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PROCEDURE FOR CALCULATING THE THERMOACOUSTIC PRESSURE FLUCTUATIONS AT BOILING SUBCOOLED LIQUID

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Проведено дослідження термоакустичних явищ в парогенеруючих каналах системи охолодження теплонавантажених пристроїв. Досліджувані режими охолодження характеризуються поверхневим кипінням теплоносія, яке виникає внаслідок високих теплових потоків на поверхні охолодження, і великими недогрівами до температури насичення ядра потоку. В таких умовах можливе виникнення в каналах охолодження високочастотних пульсацій акустичного тиску. Встановлено, що виникнення термоакустичних коливань здатне привести до утворення стоячої хвилі в каналі, однією з умов формування якої є наявність границі відображення хвиль. Представлено математичну модель, що описує генерацію термоакустичних коливань в каналі охолодження. Вважається, що коливання з високою амплітудою виникають внаслідок резонансу, що спостерігається при збігу частоти вимушених коливань парових бульбашок з власною частотою коливань пароріднинного стовпа або їх гармоніками. Для розрахунку амплітуди коливань тиску в каналі отримана залежність, яка враховує в'язкісну дисипацію енергії і втрати енергії на кінцях каналу. Показано, що при наближенні до резонансу внесок об'ємної в'язкості в коефіцієнт в'язкістного поглинання зростає. Встановлено, що для досліджуваних умов втратами енергії на стінках каналу і втратами в приграничному шарі можна знехтувати. Проведено розрахунки амплітуди термоакустичних коливань тиску для умов, що відповідають реальним процесам в каналах охолодження з поверхневим кипінням. Представлена методика пропонується до використання при проектуванні систем рідинного охолодження теплонавантажених приладів, для яких режими охолодження припускають істотний недогрів теплоносія до температури насичення і поверхневе кипіння

Ключові слова: канал охолодження, поверхневе кипіння, термоакустичне коливання тиску, резонанс, дисипація, в'язкість рідини

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1. Introduction

The development of powerful electrical devices requires complete information about operational stability of the thermal mode control system. Advancement of modern nuclear power engineering, MGD generators and high-power lasers, as well as microwave electronics, necessitated the creation of cooling systems that should operate at heat flow densities reaching 10^8 W/m^2. One of the most effective ways to cool a heating surface is to divert heat under a bubble boiling mode of the subcooled-liquid [1, 2]. Application of such a technique makes it possible to obtain high energy flow densities, and at low temperature heads [3]. However, heat exchange under such conditions is often accompanied by stable high-frequency pressure pulsations in the channel, which are self-oscillatory in character. It could be considered an established fact that the occurrence of thermoacoustic vibrations could lead to the formation of a standing wave in the channel, one of the conditions for whose formation is the wave reflection boundaries. Therefore, the boiling of an subcooled liquid at the forced convection in pipes and channels has been given much attention, which is associated with a possibility of the emergence of a high level of acoustic pressure.

2. Literature review and problem statement

Paper [4] studied the boiling of an subcooled liquid flow by visualization. It was shown that at high underheating of the liquid to a saturation temperature the steam bubbles were destroyed upon a contact with the flow core. The authors found that the surface boiling in a microchannel makes it possible to divert the heat flow reaching 14.41 MW/m^2 at a moderate increase in the temperature of the wall. This proves the feasibility of applying liquid cooling with a surface boiling. A similar approach was used by authors in [5]. It was found that the underheating significantly reduced the detachable diameter of bubbles, and at the same time there were significant fluctuations in pressure both at inlet and outlet, as well as surface temperature. However, the reported experimental data cannot be used to predict the level of fluctuations in cooled channels. Authors of [6] note that subcooling boiling has a wide scope of industrial applications, including nuclear reactors. They note that subcooling boiling has a wide scope of industrial applications, including nuclear reactors. Similar problems arise in metallurgical production when casting large-capacity ingots [7, 8]. In these processes, there were water pressure fluctuations in the cooling channels of pallet hollows when molten steel crystallized (a temperature of ~1,550 °C) that were not considered in studies [7, 8]. In the channel with a boiling heat carrier there are acoustic effects, manifested in the form of "white" noise or steady harmonic oscillations of pressure. It is noted that acoustic vibrations, or pressure waves, are characterized by high frequency (10–100 kHz) and are observed at liquid boiling, at volumetric and film boiling. By studying patterns in heat transfer during boiling, the authors of [9] found that the amplitude of pressure wave fluctuations could be much higher than stationary pressure in the system. The results reported in [9] indicate an intense vibration caused by the boiling of an subcooled liquid. The resulting vibration is associated with the rapid growth and collapse of steam bubbles at a high load of the heat flow, with bubbles located near the heated surface. It is noted in [10] that it is necessary to improve the scientific description and understanding of the vibrational aspect regarding the subcooled flow boiling to progress in the future development of compact heat exchangers, which require an appropriate level of safety during transmission of a high heat load. Spectral analysis revealed that the frequency of acoustic vibrations varied in the range of 0–1,000 Hz; resonance phenomena were observed. The findings suggest that physical perceptions of harmonic oscillations in the channel should be based on the idea of the dynamics of bubbles. In paper [11], the process of bubble generation appears intermittent and is divided into three stages – the waiting phase, the growth phase, and the collapse phase. Results reported in [12] show that even with a slight underheating of the liquid, the absolute majority of bubbles break down very close to the surface after the detachment. Under large underheating, steam bubbles are destroyed even before they break away from the heat-transmitting surface. This causes the formation of sound waves. It is assumed that the formation of a typical bubble under an subcooled boiling causes an excitatory force of the order of 10^{-4} N [12]. To assess conditions for the emergence of thermoacoustic oscillations, data on heat exchange are needed, in particular, dependences are needed to calculate the heat efficiency factor at boiling with underheating. An experimental study [13] of the heat exchange process at subcooled boiling made it possible to justify the use of correlations taking into consideration the refining parameters for calculating heat transfer coefficients. Similar studies were carried out by authors in [6]; their range of the studied heat efficiency ratios covered areas from a forced convection and boiling with underheating to the developed bubble boiling. It was noted that at large mass expenses G and low heat flows q the effect of underheating is not obvious, as the dominant mechanism of heat transfer is the forced convection. For the assigned G and q, heat transmission ratios gradually increase with a reduction in subcooling. The reported empirical dependences take into consideration the effect of an underheating temperature, heat flow, fluid consumption on the heat efficiency factor and could be used for compiling a procedure to calculate the amplitude of thermoacoustic oscillations.

The classic notions of thermoacoustic phenomena [14] imply that harmonic sound waves occur in the pipe exposed to an external harmonic force in the form of the self-coordinated vapor bubble dynamics. Theory [14] for the case of forced vibrations states that it does not matter to narrow pipes whether the source of vibrations is distributed over the cross section of the pipe or is concentrated at one point. The issue on experimental determination of the amplitude of pressure fluctuations in the channel has remained unresolved due to the complexity of a measurement system, so the analytical modelling based on the available data on the boiling subcooled liquid seems rational.

The scientific literature gives examples of applying the Maxwell's viscous-plastic relaxing environment to describe the viscous behavior of a material with energy dissipation; this model, however, yields a good convergence when there is a significant elastic component [15]. In the processes of a fluid's flow at medium pressures, elastic compression of the environment could be neglected.

Despite the large body of research into the high-frequency thermoacoustic self-oscillations, patterns of their occurrence are not sufficiently studied. There are no scientifically sound methods to control and prevent the thermoacoustic phenomena, which does not make it possible to devise a procedure for calculating the amplitude of pressure acoustic fluctuations and to assess the danger to an article. As a result, the design calculations of thermal regime systems are incomplete. All that necessitated the study into the nature of thermoacoustic phenomena at boiling subcooled liquid and compiling a calculation procedure for the amplitude of pressure fluctuations.

3. The aim and objectives of the study

The aim of this study is to devise a procedure for calculating the amplitude of pressure thermoacoustic fluctuations in the cooling channels of radio-electronic equipment under a mode of surface boiling subcooled liquid.

To accomplish the aim, the following tasks have been set:

– to construct a physical model that would describe the process of the emergence of thermoacoustic pressure fluctuations in steam-generating channels at boiling subcooled liquid;

– to build a mathematical model for calculating thermoacoustic pressure fluctuations taking into consideration energy losses and to analyze the contribution of viscous dissipative phenomena and losses at the ends of a channel;

– to run a computational experiment employing the resulting dependences to calculate the amplitude of pressure fluctuations.

4. Modeling the thermoacoustic phenomena at boiling an liquid in cooling channels

4. 1. Physical model of the evolution of thermoacoustic pressure fluctuations at boiling subcooled liquid

The following provision was accepted as the basis of a physical model. Conditions for the generation of large amplitude oscillations are the resonance phenomena observed when the frequency of forced vibrations coincides with the vapor column's natural frequency or their harmonics:

$$mf_f = nf_0, \quad (1)$$

where f_0 is the natural frequency of vibrations of the vapor-fluid column in a channel; f_f is the forced frequency of oscillations; *m* is the harmonica of forced oscillations, *n* is the harmonica of natural oscillations.

Boiling fluid bubbles that are distributed across the pipe's surface could be considered harmonic oscillators. As the heat flow increases, the sound signal spectrum changes from the shape of "white" noise to harmonic oscillations, which are caused by the self-coordinated growth and destruction of steam bubbles. To explain this phenomenon, an object under study is treated as a complex oscillatory system consisting of part *A* – a set of sound emitters determined by the number of steam centers, and part *B* – an elastic column of liquid in which acoustic vibrations propagate. The amplitude of acoustic vibrations emitted by a bubble is proportional to $\frac{R_m^3}{\tau}$, where R_m is the maximum radius of a bubble, τ is its lifetime. Thermoacoustic oscillations in subsystem *B* evolve in the form of a system of standing waves, whose amplitude is determined by the vibrational energy of the entire set of oscillators – steam bubbles. In the region of "white" noise, the subsystems *A* and *B* are not related. With the increase in heat flow *q*, the action frequency of steam centers in subsystem *A* increases, and at some value *q* the natural frequency of vibrations of subsystem *B* approaches the frequency of forced fluctuations of subsystem *A*. In this region, there occurs the synchronization of actions of the steam centers due to the effect exerted by the vibrations of subsystem *B*. The bubbles emit a positive pressure pulse into environment *B* during the compression period of a standing wave, thereby rocking it. With the further increase in *q* the frequencies begin to diverge, the connectivity of the system is disrupted. The role of bubbles in system *A* is reduced to maintaining the energy of a standing wave.

Actual cooling modes of radio-electronic equipment of the examined class are characterized by such parameters at which the vapor content in the cooling channels is small and the steam bubbles are almost absent in the stream. Even at high heat loads, a liquid's temperature in the channel remains well below the saturation temperature, and the vapor bubbles formed at the surface of heat exchange condense when in contact with the flow core. In addition, taking into consideration that the heat carrier is degassed, one could assume that a sound wave propagates in the channel with a clean liquid.

4. 2. Mathematical modeling of thermoacoustic phenomena taking into consideration energy losses

4. 2. 1. Mathematical notation of thermoacoustic oscillations in a channel

The model is based on the assumption about the resonance nature of thermoacoustic oscillations in the system with distributed parameters (a steam-generating channel) under the influence of sources of forced vibrations (acting steam centers). The main assumptions in the adopted model are:

– the movement of a heat carrier is regarded as one-dimensional;

– heat flows along the channel, thermal resistance of the wall and its deformation are not taken into consideration;

– the volume of steam bubbles varies according to the sinusoidal law:

$$V(t)=V_{\max}\cdot\exp(i\omega t);$$

– the thermal-physical properties are constant.

Propagation of waves does not depend on the orientation of a pipe or its bending degree.

The latter assumption holds for "very narrow" pipes, for which the following condition is satisfied: a pipe's diameter $d<<\lambda/2$.

The steam-generating channel is presented schematically in Fig. 1.

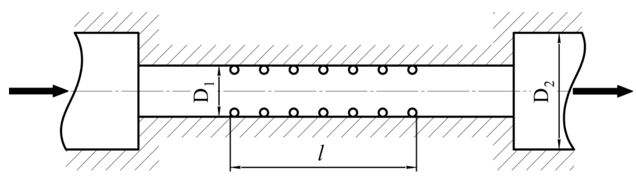

Fig. 1. Schematic of a steam-generating channel

The calculation procedure for oscillations amplitude is as follows:

1) The Reynolds number is calculated and, depending on a current mode, the heat efficiency ratio is determined according to data from [6].

2) The value of a heat flow diverted in the process of forced convection:

$$q_c = \alpha\left(T_w - T_l\right). \quad (2)$$

3) The values of a supplied heat flow *q* and q_c are compared. If $q_c>q$, the boiling does not occur, the heat is diverted through convection.

Lengthwise the channel, the liquid's temperature rises, approaching the starting boiling point $T^{s.b}$, determined from the following dependence:

$$T^{s.b} = \frac{T_w \cdot q}{\bar{\alpha}}. \quad (3)$$

In this case, a coordinate of the start of boiling is determined:

$$x_e = \frac{\left(T^{s.b} - T^{inlet}\right)\cdot G \cdot c_p}{\pi \cdot d \cdot q}, \quad (4)$$

where T^{inlet} is the temperature of a liquid at the inlet to a channel.

When $q_c<q$, it is accepted that the boiling takes place along the entire length of the heated section and temperature of the start of boiling corresponds to the liquid's temperature at the inlet:

$$T^{s.b}=T^{inlet}.$$

4) Frequency of bubbles formation is [2]:

$$f=\frac{68\left[1-\exp\left(-0.027\cdot\Delta t_w\right)\right]+132}{\rho^{0.81}\cdot d_{\max}},\qquad(5)$$

where temperature head Δt_w is the difference between the temperature of the channel's wall t_w and the mean temperature of the liquid t_l ($\Delta t_w=t_w-t_l$); $d_{\max}$ is the detachable radius of a bubble.

Circular frequency is:

$$\omega=2\cdot\pi\cdot f.\qquad(6)$$

The number of steam-generating centers, according to [16], is:

$$Z=7.5\cdot 10^{-8}\left[r\cdot\rho_g\cdot\Delta t_w/\left(\sigma\cdot t_w\right)^2\right],\qquad(7)$$

where a temperature head during water boiling Δt_w is determined from empirical dependence [17]:

$$\Delta t_w=0.33\cdot q^{0.3}\cdot p^{-0.15},\qquad(8)$$

where p is the pressure in a liquid, bar. Dimensionality of the magnitude is related to the specificity of processing experimental data.

5) Calculating acoustic characteristics.

– Calculate the mean speed of sound:

$$\bar{C}=L/\left(\frac{x_e}{C_l}+\frac{L-x_e}{C_b}\right),\qquad(9)$$

where C_l is the speed of sound in a clean liquid, C_b is the speed of sound in a two-phase environment.

A value of the speed of sound in a two-phase environment depends on the region of vapor content values.

$$C_b=\sqrt{\frac{\gamma\cdot p}{\rho_l\cdot\phi(1-\phi)}},\qquad(10)$$

where γ is the adiabatic Poisson coefficient, p is the static pressure in a channel. At $\varphi<10^{-3}$, that is in the region of small numerical values for vapor content:

$$C_b=\sqrt{\frac{1}{\beta_{ef}\cdot\rho}},\qquad(11)$$

where β_{ef} is the effective compression factor [16]:

$$\beta_{ef}=\beta_l\left(1-\phi\right)+\beta_g\cdot\phi,\qquad(12)$$

where β_l, β_g are the liquid and steam-phase compression ratios, respectively.

– Wave number K:

$$K=\omega/\bar{C}.\qquad(13)$$

– The natural frequency of vibrations of a vapor-fluid column in a channel:

$$f_o=\bar{C}/2L.\qquad(14)$$

6) Pressure oscillation amplitude:

$$p(x)=E\cdot\left\{\begin{array}{l}\sin\left(K\left(L-x\right)\right)\cdot\left[\cos\left(Kx\right)-\cos\left(Kx_e\right)\right]-\\-\sin\left(Kx\right)\cdot\left[\begin{array}{l}\cos\left(Kx\left(L-x_e\right)\right)-\\-\cos\left(K\left(l-x\right)\right)\end{array}\right]\end{array}\right\},\qquad(15)$$

where magnitude E is determined from the following expression:

$$E=\frac{\pi\cdot d\cdot\rho_l\cdot\bar{C}\cdot V_g Z}{K\cdot S\cdot\sin\left(KL\right)}.\qquad(16)$$

The dependence for calculating the amplitude of pressure fluctuations was derived by the authors of the work by assuming that the growth and destruction of vapor bubbles occur in sync, which was observed during experiments in the near-resonance region.

In the course of the computational experiment based on a given procedure, the conditions for an increase in the oscillation amplitude were established. Fig. 2 shows an example of the estimation curve in the part where the resonance phenomenon was observed, and the oscillations amplitude increased dramatically.

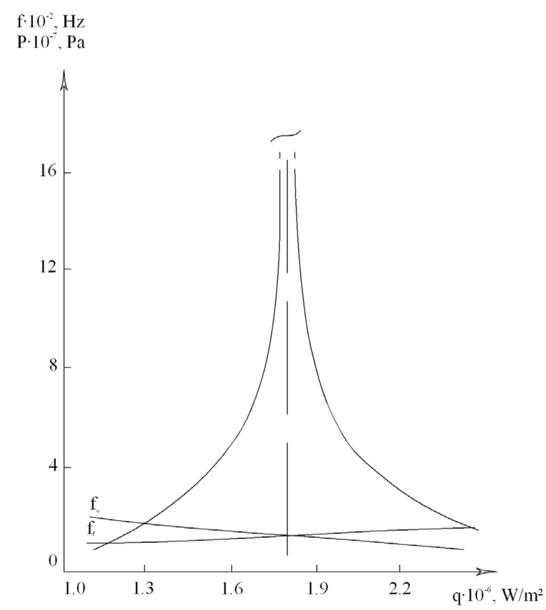

Fig. 2. Typical estimation curve of acoustic pressure dependence on heat load in the resonance region

As shown by Fig. 2, resonance arises when the values for natural f_0 and forced f_f frequencies converge. Excluding dissipative phenomena at the point of coincidence of the frequencies the amplitude of vibrations tended to infinity. That determined the need to take into consideration the main mechanisms of energy loss in the mathematical model. The principal mechanisms of energy losses in a sound wave are associated with viscous dissipation and thermal conductivity, losses at the inlet and outlet of the channel and the scattering of sound on steam bubbles.

4. 2. 2. Analysis of viscous absorption of vibrational energy

In a general from, the viscous absorption rate is determined from dependence [18]:

$$\alpha_v = \frac{\omega^2}{2\cdot\rho_l\cdot C^3}\left[\frac{4}{3}\eta+\eta'+a\left(\frac{1}{c_v}-\frac{1}{c_p}\right)\right], \tag{17}$$

where a is the thermal diffusivity factor; η is the ratio of a first (shear) viscosity; η′ is the ratio of a second (volumetric) viscosity.

A factor of dynamic viscosity is included as shear viscosity in equation (17). A second viscosity, or volumetric viscosity, is defined as internal friction when transferring the impulse in the direction of movement. In our case, its impact should be taken into consideration as the compression and expansion processes do occur. The effect of volumetric viscosity is described in detail in [18]: at compression and expansion (which is typical of sound waves) internal processes begin in a liquid, seeking to restore the thermodynamic equilibrium. These processes are irreversible and are accompanied by energy dissipation, whose intensity depends on the ratio of compression and expansion rate to relaxation time. The analytical expression that shows the wave is fading is the representation of wave number K in the form of a complex vector, whose material part determines the change in phase with distance, and imaginary part is the absorption coefficient α_η. If a sound wave period is large compared to the time of relaxation τ (ωτ<1), the equilibrium is almost keeping pace with the fluctuations in density in a sound wave and the wave number is described by the following equation:

$$K=\frac{\omega}{\bar{C}_0}+i\frac{\omega^2\tau}{2C_0^3}\left(C_\infty^2-C_0^2\right), \tag{18}$$

where C_∞ is the sound speed value, at which the influence of volumetric viscosity exceeds the shear one.

The viscous absorption coefficient is, in line with [18]:

$$\alpha_\eta=\frac{\omega^2\tau}{2C_0^3}\left(C_\infty^2-C_0^2\right). \tag{19}$$

For the case of large frequencies (ωτ>>1), the absorption coefficient increases and another dependence holds for K [18]:

$$K=\frac{\omega}{\bar{C}_\infty}+i\frac{C_\infty^2-C_0^2}{2\tau C_\infty^3}. \tag{20}$$

In some cases, the volumetric viscosity may exceed a first one. This is observed in processes characterized by a high-frequency change in fluid density. Differences in these formulae are due to that at relatively low frequencies, the equilibrium is able to follow the change in density in the sound wave. In another case, when the period of oscillations is less than the relaxation time or commensurate with it, the equilibrium does not have time to recover.

The time of acoustic relaxation in a liquid is equal to the time of free run of molecules $\tau=l/v$, and the length of free run l could be taken to be equal to the Lennard-Jones potential parameter, while the maximum valid value of particle velocity $v=1/(\rho\cdot C_0)$. The dependence to determine it then takes the following form:

$$\tau=2.91\cdot10^{-10}\cdot\rho\cdot C_0. \tag{21}$$

As shown by calculations, under the examined conditions, the relaxation time for a pure liquid (water) is $\tau\approx3\cdot10^{-4}$ s, frequency ω for a first harmonic of the forced vibrations has an order of 10^{-3} s^{-1}. Then, $\omega\cdot\tau\approx0.3$, hence, the case of relatively low frequencies is considered.

To determine C_∞ the following dependence is proposed:

$$C_\infty=1.018\cdot C_0. \tag{22}$$

Now, for the low-frequency case under consideration, the wave number takes the form:

$$K=\frac{\omega}{\bar{C}_0}+i\frac{0.018\cdot\omega^2\cdot\tau}{C_0}. \tag{23}$$

Under conditions close to resonance, the influence of viscosity is increasingly manifested, limiting the increase in amplitude due to the emergence of a complex component. In this case, the maximum value for the amplitude of pressure fluctuations in the resonance region is:

$$\left|p(x)\right|_{\max}=\frac{B}{sh\left(\alpha_\eta l\right)}. \tag{24}$$

This dependence determines the value of the maximum pressure amplitude in a channel, which cannot be exceeded due to the structural features of the liquid. In fact, the amplitude will be even less, because not all the mechanisms of a wave energy loss are taken into consideration. The need to account for energy losses due to thermal relaxation is discussed in [19]. It was shown that once viscous relaxation is taken into consideration, the thermal one can be neglected.

The scientific literature provides examples of using a Maxwell's viscous-plastic relaxing environment to describe the viscous behavior of a material with energy dissipation; this model, however, yields a good convergence when there is a significant elastic component [15]. In the processes of a fluid's flow at medium pressures the elastic compression of the environment could be disregarded.

4. 2. 3. Determining energy loss at the open ends of a channel ("soft caps")

For the case of "soft caps", the distribution of amplitudes of pressures and particle velocities along the length of a pipe is the same as for the pipe with "hard caps." However, pressure nodes should rest on absolutely soft caps. Since the actual ends of a pipe may not be completely soft, there is a phase shift in the distribution of velocities and pressures, determined by α_{cap}, which should take into consideration the loss of energy when leaving the examined channel for a pipeline:

$$p=\sin\left(Kx-\alpha_{cap}\right), \tag{25}$$

$$v=\frac{1}{i\rho C}\cos\left(Kx-\alpha_{cap}\right). \tag{26}$$

For conductivity $Y=v/p$, the same for both ends of the channel, the loss factor on caps is:

$$\alpha_{cap}=\operatorname{arcctg}\left(i\rho CY\right). \tag{27}$$

Conductivity of the open end is related to the ratio of reflection via the following dependence:

$$Y=\frac{1}{\rho C}\cdot\frac{1+\vartheta}{1-\vartheta}, \tag{28}$$

where ϑ is the reflection factor. Because a wave passes through both ends of the channel, the total reflection factor is:

$$\vartheta = \frac{S_2 - 3S_1}{S_1 + S_2}. \qquad (29)$$

Now, the factor of losses on "soft caps" takes the form suitable for calculations:

$$\alpha_{cap} = \operatorname{arcth}\left(\frac{S_2 - S_1}{2S_1}\right), \qquad (30)$$

and the maximum value for an amplitude, taking into consideration losses on caps is:

$$\left|p(x)\right|_{\max} = \frac{B}{\operatorname{sh}\left(\operatorname{arcth}\left(\frac{S_2 - S_1}{2S_1}\right)\right)}. \qquad (31)$$

Dependence (31) shows that losses at the ends of a channel are determined solely by the cross-sectional area of inlet channel S_2 and the area of the examined cooling channel S_1.

4. 2. 4. Other mechanisms of acoustic energy losses in a channel

In addition to the above processes causing energy loss, one should consider losses on the walls of the channel and losses in a boundary layer. When considering the propagation of oscillations in a channel, which has elastic properties, the dissipative effect is taken into consideration by known formula of Zhukovsky. It was derived that when taking into consideration the properties of a vapor bubble at the surface of the channel with a diameter of $4 \cdot 10^{-3}$ the deviations in the value of the speed of sound are less than 5 %. Therefore, the contribution of wall elasticity could be neglected.

Another mechanism of acoustic energy dissipation is associated with viscous effects in a boundary layer between the channel's wall and a moving liquid. The propagation of viscous shear waves is accompanied by a significant absorption of energy, and with a decrease in the diameter of the channel this magnitude increases. The thickness of the acoustic boundary layer is determined from dependence:

$$\delta_{ac} = \sqrt{\frac{2\eta}{\omega\rho}}. \qquad (32)$$

At a 1 kHz frequency, for example, δ_{ac}=4.1·10^{-3} cm. This layer could only form at a section of the channel that is not boiling. According to the results from calculations for water, at a frequency of 1 kHz the absorption in a pipe of radius 4 mm would be about 9.4 Db/cm. Therefore, the viscous absorption in a boundary layer should be taken into consideration only for long channels in which the boiling area is insignificant.

4. 2. 5. Total losses of acoustic energy on viscosity and conductivity of the ends of a channel

Joint accounting of the derived coefficients leads to the following notation of the oscillation amplitude:

$$p(x) = \frac{B}{\sqrt{\sin^2(Kl) + \operatorname{sh}^2\left(\alpha_\eta l + \alpha_{cap}\right)}}. \qquad (33)$$

Based on the resulting equation, a calculation program was drawn up in order to assess, during a computational experiment, the degree of influence exerted by the various mechanisms of acoustic energy dissipation in a channel.

4. 2. 6. Analytical study of the effect of viscous and end effects on acoustic energy dissipation

The calculations were performed for the following physical and heat hydraulic characteristics of the process:

1. The length of the channel, l=0.667 m.
2. The diameter of the channel, d=0.04 m.
3. Coordinate of the heated section's beginning, x=0.5 m.
4. Coordinate of the heated section's end, x=0.625 m.
5. Pressure at the inlet to the channel, p=2.2·10^5 Pa.
6. Water temperature at the inlet, t=78 °C.
7. Water velocity in the channel, w=6.37 m/s.
8. Specific heat flow, q=(1.5·10^6...8.0·10^6) W/m^2.

These characteristics correspond to the actual physical quantities characteristic of the cooling system channels in heat-loaded radio-electronic equipment. A given example is interesting because there were three resonance peaks in the study range: at a 179 Hz frequency at the heat flow density q=3.14·10^6 W/m^2 and q=6.7·10^6 W/m^2. The presence of two identical frequency values at different loads is explained by patterns of change in the speed of sound in a channel. At first, when the supply of a heat flow is increased (which causes the increase in the volumetric vapor content), the speed of sound decreases. Then, at vapor content φ=0.005 (in this case, such a vapor content was observed at q=5.6·10^6 W/m^2), the shape of dependence changes: an increase in vapor content leads to an increase in the speed of sound. Fig. 3 shows the dependences of natural and forced frequencies, as well as the speed of sound, on specific heat flow.

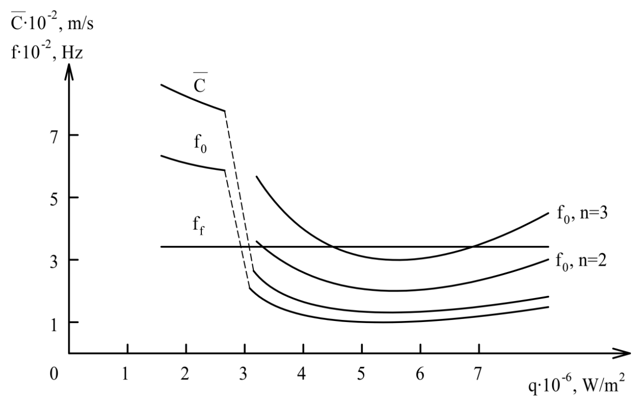

Fig. 3. Dependence of speed of sound C, natural f_0 and forced f_f frequencies on heat flow q

A first resonance emerges when the frequency of forced oscillations coincides with a second harmonica of natural frequencies, a second and a third – when a first harmonic of forced oscillations coincides with a third harmonica of its natural. As shown by calculations in line with dependences (1) to (14) and (33), at a first resonance a standing wave propagates in the channel with the second mode of vibrations, at a resonance on the third harmonica – the third mode.

Fig. 4 shows the dependence of amplitude of pressure fluctuations on heat flow when accounting for viscous absorption, losses at the ends of the channel, and joint accounting of their effect.

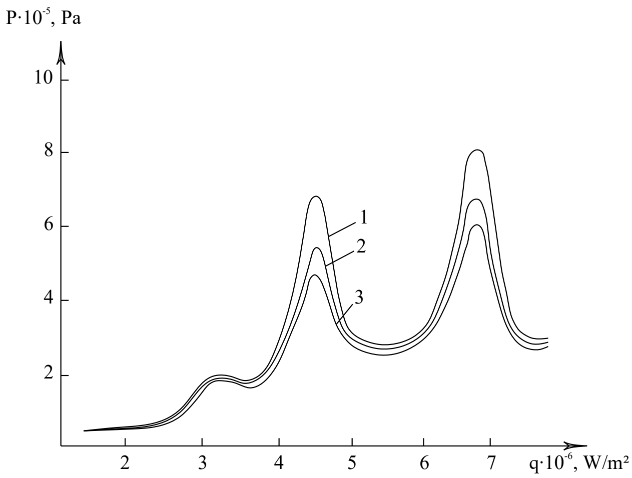

Fig. 4. Dependence of amplitude of acoustic pressure oscillations on heat flow taking into consideration the dissipative effects: 1 — accounting for viscous dissipation; 2 — accounting for ends' losses; 3 — joint effect. Distance from the inlet to the channel, x=0.1 m

It is evident that viscous dissipation contributes less to dissipative phenomena than losses at the ends of the channel. The devised procedure makes it possible to identify regions of mode parameters at which resonance phenomena arise and to assess their danger to heat-loaded radio-electronic devices.

5. Discussion of results of studying the thermoacoustic oscillations in steam-generating channels

The result from calculating the thermoacoustic vibrations (TAV) of pressure is the match between the devised procedure and the mathematical and physical models according to which, when the forced and natural frequency of vibrations coincide, there occurs resonance. However, without taking into consideration the dissipative phenomena the amplitude of pressure fluctuations in the resonance region tends to infinity (Fig. 2). When accounting for the energy loss to viscous dissipation and at the ends of the channel, it becomes possible to quantify the amplitude of fluctuations. The contribution of dissipative phenomena is taken into consideration in the dependence derived by authors (33). In this case, one should take into consideration not only the contribution of shear viscosity η, but also that of the volumetric one, which takes into consideration the dissipation of energy at volumetric deformations of the environment under conditions for the propagation of acoustic waves. Calculation of the frequency of natural and forced fluctuations makes it possible to predict the regions of resonance phenomena (Fig. 3). The above calculation example predicts the occurrence of three resonance peaks at the frequency of forced oscillations of 179 Hz at the heat flow density q=3.14·10^6 W/m^2 and q=6.7·10^6 W/m^2. A first resonance arises when the frequency of forced oscillations coincides with the second harmonica of natural frequencies, a second and a third – when the first harmonica of forced oscillations coincides with the third harmonica of its natural. The change in the amplitude of acoustic pressure, calculated from dependence (33), shows the existence of three peaks (Fig. 4) corresponding to the resonance regions in Fig. 3.

Analysis of the calculation results, illustrated in Fig. 4, shows that the dissipation of thermoacoustic oscillation energy due to viscous effects is less than the loss of energy at the ends of the channel. The proposed calculation procedure for TAV amplitude is unique and has no analogs in the scientific literature. This algorithm includes an empirical dependence to calculate the overheating temperature derived for the case of water boiling (8). When using other heat carriers, the appropriate calculation formula should be included. The mathematical model that underlies the TAV calculation procedure is limited by the conditions for surface boiling at significant underheating to the temperature of a fluid's flow core saturation. Under these conditions, vapor bubbles are located exclusively at the heat-diverting surface and are not observed in the main stream. It is also assumed that the surface section occupied by vapor bubbles is almost equal to the length of the channel. Such conditions make it possible to neglect viscous dissipation in a boundary layer. At relatively large sections of the channel that are not occupied by vapor bubbles, it is necessary to refine the dependence for calculating thermoacoustic vibrations (33) taking into consideration the thickness of the acoustic layer (32). In the future, it seems appropriate to consider conditions for the emergence of TAV in the steam-generating channels at nuclear power plants, for which both bubble boiling and emulsion modes are possible. In this case, dissipative phenomena would increase, the conditions for TAV propagation change, which requires adjustment of the procedure.

Our procedure is proposed for use in the design of liquid cooling systems for heat-loaded devices, for which cooling modes imply a significant underheating of the heat carrier to a saturation temperature. The current study could be advanced to refine the dependence for the frequency of bubble formation and the speed of sound in a two-phase environment. It is also necessary to experimentally determine the acoustic pressure along the length of a steam-generating channel. However, given the complexity of working section fabrication, this issue is still debatable.

6. Conclusions

1. A physical model has been developed describing the process of the emergence of thermoacoustic pressure fluctuations in steam-generating channels at boiling subcooled liquid. It is taken into consideration that the occurrence of thermoacoustic oscillations of pressure of high amplitude is associated with resonance phenomena. Resonance occurs when the frequency of forced oscillations coincides with the vapor column's natural frequency or their harmonics. Boiling fluid bubbles are regarded as harmonic oscillators interacting with the core of the flow.

2. We have constructed a mathematical model for calculating thermoacoustic pressure fluctuations taking into consideration energy losses, which assumed a one-dimensional movement of the heat carrier and the sinusoidal law of change in the volume of vapor bubbles. A computational experiment has shown that without taking into consideration the dissipative phenomena at the point of coincidence between the forced and natural frequencies the amplitude of vibrations tends to infinity. It was determined that the main mechanisms of energy losses in a sound wave are associated with viscous dissipation and thermal conductivity, losses at the inlet and outlet of the channel, and the scattering of

sound on steam bubbles. We have derived a viscous absorption coefficient, which takes into consideration the contribution of shear and volumetric viscosity. When approaching the resonance, the contribution of volumetric viscosity increases significantly. Energy losses at the ends of the channel are accounted for by the introduction of a loss factor on "soft" caps. The loss of energy on the walls of the channel and in a boundary layer under the examined conditions could be neglected. A dependence has been proposed to calculate the amplitude of pressure fluctuations, which takes into consideration the contribution of dissipative phenomena.

3. We have calculated the amplitude of thermoacoustic vibrations for the source data corresponding to the actual regime characteristics for cooling channels in the heat-loaded radio-electronic equipment. It was established that there were three resonance peaks in the study range: at a frequency of 179 Hz at the heat flow density q=3.14·10^6 W/m^2 and q=6.7·10^6 W/m^2. The existence of two identical frequencies at different loads is explained by the peculiarities of change in the speed of sound in the channel. The calculated amplitude takes into consideration viscous absorption and losses at the ends of the channel.

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