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THERMAL DEFORMATION OF UNEVENLY HEATED JET HEAT EXCHANGERS

S. B. Koshelev, V. V. Kharitonov, and G. É. Ter-Avakimov

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Expressions are obtained for evaluating stresses and strains in an unevenly heated jet heat exchanger having the form of a metallic disk with a cellular structure.

Jet heat exchangers are widely used in different areas of technology [1-3]. Such heat exchangers may be significantly deformed at high thermal loads and temperature gradients, and their structural materials may experience intolerably high thermal stresses. The goal of the present study is to evaluate thermal stresses and strains in jet heat exchangers having the form of a metal disk of thickness H and diameter 2b with a cellular structure (Fig. 1) in the case of nonuniform axisymmetric heating. The cellular structure makes it possible to intensify heat transfer due to the finning effect and retains the bending stiffness of the exchanger.

Formulation of the Problem. The exact simultaneous solution of the differential equations of heat conduction (second order) and thermoelasticity (fourth order) is possible only in certain simple cases [4, 5]. Thus, various approximations of the theory of thermoelasticity are used in engineering calculations [4-7]. Below we represent the resulting strain $\omega(r)$ of the heated surface ($z = H/2$) approximately in the form of the sum of the thermal expansion $\omega_{expn}(r)$ (thickening) of the heat exchanger along the z axis and the bending of its middle plane ($z = 0$), $\omega_{bnd}(r)$. We will evaluate the bending by using an approximation of plate theory for a plate with properties which vary through the thickness [5]: the radial coefficient of thermal expansion $\beta_r(z)$ and the modulus of elasticity $E_r(z)$.

According to [4, 5], the so-called thermal force and thermal moment are sources of bending stresses and strains:

$$N_t(r) = \frac{1}{1-\nu}\int\limits_{-H/2}^{H/2} \beta_r(z)\,E_r(z)\,T(r,\ z)\,dz,$$
$$M_t(r) = \frac{1}{1-\nu}\int\limits_{-H/2}^{H/2} \beta_r(z)\,E_r(z)\,T(r,\ z)\,z dz, \qquad (1)$$

which are determined by the temperature field T(r, z) in the metal. Henceforth, the temperature is reckoned from the temperature of the fluid at the inlet of the heat exchanger ($T_{in} = 0$).

After fourfold integration of the equation of thermoelasticity of circular plates [4, 5] over the radius, we obtain

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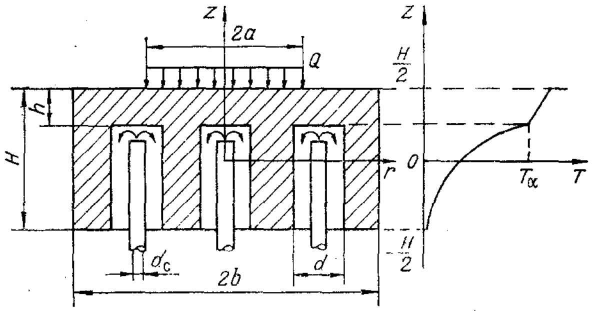

Fig. 1. Diagram of jet heat exchanger and axial temperature field in it.

$$\omega_{bnd}(r)=\frac{1}{D}\left[\int_r^b \frac{F(r)}{r}\,dr+\varkappa\frac{F(b)}{2}\left(1-\frac{r^2}{b^2}\right)\right], \tag{2}$$

where $F(r)=\int_0^r rM_t(r)dr$; $D=\frac{1}{1-\nu^2}\int_{-H/2}^{H/2} E_r(z)z^2dz$ is the bending stiffness of the heat exchanger; the coefficient $\kappa = -1$ for a heat exchanger fastened about its contour, while $\kappa = (1 - \nu)/(1 + \nu)$ for a heat exchanger which is not fastened. Equation (2) gives fundamental solutions of the theory of thermoelasticity of plates as a special case [4, 5]. A plate rigidly fastened at its edges does not bend if the temperature of the plate changes only through the thickness [the thermal moment in (2) is not dependent on the radius]; a rigidly fastened or freely supported plate does not bend if its temperature changes only over the radius ($M_t = 0$); if the temperature changes linearly through the thickness of a freely supported plate, then it bends so that there are no stresses in it. Equations (1) and (2) are valid in the case of a slight nonuniformity of the plate through its thickness. The thermal stresses are easily evaluated in the uniform heating of a heat exchanger which is not fastened at its contour [4]:

$$\sigma_{rr}=-\frac{\beta_r E_r T(z)}{1-\nu}+\frac{N_t}{H}+\frac{12M_t}{H^3}z. \tag{3}$$

The normal expansion (thickening) of the heat exchanger is of the following order of magnitude:

$$\omega_{expn}(r)\approx\varkappa_2\int_{-H/2}^{H/2}\beta_z(z)\,T(r,\ z)\,dz, \tag{4}$$

where the coefficient κ_2 may change from 1 (at low stresses) to $(1 + \nu)/(1 - \nu)$ (at high compressive stresses). Thus, to calculate the thermal stresses and strains, it is necessary to know the two-dimensional temperature field in the structural material of a jet heat exchanger and its mechanical properties.

Mechanical Properties. The heat exchanger shown in Fig. 1 is a two-layer plate consisting of a solid layer and a perforated layer made of the same material. In this case, the coefficients of thermal expansion of both layers are the same in both the radial and axial directions and are equal to the tabulated value $\beta = \beta_r = \beta_z$. The Young's moduli E_r and E_z of the perforated layer differ from the tabulated value of E: $E_z = (1 - \Pi)E$, $E_r = (1 - d/s)E$ [8]. Here the coefficients in parentheses approximately account for the reduction in the elastic modulus in proportion to the relative area over which the forces are transmitted in the axial and radial directions, respectively. In sum, the stiffness of the heat exchanger is equal to

$$D=D_0\Phi(\gamma,\ d/s);\quad \Phi=(1-d/s)+(d/s)\,\gamma\,(3-6\gamma+4\gamma^2), \tag{5}$$

where $D_0 = EH^3/12(1 - \nu^2)$ is the stiffness of the solid plate when d/s or $\gamma = 1$; Φ is the stiffness reduction factor and is dependent on the relative spacing of the perforations s/d and the relative thickness of the heated wall $\gamma = h/H$. It follows from (5) that $\Phi \approx 1 - d/s$ at $\gamma < 0.1$.

Temperature Field. First we find the temperature field in the case of uniform heating (a = b) of the entire surface z = H/2 by a heat flow with the density q, W/m². Ignoring the radial nonuniformity of the temperature of the metal, we can assume the temperature field to be linear within a heated wall having the thickness h (Fig. 1) and also within the zone of action of the heat-transfer agent $(-H/2 \leq z \leq H/2 - h)$. Here, we can ignore the heating of the heat-transfer agent relative to the temperature head within individual cells. The temperature of the metal will change exponentially with depth due to volumetric cooling, in accordance with the model of a heat exchanger in a porous body [9]. Assuming that the lower surface $z = -H/2$ is thermally insulated, we finally obtain the following expression for the unidimensional (averaged over the radius of the heat exchanger) temperature field in the metal

$$T(z) = \begin{cases} T_\alpha + \dfrac{q}{\lambda}\left(z + h - \dfrac{H}{2}\right) & \text{at } \dfrac{H}{2} - h \leqslant z \leqslant \dfrac{H}{2}, \\ T_\alpha \operatorname{ch}\left(\dfrac{z + H/2}{\delta}\right) \Big/ \operatorname{ch}\left(\dfrac{H-h}{\delta}\right) & \text{at } -\dfrac{H}{2} \leqslant z \leqslant \dfrac{H}{2} - h, \end{cases} \quad (6)$$

where $T_\alpha = q/\alpha$ is the temperature of the surface of the heat-exchanger wall undergoing cooling (see Fig. 1); $\alpha = \sqrt{(1 - \Pi)\lambda\alpha_v}$ is the effective heat-transfer coefficient; $\delta = \sqrt{(1 - \Pi)\lambda/\alpha_v}$ is the characteristic depth of heating of the perforated layer.

If the rate of volumetric cooling of the perforated layer is so great that $\delta \ll H - h$ then, in accordance with (6), the temperature of the lower surface $(z = -H/2)$ is nearly the same as the temperature of the fluid.

In the case of nonuniform heating of the heat exchangers, when the heat flux changes according to a certain law q(r) in the radial direction, the temperature of the metal changes both through the thickness and over the radius of the heat exchanger. It was shown in [10] that radial flows of heat from the heated region are small if the size of this region is greater than the thickness h of the wall undergoing rapid cooling. Also, in accordance with (1) and (2), the bending of the heat exchanger does not depend directly on the temperature field but rather indirectly, through integrals of temperature over the thickness and radius. These facts make it possible to use a simplified relation T(z, r) in the form (6) as the first approximation without a large error. Meanwhile, the radial change in temperature fully repeats the profile of the thermal load q(r).

Deformation of the Heat Exchanger. In accordance with Eq. (6), the relation T(z, r) and the thermal force and moment (1), being functions of the radius, change similarly to q(r). Insertion of (6) into (1), with allowance for the difference in elastic moduli in the layers, yields

$$\frac{1-\nu}{\beta E} N_t = q\left(\frac{1 - d/s}{\alpha_v} + \frac{h}{\alpha} + \frac{h^2}{2\lambda}\right), \quad (7)$$

$$\frac{1-\nu}{\beta E} M_t = \frac{qH}{2}\left(\frac{\varphi_1}{\alpha_v} + \frac{h\varphi_2}{\alpha} + \frac{h^2\varphi_3}{2\lambda}\right), \quad (8)$$

where $\varphi_1 = \left(1 - \dfrac{d}{s}\right)\left[(1 - 2\gamma)\operatorname{th}\dfrac{H-h}{\delta} - \dfrac{2\delta}{H}\left(1 - 1/\operatorname{ch}\dfrac{H-h}{\delta}\right)\right]$; $\varphi_2 = 1 - \gamma$; $\varphi_3 = 1 - 2\gamma/3$. If the wall of the heat exchanger is thin $(\gamma = h/H \ll 1)$ and the rate of volumetric cooling is high $(\delta \ll H)$, then $\varphi_2 \approx \varphi_3 \approx 1$, $\varphi_1 \approx 1 - d/s$, and the expressions in parentheses in Eqs. (7) and (8) nearly coincide. The first terms in parentheses in Eqs. (7) and (8) characterize the contribution of the heat-exchanger volume undergoing cooling to the thermal force and moment, while the second and third terms characterize the contribution of the heated wall.

Let us further examine the nonuniform heating of the wall by a uniform heat flow $q = Q/\pi a^2$ within a circular spot of radius $a < b$. We find the following from (4) and (6) for $r \leq a$:

$$\omega_{\text{expn}} \leqslant \varkappa_2 \frac{\beta Q}{\pi a^2}\left(\frac{1}{\alpha_v} + \frac{h}{\alpha} + \frac{h^2}{2\lambda}\right). \quad (9)$$

The first term in the right side characterizes the thermal expansion of the cooling part of the heat exchangers, while the second and third terms characterize the thermal expansion of the heated wall. It follows from Eq. (9) that with a specified thermal capacity Q, expansional strain increases with a decrease in the heated area πa^2. For a heat exchanger with

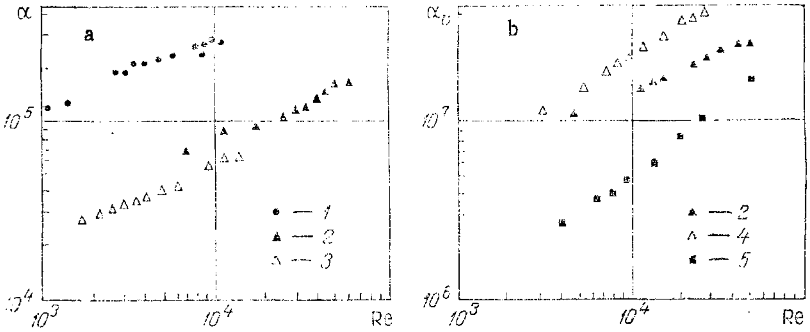

Fig. 2. Experimental dependence of the effective heat-transfer coefficient α [W/(m²·K)] (a) and the volumetric heat-transfer coefficient α_v [W/(m³·K)] (b) on the Reynolds number (for the nozzle) in copper water-cooled heat exchangers with the geometric parameters shown in the table.

TABLE 1. Geometric Dimensions of Heat Exchanger

No. of working section	Inside diam. of cell d, mm	Inside diam. of nozzle d_n, mm	Distance from nozzle to bottom of cell, mm	Porosity of heat exchanger Π
1	1,5	0,7	1,1	0,43
2	7,5	3	1	0,81
3	3	1,5	1,5	0,13
4	3	1,5	1,5	0,36
5	15	6,5	6,5	0,78

a thin wall ($h \to 0$), the strain is inversely proportional to the volumetric heat transfer and is independent of the dimensions of the heat exchanger. In the case of a thick wall, its expansion, proportional to the square of the thickness h^2, is decisive in the total expansion of the heat exchanger.

From (2), (5), (6), and (8) we find the deflection $\omega_{bnd}(0)$ of the heating surface

$$\omega_{bnd}(0) = \frac{a^2 M_t(0)}{2D}\left[\frac{1+\varkappa}{2} + \ln\frac{b}{a}\right] \approx \frac{3(1+\nu)}{\pi}\frac{\beta Q}{\Phi H^2}\left(\frac{\varphi_1}{\alpha_v} + \frac{h\varphi_2}{\alpha} + \frac{h^2\varphi_3}{2\lambda}\right)\left[\frac{1+\varkappa}{2} + \ln\frac{b}{a}\right]. \quad (10)$$

With a specified capacity Q and cooling rate (i.e., α_v and α), the bending strains, as the expansion (9), increases with an increase in the thickness of the heated wall h. Thus, to reduce the strain, it is necessary to reduce the thickness of the wall (to values roughly equal to the diameter d of the cooling channel). The term in the square brackets in Eq. (10) characterizes the effect of the nonuniformity of the heating. At $a < b$ and a specified capacity Q, bending of the heat exchanger increases logarithmically with a decrease in the size of the heated region. The least bending will occur with uniform heating, when $a = b$. If we fix the heat flux q in the heated spot rather than the total capacity Q, then in accordance with Eq. (10) an increase in the radius of the heated spot will – in contrast to the regime Q = const – be accompanied either by a monotonic increase in the bending of the heat exchanger to a maximum at $a = b$ in the case of unfastened edges or a passage through a maximum at $a/b = 1/\sqrt{e} = 0.607$ in the case of fastened edges ($\varkappa = -1$). We should note that by knowing the form of the function $\omega_{bnd}(r)$ in the heating of a circular region, it is easy to construct a solution by the superposition method for cases of heating of annular regions (see Appendix). Summing the expansional and bending strains and limiting the resulting strain to a certain maximum permissible value ω_{pr}, we obtain the maximum permissible (with respect to strain) capacity of an unsecured heat exchanger for uniform heating:

$$Q_m = \frac{\pi}{3}\frac{\omega_{pr}H^2}{\beta} \Big/ \left(\frac{f_1}{\alpha_v} + f_2\frac{h}{\alpha} + f_3\frac{h^2}{2\lambda}\right), \quad (11)$$

where $f_i = \varphi_i/\Phi + (1/3)(H/b)^2$, i = 1, 2, 3.

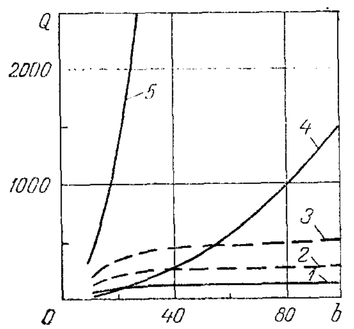

Fig. 3

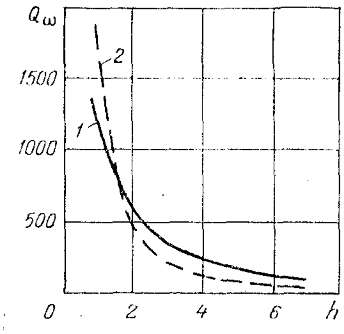

Fig. 4

Fig. 3. Effect of the radius b (mm) of a water-cooled jet heat exchanger on its thermal capacity Q (W) with a limitation on strains to 1 μm (1-4) or on stresses to 65 MPa (5) with d = 3 mm, h = 5 mm, Π = 0.1, Re = 10^4. Calculation with Eqs. (11) and (12) at H = 20 mm for structural materials Cu (1), (5), Mo (2), SiC (3), and Cu at H = b (4).

Fig. 4. Effect of the thickness of the wall h (mm) on the thermal capacity Q_ω (W) of a water-cooled jet heat exchanger made of copper, with limitation of the strain to 1 μm at b = 30 mm, H = 20 mm, Π = 0.4, and Re = 10^4. Calculation with Eq. (11) at d = 1.5 mm (1) and h = d (2).

Insertion of the expressions for the thermal force (7) and moment (8) into Eq. (3) gives an expression for the thermal capacity of the heat exchangers which is the maximum allowable with regard to the stresses on the surface z = H/2

$$Q_\sigma = \frac{(1-\nu)\sigma_{pr}}{\beta E}\pi b^2 \Big/ \left(\frac{1-h/H}{\alpha} + \frac{h(1-h/2H)}{\lambda} - \frac{\Phi}{\alpha_v H}\right), \tag{12}$$

where $\sigma_{pr} = \sigma_{rr} = \sigma_{\varphi\varphi}$ is the maximum permissible compressive stress of the material of the heat-exchanger wall.

Heat Transfer. In accordance with Eqs. (9)-(12), the thermoelastic stresses and strains in the jet heat exchanger are directly dependent on the heat-transfer coefficients α and α_V. There are no theoretical recommendations for determining them. We therefore determined them experimentally on copper specimens with different dimensions (Table 1) and using different water flow rates (Fig. 2). Here we employed the method in [3]. As can be seen (Fig. 2a), the effective heat-transfer coefficient of a finned wall cooled by jets of finite dimensions increases with an increase in water flow rate and a decrease in cell diameter, reaching $(2\text{-}3)\cdot 10^5$ W/(m^2·K) at d = 1.5 mm and Re = 10^5 without boiling of the water. The volumetric heat transfer $\alpha_V = 4\Pi\alpha_f/d$, dependent on the cell diameter d and the mean heat-transfer coefficient α_f on the lateral surface of the fins (the walls of the cell), reached $(3\text{-}4)\cdot 10^7$ W/(m^3·K) (section No. 4, Fig. 2b). The error of measurement of α is 15%, while the error of measurement of volumetric heat transfer is 20%.

Figures 3 and 4 show numerical estimates of the permissible capacities for heat exchangers unsecured at their contour in the case of uniform heating (a = b). Here we used experimental values of the heat-transfer coefficient. The value of $Q_\sigma \sim b^2$, while Q_ω is slightly dependend on the radius at $b/H \geq 2$. Meanwhile, bending strains predominate in this region of b/H. To increase the permissible (with respect to strains) thermal capacity, the radius of the heat exchanger must be increased simultaneously with its thickness H.

A second important reserve for increasing the tolerable thermal load Q_ω is reducing the thickness of the wall of the heat exchanger h. It follows from Fig. 4 that a decrease in h from 7 and 1 mm, with retention of a constant cell diameter d = 1.5 mm and constant heat-exchanger dimensions, leads to a ninefold increase in the allowable thermal load. A simultaneous decrease in the thickness of the wall and the size of the cooling cell d leads to an even greater increase in Q_ω (see Fig. 4). This is due to the fact that a decrease in the diameter of the cooling cells is accompanied by an increase in the total cooling surface and the heat-transfer coefficients α_V and α (see Fig. 2). The use of a structural material with a high thermal conductivity and low coefficient of thermal expansion also leads to an increase in capacity Q_ω. Thus, the use of molybdenum increases Q_ω by a factor of 1.5, while the use of SiC increases it by a factor of 2-2.5 relative to a similar heat exchanger made of copper (see Fig. 3) (the properties of the materials were taken from [7]).

APPENDIX

If the thermal load on the wall of the heat exchanger is confined in an annular region with an inside radius c and an outside radius $a > c$, if the thermal load is uniform within this ring and equal to q, and if q = 0 outside the ring ($r < c$, $a < r \leq b$), then in the absence of radial heat flows the thermal moment (1) repeats the profile of the thermal load and is determined by Eq. (8) at $c \leq r \leq a$. Outside the ring, $M_t = 0$. Then from (2) we obtain

$$\frac{4D}{b^2M_t}\,\omega_{\text{bnd}}(r)=\begin{cases}\mu-\eta+\mu\ln\dfrac{1}{\mu}-\eta\ln\dfrac{1}{\eta}+\varkappa(\mu-\eta)\left(1-\dfrac{r^2}{b^2}\right) & \text{at } r\leqslant c, \qquad (A.1)\\[2ex] \mu-\dfrac{r^2}{b^2}+\mu\ln\dfrac{1}{\mu}-\eta\ln\dfrac{b^2}{r^2}+\varkappa(\mu-\eta)\left(1-\dfrac{r^2}{b^2}\right) & \text{at } c\leqslant r\leqslant a, \qquad (A.2)\\[2ex] (\mu-\eta)\left[\ln\dfrac{b^2}{r^2}+\varkappa\left(1-\dfrac{r^2}{b^2}\right)\right] & \text{at } a\leqslant r\leqslant b. \qquad (A.3)\end{cases}$$

Here $\mu = a^2/b^2$; $\eta = c^2/b^2$.

In particular, when c = 0, the thermal load decreases on a circular region. When c = 0 and a = b, the thermal load is uniformly distributed over the surface of the wall. Let us compare two limiting cases: 1) heating in a circle (c = 0) and 2) heating in a ring (b = a, c > 0) and find the strains at characteristic points.

The deformation on the axis r = 0 is

$$\omega_1\frac{4D}{b^2M_t}=\mu\ln\frac{1}{\mu}+(1+\varkappa)\mu, \qquad (A.4)$$

$$\omega_2\frac{4D}{b^2M_t}=-\eta\ln\frac{1}{\eta}+(1+\varkappa)(1-\eta). \qquad (A.5)$$

The deformation on the boundary of the heated spot r = a is

$$\omega_1\frac{4D}{b^2M_t}=\mu\ln\frac{1}{\mu}+\mu[1+\varkappa(1-\mu)]. \qquad (A.6)$$

The deformation on the boundaries of the heated ring is

$$r=c\quad \omega_2\frac{4D}{b^2M_t}=\mu\ln\frac{1}{\mu}-\eta\ln\frac{1}{\eta}+(\mu-\eta)[1+\varkappa(1-\eta)], \qquad (A.7)$$

$$r=a\quad \omega_2\frac{4D}{b^2M_t}=\mu\ln\frac{1}{\mu}-\eta\ln\frac{1}{\mu}+\varkappa(\mu-\eta)(1-\mu). \qquad (A.8)$$

It follows from a comparison of (A.4) and (A.5) that a heat exchanger fastened at its edges ($\kappa = -1$) is bent (bulges) counter to the heat flow if the heating region is a circular spot, while the middle of the heat exchanger sags ($\omega < 0$) if the heating region is a ring.

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IDENTIFICATION OF THE CHARACTERISTICS OF SURFACE THERMAL INTERACTION BETWEEN MATERIALS AND GAS STREAMS

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UDC 536.24

The authors analyze the possible identification of the functional parameters in the energy balance equation on the disintegrating surface of a solid.

Mathematical modeling of processes of thermal interaction of disintegrating structural and heat shield materials with high enthalpy gas streams must be based, in general, on solving the coupled problems of unsteady heat and mass transfer. Problems of this class are formulated in the form of a single system of equations describing the whole complex of interconnected processes: the gas flow in the inviscid region; the heat and mass transfer in the high-temperature boundary layer in the presence of blowing and chemical reactions in multicomponent gas mixtures; and surface disintegration and heat transfer within the material.

Solution of the coupled heat- and mass-transfer problems in the full formulation is a complex problem, and one that is difficult to solve at present. Therefore, one must construct simplified mathematical models which describe approximately the complex processes under examination. And here one must include the basic factors influencing thermal interaction of the material with the gas stream [1, 2].

Approximate mathematical models usually contain a number of effective values of characteristics, each of which takes account of a certain set of individual phenomena and processes. Methods of parametric identification of inverse heat-transfer problems [3] have recently found widespread use in determining these characteristics.

In this paper we analyze the inverse problem of recovering the characteristics of surface thermal interaction of a disintegrating material with a high enthalpy gas stream. Here we assume that the heat-transfer process within the material is described by the homogeneous heat-conduction equation, one-dimensional in a space coordinate, with coefficients that are functions of temperature. In addition, it is assumed that the disintegration and removal of material occurs only in the gas phase. This corresponds to the mechanism of thermochemical disintegration of the surface of subliming materials. In the more general case, e.g., for composite materials, other factors [1, 2] must be accounted for.

With these assumptions the approximate mathematical model of the process of heat and mass transfer occurring in a certain time interval $(0, \tau_m]$ in the gas–solid system, allowing for disintegration of the material on the wetted surface, can be represented in the form of the following heat-conduction boundary problem:

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