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SUBCOOLED BOILING OF DOWNWARD FLOW IN A VERTICAL ANNULUS

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Summary
Measurements of steady state pressure drop were made for the downward flow of water in a vertical annulus at low pressure. The inner wall was uniformly heated to allow subcooled boiling. The minimum in the test section demand curve was compared to the onset of significant void (OSV) at the exit of the annulus. As expected, OSV was found to serve as a conservative predictor for avoiding flow excursions. However, at low flow rates the demand curve minima were significantly different from those obtained for upward flow. Similarly, conditions for OSV in downward flow were not predicted by the correlation of Saha and Zuber. More data are needed to establish a general correlation for OSV in downward flow.

Introduction
Many types of flow instability may occur in heated liquid-vapor systems. Boure et al. (1973) have classified these into static and dynamic instabilities. The latter require dynamic analysis of the flow for their description; the former may be predicted from the steady state flow characteristics alone. An example from the static category is the Ledinegg flow excursion, discussed by Bailey (1977) in terms of the supply and demand curves for flow through a heated channel. A demand curve is characteristic of the channel itself, showing the pressure drop required to produce a given flow rate. A supply curve is characteristic of the remainder of the system which supplies flow to the heated channel. This typically includes pumps, flow resistances, and often parallel channels. The intersection of supply and demand curves gives the steady state operating conditions for the heated channel.

As the flow through a heated channel is decreased, the demand curve reaches a relative minimum in pressure drop due to the presence of vapor in the liquid. Should the available driving pressure difference fall below this minimum, a flow excursion will occur. Although the prediction of the demand curve requires analysis of the entire channel, the occurrence of the minimum might be related to some observable feature of flow. A candidate would be the rapid rise in void fraction variously known as "net vapor generation", "fully developed boiling", or the "onset of significant void" (OSV).

It is unlikely that the minimum in the demand curve should occur at conditions which preclude OSV. A correlation for OSV would thus be expected to provide a conservative basis for avoiding flow excursions. In this way a local effect, OSV, could be used to indicate the behavior of the entire channel.

Previous Work
Whittle and Forgan (1967) measured the pressure drop for varying flow in several heated test sections. Rectangular channels and a tube were used, over a range of heat fluxes and inlet temperatures. For every test, the minimum of the demand curve occurred at subcooled outlet temperature. The authors characterized the minima by a ratio between the observed bulk temperature rise in the channel and the difference in exit saturation and inlet bulk temperatures. This was found to be constant for each test section, and correlated with the channel length-to-diameter

ratio.

Whittle and Forgan modified one of their test sections to include a window at the outlet. They observed that the steam void fraction increased rapidly near conditions of the minimum of the demand curve. However, this question was not explored over a wide range of conditions, and no quantitative data were presented.

Evangelisti and Lupoli (1969) used a gamma densitometer to obtain the void fraction, averaged over the channel cross-section, for heated flow in an annulus. They presented plots of void fraction as a function of subcooling in the channel. The rapid rise in void fraction characteristic of OSV generally occurred at about 1% void. Void volumes at saturation were observed to be between 5 and 7%.

Saha and Zuber (1974) used data from Evangelisti and others to construct a correlation for OSV. The data represented several geometries and fluids, and a wide range of operating conditions. The correlation was derived by plotting a Stanton number, defined in terms of the fluid subcooling, versus the Peclet number, defined in terms of the wetted equivalent diameter of the channel.

$$St = \frac{\Phi}{G\, C_p\, (T_{sat} - T_{bulk})} \tag{1}$$

$$Pe = \frac{G\, D_e C_p}{k} \tag{2}$$

An analogy between heat and momentum transfer was invoked to suggest that vapor bubbles detach from the heated wall at St = 0.0065. Their subsequent behavior depends on the prevailing Peclet number. For Pe > 70,000, the subcooling in the flow is low, and the bubbles migrate away from the wall into the bulk stream, increasing the void fraction. This was termed hydrodynamically-controlled OSV. Pe < 70,000, however, implies that the subcooling is high enough that the detached bubbles flow only in the heated boundary layer. OSV does not occur until further downstream, where the subcooling has decreased to allow the bubbles to enter the bulk fluid. This domain of low Peclet number was called thermally-controlled.

Further data for low pressure have recently been obtained by Chan (1984), and by Rogers et al. (1987). Chan raised the power supplied to a constant flow rate in a horizontal tube until visual observation of the outlet indicated a sudden increase in void fraction. The measured outlet bulk temperature for this condition was expressed as a Stanton number for comparison with the Saha-Zuber correlation. Chan found that his Stanton numbers fell consistently about 20% below the correlation over the Peclet number range in the thermally-controlled regime. This is within the bounds originally presented with the correlation. He pointed out that the OSV data of Evangelisti and Lupoli (1969), taken at atmospheric pressure and low Peclet number, were also lower than the Saha-Zuber correlation.

Rogers et al. used a gamma densitometer to determine the void fraction in a vertical annulus for upward flow of water at low velocities. In contrast to Chan's results, their data fell consistently above the Saha-Zuber correlation. They noticed as well a velocity dependence in the thermally-controlled region; this is not predicted by the correlation.

Most available OSV data have been obtained in upward or horizontal flow. Bartolini et al. (1983) obtained boiling heat transfer data for both upward and downward flow in an annulus. At the highest velocities studied, photographs indicated that the accumulation of void was not affected by flow direction. However, downward flow at low velocities exhibited a larger void fraction than was observed in upward flow. In spite of this, the flow direction made no difference in the measured heat transfer coefficient.

From Whittle and Forgan it would appear that the minimum of the demand curve is generally related to the occurrence of OSV in a heated channel. The widely accepted correlation of Saha and Zuber, altered by a safety factor to account for data which fall under it, might be used in a conservative prediction of OSV, and hence the minimum. The recent data of Chan, and Rogers et al., support such an approach. However, the Bartolini et al. data suggest that the correlation might not be valid for downward flow at low velocities. OSV data are not presently available for this case.

Equipment

Tests were conducted in a low pressure flow loop which could be arranged for either upward or downward flow. The equipment is shown schematically in Fig.1. Ball valves were used to control flow direction, and a diaphragm valve was used for throttling. The pressure in the loop was set by a liquid overflow in a standpipe connected to the pump suction piping. The test section was a heated annulus between a 5 cm pipe cross overhead and a 5 cm pipe tee underneath. The annulus was surrounded by a 15 cm diameter acrylic tube, split axially into two pieces and secured about the flanges of the terminal fittings. This served as both transparent insulation and safety shield.

The heater tube was a 0.89 mm wall stainless steel tube, 1.8 m long. Fourteen measurements with a micrometer gave an average outside diameter of 25.50 mm, with a maximum deviation of 0.025 mm. A 0.66 m length of 25.4 mm OD copper rod was brazed to each end as a bus bar. Each bus bar extended 25 mm into the steel tube for support and good electrical contact, leaving a heated length of 1.78 m. The interior of the tube was not vented; no evidence of leaks was noted during the course of the experiments. The bus bars extended through O-rings in Teflon collars which were secured in the terminal fittings. The bottom bus bar rested on a Teflon pad, leaving the top end free to grow with increases in temperature.

The heater tube was surrounded by a polycarbonate tube with 3 mm wall. Four measurements of the inside diameter averaged 38.07 mm; the cross-section was slightly oval, with the major diameter exceeding the minor by 0.30 mm, or 0.8%. The tube was flanged to the bottom tee, and free to grow within the upper cross, sealed by an O-ring in a fitting attached to the cross. Pressure taps were made in the outer tube.

Four holes, spaced at 90 degrees, were drilled at 30.5 cm intervals and tapped for number 8 screws. These were denoted 1 through 8, beginning at the bottom. After both outer tube and heater were installed and secured between the terminal fittings, screws were placed in contact with the heater at stations 1, 3, 5, and 8. These were adjusted until measurements with a depth micrometer through the remaining holes indicated that the gap was uniform within 0.25 mm. Subsequent heating revealed no bowing apparent to the eye, and additional cold measurements showed no deviation. The remainder of the holes were closed with screws arranged not to protrude into the annulus.

The test section was heated with an SCR-controlled DC rectifier, capable of 5000 A at 30 V. A calibrated shunt, 0.0102 mohm, was used to determine the current through the test section. The flow in the loop was measured with a fluidic flowmeter. For each day of experimentation, a calibrated frequency source was used to verify the conversion circuit. The static pressure was measured at the end of the heated section, the lower tap for downward flow, and the upper for upward flow, by a calibrated 0-70 kPa pressure gauge. Differential pressures were measured by strain gauge pressure transducers. These were calibrated daily by connecting them individually to a pair of water-filled reservoirs, one of which was pressurized with air, the other being open to the atmosphere. The air pressure was measured with a calibrated gauge, 0-70 kPa.

The temperature of the liquid was measured in the branch of both the upper cross and bottom tee with 3.2 mm diameter sheathed ungrounded type J thermocouples. These were calibrated in isothermal flow by a mercury-in-glass thermometer which was inserted into the branch of the bottom tee. The thermometer had been calibrated at the immersion depth used in the loop.

Data were acquired using a Hewlett-Packard HP2250 Measurement and Control Processor directed by an HP9836 microcomputer. The data acquisition software was written in HP BASIC 2.1. For each reading during experimentation, the group of channels was scanned 101 times at 20 ms intervals. From the 101 readings of each channel, the mean and the standard deviation were recorded as data. These were stored in mV units, along with the calibration constants for each channel.

Procedure

Each day of experimentation began with circulating the water in upward flow at an outlet temperature of 100 to 104°C, venting the top of the test section intermittently. At the conclusion of calibrations, a standard test was run to indicate any deterioration of the loop or test section. The valves were set for downward flow, the pump started, and the heat exchanger water turned on. The flow was set at 0.95 l/s. When the flowmeter reading was judged to have stabilized, generally after ten to fifteen seconds, all instruments were read. The flow was then reduced in steps, reading at each step after stability was observed, to 0.06 l/s. Then the flow was again increased in steps. To conclude the test, some further readings were taken at intermediate flows.

Tests were performed at fixed heat fluxes from 0 to 275 kW/m^2 and inlet temperatures from 25 to 90°C. Unheated tests were run by heating the fluid to the desired temperature, turning the power off, and varying the flow as described above for the standard test. The test was interrupted as needed for heating to keep the temperature within a 5°C band.

Heated tests were run by maintaining the inlet temperature within a 3°C range and the power within ±2 % while decreasing the flow in steps. The flow was reduced until bubbly flow gave way to a churning flow regime. From this point, the flow was increased in steps to retrace the demand curve, but in less detail than was done in descending flow. There was no appreciable hysteresis. In the course of testing, it was decided to include several tests for upward flow in the test section.

Differential pressure transducer readings were corrected for zero offset and the difference in densities between the test section fluid and that in the pressure line.

$$\Delta P_r = \Delta P_m + (\rho_l - \rho_m)g(z_i - z_o) \tag{3}$$

As is further detailed the Appendix, the reported pressure differences are due to friction alone in single phase flow; for two-phase flow they represent the sum of frictional, accelerational, and a portion of gravitational contributions to the total pressure drop. The power expended in the test section was computed from the heater and shunt voltage measurements, and the heat balance was also calculated from measured flow and temperatures.

Consistency of Data

The following uncertainty variances were assigned to measured data. These were based on repeated measurements of a quantity, or else the stated precision of calibration standards.

test section voltage	± 0.2 V
shunt voltage	± 0.1 mV
shunt resistance	± 0.1%
annulus diameters	± 0.25 mm
heated length	± 2.5 cm
flow rate	± 0.01l/s
temperatures	± 0.5°C
tap separations	± 6 mm
physical properties	± 0.1%
test section pressure drop	± 0.34 kPa
test section pressure	± 0.21 kPa

Based on these, the conventional sum-of-squares uncertainty in calculated results was computed. These were

power	± 2.5% (includes estimate of ripple in rectifier)
heat flux	± 3.0%
heat balance	± 8%
Peclet number	varies between ± 3.5% at high flow and ±20% at low
Stanton number	± 15%

The single phase data are presented in dimensionless form in Fig. 2, where it is seen that they are quite convergent. Also shown are equations for laminar flow in an annulus,

$$f = 24/Re \tag{4}$$

and a common Blasius form for turbulent flow in a smooth-walled annulus, given by Kays and Perkins (1973). This has been modified by the addition of a constant form loss.

$$f = 0.085\ Re^{-0.25} + 0.00089 \tag{5}$$

The form loss accounts for the presence of centering screws in the annulus. A set of four screws may be treated as an obstruction reducing the flow area to 84% of its normal value. For turbulent flow, the irrecoverable contraction and expansion loss coefficient is about 0.18, from Babcock and Wilcox (1972). This may be converted to a friction factor, for two sets of screws, of 0.00089.

The heater power was computed from electrical measurements as described earlier. This was compared with the change in enthalpy of the flowing fluid, computed from the flow rate and measured temperatures. The latter quantity ranged from 87 to 105% of the stated heat duty. The uncertainty in the energy balance computed from temperature differences is about 8%. For the unheated test at 90°C, the difference in inlet and outlet temperatures was about 0.3°C at high flow rates, increasing to 0.7°C at low flow. The average heat loss from the test section for this condition was thus about 1 kW, or 10 % of the power at low heat flux tests. This is consistent with the energy balance mentioned above.

Results

Fig. 3 presents a demand curve for an experiment at 205 kW/m^2 and 80°C inlet temperature. Vapor behavior at the exit of the test section was recorded on videotape, and from this the points of ONB and OSV were identified as described below. Note that OSV occurs at a higher flow rate than does the minimum pressure drop. The minimum was associated with a change in flow regime in which the bubbly flow coalesced at the exit to form large bubbles filling about three-quarters of the annular circumference. These tended to oppose the downward flow, sometimes rising up the tube, sometimes being forced down out of the test section.

Also noted in the figure is a pressure drop measured after heating had ceased. It lies on the extrapolation from higher flow rates. The hump in the demand curve between 0.44 and 0.63 l/s is thought to be due to a stable air bubble which was observed to reside below a centering screw during a portion of the test. This is not a normal feature of demand curves.

The onset of nucleate boiling (ONB) was identified as the first appearance of bubbles. Notice is taken of the study of Jiji and Clark cited by Collier (1981), which indicated that the improvement in heat transfer associated with nucleate boiling is measured before bubbles are visible in pressurized water. However, the present tests were performed at low pressure, which offers the best chance for bubbles to be visually observed at their onset.

Fig. 3 indicates that bubbles were observed well before the demand curve reached a minimum. This was typical of all experiments, and indicates that the onset of boiling does not render the heated channel susceptible to flow excursion. The conditions at which bubbles were first noticed were recorded for many of the experiments. For these conditions, the wall temperature at the exit of the test section was estimated by interpolating a heat transfer coefficient from data presented by Kays and Perkins (1973) on heat transfer in annuli. These computed wall temperatures are compared with the ONB correlation of Bergles and Rohsenow (1964) in Fig. 4.

Most of the data fall higher than the correlation, indicating that ONB is predicted to occur at higher flow rates than were actually observed. This is to be expected, because the degree of scrutiny and adjustments of flow rate were not generally directed at observing the onset of nucleation. The videotaped experiments are a different matter, because the steps in flow rate were small, and the bubbles are more apparent in the videotape than they were in direct viewing. For these data, conditions are closer to the correlation, and in two cases actually show the appearance of bubbles before the wall is calculated to reach the saturation temperature. This is most likely due to the presence of gas bubbles entrained in the liquid flow. Video images were taken at 60 Hz; this is insufficient to discriminate between growing and flowing bubbles.

Visual detection of the onset of significant void (OSV) has been employed by Hino and Ueda (1985) using criteria of size, density, and departure. In the present study, OSV was taken to be the appearance of bubbles whose diameter was about the size of the annular gap width, 6.4 mm. As it happened, this transition was easy to detect, because these were preceded at higher flow rates by bubbles of only about one-fourth that diameter. The conditions at OSV were expressed as Stanton and Peclet numbers by Eqns. (1) and (2). The bulk temperature at the exit was assumed to be equal to that read downstream of the heated section. The data are plotted in Fig. 5 with the correlation of Saha and Zuber (1974).

All data lie in the thermally-controlled region of the Peclet number. They indicate that OSV occurs at lower bulk temperature for a given heat flux than is observed in upward flow. This might be attributed to the effects of buoyancy. The thermal boundary layer is expected to be thicker when the flow direction opposes the direction of gravity; this would allow bubbles to grow larger than in a thin heated layer of liquid. In addition, larger bubbles would tend to resist the downward flow, growing in place.

It is noted that the computed Stanton numbers are in good accord with Saha and Zuber's OSV criterion for the hydrodynamically-controlled regime. Saha and Zuber did suggest that bubble detachment occurs at St = 0.0065; it might thus be concluded that OSV occurs in downward flow upon bubble detachment. However, the data do not support this hypothesis; detached, flowing bubbles were routinely observed well before the point of OSV. More data are needed to generalize the correlation for downward flow.

In Fig. 6 are presented demand curves for downward flow at 70°C inlet temperature and various values of heat flux. As expected, the flow at the minimum pressure drop increases with increasing heat flux. Also, the right hand portion of each demand curve coincides with the curve for the unheated experiment. A difference might be expected due to differences in viscosity. However, the temperature differences between inlet and outlet, and between wall and bulk fluid, are relatively low at the heat fluxes used in these experiments. It is noted that the density difference correction of Eqn. (3) does not account for the bubbles in the flow; thus the reported pressure drop is slightly higher than that actually due to friction. Were the small void fraction measured and used, the heated demand curves would fall below that for isothermal flow. This correction is estimated in the Appendix, and found to be about 0.17 kPa.

For each set of data, the outlet conditions at the minimum of the test section demand curve were expressed as Stanton and Peclet numbers by Eqns. (1) and (2). These are collected in Fig. 7 for comparison to information on OSV. The downward flow data are distinguished from those for upward flow. Also plotted are the OSV data of Fig. 5 and the correlation of Saha and Zuber (1974). For downward flow, the demand curve minima occur at higher Stanton numbers than does OSV at the exit of the annulus. This implies that a lower flow rate than that producing OSV is necessary to reach the minimum pressure drop. This was illustrated in the particular case of Fig. 3.

At low Peclet numbers the occurrence of OSV at the exit seems to be sufficient to bring about the minimum for the test section. Wallis (1969) presents criteria for the rise velocity of large bubbles in vertical channels. Using these, the rise velocity in the present test section was computed to be 0.23 m/s; a liquid velocity of this value corresponds to a Peclet number of about 17,000. At Peclet numbers in this low range, the formation of large bubbles would be sufficient to rise against the liquid and increase the pressure drop. As Pe increases, however, a greater portion of the annulus can exhibit significant void before the flow regime changes and the pressure drop requirement begins to increase. At these conditions, the large bubbles would not intrude against the flow, but would form and be flushed out the exit.

Not all demand curves showed an unambiguous minimum. Fig. 8 presents the demand curve for a single experiment at 72.6 kW/m^2 and 81°C inlet temperature. In this experiment, the pressure drop was first observed to exceed that of the unheated demand curve at a flow rate of 0.12 l/s. However, this was not the case with all measurements; not until the flow rate was reduced to 0.09 l/s was the pressure drop always higher. Such behavior was noted in several experiments at low heat flux. To assign the minimum in such experiments, the point of first rise was taken; this is appropriate for considerations of flow instability.

Assigning the minimum at the last rise (about 0.095 l/s in Fig. 8, for example), results in Stanton numbers higher by a factor of 2 to 3. These data are also included in Fig. 7. The result is a region of higher Stanton numbers at the lowest Peclet numbers. This behavior seems to be due to the evolution of noncondensable gas. As mentioned above, the system was saturated with air at 100 to 104°C; this is the temperature at which the first rise was typically observed. Visual observations in the tests at low heat flux often indicated that the first large bubbles in the annulus were noncondensable. Further data are necessary to determine the influence of gas concentration on the demand curve in the low Peclet number regime.

Fig. 7 presents data for both downward and upward flow. Below a Peclet number of about 40,000, the correlation of Saha and Zuber does not serve as a conservative indicator of flow excursions in downward flow. By contrast, the few data for upward flow minima are consistent with the correlation, falling above the predicted OSV. At higher Peclet numbers, the correlation does fall below the downward flow demand curve minima. This leads to the plausible expectation that minima for upward and downward flow would not be so different in this region. Whittle and Forgan (1967), who obtained the minima of demand curves, presented tables of heat flux and temperature data which can be converted to Stanton and Peclet numbers.

$$St = \frac{A_f (T_o - T_i)}{A_h (T_{sat} - T_o)} \tag{6}$$

$$Pe = \frac{4 A_h \Phi}{P_w k (T_o - T_i)} \tag{7}$$

They also included plots of demand curves in their paper. The Peclet number computed from the tabular data exceeds that derived from the curve for each of the eight demand curves presented, if the curves in the plots are presumed to be correctly labeled. This can be accounted for if the heat loss is presumed to range between 12 and 23%. The Stanton numbers computed from temperature data by Eqn. (6) are appropriate, because they reflect the heat flux actually delivered to the fluid; the Peclet numbers from Eqn. (7) are too high by some unknown amount.

These data are plotted in Fig. 9 with the results of the present study. The data of Whittle and Forgan, which represent minima in the demand curves for rectangular channels and a tube, are in the same Stanton number range as the data obtained in an annulus. Whittle and Forgan data for downward flow are within the uncertainty of the present data. These data are also not distinct from their data for upward flow. This indicates that the influence of flow direction is probably not important above a Peclet number of 40,000. All data but one lie above the Saha-Zuber hydrodynamically-controlled criterion for OSV. These older data thus reinforce the conclusion that OSV precedes the occurrence of the minimum in demand curves. It is noted, however, that two Whittle and Forgan upward flow observations fall below Saha-Zuber in the thermally-controlled regime. The reason for this is unknown.

The amount of scatter in the various data may indicate that the test section pressure drop is not best represented by consideration of the exit conditions alone. However, the presentation of demand curve minima in terms of the exit Stanton and Peclet numbers has been useful for comparison to information on OSV at the exit of the test section.

Conclusions

This work was performed to obtain data on the minima of demand curves for the downward flow of water in a uniformly heated annulus. These studies were augmented with data for upward flow in the same test section. Conclusions are given as follows.

(1) The onset of significant void (OSV) in downward flow, determined through visual means, occurs at a Stanton number of 0.0065 over a Peclet number domain of 18,000 to 50,000. This is in contrast to the higher Stanton numbers predicted by the Saha-Zuber correlation in the thermally-controlled regime. More OSV data are needed to make a general correlation for downward flow, but St=0.0065 is suggested for provisional use at all Peclet numbers.

(2) Nucleate boiling, prior to OSV, has little effect on the test section pressure drop. In all tests, the bubbles were observed to be detached from the heated surface well before OSV.

(3) The minimum in the test section demand curve occurs at lower flow rates than does OSV at the exit of the test section, for given heat flux and inlet temperature. Thus a correlation for OSV may be used as a conservative basis for avoiding flow excursions in parallel flow systems. For upward flow, the OSV correlation of Saha and Zuber is recommended. For downward flow, St=0.0065 should be used.

(4) Below Pe=40,000, downward flow demand curves reach the minimum at lower exit Stanton numbers than those for the more commonly studied upward flow. At greater Peclet numbers, the effect of flow direction appears to be unimportant.

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Nomenclature

A_f	cross-sectional area for flow
A_h	heated surface
C_p	liquid heat capacity
D_e	equivalent diameter based on wetted perimeter
f	Fanning friction factor
g	acceleration due to gravity
G	mass velocity
k	liquid thermal conductivity
P_i	static pressure at inlet of test section
P_o	static pressure at outlet of test section
P_w	wetted perimeter of cross-section
ΔP_a	pressure drop due to friction
ΔP_f	pressure drop due to acceleration
ΔP_m	transducer reading
ΔP_r	reported pressure drop defined in Eqn. (3)

Pe	Peclet number defined in Eqn. (2)
Re	Reynolds number based on equivalent diameter
St	Stanton number defined in Eqn. (1)
T_{bulk}	liquid bulk temperature
T_i	temperature at inlet to test section
T_o	temperature at outlet of test section
T_{sat}	saturation temperature
z_i	elevation of inlet of test section
z_o	elevation of outlet of test section
α	void fraction over the volume of the test section
ρ_l	density of liquid phase
ρ_m	density of water in measuring lines
ρ_v	density of vapor phase
Φ	heat flux into fluid

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Appendix

A differential pressure transducer is connected to taps at the inlet (z_i) and outlet (z_o) of a vertical flow channel. The reading of the transducer is due to the difference between inlet and outlet static pressures, plus the contribution of the fluid in the measuring lines.

$$\Delta P_m = P_i - P_o + \rho_m g(z_i - z_o)$$

The difference in static pressures is commonly expressed as the sum of frictional, accelerational, and gravitational contributions. The friction term is positive, the acceleration term positive for heated flow, and the gravitation term negative for downward flow. The gravitational term may be expressed in terms of the average density of the fluid in the flow channel. Making these substitutions, the pressure transducer reading is expressed as

$$\Delta P_m = \Delta P_f + \Delta P_a + (\alpha\rho_v + (1-\alpha)\rho_l)g(z_o - z_i) + \rho_m g(z_i - z_o)$$

where the average density is approximated in terms of the volume-averaged void fraction and the average liquid and vapor densities. Bringing the measuring line and liquid density terms to the left-hand-side, there results the grouping presented in Eqn.(3) as the reported pressure drop.

$$\Delta P_m + (\rho_l - \rho_m)g(z_i - z_o) = \Delta P_f + \Delta P_a + \alpha(\rho_l - \rho_v)g(z_i - z_o)$$

The reported pressure drop comprises the frictional and accelerational contributions, plus a portion of the gravitational pressure drop. For a static isothermal system, the right-hand-side vanishes, and the test section density equals that of the measuring lines, with the result that the transducer reading is zero, as is observed. For warm isothermal single phase flow the reported pressure drop is less than that read on the transducer, and represents the actual frictional loss in the flow channel. The presented demand curves for unheated flow are thus for frictional pressure drop alone.

For subcooled boiling flow, the reported pressure drop exceeds the actual frictional loss by the sum of the acceleration term and the density group. Had these latter terms been quantified by measurements of the average void fraction, demand curves for frictional loss would have fallen somewhat lower than those presented. In practice, the void fraction before OSV is of the same magnitude as the error in measurement, as shown by Rogers et al. (1987), and the correction is small. If the volume-averaged void is estimated to be about 1%, the reported pressure drop exceeds the sum of frictional and accelerational contributions by about 0.025 psi.

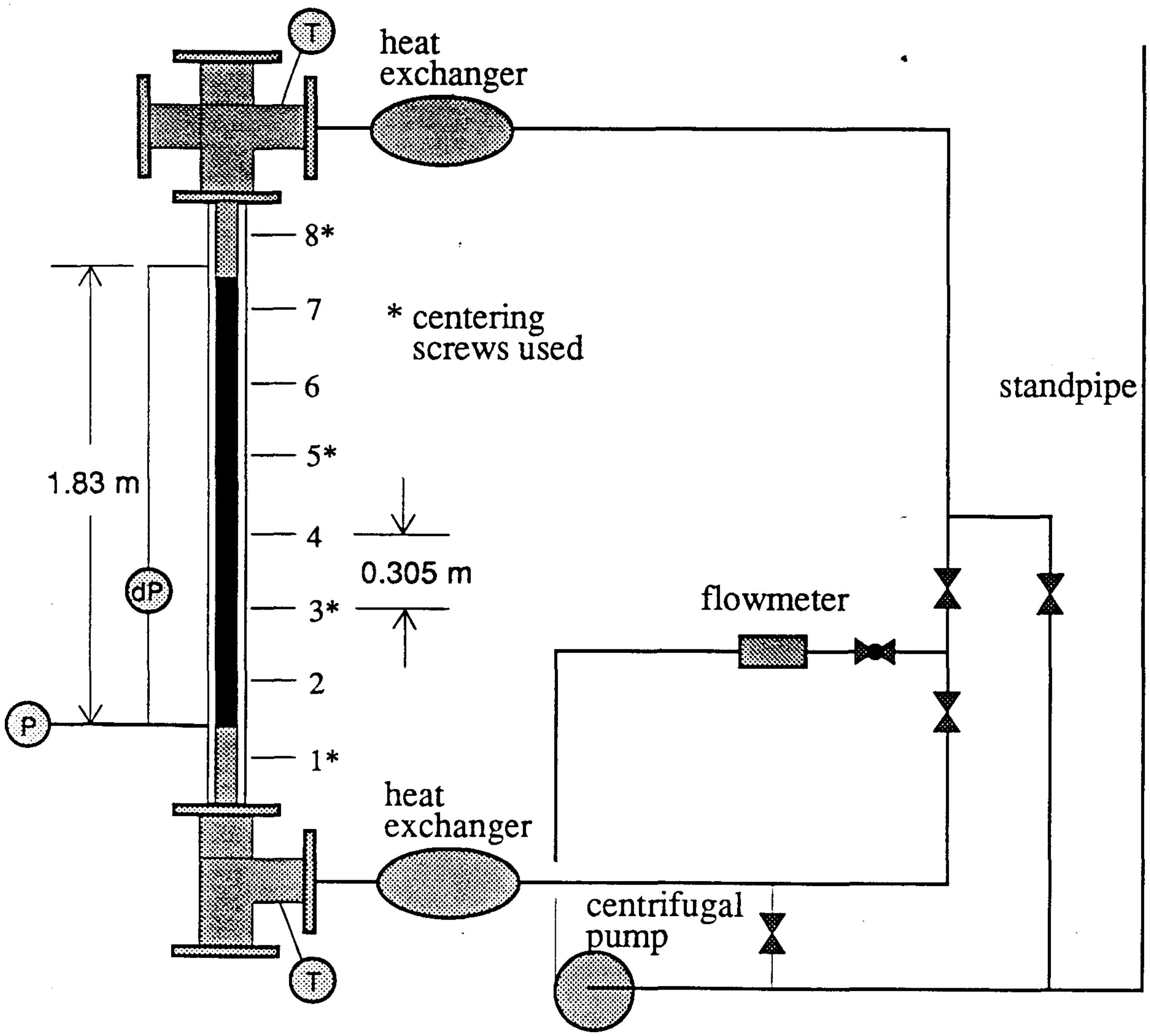

Fig. 1 Schematic of flow loop showing instrumentation and location of centering screws

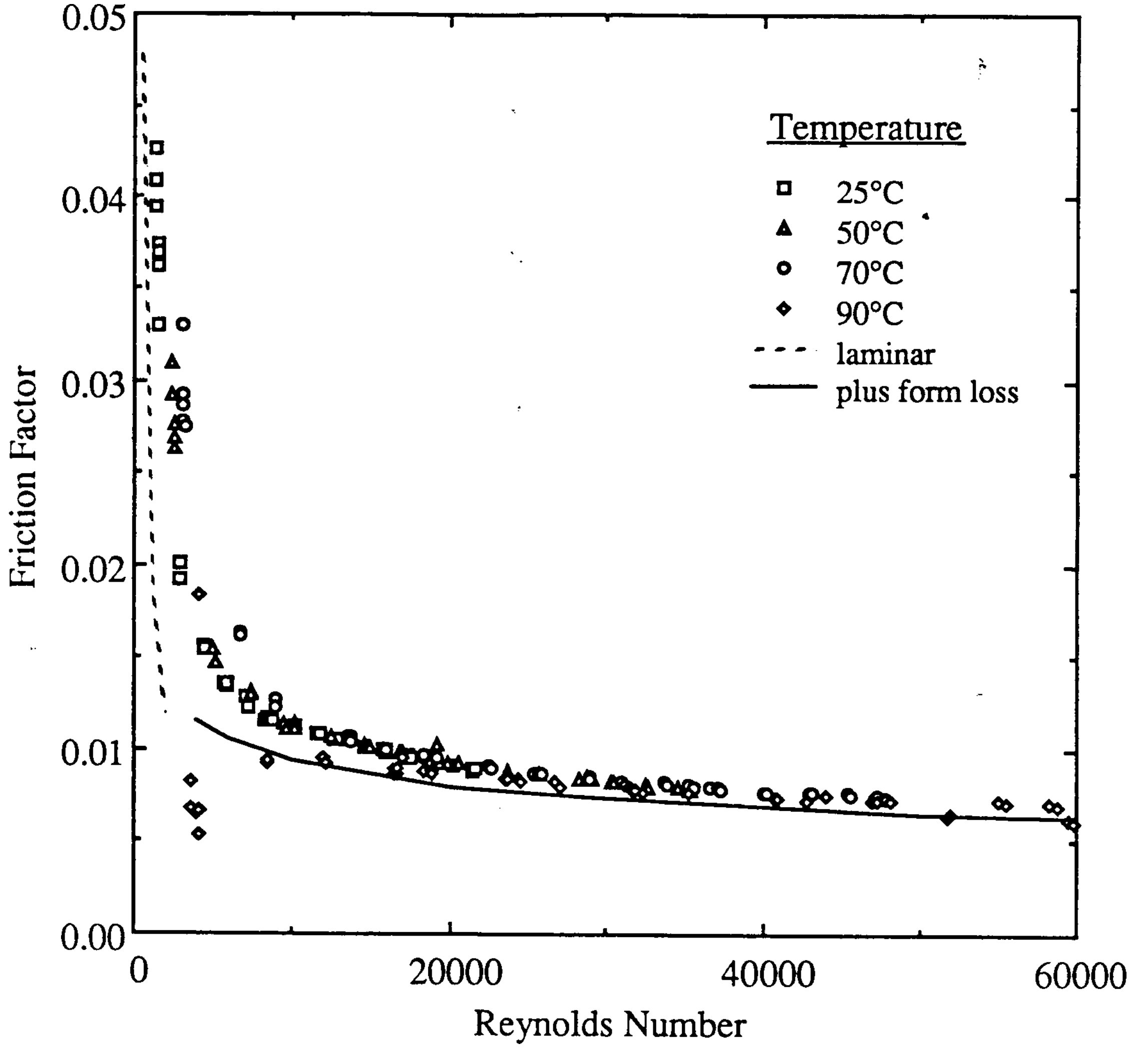

Fig. 2 Test section friction factor for unheated experiments

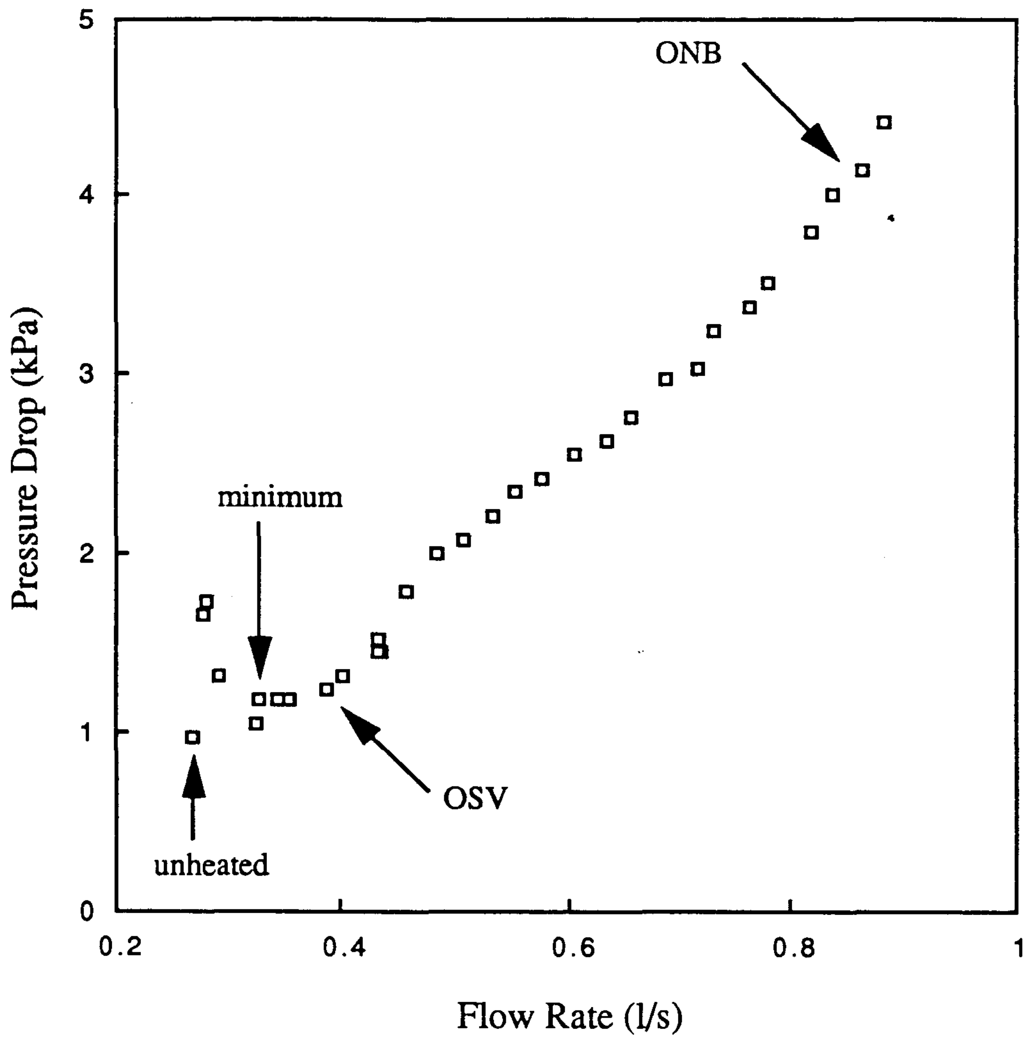

Fig. 3 Demand curve for 205 kW/m² and 80°C inlet temperature

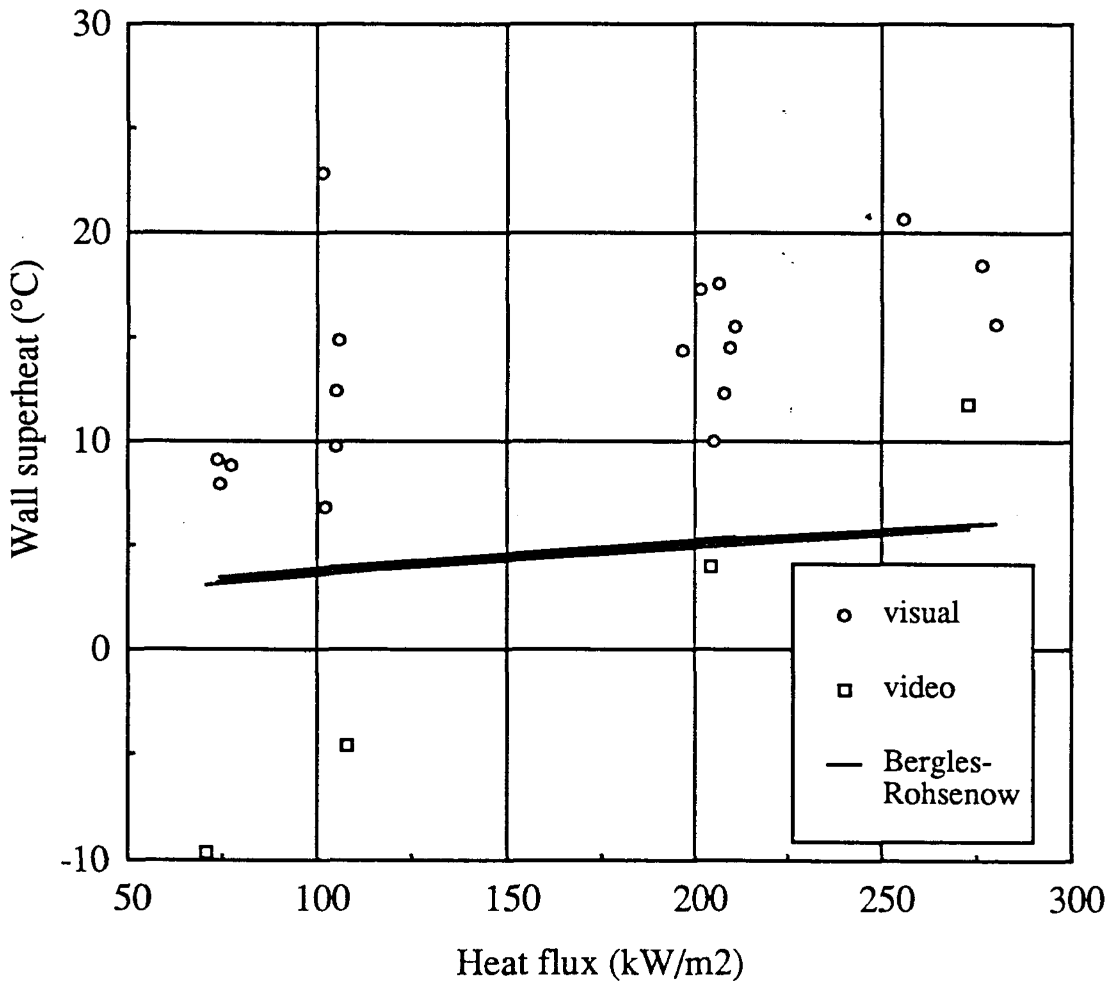

Fig. 4 Wall temperature at first appearance of bubbles compared to the correlation of Bergles and Rohsenow (1964)

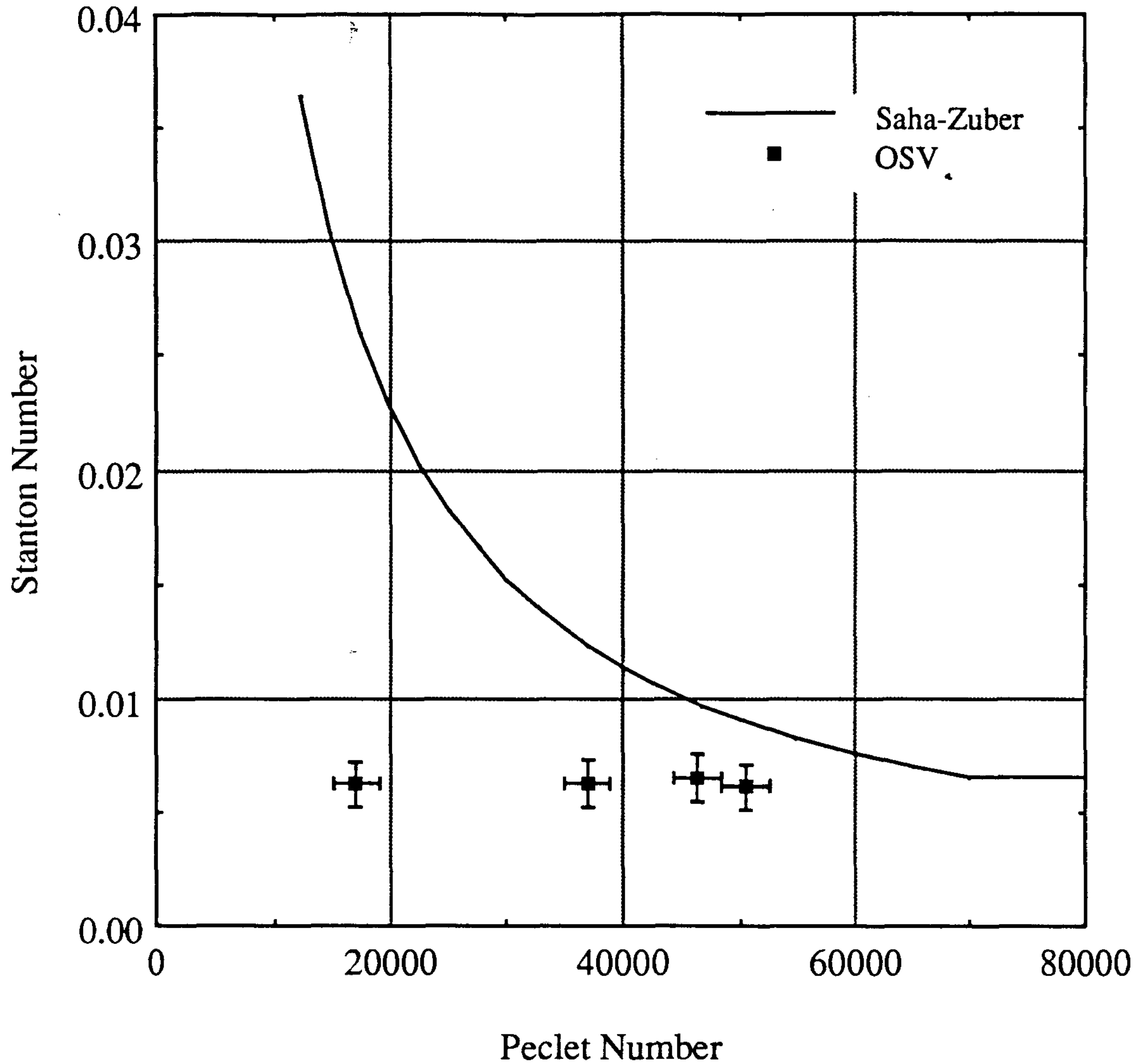

Fig. 5 Conditions at the onset of significant void (OSV) expressed as Stanton and Peclet numbers, compared to correlation of Saha and Zuber (1974)

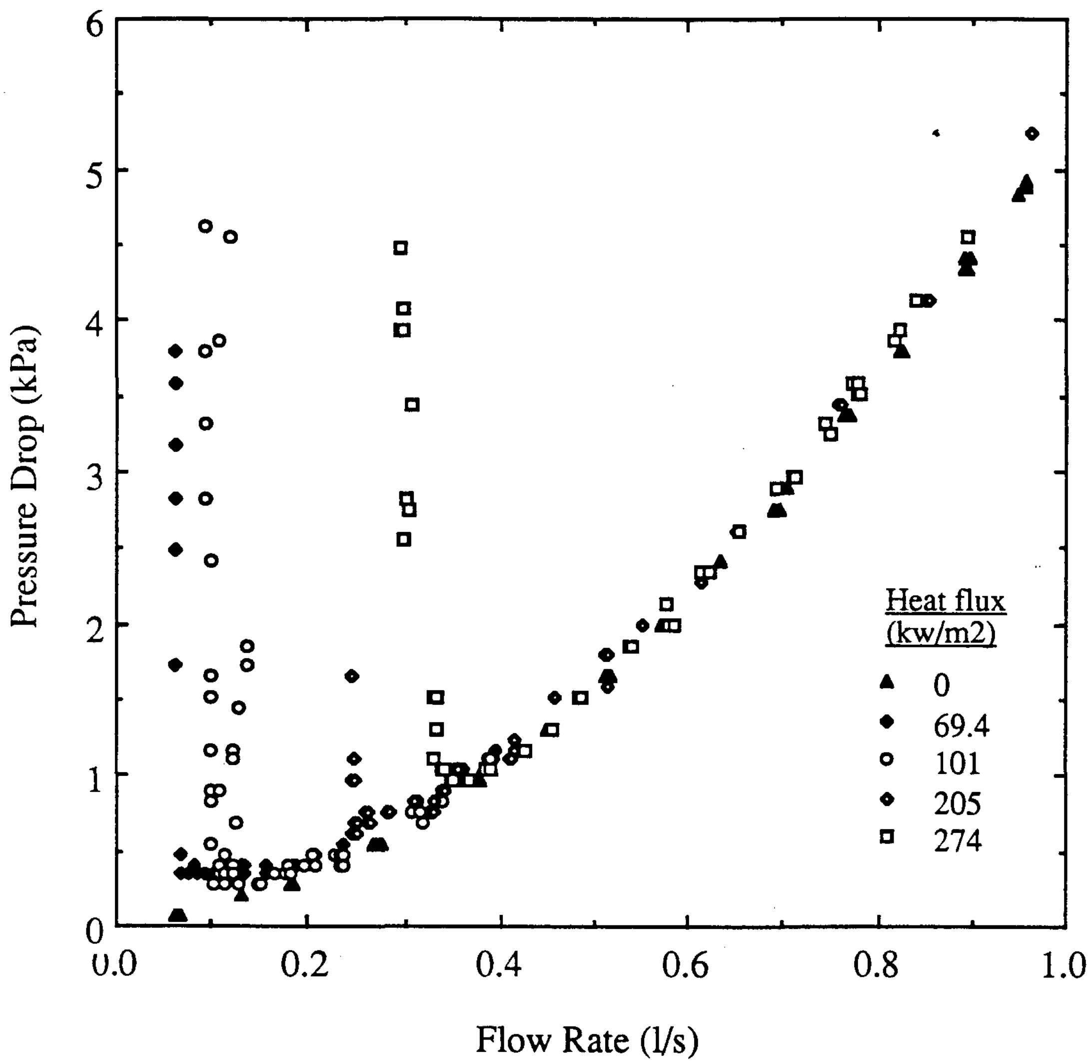

Fig. 6 Test section demand curves at 70 °C inlet temperature

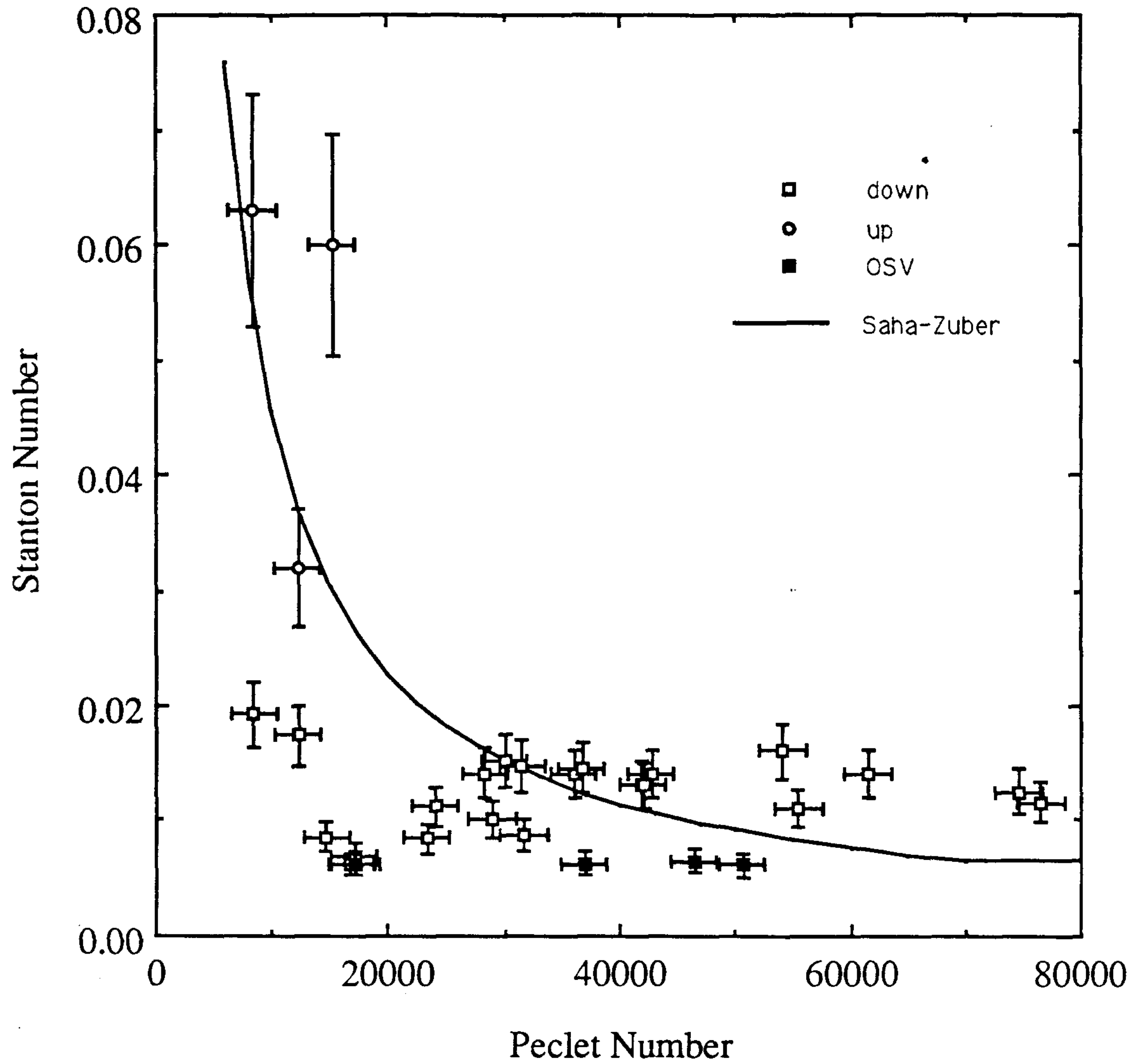

Fig. 7 Demand curve minima plotted as Stanton and Peclet numbers, compared with OSV data and the correlation of Saha and Zuber (1974)

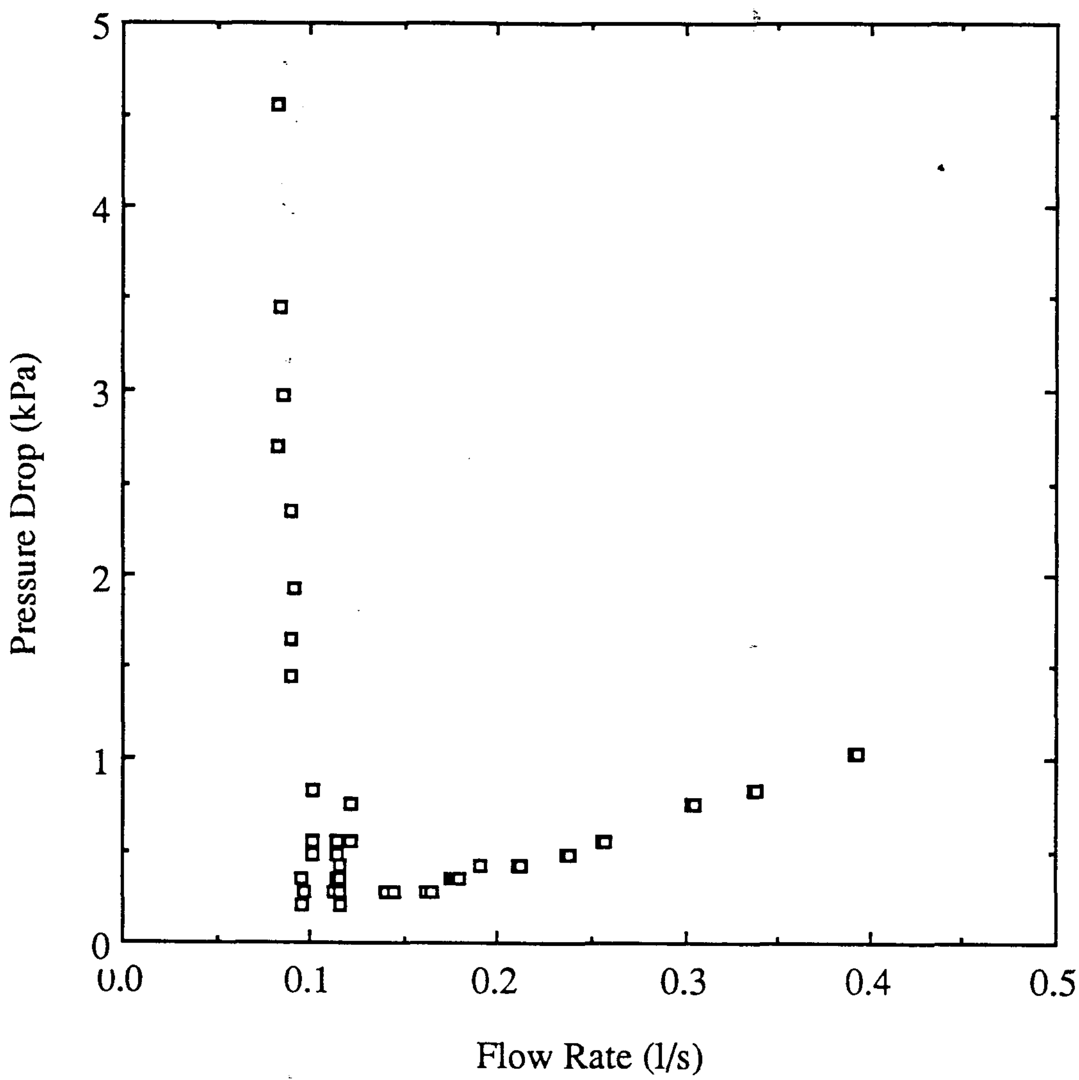

Fig. 8 Demand curve showing multiple rise in pressure drop

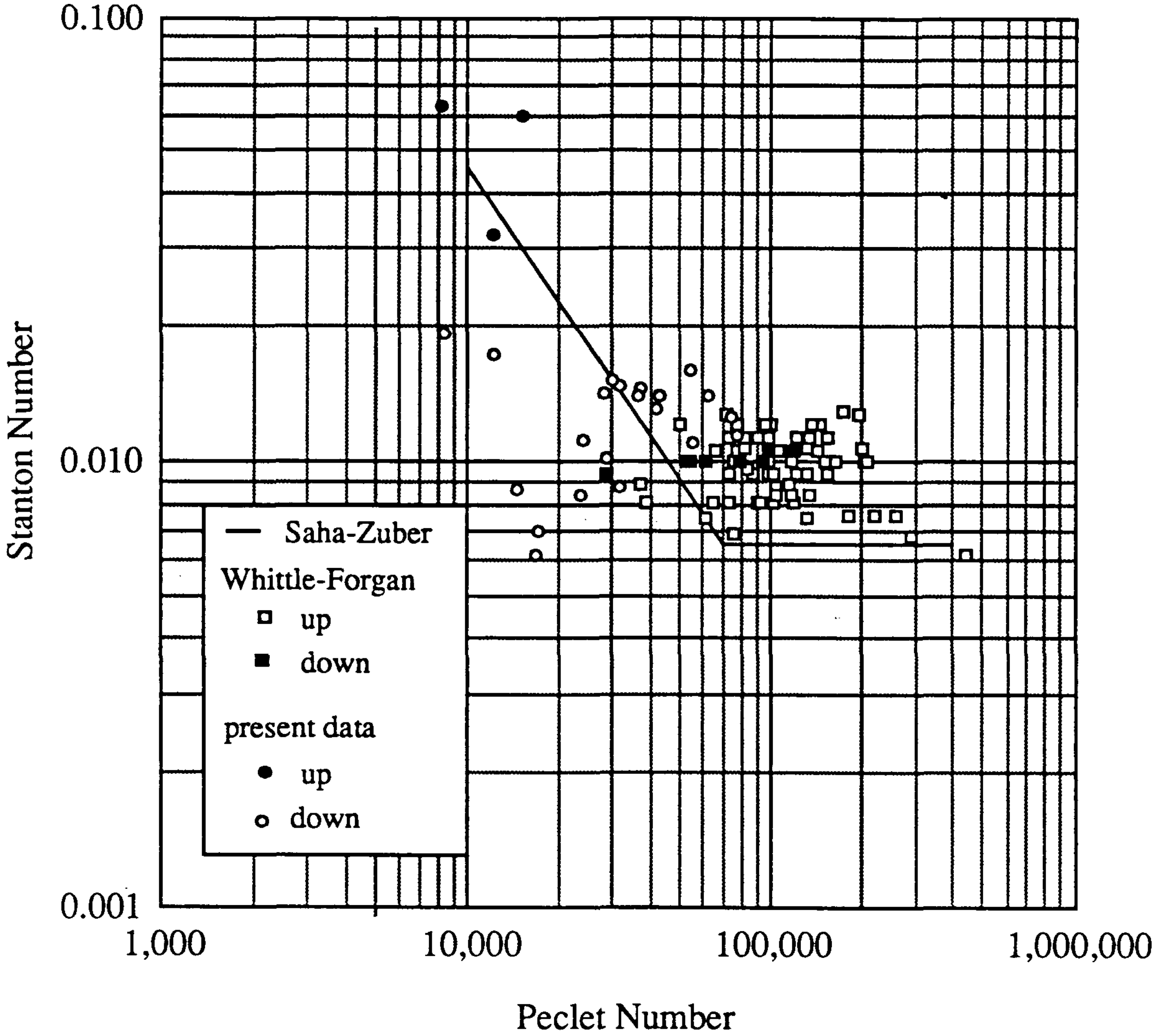

Fig. 9 Minima of demand curves from experiments of Whittle and Forgan (1967) compared to present data and correlation of Saha and Zuber (1974)