4.20 5/19/81 T.S.

(2)

Lh. 2654

R4433

SERI/TR-631-984

April 1981

Water Flows from Slotted Pipes

Douglas A. Olson

SERI

Solar Energy Research Institute
A Division of Midwest Research Institute

1617 Cole Boulevard
Golden, Colorado 80401

Operated for the
U.S. Department of Energy
under Contract No. EG-77-C-01-4042

Printed in the United States of America
Available from:
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161
Price:
Microfiche $3.00
Printed Copy $4.50

SERI/TR-631-984
UC CATEGORY: UC-64

MASTER

WATER FLOWS FROM SLOTTED PIPES

DOUGLAS A. OLSON

APRIL 1981

PREPARED UNDER TASK NO. 1110.00

Solar Energy Research Institute
A Division of Midwest Research Institute

1617 Cole Boulevard
Golden, Colorado 80401

Prepared for the
U.S. Department of Energy
Contract No. EG-77-C-01-4042

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

PREFACE

This report was prepared as part of Task 1111.00 in the Solar Thermal Research Branch. Its objective is to provide the Ocean Systems Program Office with analytical and experimental results to help them decide which Claude-cycle, open-cycle, ocean thermal energy conversion concept (OC-OTEC) performs the best. In this report we investigate the performance of a slotted pipe for generating uniform falling jets, as would be used in an OC-OTEC falling jet evaporator or condenser. I thank Dave Johnson and Abe Kogan for conceiving the experiments and critiquing the results and greatly apreciate the assistance of Brian Boyer, Jim Green, and Sung-Ho Jo.

Douglas A. Olson

Douglas A. Olson
Associate Scientist

Approved for

SOLAR ENERGY RESEARCH INSTITUTE

Frank Kreith

Frank Kreith, Chief
Solar Thermal Research Branch

Barry L Butler for DWK

D. W. Kearney, Manager
Solar Thermal, Ocean, and Wind Division

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

SUMMARY

Objective

The objective of the report is to determine the length of a slotted pipe needed to generate falling jets of water of uniform velocity, perpendicular to the pipe.

Discussion

We conducted experiments on the water flow from a pipe 6.1 m long, with a 6.3-cm inside diameter and a 0.64-cm wide slot to determine how uniformly the flow exits the slot. Dimensions are representative of those we anticipated in a falling jet evaporator or condenser that would be used in a Claude-cycle, open-cycle, ocean thermal energy conversion (OC-OTEC) plant. We measured the pressure along the pipe, flow from the slot, and angle of flow from the slot for variable slot lengths (1.5 m to 4.6 m) and variable flow rates (6.0 kg/s to 17 kg/s). We used a one-dimensional control volume analysis to model the flow from the pipe.

Conclusions and Recommendations

Experimental results show that most of the flow exits from the region of pipe next to the closed end, regardless of total flow rate or length of slot. Except near the closed end, the water exits with a very high axial component of velocity. The analysis indicates that as the ratio of jet area (slot width times length) to pipe cross-sectional area becomes much greater than two, pressure recovery will be large near the closed end, owing to the deceleration of the flow in the pipe. Pressure recovery produces the large increases in the jet flow. The region of high perpendicular jet flow we observed in the experiment is within this region of strong pressure recovery. We recommend that if this method is used in an OC-OTEC plant for generating falling jets, the slot length should be 1 m or less (for the pipe diameter and slot width noted previously). The analytical model can be used to predict the maximum practical slot length for pipes of different dimensions.

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF FIGURES (continued)

LIST OF TABLES

SECTION 1.0

INTRODUCTION

One way to distribute water in the open-cycle Ocean Thermal Energy Conversion (OTEC) falling jet evaporator and falling jet condenser is to have a number of radial manifolds feed circular coaxial slotted tubes (Kogan 1980). See Fig. 1-1. Water flowing along the tubes and out the slots becomes the jets which evaporate or condense the steam to drive the turbine. The coefficients of heat transfer for the falling jets are believed to be very sensitive to the flow properties of the jets; e.g., Reynolds number, jet width, level of turbulence, etc. In its heat and mass transfer laboratory, the Solar Energy Research Institute (SERI) is experimenting to determine the optimal flow conditions. The water distribution system should be designed so that each portion of the slotted tube will deliver the same amount of flow. Otherwise, parts of the tube will be wasted and the evaporator or condenser will have to be built larger than necessary, thus increasing cost. Senecal (1957) has shown that the flow from a slotted pipe becomes maldistributed under certain conditions. In a full-size OTEC plant, preliminary calculations based on assumptions of heat transfer coefficients and head losses indicate that an optimal jet flow rate would be 10 to 20 kg/s per m of slot. Pipes might be sized at a 5- to 8-cm diameter, with slots 0.3 to 1.0 cm wide and as long as possible. In this report we present results of experiments and analyses that indicate how successfully a slotted pipe can provide uniform flow to a falling jet.

1.1 THEORETICAL ANALYSIS

We first performed a one-dimensional analysis of the flow from a slotted pipe (see Appendix) to identify the important geometrical and flow parameters. We wrote integral momentum and continuity equations for the flow along the tube and out the slot, assuming no downward component of flow in the tube and an axial component of velocity outside the tube equal to the velocity inside the tube. The Reynolds numbers in slotted pipes sized to OTEC plant dimensions could vary from 50,000 to 1,000,000, depending on pipe length. We used a simple turbulent friction relation with a constant friction coefficient. We used friction factors of $f = 0.04$ (rough pipe with $e/D = 0.01$), $f = 0.02$, and $f = 0.01$ (smooth pipe). For a tube diameter, length, and slot width similar to those anticipated for the full-scale evaporator, results of the analysis showed that the pressure first dropped in the tube due to friction, then rose again at the closed end as the rapidly decelerating flow converted kinetic energy into pressure. This change in pressure varied the jet velocity, first constant or slightly decreasing toward the middle of the tube, then increasing again at the closed end (Fig. 1-2). For friction coefficients corresponding to smooth pipes, the decrease in pressure and jet velocity from friction was much smaller than the pressure increase resulting from flow deceleration. Increasing or decreasing the total flow through the tube also had no effect on pressure recovery length or on the relative ratios of velocities of the exiting jet. This decoupling of the velocity from the solution was due to our assumption of a simple turbulent friction correlation in which the friction factor was independent of Reynolds number. When we used the Blasius turbulent friction formula for a smooth pipe instead of the constant friction

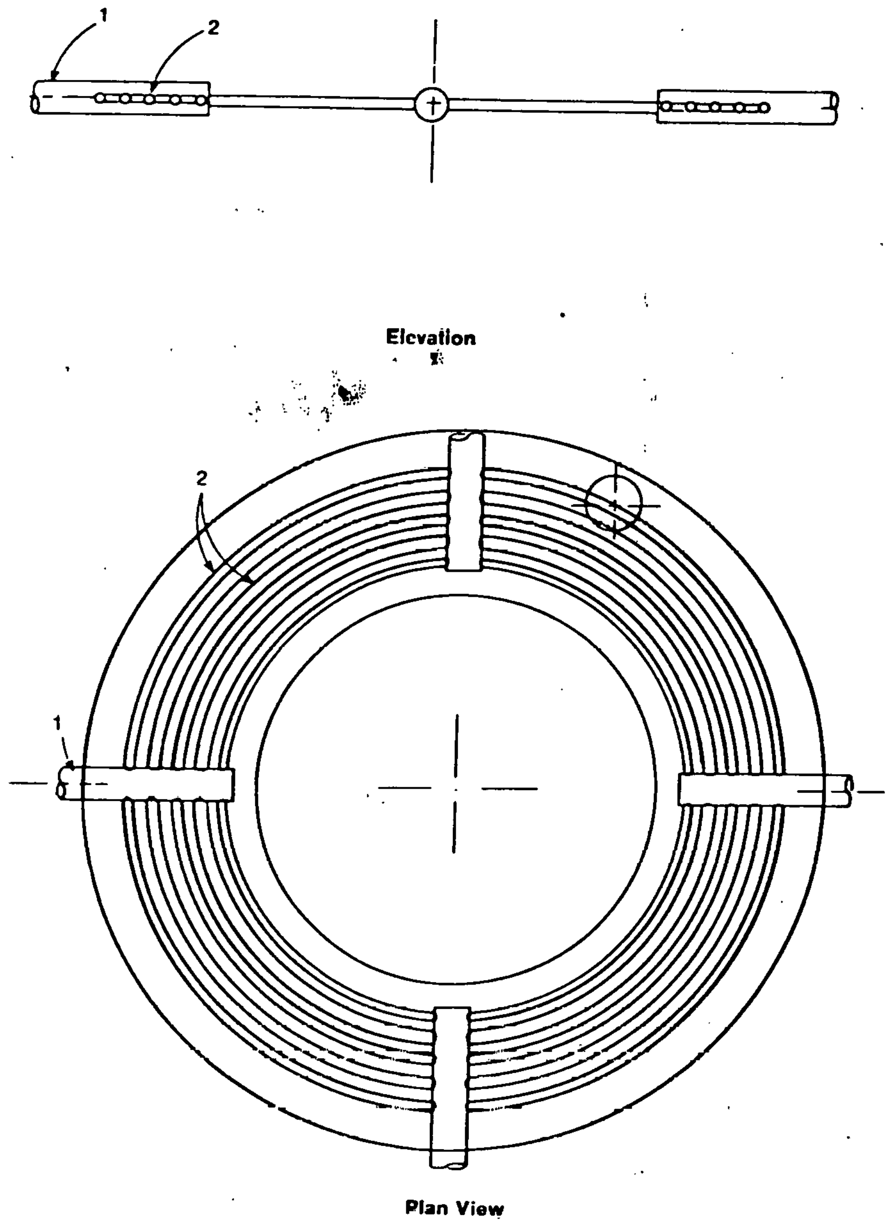

Figure 1-1. Falling Jet Evaporator Configuration

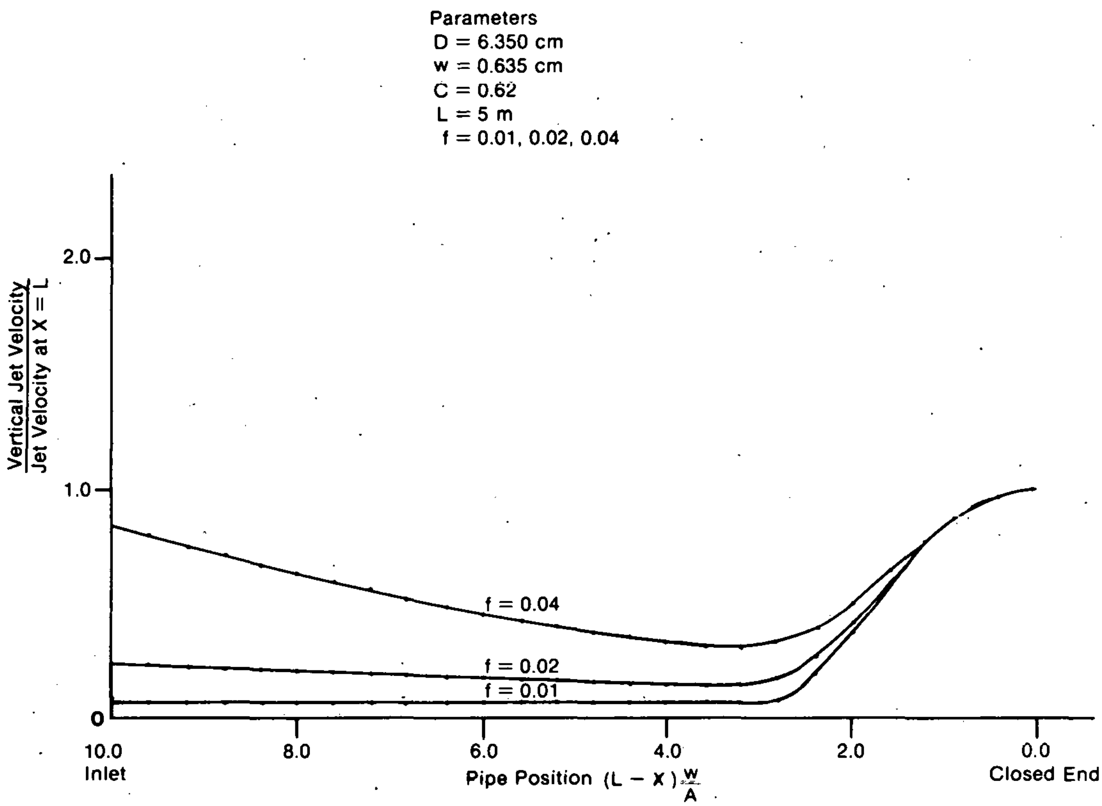

Figure 1-2. Computer Analysis of the Flow from a Slotted Pipe

coefficient, the velocity distribution was dependent on the magnitude of the flow rate. An increase in total flow increased the jet flow predominantly near the closed end.

The analysis shows that the important geometrical parameter is the ratio of slot area to flow area, which we will call the area ratio:

$$A_L = \frac{w \cdot L}{A} ,$$

where

w = slot width,
L = slot length,
A = flow area, $= \frac{\pi D^2}{4}$, and
D = inside tube diameter.

Also, let $A_x = [w(L - x)/A]$, which is a measure of the distance from the closed end of the tube. For $A_x \leq 2$, the pipe is dominated by pressure recovery; only for $A_x \leq 1$ does the jet velocity fluctuate by 10% or less of the mean. For a slotted tube with a 6.35-cm diameter and a 0.635-cm wide slot, $A_L = 2$ when the slot is 1 m long. In addition, if we assume that the jet carried the same axial velocity as the flow in the pipe at the same position, only near the closed end ($A_x \leq 1$) was the angle of the jet to the horizontal greater than 45°. See Fig. A-4 in the Appendix.

1.2 LABORATORY-SCALE EXPERIMENTS

Since the analysis was subject to several simplifying assumptions, we conducted experiments on flow through slotted tubes to confirm the accuracy of the analysis and determine the advantages or limitations of this water distribution system. We performed lab-scale experiments using a 3.18-cm inside diameter plexiglass tube with a 0.476-cm slot that was 34.3 cm long ($A_L = 2.1$). One end was closed and the other was connected to the water faucet. Results showed that most of the water exited the tube near the closed end and that the flow had a strong axial component of velocity after it left the slot.

1.3 SCOPE OF THE REPORT

Although the lab experiments qualitatively confirmed our understanding of the flow, the flow rates and pipe sizes were not the same as those anticipated for a full-size Ocean Thermal Energy Conversion (OTEC) plant. Thus, an experiment was performed using a slotted pipe of anticipated OTEC dimensions, with pipe flows and jet speeds expected in the full-scale plant. This report describes the experiments, which were performed 23-25 July 1980. We studied the flow from a slotted pipe with a 6.27-cm diameter and a 0.635-cm slot of variable length, 0 to 5.8 m ($A_L = 11.9$). From results of these studies, we suggested recommendations for the OTEC evaporator and condenser distribution system.

SECTION 2.0

EXPERIMENTAL PROCEDURE

The experimental set-up consisted of a test section of slotted piping with dimensions appropriate for modeling the OTEC distribution system. Fire hose was used to connect the pipe to a fire hydrant, with a diaphragm valve to control the water flow rate from the hydrant and instruments to measure the pressure along the pipe and the flow into the pipe and out the slots.

The slotted pipe was a schedule-40 carbon steel pipe that was 6.1 m long, with a 6.27-cm inside diameter and a 7.30-cm outer diameter. A steel plate was welded over one end, and the open end was connected through a reducing section to a 5.08-cm diaphragm flow control valve. Along one side of the pipe were cut 19 slots, each 29.2 cm long and 0.635 cm wide, with 1.27 cm spacers between slots to prevent the pipe from warping. We varied the length of exposed slots by wrapping duct tape around the pipe, closing off those unnecessary for the experiment. Eleven pressure taps were fitted along the pipe, 90° off-center from the slots (hence, when the slots were pointed vertically, the pressure ports were horizontal). The pressure taps were located at the midpoints of slots 1, 3, 5, 7, 9, 11, 13, 15, 17, 18, and 19 (slot 1 was next to the pipe inlet and slot 19 next to the closed end). See Fig. 2-1.

We conducted the experiment at the pond east of Cole Blvd. near Building 4 of the Denver West Office Park. A fire hydrant provided water to the experiment through 75 m of 7.62-cm fire hose at flow rates of 19 kg/s and less. Located between the diaphragm valve and the fire hose was a flow totalizer, provided by Consolidated Water Company, that measured the total flow through the pipe. We rested the two ends of the pipe on stepladders, 1 m above the ground to let us observe the flow of the jets and collect water in a large drum.

The following quantities were measured for each experiment:

- Total flow into the slotted pipe, which was measured using the flow totalizer provided by Consolidated Water.
- Flow out of the slots. We quickly inserted a 65-L or 200-L drum beneath each slot and measured the time necessary to fill the drum.
- Pressure variations along the pipe. The pressure taps were connected through plastic tygon tubes to two mercury manometers. One continuously monitored the pressure difference between taps 1 and 2 of the pipe. The other was connected through a manifold system to measure the pressure difference successively between tap 1 and atmosphere, tap 1 and tap 2, tap 1 and tap 3, etc. This allowed calculation of the tap pressure minus atmospheric pressure for each tap position in the pipe $[P_x - P_{atm} = (P_1 - P_{atm}) - (P_1 - P_x)]$. After the data were collected, a picture was taken of the pipe and its flow for each experiment.

We conducted seven different experiments. In the first two, 5 slots were uncovered by the tape ($A_L = 3.0$), the first with a high flow rate and the second with a low flow rate. In experiments 3 and 4, 10 slots were exposed ($A_L = 6.0$), again with a high and then a low flow rate. After the first two

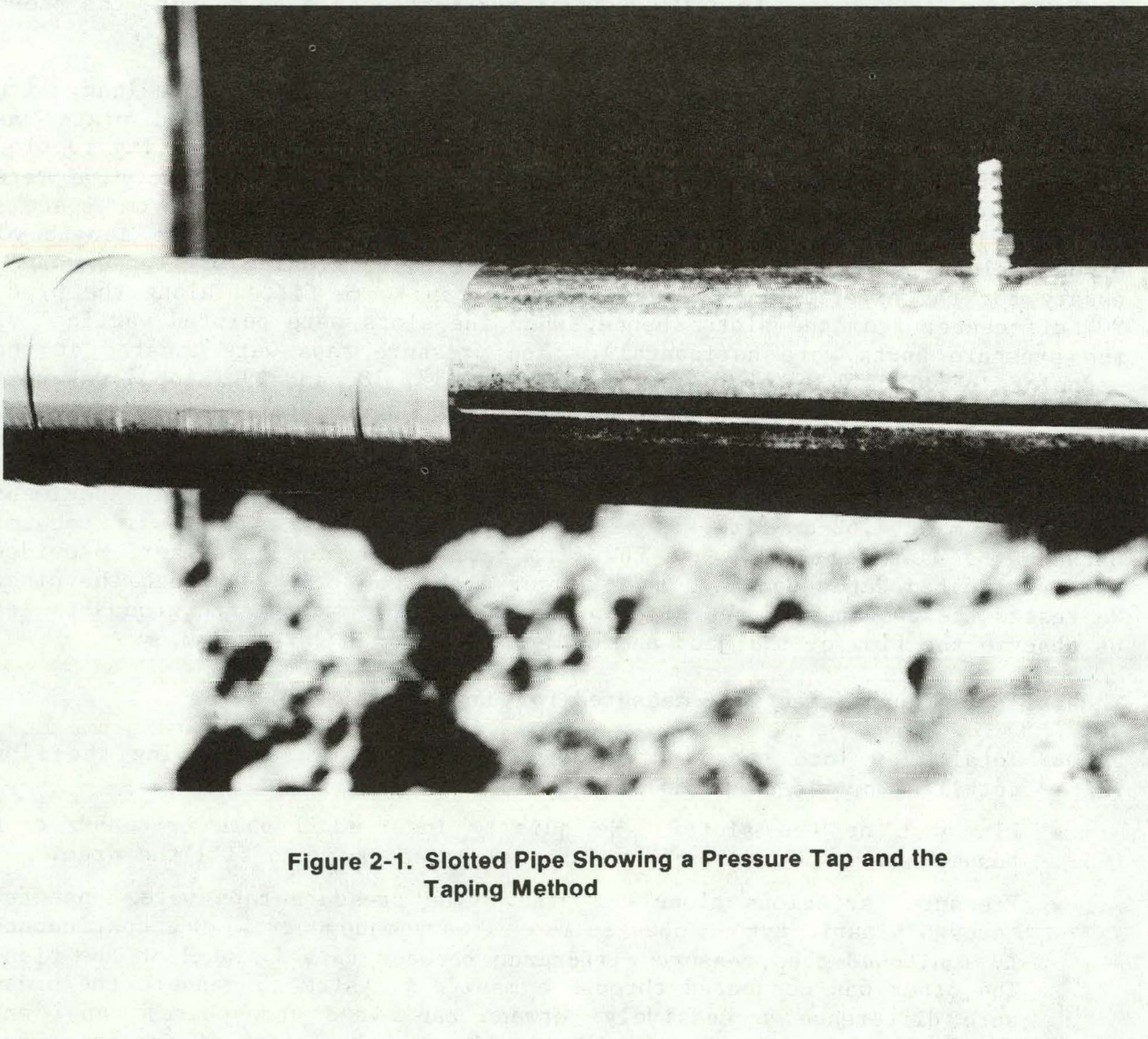

Figure 2-1. Slotted Pipe Showing a Pressure Tap and the Taping Method

experiments were completed, some errors in incoming and outgoing flow were discovered; therefore experiments 5 and 6 repeated the 5-slot experiments. Finally, experiment 7 had 15 slots exposed ($A_L = 9.0$). The fire hydrant could not deliver high enough flows to make the flow per meter of slot for the 10- and 15-slot experiments directly applicable to conditions expected in the OTEC plants. However, trends in the data were clear enough so that we could also characterize the performance of long, slotted pipes.

SERI

SECTION 3.0

RESULTS OF THE EXPERIMENTS

The data for the experiments are collected in Tables 3-1 and 3-2. For all experiments, regardless of the number of slots exposed, or the flow rate through the slots, most of the water flowed out the last two or three slots. Also, after leaving the slot, the water jets retained a large component of axial velocity of the same order as the axial velocity inside the pipe. Only at the closed end of the pipe, in the last two jets, was the angle of the jet to the ground significantly larger than 45°. These two effects limit the length of pipe that can deliver a uniform flow.

3.1 EXPERIMENTS WITH A SHORT PIPE (A_L = 3.0)

Figures 3-1a and b are pictures of the first experiment, which was a high flow through five slots (A_L = 3.0). With the jets pointed downwards, the slot next to the closed end (A_x = 0.60) delivers a fairly uniform flow. However, the successive slots further upstream deliver less and less flow, which becomes increasingly horizontal. The gaps between the jet sheets are due to the spacers separating the slots. In the furthest upstream jets, the vertical component of velocity mainly resulted from the jet hitting the spacer and bouncing downward. With the jet pointed upward, the water carries beyond the closed end of the pipe. The height of the jets gives an estimate of the vertical component of the velocity as the water leaves the slot. This height is negligible at the first slot, but increases rapidly as the flow approaches the closed end. This trend is expected from the theoretical analysis, which predicts a rapid pressure recovery as the flow decelerates, producing a larger pressure differential to drive the flow out the slot, resulting in greater mass flows and vertical jet velocities as the closed end is approached. Measurements from Fig. 3-1b of jet angles roughly correlate with the angle predicted if the jet carried with it all the axial velocity of the flow inside the pipe (see Table 3-3).

Figure 3-1c shows the measured pressure changes and slot flows for the first experiment, plotted as a function of position along the pipe. The first three-fourths of the pipe had no exposed slots, and the pressure drops uniformly owing to friction. Near the closed end and in the region of exposed slots, the pressure increases. The measured flow varies from 6.2 kg/s (20.7 kg/s-m) in the last slot to 2.5 kg/s (4.2 kg/s-m) collected in the combined first two slots. Even in the last three slots, the flow varies by ±20% from the amount collected from the penultimate slot. Hence, if no greater than a 20% variation in flow could be tolerated, the pipe would have to be limited to A_L = 1.8, or 1 meter. If no greater than a 10% variation were permissible, only a 0.3-m pipe (1-slot length) would be acceptable. The fire hydrant delivered sufficient flow for the five-slot experiment to match the flow per unit length anticipated in a full-size OTEC plant (10 to 20 kg/s-m).

Table 3-1. SLOT PRESSURE MINUS ATMOSPHERIC PRESSURE (KPa)

Run	Position*, A_x 11.11	9.91	8.71	7.51	6.31	5.10	3.90	2.70	1.50	0.90	0.30	Length of Exposed Slots**, A_L
1	13.69	10.85	8.13	6.16	4.06	3.44	1.22	-0.51	2.45	6.03	8.01	3.0
2	5.78	4.18	2.94	2.45	1.22	0.85	-0.02	-0.64	0.97	1.22	1.22	3.0
3	8.25	5.17	2.94	1.46	-0.76	-1.01	-1.01	-0.76	1.83	2.94	2.94	6.0
4	3.44	1.96	0.85	0.60	-0.39	-0.64	-0.64	-0.27	-0.39	-0.39	-0.14	6.0
5	17.64	14.18	11.09	9.37	6.40	4.80	2.20	0.35	3.93	9.24	11.59	3.0
6	3.69	2.94	2.08	1.71	0.97	0.60	0.10	-0.14	0.43	0.85	1.83	3.0
7	4.80	0.97	-2.61	-2.20	-2.24	-1.26	-1.26	-0.39	0.47	0.97	1.46	9.0

$*A_x = (L - x)\,\frac{W}{A}$

$**A_L = \frac{LW}{A}$

Table 3-2. FLOW RATES OUT SLOTS (kg/s)

Run	Position, A_x 8.71	8.11	7.51	6.91	6.31	5.70	5.11	4.50	3.90	3.30	2.70	2.10	1.50	0.90	0.30	Total	Flow Meter	Length of Exposed Slots, A_L
1											-2.541-		3.95	5.13	6.20	17.81	13.80	3.0
2											-2.09-		2.84	2.95	2.89	10.78	5.94	3.0
3						0.40	0.59	0.62	0.71	0.72	0.64	1.82	3.48	3.74	4.02	16.75	13.72	6.0
4						0.36	0.37	0.44	0.65	1.12	1.49	1.59	1.35	1.32	1.07	9.76	8.51	6.0
5											0.65	1.66	4.29	5.85	7.52	19.96	16.52	3.0
6											0.51	1.24	2.28	2.07	2.00	8.11	7.19	3.0
7	0.66	0.80	0.60	0.57	0.75	0.63	0.60	0.61	0.64	0.76	1.43	2.68	2.76	2.71	2.50	18.80	16.14	9.0

Figure 3-1a. High Flow Rate from a Slotted Pipe with 5 Slots Exposed, A_L = 3.0, 13.8 kg/s Total Flow (Slots Pointed Downward)

Figure 3-1b. High Flow Rate from a Slotted Pipe with 5 Slots Exposed, A_L = 3.0, 13.8 kg/s Total Flow (Slots Pointed Upward)

Table 3-3. ANGLES OF JETS TO HORIZONTAL

	Position, A_x				
Angle	2.70	2.10	1.50	0.90	0.30
Measured Angle[a]	8°	15°	29°	51°	74°
Predicted angle based on Vertical jet velocity and Horizontal pipe velocity	3°	9°	25°	44°	73°

[a]Run 1; High flow rate through 5 slots.

At $A_x \approx 2.7$, the measured pressure difference, $P_x - P_{atm}$, is negative, suggesting the pipe was sucking in air instead of forcing out water. We measured negative pressure differences in other experiments at locations where the jet flow rate was low (particularly experiments with low total flow rates and large numbers of exposed slots). However, we think this effect is a measurement error, since we observed flow from those slots, which indicated that the pressure was greater, rather than less, than atmospheric pressure. Three possible explanations account for the discrepancy. First, if an air bubble lodged in the tubeline to pressure tap number 1 after $P_1 - P_{atm}$ was measured (the tap from which all measurements were referenced), the difference $P_1 - P_x$ would read too large, and the difference $P_x - P_{atm} = (P_1 - P_{atm}) - (P_1 - P_x)$ would read too small. However, we believe this is unlikely, since we carefully bled the lines. Second, errors in the head difference between the manometer and the slotted pipe, resulting from incorrectly establishing the level of the slotted pipe, could explain negative pressure differences. To measure the pressure difference between pressure tap number 1 and the atmosphere, we established the relative levels of the manometer and the pipe by holding a board between the slotted pipe and the manometer (located 3 m away), and leveling it with a carpenter's level. Accurate transit equipment was not available. If this head difference were measured too small, $P_1 - P_{atm}$ would be too small and $P_x - P_{atm}$ would also be too small. Third, we established the head difference to the manometer only at the pipe inlet; it was not referenced at the closed end. We checked the levelness of the pipe with a carpenter's level. If the pipe were to slope downward, the head difference at the closed end would be smaller than at the inlet, and $P_x - P_{atm}$ would be measured too small. We believe the second explanation is most likely, since it represents the crudest measurement. Hence, the absolute magnitudes of the pressures would be offset by a few kilopascals, but the relative variations in the pressure would be unchanged.

Figure 3-2a shows the flow resulting from Experiment 2 (five slots exposed) and a reduced flow rate from Experiment 1. In comparison to Fig. 3-1b for the higher flow rate, heights were reduced at the closed end, indicating that the reduced total flow reduces the flow out the end slots. The theoretical model predicted this effect when the Blasius formula for the friction factor was used instead of constant friction factor. Figure 3-2b shows the measured flows and pressures as a function of position. The measured flow out the slot was roughly constant over the last three slots, and much lower for the first two compared with the final three slots.

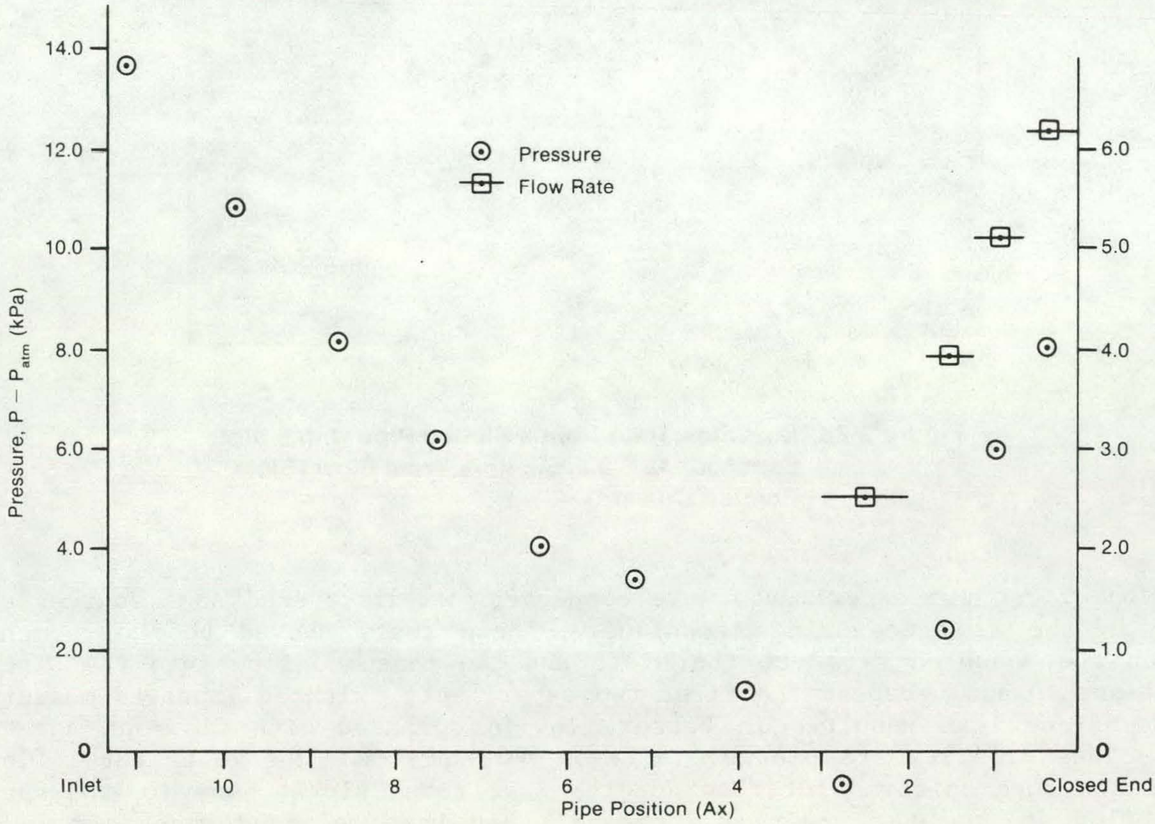

Figure 3-1c. Experiment 1 Results (5 Slots Exposed, A_L = 3.0, 13.8 kg/s Total Flow)

Figure 3-2a. Low Flow Rate from a Slotted Pipe with 5 Slots Exposed, A_L = 3.0, 5.9 kg/s Total Flow (Slots Pointed Upward)

After the first two experiments were completed, we discovered some sources of error in the flow measuring technique. These tests showed poor agreement between the measured flow out the slots and the measured flow into the pipe. Experiments 5 and 6 repeat the first two experiments, with an improved measuring technique that resulted in better flow-in compared with flow-out agreement. These results are plotted in Figs. 3-3 and 3-4. The lower total flow produces a more uniform slot flow out the last three slots; however, the specific flow is somewhat lower (6.7 kg/s-m) than that believed necessary in a full-scale OTEC plant (10-20 kg/s-m). The higher flow experiment again has a large increase in slot flow from the inlet to the closed end. We concluded that increasing the flow to a magnitude required for an OTEC plant increases the nonuniformities in the flow, and a slotted pipe with $A_L > 2$ has excessive flow variations.

3.2 EXPERIMENTS WITH A MEDIUM LENGTH PIPE (A_L = 6.0)

Figures 3-5a and 3-5b show the flow resulting from 10 slots exposed and a high total flow rate. As in the five-slot experiments, the flow is small out the beginning slots and does not increase significantly until the last three to four slots (A_x = 2.0). The theoretical analysis predicts a decreasing flow until A_x = 2.1 (the amount of decrease depending on friction) and rapidly increasing flow at the closed end. From the photograph it also appears that the jet rises higher at the entrance, then drops towards the middle and finally rises again at the closed end. We believe, however, that the heights at the beginning slots are due to the water hitting the spacer and bouncing up; greater heights result from the higher axial velocity being converted into a vertical component as it hits the spacer. The measured flow out the slot was not larger at the entrance than at the middle of the pipe.

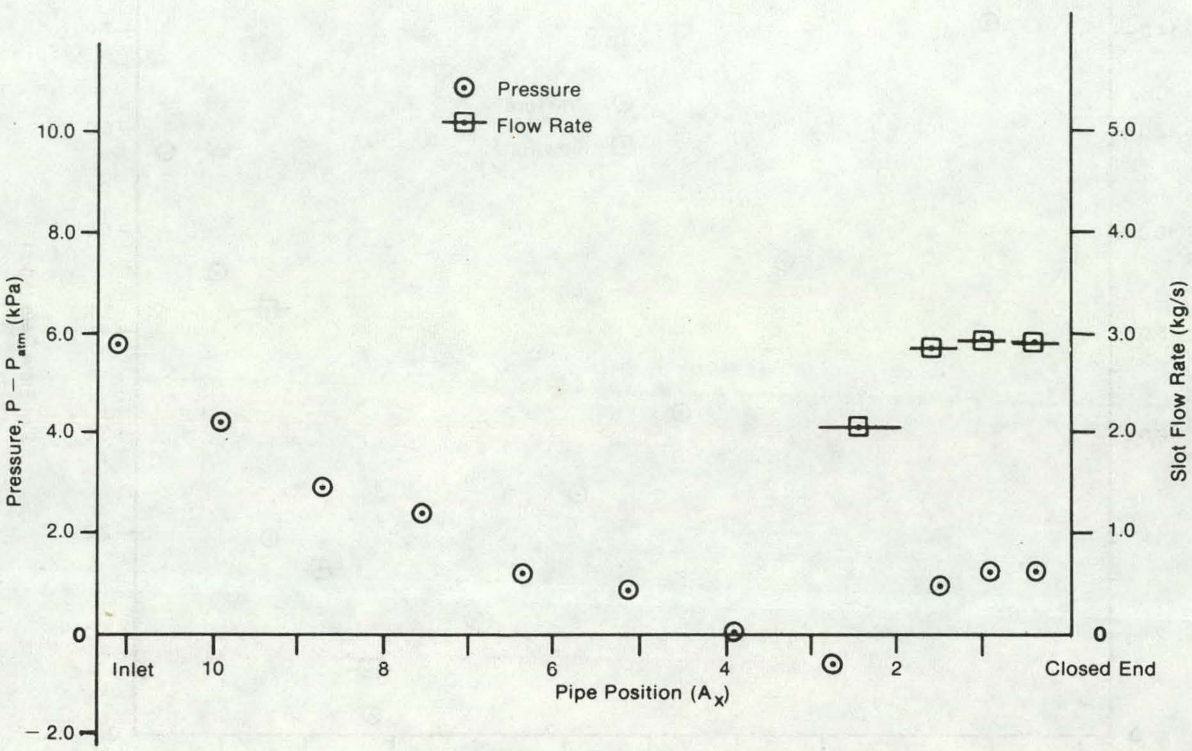

Figure 3-2b. Experiment 2 Results (5 Slots Exposed, A_L = 3.0, 5.9 kg/s Total Flow)

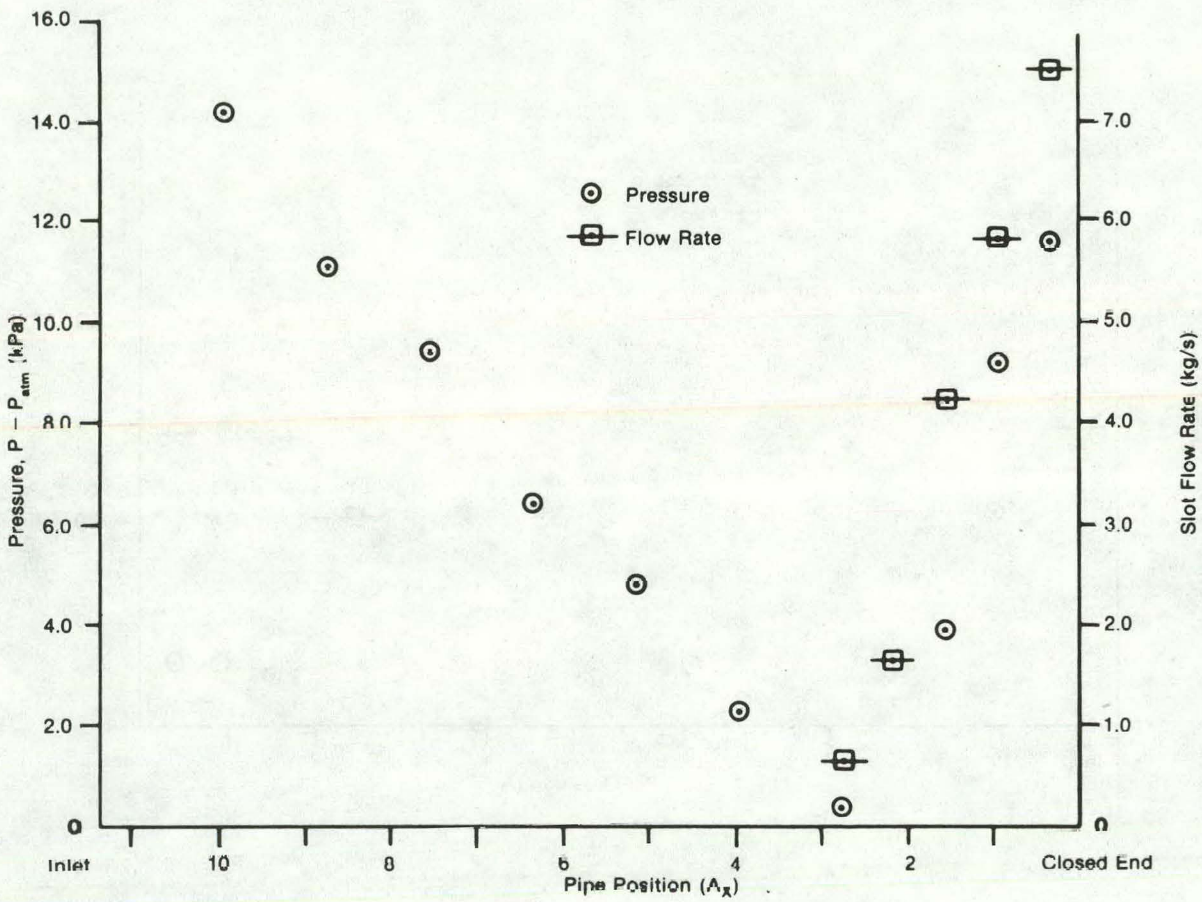

Figure 3-3. Experiment 5 Results (5 Slots Exposed, A_L = 3.0 16.5 kg/s Total Flow)

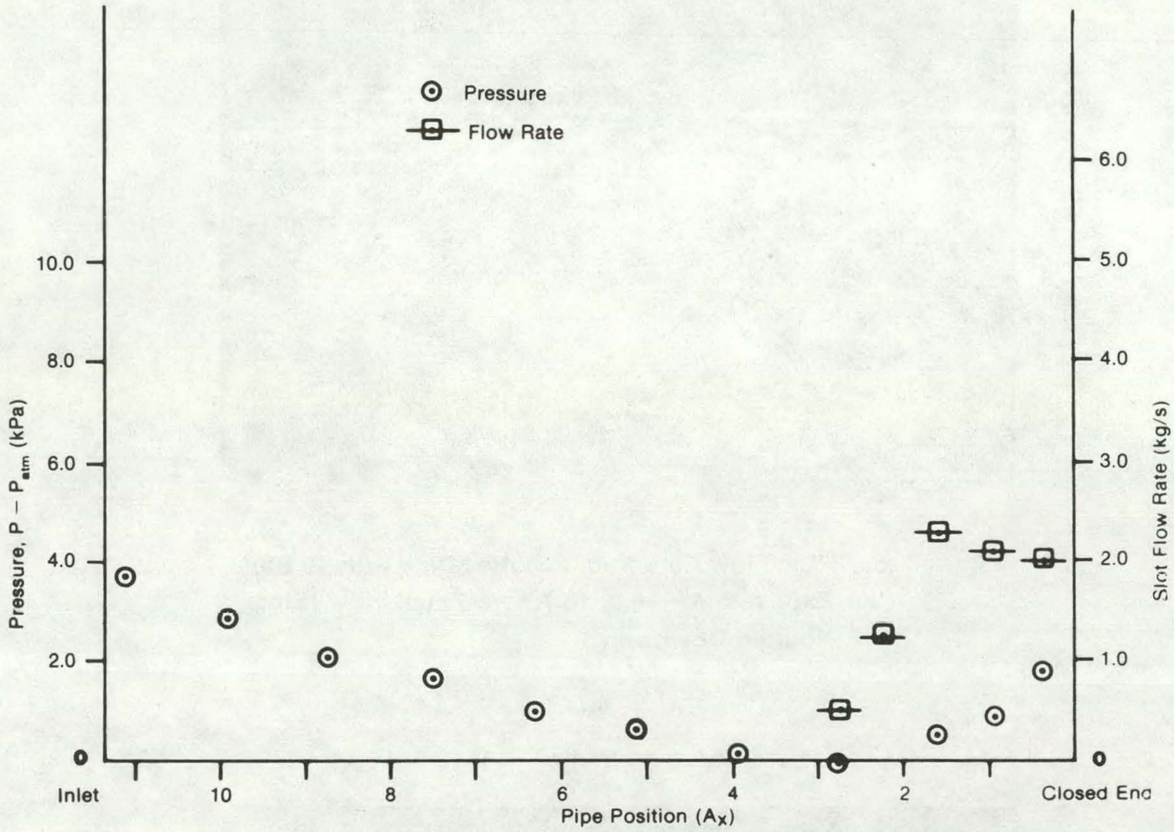

Figure 3-4. Experiment 6 Results (5 Slots Exposed, A_L = 3.0, 7.2 kg/s Total Flow)

Figure 3-5a. High Flow Rate from a Slotted Pipe with 10 Slots Exposed, A_L = 6.0, 13.7 kg/s Total Flow (Slots Pointed Downward)

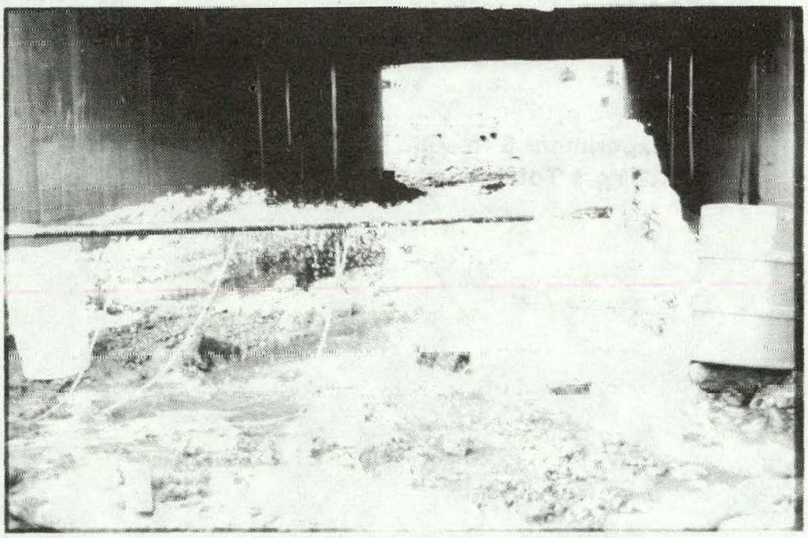

Figure 3-5b. High Flow Rate from a Slotted Pipe with 10 Slots Exposed, A_L = 6.0, 13.7 kg/s Total Flow (Slots Pointed Upward)

Figure 3-5c shows the measured flow out the slots and the pressure variations. The flow is four to five times as great out the last three slots as the first six slots. Note the marked similarity between the f = 0.02 or f = 0.01 curves on Fig. 1-2 and the slot flow rate curve of Fig. 3-5c. (The slot flow rate is directly proportional to the vertical jet velocity.) Both flows remain constant until $A_x \simeq 2.2$, then increase dramatically. The flow in the last three slots ($A_x \leq 2.0$) is within the range anticipated for OTEC.

Figure 3-6a shows the flow out the pipe with 10 slots uncovered and a low flow rate. Again, the figure indicates that most of the flow exits at the closed end of the pipe. Figure 3-6b is a plot of the measured flow and pressure variations; the pressure calibration errors on the pressure measurements are evident. The pressure measurements indicate that there was much less pressure recovery for the low flow rate test than for the higher flow rate experiment. Also, the measured flow increases at $A_x \simeq 4.0$ and then diminishes slightly at $A_x \leq 2.0$. Most of the flow, however, still exits in the final three to four slots ($A_x \leq 3.0$) . Comparing the low and high flow rate experiments for 10 slots, we see that increasing the flow rate for 10-slot experiments yields the same results as increasing it for the five-slot experiments. Hence, an increase in flow rate primarily increases the rate of flow out the end slots and leaves the inlet slot flows essentially unchanged.

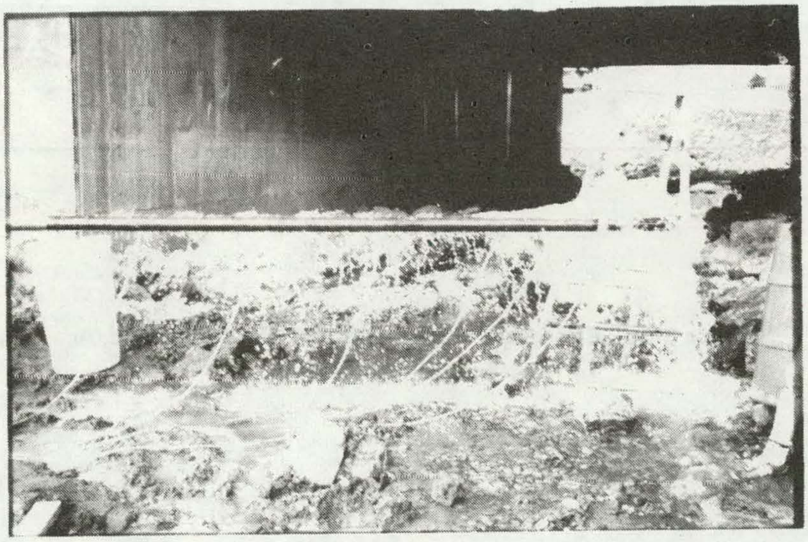

Figure 3-6a. Low Flow Rate from a Slotted Pipe with 10 Slots Exposed, A_L = 6.0, 8.5 kg/s Total Flow (Slots Pointed Upward)

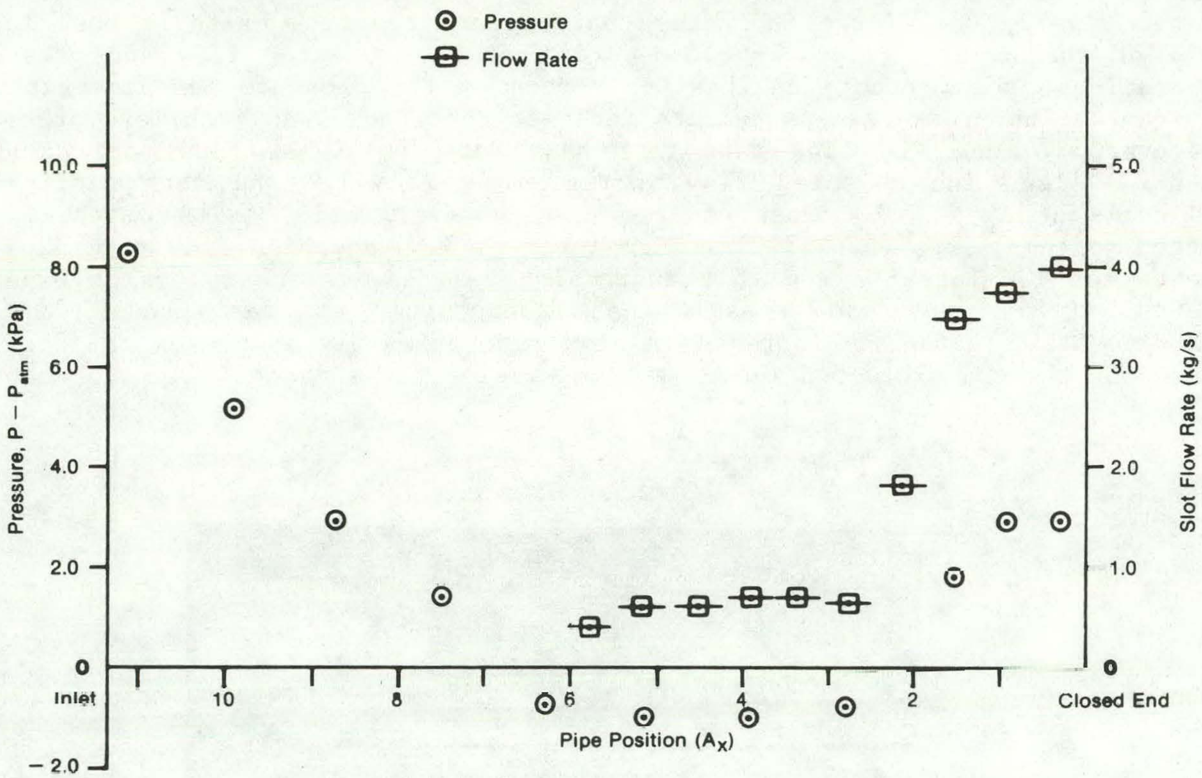

Figure 3-5c. Experiment 3 Results (10 Slots Exposed, A_L = 6.0, 13.7 kg/s Total Flow)

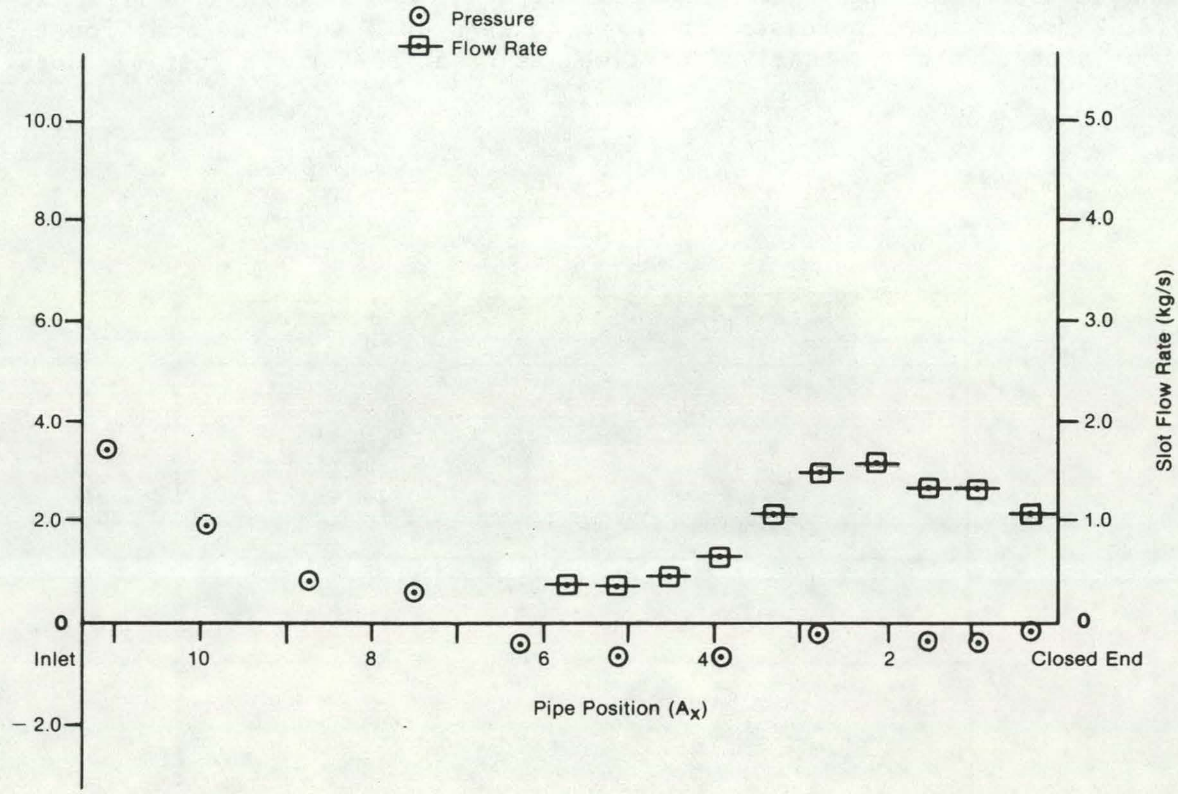

Figure 3-6b. Experiment 4 Results (10 Slots Exposed, $A_L = 6.0$, 8.5 kg/s Total Flow)

3.3 EXPERIMENTS WITH A LONG PIPE (A_x = 9.0)

The final experiment consisted of a high flow through 15 slots. Figure 3-7a shows the flow from this experiment. The results confirm the trends of the 5-and 10-slot experiments. Only the final three to four slots discharge a significant quantity of water ($A_x \leqslant 3.0$) . We observed again the large axial component of velocity in the jets. Figure 3-7b shows the measured pressure and slot flows. The jet flow remains very low through the first 10 slots ($A_x \geqslant 3.0$), then increases to 2.5 to 3 kg/s (8.2 to 10 kg/s-m) for the last four slots, which is nearly four times as great as for the initial slots.

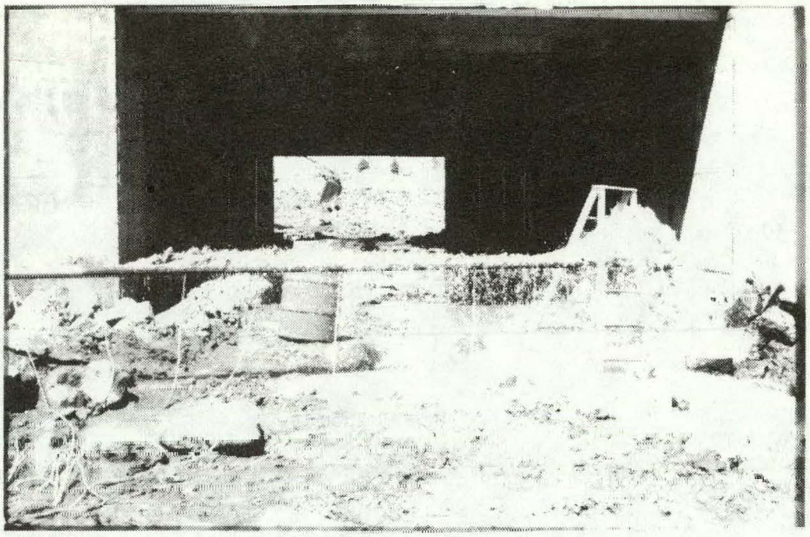

Figure 3-7a. High Flow Rate from a Slotted Pipe with 15 Slots Exposed, A_L = 9.0, 16.1 kg/s Total Flow (Slots Pointed Upward)

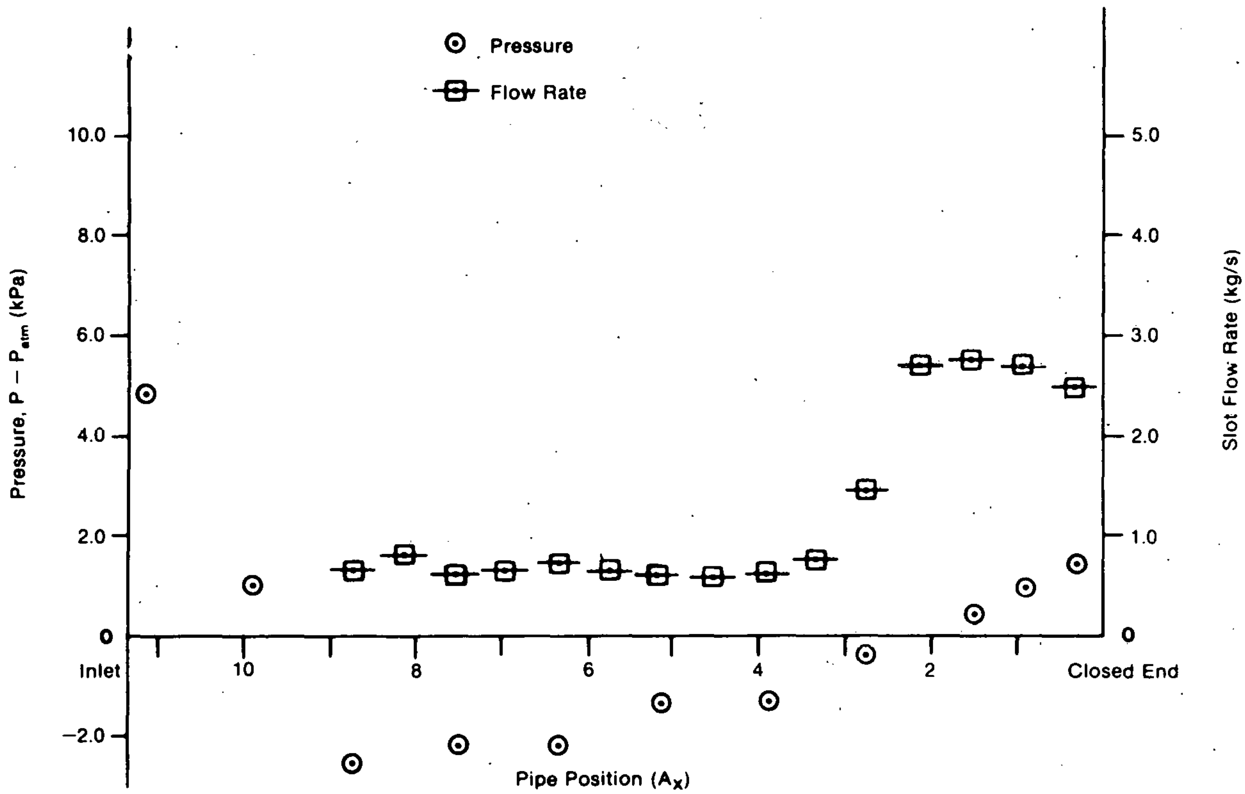

Figure 3-7b. Experiment 7 Results (15 Slots Exposed, A_L = 9.0, 16.1 kg/s Total Flow)

SERI

SECTION 4.0

CONCLUSIONS AND RECOMMENDATIONS

We make the following conclusions and recommendations, based on the results of the experiments and the theoretical analysis. First, regardless of the length of exposed slots, most of the flow exits the pipe near the closed end ($A_x < 3.0$, or the final 3 to 4 slots in this experiment). This is the same region of high vertical jet velocity predicted by the theoretical model. To keep flow variations less than 10% to 20% of the mean value, one should limit a slotted pipe to $A_L = 2.0$.

Second, the jet exiting the slots contains a large fraction of the axial velocity of the pipe flow. The angle of jet is nearly that predicted by assuming it carries all the axial velocity outside the slot. In the analytical model, where this assumption was made, only for $A_x \leqslant 1.0$, was the angle of the jet larger than 45°.

Third, the low flow rate experiments have more uniform jet flows than the higher flow experiments. Increasing the flow rate primarily increases the jet flow out the final slots, leaving the upstream jet flows unchanged. However, only the high flow rate experiments had specific flows of the magnitude anticipated for OTEC plants. If we assume a constant friction factor, the analytical model predicts that the relative values of the jet flows are independent of flow rate. Using the Blasius formula for the friction factor, the model predicts the same behavior as we observed in the experiments: increasing the total flow rate primarily increases the jet flow for $A_x \leqslant 2.0$.

Fourth, we did not perform the long slot experiments at specific flow rates simulating expected OTEC conditions, owing to water supply limitations. However, short slot experiments indicate that flow nonuniformities will be greater at the higher flow rates that simulate expected OTEC conditions.

Fifth, we constructed an analytical model of the flow through a slotted pipe. The experimental results confirm the predictions of the model, indicating that the scaling parameters have been correctly identified and that the model has not oversimplified the physical situation. The model can be used to predict the character of slotted pipe flows with dimensions different from those reported here.

Sixth, we recommend that if this configuration is used for an OTEC evaporator and condenser, the slot length should be kept to 1 m or less for a 6.27-cm diameter tube with a 0.635-cm wide slot ($A_x \leqslant 2.0$). A longer pipe can be used if the diameter is increased or the slot width decreased. One water distribution system that eliminates the jet flow variations of the slotted pipe is a box-like manifold, with a floor having the requisite number of slots. The height of the box would be adjusted to keep the area ratio small (slot area/flow area) and hence eliminate the large pressure recovery problems of the slotted pipe.

SERI

SECTION 5.0

REFERENCES

Kogan, A.; Johnson, D. H.; Green, H. J.; Olson, D. A. 1980 (June). "Open-Cycle OTEC System with Falling Jet Evaporator and Condenser." 7th Ocean Energy Conference. Washington D.C.; p. III F/3.

Senecal, V. E. 1957 (June). "Fluid Distribution in Process Equipment." Fluid Mechanics in Chemical Engineering. Vol. 49 (No. 6): pp. 993-997.

APPENDIX

ANALYSIS OF FLOW THROUGH A SLOTTED PIPE

In the appendix we derive the governing equations for flow through a slotted pipe and present some results of the analysis. The equations were integrated using a Runge-Kutta routine on SERI's CDC 170/720 computer. Although the actual flow inside the pipe is three-dimensional, the physics of the problem can be understood by using a simplified, one-dimensional model. The flow analyzed consists of a pipe of length L and diameter D (Area A), closed at the end $x = L$, and water entering at the end $x = 0$. A slot of width w runs the length of the pipe on the bottom. See Fig. A-1.

We assume the following regarding the flow:

- o $V = V(x)$ only where V is the x-component of velocity inside the pipe;
- o No y- or z-components of velocity in the pipe;
- o $P = P(x)$;
- o Gravity is unimportant;
- o $\overline{V}_j = \overline{V}_j(x)$, where $\overline{V}_j$ is the total jet velocity;
- o $V_{jx} = V$, where V_{jx} is the x-component of the jet velocity; and
- o $\tau_w = \tau_w(x)$, the flow is turbulent and friction is governed by the relation:

$$\tau_w = \frac{1}{4} f\rho \frac{V^2}{2}, \quad \text{where f is a constant.}$$

The governing equations are now derived (refer to Fig. A-2).

Continuity

The flow entering at x equals the flow leaving at x + dx plus the flow out the slot over the distance dx:

$$\rho AV = \rho A\left(V + \frac{dV}{dx} \cdot dx\right) + \overline{V}_j \cos\theta \, dx \cdot w,$$

$$V_{jy} = \overline{V}_j \cos\theta,$$

or

$$V_{jy} = -\frac{A}{w}\frac{dV}{dx} . \qquad \text{A.1}$$

x-Momentum

The change in x-momentum of the flow from x to x + dx equals the net pressure forces acting on the control volume, the wall shear, and the momentum carried out the slot through the x-component of the jet velocity:

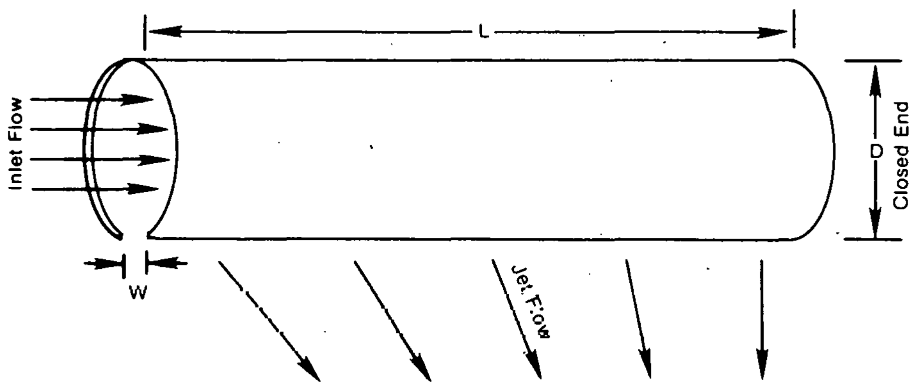

Figure A-1. Slotted Pipe

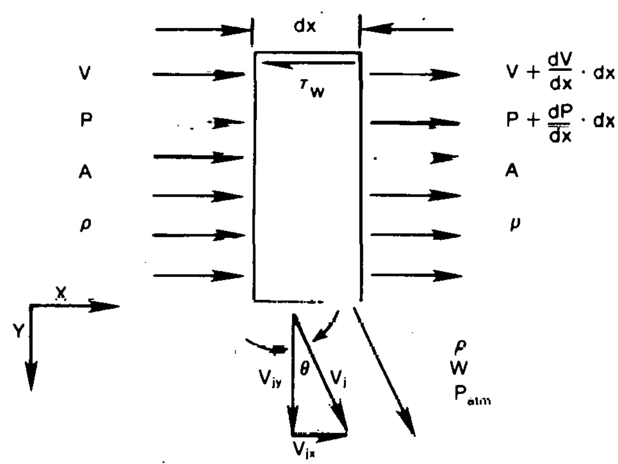

Figure A-2. Control Volume of Length dx

$$\rho AV^2 + PA = \tau_w \pi D dx + \rho A\left(V + \frac{dV}{dx} \cdot dx\right)^2 + \left(P + \frac{dP}{dx} \cdot dx\right)A$$

$$+ \rho w \, dx \cdot V_{jy} \cdot V_{jx},$$

with

$$\tau_w = \frac{1}{4} f \rho \frac{V^2}{2} .$$

We substitute, eliminate terms, and obtain

$$\frac{1}{\rho}\frac{dp}{dx} = -\frac{f}{D}\frac{V^2}{2} - 2V\frac{dV}{dx} - \frac{w}{A} \cdot V_{jy} \cdot V_{jx} . \qquad \text{A.2}$$

This equations shows that the pressure drops in the pipe owing to friction, and rises as a result of loss of mass out the slot ($\frac{dV}{dx}$ is negative and is proportional to the jet velocity). The final term is the momentum carried out the pipe by the x-component of the jet velocity, which tends to reduce the pressure.

y-Momentum

We assume a simple jet equation governs the flow of water out the slot. The equation for the acceleration of the jet through the slot in the y-direction is

$V_{jy} = c\sqrt{(P - P_a)\frac{2}{\rho}}$, where c is the discharge coefficient or

$$\frac{d}{dx}(P - P_a) = \frac{d}{dx}(V_{jy})^2 \cdot \frac{\rho}{2}\frac{1}{c^2} . \qquad \text{A.3}$$

By substituting A.1 for V_{jy} into A.2 and A.3, and by substituting A.3 into A.2, we obtain

$$\left(\frac{A}{cw}\right)^2 \frac{dV}{dx} \cdot \frac{d^2V}{dx^2} = -\frac{f}{D}\frac{V^2}{2} - \frac{dV}{dx}\left[2V - V_{jx}\right] . \qquad \text{A.4}$$

We assume the x-component of the jet velocity equals the velocity inside the pipe ($V_{jx} = V$), and nondimensionalize the variables with respect to $x^* = \frac{xw}{A}$, $V^* = V/V_o$, where V_o is the jet velocity at x = L:

$$\frac{1}{c^2}\frac{dV^*}{dx^*} \cdot \frac{d^2V^*}{dx^{*2}} = -V^*\frac{dV^*}{dx^*} - \frac{\left(\frac{f}{2}\frac{L}{D}\right)}{\left(\frac{wL}{A}\right)} V^{*2} . \qquad \text{A.5}$$

Equation A.5 can be integrated to yield V^* as a function of x^*; V_{jy} is given by A.1, and $P - P_a$ is given by A.3, once V^* is known. The boundary conditions for the integration are:

$$V = 0 \text{ at } x = L, \text{ or } V^* = 0 \text{ at } x^* = \frac{Lw}{A}$$

and

$$\frac{V_{jy}}{V_o} = 1.0 \text{ at } x = L, \text{ or } \frac{dV^*}{dX^*} = -1.0 \text{ at } x^* = \frac{Lw}{A}$$

Equation A.5 shows that there are three important parameters for the problem: the ratio at the slot area to the pipe cross-sectional area $\left(\frac{wL}{A}\right)$; the friction length of the pipe $\left(\frac{f}{2}\frac{L}{D}\right)$; and the slot discharge coefficient.* Computer solutions (Fig. A-3) show that when the flow area for the jet becomes much larger than the flow area of the pipe $\left(\frac{wL}{A} > 1\right)$, the jet velocity will vary significantly. In Fig. A-3, vertical jet velocity is plotted as a function of position along the pipe for several values of f. Specific flow rate = $\frac{\text{mass flow}}{\text{length}} = \rho\, V_{jy} \cdot w$. For $\frac{wL}{A} > 1$, the inlet pipe velocity must be larger than the average jet velocity. Hence, the amount of deceleration of the pipe flow is larger than the acceleration of the jet flow, increasing pressure in the pipe as the flow ceases. The large pressure recovery will produce large increases in jet flow as the closed end is approached.

Computer solutions also show that varying the length of the pipe does not change the region of the pipe dominated by pressure recovery. The length L can be canceled out of Eq. A.5; i.e., the solution at 1 meter from the closed end is unaffected if L is increased from 2 meters to 5 meters. This confirms the appropriateness of normalizing x to $\frac{A}{w}$, instead of L. The pipe should be measured from the closed end; therefore, we have used the parameter $A_x = (L - x)\frac{w}{A}$ in the text.

By choosing a constant friction factor, f, independent of Reynold's number, we remove the magnitude of the velocities as a parameter in the solution. The curves in Fig. A-3 are independent of the magnitude of the velocities. A slotted pipe for OTEC might be represented as a smooth pipe for which the jet flow is low and constant until pressure recovery dominates. The experimental results discussed in the text were much closer to the f = 0.02 or f = 0.01 curves than to the f = 0.04 solution.

We used a more realistic friction factor correlation to determine the effect of the magnitude of the velocity on the form of the solution. The Blasius formula for a smooth pipe, $f = 0.316\ Re^{-0.25}$, was used instead of constant f. We obtained solutions for jet velocities of 1 m/s and 5 m/s at the closed end (the velocities anticipated for falling jet evaporators and condensers fall

---

*The first and second parameters can be combined into one parameter, $f\frac{\Pi}{8} \cdot \frac{D}{w}$, but it does not have the intuitive meaning of the parameters presented here.

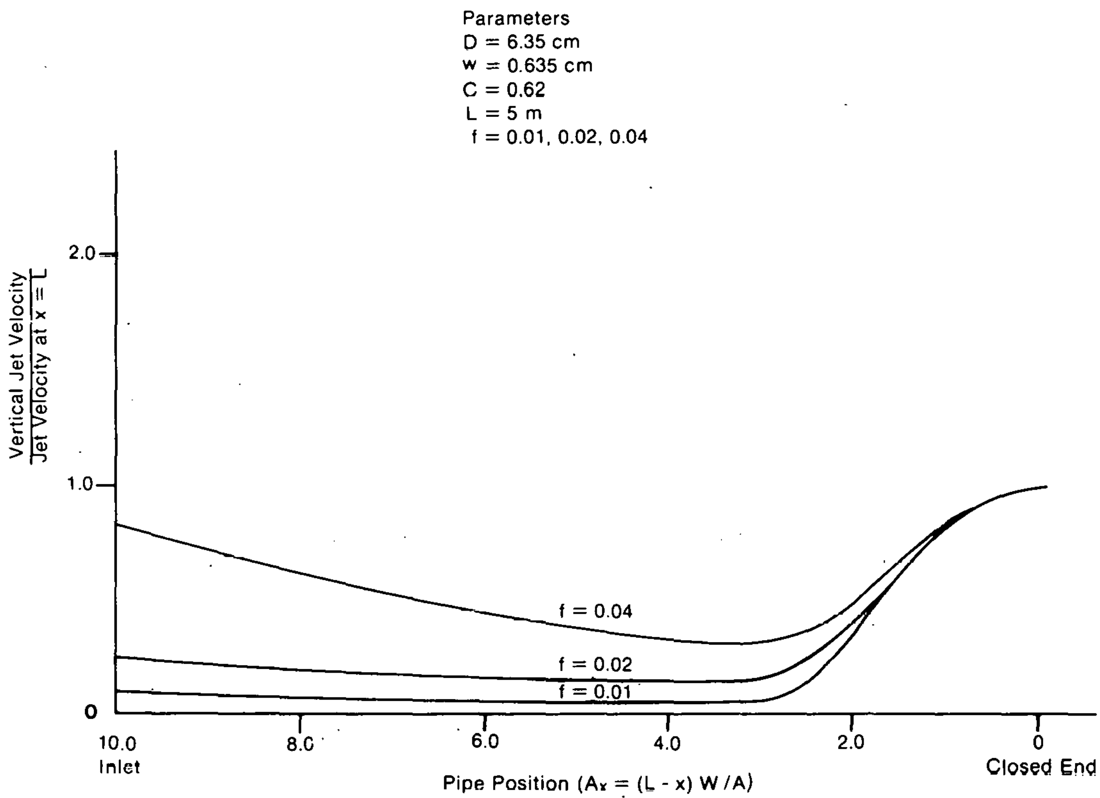

Figure A-3. Nondimensional Vertical Jet Velocity as a Function of Pipe Position for Various Friction Factors

within this range). When the solutions are normalized to the end jet velocity, they are barely distinguishable from each other and fall between the solutions for f = 0.01 and f = 0.02 of Fig. A-3. However, if we plot an absolute scale for the vertical jet velocity (Fig. A-4), we see that the velocities at the closed end are 4 to 5 times as high for the high velocity solution as for the low velocity solution. In the entry and intermediate portions of the pipe, the velocities are only 2 to 3 times as high for the high velocity solution. Hence, as the total flow rate into a section of pipe is increased, most of the increase in jet flow occurs at the closed end, which is dominated by pressure recovery.

The computer solutions indicate that the axial component of jet velocity will be high compared to the vertical component (see Fig. A-5), except for $A_x \leq 1.0$ (the closed end). The jet angles are only weakly dependent upon friction factor. Remember that this effect is due to our assumption that the jet carries the same axial velocity as is in the pipe.

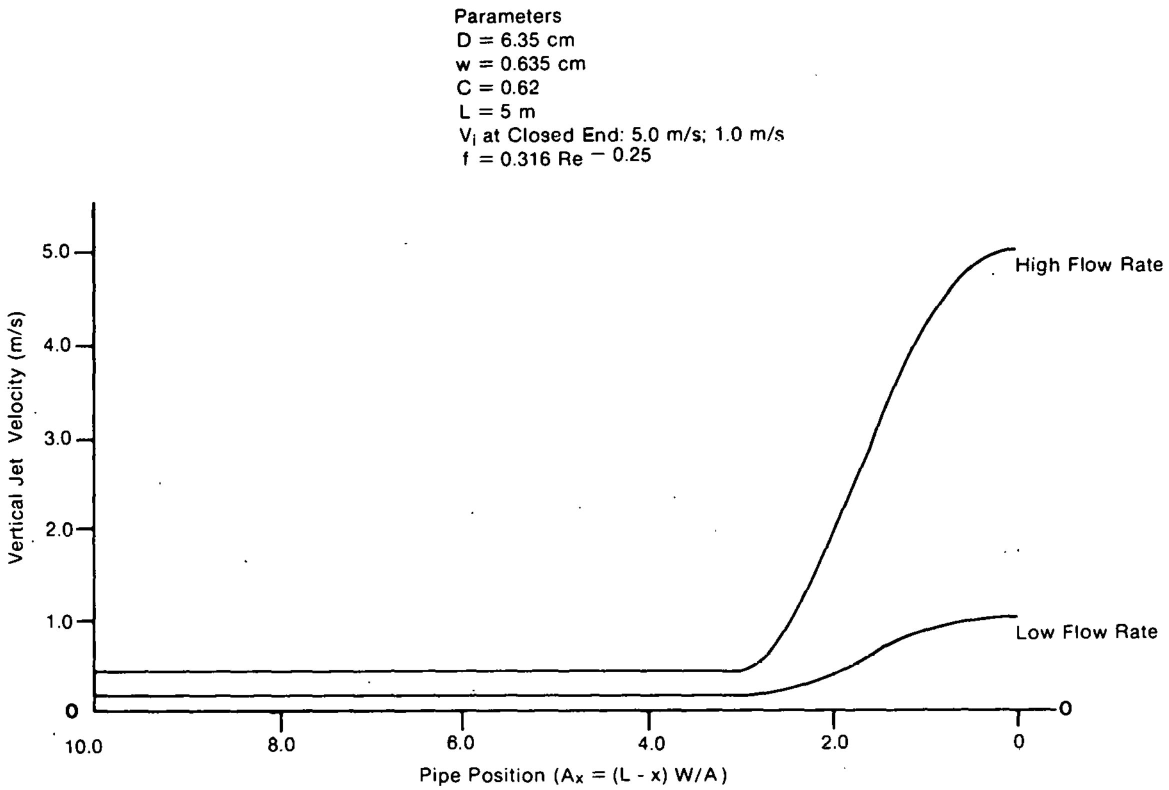

Figure A-4. Vertical Jet Velocity as a Function of Pipe Position for Low and High Flow Rate

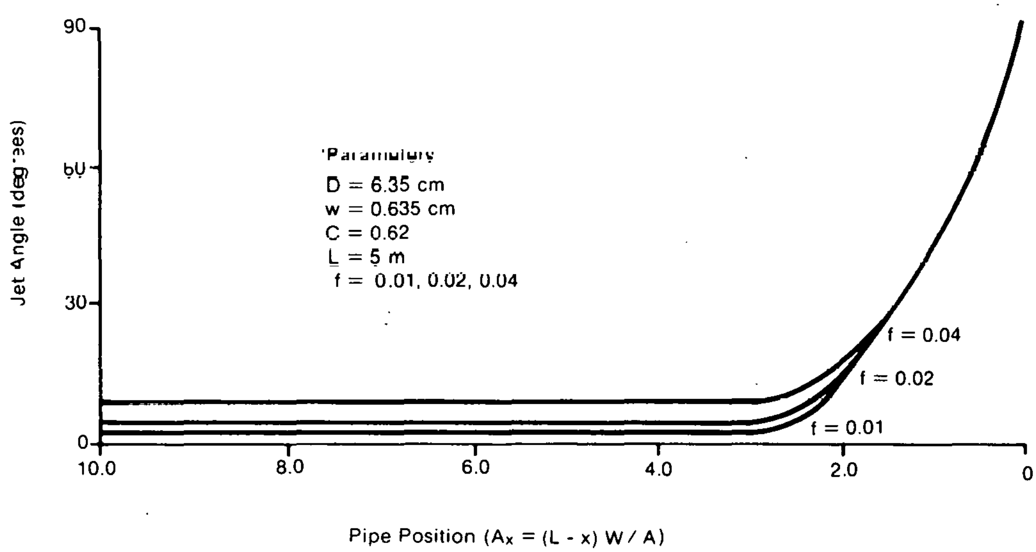

Figure A-5. Jet Angle as a Function of Pipe Position for for Several Friction Factors

Document Control Page	1. SERI Report No. TR-631-984	2. NTIS Accession No.	3. Recipient's Accession No.

4. Title and Subtitle Water Flows from Slotted Pipes	5. Publication Date March 1981
	6.
7. Author(s) Douglas A. Olson	8. Performing Organization Rept. No.
9. Performing Organization Name and Address	10. Project/Task/Work Unit No. 1111.00
	11. Contract (C) or Grant (G) No. (C) (G)
12. Sponsoring Organization Name and Address Solar Energy Research Institute 1617 Cole Boulevard Golden, Colorado 80401	13. Type of Report & Period Covered Technical Report
	14.

15. Supplementary Notes

16. Abstract (Limit: 200 words)

A. Kogan proposed slotted pipes for generating falling jets in evaporators or condensers in Claude-cycle, open-cycle ocean thermal energy conversion (OC-OTEC) power plants. This report describes results of experiments and analyses that determine jet flow distribution from slotted pipes of dimensions typical for OC-OTEC evaporators or condensers. For a pipe with a 6.3-cm inside diameter and 0.64-cm wide slot, the measured and predicted jet flow was low and nearly parallel to the pipe at the entrance, and high and perpendicular to the pipe only near the closed end. Slot lengths ranged from 1.5 m to 4.6 m, and inlet flow rates varied from 6 kg/s to 17 kg/s. Friction reduces the pressure in the entrance and intermediate portions of the pipe, while the rapidly decelerating flow produces high pressure recovery as it approaches the closed end. In the region of high flow next to the closed end, the ratio of slot area (slot length times width) to pipe cross-sectional area is less than two.To use a slotted pipe for generating falling jets in an OC-OTEC plant, the slot length should be 1 m or less (for a pipe with a 6.3-cm inside diameter and a 0.64-cm wide slot).

17. Document Analysis

a. Descriptors Ocean Thermal Energy Conversion; Open Cycle; Evaporators, Condensers, Claude-cycle; Jet Flow; Slotted pipes

b. Identifiers/Open-Ended Terms

c. UC Categories
64

18. Availability Statement National Technical Information Service U.S. Department of Commerce 5285 Port Royal Road Springfield, Virginia 22161	19. No. of Pages 41
	20. Price $4.50

Form No. 8200-13 (6-79)

☆U.S. GOVERNMENT PRINTING OFFICE:1981—777-084 / 25