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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER TO SUPERCRITICAL PRESSURE CARBON DIOXIDE IN A HORIZONTAL PIPE

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Abstract—Results obtained in an experimental investigation of heat transfer to supercritical and subcritical pressure CO_2 flowing through a uniformly heated 22.14 mm I.D. horizontal pipe are presented. The experimental work covers a flow inlet Reynolds number range of about 2×10^4 to 2×10^5. Marked peripheral temperature variations are obtained which represent the influence of buoyancy. Comparison with buoyancy free data shows that heat transfer at the bottom of the pipe is enhanced and at the top is reduced by buoyancy. Criteria proposed by Jackson [16] and by Petukhov [17] indicate that buoyancy effects would be expected under the conditions of all the experiments.

NOMENCLATURE

C_{p_b}, specific heat of fluid at bulk temperature [kJ/kgK];

$\overline{C_p}$, $= \dfrac{h_w - h_b}{T_w - T_b}$ [kJ/kgK];

D, pipe diameter [m];

h_b, enthalpy of fluid at bulk temperature [kJ/kg];

h_w, enthalpy of fluid at wall temperature [kJ/kg];

k_b, thermal conductivity of fluid at bulk temperature [kW/mK];

$\dot{m}$, mass flow rate of fluid [kg/s];

q_w, heat flux at the wall of the pipe [kW/m^2];

T_b, bulk temperature of the fluid [°C];

T_w, wall temperature of the pipe [°C];

x, distance along pipe from start of heating [m];

μ_b, dynamic viscosity of fluid at bulk temperature [kg/ms];

ν_b, kinematic viscosity of fluid at bulk temperature [m^2/s];

ρ_b, density of fluid at bulk temperature [kg/m^3];

ρ_w, density of fluid at wall temperature [kg/m^3];

Gr_b, $= \dfrac{\rho_b - \rho_w}{\rho_b} \cdot \dfrac{gD^3}{\nu_b^2}$;

Gr_b^*, $= \dfrac{\rho_b - \rho_w}{\rho_b} \cdot \dfrac{gD^4}{\nu_b^2 k_b} \cdot \dfrac{q_w}{(T_w - T_b)}$;

Nu_b, $= \dfrac{q_w D}{k_b(T_w - T_b)}$;

Pr_b, $= \dfrac{c_{p_b}\mu_b}{k_b}$;

Re_b, $= \dfrac{4}{\pi} \dfrac{\dot{m}}{D\mu_b}$.

INTRODUCTION

THIS paper reports work carried out on heat transfer to supercritical pressure CO_2 flowing in a 22.14 mm I.D. horizontal pipe. Experiments have been carried out by a number of investigators using a variety of fluids such as oxygen, hydrogen, water, carbon dioxide in the critical region. In many cases inadequate attention has been paid to the effects of buoyancy, and results have been presented for forced convection which were in reality obtained under conditions of mixed convection.

A critical review of the experimental and theoretical studies of heat transfer in the critical region is given by Hall *et al.* [1, 2]. Heat-transfer behaviour with fluids in the critical region is influenced by the very rapid variation, with temperature, of the physical and transport properties of the fluid. Of particular interest and practical significance is the role of buoyancy in sufficiently large diameter pipes. The influence of buoyancy in vertical pipe flow has been firmly established by such studies as those of Shitsman [3] and Evans-Lutterodt [4] in which marked divergences were observed in the results obtained with upflow and downflow at otherwise similar conditions. Localised temperature peaks were observed in the axial temperature distribution at sufficiently high heat fluxes in the upflow cases, whereas the downflow results generally showed improved heat transfer relative to the buoyancy free case.

The influence of buoyancy in horizontal flow, on the other hand, has not been properly investigated. When buoyancy is important in horizontal flow, there is both peripheral and axial temperature variation for a uniformly heated pipe. The extent of the peripheral temperature variation depends in part on the material and the thickness of the pipe.

Data on supercritical pressure water flowing in horizontal pipes include those of Shitsman [5] on a 16 mm I.D. pipe, Yamagata *et al.* [6] on 7.5 and 10 mm I.D. pipes, Dickenson and Welch [11] on a 7.62 mm I.D. pipe, Schmidt [7] on 5 and 8 mm I.D. pipes, Vikrev and Lokshin [8] on a 6 mm I.D. pipe. Shitsman's [5] data show significant differences between temperatures at the top and bottom of the tube. The data of Yamagata *et al.* [6] show differences between the top and bottom wall temperatures as the fluid bulk temperature approaches the pseudo-critical value. (For

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supercritical pressure fluids, a pseudo-critical temperature, T_{pc}, is defined at which the specific heat at constant pressure, C_p, attains a maximum value.) Vikrev and Lokshin [8] mention that the heat transfer is poorer at the upper surface than at the lower surface of the 6 mm I.D. horizontal tube without actually indicating the relative magnitudes. The others make no mention of buoyancy effects, or were not in a position to observe any such effects because the thermocouples were not arranged to measure peripheral temperature variation as well as axial temperature variation.

The CO_2 data on horizontal flows include those of Schnurr [9] for a 2.6 mm I.D. pipe and Koppel [10]. Schnurr's data show differences between top and bottom wall temperatures, whereas Koppel does not even state clearly the orientation of the 4.93 mm I.D. pipe used in his experiment nor is any indication given of peripheral temperature variation.

The data reported here were obtained on a 22.14 mm I.D. horizontal pipe under conditions of uniform heat flux. The tests were performed at a pressure of about 7.6 MN/m^2 and inlet temperatures to the test section in the range 10–31°C. The pseudo-critical temperature at 7.586 MN/m^2 is 32°C. (The critical pressure and temperature of CO_2 are 7.286 MN/m^2 and 31°C respectively.) The mass flow rate in the tests was varied between about 0.035 kg/s and 0.15 kg/s, and the heat flux was varied approximately between 5 kW/m^2 and 40 kW/m^2, although the maximum for each test was limited so the test section wall temperature did not exceed 100°C.

DESCRIPTION OF APPARATUS

A centrifugal pump is used to pump the CO_2 round the high pressure loop. The CO_2 flows through a parallel arrangement of pre-heater and pre-cooler which enables the inlet temperature to the test section to be varied. The CO_2 leaves the test section and is passed through an after-cooler in which the CO_2 is cooled to a "liquid" state in order to reduce compressibility effects when using an orifice plate to measure the flow rate. The pressure in the loop is maintained at a desired level by either drawing from a bank of CO_2 supply bottles when the pressure is too low, or letting off some of the CO_2 from the loop via blow off valves when the pressure is too high.

The test section was constructed from a 25.4 mm (1 in) O.D. × 1.63 mm (0.064 in) wall thickness stainless steel tube. The heated length is 2.44 m (approximately 110 diameters), preceded by an unheated length of 1.22 m (55 diameters). It was heated electrically by passing alternating current through the tube. Flexible hoses were used to connect the test section assembly to the rest of the high pressure loop, partly to facilitate the accurate setting up of the test section and also to allow for the thermal expansion of the tube.

The bulk temperature at inlet to the unheated length of the test section is measured by a set of five chromel/alumel thermo-couples immersed in the fluid. In addition, eight thermocouples were resistance welded on to the outer surface of the unheated section of the test section to indicate any changes in fluid bulk temperature prior to entry into the heated section. A total of 196 chromel/alumel thermocouples were welded onto the test section outer surface. Special care was taken to weld each pair of thermocouple wires very close together, and at the same axial location on the tube. A thermocouple is welded every 76.2 mm (3 in) along the tube at angular positions 0, 90, 180 and 270° from the top of the tube, and at 152.4 mm (6 in) intervals along the tube at angular positions of 45, 135, 225 and 315°. The thermal insulation used on the test section consisted of a layer of woven glass tape, which also held the thermocouple wires on to the tube, and about 50 mm thickness of glass wool. Heat losses were measured in a preliminary experiment in which the test section was filled with glass wool. In no case did they exceed 2% of the heat input to the carbon dioxide.

A set of five thermocouples was used to measure the fluid bulk temperature after mixing at the outlet to the test section. The mixing arrangement used initially at inlet and outlet consisted of a series of perforated copper discs. This was proved inadequate by the thermocouples located just after the mixer, which showed substantial temperature variation across the fluid flow. This problem was overcome by adding a suitable length of wire gauze to the mixer.

EXPERIMENTAL RESULTS

The experimental conditions are detailed in Table 1. The parameters varied in the tests are the mass flow rate of carbon dioxide, the inlet bulk temperature, and the wall heat flux; the pattern in which these varied will be clear from the table. The temperature distributions along the heated part of the test section are shown in Figs. 1–5; full lines refer to the top of the pipe and broken lines to the bottom of the pipe. Temperature distributions at other angular locations around the pipe for test number 3.3 are shown in Fig. 6.

Table 1. Experimental conditions

Test code	Mass flow rate (kg/s)	Inlet bulk temperature (°C)	Average heat flux (kW/m^2)	Outlet bulk temperature (°C)	Test pressure (MN/m^2)
1.1	0.151	15.9	5.3	18.1	7.586
1.2	0.148	15.4	15.1	21.3	7.59
1.3	0.146	15.7	26.9	25.6	7.586
2.1	0.0773	14.2	5.2	18.4	7.603
2.2	0.0765	13.8	15.2	24.9	7.607
2.3	0.0771	14.1	21.4	28.1	7.607
3.1	0.080	19.7	5.1	23.2	7.607
3.2	0.0778	20.7	15.4	28.9	7.607
3.3	0.0776	21.1	21.4	30.8	7.607
4.1	0.0771	29.3	5.2	30.7	7.59
4.2	0.0778	29.8	15.5	32.0	7.607
4.3	0.0748	30.1	21.0	32.3	7.61
5.1	0.0402	10.0	5.1	17.8	7.6
5.2	0.0402	10.2	15.3	29.3	7.61
5.3	0.0412	10.7	17.2	30.2	7.614

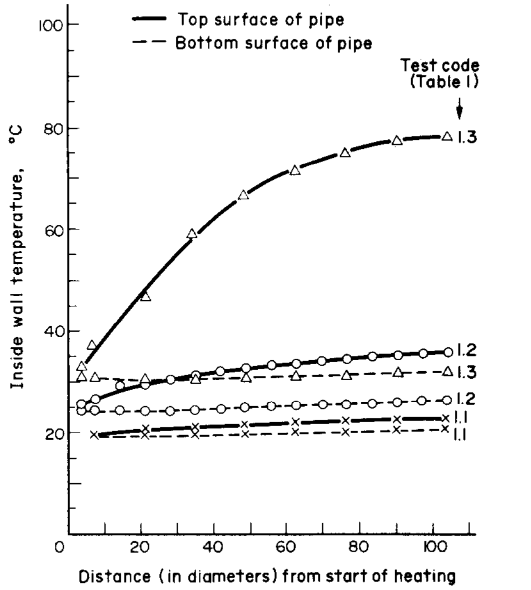

FIG. 1. Experimental temperature distribution along pipe (see Table 1 for experimental conditions).

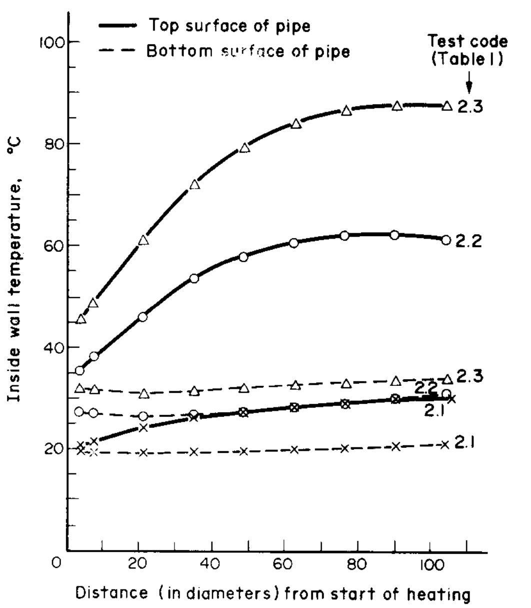

FIG. 2. Experimental temperature distribution along pipe (see Table 1 for experimental conditions).

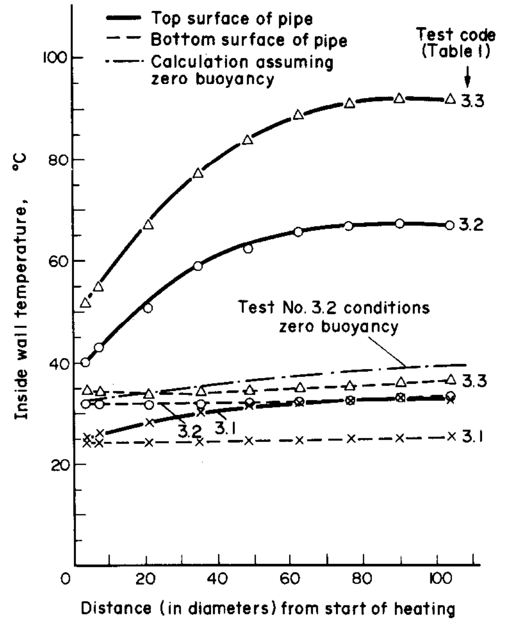

FIG. 3. Experimental temperature distribution along pipe, including a comparison with "buoyancy free" convection (equation 2) in the case of test 3.2 (see Table 1 for experimental conditions).

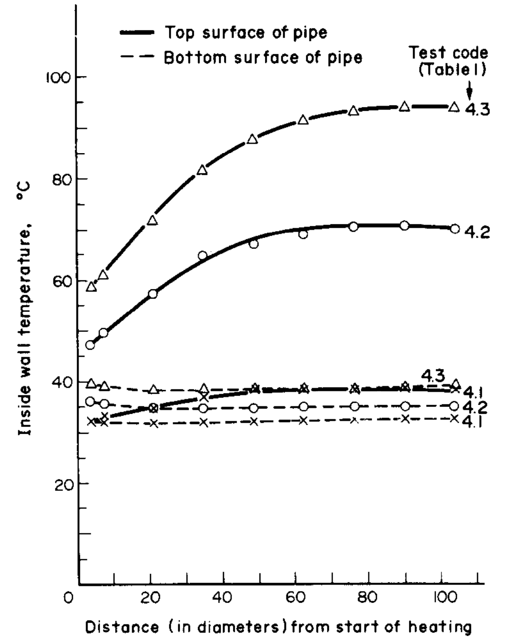

FIG. 4. Experimental temperature distribution along pipe (see Table 1 for experimental conditions).

The period of time required to scan the thermocouples measuring the pipe wall temperatures was about 5 min, and it was therefore necessary to ensure that steady conditions had been achieved. The practice was adopted of first rapidly scanning about 10 selected thermocouples along the length of the test section before making the detailed scan. A comparison of measurements on the same set of thermocouples in both groups was taken as an indication of the level of steadiness attained in the system. For all the results presented the difference between the quick scan temperature measurements and those in the detailed scan was well below 0.5°C.

The estimated errors in the measurements are: test pressure ± 0.03 bar; mass flow rate, $\pm 1.5\%$; heat flux, $\pm 3\%$; bulk inlet temperature, ± 0.5°C; tube wall temperature, ± 1°C.

DISCUSSION

Comparison with available data

There are no comparable data for CO_2 in which the effect of buoyancy in horizontal flow is systematically investigated. A qualitative comparison with the data of Shitsman [5] and Miropolskiy *et al.* [12] for supercritical pressure flow of water in horizontal pipes shows similar trends to the results of the present work, in that the temperature along the bottom of the tube is considerably lower than at the top of the tube.

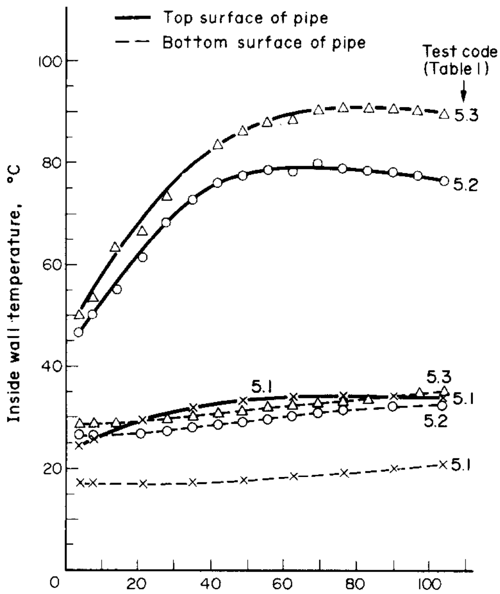

FIG. 5. Experimental temperature distribution along pipe (see Table 1 for experimental conditions).

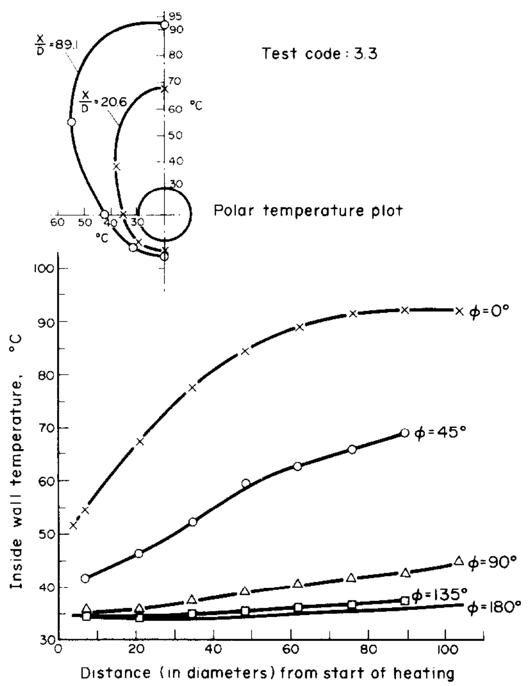

FIG. 6. Experimental axial and circumferential temperature distributions for test 3.3 (see Table 1 for experimental conditions).

Figure 7 gives a comparison between the results of Weinberg [13] for the vertical flow of supercritical pressure CO_2 and results obtained under approximately similar conditions with horizontal flow. The test section internal diameter is 19 mm in Weinberg's tests, which is slightly less than the internal diameter of the present test section. The results shown in Fig. 7 correspond to roughly the same bulk inlet temperature of 20°C and the mass flow rates indicated on the figure. Curve V_1, which is a sample of Weinberg's results, has approximately the same $(q_w \times D)$ and $(\dot{m} \times D)$ as the

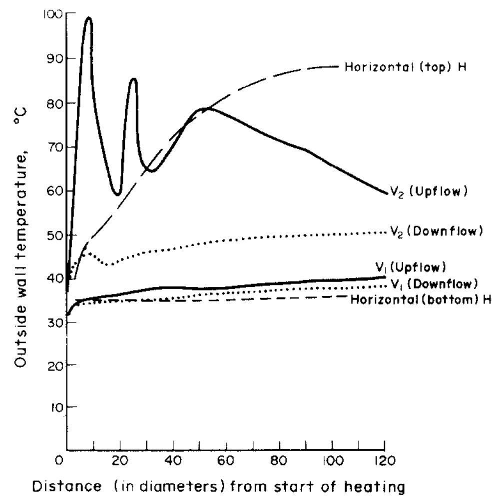

FIG. 7. Comparison of axial temperature distributions for horizontal and vertical flow. Experimental conditions as follows:

Curve	Source of data	Inlet temp. (°C)	Mass flow $\dot{m}$(kg/s)	Pipe I.D. D(m)	Heat flux q_w(kW/m^2)	$\dot{m} \times D$ (kg/s m)	$q_w \times D$ (kW/m)
V_1	Weinberg [13]	20	0.124	0.019	30	0.00238	0.570
V_2	Weinberg [13]	20	0.124	0.019	50	0.00238	0.950
H	This work	20.3	0.121	0.022	26.8	0.00270	0.596

Curves labelled V refer to vertical (upward and downward) flow and curves labelled H refer to horizontal flow.

horizontal flow result marked as curve H in Fig. 7. (Given the same inlet temperature, the wall temperature distribution along the pipe should be similar, in the absence of buoyancy effects, provided that the Reynolds number is the same and that the temperature distribution across the pipe has the same shape; this requires that the products $\dot{m} \times D$ and $q_w \times D$ should be the same.) The heat transfer is considerably worse along the top surface in horizontal flow than in the comparable vertical upflow and downflow. A second curve, V_2, is shown in Fig. 7 which exhibits the characteristic buoyancy peaks normally obtained at sufficiently high heat flux in vertical upflow.

The above comparison with vertical flow indicates that a serious reduction in heat transfer at the top of the tube with horizontal flow occurs at a lower heat flux than that required to induce buoyancy peaks with vertically upward flow.

Comparison with data obtained in the absence of buoyancy forces

In the case of a vertical tube it is possible to define a fairly precise criterion for the absence of buoyancy effects [14]; this is

$$Gr_b \, Re_b^{-2.7} < 5 \times 10^{-6}. \tag{1}$$

For upward flow, buoyancy peaks of the type shown in Fig. 7 (curve V_2) occur when this condition is not satisfied. Corresponding conditions for downward flow lead to a generally improved heat transfer when compared to conditions in which buoyancy effects are absent. Jackson and Fewster [15] have correlated data for water and CO_2 in which buoyancy effects are not present, obtaining the expression:

$$Nu_b = 0.0183\, Re_b^{0.82}\, Pr_b^{0.5} \left(\frac{\rho_w}{\rho_b}\right)^{0.5} \left(\frac{\overline{C}_p}{C_{p_b}}\right)^{0.4}. \quad (2)$$

This expression has been applied to the results shown in Table 1 and Figs. 1–5, and in all cases gives a wall temperature distribution that lies between the experimental distribution for the top and the bottom of the test section. This is illustrated in Fig. 3, when the zero buoyancy curve for test 3.2 is indicated by a chain-dotted line. Wall temperatures at $x/D = 100$ for zero buoyancy are compared with the experimental values for all the tests in Table 2. It thus appears that buoyancy in horizontal flow results not only in a lower heat transfer at the upper surface, but also an increased heat transfer at the lower surface. It should be noted that these results refer to a uniform wall heat flux boundary condition.

Table 2. Comparison between measured and calculated wall temperatures at $x/D = 100$. (Calculated wall temperature assumes zero buoyancy effects T_b calculated by heat balance)

Test code	Re_b ($\times 10^{-5}$)	Experimental temperatures T_b (°C)	T_w (Top) (°C)	T_w (Bottom) (°C)	Calc. T_w [Zero buoyancy equation (2)] (°C)
1.1	1.04	18.0	22.5	20.5	21.1
1.2	1.13	21.1	36.0	26.0	29.0
1.3	1.31	25.1	78.0	32.0	37.7
2.1	0.54	18.2	30.0	21.0	23.2
2.2	0.67	24.2	62.0	31.0	35.6
2.3	0.77	27.4	88.0	34.0	46.3
3.1	0.65	23.0	33.0	25.0	27.4
3.2	0.81	28.5	67.0	33.5	39.2
3.3	0.88	30.3	92.0	36.5	47.9
4.1	0.87	30.6	38.0	31.0	32.6
4.2	1.04	31.9	70.5	35.0	43.9
4.3	1.13	32.2	94.0	39.0	52.8
5.1	0.27	17.4	34.0	20.5	26.2
5.2	0.42	28.3	77.0	32.5	53.7
5.3	0.45	29.2	90.0	35.0	57.8

Criteria for buoyancy effects in horizontal flow

Jackson [16] has proposed a criterion for the absence of buoyancy effects in horizontal flow in the form:

$$Gr_b\, Re_b^{-2} \left(\frac{\rho_b}{\rho_w}\right)\left(\frac{x}{D}\right)^2 < 10. \quad (3)$$

It is believed that this criterion may be somewhat conservative, and that only rather small effects may be present for values of the above parameter up to about 50. The value of this parameter corresponding to all the present experimental conditions at $x/D = 100$ is greater than 1000, except for test 1.1 in which it is about 400. It is noteworthy that test 1.1 gives the least deterioration in heat transfer (about 0.7 of the zero buoyancy result) at $x/D = 100$. The results are therefore not inconsistent with the above criterion, although they cannot be regarded as a stringent test of it.

Petukhov *et al.* [17] have proposed a criterion for the buoyancy free region with horizontal flow of the form:

$$Gr_b^*\, Re_b^{-2.75}\, Pr_b^{-0.5} [1 + 2.4\, Re_b^{-1/8} (Pr_b^{2/3} - 1)]^{-1} < 3 \times 10^{-5}. \quad (4)$$

The present results in all cases exceed this value by at least a factor 30, so that they do not constitute a test of the criterion.

CONCLUSIONS

The experimental results for flow in a horizontal pipe heated with a uniform wall heat flux show a marked deterioration in heat transfer at the upper part of the pipe and an improvement at the lower part, when compared with data in which buoyancy effects are negligible.

Criteria proposed by Jackson [16] and by Petukhov [17] both predict that buoyancy should be important under the conditions of the experiments, and there is limited support for the former in that the one experiment which gave a rather small deterioration in heat transfer was that having the smallest value of the parameter in the inequality [3].

It might be noted that the criterion [3] is intended as an indication of the conditions under which buoyancy effects just begin to occur; clearly an effect of x/D would be expected in this case. Petukhov's criterion, on the other hand, is based on a "fully developed" solution of the mixed convection problem, and if property variations along the pipe are ignored this is independent of x/D. It might be expected that a criterion of the form of (4) would give an indication of the magnitude of the "fully developed" deterioration in heat transfer; however an attempt to correlate the results in these terms did not prove effective.

In many of the experiments it was clear that the buoyancy effect was still developing after 100 diameters of heated length, although there was some indication that development was complete and the heat-transfer coefficient had stabilised in the case of the lower flows and higher heat inputs. It has been suggested [11] that abnormally long thermal entry lengths are a characteristic of supercritical pressure fluids; however, they do not usually occur with vertical flow and the present results suggest that there may have been some confusion between the usual entry length and the length required to develop the temperature distribution characteristic of buoyancy effects.

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ETUDE EXPERIMENTALE DE TRANSFERT THERMIQUE AU BIOXYDE DE CARBONE A LA PRESSION SUPERCRITIQUE DANS UN TUBE HORIZONTAL

Résumé—On présente les résultats d'une étude expérimentale de transfert de chaleur à un écoulement de CO_2 aux pressions supercritiques et subcritiques dans un tube de diamètre intérieur égal à 22,14 mm, soumis à un flux thermique uniforme. Le travail expérimental recouvre une zone de nombres de Reynolds à l'admission allant de $2 \cdot 10^4$ à $2 \cdot 10^5$ environ. Des variations importantes de température sur la périphérie ont été obtenues sous l'effet de la convection naturelle. De la comparaison avec les données relatives à la convection libre pure, il ressort que la convection naturelle accroit le transfert thermique à la base du tube et le réduit à son sommet. Le critère proposé par Jackson [16] et par Petukhov [17] indique que l'effet d'Archimède doit se manifester dans les conditions correspondant à l'ensemble des expériences.

EXPERIMENTELLE UNTERSUCHUNG DES WÄRMEÜBERGANGS AN KOHLENDIOXID IN HORIZONTALEN ROHREN BEI ÜBERKRITISCHEN DRÜCKEN

Zusammenfassung—Es werden die Ergebnisse einer experimentellen Untersuchung des Wärmeübergangs an CO_2, das bei über- und unterkritischen Drücken durch ein gleichförmig beheiztes horizontales Rohr mit einem Innendurchmesser von 22,14 mm fließt, vorgestellt. Die Untersuchung umfaßt den Bereich der Reynolds-Zahlen von etwa 2×10^4 bis 2×10^5 am Eintritt des Rohres. Es wurden deutliche Temperaturänderungen über den Umfang des Rohres festgestellt, was den Einfluß des Auftriebs wiedergibt. Ein Vergleich mit Daten, die für den auftriebsfreien Fall ermittelt worden sind, zeigt, daß der Wärmeübergang durch den Auftrieb an der Unterseite des Rohres verbessert und an der Oberseite verschlechtert wird. Die von Jackson [16] und Petukhov [17] vorgeschlagenen Kriterien zeigen, daß Auftriebseffekte bei den bei diesen Experimenten zugrunde gelegenen Bedingungen zu erwarten waren.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ПЕРЕНОСА ТЕПЛА К ДВУОКИСИ УГЛЕРОДА В ГОРИЗОНТАЛЬНОЙ ТРУБЕ ПОД СВЕРХКРИТИЧЕСКИМ ДАВЛЕНИЕМ

Аннотация — Представлены результаты экспериментального исследования переноса тепла к потоку двуокиси углерода при сверх- и докритических давлениях в равномерно нагреваемой горизонтальной трубе с внутренним диаметром 22,14 мм. Диапазон значений числа Рейнольдса на входе составлял $2{,}10^4$–$2{,}10^5$. Получены значительные перепады температуры в периферической области, свидетельствующие о влиянии свободной конвекции. Сравнение с результатами, полученными при отсутствии конвекции, показывает, что в нижней части трубы благодаря конвекции происходит усиление теплообмена, а в верхней части — уменьшение. Предложенный Джексоном [16] и Петуховым [17] критерий показывает, что эффекта свободной конвекции следует ожидать в условиях всех экспериментов.