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

A. I. Leont'ev, V. A. Fedorov, and O. O. Mil'man

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 (ρw > 200 kg/m²·sec), one can use the data of [3, 4] for pressures P ≥ 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 τp is determined by three components: τ1 is dependent on the plug formation frequency, τ2 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 τ3 dependent on the length lₚ of the liquid column in the U-shaped circuit [1]. To identify the parameters that determine the τp 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 lₒ and bubble flow lₜ for q = const can be estimated from the balance equation

\[ lₒ + lₜ = \frac{ρ' \, W₀ \, n \, D}{4q} (Δtₒ + xₜ \, τₐ). \]  

(1)

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

\[ τₒ + τₜ = \frac{Δh_b \, ρ' \, D}{4q} + \frac{r \, D}{4q (v'' - v')} \ln \left( 1 + \frac{v'' - v'}{v'} \, xₜ \right), \]  

(2)

in which xₜ 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:

\[ lₚ = \sum D \exp \left( \frac{4q}{rₚ' \, D} \, i \, τₚ \right), \]  

(3)
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 ($Q \approx Q_\star$, $h_0 \approx 0.9$ m); apparatus with overflow tube: 1) water; 2) apparatus without overflow tube: 2) water; 3) freon R113; 4) freon R142; $\tau_\Sigma$ 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

$$L \leq l_0 + l_b + \sum_{i=1}^{n} l_{p_i} + (n - 1) l_{p'}.$$  

In [5, 12], in research on plug flow, it was observed that the liquid plug length $\ell_{lb}$ 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

$$\tau_1 \sim \frac{f(D)}{W}.$$  

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

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

$$W_{in} = \frac{Q_\star}{\tau_0' F_b x_{out}} \sim \left(\frac{\rho''}{\rho'} \frac{h_0}{L}\right)^{1/2} \frac{1}{x_{out}} f(D).$$

Then

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

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

The second component of $\tau_\Sigma$ increases as $q$ and $L$ increase and as $P_\sigma$ 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 $\ell_2$ and dispersed flow $\ell_1$ (Fig. 1), which also increases $\tau_2$. 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 $\rho'/\rho''$ is very large (Fig. 2).