Study heat exchange in porous structures

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Abstract. This paper shows the problem with heat exchange depending on units of thermal power plant equipment. The type of structures is determined and the heat flow for different pressures is proposed. Studies are developed for the condition of the heat exchange surface. Devices with porous coatings eliminate the development of cracks in the components and units of TPP equipment have been suggested. The research is applicable to gas turbine units of TPP. Comparable capillary-porous and flow systems have high reliability, but the former allowed the reduction of coolant consumption dozens up to 80 times. The results show that at higher heat loads it is suitable to use in porous surfaces to control the cooling surface. Evaluation of capillary-porous structures has shown their advantages over traditional cooling systems.

1. Introduction

The trends of industrial thermal power engineering in the oil industry, requires intensification of processes in new high-heat and high-pressure combustion chambers and nozzle apparatuses of TPP. The authors of contemporary works [1-18] are still interested in heat transfer of homogeneous and non-homogeneous porous coatings and from specially designed wicks to increase heat transfer. This study addresses issues of thermohydraulic characteristics of boiling in porous media - bubble size, departure frequency, and nucleation density, with the help of thermocouples and high-speed cameras. It has been found the most effective and reliable way to intensify heat exchange during boiling is the use of capillary-porous coatings. In this way, the operating mode and technological indicators are improved at a low price [19-25].

2. Experimental setup

Heat flows of the boiling process in porous structures were calculated using photo cinematography and holographic monitoring (figure 1) [19].
Figure 1. Experimental setup for studying the vaporization processes by optical methods: 1 - laser; 2 - cine-photocamera SKS-1M; 3 - cooling element with porous structure [22].

The elements from figure 2 are: the surface temperature, \( t_{st} \), liquid consumption \( m_1^l \), \( m_1 \), \( m_2 \), \( m_{dis} \), \( m_c \), \( m_{c.w} \), \( m_{air} \), steam \( m_{st} \). Where: \( t \) - tank, dis – discharge, c – condensate, c.w. – condensing water, a – air. Temperatures of liquid, \( t_w \), \( t_l^e \), \( t_l^{in} \), \( t_l^{out} \), steam \( t_{st} \), electrical insulation \( t_{el} = t_{dif} \).

Figure 2. Principal scheme: TSD – transformer, CT – universal current transformer, W – power meter, V – voltmeter, A – ammeter, VR – voltage regulator, G – galvanometer, R – rotameter, NV – needle valve

The TSD welding transformer supplied energy to the heater. The current is measured with a current transformer type CT-6M2. The voltage of the heater is measured with a voltmeter type D523. The measurement accuracy is ± 1% for the current and ±1.6% for the voltage drop, respectively. With the VR protection heater, the current is maintained from 200 to 1200 A. TL-4 thermometers with a range of 0-50° C and 50-100° C are used respectively to measurement of the liquid and the environment temperature. Drainage fluid and vapor temperatures are measured using L-type thermocouples. Its free ends are placed in melting ice. The thermocouple is connected to switches type PP-63. Consumption of coolant and/or circulating fluid is read by rotameters RED. Consumption of drained liquid and condensate is read with a pressure scale device \( 0.5 \times 10^{-3} \) l and the filling time with a stopwatch type S-P-1b with a set 0.1 s/div [4].
The error in measuring the flow rate of liquid by rotameters must not exceed $\pm$ 3% but by volume method up to $\pm$ 2\% [20]. The discrepancy between the material balance between the coolant flow rate and the drainage and condensate flow rate is up to $\pm$10\%. More information on measurement conditions and techniques is published in references [22-24].

3. Calculation of heat flows

As follows from photo-cinematic and holographic observations [20-23], the dynamic of the vapor phase after nucleation of a bubble of critical size proceeds with the participation of the evaporating micro-layer of liquid under the vapor bubble. Under certain conditions, followed by the forming of a "dry spot" [20]. The value of the detachment radius $R_0$ is calculated by the equation provided in [25]. After some "silence" of the generation, a new bubble spontaneously emerges, with growth time controlled by the coolant flow rate decreasing and occurs with a more intense heat supply from the thin superheated liquid layer. The incommensurable of the "silence" time with the growth period is also indicative of an existing superheated pulsating micro-layer of fluid, the durability, and stability of which is extended by the action of forces.

When the liquid boils in a big volume, the phase of bubble growth relate to forces $\sim (10$ up to $100)\times10^{-3}$ s, while in a porous structure this value is about ten times less. In both systems, the nucleation and detachment stages of the bubble are negligibly small and have explosion character. [24]. The silence time in the periodic cycle of bubble formation during large-volume boiling could be $\sim 0.1$ s and be commensurate with the bubble growth time.

As a result occurs the exposure of a nucleus of critical size. The increasing bubble surface slows down its expansion in the depression. Heat is transferred mostly through the micro-layer of liquid under the vapor bubble, which is cone-shaped with a "dry" spot of the bubble. The main evaporation process occurs at the base of the "dry" spot. The micro-layer of the "dry" spot change insignificantly during bubble growth [19] due to the inflow of fresh portions of coolant transported by gravitational and capillary forces, due to the lifting forces acting on the bubble. The inertial forces are hapened in the initial time of bubble grow and decrease before bursting. The bubble acquires a spherical shape of the "dry" spot and it’s reduced, and the shape of the micro-layer changes significantly. Bubble detachment is depends from surface and hydro-gasodynamic resistance forces, and the excess of liquid. Thus through the underheating and flow rate of the liquid is created and the control of thermohydraulic characteristics is realized.

When the bubble upper boundary touches the porous structure surface, the bubble collapses. According to [19], a hole appears at the point of contact, through which steam flows out of the bubble into the vapor volume. A liquid wave starts to spread along the surface. The process described proceeds $10^{-7}$ up to $10^{-6}$ s, i.e., it has an explosive wave character, character, the same as the "birth" of a vapor nucleus, with the outflow of the light phase. This phenomenon has been used to draw an analogy with explosive processes in elliptical systems [23], to separate energy processes for wave energy and compressed gases. The phenomenon described was accompanied by intense ejection of liquid droplets [22], but no disruption of the porous system was observed.

A relatively cold liquid rushes into the depression and some of the vapor there condenses until the vapor and liquid temperature at the limit of their contact is equalized. When the liquid excess is large, all the vapor can condense and the bubble-forming effect will stop. The inflow of accumulated heat in the wall will lead to a new generation cycle of nucleation. During one cycle the wall temperature below it will change which shows the high heat transfer intensity. Reducing the heat flow will return the system to its previous state.

The bubble shape growing is a form between spherical and hemispherical. The spherical bubble takes place when water boils at high pressures ($P>1$ bar), and the hemispherical one - at low pressures ($P<1$ bar). At the initial moment the surface tension forces take over and the bubble transform into cells in the ball shape.

The bubble rapid growth at the initial moment is set by a fixed value of $q$. The described model allows describing experimental the mechanism and to obtain calculated dependences for liquid boiling of a porous structure [19]:
a) surface boiling area:

\[ q = 1.1 \cdot 10^{-3} \cdot \lambda_{ef} \left( a J_{a} R_{0}^{-2} v^{-1} \right)^{0.5} \cdot \Delta TFK^{-1} \quad (1) \]

\[ F = (1 + \cos \beta)^{0.57} \cdot m^{0.25} \cdot \left( \frac{h_{v}}{h_{0}} \right)^{0.5} \cdot \left( \frac{h_{v}}{h_{0}} \right)^{0.27} \]

where \( J_{a} = \frac{c_{p}(T_{w} - T_{p})}{r_{p} v} \) is Jacob’s number and \( K = 1 + \left[ \frac{(p c \lambda)_{l}}{(p c \lambda)_{w}} \right]^{0.5} \) – wall coefficient.

This equation is useful for calculating cooling systems for steam and gas turbines in TPP [19, 23].

b) area of bubble boiling developed:

\[ q = 0.0862 \cdot r \cdot \rho \cdot \alpha \cdot J_{a} \sqrt{n \cdot F \cdot K^{-1}} \]

\[ (2) \]

c) evaporation crisis\(( \mathbf{b}_{e} > 0.28 \cdot 10^{-3} \text{ m}):\)

\[ q_{cr} = 0.0347 \cdot r \cdot \left( g \cdot (\rho_{l} - \rho_{v}) \cdot \rho_{v} \cdot D_{0}^{0.5} \cdot \left( \frac{h_{r}}{h_{0}} \right)^{0.3} \cdot \left( \frac{h_{r}}{h_{0}} \right)^{0.5} \cdot (1 + \cos \beta)^{0.6} \right) \]

The analysis shows that the steam bubble growth on the wall leads to an increase in temperature and decrease in efficiency.

In [19-25] are summarized the experimental data on different thermohydraulic processes. Calculated of heat load (equation 1-3) on temperature pressure, liquid excess, pressure, thermophysical properties of the wall, orientation, geometric characteristics of porous structures and heat exchange wall are obtained. These equations allow for calculations in the design of TPP.

4. Data analysis

Two types of evaluation criteria have been approved - dimensionless number \( \text{St} \) and dimensionless temperature. The obtained three areas were distinguished: the area of high values of the dimensionless temperature - from 1.3 to 1.7; the area of moderate values - from 1.0 to 1.3 and the area of low values - from 1.0 and less. The dimensionless temperature has been adopted as a criterion for this division.

**Area of low values.** Decreasing the temperature to such values is unacceptable and when the temperature reaches the value of 1.0 a boiling crisis occurs.

**Area of moderate values.** The operating range for capillary-porous materials (c.p.m.) from 1.0 to 1.3. At small values, the priority of single-layer c.p.m. becomes obvious, from 1.0 to 1.2. This type of heat exchange process is characteristic of combustion chambers. If the range 1.2-1.25 values is reached, the priority of one or another c.p.m. should be determined. For this purpose, an additional criterion \( \text{St} \) is used. In the range 1.25-1.3 the priority of another c.p.m. should be found on the base on the distribution of the intensity of the heat transfer process over the evaporator surface and possible wall temperature fluctuations. The area of the dimensionless temperature is divided into three-zone: from 0.1 to \( 1.0 \times 10^{-3} \text{ m}; \) from 1.0 to \( 2.0 \times 10^{-3} \text{ m} \) and from \( 2.0 \times 10^{-3} \text{ m}. \)

5. Conclusions

The correlation between the heat transfer coefficient and the wall temperature is inversely proportional. For a traditional (flow-through) cooling system, the allowable temperature fluctuations are no more than 10 K (2.5-3%), with all heat flow accumulated by the wall, leads to temperature overstress in the material. Any increase in temperature requires an increase in cooling. For a porous cooling system, the maximum allowable temperature fluctuations, the cooling system compensates for independently, and with a high degree of response to changes in the specific heat flow, are up to 138.9 K, and the loss of heat flow will be 1.174 MW/m². Considering that the working temperature is 190.15 °C for heat flow of 13 MW/m² the system's temperature reserve is 73%, and the heat flow reserve is 11%.  

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6. References

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