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TWO-PHASE AIR-WATER FLOW
IN A SLOT-TYPE DISTRIBUTOR

A Thesis
Submitted to the Faculty of Graduate Studies
Through the Department of Chemical Engineering in
Partial Fulfilment of the Requirements for the
Degree of Master of Applied Science at
The University of Windsor

by
John M. Madden

Windsor, Ontario
1968

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ABSTRACT

This study was a fundamental analysis of two-phase air-water flow through a slot-type distributor. Here a diverging manifold system was used to investigate the factors affecting crossflow. A slot was cut in the side of a vertical 1-in. diameter tube parallel to the tube axis and a controlled variable pressure drop perpendicular to the main stream flow was employed to remove a portion of the main stream. The distribution through the slot was studied with respect to the main stream quality and total mass flux, the length and width of the slot and the pressure drop across the slot. A six foot length of plexiglass tube was used as the main flow channel. Three slot lengths were used: 1.66-in., 4-in., and 6-in. while the slot width was varied from .012-in to .080-in. The total main stream mass flux was varied from 200,000 to 500,000 lbm/hr.ft² and the air quality ranged from 0.2 to 0.7 at system pressure of 50 psia. The orifice equation employing discharge coefficients was found to be successful for calculating crossflow rates. The air crossflow was virtually independent of the total mass flux while the water crossflow exhibited ogee behaviour passing from convex to concave shape with increasing mass flux. When the main stream quality was increased, while maintaining fixed values of the total mass flux and radial pressure drop, the mass flow rate of water decreased exponentially. The variation of air discharge rates with quality was much less as compared to the quality effect on water crossflow. The axial pressure gain across the slot was investigated and the value of k , the momentum correction factor, was obtained experimentally. The axial pressure drop across the test

section was also examined and it was observed that the Lockhart-Martinelli correlation [20] fits the data very well in the low quality, low mass flux region.

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I INTRODUCTION

This study was concerned primarily with the acquisition of cross-flow data for air and air-water systems flowing in a slotted pipe. Crossflow is defined here as the net mass transfer rate between two adjacent channels connected through a gap along their length when the inter-channel pressure difference is considered to be the driving force.

There are many practical engineering applications where the ability to predict crossflow is of importance: viz. in manifold design problems such as gas burners, sprinkler systems, heat exchanger headers and canal lock systems. A knowledge of the amount of and factors affecting crossflow is of particular importance in heat exchanger design when there is parallel flow in rod or tube bundles with an open matrix. (See Fig. 1.1.) Since crossflow affects the mass, momentum and energy balances in each subchannel then a more accurate prediction of local flows, enthalpies and pressures in subchannels requires a knowledge of crossflow.

For parallel flow in rod (tube) bundle heat exchangers with an open matrix a subchannel is formed by drawing a line between rod centres. Symmetrical arrays may be divided into sectors which have identical behaviour thus reducing the number of subchannels to be considered. This is also shown in Fig. 1.1.

Total Number of Subchannels = 20

Total Number of Distinct Subchannels = 5

Fig. 1.1 Typical Rod Bundle Cross Section

When the heat flux is sufficiently high in parallel flow rod bundles heat exchangers to cause boiling, then a two-phase flowing system must be analyzed. This type of coolant system is encountered in some nuclear reactor designs where a knowledge of local subchannel conditions can be related to the prediction of local critical heat flux. Critical heat flux is defined here as the heat flux at which a rapid deterioration of the cooling process occurs.

In this study a diverging manifold system is employed to investigate both qualitatively and quantitatively the factors affecting cross-flow in a flowing system. A slot was cut in the side of a circular pipe and a controlled variable pressure drop perpendicular to the main stream flow was employed to remove a portion of the main stream.

See Fig. 1.2.

Fig. 1.2 Diverging Manifold Model

The experimental data were obtained at a pressure of 50 psia on WAFER^{*}, an experimental loop for two-phase two-component (air-water) systems. A six foot length of plexiglass tube one inch in diameter was used as the main flow channel. Single phase crossflow was studied employing air as the test fluid. Single-phase and two-phase crossflows were investigated as a function of the total mainstream mass flux, quality^Δ, pressure drop across the slot, slot length and slot width. Three slot lengths were used: 1.66 in., 4 in., and 6 in. while the slot width was varied from 0.012 in. to 0.080 in. The total mainstream mass flux was varied from 200,000 to 500,000 lbm/hr-ft² and the air quality ranged from 0.2 to 0.7.

^{*} Water Air Fog Experimental Rig located at Chalk River Nuclear Laboratories (Building 456)

^Δ Quality is defined here as the ratio of air mass flow rate to the total air and water mass flow rates

The effect of crossflow on the axial pressure gradient is reported also. Since a portion of the mainstream flow is diverted through the slot, the mainstream flow is decelerated. This deceleration should be accompanied by a corresponding rise in the test section static pressure. The relative magnitude of this static pressure rise is important as it can affect the distribution of flow through the slot. Values of the mainstream static pressure rise were obtained as a function of quality, total mass flux, and pressure drop across the slot.

II LITERATURE SURVEY

A. INTRODUCTION

A search of the open literature disclosed no account of experimental or theoretical work concerning crossflow in two-phase flowing systems. However, flow redistribution in a parallel flow rod bundle heat exchanger was discussed briefly by Green et.al.[1] for two-phase flow conditions.

A number of investigations of the manifold design problem has been conducted for single-phase flow. The fluid flow mechanics for a discharging slot manifold are believed to be similar to those found in interchannel crossflow. Existing single-phase manifold data [2] have been used to predict crossflow [3] in rod bundle heat exchangers with an open matrix. There are two points of major interest in the manifold problem:

- (i) the pressure change which occurs in the main flow channel over the length of the slot, and
- (ii) the mass flow rate through the slot.

The axial static pressure rise across the slot in a discharging manifold is the result of the combined effects of frictional losses, hydrostatic head losses and momentum changes accompanying crossflow. In general, as a portion of the fluid in the main channel is diverted through the slot, the mainstream velocity decreases. This deceleration of the fluid in the main channel results in an increase in the static pressure in accordance with Bernoulli's equation. Frictional losses

have the opposite effect on the static pressure. The decreasing hydrostatic head in the downstream direction for upward flow in a vertical pipe also tends to lower the static pressure. Hence frictional and hydrostatic head effects work together while the momentum effects work in the opposite direction; the first two tending to produce a pressure drop and the last a pressure rise. However, for short manifolds, the frictional effects may be negligible [4]. Dow [5] reported that in general the pressure loss due to friction is related to the surface area of the pipe, while the pressure gain due to deceleration is related to the cross-sectional area of the pipe. As the flow area is reduced these effects become more pronounced. Only for the case where these two effects are exactly balanced will the distribution from the manifold be uniform [6].

For single-phase water flow, the gain in piezometric head has been found to be significantly large [2] and the associated head loss of the main flow stream is never as much as one-half that at an abrupt expansion. Soucek and Zelnick [7] have reported sudden axial pressure gains opposite the discharging port centre line.

Senecal reported that obtaining an optimal manifold design from theory suffers from the following aspects [8]:

"First, actual change in the velocity profile at the outlet is not considered. Second, the momentum transfer from fluid drawn off to fluid remaining in the pipe is assumed to be perfect, which is not true. Third, gen-

eral practice assumes an orifice coefficient of 0.63 which is the weakest point in the design. Pressure drop across the orifice is generally one or two orders of magnitude greater than the kinetic energy, momentum or frictional terms of the manifold pipe and therefore, a small error in the orifice coefficient is of major importance. The coefficient is a function of hole size relative to pipe diameter and wall thickness, upstream pressure, medium which the fluid discharges, flow rate from the hole and flow rate in the pipe across the hole."

B. SINGLE PHASE MANIFOLD EXPERIMENTS

Enger and Levy [9] studied the variation of the axial pressure gradient over and water discharge from a series of ten closely spaced 2-in. pipes, each 36-in. long. These pipes were connected to a manifold having a 2-in. internal diameter. Water entering one end of the manifold discharged from 1-in. holes in the caps on the ends of the laterals. A theoretical analysis was developed employing the orifice equation to predict the pressure at any point in the manifold with a long, narrow slot parallel to the flow. The head loss due to friction was assumed to be negligible. The discharge coefficient of the openings in the pipe was found to decrease as the velocity in the pipe increased.

Oakey [10] conducted a similar experiment investigating the discharge of water from short tubes (with sharp edged entrance conditions)

mounted at right angles to the side of a pipe. Tests were made with velocities in the pipe ranging up to 40 ft/sec. Oakey assumed that

$$C = \frac{Q_t^x}{Q_t} \quad 2.1$$

where Q_t^x is the volumetric rate of flow from the tube, and Q_t is the volumetric flow rate of the main flow channel, and C represents the portion of the velocity head immediately before the tube which is utilized by the flow through the tube. The coefficient of discharge was then determined by employing the equation:

$$W_t^x = C_t A_t \sqrt{2 g_c \left(\frac{C V_1^2 Q_t^2}{2 g_c} + \Delta P_r Q_t \right)} \quad 2.2$$

As long as C is constant, then the values of the loss coefficient and discharge coefficient will be the same, independent of the main pipe diameter, discharging tube diameter, the total flow in the tube and the discharge from the tube provided the ratio of discharging tube diameter to pipe diameter is constant.

Soucek and Zelnick [7] have also reported experimental data for manifold systems. These experiments were performed in conjunction with the design of the Third Locks of the Panama Canal. Soucek and Zelnick considered a horizontal tube to which a discharging lateral port is attached. Immediately in the vicinity of the port they found an abrupt pressure rise in the direction of the flow. The pressure

rise is reported to be sudden and occurring opposite the port centre line. To determine the value of the pressure rise, the upstream and downstream pressure gradients are extended until they intersect a vertical erected at the port centre line. The difference between the intersections of the two gradients at the vertical is the value of the pressure rise.

The port discharge was related to a head measurement by the use of the orifice equation. The head on the port was defined as the difference between the pressure external to the port and the intersection of the upstream gradient with the vertical erected at the centre of the port.

The data were reduced to two curves for the discharge ports. Plots of $\Delta P_a / (V_1^2 / 2g_c)$ vs. W_1^x / W_1 and the discharge coefficient vs. W_1^x / W_1 were obtained. The ratio W_1^x / W_1 was selected as the independent variable since the relative pressure changes and discharge coefficients showed no significant dependence upon other variables. However, the plots revealed that there was a significant change in discharge coefficients with different shaped ports.

Keller [6] considered the flow of a fluid through a manifold having a uniform distribution of holes along its length. This was the case of the "pipe burner" for gaseous fuels. He concluded that two dimensionless ratios were required to define the operating characteristics of this type of manifold. These are the ratio of the active length of the series

of holes to the diameter of the manifold L/D , and the area ratio

A_o/A_p or

$$\frac{\text{Sum of areas of all discharge holes}}{\text{Cross-sectional area of manifold}}.$$

The variation of discharge through the ports was taken to be dependent only upon these two ratios. Only for the case of $L/D = 70$ and $A_p/A_o > 1$ was the flow from the manifold considered to be uniform over the entire length. The discharge coefficient for the holes was assumed to be constant and the fact that the main flow Reynolds number varied along the manifold was not considered.

Dow [5] investigated the distribution of a fluid flowing through a manifold with uniformly spaced outlets, similar to the work of Keller. To achieve a uniform flame height in a gas burner manifold, it was reasoned that the frictional losses in the manifold must be exactly balanced by the pressure gain produced by a portion of the fluid leaving the manifold through the ports. Only for this case will the static pressure along the manifold be constant. It was shown that to achieve a uniform flame height along the manifold, the radius of the manifold must vary from the value at the inlet end. Design curves were presented giving the variation of the hydraulic radius with the distance along the manifold for laminar and turbulent flow.

Koestel and Tuve [11] studied the discharge of air from a long slot in a duct. The duct received air from one end only and the opp-

osite end was dead-ended. Experimentally it was found that if the ratio of the area of the slot to the area of the supply duct is held constant, then the general shape of the discharge stream will be practically independent of the pressure in the supply duct or of the width of the slot. At the slot, the discharge angle of the flow is the vector resultant of the axial component of flow velocity head and the velocity due to the pressure difference across the slot. For this study, slot lengths of 6 to 12 ft were used, the slots being 1/4-in to 1 1/2-in wide. Both rounded entrance or nozzle type slots, and square edge slots were tested using square and rectangular ducts.

Koestel and Young [12] developed a theoretical method for obtaining a uniform air distribution from a long slot in a dead-ended duct which they verified by experiment. McNown [2] conducted a fundamental study on the manifold problem employing a brass conduit of 2-in diameter. Laterals were constructed of brass tubing 1-in and 1/2-in in diameter with the junctions being machined tees with a sharp-edged intersection. Water was supplied from a constant-level tank and was used as the test fluid for all runs.

Figure 2.1 is a defining diagram for the nomenclature used in Eq. 2.3.

Fig. 2.1 Elemental Volume Over Lateral

The change in head for the flow that continues in the conduit was expressed as

$$\begin{aligned} \frac{P_2 - P_1}{\frac{\rho V_1^2}{2g_c}} &= 1 - \left(\frac{V_2}{V_1} \right)^2 - \frac{h_f}{\frac{V_1^2}{2g}} \\ &= \frac{Q_3}{Q_1} \left(2 - \frac{Q_3}{Q_1} \right) - \frac{h_f}{\frac{V_1^2}{2g}} \end{aligned} \quad 2.3$$

- where
- h_f - head loss due to friction [ft]
 - V_1 - average fluid velocity upstream of lateral [ft/sec]
 - V_2 - average fluid velocity downstream of lateral [ft/sec]
 - P - static pressure [lbf/ft²]
 - Q - volumetric flow rate [ft³/sec]

Figure 2.2 illustrates McNown's method of analysis for the discharging manifold.

Fig. 2.2 Results of a Typical Experiment for Dividing Flow - McNown [2]

The data were reduced by plotting $\frac{\Delta P_a}{\left(\frac{v_1^2 \rho}{2g}\right)}$ vs. $\frac{Q_3}{Q_1}$

after the work of Soucek and Zelnick [5]. Experimental curves were obtained for D_3/D ratios of 1, 1/2 and 1/4. Soucek and Zelnick's data were not recorded for Q_3/Q_1 ratios larger than 0.5. McNown's treatment extended the Q_3/Q_1 ratio to 1.0 which give a clearer understanding of the phenomena.

McNown concluded that:

- (i) the gain in piezometric head is large;
- (ii) the head loss is never as much as one half that at an abrupt expansion;
- (iii) for small values Q_3/Q_1 the head loss is a sizeable negative quantity.

C. SINGLE PHASE MANIFOLD THEORIES

Acrivos et. al. [13] provided a method of analyzing the mechanics of flow in a manifold. The flow through a single manifold was considered first and then the treatment was expanded to cover a manifold with several side ports.

To facilitate a momentum balance the control volume in Fig. 2.3 was considered.

Fig. 2.3 Control Volume for
Momentum Balance

The pressure gain across the slot was related to the decrease in velocity by the equation

$$P_2 - P_1 = k \rho \left[\frac{v_1^2 - v_2^2}{g_c} \right] \quad 2.4$$

where k is a factor to account for the fact that the velocity of the discharging stream is not at right angles to the main flow. It was recommended that the value of this factor k be determined experimentally. This would necessitate finding the value of the pressure gain after the method of McNown [2] and substitution in the above equation. An experimental study revealed that k decreased as D_3/D was decreased.

Cicchelli and Boucher [4] presented a design method which minimized the head pressure in a heat-exchanger and ensured uniform flow among the tubes. Flow passed from the header to the heat-exchanger tubes by making a right-angled turn. The friction losses were attributed to the fact that part of the flow had to move laterally to feed the tubes. The pressure variations caused by momentum changes were related to the Bernoulli effect. The momentum transfer of the side flow to the main header flow was assumed to be perfect. Also, the frictional loss term was neglected in the design equation as it was found that the friction term was less than 20% of the momentum term.

D. TWO-PHASE CROSSFLOW

Green et. al. [1] hypothesized that boiling in an open rod matrix could cause flow redistribution due to the resulting radial pressure

gradient setup in the matrix. Experiments were conducted to study this flow redistribution and to determine if correlations derived from tests on rectangular channels were applicable for flow parallel to rod bundles. The rod bundle test section contained nine rods enclosed in a square shroud. Water flowed parallel to the heated rods where boiling occurred. Pressure taps were located on the shroud every $1\frac{5}{8}$ -in. and installed on the inside of the rod bundles $9\frac{1}{4}$ -in. apart. The measured pressure drop across the rods was, in general, about 10 per cent less than those measured across the shroud. It was believed that flow distribution, or interchannel crossflow, caused this difference in pressure drop. Although the presence of crossflow was experimentally observed, it was concluded that the axial redistribution had little effect upon axial pressure drop.

III AIR-WATER TEST LOOP AND ASSOCIATE EQUIPMENT

A. INTRODUCTION

The air-water test loop WAFER and associated equipment used in this study were located in the Advance Engineering Building at Chalk River Nuclear Laboratories. Initially this loop was designed and constructed [14] to permit two-phase flow studies on prototype nuclear reactor rod bundle geometries. For the purposes of this cross-flow experiment minor modifications were made to the loop.

1. WAFER: Water-Air-Fog Experimental Rig

The air-water test loop used in this study consisted essentially of a regulated 100 psia air supply, a centrifugal pump for water circulation, a mixing section, a test section assembly, a cyclone separator, a downcomer and a recycle feed storage tank. This loop, which could be operated at pressures up to 65 psia, is diagrammed schematically in Fig. 3.1.

A 100 psia air service line of the plant was filtered, metered and fed through a bank of four orifices to the mixer before the test section. An air vent valve on the separator was employed to maintain a constant inlet pressure to the test section for all experimental runs.

Fig. 3.1 'WAVER' Loop with Test Section in Place

Water was supplied by a Durco (Model H7RDR74) pump which could provide 100 IGPM at a head of 107 psia. A bank of four orifice meters was employed to meter the water flows before entering the mixer.

Design of the air-water mixer before the test section was such as to promote annular flow in the test section. Specific details of the mixer employed can be obtained from Fig. 3.1. Essentially it was designed as two concentric tubes, the centre tube for water and the outer annulus for air. A piston in the water nozzle was driven by a rod connected to a diaphragm valve. This allowed the piston to be positioned in order to vary the pressure drop across the mixer which would provide the desired flow pattern. The desired air-water flow then entered the test section.

After the test section a cyclone separator with a capacity of 6,000 lbm/hr of air and 30,000 lbm/hr of water was used to separate the water from the air. The air leaving the separator passed through a pressure control valve and then was vented to the atmosphere. The water passed through a 2-in ball float trap over to a downcomer and then to a 200 I.G. feed storage tank.

2. Test Section Assembly

Two transitional sections, a 6 ft length of straight lucite pipe and a crossflow measuring station comprised the test section assembly.

(a) Transitional Sections

Two transitional sections were constructed. The first to reduce the flow cross-sectional area from the $3\frac{1}{4}$ -in I.D. at the mixer to the 1.0-in I.D. of the test section. See Fig. 3.2 for details of this transitional section. The second transitional piece served to increase the internal diameter of the flow area from 1.0-in back to the $3\frac{1}{4}$ -in which existed before the separator.

(b) Test Section

The tube used to construct the main test section was made by boring a 1.0-in diameter hole in a solid lucite rod 12-in long leaving a $\frac{1}{4}$ -in thick wall. Four of these tubes were cemented together in series using lap-joint connections as shown in Fig. 3.2.

Pressure taps were placed at intervals of approximately 5 tube diameters along the length of the test section. These pressure taps consisted of 8 equally spaced $\frac{1}{16}$ -in diameter holes drilled in the tube. An annular ring, which surrounded the pressure taps was cemented to the test section. In order to prevent any water which entered the annular ring from interfering with pressure measurements, the eight holes were located near the bottom of the ring. In addition water traps were located below the annular rings to remove any water that did enter the pressure transmission line.

REDUCING SECTION

EXIT SECTION

FLOW CHANNEL 1.000" DIAM.

SCALE: N.T.S.

TEST SECTION ASSEMBLY

Fig. 3.2 Test Section Details

(c) Crossflow Measuring Station

This crossflow measuring station (for details see Fig. 3.3.) served 2 main functions:

- (i) it provided for separation and removal of air and water which left the main channel and the subsequent measurement of their respective flow rates;
- (ii) it provided a means of controlling the pressure external to the flow test tube.

In order to allow visual observation of the crossflow stream, the measuring station was constructed mainly out of plexiglass. A baffle system immediately opposite the slot in the test section provided partial separation of the air-water crossflow stream. The water crossflow stream was collected at the bottom of the measuring station where a constant liquid level was maintained by manual valve flow control. The discharging water stream flow rates were calculated using the direct weighing technique.

Air was vented from the top of the measuring station to the gravity separator for a final separation of air and water. From the separator, the air crossflow stream was fed to an aluminum header, then to needle control valves and finally to three rotameters, connected in parallel. Manual control of the needle valves allowed the crossflow air to be metered at the proper rate while maintaining the desired constant pressure in the crossflow measuring station.

A	End piece	E	Slot
B	External pressure ring	F	Tie-rod
C	1-in. Tube pressure ring	G	Baffle
D	Flanged coupling	H	Baffle spacer

INTERNAL DIMENSIONS

LENGTH 2 ft.

DIAMETER $8\frac{3}{4}$ ins.

CROSSFLOW MEASURING STATION

Fig. 3.3 Crossflow Measuring Station

B. MEASUREMENT OF LOOP VARIABLES

1. Pressures

The static pressure at the test section entrance was maintained constant for all runs by means of an air vent control valve on the separator. The axial pressure gradient was measured by employing a manometer bank filled with manometer fluid (S.G. 2.95) under air. Care was taken to ensure that no water entered the pressure tap lines leading to the manometer panel.

Two steps were taken in order to minimize large pressure fluctuations and to reduce errors in reading the manometers. A 1/2-in long lucite rod with a bore of approximately 30 mils was inserted in the reference pressure line from the first tap. Also, for each run the manometer readings were taken by employing a Polaroid camera. The liquid levels in the manometers were observed for a time and then a picture was taken at an approximated average value of the pressure differentials indicated by the manometers. Thus, an instantaneous reading was obtained of all eight liquid levels on the manometer bank. The pressure at the first ring was used as a reference for all eight U-tubes. Therefore, each manometer indicated the pressure difference (in inches of 2.95 S.G. fluid) between the first piezometric ring and the rings located downstream on the test section.

The pressure drop across the slot in the test section to the measuring station was determined by using a U-tube manometer. Two

pressure rings on the outer wall of the measuring station were connected to the low pressure side of the manometer. A pressure tap on the main flow channel wall 3.5-in upstream from the slot was connected to the high pressure side of the manometer.

2. Air and Water Flows

(a) Main Channel

The air employed in this study was obtained from the plant power house and an automatic control loop maintained a constant test section flow. Four orifices were used as the primary flow measurement devices in conjunction with one differential pressure transmitter.

The orifices and assemblies conformed to ASME specifications [15] and were calibrated for the four following ranges as detailed in Table 3.1.

Table 3.1 Air Orifice Specifications		
Fig. 1 Designation	Orifice Diameter (in)	Range (lbm/hr)
FE - 2/1	1.941	1700-6000
FE - 2/2	1.087	500-1800
FE - 2/3	0.6455	160- 600
FE - 2/4	0.3540	40- 180

Water flows to the test section were automatically controlled over the range 50 to 50,000 lbm/hr at pressures up to 100 psig. Four orifices meeting ASME specifications were also used in conjunction with one

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differential pressure transducer to meter the water flows to the test section (See Table 3.2). All orifices were calibrated over the temperature range 45°F to 100°F.

Table 3.2 Water Orifice Specifications

Fig. 1 Designation	Orifice Diameter (in)	Range (lbm/hr)
FE - 1/1	1.665	12,000-50,000
FE - 1/2	0.9255	4,000-15,000
FE - 1/3	0.5400	1,400- 5,000
FE - 1/4	0.2960	500- 1,500

(b) Crossflow

Three calibrated rotameters were installed in parallel to meter the air crossflow rate. The respective ranges of the rotameters were 0-98.5 SCFH, 0-627 SCFH, and 0-91 SCFM. Water crossflow rates were determined by the direct weighing technique.

3. Temperature

Both the air and water inlet stream temperatures were determined using thermocouples located downstream of the orifice bank (See Fig. 3.1.). Both temperatures were recorded. The temperature of each crossflow stream was obtained using a mercury in glass thermometer.

4. Void Fraction

Initially it was planned to obtain the average void fraction in the test section employing quick closing valves. For this reason two ball valves were mounted 20-in apart in the test section downstream of the crossflow measuring station. Care was taken to ensure that the flow area remained constant and the two ball valves were mechanically linked to allow simultaneous opening and closing of the valves.

Due to the design of the void fraction measuring section the maximum void fraction which could be determined was approximately 85% and unfortunately for every run, the void fraction was greater than this figure.

IV EXPERIMENTAL PROCEDURE

A. AIR-WATER LOOP

At the beginning of an experimental run, the proper orifice for the desired air flow rate was selected and the orifice flange taps were valved into the differential pressure cell. The static pressure upstream of the orifice was adjusted to read 50 psig. At this pressure, previous calibration curves of the air flow rate vs. pressure drop across the orifice flanges had been prepared. By setting the controller at the required pressure drop across the orifice meter, the desired air flow rate was obtained. Manual control of the back pressure at the air-water separator allowed the pressure at the entrance to the test section to be maintained at 35 psig.

After the loop was pressurized with air, the desired water orifice was selected and valved into the control system. The loop pump was started and the water control valve was adjusted by manual operation of the controller until the desired water flow rate was established. At the same time the back pressure at the separator was adjusted to maintain the test section entrance pressure at 35 psig.

Steady-state conditions were assumed to be established when the air and water feed stream temperatures became constant at 80°F. The manometer lines to the piezometric rings were checked visually to ensure that any water entering the lines was removed by opening a valve in the bottom of the appropriate trap.

B. CROSSFLOW MEASUREMENTS

With no crossflow through the slot, the axial pressure drop was noted first on the manometer panel. This was the "zero" reading for each individual run. Next, by adjusting the needle valve on the air take-off line, the radial pressure difference between the flow channel and the measuring station, i.e. across the slot, was set. The air rotameter gave the crossflow rate for each radial pressure drop.

Similarly, the water flow was measured by controlling a valve on the water line to maintain a constant level of water in the crossflow measuring station. The water flow from the station was weighed directly or caught in a graduated cylinder over a known time interval to give the quantity of water discharging through the slot.

For each increase in radial pressure drop across the slot, the axial pressure drop was read from the manometer panel. This procedure gave the axial pressure variation in the main flow channel due to the crossflow through the slot.

V THEORY OF TWO-PHASE FLOW IN A MANIFOLD

Consider a vertical pipe of constant cross-sectional area with a continuous slit of width δ in which two-phases (gas-liquid) are flowing under adiabatic conditions (See Fig. 5.1).

This is termed a continuous blowing manifold which divides a large stream entering the pipe by allowing crossflow through a longitudinal slot or porous strip running continuously along the main channel.

Fig. 5.1 Continuous Blowing Manifold

A. BASIC EQUATIONS

In the one-dimensional macroscopic balance equations below it is assumed that:

- i) mean values of the density and velocity exist at any plane normal to flow;
- ii) the area occupied by the gas plus that occupied by the liquid equals the flow cross-sectional area,
- iii) uniform pressure exist at any plane normal to the conduit axis.

1. Momentum Balance

A macroscopic momentum balance over an element of length Δz can be written as:

$$\frac{d}{dt} [P_{tot}] = \left\{ e_N \langle v_N^2 \rangle A_N - e_{N+1} \langle v_{N+1}^2 \rangle A_{N+1} \right\} + \left\{ p_N A_N - p_{N+1} A_{N+1} \right\} g_c - [F] g_c + m_{tot} \vec{g} \quad (5.1)$$

where $\mathcal{L}$ = void fraction

P_{tot} = total momentum of the system

$$= [W_l + W_g] \Delta z = A[(1 - \mathcal{L}) e_l v_l + \mathcal{L} e_g v_g] \Delta z$$

$$\left\{ e_N \langle v_N^2 \rangle A_N - e_{N+1} \langle v_{N+1}^2 \rangle A_{N+1} \right\} = \text{net rate of momentum flux due to bulk fluid motion}$$

$$= - \frac{\Delta}{A} \left[\frac{W_l^2}{e_l (1 - \mathcal{L})} + \frac{W_g^2}{e_g \mathcal{L}} \right]$$

$$= - \Delta A \left[e_l (1 - \mathcal{L}) v_l^2 + e_g \mathcal{L} v_g^2 \right]$$

$$\left\{ p_N A_N - p_{N+1} A_{N+1} \right\} g_c = \text{force differential across the control element}$$

$$= - A g_c \Delta p$$

$$- [F] g_c = \text{net force of the solid surfaces}$$

$$= - \tau_t p_c \Delta z g_c$$

$$\left\{ m_{tot} \vec{g} \right\} = \text{force of gravity on the total mass of the fluid in the control volume}$$

$$= -A g [e_l (1 - \alpha) + e_g \alpha] \Delta z$$

Substituting the above terms in Eq. 5.1, the macroscopic momentum balance is obtained:

$$\begin{aligned} \frac{d}{dt} [W_l + W_g] \Delta z = & -\frac{\Delta}{A} \left[\frac{W_l^2}{e_l (1 - \alpha)} + \frac{W_g^2}{e_g \alpha} \right] - A g_c \Delta p \\ & - \tau_t p_c \Delta z g_c - A g [e_l (1 - \alpha) + e_g \alpha] \Delta z \end{aligned} \quad (5.2)$$

At steady-state conditions, the total drop in static pressure per element of length Δz becomes:

$$\begin{aligned} \frac{\Delta p}{\Delta z} = & -\tau_t \frac{p_c}{A} - \frac{\Delta}{A^2 g_c \Delta z} \left[\frac{W_l^2}{e_l (1 - \alpha)} + \frac{W_g^2}{e_g \alpha} \right] \\ & - g/g_c [e_l (1 - \alpha) + e_g \alpha] \end{aligned} \quad (5.3)$$

The total pressure drop for two-phase flow is normally separated into the sum of three components:

$$\left(\frac{\Delta p}{\Delta z} \right)_{\text{total}} = \left(\frac{\Delta p}{\Delta z} \right)_{\text{friction}} + \left(\frac{\Delta p}{\Delta z} \right)_{\text{momentum}} + \left(\frac{\Delta p}{\Delta z} \right)_{\text{gravity}}$$

The expression for each component can be obtained readily from Eq. 5.3.

In Eq. 5.3, the momentum contribution to the total pressure drop has been written assuming that the crossflows have lost all of their axial component of velocity before crossing the slot. However, to

account for the possibility that the velocity of the crossflow stream may not be perpendicular to the mainstream flow when leaving the elemental control volume, the momentum balance must be modified. Acrivos et. al.(13) have suggested the use of a momentum correction factor, k , to modify the momentum contribution of the total pressure drop as:

$$\left(\frac{\Delta p}{\Delta z} \right)_{\text{momentum}} = \frac{-k}{A^2 g_c} \frac{\Delta}{\Delta z} \left[\frac{W_l^2}{\rho_l (1 - \alpha)} + \frac{W_g^2}{\rho_g \alpha} \right]$$

This momentum correction factor is expected to be less than unity with its value obtained experimentally. Here values of k have been evaluated for single phase air flow and two-phase air-water flow.

Eq. 5.3 can be written in terms of total mass flux, G and quality, x , as:

$$\frac{\Delta p}{\Delta z} = -\tau_t \frac{P_c}{A} - \frac{k \Delta}{g_c \Delta z} \left[G^2 \left[\frac{(1-x)^2}{\rho_l (1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right] \right] \\ (g/g_c) [\rho_l (1-\alpha) + \rho_g \alpha] \quad (5.4)$$

2. Mass Balance

The change in mass flow rate of liquid in the pipe over the elemental length Δz is equal to the liquid crossflow rate.

$$-A \frac{\Delta}{\Delta z} [(1-\alpha) \rho_l v_l] = -\frac{\Delta}{\Delta z} [W_l] = \frac{W_l^x}{\Delta z} \quad (5.5)$$

Here W_l^x is mass crossflow rate.

Similarly, the mass crossflow rate for the gas over the elemental length Δz becomes:

$$-A \frac{\Delta}{\Delta z} [\mathcal{L} \rho_g V_g] = -\frac{\Delta}{\Delta z} [W_g] = \frac{W_g^x}{\Delta z} \quad (5.6)$$

The problem of relating the crossflow rates to the transverse pressure gradient, however, still remains. There are at least three possible ways of treating the resistance to transverse flow of gas and liquid. First, an orifice equation can be employed which gives [16]:

$$\begin{aligned} W_{lt}^x &= C_{lt} b \Delta z (1 - \mathcal{L}) \sqrt{2 g_c \rho_l (p_m - p^o)_t} \\ &= C_{lt} b \Delta z (1 - \mathcal{L}) \sqrt{2 g_c \rho_l (\Delta P_r)} \end{aligned} \quad (5.7)$$

$$\begin{aligned} \text{and } W_g^x &= C_{gt} b \Delta z \mathcal{L} \sqrt{2 g_c \rho_g (p_m - p^o)_t} \\ &= C_{gt} b \Delta z \mathcal{L} \sqrt{2 g_c \rho_g (\Delta P_r)} \end{aligned} \quad (5.8)$$

where C_{lt} is liquid discharge coefficient for two-phase flow conditions;

C_{gt} is the gas discharge coefficient for two-phase flow conditions;

$p_m = (p_{N+1} + p_N)/2$; and

p^o is the constant pressure outside of the pipe. The subscript, t , denotes two-phase flow conditions.

For a flowing liquid-gas mixture encountering a flow geometry change the static pressure drop can be defined in terms of the number of velocity heads as though liquid alone were flowing in the channel before expansion [17,18].

$$\begin{aligned}
 (p_m - p_o) &= K \left(\frac{1}{2} \rho_l \frac{v_l^2}{g_c} \right)^2 \\
 &= \frac{K G^2 x^2}{2 g_c \rho_l}
 \end{aligned} \tag{5.9}$$

For a two-phase pressure loss, the corresponding expression is

$$(p_m - p_o)_t = \frac{K}{2g_c} G^2 x^2 \left[\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_l (1-\alpha)} \right] \tag{5.10}$$

A third method [19] relates the frictional resistance to lateral crossflow by considering the radial component of the channel flow passing through the slot. For two-phase flow the Lockhart-Martinelli homogeneous theory multiplier [20] is employed. The equation relating the pressure drop across the slot and crossflow for single-phase liquid flow would then be:

$$(p_m - p_o) = \frac{f_l G^2 x^2 t}{\rho_l g_c D_{eq}} \tag{5.11}$$

where t is the thickness of the pipe wall and D_{eq} is the equivalent hydraulic diameter of the slot. For two-phase flow, Eq. 5.11 is modified to obtain:

$$(p_m - p_o)_t = \frac{M f_l G^2 x^2 t}{l g_c D_{eq}} \tag{5.12}$$

where M is the Lockhart-Martinelli multiplier. In this method it has been assumed that there is complete recovery of the acceleration pressure losses.

B. THEORETICAL MODELS EMPLOYED

All crossflow measurements reported here were made under separate flow conditions (i.e. two phases flowing side by side, the liquid on the wall and gas in the core with varying amounts of entrained liquid).

Fig. 5.2 Observed Flow Pattern in Adiabatic Air-Water Vertical Upflow

However, for these annular flow regimes no measurements were made of film thickness, entrainment rates or volume void fraction. An estimate of the minimum entrainment occurring here was made by comparison with the work of Gill et. al. (21) for vertical upflow of air-water in a 1 1/4-in bore tube at atmospheric pressure. At a total mass flux of $2 \times 10^5 \text{ (lbm)/(hr)(ft)}^2$ at 20% quality the minimum entrainment rate would be expected since this corresponds to the lowest gas and liquid velocities. Under these flow conditions Gill et. al. measured the entrained liquid to be approximately 28% of the total liquid flow with an annular slot type water injector. This prediction of entrainment is expected to be slightly high due to the differences in operating pressures and flow areas. For these same conditions 25% entrainment is predicted employing Fig. 11.33 in Wallis' and Collier's lecture notes (22). For

higher mass fluxes and qualities there would be considerably more entrained liquid. Under these conditions approaching mist flow the homogeneous model is recommended for predicting pressure drop (23). Also Martinelli's correlation for predicting void fraction has been successful (24).

1. Homogeneous Model

The homogeneous model was employed here to calculate the momentum correction factor, k , for two-phase flow conditions. In this simple model, it is assumed that:

- i) the gas and liquid velocities are equal (no slip);
- ii) thermodynamic equilibrium exists;
- iii) a single-phase friction factor (suitably defined) can be used for two-phase flow.

Since

$$V_l = \frac{W_l}{\rho_l A_l} = \frac{G(1-x)A}{\rho_l A_l}, \quad (5.13)$$

$$V_g = \frac{W_g}{A_g \rho_g} = \frac{G x A}{\rho_g A_g}, \quad \text{and} \quad (5.14)$$

$$A_l/A = 1 - \alpha, \quad A_g/A = \alpha \quad (5.15)$$

then from Eqs. 5.13, 5.14 and 5.15 setting $V_l = V_g$

$$(1 - \alpha) = \frac{(1 - x) \bar{\rho}}{\rho_g} \quad \text{and} \quad (5.16)$$

$$\mathcal{L} = \frac{x \bar{\mathcal{E}}}{e_g} \quad (5.17)$$

Here $\bar{\mathcal{E}} = 1/\bar{v}$ where $\bar{v} = x \bar{v}_g + (1 - x) \bar{v}_l$. Substituting Eqs. 5.16 and 5.17 into the previous expression for momentum pressure loss (Eq. 5.4)

$$\left(\frac{\Delta P}{\Delta z} \right)_{\text{momentum}} = - \frac{k}{g_c} \frac{\Delta}{\Delta z} \left[\frac{G^2}{\bar{\mathcal{E}}} \right] \quad (5.18)$$

Values of k were determined from

$$k = \frac{(\Delta P_a)_{\text{measured}}}{\Delta [G^2 / \bar{\mathcal{E}} g_c]} \quad (5.19)$$

2. Lockhart-Martinelli Correlation

The Lockhart-Martinelli correlation was developed initially from a series of studies of isothermal two-phase two-component flow in horizontal pipes. A generalized method was proposed for calculating the frictional pressure drop for isothermal two-component flow which was later extended to include acceleration and gravitational effects for the prediction of pressure drops during diabatic flow. The basic assumptions in this correlation are the following. Four flow categories exist depending on whether laminar or turbulent flow conditions are predicted when the single phase (gas or liquid) is flowing alone in the pipe. Secondly, there are equal pressure drops for the liquid and gas phases independent of the flow categories. A particular portion of the total cross-sectional flow area is assigned to each phase and the conventional friction pressure drop equations are assumed to be valid for each phase. The

ratio of the actual two-phase frictional pressure gradient to that which would result if the liquid flows alone in the pipe was reasoned to be only a function of the fraction and shape of flow area occupied by the liquid phase. Parameters

$$Q_l = \sqrt{(\Delta P)_t / (\Delta P)_l} \quad \text{and} \quad Q_g = \sqrt{(\Delta P)_t / \Delta P_g} \quad \text{thus}$$

were functions of $M = \sqrt{\frac{(\Delta P)_l}{(\Delta P)_g}}$ only. This unique correlation of

Q_l and Q_g by M was verified by their experimental data. Therefore, to use the Lockhart-Martinelli correlation only the single-phase frictional pressure drops have to be calculated.

The liquid holdup or liquid volume void fraction, $(1 - \alpha)$, is also a function of the single phase equivalent diameter and shapes which are also functions of M only. Lockhart-Martinelli also correlated $(1 - \alpha)$ uniquely by employing M . Hewitt et. al. (25) have fitted the liquid holdup correlation in the form $\ln(1 - \alpha) = \sum_{n=0}^5 a_n (\ln M)^n$ which greatly facilitates its use in a computer programme.

VI EXPERIMENTAL RESULTS

A. INTRODUCTION

The effect of five independent variables - mass flux, quality, slot length, slot width and pressure drop across the slot - on single-phase air and air-water crossflows in a flowing two-phase stream are reported. In addition the variation of the static pressure rise in the test section over the slot length due to crossflow is given as a function of the same independent variables.

Representative curves only are given here. The slot dimensions are listed below for use with all figures. All data are tabulated in Appendix I.

Slot Dimensions

Slot Designation	Length Inches	Width mils	β , $\frac{\text{Slot Area}}{\text{Pipe Area}}$
11	1.66	12	.0255
21	3.98	12	.0612
31	5.98	12	.0917
32	6.00	40	.306
33	6.00	80	.619

B. AIR CROSSFLOW IN SINGLE-PHASE FLOW

The mass flow rate of air, W_g^x , discharging through the slot in the pipe was found to be directly proportional to the square root of the pressure drop across the slot, $\sqrt{\Delta P_r}$, for all values of the main-stream mass flux, G . This has been shown in Fig. 6.1 for data obtained

Fig. 6.1 Single-Phase Air Crossflow

with the 1.66-in long by 12-mil wide slot and 5.98-in by 12-mil slot. The effect of the total mass flux on W_g^x is evident also in Fig. 6.1. By crossplotting the air crossflow rate against G for fixed values of $\sqrt{\Delta Pr}$ the air crossflow rate was found to be inversely proportional to the total air mass flux. (Note that $\Delta P_r = \Delta P_x + 1/2 \Delta P_a$.)

Since the variation of single-phase air discharge through the slot as a function of slot length, slot width and total mainstream mass flux was very similar to the two-phase air crossflow data, only the two-phase data is given here in graphical form. Air crossflow rates were found to be directly proportional to the slot width for fixed slot lengths and mass flux. Also the air discharge rates varied with slot length to the two-thirds ($2/3$) power for a fixed width and mass flux.

C. TWO-PHASE FLOW

1. Water Crossflow

In Fig. 6.2 a representative plot of the water crossflow rate W_l^x as a function of $\sqrt{\Delta Pr}$ is given for a fixed value of the total (air and water) mass flux at various qualities. As was the case for single-phase air, the water crossflow was found to be directly proportional to $\sqrt{\Delta Pr}$. When the mainstream quality is increased, while maintaining fixed values of the total mass flux and radial pressure drop, the mass flow rate of water decreased exponentially (Fig. 6.3).

Fig. 6.2 Air-Water Crossflow

Fig. 6.3 Air-Water Crossflow - Quality Varied

Fig. 6.4 Air-Water Crossflow - Mass Flux Varied

If the pressure drop across the slot and the quality of the main flow stream is fixed then the variation of water discharge rates with mass flux can be studied. In Fig. 6.4, the water crossflows are seen to decrease continuously with increasing mass flux at high qualities. At low qualities the water crossflow initially increases, reaches a maximum at a G of approximately $3.5 \times 10^5 \text{ (lbm)/(hr) (ft)}^2$, and then begins to decrease. At the intermediate qualities the water crossflow-mass flux curve exhibits ogee behaviour passing from convex to concave shape with increasing mass flux.

The water crossflows were found to be directly proportional to the slot width (Fig. 6.5) and to vary with slot length to the two thirds ($2/3$) power (Fig. 6.6) when the appropriate independent variables were maintained constant. This behaviour was similar to that exhibited by the single phase air.

2. Air Crossflow

Air crossflow rates were found (Fig. 6.2) to be directly proportional to $\sqrt{\Delta P}$ under two-phase flow conditions. The variation of air discharge rates with mainstream quality was much less as compared to the quality effect on water crossflow. The major portion of the gas crossflow rate increase occurred when the quality was increased from 20% to 30%. At the higher qualities (50-70%) the mainstream mass flux did not affect the air crossflow noticeably (Fig. 6.4). However, at the low qualities, the air crossflow rates began to decrease as the mass flux was increased.

Fig. 6.5 Air-Water Crossflow - Slot Width Varied

Fig. 6.6 Air-Water Crossflow - Slot Length Varied

Fig. 6.7 Axial Static Pressure Variation with Crossflow

At a fixed pressure drop of 0.25 p.s.i. across the slot in the test section, the air crossflow was found to be proportional to the slot width (Fig. 6.5). For all values of quality (less than 1), all values of total mainstream mass flux and all slot lengths, if the air crossflow vs. slot width curve were extrapolated to zero air crossflow, then a critical slot width of approximately $7 \frac{1}{2}$ mils was obtained. This behaviour was not observed under single-phase air conditions nor for the water crossflow in two-phase flow. As the pressure drop across the slot was increased, then the critical slot width decreased.

D. Axial Pressure Recovery

Figure 6.7 illustrates the main flow axial pressure recovery. The continuous line is the pressure gradient in the pipe for the given flow conditions when there is no crossflow. In order to obtain the static pressure increase over the slot length the pressure readings downstream from the slot with crossflow were extrapolated back to the slot centre line. With the pressure gradient without crossflow used as a reference, the difference between the reference gradient and the extrapolated gradient at the slot centre line gave the recorded value of the pressure rise.

In Fig. 6.8 the value of the momentum correction factor, k , for single phase air was obtained as 0.708. This is the slope of the straight line of ΔP_a vs $-\Delta(G\bar{V})$, the theoretical axial pressure recovery.

Fig. 6.8 Momentum Correction Factor for Single-Phase Air

Fig. 6.9 Momentum Correction Factor for Two-Phase Air-Water Mixtures

When this same curve was plotted for the two-phase data, the momentum correction factor was found to be a function of the mainstream quality. In Fig. 6.9, the mean velocity of the two phase mixture was obtained by using a homogeneous specific volume evaluated $\bar{v} = x \bar{v}_g + (1 - x)\bar{v}_l$. The value of k increased with quality and k was found to be greater at a quality of 70% than at 100% quality (or single-phase air conditions). The values of k noted on the curves (Fig. 6.8 and 6.9) were those obtained employing the least squares method.

VII DISCUSSION OF RESULTS

A. HYDRAULIC RESISTANCE TO CROSSFLOW

1. Single Phase Air

The successful use of the orifice equation by previous investigators [7,9,10] to correlate manifold data made it a logical choice for correlating our data. Also, the linear dependency of the air crossflow on the square root of the pressure gradient ΔP_r , indicated that an orifice type equation might be successful (See Fig. 6.1). Fig. 7.1 is a plot of the air discharge coefficients vs the ratio of the velocity of the crossflow stream through the slot to the main flow velocity. The dimensionless velocity ratio $V_g^x/\bar{V}$ was found to be the most valuable variable for correlation purposes. A definite geometry effect on the discharge coefficient was obtained which agrees with the results of previous investigators, notably McNown [2] and Soucek and Zelnick [7]. Discharge coefficients for slots 21, 32 and 33 exhibited the same behaviour, while the smallest slot 11 had the highest discharge coefficient. Slot 31 had a much lower discharge coefficient at the same value of $V_g^x/\bar{V}$.

Oakey's [10] water data had C_d exhibit the same behaviour with $\bar{V}_l^x/\bar{V}$ but there was no noticeable effect of geometry for β varied from .054 to .685. Soucek's and Zelnick's [7] water data for β values of 0.194, 0.268 and 0.34 also had C_d increasing with $\bar{V}_l^x/\bar{V}$. There was a slight increase in the value of C_d as β decreased from 0.34 to 0.194.

Fig. 7.1 Single-Phase Air Discharge Coefficients

The geometry effect is illustrated in Fig. 7.2 where C_g decreases linearly with β until $\beta = 0.1$. No definite conclusions can be made here for $0.1 \leq \beta \leq 0.3$. However, for $\beta > 0.3$, C_g remained constant. It is of interest to note that data points labelled 1, 2 and 3 are for increasing slot length and with a fixed slot width. Also, data points labelled 3, 4 and 5 are for increasing slot width with the length held constant. In general, [25] for concentric orifices, as the β ratio is increased, the value of the discharge coefficient increases. In Fig. 7.1 it is evident that the greatest resistance to crossflow (which corresponds to low values of C_g) occurs at small values of $\bar{V}_g^x / \bar{V}$. Thus, there is a high resistance to crossflow at low crossflow rates for a fixed total mass flux.

This behaviour can be linked to the momentum of the mainstream. The principal effect of high mainstream velocity approaching the slot would appear to be such as to reduce the port discharge. McNown's experiment [2] has indicated that the effective area in a lateral may be considerably less than the total area of the lateral for high mainstream velocities. Also, the crossflow velocity is dependent upon the static head within the tube rather than upon the total head. Therefore, the effect of high mainstream velocity is to reduce the rate of crossflow because of larger losses and reduced effective area rather than to increase the crossflow because of the additional kinetic energy. This can be discussed further in the light of Soucek and Zelnick's comments [7]. They suggested that the jet discharging from a continuous longitudinal slot incurred no loss of momentum in the direction parallel to the mainstream flow, but that each element retains a longitudinal velocity component equal to the

Fig. 7.2 Discharge Coefficients for β Varied

pipe velocity at the point of issue. The normal component of velocity appears to take place at the expense of the pressure head only. The angular direction of the discharge stream with respect to the slot axis, Θ , was related by Koestel and Tuve [11] to a vertical sum of two velocities as shown in Fig. 7.3.

Fig. 7.3 Velocity Vectors after Koestel and Tuve [11]

It was observed here that the air crossflow rate is inversely proportional to the mass flux, in agreement with McNown's discussion. (See Fig. 6.1.) Koestel and Tuve also found that if the β ratio is held constant, the general shape of the stream will be practically independent of the pressure in the conduit or the width of the slot. The stream discharge velocity along the length of the slot remains almost constant, but its direction changes (See Fig. 7.4.)

Fig. 7.4 Discharge Velocity Vectors

There have been several different methods attempted to describe the crossflow driving force, ΔP_r , in the orifice equation. Soucek and Zelnick [7] defined ΔP_r as the difference between the external pressure and the intersection of the hydraulic gradient at a vertical erected at the center of the slot. Oakey [10] defined a constant C , equal to Q^x/Q , to account for that portion of the mainstream velocity head which is used to produce the crossflow. The method of defining ΔP_r used here was to extrapolate the upstream and downstream gradients back to the vertical erected at the slot mid point after the method of McNown. Then ΔP_r is equal to the arithmetic mean static pressure at the intersection of the upstream and downstream gradients minus the external reference pressure in the crossflow measuring station. Soucek and Zelnick neglected the action of the mainstream momentum effects in their definition of ΔP_r . Oakey added a portion of the velocity head equal to $(\bar{V}^x/\bar{V})$ to the static pressure just before the lateral to obtain ΔP_r .

One further aspect to be considered is the medium into which the crossflow stream discharges. One might expect that the values obtained here for C_g could be different if the stream discharged into water or another flowing system instead of the static measuring station employed here.

2. Two-Phase Flow

(a) Water Crossflow

There were three possible methods considered in Chapter V for relating the hydraulic resistance to crossflow to the actual crossflow. One method

was the use of discharge coefficients which gives:

$$C_{lt} = \frac{W_l^x}{b \Delta Z (1 - \alpha) \sqrt{2g_c \rho_l (p_m - p^o)_t}} \quad (7.1)$$

The liquid discharge coefficients measured here are shown in Fig. 7.5 as a function of $V_l^x/\bar{V}$. All values of $\mathcal{L}$ used here were computed employing the Martinelli-Lockhart correlation [20] based upon crossflow properties. Slot 31 was omitted from the plot since it was consistently lower than values determined for the other slots. A slight geometry effect which is within the experimental scatter of the data, still persists as was the case for single phase air. The correlation obtained for slots 11, 21, 32, and 33 is of the form:

$$\ln C_{lt} = 0.0170 + 0.1496 \ln (V_l^x/\bar{V}) - 0.0281 [\ln(V_l^x/\bar{V})]^2 \quad (7.2)$$

with a standard deviation of $\pm 5\%$. For slot 31 equation 7.3 was obtained.

$$\ln C_{lt} = -0.2375 + 0.112 \ln (\bar{V}_l^x/\bar{V}) - 0.0216 [\ln(\bar{V}_l^x/\bar{V})]^2 \quad (7.3)$$

with a standard deviation of $\pm 1.4\%$.

The second method was the use of a loss coefficient, K , defined by

$$(p_m - p_o) = \frac{KG^x^2}{2 g_c \rho_l} \quad (7.4)$$

Fig. 7.5 Two-Phase Flow Discharge Coefficients

Here the total two-phase mass flux is considered in terms of liquid alone flowing through the slot. In Fig. 7.6 the loss coefficient in terms of number of water velocity heads is shown to be strongly dependent on the crossflow quality. As the crossflow quality increases from 0 to 100% the loss coefficient increases approximately three decades. The single phase water data ($X^x = 0$) was obtained from Oakey [10]. Here, the dimensionless velocity ratio, $\bar{V}^x/\bar{V}$, was once more found to be a significant correlation parameter. A final correlation of the data for slots 21, 32 and 33 was obtained:

$$\ln [K/250(X^x)^{1.4}] = 0.87363 - 0.18280 [\ln(\bar{V}^x/\bar{V})] + 0.14818 [\ln(\bar{V}^x/\bar{V})]^2 \quad (7.5)$$

with a standard deviation of 0.044.

Fig. 7.7 gives a plot of the final correlation as a function of $\bar{V}^x/\bar{V}$. The difficulty in employing this model is that one further correlation is required to relate the crossflow quality to $\bar{V}^x/\bar{V}$. One could as a first approximation assume that the crossflow quality is identical to the mainstream quality.

The third method employed the Martinelli-Lockhart homogeneous theory multiplier [20]. The equation relating the pressure drop across the slot and crossflow is:

$$(p_m - p_o)_t = \frac{MF \cdot G^x{}^2 \cdot t}{Q \cdot g_c \cdot D_{eq}} \quad (7.6)$$

Fig. 7.6 Loss Coefficient in Terms of Number of Water Velocity Heads

Fig. 7.7 Corrected Loss Coefficient.

A comparison with this method indicated that the predicted hydraulic losses were considerably less than that actually measured especially for the high quality runs. Once more a correlation of X^x with some other flow parameter is required to enable the individual crossflow rate (air and water) to be predicted. The orifice equation employing the discharge coefficients is the method recommended here for calculating crossflow rates.

(b) Air Crossflow

The three methods of calculating water crossflow rates apply as well to air crossflow rates and the same conclusions hold true. Fig. 7.5 is the plot of C_{gt} vs $\bar{V}^x/\bar{V}$ where

$$C_{gt} = \frac{W_g^x}{b \Delta Z \sqrt{2g_c} \rho_g (p_m - p_o)_t} \quad (7.7)$$

Here again a slight geometry effect is evidenced but the error is within the experimental data scatter. A correlation was obtained using slots 11, 21, 32, and 33:

$$\ln C_{gt} = -0.8311 + 0.2674 \ln (\bar{V}_g^x/\bar{V}) - 0.07163 [\ln (\bar{V}_g^x/\bar{V})]^2 \quad (7.8)$$

with standard deviation = 5%.

For slot 31, Equation 7.9 was obtained:

$$\ln C_{gt} = -1.166 + 0.262 \ln (\bar{V}_g^x/\bar{V}) - 0.07174 [\ln (\bar{V}_g^x/\bar{V})]^2 \quad (7.9)$$

with standard deviation = 5.4%.

One further point for discussion is that for all values of quality less than 100% all values of total mass flux and all slot lengths, if the air crossflow vs slot width curve were extrapolated to zero air crossflow, then a critical slot width of 7-1/2 mils was obtained at a value of $\Delta P_r = 0.25$ psi (See Fig. 6.5.). This behaviour was not observed under single-phase air conditions nor for water crossflow in two-phase flow. It is believed here that surface tension forces must be included in any correlation which seeks to predict this critical slot width.

B. AXIAL PRESSURE RECOVERY

1. Single Phase Flow

Fig. 6.8 compared the measured axial pressure recovery, ΔP_a , with the theoretical axial pressure recovery when the momentum correction factor, k , was defined as:

$$\Delta P_a = k \left[\frac{G_1 \bar{V}_1 - G_2 \bar{V}_2}{g_c} \right] \quad (7.10)$$

For ideal axial pressure gain, the value of k would be 1.00. The value obtained here by a least squares fit of the data is 0.708 which is of the same order of magnitude as values reported by Acrivos et. al. [13]. Values of ΔP_a for slot 11 could not be measured as the pressure rise across the slot was too small.

2. Two-Phase Flow

Fig. 6.9 is the plot of the measured axial pressure recovery ΔP_a vs the theoretical axial pressure recovery where k is again defined by the homogeneous model:

$$\Delta P_a = k \left[\frac{G_1 \bar{V}_1 - G_2 \bar{V}_2}{g_c} \right] \quad (7.6)$$

The quality of the main flow stream now becomes a parameter. Here k increases with increasing quality until $x = 0.7$ where it reaches a maximum value. In Fig. 7.8 a plot of measured axial pressure recovery vs predicted axial pressure recovery is given where k has been redefined as

$$\Delta P_a = k \left(\Delta \left\{ \frac{G^2}{g_c} \left[\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_l (1-\alpha)} \right] \right\} \right) \quad (7.7)$$

This method of predicting k eliminated quality as a parameter. The value of k was obtained as 1.0 by a least squares fit. This would seem to indicate that the crossflow stream discharged from the slot at right angles to the pipe axis. This was not visually observed to be true, so this definition of the momentum correction factor gives values of k larger than the actual. Here the Lockhart-Martinelli model [20] was employed to predict all α values. This slip model for correlating the two-phase momentum correction factor, k , considers the total pressure rise to be made up of the sum of the liquid phase rise plus the gas phase rise. The slip model is recommended for use in predicting two-phase flow axial pressure rises under conditions similar to those found here. In order

Fig. 7.8 Momentum Correction Factors for Two-Phase Air-Water Mixtures at 50 psia

to predict the value of the axial pressure recovery Fig. 7.9 is employed.

No case of "zero" crossflow was observed. As the ΔP_x (pressure drop actually measured experimentally) was reduced to zero, oscillations in pressure between the inside pipe pressure and the pressure in the measuring station were set up. Consequently, a continuous sucking and blowing manifold was observed when "zero" ΔP_r was attempted. The mean value on the manometer was used as the "zero" ΔP_r reading. These oscillations did not affect the axial pressure gradient. When the value of the axial pressure gradient was compared for the same two-phase conditions both before and after the slot was inserted, it was found that the two agreed to within 4% (See Table II-1, II-2, Appendix 2.). The axial pressure gradients were determined for the flow under study before the slot was cut in the tube to insure that a reference axial pressure gradient would be available for the case of "zero" crossflow.

C. FLOW DISTRIBUTION THROUGH SLOT

As was discussed above, the flow through the slot under two-phase conditions maintained a constant appearance of a mist flow no matter how the system parameters were varied in the range of interest. It is important to be able to predict the quality of the crossflow stream. This would have applications in heat exchanger design with regards to unequal heat loads for the tubes. This aspect of rod spacing is of special importance in coolant fuel assemblies for nuclear reactors since one does not want to pass a low quality crossflow from a channel that already has a high quality because of the danger of burnout. It was

observed that larger slots pass a greater quality crossflow stream than the smaller slot. However, they both converge to $\bar{x}^x/\bar{x} = 1$ at $\bar{V}^x/\bar{V} = .8$. By use of a design curve and the iteration technique, the desired slot dimensions and $\bar{V}^x/\bar{V}$ ratios could be determined to enable the crossflow stream to have the same quality as that possessed by the main flow stream. Since it was not possible in the study to determine the angle of the crossflow air stream, single-phase water was run in the system to enable the flow characteristics of the crossflow stream to be visually observed. Fig. 7.9 shows the results and a two-phase crossflow discharge for comparison.

Fig. 7.9 Discharge Characteristics

Considering first the two-phase flow distribution, the general shape of the flow through the slot was constant for all flow conditions

in the main flow channel as far as attitude and angle of emittance are concerned. This agrees with the work of Koestel and Tuve [11]. The crossflow stream was observed to be a homogeneous mixture of air and water similar to mist flow for all qualities. It was hoped that the distribution angle could be photographed. To this end, the crossflow measuring station was constructed of clear plexiglass. However, during each run the glass misted up considerably and a photograph was unobtainable. From Fig. 7.9(b) the effect of surface tension is evident at low ΔP_r (corresponding to low $\bar{V}_g^x/\bar{V}$ in Fig. 7.1). In Fig. 7.9(c) the ΔP_r is large enough to overcome the surface tension forces somewhat. It is postulated here that surface tension forces are responsible for the critical slot width of 7-1/2 mils observed for the air crossflow in two-phase flow at a ΔP_r equal to 0.25 psi. For the purpose of this discussion one could use the analogy of bubble growth. In the beginning, the bubble exists in a thermodynamic equilibrium between its vapour pressure and surface tension forces. Similarly, if we take a flow particle that is hovering on the verge of becoming a crossflow particle it is in a dynamic equilibrium between the radial pressure gradient, ΔP_r , and the surface tension forces. Carrying the analogy further, for the bubble to grow, its vapour pressure must overcome the surface tension forces and the inertia of the liquid around it. The potential crossflow particle must be acted upon by a large enough ΔP_r to overcome the surface tension forces at the slot and its own inertia due to the fact that it is moving with main stream velocity and must now move in a direction at right angles to its present path. This latter point may be an important factor, especially when considering a small slot. However, it has been shown experimentally by McNown [2] that the position of the flow in-

volved in crossflow will come from the slower moving flow closer to the wall.

D. AXIAL PRESSURE GRADIENTS

1. Single-Phase Air

The Fanning equation was employed to find the friction factor for the plexiglass flow channel. As would be expected, the data fall on the curve for a smooth pipe. Richardson's data [27] for single-phase water in a plexiglass test section is also included in Fig. 7.10 for comparison.

2. Two-Phase Air-Water Flow

The two-phase pressure drop data over the test section when no crossflow occurred is listed in Appendix II. The data is very reproducible even though it was taken months apart and under slightly different ambient conditions.

The Martinelli-Lockhart correlation [20] was employed (See Fig. 7.11) and fits the data very well in the low quality, low mass flux region but falls down in the upper regions. This is to be expected, since the correlation was intended for annular flow and the upper quality, mass flux runs were visually observed to be in the annular flow with high entrainment and mist flow regions. Owens Model [28] was tested but was not successful.

Fig. 7.10 Fanning Friction Factor vs Reynolds Number for Lucite Test Sections

Fig. 7.11 Comparison of Lockhart-Martinelli Model with Experimental Data

VIII CONCLUSIONS

A. SINGLE-PHASE AIR

The orifice equation proved to be successful. The general shape of the curves is consistent for each slot studied although a definite geometry effect is evidenced. It may be concluded that:

1. Since the value of the discharge coefficient changes rapidly over the range $0 \leq \bar{v}_g^x / \bar{v} \leq 0.5$, the experimental determination of the discharge coefficient is necessary.
2. The use of a constant value for C_g in the orifice equation for design purposes would not be feasible. However, for values of $\bar{v}_g^x / \bar{v} > 0.5$, the C_g approaches a constant value of about 0.63.
3. Air crossflow rate is inversely proportional to the mass flux.
4. The value of k , the momentum correction factor, was obtained here as 0.708.

5. The friction factor commonly used for smooth pipes is also applicable to plexiglass test sections.

B. TWO-PHASE FLOW

1. Water Crossflow

1. A slight geometry effect, which is within the experimental scatter of the data, is evidenced when the orifice equation is applied.
2. The orifice equation employing discharge coefficients is the method recommended here for calculating crossflow rates.

2. Air Crossflow

1. The same conclusions hold true here as for the water crossflow rates.

3. Axial Pressure Recovery

1. The slip model is recommended for use in predicting two-phase flow axial pressure rises under conditions similar to those found here.
2. The value of k obtained employing the slip model is concluded to be 1.0.

3. The presence of the slot in the tube test section did not affect the axial pressure gradients for the case of "zero" crossflow.

4. Axial Pressure Gradients

1. The Lockhart-Martinelli correlation [20] fits the data very well in the low quality, low mass flux region.

NOMENCLATURE

Dimensions are given in terms of mass (M), length (L), time (t) and temperature (T).

A_d	Discharging tube area	L^2
A_o	Sum of areas of all discharge holes	L^2
A_p	Cross-sectional area of manifold	L^2
b	Slot width	L
C	Oakey's constant Q_d^x/Q	
C_g	Single-phase air discharge coefficient	
C_l	Single-phase water discharge coefficient	
C_{gt}	Two-phase air discharge coefficient	
C_{lt}	Two-phase water discharge coefficient	
D	Diameter of flow channel	L
f	Fanning friction factor	
G	Total mass flux	M/tl^2
g_c	Gravitational conversion factor	
h_f	Head loss due to friction	L
K	Loss coefficient	
k	Momentum correction factor	
M	Lockhart-Martinelli parameter	
m	Mass	M
N_{Re}	Reynold's Number	
P_c	Perimeter of channel	L
P	Static pressure	M/L^2

$\frac{\Delta P}{\Delta L}$	Pressure drop per unit length	M/L^3
$\frac{\Delta P}{\Delta Z}$	Pressure drop across the slot	M/L^3
ΔP_a	Axial pressure recovery across the slot	M/L^2
ΔP_r	Radial pressure drop through the slot $= \Delta P_x + \Delta P_a/z$	M/L^2
ΔP_x	Measured pressure drop through the slot	M/L^2
Q	Volumetric flow rate	L^3/t
t	Test section wall thickness	L
$\bar{V}$	Average velocity	L/t
$\bar{v}$	Average specific volume	L^3/t
W	Mass flow rate	M/t
X	Quality	Dimensionless
Z	Slot length	L

Greek Symbols

$\mathcal{L}$	Void fraction	
β	$\frac{\text{Slot area}}{\text{Pipe area}}$	
Q	Function of M utilized in calculating two-phase pressure drop	
μ	Viscosity	M/Lt
ρ	Density	M/L^3
$\bar{\rho}$	$\mathcal{L}\rho_g + (L - \mathcal{L})\rho_l$	M/L^3
γ	Shear stress tensor	M/t^2L

Subscripts

a	Axial
g	Gas or air phase
l	Liquid or water phase

m	internal
r	radial
t	two-phase
x	measured
1	upstream from slot
2	downstream from slot
3	slot condition

Superscripts

o	Conditions external to the tube
x	Crossflow stream conditions

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APPENDIX I

SINGLE-PHASE AIR AND AIR-WATER

CROSSFLOW DATA

APPENDIX I

TABLE I.1

Water and Air Crossflow Rates at 50 psia

Slot Length = 1.66 in.
Slot Width = 0.012 in.

Run	X Quality	G _T Total Mass Flux lbm/hr-ft ² x 10 ⁻⁶	Δ P _x Pressure Drop Across Slot in. water	W _G ^x Air Phase Crossflow lbm/hr	W _L ^x Water Phase Crossflow lbm/hr
L-W-X-G					
1-1-1-1	0.200	0.205	9.30 14.82 20.88 28.09	4.43 6.35 8.64 10.57	37.8 46.0 54.5 58.2
1-1-2-1	0.303	0.201	5.91 14.29 20.43 32.95	3.69 7.08 9.86 14.07	24.0 32.8 56.6 40.8
1-1-3-1	0.500	0.199	3.73 10.11 21.16 30.58	3.32 7.00 12.65 15.93	6.5 10.8 17.1 17.8
1-1-4-1	0.705	0.200	5.02 9.61 21.53 34.81	3.67 6.98 14.03 18.71	5.3 7.2 9.8 12.8
1-1-1-2	0.212	0.279	13.62 16.00 27.25 49.56	2.20 5.88 11.69 18.71	25.8 40.6 51.7 66.7

Run	X	G _T	ΔP_x	W_G^x	W_L^x
1-1-2-2	0.294	0.303	1.82	1.84	11.0
			13.10	5.95	24.0
			24.80	12.86	33.5
			35.64	16.83	44.4
1-1-3-2	0.495	0.302	3.03	2.06	3.6
			10.05	6.98	6.1
			18.32	11.22	9.8
			31.70	16.37	12.6
1-1-4-2	0.698	0.299	2.69	2.20	2.3
			11.16	7.20	3.5
			20.88	12.16	4.9
			33.06	17.31	6.3
1-1-1-3	0.207	0.485	8.53	1.47	17.4
			17.64	5.88	34.8
			27.35	9.36	42.5
			39.31	13.10	51.5
1-1-2-3	0.307	0.504	5.86	1.47	5.4
			12.11	6.25	12.2
			18.75	9.36	14.6
			33.29	14.98	19.4
1-1-3-3	0.492	0.498	4.41	3.31	4.8
			13.99	8.89	8.6
			27.98	13.81	12.2
			43.56	18.96	14.5
1-1-4-3	0.702	0.509	1.66	1.48	0.8
			8.07	5.42	2.0
			21.53	11.98	3.6
			42.77	19.03	5.0
1-1-1-4	0.202	0.698	4.28	2.20	19.4
			9.86	5.15	31.1
			21.62	9.60	39.6
			38.94	14.20	60.8
1-1-2-4	0.306	0.683	1.46	1.84	4.6
			7.40	5.16	8.1
			18.92	11.49	13.2
			36.60	17.12	16.9

TABLE I.2

Water and Air Crossflow Rates at 50 psia

Slot Length = 3.985 in.
 Slot Width = 0.012 in.

Run	X Quality	G_T Total Mass Flux lbm/hr-ft ² $\times 10^{-6}$	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Phase Crossflow lbm/hr	W_L^x Water Phase Crossflow lbm/hr	ΔP_a Axial Pressure Recovery in. water
L-W-X-G						
2-1-1-1	0.199	0.204	2.69 6.35 18.49 34.81	2.21 6.62 15.91 24.34	33.5 56.5 91.1 105.0	0.11 0.21 0.55 0.70
2-1-2-1	0.302	0.201	2.28 5.66 17.64 30.25	2.20 6.98 17.77 26.19	23.8 39.2 66.1 78.4	0.42 1.01 1.37
2-1-3-1	0.502	0.200	2.22 6.50 19.80 31.36	4.41 10.30 23.41 31.84	12.6 20.5 33.4 33.8	0.50 0.92 1.37 1.90
2-1-4-1	0.705	0.200	2.72 8.64 19.36	5.15 14.05 24.35	7.6 12.2 15.6	
2-1-1-2	0.193	0.298	4.12 8.94 19.54 31.25	2.58 6.99 14.99 22.49	40.3 64.4 92.5 104.0	0.18 0.78 0.99 1.54
2-1-2-2	0.295	0.304	2.19 9.06 19.80 32.04	2.94 11.71 22.48 30.91	24.6 45.9 61.8 70.4	1.09 1.59 1.97 3.08

Run	X	G _T	ΔP_x	W_G^x	W_L^x	ΔP_a
2-1-3-2	0.496	0.302	1.93	5.15	8.1	0.50
			8.18	14.05	14.9	1.40
			21.44	26.24	21.8	2.01
			30.25	32.79	23.8	2.58
2-1-4-2	0.698	0.300	3.61	4.72	2.2	0.36
			8.64	11.75	5.8	1.73
			19.18	24.43	10.6	4.75
			31.25	32.89	13.3	5.60
2-1-1-3	0.203	0.406	3.39	2.21	42.5	0.33
			8.53	6.63	71.0	0.65
			19.36	14.07	90.0	0.97
			47.61	23.46	104.0	1.42
2-1-2-3	0.299	0.401	3.84	5.17	24.7	0.65
			12.32	14.09	39.7	2.15
			21.16	21.61	49.0	3.90
			33.41	29.13	55.5	
2-1-3-3	0.495	0.408	2.99	5.17	5.4	0.90
			9.12	14.09	10.05	1.90
			13.47	23.49	14.30	3.18
			33.18	32.89	17.10	4.40
2-1-4-3	0.702	0.399	2.22	6.04	4.24	
			13.91	18.77	8.85	
			26.21	28.15	12.45	
			35.05	33.31	14.70	
2-1-1-4	0.207	0.485	3.46	2.57	35.8	0.34
			8.07	6.99	55.9	1.15
			20.43	16.86	79.2	1.98
			31.81	24.35	93.0	2.73
2-1-2-4	0.307	0.504	2.43	3.68	12.1	0.75
			10.24	14.05	24.0	1.60
			19.36	21.54	30.8	2.25
			33.18	30.91	36.0	3.95
2-1-3-4	0.492	0.499	2.28	5.15	4.1	1.48
			8.47	14.05	8.0	2.57
			19.89	24.36	11.7	3.55
			30.25	30.92	14.6	6.53

TABLE I.3

Water and Air Crossflow Rates at 50 psia

Slot Length = 5.98 in.
Slot Width = 0.012 in.

Run	X Quality	G _T Total Mass Flux lbm/hr-ft ² x 10 ⁻⁶	Δ P _x Pressure Drop Across Slot in. water	W _G ^x Air Phase Crossflow lbm/hr	W _L ^x Water Phase Crossflow lbm/hr	Δ P _a Axial Pressure Recovery in. water
3-1-1-1	0.200	0.203	3.39 13.25 21.81 36.60	3.69 14.11 21.16 28.22	46.5 96.4 115.8 129.9	0.49 0.91 1.03 1.79
3-1-2-1	0.302	0.201	3.17 9.49 21.16 31.36	2.95 12.20 23.45 31.87	32.9 59.4 83.7 92.5	0.44 0.85 1.69 2.00
3-1-3-1	0.500	0.200	3.17 9.12 21.16 32.72	5.15 14.06 28.12 38.43	16.4 26.8 38.1 44.2	0.58 1.16 2.24 2.24
3-1-4-1	0.705	0.200	3.35 8.53 18.06 29.54	5.15 14.05 25.76 37.46	8.5 13.1 17.9 21.4	0.89 1.79 2.93 3.57
3-1-1-2	0.202	0.301	3.10 7.95 18.49 32.26	2.21 7.00 16.87 27.19	50.6 80.5 112.0 138.4	0.32 0.71 1.21 1.69
3-1-2-2	0.295	0.304	3.35 7.08 19.62 32.38	2.21 6.99 20.59 31.83	29.4 48.0 74.9 86.0	1.17 1.50 1.91

Run	X	G _T	ΔP_x	w_G^x	w_L^x	ΔP_a
3-1-3-2	0.493	0.302	2.82	3.68	8.9	0.76
			9.30	14.04	16.1	1.35
			19.10	25.28	22.5	2.28
			33.76	39.32	28.5	3.44
3-1-4-2	0.698	0.299	3.10	5.15	4.4	1.08
			9.12	14.04	8.2	2.93
			20.70	28.08	12.8	4.00
			33.76	40.25	16.3	5.15
3-1-1-3	0.203	0.406	3.84	2.21	49.5	0.49
			9.12	7.00	80.4	1.07
			22.75	18.76	97.9	1.70
			35.28	28.14	137.0	2.57
3-1-2-3	0.298	0.400	3.53	4.42	28.3	0.63
			10.89	14.06	43.7	1.42
			20.25	23.44	54.7	2.08
			31.25	32.82	63.7	2.55
3-1-3-3	0.495	0.406	2.22	3.68	6.2	0.82
			8.64	14.04	11.4	2.02
			20.25	27.15	17.3	3.72
			31.36	37.44	21.6	5.12
3-1-4-3	0.697	0.397	2.66	4.42	3.0	1.40
			8.76	14.07	6.5	5.30
			19.10	26.26	9.5	7.90
			31.25	37.51	12.4	7.90
3-1-1-4	0.207	0.485	3.76	2.21	52.5	0.88
			9.55	6.99	78.4	1.90
			28.73	18.72	111.2	2.96
			40.58	28.08	128.8	3.68
3-1-2-4	0.307	0.504	3.46	3.31	16.9	2.25
			8.41	10.30	28.5	3.15
			20.70	23.41	40.6	4.35
			37.95	33.72	48.0	5.30
3-1-3-4	0.492	0.497	3.17	5.14	6.3	2.00
			7.95	14.04	10.4	2.90
			17.89	25.26	15.2	5.70
			30.80	37.43	20.1	7.80

TABLE I.4

Water and Air Crossflow Rates at 50 psia

Slot Length = 6.00 in.
Slot Width = 0.04 in.

Run	X Quality	G_T Total Mass Flux lbm/hr-ft ² x 10 ⁻⁶	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Phase Crossflow lbm/hr	W_L^x Water Phase Crossflow lbm/hr	ΔP_a Axial Pressure Recovery in. water
3-2-1-1	0.201	0.203	2.72 8.64 18.66 35.64	44.62 82.67 115.73 148.73	159.0 230.0 279.0 427.5	2.54 3.65
3-2-2-1	0.303	0.201	2.59 8.41 15.84 31.25	44.63 82.59 115.57 156.84	103.2 126.0 185.0 225.0	2.05 3.38 4.36
3-2-3-1	0.501	0.200	2.16 6.66 17.64 33.18	44.59 82.51 131.95 173.19	46.5 66.4 88.7 108.0	3.20 5.55 7.95 10.05
3-2-4-1	0.705	0.200	1.08 6.86 20.25 29.81	44.59 90.76 148.52 173.27	22.4 35.5 48.4 54.5	3.92 6.62 10.52 11.12
3-2-1-2	0.202	0.302	2.07 8.53 20.88 36.84	44.63 82.59 123.82 156.77	166.5 240.0 302.0 354.0	2.85 4.50 5.50 6.40
3-2-2-2	0.294	0.304	1.99 8.88 20.61 31.36	44.57 90.72 131.95 156.70	91.0 141.0 176.1 199.5	3.60 6.25 8.15 9.30

Run	X	G _T	ΔP_x	W_G^x	W_L^x	ΔP_a
3-2-3-2	0.494	0.302	2.37	44.53	28.0	4.70
			9.49	98.87	51.6	9.10
			22.47	148.31	63.5	13.50
			33.06	173.03	68.0	15.50
3-2-4-2	0.698	0.300	2.53	44.57	15.8	5.71
			8.88	98.97	30.1	12.89
			20.79	148.45	40.2	17.75
			30.25	173.19	46.1	20.85
3-2-1-3	0.203	0.406	2.72	44.57	160.0	3.35
			9.55	82.47	242.0	6.45
			19.71	115.43	300.0	8.66
			35.40	148.38	354.0	10.35
3-2-2-3	0.298	0.400	1.46	44.57	70.5	4.45
			7.08	82.47	107.0	7.60
			19.80	131.95	143.0	10.95
			30.64	156.70	164.0	13.00
3-2-3-3	0.495	0.407	1.93	44.53	21.2	7.10
			8.88	98.85	42.4	13.55
			19.71	140.04	58.5	18.55
			32.38	172.99	70.6	22.70
3-2-4-3	0.702	0.399	2.69	44.61	11.2	13.80
			9.61	99.06	24.6	23.00
			19.19	140.33	33.2	32.20
			31.92	173.35	40.9	40.70
3-2-1-4	0.207	0.485	2.79	44.67	166.0	5.15
			9.61	82.67	244.0	8.70
			20.16	115.73	297.0	
			31.02	140.46	338.0	
3-2-3-4	0.307	0.504	2.22	44.63	57.2	5.02
			7.51	82.59	92.0	10.57
			20.70	132.14	130.5	14.34
			36.24	165.17	153.0	18.48
3-2-3-4	0.492	0.497	2.22	44.57	19.2	8.19
			7.73	90.72	38.6	16.09
			20.43	140.20	57.5	23.19
			31.02	164.94	69.2	26.39

TABLE I.5

Water and Air Crossflow Rates at 50 psia

Slot Length = 6.00 in.
Slot Width = 0.08 in.

Run	X Quality	G_T Total Mass Flux lbm/hr-ft ² $\times 10^{-6}$	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Phase Crossflow lbm/hr	W_L^x Water Phase Crossflow lbm/hr	ΔP_a Axial Pressure Recovery in. water
L-W-X-G						
3-3-1-1	0.200	0.203	3.24 7.18	123.59 164.79	289.0 366.0	1.92 4.01
3-3-2-1	0.303	0.201	2.37 7.67	123.71 190.51	186.5 280.0	4.45 5.44
3-3-3-1	0.501	0.200	1.74 5.86	123.71 188.86	95.4 130.0	7.03 10.30
3-3-4-1	0.705	0.201	1.32 4.41	123.71 186.80	48.7 68.2	9.27 14.61
3-3-1-2	0.202	0.302	2.37 8.12	123.71 189.68	294.0 392.0	5.29 7.98
3-3-2-2	0.295	0.304	1.77 6.35	123.62 185.84	202.0 300.0	8.05 11.73
3-3-3-2	0.496	0.302	2.07 4.93	123.62 183.0	89.9 124.2	9.32 17.15
3-3-4-2	0.698	0.300	1.19 3.98	123.71 181.85	89.9 124.2	15.05 22.45
3-3-1-3	0.203	0.406	2.13 6.86	123.65 185.47	230.0 314.0	7.15 10.72
3-3-2-3	0.298	0.400	1.61 5.02	123.71 185.47	120.1 169.0	10.84 15.97

Run	X	G _T	ΔP_x	W_G^x	W_L^x	ΔP_a
3-3-3-3	0.495	0.407	1.32	123.71	66.1	18.60
			3.69	180.61	93.4	26.20
3-3-4-3	0.703	0.399	1.54	123.88	36.0	
			3.17	175.50	49.5	
3-3-1-4	0.207	0.485	1.30	123.68	252.0	11.05
			2.47	182.63	324.0	16.75
3-3-2-4	0.307	0.504	2.07	123.71	96.7	14.68
			5.02	179.38	133.0	20.98
3-3-3-4	0.492	0.498	1.77	123.71	60.5	17.90
			3.76	174.01	85.5	28.70

TABLE I.6

Single Phase Air Crossflow Rates at 50 psia

Slot Length = 1.66 in. Slot Width = 0.012 in.					
Run	W_G Mass Flow Rate	G_T Mass Flux 1bm/hr-ft ² x 10 ⁻⁶	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Crossflow 1bm/hr	ΔP_a Axial Pressure Recovery in. water
L-W-G	1bm/hr				
1-1-1	769.4	0.141	2.37 8.18 19.54 31.14	4.05 9.38 15.95 21.11	
1-1-2	1095.9	0.201	3.10 9.00 20.43 32.83	4.05 9.38 15.95 21.11	
1-1-3	1399.6	0.256	3.24 9.86 22.28 34.22	4.05 9.38 16.42 21.11	
1-1-4	1652.1	0.303	4.12 14.75 24.40 36.48	4.05 11.73 16.42 21.11	
1-1-5	2216.1	0.406	5.15 17.89 28.41 41.34	4.05 11.73 16.42 21.11	

TABLE I.7

Single Phase Air Crossflow Rates at 50 psia

Slot Length = 3.98 in.
 Slot Width = 0.012 in.

Run	W_G Mass Flow Rate	G_T Mass Flux lbm/hr-ft ² x 10 ⁻⁶	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Crossflow lbm/hr	ΔP_a Axial Pressure Recovery in. water
L-W-G	lbm/hr				
2-1-1	769.4	0.141	1.77 5.11 17.31 28.62	6.61 14.03 28.99 38.35	0.62 0.94 1.88 2.43
2-1-2	1097.0	0.201	2.22 5.57 17.14 30.91	6.61 14.03 28.06 39.28	0.85 1.25 2.65 3.50
2-1-3	1400.9	0.256	2.53 6.20 18.23 30.80	6.61 14.03 28.06 38.35	
2-1-4	1652.1	0.303	2.79 6.76 18.06 30.80	6.61 14.03 27.12 38.35	2.65 3.50 4.55 6.15
2-1-5	2191.5	0.402	3.61 8.12 20.52 34.34	6.61 14.03 27.12 38.35	2.48 3.50 6.49 8.59

TABLE I.8

Single Phase Air Crossflow Rates at 50 psia

		Slot Length	=	5.98 in.		
		Slot Width	=	0.012 in.		
Run	W_G Mass Flow Rate	G_T Mass Flux 1bm/hr-ft ² x 10 ⁻⁶	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Crossflow 1bm/hr	ΔP_a Axial Pressure Recovery in. water	
L-W-G	1bm/hr					
3-1-1	774.1	0.142	2.22 7.08 16.97 30.25	7.01 18.80 32.90 46.04	0.50 0.71 1.48 1.88	
3-1-2	1101.6	0.201	2.07 8.18 15.92 30.03	7.02 19.75 30.57 44.67	0.42 1.47 2.20 3.07	
3-1-3	1405.5	0.256	2.79 8.07 19.54 30.25	7.02 18.80 32.90 43.24	0.72 2.00 3.26 4.52	
3-1-4	1657.2	0.304	3.35 9.67 21.62 30.25	7.01 18.79 30.06 42.26	1.08 2.78 4.58 5.68	
3-1-5	2197.7	0.403	4.28 9.86 19.62 34.22	7.01 16.44 28.18 42.53	2.35 3.65 6.45 9.55	

TABLE I.9

Single Phase Air Crossflow Rates at 50 psia

Slot Length = 6.00 in.					
Slot Width = 0.040 in.					
Run	W_G Mass Flow Rate	G_T Mass Flux lbm/hr-ft ² $\times 10^{-6}$	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Crossflow lbm/hr	ΔP_a Axial Pressure Recovery in. water
L-W-G	lbm/hr				
3-2-1	767.0	0.141	1.17	44.53	2.33
			7.45	74.18	5.53
			13.10	136.0	7.12
			22.37	168.99	
			31.25	189.68	
3-2-2	1095.9	0.201	1.32	44.57	3.73
			6.05	94.05	7.47
			17.31	168.33	12.38
			30.69	214.72	15.03
3-2-3	1399.6	0.256	1.82	44.59	4.97
			6.40	100.99	10.00
			14.14	140.33	14.45
			20.61	165.10	18.00
			30.25	195.79	21.80
3-2-4	1652.1	0.304	1.90	44.63	5.80
			5.90	90.85	13.65
			15.05	144.53	19.45
			20.61	165.17	22.15
			31.14	203.55	25.80
3-2-5	2195.7	0.403	2.50	44.63	10.35
			7.95	103.23	19.85
			14.75	140.46	26.55
			24.21	173.60	32.75
			33.06	214.38	38.75

TABLE I.10

Single Phase Air Crossflow Rates at 50 psia

		Slot Length	= 6.00 in.		
		Slot Width	= 0.08 in.		
Run	W_G Mass Flow Rate	G_T Mass Flux lbm/hr-ft ² x 10 ⁻⁶	ΔP_x Pressure Drop Across Slot in. water	W_G^x Air Crossflow lbm/hr	ΔP_a Axial Pressure Recovery in. water
L-W-G	lbm/hr				
3-3-1	769.4	0.141	1.32 2.79 4.28	123.82 165.1 193.99	6.57 8.50 10.03
3-3-2	1095.9	0.201	1.19 2.46 3.84	123.9 165.1 190.7	10.0 13.77 16.44
3-3-3	1399.6	0.256	1.32 2.37 3.65	123.9 165.2 190.0	13.81 18.40 21.38
3-3-4	1652.1	0.303	1.69 2.56 3.61	123.94 165.3 186.8	17.42 22.72 26.67
3-3-5	2194.6	0.406	2.59 3.61 4.08	123.9 165.3 179.8	25.4 33.5 37.2

APPENDIX II

TABLE II.1

Two-Phase Pressure Drop Data at 50 psia

1-in. Tube Flow Section

G _T Total Mass Flux lbm/hr-ft ² x 10 ⁻⁶	X Quality	Slot Number					
		0	11	21	31	32	33
		$\frac{\Delta P}{\Delta L}_{TP}$ Exptl psi/ft	$\frac{\Delta P}{\Delta L}_{TP}$ Exptl psi/ft	$\frac{\Delta P}{\Delta L}_{TP}$ Exptl psi/ft	$\frac{\Delta P}{\Delta L}_{TP}$ Exptl psi/ft	$\frac{\Delta P}{\Delta L}_{TP}$ Exptl psi/ft	$\frac{\Delta P}{\Delta L}_{TP}$ Exptl psi/ft
.2	.2	.176	.162	.159	.157	.147	.155
.2	.3	.201	.193	.195	.222	.189	.189
.2	.5	.258	.252	.281	.262	.266	.261
.2	.7	.293	.312	.326	.318	.332	.326
.3	.2	.248	.249	.269	.263	.268	.246
.3	.3	.308	.314	.323	.317	.323	.316
.3	.5	.416	.420	.420	.400	.422	.490
.3	.7	.601	.575	.584	.589	.602	.615
.4	.2	-	-	.430	.396	.396	.405
.4	.3	-	-	.465	.420	.443	.430
.4	.5	-	-	.648	.600	.626	.666
.4	.7	-	-	1.090	1.020	1.060	1.060
.5	.2	.480	.455	.505	.505	.504	.450
.5	.3	.595	.558	.613	.625	.629	.640
.5	.5	.895	.856	.847	.848	.884	.980
.5	.7	1.68	1.87	-	-	-	-
.7	.2	.775	.806	-	-	-	-
.7	.3	.920	.990	-	-	-	-
.7	.5	1.21	-	-	-	-	-

TABLE II.2

Single-Phase Air Pressure Drop Data at 50 psia

1-in. Tube Flow Section

G_T Mass Flux lbm/hr-ft ² $\times 10^{-6}$	Slot Number					
	0	11	21	31	32	33
	$\frac{\Delta P}{\Delta L}$ g Exptl psi/ft	$\frac{\Delta P}{\Delta L}$ g Exptl psi/ft	$\frac{\Delta P}{\Delta L}$ g Exptl psi/ft	$\frac{\Delta P}{\Delta L}$ g Exptl psi/ft	$\frac{\Delta P}{\Delta L}$ g Exptl psi/ft	$\frac{\Delta P}{\Delta L}$ g Exptl psi/ft
.100	.078	-	-	-	-	-
.141	.132	.122	.123	.118	.120	.133
.201	.238	.230	.235	.224	.229	.248
.256	.392	.391	.384	.386	.387	.403
.303	.559	.564	.547	.545	.571	.576
.350	.708	-	-	-	-	-
.406	1.01	1.02	1.035	1.02	1.02	1.10
.500	1.57	-	-	-	-	-

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