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HYDRAULIC TESTS AND DEVELOPMENT OF MULTIJET SLEEVE VALVES

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16. ABSTRACT

Hydraulic laboratory studies were conducted to test a 200-mm inline polyjet valve and to develop and test a 200-mm horizontal multijet sleeve valve and stilling chamber. The study results demonstrate the capability of the horizontal multijet sleeve valve and stilling chamber to perform well as an energy dissipator and also deliver design discharges with minimum head loss. A computer program was developed to size and locate the nozzles and slots used in the ported sleeve of the multijet sleeve valve. This multijet concept of valve control which results in the production of cavitation in the water of the stilling chamber, away from structural members, permits design consideration of high head (100 to 300 m), one-stage flow control installations.

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HYDRAULIC TESTS AND DEVELOPMENT OF MULTIJET SLEEVE VALVES

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December 1977

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UNITED STATES DEPARTMENT OF THE INTERIOR * BUREAU OF RECLAMATION

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Messrs R. E. Thibault, Water Conveyance Branch, and E. O. Green, Mechanical Branch, have followed the progress of this study and offered many helpful suggestions throughout the course of the investigation. R. E. Thibault developed the computer program included as the appendix. The 200-mm polyjet valve was loaned for test purposes by the Chas M. Bailey Co., Inc., Emeryville, Calif.

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PURPOSE

The USBR (Bureau of Reclamation) has been involved in the design and construction of irrigation and hydroelectric power systems since its inception in 1902. In recent years, there has been an increase in the design and construction of high head municipal and industrial water supply systems. These new systems often have long aqueducts which require regulating valves and energy dissipators that will safely deliver high energy flows over a range of discharges. The purpose of the research program reported herein was to develop a regulating valve and associated energy dissipator which will efficiently perform the function of flow control.

CONCLUSIONS

The 200-mm polyjet valve tested in the laboratory performed well as an inline energy dissipation valve. Evidence of paint removal due to cavitation was noted when the valve was operated in a severe cavitation range. The paint removal appeared to be related to the quantity of vapor bubbles produced (percent valve opening) and the cavitation index σ. In general, the polyjet valve performed quite well when it was operated within the manufacturer's suggested head ranges.

The 200-mm horizontal multijet sleeve valve demonstrated in laboratory tests its capability to perform as an energy dissipation device and also to deliver design flows with a minimum of head loss. The computer program developed for the study effectively analyzes the valve flow characteristics and locates the multijet ports to produce a nearly linear relationship between control sleeve travel and valve discharge.

Of the two valve sleeves tested, the ported sleeve with nozzles and slots provided valve characteristics similar to those desired and was selected as the best valve design for municipal and industrial water supply aqueduct systems.

The use of horizontal multijet sleeve valves on municipal and industrial water supply aqueducts can save hundreds of thousands of dollars with the elimination of each conventionally designed large flow control station.

APPLICATION

Several types of multijet sleeve valves have been studied by others with recommendations made regarding their use. The development of the horizontal multijet sleeve valve and energy dissipator specifically meets the need expressed by the USBR's Division of Design for a regulating valve and energy dissipator capable of controlling high energy aqueduct flows. The first horizontal multijet sleeve valve has been installed on the Frederick Aqueduct, Mountain Park Project, Okla.

INTRODUCTION

With the design and construction of long aqueducts to supply municipal and industrial water, the need arose to develop a valve and energy dissipator system which would be compatible with the aqueduct system. A system was needed that would (1) adequately dissipate high energy flow at small discharges, and (2) pass design flows with a minimum of energy loss. The aqueduct would dissipate the majority of available energy in line losses when operating under design flow conditions. When operating under a throttled condition, the aqueduct would dissipate the majority of excess energy at the control valve and energy dissipator. Thus, the valve would serve two functions: (1) pressure reduction at small flows, and (2) delivery of design flows with minimal head loss.

Flow control stations on long aqueducts have been limited to pressure head differentials of approximately 15 m. This head restriction resulted from the cavitation damage associated with the use of butterfly valves at the higher head differentials. If a valve and associated energy dissipator could be developed to accommodate pressure head differentials exceeding 15 m, the number of flow control stations required along an aqueduct could be reduced, resulting in significant cost savings.

With these needs in mind, the Divisions of General Research and Design at the Bureau of Reclamation's Engineering and Research Center, Denver, initiated a research program in 1972 to develop a suitable control valve and energy dissipator.

PREVIOUS INVESTIGATIONS

Previous studies conducted by the author[1][1] on a 50-mm model of a multiport sleeve valve in a vertical stilling well indicated that such a valve could fulfill the control requirements stated previously. In a study reported by Miller[2], Glenfield and Kennedy, Ltd., a submerged discharge valve (sleeve-type) similar to the Wanship sleeve valve[3] was described. Glenfield and Kennedy developed a new device which could be attached to their standard sleeve valve resulting in a ported sleeve valve. The advantage of the ported sleeve is that port shape and area can be designed to accommodate the specific hydraulic characteristics desired.

[1] Numbers in brackets refer to items in the bibliography.

Miller also noted an improvement in energy dissipation resulting from the small individual jets leaving the valve. The MWD (Metropolitan Water District) of Southern California[4] conducted tests on an improved submerged discharge valve (without ports), but found that at heads in excess of 30 m, cavitation damage occurred on the bottom plate of the valve and edges of the control sleeve. To develop a valve which could control high energy flows, the concept of flow nozzles was used. MWD engineers developed an outer sleeve containing a large number of small nozzles and attached it to the sleeve valve under study. The nozzles accelerate the flow through the outer sleeve, thus, no cavitation can occur in the metal flow passages. As the jets exit the valve, cavitation occurs in the water surrounding the valve and not against the flow surfaces.

The multijet concept of valve control has permitted designers to consider controlling high head flows, ranging from 150 to 300 m, at one installation. Previously, such high head flow was controlled using several steps of energy reduction to prevent cavitation damage of the control valves.

HYDRAULIC CONSIDERATIONS

Two appealing flow characteristics of a valve using the multijet concept are:

1. The flow energy is dissipated quite rapidly upon leaving the valve, thus requiring a relatively small energy dissipation structure.

2. The inevitable process of formation and collapse of cavitation bubbles, which occurs during throttling of high energy flow, can be controlled to occur in the water surrounding the valve and not against the flow surfaces of the valve or energy dissipation chamber.

Jet velocity deceleration rate is related to port size. The location of the "cavitation bubble collapse zone" is related to the multijet exit port shape and its relative proximity to other ports. Figure 1 illustrates the flow pattern developed by a submerged jet[5]. Albertson[6] and Yevdjevich[7] discussed the characteristics of a submerged jet. Albertson described the phenomena as two stages: zone of flow establishment and zone of established flow. In the zone of flow establishment, the core of the submerged jet is penetrated by viscous shear until the centerline velocity begins to decrease. Thus, the fluid in the jet decelerates while the fluid surrounding the jet gradually accelerates. The zone of established flow is defined as that zone where the entire jet becomes turbulent and the centerline velocity begins

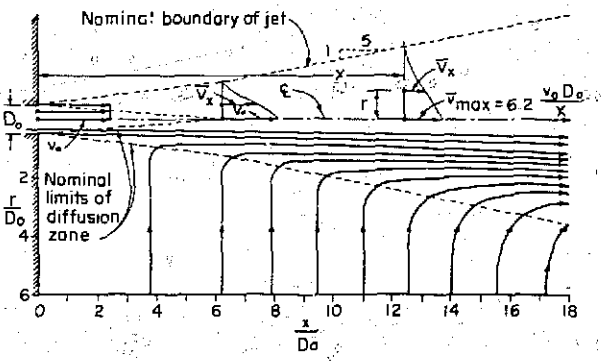


Figure 1.—Submerged jet flow patterns.

to decelerate. The centerline velocity V_m is defined by Albertson as:

$$V_m = 2.28\, V_o \sqrt{\frac{B_o}{X}} \text{ (slotted port)} \quad (1)$$

$$V_m = 6.2\, V_o \left(\frac{D_o}{X}\right) \text{ (circular port)} \quad (2)$$

$$V_o = C\sqrt{2g(\Delta H)}$$

where: V_m = jet centerline velocity at distance X,
V_o = jet exit velocity,
B_o = slot width,
D_o = circular port diameter,
X = distance from exit port along jet, centerline,
C = discharge coefficient of port, and
ΔH = head differential across port.

From equation (2), it is evident that for a circular port, the distance X from the port needed to reduce the jet velocity V_m to a certain fraction of the jet exit velocity V_o is directly related to the port diameter D_o. Thus, a reduction in the port diameter by one-half would reduce the distance X required to produce the same centerline velocity V_m by one-half. Ports as small as 3.2 mm have been used on some multijet valves. Therefore, for a head of 150 m, the jet velocity could be reduced from 51 m/s at the exit port, to 1.0 m/s in 1.0 m, using a 3.2-mm diameter port.

With regard to cavitation bubble formation and collapse, Rouse[8] made the following observation related to the mixing zone of a submerged jet: "Noteworthy is the fact that—although the zone of maximum cavitation coincides in general with the zone of maximum turbulence—no vapor formation is evident over an initial length of about one diameter." Appel[9] further added to the understanding of bubble collapse near a submerged jet with his studies, indicating that the maximum noise level and thus the greatest level of bubble collapse occurs approximately five nozzle diameters from

the exit port. The greatest level of bubble collapse coincides with the zone in which the turbulence reaches the centerline of the submerged jet. Figure 2 illustrates these findings superimposed on a copy of a photograph taken during the Iowa (Appel) studies.

HIGH HEAD TEST FACILITY

The facility consists of a seven-stage vertical turbine pump driven by a 186-kW, d-c motor. A rectifying unit and motor-speed control converts a.c. into the d.c. needed for the motor and provides speed selection from 200 to 1800 r/min Rate of flow is measured with a 200-mm venturi meter permanently installed 3.0 m downstream from the pump outlet. A 200-mm motor-operated valve is used to control the downstream pressure on the test valve. Figure 3 illustrates the laboratory test facility and performance characteristics of the high head pump.

INVESTIGATIONS AND RESULTS

The Bailey Polyjet Valve

Investigation.—Tests were conducted on a 200-mm test valve provided by the Chas. M. Bailey Company, Inc. (fig. 4). The purpose of the test was to determine the performance characteristics of the valve under the operating conditions previously discussed, namely, energy dissipation at throttled flows and minimal energy loss at design flows.

The control for the valve consists of a cylindrical sleeve, located in the annular chamber of the valve which travels over the multijet ports, controlling the open port area and, thus, the valve discharge. The flow passes through the 1835-4.78 mm ports, then through the inside of the perforated sleeve into the downstream pipe, which is the same diameter as the inlet pipe.

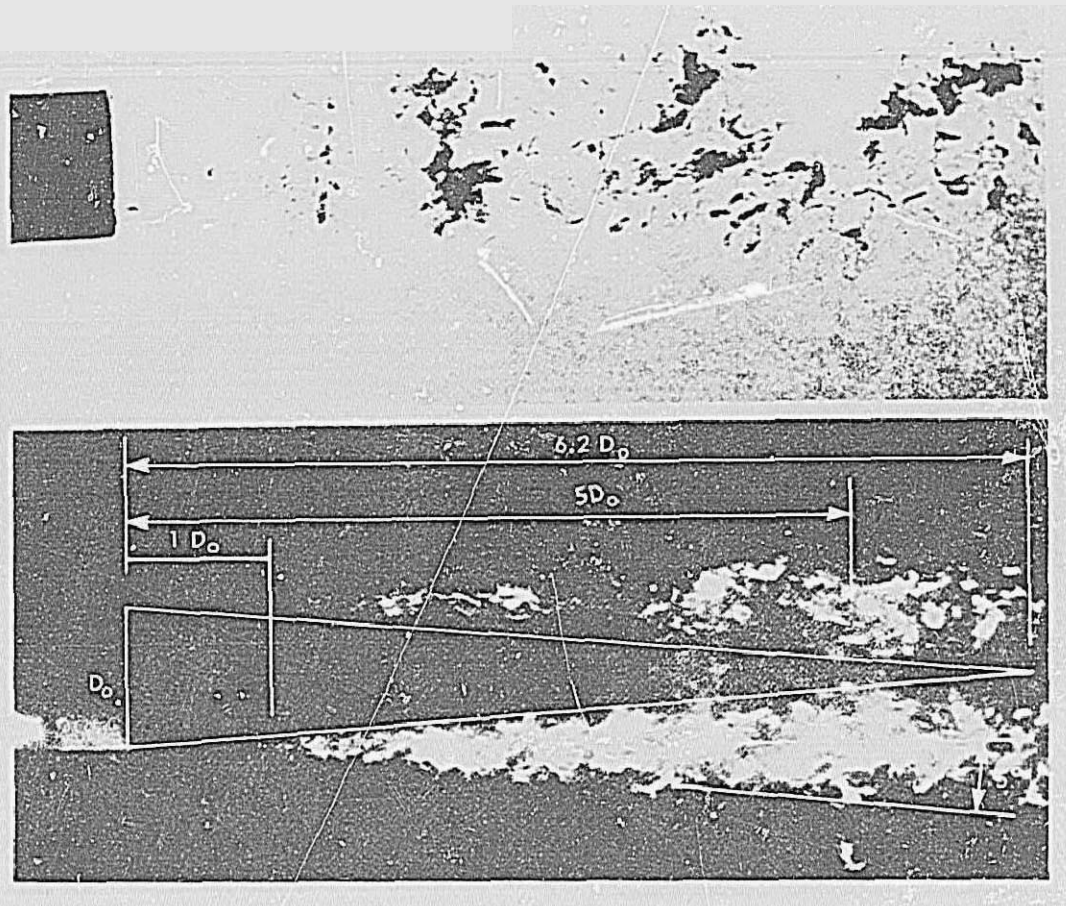


Figure 2.—Cavitation in a submerged jet. (from Reference [5])
(upper photo--rear illumination)
(lower photo—center plane illumination)
Photo P801-D-78313

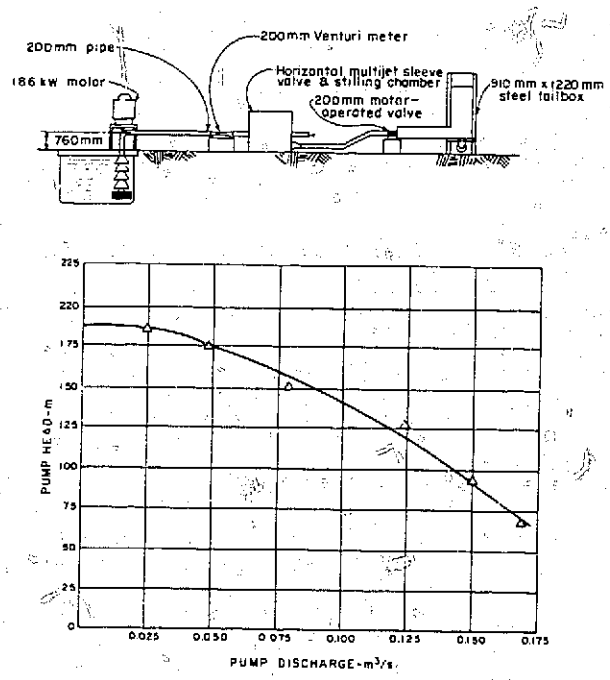
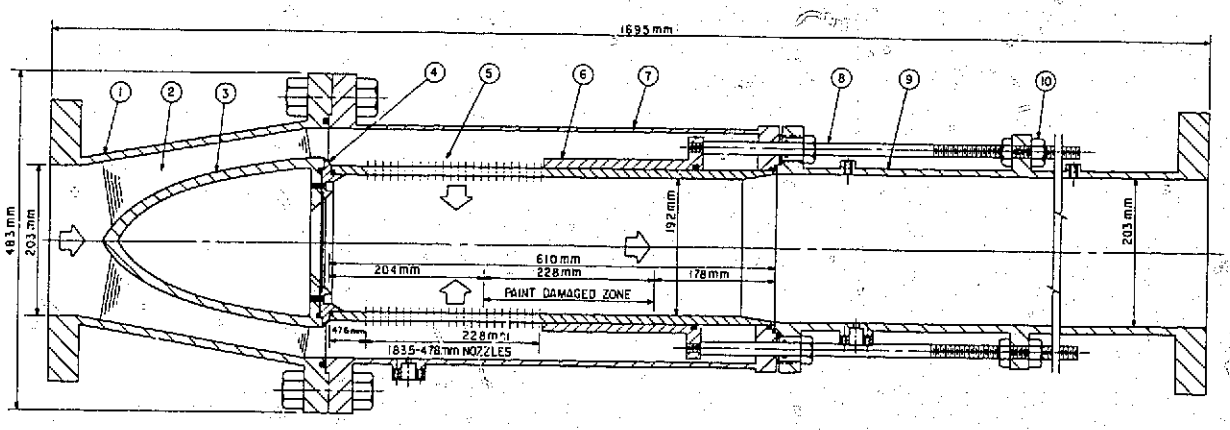


Figure 3.—200-mm high head test facility.



VALVE PARTS LIST

1	UPSTREAM BODY
2	SUPPORT FIN
3	CONE
4	SEAT RING
5	PERFORATED SLEEVE
6	CONTROL SLEEVE
7	MAIN BODY
8	OPERATING STEM
9	DOWNSTREAM BODY
10	OPERATING NUTS

Figure 4.—200-mm Bailey polyjet valve.

Test results.—Test data were measured using a Venturi meter, pressure cells, mercury and water manometers, and a sound level meter. The interior surface of the valve was coated with concrete curing compound to determine cavitation damage potential. Tests were conducted to determine the discharge coefficient C_d for the Bailey valve. The discharge and head loss coefficient (C_d and K) curves based on the 203-mm diameter pipe inlet area and the 192-mm diameter perforated sleeve area are shown on figures 5 and 6. Since all 1835 ports were the same size and configuration, it would appear that the overall valve flow characteristics would be the same as the local flow characteristics of each port. This would result in a linear relationship between the valve coefficient of discharge C_d and area ratio A_{port}/A_{pipe}. As indicated on figure 5, this was not the case. Test results conducted by the MWD of Southern California[10] on a similar 200-mm Bailey polyjet valve are included on figure 5.

As the control sleeve is opened, exposing more ports, the flow near the upstream end of the perforated sleeve passes the upstream ports in an effort to satisfy the downstream flow demand. The resultant approach velocity and pressure drop near the upstream ports causes a reduction in the flow through these ports. This phenomenon is similar to that which occurs in manifold pipes. Enger and Levy[12], in a discussion of pressures in manifold pipes, explain that a limiting area of ports can be developed beyond which there will be no flow through some of the initial ports.

The head loss coefficient $K = \frac{2g\Delta H}{V^2} = \frac{1}{C_d^2}$, is shown on figure 6. The test valve had an inlet diameter equal to 203 mm, and an internal sleeve diameter of 192 mm. The two USBR K valve curves plotted on figure 6 reflect the two pipe areas used for a given discharge Q and head loss ΔH. The design head loss coefficient K for a polyjet valve with internal sleeve diameter equal to the inlet pipe diameter and the total port area equal to the pipe area should have a value close to 2.21.

Cavitation damage potential.—To evaluate potential cavitation damage to the 200-mm polyjet test valve,

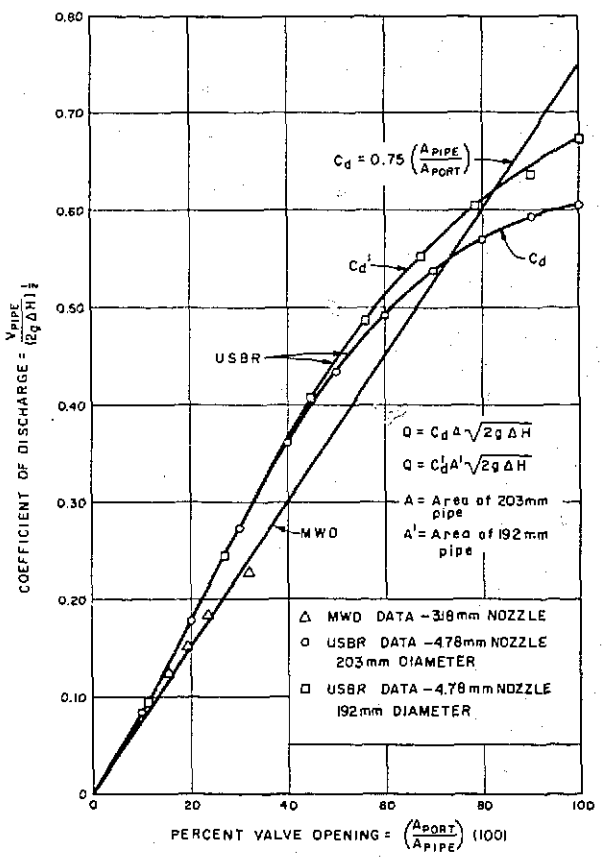


Figure 5.—Bailey polyjet valve discharge coefficient.

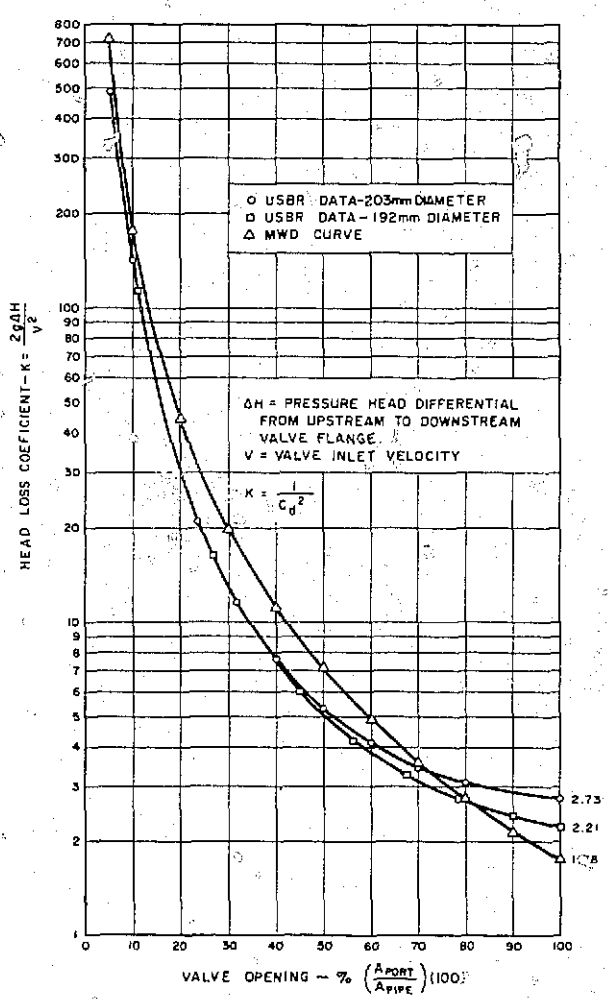


Figure 6.—Bailey polyjet valve head loss coefficient.

the inside flow surface of the ported sleeve was painted with a concrete curing compound. The concrete curing compound has proven to be an excellent indicator of the location and degree of pitting resulting from bubble collapse. Table 1 presents the test data for valve openings of 5, 10, 15, 20, and 30 percent. The pressure head was measured 1.83 m upstream (H_u) and 1.73 m downstream (H_d) from the valve. The vapor pressure H_v at the laboratory elevation is equivalent to −8.47 m of water.

The high head pump was designed to deliver approximately 0.17 m^3/s. Therefore, the cavitation damage tests were limited to valve openings between 5 and 30 percent and cavitation index $\sigma = \frac{H_d - H_v}{H_u - H_d}$ values ranging from 0.08 to 0.59 respectively. Figure 7 illustrates the resulting paint removal on the internal flow surface of the ported sleeve under the test conditions described in table 1. The photographic sequence shows accumulative damage from the 5 percent open up to and including the 30 percent open level. The scratch lines on the paint in the damaged area were caused by insertion of the mirror for photographs. All tests, except 7d, were operated for 2 h. Test 7d was operated for 1½ h.

The cavitation damage shown on figure 7b is typical of locations where individual cavities imploded near the flow surface. It was noted that, although the valve was operated at an extremely low sigma value (σ = 0.08) for the 5 percent test, there was no apparent cavitation damage. The majority of paint removal occurred at the 10 and 15 percent valve openings. The results indicate that cavitation damage in this particular valve is related to the quantity of vapor bubbles produced as well as the pressure head relationships, $\sigma = \frac{H_d - H_v}{H_u - H_d}$. The damage was located in a zone which extended 152 mm downstream from the ports to 76 mm into the port zone, figure 4.

To determine potential cavitation damage of the polyjet valve under the conditions tested, a 75-mm by 270-mm steel specimen was coated with the concrete curing compound and tested in the laboratory's Venturi cavitation test facility for 1 hour. Photographs of the specimen before and after the test are shown on figure 8. The amount of paint removal from the interior of the ported sleeve was much less than that experienced on the steel specimen tested in the cavitation facility. The Venturi cavitation test facility produces mild cavitation. Based on these test results, the mild pitting of the perforated sleeve of the polyjet valve that occurred would take years to cause significant damage.

The amount of paint removal would increase for greater valve openings at low sigma values; however, on long pipelines, the majority of the energy head would be dissipated in upstream pipe losses at larger valve openings resulting in higher sigma values with less cavitation potential. It is most likely that the critical point, with respect to cavitation damage for these values on long aqueducts with high friction losses, will be in the range of 10 to 15 percent open. In general, the polyjet valve performed well when operated within the manufacturer's suggested pressure head ranges.

Noise level.—A sound level meter was used to determine the noise level at various distances from the valve and the location of the maximum noise level along the valve.

Table 1.—*Bailey polyjet valve test data*

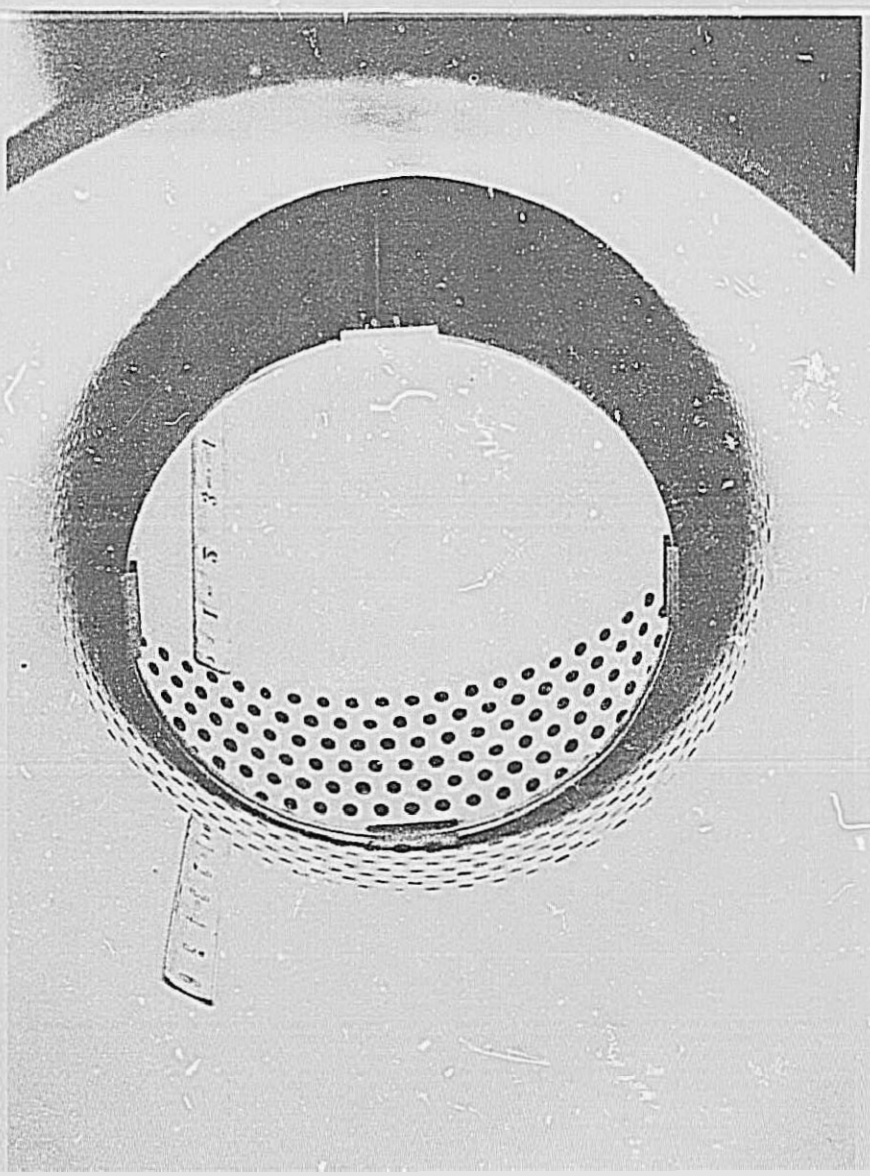
Valve opening, %	H_u, m	H_d, m	Q, m^3/s	V, m/s	[1] Cavitation index = σ	Time, h
5	136	2.19	0.066	2.04	0.08	2.0
10	107	5.85	.122	3.78	.14	2.0
15	70.1	4.18	.151	4.66	.19	2.0
20	46.0	4.15	.167	5.15	.30	1.5
30	25.3	4.05	.171	5.27	.59	2.0

[1] $\sigma = \frac{H_d - H_v}{H_u - H_d}$

H_v = −8.47m

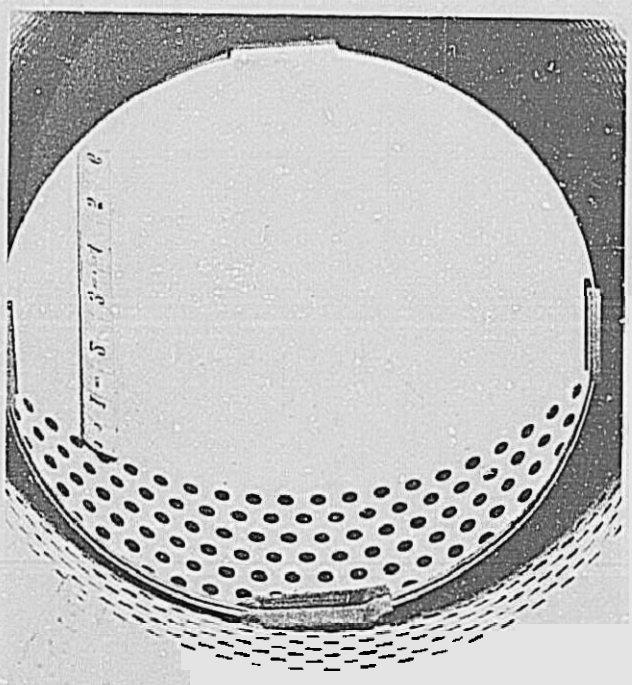
H_d = Downstream pressure head

H_u = Upstream pressure head

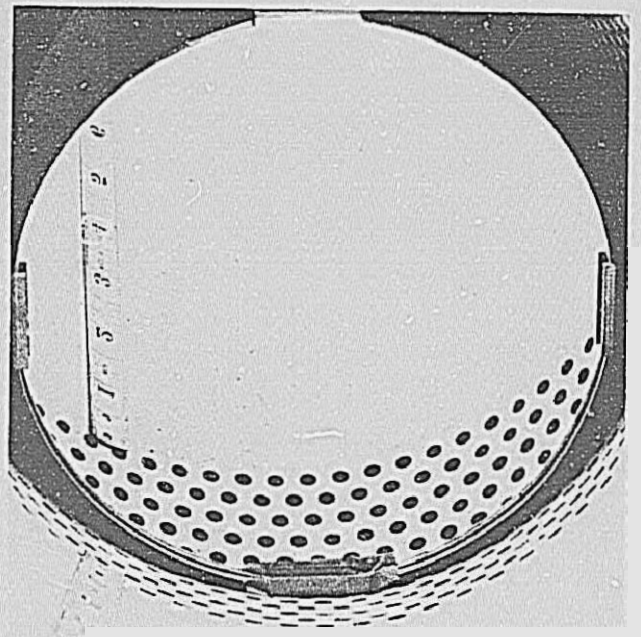


n. Valve operated at 5 percent open for 2 h.
H_u = 136 m σ = 0.08
H_d = 2.19 m Q = 0.066 m^3/s

Figure 7.—Looking upstream at ported sleeve flow surfaces. Photo P801-D-78314

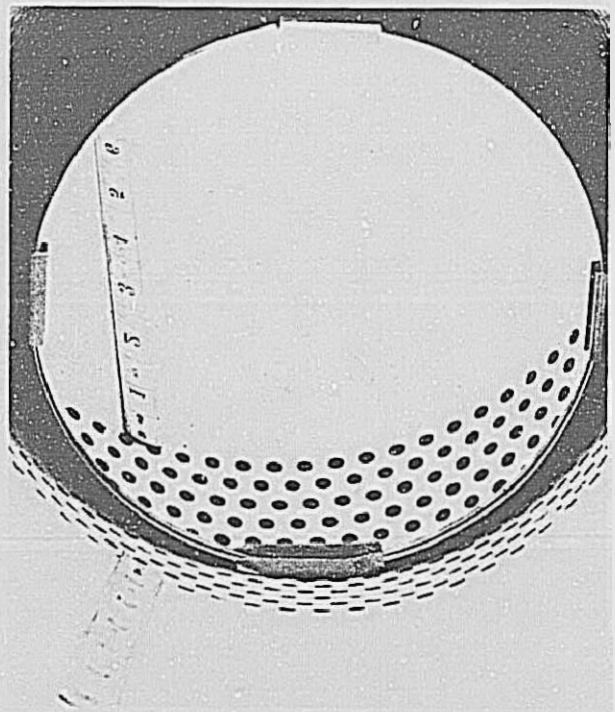


b. Valve operated at 10 percent open for 2 h.
H_u = 107.0 m σ = 0.14
H_d = 5.85 m Q = 0.122 m^3/s

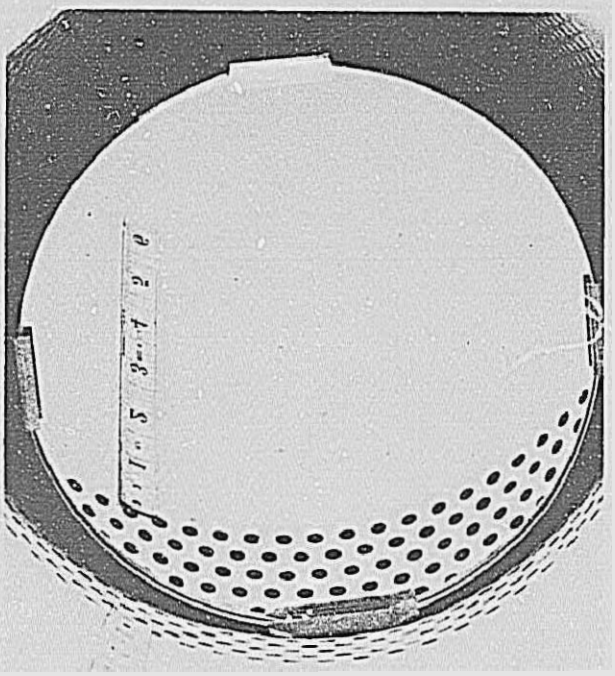


c. Valve operated at 15 percent open for 2 h.
H_u = 70.1 m σ = 0.19
H_d = 4.18 m Q = 0.151 m^3/s

Figure 7.—Looking upstream at ported sleeve flow surfaces. (Continued) Photos P801-D-78315 and P801-D-78320

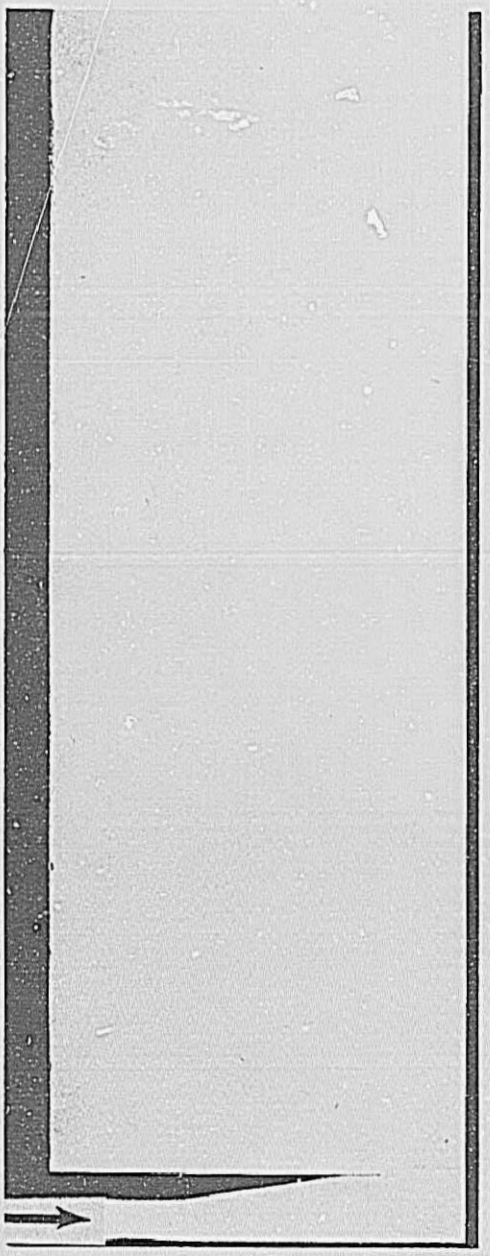


d. Valve operated at 20 percent open for 1-½ h.
H_u = 46.0 m σ = 0.30
H_d = 4.15 m Q = 0.167 m^3/s

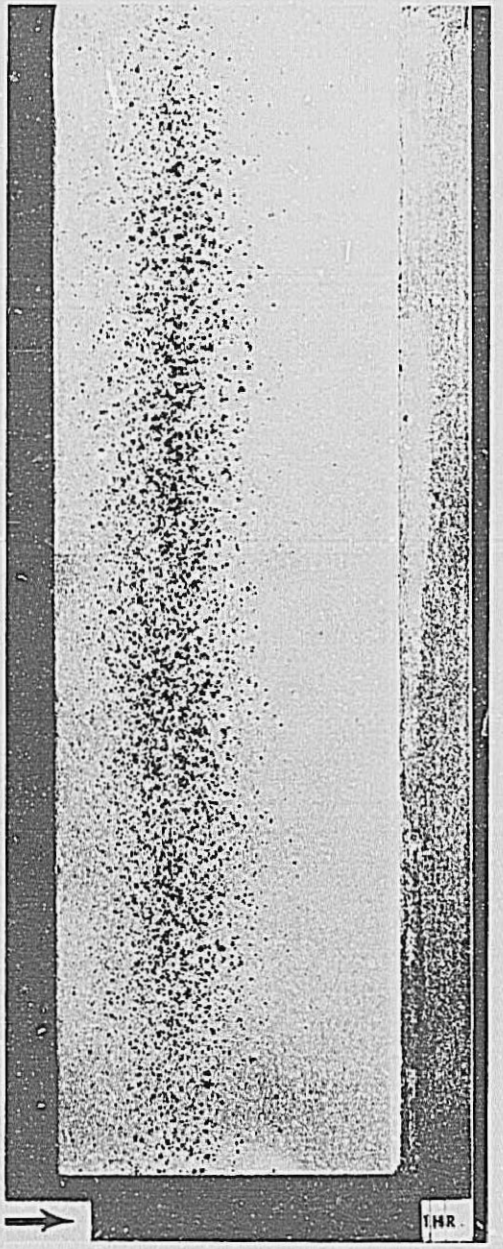


e. Valve operated at 30 percent open for 2 h.
H_u = 25.3 m σ = 0.59
H_d = 4.05 m Q = 0.171 m^3/s

Figure 7.—Looking upstream at ported sleeve flow surfaces. (Continued) Photos P801-D-78321 and P801-D-78322



a. Before. Photo P801-D-78316



b. After 1 hour. Photo P801-D-74770

Figure 8.—Cavitation damage on steel plate.

At 300 mm from the valve, the maximum noise level of 92 dBA occurred at 10 percent open ($\sigma = 0.14$). The maximum noise level at 50 mm from the valve occurred in the ranges 280 to 400 mm and 225 to 425 mm from the upstream end of the perforated sleeve, with 10 and 15 percent valve openings, respectively. The location of the maximum noise level measurements was the same as the paint damaged zone shown on figure 4. Table 2 shows the maximum sound level measurements at 50 and 300 mm from the valve body. There is also a good correlation between the degree of damage and the high sound level measurements, with greater damage occurring in those tests experiencing high sound levels.

Table 2.—*Sound level measurements*

Valve opening, %	Sound level measurements, dBA: 50 mm from valve	Sound level measurements, dBA: 300 mm from valve
5	98	90[1]
10	103	97
15	103	96
20	95	90[1]
30	92	90[1]

[1] Sound level of background noise

Horizontal Multijet Sleeve Valves

Introduction.—Previous multijet sleeve valve designs did not fully satisfy the design criteria desired by the USBR, that is, a valve which will dissipate high energy flows at throttled discharges and deliver design flows with a minimum head loss at the valve. Although the previous designs function quite well as pressure reducing valves, most do not emphasize minimal head loss when delivering design flows.

The concept of a linear relationship between sleeve travel and valve discharge was another desired characteristic sought in the new design. Such a valve would provide better control characteristics on long aqueducts where waterhammer presents a potential problem. Figure 9 illustrates ideal valve characteristics as a function of sleeve travel for a 200-mm pipeline where the static upstream pressure head is 137 m. As the valve is opened and the port area slowly increases, the valve discharge increases linearly with sleeve travel. This increase in valve discharge results in a reduced upstream pressure head due to friction losses in the long aqueduct. When the valve has opened 230 mm, it has performed the function of a pressure reducing valve and assumes the role of a low head-loss control valve. At this point, the valve port area increases rapidly, but with little increase in discharge due to the low available pressure head across the valve. When the valve port area equals

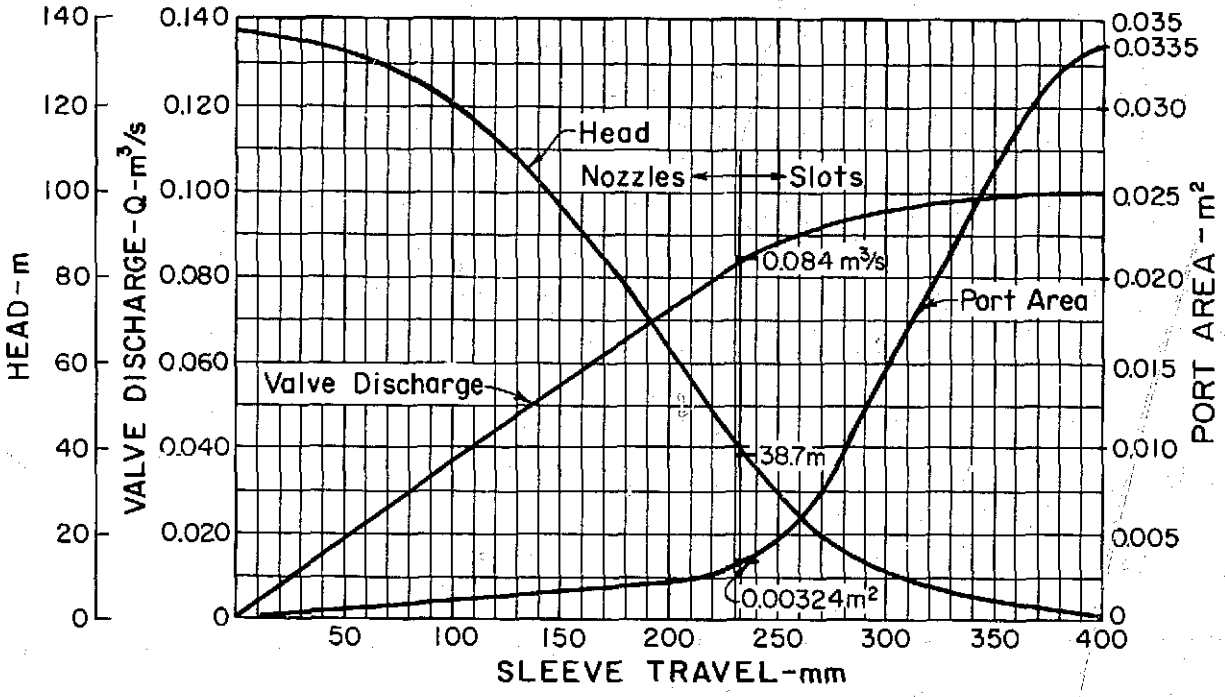


Figure 9.—Ideal multijet sleeve valve characteristics.

the 200-mm pipe area, the pressure head ΔH has decreased to 1.1 m across the valve, resulting in a discharge coefficient C_d of approximately:

$$C = \frac{Q/A}{\sqrt{2g\Delta H}} = 0.68$$

where: Q = valve discharge,
A = port area, and
ΔH = pressure head differential across valve.

The key to the multijet design is the proper placement of the ports along the spiral length of the valve to produce a near linear relationship between the valve discharge and sleeve travel. A computer program (see appendix) was developed to calculate the flow passing through the nozzle ports in each successive spiral quadrant along the length of the helix. The program locates the ports in a manner which nearly equalizes the flow through each quadrant. This results in a linear relationship between sleeve travel and valve discharge. When the required nozzle diameter is greater than the valve wall thickness, the program automatically changes the port configuration from nozzles to slots. Laboratory tests with the 200-mm valve were used to define the change in discharge coefficient for the nozzles and slots as the control sleeve varied the open port area. The flow chart shown on figure 10 presents the basic logic for the hydraulic considerations in the program (note that the computer program is written in U.S. customary units).

High head pump facility.—The horizontal multijet sleeve valve was tested in the same facility as the Bailey polyjet valve. Test data were measured using a Venturi meter, pressure transducers, and mercury and water manometers.

Investigation.—The 200-mm laboratory test valve discharged into a 1370-mm diameter, 1220-mm stilling chamber (fig. 11a). The basic concept of the valve and stilling chamber is illustrated on figure 11b. Flow enters the valve from the high pressure side and is discharged through the perforated body of the valve into the stilling chamber. A cylindrical sleeve, located inside the valve body, travels over the perforated section of the valve, controlling the port area and thus the valve discharge. The flow enters the downstream pipeline at the lower end of the stilling chamber. Pressure heads were measured at pressure taps P_1, P_2, and P_3, and corrected for a pressure differential ΔH between the upstream flange of the sleeve valve H_1 and the downstream 200-mm pipe flange H_2.

Equations (3) and (4) were used to calculate the discharge through the nozzles and slots respectively.

$$\text{Nozzle Discharge} = Q_n = C_n A_n \sqrt{2g\Delta h} \quad (3)$$

$$\text{Slot Discharge} = Q_s = C_s A_s \sqrt{2gH} \quad (4)$$

where: $C_n = 0.94\,[1 - ((V+V')/2)^2/(2g\,H)]$, (5)
$C_o = 0.85\,[1 - (V'/2)^2/(2g\,H)]$, (6)
A_n = open nozzle area,
A_s = open slot area,
Δh = pressure head difference between valve and stilling chamber,
A_p = pipe inlet area,
$V = Q/A_p$,
$V' = (Q - Q_n)/A_p$, and
$H = \Delta h + V^2/2g$.

When the nozzles alone are exposed, the velocity term on the right side of equation (5) is negligible since the velocity head is small with respect to the total available energy H. As the control sleeve opens, the total energy term decreases due to friction losses in the upstream aqueduct and the velocity term increases. Thus, the nozzle and slot coefficients C_n and C_s decrease in value to reflect the phenomenon of a manifold where the majority of the valve discharge is released through the downstream ports of the valve.

Equations (5) and (6) are similar to the equations presented by Vigander[11] and Enger[12] dealing with large diffusers and manifolds. They are empirical equations based on results of the laboratory studies conducted on the 200-mm test valve.

Port configurations tested.—The investigation included studies conducted on the two ported valve sleeves shown on figure 12. Dimensional sketches of the two sleeves are shown on figure 13. The sleeve travel versus port area relationship curves for the two configurations and the polyjet valve are shown on figure 14. The polyjet valve has a linear relationship between port area and sleeve travel. The area for the slotted port configuration increases at a somewhat slower rate, initially. The area for the valve with nozzles and slots increases very slowly at the start of the sleeve travel and then rapidly increases once the slots are exposed, yielding a linear relationship between sleeve travel and valve discharge.

The use of nozzles or orifices for the discharge ports should be carefully considered. The present spiral arrangement of the ports results in the partial blockage of some of the ports when the valve is used to control the flow. Figure 15 illustrates a typical 25.0-mm diameter port as a nozzle and an orifice with the control sleeve partially blocking the port. Although the nozzle-port has the advantages of a higher discharge coefficient and structurally requires a smaller port cross section in

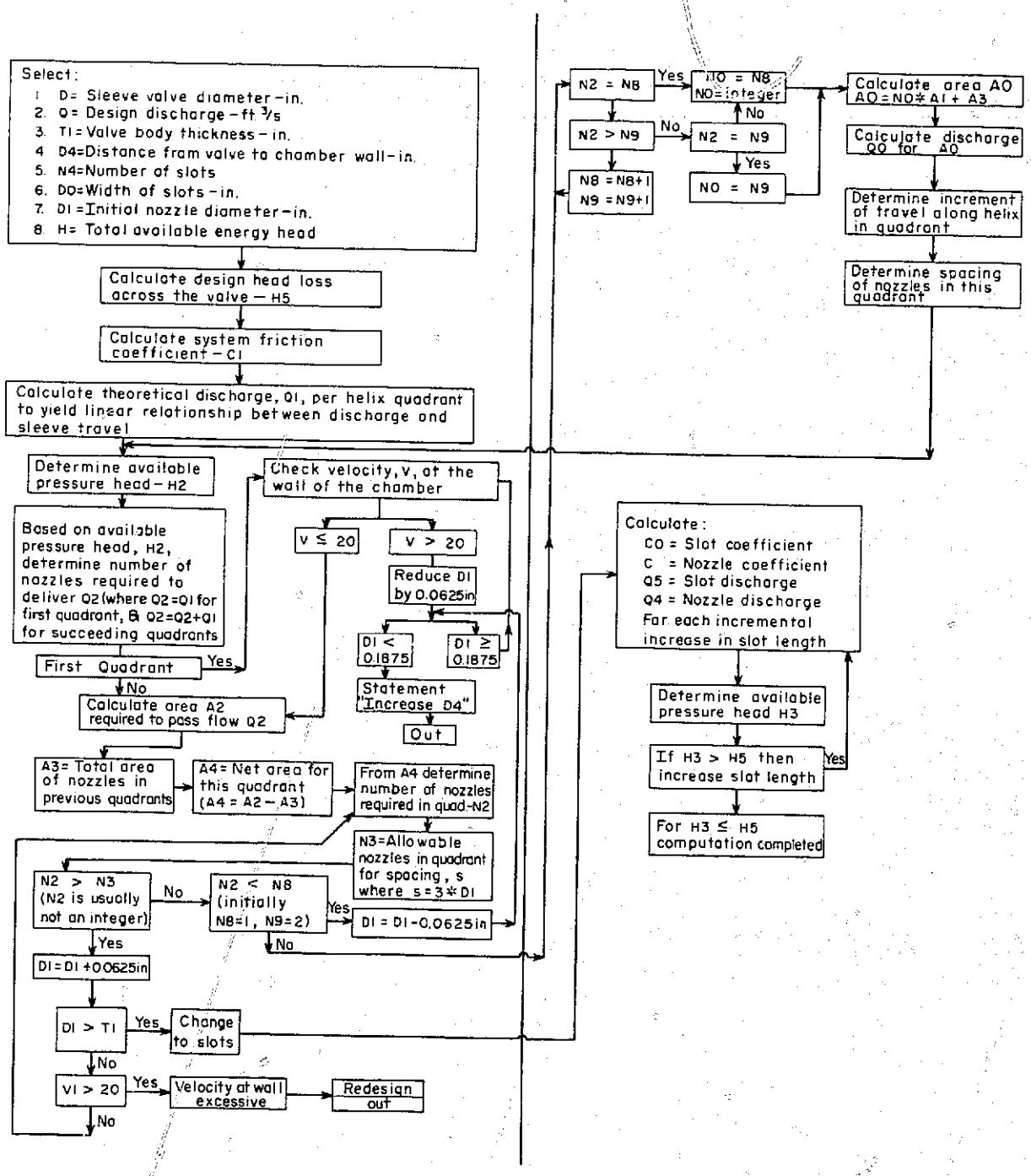
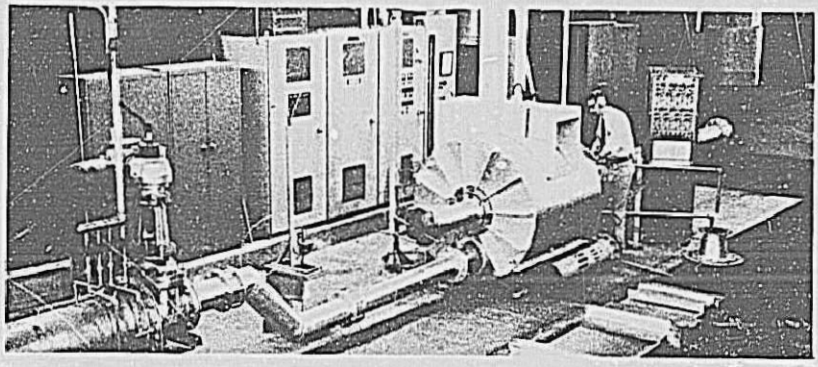
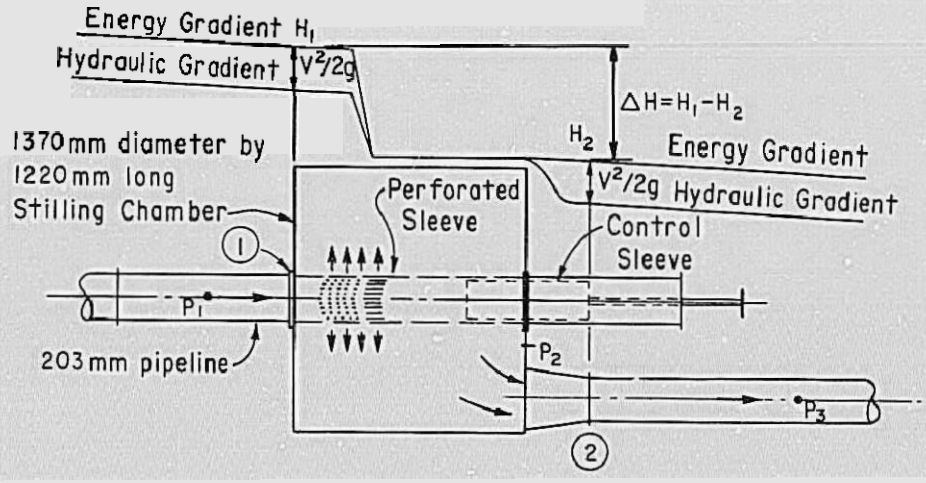


Figure 10.—Flow chart for hydraulic computations.

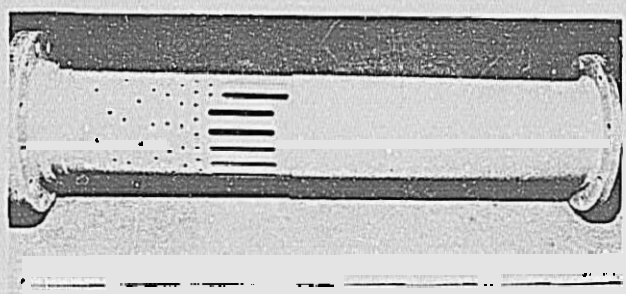


a. Laboratory test valve and chamber



b. Schematic of valve chamber

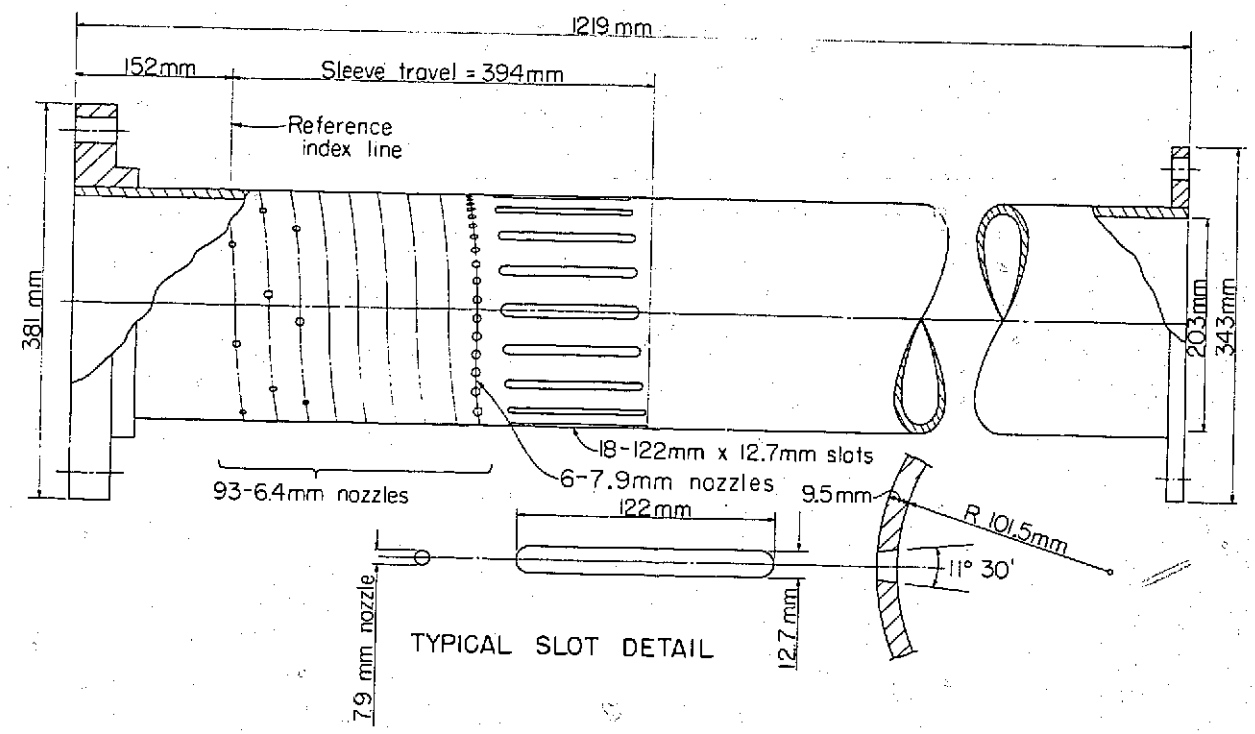
Figure 11.—200-mm horizontal multijet sleeve valve.



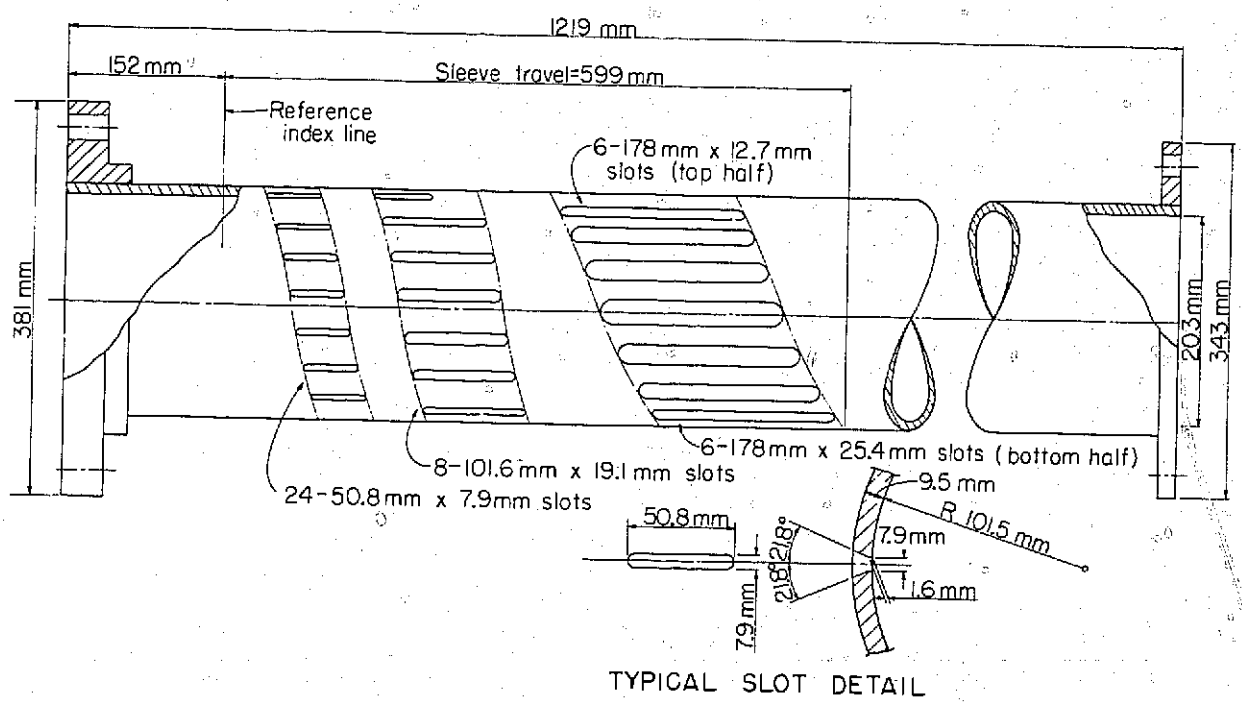
a. Nozzles and slots. Photo P801-D-78318

b. Slots. Photo P801-D-78319

Figure 12.—Ported valve sleeves.



A. NOZZLES AND SLOTS



B. SLOTS

Figure 13.—Dimensional sketches of ported valve sleeves.

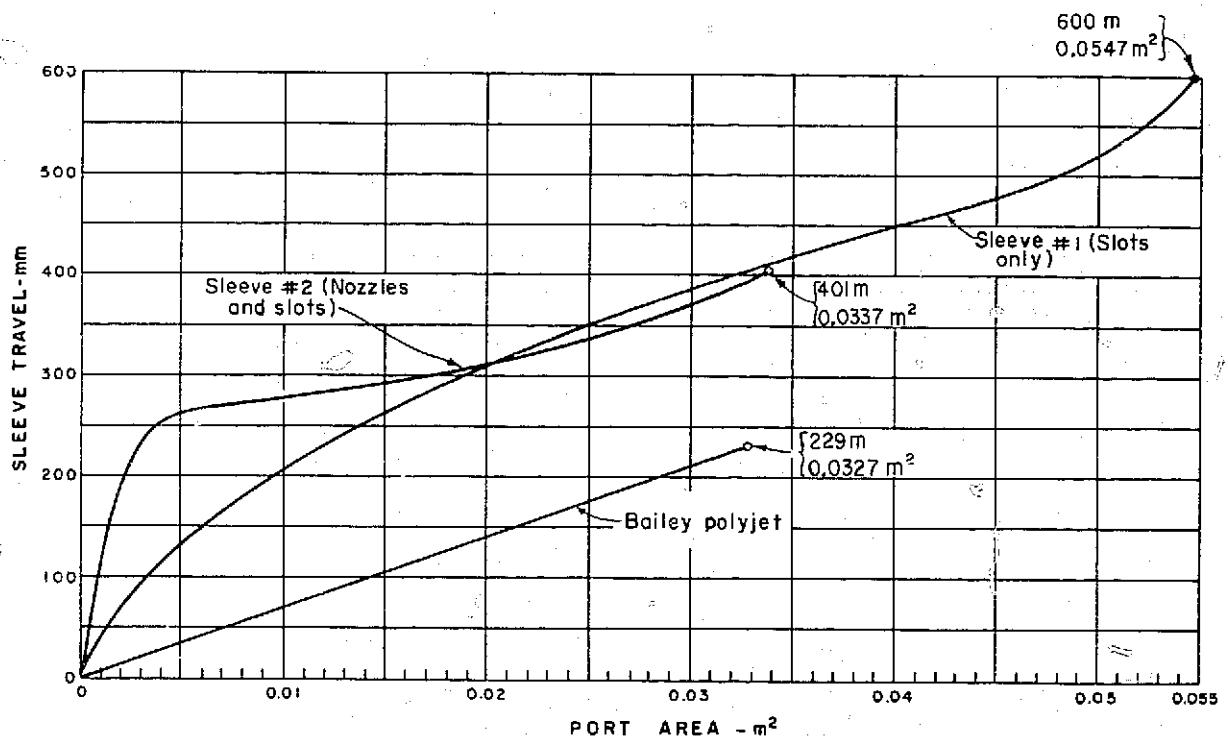


Figure 14.—Sleeve travel versus port area.

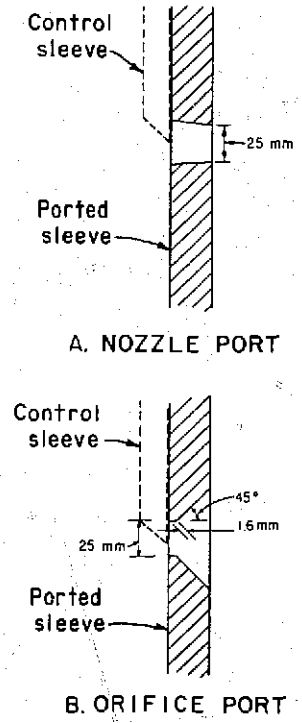


Figure 15.—Typical port designs.

the ported sleeve, it has the disadvantage that during partial blockage, the control point for the nozzle can move from the exterior surface of the ported sleeve to an internal control at the control sleeve. This change in control position can cause cavitation damage to the flow surface of large nozzle-ports during high pressure control (greater than 31 m). Laboratory tests with nozzle-ports up to 7.9 mm in diameter and studies conducted by the MWD of Southern California[13,14] with nozzle-ports up to 19 mm in diameter show no signs of cavitation damage to the nozzle flow surfaces. Further laboratory investigations should be conducted to determine the pressure head, nozzle diameter relationship where cavitation will occur in large nozzle flow passages.

Until such laboratory investigations have been conducted, it is suggested that an orifice design similar to that shown on figure 15b be considered for ports 19 mm in diameter or larger. The orifice-port has a lower discharge coefficient C_d due to the vena contracta and, thus, requires a larger total port area for the ported sleeve. There is no change in the control point for a partially closed orifice port. The control for the jet is at the interior surface of the ported sleeve when the orifice is completely open or partially blocked by the control sleeve. The 1.6-mm control surface and 45° taper provide a clean control point for the jet with adequate circulation.

The data for the head loss coefficient K are plotted with respect to percent valve opening (100 X port area/pipe area) on figure 16 for both port configurations shown on figure 12. The head loss coefficient is based on the pressure head differential ΔH between the upstream valve flange and the 200-mm pipe flange downstream of the stilling chamber. The loss coefficient K, therefore, includes the total system loss for the control structure. The polyjet valve data are also plotted on figure 16.

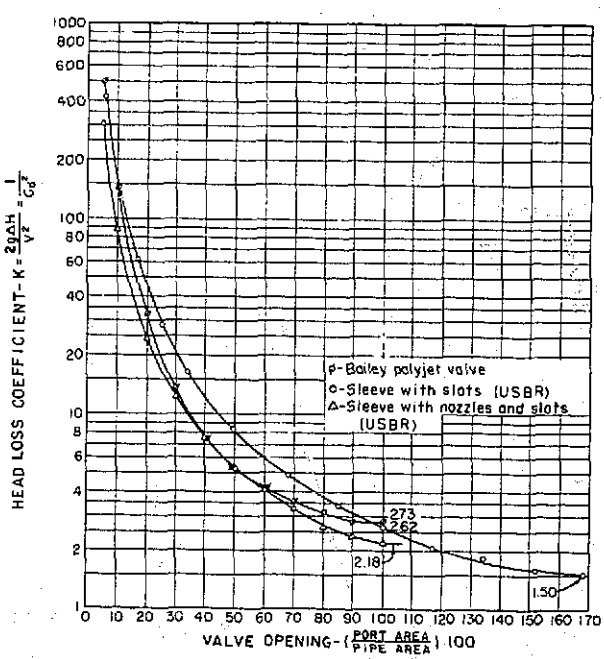


Figure 16.—Head loss coefficient.

Jet velocity measurements in stilling chamber.—As part of the horizontal multijet sleeve valve test program, a limited number of velocity measurements were made to determine the velocity characteristics of the issuing jets. A pitot cylinder probe was used to measure the dynamic and static pressure in a horizontal plane normal to the axis of the sleeve valve. Velocity traverses were made at distances of 152, 305, and 457 mm from the sleeve valve.

Figure 17 illustrates the pitot cylinder probe and its installation in the stilling chamber. The pitot cylinder consists of three small ports located in a plane normal to the axis of the probe and spaced at an angular distance of 39¼°. Investigations conducted by Binder[15] and Winternitz[16] have verified that the two outside

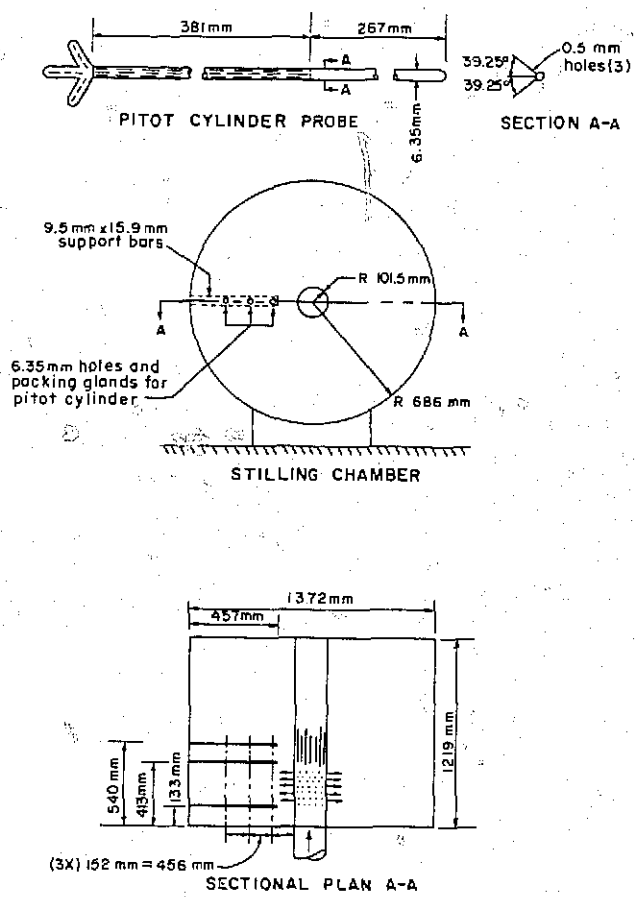


Figure 17.—Pitot cylinder probe and stilling chamber.

ports separated by an angular distance of 78½° will measure static pressure when the flow direction bisects the angle resulting in equal pressure at both ports. The total energy head of the jet is given by the central port which is directed into the flow. Thus, the jet velocity head is the difference between the total energy head measured by the central port and the static head measured by either side port once it has been determined the side port pressures are the same. The pitot cylinder can also be used to determine the direction of flow in the plane normal to the pitot cylinder.

Differential pressures were measured using a 172-kPa pressure transducer. Figure 18 illustrates the average and maximum measured velocities for pressure head differentials of 44.5 and 104 m. The maximum average velocity at distances of 152 and 305 mm from the valve are plotted on figure 19. The heavy solid line describes equation (2) proposed by Albertson[6]. The dashed line identifies data measured by the MWD of Southern California[13].

Extension of the USBR and MWD data on figure 19 for multijet stilling chambers indicates the centerline jet velocity V_m will remain the same as the exit velocity V_o for a distance of approximately 10 nozzle diameters (10 D_o). Beyond 10 nozzle diameters, the centerline jet velocity decreases at a rate equivalent to the 1.4 power of the ratio-nozzle diameter D_o to distance from valve X, that is, $(D_o/X)^{1.4}$.

The velocity data for the slots were considerably lower in value than those reported by other investigators. The velocity probe was apparently not in the center plane of the discharging slot and, therefore, these data were not summarized on figure 19.

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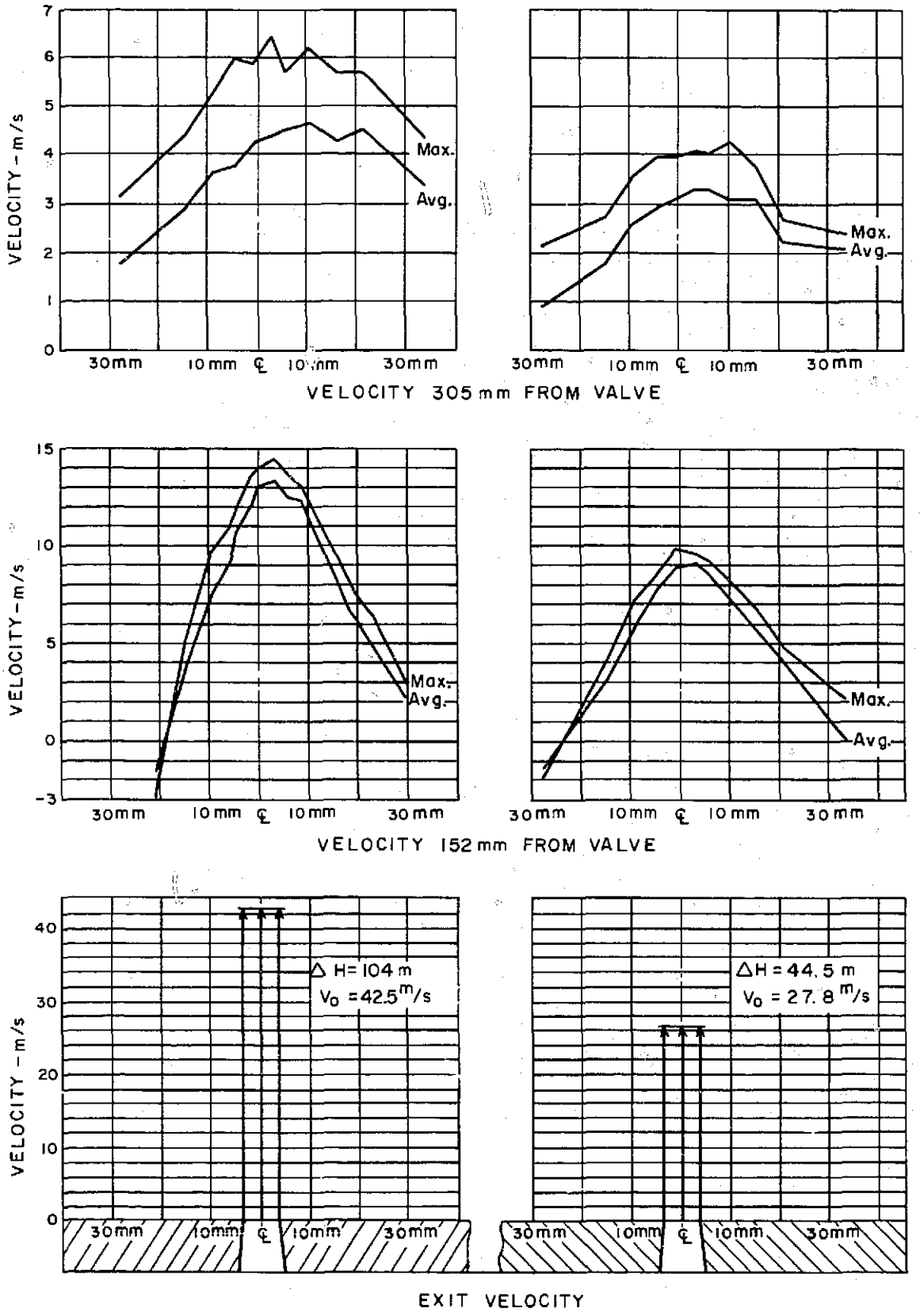

Figure 18.—Multijet velocity measurements.

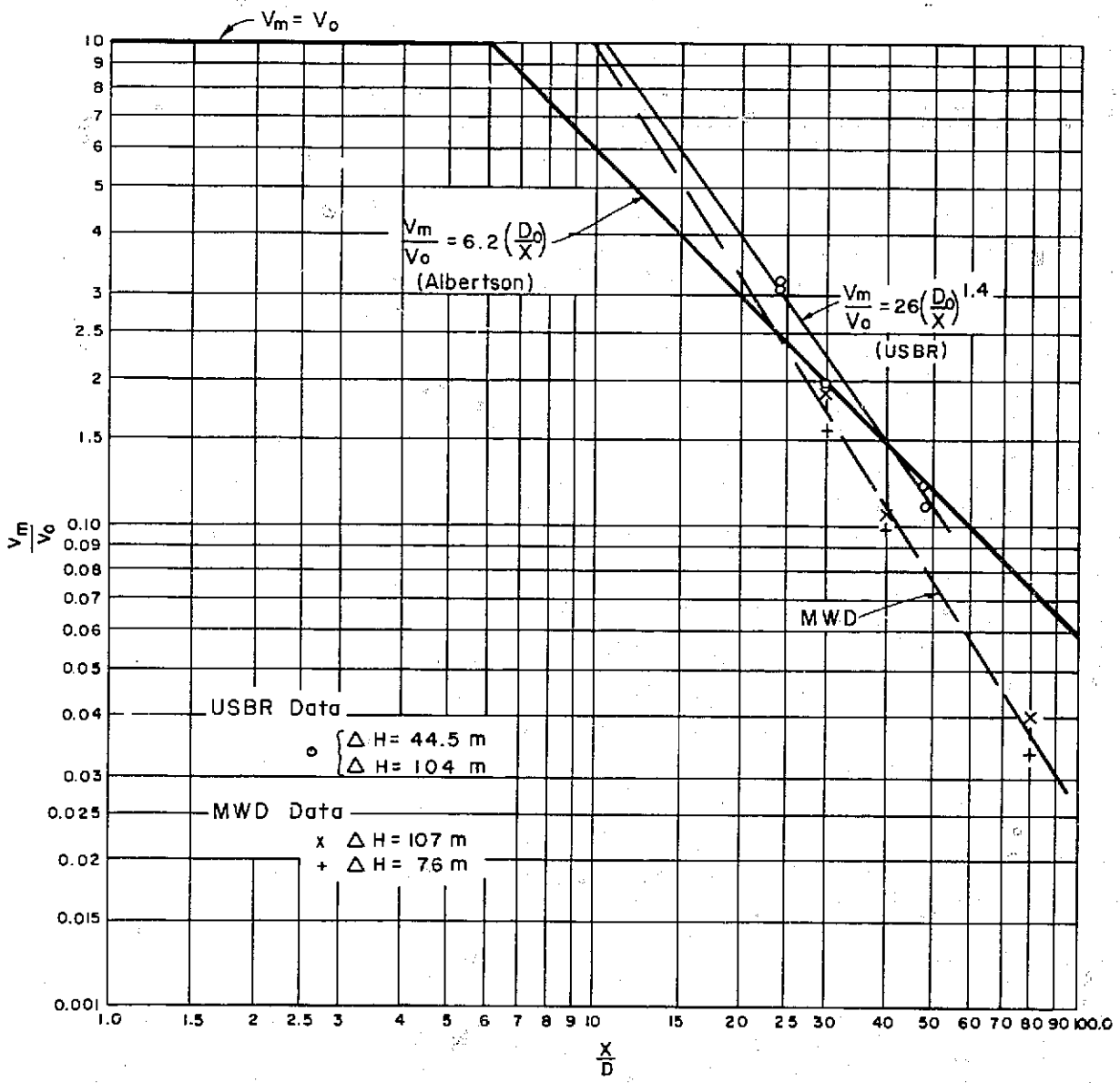

Figure 19.—Relationship of jet velocity to distance from valve.

APPENDIX

APPENDIX

Computer programs developed by the Bureau of Reclamation are subject to the following conditions. Consulting service and assistance with conversion to other computers cannot be provided. The programs have been developed for use at the USBR and no warranty as to accuracy, usefulness, or completeness is expressed or implied.

Permission is granted to reproduce or quote from the program; however, it is requested that credit be given to the Bureau of Reclamation, U.S. Department of Interior, as the owner.

PROGRAM TITLE – HYDRAULIC COMPUTATION FOR MULTIJET SLEEVE VALVE

GENERAL INFORMATION

This computer program was developed to determine the size, number, and location of discharge ports for multijet sleeve valves placed in horizontal stilling chambers on municipal and industrial water supply aqueducts.

The program is written in Basic language for a CDC computer. The input and output data, as well as all computations, are in U.S. customary units.

The program is designed to calculate and compare with an established standard several design calculations, including: jet velocity at the stilling chamber wall, steel stresses in the ported body of the valve, rate of control-sleeve travel, size and proximity of exit ports, and discharge coefficient for the individual ports. A major program function is to calculate valve discharge in each spiral quadrant as the control sleeve opens. Since a near linear relationship is desired between sleeve travel and discharge, a theoretical estimate of discharge per quadrant is made, and the size and number of nozzles is calculated based on available head at that particular quadrant. The jet velocity at the chamber wall is checked; if it exceeds 20 ft/s, the nozzle diameter is reduced by 1/16 inch. The center-to-center nozzle spacing is checked, and if it is less than three nozzle diameters, the nozzle diameter is increased by 1/16 inch. The program will change the port configuration from nozzles to slots when the calculated nozzle diameter exceeds the valve wall thickness.

The number and width of slots is specified with the input data. As the total discharge through the valve increases, the discharge through the initial nozzles decreases. This is taken into consideration in the computations by correction coefficients given in equations (5) and (6) of the text. Incremental slot lengths (1/4 inch) are considered until a predetermined minimum design head loss for the valve is reached.

INPUT

The input data begins with the structure's title at line 900. The second data line includes the following six items:

1. Number of pipe reaches of various diameters between the upstream storage tank and the multijet valve.
2. Desired minimum closing time (full-open to full-closed) of the multijet valve, in seconds.
3. Diameter of the pipe from the stilling chamber to the downstream storage tank, in inches.
4. Upstream storage tank water surface elevation.
5. Centerline elevation of the multijet sleeve valve.
6. Downstream storage tank water surface elevation.

The third data line (this line is repeated for each pipe reach listed in No. 1 above) includes the following four items:

1. Diameter of first pipe reach, in inches (beginning at upstream tank).
2. Length of pipe, in feet.
3. Scobey's friction coefficient.
4. Minimum friction coefficient.

The last data line includes eight items dealing with the multijet sleeve valve and stilling chamber:

1. Sleeve valve diameter, in inches.
2. Design flow through valve in ft^3/s.
3. Sleeve valve body wall thickness, in inches.
4. Distance from the outside wall of the valve to the stilling chamber wall, in inches.
5. Number of slots in sleeve.
6. Width of slots, in inches.
7. Diameter of first nozzle in sleeve body.
8. KO-coefficient for head loss from sleeve valve stilling chamber to downstream storage tank.

OUTPUT

The computer program output lists a number of calculated results including:

- Maximum jet velocity at the stilling chamber wall.
- Steel stresses in the valve body.
- Axial length of slots and total control sleeve length required for nozzles and slots.

Each of the nozzles and slots is identified by a distance from the index circle and an angle from the index line. The discharge, pressure head, and flow area are also given for each increment of opening. In the slotted portion of the valve, the nozzle and the slot coefficients are given with a calculated discharge through the nozzles and slots.

EXAMPLE

The following example problem is given to illustrate how to use the program. The system illustrated is the Deep Red Run rate-of-flow control station (RFC) on the Mountain Park Project in Oklahoma. There are three upstream pipe reaches with appropriate data listed in the table below:

Reach No.	Pipe diameter, in	Pipe length, ft	Friction coefficient	
			normal	minimum
1	24	18,118	0.370	0.410
2	18	21,030	0.345	0.400
3	16	12,370	0.345	0.400

Other data includes:

1. Pipe diameter between stilling chamber and downstream tank—24 inches.
2. Water surface elevation, upstream tank—1443 feet.
3. Water surface elevation, downstream tank—1214 feet.
4. Centerline elevation for multijet valve—1193 ft.
5. Valve closing time—400 seconds.
6. Sleeve valve diameter—14 inches.
7. Design discharge—8.1 ft^3/s.
8. Valve wall thickness—0.50 inches.
9. Distance from valve to chamber wall—19.5 inches.
10. Number of slots—20.
11. Width of slots—0.75 inch.
12. First nozzle diameter—0.375 inch.
13. KO-head loss coefficient from chamber to downstream tank—1.340.

GPO 846-051

EXAMPLE OUTPUT

DEEP RED RUN RFC VALVE STRUCTURE
DIA. OF JETS AND SPACING FOR MULTI-JET SLEEVE VALVE
VALVE DIAMETER= 14 INCHES

UPS TANK WS ELEV	SLEEVE CL ELEV	RFC TANK WS ELEV	PIPE FRICTION CONSTANTS NORMAL	MINIMUM	TOTAL LESS SLEEVE VAL
1443.00	1193.00	1214.00	3.44436218	2.58007641	3.46317816

SLEEVE WALL THICKNESS= .5 INCHES

PITCH OF HELIX= 1.5 INCHES

LENGTH REQUIRED FOR 10 TURNS= 15 INCHES

JET VELOCITY AT WALL= 11.7469 FEET PER SECOND

JET NO	DIA OF JET	DIST FROM INDEX CIR	ANGLE FROM INDEX LINE	QUAD NO	FLOW IN CFS	HEAD IN FT	REQUIRED AREA SF
1	.3750	.162	38.88	1			
2	.3750	.324	77.76	1	.175	228.89	.0015
3	.3750	.486	116.59	2			
4	.3750	.648	155.41	2	.350	228.57	.0031
5	.3750	.809	194.13	3			
6	.3750	.970	232.85	3	.524	228.04	.0046
7	.3750	1.131	271.35	4			
8	.3750	1.291	309.85	4			
9	.3750	1.451	348.36	4	.784	226.85	.0069

ALLOWABLE STEEL STRESS IN SLEEVE=15000 PSI

ACTUAL STEEL STRESS IN SLEEVE= 1301.23 PSI

10	.3750	1.611	26.57	5			
11	.3750	1.770	64.79	5	.956	225.81	.0084
12	.3750	1.928	102.64	6			
13	.3750	2.085	140.48	6			
14	.3750	2.243	178.33	6	1.211	223.88	.0107
15	.3750	2.399	215.73	7			
16	.3750	2.555	253.14	7	1.380	222.36	.0123
17	.3750	2.709	290.14	8			
18	.3750	2.863	327.14	8	1.546	220.66	.0138
19	.3750	3.015	3.57	9			
20	.3750	3.167	40.01	9			
21	.3750	3.319	76.44	9	1.792	217.80	.0161
22	.3750	3.468	112.25	10			
23	.3750	3.617	148.07	10	1.953	215.69	.0176
24	.3750	3.763	183.20	11			
25	.3750	3.910	218.34	11			
26	.3750	4.056	253.47	11	2.190	212.27	.0199
27	.3750	4.199	287.73	12			
28	.3750	4.342	321.98	12			
29	.3750	4.484	356.24	12	2.422	208.55	.0222
30	.3750	4.624	29.73	13			
31	.3750	4.763	63.21	13	2.572	205.93	.0238

32	.3750	4.900	35.88	14			
33	.3750	5.036	128.56	14			
34	.3750	5.172	161.23	14	2.793	201.80	.0261
35	.3750	5.304	192.89	15			
36	.3750	5.436	224.56	15			
37	.3750	5.568	256.22	15	3.006	197.49	.0284
38	.3750	5.695	286.85	16			
39	.3750	5.823	317.47	16			
40	.3750	5.950	348.10	16	3.213	193.01	.0307
41	.3750	6.074	17.66	17			
42	.3750	6.197	47.22	17			
43	.3750	6.320	76.79	17	3.413	188.41	.0330
44	.3750	6.439	105.28	18			
45	.3750	6.557	133.77	18			
46	.3750	6.676	162.26	18	3.605	183.71	.0353
47	.3750	6.790	189.67	19			
48	.3750	6.905	217.08	19			
49	.3750	7.019	244.50	19	3.790	178.95	.0376
50	.3750	7.128	270.66	20			
51	.3750	7.237	296.81	20			
52	.3750	7.346	322.97	20			
53	.3750	7.455	349.13	20	4.026	172.55	.0407
54	.3750	7.558	13.87	21			
55	.3750	7.661	38.62	21			
56	.3750	7.764	63.36	21			
57	.3750	7.867	88.10	21	4.248	166.14	.0437
58	.3750	7.965	111.64	22			
59	.3750	8.063	135.17	22			
60	.3750	8.161	158.71	22	4.407	161.37	.0460
61	.3750	8.254	180.89	23			
62	.3750	8.346	203.07	23			
63	.3750	8.439	225.25	23			
64	.3750	8.531	247.44	23			
65	.3750	8.623	269.62	23	4.657	153.51	.0499
66	.3750	8.710	290.34	24			
67	.3750	8.796	311.07	24			
68	.3750	8.882	331.79	24			
69	.3750	8.969	352.52	24	4.843	147.36	.0529
70	.3750	9.049	11.85	25			
71	.3750	9.130	31.18	25			
72	.3750	9.210	50.52	25			
73	.3750	9.291	69.85	25			
74	.3750	9.372	89.18	25	5.061	139.88	.0568
75	.3750	9.446	107.05	26			
76	.3750	9.521	124.93	26			
77	.3750	9.595	142.80	26			
78	.3750	9.669	160.68	26			
79	.3750	9.744	178.55	26	5.262	132.68	.0606
80	.3750	9.813	195.06	27			
81	.3750	9.882	211.57	27			
82	.3750	9.950	228.08	27			
83	.3750	10.019	244.60	27			
84	.3750	10.088	261.11	27	5.447	125.79	.0644
85	.3750	10.151	276.22	28			
86	.3750	10.214	291.34	28			

87	.3750	10.277	306.46	28			
88	.3750	10.340	321.58	28			
89	.3750	10.403	336.70	28			
90	.3750	10.466	351.81	28	5.652	117.94	.0690
91	.3750	10.523	5.43	29			
92	.3750	10.579	19.05	29			
93	.3750	10.636	32.67	29			
94	.3750	10.693	46.28	29			
95	.3750	10.750	59.90	29			
96	.3750	10.806	73.52	29			
97	.3750	10.863	87.14	29	5.866	109.39	.0744
98	.3750	10.914	99.31	30			
99	.3750	10.964	111.48	30			
100	.3750	11.015	123.65	30			
101	.3750	11.066	135.82	30			
102	.3750	11.117	147.99	30			
103	.3750	11.167	160.16	30			
104	.3750	11.218	172.33	30	6.058	101.48	.0798
105	.3750	11.263	183.05	31			
106	.3750	11.307	193.76	31			
107	.3750	11.352	204.47	31			
108	.3750	11.397	215.18	31			
109	.3750	11.441	225.89	31			
110	.3750	11.486	236.60	31			
111	.3750	11.530	247.31	31			
112	.3750	11.575	258.03	31			
113	.3750	11.620	268.74	31	6.275	92.22	.0867
114	.3750	11.658	278.03	32			
115	.3750	11.697	287.33	32			
116	.3750	11.736	296.63	32			
117	.3750	11.775	305.93	32			
118	.3750	11.813	315.22	32			
119	.3750	11.852	324.52	32			
120	.3750	11.891	333.82	32			
121	.3750	11.930	343.11	32			
122	.3750	11.968	352.41	32	6.463	83.94	.0936
123	.5000	12.028	6.61	33			
124	.5000	12.087	20.80	33			
125	.5000	12.146	35.00	33			
126	.5000	12.205	49.20	33			
127	.5000	12.264	63.39	33			
128	.5000	12.323	77.59	33	6.655	75.25	.1018

CHANGE FROM HOLES TO SLOTS

NUMBER OF SLOTS 20

SLOT WIDTH= .75 INCHES

DIST FROM INDEX CIR	HEAD IN FT	FLOW IN CFS	AREA IN SF	COEFFICIENTS CD	C	ORIF QN	SLOT QS
12.823	66.31	6.85	.113	.850	.940	6.216	.630
13.198	53.15	7.12	.133	.850	.937	5.540	1.579
13.573	39.30	7.40	.163	.850	.935	4.740	2.656
13.948	27.71	7.62	.203	.849	.932	3.945	3.675

14.198	22.72	7.71	.229	.848	.923	3.525	4.190
14.448	18.92	7.79	.255	.847	.917	3.184	4.602
14.698	16.00	7.84	.281	.846	.911	2.895	4.945
14.948	13.70	7.88	.307	.845	.904	2.646	5.237
15.198	11.88	7.92	.333	.844	.897	2.430	5.486
15.448	10.40	7.94	.359	.842	.889	2.240	5.703
15.698	9.18	7.97	.385	.841	.880	2.072	5.894
15.948	8.18	7.98	.411	.839	.871	1.922	6.063
16.198	7.34	8.00	.437	.837	.861	1.786	6.213
16.448	6.63	8.01	.463	.834	.850	1.664	6.349
16.698	6.02	8.02	.489	.832	.839	1.552	6.471
16.948	5.50	8.03	.515	.830	.827	1.450	6.583
17.198	5.05	8.04	.541	.827	.814	1.356	6.685
17.448	4.65	8.05	.568	.824	.800	1.269	6.779
17.698	4.31	8.05	.594	.821	.786	1.188	6.866
17.948	4.01	8.06	.620	.818	.772	1.113	6.947
18.198	3.74	8.06	.646	.814	.756	1.043	7.021
18.448	3.51	8.07	.672	.811	.740	.978	7.091
18.698	3.29	8.07	.698	.807	.723	.916	7.157
18.948	3.11	8.08	.724	.803	.706	.858	7.218
19.198	2.94	8.08	.750	.799	.688	.803	7.276
19.448	2.78	8.08	.776	.795	.669	.751	7.331
19.698	2.65	8.08	.802	.790	.649	.702	7.382
19.948	2.52	8.09	.828	.786	.629	.656	7.431
20.198	2.41	8.09	.854	.781	.608	.612	7.477
20.448	2.31	8.09	.880	.776	.586	.570	7.521
20.698	2.22	8.09	.906	.771	.564	.530	7.562
20.948	2.13	8.09	.932	.766	.541	.492	7.602
21.198	2.06	8.10	.958	.760	.517	.455	7.640
22.623	1.77	8.10	1.059	.743	.469	.358	7.742

SLOT NUMBER	DISTANCE IN INCHES FROM INDEX CIRCLE TO FIRST RADII POINT	ANGLE FROM INDEX LINE
1	12.523	94.78
2	12.598	112.78
3	12.673	130.78
4	12.748	148.78
5	12.823	166.78
6	12.898	184.78
7	12.973	202.78
8	13.048	220.78
9	13.123	238.78
10	13.198	256.78
11	13.273	274.78
12	13.348	292.78
13	13.423	310.78
14	13.498	328.78
15	13.573	346.78
16	13.648	4.78
17	13.723	22.78
18	13.798	40.78
19	13.873	58.78
20	13.948	76.78

AXIAL LENGTH-CTR TO CTR-OF SLOT RADII= 8.75 INCHES

SLEEVE LENGTH REQD BY ORIFICES AND SLOTS= 23.0238 INCHES

FOR MINIMUM PIPE FRICTION AT Q= 8.1 CFS
PISTON TRAVEL= 13.5733 INCHES IN 400 SECONDS
PERCENT OF FLOW RANGE TRAVEL= 58.9533

DATA FOR NORMAL PIPE FRICTION
MAX FLOW AT END OF HOLES= 6.65458 CFS
THIS IS 53.5242 PERCENT OF TOTAL FLOW RANGE TRAVEL
PISTON TRAVEL TO LAST HOLE= 12.3233 INCHES IN 363.163 SECONDS

TOTAL PISTON TRAVEL TIME IN FLOW RANGE= 678.503 SECONDS
DESIGN RATE FOR PISTON TRAVEL= 29.4696 SECONDS PER INCH

MODULATING RANGE VOLUME= 3958.38 CUBIC FEET

ALLOWABLE STEEL STRESS=15000 PSI

ACTUAL STEEL STRESS AT SLOTS= 153.24 PSI

MULTIJET SLEEVE VALVE COMPUTER PROGRAM "SLEVAL"


```
100 DIM D9(50),R9(50),C9(50),K9(50),C8(50),K8(50),S9(50),O9(50)
102 DIM F4(400),F5(400),X8(400)
108 READ A$
110 PRINT A$;" VALVE STRUCTURE"
112 READ M9,T9,D6,E1,E2,E3
114 Z4=3.14159265359/576
116 K3=0
118 K4=0
120 H=E1-E2
122 H4=E3-E2
123 PRINT " DIA. OF JETS AND SPACING FOR MULTI-JET SLEEVE VALVE"
124 FOR J=1 TO M9 STEP 1
126 READ D9(J),R9(J),C9(J),C8(J)
128 K9(J)=R9(J)/((Z4*C9(J)*D9(J)!2.625)!2*1000)
130 K3=K3+K9(J)
132 K8(J)=R9(J)/((Z4*C8(J)*D9(J)!2.625)!2*1000)
134 K4=K4+K8(J)
136 NEXT J
137 C=0.940
138 C0=0.850
139 O8=0
140 O7=0
142 F7=0
144 Y9=0
146 READ D,Q,T1,D4,N4,D0,D1,K0
147 PRINT " VALVE DIAMETER=";D"INCHES"
148 PRINT
149 P=3*T1
150 N=10
152 G=32.16
154 L=N*P
156 A6=3.14159*(D/2)!2/144
158 V2=Q/A6
160 B9=3.14159*(D6/2)!2/144
162 V4=Q/B9
164 H5=1.84*V2!2/(2*G)+K0*V4!2/(2*G)
166 H9=H-H4-H5-(K4*Q!2)
168 O9=Q/(C*(2*G*H9)!0.5)
170 C1=K3+(((H-H4)-K3*Q!2-H5)/Q!2)
172 L1=N*((3.14159265*(D+2*T1))!2+P!2)!0.5
174 C4=C*(1-(V2)!2/(2*G*H5))
176 C7=C0*(1-(V2/2)!2/(2*G*H5))
178 Q1=P*Q/(L*4)
180 L2=L1/(N*4)
182 N1=4*N
184 N2=0
186 N3=0
188 S8=0
190 X=0
192 Q6=0
194 Z9=0
196 PRINT  USING 198
198: UPS TANK   SLEEVE   RFC TANK  PIPE FRICTION CONSTANTS  TOTAL LESS
200 PRINT  USING 202
202: WS  ELEV  CL  ELEV  WS  ELEV    NORMAL     MINIMUM     SLEEVE VAL
```

```
204 PRINT  USING 206,E1,E2,E3,K3,K4,C1
206: #####.##  #####.##  #####.##  #.########  #.########  ##.########
207 PRINT
208 PRINT " SLEEVE WALL THICKNESS=";T1"INCHES"
209 PRINT
210 PRINT " PITCH OF HELIX=";P "INCHES"
212 PRINT
214 PRINT " LENGTH REQUIRED FOR ";N; "TURNS=";L; "INCHES"
216 PRINT
218 FOR I=1 TO N1 STEP 1
219 N8=1
220 N9=2
221 IF I > 1 THEN 223
222 Q2=Q1
223 V2=Q2/A6
224 V4=Q2/B9
225 H5=1.84*V2!2/(2*G)+K0*V4!2/(2*G)
228 C1=K3+(((H-H4)-K3*Q!2-H5)/Q!2)
229 H1=C1*Q2!2
230 H2=H-H1-H4
232 IF H2 < 1 THEN 999
234 A1=.7854*D1!2/144
236 S=3*D1
238 IF N2 > N3 THEN 272
240 IF I > 1 THEN 268
242 A3=0
244 N6=0
246 H3=0
248 L3=0
250 V=26*(D1/D4)!1.4*0.94*(64.4*H2)!0.5
252 IF V > 20 THEN 286
254 PRINT " JET VELOCITY AT WALL=";V"FEET PER SECOND"
256 PRINT
258 PRINT
260 PRINT  USING 262
262: JET  DIA OF  DIST FROM  ANGLE FROM  QUAD   FLOW    HEAD  REQUIRED
264 PRINT  USING 266
266: NO    JET    INDEX CIR  INDEX LINE   NO   IN CFS  IN FT   AREA SF
268 A2=Q2/(C*(2*G*H2)!0.5)
270 A4=A2-A3
272 N2=A4/A1
274 N3=(L2-(1.5*D1/4))/S
276 IF N2 > N3 THEN 434
278 IF N2 < N8 THEN 286
280 IF N2=N8 THEN 300
282 IF N2 > N9 THEN 294
283 IF N2=N9 THEN 302
284 GO TO 300
286 D1=D1-.0625
288 GO TO 462
290 IF N2 > N9 THEN 294
292 GO TO 300
294 N8=N8+1
296 N9=N9+1
298 GO TO 282
```

```
300 N0=N8
301 GO TO 303
302 N0=N9
303 N6=N6+N0
304 A0=N0*A1+A3
306 O6=A3
308 B0=C!2*(N0*A1+A3)!2*2*G
310 Q0=(B0*(H-H4)/(1+B0*C1))!0.5
312 H0=C1*Q0!2
314 H3=H-H0-H4
316 L5=Q0*L1/Q
318 L4=L5-L3
320 S0=L4/N0
322 IF S0 < S THEN 434
324 L3=L5
326 IF I=4 THEN 330
328 GO TO 348
330 D2=2*T1*.100694705+D1
332 P0=62.4*H3/144
334 A9=N6*T1*(D1+D2)/2
336 A=.7854*(D+2*T1)!2-.7854*D!2
338 T0=T1*(A-A9)/A
340 S3=P0*D/(2*T0)
342 S4=P0*D/(4*T0)
344 S5=(S3!2+S4!2-S3*S4)!0.5
346 IF S5 > 15000 THEN 462
348 A3=A0
350 Q8=Q0-Q6
352 Q7=Q8/N0
354 X0=Q7*L/Q
356 Z8=(360*N*L4)/(N0*L1)
358 FOR J=1 TO N0 STEP 1
360 S8=S8+1
362 X=X+X0
364 IF I>1 THEN 376
366 IF J>1 THEN 376
368 IF D1/2 > X0 THEN 374
370 D5=0
372 GO TO 376
374 D5=D1/2-X0
376 IF O9>=A0 THEN 390
378 IF O7>=O9 THEN 390
380 O8=O8+A1
382 O7=O8+O6
384 IF O7>=O9 THEN 388
386 GO TO 390
388 X9=X
390 Z9=Z8+Z9
392 IF Z9 > 360 THEN 396
394 GO TO 398
396 Z9=Z9-360
398 IF J=N0 THEN 402
400 GO TO 408
402 PRINT  USING 404,S8,D1,X,Z9,I,Q0,H3,A3
404: ###  #.####  ###.###        ###.##   ###  ###.###  ###.## ###.####
```

```
406 GO TO 414
408 PRINT  USING 410,S8,D1,X,Z9,I
410: ###  #.####  ###.###        ###.##   ###
412 NEXT J
414 IF I=4 THEN 418
416 GO TO 428
418 PRINT
420 PRINT " ALLOWABLE STEEL STRESS IN SLEEVE=15000 PSI"
422 PRINT
424 PRINT " ACTUAL STEEL STRESS IN SLEEVE=";S5; "PSI"
426 PRINT
428 Q2=Q2+Q1
430 Q6=Q0
432 GO TO 476
434 D1=D1+.0625
436 IF D1 > T1 THEN 450
438 V1=26*(D1/D4)!1.4*0.94*(64.4*H3)!0.5
440 IF V1 > 20 THEN 443
442 GO TO 234
443 PRINT " VELOCITY AT WALL IS EXCESSIVE-V1=";V1"FT/SEC-REDESIGN"
444 PRINT
445 GO TO 999
446 PRINT " INCREASE D4 BECAUSE D1 IS LESS THAN .1875 INCHES"
448 GO TO 999
450 L6=L1/N
452 PRINT
454 PRINT " CHANGE FROM HOLES TO SLOTS"
456 PRINT
458 F6=Q0
460 X6=X
462 I=N1
463 IF D1<.1875 THEN 446
464 IF S5 > 15000 THEN 470
466 IF N2 < N8 THEN 478
468 GO TO 480
470 PRINT " STEEL STRESS=";S5;"PSI-PROGRAM WILL INCREASE T1"
472 T1=T1+.0625
474 GO TO 148
476 NEXT I
478 GO TO 218
480 S1=3*D0*L6/(3.14159265*(D+2*T1))
482 S2=(D0-T1)/2
484 Z1=1
486 Z2=2
488 M0=0
490 L0=0
492 PRINT " NUMBER OF SLOTS";N4
494 PRINT
496 PRINT " SLOT WIDTH=";D0;"INCHES"
498 PRINT
500 PRINT  USING 502
502: DIST FROM   HEAD   FLOW     AREA  COEFFICIENTS    ORIF     SLOT
504 PRINT  USING 506
506: INDEX CIR  IN FT  IN CFS   IN  SF   CO    C        QN       QS
508 C3=C
```

```
510 C2=C0
512 W4=0
514 U8=N4/4
516 W4=W4+1
518 IF U8>W4+1 THEN 516
520 FOR K=1 TO 4
522 IF U8=W4 THEN 526
523 IF U8=W4-0.75 THEN 529
524 IF U8=W4-0.50 THEN 533
525 IF U8=W4-0.25 THEN 537
526 U(K)=U8
527 IF K=1 THEN 549
528 GO TO 551
529 IF K=1 THEN 541
530 IF K=2 THEN 541
531 IF K=3 THEN 544
532 IF K=4 THEN 544
533 IF K=1 THEN 546
534 IF K=2 THEN 544
535 IF K=3 THEN 546
536 IF K=4 THEN 544
537 IF K=1 THEN 546
538 IF K=2 THEN 546
539 IF K=3 THEN 541
540 IF K=4 THEN 541
541 U(K)=U8+.5
542 IF K=1 THEN 549
543 GO TO 551
544 U(K)=U8
545 GO TO 551
546 U(K)=U8+1
547 IF K=1 THEN 549
548 GO TO 551
549 A5(K)=U(K)*0.7854*D0!2/288+((U(K)-1)*P/N4*(U(K)-1)*D0)/288
550 GO TO 576
551 A5(K)=U(K)*0.7854*D0!2/288+(U(K)*P/N4*U(K)*D0)/288
576 X2(K)=U(K)*P/N4
578 IF K=2 THEN 590
580 IF K=3 THEN 594
582 IF K=4 THEN 598
584 A7=A5(K)
586 X=X2(K)+X6+S2
588 GO TO 642
590 A7=A7+A5(K)+(U(K-1)-1)*X2(K)*D0/144
592 GO TO 600
594 A7=A7+A5(K)+(U(K-1)+(U(K-2)-1))*X2(K)*D0/144
596 GO TO 600
598 A7=A7+A5(K)+(U(K-1)+U(K-2)+(U(K-3)-1))*X2(K)*D0/144
600 X=X+X2(K)
602 IF K=4 THEN 606
604 GO TO 642
606 Y9=1
608 X3=X
610 D7=A7
612 GO TO 642
```

```
613 Y9=2
614 A7=A7+D7
615 B1=(C4*A3+C7*A7)!2
616 Q3=(2*G*B1*(H-H4)/(1+2*G*C1*B1))!0.5
617 V3=Q3/A6
618 V2=V3
619 V4=Q3/B9
620 H5=1.84*V2!2/(2*G)+K0*V4!2/(2*G)
622 C1=K3+(((H-H4)-K3*Q!2-H5)/Q!2)
624 H3=H-C1*Q3!2-H4
626 IF H3>H5 THEN 640
628 Q4=C4*A3*(2*G*(H3-V3!2/(2*G)))!0.5
630 Q5=Q3-Q4
632 X=X+X3-X6-P/N4-S2
634 C0=C7
636 C=C4
638 GO TO 686
640 A7=D7+X4*N4*D0/144
642 C=C3*(1-V6!2/(2*G*H3))
643 C0=C2*(1-(V5/2)!2/(2*G*H3))
644 B1=(C*A3+C0*A7)!2
645 Q3=(2*G*B1*(H-H4)/(1+2*G*C1*B1))!0.5
646 V3=Q3/A6
647 V2=V3
648 V4=Q3/B9
649 H5=1.84*V2!2/(2*G)+K0*V4!2/(2*G)
651 C1=K3+(((H-H4)-K3*Q!2-H5)/Q!2)
652 H3=H-C1*Q3!2-H4
654 Q4=C*A3*(2*G*(H3-V3!2/(2*G)))!0.5
656 Q5=Q3-Q4
658 V5=Q5/A6
660 V6=(V5+V3)/2
662 IF Y9<2 THEN 670
664 X=X+M0
666 X4=X-X3+0.25
668 GO TO 672
670 X4=0.25
672 IF F7>=0 THEN 686
674 F9=C*A3*((2*G*H9)!0.5)
676 F8=C0*A7*((2*G*H9)!0.5)
678 F7=F8+F9
680 IF F7>=0 THEN 684
682 GO TO 686
684 X9=X
686 A8=A3+A7
688 IF K=1 THEN 694
690 F4(K)=F5(K-1)
692 GO TO 698
694 F4(K)=F6
696 C5=1
698 PRINT  USING 700,X,H3,Q3,A8,C0,C,Q4,Q5
700:  ###.###  ###.## ####.##  ###.###  .###  .###  ###.### ####.###
702 IF K>1 THEN 710
704 X8(K)=X-X6
706 X7=X8(K)+X6
```

```
708 GO TO 714
710 X8(K)=X-X7
712 X7=X7+X8(K)
714 F5(K)=Q3
716 C5=C5+1
718 IF Y9=1 THEN 722
720 NEXT K
722 K=K+1
724 IF C=C4 THEN 732
726 M0=0.25
728 IF Y9=2 THEN 614
730 GO TO 613
732 Z7=360*S1/L6
734 Z9=Z9+Z7
736 IF Z9 > 360 THEN 740
738 GO TO 742
740 Z9=Z9-360
742 PRINT
744 PRINT  USING 746
746:  SLOT   DISTANCE IN INCHES FROM INDEX   ANGLE FROM
748 PRINT  USING 750
750: NUMBER   CIRCLE TO FIRST RADII POINT    INDEX LINE
752 Z6=360/N4
754 X1=P/N4
756 X5=X6+S2+P/N4
758 FOR J=1 TO N4 STEP 1
760 IF J > 1 THEN 764
762 GO TO 778
764 Z9=Z9+Z6
766 IF Z9 > 360 THEN 770
768 GO TO 776
770 Z9=Z9-360
772 IF J>1 THEN 776
774 GO TO 778
776 X5=X5+X1
778 PRINT  USING 780,J,X5,Z9
780:   ###              ###.###                    ###.##
782 NEXT J
784 PRINT
786 L8=X-X6-S2-(P*(N4-1)/N4)
788 PRINT " AXIAL LENGTH-CTR TO CTR-OF SLOT RADII=";L8; "INCHES"
790 PRINT
792 L7=D5+X+D0/2
794 PRINT " SLEEVE LENGTH REQD BY ORIFICES AND SLOTS=";L7;"INCHES"
796 PRINT
798 P7=X6/L7*100
800 T8=T9*X6/X9
802 T6=T9/X9
804 P9=X9/L7*100
806 T5=T6*L7
808 PRINT " FOR MINIMUM PIPE FRICTION AT Q=";Q "CFS"
810 PRINT " PISTON TRAVEL=";X9"INCHES IN";T9"SECONDS"
812 PRINT " PERCENT OF FLOW RANGE TRAVEL=";P9
814 PRINT
816 PRINT " DATA FOR NORMAL PIPE FRICTION"
```

```
818 PRINT " MAX FLOW AT END OF HOLES=";F6"CFS"
820 PRINT " THIS IS";P7"PERCENT OF TOTAL FLOW RANGE TRAVEL"
822 PRINT " PISTON TRAVEL TO LAST HOLE=";X6"INCHES IN";T8"SECONDS"
824 PRINT
826 PRINT " TOTAL PISTON TRAVEL TIME IN FLOW RANGE=";T5"SECONDS"
828 PRINT " DESIGN RATE FOR PISTON TRAVEL=";T6"SECONDS PER INCH"
830 PRINT
832 V8=0
834 V9=X6*T6*F6/2
836 FOR K=1 TO C5
838 V7=(F4(K)+F5(K))/2*T6*X8(K)
840 V8=V8+V7
842 NEXT K
844 V0=(V8+V9)*1.10
846 PRINT " MODULATING RANGE VOLUME=";V0"CUBIC FEET"
848 PRINT
850 D3=2*T1*.100694705+D0
852 B8=N4*T1*(D0+D3)/2
854 T=(A-B8)*T1/A
856 P1=62.4*H3/144
858 S6=P1*D/(4*T)
860 B=2*D0
862 B2=B-2*T1*.100694705
864 W=P1*B
866 L9=L8+DC
868 M=W*L9!2/12
870 C6=(T1*(2*B+B2))/(3*(B+B2))
872 I1=(T1!3*(B!2+4*B*B2+B2!2))/(36*(B+B2))
874 F=M*C6/I1
876 S7=S6+F
878 IF S7 > 15000 THEN 890
880 PRINT " ALLOWABLE STEEL STRESS=15000 PSI"
882 PRINT
884 PRINT " ACTUAL STEEL STRESS AT SLOTS=";S7; "PSI"
886 PRINT
888 GO TO 999
890 PRINT " STEEL STRESS AT SLOTS=";S7;"PSI-THIS IS TOO HIGH"
892 PRINT
894 PRINT " INCREASE THICKNESS OF SLEEVE AND RERUN PROGRAM"
896 GO TO 999
900 DATA DEEP RED RUN RFC
901 DATA 3,400,24,1443.0,1193.0,1214.0
902 DATA 24,18118,.370,.410
903 DATA 18,21030,.345,.400
904 DATA 16,12370,.345,.400
905 DATA 14,8.1,0.5,19.5,20,0.75,.375,1.340
999 END
```

ABSTRACT

Hydraulic laboratory studies were conducted to test a 200-mm inline polyjet valve and to develop and test a 200-mm horizontal multijet sleeve valve and stilling chamber. The study results demonstrate the capability of the horizontal multijet sleeve valve and stilling chamber to perform well as an energy dissipator and also deliver design discharges with minimum head loss. A computer program was developed to size and locate the nozzles and slots used in the ported sleeve of the multijet sleeve valve. This multijet concept of valve controls which results in the production of cavitation in the water of the stilling chamber, away from structural members, permits design consideration of high head (100 to 300 m), one-stage flow control installations.

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