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HYDRODYNAMIC AND THERMAL CHARACTERISTICS OF A TWO-PHASE TRANSPIRATION COOLING SYSTEM

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Abstract—This paper generalizes static characteristics of heat engineering processes which tend to instability. Hydrodynamic and thermal characteristics of a two-phase transpiration cooling system are plotted based on the solution of a nonlinear closed system of differential equations describing the hydrodynamics and heat transfer of a filtering coolant with unknown position of the surface of its equilibrium phase transformation. The study of aperiodic stability of two-phase transpiration cooling with different static characteristics yields the same results, while the use of each characteristic separately makes it possible to detect permissible disturbances of an appropriate characteristic parameter. With the use of a set of static characteristics, permissible disturbances of all of the characteristic parameters can be found and the peculiarities of stable two-phase transpiration cooling can be revealed.

INTRODUCTION

Improving the operation of different installations requires new and reliable heat protection systems. Among those under development, the system of cooling a heated porous surface with a fluid undergoing phase transformation seems to be the most promising. However, the essential advantages of the method remain to be incompletely utilized because of its appreciable shortcoming, *viz.* the potential instability of two-phase transpiration cooling (Duwez & Wheeler 1948; Polyaev & Sukhov 1969; Gomes, Curry & Johnston 1971; Frost, Reth & Buchanan 1972; Reth & Frost 1972; Wayner & Bankoff 1965; Pai & Bankoff 1965, 1966; Koh & del Casal 1968; Koh *et al.* 1970).

Maiorov & Vasiliev (1973), analyzed the solution of a nonlinear closed system of differential equations describing a physical model of two-phase transpiration cooling, and found one of the reasons for its instability. Furthermore, the peculiarities of this process were studied using static characteristics similar to the characteristics of other unstable heat engineering systems.

A two-phase transpiration cooling system is considered unstable if after a small perturbation from a developed state, it does not achieve a new developed state near the initial one, and its parameters undergo monotonic change of large amplitude. The reason for this phenomenon lies in steady state laws described by equations containing no time derivatives. Static characteristics reflect these laws and their study makes it possible not only to draw a conclusion as to aperiodic stability of the system, but also to find the values of permissible slow variations of the parameters necessary for the stability of the process to survive.

STATIC CHARACTERISTICS OF UNSTABLE HEAT ENGINEERING SYSTEMS

At present, aperiodic instability of heat engineering devices is known to be of two types, *viz.* boiling crisis and abrupt high amplitude variation of the working fluid flow rate in heated channels. Static characteristics of both processes are peculiar for their ambiguity. Liquid pool boiling is characterized by a boiling curve of the density of heat flux from a heating surface to fluid vs the difference of surface and saturation temperatures. The

† Professor A. V. Luikov passed away on 28 June 1974.

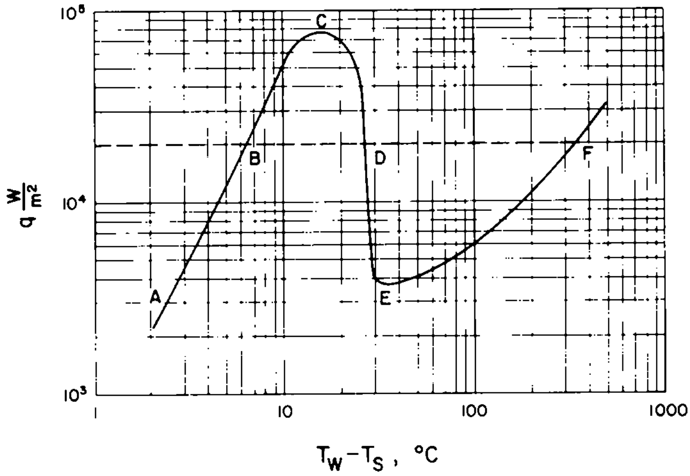

Figure 1. Characteristic curve of liquid nitrogen pool boiling (Mert & Clark 1964).

curve for nitrogen boiling is shown in figure 1 (Mert & Clark 1964). A hydrodynamic characteristic of a heated channel, its resistance vs the flow rate of working fluid, is plotted in figure 2 (Petrov 1960; Tong 1967). Both curves have a descending section CE. For liquid boiling the curves show an increase of heat flux from the surface with decreasing temperature difference during transition from film boiling to nucleate boiling. For hydrodynamic characteristic *I*, the increase corresponds to an increase in the channel resistance due to the higher vapour content of a two-phase flow.

The stability of wall cooling with boiling is characterized by the type of intersection of the boiling curve and the curve giving the relationship between the density of the heat flux from external sources, q_{sup}, and the wall temperature, T_w. The analytical form of the stability condition is (Adiutori 1964):

$$\frac{dq_{sup}}{dT_w} < \frac{dq_{removed}}{dT_w}. \qquad [1]$$

In practice, processes are frequently used where the heat flux to a surface is quite or almost independent of its temperature $dq_{sup}/dT_w = 0$. Examples of such processes are electric and radiative heating and heat generation in fuel cells.

In such cases, transient boiling with $dq_{rem}/dT_w < 0$ cannot be realized because of disturbance of the stability. The critical heat flux (corresponding to the point *c*) is the boundary of stable and reliable work.

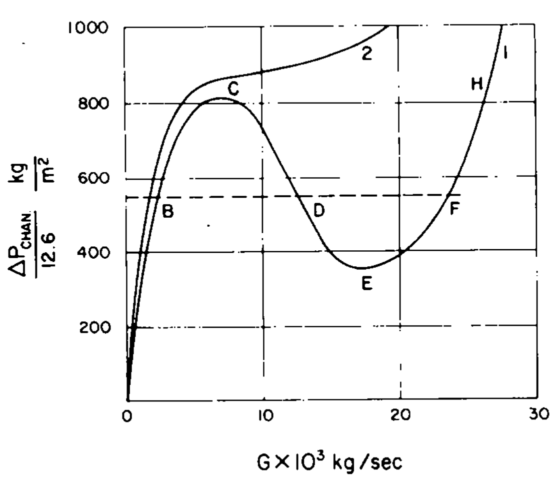

Figure 2. Hydrodynamic characteristics of a uniformly heated channel (Petrov 1960).

A steady state for transient boiling can be achieved only when a heat supply is provided to a plate with heat transfer coefficient h higher than the slope of the boiling curve

$$h = -\frac{dq_{sup}}{dT_w} = -\frac{d}{dT_w}[h(T_\infty - T_w)] > -\frac{dq_{rem}}{dT_w}.$$

Vapour condensation provides such a heat source. However, due to a limited heat transfer coefficient and the essentially destabilizing effect of the wall thermal resistance, the steady states are achieved only at the beginning and end of the transition region, where the slope of the boiling curve is slight (Berenson 1962; Hale & Wallis 1972). The experimental data available for the whole transient process were obtained by the method of unsteady cooling by boiling of an initially heated fluid, resulting in continuous variation of the temperature of its surface (Hert & Clark 1964).

In a heated channel, fluid motion is stable if the slope of its hydrodynamic characteristic at the working point is greater than that of the characteristic of the pumping device (Boure *et al.* 1971)

$$\frac{d\Delta P_{channel}}{dG} > \frac{d\Delta P_{ext}}{dG}. \qquad [2]$$

where G is the flow rate, ΔP_{ext} is the pressure drop created by an external source on the channel, and $\Delta P_{channel}$ is the internal demand pressure drop across the channel.

If a constant pressure drop $d\Delta P_{ext}/dG = 0$ is maintained on the heated channel, then the descending section of the channel characteristic $d\Delta P_{channel}/dG = 0$ is a region of unstable operation. Under these conditions aperiodic instability can be avoided by eliminating the ambiguity of the hydrodynamic characteristics. This is achieved by adding additional resistance in the form of a choke plate at the inlet to the channel, where single phase fluid flows. The plate resistance adds stability to the process. The hydrodynamic characteristic of a stable system is of the form of curve 2 in figure 2.

It is very important to note that the descending section on the hydrodynamic curve is characteristic not only of systems with phase transformation of the working fluid. It appears that the shape similar to curve 1 in figure 2 is also peculiar to the hydrodynamic characteristic of a gas-cooled plate subject to surface heating (Gomez *et al.* 1971). Here ambiguity of the characteristic is due to the temperature variation of dynamic and kinematic viscosities of a gaseous coolant resulting in the fact that three possible mass flow rates of injected fluid correspond to some definite pressure drop on the plate with considerably different mean wall temperatures. The region of stable and reliable work is presented by branch EH on the curve where effective control of the coolant flow is possible due to the change in the pressure drop across the plate. The value of the flow rate is sufficient to keep the temperature of the material within the permissible range. Decreasing the pressure drop below the value corresponding to the point E leads to a sharp decrease in the coolant flow rate, causing burnout of the wall. Unlike the system with phase transformation of a working fluid, no artificial measures can add unambiguity to the hydrodynamic characteristic of a gas-cooled porous plate. The ambiguity of the hydrodynamic characteristic of a system with gaseous heat transfer agent was first discovered during the investigation of helium cooling of porous heat generating elements (Green 1952).

It follows from the above that the static characteristics allow determination of the value of permissible slow fluctuations of the parameters (density of the heat flux supplied, for boiling and pressure drop on the plate, for gas transpiration cooling) at which stable and reliable operation of the system lasts.

As to the two-phase transpiration cooling, only permissible fluctuations of the initial temperature of the coolant can easily be found using the static characteristic plotted by (Maiorov & Vasiliev 1973). Temperature dependence of the pressure in an expected region of phase transformation is the most natural result of solution of the system of equations describing the process. Due to the specific operation of the system, the estimation of permissible fluctuations of other parameters such as the density of an external heat source and pressure drop across the inlet plate are of equal importance. Such a problem can be solved with the use of the characteristics including the parameter whose permissible variation is to be found.

HYDRODYNAMIC CHARACTERISTIC OF TWO-PHASE TRANSPIRATION COOLING

This characteristic relates the pressure drop across the plate and coolant flow rate at different values of the coordinate of the surface of equilibrium phase transformation. This relationship is to be found for the case of boiling equilibrium.

The pressure for the interface region is sought through the resistance of the vapor section

$$P_l - P_1 = \delta\left[\alpha G v''(1 - l) + \beta \frac{G^2}{\rho''}(1 - l)\right] \quad [3]$$

where P_l is the supply pressure; P_1 is the ambient pressure; δ is the wall thickness; α and β are viscous and inertia resistance coefficients, respectively; v is the kinematic viscosity; l is the dimensionless coordinate of the phase transformation region; ρ is the density and $''$ denotes physical properties of saturated vapour.

In the region of equilibrium phase transition, instead of calculating the temperature T_l

$$(T_l - T_\infty)\frac{c'}{r} = \frac{q\exp[k_2(l - 1)]}{Gr} - 1 \quad [4]$$

it is more convenient to use the enthalpy of generated dry saturated vapour i_l'':

$$i_l'' - c_m T_\infty = \frac{q\exp[k_2(l - 1)]}{G} \quad [5]$$

where T_l is the temperature in the phase transition region; T_∞ is the fluid temperature; c' is the specific heat of the saturated liquid; c_m is the heat capacity of a liquid; G is the coolant mass flow rate; r is the latent heat; k_2 is the permeability and q is the density of the external heat flux. The use of the vapour enthalpy and mean heat capacity of fluid in the temperature range of 0°C–T_∞ simplifies the expression and increases its accuracy.

The expression obtained is rewritten as

$$i_l'' - i_{l=1}'' = \frac{q\exp[k_2(l - 1)]}{G} - (i_{l=1}'' - c_m T_\infty). \quad [6]$$

This expression can be used to calculate the enthalpy of the saturated vapour and the pressure in the region of phase transformation, based on the saturation parameters at the external surface of the wall where the ambient pressure is prescribed. Since both states are equilibrium states of saturation, the differences of their pressures and enthalpies of dry vapour are related by a single-valued expression:

$$i_l'' - i_{l=1}'' = \psi(P_l - P_1) \quad [7]$$

depending on the kind of coolant.

A linear approximation of this relationship at the point with the pressure equal to the ambient pressure

$$i_l'' - i_{l=1}'' = \left.\frac{\mathrm{d}i''}{\mathrm{d}P}\right|_{P=P_1}(P_l - P_1) \quad [8]$$

allows [3] and [6] to be combined into one analytical transcendental equation to determine the coolant flow rate G with equilibrium phase transformation in the region with the characteristic length l:

$$q\frac{\exp[k_2(l-1)]}{G} - (i''_{l=1} - c_m T_\infty) = \left.\frac{di''}{dP}\right|_{P_1} \delta\left[\alpha G v''(1-l) + \beta\frac{G^2}{\rho''}(1-l)\right]. \qquad [9]$$

It should be noted that the value of the coolant flow rate enters into the criterion of transpiration cooling.

At the plate, the necessary pressure drop is calculated by the known flow rate from the expression

$$P_0 - P_1 = \delta\left\{\alpha G[v'l + v''(1-l)] + \beta G^2\left[\frac{l}{\rho'} + \frac{(1-l)}{\rho''}\right]\right\}. \qquad [10]$$

It is the successive solution of [9] and [10] for all values of the parameter l with its further elimination that specifies the shape of the hydrodynamic characteristic of the two-phase transpiration cooling system. Physical properties of the coolant, external heat flux and initial fluid temperature are included as constant parameters of the process.

Solution of [7] or its simplified analytical variant [9] involves a considerable amount of calculation. However, there is one particular case, which is rather promising for practical application, when the amount of calculation may be reduced. In accordance with available data (Vukalovich *et al.* 1969), the enthalpy of saturated water vapour remains constant within 4.5 per cent for a wide range of pressure from 0.1 to 12 MN/m^2 (i'' = const). The governing relationship [9] for determination of the flow rate in the water-cooled system is simplified to

$$q\frac{\exp[k_2(l-1)]}{G} = (i'' - c_m T_\infty). \qquad [11]$$

The results of solution of [10] and [11] are presented in figure 3. For comparison with the data (Maiorov & Vasiliev 1973) the same constant parameters of systems 1–3 are used: water is the coolant; ambient pressure $P_1 = 1.0$ MN/m^2; wall thickness $\delta = 5 \times 10^{-3}$ m; porosity $\pi = 0.2$; resistance coefficients $\alpha = 3.5 \times 10^{13}$ m^{-2} and $\beta = 1.2 \times 10^8$ m^{-1}. No changes are introduced into the heat flux density and effective thermal conductivity of the vapour section.

The hydrodynamic characteristics are plotted in normalized coordinates. The pressure drop is referred to the value $(P_0 - P_1)^* = 0.125$ MN/m^2 that provides the equilibrium phase transition in the region with the coordinate $l = 0.95$. The appropriate Reynolds number of the coolant flow is Re = 0.1. The coolant flow rates are compared with the fluid flow rate $G_1 = 4.34$ kg/m^2 sec under the pressure drop $(P_0 - P_1)^* = 0.125$ MN/m^2 in a viscous flow (Re = 0.0).

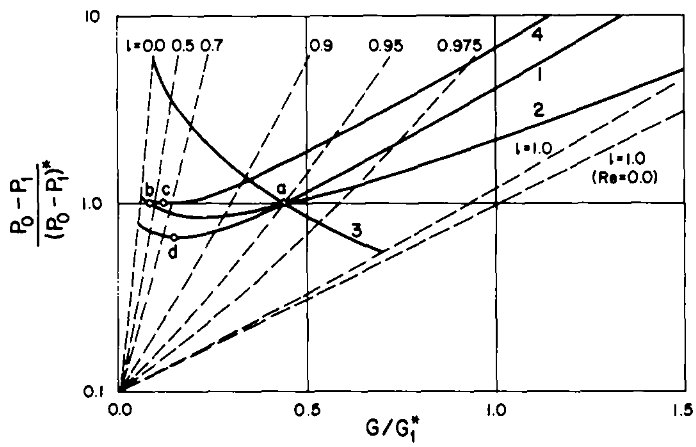

Figure 3. Hydrodynamic characteristics of system of two-phase transpiration cooling: 1 – $q = 3.5 \times 10^7$ W/m^2; $\lambda_2 = 0.65$ W/m deg; 2 – $q = 1.8 \times 10^7$ W/m^2; $\lambda_2 = 1.0$ W/m deg; 3 – $q = 8.3 \times 10^6$ W/m^2; $\lambda_2 = 2.6$ W/m deg; 4 – $q = 8.3 \times 10^7$ W/m^2; $\lambda_2 = 0.65$ W/m deg.

Inclined dashed lines l = const relate the coolant flow rate and pressure drop across the plate with the fixed position of the region of phase transformation. In particular, the line $l = 1.0$ determines the resistance of the plate to a single-phase fluid flow, and the line $l = 0.0$, to a dry vapour flow. Smooth deviation of dashed lines from the initial rectilinear shape manifests the increase of the effect of inertia resistance when the flow changes from viscous (Darcy law) to transient one.

Hydrodynamic characteristics of all of the three systems intersect at one working point a, which is due to the appropriate choice of the parameters λ_2, q. In systems 1 and 2, however, the working point lies at the ascending section, in system 3, at the descending one. As prescribed, a constant pressure drop $P_0 - P_1$ is maintained across the plate. Therefore, in accordance with stability condition [2], systems 1 and 2 are stable at the working point a, while system 3 is unstable. There is one more working point b in system 2 corresponding to unstable operation.

The conclusions as to the stability of the system of two-phase transpiration cooling obtained from the studies of its hydrodynamic characteristic coincide with the results of the analysis of the intersection of the saturation curve and the line of pressure vs temperature in the supposed region of phase transformation.

In addition, the hydrodynamic characteristic of a stable system may be used to find the value of permissible slow fluctuations of the pressure drop at the plate. Thus, in system 1 a decrease in the supply pressure involves smooth regression of the boiling region up to the state corresponding to the stability boundary (point d). Thereafter, a boiling region rapidly shifts to the internal surface of the plate resulting in a considerable decrease of the coolant flow rate. The point d stands for the minimum pressure drop for system 1.

Strictly speaking, the pressure drop due to supply pressure fluctuations only can be accurately determined within the frames of the assumed model where physical properties of both coolant phases are constant and chosen in a state of saturation with the ambient pressure prescribed. Variation of the ambient pressure changes the physical properties of the coolant and, hence, the shape of the hydrodynamic characteristic. We may assume, however, that the error due to permissible fluctuations of the dominant state does not exceed the error inherent in the model itself. Therefore, based on the characteristic derived, permissible fluctuations of the ambient pressure may also be estimated within the same accuracy.

THERMAL CHARACTERISTIC OF A TWO-PHASE TRANSPIRATION COOLING SYSTEM

Permissible fluctuations of the external heat flux in the two-phase transpiration cooling system may be found using the thermal static characteristic, the density of external heat flux vs the coordinate of the surface of equilibrium phase transformation. Transformation of [5] yields the following basic relation

$$q = (i_l'' - c_m T_\infty) G \exp[k_2(1 - l)] \qquad [12]$$

which includes the coolant flow rate G and enthalpy of saturated vapour i_l'' depending on the coordinate l of the phase transition surface.

The pressure drop at the plate being constant, the coolant flow rate changes sharply with motion of the phase transition surface, and the expression defining its value is of the form

$$G = \frac{\mu'}{\beta/\alpha}\frac{m}{2n}\left(-1 + \sqrt{1 + 4\,\mathrm{Re}\frac{n}{m^2}}\right) \qquad [13]$$

where $\mathrm{Re} = [(P_0 - P_1)/\delta v'\alpha][(\beta/\alpha)/\mu']$ is Reynolds number of the coolant flow; $m = [l + (v''/v')(1 - l)]$ and $n = [l + (\rho'/\rho'')(1 - l)]$ are auxiliary complices and where v' and

ν'' are the kinematic viscosities of the saturated liquid and vapour, respectively; ρ' and ρ'' are the respective densities; μ' is the dynamic viscosity.

The change of the enthalpy of dry vapour with motion of the boiling region into the porous wall is attributed to the increase in the saturation pressure. Relations for estimation of an increase in the saturation pressure [3] and appropriate change in the enthalpy of saturated vapour [7] or [8] are already available but there are similar equations for saturation parameters of the external surface. Linear approximation of the dependence of the enthalpy of saturated vapour on pressure provides the analytical shape of the unknown static characteristic

$$q = G\exp[k_2(1-l)]\left\{(i''_{l=1} - c_m T_x) + \left.\frac{di''}{dP}\right|_{P_1} \delta\left[\alpha G\nu''(1-l) + \beta\frac{G^2}{\rho''}(1-l)\right]\right\}. \quad [14]$$

If water is the coolant, the second summand in brackets turns to zero since the enthalpy of saturated water vapour remains constant in a wide range of pressures.

Figure 4 furnishes thermal characteristics 1–3 of those systems whose hydrodynamic characteristics 1–3 are plotted in figure 3. In all of the systems pressure drop is kept constant and equal to $P_0 - P_1 = 0.125\,\mathrm{MN/m^2}$. In each of the systems the heat flux is referred to the value q^* at which the phase transition region has the coordinate $l = 0.95$. The appropriate dimensionless values are presented in figure captions.

Mutual disposition of all of three curves at the working point a is similar to the intersection of hydrodynamic characteristics. Here, the working point a lies within the descending section of curves 1 and 2 and ascending section of curve 3. System 2 has the second working point b. For thermal characteristics the descending section is a stability region, which follows from the physical nature of the process. Thus, in stable system 1 a small increase in the heat flux over the developed state a causes regression of the interface which recovers its initial state with removal of the disturbance. In unstable system 3, an initial increase of external heat flux results in continuous motion of the boiling region to the internal surface of the plate, since in all of the intermediate states a smaller amount of heat supplied to the external surface is needed for equilibrium phase transformation. The

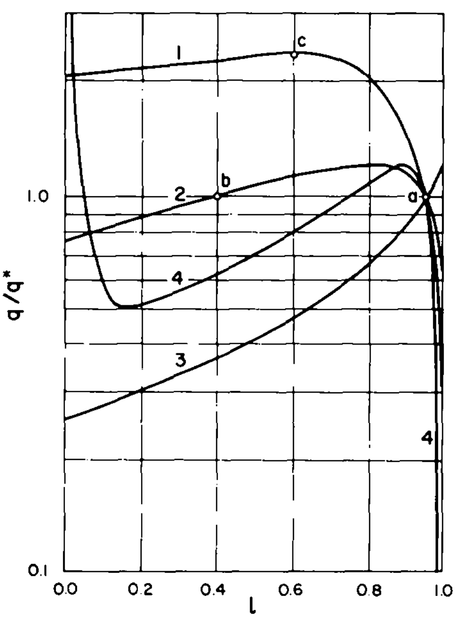

Figure 4. Thermal characteristic (density of external heat flux versus position of surface of equilibrium phase transformation) of a system of two-phase transpiration cooling: (1) $q^* = 3.5 \times 10^7\,\mathrm{W/m^2}$; $\lambda_2 = 0.65\,\mathrm{W/m\,deg}$; (2) $q^* = 1.8 \times 10^7\,\mathrm{W/m^2}$; $\lambda_2 = 1.0\,\mathrm{W/m\,deg}$; (3) $q^* = 8.3 \times 10^6\,\mathrm{W/m^2}$; $\lambda_2 = 2.6\,\mathrm{W/m\,deg}$; (4) data presented by (Rubin & Schweitzer 1972).

thermal characteristic being applied, the condition of stability of two-phase transition cooling assumes the form

$$\frac{dq}{dl} < 0. \qquad [15]$$

It should be pointed out that this condition works well for the system with a coolant of any kind. Plotting of the characteristic is simplified for water. Similar condition for this particular case is obtained without its generalization from the solution of a linearized unsteady system of equations but with a more simple statement of the whole problem (Rubin & Schweitzer 1972). One of these results is presented at curve 4 of figure 4. The presence of one more stability region for small values of l at curve 4 is due to an unrealizable boundary condition of constant temperature at the internal surface of a permeable wall.

Using the thermal characteristic, the limited value of external heat flux in a stable system can easily be found. In system 1 smooth increase of the heat flux involves gradual regression of the boiling region to the position with the coordinate $l = 0.6$ (point c). Thereafter even small disturbance of the external heat flux results in boiling at the internal surface. The hydrodynamic characteristic of system 1 with limited heat flux also reflects the fact that the system is at the stability boundary (curve 4 in figure 3 is tangential to the line of constant pressure drop at the point c).

The study of aperiodic stability of the system of two-phase transpiration cooling with the use of different static characteristics gives similar results. Each of these characteristics makes it possible to reveal readily the disturbance of one of the characteristic parameters permissible in a stable system. Simultaneous application of the characteristics allows the permissible fluctuations of all of the characteristic parameters to be found and the peculiarities of the process to be revealed comprehensively.

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Résumé—On généralise dans cet article les caractéristiques statiques des processus de génie thermique sujets à instabilité. On trace les caractéristiques thermique et hydrodynamique d'un système de refroidissement par transpiration diphasique, basées sur la solution d'un système formé d'équations différentielles non linéaires décrivant l'hydrodynamique et le transfert de chaleur pour un réfrigérant filtrant, la position de la surface du changement de phase à l'équilibre étant inconnue. L'étude des instabilités apériodiques du refroidissement par transpiration diphasique conduit aux mêmes résultats, cependant que l'utilisation séparée de chaque caractéristique rend possible la détermination des perturbations tolérables d'un paramètre caractéristique approprié. L'utilisation d'un jeu de caractéristiques statiques permet de déterminer les perturbations tolérables de tous les paramètres caractéristiques et de mettre en évidence les particularités du refroidissement par transpiration diphasique stable.

Auszug—In dem Ausfsatz werden statische Charakteristiken von waermetchnischen Prozessen mit einer Neigung zu instabilem Verlauf verallgemeinert. Hydrodynamische und thermische Charakteristiken eines Systems der Zweiphasen-Verdunstungskuehlung werden aufgezeichnet, gemaess den Ergebnissen der Loesung eines geschlossenen nichtlinearen Systems von Differentialgleichungen. Diese beschreiben das hydrodynamische und das Waermeuebergangsverhalten in einem Filtrierkuehler mit unbekannter Lage der Oberflaeche der Gleichgewichtsphasenumwandlung. Das Studium der aperiodischen Stabilitaet der Zweiphasen-Verdunstungskuehlung mit verschiedenen statischen Charakteristiken ergibt die gleichen Resultate, waehrend Anwendung jeder einzelnen Charakteristik fuer such allein es ermoeglicht, zulaessige Schwankungen eines passend gewaehlten Kennparameters festzustellen. Mit Hilfe einer Reihe von statischen Charakteristiken koennen die zulaessigen Schwankungen aller Kennparameter gefunden, und so die Besonderheiten stabiler Zweiphasen-Verdunstungskuehlung klargelegt werden.

Резюме—Обобщены статические характеристики теплотехнических процессов, известных склонностью к неустойчивой рабсте.

Построены гидродинамиуеская и тепловая характеристики системы двухфазного пористого охлаждения на основе результатов решения нелинейной замкнутой системы дифференциальных уравнений, описывающей гидродинамику и теплообмен фильтрующегося охладителя при неизвестном положении поверхности его равновесного фазового превращения.

Исследование апериодической устойчивости двухфазного пористого охлаждения с помощьы разлических статиуеских характеристик приводит к одинаковым результатам, но в тоже время использование по отдельности каждой характеристики позволяет обнаружить допустимые колебания соответствующего определяющего параметра. Применение совокупности статических характеристик дает возможность найти допустимые возмущения всех определяющих параметров и выявить особенности устойчивого двухфазного пористого охлаждения.