HYDRAULIC RESISTANCE AT HIGH FLOW SPEEDS IN THE BOILING OF ETHANOL BELOW THE SATURATION TEMPERATURE

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Results are given on the hydraulic resistance to a flow of unsaturated liquid with boiling at the wall at speeds of 50–210 m/sec.

There have recently been papers [1–4] on the effects of boiling on liquid flow, most of which have been done with water at low flow rates ($W_g \leq 10^4$ kg/m^2 · sec).

Here we present results for flow of 96% ethanol in tubes with boiling at the wall over a very large range of speeds.

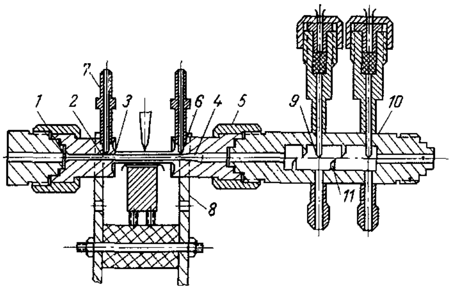

Fig. 1

The apparatus was a closed high-pressure loop with an auxiliary heater, working section, mixer (with thermocouples), cooler, control valves, and sump. The liquid was circulated by two three-piston pumps in series with an auxiliary centrifugal pump.

The working section was made of 1Cr18Ni9Ti stainless steel or alloyed copper with a diameter of 2 or 3 mm and a length l = 15D. The surface finish of the smooth tubes was class 8–9, while the roughened tubes had D/Δ from 3000–12 000 (Δ of 0.1–0.6 μm).

Figure 1 shows the working section, which consists of the confuser 1, the hydrodynamic stabilization section 2, heated part 3, and diffuser 4. The input and output parts had very thick walls with pressure nuts 5 to connect to the rest of the loop. These thick parts had two radial holes 6 each 0.4 mm in diameter at 0.5 mm from the inlet and outlet of the heated part, in order to measure the pressure and hydraulic resistance. These holes led to the tubes 7, which were connected to gauges. The confuser and diffuser provided flow without detachment in the heated part.

The high flow speeds meant that the pressure at the inlet sometimes exceeded 300 bar, whereas the static pressure in the heated part was only 20–50 bar. Under these conditions, even a small effect from the dynamic pressure on the measured static pressure could lead to a considerable error in determining the hydraulic resistance and the deviation from the saturation temperature. For this reason, especial attention was given to the internal finish, especially at points where the pressure was measured. To eliminate burrs on the inner wall, holes 6 were drilled before the final surface finishing of the heated part. The bore of part 3 was finished correct to 0.005 mm. As each section had two holes each with its own pressure gauge, the gauge readings could be compared in order to establish whether the pressure was read correctly. The components were considered acceptable if the readings agreed exactly.

The heated part was heated by low-voltage ac brought in via the copper leads 8 from an OSU-80 stepdown transormer. An ROT-25 variac was used to control the heat input.

Measurements were made of the heat input, hydraulic resistance, pressure, flow rate, and exit temperature. The heat input was deduced from the electrical input to the heated part, which was measured with an astatic wattmeter of accuracy class 0.2. The pressure and hydraulic resistance were measured with class 0.4 gauges whose accuracy had been further improved by previous calibration against an MP-60 piston gauge of class 0.05.

The flow rate was deduced from the pressure drop across a throttle as measured with a differential mercury manometer. This manometer had previously been adjusted to give the optimal difference between the mercury levels and was calibrated by use of a measuring vessel, a three-way stopcock, and an electric clock. Check calibrations were performed after each run. Runs were accepted if the scale of the differential manometer had altered less than 1% between repeat calibrations. The exit temperature was measured by the three chromel-alumel couples 9 (Fig. 1), whose hot junctions were in the liquid within the mixer 10. The emf was measured with an R 2/1 semiautomatic potentiometer. The mixer produced a uniform temperature distribution across the cross-section; it was a steel tube fitted with semicircular baffles 11. The couples were placed in the gaps between the baffles; their readings were compared to establish the performance of the mixer.

The tests were done at 20, 30, 40, and 50 bar with flow rates of $1.75 \cdot 10^4$, $4 \cdot 10^4$, $5 \cdot 10^4$, $5.3 \cdot 10^4$, $5.8 \cdot 10^4$, $7 \cdot 10^4$, $8 \cdot 10^4$, and $13.5 \cdot 10^4$ kg/m^2 · sec and deviations from saturation of 10°–190° K. These conditions corresponded to Reynolds numbers $R = 2 \cdot 10^4 - 2 \cdot 10^6$.

Some of the measurements were made on a tube of 1Cr18Ni9Ti steel whose internal surface had been buffed. The results for isothermal flow up to R of $2 \cdot 10^6$ agreed with Nikuradse's results for smooth tubes.

The effects of surface roughness were examined by using two stainless-steel tubes whose internal surfaces had been treated with abrasive powders of different grain sizes. Resistance measurements under isothermal conditions showed that the results were as for rough tubes with uniform roughness for $R > 3.2 \cdot 10^5$ in one case and $R > 5.7 \cdot 10^5$ in the other. The heights of the surface projections were 0.25 and 0.60 μm.

We also used tubes of alloy copper with D of 1.5 and 2 mm and roughness 0.2–0.6 μm.

The runs with heating were performed as series; in each series, the pressure and mass flow rate remained constant, the variable parameter in the series being either the heat input or the deviation from saturation. The former was varied from zero up to the critical loading. The results were presented as ξ^0 (reduced hydraulic-resistance coefficient) as a function of q^0 (reduced heat flux) and K_* (dimensionless deviation from the saturation temperature), in which ξ^0 is the ratio of ξ (coefficient of hydraulic resistance for temperature-varying flow) to ξ_0 (the same for isothermal flow).

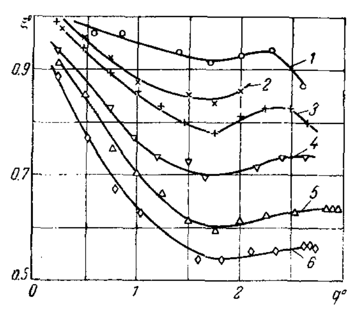

Fig. 2

It has been shown [5] that the relation of critical heat flux to deviation from saturation can be extended if we take the following dimensionless quantity as the parameter defining the effects of that deviation on the critical flux:

$$K_* = (i' - i) r^{-1} \sqrt{\rho' / \rho''}.$$

Smooth tubes			Rough tubes			
point	p, bar	W_g, kg/m² · sec	point	p, bar	W_g, kg/m² · sec	D/Δ
1	20	$4 \cdot 10^4$	6	30	$5.3 \cdot 10^4$	5000
2	30	$4 \cdot 10^4$	7	50	$8 \cdot 10^4$	3000
3	30	$5 \cdot 10^4$	8	30	$8 \cdot 10^4$	3000
4	30	$5 \cdot 10^4$	9	20	$8 \cdot 10^4$	3000
5	40	$7 \cdot 10^4$	10	30	$13.5 \cdot 10^4$	5000

It has been found [6] that, at high flow speeds, there is a direct relation between the critical heat flux and the tangential stress at the wall. For this reason, and by analogy with the boiling crisis in forced flow, we used K_* as the generalized parameter defining the effects of deviation from saturation on ξ^0.

The effects of heat load on ξ^0 with boiling were evaluated via the above q^0, which is the ratio of the actual heat flux q to the heat flux q_0 corresponding to the onset of boiling at the heated wall, as deduced from the formula for convective heat transfer without boiling

$$q_0 = 0.023 \frac{\lambda}{D} R^{0.8} P^{0.4} \left(\frac{\mu'}{\mu}\right)^{0.14} (T' - T).$$

The results were first worked up as ξ^0 against q^0 for each pressure and W_g. Figure 2 shows a set of curves for p = 30 bar and $W_g = 5.8 \cdot 10^4$ kg/m² · sec, with K_* as parameter taking the following values: 1) 2.34, 2) 2.6, 3) 2.9, 4) 3.5, 5) 4, 6) 4.3. There is a marked fall in ξ^0 as q^0 increases from 0 to 1, with a minimum and a maximum in the region $q^0 > 1$. The maximum corresponds to the precrisis state. There is less than 10% difference between the maximum and minimum values of ξ^0 for $W_g \geq 4 \cdot 10^4$ kg/m²· sec and $q^0 > 1$.

Figure 3 shows $\xi^0(K_*)$ for the maximum on $\xi^0(q^0)$, while Table 1 gives the p, W_g, and D/Δ for Fig. 3. There are three distinct regions in K_*: 1) < 0.2, 2) 0.2 < K_* < 2, 3) >2. Here we consider only the latter two regions; the region K_* < 0.2 is one of extensive evaporation and requires further study. Figure 3 shows that ξ^0 is constant at about 1 in the region 0.2 < K_* < 2 and is independent of the heat flux, K_*, wall roughness, p, and W_g. As $K_* \to 2$, ξ^0 falls to 0.9 and remains constant in the range 2 < K_* < 2.5. There is a further fall in ξ^0 for K_* > 2.5, where for W_g between $4 \cdot 10^4$ and $6 \cdot 10^4$ kg/m² · sec all the points lie on the single straight line

$$\xi^0 = 1.36 - 0.18\, K_*$$

no matter what the value of q, p, or W_g.

Comparison of theory with experiment shows that the fall in ξ^0 for K_* > 2 cannot be due solely to the nonisothermal nature of the flow. The theoretical $\xi^0(K_*)$ curves for nonisothermal flow without boiling should also show an appreciable dependence on p and W_g, but Fig. 3 does not reveal this. We may therefore suppose that the fall in ξ^0 for K_* > 2 is due to an effect of boiling on the hydraulic resistance at high speeds which is specific to large deviations from saturation.

It has been shown [5] that the region K_* < 2 is characteristic of the occurrence of normal boiling crises in forced motion of a liquid. For K_* > 2 there are extended boiling crises that differ substantially from normal ones not only in external features but also in the relation of critical heat flux to deviation from saturation. There is a common

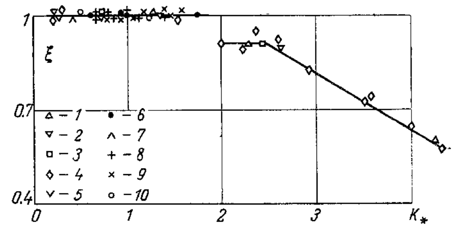

Fig. 3

boundary for 1) transition from normal crises to extended ones, 2) transition from the region $\xi^0 \approx 1$ to the region of lower ξ^0. This indicates that there is some deeper and hitherto unsuspected relation between boiling crises and hydrodynamic phenomena in liquid flows, which here makes itself felt in the transition to the region of extended boiling crises.

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