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CRITICAL AND NEAR CRITICAL TWO PHASE FLOW

by

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SUMMARY

If a loss of coolant accident was to occur in a water cooled nuclear power reactor, critical flow of two phase fluid would occur at the break in the coolant circuit for most of the resulting blowdown. Usually special slip and friction models are used to calculate the critical flow rate and pressure profile upstream of the break. The present paper confirms the finding of Nahavandi and Von Hollen (9) that special flow models are not needed for analysis of critical flow at qualities above a few per cent. It is shown that a flow model comprising Jones' (6) slip and Beattie's (8) annular friction gives better agreement with Fauske's experiments (3) than do the usual special critical flow correlations.

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1. INTRODUCTION

Part of the safety analysis of a water cooled nuclear power reactor is the estimation of the consequences of a loss of coolant accident. This accident is assumed to be initiated by a break in one of the pressure vessels or pipes containing the pressurised water which cools the nuclear fuel. Much of the work of developing and improving methods of estimating the consequences of such accidents has been carried out in the U.S.A. However a small effort in the AAEC Physics Division has resulted in a computer code NAIAD (10) capable of handling loss of coolant accidents in simple pipe networks. This code has been used to simulate loss of coolant and channel blockage experiments carried out by Premoli (11).

One of the problems that arose in simulating these experiments was the calculation of the critical flow of water at qualities above a few per cent from a completely severed pipe. In all the methods given in the literature except one, special flow models applicable only to critical flow are used e.g. Moody (1), Fauske (3,4). These methods are reviewed by Hsu (2). In the remaining method described by Nahavandi and Von Hollen (9), a flow model that is applicable to non critical flow is used and gives reasonable agreement with the critical flow data. This paper was overlooked by Hsu (2) in his review.

The special flow model approach is the one usually used in loss of coolant analysis. In the U.S.A. at present, the Moody (1) critical flow model is specified for use in any loss of coolant analysis presented to the USAEC for reactor licensing. This model assumes that the expansion to the critical or choke conditions begins from stagnation and occurs through an isentropic nozzle, the slip ratio being chosen to give the maximum flow for fixed stagnation properties. This obviously does not correspond to the real break situation, hence the flow predicted by the Moody model is adjusted by a Moody multiplier between 0 and 1. The analysis is repeated for a range of multipliers to ensure that the actual flow rate is covered.

However, use of a special flow model for critical flow means that the point of change over from the normal to the special flow model must be determined. This problem is avoided if the Nahavandi and Von Hollen approach is used, as the normal flow model applies right up to the critical flow plane. The flow model in NAIAD is Jones (6) slip in combination with one of a number of friction models from Beattie (8). Jones obtained his slip model by generalising the Bankoff slip model with which the Armand void fraction correlation (used by Nahavandi and Von Hollen) gives fairly good agreement (9). Thus a flow model based on Jones slip model might also be expected to give reasonable agreement with critical flow data. This has been tested using the Fauske experiments (3,5), the same experiments as used by Nahavandi and Von Hollen. However we have analysed almost all the Fauske experiments rather than a third and have used all five measured pressure drops from each experiment, rather than just the overall pressure drop as did Nahavandi and Von Hollen.

2. FAUSKE'S EXPERIMENTS (3)

Measured flows of superheated steam and cold water were mixed together and directed along a horizontal uniform tube so that critical flow occurred at the tube exit. The stagnation enthalpy and flow of the two phase water entering the tube were measured. The pressure profile along the tube was obtained by measuring the pressures at six points along the tube at distances of one diameter plus 0, 152, 305, 610, 914 and 1219 mm from the exit. The pressures were measured with Bourdon pressure guages accurate to about 70 kPa. Fluctuations in pressure up to 30 kPa were reported. Thus errors of up to about 30 per cent in the pressure drop between pressure taps seem possible. Fauske reported 91 measurements with a 6.83 mm tube and 55 with a 3.17 mm diameter tube. The tubes were of cold drawn seamless steel, sanded with fine emergy cloth. Measurements were made over qualities ranging from 1 to 60 per cent, flows from 2.5 to 20 Mg m^{-2} s^{-1}, and pressures from 0.3 to 6.5 MPa. The throat pressure at the tube exit was obtained by graphical extrapolation of the pressure profile over one tube diameter from the last pressure tap. The stagnation enthalpy was not given in the report; instead the exit quality corresponding to the throat pressure assuming homogeneous flow was given. For each of the tests we have calculated the stagnation enthalpy from this exit quality, the throat pressure and the mass velocity.

3. ANALYSIS OF PRESSURE PROFILES

Henry (7) found that the absence of thermodynamic equilibrium between the phases was only important for qualities below a few per cent. As we are not concerned with this quality region we assumed thermodynamic equilibrium. We also assumed a separated flow model, i.e. that all liquid at one cross section moves with speed u_L and all vapour at speed u_V. Then the equations of conservation of energy and momentum for adiabatic flow along a horizontal uniform tube in the positive Z direction are:

$$\text{stagnation enthalpy } H_o = (1-\alpha)\,\rho_L\,u_L\left[H_L + \frac{u_L^2}{2}\right] + \alpha\rho_v\,u_v\left[H_v + \frac{u_v^2}{2}\right] = \text{constant},$$

$$\text{and} \quad \frac{dM}{dZ} + F + \frac{dP}{dZ} = 0 \quad ;$$

where $M = (1-\alpha)\,\rho_L\,u_L^2 + \alpha\rho_v\,u_v^2$,

P = pressure,
α = void fraction,
ρ = density,
H = enthalpy,
u = speed,
subscript L indicates saturated liquid,
subscript v indicates saturated vapour, and
F = frictional force per unit volume.

The three terms in the momentum equation are the expansion pressure gradient or momentum flux, the friction pressure gradient and the pressure gradient itself, respectively.

These two equations are not sufficient to determine the flow; a slip model which determines the ratio u_v to u_L and a friction model are required. Given a slip model and using the fact that H_o is constant, M may be expressed as a function of pressure only. Therefore

$$M + P = P_M(P) \quad .$$

Let P_1 and P_2 be the pressures measured at distances Z_1 and Z_2 (Z_2 greater than Z_1) from the tube inlet in one of the Fauske tests. Integrating the momentum equation we obtain the tube length ΔZ_c required to give the measured pressure drop.

$$\text{Calculated length } \Delta Z_c = -\int_{P_M(P_1)}^{P_M(P_2)} \frac{dP_M}{F} \quad .$$

In general, P_M decreases with P until the critical flow plane is reached, then rises due to the more rapid rise in the momentum flux or expansion term. At the point where

$$\left(\frac{dP_M}{dP}\right)_{H_o} = 0$$

the calculated pressure gradient becomes infinite. As is discussed later, this is the point at which critical flow is calculated to occur and the pressure at this point is the calculated throat pressure. If P_1 and P_2 are above the calculated throat pressure, ΔZ_c is positive; for P_1 and P_2 below it, ΔZ_c is negative and the calculated throat pressure is too high.

The tube lengths to give the five measured pressure drops from each Fauske experiment were calculated for each slip model friction model combination. This was done by calculating P_M and other coolant properties for pressures P_1, P_2 and four intermediate pressures. Using these properties, the friction was calculated from the friction model and the integral calculated by the trapezoidal rule. The ratio of this calculated length to the measured length was plotted against the mean of the calculated qualities for pressures P_1 and P_2. This technique gave five points from each experiment rather than one as in earlier analyses (3,9). All experiments were analysed except the nineteen with the smaller test section, which were subcooled at the inlet.

An annular flow friction model, recommended by Beattie (8), was used with zero roughness in all the calculations reported here. Other friction models were tried and, in general, gave decreased calculated lengths. A number of slip models were used:

(a) Moody (1), $\dfrac{u_v}{u_L} = \left(\dfrac{\rho_L}{\rho_v}\right)^{1/3}$,

(b) Fauske (5), $\dfrac{u_v}{u_L} = \left(\dfrac{\rho_L}{\rho_v}\right)^{1/2}$ used by Fauske in his critical flow model,

(c) Jones (7), and

(d) Homeogeneous flow, $u_v = u_L$.

The results are shown in figures 1 and 2, and in table 1. In general, the scatter in the data is very large. No error analysis was given by Fauske (5), but in his analysis (4) some of his measured lengths are adjusted by at least 30 per cent. From the table it is clear that both the Moody and Fauske models give very poor agreement with experiment. Also, as the tube exit is approached where these models are supposed to apply, the disagreement becomes greater. The Fauske slip model was formulated to minimise the pressure gradient due to the expansion term, thus the pressure drop left for friction is a maximum leading to very long calculated lengths. If Fauske's critical flow friction factor correlation (4) is used this would not occur. However this correlation is unsuitable for reasons discussed by Nahavandi and Von Hollen (9). The Moody model errs in the other direction; nearly all the measured pressure drop is required for the expansion. Both the Moody and the homogeneous slip models give some negative calculated lengths. This indicates that the throat pressure corresponding to the slip model is higher than P_2. This result is independent of the friction model as F is always positive. The Jones correlation gives considerably better agreement with the data than the other two correlations.

TABLE 1

MEAN RATIO OF CALCULATED TO MEASURED LENGTHS TO GIVE MEASURED PRESSURE DROP

	Slip Model				
	Moody	Fauske	Homogeneous	Jones	
Tube Diameter (mm)	6.83	6.83	6.83	6.83	3.17
Ratio for segments in order from inlet to exit	0.65	0.69	0.75	1.05	0.76
	0.46	0.60	0.53	0.85	0.70
	0.36	0.75	0.37	0.86	0.85
	0.24	1.04	0.06	0.78	1.17
	0.27	3.02	-1.30	0.70	1.14
Mean	0.40	1.22	0.08	0.85	0.93

An evaluation of the effect of the assumption of separated flow has been made. Let u_v and U_L now be the local vapour and liquid velocities respectively, and $\bar{u}_v$ and $\bar{u}_L$ be averages over the cross section of the tube. In the separated flow model

$$\overline{u_v^2} = \bar{u}_v^2 \quad \text{and} \quad \overline{u_L^2} = \bar{u}_L^2 \quad .$$

We now introduce M_v and M_L defined by

$$\overline{u_v^2} = M_v \, \bar{u}_v^2 \text{ and } \quad \overline{u_L^2} = M_L \, \bar{u}_L^2 .$$

The conservation equations then become

$$H_o = (1-\alpha) \, \rho_L \, \bar{u}_L \left[H_L + \left(\frac{3}{2} M_L - 1 \right) \bar{u}_L^2 \right]$$

$$+ \alpha \rho_v \, \bar{u}_v \left[H_v + \left(\frac{3}{2} M_v - 1 \right) \bar{u}_v^2 \right] = \text{constant,}$$

and
$$\frac{dM}{dZ} + F + \frac{dP}{dZ} = 0$$

with
$$M = (1-\alpha) \, \rho_L \, \overline{u_L}^2 M_L + \alpha \rho_v \, u_v^2 \, M_v \; .$$

The ratios of calculated to measured lengths using these equations, Jones' slip and Beattie's annular friction have been computed for various values of M_v and M_L. The results are shown in table 2.

FIGURE 1 - COMPARISON OF CALCULATED AND MEASURED PRESSURE PROFILES
FAUSKE TEST SECTION II DIAMETER 6.833 mm

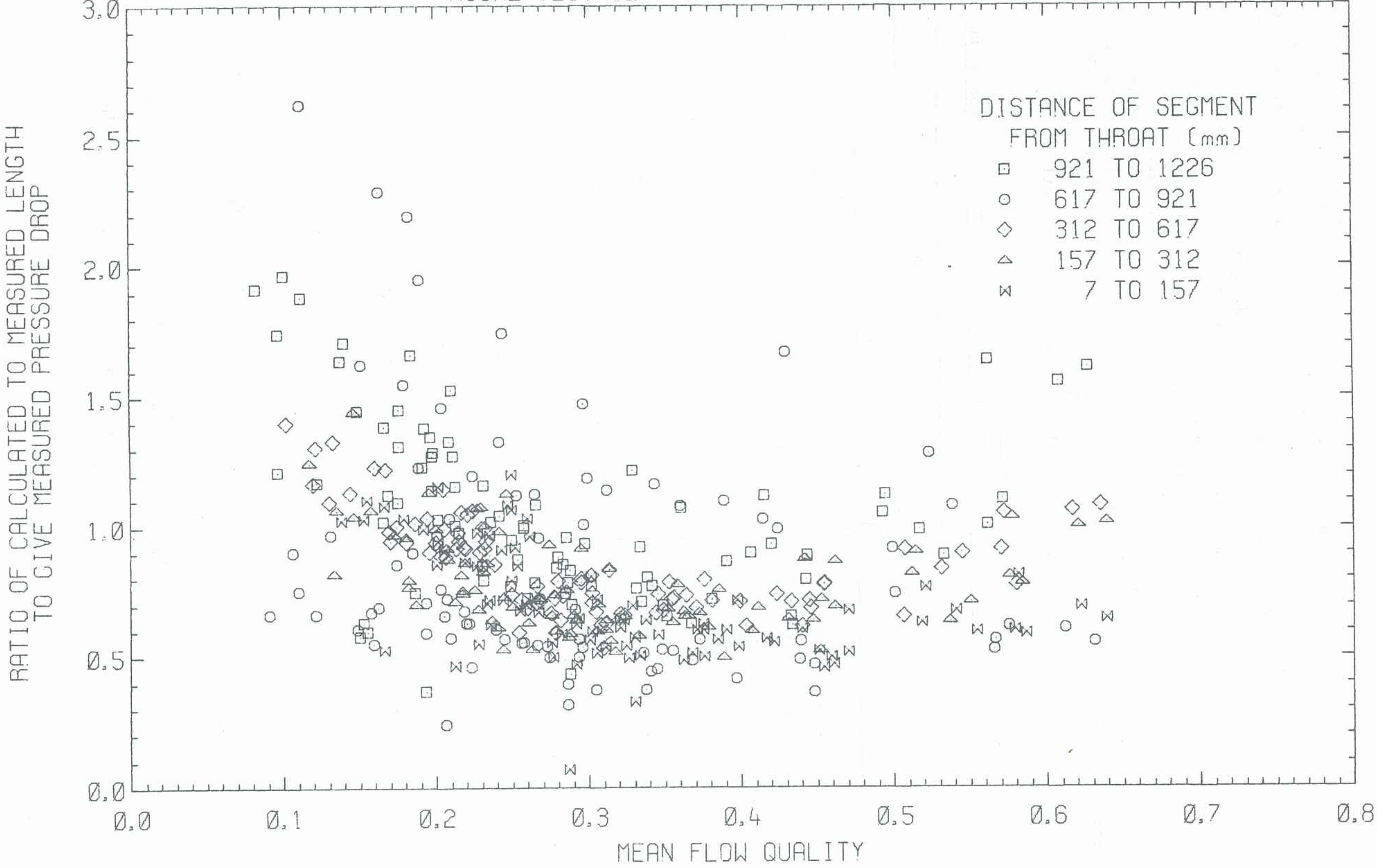


FIGURE 2 - COMPARISON OF CALCULATED AND MEASURED PRESSURE PROFILES
FAUSKE TEST SECTION IV DIAMETER 3.175 mm

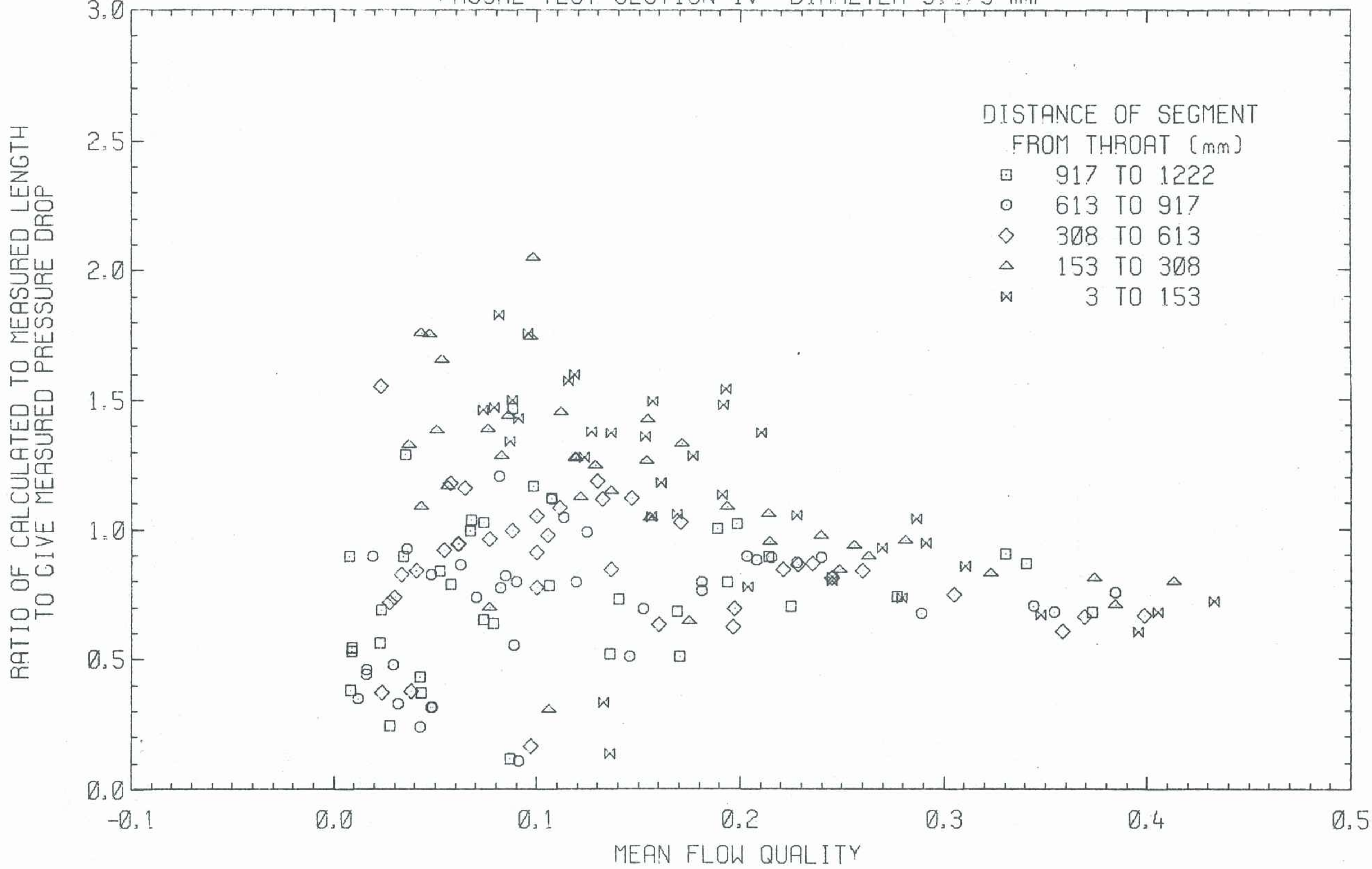


TABLE 2

MEAN* RATIO OF CALCULATED TO MEASURED LENGTHS TO GIVE MEASURED PRESSURE DROP

	M_V		
M_L	1.0	1.1	1.5
1.0	0.73	0.65	0.37
1.1	0.71	0.63	0.35
1.5	0.63	0.55	0.27
2.0	0.53	0.45	0.17

*Mean over exit segments of Fauske's experiments TSII, 2, 4, 20, 30

It is clear that the effect of relaxing the separated flow model assumption is to reduce the calculated lengths and to increase the difference between the measured and calculated lengths. This is because the effect of increasing either parameter is principally to increase the momentum flux term leaving less pressure gradient for friction. The quality is slightly changed, but not enough to have a significant effect on the momentum flux.

4. ANALYSIS OF CRITICAL FLOW

The pressure profile is determined by the conservation equations, the assumptions given above, and the slip and friction models. In single phase flow or homogeneous two phase flow, critical flow occurs when

$$\left(\frac{\partial S}{\partial H}\right)_{H_o} = 0 \quad , \qquad \ldots(1)$$

or when the entropy S of the fluid as it moves down the tube stops increasing. Also, at this point

$$\left(\frac{\partial P_M}{\partial P}\right)_{H_o} = 0 \qquad \ldots(2)$$

i.e. all the pressure gradient is required to produce the acceleration needed for the fluid expansion resulting from the pressure gradient. In general, these points do not coincide in two phase flow with slip. We have determined the critical flow from equation 2 for the stagnation enthalpy and throat pressure from Fauske's tests. This was done by numerically evaluating P_M and its derivative for various mass velocities until the mass velocity for zero slope was located. The ratio of the calculated to measured critical flow is shown in figure 3. The agreement is excellent considering the extrapolation method used by Fauske to determine the critical pressure and is as good as that for any other critical flow model. The value of $(\partial S/\partial P)_{H_o}$ at the critical plane was calculated. This was negative for all tests except two, where it was very small, thus showing that entropy increased up to the critical plane.

5. CONCLUSION

It has been shown that a flow model comprising the following:

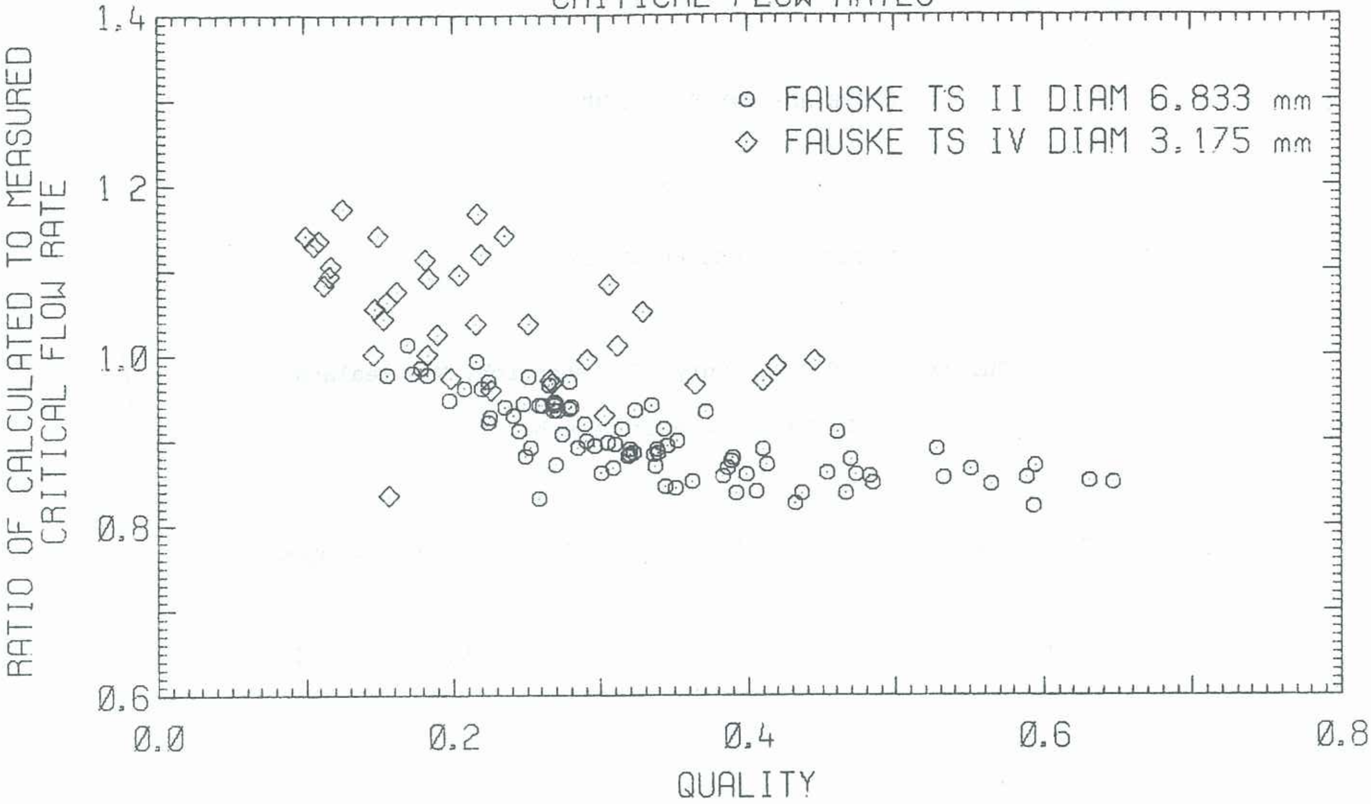
(a) thermodynamic equilibrium,

(b) Jones' slip, and

(c) Beattie's annular friction,

gives pressure profiles and critical flow rates which are not in conflict with the data from Fauske's critical flow measurements. The critical flow rates predicted by the model are in reasonable agreement with Fauske's experiments. The Moody and Fauske slip models which were developed for critical flow, give pressure profiles very different from the measured ones, the difference increasing as the critical flow plane is approached. This confirms the finding of Nahavandi and Von Hollen (9) that special flow models are not needed for analysis of critical flow at qualities above a few per cent.

FIGURE 3 - COMPARISON OF CALCULATED AND MEASURED CRITICAL FLOW RATES



6. ACKNOWLEDGEMENT

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