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THE EFFECT OF LONGITUDINAL SPACER RIBS ON THE MINIMUM PRESSURE DROP IN A HEATED ANNULUS

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SUMMARY

When evaluating a heated flow passage for vulnerability to static flow excursions, special note should be taken of flow restrictions which might allow premature vapor generation. In this study, measurements of steady state pressure drop were made for the downward flow of water in a vertical annulus. The outer wall was uniformly heated to allow subcooled boiling. Minima in the pressure drop characteristics were compared for test sections with and without longitudinal spacer ribs. For a given power and inlet temperature, the minimum occurred at a higher flow rate in the ribbed test section. This is attributed to vapor generation at the ribs.

INTRODUCTION

The Ledinegg flow excursion is one of several flow instabilities which may be observed in a heated channel (Ishii, 1982). It requires a system whose resistance to flow may increase without a corresponding increase in the pressure difference producing the flow. If vapor is generated, the result is an excursive reduction in channel flow rate. In some systems, this may lead to overheating and damage.

A common candidate for this type of flow excursion is a group of parallel channels joining common headers. The resistance to flow is determined by the channels as a group; an increase in resistance in a single channel will have little effect on the overall pressure drop. Increased flow resistance may be caused by a piece of debris becoming lodged in one channel. Even filtered systems, however, can experience flow excursions if the heating is not evenly distributed to the channels. This is often the case in a fired boiler or nuclear reactor. The formation of vapor in the hottest channel can provide the increased flow resistance necessary to initiate the excursion. This might occur in an upset to a normally all-liquid system, or during startup of a system which is designed to run at high vapor fraction.

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Flow excursions are usually analyzed from consideration of the steady state behavior alone (Boure, et al., 1973). The pressure drop required for flow through a single channel is plotted versus its flow rate. For an unheated channel, this "demand curve" is a monotonically increasing relationship. When heating a fluid which may boil, however, there exists a domain of flow rate in which the pressure drop actually increases for decreases in flow. This implies a local minimum in the demand curve; it signifies the conditions at which a flow excursion would occur, were this channel placed in parallel with others and the driving pressure drop reduced. Demand curves for short tubes have been obtained by Maulbetsch and Griffith (1965) and for narrow rectangular channels by Whittle and Forgan (1967).

In design or evaluation, the channel pressure drop must be estimated by considering the entire channel, part of which features two-phase flow. As a simpler means of predicting flow excursion conditions, a local indicator of the onset of flow excursion might be used. Whittle and Forgan (1967) tried to relate their minima data to a criterion for the departure of bubbles from the heated surface. Saha and Zuber (1974) have discussed the role of bubble departure in determining the Onset of Significant Void (OSV). OSV, by various names, is a recognized occurrence in subcooled nucleate boiling (Collier, 1981). Analyses and correlations for OSV have been presented by Levy (1967), Saha and Zuber (1974), and Rogers, et al, (1987), among others. Qureshi, et al, (1989) have noted that OSV is expected to occur in a heated channel at a flow rate somewhat higher than that at the demand curve minimum. This would provide a conservative estimation of flow excursion conditions.

The use of OSV as an indicator seems a plausible scheme. However, the correlations for OSV were formed for channels in which the flow of fluid and heat were unobstructed. A solid object placed adjacent to a heated wall in a flow channel will have the effect of producing a hot spot on the wall at the same place that fluid flow is restricted. This might be expected to lead to vapor formation at conditions where there would be none without the restriction. Were the vapor formed in sufficient quantity, the demand curve for the channel might reach its minimum before OSV would be expected in the unobstructed channel. The use of an OSV criterion in the hope of avoiding a flow excursion would then be poorly repaid.

It is well known that narrowing a flow passage can improve heat transfer. Agrawal and Sengupta (1987) performed calculations for laminar single phase flow with constant temperature boundary conditions in an obstructed annulus, finding that the Nusselt numbers were increased in the region of the obstruction. Ishibashi and Nishikawa (1969) studied pool boiling in a vertical concentric annulus. They determined that heat transfer improved as the gap size was decreased. Jensen, et al, (1977), in an experimental study of boiling in a narrow horizontal annulus, also showed that heat transfer was improved in the confined region. Chyu and Mghamis (1989) presented data for pool boiling between cylindrical heaters in line contact. This arrangement, compared to a single tube, produced ten-fold increases in heat transfer coefficient.

However, these boiling studies also demonstrated that narrow spaces can increase vapor production. Chyu and Mghamis (1989) observed nucleate boiling in the heater contact region at lower superheat than required for a single tube. Jensen, et al, (1977) noted that boiling was initiated in the annular crevice prior to the unobstructed heater surface. Both Jensen, et al, (1977) and Ishibashi and Nishikawa (1969) also found that the critical heat flux was reduced by the narrow gap.

In another study of restricted boiling, Baum and Curlee (1980) examined eccentric crevices of various shapes appropriate to steam generator tube supports. A heated tube was pressed against a surrounding wall; high tube wall temperatures indicated the presence of a stable dry region, and thus vapor production, about the line of contact. Separate experiments related the regions of high temperature to chemical deposits on the heater surface. Sharon, et al, (1983) described experiments with boiling flow in narrow concentric gaps formed by passing a heated tube through drilled plates. At bulk fluid temperatures far below saturation, the pressure drop across a plate was found to exceed that for non-boiling flow.

The works cited above show how a restriction in a heated channel can produce vapor which would not be observed in the absence of the restriction. In addition, the paper by Sharon, et al, (1983) indicates that this vapor production increased the pressure drop at conditions under which the unobstructed channel would have shown no increase. In the present study, the effect of a flow restriction on the tendency to flow excursion is explored by finding demand curves for a heated annulus in subcooled boiling flow. The annulus is heated from the outside, and alternately equipped with and without longitudinal spacer ribs. These ribs separate the heated and unheated walls; in pressing against the heated wall they provide a means for premature vapor production. Spacer ribs are used in nuclear reactor assemblies (Hodges and Knoebel, 1973), extended-surface heat transfer (Fraas, 1989), and for high pressure applications (Purohit, 1986).

EQUIPMENT

A conventional piping loop was used to circulate water in downward flow through the heated test section. The flow was throttled at pump discharge and measured with a turbine meter. The pressure in the loop was set by compressed air in a surge tank connected to the pump suction piping. Power was supplied by four SCR-controlled DC rectifiers, each capable of 5000 A at 30 V. These were connected to the test section via water-cooled aluminum buswork.

The test section is illustrated in Figure 1. An annulus was formed between a heated stainless steel tube and an unheated insert. This insert was composed of aluminum segments separated by polyamide spacers, the latter used to support thermocouples and pressure taps. The annulus was separated into four channels by fiberglass ribs attached to the unheated inner wall. For a second series of tests, the ribs in the heated section were filed down flush with the inner wall. Test section dimensions are listed in the table below.

Quantity	Measurements	Mean (mm)	St Dev (mm)
Heater OD	14	83.2	0.5
Heater ID	20	80.1	0.4
East-West	1	79.6	–
North-South	1	80.5	–
Rib circle OD	24	79.5	0.2
Insert OD	24	70.9	0.1
Rib thickness	24	3.5	0.1

The outer tube was found to be slightly oval in cross-section. The radial clearance over the north and south ribs was thus about 0.5 mm, while that over the east and west ribs was about one tenth of that, which lies within the uncertainty of the measurement.

Pressure taps and thermocouples were led out through the insert to the bottom of the test section. Copper rings were silver-soldered to the outer stainless steel tube; bus bars were clamped to these for power distribution. Sheathed thermocouples were mounted about the periphery in polycarbonate ring supports; the sheaths were separated from the heater surface by sheets of mica. The entire test section was covered with about 80 mm of fiberglass insulation.

INSTRUMENTATION AND DATA ACQUISITION

The pressure drop was measured over a 0.30 m portion of the heated length, as shown in Figure 1. The differential pressure gauge zero and span were checked daily against a calibrated gauge. The static pressure was measured at the bottom of the 0.30 m measurement section. Loop flow was measured by a turbine flowmeter, calibrated by mass scale and timer. Thermocouple circuits were calibrated by comparing the readings to that of a calibrated mercury-in-glass thermometer with the loop in isothermal flow. Signals from these instruments were acquired using a minicomputer with A/D conversion boards. For each reading during experimentation, the group of channels was scanned 15 times at a frequency of 5 Hz.

The heater voltage was measured as the sum of the readings of two digital voltmeters connected between each bus bar and ground. This method allowed observation of shifts in the ground plane between the bus bars. A measurable shift was usually associated with significant boiling within the test section. The current was inferred from measurements of voltage across the shunts associated with each of the four rectifiers used.

PROCEDURE

For each day of testing, the pressure transducers were calibrated. The loop was then run at boiling to reduce dissolved gas concentration in the water. Tests were performed over a range of heat flux from 0 to 1.55 MW/m^2 and inlet temperature between 25 and 115 °C, with a nominal 330 kPa exit pressure. Heated tests were run by maintaining the inlet temperature and power while decreasing the flow in steps. After each change in flow rate, from 30 to 120 seconds were allowed to pass to achieve stability of the flowmeter and temperatures before making measurements. The flow was reduced until the test section was visibly shaking. Typically, the distribution of voltage between positive and negative would shift and vary, the total remaining about constant. On several occasions in ribbed tests, smoke was noticed issuing from the insulation. From this point, the flow was increased in steps to retrace the demand curve, but in less detail than was done in descending flow. The inlet temperature was kept within the prescribed limits by making adjustments to the coolant water flow rate as the loop flow was decreased.

DATA REDUCTION

The data were time-averaged. The thermocouple readings were corrected by the calibration constants and used to calculate physical properties of water at the outlet temperature. Heated wall temperatures were derived by radial conduction from the outer wall measurements. The absolute pressure was calculated by correcting the gauge pressure measurement for prevailing atmospheric pressure and the elevation of the gauge with respect to the tap. No measuring line correction was necessary for the differential pressure transducer readings because the taps were inside the insert and were thus nearly at operating temperature. 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.

Comparison of the power determined from electrical measurements with that obtained by heat balance gave agreement within ±5% over the full range of flow rates used. The reported heat flux is calculated from the power, based on the inside area of the heated tube.

Uncertainty variances are estimated as follows.

Annulus Diameters	± 0.3 mm
Heated Length	± 1.3 mm
Tap Spacing	± 2.5 mm
Temperature	±1°C
Pressure Drop	±4% of reading
Gauge Pressure	±0.5% of reading
Flowmeter	±2% of reading
Heater Voltage	±0.2 V
Heater Current	±4% of reading

Based on these, the standard sum-of-squares uncertainty in calculated results is computed.

Heat Flux	± 4.4%
Peclet Number	± 5.3%
Stanton Number	± 10%

The uncertainty in the Stanton number includes some uncertainty in determining the flow rate at the minimum pressure drop.

OBSERVATIONS OF PHYSICAL CONDITION OF TEST SECTION

After the first series of tests, the test section was dismantled so that the ribs could be removed. Opening the insulation revealed that the inner portion of the polycarbonate thermocouple support rings had been melted, and over the west rib were mixed with scorched fiberglass insulation. The outer surface showed marks of overheating, symmetric about the east rib, and on the north side of the west rib. The thermocouples had not remained in place; half of the twelve in use had slipped from their pinned positions.

The aluminum insert was corroded in two distinct patterns, corresponding to the prevailing voltage field. The upper portion, negative with respect to the heater, was covered with a smooth black deposit. The lower portion, positive with respect to the heater, was pitted and covered with powdery white deposits. The plastic spacers were protruding above the aluminum surface by as much as 0.8 mm. The lower spacers were cracked and flaking, as well.

By contrast, the inner surface of the stainless steel heater tube was generally smooth and shiny. Deposited solids were found marking the positions of the ribs on the heater. These were most pronounced for the east and west ribs, and lightest for the south rib. The deposit was thin in a stripe where each rib had been, and heavier to either side. Scanning electron microscope analysis of this residue revealed the presence of aluminum, silicon, calcium, magnesium, and iron. The deposits were firmly attached; fine steel wool was required to remove them from the heater tube.

In preparation for the second series of tests, the ribs were filed flush with the surface of the insert over the heated section, leaving portions intact up and downstream for centering, as shown in Figure 1. The aluminum insert was smoothed with emery cloth. A second series of tests was then run with no ribs in the measurement section. Dismantling the test section after this second series revealed that the insert was corroded in a similar manner to that first observed. Once again, the spacers had swelled and cracked, protruding above the aluminum surface, and lifting the embedded ribs up slightly. After wiping with a cloth, the inside of the heater was smooth and shiny; the former crusty rib deposits were not in evidence.

RESULTS

Figure 2 compares test section demand curves for ribbed and unribbed test sections. Also shown for reference is a demand curve for unheated flow. The two heated tests were run at the same inlet temperature and same total power. For the ribbed test section, the upturn in the demand curve occurs at a higher flow rate; this implies a lower exit temperature at the minimum than is observed without ribs. Had two such test sections been put in parallel, and the driving pressure drop reduced, the ribbed section would have been the first to experience flow excursion.

More experiments of the sort illustrated in Figure 2 were run at various power and inlet temperature conditions. The flow rate at the minimum was determined for each. To present the data, it is necessary to find some means of combining the effects of power and temperature. Whittle and Forgan (1967) accomplished this by forming the ratio of actual temperature rise to inlet subcooling.

$$R = \frac{T_o - T_i}{T_s - T_i} \qquad (1)$$

These authors then used an overall energy balance to express R in terms of the outlet conditions alone. This can be written

$$R = \frac{1}{1 + \frac{A_f}{A_h} \frac{G\, C_p\, (T_s - T_o)}{\phi}} \qquad (2)$$

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In this form, R combines the power, inlet temperature, and flow rate with the dimensions of the heated section. The ratio in the denominator recalls the modified Stanton number used by Saha and Zuber (1974) for their correlation of the onset of significant void (OSV).

$$St = \frac{\phi}{G\, C_p\, (T_s - T_o)} \qquad (3)$$

Whittle and Forgan (1967) had some success using Equation (2) to correlate R with length-to-dia neter ratio. This suggests that the minima of demand curves might be usefully presented in terms of the Stanton number, calculated at the pressure and average temperature at the exit of the channel. The Stanton number which specifically characterizes the power, flow, pressure, and liquid temperature at the demand curve minimum will be termed St*. A large St* indicates that the minimum features some combination of high power, low flow, and low subcooling. By contrast, a low St* indicates that flow excursions would occur at lower power and higher flow and subcooling.

Figure 3 collects the minima data for both test sections. Here the exit Stanton number has been plotted against the Peclet number, which indicates primarily flow variation between experiments. The data fall into two distinct groups with little overlap. A line representing the mean of the observations is set through each group. As was discussed above, the lower St* of the ribbed test section indicates that it reaches its pressure drop minimum at milder conditions, or equivalently, that it is more prone to flow excursion.

As a convenient reference, the OSV correlation of Saha and Zuber (1974) has been plotted as well. With few exceptions, this correlation divides the ribbed and unribbed data. This indicates that the minimum pressure drop in the unribbed test section would not occur unless OSV had been predicted in the test section. Conversely, the ribbed test section would reach its minimum at conditions predicted to be insufficient for OSV. Alternately stated, the ribbed test section shows the effects of vapor production before significant vapor is predicted to be present, if the prediction is based on a correlation for an unobstructed channel.

Figure 4 presents data of Maulbetsch and Griffith (1965) and Whittle and Forgan (1967). Equation (2) implies that demand curve minima have a certain dependence on the ratio of heated area to flow area; this is borne out by the tube data. St* from the three tubes of Maulbetsch and Griffith (1965) overlap, and they match the Whittle and Forgan (1967) tube data, as well. However, data from the three shorter rectangular test sections of Whittle and Forgan (1967) tend to be separated from the tube data, even though they are in the same range of heated area/flow area. On the other hand, data from the long rectangular test section of Whittle and Forgan (1967) are closer to the tube data. The data for the unribbed annulus in the present study also fall in with the tube data, while the ribbed data lie below the others. Wide scatter is observed in all data sets, indicating that use of the outlet conditions alone may not be a sufficient means of characterizing the channel pressure drop.

MINIMA IN THE RIBBED TEST SECTION

It was noted above that Baum and Curlee (1980) observed regions of stable dryout about the region of contact between a heated surface and another object. They showed that solids deposition can be associated with such a dryout, and that stable high temperature gradients are maintained from the dry patch to the surrounding well-cooled areas. In the present study, the test section was installed in an existing piping loop, which through years of use was no longer suitable for maintaining water purity. As described above, deposited solids were found in regions of rib contact. Here expediency poses as virtue, or at least good fortune, because the deposited solids are evidence of prevailing dryout, and hence vapor production, near the ribs. Although dryout in channel centers was indicated by high wall thermocouple readings on several occasions, these incidents were too few and short in duration to account for the solids buildup. The deposits were observed only at ribs, particularly at those with the tightest clearance.

It is not surprising that vapor was routinely produced at the ribs. Chyu and Mghamis (1989) have shown that unobstructed portions of the heated wall need not be above the saturation temperature to have boiling take place in the confined regions. A simple two-dimensional conduction calculation for the heater tube indicates that the wall near the rib will be superheated for the range of heat flux used in this test. Contact dryout occurs at very low heat fluxes, so that vapor production is realized well before the central portion of the channel is expected to exhibit boiling.

Vapor which forms at the ribs in sufficient quantity is expected to force redistribution of flow among the channels. In effect, a Ledinegg flow excursion may occur within the test section itself. Figure 5 plots the temperature of the unheated wall at the inlet and outlet of a single channel. The difference in these is indicative of the fluid temperature rise in the channel. As the flow is decreased, this difference rises in the expected manner. At the point of the minimum pressure drop, however, the temperatures increase and become nearly equal, indicating that flow in that channel has stalled. The corresponding wall superheat at the outlet of the measuring section is shown in Figure 6. Until the flow diversion resulted in widespread drying and high temperatures, the wall itself was subcooled.

PERTINENCE TO THE TYPICAL FINNED TUBE

It must be emphasized that the vapor-producing characteristics of this ribbed test section were more severe than they might be. First of all, the ribs were not thermally conductive; in non-electrical applications, the heat produced in the tube wall can be conducted through more congenial rib material (as in a cooling fin) to the flowing liquid. Second, because the outer wall was insulated, the heat generated under the insulating rib had to be conducted azimuthally to either side. Were the wall cooled on two sides, the overheating under the rib would be considerably less. In addition, a narrower rib would also provide less blockage to heat flow. Finally, the rib clearance was small; a larger clearance would be less susceptible to dryout.

CONCLUSION

This work was performed to relate an observed phenomenon - enhanced vapor production near restrictions - to an exacerbated tendency to flow excursion. To this end, the minima of demand curves were obtained for the downward flow of water in a uniformly heated annulus. Both ribbed and non-ribbed test sections were used. For a given power and inlet temperature, the ribbed test section reached its pressure drop minimum at a higher flow rate than did the unribbed test section. In addition, the ribbed test section reached its minimum before any significant vapor formation was predicted to occur in the unobstructed channel. This is attributed to vapor generated near the ribs. Prevailing dryout and vapor production were indicated by the buildup of deposited solids about the ribs. Such deposits were not observed in the test section without ribs.

It is concluded that obstructions in heated flow channels should indeed be given special consideration when evaluating the possibility of flow excursion. However, the test section used in this study is perhaps extreme; its wide, insulating, tight-fitting ribs will not be found in many applications. Damage suffered by the test section in the course of the experiment will motivate alterations in design and material selection for any future work in this area.

ACKNOWLEDGEMENTS

The test section was designed by T. J. Steeper and constructed by R. S. Riley. R. Keklak provided critical review of equipment and procedures. M. A. Lee operated the loop, and modified the test section for the second series.

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
G	mass velocity
k	liquid thermal conductivity
Pe	Peclet number, D_e G Cp/k
R	normalized liquid temperature rise
St	Stanton number defined in Equation (3)
St*	Stanton number representing the demand curve minimum
T_i	liquid temperature at inlet to test section
T_o	liquid temperature at outlet of test section
T_s	saturation temperature
Φ	heat flux into fluid

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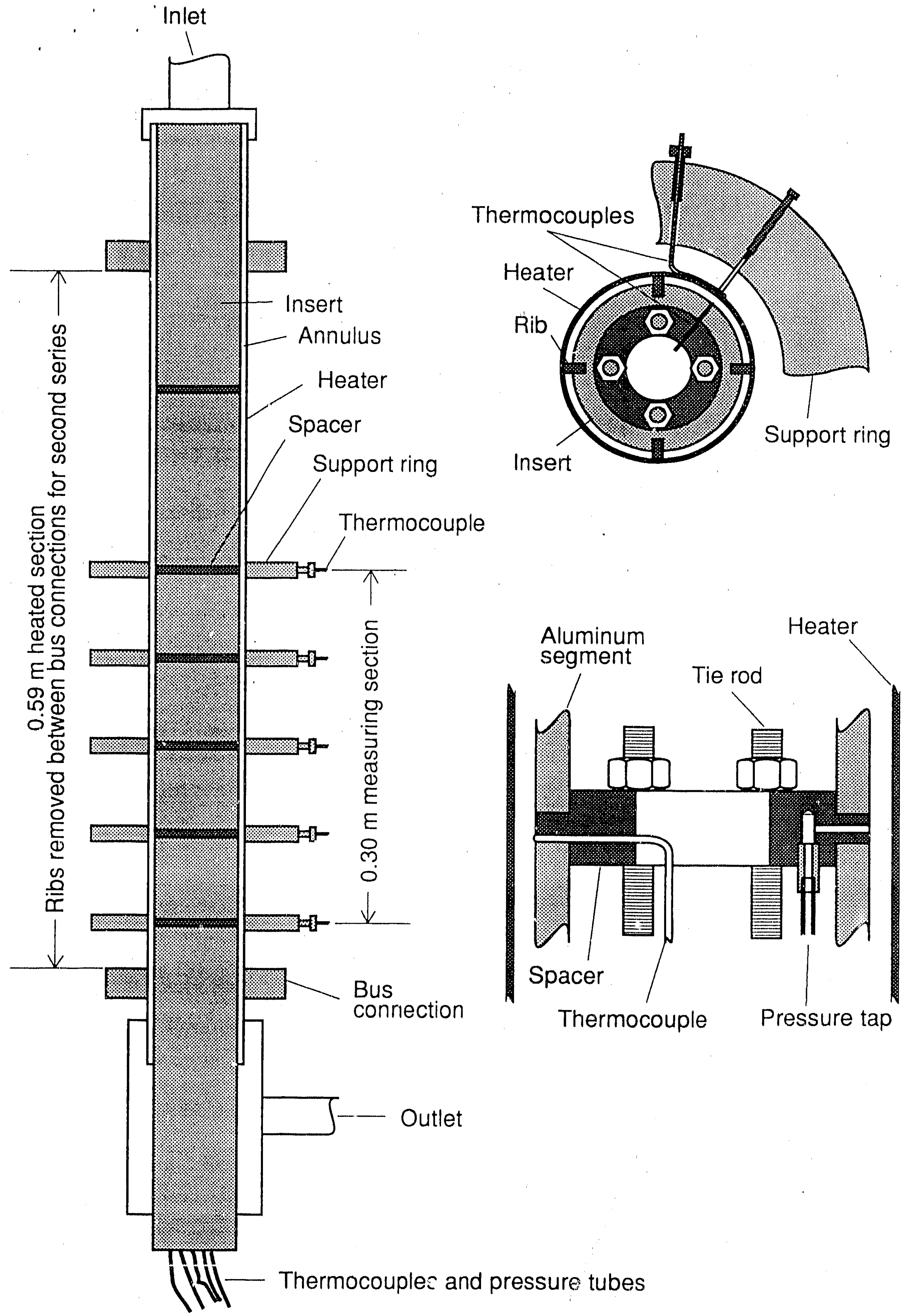
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Thermocouples and pressure tubes
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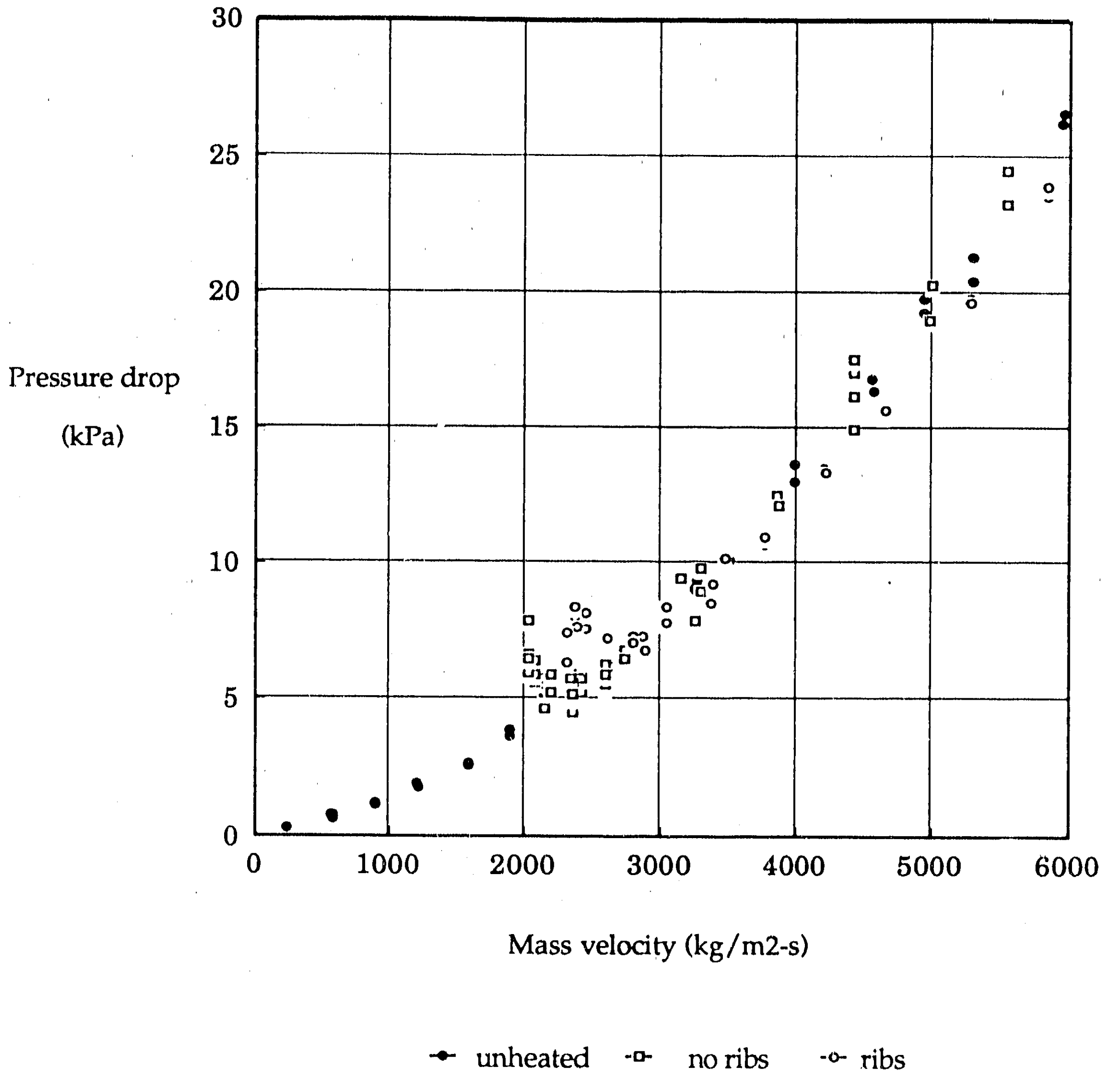

Figure 2. Demand curves for test section, with and without ribs. In heated tests, average heat flux 1.10 MW/m2. Inlet temperature 105°C. Exit pressure 331 kPa.

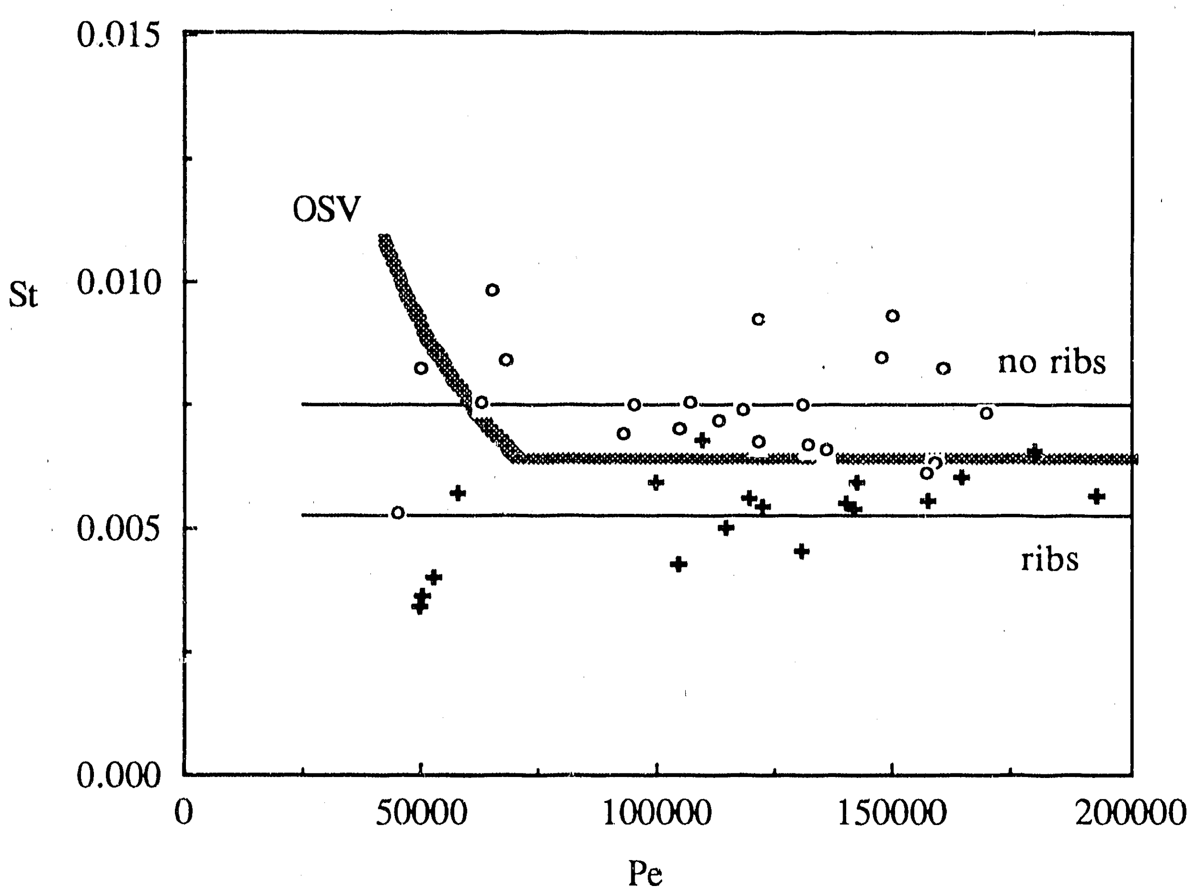

Figure 3. Minima of demand curves for heated test section, with and without ribs. Conditions at minimum expressed as Stanton and Peclet numbers and compared with the OSV correlation of Saha and Zuber (1974).

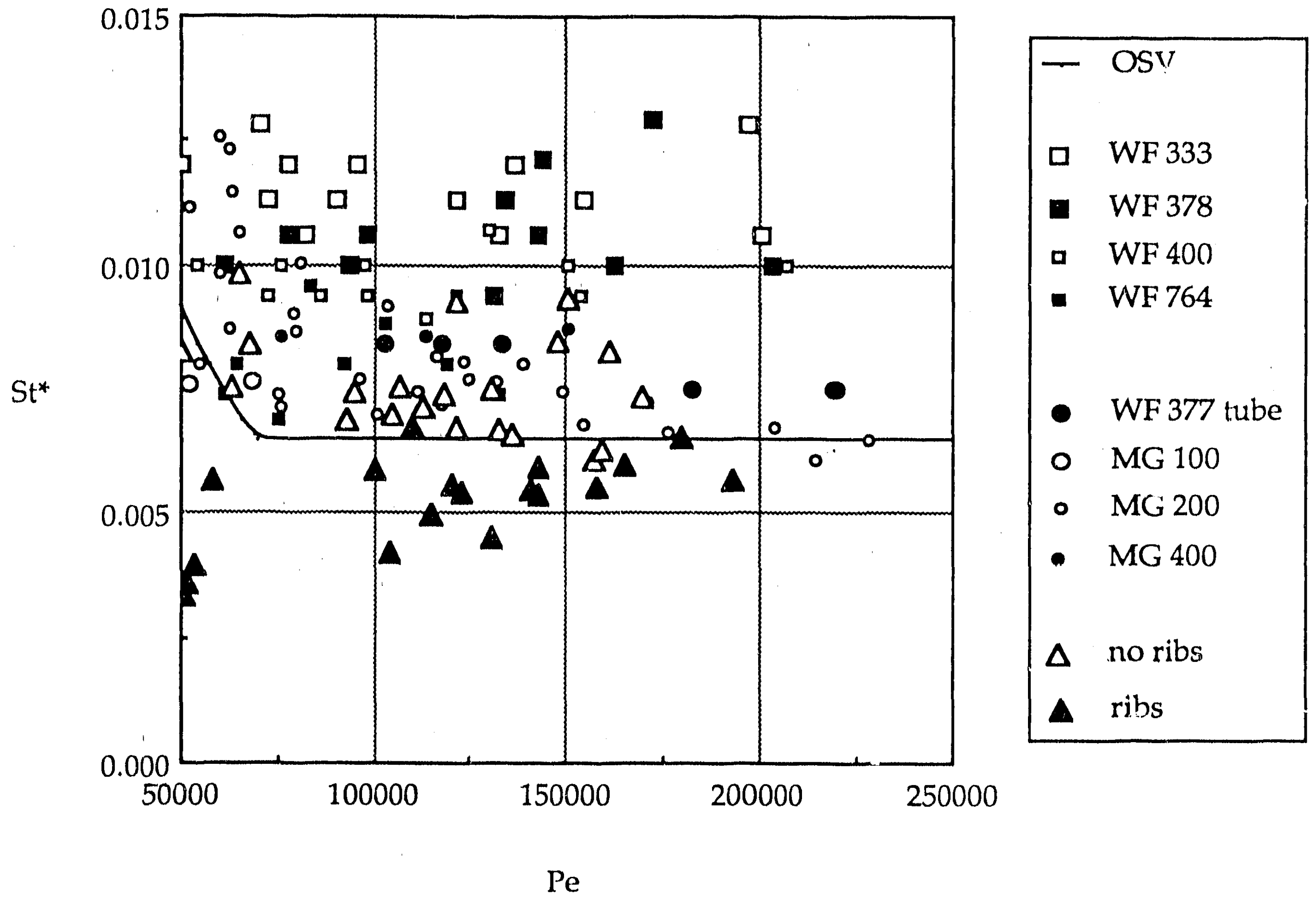

Figure 4. Minima of demand curves from several geometries and heated/flow area ratios.

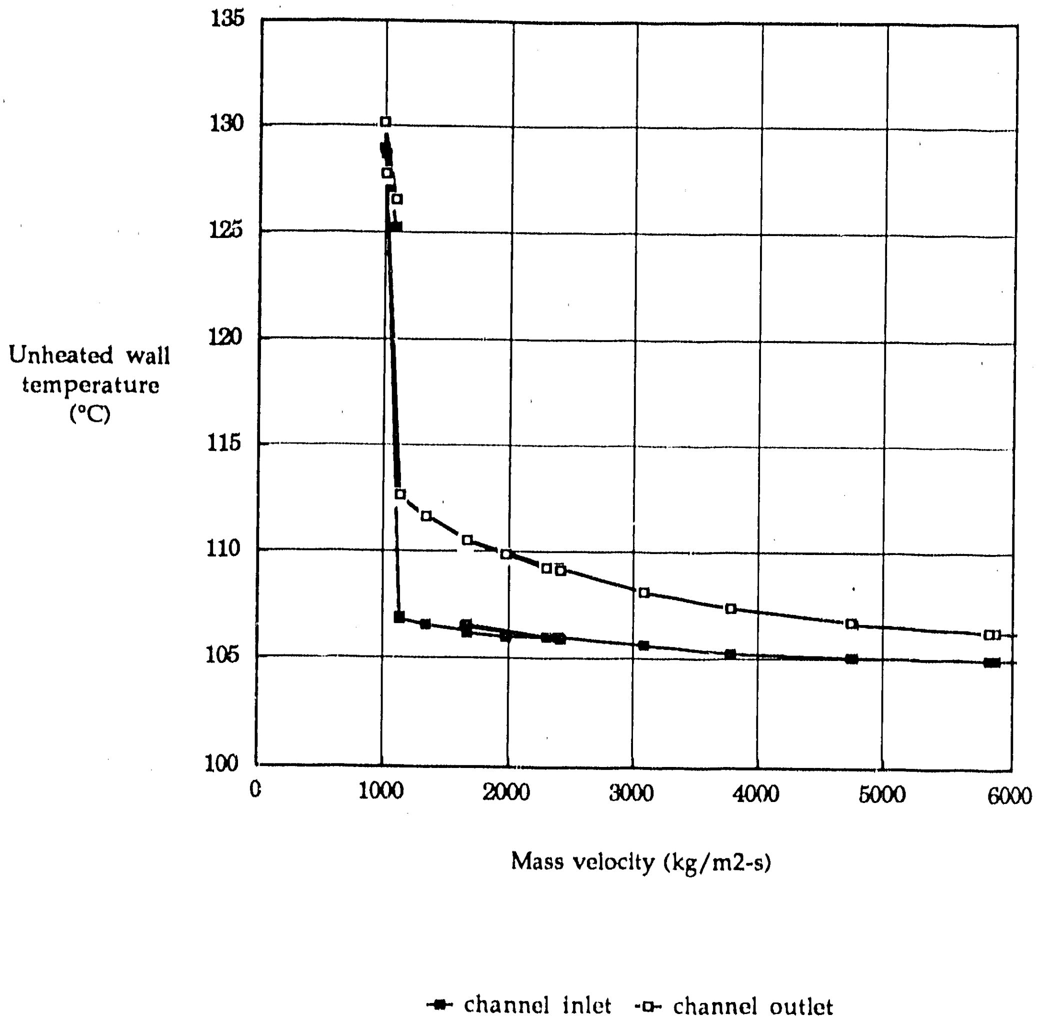

Figure 5. Variation of unheated wall temperature with mass velocity. Data from northwest channel. Average heat flux 360 kW/m2. Inlet temperature 105°C. Exit pressure 352 kPa.

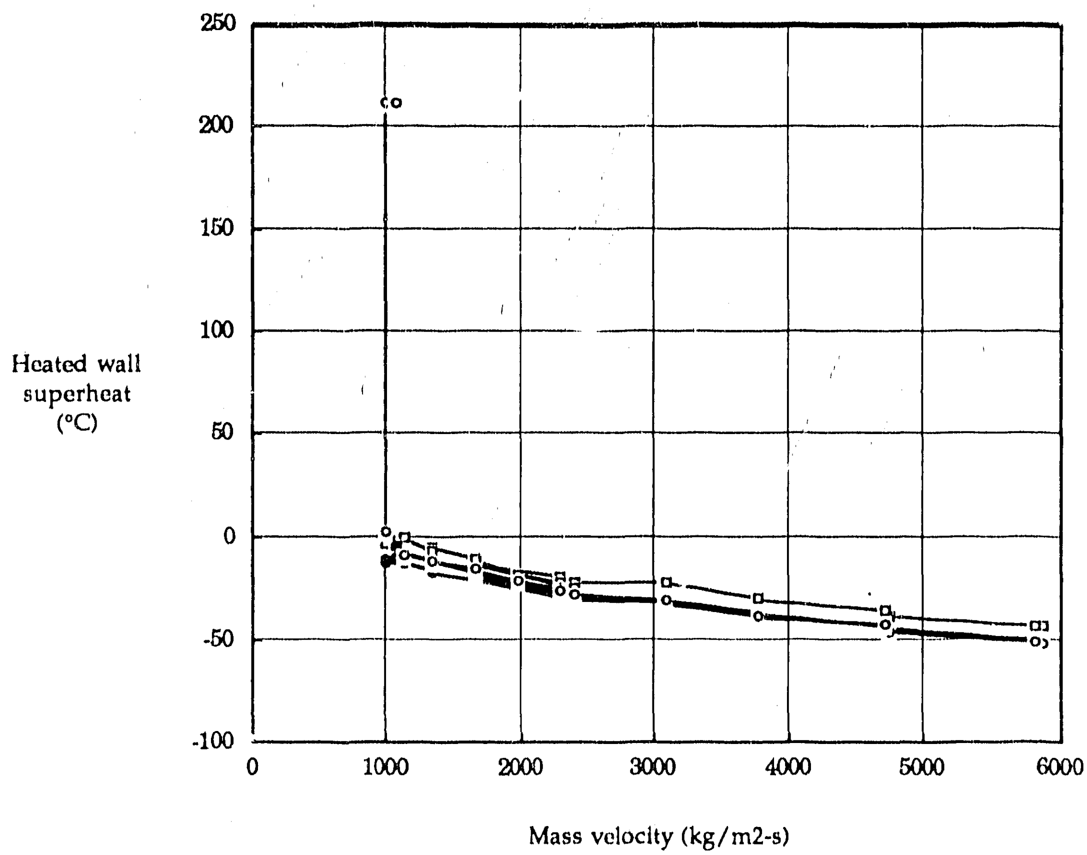

Figure 6. Variation of channel outlet wall superheat with mass velocity. All four channels shown; northwest channel is hot. Average heat flux 360 kW/m2. Inlet temperature 105°C. Exit pressure 352 kPa.

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