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THERMAL ENERGY LABORATORY

FINAL REPORT

ANALYSIS OF A FLOW METERING DEVICE
FOR LOW-QUALITY STEAM-WATER FLOWS

PO 7659303

MECHANICAL ENGINEERING DEPARTMENT

WASHINGTON STATE UNIVERSITY

PULLMAN, WA.

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Final Report

Analysis of a Flow-Metering Device for
Low-Quality Steam-Water Flows

University of California
Contract No. 7659303

C. T. Crowe

Washington State University
Pullman, Washington 99164

submitted to

Lawrence Livermore Laboratory
Livermore, California

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Project Monitor: H. Weiss

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INTRODUCTION

Measuring the mass flow rate and quality of fluid issuing from a geothermal well is essential to determining the potential of the energy resource and selecting the most appropriate energy conversion scheme. Also, this information is needed to establish the performance efficiency of conversion machinery, both in the laboratory and in the field. Durability and reliability are essential characteristics of the metering device to be used on site. For this reason, pressure inferential devices, such as orifices, flow nozzles, and venturis have appeared most suitable.

A review of the literature shows that considerable effort has been devoted to the use of orifices to meter the flow of steam-water mixtures. In particular, Murdock [1] and Chisholm [2] have developed similar empirical relationships for the pressure drop across an orifice as a function of the Lockhart-Martinelli parameter. These empirical relationships are used with the orifice equation to yield an expression for mass-flow rate as a function of pressure drop across the orifice and the upstream quality. James [3] has proposed an orifice equation with a modified mixture density. Several other correlations can be found in the literature.

Common to every orifice equation proposed in the literature is the need to know the quality upstream of the mixture. This information is generally unavailable for fluid issuing from a geothermal well. Consequently, the metering equations in their present form are of little value.

James [3] has developed an empirical relationship for determining the flow rate by exhausting the low quality effluent through a sonic nozzle and measuring upstream enthalpy and pressure drop. He claims the empirical relationship provides reliable results. Of course, exhausting through a sonic nozzle to measure flow rate on-line would result in a sizeable penalty in pressure loss.

Collins and Gacesa [4] have investigated the possibility of using a venturi and orifice in series (orifice downstream) to measure both quality and mass flow rate. Their analysis is based on an extension of the empirical relationship developed by Murdock. Another empirical term had to be added to the expression to account for the affect of quality before an acceptable set of the data could be achieved. This scheme appeared acceptable for high quality flows.

The possibility of using two orifices, separated by a valve, for the concurrent measurement of mass flow and quality was investigated by LLL [5]. The pressure drop across the valve gave rise to slightly higher quality upstream. The data were reduced assuming an isoenthalpic flow between the two orifices. It was found that the pressure drop across the valve gave rise to different empirical constants in Murdock's correlation for the upstream and downstream orifices. There appeared to be no way in which the adjustment of the empirical constants could be correlated with the experimental conditions. Thus, the technique was not reliable. The observed dependence of empirical constants in Murdock's equation also sheds doubt on the universality of Collins and Gacesa's empirical equation for quality.

The goal of this project is to investigate the potential of the meter configuration shown in Fig. 1. It consists of a sharp-edged contraction section followed by an extended length of constant area duct and, finally, a diffuser section for pressure recovery. The principle underlying the scheme is straightforward. The sharp-edged entry section will behave similar to an orifice and the mass flow rate will be a function of the pressure drop between stations 1 and 2 and the upstream quality. After a recovery length, the steam-droplet flow should behave as a homogeneous mixture, flowing steadily in a constant area duct in which pressure gradient is dependent on the mass-flow rate and quality (mixture density). Using the two sets of data from the sudden contraction and constant area duct allows one, in principle, to solve for the quality and mass-flow rate.

Besides the configuration shown in Fig. 1 (Design A), two other configurations were tested for this study. These configurations and the reasons underlying their selection are described and discussed in the body of this report.

FUNDAMENTAL EQUATIONS RELATING TO METER OPERATION

As discussed above, the concept underlying the meter design was to use two sets of data, from the sudden contraction section and the constant area duct, to correlate the mass-flow and quality.

Murdock's correlation for mass-flow rate through an orifice (or venturi meter) as a function of pressure drop and quality is given by

DESIGN A (NOMINAL DESIGN)

Figure 1 - Geometric configuration of Design A (Nominal Design)

$$\dot{m}/A_o = K\sqrt{2\Delta P_{tp}\rho_v} / [x + \lambda(1 - x)\sqrt{\rho_l/\rho_v}] \quad (1)$$

where A_o is the orifice area, K is the flow coefficient, ΔP_{tp} is the pressure drop across an orifice, ρ_v is the vapor density, ρ_l the liquid density, and x the upstream quality. The factor λ is an empirical factor that must be determined from experiment. The method by which the factor is established is to plot data for the ratio $\sqrt{\Delta P_{tp}/\Delta P_v}$ versus the Lockhart-Martinelli parameter, $\sqrt{\Delta P_l/\Delta P_v}$. The pressure difference ΔP_v is the pressure which would exist if the vapor only of vapor-droplet mixture were flowing through the orifice. Correspondingly, ΔP_l is the pressure drop if the liquid only were to flow through the orifice. The slope of the line which best fits the data is λ . Murdock found, for the quality range and conditions of his and others' experiments, that the correlation factor should be 1.26. Subsequent studies at LLL of low quality steam-water flow showed considerable scatter of the data for $\sqrt{\Delta P_{tp}/\Delta P_v}$ versus $\sqrt{\Delta P_l/\Delta P_v}$. The better correlations were obtained with low β -ratio orifices, particularly for an orifice with a β -ratio of .57. The major difficulty with the LLL orifice-valve-orifice arrangement was the unpredictable shift in the correlating parameter from the upstream to the downstream orifice. It was assumed that the Murdock equation would also be valid for a sudden contraction section with a modified correlation parameter λ . One object, then, of the experimental program was to measure λ for a sudden contraction section.

After a calming length, the two-phase mixture should achieve equilibrium and yield a pressure gradient corresponding to a homogeneous flow in a constant area duct. Because of the high velocities in the duct, the two-phase flow should be in the mist-flow regime. The equation for pressure loss in a duct, as related to the Darcy-Weisbach friction factor, f , is

$$\Delta P/\Delta L = (f/D)(\rho_m U^2/2) \quad (2)$$

where ρ_m is the mixture density and D is the duct diameter. The equation can be re-expressed in terms of the mass-flow and duct area, A_o , by

$$(\dot{m}/A_o)^2 = (2D\rho_m/f)(\Delta P/\Delta L) \quad (3)$$

Finally, using the relationship between quality and mixture density, one has

$$(1/\rho_m) = (x/\rho_v) + (1-x)/\rho_l \quad (4)$$

so that the pressure gradient becomes

$$\Delta P/\Delta L = (f/2D) (\dot{m}/A_o)^2 [x/\rho_v + (1-x)/\rho_l] \quad (5)$$

Combining Murdock's equation and the above equation to eliminate mass-flow yields the following implicit equation for quality:

$$(\Delta P_{tp}) (\Delta P/\Delta L) = (D/K^2 f) [x + \lambda(1-x)(\rho_v/\rho_l)^{1/2}]^2 / [x + (1-x)(\rho_v/\rho_l)] \quad (6)$$

The objective of the present study is to determine λ and f from experimental data such that the above equation could be used to first determine for quality then, using Murdock's equation, calculate the mass-flow rate.

EXPERIMENTAL PROGRAM

The test program was carried out at the LLL geothermal test facility. This unique facility is designed to evaluate the performance of nozzles, turbines, and other energy conversion devices operating in a two-phase working fluid. The gas-fired hot water generator is capable of supplying 7900 lbm/hr at 5.5×10^6 BTU's. The water is heated to 550°F and pressurized sufficiently to remain in the compressed liquid regime. The total mass-flow rate of the fluid is measured by an orifice in the line conveying the compressed liquid. The two-phase fluid is obtained by flashing the compressed liquid through a throttling valve upstream of the test section. The quality is determined by measuring the temperature downstream and upstream of the throttling valve and assuming an isoenthalpic process. The single-phase fluid was flashed to a maximum test section pressure of 150 psia. This pressure was controlled by a downstream gate valve opening into a larger tube-and-shell condenser maintained at atmospheric pressure.

A detailed drawing of the nominal contraction-tube configuration is shown in Fig. 2. The upstream section is a one-inch schedule 40 pipe with an inside diameter of 1.049 inches. The downstream tube is a nominal half-inch pipe schedule 40, with an inside diameter of .622 inches. Thus, the effective β ratio of the sudden contraction is .593. The pressure taps for the sudden contraction section are located one-half inch upstream and one-half inch downstream of the contraction section. Pressure taps to measure the pressure gradient are located at

three inch intervals along the tube. A diffuser section is attached to the downstream end of the half-inch pipe. A series of valves and 1/8-inch O.D. stainless steel tubing were used to connect each tap to a 500 psi Validyne pressure transducer. The signal from the transducer was amplified and displayed on a strip chart recorder using a scale of 10 psi per inch.

The procedure was to first control the flow rate by adjusting the throttling valve and then to set the back pressure by operation of the back pressure gate valve. The operating pressure was recorded on a single point Hewlett-Packard strip chart analog recorder. The valve for each pressure tap was opened sequentially by hand and the pressure was recorded while continuously monitoring the system pressure. A summary of the data is provided in Appendix A.

A small experimental program was undertaken at Washington State University to measure the flow coefficient of the sudden contraction section. The test was performed with one of the test sections used in the LLL tests. The configuration consisted of an orifice mounted in a straight duct 20 diameters upstream of the sudden contraction. This orifice was used to measure the mass flow rate. The mass flow-rate data were correlated and the pressure drop across the orifice was correlated with the pressure drop across the sudden contraction section yielding a flow coefficient for the sudden contraction of $.867 \pm .016$ (1 σ limit) at a Reynolds number of 2×10^5 . Because flow coefficient is insensitive to Reynolds numbers in this range, this value was used in the reduction of the LLL data.

DATA REDUCTION

The data from each run were reduced using the computer program listed and documented in Appendix B. The data read into the program consisted of:

- 1) Mass-flow obtained from a chart for the calibrated orifice provided by LLL
- 2) Pressure difference across the sudden contraction section
- 3) Pressure distribution along the constant area duct. All pressures were referenced with respect to the pressure upstream of the sudden contraction section.
- 4) Temperatures upstream and downstream of the throttling valve.

The data reduction program first used the temperature data across the throttling valve to calculate the quality downstream of the valve (and upstream of the test section). This quality was then used to determine the flow rates of vapor and liquid phases. From these data and the measured pressure drop across the sudden contraction section, the Lockhart-Martinelli parameter $\sqrt{\Delta P_l / \Delta P_v}$ was then calculated. Also, the square root of the pressure drop across the orifice divided by the pressure if vapor only were going through the orifice was also calculated. These parameters constitute those necessary to determine the empirical factor λ in Murdock's equation. A least square linear fit is then performed for the pressure gradient along the constant area duct. A friction factor is calculated using this pressure gradient and the mass-flow through the duct. The calculated parameters are then printed out and a new group of data are selected for further reduction. As pointed out above, the details of the data reduction program are to be found in Appendix B.

RESULTS

The first series of tests were performed with the nominal design as formulated by the proposed plan of study. The data for the pressure ratio $\sqrt{\Delta P_{tp} / \Delta P_v}$ versus the Lockhart-Martinelli parameter as required to determine the empirical parameter λ in Murdock's equation are shown in Fig. 3. One notes that the data appears to vary in a linear fashion. However, visual extrapolation of the line to pure vapor flow (Lockhart-Martinelli parameter approaching zero) indicates the pressure ratio does not approach unity -- a condition necessary to utilize Murdock's equation. Apparently Murdock's equation is inapplicable to correlate pressure, flow and quality data for this type of configuration.

Chisholm's empirical equation suggests that the pressure ratio $\sqrt{\Delta P_{tp} / \Delta P_v}$ should vary with the Lockhart-Martinelli parameter X as

$$\Delta P_{tp} / \Delta P_v = 1 + CX + X^2 \quad (7)$$

where C is an empirically determined constant. This equation, by its nature, yields $\Delta P_{tp} / \Delta P_v \rightarrow 1$ as X approaches zero. The equation also gives the correct limit for pure liquid flow; that is, $\Delta P_{tp} / \Delta P_l$ approaches unity as X approaches infinity. Fitting Chisholm's equation with the data on Fig. 3, yields a value for C of 14.9 which is considerably larger than 2 which corresponds to equilibrium flow and values of 4 - 6

Figure 3 - Variation of pressure ratio with Lockhart-Martinelli parameter for Design A.

characteristic of orifices of venturis. The best fit of Chisholm's correlation is also shown on Fig. 3.

The Darcy-Weisbach friction factor obtained for the sudden contraction-constant area duct configuration is shown in Fig. 4. These data are plotted versus the Reynolds number corresponding to homogeneous flow in the duct. One notes a rather large scatter in the experimental results, although the values are within a reasonable range. The source of the data scatter is not known. However, it cannot be attributed entirely to experimental uncertainties. It is very likely that the Reynolds number is not the single correlating parameter and that other factors, such as the Lockhart-Martinelli parameter, and the mass-flow rate itself probably influence the value of the friction factor. The flow field is so complex, that it is very difficult to establish reasonable empirical correlations.

The inapplicability of Murdock's equation for a sudden-contraction configuration led to consideration of another design. A lip was added downstream of the sudden contraction section to make the section itself function more as an orifice. A diagram of the modified design (Design B) is shown in Fig. 5. Only three data points were obtained for this configuration and they are plotted on Fig. 6. The variation of the pressure ratio as a function of the Lockhart-Martinelli parameter tends to show the same trend as for the nominal design; that is the extrapolated line does not pass through unity for pure vapor flow. Fitting Chisholm's equation to these data, although they are few, indicates that the value for C is 6.8, which is more in line with the type of value expected for an orifice or a venturi.

The friction factor as a function of Reynolds number in the constant area duct corresponding to the design with the lip at the sudden contraction section is shown in Fig. 4. One notes inordinately low friction factors, less than .01, which is not anticipated for flow in a duct. The reason for these very low friction factors is not known. However, it is possible that the lip on the sudden contraction section may be responsible for separating the steam/water flow to the point that much of the liquid phase ends up on the duct wall. In such a case, the mass-flow rate through the duct itself in the core region would be significantly less, which possibly would reduce the pressure gradient and the Darcy-Weisbach friction factor. The observation that Murdock's equation

Figure 4 - Darcy-Weisbach friction factor for flow in duct

DESIGN B

Figure 5 - Dimensions and pressure tap locations for Design B

Figure 6 - Variation of pressure ratio with Lockhart-Martinelli parameter for Design B

DESIGN C

Figure 7 - Dimensions and pressure tap locations for Design C

is invalid for this type of configuration and the unexplainably low friction factors led to abandoning this specific design.

The third and final design (Design C) consisted simply of an orifice in a constant area duct as shown in Fig. 7. With this design, it was anticipated that the unfavorable effects of the sudden contraction at the entrance to the constant area duct would be eliminated. The data for the pressure ratio as a function of the Lockhart-Martinelli parameter for this configuration are shown in Fig. 8. Here one notes as before that an extrapolation of a linear curve to pure vapor flow does not go through a pressure ratio of unity. Fitting Chisholm's equation to these data indicates that the value of C is 4.2 which is within the range that one would expect.

The data for the friction factor as a function of Reynolds number in the duct is shown in Fig. 4. Here one notes that the friction factors are decidedly higher than those for the nominal design. The scatter in the data are comparable. One would anticipate that the friction factor would be higher at a lower Reynolds number but not of the magnitude observed on the figure.

In summary, the data from the three configurations tested indicate that Murdock's equation is inapplicable for correlating quality and mass-flow rate data for low quality flow through a sudden contraction section or solitary orifices. The primary problem is that the data do not extrapolate to the pure vapor case so the λ in Murdock's equation cannot be evaluated. The other difficulty with the experimental results is the spread in the data for the friction factor. It is very difficult to establish a viable value for the friction factor from the experimental results. Chisholm's equation, on the other hand, does provide a better correlation for the flow through the sudden contraction and the other configurations.

The observation that Chisholm's correlation can be used to fit experimental data leaves the question as to its applicability for obtaining mass-flow quality data. For the orifice-pipe combination, Design C, the pressure ratio across the orifice is correlated by

$$\Delta P_{tp} / \Delta P_v = 1 + 4.2(\Delta P_\ell / \Delta P_v)^{1/2} + (\Delta P_\ell / \Delta P_v) \quad (8)$$

The Lockhardt-Martinelli parameter can be written as

Figure 8 - Pressure ratio variation with Lockhart-Martinelli parameter for Design C

$$(\Delta P_{\ell}/\Delta P_v)^{\frac{1}{2}} = (\rho_v/\rho_{\ell})^{\frac{1}{2}}(1-x)/x \quad (9)$$

where x is the quality. Thus, one has

$$\Delta P_{tp}/\Delta P_v = 1 + 4.2(\rho_v/\rho_{\ell})^{\frac{1}{2}}(1-x)/x + (\rho_v/\rho_{\ell})[(1-x)/x]^2 \quad (10)$$

or

$$\Delta P_{tp}/\Delta P_v = Z(x) \quad (11)$$

The mass-flow rate can be related to ΔP_v by

$$\dot{m} = K_v Y_v A (2\rho_v \Delta P_v g_c)^{\frac{1}{2}} \quad (12)$$

or

$$\Delta P_v = (\dot{m}/K_v Y_v A)^2 (1/2\rho_v g_c) \quad (13)$$

where $\dot{m}$ is the mixture mass-flow rate. Substituting back into Eqn. (11) one finds

$$\Delta P_{tp} = (1/2\rho_v g_c) Z(x) \dot{m}^2 / (K_v Y_v A)^2 \quad (14)$$

Since ρ_v , K_v and Y_v as well as A are constants, one can rewrite this equation in the form

$$\Delta P_{tp} = B \dot{m}^2 x^2 Z(x) \quad (15)$$

where B is now a new constant. Thus, for a measured value of ΔP_{tp} across the orifice, the mass-flow and quality must be such that Eqn. 15 is satisfied. Consider a specific example using the following representative values:

$$\begin{aligned} K_v &= .87 & Y_v &= 1 & A &= .00211 \text{ ft}^2 \\ \rho_v &= 2.3 \text{ lbm/ft}^3 & \rho_{\ell} &= 56.3 \text{ lbm/ft}^3 \end{aligned}$$

Evaluating the constant B and substituting in Eqn. 15 yields

$$\Delta P_{tp} = 2.00 \times 10^4 Z(x) \dot{m}^2 x^2 \quad (16)$$

where $\dot{m}$ has the dimensions of lbm/sec and ΔP_{tp} has the units of lbf/ft². Rewriting Eqn. (16) by writing out the expression for $Z(x)$ one has

$$\Delta P_{tp} = 2.00 \times 10^4 \dot{m}^2 x^2 [1 + .267(1-x)/x + .0041(1-x)^2/x^2] \quad (17)$$

Assume now that a value of 1.5 psid is measured for ΔP_{tp} (216 psf). The relationship between mass-flow and quality to satisfy Eqn. 17 is plotted in Fig. 9.

Figure 9 - Example curve for determining mass flow and quality using Chisholm's correlation

The pressure gradient in the constant area duct is related to the mass-flow rate and quality by Eqn. 5. Taking a value of .04 for the Darcy-Weisbach friction factor and using the parameters presented above one finds,

$$\Delta P/\Delta L = 11.7 \times 10^3 \dot{m}^2 x [1 + .0041(1 - x)/x] \quad (18)$$

where $\dot{m}$ is given in lbm/sec and $\Delta P/\Delta L$ in lbf/ft³. Thus, the mass-flow rate and quality must vary for a given $\Delta P/\Delta L$ as given by Eqn. (18). Continuing the above example, assume that $\Delta P/\Delta L$ is .15 psi/inch (259 lbf/ft³). The mass flow-quality relationship corresponding to this measured pressure gradient is also plotted in Fig. 9. One notes that the mass-flow and quality through the orifice-pipe combination is given by the intersection of the two lines. One notes that the intersection point corresponds to .3 lbm/sec at a quality of 25%.

The primary difficulty with the method lies in the similarity of the curves corresponding to the orifice and to the constant area duct. Slight inaccuracies in the data or in the evaluation of the friction factor will cause a shift in the curves and significantly affect the results. For example, a 10% error above the true value of the measured pressure gradient will shift the quality to 32%. That is, a 10% error will change the results by 30%. Unfortunately, the data derived from such an experiment cannot be measured with accuracies sufficient to make this method viable.

A numerical study was also performed as part of the program to try to model the steam-water flow in the sudden-contraction to assess the viability of the scheme at larger scale. The numerical program functioned well at high qualities (90%) but exhibited uncontrollable instabilities at low qualities (20%). The results of this study were prepared as a paper presented at the meeting of the International Centre for Heat and Mass Transfer. The paper is included in Appendix C.

CONCLUSION

Murdock's correlation for steam/water flow through orifices and sudden contraction sections at low qualities is invalid. The data do not approach a correct limit for qualities approaching zero. Therefore, the empirical parameter in Murdock's equation cannot be evaluated. Thus, the metering scheme proposed in this study based on Murdock's correlation

is inoperative. Chisholm's correlation, on the other hand, provides a better fit for the data and does provide the correct limiting values. Using Chisholm's correlation for the orifice and the pressure drop equation for homogeneous flow in a constant area duct shows that the variation of mass flow rate and quality are very similar. The similar trends of these two curves precludes obtaining a good measurement of mass-flow rate and quality.

SUGGESTION FOR FURTHER STUDY

It would appear that the scheme proposed here would have some merit if another method could be used to measure mass-flow rate and quality within the constant area duct. One idea would be the use of a laser to project a beam across the flow (probably just downstream of the orifice) and measure the attenuation. This measurement may provide an indication of the local liquid bulk density and, possibly, the quality. This information together with the pressure drop across the orifice would lead to the mass-flow rate and quality of the flow.

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Appendix A

SUMMARY OF DATA

The data consist of mass flow, pressure and temperature measurements.

All pressures were measured with respect to the pressure tap upstream of the sudden contraction (or orifice). The pressure $P(1)$ is the first tap downstream of the contraction. The following eight pressures, $P(2)$ through $P(9)$, were measured along the duct at three-inch intervals. The final pressure, $P(10)$, was taken just downstream of the diffuser.

The temperatures $T(1)$ and $T(2)$ were measured by thermocouples just upstream and downstream of the throttling valve and used for quality determination. The third temperature, $T(3)$, was measured just upstream of the test section.

Data Summary

Nominal Design: Design A

Run No.	m lbm/sec	P(1) psid	P(2) psid	P(3) psid	P(4) psid	P(5) psid	P(6) psid	P(7) psid	P(8) psid	P(9) psid	P(10) psid	T(1) °C	T(2) °C	T(3) °C
1	.52	-11.0	-12.4	-14.4	-16.2	-16.8	-18.6	-19.4	-21.6	-21.6	-20.4	235.9	153.9	152.6
2	.74	-13.3	-14.9	-16.3	-18.3	-19.3	-21.0	-22.4	-25.3	-25.8	-24.3	238.0	166.3	165.0
3	.97	-11.8	-13.6	-14.3	-17.0	-18.4	-20.5	-22.4	-24.9	-25.7	-24.5	229.7	174.6	173.2
4	1.22	-15.0	-16.5	-14.7	-16.2	-16.9	-18.0	-19.7	-21.0	-20.7	-20.4	231.3	190.8	190.5
5	1.38	-14.8	-15.7	-15.2	-18.7	-20.3	-21.6	-22.1	-24.8	-24.4	-23.5	232.0	194.7	193.9
6	1.71	-24.7	-27.2	-27.5	-32.7	-35.9	-41.7	-42.9	-47.8	-48.1	-46.9	231.6	195.3	194.2
7	1.33	-21.7	-24.5	-24.2	-28.3	-31.2	-33.5	-35.7	-41.4	-42.5	-41.0	230.6	186.1	184.9
8	1.03	-13.5	-15.2	-15.3	-16.6	-17.5	-18.7	-19.7	-21.0	-22.3	-21.8	235.0	186.0	185.8
9	1.10	-13.8	-14.2	-13.8	-15.0	-15.8	-16.7	-16.8	-17.5	-16.9	-15.6	237.0	188.3	187.4
10	1.36	-16.5	-18.8	-19.2	-22.2	-23.3	-25.5	-27.1	-28.5	-29.3	-28.1	238.0	195.0	194.2
11	1.61	-25.4	-28.2	-30.0	-34.3	-37.8	-41.8	-44.8	-49.8	-52.8	-49.8	239.0	195.0	194.0
12	1.32	-21.0	-23.2	-24.0	-27.1	-29.7	-32.8	-35.6	-38.5	-40.3	-38.9	230.0	186.6	185.3
13	.97	-15.2	-17.2	-18.4	-20.6	-22.0	-24.5	-26.3	-	-30.2	-28.7	231.2	175.6	174.2
14	.71	-14.7	-16.3	-18.3	-19.8	-21.1	-22.7	-24.1	-	-26.9	-25.8	232.0	163.8	162.4
15	.49	- 3.5	- 4.9	- 4.2	- 5.2	- 5.8	- 6.6	- 8.0	- 8.0	- 8.9	- 9.0	230.0	180.0	179.0
16	.72	- 8.8	- 9.2	-10.0	-11.0	-11.9	-12.6	-13.5	-14.2	-14.9	-14.1	232.0	174.0	173.8

Data Summary

Design B	Run No.	$\dot{m}$ lbm/sec	P(1) psid	P(2) psid	P(3) psid	P(4) psid	P(5) psid	P(6) psid	P(7) psid	P(8) psid	P(9) psid	P(10) psid	T(1) °C	T(2) °C	T(3) °C
Design B	1	1.0	-16.0	- 3.0	- 3.0	- 2.1	- 1.7	- 2.0	- 2.3	- 2.8	- 3.3	-	231.7	169.3	167.2
	2	1.18	-19.0	- 2.5	- 2.5	- 1.5	- 1.0	- 1.3	- 2.0	- 2.3	- 3.0	-	232.2	175.8	173.7
	3	1.31	-12.0	- 3.0	- 3.6	- 3.8	- 4.1	- 4.7	-	-	-	-	231.0	188.1	186.9

Design C	1	.70	- 8.5	-11.5	-16.5	-17.0	-18.5	-20.0	-22.0	-22.0	-	-	234.9	164.7	-
	2	.84	-10.5	-13.5	-18.2	-20.5	-21.0	-25.5	-27.0	-	-	-	235.3	171.3	-
	3	.66	- 4.5	- 5.0	- 7.0	- 7.5	- 8.5	- 9.0	- 9.0	-	-	-	234.2	180.2	-
	4	.84	- 7.0	- 8.5	-11.0	-12.3	-13.5	-15.0	-15.5	-	-	-	235.6	180.1	-
	5	.85	- 7.2	- 9.0	-11.5	-12.5	-13.5	-14.5	-14.5	-	-	-	236.0	180.0	-
	6	.85	-11.0	-15.0	-21.0	-22.0	-24.0	-26.5	-29.0	-	-	-	236.5	171.0	-
	7	.68	- 4.7	- 5.3	- 7.4	- 7.4	- 8.0	- 9.0	-10.0	-	-	-	236.5	180.1	-
	8	.68	- 9.0	-12.5	-17.5	-19.2	-10.5	-22.5	-24.0	-	-	-	235.9	163.6	-
	9	.97	- 9.6	-12.0	-15.6	-16.0	-18.0	-20.0	-21.5	-	-	-	234.2	180.4	-
	10	.99	-10.2	-11.0	-15.5	-17.0	-17.5	-19.0	-21.0	-	-	-	235.1	180.8	-

Appendix B

DATA REDUCTION PROGRAM

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C****PROGRAM TO REDUCE LLL FLOW METERING DATA
      DIMENSION P(10),T(3),DELTAP(10),PSAT(3),AL(9)
      J=1
C****READ IN DATA/ DP-PIPE DIAMETER(IN), DO-ORIFICE DIA(IN),
C      DL-LENGTH BETWEEN TAPS(IN), YG-COMPRESSIBILITY FACTOR
C      AKG-VAPOR FLOW COEFFICIENT, AKL-LIQUID FLOW COEFFICIENT,
C****PFAC-FACTOR TO CONVERT TO PSI
      READ(5,200) DP,DO,DL,YG,AKG,AKL,PFAC
      5 CONTINUE
C****READ P-T DATA/ NR-RUN NUMBER, NT1-LEAST SQUARE LOWER
C      LIMIT, NT2-LEAST SQUARE UPPER LIMIT, AMDOT-MASS FLOW
C      RATE(LBM/SEC), P(I)-PRESSURE DATA FROM TAPS 1 THRU 10
C****T(I)-TEMPERATURE DATA FROM LOCATIONS 1 THRU 3
      READ(5,150)NR,NT1,NT2,AMDOT,(P(I),I=1,10),(T(I),I=1,3)
C****CALCULATE ORIFICE AREA
      AT=0.785*(DO/12.)**2
      WRITE(6,60) J
      WRITE(6,150) NR,NT1,NT2,AMDOT,(P(I),I=1,10),(T(I),I=1,3)
C****CONVERT PRESSURES TO PSI
      DO 10 I=1,10
      10 DELTAP(I)=PFAC*P(I)
C****DETERMINE SATURATION PRESSURE FROM TEMPERATURE
      DO 20 I=1,3
      PSAT(I)=2.39929+0.108172E-06*T(I)**4
      20 PSAT(I)=PSAT(I)+0.155821E-09*T(I)**5
C****DETERMINE QUALITY AT METER
      P1=PSAT(1)
      HFP1=HFP(P1)
      P2=PSAT(2)
      HFP2=HFP(P2)
      HGP2=HGP(P2)
      XM=((HFP1)-HFP2)/(HGP2-HFP2)
C****CALCULATE MURDOCH PRESSURE PARAMETER
      P3=PSAT(3)
      GMDOT=AMDOT*XM
      DPG=(GMDOT/(YG*AKG*AT))**2/(RHOG(P3)*9273.6)
      DPTP=SQRT(DELTAP(1)/DPG)
C****CALCULATE LOCKHART-MARTINELLI PARAMETER
      EMDOT=AMDOT*(1.-XM)
      DPF=(EMDOT/(AKL*AT))**2/(9273.6*RHOF(P3))
      DPGF=SQRT(DPF/DPG)
C****LEAST SQUARE FIT FOR PRESSURE GRADIENT IN DUCT
      AL(1)=0.0
      ALPP=0.0
      AL2=0.0
      ALS=0.0
      PS=0.0
      DO 50 I=2,9
      50 AL(I)=AL(I-1)+DL
      DC 100 I=NT1,NT2
      ALPP=ALPP+DELTAP(I)*AL(I)
      AL2=AL2+AL(I)**2
      ALS=ALS+AL(I)
      100 PS=PS+DELTAP(I)
      NI=NT2-NT1+1
      PA=PS/NI

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        GRA=((NI*ALPP)-(ALS*PS))/((NI*AL2)-(ALS*ALS))
C****CALCULATION OF DARCY-WEISBACH FRICTION FACTOR
        PTUBE=P3-PA
        RHOM=RHOG(PTUBE)/(XM+(1.-XM)*RHOG(PTUBE)/RHOF(PTUBE))
        FFAC=772.8*DP*RHOM*GRA/(AMDOT/AT)**2
C****PRINTOUT OF CALCULATED VALUES
        WRITE(6,160) J
        WRITE(6,210) DPTP,DPPG,AMDOT,XM,GRA,FFAC
        WRITE(6,180)
        WRITE(6,210) RHOM,PA,DPG,DPF
C****PROCEED TO NEXT DATA SET OR, IF DEPLETED, STOP
        K=NR-J
        IF(K.EQ.0) GO TO 500
        J=J+1
        GC TO 5
500 CONTINUE
        STOP
        60 FORMAT(1H,2(/),' INPUT DATA FOR RUN NO.',I3)
150 FORMAT(3I3/8F8.3/7F8.3)
160 FORMAT(1H,3(/),' CALCULATED VALUE FOR RUN NO.',I3)
        WRITE(6,170)
170 FORMAT(1H,2(/),5X,'DPTP',6X,'DPPG',5X,'AMDOT',8X,'XM'
/,7X,'GRA',6X,'FFAC',2(/),62(1H*),2(/))
180 FORMAT(1H,2(/),5X,'RHOM',8X,'PA',7X,'DPG',7X,'DPF'
/,2(/),41(1H*),2(/))
200 FORMAT(7F9.4)
210 FORMAT(5F10.4,F10.6)
        END

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Appendix C

Two-Dimensional Numerical Model for Steam-Water Flow in a Sudden Contraction

TWO DIMENSIONAL NUMERICAL MODEL FOR
STEAM-WATER FLOW IN A SUDDEN CONTRACTION

C. T. Crowe and H. N. Choi

Washington State University, Pullman, WA, USA

ABSTRACT

A computational model developed for two-dimensional dispersed two-phase flows is applied to steam-water flow in a sudden contraction. The calculational scheme utilizes the cellular approach in which each cell is regarded as a control volume and the droplets are regarded as sources of mass, momentum and energy to the conveying (steam) phase. The predictions show how droplets channel in the entry region and affect the velocity and pressure distributions along the duct.

NOMENCLATURE

A	cross-sectional area (m^2)
A_d	droplet area (m^2)
C_D	drag coefficient
c_d	specific heat of droplet (kJ/kg K)
d	droplet diameter (m)
G	mass flux (kg/s)
L	latent heat of vaporization (kJ/kg)
M^x	momentum flux in x-direction (kg m/s ²)
$\dot{m}_d$	net mass efflux rate of droplets (kg/s)
ΔM_d^x	net x-momentum efflux of droplets from cell (kg m/s ²)
m_d	mass of droplet (kg)
P	pressure (Pa)
Re	Reynolds number
S_P^x	momentum source in x-direction (kg m/s ²)
T	temperature (K)
t	time (s)
$\vec{U}$	vapor velocity (m/s)
u	x-component of vapor velocity (m/s)
$\vec{v}$	droplet velocity (m/s)
X_i	fraction of droplet mass entering at port i

- x_i space coordinate (m)
 y_i fraction of droplet mass associated with initial diameter d_i
 η_i number flow rate of droplets along trajectory i (s^{-1})
 μ coefficient of viscosity (Ns/m^2)
 ρ density (kg/m^3)
 τ characteristic time (s)

Subscripts

- d droplet
 i, J index to indentify grid node
 v vapor

INTRODUCTION

The flow of steam-water mixtures through ducts is of interest in power systems using water as the energy transfer medium such as fossil fuel and nuclear power plants. Also, the utilization of geothermal resources is dependent on the conveyance of steam-water mixtures through ducts.

The design of piping systems to carry steam-water mixtures is primarily empirical. The losses associated with bends, tees and other fittings are estimated using results from similar designs, or predicted based on a correlation parameter established by experimentation. The complexity of the phenomena constituting the flow of steam-water mixtures has precluded the development of analytic models to predict flow characteristics and head losses.

The purpose of this paper is to illustrate the application of the Particle-Source-In-Cell (PSI-Cell) model to analyze disperse steam-water (vapor-droplet) flow in a sudden contraction section. The generality of the model suggests its adaptability to other configurations and ultimately a better understanding of, and predictive capability for, steam-water flow in ducts.

BASIC CONCEPT

In order to apply the PSI-Cell model, it is first necessary to sub-divide the flow field into a series of cells, as shown in Fig. 1. Each cell is regarded as a control volume for the vapor phase. As droplets traverse a given cell in the flow field, they may be:

1. evaporating or condensing, resulting in a source (or sink) of vapor mass to the fluid in the cell,
2. accelerating or decelerating, resulting in a momentum augmentation or deficiency in the vapor in the cell in the direction of droplet motion.

Finite-difference equations for mass and momentum conservation are written for each cell, incorporating the contribution due to the droplets. The continuum flow field thus is analyzed utilizing the Eulerian approach, which is the most straightforward approach for analyzing continuum flows. The entire flow-field solution is obtained by solving the system of algebraic equations for each cell.

The droplet trajectories, size and temperature history are obtained by integrating the equations of motion for the droplets in the vapor flow field. Solving for the droplet velocity, size and temperature along droplet trajectories is done using the Lagrangian approach, which is the most straightforward approach

Fig. 1. Flow Field Grid System

for the droplet phase. Recording the mass and momentum of the droplets on crossing cell boundaries provides the droplet source terms for the vapor flow equations.

The flow of steam-water mixture is assumed adiabatic. Also, the velocities are sufficiently low that the kinetic energy can be neglected compared to the thermal energy. Hence, the flow is taken as isoenthalpic. Best-fit curves are used to relate the steam density and temperature to the local pressure along the vapor saturation line.

The complete solution for a vapor-droplet flow field is executed as follows: The calculation is begun by solving the vapor flow field assuming no droplets are present. Using this flow field, droplet trajectories together with size and temperature histories along the trajectories are calculated. The mass and momentum source terms for each cell throughout the flow field then are determined. The vapor flow field is solved again, incorporating these source terms. The new vapor flow field is used to establish new droplet trajectories and temperature histories, which constitute the effect of the vapor phase on the droplets. Calculating new source terms and incorporating them into the vapor flow field equations constitutes the effect of the droplet cloud on the vapor phase, thereby completing the cycle of mutual interaction or "two-way" coupling. After several iterations, the flow field equations are satisfied to within a predetermined value and the solution which accounts for the mutual interaction of the droplets and vapor is obtained.

VAPOR FLOW EQUATIONS

The basic equations governing the droplet-vapor flow field for the computational model described here are derived using the principles of conservation of mass and momentum. The flow field of interest is subdivided into rectangular cells, and each cell is treated as a control volume for the analysis. The technique adopted here is an extension of the TEACH program [1] developed at Imperial College of Science and Technology in London.

Refer to the cell shown in Fig. 2, which encloses a typical node (I,J). The four faces of this cell are identified as points on a compass, -- N, S, E, W. The continuity equation for the steady flow of a vapor-droplet mixture through this cell (control volume) is

$$G_N - G_S + G_E - G_W + \Delta \dot{m}_d = 0 \quad (1)$$

where G_i is the vapor mass flow rate through face "i" and $\Delta \dot{m}_d$ is the net mass efflux rate of the droplets for the cell. The vapor mass flux through the west face is given by

$$G_W = u_W A_W \frac{\rho_{I,J} + \rho_{I-1,J}}{2} \quad (2)$$

where u_W is the velocity component in the x-direction on the west face and A_W is the cross-sectional area of the cell. A linear variation in density between the (I,J) and (I-1,J) nodes is assumed. The vapor mass flow through the remaining faces is calculated in the same way.

Fig. 2. Typical Computational Cell.

One notes that the droplet mass term, $\Delta \dot{m}_d$, in Eqn. 1 can be regarded as a source (or sink) of mass to the gaseous phase, which is the basis of the PSI-Cell model.

Momentum Equations

Having located the cells for the continuity equation as enclosing the nodal points, the cells for the momentum equations must be located between nodes. Referring to Fig. 2, one notes that the u component velocity in the continuity equation is not the velocity at the nodal point, but the velocity halfway between nodes. Consequently, the control volume used to solve for u from the x-momentum equation is displaced from that used for the continuity equation and lies between grid lines I and I-1. The velocity, however, is identified by the node I,J.

The momentum equation in the x-direction for steady flow requires that

$$-M_W^x + M_E^x - M_S^x + M_N^x + \Delta M_P^x = (P_{I-1,J} - P_{I,J}) A_W + S_P^x \quad (3)$$

where M_i^x is the momentum of the vapor in the x-direction across the "i-th" face. ΔM_P^x is the net efflux of x-momentum from the cell due to the droplets, $P_{I,J}$ is the pressure at node (I,J) and S_P^x arises from the variation in effective viscosity in the flow field [2] and from body forces such as gravity.

The momentum flux is due to two mechanisms: convection and diffusion. The momentum flux in the x-direction across the "i-th" face can be written as

$$M_i^x = G_i u - \mu A_i (\partial u / \partial x_i) \quad (4)$$

where A_i is the area of face "i" and x_i is the spatial coordinate in the i-direction (normal to the area A_i). This differential expression is expressed in a finite difference form depending on the magnitude of the cell Reynolds number. If the magnitude of the cell Reynolds number is two or greater, upwind differencing is used; if not, central differencing is used.

Substituting the finite difference equations for momentum flux into Eqn. (3) yields

$$u \sum_k a_k = \sum_i a_i u_i + A_w (P_{I-1,J} - P_{I,J}) - \Delta M_p^x + S_p^x \quad (5)$$

where a_i are the resulting velocity coefficients, and the summation is performed over all faces of the cell. Note that the net efflux of particle momentum can be regarded as a body force acting on the gaseous phase.

The cell used for the y-momentum equation is located below and midway between the nodes (I,J) and (I,J-1). Equivalent finite-difference equations are derived for the y-component of velocity v as a function of the velocities v_i , the pressure gradient and the droplet momentum efflux in the y-direction. The resulting velocity component is identified by the node (I,J).

Turbulence Equations

Two more equations are used in the computer model to account for turbulence and to determine the effective viscosity. The turbulence field is described by the local intensity of turbulence and the dissipation rate. The finite-difference form of the equations is incorporated into the program to yield the kinetic energy of turbulence and the dissipation rate as related to the local mean-velocity field according to the scheme proposed by Launder and Spalding [3]. Having evaluated the turbulence intensity and dissipation rate in a cell, the effective viscosity is determined, using the Prandtl-Kolmogorov formula.

No attempt is made to include the effect of the droplets on the turbulence field, in that little is known as to the quantitative effect of particle size and concentration on the turbulence parameters.

Solution of the Vapor Flow Equations

The solution scheme is initiated by assuming a pressure and velocity field. The initial density and temperature field correspond, though the vapor saturation conditions, to the assumed pressure field. The momentum equations are then solved to yield a new velocity field. The new velocities are then used to evaluate mass fluxes and solve for a new pressure field. The density and temperature are updated to correspond to the new pressure field. The cycle is repeated by returning to the momentum equations to solve for a new velocity field. The solution is complete when the finite difference equations are satisfied to within a predetermined value. The reader is referred to the original reference [1] for more complete details on the solution scheme.

DROPLET EQUATIONS

In order to evaluate the source terms in the vapor phase flow equations due to the presence of droplets, it is necessary to establish droplet trajectories, size and temperature histories. This is accomplished by integrating the droplet equation of motion and using equations for droplet temperature and size. The velocity, pressure and temperature field of the gas is used in these calculations.

The equation of motion of a droplet is given by

$$m_d \frac{d\vec{v}}{dt} = C_D \rho_g (\vec{U} - \vec{v}) |\vec{U} - \vec{v}| \frac{A_d}{2} \quad (6)$$

where C_D is the drag coefficient and A_d is the droplet area. The other terms contributing to aerodynamic forces on the droplet -- namely, the pressure gradient, virtual mass and Basset term -- are neglected because they are of the order of the vapor/droplet material density ratio which is small. The Saffman lift and Magnus forces also are neglected because the droplets are not in a high-shear region of the gas flow.

The drag coefficient for a droplet depends primarily on the Reynolds number based on the gas-droplet relative velocity,

$$Re = \rho_g \frac{|\vec{U} - \vec{v}| d}{\mu} \quad (7)$$

where d is the droplet diameter. For a non-evaporating droplet, the drag coefficient can be represented reasonably well by [4]

$$C_D = \left(\frac{24}{Re} \right) (1 + 0.15 Re^{0.687}) \quad (8)$$

for Reynolds numbers up to 1,000. Evaporation can reduce the drag coefficient due to mass flux from the surface, but its effect is negligible for the application considered here.

Considerable economy in computing time is realized if the droplet trajectory equation is integrated once analytically. Rewriting the above equation for droplet motion, one has

$$\frac{d\vec{v}}{dt} = \left(\frac{18\mu f}{\rho_d d^2} \right) (\vec{U} - \vec{v}) \quad (9)$$

where $f = C_D Re/24$. Integrating the equation, assuming the gas velocity is constant over the time of integration, yields

$$\vec{v} = \vec{U} + (\vec{U} - \vec{v}_0) \exp\left(\frac{-\Delta t}{\tau}\right) \quad (10)$$

where $\vec{v}_0$ is the initial droplet velocity, Δt is the time interval and τ is the characteristic time defined by

$$\tau = \frac{\rho_d d^2}{18 \mu f} \quad (11)$$

After determining the new droplet velocity at time Δt , the droplet position at time Δt is determined from

$$\vec{x}_d = \vec{x}_{d,0} + (\vec{v} + \vec{v}_0) \frac{\Delta t}{2} \quad (12)$$

where $\vec{x}_{d,0}$ is the droplet position at the beginning of the time increment.

Because the steam-water mixture constitutes a single component, the temperature at the droplet surface must be the local vapor temperature. Hence, there is no (or negligible) heat transfer from the droplet surface to the vapor phase. Thus, the energy associated with a decrease in droplet temperature is absorbed by a change of phase at the vapor-droplet interface, therefore we have

$$\frac{dm_d}{dt} = \frac{m_d c_d}{L} \frac{dT_v}{dt} \quad (13)$$

or

$$m_d = m_{d_0} \exp \left(-\frac{c_d \Delta T_v}{L} \right) \quad (14)$$

where m_{d_0} is the droplet mass at the beginning of the time step and ΔT_v is the change in vapor (and droplet) temperature over the time step.

DROPLET SOURCE TERMS

In the philosophy of the PSI-Cell approach, the droplets are regarded as sources of mass and momentum to the conveying vapor phase. The source terms are incorporated into the gas flow equations, providing the influence of the droplets on the vapor velocity and temperature fields.

As shown in the previous section, once the vapor flow field has been established, the droplet mass and momentum equations can be integrated using the Lagrangian approach to yield the droplet trajectories and the droplet diameter history along each trajectory.

The entry of the droplets is represented by a finite number of entry ports. The mass of droplet size, d_i , which enters per unit time at port j is given by

$$\dot{m}_{p_j}(d_i) = \dot{m}_p X_j Y_i \quad (15)$$

where $\dot{m}_p$ is the total droplet mass inflow rate. The number flow rate of droplets of initial diameter d_i along a given trajectory is determined by

$$\dot{n}_j(d_i) = 6\dot{m}_{p_j} Y_i / m_d \quad (16)$$

If the Weber number, based on the relative velocity, exceeds six [5], breakup can occur. The droplet size is readjusted to yield a Weber number of six and the number flow rate changed accordingly.

The source term in the continuity equation, $\Delta \dot{m}_d$, is the net efflux rate of droplet mass from a computational cell (control volume). The mass source due to trajectory "j" is given by

$$(\Delta \dot{m}_d)_j = \dot{n}_j (m_{out} - m_{in}) \quad (17)$$

where m is the mass of an individual droplet traveling along trajectory "j". The net source term for a cell is obtained by summing over all trajectories which traverse that cell.

The corresponding source terms for momentum are obtained in the same fashion.

APPLICATION

The numerical model is applied to a steam-water flow into a sudden contraction as shown in Fig. 1. The flow is assumed to enter the section at 20 MPa with a quality of 90%. The upstream tube diameter is 2.5 cm and the velocity is 20 m/s. The contraction ratio is .62. The initial droplet diameter is taken as 50 micrometers.

The predicted droplet trajectories are shown in Fig. 3. Sixteen starting

locations were used to represent the incoming droplet phase. One notes that the droplets tend to channel, leaving a droplet-free region adjacent to the wall in the sudden contraction section. This region is delineated by the limiting droplet trajectory. Droplets near the periphery of the duct impinge on the upstream face, after which it is assumed they no longer influence the flow field.

Fig. 3. Droplet Trajectories Near Transition Region

Fig. 4. Droplet-Vapor Velocity Distribution Near Centerline

The predicted droplet velocity for a droplet near the centerline is shown in Fig. 4. One notes the droplets cannot maintain kinetic equilibrium with the vapor in the transition region. The droplet velocity continues to increase as the mixture proceeds downstream.

The predicted pressure coefficient distribution across the transition is shown in Fig. 5. The pressure coefficient is referenced to the upstream pressure and based on the upstream kinetic pressure. The pressure drop is larger for the two-phase mixture, as expected, because of the augmented pressure needed to balance the droplet momentum sink. Also, the trend toward pressure recovery corresponds with preliminary data currently being analyzed by the authors.

Fig. 5. Pressure Coefficient Distribution

At the present time, the solution scheme is being extended to low quality mixtures ($\sim 10\%$).

CONCLUSION

The PSI-Cell model represents a viable numerical model for two-dimensional single-component two-phase flows. The solution scheme provides a means whereby mass, momentum and energy coupling between phases can be adequately accounted for.

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