e := 1.0$$

Pipe 7 Summation of losses

$$H_7(Q) := h_{f7}(Q) + (K_{b15.1} + K_{b15.2} + K_{b90} + K_e) \cdot \frac{V(Q, D_{h7}, D_{w7})^2}{2 \cdot g}$$

$$H_7(Q_{7'}) = 10.454\,\text{ft} \qquad V(Q_{7'}, D_{h7}, D_{w7}) = 9.688\,\frac{\text{ft}}{\text{s}}$$

Solve for available flow

Trial Flow Rates

$Q_{1'} = 300\cdot cfs$

$Q_{2'} = 130\cdot cfs$

$Q_{3'} = 100\cdot cfs$

$Q_{4'} = 90\cdot cfs$

$Q_5(Q_3, Q_4) := Q_3 + Q_4$

$Q_6(Q_3, Q_4, Q_2) := Q_2 + Q_3 + Q_4$

$Q_7(Q_1, Q_2, Q_3, Q_4) := Q_1 + Q_2 + Q_3 + Q_4$

Total available driving head

$H_a := 12.0ft$

Q 5, 6 & 7 are function of flow through pipes 1, 2, 3, & 4

Set Solve Block for Equalization of Head losses

Given

$H_1(Q_{1'}) = H_2(Q_{2'}) + H_6(Q_{3'} + Q_{4'} + Q_{2'})$

$H_1(Q_{1'}) = H_3(Q_{3'}) + H_5(Q_{3'} + Q_{4'}) + H_6(Q_{3'} + Q_{4'} + Q_{2'})$

$H_1(Q_{1'}) = H_4(Q_{4'}) + H_5(Q_{3'} + Q_{4'}) + H_6(Q_{3'} + Q_{4'} + Q_{2'})$

$H_2(Q_{2'}) = H_3(Q_{3'}) + H_5(Q_{3'} + Q_{4'})$

$H_2(Q_{2'}) = H_4(Q_{4'}) + H_5(Q_{3'} + Q_{4'})$

$H_3(Q_{3'}) = H_4(Q_{4'})$

$H_a = H_7(Q_{3'} + Q_{4'} + Q_{2'} + Q_{1'}) + H_1(Q_{1'})$

$H_a = H_7(Q_{3'} + Q_{4'} + Q_{2'} + Q_{1'}) + H_6(Q_{3'} + Q_{4'} + Q_{2'}) + H_2(Q_{2'})$

$H_a = H_7(Q_{3'} + Q_{4'} + Q_{2'} + Q_{1'}) + H_6(Q_{3'} + Q_{4'} + Q_{2'}) + H_5(Q_{3'} + Q_{4'}) + H_3(Q_{3'})$

$H_a = H_7(Q_{3'} + Q_{4'} + Q_{2'} + Q_{1'}) + H_6(Q_{3'} + Q_{4'} + Q_{2'}) + H_5(Q_{3'} + Q_{4'}) + H_4(Q_{4'})$

$$\begin{pmatrix} Q_1 \\ Q_2 \\ Q_3 \\ Q_4 \end{pmatrix} := \mathrm{Find}(Q_{1'}, Q_{2'}, Q_{3'}, Q_{4'})$$

$Q_1 = 303.813\cdot cfs$ $Q_2 = 124.413\cdot cfs$ $Q_3 = 108.289\cdot cfs$ $Q_4 = 84.052\cdot cfs$

$Q_5(Q_3, Q_4) = 192.341\cdot cfs$ $Q_6(Q_3, Q_4, Q_2) = 316.754\cdot cfs$ $Q_7(Q_1, Q_2, Q_3, Q_4) = 620.567\cdot cfs$

Back check of the solve block

$H_1(Q_1) = 1.527\,ft$ $\quad H_2(Q_2) + H_6(Q_3 + Q_4 + Q_2) = 1.527\,ft$

$H_3(Q_3) + H_5(Q_3 + Q_4) + H_6(Q_3 + Q_4 + Q_2) = 1.527\,ft$

$H_4(Q_4) + H_5(Q_3 + Q_4) + H_6(Q_3 + Q_4 + Q_2) = 1.527\,ft$

$H_2(Q_2) = 1.344\,ft$ $\quad H_3(Q_3) + H_5(Q_3 + Q_4) = 1.344\,ft$

$H_4(Q_4) + H_5(Q_3 + Q_4) = 1.344\,ft$

$H_3(Q_3) = 0.884\,ft$ $\quad H_4(Q_4) = 0.884\,ft$

$H_a = 12\,ft$ $\quad H_7(Q_3 + Q_4 + Q_2 + Q_1) + H_1(Q_1) = 12\,ft$

$H_7(Q_3 + Q_4 + Q_2 + Q_1) + H_6(Q_3 + Q_4 + Q_2) + H_2(Q_2) = 12\,ft$

$H_7(Q_3 + Q_4 + Q_2 + Q_1) + H_6(Q_3 + Q_4 + Q_2) + H_5(Q_3 + Q_4) + H_3(Q_3) = 12\,ft$

$H_7(Q_3 + Q_4 + Q_2 + Q_1) + H_6(Q_3 + Q_4 + Q_2) + H_5(Q_3 + Q_4) + H_4(Q_4) = 12\,ft$

Units Definition $cfs := ft^3 \cdot s^{-1}$ Units Coefficient $C_u := 1.486\sqrt[3]{ft \cdot s^{-1}}$ Energy Coefficient $\alpha := 1.0$

Culvert Computations **Circular Culvert (Clean)** **Input** **Output**

Data Input

Pipe Diameter $D := 6ft = 72 \cdot in$

Pipe Manning's n $n := 0.01$

Pipe Slope $S_o := 0.001$ Flow Rate $Q := 400cfs$ Per culvert

FHWA Table 9 Coefficients

$K := 0.0098$ $M := 2.0$ $c := 0.0399$ $Y := 0.67$ Circular, concrete with headwall entrance

Culvert Length $L := 50ft$

Outlet Invert Ele. $OInv_{el} := 90.5ft$ TW Elevation $TW_{el} := 90.5ft$

Inlet Invert Ele. $InInv_{el} := OInv_{el} + S_o \cdot L$ $InInv_{el} = 90.55\ ft$

Loss Coefficients, Sturm pg 226

$k_e := 0.5$ Entrance loss $k_o := 1.0$ Exit loss

Hydraulic Geometry

Angle Functions $\theta(y) := 2 \cdot acos\left(1 - 2 \cdot \frac{y}{D}\right)$ $A_f := \frac{\pi \cdot D^2}{4}$

Area Functions $A(\theta) := \frac{D^2}{8} \cdot (\theta - \sin(\theta))$

Perimeter Functions $P(\theta) := \frac{D}{2} \cdot (\theta)$

Hydraulic Radius $R_H(\theta) := A(\theta) \cdot P(\theta)^{-1}$

Top Width $T(\theta) := D \cdot \sin\left(\frac{\theta}{2}\right)$

Full Pipe Condition

$y_f := 0.90D$ $y_f = 5.4\,ft$ $\theta_f := 2 \cdot acos\left(1 - 2\frac{y_f}{D}\right)$ $\theta_f = 4.996$

Inlet Control Computations

Critical Depth Computations

$N_f = \dfrac{V}{\sqrt{\alpha \cdot g \cdot D}}$ $\quad Z_c := \dfrac{Q^2}{g \cdot \alpha}$ $\quad Z_c = 4.973 \times 10^3\ ft^5$ Critical Section Factor

$\theta := 1.1\pi$ Trial value

Given Solve block for critical depth angle

$\dfrac{A(\theta)^3}{T(\theta)} = Z_c$ $\quad \theta_c := Find(\theta)$ $\quad \theta_c = 4.925$ $\quad \theta_c := \begin{cases} (2\cdot\pi) & \text{if } \theta_c > 2\cdot\pi \\ \theta_c & \text{otherwise} \end{cases}$ $\quad \theta_c = 4.925$

$y_c := \begin{cases} D & \text{if } \theta_c > \theta_f \\ \dfrac{D}{2}\cdot\left(1 - \cos\left(\dfrac{\theta_c}{2}\right)\right) & \text{otherwise} \end{cases}$ $\quad y_c = 5.335\ ft$ Critical Depth

Hydraulic Radius $\quad R_H(\theta_c) = 1.798\ ft$

Percent Full $\quad Per_{full}(y) := \dfrac{y}{D}$ $\quad Per_{full}(y_c) = 88.909\cdot\%$

Velocity $\quad V_c := Q\cdot A(\theta_c)^{-1}$ $\quad V_c = 15.059\ \dfrac{ft}{s}$ Critical Velocity

Top Width $\quad T(\theta_c) = 3.768\ ft$

Critical Slope $\quad S_c := \dfrac{Q^2\cdot n^2}{C_u^2\cdot A(\theta_c)^2\cdot\sqrt[3]{R_H(\theta_c)^4}}$ $\quad S_c = 0.47\cdot\%$

Inlet Condition Factor $\quad N := \dfrac{Q\cdot cfs^{-1}}{A_f\cdot ft^{-2}\cdot\sqrt{D\cdot ft^{-1}}}$ $\quad N = 5.776$

Specific Head at Critical Depth

$H_c := y_c + \dfrac{V_c^2}{2\cdot g}$ $\quad H_c = 8.859\ ft$

Calculate Headwater For Inlet Control, feet above culvert invert at the inlet:

$HW_{inlet} := \begin{cases} \left[H_c + D\cdot\left(K\cdot N^M - 0.5\cdot S_o\right)\right] & \text{if } N < 3.5 \\ \left[D\cdot\left(c\cdot N^2 + Y - 0.5\cdot S_o\right)\right] & \text{otherwise} \end{cases}$ $\quad HW_{inlet} = 12.003\ ft$ $\quad \dfrac{HW_{inlet}}{D} = 2$

Outlet Control Computations

Normal Depth Computation

Trial depth angle $\theta := 1.5\pi$

Given $Q = \frac{C_u}{n} \cdot A(\theta) \cdot R_H(\theta)^{\frac{2}{3}} \cdot \sqrt{S_o}$ $\theta_n := \text{Find}(\theta)$ $\theta_n = 15.215$

$$\theta_n := \begin{cases} (2\cdot\pi) & \text{if } \theta_n > 2\cdot\pi \\ \theta_n & \text{otherwise} \end{cases}$$ $\theta_n = 6.283$

	Normal Depth	Critical Depth
$y_n := \begin{cases} D & \text{if } \theta_n > \theta_f \\ \frac{D}{2}\cdot\left(1 - \cos\left(\frac{\theta_n}{2}\right)\right) & \text{otherwise} \end{cases}$	$y_n = 6\,\text{ft}$	$y_c = 5.335\,\text{ft}$

Flow Area $A(\theta_n) = 28.274\,\text{ft}^2$

Hydraulic Radius $R_H(\theta_n) = 1.5\,\text{ft}$

Percent Full $Per_{full}(y_n) = 100\cdot\%$

Velocity $V_n := Q\cdot A(\theta_n)^{-1}$ $V_n = 14.147\,\frac{\text{ft}}{\text{s}}$

Top Width $T(\theta_n) = 0\,\text{ft}$

Hydraulic Depth $D_{hn} := \frac{A(\theta_n)}{T(\theta_n)}$ $D_{hn} = 3.848 \times 10^{16}\,\text{ft}$

Calculate Headwater For Outlet Control, feet above culvert invert at the inlet:

$k_f := \frac{2\cdot g\cdot n^2\cdot L}{C_u^{\,2}\cdot\sqrt[3]{R_H(\theta_f)^4}}$ Friction loss based on full pipe $k_f = 0.067$

$TW := TW_{el} - OInv_{el}$ $TW = 0\,\text{ft}$ Tailwater depth $\frac{TW}{D} = 0$

$$\text{PipeCondition} := \begin{cases} \text{"pipe full"} & \text{if } TW > D \\ \text{"part-full pipe"} & \text{otherwise} \end{cases}$$ PipeCondition = "part-full pipe"

Calculation of Energy Loss

$H := (k_e + k_o + k_f)\cdot\frac{V_n^{\,2}}{2\cdot g}$ $H = 4.874\,\text{ft}$

Outlet Head

$$h_o := \begin{cases} TW & \text{if } TW > \dfrac{y_c + D}{2} \\ \dfrac{y_c + D}{2} & \end{cases}$$

$h_o = 5.667\ \text{ft}$

$HW_{out} := h_o - S_o \cdot L + H$

$HW_{out} = 10.491\ \text{ft}$

Control and Headwater

$$HW\_Condition := \begin{cases} \text{"Inlet"} & \text{if } HW_{inlet} > HW_{out} \\ \text{"Outlet"} & \text{otherwise} \end{cases}$$

$$HW := \begin{cases} HW_{inlet} & \text{if } HW_{inlet} > HW_{out} \\ HW_{out} & \text{otherwise} \end{cases}$$

$HW_{inlet} = 12.003\ \text{ft}$

$HW_{out} = 10.491\ \text{ft}$

$HW\_Condition = \text{"Inlet"}$

$HW = 12.003\ \text{ft}$

Design Summary - (Project)

$D = 6\ \text{ft}$	Culvert Diameter	$L = 50\ \text{ft}$	Culvert Length
$S_o = 1 \times 10^{-3}$	Invert Slope	$n = 0.01$	Manning's n for pipe
$Q = 400 \cdot \text{cfs}$	Discharge		
$y_c = 5.335\ \text{ft}$	Critical Depth		
$y_n = 6\ \text{ft}$	Normal Depth		

$HW\_Condition = \text{"Inlet"}$ Headwater Condition

$HW = 12.003\ \text{ft}$ Headwater Depth

$WSE_{inlet} := HW + InInv_{el}$

$WSE_{inlet} = 102.553\ \text{ft}$ Water surface elevation in FLAC

Custom Units Definition

$fps := ft \cdot s^{-1}$ feet per second

$cfs := ft^2 \cdot fps$ cubic feet per second

Fluid Properties

$\rho := 1000 \frac{kg}{m^3}$ Assumed density

Assumed temperature deg. F

$T_f := 50$ $T_c := (T_f - 32) \cdot \frac{5}{9}$ $T_c = 10$ Temp. deg. C

$$\nu := \frac{1.792 \cdot 10^{-6}}{1.0 + \left(0.0337 \cdot T_c + 0.000221 \cdot T_c^{\,2}\right)} \cdot \frac{m^2}{s}$$

$\nu = 1.319 \times 10^{-6} \cdot \frac{m^2}{s}$ Kinematic viscosity of water from temp. relationship

Area function | Reynolds number | Average velocity

$A(d) := \frac{\pi d^2}{4}$ $Re(Q,d) := \frac{Q \cdot d}{A(d) \cdot \nu}$ $V(Q,d) := \frac{Q}{A(d)}$

Jain's equation for friction factor

$$f(Q,d,k_s) := \frac{0.25}{\log\left(\frac{k_s}{3.7 \cdot d} + \frac{5.72}{Re(Q,d)^{0.9}}\right)^2}$$

Ref: Swamee and Jain, 1976, "Explicit equations for pipe-flow problems," Journal of Hydr. Div. ASCE, Vol. 102, No. HY5, pp. 657-664

Design Parameters

$Q := 1400\text{cfs}$ Design flow rate

$d := 7\text{ft}$ Trial Diameter

$R_h := 160\text{ft}$ Maximum forebay operating range

$R_l := 155\text{ft}$ Minimum forebay operating range

$TW := 102.5\text{ft}$ Design tailwater for stilling basin

$H_h := R_h - TW$ $H_h = 57.5\,\text{ft}$ Maximum driving head

$H_l := R_l - TW$ $H_l = 52.5\,\text{ft}$ Minimum driving head

$H := 40.25\text{ft} + \frac{V(Q,d)^2}{2 \cdot g}$ Minium head at valve with friction losses

Basin design sizing and valve selection

$C := 0.7$ Typical hollow-jet valve discharge coefficient

$A := \frac{Q}{C \cdot \sqrt{2g \cdot (H)}}$ $A = 31.971\,\text{ft}^2$ Required area

$d := \sqrt{\frac{4A}{\pi}}$ $d = 6.38\,\text{ft}$ $d_o := d$ Recommended diameter

$d := 7\text{ft}$ Selected diameter

$\text{Check\_d} := \begin{cases} \text{"ok"} & \text{if } d_o \le d \\ \text{"fix d"} & \text{otherwise} \end{cases}$ $\text{Check\_d} = \text{"ok"}$ Diameter selection check

$A := \pi \cdot \frac{d^2}{4}$ Cross sectional valve area

$C := \frac{Q}{A \cdot \sqrt{2g \cdot (H)}}$ $C = 0.582$ Coefficient of discharge needed

Note: 7-foot Ring-Jet valve provides more efficient C value of 0.78, it should be capable of achieving design discharge

$\frac{H}{d} = 8.688$ Head to diameter ration

$\frac{D}{d} = 3.4$ Taiwater depth ratio from Fig 12, ASCE Journal of Hydraulics Division

$D := d \cdot 3.4$ $D = 23.8\,ft$ Ideal tailwater depth

$\frac{L}{d} = 10.8$ Length ratio from Fig 14, ASCE Journal of Hydraulics Division

$L := 10.8 \cdot d$ $L = 75.6\,ft$ Minium length of basin

$\frac{W}{d} = 2.5$ Width ratio from Fig 15, ASCE Journal of Hydraulics Division

$W := 2.5 \cdot d$ $W = 17.5\,ft$ Minimum width

$\frac{D_s}{d} = 2.8$ Sweepout depth ratio from Fig 13, ASCE Journal of Hydraulics Division

$D_s := 2.8 \cdot d$ $D_s = 19.6\,ft$ Downstream minimum sweepout depth

$A_o := (D - 0.125 \cdot d) \cdot W$ $A_o = 401.187\,ft^2$ Cross section area of flow at the end of the stilling basin

$V_o := \frac{Q}{A_o}$ $V_o = 3.49\,\frac{ft}{s}$ Velocity at exit of stilling basin

This fish lock approach channel fits these dimesnions and is oversized in width, depth and length.

ASCE Journal Reference

Beichley, G. L. and Peterka, A. J., 1961, "Hydraulic Design of Hollow-Jet Valve Stilling Basins," *Jour. of Hydr. Div.*, ASCE, No. HY5

CORPS OF ENGINEERS, U.S. ARMY | WALLA WALLA DISTRICT

DIVISION | BRANCH | SECTION

PROJECT DALLES EFL EMERGENCY AWS

SUBJECT THRUST RESTRAINT FIRST 90°

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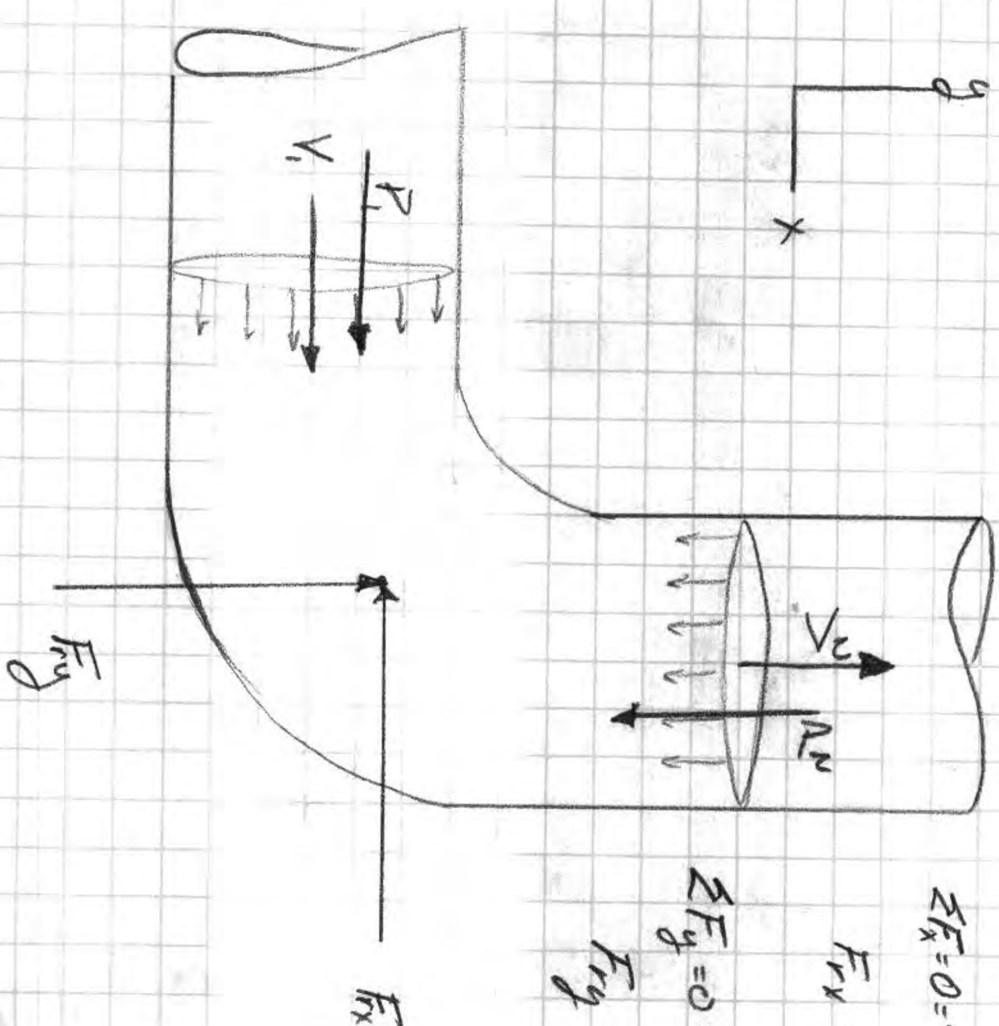

$$\Sigma F_x = 0 = -F_{rx} + P_1 + \cancel{P_2}^{0} - \rho Q(\cancel{V_{2x}}^{0} - V_{1x})$$

$$F_{rx} = P_1 + \rho Q V_{1x}$$

$$\Sigma F_y = 0 = F_{ry} + \cancel{P_1}^{0} - P_2 - \rho Q(V_{2y} - \cancel{V_{1y}}^{0})$$

$$F_{ry} = P_2 + \rho Q V_{2y}$$

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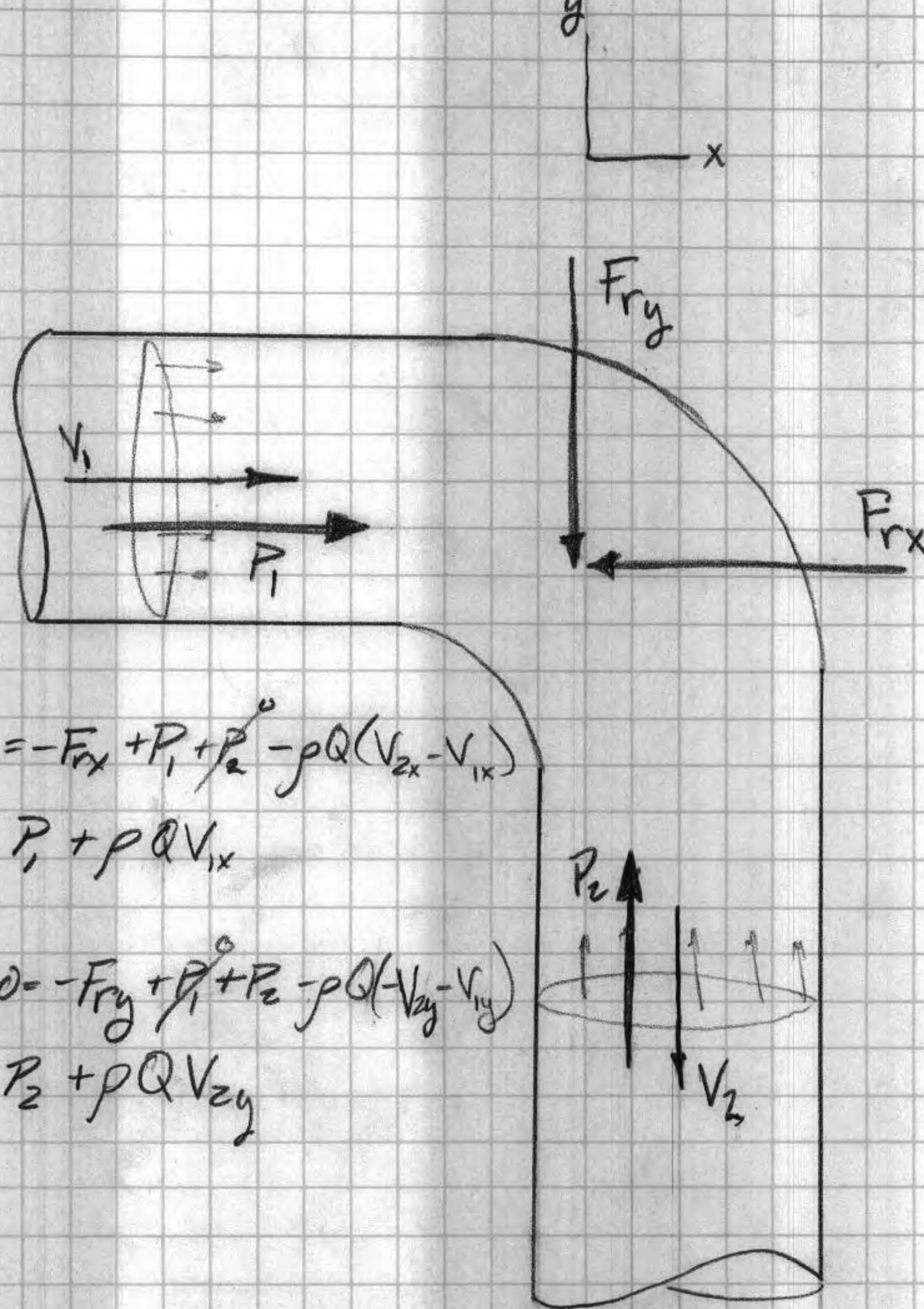
PROJECT DALLES EFL EMERGENCY AWS

SUBJECT THRUST RESTRAINT SECOND 90°

BY DATE CHECKED PART PAGE OF



$$\sum F_x = 0 = -F_{rx} + P_1 + \cancel{P_2}^{0} - \rho Q(V_{2x} - V_{1x})$$

$$F_{rx} = P_1 + \rho Q V_{1x}$$

$$\sum F_y = 0 = -F_{ry} + \cancel{P_1}^{0} + P_2 - \rho Q(-V_{2y} - V_{1y})$$

$$F_{ry} = P_2 + \rho Q V_{2y}$$

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DIVISION — BRANCH — SECTION

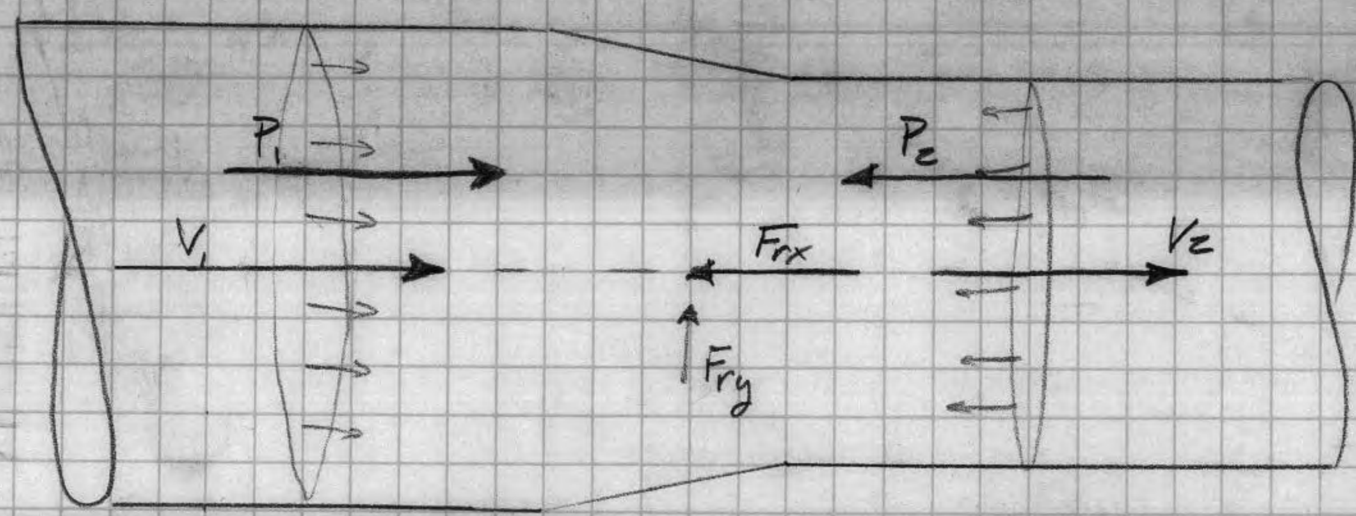
PROJECT: DALLES EFL EMERGENCY AWS

SUBJECT: THRUST RESTRAINT FOR 10'ϕ → 8'ϕ CONTRACTION

BY — DATE — CHECKED — PART — PAGE — OF

$$\Sigma F_x = 0 = -F_{rx} + P_1 - P_2 + \rho Q(V_{2x} - V_{1x})$$

$$F_{rx} = P_1 - P_2 + \rho Q(V_{2x} - V_{1x})$$



$$\Sigma F_y = 0 = F_{ry} + \cancel{P_1}^{0} + \cancel{P_2}^{0} - \rho Q(\cancel{V_{2y}}^{0} - \cancel{V_{1y}}^{0})$$

$$F_{ry} = 0$$

USE SAME FOR
8'ϕ → 7'ϕ CONT.

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DIVISION — BRANCH — SECTION

PROJECT: DALLES EFL EMERGENCY AWS

SUBJECT: THRUST RESTRAINT 40° BEND

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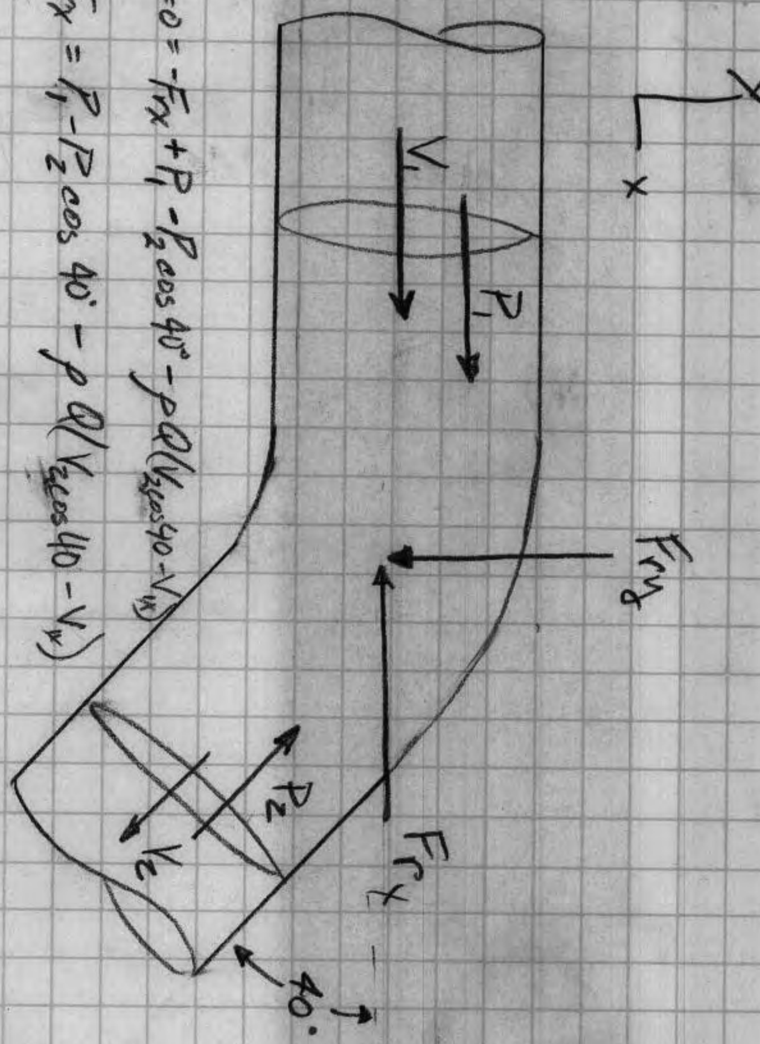

$$\Sigma F_x = 0 = -F_{rx} + P_1 - P_2\cos 40° - \rho Q(V_{2}\cos 40 - V_{1x})$$

$$F_{rx} = P_1 - P_2\cos 40° - \rho Q(V_{2}\cos 40 - V_{x})$$

$$\Sigma F_y = 0 = -F_{ry} + \cancel{P_1}^{0} + P_2\sin 40° - \rho Q(-V_2\sin 40° - \cancel{V_{1y}}^{0})$$

$$F_{ry} = P_2\sin 40° + \rho Q V_2 \sin 40°$$

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Custom Units Definition

$fps := ft \cdot s^{-1}$ feet per second

$cfs := ft^2 \cdot fps$ cubic feet per second

Fluid Properties

$\rho := 1000 \frac{kg}{m^3}$ $\gamma := 62.41 \frac{lbf}{ft^3}$

Assumed temperature deg. F

$T_f := 50$ $T_c := \left(T_f - 32\right) \cdot \frac{5}{9}$ $T_c = 10$ Temp. deg. C

$$\nu := \frac{1.792 \cdot 10^{-6}}{1.0 + \left(0.0337 \cdot T_c + 0.000221 \cdot T_c^{\,2}\right)} \cdot \frac{m^2}{s}$$

$\nu = 1.319 \times 10^{-6} \cdot \frac{m^2}{s}$ Kinematic viscosity of water from temp. relationship

Global Functions

Area function

$A(d) := \frac{\pi d^2}{4}$

Reynolds number

$Re(Q,d) := \frac{Q \cdot d}{A(d) \cdot \nu}$

Average velocity

$V(Q,d) := \frac{Q}{A(d)}$

Jain's equation for friction factor

$$f\left(Q,d,k_s\right) := \frac{0.25}{\log\left(\frac{k_s}{3.7 \cdot d} + \frac{5.74}{Re(Q,d)^{0.9}}\right)^2}$$

Ref: Swamee and Jain, 1976, "Explicit equations for pipe-flow problems," Journal of Hydr. Div. ASCE, Vol. 102, No. HY5, pp. 657-664

Design Parameters

$Q := 1400\text{cfs}$ Design flow rate

	Diameter	Length	Roughness	
Pipe 1	$D_1 := 10\text{ft}$	$L_1 := 225\text{ft}$	$k := 0.0002\text{ft}$	Through new pipe
			$k_{sr} := 0.001\text{ft}$	Fully rough
Pipe 2 - contraction	$D_2 := 8\text{ft}$	$L_2 := 50\text{ft}$		
			$k_{ss} := 0.00015\text{ft}$	Fully smooth
Final diameter before valve	$D_3 := 7\text{ft}$			

Pipe 1 Losses

Trash rack - See Trashrack Calculations.xcmd

Entrance loss

Isolation BV

90 deg bend loss 1 & 2

Contraction

Minor bend - 40 deg

Contraction

Friction Losses

$\text{Re}(Q, D_1) = 1.256 \times 10^7$ Reynolds number

Trash rack loss from other worksheet $K_t := 0.82$ ---> $h_t := 0.319\text{ft}$

Entrance loss $K_e := 0.5$ Assumed loss till further inlet works design is complete

90 deg bend loss

90 deg bend made up of three 30 deg mitered bends

$\frac{r}{d} = 2$

$k'_b := 0.275$ From Miller Fig. 9.10

$C_{Re} := 1.0$ From Miller Fig. 9.3

$C_o := 1.0$ No outlet, Miller Fig. 9.4

$C_f := \frac{f(Q, D_1, k_{sr})}{f(Q, D_1, k_{ss})}$ $C_f = 1.313$ From Miller Eq. 9.3

$K_{b90} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{b90} = 0.361$ From Miller Eq. 9.4

Contraction loss

Length of contraction over contracted radius

$A(D_1) = 78.54\,ft^2$

$A(D_2) = 50.265\,ft^2$ $\frac{A(D_1)}{A(D_2)} = 1.563$ $\frac{N}{R} = 2$ $N = 8ft$ $K_{c1} := 0.05$ From Miller Fig. 14.14(1)

$A(D_3) = 38.485\,ft^2$ $\frac{A(D_2)}{A(D_3)} = 1.306$ $\frac{N}{R} = 1.15$ $N = 4ft$ $K_{c2} := 0.035$ From Miller Fig. 14.14(1)

40 deg bend loss

triple-mitered bend

$\frac{r}{d} = 2$

$k'_b := 0.22$ From Miller Fig. 9.9, conservatively based on single miter

$C_{Re} := 1.0$ From Miller Fig. 9.3

$C_o := 0.5$ No outlet, Miller Fig. 9.4

$C_f := \frac{f(Q, D_2, k_{sr})}{f(Q, D_2, k_{ss})}$ $C_f = 1.344$ From Miller Eq. 9.3

$K_{b40} := k'_b \cdot C_{Re} \cdot C_o \cdot C_f$ $K_{b40} = 0.148$ Miller Eq. 9.4

Isolation Butterfly Valve Loss

$K_v := 0.2$ From Miller Fig. 14.19

Friction loss

$f_1 := f(Q, D_1, k)$ $f_1 = 9.571 \times 10^{-3}$

$f_2 := f(Q, D_2, k)$ $f_2 = 9.757 \times 10^{-3}$

Total losses

Velocity head $H_{v1} := \frac{V(Q, D_1)^2}{2g}$ $H_{v1} = 4.938\ ft$ $V(Q, D_1) = 17.825\ \frac{ft}{s}$

$H_{v2} := \frac{V(Q, D_2)^2}{2g}$ $H_{v2} = 12.055\ ft$ $V(Q, D_2) = 27.852\ \frac{ft}{s}$

$H_{v3} := \frac{V(Q, D_3)^2}{2g}$ $H_{v3} = 20.566\ ft$ $V(Q, D_3) = 36.378\ \frac{ft}{s}$

$$H_{p1} := \left(f_1 \cdot \frac{L_1}{D_1} + K_e + K_v + 2 \cdot K_{b90}\right) H_{v1}$$

$H_{p1} = 8.087\ ft$ Head loss through 10-ft diameter conduit

$$H_{p2} := \left(f_2 \cdot \frac{L_2}{D_2} + K_{c1} + K_{b40}\right) H_{v2}$$

$H_{p2} = 3.12\,ft$ Head loss through 8-ft diameter conduit

$H_3 := K_{c2} \cdot H_{v3}$

$H_3 = 0.72\,ft$ Head loss through final contraction

Maximum Operating Forebay

$R_h := 160ft$ $FL_h := R_h - h_t - H_{p1} - H_{p2} - H_3$ $FL_h = 147.754\,ft$

Miniumum Operating Forebay

$R_l := 155ft$ $FL_l := R_l - h_t - H_{p1} - H_{p2} - H_3$ $FL_l = 142.754\,ft$

$WSE_{fl} := 102.5ft$ Water surface in FLAC

$H_{vH} := FL_h - WSE_{fl}$ $H_{vH} = 45.254\,ft$ Energy to dissipate at high pool

$H_{vL} := FL_l - WSE_{fl}$ $H_{vL} = 40.254\,ft$ Energy to dissipate at low pool

Thrust force calculations for new bypass - First 90 degree bend

$V(Q, D_1) = 17.825\ \frac{ft}{s}$ Thrust velocity

$V_1 := V(Q, D_1)$ $V_{1x} := V_1 \cdot \cos(0)$ $V_{1x} = 17.825\ \frac{ft}{s}$

$V_{1y} := V_1 \cdot \sin(0)$ $V_{1y} = 0\ \frac{ft}{s}$

$V_2 := V(Q, D_1)$ $V_{2x} := V_2 \cdot \cos\left(\frac{\pi}{2}\right)$ $V_{2x} = 1.091 \times 10^{-15}\ \frac{ft}{s}$

$V_{2y} := V_2 \cdot \sin\left(\frac{\pi}{2}\right)$ $V_{2y} = 17.825\ \frac{ft}{s}$

$z := 104.5ft$ Approximated center of pipe elevation

$R_1 := 160ft$

$$H_z := R_1 - h_t - \left(f_1 \cdot \frac{50ft}{D_1} + K_e + K_v\right) H_{v1}$$

$H_z = 155.988\ ft$ Resulting hydraulic grade with entrance and friction loss assumption

$p := (H_z - z) \cdot \rho \cdot g$ $p = 22.322\ psi$

$A_1 := A(D_1) = 78.54\ ft^2$

$A_2 := A(D_1) = 78.54\ ft^2$

Cavitation check

$h_u := p$ $h_v := 0.18psi + 1atm$

$$\sigma_b := \frac{h_u - h_v}{\gamma \cdot \frac{V(Q, D_1)^2}{2 \cdot g}}$$ $\sigma_b = 3.479$

Cavitation parameter is greater than incipient cavitation for r/d = 1

$\sigma_{bi} := 2.2$ Incipient cavitation parameter from Miller Fig 6.10 with r/d = 1

$p_1 := p$ $\quad p_{1x} := p_1 \cdot \cos(0)$ $\quad p_{1x} = 22.322\,\text{psi}$

$p_{1y} := p_1 \cdot \sin(0)$ $\quad p_{1y} = 0\,\text{psi}$

$p_2 := p - K_{b90} \cdot H_{v1} \cdot \rho \cdot g$ $\quad p_{2x} := p_2 \cdot \cos\left(-\frac{\pi}{2}\right)$ $\quad p_{2x} = 1.319 \times 10^{-15}\,\text{psi}$

$p_{2y} := p_2 \cdot \sin\left(\frac{-\pi}{2}\right)$ $\quad p_{2y} = -21.548\,\text{psi}$

$$0 = -F_{rx} + p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$$0 = F_{ry} + p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

Thrust Restraint Force

$$F_{rx} := p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$F_{rx} = 300.872 \cdot \text{kip}$ Force in the plane of the bend acting towards the dam

$$F_{ry} := -\left[p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)\right]$$

$F_{ry} = 292.128 \cdot \text{kip}$ Force in the plane of the bend acting in line with the downstream flow $\quad \sqrt{F_{rx}^{\;2} + F_{ry}^{\;2}} = 419.36 \cdot \text{kip}$

Thrust force calculations for new bypass - Second 90 degree bend

$V(Q, D_1) = 17.825\,\frac{ft}{s}$ Thrust velocity

$V_1 := V(Q, D_1)$ $V_{1x} := V_1 \cdot \cos(0)$ $V_{1x} = 17.825\,\frac{ft}{s}$

$V_{1y} := V_1 \cdot \sin(0)$ $V_{1y} = 0\,\frac{ft}{s}$

$V_2 := V(Q, D_1)$ $V_{2x} := V_2 \cdot \cos\left(\frac{-\pi}{2}\right)$ $V_{2x} = 1.091 \times 10^{-15}\,\frac{ft}{s}$

$V_{2y} := V_2 \cdot \sin\left(\frac{-\pi}{2}\right)$ $V_{2y} = -17.825\,\frac{ft}{s}$

$z := 104.5ft$ Approximated center of pipe elevation

$R_1 := 160ft$

$$H_z := R_1 - h_t - \left(f_1 \cdot \frac{220ft}{D_1} + K_e + K_v + K_{b90}\right) H_{v1}$$

$H_z = 153.401\,ft$ Resulting hydraulic grade with entrance and friction loss assumption

$p := (H_z - z) \cdot \rho \cdot g$ $p = 21.2\,psi$

$A_1 := A(D_1) = 78.54\,ft^2$

$A_2 := A(D_1) = 78.54\,ft^2$

$h_u := p$ $h_v := 0.18psi + 1atm$

$$\sigma_b := \frac{h_u - h_v}{\gamma \cdot \frac{V(Q, D_1)^2}{2 \cdot g}}$$ $\sigma_b = 2.955$

$\sigma_{bi} := 2.2$ Incipient cavitation parameter from Miller Fig 6.10 with r/d = 1

Cavitation parameter is greater than incipient choking for r/d = 1

$p_1 := p$ $\quad p_{1x} := p_1 \cdot \cos(0)$ $\quad p_{1x} = 21.2\,\text{psi}$

$p_{1y} := p_1 \cdot \sin(0)$ $\quad p_{1y} = 0\,\text{psi}$

$p_2 := p - K_{b90} \cdot H_{v1} \cdot \rho \cdot g$ $\quad p_{2x} := p_2 \cdot \cos\left(\frac{\pi}{2}\right)$ $\quad p_{2x} = 1.251 \times 10^{-15}\,\text{psi}$

$p_{2y} := p_2 \cdot \sin\left(\frac{\pi}{2}\right)$ $\quad p_{2y} = 20.427\,\text{psi}$

$$0 = -F_{rx} + p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot (V_{2x} - V_{1x})$$

$$0 = -F_{ry} + p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot (V_{2y} - V_{1y})$$

Thrust Restraint Force

$$F_{rx} := p_{1x} \cdot A_1 + p_{2x} \cdot A_2 + \rho \cdot Q \cdot (V_{2x} - V_{1x})$$

$F_{rx} = 191.345 \cdot \text{kip}$ Force in the plane of the bend acting away from the dam

$$F_{ry} := p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot (V_{2y} - V_{1y})$$

$F_{ry} = 279.443 \cdot \text{kip}$ Force in the plane of the bend acting against with the upstream flow $\quad \sqrt{F_{rx}^2 + F_{ry}^2} = 338.676 \cdot \text{kip}$

Thrust force calculations for contraction

$V(Q,D_1) = 17.825\,\frac{ft}{s}$ Thrust velocity

$V_1 := V(Q,D_1)$ $V_{1x} := V_1 \cdot \cos(0)$ $V_{1x} = 17.825\,\frac{ft}{s}$

$V_{1y} := V_1 \cdot \sin(0)$ $V_{1y} = 0\,\frac{ft}{s}$

$V_2 := V(Q,D_2)$ $V_{2x} := V_2 \cdot \cos(0)$ $V_{2x} = 27.852\,\frac{ft}{s}$

$V_{2y} := V_2 \cdot \sin(0)$ $V_{2y} = 0\,\frac{ft}{s}$

$z := 104.5\,ft$ Approximated center of pipe elevation

$R_1 := 160\,ft$

$$H_z := R_1 - h_t - \left(f_1 \cdot \frac{230\,ft}{D_1} + K_e + K_v + 2K_{b90}\right) H_{v1}$$

$H_z = 151.57\,ft$ Resulting hydraulic grade with entrance and friction loss assumption

$p := (H_z - z) \cdot \rho \cdot g$ $p = 20.406\,psi$

$A_1 := A(D_1) = 78.54\,ft^2$

$A_2 := A(D_2) = 50.265\,ft^2$

$p_1 := p$ $\qquad p_{1x} := p_1 \cdot \cos(0)$ $\qquad p_{1x} = 20.406\,\text{psi}$

$p_{1y} := p_1 \cdot \sin(0)$ $\qquad p_{1y} = 0\,\text{psi}$

$p_2 := p - K_{c1} \cdot H_{v2} \cdot \rho \cdot g$ $\qquad p_{2x} := p_2 \cdot \cos(\pi)$ $\qquad p_{2x} = -20.145\,\text{psi}$

$p_{2y} := p_2 \cdot \sin(\pi)$ $\qquad p_{2y} = 2.467 \times 10^{-15}\,\text{psi}$

$$0 = -F_{rx} + p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$$0 = F_{ry} + p_{1y} \cdot A_1 + p_{2y} \cdot A - \blacksquare + \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

Thrust Restraint Force

$$F_{rx} := p_{1x} \cdot A_1 + p_{2x} \cdot A_2 + \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$F_{rx} = 112.213 \cdot \text{kip}$ Force in the plane of the bend acting towards the dam

$$F_{ry} := p_{1y} \cdot A_1 + p_{2y} \cdot A_2 + \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

$F_{ry} = 1.786 \times 10^{-14} \cdot \text{kip}$ Force in the plane of the bend acting opposite the direction of flow $\qquad \sqrt{F_{rx}^{\,2} + F_{ry}^{\,2}} = 112.213 \cdot \text{kip}$

Thrust force calculations for 40 degree bend

$V_1 := V(Q, D_2)$ $\quad V_{1x} := V_1 \cdot \cos(0)$ $\quad V_{1x} = 27.852 \frac{ft}{s}$

$V_{1y} := V_1 \cdot \sin(0)$ $\quad V_{1y} = 0 \frac{ft}{s}$

$V_2 := V(Q, D_2)$ $\quad V_{2x} := V_2 \cdot \cos(-40deg)$ $\quad V_{2x} = 21.336 \frac{ft}{s}$

$V_{2y} := V_2 \cdot \sin(-40deg)$ $\quad V_{2y} = -17.903 \frac{ft}{s}$

$z := 104.5ft$ Approximated center of pipe elevation

$R_1 := 160ft$

$$H_z := R_1 - h_t - \left(f_1 \cdot \frac{230ft}{D_1} + K_e + K_v + 2K_{b90} + K_{c1}\right) H_{v1} - f_2 \cdot 20 \frac{ft}{D_2} \cdot H_{v2}$$

$H_z = 151.029\ ft$ Resulting hydraulic grade with entrance and friction loss assumption

$p := (H_z - z) \cdot \rho \cdot g$ $\quad p = 20.172\ psi$

$A_1 := A(D_2) = 50.265\ ft^2$

$A_2 := A(D_2) = 50.265\ ft^2$

$h_u := p$ $\quad h_v := 0.18psi + 1atm$

$$\sigma_b := \frac{h_u - h_v}{\gamma \cdot \frac{V(Q, D_2)^2}{2 \cdot g}}$$

$\sigma_b = 1.014$

$\sigma_{bi} := 0.75$ Incipient cavitation parameter from Miller Fig 6.10 with r/d = 2

Cavitation parameter is greater than incipient cavitation for r/d = 2 with a 8 ft conduit -> ok

$p_1 := p$ $\quad p_{1x} := p_1 \cdot \cos(0)$ $\quad p_{1x} = 20.172\,\text{psi}$

$p_{1y} := p_1 \cdot \sin(0)$ $\quad p_{1y} = 0\,\text{psi}$

$p_2 := p - K_{b40} \cdot H_{v2} \cdot \rho \cdot g$ $\quad p_{2x} := p_2 \cdot \cos(140\text{deg})$ $\quad p_{2x} = -14.861\,\text{psi}$

$p_{2y} := p_2 \cdot \sin(140\text{deg})$ $\quad p_{2y} = 12.47\,\text{psi}$

$$0 = -F_{rx} + p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$$0 = -F_{ry} + p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

Thrust Restraint Force

$$F_{rx} := \left[p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)\right]$$

$F_{rx} = 56.145 \cdot \text{kip}$ Force in the plane of the bend acting towards the dam

$$F_{ry} := p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

$F_{ry} = 138.89 \cdot \text{kip}$ Force in the plane of the bend acting perpendicular with the upstream flow $\sqrt{F_{rx}^{\ 2} + F_{ry}^{\ 2}} = 149.808 \cdot \text{kip}$

Thrust force calculations for contraction

$V(Q, D_2) = 27.852 \frac{ft}{s}$ Thrust velocity

$V_1 := V(Q, D_2)$ $V_{1x} := V_1 \cdot \cos(0)$ $V_{1x} = 27.852 \frac{ft}{s}$

$V_{1y} := V_1 \cdot \sin(0)$ $V_{1y} = 0 \frac{ft}{s}$

$V_2 := V(Q, D_3)$ $V_{2x} := V_2 \cdot \cos(0)$ $V_{2x} = 36.378 \frac{ft}{s}$

$V_{2y} := V_2 \cdot \sin(0)$ $V_{2y} = 0 \frac{ft}{s}$

$z := 104.5 ft$ Approximated center of pipe elevation

$R_1 := 160 ft$

$$H_z := R_1 - h_t - \left(f_1 \cdot \frac{L_1}{D_1} + K_e + K_v + 2K_{b90} + K_{c1}\right) H_{v1} - \left(f_2 \cdot \frac{L_2}{D_2} + K_{b40}\right) \cdot H_{v2}$$

$H_z = 148.83 ft$ Resulting hydraulic grade with entrance and friction loss assumption

$p := (H_z - z) \cdot \rho \cdot g$ $p = 19.218 psi$

$A_1 := A(D_2) = 50.265 ft^2$

$A_2 := A(D_3) = 38.485 ft^2$

$p_1 := p$ $\quad$ $p_{1x} := p_1 \cdot \cos(0)$ $\quad$ $p_{1x} = 19.218\,\text{psi}$

$p_{1y} := p_1 \cdot \sin(0)$ $\quad$ $p_{1y} = 0\,\text{psi}$

$p_2 := p - K_{c2} \cdot H_{v1} \cdot \rho \cdot g$ $\quad$ $p_{2x} := p_2 \cdot \cos(\pi)$ $\quad$ $p_{2x} = -19.143\,\text{psi}$

$p_{2y} := p_2 \cdot \sin(\pi)$ $\quad$ $p_{2y} = 2.344 \times 10^{-15}\,\text{psi}$

$$0 = -F_{rx} + p_{1x} \cdot A_1 + p_{2x} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$$0 = F_{ry} + p_{1y} \cdot A_1 + p_{2y} \cdot A_2 - \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

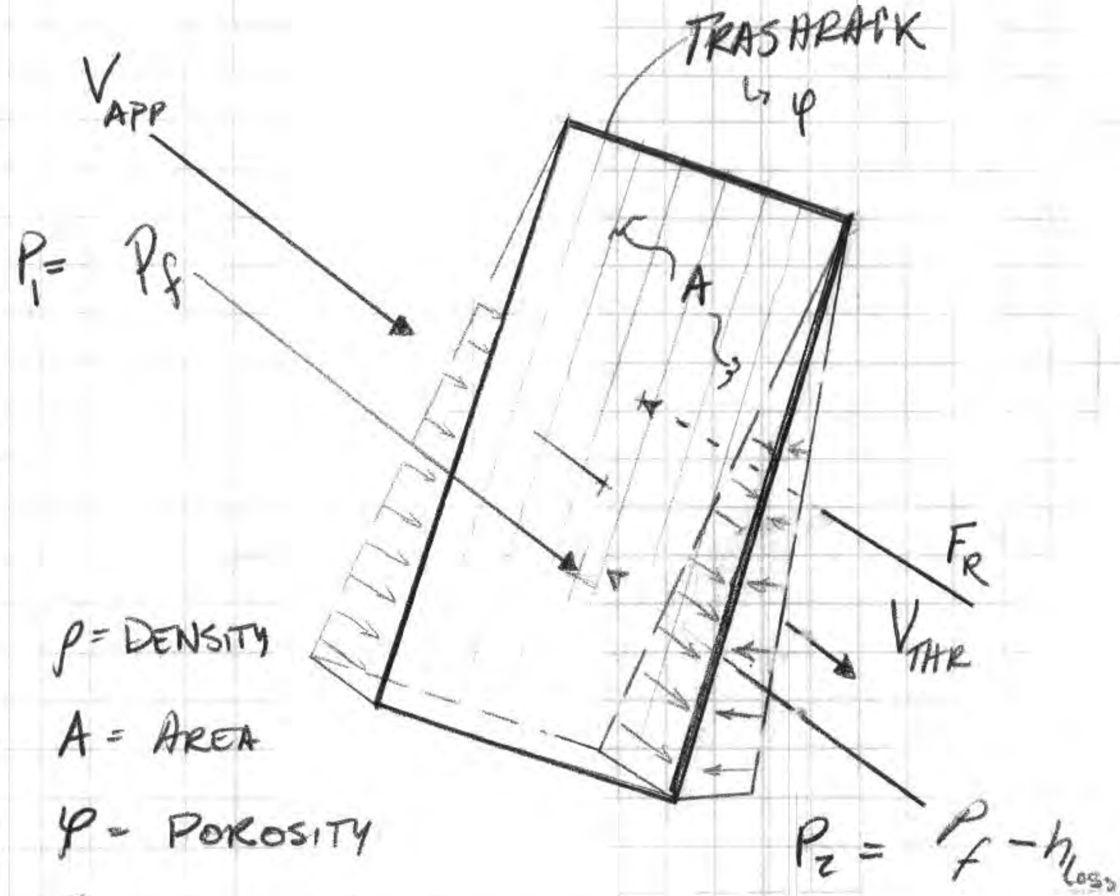
Thrust Restraint Force

$$F_{rx} := p_{1x} \cdot A_1 + p_{2x} \cdot A_2 + \rho \cdot Q \cdot \left(V_{2x} - V_{1x}\right)$$

$F_{rx} = 56.179 \cdot \text{kip}$ Force in the plane of the bend acting towards the dam

$$F_{ry} := p_{1y} \cdot A_1 + p_{2y} \cdot A_2 + \rho \cdot Q \cdot \left(V_{2y} - V_{1y}\right)$$

$F_{ry} = 1.299 \times 10^{-14} \cdot \text{kip}$ Force in the plane of the bend acting perpendicular with the upstream flow $\sqrt{F_{rx}^{\;2} + F_{ry}^{\;2}} = 56.179 \cdot \text{kip}$



ρ = DENSITY

A = AREA

φ - POROSITY

V_{APP} = APPROACH VELOCITY

V_{THR} = THRU VELOCITY

P_f = STATIC PRESSURE FORCE

$P_f - h_L$ = STATIC PRESSURE MINUS HEAD DIFFERENTIAL

F_R = RESULTANT FORCE

$$F_R = (P_1 - P_2) \cdot A(1-\varphi) + \rho \cdot Q(V_2 - V_1)$$

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Translucent Grid

Units definition

$cfs := ft^3 \cdot s^{-1}$ cubic feet per second

$fps := ft \cdot s^{-1}$ feet per second

Hydraulic Properties

$\rho := 1000 \frac{kg}{m^3}$ Fluid density

Assumed temperature deg. F

$T_f := 50$ $T_c := (T_f - 32) \cdot \frac{5}{9}$ $T_c = 10$ Temp. deg. C

$\nu := \frac{1.792 \cdot 10^{-6}}{1.0 + \left(0.0337 \cdot T_c + 0.000221 \cdot T_c^{\,2}\right)} \cdot \frac{m^2}{s}$ $\nu = 1.319 \times 10^{-6} \cdot \frac{m^2}{s}$ Kinematic viscosity of water from temp. relationship

Design Parameters

$Q := 1400cfs$ Design flow rate

$V := 3fps$ Velocity limitation for trashrack approach velocity - EM 1110-2-1602

$V_{thr} := 5fps$ Recommended thru velocity maximum for cleaning accessible trashracks from Bureau of Reclamation - Design of Small Dams

$A_{req} := \frac{Q}{V}$ $A_{req} = 466.667\ ft^2$ Area required to meet trashrack approach velocity limitation

$h_t = K_t \cdot \frac{v_n^2}{2 \cdot g}$ $K_t = 1.45 - 0.45 \cdot \frac{a_n}{a_g} - \left(\frac{a_n}{a_g}\right)^2$ Equation 11, Design of Small Dams - BoR

$a_n := 0.75in$ Design bar spacing per EDR

$a_g := \frac{3}{16} in + a_n$ Assummed unit thickness for bar and space

$\frac{a_n}{a_g} = 0.8$ Resultant porosity

$K_t := 1.45 - 0.45 \cdot \frac{a_n}{a_g} - \left(\frac{a_n}{a_g}\right)^2$ $K_t = 0.45$ Resultant loss coefficient

$v_n := V_{thr}$ Thru velocity for head loss differential

$h_t := K_t \cdot \frac{v_n^2}{2 \cdot g}$ $h_t = 0.175\ ft$ Resultant head differential

$A_{req} := \frac{Q}{v_n} \cdot \frac{a_g}{a_n}$ $A_{req} = 350\ ft^2$ Based on thru velocity limitations $\boxed{A_{req} := 466ft^2}$ Area required based on approach velocity limitations - Controlling

Required trashrack height based on 15 foot width

$H := \frac{A_{req}}{15ft}$ $H = 31.067\ ft$

Required trashrack height based on 20 foot width

$H := \frac{A_{req}}{20ft}$ $H = 23.3\ ft$

Trashracks for the intake are sized with a 3 fps approach velocity and a flow of 1400 cfs. Velocity criterion was determined during the EDR phase of design and based off of EM 1110-2-1602. A through bar velocity of 5 fps is recommended by the Bureau of Reclamation *Design of Small Dams* publication. An assumed porosity of 80 percent for the trashrack results in a required gross area of 350 square feet; however, in order to meet the approach velocity a required gross area of trashrack is required to be 466 square feet.

$A_{req} = 466\,ft^2$

$R_h := 160ft$ $R_l := 155ft$ $CL := 116.5ft$ $p_t := h_t \cdot \rho \cdot g$ $p_t = 0.076\,psi$

$P_1 := (R_h - CL) \cdot g \cdot \rho$ $P_1 = 18.858\,psi$ $p_1 := R_h - CL$

$P_2 := P_1 - h_t \cdot g \cdot \rho$ $P_2 = 18.783\,psi$ $p_2 := p_1 - h_t$

$V_1 := V = 3\,\frac{ft}{s}$ $V_2 := V_{thr} = 5\,\frac{ft}{s}$

$F_r := (P_1 - P_2) \cdot A_{req} \cdot \left(1 - \frac{a_n}{a_g}\right) + \rho \cdot Q \cdot (V_2 - V_1)$ Equation for force imparted by momentum and pressure differential

$F_r = 6.45 \cdot kip$ Resultant force from momentum and pressure differential

$\frac{F_r}{A_{req}} = 13.841 \cdot psf$ Resultant pressure resistance from momentum and pressure differential

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HYDRAULIC DESIGN OF HOLLOW-JET VALVE STILLING BASINS

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SYNOPSIS

Hydraulic model and prototype tests made to generalize and prove the hydraulic design of a new type of stilling basin which utilizes the hollow-jet valve for discharge control are described. Dimensionless curves are derived from model data and are used to define the important dimensions of the basin for the usual combinations of valve size, operating head, and discharge. Sample problems are presented to illustrate the use of the design curves and the general hydraulic design procedures. Prototype tests on the Boysen and Falcon Dam stilling basins are described and analyzed to help establish the reliability of the recommended basins. Basin dimensions obtained from individual model tests on six stilling basins are shown to compare favorably with the dimensions obtained from the dimensionless curves and methods given in this paper.

INTRODUCTION

The hollow-jet valve stilling basin described in this paper is of a new type and is used to dissipate hydraulic energy at the downstream end of an outlet works control structure. To reduce cost and save space, the stilling basin is usually constructed within or adjacent to the powerhouse structure as shown in Figs. 1 and 2. The hollow-jet valve, Fig. 3, controls and regulates the flow.

Note.—Discussion open until February 1, 1962. To extend the closing date one month, a written request must be filed with the Executive Secretary, ASCE. This paper is part of the copyrighted Journal of the Hydraulics Division, Proceedings of the American Society of Civil Engineers, Vol. 87, No. HY 5, September, 1961.

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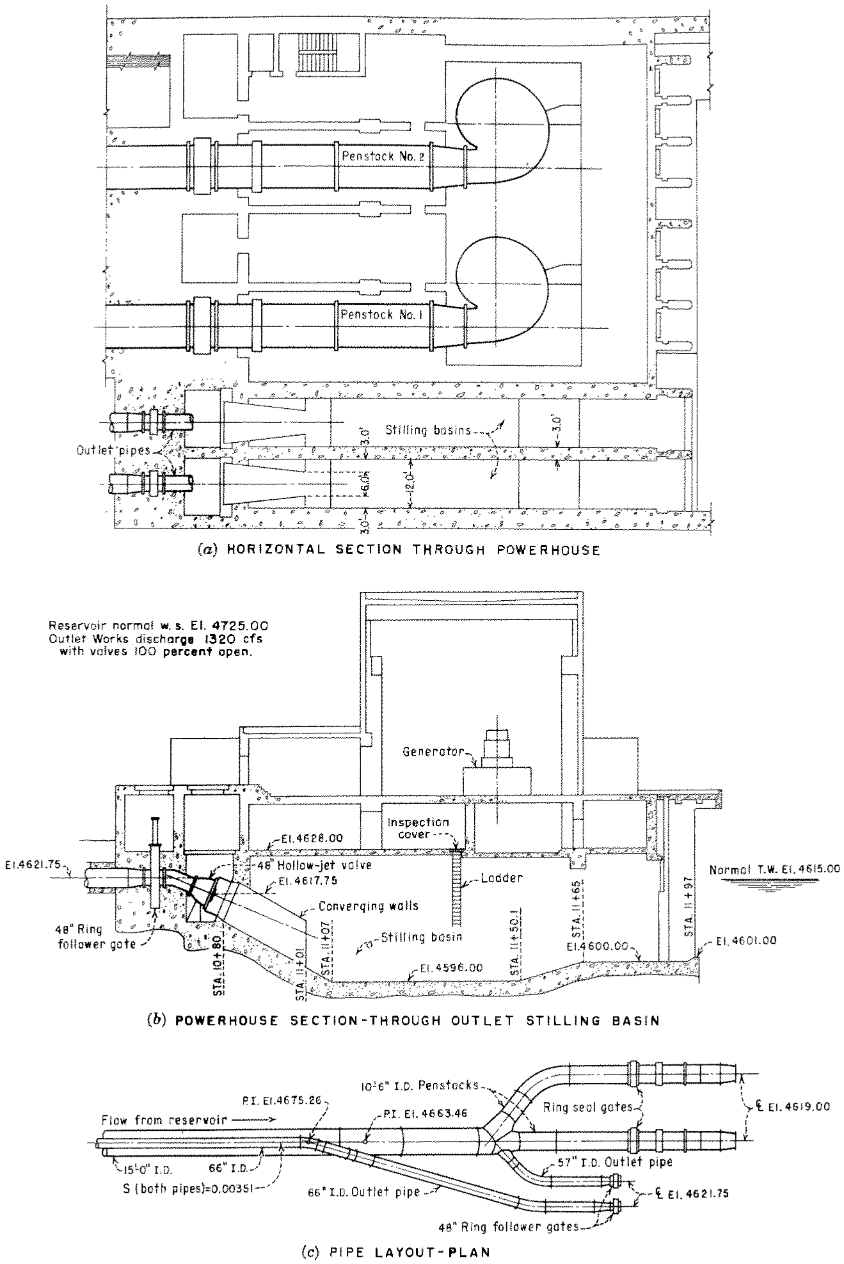


FIG. 1.—BOYSEN DAM OUTLET WORKS STILLING BASIN AND POWER PLANT

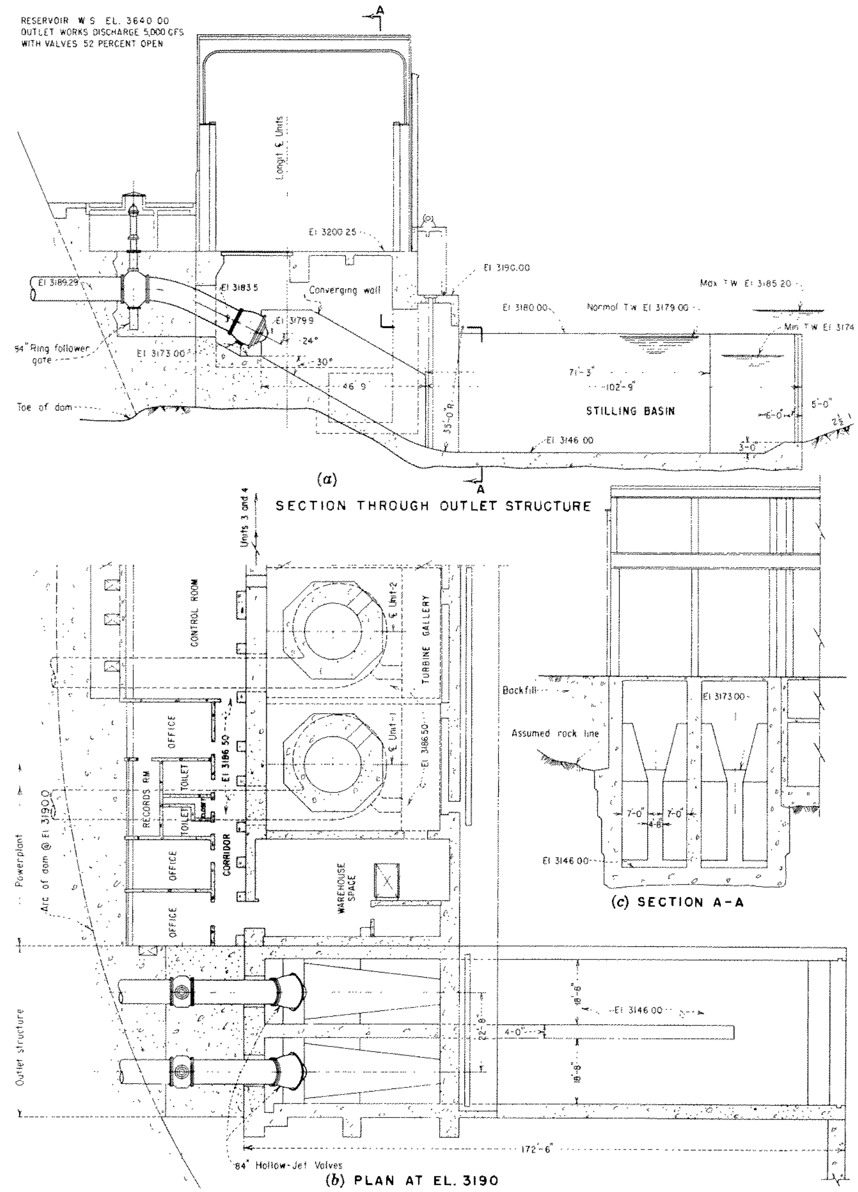


FIG. 2.—YELLOWTAIL DAM PROPOSED OUTLET WORKS STILLING BASIN AND POWERPLANT

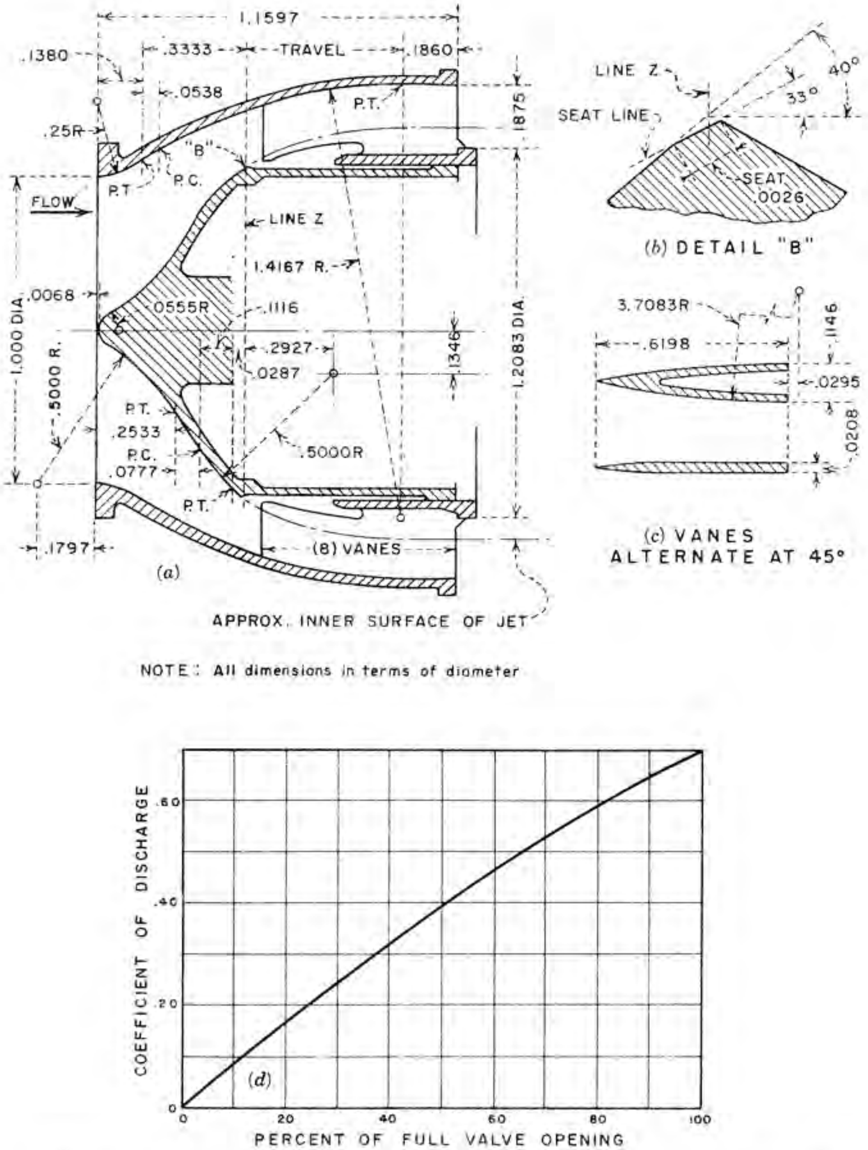


FIG. 3.—HOLLOW-JET VALVE DIMENSIONS AND DISCHARGE COEFFICIENTS

Regardless of the valve opening or head, the outflow has the same pattern, an annular or hollow jet of water of practically uniform diameter throughout its length, Fig. 4. The stilling basin is designed to take advantage of the hollow-jet shape; solid jets cannot be used in this basin.

The hollow-jet valve was developed by the Bureau of Reclamation in the early 1940's to fill a need for a dependable regulating valve. The design was accomplished with the aid of a complete 6-in.-diameter hydraulic model and a sectional 12-in.-diameter air model. These models were tested in the Bureau of Reclamation Hydraulic Laboratory. To evaluate the valve characteristics at greater than scale heads, a 24-in.-diameter valve was tested at Hoover Dam under heads ranging from 197 ft to 349 ft.

(a) Valve fully open

(b) Valve 50 percent open

FIG. 4.—SIX-INCH HOLLOW-JET VALVE DISCHARGING

Piezometer pressure measurements, thrust determinations on the valve needle, and rates of discharge were studied in both field and laboratory tests. It was found that the hydraulic characteristics of the larger valves could be predicted from the performance of the smaller model valves. From these tests and investigations of prototype valves up to 96 in. in diameter, the valve has been proved to be a satisfactory control device.

Cavitation damage, found on a few of the many prototype valves in use, was minor in nature and was caused by local irregularities in the body casting and by misalinement of the valve with the pipe. These difficulties have been eliminated by careful foundry and installation practices. On one installation, damage that occurred on the cast iron valve support vanes may have been caused by abrasive sediment in the water. The design itself is cavitation free.

Because a large valve operating at high heads can discharge flows having an energy content of up to 150,000 hp, a stilling basin is usually required downstream from the valve. In early designs, the valve was discharged horizontally onto a trajectory curved floor which was sufficiently long to provide a uniformly distributed jet entering the hydraulic jump stilling pool. This resulted in an extremely long structure, twice or more the length of the basin recommended herein. When two valves were used side by side, a long, costly dividing wall was also required. Hydraulic model tests showed that the basin length could be reduced more than 50% by turning the hollow-jet valves downward and using a different energy dissipating principle in the stilling basin. The first stilling basin of this type was developed for use at Boysen Dam, a relatively low-head structure. Basins for larger discharges and higher heads were later developed from individual hydraulic models of the outlet works at Falcon, Yellowtail, Trinity, and Navajo Dams. It became apparent at this time that generalized design curves could be determined to cover a wide range of operating heads and discharges. Therefore, a testing program was initiated to provide the necessary data. A brief description of the individual model tests made to develop the basin type is given in the following section. Table 1 gives a summary of basin dimensions, valve sizes, test heads, and discharges for these structures.

DEVELOPMENT OF BASIN FEATURES

Boysen Dam.—In the Boysen Dam model studies, a series of basic tests was made to determine the optimum angle of entry of a hollow-jet into the tail water. For flat angles of entry, the jet did not penetrate the pool but skipped along the tail water surface. For steep angles, the jet penetrated the pool but rose almost vertically to form an objectionable boil on the water surface. When the valves were depressed 24° from the horizontal, Fig. 1, and a 30° sloping floor was placed downstream from the valve to protect the underside of the jet from turbulent eddies, optimum performance resulted. The submerged path of the valve jet was then sufficiently long that only a minimum boil rose to the surface. The size and intensity of the boil were further reduced when converging walls were placed on the 30° sloping floor to protect the sides of the jet until it was fully submerged. The converging walls have another function, however; they compress the hollow-jet between them to give the resulting thin jet greater ability to penetrate the tail water pool. Sudden expansion of the jet as it leaves the converging walls plus the creation of fine grain turbulence in the basin account for most of the energy losses in the flow. Thorough breaking-up of the valve jet within the basin and good velocity distribution over the entire flow cross section account for the low velocities leaving the basin. Fig. 5 shows the performance of a hollow-jet basin both with and without the converging walls.

Pressures on the inside face and downstream end of the converging walls were measured to determine whether low pressures which might induce cavitation were present. The lowest pressure, measured on the end of the wall,

was 3 ft of water above atmospheric; therefore, cavitation should not occur. Pressures measured on the sloping floor, and under and near the impinging jet, were all above atmospheric. Maximum pressures did not exceed one-fourth of the total head at the valve.

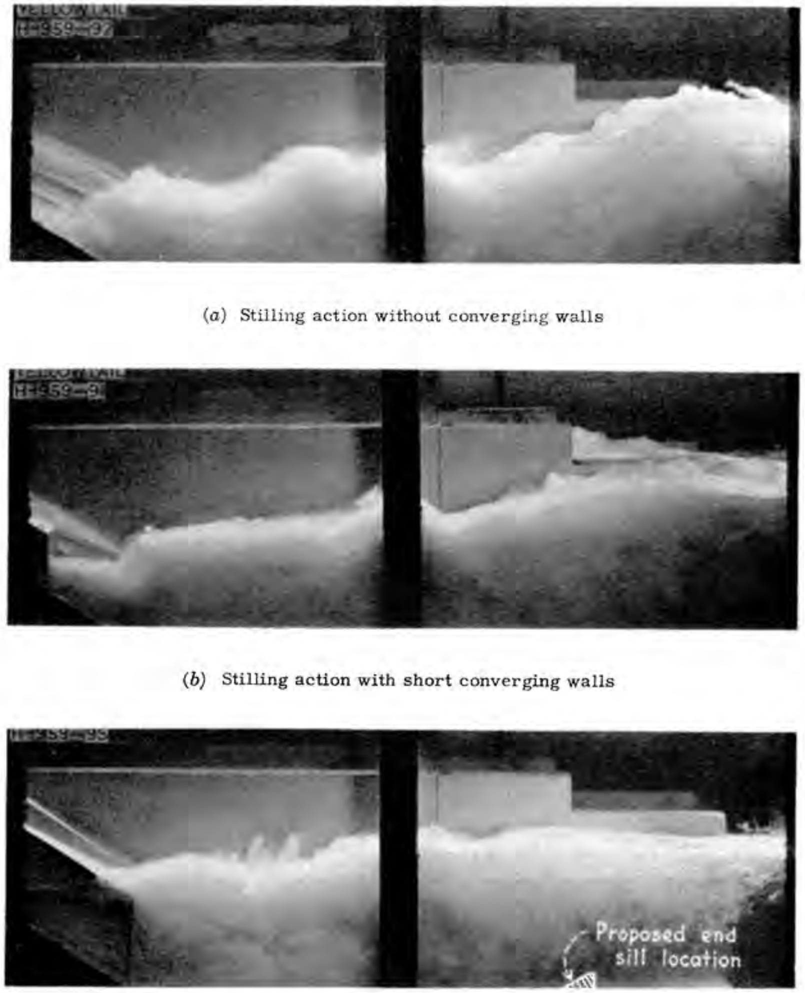
TABLE 1.—COMPARISON OF BASIN DIMENSIONS[a, b, c]

Basin Dimensions (1)	Boysen (2)	Falcon, U. S. (3)	Falcon, Mexico (4)	Yellowtail (5)	Trinity (6)	Navajo (7)
Valve diameter, in ft	4	6	7.5	7	7	6
Head at valve, in ft	86	81.5	81.9	380	315	217
Design Q, in cfs	660	1,460	2,285	2,500	3,835	2,340
Coefficient C	0.70	0.70	0.70	0.41	0.70	0.70
Percentage valve open	100	100	100	52	100	100
Depth D, in ft	16.2 19	21.0 22.5	24.7 25.2	31.5 32.6	38.5 38	30.0 35[e]
Depth D_s, in ft	13.6 14	17.4 17.5	20.2 19.5	25.9 25.6	31.5 31.8	24.6 24
Length L, in ft	60.4 58	74.4 73.9	86.2 94	104 102.8	129 123	103 110[e]
Width W, in ft	10.2 12	14.7 16.2	18 16.2	19.2 18.7	19.6 18.9	16.2 18.0[e]
End sill height	3 4	3 3	3.1 3	3.9 3	4.8 5	. . .[e] . . .[e]
End sill slope	3.3:1[d]	2:1	2:1	2:1	2:1	. . .[e]
Converg wall height	3.0 d	4.5 d	3.9 d	3.1 d	3.5 d	3.4 d
Converg wall gap	0.50 W	0.52 W	0.65 W	0.25 W	0.25 W	0.23 W
Center wall length	1.5 L[d]	0.5 L	0.4 L	0.7 L	0.3 L	0.5 L
Channel slope	. . .[d]	4:1	4:1	2.5:1	2:1	6:1[e]

[a] Upper values in each box were calculated from Figs. 11 through 15; lower values in each box were developed from individual model studies.
[b] Valve tilt 24°; inclined floor 30° in all cases.
[c] See Figs. 1, 2, 6, 7, 8, 9, and 11.
[d] Special case, for structural reasons.
[e] Special case, for diversion flow requirements (dentated sill used and basin size increased).

Scour downstream from the end sill was mild and prototype wave heights were only 0.5 ft in the river channel. A vertical traverse taken near the end sill showed surface velocities to be about 5 fps, decreasing uniformly to about 2 fps near the floor.

Falcon Dam.—In the Falcon Dam tests, two separate basins were developed, one for the United States outlet works and one for the Mexican outlet works, Figs. 6 and 7. In these tests, the basic concepts of the Boysen design were proved to be satisfactory for greater discharges. In addition, it was confirmed



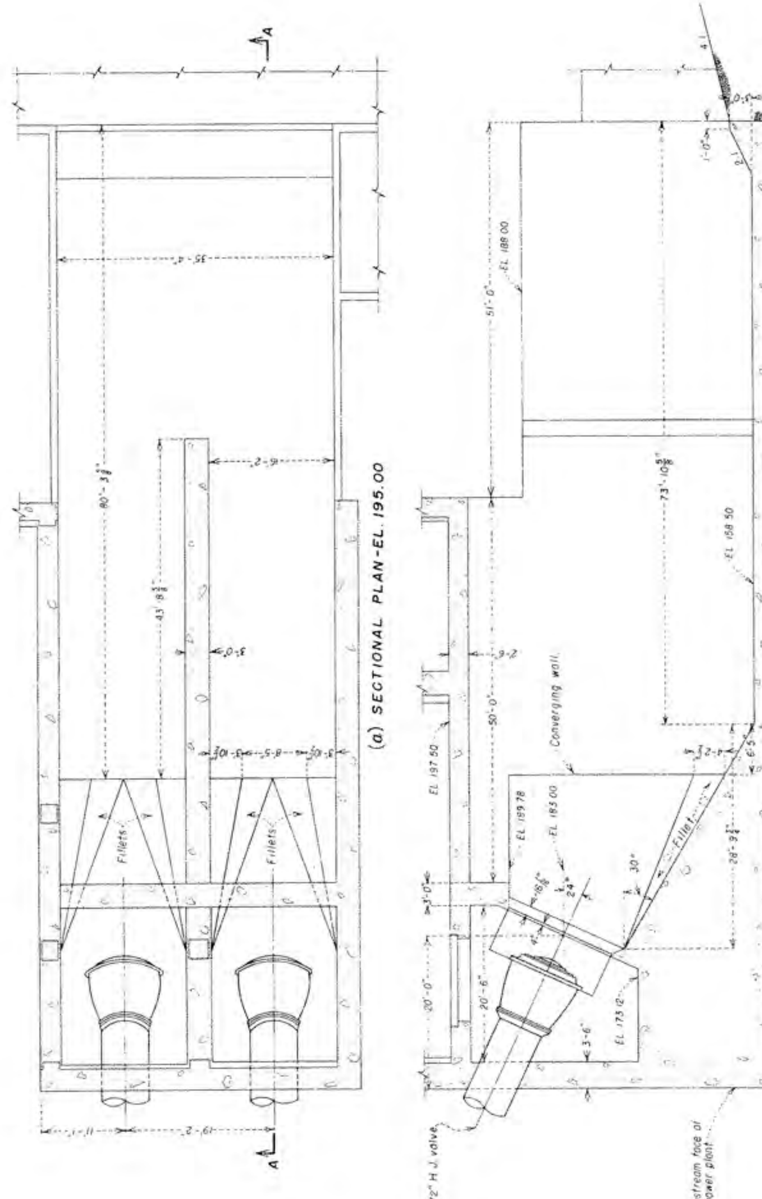
(*a*) Stilling action without converging walls

(*b*) Stilling action with short converging walls

(*c*) Stilling action with recommended converging walls

FIG. 5.—HOLLOW-JET VALVE STILLING BASIN WITH AND WITHOUT CONVERGING WALLS

that dentils on the end sill were not necessary and that the center dividing wall need not extend the full length of the basin. A low 2:1 sloping end sill was sufficient to provide minimum scour and wave heights. Maximum pressures on



FALCON DAM

FIG. 6.—UNITED STATES OUTLET WORKS

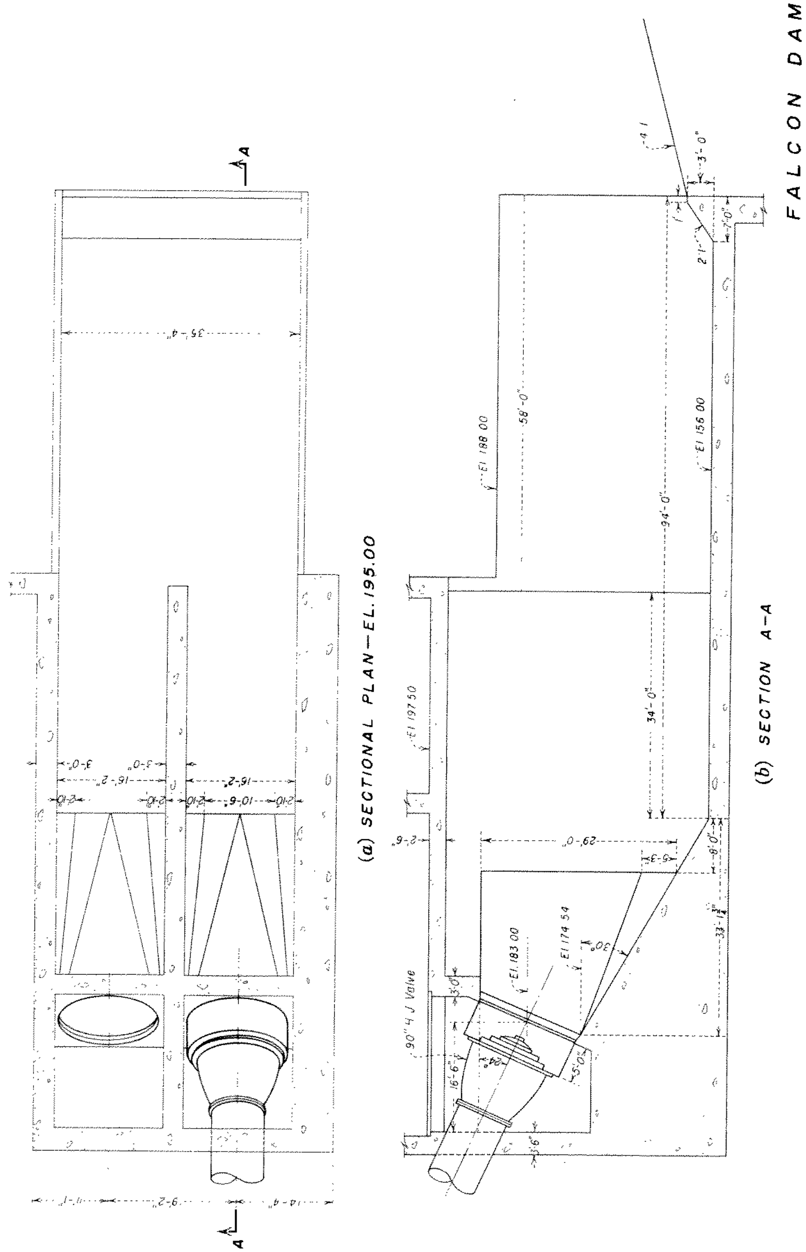


FIG. 7.—MEXICAN OUTLET WORKS

the floor beneath the impinging jet were found to be about one-third of the total head at the valve, somewhat greater than found in the Boysen tests, but still not excessive.

Yellowtail Dam.—In the Yellowtail Dam model studies, the head and discharge were both considerably higher than in the Boysen and Falcon tests. Because of the high velocity flow from the valves, it was found necessary to extend the converging walls to the downstream end of the sloping floor, Fig. 2, and to reduce the wall gap to about one-quarter of the basin width. These refinements improved the stilling action within the basin, Fig. 5 (c), and made it possible to further reduce the basin length. Scour was not excessive, and the water surface in the downstream channel was relatively smooth. Pressures on the converging walls and other critical areas in the basin were found to be above atmospheric.

Trinity Dam.—The Trinity Dam outlet works utilized a head almost 4 times greater and a discharge 5 times greater than at Boysen Dam. In the development tests, it was found that the performance of this type of basin would be satisfactory for extremely high heads and discharges. Although several variations in the basin arrangement were investigated, no new features were incorporated in the design. Fig. 8 shows the developed design.

Navajo Dam.—The experimental work on the Navajo outlet works was complicated by the fact that the hollow-jet valve basin, Fig. 9, had to first serve as a temporary diversion works stilling basin. Since the diversion works basin was larger than required for the outlet works basin, it was possible to insert the proper appurtenances in the temporary basin to convert it to a permanent outlet works basin. The development tests indicated that a larger than necessary basin does not in itself guarantee satisfactory performance of the hollow-jet valve basin. Best outlet works performance was obtained when the temporary basin was reduced in size to conform to the optimum size required for the permanent structure. Since the Navajo Dam outlet works model was available both during and after the generalization tests, the model was used both to aid in obtaining the generalized data and to prove that the design curves obtained were correct.

GENERALIZATION STUDY

Because development work on individual basins had reached a point where the general arrangement of the basin features was consistent, and because the basin had been proved satisfactory for a wide range of operating conditions, a testing program was inaugurated to provide data for use in generalizing the basin design. The purpose of these tests was to provide basin dimensions and hydraulic design procedures for any usual combinations of valve size, discharge, and operating head. The main purpose of this paper is to describe these tests, to explain the dimensionless curves which are derived from the test data, and to show, by means of sample problems, the procedures which may be used to hydraulically design a hollow-jet valve stilling basin. Prototype tests on the Boysen and Falcon basins are included to demonstrate that hollow-jet valve basins, that fit the dimensionless curves derived in the general study, will perform as well in the field as predicted from the model tests.

Test Equipment.—The outlet works stilling basin model shown in Fig. 10 was used for the generalization tests. The glass-walled testing flume contained two stilling basins separated by a dividing wall. The right-hand basin

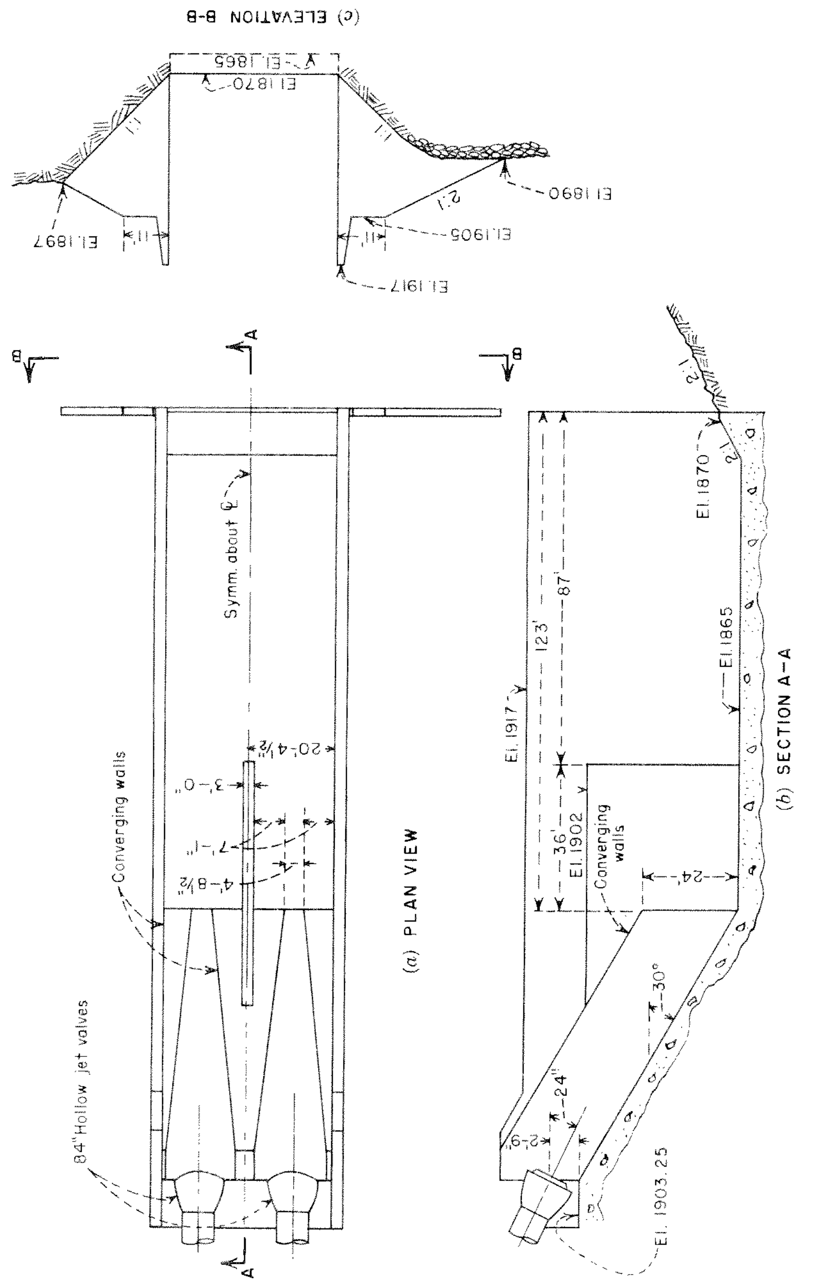


FIG. 8.—TRINITY DAM OUTLET WORKS

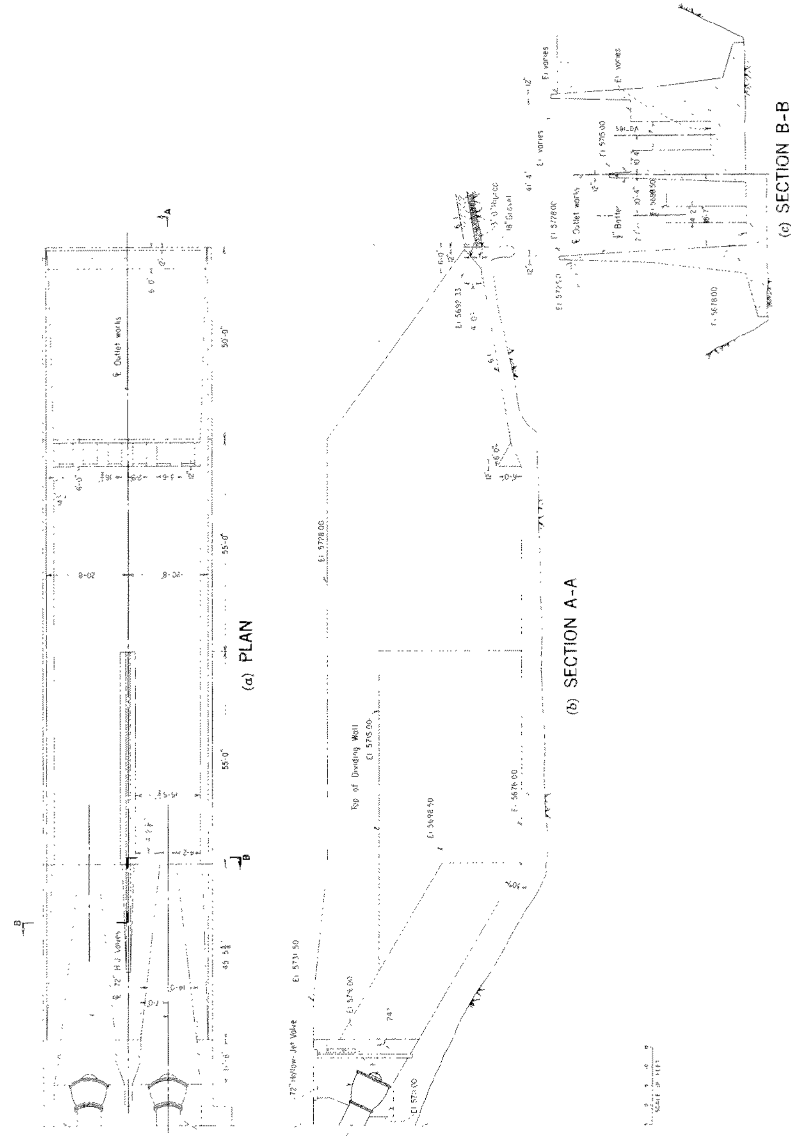


FIG. 9.—NAVAJO DAM OUTLET WORKS STILLING BASIN

having the glass panel as one wall was operated singly to determine the basin length, width, and depth requiremetns; both basins were used to study the performance with and without flow in an adjacent basin.

The glass panel permitted observation of the stilling action and the flow currents within and downstream from the basin. The length, width, and depth of the basin were varied by inserting false walls or by moving the basin within the test box. The tail box contained an erodible sand bed to represent the discharge channel bed.

The test valves were exact models of a prototype valve in that the flow surfaces were exactly reproduced, and could be opened and closed to any partial opening. The models were 3-in. valves machined from bronze castings.

The pressure head at each model valve was measured using a piezometer located in the 3-in. supply pipe 1 diameter upstream from the valve flange.

FIG. 10.—HOLLOW-JET VALVE STILLING BASIN MODEL USED FOR GENERALIZATION TESTS

Discharges were measured using calibrated venturi meters permanently installed in the laboratory. The tail water elevation in the discharge channel was controlled with a hinged tailgate in the tail box. Tail water elevations were determined visually from a staff gage on the tail box wall located approximately 62 valve diameters downstream from the valves.

Preliminary Procedures.—The investigation was begun by tabulating the important dimensions of the Boysen, Falcon, Yellowtail, and Trinity outlet works basins and expressing them in dimensionless form, as shown in Table 1. Based on these dimensions, a model was constructed as shown in Fig. 11, using the 3-in. valve dimension to establish the absolute model size. More weight was given to the Yellowtail and Trinity basins because they were developed for higher heads and contained refinements in the converging wall design which improved the basin performance at high heads. Also, the latter basins had

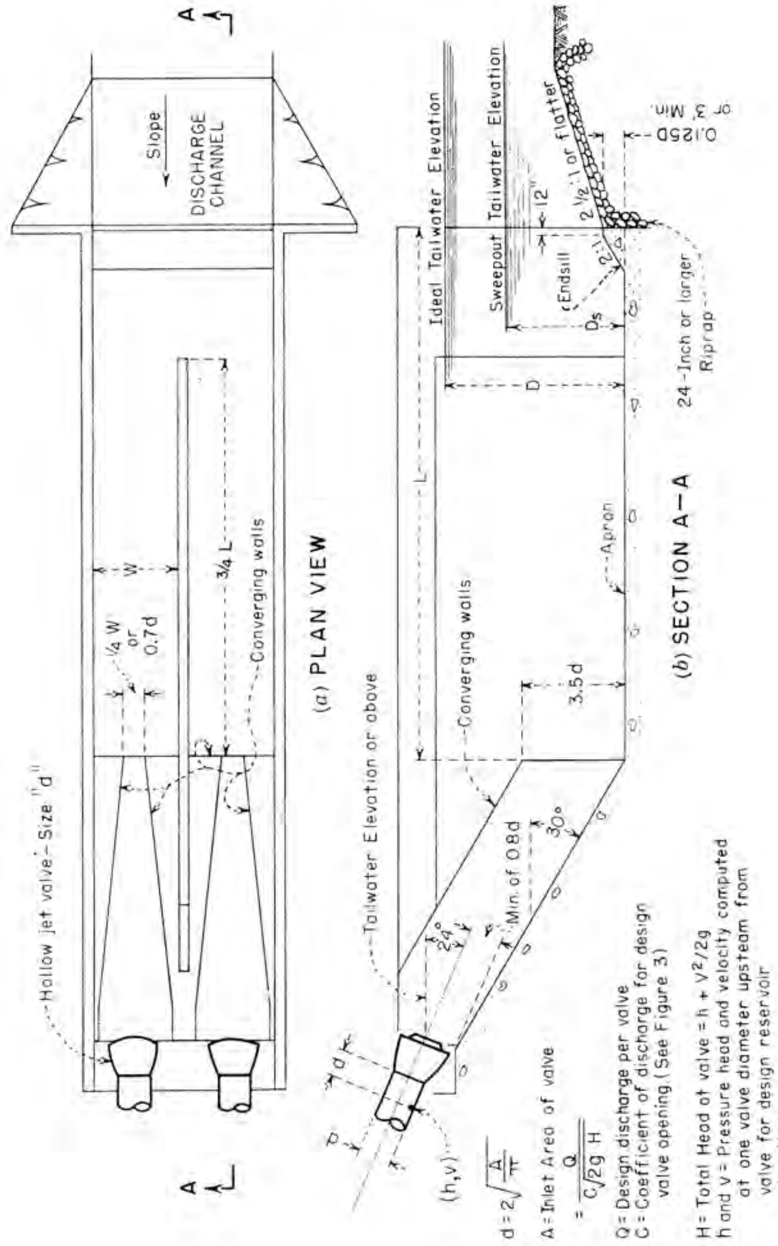


FIG. 11.—GENERALIZED DESIGN

been model tested over a greater operating range than were the earlier low-head basins.

To provide practical dishcarge limits for the tests, the 3-in. model was assumed to represent an 84-in. prototype valve, making the model scale 1:28. Discharges of 2,000 sec-ft to 4,000 sec-ft with one valve open 100% were considered to be the usual design discharges for a valve of this size. To produce these discharges, heads of 100 ft to 345 ft of water at the valve would be required.

Initial tests were made with the stilling basin apron longer than necessary and with no end sill in place. For a given discharge, the ideal depth of tail water was determined from visual inspection of the stilling action as it occurred over a range of tail water elevations. For each ideal tail water determination, the minimum length of concrete apron was estimated after an inspection of the flow currents in the model had indicated where an end sill should be placed in the prototype. Confirming tests were then conducted successively on a representative group of basins having the apron lengths previously determined and having an end sill at the end of the apron. Adjustments were then made as necessary to the preliminary values to obtain final ideal tail water depths and apron lengths. In the latter tests, the height of the valve above the maximum tail water elevation was adjusted to simulate a typical prototype installation. Similar tests were then made with the valve open 75% and 50%. Finally, a series of tests was made to determine the ideal width of stilling basin and the range of widths over which satisfactory performance could be expected.

Preliminary Tests.—In a typical test, the desired discharge was set by means of the laboratory venturi meters and passed through the hollow-jet valve or valves opened 100%. The tail water elevation was adjusted to provide the best energy dissipating action in the basin. The optimum value, tail water depth D in Fig. 11, was judged by the appearance and quality of the stilling action in the basin and on the smoothness of the tail water surface.

For discharges of 2,000 sec-ft to 4,000 sec-ft, it was found that the tail water could be raised or lowered about 3 ft (0.1 ft in model) from the ideal tail water elevation without adversely affecting the basin performance. Increasing the tail water depth beyond this margin reduced the efficiency of the stilling action and allowed the jet to flow along the bottom of the basin for a greater distance before being dissipated. This also produced surges in the basin and increased the wave heights in the discharge channel. Decreasing the tail water depth below the 3-ft margin moved the stilling action downstream in the basin and uncovered the valve jets at the end of the converging walls. This increased the flow velocity entering the discharge channel and increased the tendency to produce bed scour. Uncovering of the stilling action also produced objectionable splashing at the upstream end of the basin. If the tail water depth was decreased further, the flow swept through the basin with no stilling action having occurred. The latter tail water depth was measured and recorded as the sweep-out depth D_S. These tests were made with the dividing wall extended to the end of the basin, since this provided the least factor of safety against jump sweep out. With a shorter dividing wall, sweep out occurs at a tail water elevation slightly less than D_S.

With the ideal tail water depth set for a desired flow, the action in the basin was examined to determine the ideal length, L, of the basin apron, Fig. 11. The apron length was taken to the point where the bottom flow currents began to rise from the basin floor of their own accord, without assistance from an end sill, Fig. 5 (c). The water surface directly above and downstream from

this point was fairly smooth, indicating that the stilling action had been completed and that the paved apron and training walls need not extend farther. In the preceding individual model studies, it had been found that when the basin was appreciably longer than ideal, the ground roller at the end sill carried bed material from the discharge channel over the end sill and into the basin. If this action occurred in a prototype structure the deposited material would swirl around in the downstream end of the basin and cause abrasive damage to the concrete apron and end sill. It had also been found that scour tendencies in the discharge channel were materially increased if the basin was appreciably shorter than ideal. Therefore, the point at which the currents turned upward from the apron, plus the additional length required for an end sill, was determined to be the optimum length of apron. At this point, the scouring velocities were a minimum and any scouring tendencies would be reduced by the sloping end sill to be added later.

Practical difficulties were experienced in determining the exact length of apron required, however. Surges in the currents flowing along the basin floor caused the point of upturn to move upstream and downstream a distance of 1/4 to 1/2 D in a period of 15 sec to 20 sec in the model. An average apron length was therefore selected in the preliminary tests. For this reason, too, the end sill would help to neutralize the scouring tendencies which increased as the bottom currents surged downstream.

The depth D, sweep-out depth D_S, and length L were then determined for the range of discharges possible with the hollow-jet valve open 75%, and finally 50%, using the testing methods described in the preceding paragraphs. Partial openings were investigated because the valve size is often determined for the minimum operating head and maximum design discharge. When the same quantity is discharged at higher heads, the valve opening must be reduced. It may be necessary, therefore, to design the basin for maximum discharge with the valves opened less than 100%. When the relation between head and velocity in the valve is changed materially, the minimum required basin dimensions will be affected. The data for the partially opened valves are also useful in indicating the basin size requirements for discharges greater or less than the design flow conditions.

Final Tests and Procedures.—The final tests were made to correct or verify the dimensions obtained in the preliminary tests and to investigate the effect of varying the basin width. Scour tendencies were also observed to help evaluate the basin performance. D, D_S, and L for the three valve openings are functions of the energy in the flow at the valve. The energy may be represented by the total head, H, at the valve, Fig. 11. Therefore, to provide dimensionless data which may be used to design a basin for any size hollow-jet valve, D, D_S, and L values from the preliminary tests were divided by the valve diameter d, and each variable was plotted against H/d. The resulting curves, similar to those in Figs. 12, 13, and 14, were used to obtain dimensions for a group of model basins which were tested with the end sill at the end of the apron and with the valves placed to give the proper vertical distance between the valve and the tail water. For each model basin, a 3:1 upward sloping erodible bed, composed of fine sand, was installed downstream from the end sill. The bed was kept sufficiently low that it did not interfere with tail water manipulation, even when the tail water was lowered for the sweep-out tests. Test procedure was essentially as described for the preliminary tests.

Basin Depth and Length.—The preliminary depth curves for both ideal tail water depth and sweep-out tail water depth needed but little adjustment. The preliminary basin lengths were found to be too long for the high heads and too short for the lower heads, although both adjustments were relatively minor. The adjusted and final curves are shown in Figs. 12, 13, and 14.

It was observed that a longer apron than indicated by Fig. 14 was necessary when the tail water depth exceeded the tail water depth limit in Fig. 12. As the stilling action became drowned, the action in the basin changed from fine-grain turbulence to larger and slower moving vertical eddies. The bottom flow currents were not dissipated as thoroughly or as quickly and were visible on the apron for a greater distance, thereby increasing the necessary length of basin. The action is similar to that observed in hydraulic jumps which are drowned by excessive tail water depths. A moderate amount of drowning is tolerable, but it is important that the ideal tail water depth be maintained within stated limits if the best performance is desired. The tail water depth limits, 0.1 ft above and below the ideal depth, expressed in dimensionless form is 0.4 d. If this limit is exceeded, a model study is recommended.

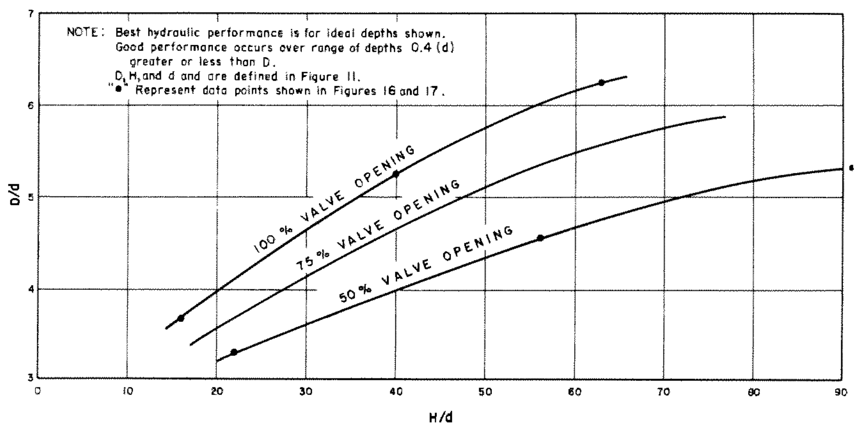


FIG. 12.—IDEAL TAIL WATER DEPTH

Basin Width.—To determine the effect of basin width, tests on several basins were made in which only the basin width was varied. It was found that the width could be increased to 3.0 times the valve diameter before the action became unstable. The width could be decreased to 2.5 times the valve diameter before the stilling action extended beyond the ideal length of basin. However, the H/d ratio and the valve opening were found to affect the required basin width as shown for 100%, 75%, and 50% valve openings in Fig. 15.

Basin width is not a critical dimension but certain precautions should be taken when selecting a minimum value. If the tail water is never to be lower than ideal, as shown by the curves in Fig. 12, the basin width may be reduced to 2.5 d. If the tail water elevation is to be below ideal, however, the curve values for width in Fig. 15 should be used. In other words, the lower limits for both tail water and basin width should not be used in the same structure. The combined minimums tend to reduce the safety factor against jump sweep-out and poor overall performance results. The basin width should not be increased above 3.0 d to substitute for some of the required length or depth of

the basin. If unusual combinations of width, depth, and length are needed to fit a particular space requirement, a model study is recommended.

Basin Performance.—The six model basins shown operating in Figs. 16 and 17 illustrate the performance to be expected from the recommended structures. The operating conditions in Figs. 16 and 17 correspond to points shown in Figs. 12, 14, and 15. Fig. 16 shows the operation for 100% valve opening; Fig. 17 shows the operation for 50% opening. The photographs may be used to determine the model appearance of the prototype basin and may help to provide a visual appraisal of the prototype structure. Wave heights, boil heights, or other visible dimensions may be scaled from the photographs (using the scale shown in the photographs) and converted to prototype dimensions by multiplying the scaled distances by the model scale. To determine the model scale, the prototype valve diameter in inches should be divided by 3 (the model valve diameter). To determine which of the six photographs represents the prototype in question, the H/d ratio should be used to select the photograph which most nearly represents the design problem. It is permissible to interpolate between photographs when necessary.

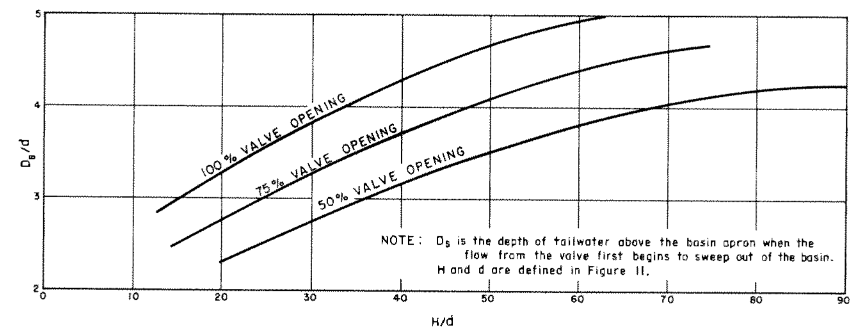


FIG. 13.—TAIL WATER SWEEPOUT DEPTH

Center Dividing Wall.—Prototype stilling basins usually have two valves placed a minimum distance apart, and alined to discharge parallel jets. It is necessary, without exception, to provide dividing walls between the valves for satisfactory hydraulic performance. When both valves are discharging without a dividing wall, the flow in the double basin sways from side to side to produce longitudinal surges in the tail water pool. This action occurs because the surging downstream from each valve does not have a fixed period, and the resulting harmonic motion at times becomes intense. When only one valve is discharging, conditions are worse. The depressed water surface downstream from the operating valve induces flow from the higher water level on the non-operating side. Violent eddies carry bed material from the discharge channel into the basin and swirl it around. This action in the prototype would damage the basin as well as the discharge channel. In addition, the stilling action on the operating side is impaired.

To provide acceptable operation with one valve operating, the dividing wall should extend to three-fourths of the basin length or more. However, if the two adjacent valves discharge equal quantities of flow at all times, the length

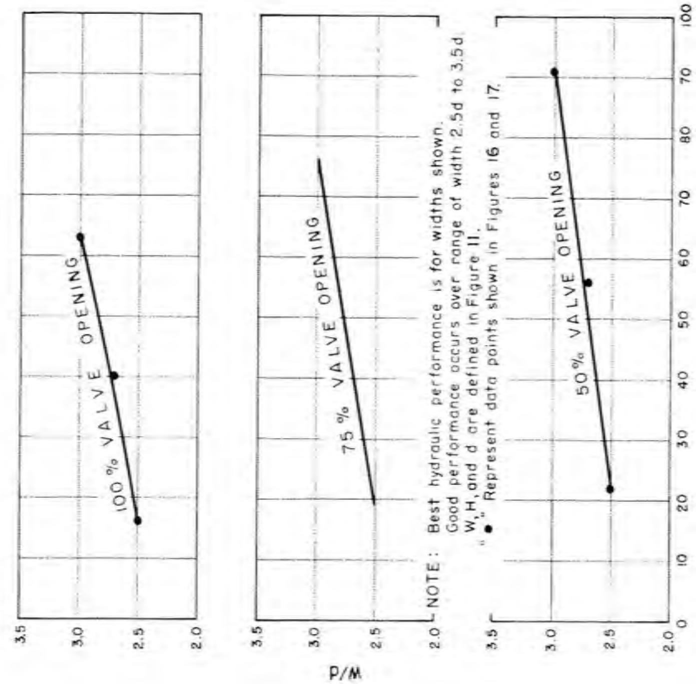


FIG. 15.—BASIN WIDTH PER VALVE

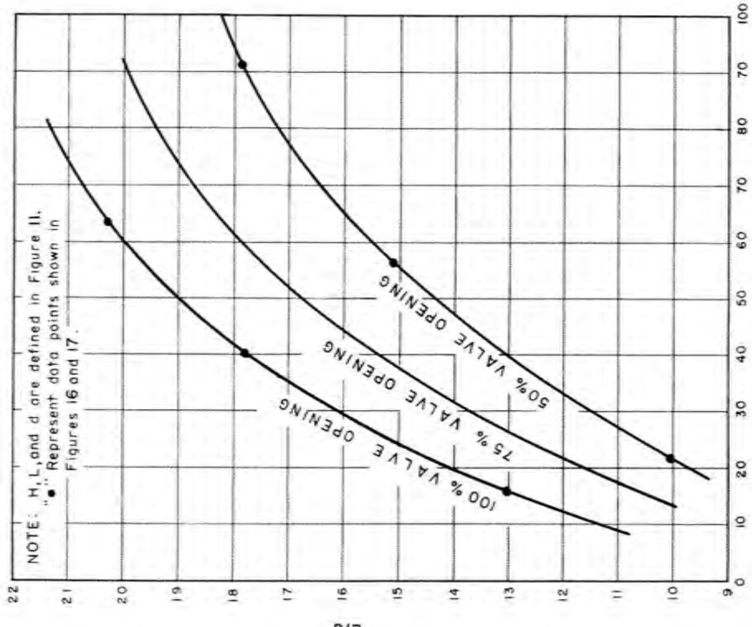
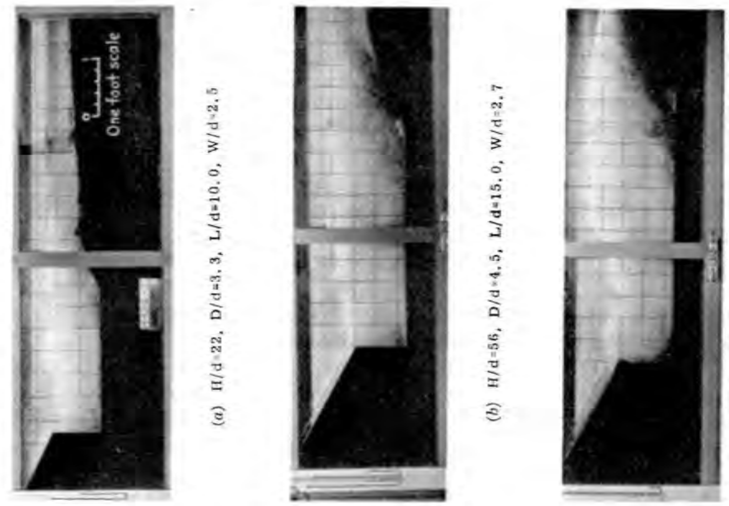


FIG. 14.—STILLING BASIN LENGTH

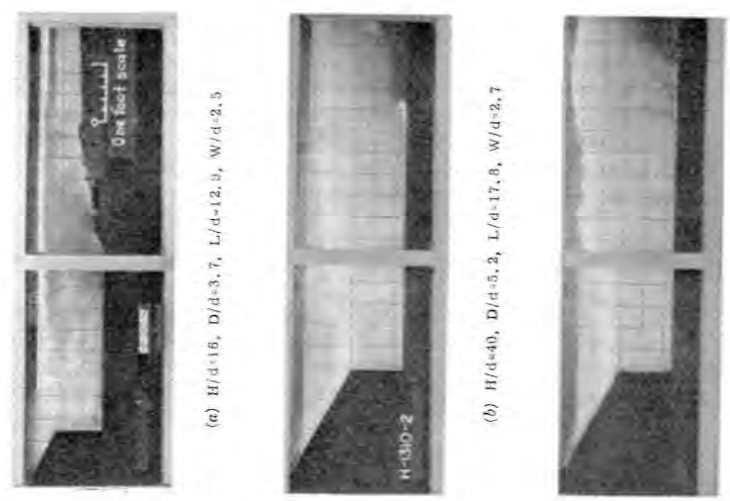


(a) H/d=22, D/d=3.3, L/d=10.0, W/d=2.5

(b) H/d=56, D/d=4.5, L/d=15.0, W/d=2.7

(c) H/d=91, D/d=5.3, L/d=17.8, W/d=3.0

FIG. 17.—HOLLOW-JET VALVE STILLING BASIN PERFORMANCE VALVE 50% OPEN



(a) H/d=16, D/d=3.7, L/d=12.9, W/d=2.5

(b) H/d=40, D/d=5.2, L/d=17.8, W/d=2.7

(c) H/d=63, D/d=6.2, L/d=20.3, W/d=3.0

FIG. 16.—HOLLOW-JET VALVE STILLING BASIN PERFORMANCE, VALVE 100% OPEN

of the center dividing wall may be reduced to one-half of the basin length. The margin against sweep out is increased, but the stability of the flow pattern is decreased as the dividing wall is shortened. In some installations, a full-length wall may be desirable to help support the upper levels of a powerplant, Fig. 1. If other arrangements of the center wall are required a model study is recommended.

Valve Placement.—A hollow-jet valve should not operate submerged because of the possibility of cavitation occurring within the valve. However, the valve may be set with the valve top at maximum tail water elevation, and the valve will not be underwater at maximum discharge. The valve jet sweeps the tail water away from the downstream face of the valve sufficiently to allow usual ventilation of the valve. However, as a general rule, it is recommended that the valve be placed with its center (downstream end) no lower than tail water elevation.

Riprap Size.—A prototype basin is usually designed for maximum discharge, but will often be used for lesser flows at partial and full valve openings. For these lesser discharges, the basin will be larger than necessary, and in most respects, the hydraulic performance will be improved. However, at less than design discharge, particularly those close to the design discharge, the ground roller will tend to carry some bed material upstream and over the end sill into the basin. The intensity of this action is relatively mild over most of the discharge range, and movement of material may be prevented by placing riprap downstream from the end sill. Riprap, having 50% or more of the individual stones 24 in. to 30 in. or larger in diameter, should provide a stable channel downstream from the end sill. The riprap should extend a distance D, or more, from the end sill. If the channel is excavated and slopes upward to the natural river channel, the riprap should extend from the end sill to the top of the slope, or more. The riprap should not be terminated on the slope.

The justification for choosing riprap as described is as follows: Because of the fixed relationships between depth and width of basin, the average velocity leaving the basin will seldom exceed 5 fps, regardless of structure size. Surface velocities will therefore seldom exceed 7 fps to 8 fps and bottom velocities 3 fps to 4 fps. To protect against these velocities, stones 10 in. to 12 in. in diameter would be ample. However, the critical velocity for riprap stability is the upstream velocity of the ground roller which has a curved path and tends to lift the stones out of place. Model tests showed that graded riprap up to 24 in. to 30 in. in diameter was sufficient to provide bed stability.

APPLICATION OF RESULTS

Problems.—Design a stilling basin for (a) 1 hollow-jet valve discharging 1,300 cfs, and (b) a double basin for 2 valves discharging 650 cfs each. In both problems, the reservoir is 108 ft above maximum tail water elevation.

One-valve Stilling Basin Design.—The valve size should be determined from the equation:

$$Q = C\,A\,\sqrt{2\,g\,H} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

in which Q is the design discharge, C is the coefficient of discharge, A is the inlet area to the valve, g is the acceleration of gravity, and H is the usable or total head at the valve with the valve center placed at maximum tail water elevation. In this example, the usable head at the valve is estimated to be 80% of the total head of 108 ft, or 86 ft.

From Fig. 3, for 100% valve opening:

$$C = 0.7$$

Then, from Eq. 1

$$A = 25 \text{ sq ft}$$

and

$$d = 5.67 \text{ ft}$$

in which d is the inlet diameter of the valve and also the nominal valve size. Since nominal valve sizes are usually graduated in 6-in. increments,

$$d = 6 \text{ ft}$$

would be selected. Because the selected valve is larger than required, it would not be necessary to open the valve fully to pass the design flow at the maximum head.

Having determined the valve size and therefore the diameter of the supply conduit, the probable head losses in the system from reservoir to valve may be computed. In this example, the computed losses are assumed to be 20 ft, which leaves 88 ft of head at the valve. Using Eq. 1, C is computed to be 0.61; from Fig. 3, the valve opening necessary to pass the design discharge at the design head is 83%.

The basin depth, length, and width may be determined from Figs. 12, 13, 14, and 15 using the head ratio

$$\frac{H}{d} = \frac{88}{6} = 14.67$$

For 83% valve opening, Fig. 12 shows the depth ratio

$$\frac{D}{d} = 3.4$$

The depth of the basin is

$$D = 20.4 \text{ ft}$$

therefore, the apron is placed 20.4 ft below the maximum tail water elevation.

For 83% valve opening, Fig. 14 shows the length ratio

$$\frac{L}{d} = 11.2$$

The length of the basin is

$$L = 67 \text{ ft}$$

For 83% valve opening, Fig. 15 shows the width ratio

$$\frac{W}{d} = 2.5$$

The width of the basin is

$$W = 15 \text{ ft}$$

The dimensions of other components of the basin may be determined from Fig. 11.

The tail water depth at which the flow will sweep from the basin may be determined from Fig. 13. For 83% valve opening, the depth sweep-out ratio

$$\frac{D_S}{d} = 2.7$$

The sweep-out depth is

$$D_S = 16.2 \text{ ft}$$

Since 20.4 ft of depth is provided, the basin has a safety factor against sweep-out of 4.2 ft of tail water depth. In most installations this is sufficient, but if a greater margin of safety is desired, the apron elevation may be lowered

$$0.4 \text{ (d)} = 2.4 \text{ ft}$$

If greater economy and less margin of safety are desired, the basin floor may be placed 2.4 ft higher to provide only 18 ft of depth

If the tail water depth from Fig. 12 is adopted, the water surface profile will be similar to that shown in Fig. 16 (a), since the H/d value of 16 in Fig. 16 (a) is comparable to 14.67 in this example. If tail water depth 2 ft greater or less than the ideal is adopted for the prototype, the water surface profile will be moved up or down accordingly. Water surfaces may be estimated by multiplying the variations shown in Fig. 16 (a) by the quotient obtained by dividing the prototype valve diameter of 72 in. by the mdoel valve diameter of 3 in. Wave heights in the downstream channel will be considerably less as indicated in other photographs showing downstream conditions.

Two-valve Stilling Basin Design.—If two valves are to be used to discharge the design flow of 1,300 sec-ft, a double basin with a dividing wall is required. The discharge per valve is 650 cfs, and at 100% valve opening the valve coefficient is 0.7, Fig. 3. The head on the valve is estimated to be 86 ft as in the first example. From Eq. 1, the inlet area of the valve is found to be 12.48 sq ft. A 48-in. valve provides practically the exact area required.

For this example, it is assumed that the computations to determine head losses have been made and that the estimated head of 86 ft at the valves is correct. Therefore, 100% valve opening will be necessary to pass the design flow.

Using the methods given in detail in the first example:

$$\frac{H}{d} = 21.5$$

$$\frac{D}{d} = 4.06, \text{ from Fig. 12}$$

and

$$D = 16.2 \text{ ft}$$

$$\frac{D_S}{d} = 3.3, \text{ from Fig. 13}$$

then

$$D_S = 13.2 \text{ ft}$$

The tail water depth for sweep out is therefore 3.0 ft below the ideal tail water depth. If more or less insurance against the possibility of sweep out is desired, the apron may be set lower or higher by the amount

$$0.4 \text{ (d)} = 1.6 \text{ ft}$$

To aid in determining the apron elevation, the effect of spillway, turbine, or other discharges on the tail water range may need to be considered.

$$\frac{L}{d} = 14.4, \text{ from Fig. 14}$$

then

$$L = 58 \text{ ft}$$

$$\frac{W}{d} = 2.6, \text{ from Fig. 15}$$

then

$$W = 10.4 \text{ ft}$$

Since two valves are to be used, the total width of the basin will be 2(W) plus the thickness of the center dividing wall. The length of the center dividing wall should be three-fourths of the apron length or 43.5 ft long, Fig. 11. If it is certain that both valves will always discharge equally, the wall need be only one-half the apron length or 29 ft long. The hydraulic design of the basin may be completed using Fig. 11.

If the tail water depth determined from Fig. 12 is adopted, the water surface profile for determining wall heights may be estimated by interpolating between Fig. 16 (a) and (b). Water surface variations may be predicted by multiplying values scaled from the photographs by the ratio 48/3.

PROTOTYPE PERFORMANCE

The Boysen Dam and Falcon Dam outlet works stilling basins, Figs. 1, 6, and 7, fit the design curves derived from the generalized study quite well, and have been field tested and found to perform in an excellent manner. Table 1 shows the important dimensions of these basins and indicates that the values computed from the design curves of this paper are in good agreement with those obtained from the individual model tests.

Boysen Dam.—The outlet works basin at Boysen Dam is designed for 1,320 cfs from two 48-in. hollow-jet valves 100% open at reservoir elevation 4725.00. Design tail water elevation at the basin is 4616.00. The model performance of this basin is shown in Figs. 18 and 19.

The prototype tests, Figs. 20, 21, and 22, were conducted with the reservoir at elevation 4723.5 and with the powerplant both operating and shut down. The spillway was not operating. The outlet works discharge was measured at a temporary gaging station located about 1/2 mile downstream from the dam using a current meter to determine the discharge. Tail water elevations were read on the gage located in the powerhouse.

The prototype performed as well as predicted by the model and was considered satisfactory in all respects. However, the field structure entrained more air within the flow than did the model. This caused the prototype flow to appear more bulky, and "white water" extended farther into the downstream channel than was indicated in the model. A comparison of the model and prototype photographs, Figs. 19 and 22, illustrates this difference. Greater air

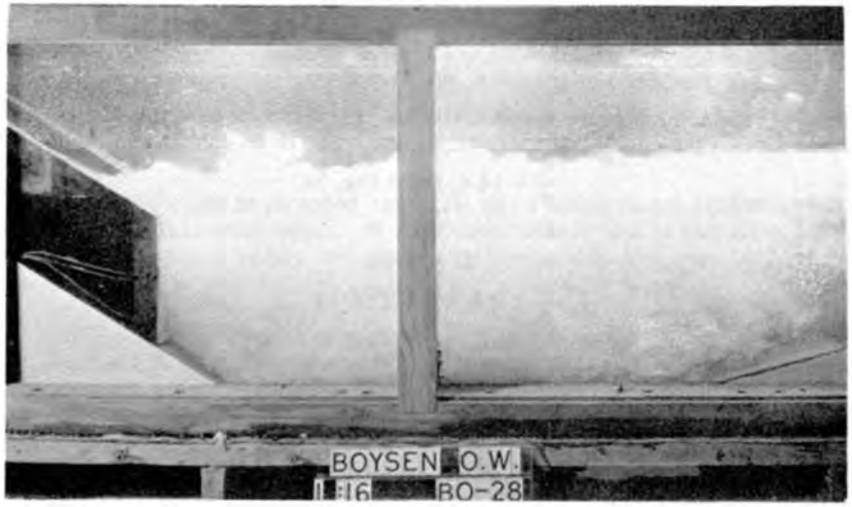


FIG. 18.—BOYSEN DAM, LEFT VALVE OF OUTLET WORKS BASIN, DISCHARGING 660 CFS 1:16 SCALE MODEL

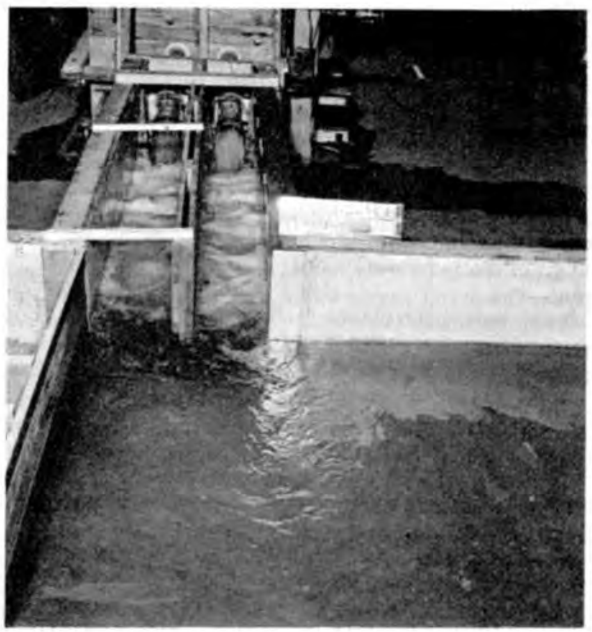


FIG. 19.—BOYSEN DAM, OUTLET WORKS DISCHARGING 1320 CFS 1:16 SCALE MODEL

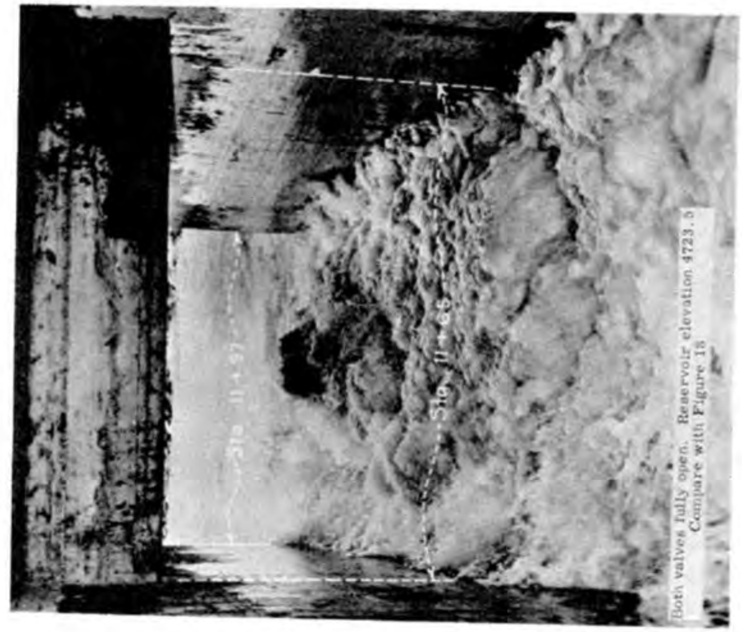


FIG. 21.—BOYSEN DAM, LEFT VALVE OF OUTLET WORKS BASIN DISCHARGING 732 CFS LOOKING DOWNSTREAM

FIG. 20.—BOYSEN DAM, LEFT VALVE OF OUTLET WORKS BASIN DISCHARGING 732 CFS LOOKING UPSTREAM

FIG. 22.—BOYSEN DAM OUTLET WORKS DISCHARGING 1344 CFS

FIG. 23.—MEXICAN OUTLET WORKS - FALCON DAM

entrainment in the prototype is usually found when making model prototype comparisons, particularly when the difference between model and prototype velocities is appreciable. In other respects, however, the prototype basin was as good or better than predicted from the model tests.

For the initial prototype test, only the left outlet valve was operated; the powerhouse was not operating. At the gaging station, the discharge was measured to be 732 cfs after the tail water stabilized at elevation 4614.5. (This is a greater discharge than can be accounted for by calculations. It is presumed that valve overtravel caused the valve opening to exceed 100% even though the indicator showed 100% open.) It was possible to descend the steel ladder, Fig. 1, to closely observe and photograph the flow in the stilling basin, Figs. 20 and 21. The basin was remarkably free of surges and spray; the energy dissipating action was excellent. There was no noticeable vibration at the valves or in the basin. The flow leaving the structure caused only slightly more disturbance in the tailrace than the flow from the draft tubes when the turbines were operating at normal load.

Operation of the prototype provided an opportunity to check the air requirements of the structure, which could not be done on the model. With the inspection cover removed, Fig. 1, the basin was open to the rooms above. Air movements through the inspection opening and in the powerplant structure were negligible, which indicated that ample air could circulate from the partially open end of the stilling basin, Fig. 21.

When both valves were discharging fully open, the tail water stabilized at elevation 4615. A discharge measurement at the gaging station disclosed that both valves were discharging 1,344 cfs. Since the left valve had been found to discharge 732 cfs, the right valve was discharging 612 cfs.

The reason for the difference in discharge is that the 57-inch-inside-diameter outlet pipe to the left valve is short and is connected to the 15-foot-diameter header which supplies water to the turbines, Fig. 1. The right valve is supplied by a separate 66-inch-diameter pipe extending to the reservoir. Therefore, greater hydraulic head losses occur in the right valve supply line, which accounts for the lesser discharge through the right valve. Although it was apparent by visual observation that the left valve was discharging more than the right valve, Fig. 22, no adverse effect on the performance of the outlet works stilling basin or on flow conditions in the powerhouse tailrace could be found.

The outlet works basin performance was also observed with the turbines operating and the tail water at about elevation 4617. No adverse effects of the outlet works discharge on powerplant performance could be detected. Flow conditions in the tailrace area were entirely satisfactory, Fig. 22. Since the tests were made at normal reservoir level and maximum discharge, the stilling basin was subjected to a severe test.

Falcon Dam.—The outlet works basin on the Mexico side at Falcon Dam is designed to accommodate 4,570 cfs from two 90-in. valves or 2,400 cfs from one valve, with the valves 100% open and the reservoir at elevation 300. The tail water elevation is 181.2 when the powerplant is discharging 5,400 cfs in conjunction with both valves. The model performance of this basin is shown in Figs. 23 and 24.

The outlet works basin on the United States side at Falcon Dam is designed to discharge 2,920 cfs from two 72-in. valves, or 1,600 cfs from one valve, with the valves 100% open and the reservoir at elevation 310. Tail water is at

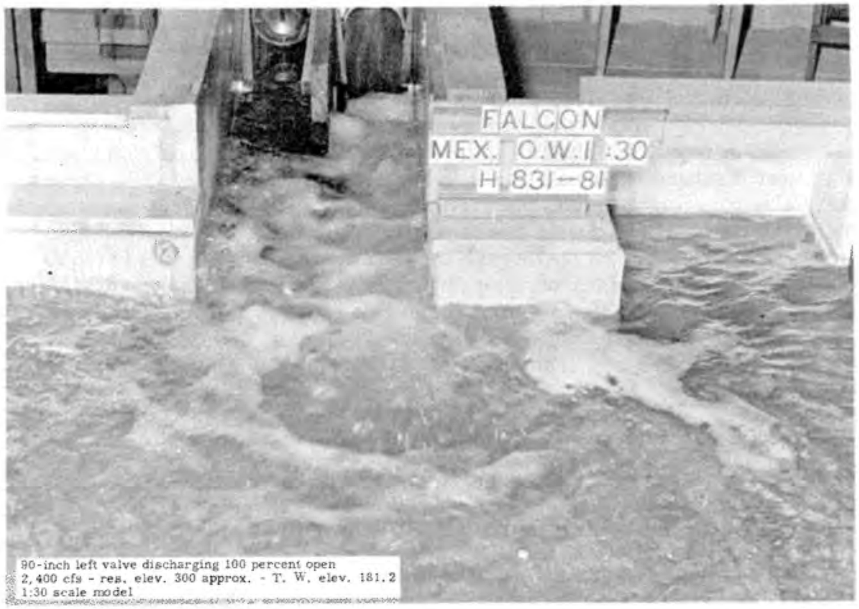


FIG. 24.—MEXICAN OUTLET WORKS - FALCON DAM

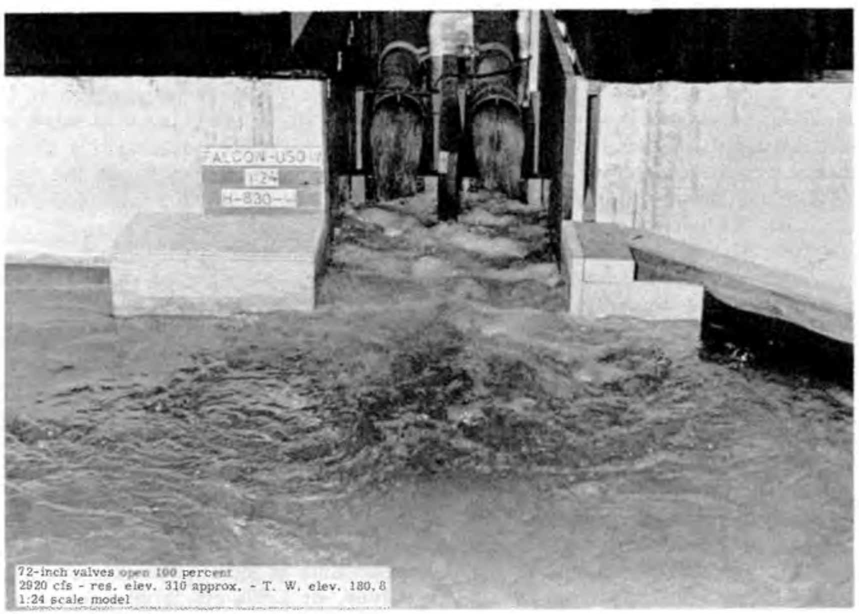


FIG. 25.—UNITED STATES OUTLET WORKS - FALCON DAM

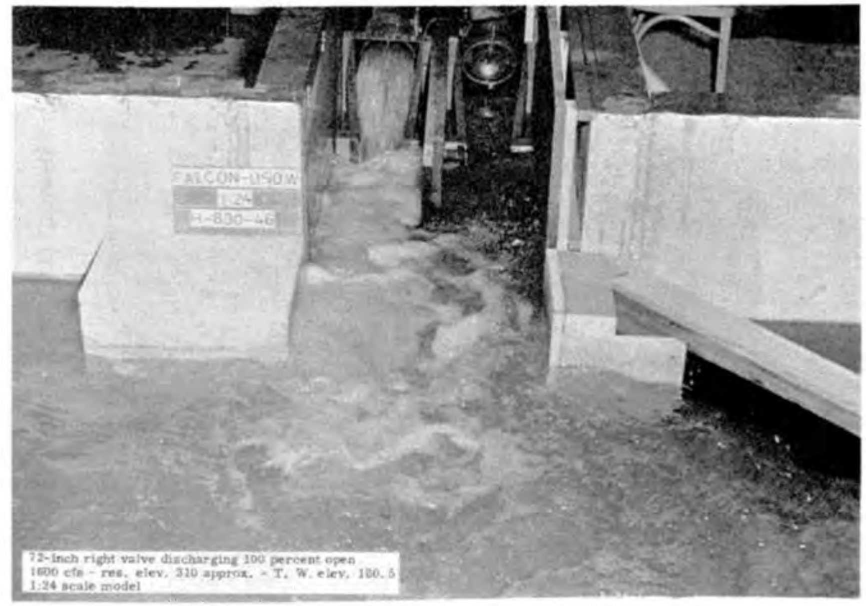


FIG. 26.—UNITED STATES OUTLET WORKS - FALCON DAM

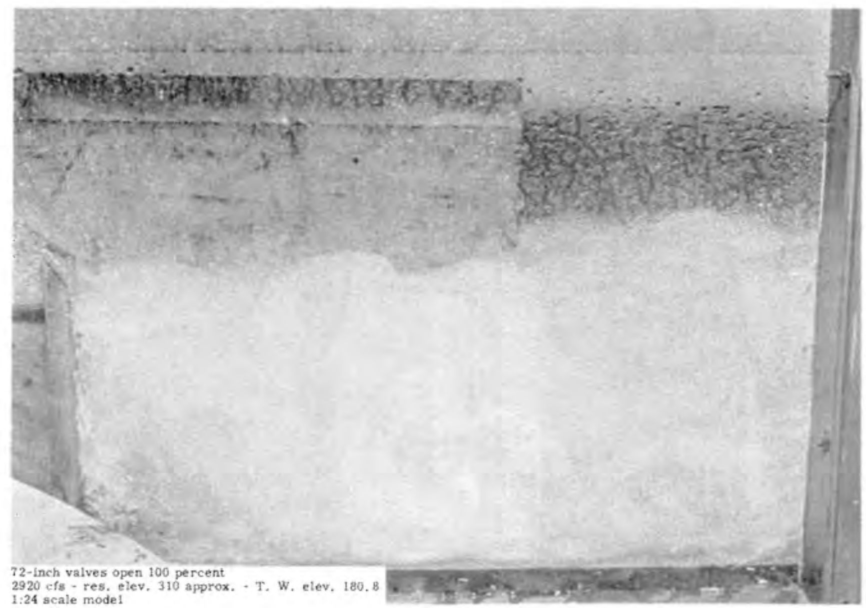


FIG. 27.—UNITED STATES OUTLET WORKS - FALCON DAM

(a)

(b)

90-inch left valve discharging - 100 percent open
2,300 cfs approx. - res. elev. 301.83 - T. W. elev. 183.0
Compare with Figure 24

FIG. 28.—MEXICAN OUTLET WORKS - FALCON DAM

(a)

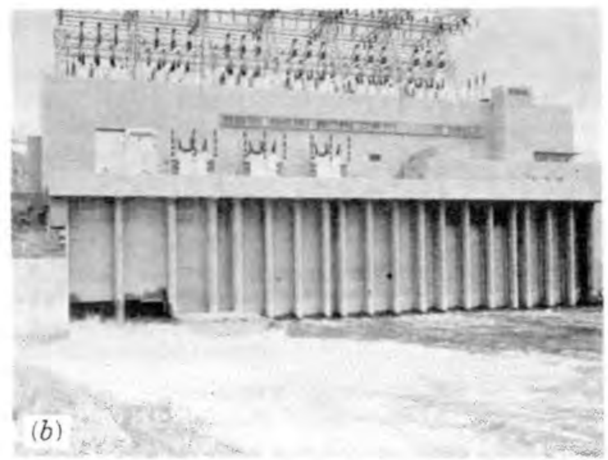
(b)

72-inch left valve discharging 100 percent open
1,750 cfs approx. - res. elev. 301.83 - T. W. elev. 182.7
Compare with Figures 26 & 27

FIG. 29.—UNITED STATES OUTLET WORKS - FALCON DAM

(a)

90-inch outlet works valves open 100 percent discharging
4,500 cfs approx. - T. W. elev. 183.6.
Turbine gates 72 percent open - 100 percent load.



(b)

72-inch outlet works valves open 100 percent discharging
3,000 cfs approx. - T. W. elev. 184.1.
Turbine gates 72 percent open - 100 percent load.

FIG. 30.—FALCON DAM MEXICAN & UNITED STATES POWER-PLANTS & OUTLET WORKS DISCHARGING AT RESERVOIR ELEVATION 301.83.

elevation 180.8 when two valves are operating and 180.5 when one valve is operating. The model performance of this basin is shown in Figs. 25, 26, and 27.

The prototype tests at Falcon, Figs. 28, 29, and 30, were conducted at near maximum conditions; the reservoir was at elevation 301.83, and the valves were 100% open. In each outlet works, the valves were operated together and individually. Single-valve operation represents an emergency condition and subjects the stilling basin to the severest test, Figs. 28 and 29. All turbines at both powerplants were operating at 72% gate and 100% load during all tests. The prototype valve discharges were determined from discharge curves based on model test data.

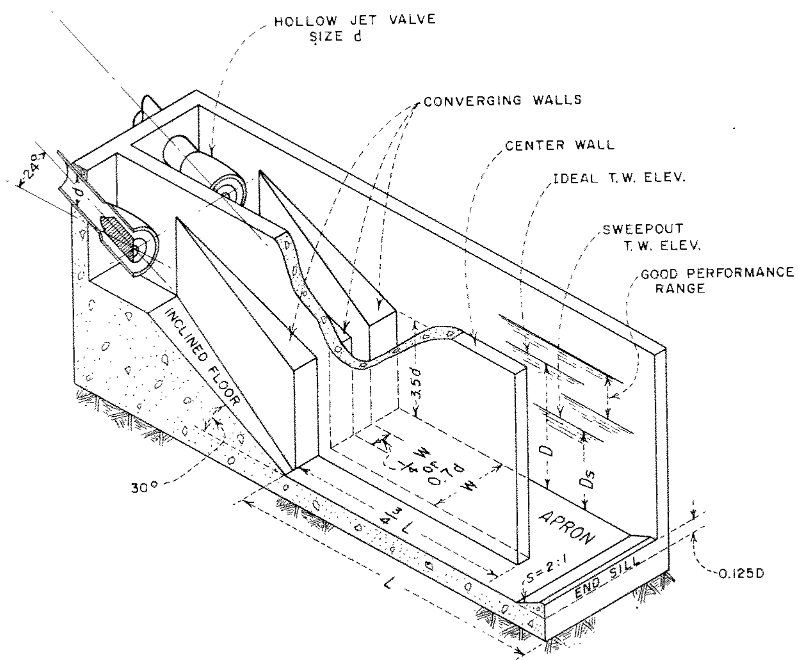


FIG. 31.—DEVELOPED BASIN

Here, too, more white water was evident in the prototype than in the model. The greater amount of air entrainment in the prototype, evident in the photographs, caused bulking of the flow at the end of the stilling basin and a higher water surface than was observed in the model. However, the prototype tail water is 3 ft to 4 ft higher than shown in the model photograph, and this probably helps to produce a higher water surface boil at the downstream end of the basin by reducing the efficiency of the stilling action. In other respects, the prototype basin performed as predicted by the model.

CONCLUSIONS

The schematic drawing, Fig. 31, shows the developed basin and the relationships between important dimensions.

A brief description of the seven steps required to design a stilling basin is given below:

1. Using the design discharge Q, the total head at the valve H, and the hollow-jet valve discharge coefficient C from Fig. 3, solve the equation $Q = C\,A\,\sqrt{2\,g\,H}$ for the valve inlet area A and compute the corresponding diameter d which is also the nominal valve size.
2. Use H/d in Fig. 12 to find D/d and thus D, the ideal depth of tail water in the basin. Determine the elevation of the basin floor, tail water elevation minus D. It is permissible to increase or decrease D by as much as 0.4 (d).
3. Use H/d in Fig. 14 to find L/d and thus L, the length of the horizontal apron.
4. Use H/d in Fig. 15 to find W/d and thus W, the width of the basin for one valve.
5. Use H/d in Fig. 13 to find D_s/d and thus D_s, the tail water depth at which the action is swept out of the basin. D minus D_s gives the margin of safety against sweep out.
6. Complete the hydraulic design of the basin from the relationships given in Fig. 11.
7. Use the H/d ratio to select the proper photograph in Figs. 16 and 17 to see the model and help visualize the prototype performance of the design. The water surface profile may be scaled from the photograph using the scale on the photograph. To convert to prototype dimensions, multiply the scaled values by the ratio d (in.)/3.

Stilling basin dimensions calculated as indicated above are in close agreement with the dimensions obtained from individual model tests of the basins for Boysen, Falcon, Yellowtail, Trinity, and Navajo Dams, Table 1. Since the Boysen and Falcon basins performed satisfactorily during prototype tests, it is believed that satisfactory future projects may be hydraulically designed from the material presented herein.

ACKNOWLEDGMENTS

Data and material used in this paper were obtained through cooperation of individuals too numerous to acknowledge singly, yet their wholehearted interest aided materially in providing a complete analysis of the problem. Their assistance is gratefully acknowledged.

The hollow-jet valve stilling basin was developed in the Hydraulic Laboratory, Division of Engineering Laboratories, through close coordination with the Mechanical Branch and the Dams Branch of the Divsioni of Design, all of the Bureau of Reclamation, Assistant Commissioner and Chief Engineer's Office, Denver, Colorado.

The prototype tests at Boysen Dam were made with the cooperation of the Bureau's Region 6 office in Billings, Montana, and the Yellowstone-Bighorn Projects Office, Cody, Wyoming. Bureau personnel at Boysen Dam operated the hydraulic structures and assisted in obtaining data. United States Geological Survey personnel at Riverton, Wyoming, made the river gagings, and State of Wyoming personnel made downstream river adjustments to permit above-normal discharges.

The prototype tests at Falcon Dam, which included tests on both the United States and Mexico outlet works and powerplants, were conducted by personnel at Falcon Dam through arrangements with the International Boundary and Water Commission, El Paso, Texas.

HOWELL-BUNGER & RING JET VALVE SIZING

H = Net Head at HBV/RJV(Minimum) - (FT) = 60

$$Q(CFS) = C_d \times \sqrt{(2 \times g \times \Delta H)} \times \left(\frac{\pi \times D^2}{4}\right)$$

Q = Flow(Maximum) - (CFS(FT^3/SEC)) = 1400

$$D(FT) = \sqrt{\frac{4 \times Q}{\pi \times C_d \times \sqrt{(2 \times g \times \Delta H)}}}$$

D(HBV)(IN) = 69.70 C_d = 0.85 VALVE SIZE = 72 INCHES

D(RJV)(IN) = 72.76 C_d = 0.78 VALVE SIZE = 78 INCHES

$$\Delta H = \frac{Q^2}{2 \times g \times \left(Cd \times \frac{\pi}{4} \times \left(\frac{D}{12}\right)^2\right)^2}$$

H (FT) =	52.7	Howell-Bunger Valve	Minimum head required to obtain maximum flow from selected HBV/RJV size	72
H (FT) =	62.6	Ring Jet Valve		78

$$Q(CFS) = C_d \times \sqrt{(2 \times g \times \Delta H)} \times \left(\frac{\pi \times D^2}{4}\right)$$

Q (CFS(FT^3/SEC)) =	1,494	Howell-Bunger Valve	Maximum obtainable flow at minimum net head from selected HBV/RJV size	72
Q (CFS(FT^3/SEC)) =	1,609	Ring Jet Valve		78

V:\Valve Standards\RJV\[HBV&RJV Sizing.xls]Sheet1

HOWELL-BUNGER & RING JET VALVE SIZING

H = Net Head at HBV/RJV(Minimum) - (FT) = 40

$$Q(CFS) = C_d \times \sqrt{(2 \times g \times \Delta H)} \times \left(\frac{\pi \times D^2}{4} \right)$$

Q = Flow(Maximum) - (CFS(FT^3/SEC)) = 1400

$$D(FT) = \sqrt{\frac{4 \times Q}{\pi \times C_d \times \sqrt{(2 \times g \times \Delta H)}}}$$

D(HBV)(IN) = 77.14 C_d = 0.85 VALVE SIZE = 78 INCHES

D(RJV)(IN) = 80.52 C_d = 0.78 VALVE SIZE = 84 INCHES

$$\Delta H = \frac{Q^2}{2 \times g \times \left(Cd \times \frac{\pi}{4} \times \left(\frac{D}{12} \right)^2 \right)^2}$$

H (FT) =	38.3	Howell-Bunger Valve	Minimum head required to obtain maximum flow from selected HBV/RJV size	78
H (FT) =	45.4	Ring Jet Valve		84

$$Q(CFS) = C_d \times \sqrt{(2 \times g \times \Delta H)} \times \left(\frac{\pi \times D^2}{4} \right)$$

Q (CFS(FT^3/SEC)) =	1,432	Howell-Bunger Valve	Maximum obtainable flow at minimum net head from selected HBV/RJV size	78
Q (CFS(FT^3/SEC)) =	1,524	Ring Jet Valve		84

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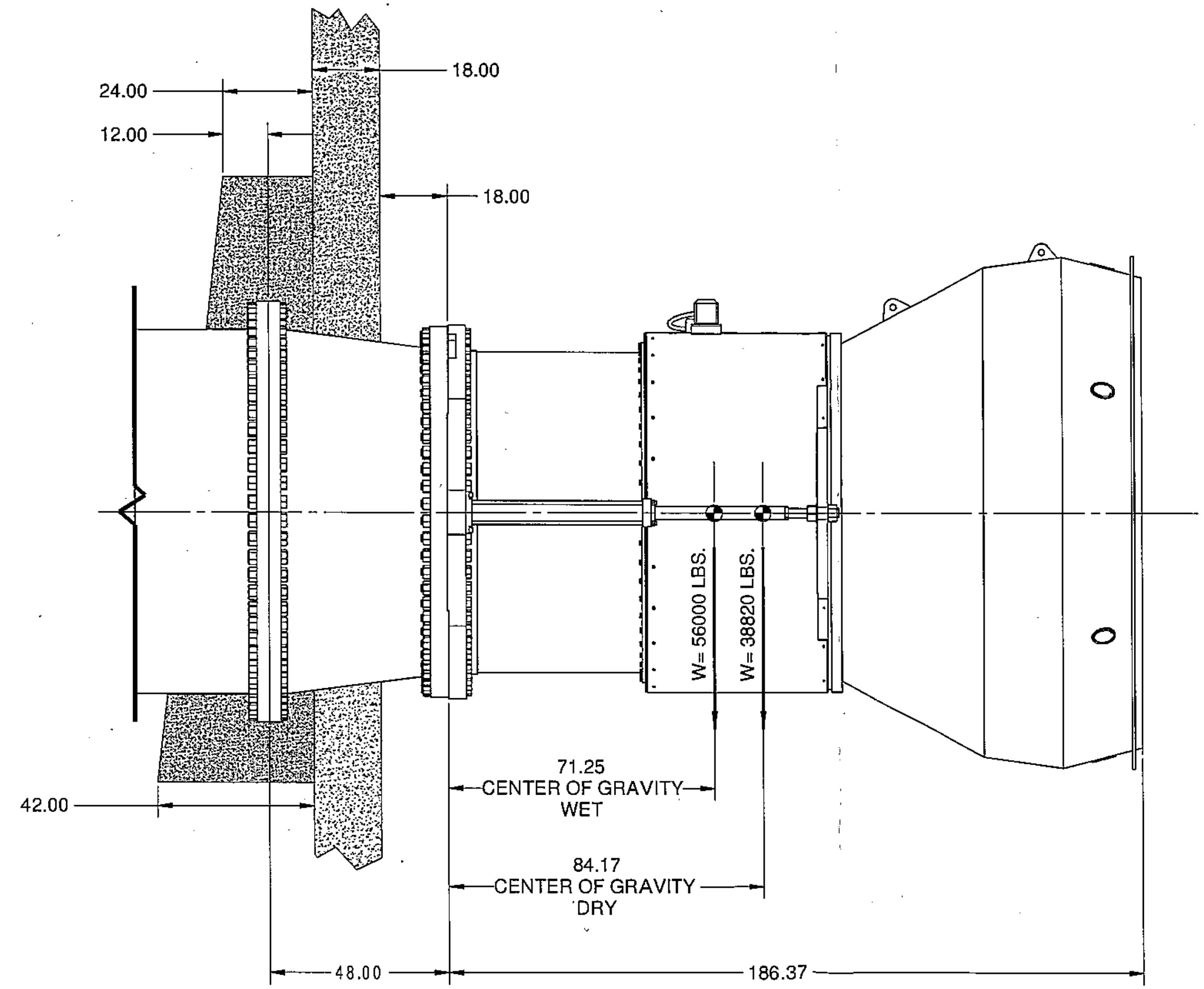

PROJECT: CLARK CANYON HYDRO.
EAST BENCH UNIT, MONTANA

QUANTITY: (1) RING JET VALVE
(1) 96" TO 84" REDUCER

LOCATION: VALVE HOUSE

SEE MATERIAL SPEC SHEET FOR PAINT AND TESTING DETAILS

WEIGHTS

VALVE: 38820 LBS.
VALVE WITH WATER: 56000 LBS.
REDUCER: 10045 LBS.

RODNEY HUNT COMPANY		ORANGE, MA. 01364-1251	
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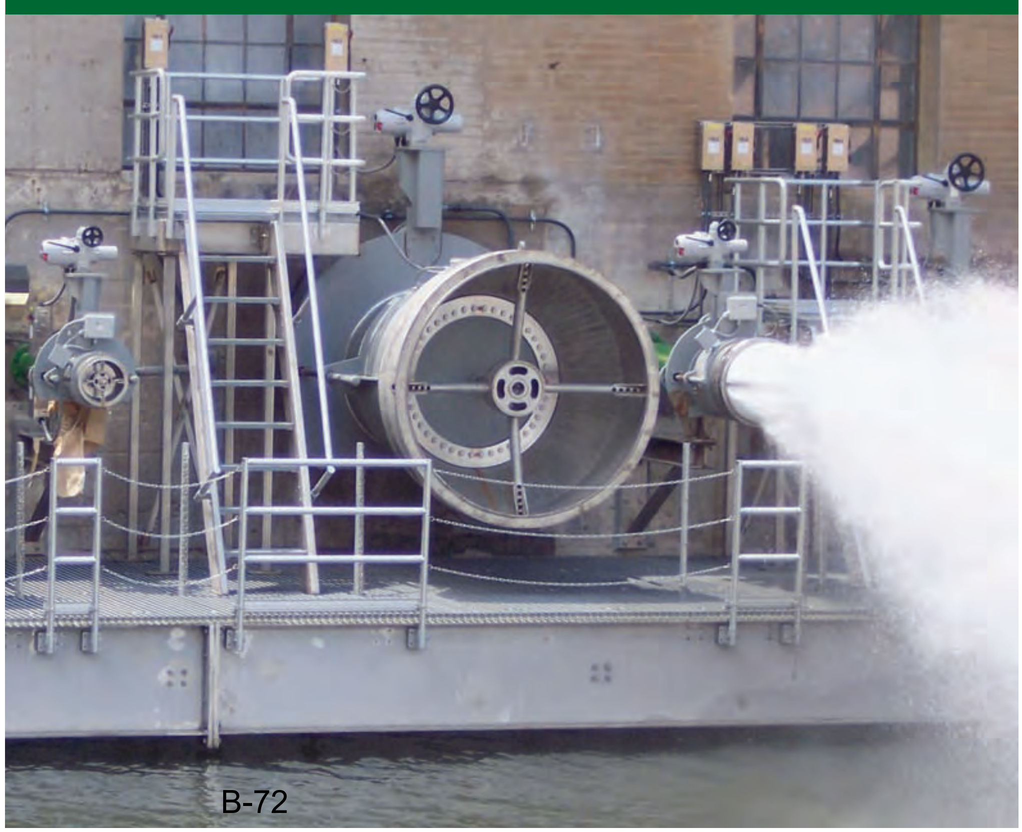
RODNEY HUNT®

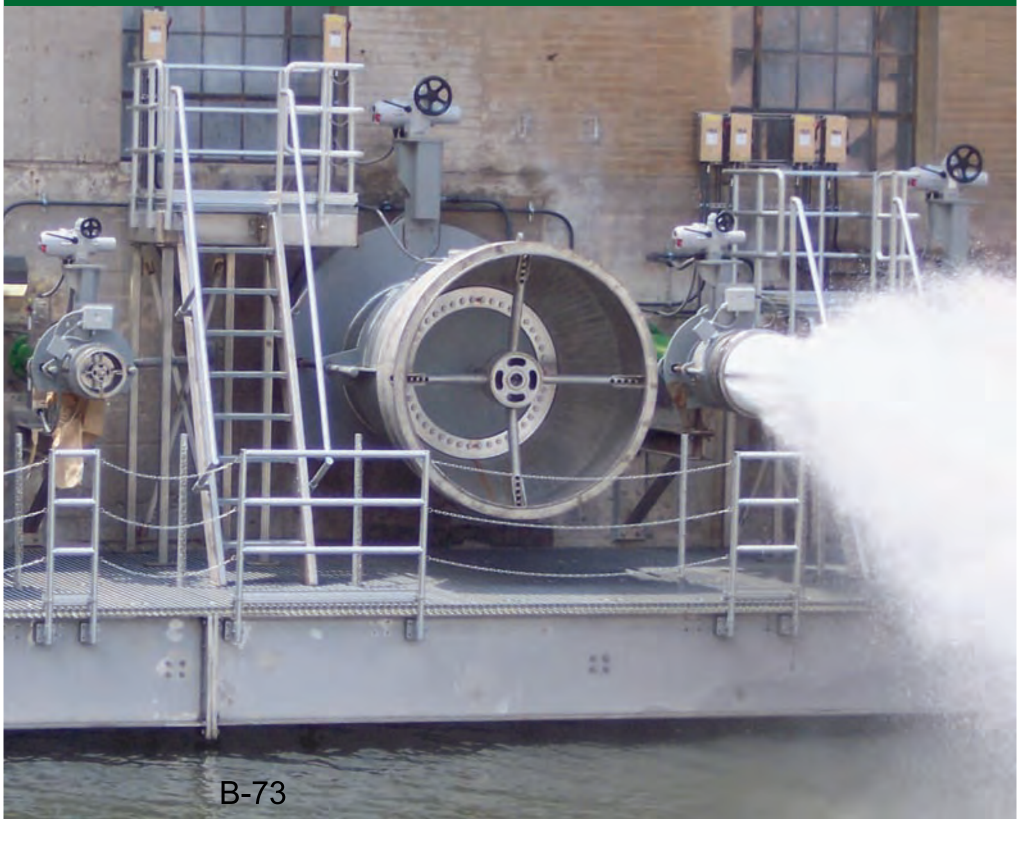
Howell-Bunger® and Ring Jet® Fixed Cone Valves

Hydraulic energy dissipation under free discharge



Howell-Bunger® and Ring Jet® Fixed Cone Valves

Hydraulic energy dissipation under free discharge



RODNEY HUNT®

Excellence in Engineering

For over 170 years, the engineering team at Rodney Hunt has pioneered safe and reliable flow control systems in thousands of applications around the world. We have worked with municipalities, utilities, contractors, consulting engineers, and plant operators to meet their flow control needs and solve some of their toughest design, operation, and application problems.

Superior Quality

Rodney Hunt brings exceptional quality to every project with one of the most flexible and comprehensive metal casting, fabrication, machining, assembly, and testing operations in North America. This allows us to monitor and ensure quality in all aspects of production and provide consistent, reliable and superior products. We are also ISO-9001 certified and made in the USA.

Comprehensive Product Offering

Our total product offering is among the most comprehensive in the flow control industry. From all types and sizes of cast and fabricated gates — including Fontaine standard designs — to custom valves, gates, and actuation options, Rodney Hunt brings a total solution to your project. Our capability to design, manufacture, and test large custom valves is unrivaled in the world.

Responsive Service

Rodney Hunt brings not only an incredible wealth of knowledge and expertise to your project, but also a genuine responsiveness to your needs throughout the design, manufacturing, and installation process. From the factory to the field, we offer the most experienced and knowledgeable service team in the industry.

Shown here is the O-ring (Buna-N) being installed on a 42" Ring Jet Valve destined for Bardella in Sao Paulo, Brazil. The valve is powered by an oil hydraulic power unit, designed and built by Rodney Hunt.

On the cover: Eleven Mile Dam, Colorado, USA—5 Ring Jet Valves (8", 16", 30", and two 48"), five Rotovalve® Cone Vales (8", 16", 30", and two 48").

Howell-Bunger® and Ring Jet® Fixed Cone Valves

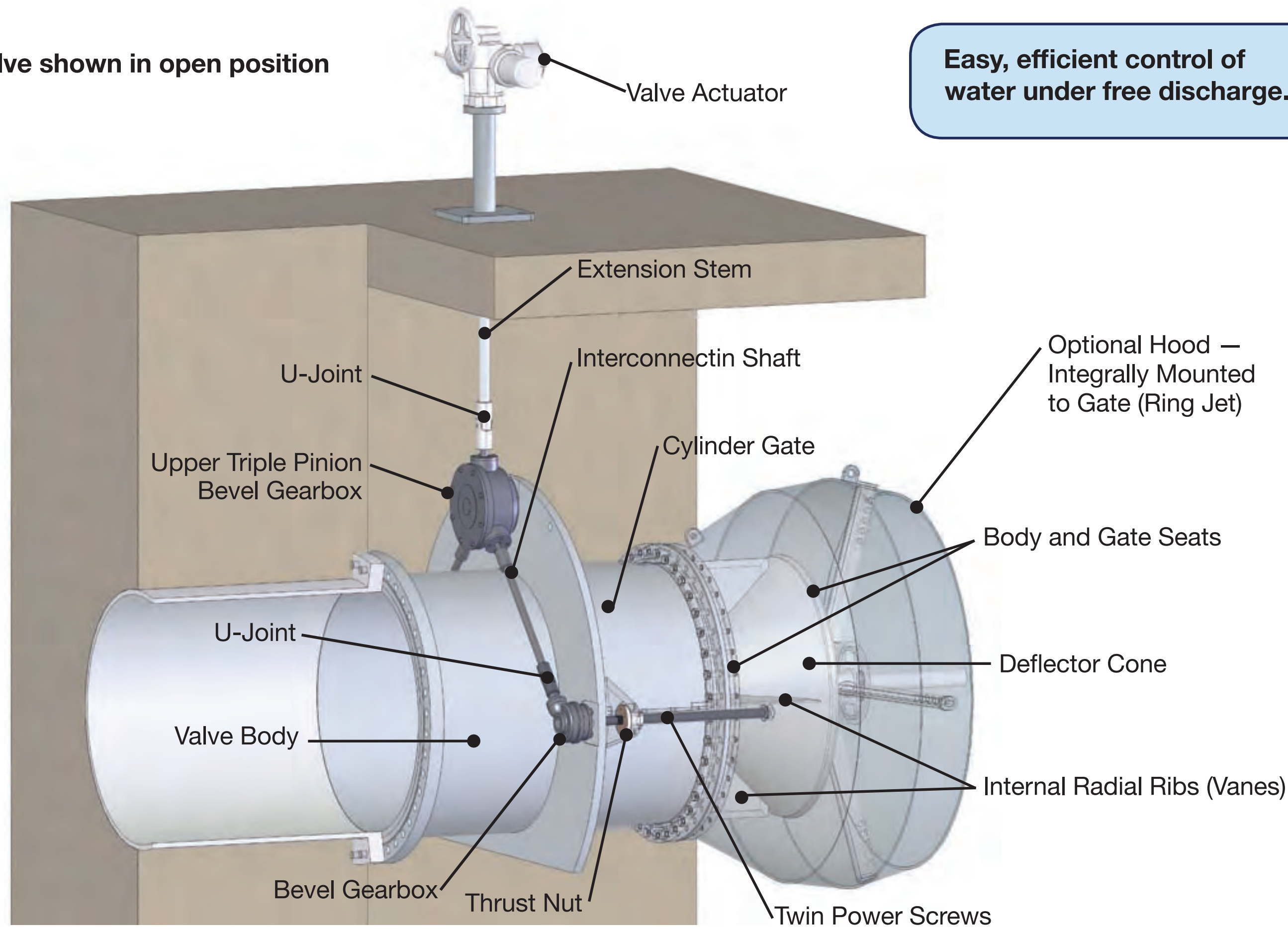
Hydraulic energy dissipation under free discharge

Howell-Bunger and Ring Jet Fixed Cone valves provide controlled discharge of water while protecting the downstream environment. They are ideally suited for low level outlet works for power projects, turbine bypass, flood control systems, irrigation facilities, and draining reservoirs or ponds.

Howell-Bunger valves break up the discharge water into a large, hollow, expanding spray and can be used in most situations, including submerged applications. Ring Jet valves incorporate a steel hood that concentrates the discharge spray into a "jet" and are more frequently used in cold climates.

Easy, efficient control of water under free discharge.

Valve shown in open position



Proven Performance

Howell-Bunger and Ring Jet valves are proven performers in applications requiring control of water under free discharge.

Smooth, Vibration Free Operation

Efficient, free-discharge operation for high and low heads, operating through the entire stroke range without vibration or pitting.

Remote Control Capabilities

The valves can be equipped with remote control devices that open or close the valve to hold a pre-determined level upstream or downstream of the valve.

Stainless Steel Construction

Rugged stainless steel, bronze, and steel construction ensures long life.

Actuation

Adaptable to almost any type of actuator, Rodney Hunt can provide manual, hydraulic, or electric options.

Easy to Operate and Maintain

The cylinder gate that seats against the valve requires little effort to operate and is the only moving part of the assembly in contact with the water flow.

Certified to ISO 9001:2008

Sizing and Dimensions

The size of the valve is determined by the maximum available net head at the valve. Net head is the distance between the head water elevation and the centerline of the valve — or if the valve is submerged, the tail water elevation — less the inlet, conduit, bend or friction losses.

The graph below shows the maximum calculated discharge for valve sizes 8 to 108 inches, based on net heads up to 500 feet.

This graph is based on an average coefficient of discharge of .85. Maximum discharge values for other heads can be determined from the formula:

$Q = C \times \sqrt{2gH} \times A$

where Q = Cubic feet per second (cfs)
C = coefficient of discharge with valve full open = .85
g = acceleration due to gravity = 32.174
H = net head in feet
A = area of valve in square feet (based on nominal inside diameter)

Using a coefficient of discharge of .85, this formula can be expressed as:

$$Q = .85 \times \frac{\pi D^2}{4} \times \sqrt{2gH}$$

Ring Jet Sizing

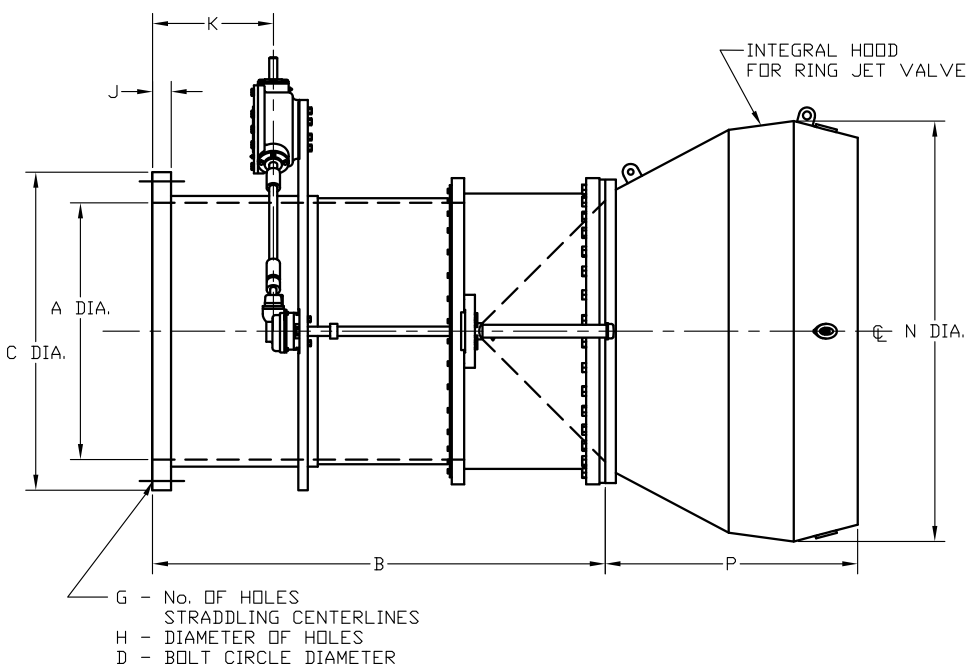
Use the following formula to size Ring Jet Valves (C = .78):

$$Q = .78 \times \frac{\pi D^2}{4} \times \sqrt{2gH}$$

where Q = discharge in cfs
D = diameter in feet

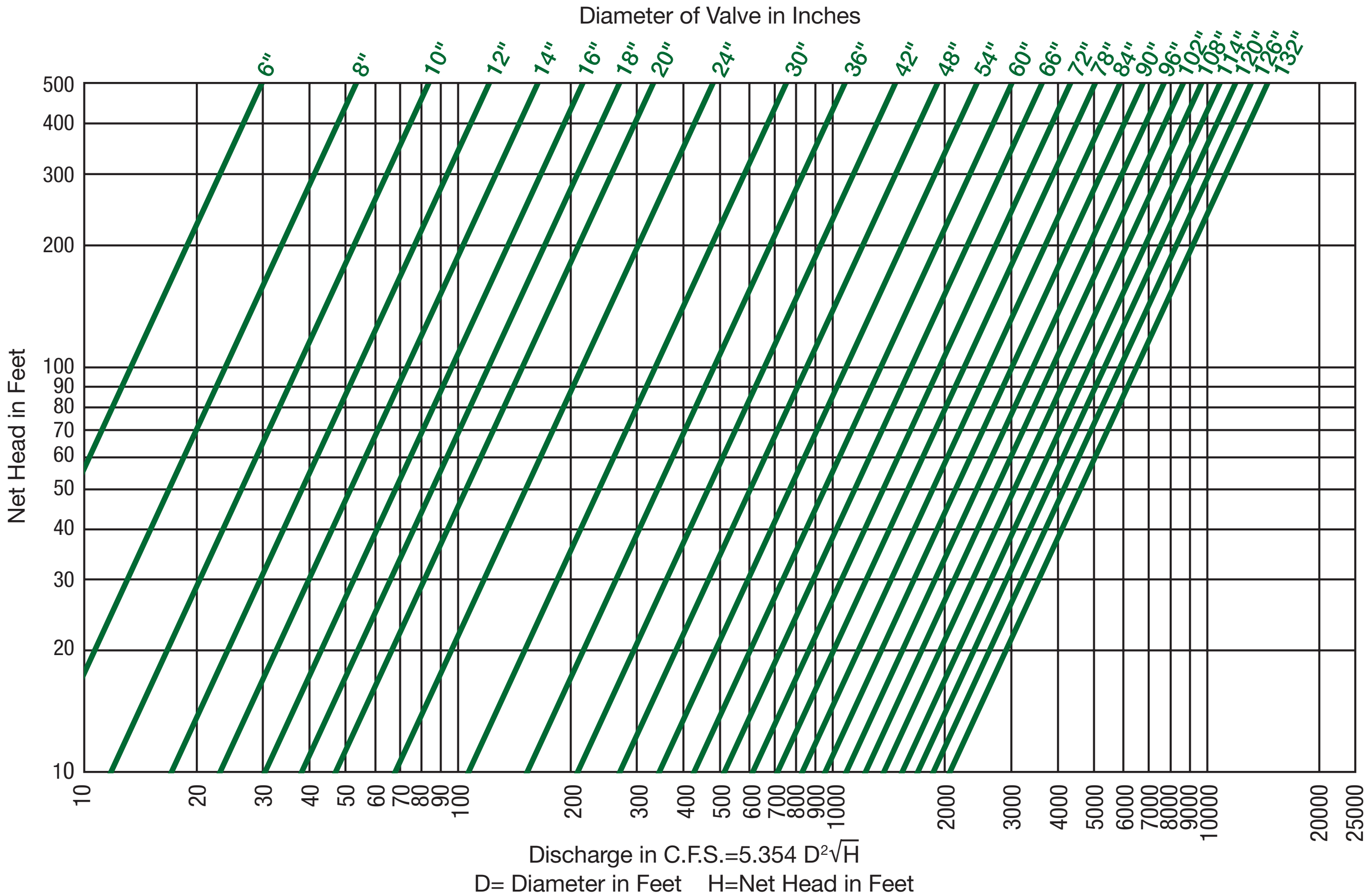
Dimensions								Ring Jet Valve	
A	B	*C	*D	*G	*H	*J	K	N	P
6	28	11.00	9.50	8	0.87	1.31	8	11	6
8	30	13.50	11.75	8	0.87	1.50	8	14	8
10	32	16.00	14.25	12	1.00	1.56	8	18	10
12	38	19.00	17.00	12	1.00	1.75	10	21	12
14	40	21.00	18.75	12	1.12	1.88	10	25	14
16	42	23.50	21.25	16	1.12	2.00	10	29	16
18	48	25.00	22.75	16	1.25	2.13	12	32	18
20	50	27.50	25.00	20	1.25	2.38	12	35	20
24	54	32.00	29.50	20	1.37	2.63	12	42	24
30	64	38.75	36.00	28	1.37	2.88	14	52	30
36	70	46.00	42.75	32	1.62	3.13	14	62	36
42	76	53.00	49.50	36	1.62	3.38	14	72	42
48	86	59.50	56.00	44	1.62	3.50	16	82	48
54	92	66.25	62.75	44	1.87	3.75	16	92	54
60	102	73.00	69.25	52	1.87	3.88	18	102	60
66	108	80.00	76.00	52	1.87	4.25	18	112	66
72	118	86.50	82.50	60	1.87	4.38	20	122	72
78	124	93.00	89.00	64	2.12	4.75	22	129	78
84	134	99.75	95.50	64	2.12	4.75	22	USE SEPARATE HOOD	
90	140	106.50	102.00	68	2.37	5.13	22		
96	150	113.25	108.50	68	2.37	5.13	24		

* C207 CLASS E (275 PSI)



Valve Selection Chart

To determine the discharge of any size Howell-Bunger Valve, follow the horizontal line for a given head (net head at the valve) to the point where it crosses the diagonal line representing valve size. From this point, follow the vertical line to the bottom of the chart, and read the discharge cfs. The chart is specific to the traditional unhooded Howell-Bunger Valve with a C_d of 0.85, and is not suitable for sizing or selecting a Ring Jet Valve.



Valve Operation

Howell-Bunger and Ring Jet Valves are typically operated by a manual, electric, or hydraulic actuator mounted above a triple pinion bevel gearbox. The gearbox transmits torque to the twin power screws (one on each side of valve) which in turn engage bronze thrust nuts to open (retract) or close (extend) the cylinder gate.

Howell-Bunger – In open positions, flow is directed outward around the deflector cone at a 45° angle (approximate) in a wide spray pattern into the atmosphere, dissipating remaining hydraulic energy without erosion of the surrounding area.

Ring Jet – In open positions, flow is directed outward around the deflector cone at a 45° angle (approximate). The wide spray pattern is then redirected in the hood (integrally attached to traveling gate) to create a jet-like stream. This is often helpful where the wide spray pattern of a Howell-Bunger valve is objectionable or subject to freezing.

Eleven Mile Dam, Colorado, USA—Five Ring Jet Valves (8", 16", 30", and two 48") and Five Rotovalve® Cone Vales (8", 16", 30", and two 48").

Projects

Since 1935, over 400 Howell-Bunger and Ring Jet Valves have been installed in applications ranging from an 8" valve with 1400' of head, to a 108" valve with 471' of head. In settings throughout the world, Howell-Bunger and Ring Jet valves have been the valve of choice wherever the control of water flow under free discharge is needed. The photos below show a few of our recent installations.

Williams Fork Dam, Colorado, USA—Three Ring Jet Valves (12", 20", and 36") and three Rotovalve® Cone Valves (16", 24", and 42").

Berlin Lake Dam, Ohio, USA—Two Ring Jet Valves (36") and Eight Ball Valves (36").

Pueblo Dam, Colorado, USA—One Ring Jet Valve (60") and One Rotovalve® Cone Valve (60").

Salt River and Lower Bear Reservoirs, California, USA—Four Howell-Bunger Valves, with stationary hoods (18", 24", 60", 78").

Guarulhos San Paulo, Brazil—One Ring Jet Valve (42").

Eleven Mile Dam, Colorado, USA—View from downstream.

Specifications

Fixed Cone Valve

General: The fixed cone valve(s) will be of the Howell-Bunger type as manufactured by Rodney Hunt Company, Orange, Massachusetts. The valve will be __ inches in diameter and will be designed to discharge __ cfs at __ feet of net head. The valve will be used to control the free discharge of water into the atmosphere and will be designed to operate at any position between fully open and fully closed without damaging vibration.

Design: The construction of the valve(s) will be sufficiently rugged and all parts will be designed for safe and satisfactory operation within the specified operating conditions. Liberal factors of safety will be used throughout, especially in the design of parts subject to intermittent and/or alternating stresses. In general, working stresses will not exceed one-third of the yield strength or one-fifth of the ultimate strength of the material.

Valve Body: The valve body will consist of a cylinder with a conical deflector head on the downstream end, internal radial ribs and an upstream, mounting flange for attachment to a conduit liner or penstock. The internal ribs and deflector head will extend beyond the downstream end of the valve body a sufficient distance to permit the rated discharge capacity. The sealing and sliding surfaces of the valve body will be stainless steel. The mounting flange will be in accordance with AWWA C207 Class "E" and will be provided with an O-ring gasket. The valve body will be constructed of steel plate conforming to ASTM A516 grade 70.

Valve Gate: The valve gate will consist of a cylinder designed to slide over the valve body. The gate will slide upstream to open and downstream to close off the valve ports. The upstream end of the will be counter-bored to receive the body seal. The downstream end will have a stainless steel seat machined to fully contact the valve body seat, The Interior sliding surface of the gate will be bronze. The valve gate will be constructed of steel plate conforming to ASTM A516 grade 70.

Seals: The valve body shall have a removable seat attached to the downstream end of the valve body with bolts and a gasket. The sealing contact surface of the seat shall be stainless steel. The downstream end of the gate shall have a removable seat attached to the gate with bolts and gasket. The sealing contact surface shall be stainless steel and machined to a contour to provide a satisfactory hydraulic profile. The upstream end of the gate shall be counter-bored to receive a U shape packing to seal between the gate and the stainless steel outside the valve body. The U packing shall be retained by a bronze or stainless steel gland and fasteners.

Hood (Ring Jet Valve only): A steel jet deflector hood will be bolted to the downstream end of the cylinder gate. The hood will reduce the discharge spray by confining the exiting water jet. The hood will have several radial internal rib supports coming together at the valve centerline to form a support ring fitted with a self-lubricated sleeve bushing. The rib supports shall be hollow with an opening of sufficient size to provide aeration of the jet. The upstream edge of the ribs shall be contoured to provide proper hydraulic shape. The center support ring shall ride on a stainless steel guide rod attached to the downstream end of the valve body.

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The mechanical screw stem actuating system will consist of two screw stem actuators mounted diametrically opposite and connected to a miter gearbox. Interconnecting shafting shall be stainless steel and shall be connected by flexible couplings. Screw stems shall be type 304 stainless steel and drive nuts shall be bronze.

If the hydraulic cylinders are used, the two cylinders shall be mounted diametrically opposite. Hydraulic valving and plumbing shall be arranged to provide synchronous operation of the two hydraulic cylinders. The hydraulic cylinders shall be of materials and seals suitable for submergence. Piston rods shall be stainless steel and hard chrome plated. Piston seals shall be of the lip seal type. Rods shall be equipped with rod scrapers.

Electric Motor Actuator: The electric motor actuator shall operate from __ volt, __ phase, __ hertz electric power. The electric motor actuator shall be manufactured by Limitorque, EIM, Auma, Rotork, or approval equal.

The actuator will include: electric motor, gearing, limit switches, torque switches, control transformer, reversing starter, overload relays, "open" – "stop" – "close" push-button station, "open" and "close" indicating lights, lockable "local" – "off" – "remote" selector switch, and auxiliary hand wheel. All electrical controls shall be integrally mounted in a NEMA 4 enclosure mounted directly on the valve actuator housing.

The motor shall be specifically designed for valve service, and be of high torque, totally enclosed, non-ventilated construction. The motor shall be of sufficient size to open and close the valve against the maximum differential pressure when the voltage is 10 percent above or below the nominal voltage.

An auxiliary hand wheel shall be provided for manual oper-ation. The hand wheel shall not rotate during electric operation. The maximum hand wheel effort shall not exceed 60 pounds.

Four sets of independently adjustable limit switches shall be provided.

A mechanical dial position indicator shall be provided. A slide wire type, 2-wire transmitter, 4 to 20 mA output potentiometer shall be provided for remote valve position indication.

The motor and control compartments shall have heaters.

Shop Testing: The fully assembled valve shall be hydro-statically tested at a pressure of two times the rated valve pressure for 30 minutes. There shall not be any evidence leakage except at the valve seats.

The fully assembled valve shall be leak tested at the rated pressure for 5 minutes. The allowable leakage through the seats shall not exceed 0.4 ounces per minute per inch of valve diameter. The valve shall be opened and closed three times using the actuating mechanism.

Painting: All unmachined portions of the valve shall be blast cleaned per SSPC-SP 10 (near white) and shall receive two coats of high solids epoxy paint.

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- Jet Flow Valves
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- Streamseal® Circular and Rectangular Butterfly Valves

Actuation

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For more information about Rodney Hunt products or to contact a sales representative, visit the Rodney Hunt website (www.rodneyhunt.com).

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Rodney Hunt is part of the Rexnord Water Management Group (**www.rexnord.com**)

VAG GA INDUSTRIES Fontaine RODNEY HUNT®

RODNEY HUNT®

Excellence in Engineering

For over 170 years, the engineering team at Rodney Hunt has pioneered safe and reliable flow control systems in thousands of applications around the world. We have worked with municipalities, utilities, contractors, consulting engineers, and plant operators to meet their flow control needs and solve some of their toughest design, operation, and application problems.

Superior Quality

Rodney Hunt brings exceptional quality to every project with one of the most flexible and comprehensive metal casting, fabrication, machining, assembly, and testing operations in North America. This allows us to monitor and ensure quality in all aspects of production and provide consistent, reliable and superior products. We are also ISO-9001 certified and made in the USA.

Comprehensive Product Offering

Our total product offering is among the most comprehensive in the flow control industry. From all types and sizes of cast and fabricated gates — including Fontaine standard designs — to custom valves, gates, and actuation options, Rodney Hunt brings a total solution to your project. Our capability to design, manufacture, and test large custom valves is unrivaled in the world.

Responsive Service

Rodney Hunt brings not only an incredible wealth of knowledge and expertise to your project, but also a genuine responsiveness to your needs throughout the design, manufacturing, and installation process. From the factory to the field, we offer the most experienced and knowledgeable service team in the industry.

Shown here is the O-ring (Buna-N) being installed on a 42" Ring Jet Valve destined for Bardella in Sao Paulo, Brazil. The valve is powered by an oil hydraulic power unit, designed and built by Rodney Hunt.

On the cover: Eleven Mile Dam, Colorado, USA—5 Ring Jet Valves (8", 16", 30", and two 48"), five Rotovalve® Cone Vales (8", 16", 30", and two 48").

Howell-Bunger® and Ring Jet® Fixed Cone Valves

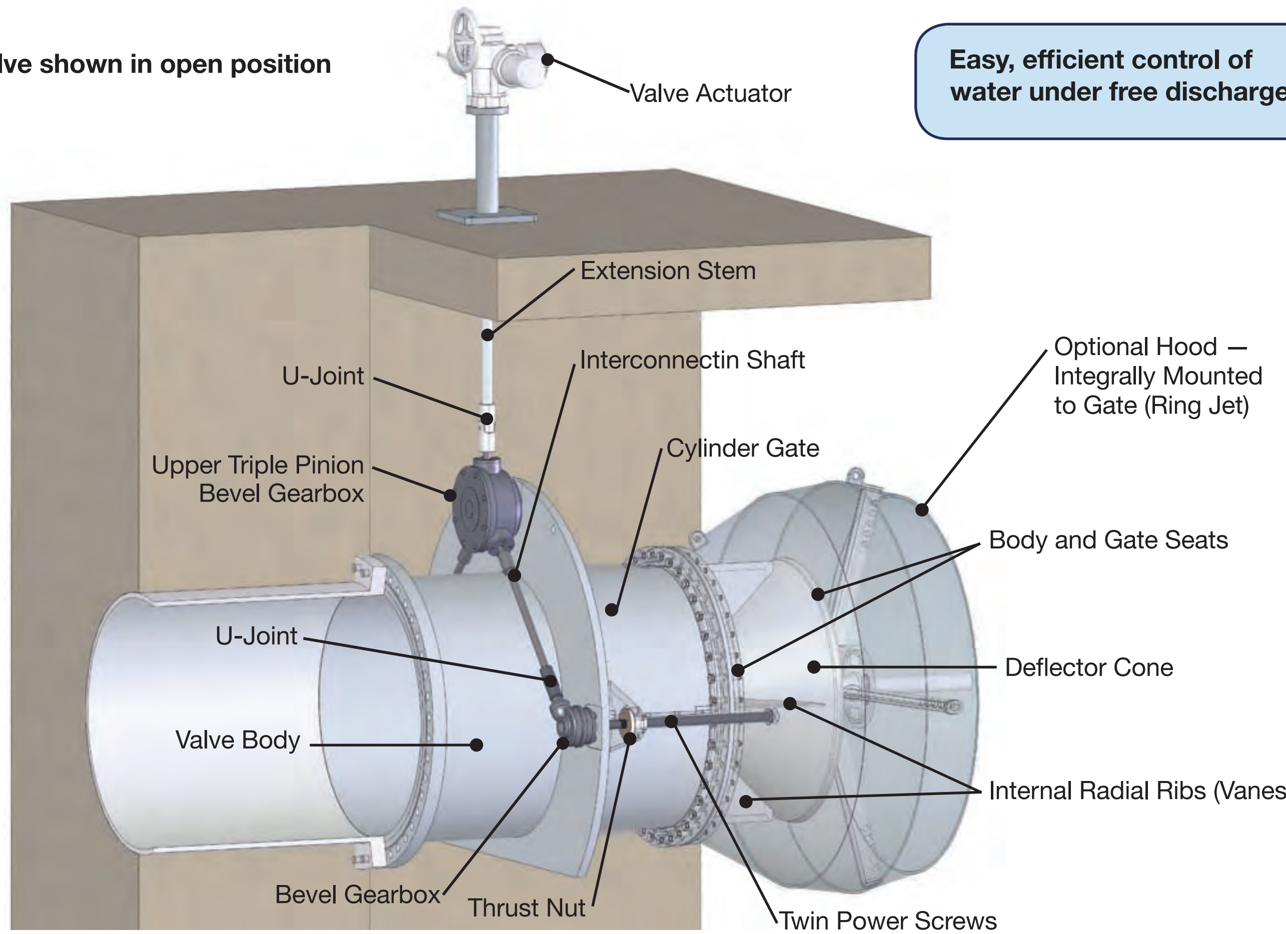
Hydraulic energy dissipation under free discharge

Howell-Bunger and Ring Jet Fixed Cone valves provide controlled discharge of water while protecting the down-stream environment. They are ideally suited for low level outlet works for power projects, turbine bypass, flood control systems, irrigation facilities, and draining reservoirs or ponds.

Howell-Bunger valves break up the discharge water into a large, hollow, expanding spray and can be used in most situations, including submerged applications. Ring Jet valves incorporate a steel hood that concentrates the dis-charge spray into a "jet" and are more frequently used in cold climates.

Valve shown in open position

Easy, efficient control of water under free discharge.



Proven Performance

Howell-Bunger and Ring Jet valves are proven perform-ers in applications requiring control of water under free discharge.

Smooth, Vibration Free Operation

Efficient, free-discharge operation for high and low heads, operating through the entire stroke range without vibration or pitting.

Remote Control Capabilities

The valves can be equipped with remote control devices that open or close the valve to hold a pre-determined level upstream or downstream of the valve.

Stainless Steel Construction

Rugged stainless steel, bronze, and steel construction ensures long life.

Actuation

Adaptable to almost any type of actuator, Rodney Hunt can provide manual, hydraulic, or electric options.

Easy to Operate and Maintain

The cylinder gate that seats against the valve requires little effort to operate and is the only moving part of the assembly in contact with the water flow.

Certified to ISO 9001:2008

Sizing and Dimensions

The size of the valve is determined by the maximum available net head at the valve. Net head is the distance between the head water elevation and the centerline of the valve — or if the valve is submerged, the tail water elevation — less the inlet, conduit, bend or friction losses.

The graph below shows the maximum calculated discharge for valve sizes 8 to 108 inches, based on net heads up to 500 feet.

This graph is based on an average coefficient of discharge of .85. Maximum discharge values for other heads can be determined from the formula:

$$Q = C \times \sqrt{2gH} \times A$$

where Q = Cubic feet per second (cfs)
C = coefficient of discharge with valve full open = .85
g = acceleration due to gravity = 32.174
H = net head in feet
A = area of valve in square feet (based on nominal inside diameter)

Using a coefficient of discharge of .85, this formula can be expressed as:

$$Q = .85 \times \frac{\pi D^2}{4} \times \sqrt{2gH}$$

Ring Jet Sizing

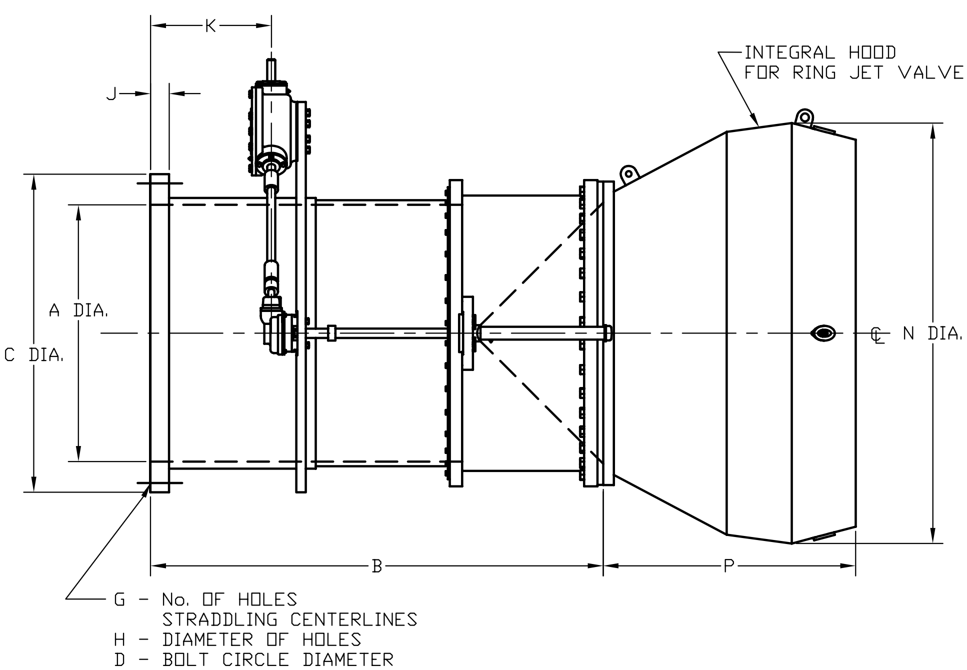
Use the following formula to size Ring Jet Valves (C = .78):

$$Q = .78 \times \frac{\pi D^2}{4} \times \sqrt{2gH}$$

where Q = discharge in cfs
D = diameter in feet

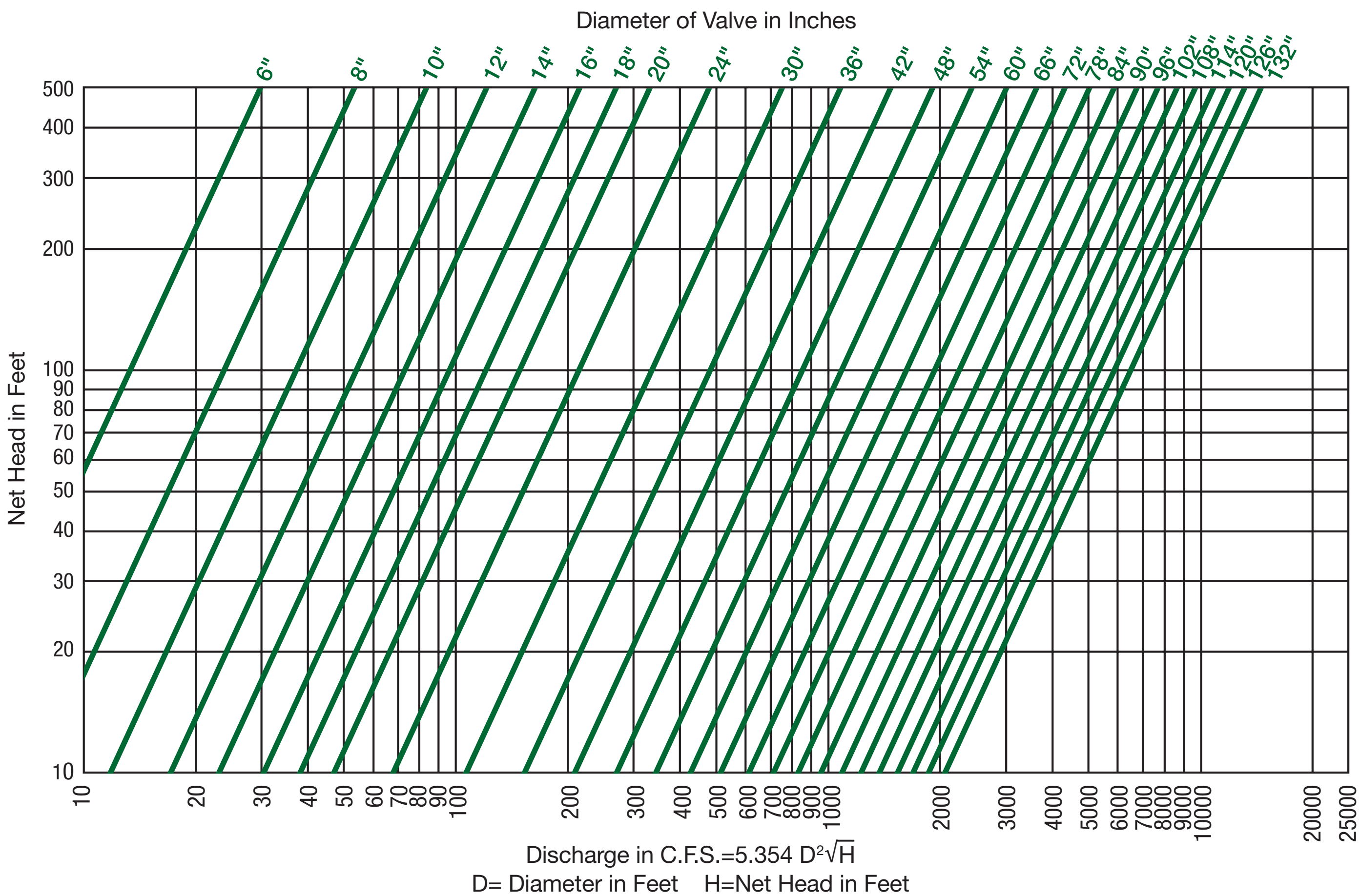
Dimensions								Ring Jet Valve	
A	B	*C	*D	*G	*H	*J	K	N	P
6	28	11.00	9.50	8	0.87	1.31	8	11	6
8	30	13.50	11.75	8	0.87	1.50	8	14	8
10	32	16.00	14.25	12	1.00	1.56	8	18	10
12	38	19.00	17.00	12	1.00	1.75	10	21	12
14	40	21.00	18.75	12	1.12	1.88	10	25	14
16	42	23.50	21.25	16	1.12	2.00	10	29	16
18	48	25.00	22.75	16	1.25	2.13	12	32	18
20	50	27.50	25.00	20	1.25	2.38	12	35	20
24	54	32.00	29.50	20	1.37	2.63	12	42	24
30	64	38.75	36.00	28	1.37	2.88	14	52	30
36	70	46.00	42.75	32	1.62	3.13	14	62	36
42	76	53.00	49.50	36	1.62	3.38	14	72	42
48	86	59.50	56.00	44	1.62	3.50	16	82	48
54	92	66.25	62.75	44	1.87	3.75	16	92	54
60	102	73.00	69.25	52	1.87	3.88	18	102	60
66	108	80.00	76.00	52	1.87	4.25	18	112	66
72	118	86.50	82.50	60	1.87	4.38	20	122	72
78	124	93.00	89.00	64	2.12	4.75	22	129	78
84	134	99.75	95.50	64	2.12	4.75	22	USE SEPARATE HOOD	
90	140	106.50	102.00	68	2.37	5.13	22		
96	150	113.25	108.50	68	2.37	5.13	24		

* C207 CLASS E (275 PSI)



Valve Selection Chart

To determine the discharge of any size Howell-Bunger Valve, follow the horizontal line for a given head (net head at the valve) to the point where it crosses the diagonal line representing valve size. From this point, follow the vertical line to the bottom of the chart, and read the discharge cfs. The chart is specific to the traditional unhooded Howell-Bunger Valve with a C_d of 0.85, and is not suitable for sizing or selecting a Ring Jet Valve.



Valve Operation

Howell-Bunger and Ring Jet Valves are typically operated by a manual, electric, or hydraulic actuator mounted above a triple pinion bevel gearbox. The gearbox transmits torque to the twin power screws (one on each side of valve) which in turn engage bronze thrust nuts to open (retract) or close (extend) the cylinder gate.

Howell-Bunger – In open positions, flow is directed outward around the deflector cone at a 45° angle (approximate) in a wide spray pattern into the atmosphere, dissipating remaining hydraulic energy without erosion of the surrounding area.

Ring Jet – In open positions, flow is directed outward around the deflector cone at a 45° angle (approximate). The wide spray pattern is then redirected in the hood (integrally attached to traveling gate) to create a jet-like stream. This is often helpful where the wide spray pattern of a Howell-Bunger valve is objectionable or subject to freezing.

Eleven Mile Dam, Colorado, USA—Five Ring Jet Valves (8", 16", 30", and two 48") and Five Rotovalve® Cone Vales (8", 16", 30", and two 48").

Projects

Since 1935, over 400 Howell-Bunger and Ring Jet Valves have been installed in applications ranging from an 8” valve with 1400’ of head, to a 108” valve with 471’ of head. In settings throughout the world, Howell-Bunger and Ring Jet valves have been the valve of choice wherever the control of water flow under free discharge is needed. The photos below show a few of our recent installations.

Williams Fork Dam, Colorado, USA—Three Ring Jet Valves (12", 20", and 36") and three Rotovalve® Cone Valves (16", 24", and 42").

Berlin Lake Dam, Ohio, USA—Two Ring Jet Valves (36") and Eight Ball Valves (36").

Pueblo Dam, Colorado, USA—One Ring Jet Valve (60") and One Rotovalve® Cone Valve (60").

Salt River and Lower Bear Reservoirs, California, USA—Four Howell-Bunger Valves, with stationary hoods (18", 24", 60", 78").

Guarulhos San Paulo, Brazil—One Ring Jet Valve (42").

Eleven Mile Dam, Colorado, USA—View from downstream.

Specifications

Fixed Cone Valve

General: The fixed cone valve(s) will be of the Howell-Bunger type as manufactured by Rodney Hunt Company, Orange, Massachusetts. The valve will be __ inches in diameter and will be designed to discharge __ cfs at __ feet of net head. The valve will be used to control the free discharge of water into the atmosphere and will be designed to operate at any position between fully open and fully closed without damaging vibration.

Design: The construction of the valve(s) will be sufficiently rugged and all parts will be designed for safe and satisfactory operation within the specified operating conditions. Liberal factors of safety will be used throughout, especially in the design of parts subject to intermittent and/or alternating stresses. In general, working stresses will not exceed one-third of the yield strength or one-fifth of the ultimate strength of the material.

Valve Body: The valve body will consist of a cylinder with a conical deflector head on the downstream end, internal radial ribs and an upstream, mounting flange for attachment to a conduit liner or penstock. The internal ribs and deflector head will extend beyond the downstream end of the valve body a sufficient distance to permit the rated discharge capacity. The sealing and sliding surfaces of the valve body will be stainless steel. The mounting flange will be in accordance with AWWA C207 Class “E” and will be provided with an O-ring gasket. The valve body will be constructed of steel plate conforming to ASTM A516 grade 70.

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Gates

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- Timber Gates
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- Bulkhead Gates
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The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX C

Structural

The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX D

Electrical

FISH LADDER			**FCQ09**				*FISH AWS*	
CIR #	LOCATION/DESCRIPTION	CB AMPS	VOLT-AMPS OR WATTS	PH		CB AMPS	LOCATION/DESCRIPTION	#
1	HYDRAULIC POWER UNIT, PORTABLE	3P	3878	A	4210	3P	VALVE ACTUATOR, 6'ID (2 EACH)	2
3	" " " " (10 HP)	X	3878	B	4210	X	" " " " (5 HP, 2 EACH)	4
5	" " " "	25	3878	C	4210	20	" " " "	6
7	PANEL, LIGHTING (208/120V/3PH)	3P	5000	A	2105	3P	VALVE ACTUATOR, JET VALVE	8
9	" " " "	X	5000	B	2105	X	" " " " (5 HP, 1 EACH)	10
11	" " " "	20	5000	C	2105	15	" " " "	12
13				A				14
15				B				16
17				C		20		18

PANEL DESCRIPTION:

3 PHASE, 4 WIRE WYE, 277Y/480V
100 AMPS, 18,000 AIC BUS ,100 AMP MAIN
BOLT-TO-BUS BREAKERS, 10,000 AIC
SURFACE MOUNT, NEMA 3R, DOOR-IN-DOOR STYLE REQUIRED
FED BY: (feeder circuit)

CONNECTED LOAD:

PHASE A	55	AMPS
PHASE B	55	AMPS
PHASE C	55	AMPS
ALL	46	KVA

Figure D-1. Electrical Equipment near Fishlock

Figure D-2. Electrical Equipment, Fishlock Approach Channel

Figure D-3. Electrical Equipment, Fishlock Entrance Gate

The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX E

Mechanical

This worksheet is to calculate the force required to rotate gate wheels against friction forces while the gate is under flow.

Variables

$H_1 := 50ft$		Depth of the bottom of the gate below water surface
$Height_g := 14ft$		Height of the gate
$Width_g := 14ft$		Width of the gate
$Num_w := 8$		Number of wheels
$Wheel_{od} := 12in$		Outside diameter of the wheel
$Wheel_{sp} := 9in$		Diameter of the spherical sliding surface of the wheel
$mu_s := .1$		Coefficient of sliding friction of the sliding surface.
$rho_{wat} := 62.4\dfrac{lbf}{ft^3}$		Density of water

Calculations

$A_{gate} := Height_g \cdot Width_g$	$A_{gate} = 196 \cdot ft^2$	Area of the gate
$P0_{gate} := (H_1 - Height_g) \cdot rho_{wat}$	$P0_{gate} = 2.246 \times 10^3 \cdot psf$	Pressure at the top of the gate
$Pb_{gate} := H_1 \cdot rho_{wat}$	$Pb_{gate} = 3.12 \times 10^3 \cdot psf$	Pressure at the bottom of the gate.
$Space_{wh} := \dfrac{Height_g}{\left(\dfrac{Num_w}{2}\right)}$	$Space_{wh} = 3.5 \cdot ft$	Wheel spacing
$Force_b := Pb_{gate} \cdot Width_g \cdot Space_{wh} \cdot .5$	$Force_b = 7.644 \times 10^4 \cdot lbf$	Force on each of the bottom pair of wheels
$M_{frictm} := Force_b \cdot mu_s \cdot \dfrac{Wheel_{sp}}{2}$	$M_{frictm} = 3.44 \times 10^4 \cdot in \cdot lbf$	Max Moment requred to turn each wheel under load

$$F_{wheelm} := \frac{M_{frictm}}{\left(\frac{Wheel_{od}}{2}\right)}$$ $F_{wheelm} = 5.733 \times 10^{3} \cdot lbf$ Max Force applied to wheel OD required to turn wheel

$F_{const} := P0_{gate} \cdot A_{gate}$ $F_{const} = 4.403 \times 10^{5} \cdot lbf$ Total constant force on gate

$$F_{grad} := A_{gate} \cdot \frac{\left(Pb_{gate} - P0_{gate}\right)}{2}$$

$F_{grad} = 8.561 \times 10^{4} \cdot lbf$ Total force on gate due to gradient.

$F_{tot} := F_{const} + F_{grad}$ $F_{tot} = 5.259 \times 10^{5} \cdot lbf$ Total force acting on gate due to water pressure.

$$F_{avg} := \frac{F_{tot}}{Num_{w}}$$ $F_{avg} = 6.574 \times 10^{4} \cdot lbf$ Average force acting on each wheel

$$M_{fricta} := F_{avg} \cdot mu_{s} \cdot \frac{Wheel_{sp}}{2}$$

$M_{fricta} = 2.958 \times 10^{4} \cdot in \cdot lbf$ Avg Moment requred to turr each wheel under load.

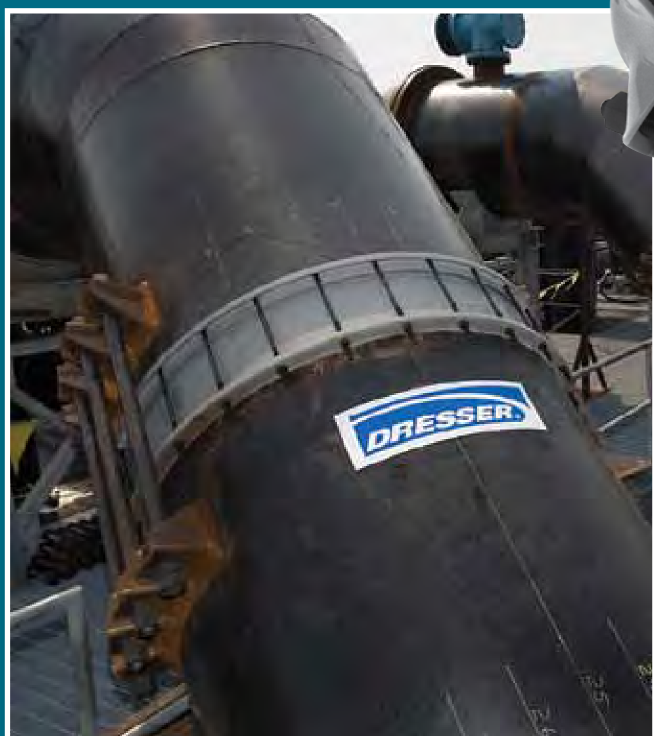
$$F_{wheela} := \frac{M_{fricta}}{\left(\frac{Wheel_{od}}{2}\right)}$$ $F_{wheela} = 4.93 \times 10^{3} \cdot lbf$ Avg Force applied to wheel OD required to turn wheel

$F_{frict\_total} := F_{wheela} \cdot Num_{w}$

$F_{frict\_total} = 3.944 \times 10^{4} \cdot lbf$ Total downward force requried to turn wheels under load.

DRESSER®

Piping Specialties



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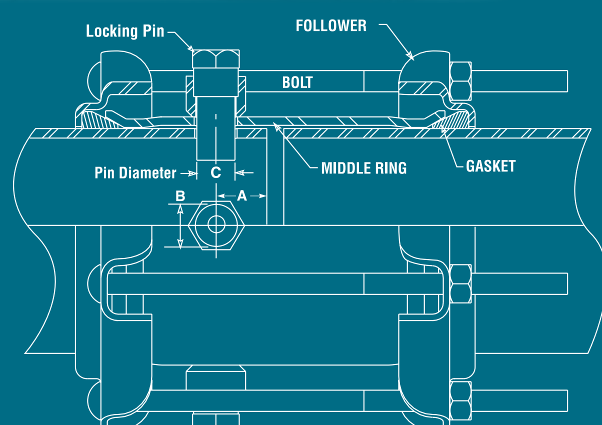
- Expansion Joints
- Dismantling Joints
- Joint Harnesses
- Custom Fabrication



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DRESSER®

Piping Specialties
Bradford, PA

Dresser water market products you'll find in this catalog…

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Sales Fax: 800-362-9363
email: dmdsales@dresser.com

AL-CLAD™ Coating offered as standard

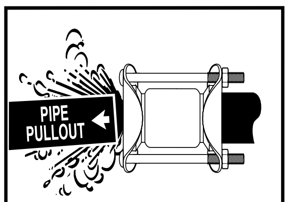
Dresser AL-CLAD fusion-bonded epoxy coating is offered as standard on the most common Dresser pipe joining products in the most popular sizes featured in this catalog.*

Tough, corrosion-resistant, factory-applied Dresser AL-CLAD coating has been developed through years of exhaustive testing and field application.

AL-CLAD epoxy coating is a fusion-bonded coating applied under rigidly controlled factory conditions and offers smoother flow in wetted waterways and provides protection against corrosive or aggressive conditions.

*Excludes Style 63 Expansion Joints where AL-CLAD coating is optional. Please consult factory for other products and sizes where AL-CLAD coating may be optional.

⚠WARNING

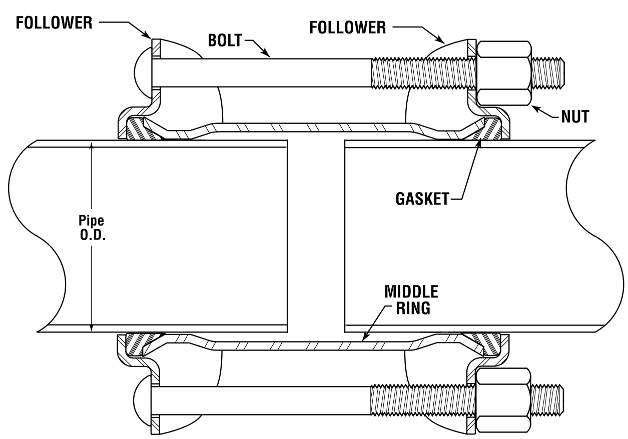


When pipe pullout could occur, pipe joint MUST be anchored. Failure to anchor pipe joint could result in escaping line content that could ignite and cause property damage, serious injury or death.

DRESSER® **Steel Products for Water and Industrial Piping Systems**

Why are DRESSER® couplings used more than any other coupling?

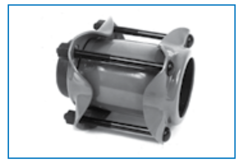
- Dresser offers the broadest line of couplings, including long body, insulating, reducing and transition types.
- Products feature Dresser AL-CLAD™ coating as standard in the most popular sizes. Our epoxy coating offers optimum protection against highly corrosive soil or aggressive water conditions and for handling brine, brackish water, most acids, alkalies, oil, chemical particulates and gases.
- Sizes range from 3/8” through 405” to cover every application including high temperature and abrasion.
- Dresser couplings are fast and easy to install with any size pipe or tubing.
- Wide temperature range from -20°F to +1200°F, with pressure ratings to 1500 psi.
- Available in rugged welded steel construction, stainless or carbon steel, titanium, monel or other alloys for special applications.
- Use a Dresser coupling and your pipeline joint is non-rigid, accepting expansion, contraction, vibration and line deflection.
- Special elastomer formulations are provided custom-matched to specific fluid process or application requirements.

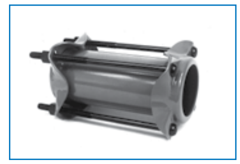


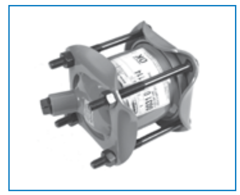
Cutaway view shows components of a basic Dresser Style 38 Coupling

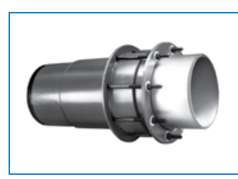
The Basic Working Principle of Dresser Couplings...

The **Dresser** coupling consists of one cylindrical middle ring, two follower rings, two resilient gaskets of special Dresser compound, and a set of steel trackhead bolts. The middle ring has a conical flare at each end to receive the wedge portion of the gaskets. The follower rings confine the outer ends of the gaskets. As the nuts are tightened, the bolts draw the follower rings toward each other, compressing the gaskets in the spaces formed by follower rings, middle ring flares and pipe surface thus producing a flexible, leak-proof seal on the pipe joint.









How to Specify Dresser Products

For those who may wish to draw up specifications of a general nature covering Dresser Style 38 couplings, this suggested form is offered:

1.) The pipe coupling shall be of a gasketed, sleeve-type design with diameter to properly fit the pipe. Each coupling shall consist of one (1) steel middle ring, of thickness and length specified, two (2) steel followers, two (2) rubber-compounded wedge section gaskets and sufficient track-head steel bolts to properly compress the gaskets.

The middle ring and followers of the coupling shall be true circular sections free from irregularities, flat spots or surface defects. They shall be formed from mill sections with the follower-ring section of such design as to provide confinement of the gasket. After welding, they shall be tested by cold expanding a minimum of 1% beyond the yield point. The middle ring, inside and out, and followers shall be coated with AL-CLAD™ thermosetting, fusion-bonded epoxy coating material that provides disbondment resistance in cathodically-protected systems and resistance to soil stresses and fungi. All constituents of the cured film are FDA and NSF-61 approved for exposure to fluids for human consumption and potable water.

The coupling bolts shall be of the elliptic-neck, track-head design with rolled threads. The manufacturer shall supply information as to the recommended torque to which the bolts shall be tightened. All bolt holes in the followers shall be oval for greater strength.

The coupling gaskets shall be composed of a crude or synthetic rubber base compounded with other products to produce a material that will not deteriorate from age, heat, or exposure to air under normal storage conditions. It shall also possess the quality of resilience and ability to resist cold flow of the material so that the joint will remain sealed and tight indefinitely when subjected to shock, vibration, pulsation and temperature or other adjustments of the pipeline.

2.) The couplings shall be assembled on the job in a manner to ensure permanently tight joints under all reasonable conditions of expansion, contraction, shifting and settlement, unavoidable variations in trench gradient, etc. The coupling shall be Dresser Style 38, as manufactured by Dresser Piping Specialties, Bradford, PA, and the necessary quantity shall be furnished.

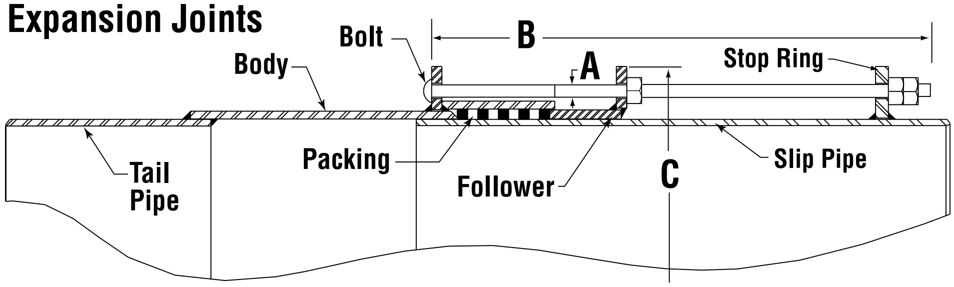
When Ordering Dresser Expansion Joints

Inquiries or orders for Dresser Style 63 Expansion Joints should contain the following information:

- **(1) Quantity**
- **(2) Type of pipe: ductile iron, steel, etc.**
- **(3) Style number and type**
- **(4) Service: Water, Industrial, etc.**
- **(5) Maximum working pressure**
- **(6) Amount of movement to be taken care of by each joint**
- **(7) Temperature limitations and ranges**
- **(8) Frequency of cycling;**
- **(9) End preparation of slip or tail pipe—beveled for welding, flanged, other**
- **(10) Remarks, unusual installations, and list support methods of line and joint**

The proper type of expansion joint to use and the method of anchoring and connecting it into a line depend upon the conditions of service and type of installation, as well as other joints in the line. The most effective use of Style 63 expansion joints usually requires an engineering recommendation. For that reason, a complete description of the installation should be submitted, with sketches or working drawings, if possible. Special joints may also be made for unusual conditions.

Expansion Joints



How to Specify Pipe Ends for Dresser Couplings

How to Specify Ends* on Steel Pipe

On orders and in specifications, the ends on steel pipe to be used with Dresser couplings may be specified briefly as follows:

- The pipe shall be furnished with plain ends for Dresser couplings in accordance with **A.W.W.A.** (American Water Works Association) Steel Water Pipe Specifications;

OR:

- The pipe shall be furnished with plain ends for Dresser couplings in accordance with A.P.I. (American Petroleum Institute) Line Pipe Specifications.

If specifications are to be detailed, the following may be used:

For Pipe Above 5” OD to 10-3/4” OD inclusive:

- The pipe shall be sufficiently free from indentations, projections or roll marks for a distance of 8” from the end of the pipe to make a tight joint with the rubber-gasket type of coupling. The outside diameter of the pipe shall not be more than 1/64” smaller than the nominal outside diameter for a distance of 8” from the end of the pipe and shall permit the passing for a distance of 8” of a ring gauge which has a bore 1/16” larger than the nominal outside diameter of the pipe. The minimum outside pipe diameter shall be determined by the use of a steel tape circumferentially applied to prevent the shipment of undersize, out-of-round pipe which, if measured diametrically through the maximum diameter or checked with a No-Go ring gauge, might appear within the specified tolerance.

For Pipe Larger than 10-3/4” OD:

- The pipe shall be sufficiently free from indentations, projections or roll marks for a distance of 8” from the end of the pipe to make a tight joint with the rubber-gasket type of coupling. The outside diameter of the pipe shall not be more than 1/32” smaller than the nominal outside diameter for a distance of 8” from the end of the pipe and shall permit the passing for a distance of 8” of a ring gauge which has a bore 3/32” larger than the nominal outside diameter of the pipe. The minimum outside pipe diameter shall be determined by the use of a steel tape circumferentially applied to prevent the shipment of undersize, out-of-round pipe which, if measured diametrically through the maximum diameter or checked with a No-Go ring gauge, might appear within the specified tolerance.

*While Dresser couplings require only plain-end pipe, other kinds of pipe ends (such as threaded, beveled or grooved) can be used if such pipe is already on hand.

How to Specify Ends on Cast/Ductile Iron Pipe

On orders and in specifications, the ends on cast or ductile iron pipe to be used with Dresser couplings may be specified briefly as follows:

- The pipe shall be furnished with plain ends for Dresser couplings in accordance with **A.W.W.A.** (American Water Works Association) specifications on tolerances;

OR:

- The pipe shall be furnished with plain ends for Dresser couplings in accordance with A.G.A. (American Gas Association) specifications on tolerances.

If further specifications are desired, the following may be added:

- The pipe shall be smooth and round for a distance of 8” from each end. The maximum plus or minus variation from nominal outside diameters for each size shall not exceed dimensions as shown in chart shown below.
- The maximum outside pipe diameter shall be such as to permit the passing of a ring gauge having an internal bore not greater than .01” larger than the maximum allowable outside diameter of the pipe. This ring gauge shall go over the end of the pipe for a distance of 8” for all sizes up to and including 24” and for a distance of 12” on sizes above 24”.
- The minimum outside diameter shall be determined by use of a steel tape circumferentially applied to prevent the shipment of undersized, out-of-round pipe which, if measured diametrically through the maximum diameter or checked with a No-Go ring gauge, might appear within the specified tolerance.

Size	Maximum Variation
3” - 16”	.06”
18”- 24”	.08”
30”- 42”	.10”
48”	.12”
54”- 60”	.15”

Coupling Deflection, Movement, Expansion and Contraction

Laying out curves with standard Dresser couplings and straight sections of pipe

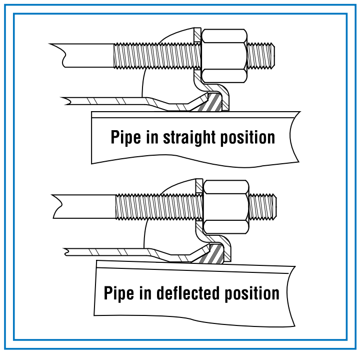
Presented in tabular form in the table at right entitled "Radius of Curve and Deflection of Pipe in Feet", this chart indicates (1) radius of circle for any given degrees of deflection and pipe length, (2) length of pipe for any given radius and deflection or (3) degrees deflection necessary for any given pipe length and radius. This information is worked out for the more commonly used pipe lengths and degrees deflection.

Expansion & Contraction

Each coupling 10" ID and larger will safely accommodate up to 3/8" longitudinal pipe movement. This is equivalent to the amount of movement resulting from a 120° temperature variation in a 40-foot length of steel pipe. If pipe is not buried, anchorage should be provided to prevent excessive accumulation of movement. For repeated movements such as on a bridge or above ground, or if expansion exceeds 3/8" per joint, a Dresser Style 63 expansion joint should be used.

RADIUS OF CURVE AND DEFLECTION OF PIPE IN FEET

Length of Pipe Sec. (feet)	Radius of Curve (Feet) Varying degrees deflection in each coupling						Deflection of Pipe (Feet/Inches) Varying degrees deflection in each coupling					
	1°	2°	3°	4°	5°	6°	1°	2°	3°	4°	5°	6°
6	344	172	115	84	66	57	1/4"	2-1/2"	3-3/4"	5"	6-1/4"	7-1/2"
12	687	344	229	172	138	114	2-1/2	5	7-1/2	10	1' 5/8	1' 3
16	916	458	306	229	183	153	3-3/8	6-3/4	10	1' 1-1/2	1' 4-3/4	1' 8
18	1031	516	344	258	206	172	3-3/8	7-1/2	1'1-1/4	1' 3-1/8	1' 6-7/8	1'10-1/2
20	1145	573	382	286	229	191	4-1/4	8-3/8	1' 5/8	1' 4-3/4	1' 8-7/8	2' 1
30	1718	860	573	430	344	286	6-1/4	1' 5/8	1' 6-7/8	2' 1	2' 7-7/8	3' 1-5/8
40	2291	1146	764	573	458	382	8-3/8	1' 4-3/4	2' 1	2' 9-1/2	3' 5-7/8	4' 2-1/8



Maximum Recommended Laying Deflection Dresser Style 38 Couplings

From 3/8" ID to 2" ID Inclusive..................6°
From 2" ID to 14" OD Inclusive..................4°

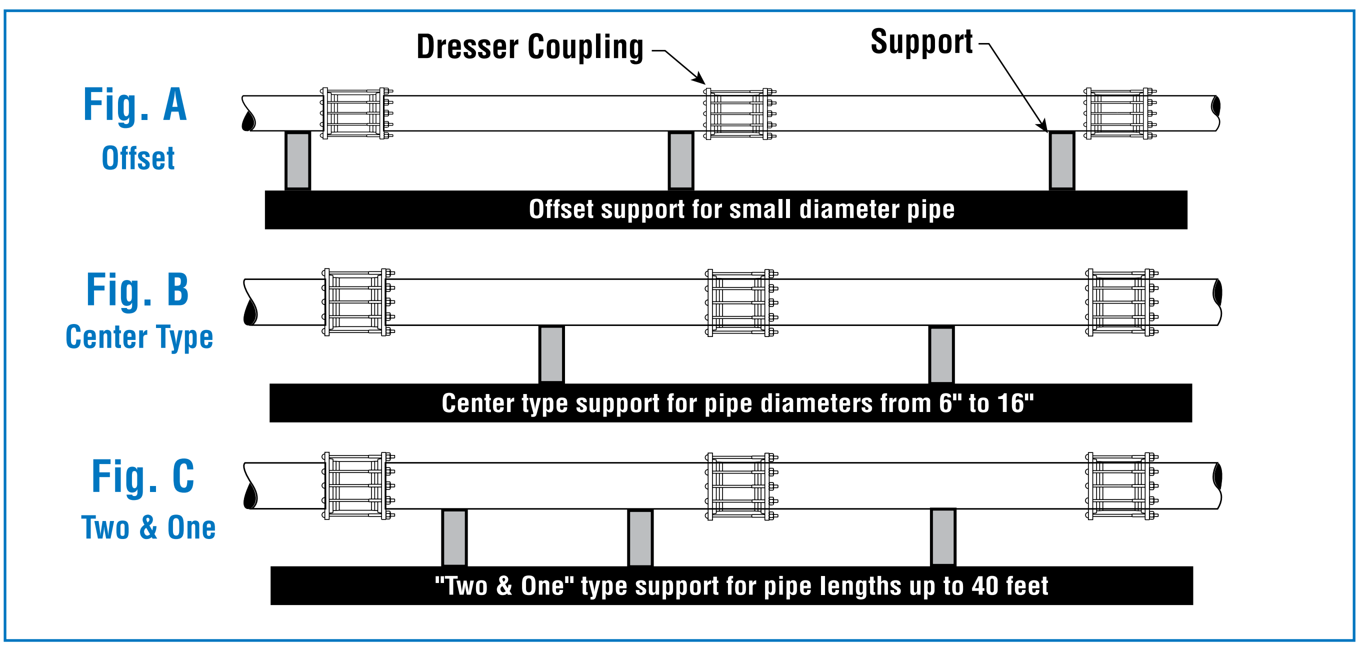
With Middle Ring Lengths:	5"	7"	10"
14" OD - 20" OD Inclusive	2-1/2°	4°	4°
20" OD - 30" OD Inclusive	2°	4°	4°
30" OD - 37" OD Inclusive	1-1/2°	3°	3-1/2°
37" OD - 42" OD Inclusive		2-1/2°	3-1/2°
42" OD - 49" OD Inclusive		2°	3°
49" OD - 54" OD Inclusive		2°	3°
54" OD - 66" OD Inclusive		2°	2-1/2°
66" OD - 78" OD Inclusive			2°
78" OD - 90" OD Inclusive			1-1/2°

Methods of Supporting Coupled Lines

Shown below are three options for supporting pipeline connections when using Dresser couplings. **Figure A** shows the offset method near the pipe joint for diameters 6" and smaller with pipe lengths up to 20 feet. Suitable for any pressure providing pipe is anchored to support for high pressure. **Figure B** indicates the center-type support for diameters from 6" to 16" and lengths not over 20 feet.

This method is suitable for pressures up to 25 lb. maximum with pipe fully anchored to supports.

Figure C shows the "Two & One" method for all sizes and any length of pipe up to 40 feet. Suitable for any pressure providing pipe is adequately anchored. When utilizing this method each length of pipe must be anchored to one (and ONLY one) support.



For absorbing concentrated pipe movement

NOTE:
See Page 2 for Style 63 ordering information

Style 63 Expansion Joints

Dresser offers the broadest line of **Style 63 Expansion Joints** including single-end (Type 1 and Type 3 shown below), and double-end (Type 2 & 4), limited-movement types, flanged, lock coupled, or weld ends. Aggressive wear and pipe wall failure caused by fatigue of the convoluted surfaces present in rubber accordion or metal bellows types is eliminated with Dresser expansion joints. There is no need for expensive pipe loop systems.

Dresser expansion joints are built to order and are available up to 120” in diameter. Provided with rugged welded steel construction, the Style 63 is available in stainless or carbon steel, monel or other alloys for special applications. Single-end expansion joints permit up to 10” of concentrated pipe movement. Larger amounts of movement are available per application.

Special packing and lubrication requirements are custom-matched to specific fluid processes or application requirements. Temperature ratings to 800°F and pressure ratings to 1200 psi.

Available with Dresser AL-CLAD™ coating for optimum protection against aggressive water conditions and for handling brine, brackish water, coke oven gas, petroleum and other line content.

Materials of Construction

Body: AISI C1006, C1010, C1015, C1025 or ASTM A513 Carbon Steel

Follower: AISI C1012, C1021, ASTM A20 or A36 Carbon Steel

Slip Pipe: Chrome plated

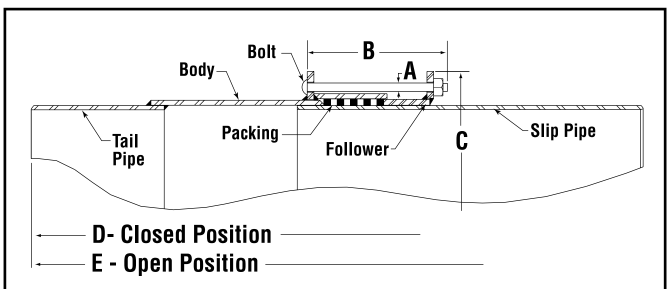
Tail Pipe: AISI C1006, C1010, C1015, C1025 or ASTM A513 Carbon Steel

Bolts & Nuts: ANSI/AWWA C111/ANSI A21.11

Packing: Standard packing is alternate rings of Buna-S and lubricating split jute

Style 63 Type 1 Sizes and Specifications

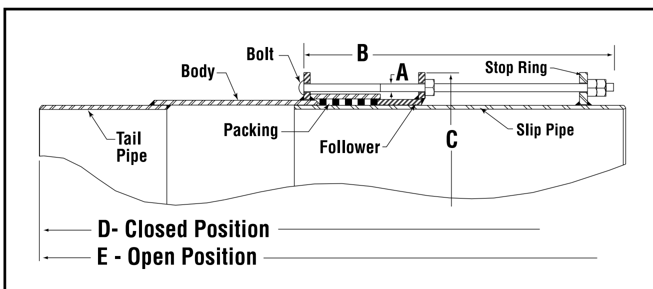
Pipe Nominal Size (In)	Outside Diameter (OD)	Bolts No./Diam. x Length (A&B)	Overall Dimensions Diam. (C)	Overall Dimensions Length (D) (E)	Weight Per Joint (Lbs)
3	3.500	4–5/8 x 11	8-1/2	CONSULT FACTORY PER ORDER	65
4	4.500	4–5/8 x 11	9-1/2		75
5	5.563	4–5/8 x 11	10-5/8		110
6	6.625	6–5/8 x 11	11-3/4		130
8	8.625	6–5/8 x 11	13-3/4		180
10	10.750	8–5/8 x 11	15-7/8		250
12	12.750	8–5/8 x 11	17-7/8		315
	14.000	8–5/8 x 11	19-1/2		340
	16.000	10–5/8 x 11	21-1/2		380
	18.000	10–5/8 x 11	23-1/2		415
	20.000	12–5/8 x 11	25-1/2		470
	22.000	14–5/8 x 11	27-1/2		525
	24.000	14–5/8 x 11	29-1/2		565



Type 1 is a single-end expansion joint permitting up to 10” of concentrated pipe movement. Standard packing consists of alternate layers of split resilient sealing rings and jute lubricating rings. Other packing for special conditions can be supplied.

Style 63 Type 3 Sizes and Specifications

Pipe Nominal Size (In)	Outside Diameter (OD)	Bolts No./Diam. x Length (A&B)	Overall Dimensions Diam. (C)	Overall Dimensions Length (D) (E)	Weight Per Joint (Lbs)
3	3.500	4–5/8 x 24	8-1/2	CONSULT FACTORY PER ORDER	80
4	4.500	4–5/8 x 24	9-1/2		90
5	5.563	4–5/8 x 24	10-5/8		125
6	6.625	6–5/8 x 24	11-3/4		155
8	8.625	6–5/8 x 24	13-3/4		205
10	10.750	8–5/8 x 24	15-7/8		285
12	12.750	8–5/8 x 24	17-7/8		350
	14.000	8–5/8 x 24	19-1/2		385
	16.000	10–5/8 x 24	21-1/2		430
	18.000	10–5/8 x 24	23-1/2		470
	20.000	12–5/8 x 24	25-1/2		530
	22.000	14–5/8 x 24	27-1/2		590
	24.000	14–5/8 x 24	29-1/2		635



Type 3 is a single-end expansion joint equipped with a limited movement feature to limit the maximum amount of pipe withdrawal. Slip pipes are regularly furnished for Type 3 expansion joints.

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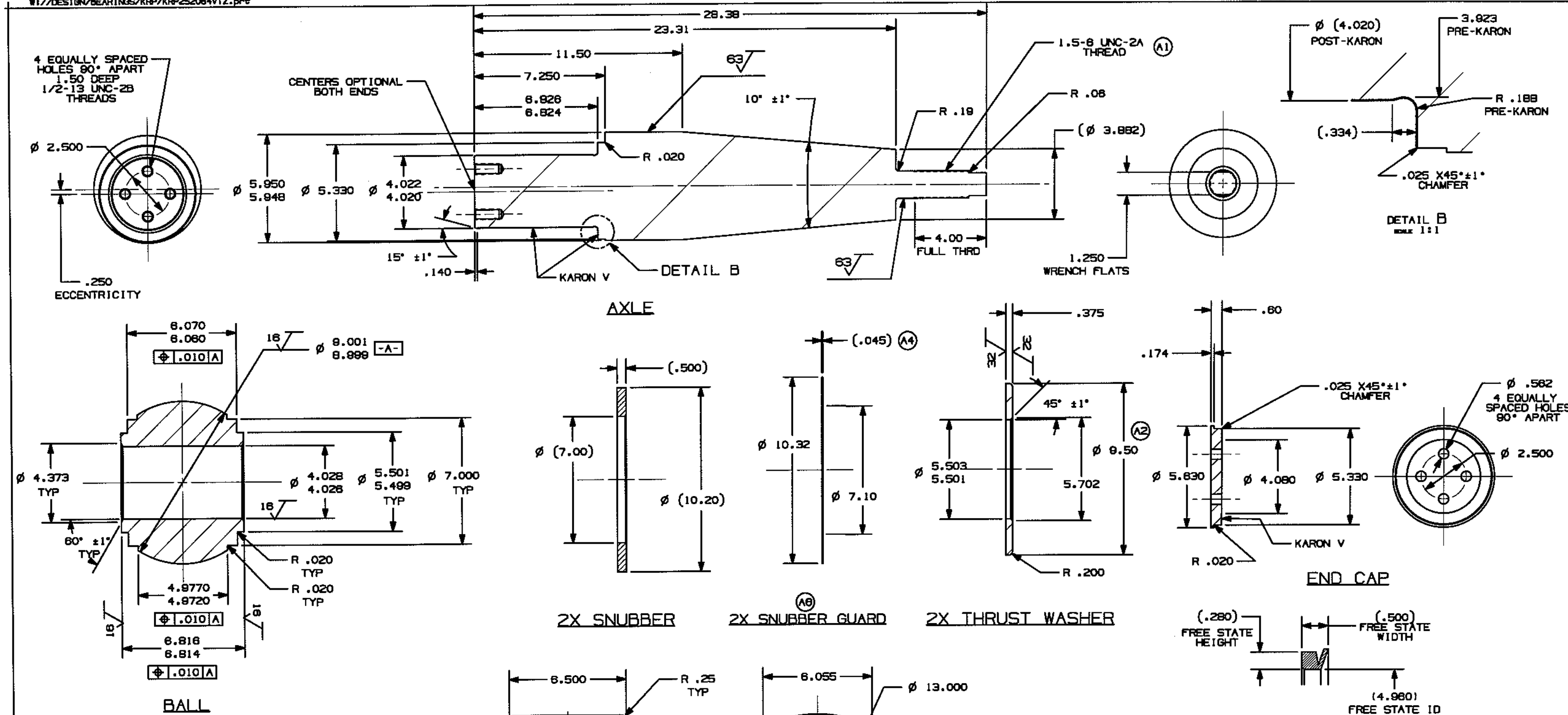
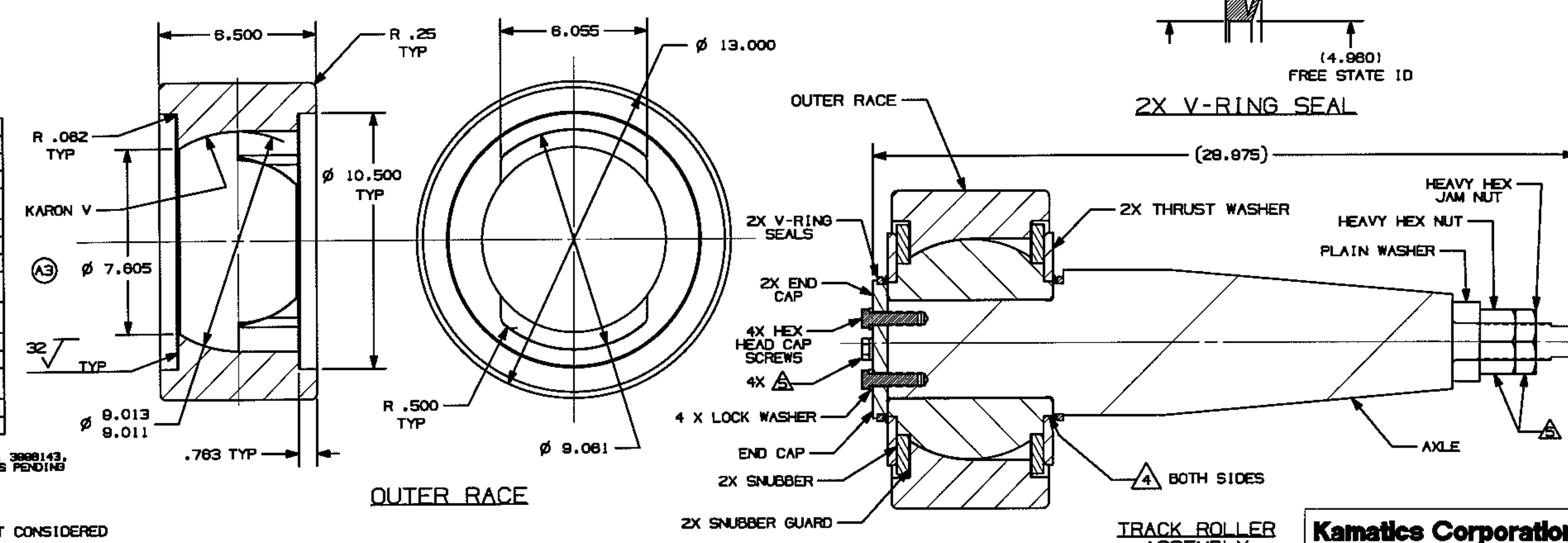
KRP252064VTZ REV A



MATERAIALS	
OUTER RACE	17-4 CRES (UNS S17400), RC 38 MIN
BALL	17-4 CRES (UNS S17400), RC 40 MIN
THRUST WASHERS	17-4 CRES (UNS S17400), RC 40 MIN
AXLE	CRES PER ASTM A 564, UNS S17400, TYPE 630, CONDITION H1150
(A7) SNUBBER	ELASTOMER (NITRILE/BUNA N OR EQ.)
(A5) SNUBBER GUARD	P54 WEAR STRIP
V-RING SEALS	NITRILE (GENERAL PURPOSE SEAL)
END CAP	17-4 STAINLESS STEEL (UNS S17400) HEAT TREATED TO RC 30 MIN
HEX HEAD CAP SCREWS	18-8 OR 300 SERIES STAINLESS STEEL (2-INCH LONG)
LOCK WASHER	18-8 OR 300 SERIES STAINLESS STEEL
HEAVY HEX NUT	410 STAINLESS STEEL
(A9) HEAVY HEX JAM NUT	410 STAINLESS STEEL
PLAIN WASHER	18-8 OR 300 SERIES STAINLESS STEEL
LINER	KARON V (.010/.015 THK)

U.S. PATENT NO. 3806218, 3904731, 3929398, 3998143, 4033019, 4048370, 4053665, 4080287, OTHERS PENDING



NOTES:
1. RADIAL STATIC LIMIT LOAD = 1,000,000 LB (BASED ON KARON COMPONENT ONLY, AXLE BENDING NOT CONSIDERED
2. RADIAL DYNAMIC LOAD RATING = 365,500 LB (BASED ON KARON COMPONENT ONLY, AXLE BENDING NOT CONSIDERED)
3. BEARING MISALIGNMENT = ±2.5°
4. WELD THRUST WASHER TO BALL BOTH SIDES AROUND ENTIRE CIRC (.060 MIN DEEP)
5. ALL MECHANICAL FASTENERS SHALL BE INSTALLED WITH ANTI SEIZE COMPOUND (A8) (SCREWS, NUTS, THREADS, ETC...)

32.5427.01

REVISIONS	
A	1. ADDED THREAD CALL OUT 2. WAS 9.75 IS 9.50 3. ADDED SPHERICAL "WINDOW" DIA - 7.605 4. ADDED REF SNUBBER GUARD THK 5. ADDED SNUBBER GUARD TO BOM 6. RENAMED P54 WASHER TO SNUBBER GUARD 7. ADDED ELASTOMER CALLOUT TO SNUBBER 8. ADDED NOTE 5 "ANTI SEIZE COMPOUND REQ" 9. ADDED LOCK WASHER, HEAVY HEX NUT, HEAVY HEX JAM NUT, AND PLAIN WASHER TO BOM TPR 07/13/07 7/19/07

TOLERANCE
UNLESS OTHERWISE SPECIFIED
All dimensions in inches
Break sharp edges; .015 max.
Single place decimals ± .125
Two place decimals ± .030
Three place decimals ± .010
Angles ± 2° (degrees)
Surface roughness; 125
per MIL-STD-10

Sheet 1 OF 1		
PLOT SCALE: NA		
DWN	TPR	7/02/07
APPVD		

Kamatics Corporation
KAMAN
MANUFACTURING ENGINEERING
TOOLING DRAWING

JOHN DAY TEMPORARY SPILLWAY WEIR GATE ROLLER

KRP252064VTZ REV. A

CORPS OF ENGINEERS 02-03

The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX F

Cost Estimates

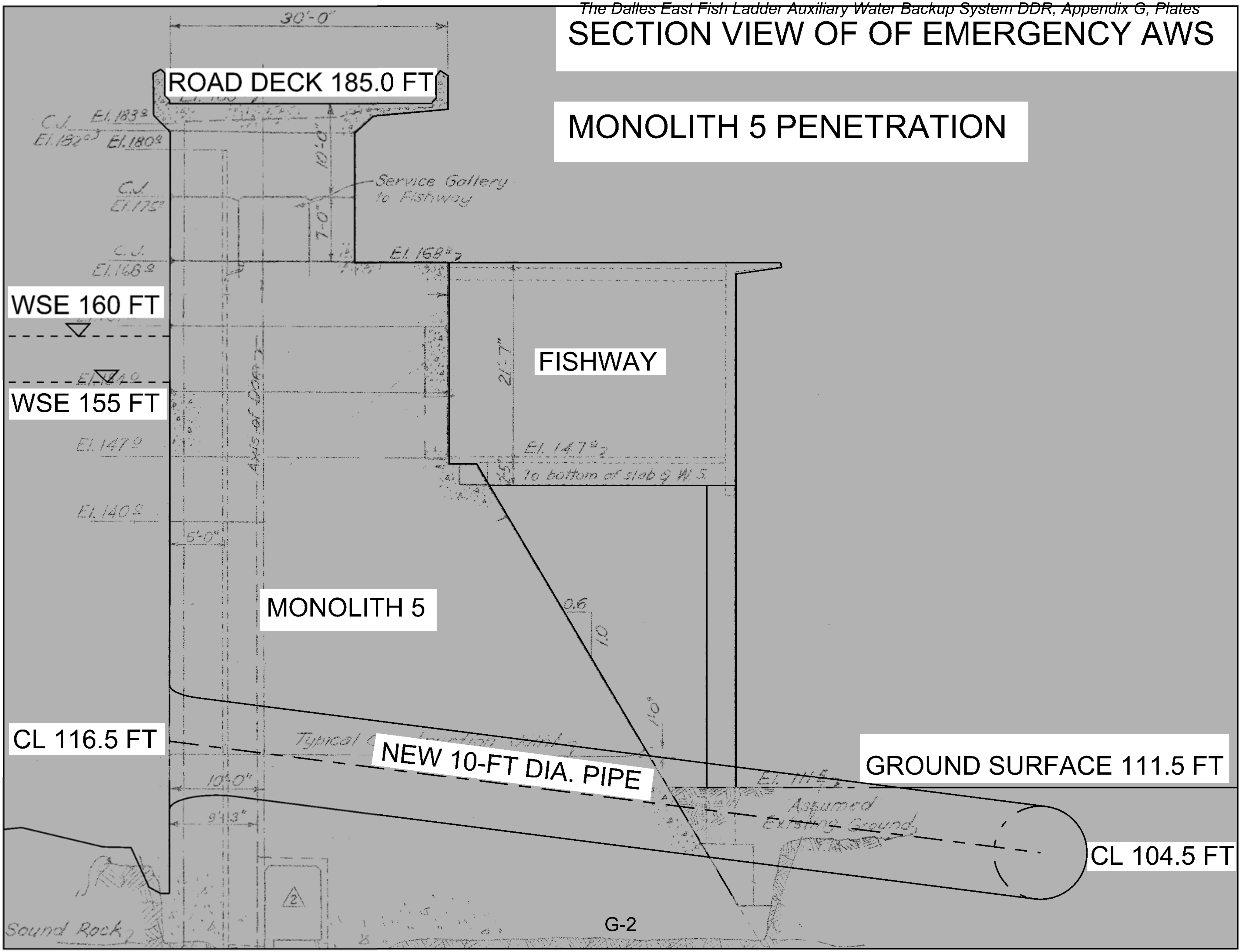
The Dalles East Fish Ladder Auxiliary Water Backup System

30 Percent Documentation Design Report

APPENDIX G

Plates

PLAN VIEW OF EMERGENCY AWS


SECTION VIEW OF OF EMERGENCY AWS
MONOLITH 5 PENETRATION
30'-0"
ROAD DECK 185.0 FT
C.J.
El. 183.0
El. 182.0
El. 180.0
10'-0"
C.J.
El. 175.0
Service Gallery to Fishway
7'-0"
C.J.
El. 168.0
El. 168.0
WSE 160 FT
WSE 155 FT
FISHWAY
21'-7"
El. 147.0
El. 147.0
To bottom of slab & W.S.
El. 140.0
5'-0"
Axis of Dam
MONOLITH 5
0.6
1.0
CL 116.5 FT
NEW 10-FT DIA. PIPE
GROUND SURFACE 111.5 FT
10'-0"
9'-3"
Assumed Existing Ground
CL 104.5 FT
Sound Rock