PSEUDOBOILING MECHANISM FOR n-HEPTANE

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It is shown that the improved heat transfer in pseudoboiling is due to self-excited thermoacoustic oscillations.

Much evidence has now been accumulated to show that heat transfer to many different liquids at supercritical pressures can be improved considerably under certain conditions [1, 2]. However, the physical nature of the improved heat-transfer mechanism, which is called pseudoboiling, has not yet been elucidated. Many suggestions have been made, but none of them has a particularly sound basis, and thus then cannot be used in working formulas.

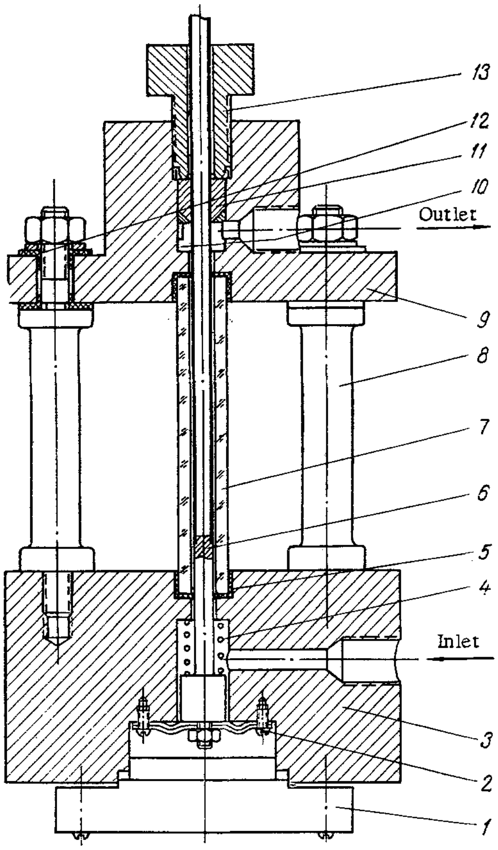

Fig. 1. Working section of heat exchanger: 1) cover; 2) current lead; 3) base; 4) spring; 5) seal; 6) heated tube; 7) glass tube; 8) spacer rod; 9) top plate; 10) sealing section; 11) sealing gland; 12) insulating washer; 13) nut.

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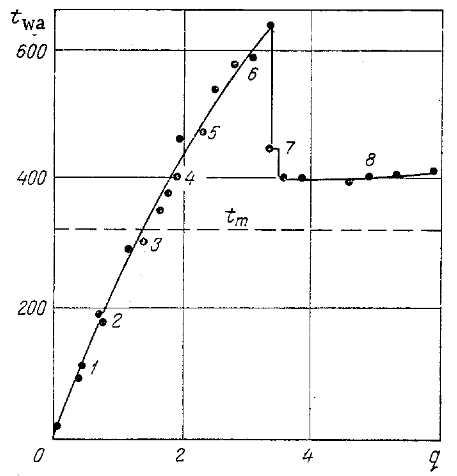

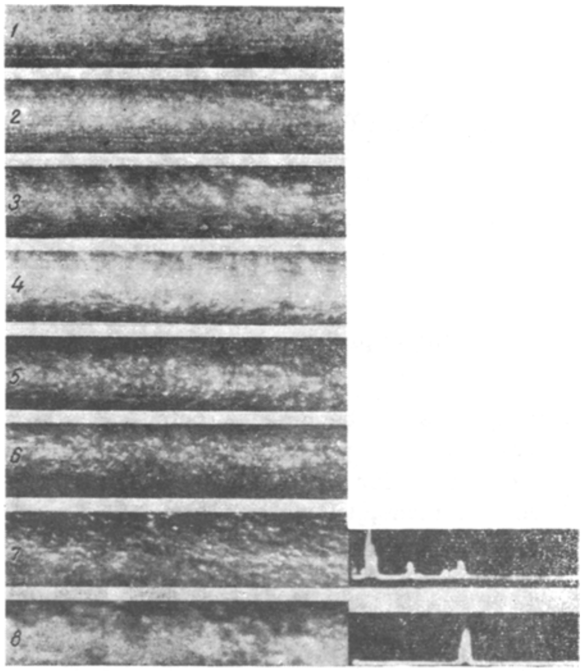

Fig. 2

Fig. 3

Fig. 2. Wall temperature as a function of heat flux for n-heptane: P = 40 atm; P/P_{cr} = 1.48; w_γ = 2500 kg/m^2·sec; $t_{l.in}$ = 20°C; d_1 = 4 mm; d_2 = 3 mm; l = 60 mm; t_{wa} = °C; q, 10^6·W/m^2.

Fig. 3. Part of the surface of heated tube (magnification about 20) and recordings from spectrum analyzer. The numbers are those of the runs in order of increasing heat flux.

Some workers ascribe the improved heat transfer to pseudoboiling, while others ascribe it to thermoacoustic pressure variations on the basis of Rayleigh's theory [3].

Here we report measurements on the mechanism, which have involved photographing pseudoboiling and measuring the heat-transfer rate.

We used an apparatus with a closed loop consisting of stainless-steel tubes, in which the liquid was circulated by a pump. The working section was an annular channel (Fig. 1).

The liquid flowed through this channel between the glass tube 7 (internal diameter 4 mm) and the concentric heater, which was supplied electrically, this being a tube of 1Kh18N10T steel of outside diameter 3 mm, wall thickness 0.5 mm, and length 60 mm. This tube was attached at one end to a copper tail, which could move freely within the base 3. Electrical contact between the base and the tail was provided by the lead 2 made of copper foil. The spring 4 prevented the tube from bending during heating. The wall temperature was measured with thermocouples. The pressure and temperature of the liquid at the inlet and outlet were measured, together with the flow rate, heat flux, and temperature of the internal surface in the tube; the temperature of the outer (cooled) surface of the tube was calculated from the temperature drop across the wall thickness.

These normal measurements were accompanied by piezoelectric measurements with a frequency analyzer to define the frequency spectrum of the thermoacoustic pressure oscillations; the behavior was also recorded by photography in reflected light. The light source was an ISSh-100 lamp giving a flash of duration about 10^{-5} sec. The SKS-1M camera operated at 3500-3900 frames/sec.

The measurements were made with n-heptane, for which P_{cr} = 27.01 atm and t_{cr} = 267.01°C; photographs were taken in 30 runs, including ones involving convection without boiling, boiling with the liquid initially below the boiling point (at 20 and 10 atm), and with pseudoboiling (30, 40, 50, and 60 atm); the mass flow rate $w\gamma$ was either 2500 or 5000 kg/m^2·sec. The inlet temperature in all cases was about 20°C, while the outlet temperatures ranged from 20 to 100°C in accordance with the heat flux and flow rate of the n-heptane. The Reynolds numbers for isothermal flow were in the range from 3500 to 7000, while during heat transfer it was larger on account of the heating, which reduced the viscosity of the n-heptane correspondingly.

Here we describe one characteristic series of experiments at supercritical pressures; Fig. 2 shows the temperature of the cooled wall as a function of the heat-flux density for n-heptane flowing turbulently at 40 atm. The part of the $t_{wa} = \varphi(q)$ curve up to $q = 3.3 \cdot 10^6$ W/m^2 corresponds to convective heat transfer, where the temperature rises smoothly with the heat flux. At a certain value of q, the wall temperature falls sharply and then remains almost constant and independent of q. This state is called pseudoboiling or improved heat transfer. If the improvement is marked and sharp, the pseudoboiling is accompanied by thermoacoustic oscillations and increase in the hydraulic resistance.

Figure 3 shows photographs of the tube during the runs represented in Fig. 2; the numbers on the two figures denote the runs. Photograph 1 shows the surface of the tube at t_{wa} = 110°C, while photographs 2 and 3 show the same for 190 and 300°C. Since the flow was not isothermal, the angle of refraction in the liquid varied, which revealed the turbulence in the flow as eddies; photograph 4 (wall temperature 400°C) shows the turbulence more clearly, while photographs 5 and 6 (wall temperatures of 470 and 590°C) show that the turbulent eddies take a form very similar to that for gas bubbles in a liquid. These photographs are very similar to photographs for heat transfer during surface boiling at subcritical pressures.

The change in structure of the flow turbulence as the wall temperature rose (photographs 1-6, with onset of pseudoboiling) did not affect the heat-transfer rate [the $t_{wa} = \varphi(q)$ relation remained as before].

The improved heat transfer (runs 7 and 8) occurred only when thermoacoustic pressure oscillations occurred, which were recorded by the frequency analyzer (photographs on the right in Fig. 3). In run 7, the dominant frequency was about 4 kHz, but there were also oscillations at 8 and 12 kHz. When the heat flux was increased (run 8), the dominant frequency jumped to one of the harmonics, namely, about 12 kHz. It was difficult to establish the effects of the thermoacoustic oscillations on the structure of the pseudoboiling simply from single photographs.

A much clearer picture of the change in structure of the turbulence as the wall temperature rose could be obtained from high-speed films; the density difference between the n-heptane at the core of the flow and at the wall increased with the heat flux, namely, with the wall temperature, and the turbulent eddies became sharper, with the shape approximating to that of gas bubbles in a liquid. A wall temperature of 500°C caused the entire heat-transfer surface to be covered with bubbles of size 0.01-0.1 mm, which moved along the flow direction. Superficially, this process was similar to that of boiling in a liquid initially below its boiling point and at a subcritical pressure.

The pseudoboiling changed substantially when the thermoacoustic oscillations arose; the gas bubbles appeared and disappeared simultaneously throughout the surface of the tube, occurring at the dominant frequency.

The pseudoboiling mechanism may be represented as follows. The bubbles are turbulent eddies of spiral structure,* which are visible on account of the density difference. The largest density occurred for $t_l \ll t_m$ in conjunction with $t_{wa} > t_m$, i. e., when the layer near the wall was effectively a hot gas, while the core of the flow was a cold liquid.

The turbulent eddies became detached from the gas layer near the wall and entered the cold core of the liquid on account of the surface tension, and during this process they became nearly spherical. These bubbles were rapidly cooled on entering the core, and therefore they collapsed, which was accompanied by the emission of pressure pulses, which were the larger, the larger the bubble and the more rapid the cooling. Conversely, eddies of cold liquid passing from the core to the hot wall result in droplets. The rapid expansion of the droplets also results in pressure pulses.

This process thus differs little from ordinary turbulence and thus does not affect the heat transfer and hydraulic distance; this is the major difference between pseudoboiling and boiling with the liquid initially below the boiling point.

The picture changes considerably when thermoacoustic oscillations occur; the pressure pulses resulting from the formation and collapse of the gas bubbles propagate in the flow with the speed of sound, which in this case is very much dependent on the compressibility (temperature and thickness of the gas layer). When the pressure pulses are reflected from any acoustic obstacle, standing waves are set up, whose frequency is equal to the natural frequency of a section of the channel of length l:

*These represent large-scale turbulence, where the size of the eddies is comparable to the thickness of the layer near the wall.

$$f = \frac{c}{2l} n,$$

where n = 1, 2, 3, etc.

The amplitude of the pressure oscillations in a standing wave is defined by

$$A = A_0 \sin 2\pi f\tau \cos 2\pi \frac{x}{\lambda}.$$

The standing pressure wave accentuates the formation and collapse of the bubbles; also, the oscillations occur in the same phase along the channel, so the bubbles will be formed and collapse simultaneously. This process was clear from the films. The amplitude of the pressure oscillation varied along the channel (zero amplitude at the nodes and maximum at the antinodes), so the heat-transfer rate should vary correspondingly, as has been observed [2].

The macroscopic characteristics (heat transfer, hydraulic resistance, thermoacoustic oscillations) make pseudoboiling similar to boiling in the presence of a cool liquid; however, the two processes differ in physical nature. In boiling, the bubbles are formed at the wall at nucleation centers, whereas in pseudoboiling they are formed from turbulent eddies in a uniform flow in the presence of large temperature and density gradients. In the first case, boiling sets in when the wall temperature exceeds the boiling point by a certain amount, and it is always accompanied by improved heat transfer. The heat transfer in pseudoboiling is improved only if thermoacoustic oscillations occur; i. e., the result is dependent on the acoustic parameters of the channel and of the liquid.

To a first approximation, the conditions for thermoacoustic oscillation can be represented as follows. These oscillations require a resonant condition, i. e., coincidence between the frequency f_n of the turbulent pulsations (in our case the frequency of bubble formation) and the natural frequency f for a column of liquid in the pipe:

$$f_n = f = \frac{c}{2l} n.$$

This condition is readily met, since the spectrum of the turbulent pulsations for a liquid in a tube is very broad (from 1 to about 10,000 sec^{-1} [4]), while the number of harmonics for the natural vibrations of the liquid column (n = 1, 2, 3, etc.) is also large, and the speed of sound may vary greatly on account of the compressibility variation.

A second condition is that the flow must receive the energy required to maintain the oscillation. This energy flux may be estimated from an equation that defines the energy arising from collapse of a cavitation bubble [5]:

$$J_n = \frac{4}{3} \pi (R_{max}^3 - R_{min}^3) PNdl.$$

The energy flux in the sound wave is defined by the acoustic equation

$$J_w = \frac{1}{2} \rho_l (Af)^2 c \frac{\pi d^2}{4}.$$

These oscillations will be stable for $J_n \geq J_w$; however, it is very difficult to derive from theory the quantities appearing in these equations, namely, the bubble size, frequency of formation, and speed of sound. Therefore, it is at present impossible to use this description.

NOTATION

A, pressure oscillation amplitude at distance x from start of section; A_0, maximum amplitude; c, velocity of sound; d, tube diameter; l, channel length; P, pressure; P_{cr}, critical pressure; q, heat flux; J_n, energy flux; R_{max}, R_{min}, maximum and minimum radii of bubbles; N, number of bubbles formed per unit area per unit time; t_l, liquid temperature in flow core; t_{cr}, critical temperature; t_{wa}, wall temperature; t_m, temperature at which physical properties change sharply for $P > P_{cr}$ and the specific heat is maximal; w_γ, mass velocity; ρ_l, liquid density; λ, wavelength; τ, time.

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MODEL OF HEAT TRANSFER ASSOCIATED WITH THE SURFACE BOILING OF LIQUID IN TUBES

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A model is proposed for the heat-transfer process associated with the surface boiling of liquid in tubes. An analysis of the various constituents of the heat-transfer mechanism distributed over the thickness of the intermediate boiling layer is presented.

Heat transfer at various points within the thickness of the evaporating layer created by the surface boiling of liquids in tubes is made up of a number of components differing essentially in mechanism. We shall here make a further attempt at analyzing these constituents, and after allowing for their interactions we shall present a method of analyzing and generalizing experimental data regarding the heat transfer associated with the boiling of an underheated liquid.

A detailed analysis of the intensity of heat transfer in the course of boiling should allow for the effect of the conditions at the liquid—vapor—solid boundary. However, there is as yet insufficient information available to be able to formulate reliable initial data for constructing an analytical model of the heat-transfer process.

In our subsequent analysis, instead of considering the characteristics of the vaporization mechanism (number of vaporization centers, separation frequency, and separation diameter of the bubbles) separately, we shall introduce an integrated characteristic in the form of the true volumetric vapor content. In order to calculate this quantity we may, for example, make use of the empirical formula of [1] or the method recommended in [2].

Following [2-4], we distinguish two zones in the flow: the zone of wall (boundary) heating and the underheated core of the flow. Inside the boundary zone the enthalpy of the flow is greater than the saturation enthalpy of the liquid, while its boundary with the core of the flow may be found by using equations derived for calculating heat transfer during the flow of a single-phase heat carrier, i. e., on the assumption that no boiling occurs in the channels. This will give the isothermal surface in the flow corresponding to temperature t_s.

An analysis of experimental data in [2] showed that the ratio of the area of the boundary zone to the true volumetric vapor content was constant along the whole length of the channel:

$$\varphi = \varkappa\varphi', \tag{1}$$

where $\varphi' = [1-(r_c^2/R^2)]$ is the relative area of the heated zone, and $\varkappa$ is a proportionality factor depending in the following way on the operating parameters:

$$\varkappa = 30\left(\frac{q\lambda'}{\alpha_{d.b}\, r_q \mu'}\right)^{1.15} \tag{2}$$

If the true volumetric vapor content is known, the radius of the underheated core of the flow may be calculated from the equation

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