THERMOHYDRAULIC OSCILLATIONS IN A SYSTEM SHOWING NATURAL CIRCULATION OF A TWO-PHASE COOLANT

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Thermohydraulic oscillations are considered for a wide range in coolant thermophysical parameters.

A natural-circulation heat exchanger operating with the liquid boiling in the tubes may [1, 2] show thermohydraulic oscillations, whose period is dependent on the type of liquid, pressure, thermal loading, and other parameters. To determine the stability limits and the oscillation characteristics in steam-generating tubes with forced water flow ($\rho\bar{w} > 200$ kg/m^2·sec), one can use the data of [3, 4] for pressures $P_s \geq 5$ MPa, but they cannot be used to examine the steam generation in a natural-circulation system because the dimensions of the part of the steam-generation tube with deteriorated heat transfer will not be known, nor will the circulation velocity, or else they will not be the decisive parameters, while the liquid underheating at the inlet to the steam-generating part may be zero. Also, at pressures less than 2 MPa and low circulation rates, there is [5] a rearrangement in the flow structure in the steam-liquid mixture, and the predominant mode becomes of plug type. The latter is confirmed by the [6-8] results, where it was considered that oscillations in that range are associated with the plug flow, with the period determined by the passage frequency for the single steam plugs [6, 7] or groups of them [8]. There is no unified approach to analyzing the causes and characteristics of these oscillations in low-pressure natural-circulation systems.

Here we consider an approximate physical model for the flow of a boiling liquid having a high flow vapor content at the outlet from the vapor-generating tube, which explains the oscillations and enables one to identify the main components in the period and which thermophysical parameters govern the amplitude and frequency. This model has been applied to measurements on instability in the boiling of water and the freons R22, R113, and R142 in vertical tubes heated by condensing steam in application to low-pressure natural circulation systems at heat loads close to the limiting values [2].

The following assumptions are made in the model for these oscillations. Firstly, the oscillations are related to plug flow. Secondly, the period τ_Σ is determined by three components: τ_1 is dependent on the plug formation frequency, τ_2 being related to the length of the emerging plugs and the time for rapid vapor formation in annular flow arising after the emergence of the plugs from the vapor-generating channels, and τ_3 dependent on the length ℓ_Σ of the liquid column in the U-shaped circuit [1]. To identify the parameters that determine the τ_Σ components separately and also the oscillation period as a whole, we consider the flow structure in a vapor-generating channel.

The lengths in the tube with single-phase flow ℓ_0 and bubble flow ℓ_b for q = const can be estimated from the balance equation

$$l_0 + l_b = \frac{\rho' W_{in}' D}{4q} (\Delta i_0 + x_b r). \tag{1}$$

The time taken by the liquid with homogeneous structure to attain a flow bubble content x_b is readily determined from the [9] expression:

$$\tau_0 + \tau_b = \frac{\Delta i_b \rho' D}{4q} + \frac{rD}{4q(v'' - v')} \ln\left(1 + \frac{v'' - v'}{v'} x_b\right), \tag{2}$$

in which x_b is the vapor bubble content at which there is a transition from bubble mode to plug. According to [10], bubble mode can exist for volume vapor contents not exceeding 0.3. The vapor plugs have initial length equal to the internal diameter of the tube D, while the heat flux density q is constant over the height, which can be determined from a formula analogous to that in [11] for a slot channel:

$$l_{\rho i} = \sum D \exp\left(\frac{4q}{r\rho'' D} \, i \, \tau_1\right), \tag{3}$$

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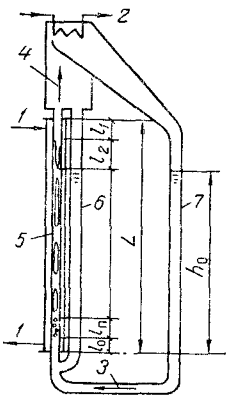

Fig. 1

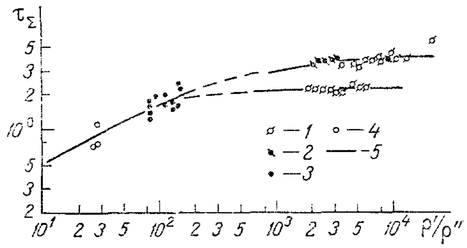

Fig. 2

Fig. 1. Apparatus and two-phase flow structure for liquid boiling in a tube: 1) heating medium; 2) cooling water; 3) liquid heat carrier; 4) heat carrier vapor; 5) vapor-generating channel; 6) overflow tube; 7) descending tube.

Fig. 2. Oscillation period as affected by type of liquid and pressure ($Q \approx Q_*$, $h_0 \approx 0.9$ m); apparatus with overflow tube: 1) water; 2) apparatus without overflow tube: 2) water; 3) freon R113; 4) freon R142; τ_Σ in sec.

where i is the sequence number of the plug along the height. The number of them n is related to the tube length L by

$$L \leqslant l_0 + l_b + \sum_{i=1}^{n} l_{p\,i} + (n-1)\, l_{lp}. \tag{4}$$

In [5, 12], in research on plug flow, it was observed that the liquid plug length $\ell_{\ell p}$ is only slightly dependent on the working parameters over a wide range and is determined in the main by the tube diameter, so one can assume that the plug formation period is

$$\tau_1 \sim \frac{f(D)}{\overline{W}}, \tag{5}$$

in which $\overline{W}_m = W'_{in}\left(1 + \frac{\rho' - \rho''}{\rho''} x_b\right)$. For low pressures, $\left(1 + \frac{\rho' - \rho''}{\rho''} x_b\right)$ varies little, and one can take $\overline{W}_m \approx W'_{in} \times \mathrm{const}$ for the bubble flow region.

In the region of limiting thermal loads Q_*, a low-pressure natural-circulation system with relative filling level h_0/L will have $Q_* \sim (\rho'\rho'')^{1/2} r (h_0/L)^{1/2}$ [2], and

$$W'_{in} = \frac{Q_*}{r\rho' F_b x_{out}} \sim \left(\frac{\rho''}{\rho'}\,\frac{h_0}{L}\right)^{1/2} \frac{1}{x_{out}} f(D). \tag{6}$$

Then

$$\tau_1 \sim x_{out} \left(\frac{\rho'}{\rho''}\right)^{1/2} f_2(D) \left(\frac{h_0}{L}\right)^{-1/2}, \tag{7}$$

in which x_{out} is the mass vapor content at the outlet from the vapor-generating channels for $Q \approx Q_*$.

The second component of τ_Σ increases as q and L increase and as P_s decreases because the exit plug length rises. At the same time, there are increases in the periodic parts of the tube having dispersed annular flow length ℓ_2 and dispersed flow ℓ_1 (Fig. 1), which also increases τ_Σ. There is rapid evaporation in the dispersed annular region and high pressure loss from friction in the dispersed one, which may reduce the inlet flow rate and even reverse the flow direction in the economizer section. This effect is most prominent for low pressures, where ρ'/ρ'' is very large (Fig. 2).

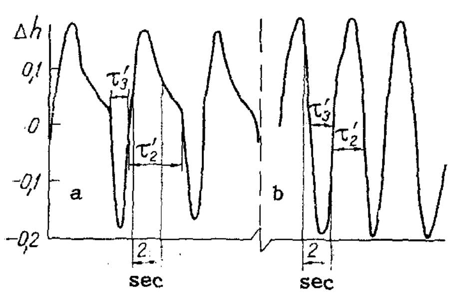

Fig. 3

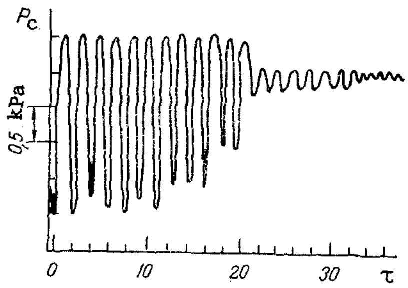

Fig. 4

Fig. 3. Level oscillations in the drainage tube for boiling water ($h_0 \approx 0.9$ m): a) $Q = 54.6$ kW, $\overline{P}_c = 12.1$ kPa, $\tau_\Sigma = \tau_2' + \tau_3' = 4.6$ sec; b) $Q = 67.3$ kW, $P_c = 35$ kPa, $\tau_\Sigma = 3.4$ sec. Δh, m.

Fig. 4. Oscillations in the pressure P_c as a function of tube length heated by condensing steam for R113: $h_0/L = 0.79$, $\overline{P}_c = 185$ kPa, $Q = 30.5$ kW, $\tau = 0$, $(L—h_c) = 1.52$ m; $\tau = 35$ sec, $(L—h_c) = 0.76$ m. t, sec.

Equations (3) and (4) show that the plug length at the exit ℓ_p^{out} for $Q \approx Q_*$ is proportional to the following:

$$\ell_p^{out} \sim f\left(\frac{q}{r\rho'' D},\ L - l_0 - l_b,\ D\right) \sim f\left[\left(\frac{h_0}{L}\frac{\rho'}{\rho''}\right)^{1/2},\ L - l_0 - l_b,\ D\right]. \tag{8}$$

The liquid film vanishes at a certain rate from the tube surface as it moves along with the vapor, and this is dependent on the boundary vapor content, as is the time for rapid vapor formation [13], so one has

$$\tau_2 \sim x_f \sim \frac{\sigma\rho'}{\mu'}\frac{1}{\rho W}\left(\frac{\rho''}{\rho'}\right)^{1/3}, \tag{9}$$

or for $Q \leq Q_*$

$$\tau_2 \sim \frac{\sigma\rho'}{\mu'}\frac{x_{out}\, r F_b}{Q}\left(\frac{\rho''}{\rho'}\right)^{1/3} \sim \frac{\sigma}{\mu'} x_{out}\left(\frac{\rho'}{\rho''}\right)^{1/6}. \tag{10}$$

Then τ_2 is dependent on the coolant properties and channel geometry:

$$\tau_2 \sim f\left(\frac{\rho'}{\rho''},\ \frac{h_0}{L},\ \frac{\sigma}{\mu'},\ L - l_0 - l_b,\ D\right). \tag{11}$$

For $\rho'/\rho'' \geq 10^4 = 10^5$ [1], the boiling becomes pulsating and is characterized by a frequency close to the natural frequency of the liquid column in the U-tube system, with extensive flow pulsations at the inlet. That state has been observed with water boiling at 8-90 kPa and is accompanied by fluctuations in liquid level in the drainage tube, with amplitude up to 0.3 m [1]. A similar phenomenon occurs in the boiling of sodium [9]. The level fluctuations, which influence τ_Σ, for a known pulsation frequency in the two-phase flow can be determined from the [3] equation

$$\ddot{y} + 2h\dot{y} + w_0^2 y = F(w), \tag{12}$$

in which $F(\omega)$ is the reduced force arising in the formation and growth of the vapor plugs. In a natural-circulation system, where there is no flow throttling, the effective cross sections of the vapor-generating and drainage parts are

$$w_0 = \sqrt{\frac{2g}{l_\Sigma}},\quad h = \frac{F_f}{2m},\quad w_0 = \frac{2\pi}{\tau_0}, \tag{13}$$

in which ℓ_Σ, m, and τ_0 are the length, mass, and period for the liquid column oscillations in the U-tube system, g the acceleration due to gravity, and F_f the frictional force arising on displacing the liquid column during the oscillations. Theory and experiment show that such oscillations are possible for $\tau \approx \tau_0$.

Equations (7), (11), and (13) apply for the individual components of τ_Σ for a liquid boiling in a low-pressure natural-circulation system at thermal loads close to the limiting ones; the oscillation period and amplitude increase as the pressure decreases and as the filling level rises and the length of the heated part of the vapor-generating tubes increases. In general, τ_Σ is determined by the following:

$$\tau_\Sigma = f\left(\frac{\rho'}{\rho''},\ \frac{h_0}{L},\ l_\Sigma,\ g,\ D,\ x_{out},\ \sigma,\ \mu'\right). \tag{14}$$

The ratio $q/(r\rho''D)$ does not appear explicitly in the list of independent variables because it can be determined from (8) in the limiting heat load range.

We now examine observed effects with this model. The experiments were performed with an apparatus having natural circulation for water and the freons R22, R113, and R142 having saturation temperatures in the range 30-100°C. The tubes were heated by condensing steam. Tube diameters 6 × 1, length 1.52 m, 101 tubes. The apparatus has been described in more detail in [1]. Figures 2-4 give the results.

There is a fall in τ_Σ as the mean saturation pressure above the vapor-generating tubes $\overline{P}_c$ increases under conditions of pulsating boiling for water ($\rho'/\rho'' > 2\cdot10^3$), which is explained by Fig. 3, which shows the oscillations in the liquid column in the drainage tube. A period consists of a constant component τ_3' related to the free motion of the liquid column in the U-tube system and a time-varying part τ_2', which is governed by the time for rapid vapor production after the emergence of a vapor plug from the generating channel. As $\overline{P}_c$ and q increase, the lengths of the emerging plugs decrease, as does τ_2', while $\tau_\Sigma = \tau_2' + \tau_3'$ also decreases. With certain combinations of $\overline{P}_c$, q, l_Σ in the U-tube system, when $\tau_\Sigma \approx \tau_0$, one gets a form of thermohydraulic resonance, which has been observed for water boiling at $2\cdot10^3 \le \rho'/\rho'' \le 4\cdot10^3$ (Fig. 2). The pressure amplitude P_c within the circuit attained 5 kPa, while the pressure of the heating steam P_{st} was up to 3 kPa, and the level in the drainage tube was up to 0.3 m. As P_c and q increase in the resonance range, the frequency characteristic can alter in two senses. In the first case, where the U-tube system is a single one in the natural circuit (apparatus without overflow tube), increase in $\overline{P}_c$ and q produces a monotone decrease in the frequency. If the circuit contains a U tube with a smaller ℓ_Σ (with overflow tube), there is a stepwise reduction in the period. In that transition, the amplitudes of the oscillations in P_c and P_{st} are almost halved. The oscillations in the liquid level in the drainage tube vanish, while those in the overflow tube persist. In the pulsating boiling state, there are also considerable effects from h_0/L, l_Σ, whose increase causes τ_Σ to rise [1]. Then at low pressures, the main parameters influencing τ_Σ are the pressure (ρ'/ρ''), the design of the drainage system, and the liquid filling level.

With R22, R113, and R142, the experiments were performed without the overflow tube. Oscillations occurred for $\tau_\Sigma < \tau_0$, but there were virtually no liquid level oscillations in the drainage tube. Consequently, they were not dependent on the design of the drainage system and were determined only by boiling in the tubes. When τ_Σ was independent of ℓ_Σ, the period and extent of the oscillations decreased monotonically as the ratio of $\overline{P}_c$ to the critical pressure P_{cr} increased because of the reduction in the plug length at the exit. For example, with R113, the amplitude ΔP_c for $100 \le \overline{P}_c \le 260$ kPa ($0.03 \le \overline{P}_c/P_{cr} \le 0.08$) did not exceed 2 kPa. With R142 at $440 \le \overline{P}_c \le 1.1$ MPa ($0.1 \le P_c/P_{cr} \le 0.25$), the amplitude did not exceed 0.4 kPa. In this oscillation range, reducing the heated length $L - h_c$ reduced the oscillation intensity because of the continuous flooding of the lower part with condensate (Fig. 4), although the frequency remained almost constant. This confirms the theoretical conclusion that the heated tube length affects the oscillation intensity and frequency.

With further increase in (P_c/P_{cr}), there is an increase in the total mass flow rate of the liquid through the channel, with simultaneous reduction in the volume flow rate for the vapor phase for a particular Q. The probability of plug flow is decreased, and the bubble state goes over directly to the dispersed annular or dispersed one. That change in state over the height cannot lead to substantial oscillations in the average heat-transfer coefficient, since there was no visible instability. Pulsation-free operation was observed (on manometer readings) with R142 for $\overline{P}_c > 1.1$ MPa and for R22 for $\overline{P}_c > 1.5$ MPa.

The following conclusions are drawn from the averaged curve in Fig. 2. With R113 and R142, where τ_Σ is independent of the drainage system, the measurements agree well with the theory, i.e., $\tau_\Sigma \sim (\rho'/\rho'')^{1/2}$. The slope of the average curve alters at low pressures because ℓ_Σ affects the period, as does the design of the drainage system. There is a good fit to the measurements on τ_Σ as a function of ρ'/ρ'' if one neglects σ/μ', which suggests that the rate of loss of the film in the dispersed annular state has little effect on τ_Σ. The theory indicates that the boiling of a saturated liquid in the limiting heat load range is such that the latent heat of evaporation r should not affect the oscillation period, which is confirmed by the observation that although r varied considerably (from 2300 to 100 kJ/kg), there was no substantial spread in the measurements about the averaging curve in Fig. 2. We did not determine the effects of g and D on τ_Σ here. The vapor content x_{out} was close to one [2] and therefore did not affect the data fitting by means of the ρ'/ρ'' simplex.

There are three characteristic thermohydraulic oscillation states with low-pressure natural-circulation heat exchangers operating at $Q \approx Q_*$, and conclusions can be drawn on the correct model for two-phase flow of a boiling liquid in that range.

In the first state, τ_Σ is dependent on the boiling processes and on the hydraulic phenomena associated with the design of the drainage tube. The transition to the second state is via thermohydraulic resonance, where the oscillation amplitude is largest. In the second state, the oscillations are governed by the liquid boiling only in the vapor-generating tubes. As $\bar{P}_c$ increases, there is a monotone decrease in the oscillation intensity and period. In the third state (with $\bar{P}_c/P_{cr} \geq 0.2 = 0.3$), the oscillations have practically no effect on the operation of the system or are absent completely.

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HYDRODYNAMICS OF AN ASCENDING LIQUID FILM FLOW AND VAPOR FLUX IN A VERTICAL ANNULAR CHANNEL

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Results are represented of an experimental investigation of crisis phenomena constraining different modes of an ascending liquid film flow under the action of a vapor flux on the basis of liquid film parameter measurements.

A significant quantity of papers [1-16] is devoted to investigation of the phenomenon of destruction of the stability of different flow modes of a two-phase stream in vertical channels. Meanwhile the results of investigations characterizing the stability of film flow in an ascending coflow are of limited nature [14-16]. Consequently, there are uninvestigated areas on maps of two-phase flow modes in vertical tubes and a number of characteristic transitions between the flow modes is indicated provisionally. In addition, available experimental data in the area of individual transitions between different flow modes are obtained on the basis of different methods, which makes their comparison difficult.

The available boundaries of the annular flow mode are obtained principally in a study of a descending liquid film coflow and vapor flux [5-13]. L Ya. Zhivaikin [14] and B. I. Nigmatullin et al. [15] performed a direct experimental investigation of liquid film disruption phenomenon in a vertical ascending flow. The equations obtained that characterize the phenomenon of film rupture differ radically in structure, which is apparently explained by the determination of different physical effects in the experiments during interaction of the light phase flux and the liquid film as well as by utilization of different methodological approaches.

Investigation of the film flow modes was performed on an experimental set-up (Fig. 1) consisting of the following main components: a vapor generator chamber, a working section, a condenser, chambers for condensate collection, apparatus for fluid insertion into the annular channel of the working section, and a measuring part. The

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