Int. J. Heat Mass Transfer. Vol. 16, pp. 257–270. Pergamon Press 1973. Printed in Great Britain

THERMO-ACOUSTIC OSCILLATIONS IN FORCED CONVECTION HEAT TRANSFER TO SUPERCRITICAL PRESSURE WATER

E. STEWART, P. STEWART and A. WATSON

Simon Engineering Labs., University of Manchester, Manchester, England

(*Received* 7 *June* 1971 *and in revised form* 27 *April* 1972)

Abstract—Heat-transfer measurements are presented for the case of supercritical pressure water flowing through horizontal small bore tubes under conditions of uniform heat flux. The experiments were designed to produce the high frequency oscillations which had been previously observed to occur spontaneously in heat transfer to supercritical pressure hydrogen and to supercritical pressure water. Oscillation frequencies were measured and are shown to be associated with pressure oscillations in the test section resulting from a standing pressure wave between entry and exit. Several modes of standing wave were identified. Detailed wall temperature distributions were measured and heat-transfer coefficients deduced. The heat-transfer coefficient distribution along the test section is seen to have a variation which shows that it is strongly influenced by local pressure fluctuations.

NOMENCLATURE

A, constant;
B, constant defining variation of c with z;
E, F, constants;
I, electrical current flow per unit cross sectional area of tube wall;
L, total length of test section between mixing chambers;
L_h, heated length of test section;
P, pressure;
R, electrical resistivity of tube wall;
Re, Reynolds number;
T, temperature;
V, axial velocity;
a, b, location of exit and entry on z axis;
c, sound velocity;
d, test section bore;
f, frequency;
k, thermal conductivity of tube;
m, mass flow in test section;
n, mode of oscillation;
p, amplitude of pressure fluctuations, $P' = pe^{i\omega t}$;
q, heat flux;
r, radius;
t, time;
z, axial coordinate;
α, heat-transfer coefficient;
Δh, defined by equation (4);
γ, defined by equation (4);
ρ, density;
ω, angular frequency.

Superscripts
$'$, fluctuating;
$*$, reference value.

Subscripts
b, bulk fluid;
c, critical;
h, refers to heated length;
in, inlet condition;
0, steady state value;

pc, pseudo-critical;
w, tube inner wall;
s, constant entropy;
n, refers to mode.

INTRODUCTION

AN INCREASING amount of effort has, in recent years, been applied to understanding heat transfer to fluids in the region of their thermodynamic critical point. This has resulted mainly from attempts to collect design data for applications such as superconducting cables (employing helium), rocket motors (hydrogen and hydrocarbon fuels) and electricity generating plant (water). In each of these applications the fluid used is at supercritical pressure and transfers heat at temperatures in the region of its pseudo-critical temperature. Here, both thermodynamic and transport properties vary sharply with temperature, and the equations governing heat transfer and fluid flow are in their most complicated form. The early experimental work was often correlated in the same manner as constant property heat transfer, resulting, as the surveys by Hall, Jackson and Watson [1] and Hendricks and Simoneau [2] show, in the appearance of apparently contradictory sets of data. More recently, experiments have been designed to demonstrate how such contradictions arise.

Fluctuations of pressure and temperature over a wide range of frequencies have been a familiar feature of many of the experimental investigations. Particularly in some forced convection investigations in pipes the pressure fluctuations were of high intensity, sufficient to communicate themselves as a clearly audible sound—"whistling"—to bystanders. Relatively few experiments had been carried out with the aim of defining the nature of the oscillations at the time when the work reported here was begun, and this was the object of the experiments.

LITERATURE SURVEY

Although this paper is primarily concerned with forced convection it is interesting to consider also the oscillations occurring in natural circulation loops such as that used by Schmidt, Eckert and Grigull [3] containing near critical ammonia. Cornelius and Parker [4] performed such a set of experiments which, although preceded by others [5, 6] is particularly relevant. The flow circuit was a simple one in which slightly supercritical Freon 114 was circulated by heating and cooling in different sections. The test section was a straight, heated length of vertical pipe. Forced circulation by pump was also possible. Oscillations of 5–30 Hz were observed in pressure when $T_b < T_{pc}$ and $T_w > T_{pc}$. The oscillations were sinusoidal and of discrete frequency, and it was deduced that they were the result of a pressure wave travelling around the circuit (length l) with sonic velocity. The frequency of the oscillations was thus $f = c/l$. where c varied around the circuit with fluid temperature. A series of experiments was carried out for constant l in which the bulk temperature level of the fluid in the loop was varied, being kept always below T_{pc}. The change in f with T_b which resulted followed a curve similar to the variation of c with temperature. Further experiments in which l was varied consolidated the idea of a standing wave in the loop. By introducing a restriction in the flow circuit, Cornelius and Parker found that the oscillations could be prevented, presumably because of the uncoupling of the path taken by the wave. Using a test section of much smaller bore than the rest of the circuit produced a higher frequency range of oscillations under the same temperature conditions. This was analyzed by considering the test section as a pipe in a state of resonance with either end open, and the agreement obtained with measured frequencies suggested that this was a satisfactory interpretation. Wall temperature distributions varied in shape between different oscillatory states and localised hot regions were observed in some cases. Chemical deposits on the pipe wall were mentioned as a possible cause. In other cases a general lowering of wall temperature level accompanied the presence of oscillations. Another range of frequencies of the order 0·1–0·5 Hz occurred in the loop in natural convection experiments in which the bulk temperature at test section outlet approached T_{pc}. Associated with pressure oscillations were fluctuations in flow rate, fluid temperature and wall temperature of the same frequency. The authors explained these phenomena as the effect of large specific volume changes in the pseudocritical temperature region.

Oscillations have occurred in many supercritical pressure fluids in purely forced convection experiments. Goldman [7] reported experiments with turbulent flow of supercritical pressure water ($1{\cdot}24 \leqslant P/P_c \leqslant 2{\cdot}03$) in small bore tubes, and observed frequencies in the region of 2 kHz. For the conditions of the experiments the bulk temperature was always less than T_{pc} whilst the wall temperature was above T_{pc}. Oscillations occurred only at a high heat flux level, higher than 3·15 MW/m^2, the level depending upon inlet bulk temperature and mass flow rate. Wall temperature measurements were presented for one "oscillating" run in which temperatures near the outlet end of the tube dropped below their "non-oscillating" level indicating an increase in heat transfer coefficient. Hines and Wolf [8] using hydrocarbon rocket fuel at $P/P_c = 2$ flowing turbulently in small bore tubes also found that the presence of high frequency oscillations improved the heat transfer. Constant flux heating up to the extremely high level of 15 MW/m^2 was used. Although heat transfer was improved by the oscillations the pressure fluctuations reached large amplitude in the test section resulting in tube failure in one case. The frequency of the oscillations was in most cases much higher than the value presented as the fundamental frequency for "open-open" pipes, however, the paper is not clear regarding the way in which the fundamental was calculated.

In 1964, Bishop, Sandberg and Tong [9] reported experiments with water at $1{\cdot}03 \leqslant P/P_c \leqslant 1{\cdot}25$ in turbulent forced convection in small bore test sections. Heat fluxes up to 4·8 MW/m^2 were used. No general conclusions regarding the oscillations were reached. As with the experiments of Hines and Wolf [8] the fundamental pipe resonance frequency was lower than the frequencies of the sound heard (1–10 kHz) but contrary to the latter and Goldman's [7] report, heat transfer deteriorated locally when the oscillation occurred.

Experiments with hydrogen revealed that oscillations can occur over a very wide range of frequencies. The most relevant sets are those carried out by Thurston [10] and by Thurston, Rogers and Skogland [11]. These were experiments in which hydrogen at up to $P/P_c = 1{\cdot}4$ was discharged through a small bore tube. In the earlier reference, thermal boundary conditions were rather ill-defined, but in the other, a constant heat flux boundary condition was maintained. The modes of oscillation were identified as "Helmholtz" and "open-open" pipe resonance. Variation in frequency of the pipe resonance with changes in bulk fluid temperature were noticed in the experiments and allowed for in supporting calculations by evaluating the frequency of pipe modes numerically for each experimental run, supplying the appropriate variation of bulk temperature along the test section. It was found that oscillations only occurred when the wall temperature was above T_{pc} and bulk temperature below, in line with previous experiments. The lowest pressure used in these experiments was $P/P_c = 0{\cdot}6$ and oscillations were observed at this subcritical pressure also.

There have been a number of occurrences of audio-frequency oscillations in two-phase forced convection heat transfer to water. These occur particularly in film boiling experiments, as reviewed by Goldman [12] for example and also Kafengauz and Fedorov [13]. Film boiling certainly has one of the principal features of the supercritical pressure experiments in which oscillations occur, a dense "liquid" core separated from the tube wall by a "gaseous" layer, but surface tension is an added complication.

For the present experiments it was decided to operate under conditions similar to those of Bishop, Sandberg and Tong [9], and to apply more effort to making detailed observations of oscillations.

EXPERIMENTAL ARRANGEMENT

A double-acting ram pump driven by compressed air accepted water from a storage tank and supplied it through the circuit of Fig. 1 at pressures up to 300 bars. Nominal working pressure in the experiments reported here was 250 bars and this was maintained to about ± 2 bars. The pump delivery was at ambient temperature and was used to cool the return water from the test section in a simple heat exchanger. At the outlet from this the delivery water temperature was typically 200°C. This was increased in a preheater to the required test section bulk inlet temperature. Both preheater and test section were horizontal tubes heated directly by the passage of alternating current (50 c/s). The test section was connected to the main loop by means of flanges with mixing chambers inside as shown in Fig. 1. Water from the test section outlet flowed through the heat exchanger and was further cooled in stages to 20°C before being reduced in pressure to atmospheric. Flow measurement was then made with a rotameter. The whole of the pressurised circuit was made from stainless steel (AISI 321). The purity of the water was maintained by a separate circuit; the pH value was kept slightly above 7 and the water was degassed by boiling before being stored under a Nitrogen blanket. The electrical power inputs to preheater and test section were measured using current transformers and voltage tappings. In order to keep the two circuits separate, thus avoiding heat generation in the heat exchanger, an electrical break in the form of a PTFE gasket was inserted in the heat exchanger flange. Temperature measurements were made with chromel–alumel thermocouples and a digital voltmeter, the two wires being welded side by side to the stainless steel tubes. Two test sections were used. The smaller was of 1·524 mm bore and 0·203 m heated length and the larger of 3·1 mm bore and 0·61 m heated length. Both had thermocouples

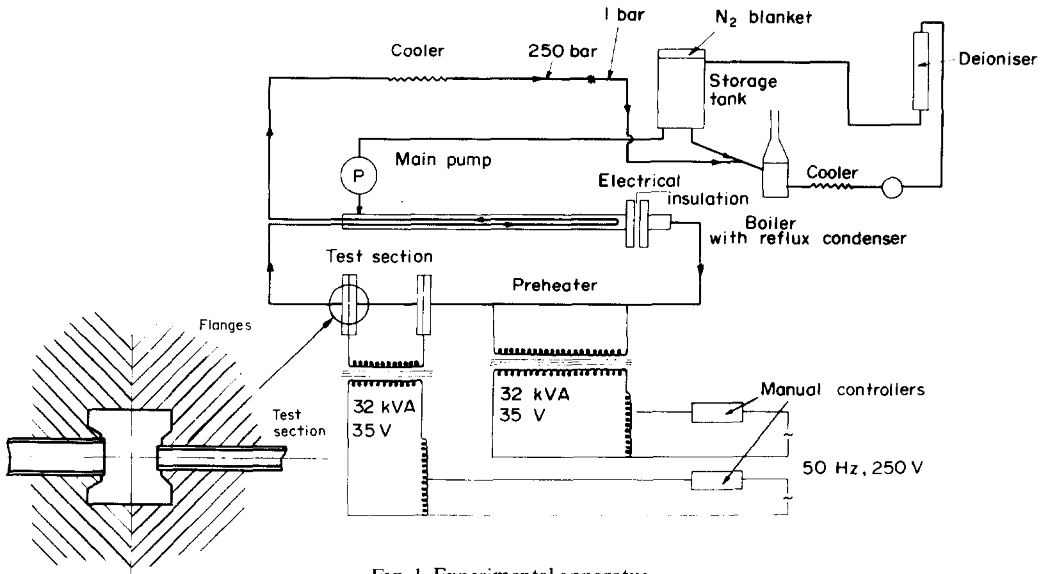

FIG. 1. Experimental apparatus.

uniformly spaced at 12·5 mm intervals. Bulk inlet and outlet fluid temperatures were deduced by measuring wall temperature distributions along the unheated, thermally insulated tube immediately before and after the test section.

Sound frequency measurements were made by means of a microphone suspended above the test section. Since conditions were stable in time, frequency analysis of the microphone output was carried out by a frequency analyser using a manual scan. Pressure fluctuations were measured in the fluid after the heat exchanger, as no transducer capable of withstanding temperature conditions in the test section could be obtained.

REDUCTION OF EXPERIMENTAL DATA

Because of the thickness of the test section tube walls, the temperature difference across the wall was of the same order of magnitude as the difference between inside surface temperature of the tube and bulk temperature. An additional feature was the variation of thermal conductivity and electrical resistivity with temperature which were significant at the high temperatures involved. Calculation of the inside surface temperature was undertaken by solving the steady state conduction equation with temperature dependent properties.

$$0 = \frac{1}{r}\frac{\mathrm{d}}{\mathrm{d}r}\left(kr\frac{\mathrm{d}T}{\mathrm{d}r}\right) + I^2R. \qquad (1)$$

The outer surface of the tube was well insulated thermally and was assumed to be adiabatic and at the temperature indicated by the thermocouples. Current density was assumed to be independent of radius. Local heat flux, which varied along the test section by up to 10 per cent of the mean, was also determined. Local bulk temperature was determined from a knowledge of inlet temperature, heat flux distribution and mass flow rate, taking into consideration the temperature variation of specific heat. Hence local heat transfer coefficient was found. Uncertainty in local heat transfer coefficient away from the tube ends was estimated to be about ± 10 per cent maximum. Errors in local heat flux determination resulting from the neglect of axial conduction terms in equation (1) were of order 2 per cent at most.

EXPERIMENTAL RESULTS

In discussing the results mean heat fluxes are quoted.

(a) *Small test section*

The range of conditions for the small test section experiments is shown in Table 1. The procedure adopted was to fix mass flow and inlet temperature to the test section and examine the influence of heat flux on wall temperature distribution and sound production. The behaviour with the lowest mass flow rate and an inlet bulk temperature of 290°C demonstrates the principal effects observed and is described first.

Heat flux was increased in steps up to $q = 2·96$ MW/m^2 before any oscillations were heard. The change in shape which the wall temperatures underwent in passing this threshold flux is shown in Fig. 2. The microphone output indicated a single frequency of 1926 Hz. Further increase in q to 3·85 MW/m^2 (Fig. 2 curve (c)) changed this frequency steadily to 1863 Hz and an additional frequency of 2700 Hz was found to be present at this stage. When the heat flux was decreased to $q = 3·16$ MW/m^2 the microphone output changed in a step to a single frequency of 1100 Hz. There was in fact a pause marking this step, when no audible sound was produced. The wall temperature distribution was then that of Fig. 2 (d). Further reduction to $q = 2·17$ MW/m^2, well below the threshold necessary to initiate oscillations, did not cause oscillations to cease. The frequency was then 1203 Hz and the wall temperature distribution was fairly uniform. The heat flux was again increased in steps to $q = 4·05$ MW/m^2. Between 3·1 MW/m^2 and 3·3 MW/m^2 the microphone output changed frequency from 1100 Hz to 2033 Hz and the wall temperature distributions showed a corresponding change

Table 1. Range of parameters covered in small test section experiments

Mass flow (g/s)	Test section inlet temperature (±5°C) (°C)	Maximum test section outlet temperature (°C)	Bulk inlet Reynolds number
	290	381	$4{\cdot}80 \times 10^4$
5·65	315	380	$5{\cdot}35 \times 10^4$
	335	382	$5{\cdot}95 \times 10^4$
	290	376	$5{\cdot}90 \times 10^4$
6·94	315	379	$6{\cdot}5 \times 10^4$
	340	382	$7{\cdot}12 \times 10^4$
	290	369	$7{\cdot}05 \times 10^4$
8·2	315	379	$7{\cdot}6 \times 10^4$
	340	382	$8{\cdot}55 \times 10^4$
	290	366	$7{\cdot}35 \times 10^4$
9·5	315	375	$8{\cdot}04 \times 10^4$
	340	379	$9{\cdot}05 \times 10^4$

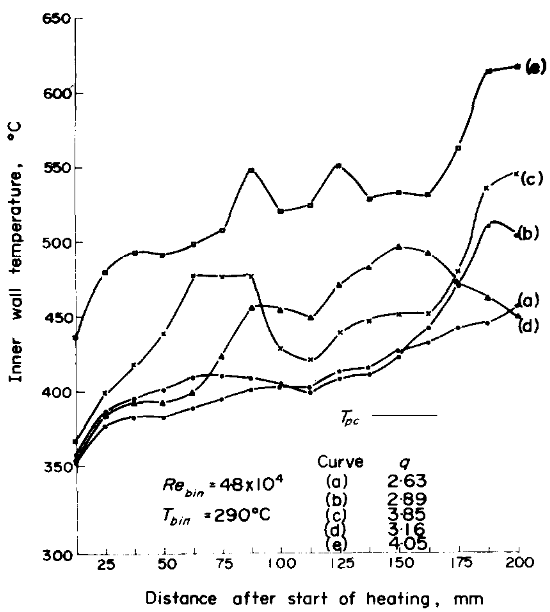

FIG. 2. Wall temperature distribution in the presence of oscillations.

back to the form of Fig. 2(b) and 2(c). Above $q = 3{\cdot}85$ MW/m^2 several discrete frequencies were present, with predominant peaks in the spectrum at 2750 Hz and 1840 Hz; a typical wall temperature distribution is shown as Fig. 2(e). Reducing heat flux from this high level to zero did not introduce any changes significantly different from those described above.

The same general pattern of events took place at the other mass flow rates for the lowest inlet bulk temperatures. Figure 3 shows the variation of the threshold heat flux with mass flow rate

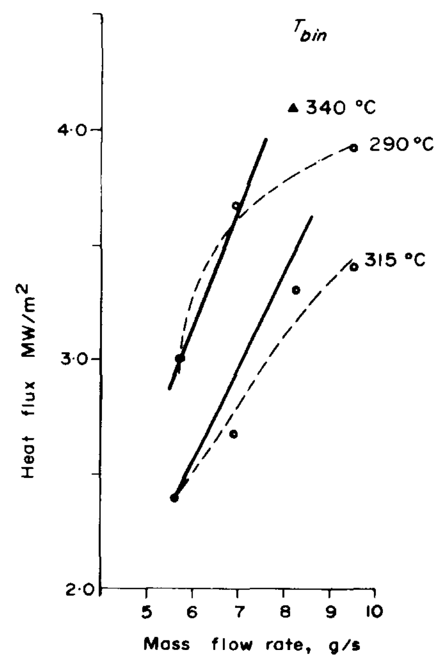

FIG. 3. Variation of heat flux at onset of oscillations with mass flow rate.

and inlet bulk temperature. The threshold increased as mass flow increased for fixed bulk inlet temperature. Once oscillations had begun they changed in frequency as the heat flux was varied, either gradually or in well defined large steps at some heat flux levels. Figure 4 shows a plot of frequency versus mean bulk temperature along the test section. The points fall into three distinct groups each of which has a linear trend with mean bulk temperature. In general the frequency of oscillations at their onset lay in the middle group of Fig. 4, i.e. between 1700 and 2500 Hz. An exception to this was the run at a mass flow of 8·2 g/s with an inlet temperature of 340°C, in which whistling occurred at a heat flux of 4·1 MW/m^2 with a frequency lying in the highest group, of 2780 Hz. This threshold flux is shown as a single point in Fig. 3. No oscillations occurred for the other three mass flows at such high bulk fluid inlet temperatures although the heat flux was increased to between 3·1 and 3·7 MW/m^2 in all cases. This suggests that the trend of decreasing threshold flux with increasing inlet bulk temperature suggested by Fig. 3 does not continue as T_{bin} approaches T_{pc}.

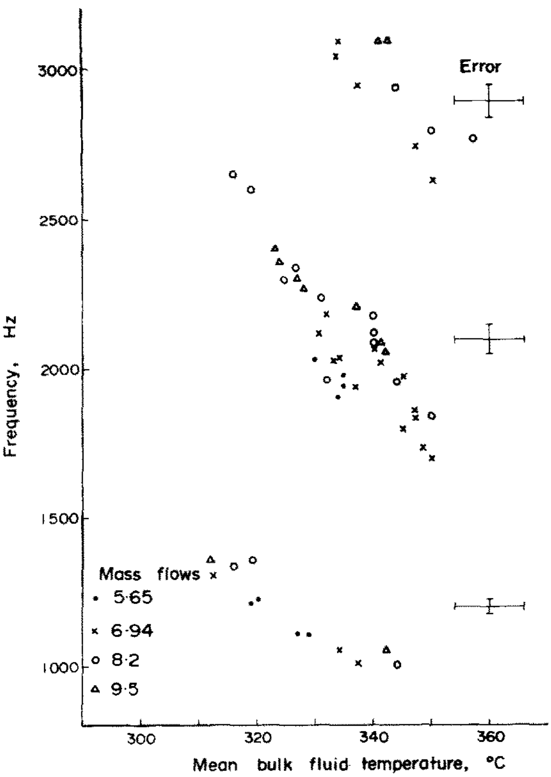

FIG. 4. Dependence of frequency on bulk fluid temperature.

No clear heat flux data can be presented to to mark the step changes in frequency between the groups of points in Fig. 4, because of the tendency for frequencies at all three levels to occur simultaneously once oscillations had begun and the heat flux level had been changed from the threshold. At the highest mass flow rate a frequency of order 5000 Hz was observed. In some cases at the highest heat flux levels the oscillations were sinusoidal of high frequency ($\sim$3000 Hz) but were amplitude modulated in a non-sinusoidal manner with a period of $\sim$2 s. Wall temperatures were observed to fluctuate with the same period, especially in the vicinity of wall temperature peaks. The occurrence of oscillations in the lowest frequency group was in some cases in conjunction with higher frequencies, but sometimes as a single frequency when the heat flux was taken below the threshold level after oscillations had begun. This latter occurred on runs with the lowest bulk inlet temperature.

Wall temperature distributions changed in the manner of Fig. 2 for all runs in which oscillations occurred; further examples are shown in Fig. 5. Often the initiation of oscillations did not produce an immediate change in wall temperature distribution but, as shown in Figs. 5(b) and (c), followed a flux increase (no detailed record of the amplitude of the oscillations is available to compare the amplitudes of cases 5(b) and (c)). Likewise, the temperature distributions did not always respond immediately to the step changes in frequency which took place, but did so after a pause of up to several minutes.

(b) *Large test section*

Experiments were carried out with bulk fluid inlet temperature ranging from 100 to 290°C and inlet Reynolds numbers from $2{\cdot}2 \times 10^4$ to $7{\cdot}48 \times 10^4$. Heat fluxes up to 3·1 MW/m^2 were applied, giving maximum bulk fluid outlet temperatures of about 384°C i.e. very close to

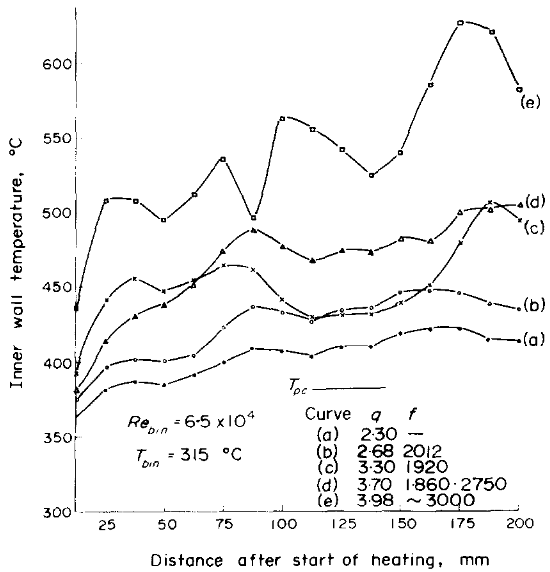

FIG. 5. Wall temperature distribution in the presence of oscillations.

T_{pc}. Oscillations were observed for one case only, when $T_{bin} = 100$°C and $Re = 2{\cdot}2 \times 10^4$, with $f = 1000$ Hz. The heat flux was 3·07 MW/m^2 and bulk fluid outlet temperature was 363°C. Tube wall temperature was above T_{pc} over the the downstream half of the tube.

DISCUSSION

Nature of the oscillations

In a cylindrical geometry there are essentially two uncoupled modes of vibration, radial and longitudinal. There are further, two possible media which might act as a vehicle for such modes, the metal pipe or the fluid within. Calculations eliminate the pipe wall radial and longitudinal modes, and radial modes in the fluid are orders of magnitude higher in frequency than the oscillations observed. The construction of the test sections used in the present work is such that they could encourage "open-open" pipe resonance, with a node at each end (Fig. 1). If the bulk temperature of the core of dense fluid was to remain constant with z, the resonant frequencies of such an oscillation would be given by

$$f = \frac{nc}{2\pi L} \text{where} \quad n = 1, 2, 3.... \tag{2}$$

$n = 1$ indicates a half wavelength filling the tube, $n = 2$ a full wavelength with a node at the mid-point, etc. The effect of changing the level of temperature of the fluid core would be proportional to $\mathrm{d}f/\mathrm{d}c = (n/2\pi L)$ since c is a function of temperature, that is, proportional to n. This accounts for the slope of the three lines through the points in Fig. 4. The central group is for $n = 2$ and has a slope twice that of the lower group which is for $n = 1$. In fact bulk fluid temperature varies axially, the velocity of sound therefore varies axially and equation (2) is not the correct solution for this case.

If it is assumed that the wave motions produce small perturbations of the fluid pressure, that the bulk fluid motion and viscous effects are not important to the wave motion and that pressure fluctuations in the fluid occur adiabatically, then we have from first order perturbation analysis the following wave equation:

$$\frac{\mathrm{d}^2 p}{\mathrm{d}z^2} + \frac{\omega^2 p}{c^2} = 0 \tag{3}$$

where p is pressure fluctuation amplitude, c^2 may be a function of z, and ω is the wave frequency.

A linear variation of c with distance along the test section was found to be a reasonable assumption after considering the fluid properties and the variation of bulk fluid temperature. Thus $c = Bz$ may be introduced and the test section entry and exit considered to be at $z = a$ and $z = b$ respectively. We require then the solution of the equation:

$$z^2 \frac{\mathrm{d}^2 p}{\mathrm{d}z^2} + \left(\frac{\omega}{B}\right)^2 p = 0.$$

By putting $z = e^t$ the above equation may be reduced to one with linear coefficients and we finally obtain:

$$p = (\sqrt{z})(Ez^{\gamma} + Fz^{-\gamma}) \quad (4)$$

where

$$\gamma = \tfrac{1}{2}\sqrt{[1 - 4(\omega/B)^2]}.$$

The boundary conditions for an open ended pipe are, at $z = a$ and $z = b$, $p = 0$, and give the following:

$$f_n = \frac{B}{4\pi}\sqrt{\left[1 + \left(\frac{2n}{\ln(b/a)}\right)^2\right]} \quad n = 1, 2, 3 \ldots \quad (5)$$

f_n was evaluated for several experimental runs and it was found that the calculated frequencies agreed with those observed to within 10–15 per cent.

Relationship between wall temperature distribution and frequency

Figures 2 and 5 show that the presence and frequency of oscillations influence the wall temperature distribution. Interpretation of these plots is complicated by the fact that one is dealing with heat transfer in the presence of highly variable properties, so that even in the absence of oscillations the heat-transfer coefficient varies along the test section, and with Re and T_{bin}, in a manner which does not comply with existing forced convection correlations and cannot yet be predicted. Some of this variation can be removed by plotting on a normalized basis, as Fig. 6, in which a normalized heat-transfer coefficient is defined as $(\alpha - \alpha^*)/\alpha^*$. α^* is arbitrarily taken as the value midway between the minimum and maximum in α. Figure 6 contains

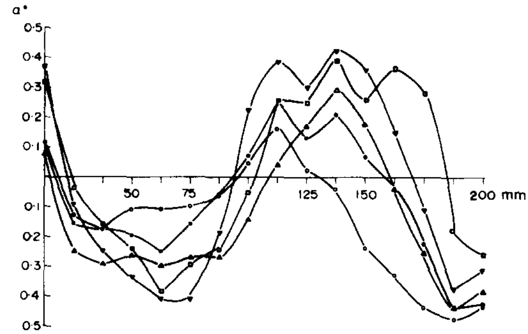

FIG. 6. Heat-transfer coefficient distributions in the presence of oscillations.

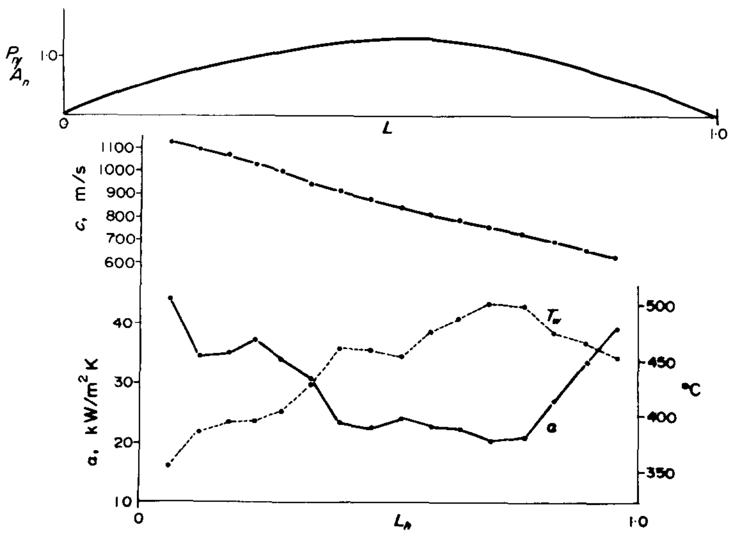

FIG. 7. Comparison of measured wall temperature and heat transfer coefficient distributions with fluctuating pressure distribution, $n = 1$.

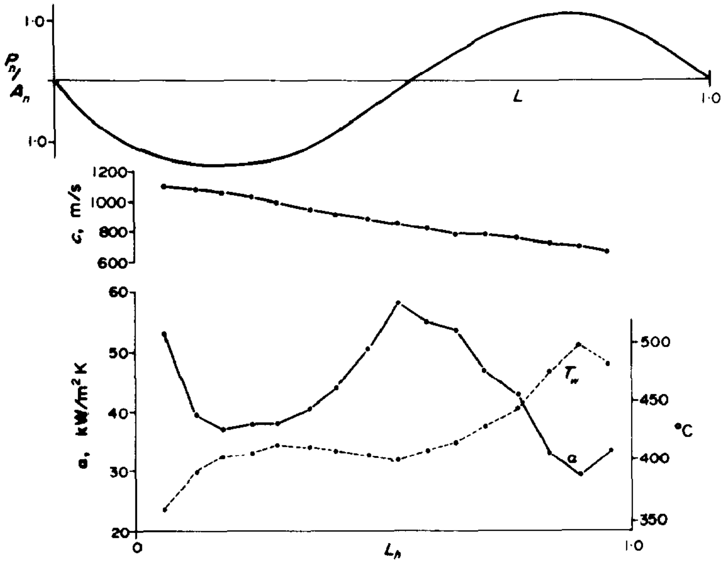

FIG. 8. Comparison of measured wall temperature and heat transfer coefficient distributions with fluctuating pressure distribution, $n = 2$.

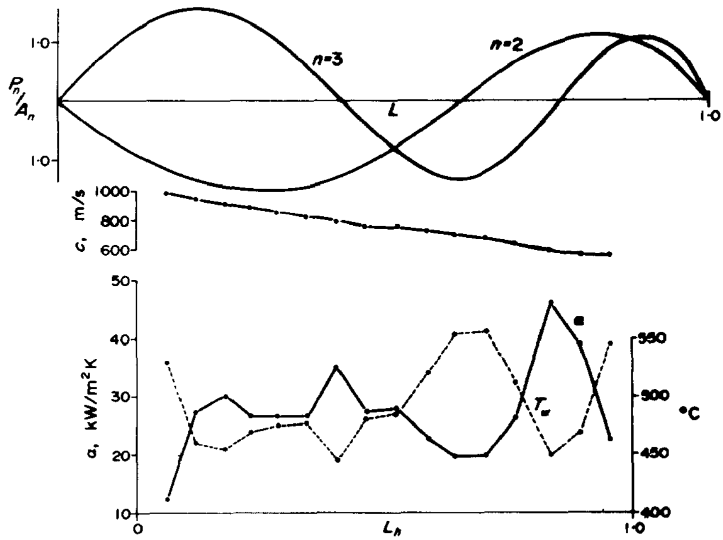

FIG. 9. Comparison of measured wall temperature and heat-transfer coefficient distributions with fluctuating pressure distribution, $n = 2$ and $n = 3$ simultaneously.

data from runs in which mode $n = 2$ only occurred. The runs shown were chosen irrespective of mass flow, inlet temperature and heat flux. The close agreement between all curves suggests that a particular mode of oscillation has associated with it a particular heat-transfer coefficient distribution.

From equation (4) and (5) the pressure fluctuation of the nth mode may be written.

$$P'_n = A_n\left(\frac{z}{a}\right)^{\frac{1}{2}} \sin\left(n\pi \frac{\ln(z/a)}{\ln(b/a)}\right) \tag{6}$$

and Figs. 7–9 show P'_n/A_n superimposed on wall temperature distribution and heat-transfer coefficient distribution for selected runs. Figures 7 and 8 show runs in which a single mode existed, $n = 1$ and $n = 2$ respectively. In both cases the local maxima in wall temperature (which give local minima in α) are located axially very close to the maxima in pressure amplitude.

Zeros in pressure amplitude are associated with maxima in heat transfer coefficient. Figure 9 shows a run in which modes $n = 2$ and $n = 3$ occurred simultaneously. The relative amplitudes are unknown but on the present interpretation mode $n = 3$ seems to be dominant, zeros in pressure amplitude corresponding to local maxima in α and maxima in pressure amplitude corresponding to minima in α. The wavelength of the axial variation of heat transfer coefficient is thus half of the acoustic wavelength.

The work of Jackson and Purdy [14] is particularly relevant to these results. They performed a series of experiments in which a pipe with constant wall temperature was cooled by forced convection of atmospheric pressure air at Reynolds numbers between 2×10^3 and 5×10^4. A standing wave was produced in the pipe by means of a microphone tuned to the frequency corresponding to $n = 4$ in equation (2). At high sound levels, measurements of local heat-transfer coefficient revealed that it had a sinusoidal axial variation, following the pattern of the standing wave but having half the wavelength. For laminar flow their results were supported by some theoretical work, which predicted the velocity distributions in the pipe and allowed qualitative comparisons with the measured heat-transfer coefficient distributions.

Cause of oscillations

The verification that a longitudinal standing wave occurs in the tube allows speculation that the energy source driving the oscillation arises in the low density wall layer, for equation (3) for the high density core of the flow fits the experimental facts quite well and is without a source term.

The variation of threshold values of heat flux with mass flow rate in Fig. 3 is consistent with the physical interpretation of the conditions necessary to support oscillations i.e. with a low density wall layer separating the high density core of fluid from the wall. The bulk enthalpy rise of the fluid between entry and exit of the test section, Δh, is given by

$$\Delta h = q \cdot \pi L_h \cdot d/m \tag{7}$$

assuming q to be uniform. The value of Δh will have a strong influence on the distribution of the fluid between low and high density regions, that is, upon the thickness of the wall layer. The lines drawn in Fig. 3 represent values of q necessary to maintain a fixed Δh for varying mass flow rates. The experimental points at the lowest mass flow have been arbitrarily chosen to define Δh. In fact, the radial distribution of heat flux in the fluid will depend upon local Reynolds number and will also influence the thickness of the low density layer, however the general trend of points is in agreement with the simple picture.

Part of the work of Stewart [15] upon which this paper is based, was direct to an examination of the alternate compression and expansion of the wall layer. This was based on a suggestion by Hall [16] that if the low density layer is subjected to cyclic pressure fluctuations whilst being supplied with heat from the tube wall, a net work output could result. The model depends upon the variation of thermal conductivity with pressure and temperature for near-critical fluids. An idealized situation was studied in which two planes, one representing the tube wall and the other the interface between high and low density fluids, varied their distance apart in a stepwise cyclic manner. Between each of the steps the fluid was allowed to reach thermal equilibrium. Two different sets of boundary conditions were applied. Firstly, both planes were held isothermal but in later calculations the tube wall was maintained at constant heat flux and constant temperature, more nearly representing the experiments. The cycle starts with the gap between the planes at its maximum and a temperature distribution due to steady conduction between the planes. Compression occurs isentropically and the temperature distribution changes. A period of conduction at constant pressure again occurs until a steady state is reached. In the instantaneous expansion the temperature distribution again changes isentropically, and is allowed subsequently to return, by thermal conduction, to the steady state corresponding to the start of the cycle. Because of the rapid variation of thermal conductivity with pressure and temperature the local heat flow during the cycle is difficult to visualize. In fact heat can flow locally to the fluid after compression and from the fluid after expansion, the requirements for a work output. The calculations were aimed at determining whether the net output could be positive. It was found to be positive for cases in which the wall temperature was above 400°C and heat flux was between 1·5 and 4·5 MW/m^2. The applied pressure fluctuation was between 5 and 30 bars on a steady level of 260 bars. This maintained the low density region very close to the wall.

There is a considerable amount of literature on thermoacoustic phenomena in less highly variable property fluids, e.g. the reviews by Feldman [17, 18] and the work of Rott [19], which is relevant to the present study but remains difficult to connect in a definite manner.

CONCLUSIONS

Forced convection experiments have been carried out using supercritical pressure water at 250 bars ($P/P_c = 1{\cdot}18$) flowing in small bore tubes at inlet bulk Reynolds numbers between $4{\cdot}8 \times 10^4$ and 9×10^4. The heat flux on the tube wall was approximately uniform. Specific conclusions may be drawn as follows:

(1) The high frequency audio oscillations appeared when the inside wall temperature over a large part of the test section exceeded the pseudocritical temperature and mean flux generated in the test section reached a critical value.

(2) The oscillations were identified with standing pressure waves set up in the test section.

(3) The test section axial wall temperature profiles were observed to change from almost linear before oscillations to ones with significant maxima and minima when oscillations occurred. Local heat-transfer coefficients varied by up to 100 per cent between these maxima and minima.

(4) Calculations indicate that in the vicinity of maxima in the pressure fluctuation there are maxima in wall temperature and minima in heat-transfer coefficient, and that zeros in pressure fluctuation are associated with minima in wall temperature and maxima in heat-transfer coefficient.

ACKNOWLEDGEMENTS

This work forms part of a programme on heat transfer to supercritical pressure fluids being carried out in the Simon Engineering Laboratories, University of Manchester, with the financial assistance of the Central Electricity Generating Board and under the direction of Professor W. B. Hall.

REFERENCES

1. W. B. HALL, J. D. JACKSON and A. WATSON. A review of forced convection heat transfer to fluids at supercritical pressures, Symp. Ht. Trans. and Fluid Dynamics of Near Critical Fluids, Proc. Inst. Mech. Eng. 182 Part 31 (1968).
2. R. C. HENDRICKS and R. J. SIMONEAU, Survey of heat transfer to near critical fluids, NACA Tech. Memo. NACA TM X'52612 (1969).
3. E. SCHMIDT, E. R. G. ECKERT and U. GRIGULL, Heat transfer by liquids near the critical state, Air Matls. Command, AAF Translation 527 (1939).

4. A. J. CORNELIUS and J. D. PARKER, Heat transfer instabilities near the thermodynamic critical point, Ht. Trans. and Fluid Mechs. Inst., U.C.L.A. (1965).
5. D. G. HARDEN, Transient behaviour of a natural circulation loop operating near the thermodynamic critical point, ANL 6710 (1963).
6. B. J. WALKER and D. G. HARDEN, Heat driven pressure and flow transients in the supercritical thermodynamic region, ASME paper 54-WA/HT-37 (1964).
7. K. GOLDMAN, Heat transfer to supercritical water at 5000 psi flowing at high mass flow rates through round tubes, Int. Dev. Ht. Trans., Proc. Ht. Trans. Conf. Pt. II, Paper 66 (1961).
8. W. S. HINES and H. WOLF, Pressure oscillations associated with heat transfer to hydrocarbon fluids at supercritical pressures and temperatures, ARS 32 (March 1962).
9. A. A. BISHOP, R. O. SANDBERG and L. B. TONG, Forced convection heat transfer to water at near critical temperatures and supercritical pressures, WCAP 2056 Pt. 4 (1964).
10. R. S. THURSTON, Pressure oscillations induced by forced convection heat transfer to two phase and supercritical hydrogen, LAMS 3070 (1964).
11. R. S. THURSTON, J. D. ROGERS and V. J. SKOGLUND, Pressure oscillations induced by forced convection heating of dense hydrogen, Advances in Cryo. Eng. Vol. 12, Proc. Conf. Crys. Eng., Colorado (1966).
12. K. GOLDMAN, Boiling songs, NDA 10-68 (1953).
13. N. L. KAFENGAUZ and M. J. FEDOROV, Investigation of high frequency pressure fluctuations occuring with heat transfer to water, *Teploenergetica* **15**, 1 (1968).
14. T. W. JACKSON and K. R. PURDY, Resonant pulsating flow and convective heat transfer, *Trans. Am. Soc. Mech Engrs* **87C**, 507 (1965).
15. E. STEWART, Thermally driven vibrations in a supercritical fluid. Ph.D. Thesis, University of Manchester (1969).
16. W. B. HALL, Private communication.
17. K. T. FELDMAN, Review of the literature on Rijke thermoacoustic phenomena, *J. Sound Vib.* 7(1), 83 (1968).
18. K. T. FELDMAN, Review of the literature on Sondhauss thermoacoustic phenomena, *J. Sound Vib.* 7(1), 71 (1968).
19. N. ROTT, Damped and thermally driven acoustic oscillations in wide and narrow tubes, *Z. Angew. Math. Phys.* **20**, 230–243 (1969).

OSCILLATIONS THERMOACOUSTIQUES LORS DE LA CONVECTION THERMIQUE DANS L'EAU SOUS PRESSION SUPERCRITIQUE

Résumé— Des mesures de transfert thermique sont présentées dans le cas d'eau sous pression supercritique qui s'écoule à travers des canaux horizontaux de petit diamètre sous la condition d'uniformité du flux thermique. Les expériences ont été conduites de façon à produire les oscillations à haute fréquence précédemment observées et qui étaient spontanément produites lors du transfert thermique à l'hydrogène ou à l'eau sous pression supercritique. On a mesuré les fréquences d'oscillation et on montre qu'elles sont associées aux oscillations de pression dans la section d'essai résultant d'une onde de pression existant entre l'entrée et la sortie. Plusieurs modes d'ondes stationnaires ont été identifiées. On a mesuré avec soin les distributions de température pariétale et les coefficients de transfert thermique ont été déduits. On constate que la distribution du coefficient de transfert thermique le long de la section de mesure a une variation fortement influencée par les fluctuations de pression locales.

THERMO-AKUSTISCHE SCHWINGUNGEN BEIM WÄRMEÜBERGANG AN WASSER BEI ÜBERKRITISCHEM DRUCK UND ERZWUNGENER KONVEKTION

Zusammenfassung— Diese Arbeit beschäftigt sich mit dem Wärmeübergang an Wasser bei überkritischem Druck, das unter den Bedingungen konstanten Wärmestroms durch horizontale Rohre kleiner Bohrung fliesst. Die Versuche waren dazu bestimmt, Schwingungen hoher Frequenzen zu erzeugen, deren plötzliches Auftreten schon früher beim Wärmeübergang an Wasserstoff und Wasser bei überkritischem Druck beobachtet worden war. Es zeigte sich, dass die Schwingungsfrequenzen, die gemessen wurden, von Druckschwankungen in der Teststrecke begleitet waren, die aus einer stehenden Welle zwischen Ein- und Austritt resultierten. Verschiedene Arten stehender Wellen wurden festgestellt. Aus genauen Messungen der Wandtemperaturverteilung wurden die Wärmeübergangskoeffizienten berechnet. Die Verteilung des Wärmeübergangskoeffizienten entlang der Mess-Strecke scheint zu variieren, was zeigt, dass diese Verteilung sehr stark durch örtliche Druckschwankungen beeinflusst wird.

ТЕРМОАКУСТИЧЕСКИЕ КОЛЕБАНИЯ ПРИ ВЫНУЖЕННОЙ КОНВЕКЦИИ В ВОДЕ В УСЛОВИЯХ СВЕРХКРИТИЧЕСКОГО ДАВЛЕНИЯ

Аннотация—В статье приводятся данные по измерению теплообмена при течении воды в горизонтальных тонких трубках в условиях сверхкритического давления и равномерного теплового потока. Эксперименты проводились с целью создания высокочастотных колебаний. Ранее бвло замечено, что эти колебания возникают самопроизвольно при переносе тепла к водороду и воде в условиях сверхкритического давления. Показано, что измеренные частоты колебаний связаны с изменениями давления в рабочем участке в результате возникновения стоячей волны между входом и выходом. Обнаружено несколько видов стоячих волн. Измерены распределения пристенной температуры, и рассчитаны коэффициенты теплообмена. Установлено, что колебания локального давления оказывают сильное влияние на распределение коэффициентов теплообмена в рабочем участке.