Two-Phase Critical Flow of Steam-Water Mixtures

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Critical flow of two-phase steam-water mixtures in annuli has been studied, with a cylindrical test section 0.574 in. in diameter with an axially centered rod, 0.187 or 0.375 in. O.D., as a pressure probe. Pressure taps on the wall and the center rod permitted an accurate determination of the pressure profile over the entire length of the section, and, in cases where a movable probe was used, a short distance into the downstream exhaust chamber. Results were correlated by plotting the ratio of the observed critical mass velocity to the theoretical critical mass velocity for homogeneous flow as a function of quality.

Exit pressures were found to be lower than most values previously reported. The ratio G_o/G_{TH} was found to be independent of the probe diameter, the manner of upstream steam-water mixing, and, for test sections greater than 9 in. in length, the test section length. Since probe diameter had no observable effect on G_o/G_{TH}, the correlation may be applicable to full bore pipes near ½ in. in diameter. In the range of qualities from 2 to 15% the critical G_o/G_{TH} ratio was depressed with increasing exit (throat) pressure, but at other qualities no pressure dependency was noted. The effect of changing the downstream exhaust chamber pressure was found to influence the exit pressure but to have little effect on the observed critical mass velocity. Addition of surface active agents to the steam-water mixture did not affect the mass velocity but did result in increased exit pressure.

INTRODUCTION AND THEORY

Advances in nuclear reactor technology have given new impetus to the study of two-phase critical flow. In order to predict the rate at which coolant will leave a reactor in the case of a ruptured line or the maximum rate of power removal by the coolant in a reactor for example, a knowledge of the critical flow rate is essential.

Critical mass velocity of a fluid is defined as the maximum mass velocity obtainable with the fluid in a given thermodynamic state. For a single-phase fluid the theory has been well established, and by combining a continuity relation with Bernoulli's equation the following expression may be derived:

$$G_c = \left[- g_c \left(\frac{\partial P}{\partial V} \right)_s \right]^{0.5} \qquad (1)$$

This equation may be applied to a flowing two-phase mixture if it is regarded as a homogeneous fluid, that is that the linear velocity of each of the phases is identical, thermodynamic phase equilibrium exists, and the specific volumes of vapor and liquid are additive. If thermodynamic data are available, the derivative may be evaluated by appropriate numerical methods and the theoretical critical mass velocity G_{TH} found as a unique function of mixture quality and pressure:

$$G_{TH} = \left[- g_c \left(\frac{\Delta P}{\Delta V} \right)_s \right]^{0.5} \qquad (2)$$

A large amount of experimental evidence however has shown that such a model is inadequate. The flow of saturated water through sharp-edged orifices has been found to be similar to that predicted for cold water (*5, 6, 27*), while the flow through nozzles was less than that predicted for cold water but greater than predicted for a homogeneous, equilibrium mixture (*1, 4, 8, 23, 24*) and could be explained only by assuming a rate process limited either by heat or mass transfer from a centrally located cylinder of liquid. In studying the flow of saturated steam, evidence of metastability or delayed droplet formation has been found (*18, 20, 21, 25, 28*). These results all tend to invalidate the assumption of a homogeneous mixture of steam and water in a critical flow process.

Experimental results from studies of two-phase critical flow in pipes at low vapor qualities show large deviations from homogeneous theory (*8, 9, 11, 14, 15, 17, 19, 29*) contrary to the results of early workers who studied flow in large pipes (*2, 5*). Both metastability and the existence of different vapor and liquid velocities have been used to explain the deviations encountered, but the lack of knowledge about both phenomena at the high mass velocities encountered has handicapped attempts at theoretical formulation (*11, 12, 17, 26*). The most successful theoretical model appears to be one due to Isbin and Fauske (*11*).

The lack of accurate pressure profiles extending to the actual exit plane of the test section and the generally poor agreement between different results prompted a careful study of this type of flow. Special attention was given to the behavior near the exit plane in order to achieve a better understanding of the local phenomena and their effects on the system behavior.

EXPERIMENTAL APPARATUS AND PROCEDURE

The apparatus is shown diagrammatically in Figure 1 and is fully described elsewhere (*10*). High pressure, saturated steam leaving the separator was mixed with water in the mixing section. The water, before entering the mixing section, could be heated to nearly its saturation temperature by a diverted portion of the steam. Mixing of the water and vapor streams was accomplished by discharging the water through either a 1-in. nipple or a perforated pipe cap (nozzle C) into the vapor stream, as shown in Figure 2. The resulting mixture flowed through the converging portion of the mixing section and the annular test section to the expansion chamber. Flow rates of cold water and steam were measured with orifice meters. Temperatures were taken with accurate thermometers or copper-constantan thermocouples, while high quality Bourdon tube gauges were used to measure pressures. The entire apparatus was well insulated so that it was possible to make energy balances as a check on the instrument readings.

Several test sections were used, the pertinent dimensions of which are given in Table 1. Each of these consisted of a length of tubing with an axially centered probe running its full length and out into the expansion chamber. The probes were held in place by a tripod at the downstream end and by four welded fins at the upstream end and were equipped with four pressure taps at 90 deg. intervals in a plane perpendicular to the probe axis. Probes of TS IV long and IV short could be moved axially and positioned accurately by means of a trombone arrangement which exitted through a packing gland. In this way pressure profiles extending into the expansion chamber could be obtained. Other test sections were equipped with fixed probes whose pressure taps were located in the exit (throat) plane. With the exception of TS VIII all test sections were equipped with wall pressure taps following the specifications of Allen and Hooper (3). The locations of these is given in Table 2.

TS VIII, the shortest test section, was a modification of the mixing section rather than a distinct section. The 2½ in. O.D. flange at the exit end of the mixing section was fitted with a 8.25 in. O.D. x 2.5 in. I.D. annular ring of mild steel so that it could be mated with the expansion chamber, and the last 0.531 in. of the inside of the flange reamed to give a uniform bore. The central probe was held in place by a tripod located at the downstream end and by three ⅛-in. rods spaced 120 deg. apart in a plane 9.38 in. upstream of the exit plane.

TABLE 1. DIMENSIONS OF TEST SECTIONS

Test section	I.D. of tube (in.)	O.D. of probe (in.)	Length (in.)
IV long	0.574	0.1879	35.14
IV short	0.574	0.1879	21.31
V	0.574	0.3754	21.31
VI	0.574	0.3754	8.98
VII	0.574	0.1860	8.98
VIII	0.570	0.3755	0.5313

EXPERIMENTAL RESULTS

The results obtained with the above apparatus consisted principally of pressure profiles and critical mass velocities and their dependence on several system parameters. The reliability of the pressure profiles measured with the central probe and the disturbance due to its supporting structure in the exhaust chamber were of particular concern. To determine the effect of the supporting fins at the upstream end of the probe several checks were made, among them the agreement between wall and center probe pressure measurements, and changes in pressure as measured by the wall taps when the probe was moved axially. In all cases it was found that the probe did not influence the pressure profile in the pertinent area near the exit plane. Some effect on pressures in the expansion chamber may have been caused by the tripod supporting the downstream end of the probe, but the extent was not determined and was not pertinent to the critical measurements in the test section itself. A complete analysis of experimental errors is available (*10*).

Effect of Expansion Chamber Pressure

A characteristic of single-phase critical flow in pipes is that both the mass velocity and exit plane pressure are independent of the expansion chamber pressure, as long as it is equal to or less than the critical exit pressure. In this case the critical exit pressure is well defined and is that pressure at which sonic velocity occurs at the pipe exit. With two-phase flow however the critical exit pressure is not well defined.

In this work conditions were regarded as critical only when the expansion chamber pressure was below the point at which changes in it caused changes in either the exit plane pressure or mass velocity in the test section. The pressure at the exit plane of the test section P_{th} was then taken as the critical pressure.

Two-phase critical flow defined in this manner was found to differ from single-phase critical flow in two respects. The first of these is the effect of expansion chamber pressure on the throat (exit plane) pressure. For any set of critical conditions there is a maximum allowed expansion chamber pressure (MAECP) which, if exceeded, will make the flow noncritical. This pressure, if exceeded, will affect either the throat pressure (until then critical) or the flow rate or both. For single-phase flow the MAECP, when exceeded, will affect both the flow rate and the exit pressure, and will be equal to or, in the event of a pressure recovery downstream of the throat, greater than the critical throat pressure. However for two-phase flow with qualities less than 50% it was observed that the MAECP was less than the critical pressure (see Table 3). This is probably the result of the asymptotic approach to critical flow found by Zaloudek (29) and described later in this paper.

The second observation of interest is that the mass velocity is often insensitive to large increases in the expansion chamber and throat pressure even as they reach values far exceeding the critical pressure. The insensitivity of

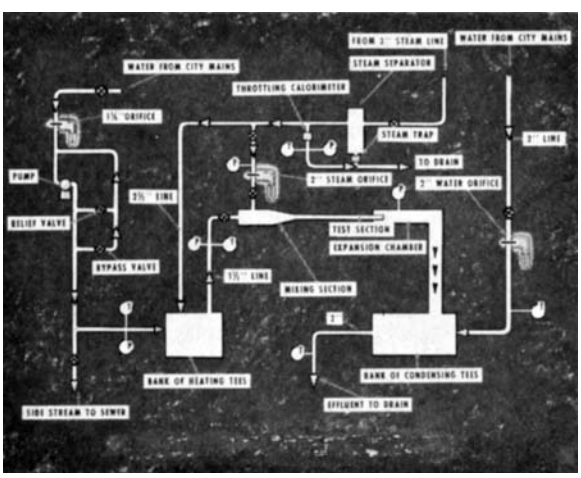

Fig. 1. Schematic diagram of equipment.

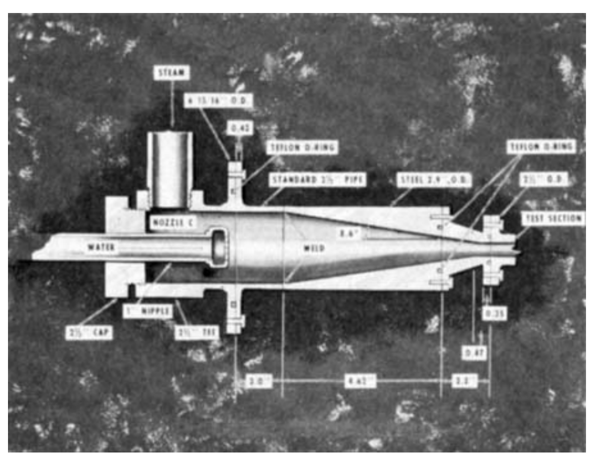

Fig. 2. Cutaway of the mixing section schematic.

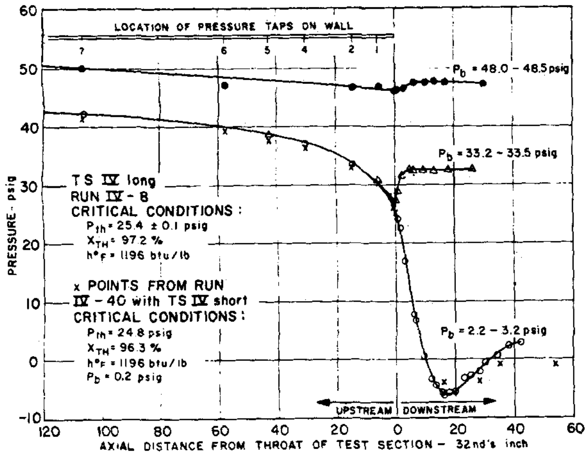

Fig. 3. Pressure profiles, 97.2 and 96.3% steam.

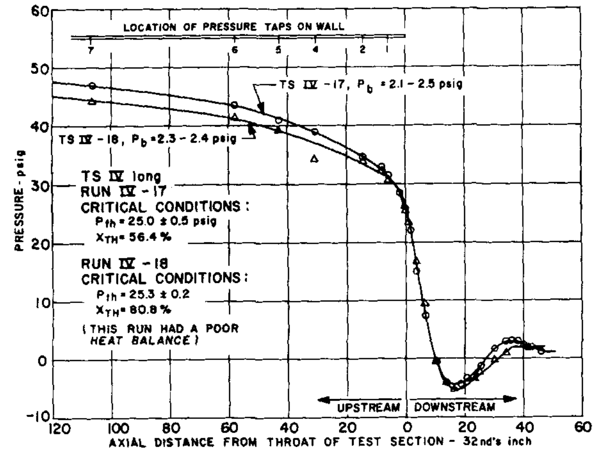

Fig. 4. Comparison of pressure profiles, 56.4 and 80.8% steam.

mass velocity to changes in expansion chamber pressure has been previously observed (*1*, *8*) but never thoroughly studied. It is important not only because of its theoretical significance, but also because design techniques based on simplified models, such as the homogeneous model, which may be in error in predicting the critical pressure, may still give reasonably accurate results when used to calculate flow rates.

Pressure Profiles

Profiles were obtained with pressure taps in the central probe and on the wall of the test section. Representative curves obtained with test sections TS IV long and TS IV short are shown in Figures 3 through 7. The pressure readings fluctuated with time about their mean values with an amplitude of 1 to 2 lb./sq.in. upstream and ½ to 1 lb./sq.in. downstream from the exit plane. A comparison of the profiles shows that both the gradient of the profile as it approaches the exit and its shape downstream from the exit were functions of the quality. Standing pressure waves, which, as Stodola (*25*) states, have the form of damped oscillations and occur only when the back pressure is less than the critical pressure, were observed but were markedly reduced in magnitude with decreasing quality. The small but reproducible pressure recovery observed at 5% quality may or may not be evidence of this phenomenon, since an interaction of the jet of fluid leaving the test section with the supporting tripod could have led to a pressure rise in the expansion chamber.

The gradient of the profile changed rapidly during the first 5/16 in. upstream from the exit, then tapered off further upstream. Since the equivalent diameter of sections TS IV long and TS IV short was 0.387 in., the 5/16-in. distance would correspond to 0.81 equivalent diameters. These results should be helpful in interpreting the validity of using an extrapolative technique to determine the outlet pressure. The gradient of the pressure profile downstream from the exit was the same for all runs at qualities equal to or greater than 56% but at lower qualities decreased with decreasing quality.

Critical Mass Velocity

The critical mass velocity was measured and the results correlated by plotting the ratio of the observed critical mass velocity G_o to the theoretical critical mass velocity for homogeneous flow G_{TH} as a function of quality. This method of correlation is similar to that used by Cruz, Moy, and Isbin (*9*, *15*, *19*), who plotted $(G_o - G_{TH})/G_o$ vs. quality. The use of G_o/G_{TH} has the advantage of not reducing the amount of scatter at low qualities nor magnifying it at high qualities. The correlations shown on Figures 8 and 9 have been constructed to indicate not only the general behavior but also differences between high and low quality regions and effects of using a very long or a very short test section, sections TS IV long or TS VIII, respectively. The magnitude of time variations in the experimental results can be inferred from those points connected with a dotted line, indicating that they were obtained during the same run with identical valve settings. In cases where many subruns were made, average values have been plotted.

TABLE 2. LOCATION OF WALL PRESSURE TAPS ON TS IV SHORT, IV LONG, V, VII, AND VI

Pressure tap	Distance from exit end (in.)
1	12/64
2	29/64
3	43/64
4	61/64
5	86/64
6	116/64
7	213/64
8	447/64
9	896/64

In the quality region from 10 to 20% flow rate varied significantly with time, and data were obtained only with difficulty. The possibility exists that the oscillations were due to upstream conditions, but it seems more likely that the cause was a change in flow pattern. At 40 lb./sq.in.abs. Hoogendorn's flow pattern correlation (*13*) indicates a transition from slug to mist-annular flow in the 5 to 10% quality range. The validity of this correlation at critical mass velocities however is questionable.

The type of steam-water mixing did not affect the critical flow, although nozzle C tended to make the mixing process less violent that it would be if a nipple were used. Similarly the use of preheated water as opposed to cold water had no effect on the critical flow correlation.

At qualities less than 2%, as shown in Table 4, G_o/G_{TH} increased rapidly with decreasing quality, and the correlation is of limited value in this region. The scatter shown in Figure 8 is due in part to the extreme sensitivity of G_o to small changes in quality.

The relation between G_o/G_{TH} and quality was independent of pressure for qualities equal to or greater than 15%. At lower qualities however the ratio was depressed with increasing pressure, as indicated on Figure 8. The exact nature of the pressure dependency was not determined, but the dashed line is an indication of its magnitude.

The critical mass velocity ratio was not affected by the substitution of a

Table 3. Effect of Exhaust Chamber Pressure on Throat Pressure

Test section	Run no.	Quality, X_{TH} (%)	Critical pressure, (lb./sq. in. abs.)	Lowest value of exhaust chamber back pressure causing a change in exit plane pressure (lb./sq. in. abs.)
IV short	57	1.1	26.5	18.2
IV long	11	1.3	40.0 ± 0.8	36.2
IV long	15	1.6	55.0 ± 1.3	39
IV long	16	1.8	75.0 ± 0.5	58
IV short	59	1.9	73.1	55-60
IV long	14	3.4	75.0 ± 1.0	46
IV short	55	3.8	27.7	18-20
IV long	13	4.4	55.5 ± 0.5	32.75
IV long	12	6.5	40.0 ± 1.0	26.5
IV short	51	25.8	27.2	18.7
IV short	54	52.6	26.2	27
IV long	17	56.4	39.7 ± 0.5	36.8
IV long	18-02	80.8	40.0	40.0

3/8-in. probe for the 3/16-in. probe in all cases where the substitution was made, that is all test sections except the shortest, TS VIII. Since the range of equivalent diameters therefore was from 0.1985 to 0.386 in., it seems likely that changes in the equivalent diameter have only a slight, if any, effect on the critical mass velocity ratio. It would seem safe to assume that the ratio would not be affected if the probe was completely removed, making the correlation valuable for critical flow in full bore pipes approximately ½ in. in diameter. It also seems probable that the critical mass velocity ratio for full bore pipes would not be a strong function of the diameter, but no evidence is available to substantiate this hypothesis.

Effect of Surface Tension

A knowledge of the effect of liquid surface tension on two phase critical flow phenomena is important since the surface tension is intimately associated with the work expended in new surface formation and the problem of metastability. Consequently a series of experiments was run with test section TS VIII in which a commercial detergent was injected into the entering hot water stream. The detergent used was "Liquid Vel."

The results of eight runs conducted both with and without detergent are shown in Table 5. With the first exploratory runs (VIII-2 through VIII-5) the detergent was injected during a very short time (about 10 sec.), and it was difficult to obtain good data. In later runs however the injection time was increased to 3 min. or more, and steady conditions were attained.

The surface tension of 77°F. water was nearly constant over the range of detergent concentration used (see column nine of Table 5). The surface tension of such detergent solutions would not be expected to decrease by more than 10% between 77° and 321°F., the highest saturation temperature (based on throat pressures) of the detergent runs (7).

A more serious complicating factor is that a finite time is required after the injection of a surface active agent for the equilibrium surface tension to become established (22). For extremely short times the surface tension may approach that of pure water. Consequently, since the rapid expansion occurring in the test section almost certainly results in the rapid destruction and formation of fluid interfaces, it would be extremely difficult to estimate accurately the magnitude of this effect on the surface tension.

The addition of detergent has little or no effect on the flow rate over the quality range studied as the small changes observed in flow rates probably resulted from changes in upstream conditions. Throat pressure however increased markedly with the addition of detergent, but the effect appeared to decrease with increasing quality. The increase in throat pressures resulted in larger calculated values of G_{TH} and therefore reduced the G_o/G_{TH} ratio. The insensitivity of G_o to changes in the exit pressure agreed with results obtained in the study of expansion chamber pressure discussed above.

The demonstrated existence of a critical pressure dependence on surface tension in steam-water mixtures may be of important theoretical significance, since it indicates that surface tension plays an important role in two-phase critical flow and implies the existence of vapor-liquid metastability.

DISCUSSION OF RESULTS

The data obtained in this investigation have resulted in values of G_o/G_{TH} considerably higher than most values previously reported. This difference can in general be attributed to the care that was taken in measuring the pressure at the exit plane of the test section. Previous investigators have usually relied on an extrapolation of the pressure profile taken along the test section to the exit plane to determine the critical exit plane pressure. Since the experimental results reported here show that the pressure gradient continues to change significantly even within ½ pipe diameter of the exit plane, extrapolation will generally give an overestimate of the critical pres-

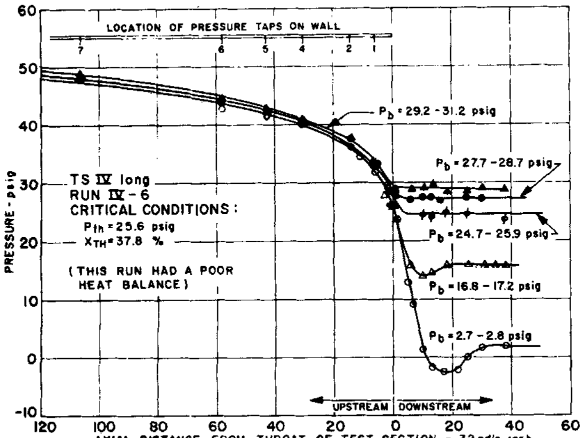

Fig. 5. Pressure profiles, 37.8% steam.

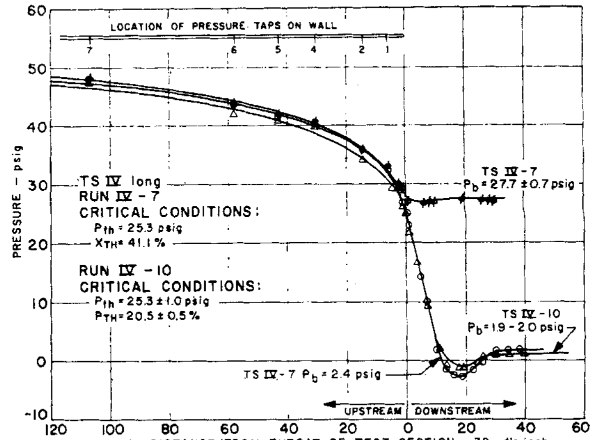

Fig. 6. Comparison of pressure profiles, 41.1 and 20.5% steam.

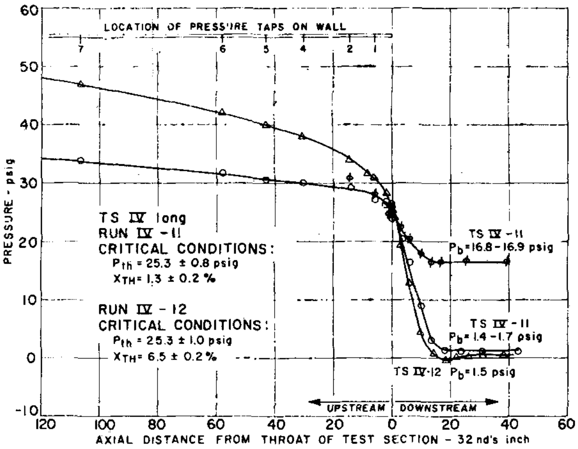

Fig. 7. Pressure profiles, 1.3 and 6.5% steam.

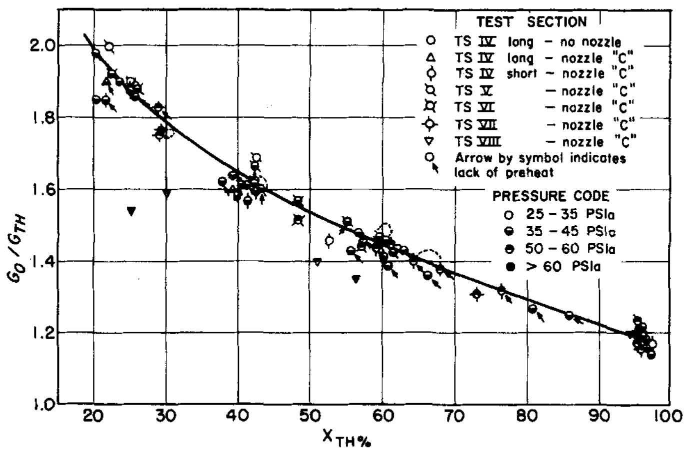

Fig. 9. Correlation of experimental results, high quality.

sure, resulting in higher values of G_{TH} and a reduced G_o/G_{TH} ratio.

Recently published data of Zaloudek (*29*) are in excellent agreement with the authors. Zaloudek used an extrapolative technique but had a pressure tap located within 0.07 diameters of the exit, which minimized any error. At qualities greater than 20% the results agree to within 2%, those of Zaloudek falling slightly below those reported here, and therefore in the direction that would be expected. In the 5 to 20% quality region however the discrepancy is greater, reaching a maximum of 16% at a steam quality of 12%. The larger disagreement may be due to differences in the upstream conditions, for example differences in mixing, although the independence of G_o/G_{TH} with test section length cited above implies that mixing is not an important variable.

By using very sensitive instrumentation Zaloudek was able to detect an asymptotic approach to critical conditions. At qualities less than 75% reductions in expansion chamber pressure continued to affect test section pressures even after a large difference between exit plane and expansion chamber pressure was established. These observations corroborate those in this study on the effects of back pressure.

Fauske (*11*) has recently reported results obtained from two-phase critical flow experiments using test sections of 0.483, 0.269, and 0.125 in. in diameter, and an extrapolation of pressures from upstream pressure taps, the last one located 1 equivalent diameter upstream from the exit plane. The only results that can be compared are those from his 0.269-in. section, which are in substantial agreement with those reported here, but with G_o/G_{TH} ratios about 5% lower. Fauske has derived a theoretical model for the prediction of critical flow which appears to be the most successful of any now in use. The agreement between it and the experimental results found in this investigation has been reported by Zaloudek (*29*).

The extensive results of Cruz, Moy, and Isbin (*15*) from two-phase critical flow studies, data of Burnell (*8*) from studies of saturated water flow through pipes and nozzles, that of Linning (*17*) based on adiabatic flow of evaporating fluids, and that of Agostinelli and Saleman (*1*) from flow of saturated water through small annular clearances have been compared in detail elsewhere (*10*) with results from the current investigation. In each case the critical pressure was far higher than that observed here and/or G_o/G_{TH} was far lower. Investigation of Benjamin and Miller (*5*) and Bottomley (*6*) into critical flow in large pipes give G_o/G_{TH} very close to, or less than, unity, which is not in agreement with these results.

Mellanby and Kerr (*18*) have studied the flow through nozzles of steam undergoing an expansion into the wet region, using search tubes to determine throat pressure. From the results they obtained one can calculate G_o/G_{TH} and X_{TH} and make a comparison. When one uses the data obtained with the shorter of two nozzles of interest (nozzle A, 0.25 in. long), the calculated G_o/G_{TH} ratios fall far below those observed here, but with the longer nozzle (nozzle B_1, 0.500 in. long) the calculated results are in reasonable agree-

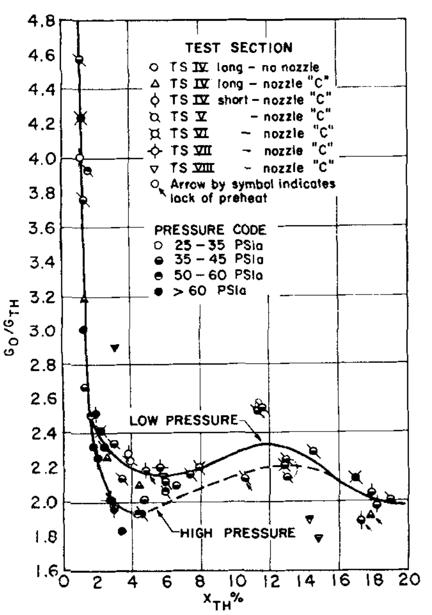

Fig. 8. Correlation of experimental results, low quality.

TABLE 4. G_o/G_{TH} VALUES AT LOW QUALITIES

Test section	Run no.	P_{TH}, lb./sq. in. abs.	$h_F°$, B.t.u./lb.	X_{TH}, %	G_o, lb.-m/sq. ft.-sec.	G_o/G_{TH}, (exper.)
TS IV short	59	73.1	294.0	1.9	1967.8	2.51
TS IV long	16	75.0 ± 0.5	294.5	1.80 ± .11	1879.0	2.32
TS IV long	15	55.0 ± 1.3	272.5 ± 1.0	1.63 ± 0.1	1555.0	2.50
TS V	21	53.8 ± 0.5	271.0	1.6	2423.6	3.93
TS IV long	3	80.0	294.5	1.4 ± 0.3	2618.0	3.00
TS IV long	11	40.0 ± 0.8	249 ± 2.0	1.3 ± 0.2	477.0	2.67 ± 0.03
TS IV long	37	40.2 ± 0.3	249.5	1.3	1535.0	3.19
TS V	7-04	40.5 ± 0.2	250.0	1.3	1824.7	3.76
TS IV short	57	26.5	223.0	1.1	1362.5	4.01
TS VI	10	78.8 ± 0.5	292.0	1.1	3676.8	4.23
TS V	7-01	38.6 ± 0.1	244.0	1.0	2177.0	4.57
TS VI	11	40.3 ± 0.5	237.3	0.9	4753.0	9.67
TS V	8-02	27.7	221.5	0.7	2110.5	5.82
TS VIII	9-01	39.4 ± 0.2	237.8	0.2	6248.0	12.29
TS VIII	9-02	44.5 ± 0.5	243.5	0.1	6214.5	11.02

ment, falling about 5% below results of the current study. Nozzle B_1 had a radius of entry of 7/32 in., a 0.2513 in. I.D. and utilized a search tube 0.1318 O.D. Some inaccuracy was unavoidable in treating their data. For example with slightly superheated steam the calculated G_o/G_{TH} value was found to be 1.02, while it should have been exactly unity. Although nozzle B_1 gave lower values of G_o/G_{TH} than TS VIII, which has a comparable L/D ratio, the discrepancy may be due either to the approximations used in reducing the data of Mellanby and Kerr or to a convergent flow pattern in their nozzle B_1.

$$X_{TH} = 100 \left\{ \frac{-h_{fg} + \left[h^2_{fg} - 4(h_f - h_F{^\circ}) \dfrac{G^2V^2_g}{2gcJ} \right]^{0.5}}{2\dfrac{G^2V^2_g}{2gcJ}} \right\} \quad (5)$$

Attempts were made to theoretically explain the high values of G_o/G_{TH} found at 96% quality. Models based on slip, increased vapor pressure of small droplets, and surface energy in the droplet did not explain the data. It is possible that the supersaturated steam forms dimers which release enough latent heat to satisfy the required energy and mass velocity relationships.

AN EQUATION FOR CALCULATING CRITICAL MASS VELOCITY

If the steam tables (*16*) are used to obtain specific volume as a function of pressure at constant entropy, Equation (2) may be approximated by

$$G_{TH} = \frac{96.2604}{(A'X' + B')^{0.5}} \quad (3)$$

A' and B' can be expressed as power functions of pressure to give an equation for G_{TH} having an error of less than 0.75% at any pressure within the range from 28 to 100 lb./sq.in.abs.:

$$G_{TH} = 96.2604\ (6.1602P_c^{-1.974}\,X_{TH} + 17.198P_c^{-1.684})^{-0.5} \quad (4)$$

X_{TH} is calculated from the following relation derived from a total energy balance written for a homogeneous fluid with equal vapor and liquid velocities undergoing adiabatic flow at equilibrium conditions, where the liquid volume has been neglected:

As already indicated, experimental values of G_o/G_{TH} were found to be independent of pressure at qualities greater than 15%. For qualities between 25 and 95% G_o/G_{TH} can be correlated with quality by the following equation:

$$\frac{G_o}{G_{TH}} = [1 - 0.566 + 4.26\ (10)^{-3}\,X_{TH}]^{-1} \quad (6)$$

Combining Equations (4) and (6) one gets the following relation which is accurate to within a few percent over the range of pressures studied (25 to 60 lb./sq.in.abs.:

$$G_o = \frac{(96.26)\ (6.16X_{TH}P_c^{-1.974} + 17.20P_c^{-1.684})^{-0.5}}{0.434 + (4.26)\ (10)^{-3}\,X_{TH}} \quad (7)$$

The error in this correlation increases rapidly outside the limits of the recommended quality range, 25 to 95%. Critical flow data at higher pressures are needed to determine if Equation (6) is valid [and therefore Equation (7)] above 60 lb./sq.in.abs.

CONCLUSIONS

Experiments with steam-water critical flow in straight annuli, with pressures obtained from wall taps and a movable, center-line probe, have shown that pressure gradients near the exit are not only very steep but also continue to change within one-half the equivalent diameter of the flow channel exit. Correlations of two-phase critical flow rates are often based on a ratio of the observed flow rate to that predicted with a homogeneous (no slip) model G_o/G_{TH}. Since G_{TH} depends strongly on the critical exit pressure, so will the ratio, and care must be taken in using existing correlations and in determining exit pressures.

Critical exit pressure with two-phase flow must be defined with care, because unlike single phase critical flow, a reduction in exit chamber pressure below the point where mass velocity is apparently constant under some conditions results in significant changes in upstream pressures. Such changes however are minor at high qualities.

The relation of G_o/G_{TH} to exit steam quality was not affected by length of test section (for lengths greater than 9 in.), probe diameter, method of mixing, or degree of liquid preheat and was independent of pressure except at qualities below 15%. The addition of surface active agents had a significant effect on the pressure profile but not on the critical flow rate.

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NOTATION

A', B' = functions of pressure
g_c = gravitational constant
G = mass velocity, lb.-m/sq.ft.-sec.
G_c = critical mass velocity for single phase fluid, lb.-m/sq.ft.-sec.
G_o = observed critical mass velocity, lb.-m/sq.ft.-sec.
G_{TH} = theoretical mass velocity based on the homogeneous model
h_f = enthalpy of saturated liquid, B.t.u./lb.-m

TABLE 5. CRITICAL FLOW WITH AND WITHOUT THE INJECTION OF DETERGENT

	Conditions without detergent				Conditions with detergent							
Run	P_{th} (lb./sq. in. abs.)	X_{TH} (lb./sq. in. abs.)	G_o (lb.-m/sq. ft.-sec.)	G_o/G_{TH}	Conc. of Liq. Vel*	P_{th} (lb./sq. in. abs.)	X_{TH} (%)	G_o (lb.-m/sq. ft.-sec.)	G_o/G_{TH}	Surface tension at 77°F (7) (dynes/ cm.)	Change in G_o/G_{TH}† (%)	Duration of detergent portion of run (sec.)
VIII-9	39.4 ± 0.2	0.2	6,248	12.29	0.16	44.3 ± 0.5	0.1	6,215**	11.02	27.2	−10.32%	200.0
VIII-10	91 ± 1.3	3.1	2,630	2.90	0.29	99.8	2.1	2,659	2.61	27.0 +	−10.3%	187.0
VIII-8	45.3 ± 1.5	5.7	967	2.15	0.096	51.3 ± 0.5	3.9	1,012**	1.90	27.5	−11.6%	1,500 ± 60
VIII-2	57.8 ± 3.0	14.3	839	1.91	1.76	62.7 ± 3.0	14.1	839	1.76	26.5	−7.85%	8.0
VIII-4	46 ± 1.3	14.7	631	1.79	2.30	49.7 ± 1.0	14.5	631	1.65	26.0	−7.83%	8.0
VIII-3	81.7 ± 1	25.3	774	1.54	2.14	no effect			1.54	26.0 +	0.0%	8.0
VIII-11	57.8 ± 2.5	30.1	539	1.59	0.88	60.7 ± 1.0	30.8	526**	1.50	27.0 +	−5.66%	300.0
VIII-5	57.3 ± 1.5	50.9	381	1.40	3.64	no effect			1.40	26.0 +	0.0%	14.5

* Expressed as lb. Liq. Vel/100 lb. of liquid at the throat.
† No significant change in G_o occurred. Change in G_o/G_{TH} mainly due to increase in P_{th}.
** Changes in G_o caused by changes in steam supply pressure.

h_{fg} = enthalpy of vaporization, B.t.u./lb.-m
$h_F{}^{\circ}$ = stagnation enthalpy of mixture entering test section, B.t.u./lb.-m
J = mechanical equivalent of heat
P = pressure, lb./sq.in.abs.
P_c = critical pressure, lb./sq.in.abs.
P_{th} = pressure at test section exit, lb./sq.in.abs.
P_b = expansion chamber pressure, lb./sq.in.abs.
V = specific volume, cu.ft./lb.-m
V_g = vapor specific volume, cu.ft./lb.-m
X = vapor quality, %
X' = weight fraction steam
X_{TH} = vapor quality at exit based on an energy balance with the homogeneous model [Equation (5)]

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A Study of Carry-Under Phenomena in Vapor Liquid Separation

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In systems where the circulating coolant is boiled, it becomes increasingly difficult to achieve effective primary separation of the phases as the systems volumetric capacity decreases and power density increases. When it is desired to effect this separation by purely natural means (gravity), the problem becomes acute. It is virtually impossible to effect a complete separation of the phases without the use of mechanical devices. A certain fraction of the vapor phase will be entrained in the downcomer with the recirculating liquid phase. The fraction of the vapor phase that is entrained in the downcomer is generally referred to as the *percent carry under*.

In boiling systems where the fluid flow is derived by means of natural convection, the carry-under problem can become especially crucial. This stems from the fact that the recirculation flow rates are a function of the density difference existing between the upcomer (riser) and downcomer segments of the system. Should substantial quantities of vapor be entrained by the circulating fluid in the downcomer, the performance of the system could suffer significantly. Forced circulation systems could also be adversely affected, since excessive carry under into the suction lines could lead to cavitation problems.

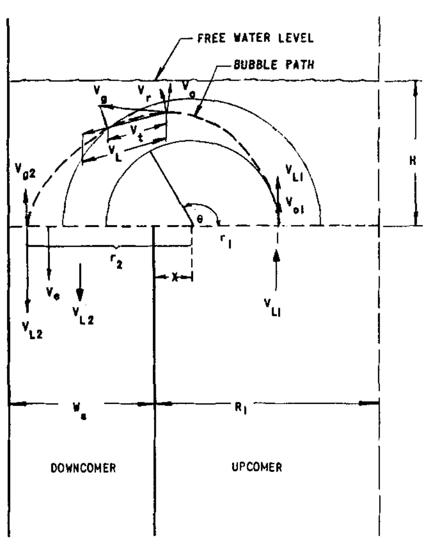

Fig. 1. Bubble trajectory in the separation plenum.

A schematic of a typical plenum where the separation process takes place is shown in Figure 1. The two-phase mixture enters the plenum through an upcomer and the recirculating coolant leaves the plenum through the downcomer. The separation of the vapor phase from the liquid phase must therefore occur by means of buoyant and hydrodynamic forces in the time interval between entrance and discharge of the coolant into the plenum. The average time interval can

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