EVAPORATION OF METHANOL IN AN IRRIGATED TUBE UNDER SMALL HEAT LOADS

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The authors analyze the process of methanol evaporation and heating when film of this substance flows downward along a vertical tube under a small heat load ($q \cong 8000$ W/m^2). The tests were performed in a quartz column with an inside diameter of 0.0145 m and a height of 0.4 m. The irrigation intensity was varied from 0.0025 to 0.15 kg/m·sec. The Reynolds number varied from 50 to 2000. Ethyl alcohol, trichloroethylene, and isopropyl alcohol vapors were used as the heat carrier.

The values of the heat transfer coefficient obtained here for methanol during evaporation were close to those corresponding to boiling in a large vessel or in a tube, but were much lower than those corresponding to condensation. The heat transfer coefficient as a function of the irrigation intensity first drops distinctly to a certain minimum. At a Reynolds number above 400, the heat transfer coefficient as a function of the irrigation intensity and of other parameters is described by an equation analogous to the McAdams equation for free laminar flow downward – but with the coefficient 0.9 instead of 0.67.

On the same apparatus the authors also tested the heating of methanol under conditions of free downward flow. Methanol vapor was used as the heat carrier. The heat transfer coefficient could be calculated from the measured temperature profiles of the film along the column height. Calculations according to McAdams agree closely with the test results. The minimum on the curve of heat transfer coefficient versus irrigation intensity does not pass through a minimum but, instead, follows a trend similar to that for the case of methanol boiling under a small heat load. The heat transfer rate during evaporation is 35% higher than during heating. These observations lead to the conclusion that, under a small heat load, the mechanism of heat transfer in an evaporator with a downward flowing liquid film is more similar to the mechanism of convective heat transfer than to the mechanism of film condensation.

It is also concluded here that vapor generation at the tube walls, rather than wave generation, has some turbulizing effect on the process of heat transfer during evaporation of methanol.

INFRARED HEATING OF THE HEARTH GRATING AND OF DOUGH LOAVES IN THE FIRST STAGE OF A TUNNEL OVEN

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The article deals with the heating of the spiral-rod hearth grating and of dough loaves under a constant density of thermal flux supplied by an infrared radiation source underneath.

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With typical thermophysical properties of a dough during the initial stage of baking, and with typical structural features of the hearth grating, the heating of one grating wire and of the dough lying on it is treated here as a one-dimensional problem in heat transfer involving a finite-size and a semiinfinitely large body.

The expressions for the temperature of wire and dough as well as for the ratio of heat absorbed by the dough to heat received by the radiation-sensitive surface of a wire are:

$$T_1(x, \tau) - T_{01} = 2\frac{\sqrt{a_1}}{\lambda_1} q \sqrt{\tau}\, i\,\mathrm{erfc}\left(\frac{x}{2\sqrt{a_1\tau}}\right), \tag{1}$$

$$T_2(x, \tau) - T_{02} = b\Gamma(m+1)(4\tau)^m i^{2m}\,\mathrm{erfc}\left(\frac{x-R}{2\sqrt{a_2\tau}}\right), \tag{2}$$

$$\frac{Q_2}{Q} = 1 - 4\frac{R}{\sqrt{a_1\pi}}\left[\frac{1}{\sqrt{\tau_1}}\exp\left(-\frac{R^2}{4a_1\tau_1}\right) - \frac{R}{2\tau_1}\sqrt{\frac{\pi}{a_1}}\,\mathrm{erfc}\,\frac{R}{2\sqrt{a_1\tau_1}}\right]$$
$$-\mathrm{erf}\left(\frac{R}{2\sqrt{a_1\tau_1}}\right) - \frac{2R}{\sqrt{a_1\pi}}\left[\frac{1}{\tau_1}\exp\left(-\frac{R^2}{4a_1\tau} - \frac{R}{2\tau_1}\right)\sqrt{\frac{\pi}{a_1}}\,\mathrm{erfc}\left(\frac{R}{2\sqrt{a_1\tau_1}}\right)\right]. \tag{3}$$

The energy radiated toward the grating is absorbed by the wire and by the dough loaves, but part of it is dissipated into the ambient medium.

The following ratios are useful for calculating the heat transfer within the first stage of a tunnel oven with infrared radiation sources:

$$\frac{q_{im}}{q_{fr}} = \frac{K_3\left(\varepsilon_{tr.1}\frac{1-f}{f} + \varepsilon_{tr.2}\right)}{(1-k_3)\left(\varepsilon_{tr.1}\frac{1-f}{f} + \varepsilon_{tr.3}\right)}, \tag{4}$$

$$\frac{q_{tr.2}}{q_a} = \frac{\varepsilon_{tr.1}(1-f)}{\varepsilon_{tr.3} f}, \tag{5}$$

$$\frac{q_{1\text{-}tr.1}}{q_T} = \frac{\varepsilon_{tr.1}}{\varepsilon_{tr.2}}\cdot\frac{1-f}{f}. \tag{6}$$

Calculated values agree closely enough with test data.

NOTATION

q	is the thermal flux at plane x = 0;
q_{im}	is the energy flux radiated from the reference surface 1 toward dough loaves and wires of the immersed grating segments;
q_{fr}	is the energy flux radiated from surface 1 toward wires of the free grating segments and to the ambient medium through gaps in the grating;
$q_{1\text{-}tr.1}$	is the thermal flux transmitted through the radiation-sensitive surface of immersed grating wires;
q_T	is the thermal flux transmitted through the radiation-sensitive portion of the lower dough loaf surfaces;
$q_{T\text{-}tr.1}$	is the thermal flux transmitted through the contact surface between wires and dough loaves;
Q	is the quantity of energy received by the radiation-sensitive surface of a wire;
Q_2	is the quantity of energy received by the dough;
$q_{tr.2}$	is the thermal flux radiated from surface 1 and expended on heating the wires in the upper layer of the grating free of dough loaves;
q_a	is the thermal flux transmitted through the active cross section of the upper grating layer free of dough loaves;
τ	is the time;
x	is the space coordinate;
R	is the wire thickness;
T_{01}	is the initial wire temperature;
T_{02}	is the initial dough temperature;
$T_1(x, \tau)$	is the wire temperature at a point x_1 at a time τ;
$T_2(x, \tau)$	is the dough temperature at a point x_1 at a time τ;
a_1, a_2	is the thermal diffusivity of metal and dough respectively;
b	is the constant, proportional to the temperature at the loaf surface;

m	is the power exponent;
ε_{tr_1}, ε_{tr_2}, ε_{tr_3}	are the referred emissivity of systems participating in the heat transfer and consisting of surface 1, the wire and load surfaces, and the medium in plane 2;
k_3	is the hearth loading factor;
f	is the free cross section in plane 2;
λ_1, λ_2	is the thermal conductivity of metal and dough respectively.

THERMAL CONDUCTIVITY AND ELECTRICAL RESISTIVITY OF TITANIUM MEASURED BY THE KOHLRAUSCH METHOD OVER THE 400-1100°K TEMPERATURE RANGE

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UDC 536.2

Both the thermal conductivity and the electrical resistivity of 99.7% pure titanium were measured under 10^{-4} mm Hg vacuum. The specimen, 120 mm long and 5 mm in diameter, was soldered to copper terminals and fastened between electrodes with water cooling to a constant temperature. The specimen was enclosed inside a molybdenum shielding tube 8/7 mm in diameter. The ends of this shield were soldered to water-cooled copper flanges. Thermal flux leakage from the lateral surface of the specimen was overcome by holding the temperatures at facing one another points of the specimen and the shield equal along the active specimen length. Such a compensation was achieved by a separate regulation of the alternating electric current through the specimen, the shield, and the center heater. According to calculations, an approximately 1 mm wide gap between specimen and shield reduced to a negligible level any heat loss due to radiation. The apparatus had been designed so as to allow for free expansion of both specimen and shield during heating. The temperature was measured with Chromel–Alumel thermocouples. The thermocouple electrodes were used also for measuring the voltage drop along the active specimen segment (l = 15 mm). This voltage drop was measured by the compensation method with a model R-56 ac potentiometer. The thermal conductivity was then determined from the Kohlrausch formula. The accuracy of measurements was estimated, indicating a 2-3% mean error in the thermal conductivity, a 1% error in the electrical resistivity, and a 2-3% error in the Lorentz function.

According to the authors' data, the thermal conductivity of titanium is 20.5 W/m · °C at T = 400°K and then decreases slightly (by approximately 5%) over the 400-1100°K temperature range. The results of this study concerning the temperature characteristic of electrical resistivity of titanium agree closely with published data. The subsequently calculated value of the Lorentz function for titanium exceeds its theoretical value by 30-35% and decreases with the temperature from $L = 3.4 \cdot 10^{-8}$ (W/°C)^2 at T = 400°K to $L = 3.2 \cdot 10^{-8}$ (W/°C)^2 at 1100°K.

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AVERAGED CHARACTERISTICS OF A WHIRLED STREAM

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UDC 532.55

The authors describe a whirled stream by resolving the flow of a characteristic fluid element into two components: circular motion in the force field N at some angular velocity and motion of the curl center along the pipe axis in the force field M.

A relation is established between the whirling force N, the stream parameters, and the turbulizer design parameters. It is shown that the quantity

$$N = \frac{\rho_s (R_1^2 + R_2^2)\,\omega}{4R_1^2}, \tag{1}$$

equal to the ratio of the principal momentum in the turbulizer ducts to the pipe radius squared may serve as the criterial whirl parameter.

A formula is derived for calculating the hydraulic drag of axially symmetrical blade-type turbulizers:

$$\Delta P = \frac{\rho_s (R_1^2 + R_2^2)^2 V_s^2}{32 R_1^4 \left[\pi (R_1^2 - R_2^2) - nab\right]^2 \sin^2 \varphi}. \tag{2}$$

Relation (2) has been verified experimentally in turbulizers of various designs for velocities ranging from 10 to 35 m/sec. The agreement between calculated and measured values of the hydraulic drag in multiturn blade-type turbulizers appears satisfactory.

NOTATION

R_1	is the pipe radius;
R_2	is the radius of turbulizer hub;
n	is the number of turbulizer turns;
a,b	are the height and width of turbulizer blade;
φ	is the pitch angle;
ω	is the angular velocity;
ρ_S	is the gas density under standard conditions;
V_S	is the gas flow rate under standard conditions.

STEADY-STATE OVERLOAD CAPACITY OF THYRISTORS IN A STRUCTURE OVERHEATED BECAUSE OF INCOMPLETE CONDUCTION

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Under specific conditions, the structure of a thyristor may conduct direct electric current nonuniformly. This may be caused by inhomogeneities (constructional, technological, and physical) of the

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structure itself, a low quality of contact tabs (burrs and pits due to treatment), or the particular conditions of the converter operation (e.g., at higher frequencies).

The two-dimensional linear equation of steady-state heat conduction is solved on an analog R-network, whereupon a relation is derived between the steady-state thermal resistance of a TT-2 thyristor with impressed contact tabs and the area of the annular local conduction zone in the structure – when the latter varies from 5 to 100% of the active structure area. This relation is accurately enough represented by a formula where the thermal resistance is inversely proportional to the square root of the area of the local conduction zone. Solutions are obtained for a thyristor operating with bilateral and with unilateral cooling. A method is proposed for estimating the superheat in the structure at any radial coordinate point when the heat is not completely conducted away. It has been demonstrated experimentally that the residual electrical resistance of a thyristor is inversely proportional to the area of the local conduction zone in the structure. The dependence of the device on the area of the local conduction zone is used for estimating the overload capacity of a thyristor.

SOLIDIFICATION OF A CONDENSED METAL FILM DURING THERMAL EVAPORATION UNDER VACUUM

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UDC 536.425.539.234

Studies concerning the structural characteristics of metallic condensate films which form during thermal evaporation under vacuum and studies of transient temperature fields in semiconductor films yield much useful information about both processes.

Meanwhile, the pattern of condensation of metal vapor on a cooled substrate – a process characterized by a change of the aggregate state – can be explained fully only if changes occurring in the process are taken into account.

In analyzing the process of heat transfer from a metal film on a neutral substrate, therefore, it is appropriate to consider the changes of the aggregate state of the film in the sequence: vapor to liquid and liquid to solid. Such a sequence may properly be assumed for determining the deposition parameters (at the evaporation temperature T_e and the substrate temperature T_b): $T_e \simeq T_S$ and $T_b \simeq 0.33T_S$, with T_S denoting the solidification point of a given metal.

The problem of film solidification is formulated as a problem of coupling two temperature fields $T_1 = T_e \simeq T_S$ of the not yet solid film and $T_2 = T_b + (T_S - T_b)X/\xi$ of the already solid film, with a special boundary condition given at the moving interface ξ between zones.

With functions $T_1(X, \tau)$ and $T_2(X, \tau)$ which satisfy the initial and the boundary conditions of the given problem as well as the differential equation of heat conduction, it is possible to solve the nonlinear equation of the moving interphase boundary between the already solid and the not yet solid zone, then to find expressions for the proportionality factor $\beta^2 = \xi^2/\tau$ which characterizes the velocity of the solidification front and thus constitutes a measure of the solidification rate, and also to determine the velocity $v = d\xi/d\tau$ of the solidification front in the condensate film.

One can determine the numerical values of ξ^2/τ and v while calculating the heat of phase transformation r removed from the molecular stream of metal vapor (at a constant volume) as a result of cooling down to the substrate temperature: $r = c_v(T_e - T_b)$ with c_v denoting the specific heat of the given metal vapor.

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The study has shown:

an agreement between calculated values of ξ^2/τ, which measures the cooling rate of condensed metal films, and the values of thermal diffusivity a for the respective metals (which measures their cooling rate), which confirms the validity of the original problem formulation;

an agreement between the values of the fusion heat r* of bulk metal and calculated values r, which indicates that the internal energy of phase transformation during film solidification may be expressed as the change in internal energy per unit mass of vapor during cooling on a substrate;

a decrease in the solidification rate v of condensed metal films with a rise in the boiling point of metal vapor, within the range of relevant groups of metals (light, refractory, etc.) in the Periodic Table.

CALCULATING THE TEMPERATURE CHARACTERISTICS OF HIGH-CURRENT THERMISTORS

V. V. Popivnenko

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A study was made of high-current semiconductor thermistors with often complex geometries, a large mass (up to 12.5 g), wide ranges of temperature variation (up to 750°K), massive contact tabs, and other distinguishing features (not found in low-power semiconductor thermistors), the results of these studies indicating that the thermal capacity C, the power dissipation P_α, the static and the dynamic dissipation factors k_{st} and k_{dyn}, the electrical and the thermal time constants τ_θ and τ_e, and also the dynamic performance coefficient D of a high-current semiconductor thermistor depend largely on its temperature. For calculating the steady-state and the transient performance modes of a circuit with high-current semiconductor thermistors, therefore, one must know not the thermistor constants and parameters but their temperature characteristics $C = f(\theta)$, $P_\alpha = f(\theta)$, $k_{st} = f(\theta)$, $k_{dyn} = f(\theta)$, $\tau_\theta = f(\theta)$, $\tau_e = f(\theta)$, and $D = f(\theta)$ (θ denoting the superheat temperature of the semiconductor thermistor).

Calculation of the $P_\alpha = f(\theta)$ and the $k_{st} = f(\theta)$ characteristics is based on tests and thus accounts for all energy processes occurring in a semiconductor thermistor, and calculation of its static volt-ampere characteristic accounts for the conditions of heat transfer.

It is suggested that the $C = f(\theta)$ characteristic be calculated from static and dynamic volt-ampere data as well as from oscillograms of current as a function of time $i = f(t)$, without the need for intermediate graphicoanalytical transformations.

The heat capacity of a semiconductor thermistor is a function of the specific heat of the specimen material and of its mass (the specific heat is a function of the temperature), while the dissipation factor k_{st} depends on the geometry, the size, the temperature, and the spatial orientation of the device as well as on the thermodynamic properties and the state of the ambient medium. The studies have shown that dimensional and mass variances from nominal values in a large-scale production of a specific type semiconductor thermistor are insignificant (less than 4%), while points on $k_{st} = f(\theta)$, $P_\alpha = f(\theta)$, and $C = f(\theta)$ curves for specimens of various ratings are dispersed from the mean curves by not more than 5%. These curves are averaged temperature characteristics of given types of semiconductor thermistors, necessary for the determination of their $k_{dyn} = f(\theta)$, $\tau_\theta = f(\theta)$, $\tau_e = f(\theta)$, and $D = f(\theta)$ characteristics.

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The formulas given here for calculating these characteristics apply to any type of semiconductor thermistors. In many cases such as, for example, high-current semiconductor thermistors of the "honeycomb" construction the most often used $k_{st} = f(\theta)$, $k_{dyn} = f(\theta)$, and $C = f(\theta)$ characteristics can be expressed in analytical form and thus in a form suitable for automated computer-aided design procedures.

An analysis of the results has shown that the temperature characteristics of high-current semiconductor thermistors operating under a heavy load vary over wide ranges. When temperature θ changes from 50 to 450°K, for example, the value of k_{st} becomes 1.9 higher, the value of k_{dyn} becomes 2.7 times higher, and the value of C becomes 3.0 times higher in the case of a "honeycomb" semiconductor thermistor.

Knowing the temperature characteristic of a semiconductor thermistor, one can not only quickly, simply, and rather accurately calculate its steady-state temperature but also refine the transient calculations for circuits with semiconductor thermistors.

TEMPERATURE FIELD OF THE STATOR OF AN ENCLOSED AND FINNED INDUCTION MOTOR WITH ASYMMETRICAL COOLING

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The article deals with the temperature profile of a stator section underneath its finned housing. The core is subdivided into two zones (Fig. 1): the yoke or zone 1 and the slotted core (teeth and coils) or zone 2. The temperature $T_i(r, \varphi)$ in these zones is described by the Poisson equation in cylindrical coordinates:

$$\frac{1}{r}\cdot\frac{\partial}{\partial r}\left(r\frac{\partial T_i}{\partial r}\right)+\frac{1}{r^2}\cdot\frac{\partial^2 T_i}{\partial\varphi^2}=-\left(\frac{\rho}{\lambda}\right)_i \quad (i=1,\ 2).$$

The boundary conditions are: convective heat transfer ($0 \le \varphi \le \varphi_1$)

$$-\lambda_{sta}\frac{\partial T_1}{\partial r}\Big|_{r=R_1} S_1=\alpha_f S_{fin}(T_f-T)\Big|_{r=R_1};$$

thermal flux through the base between feet ($\varphi_1 \le \varphi \le \pi$)

$$-\lambda_{sta}\frac{\partial T_1}{\partial r}\Big|_{r=R_1}=g_1;$$

thermal flux from rotor to stator through the airgap

$$-\lambda_{sta}\frac{\partial T_2}{\partial r}\Big|_{r=R_2}=g_2.$$

The condition of symmetry with respect to the OO_1 axis is

$$\frac{\partial T_i}{\partial\varphi}\Big|_{\varphi=0}=\frac{\partial T_i}{\partial\varphi}\Big|_{\varphi=\pi}=0.$$

Continuity of the temperature function and of the thermal flux function at $r = R_2$ is expressed as

$$T_1(R_2,\ \varphi)=T_2(R_2,\ \varphi);\quad \frac{\partial T_1}{\partial r}(R_2,\ \varphi)=\frac{\partial T_2}{\partial r}(R_2,\ \varphi),$$

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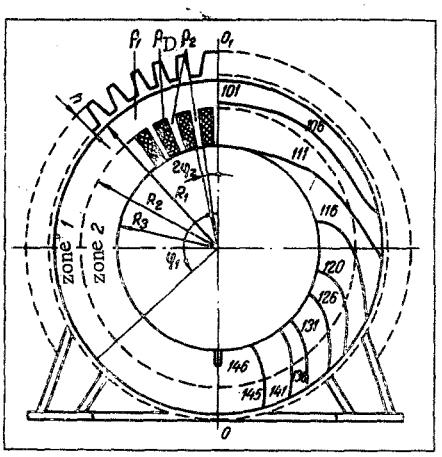

Fig. 1. Schematic diagram of a stator section and its temperature profile, for a model A4-100L4 electric motor. Numbers at the curves denote temperatures in °C.

with the area of the heat emitting core surface S_1, the surface of fins S_{fin}, the coefficient of heat transfer between fins α_f, the fin temperature at the root T_f, the ambient air temperature T_A, the density of thermal flux from core through base g_1, the density of thermal flux from rotor to stator g_2, and

$$\left(\frac{\rho}{\lambda}\right)_1 = \frac{\rho_1}{\lambda_{sta}}.$$

Here ρ_1 denotes the specific losses in the stator yoke region and λ_{sta} denotes the thermal conductivity along the stack

$$\left(\frac{\rho}{\lambda}\right)_2 = \begin{cases} \dfrac{\rho^2}{\lambda_{sta}} & \text{in the stator teeth region;} \\ \dfrac{\rho_{Cu}}{\lambda_{Cu}} & \text{in the stator yoke region,} \end{cases}$$

with the specific losses in the stator teeth region ρ_2, the specific stator copper losses ρ_{Cu}, and the thermal conductivity of copper λ_{Cu}.

The problem is solved by the Fourier method of separating the variables. In order to determine the integration constants, from the boundary condition the authors derive an infinite system of equations which is then solved by successive approximations. The temperature field of the stator of a model A4-100L4 electric motor is shown in Fig. 1.

SOME CONCEPTS IN THE THEORY OF THERMAL CIRCUITS

Yu. A. Gavrilov

UDC 536.21

In order to find approximate solutions to problems of heat transfer within a system of bodies, one often makes assumptions which allow the temperature field of each body to be considered uniform or variable in one direction only. In such cases the system of bodies may be treated as a thermal circuit and represented graphically by an equivalent circuit diagram. Thermal circuits without active elements but consisting of thermal resistances and conductances only have been described in the technical literature

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[1, 2]. The boundary conditions of a problem and the presence of either heat sources or heat sinks may be represented by active circuit elements in the form of temperature or thermal flux sources. The properties of and the symbols for these elements are analogous to those of and for conventional voltage and current sources in electrical circuit diagrams. The physical analog of an active element in a linear thermal circuit would be an ideal heat pump which maintains a stipulated temperature difference or thermal flux level regardless of the load. The effect of a heat carrier on the temperature of bodies can be accounted for by a resistance whose reciprocal value is proportional to the product of thermal capacity and mass flow rate of the heat carrier. For a transient process, furthermore, the thermal circuit must contain thermal capacities analogous to electrical capacitances in electrical circuits.

Thermal circuits can be analyzed by methods and laws derived in the electrical circuit theory (theorem of variations, method of the equivalent source, complex-numbers notation of quasisteady process quantities, trend toward automated computer-aided designs, etc.). The validity of this approach can be demonstrated if the laws of Fourier, Newton–Riechmann, Stefan–Boltzmann, and energy conservation are expressed in a form analogous to Ohm's and Kirchhoff's laws for electric circuits, and if an open thermal system can be transformed into a closed one. In order to apply the methods, the formulas, and the ready solutions from circuit theory to thermal circuits, it is sufficient to replace currents, voltages, electrical resistances and capacitances by their respective thermal analogs. Moreover, the inductance as well as the power of active elements and the heat dissipated in resistances are all assumed equal to zero.

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APPLICATION OF THE EXPANSION THEOREM TO THE DETERMINATION OF THE TEMPERATURE IN MULTILAYER BODIES

M. I. Dubovis

A solution is obtained to the equation of heat conduction

$$\frac{\partial T_i}{\partial \tau} = a_i \left(\frac{\partial^2 T_i}{\partial r^2} + \frac{\omega \partial T_i}{r \partial r} \right) + w_i(\tau), \qquad (1)$$

where $\omega = 0, 1, 2$. The initial temperature of each layer is constant. The boundary conditions at the outside surfaces are of the third kind, functions of time. Conditions of the first and of the second kind represent special cases. Perfect contact conditions are assumed at the inside boundary surfaces. A Laplace transformation of Eq. (1) into an ordinary differential equation is performed and the solution to the latter is found as a linear combination of two eigenfunctions. The arbitrary constants are determined from a system of linear equations corresponding to the boundary conditions for the layers. The original function is found in the form of a series of residues at the poles. If the transforms of the given functions are not fractional-rational, then one also uses the Convolution Theorem. Discussed are problems involving the application of the Expansion Theorem to the solution of this problem. Shown are two forms of the solution

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in transforms, the rule of reducing the characteristic determinant and the particular determinants to a power series, the rule for writing in complete form the elements of the characteristic determinant and, in the case of an infinitely large plate, the rule for writing in complete form the determinant elements obtained by the first differentiation of the characteristic determinant, also a calculation scheme is shown in which the characteristic determinant and the particular determinants are expressed as sums of products of second-order determinants, and formulas are derived for calculating the first terms (coefficients) of power series which represent those determinants. These coefficients are then used in calculations for determining the order of the zero pole of the solution-transform. A relation is established between the zero pole and the kind of boundary conditions. As an example, calculations are shown (in general form) with a double zero pole.

ONE APPROXIMATE SOLUTION TO THE SOLIDIFICATION PROBLEM

V. I. Lozgachev

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When a melt is bounded by an isothermal plane and the solidification front propagates inward in the direction normal to that plane, then the thickness of the solidified layer is a following function of time:

$$y = \beta \sqrt{t}, \tag{1}$$

with constant β determined from the well known relation in [1].

Formula (1) is valid for a constant amount of subcooling r_s. We now find an expression for y when the amount of subcooling is given as a certain function of time $r_s = f(t)$ at a constant superheat of the melt.

Let the following jumps in the amount of subcooling at the boundary plane $r_{s1} < r_{s2} < \ldots < r_{sn}$ occur consecutively within respective time intervals Δt_1, Δt_2, . . . , Δt_n. We assume that the corresponding smoothing of the temperature curves according to formula (1) is effected by those jumps without any inertia, and that then Δt_i will represent exactly the buildup time of a solid layer under the condition r_{si}. Consequently,

$$\Delta t_1 + \Delta t_2 + \cdots + \Delta t_n = t \tag{2}$$

and formula (1) will apply to every segment.

The thickness of a built up layer after n jumps is, according to formula (1),

$$y_{1,2,\ldots,n} = \beta_n \sqrt{t_{n-1} + \Delta t_n}, \tag{3}$$

with t_{n-1} denoting some fictitious buildup time of layer $y_{1,2,\ldots,n-1}$ as if condition β_n prevailed all the time.

Analogously, we can write

$$y_{1,2,\ldots n+1} = \beta_{n+1} \sqrt{t_n + \Delta t_{n+1}}. \tag{4}$$

When $\Delta t_{n+1} = 0$, then evidently $y_n = y_{n+1}$. Equating (3) and (4) at $\Delta t_{n+1} = 0$ yields recurrence relations for the fictitious time t_n:

$$t_n = \frac{\beta_n^2}{\beta_{n+1}^2}(t_{n-1} + \Delta t_n), \quad t_{n-1} = \frac{\beta_{n-1}^2}{\beta_n^2}(t_{n-2} + \Delta t_{n-1}), \ \ldots$$

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Inserting the thus obtained values of t_{n-2}, t_{n-3}, . . . , consecutively yields, in the final analysis,

$$t_{n-1}=\frac{1}{\beta_n^2}\sum_1^{n-1}\beta_i^2\Delta t_i.$$

Inserting this into formula (3), we find

$$y_n=\sqrt{\sum_1^n\beta_i^2\Delta t_i}$$

or, inasmuch as an integral sum is under the square-root here, at the limit $\Delta t_i \to 0$ formula (2) yields finally

$$y(t)=\sqrt{\int_0^t\beta^2(\tau)\,d\tau}. \tag{5}$$

The relation $\beta = f(\tau)$ is determined according to the formula in [1] for a given function $r_s(t)$. For instance, for small values of β we have

$$\beta(\tau)=\frac{\lambda_s\sqrt{\pi a_l}}{\lambda_l r_l}\,r_s(\tau).$$

Expression (5) is valid for any β within the indicated accuracy. Formula (5) is valid also when the melt is not superheated ($r_l = 0$). For this case an equation has been derived in [2] which relates $r_s(t)$ to y(t) at any time. If the value for y(t) from formula (5) is inserted into this equation, for small values of y, then the expression for β in [1] (in the form of a series) is obtained with $r_l = 0$. For instance, with small values of β the formula for β in [1] yields when $r_l = 0$:

$$\beta=\sqrt{\frac{2\lambda}{q}}\sqrt{r_s}.$$

Letting $r_s = kt^n$, moreover, we obtain from formula (5)

$$y^2=\frac{2k\lambda}{q}\cdot\frac{t^{n+1}}{n+1}.$$

Inserting these values of r_s and y^2 into the Lyubov formula, we obtain an identity at small values of t.

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PECULIARITIES OF TEMPERATURE CALCULATIONS IN THE THEORY OF ULTRASONIC WELDING

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For calculating the temperature field of a polymer welded ultrasonically, a mathematical model has been developed which describes the effect of ultrasonic vibrations on the plastic mass and which is based on

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the Rzhevkin representation [1]. According to this model, directional ultrasonic radiation is treated as a "sound beam" emitted from a unique "horn."

Pursuing this geometrical approach, we assume that the "beam" is made up of tubes 0.1λ in diameter, with λ denoting the wavelength of longitudinal vibrations in the welded polymer. This makes it feasible to apply here the acoustics problem of a finite-length tube segment [1] and thus to take into account the finite dimensions of the welded package. In solving the problem, one calculates the acoustic pressure along a tube. With the acoustic pressures given at the entrance to the tubes, corresponding to the wave radiation function along a spout radius, it becomes possible to determine the acoustic pressure field of the section plane within the sound zone.

From the acoustic pressure we calculate the energy of the ultrasonic field at a sepcific point. This energy is then multiplied by the ultrasonic absorptivity and this yields the energy lost on heating the polymer and on irreversible deformations in it. Assuming that the heat released in the polymer by ultrasonic vibrations is maximum during resonance between thermal and acoustic waves at frequencies which are harmonics of the operating ultrasonic frequency, we then determine the energy lost on heating. This quantity is found, according to the Zener theory [2], as the product of the total energy absorbed in the polymer and the loss factor. From the heating energy thus determined, we can easily calculate the temperature change at the specific point.

The variation in the temperature field of polystyrene during ultrasonic welding was calculated on a "Nairi" computer and compared with temperature measurements made with model ÉPP-09 electronic potentiometers. The difference between theoretical and experimental thermograms amounted to 5-7% in the time and 3-19% in the location of a specific point.

Thus, the problem of sound propagation along a finite-length tube segment and the Zener theory are both applicable for rough theoretical calculations of the temperature field.

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CHARACTERISTICS OF HEAT REGENERATORS WITH THE GAS PRESSURE VARYING IN TIME

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UDC 536.27

The most characteristic feature of the process in Stirling and Ericson regenerative heating and refrigerating engines with a pulsation tube and similar devices operating on an unsteady flow of the working medium is that the gas pressure varies in time. The classical Nusselt–Gausen theory of a heat regenerator is not applicable under such conditions, because the number of differential equations describing the mathematical model of a regenerator increases and they become more complex:

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$$\begin{cases} \dfrac{RT}{A}\cdot\dfrac{\partial q_m}{\partial x}+\dfrac{dp}{dt}-\dfrac{p}{T}\cdot\dfrac{\partial T}{\partial t}=0, & (1)\\ \dfrac{\gamma}{\gamma-1}\cdot\dfrac{p}{T}\cdot\dfrac{\partial T}{\partial t}+\dfrac{c_p}{A}\cdot q_m\cdot\dfrac{\partial T}{\partial x}+\varkappa(T-T_w)-\dfrac{dp}{dt}=0, & (2)\\ \dfrac{\partial T_w}{\partial t}+\delta(T_w-T)=0, & (3)\end{cases}$$

where

$$\varkappa=\frac{KA_\varphi}{V};\qquad \delta=\frac{KA_\varphi}{\rho_w c_w V_w}.$$

Most interesting are those solutions to system (1)-(3) which do not depend on the initial conditions and are periodic in time, corresponding to the steady-state operation of a regenerator. The approximate periodic solutions are sought in the form

$$T(x,\ t)=T_e(x)+T_s(x)\sin\omega t+T_c(x)\cos\omega t, \tag{4}$$

$$T_w(x,\ t)=T_{we}(x)+T_{ws}(x)\sin\omega t+T_{wc}(x)\cos\omega t, \tag{5}$$

$$q_m(x,\ t)=q_{ms}(x)\sin\omega t+q_{mc}(x)\cos\omega t. \tag{6}$$

with the assumption that

$$p=p_e+p_s\sin\omega t. \tag{7}$$

Relations (4)-(6) represent the simplest functions of the variables x and t, automatically satisfying the condition of periodicity in t. An analysis of the solutions based on these relations will reveal the most characteristic features of the process in heat regenerators with a variable-pressure gas flow.

In order to determine the coefficients in Eqs. (4)-(6), the author uses the variational method of "least squares" and thus ensures the best "overall" accuracy of the sought approximate solution.

The obtained solution is in satisfactory agreement with test data. On this basis, then, the author examines how the regenerator efficiency depends on its design parameters and on the phase relations between temperature, pressure, and flow rate.

It is shown that pressure fluctuations in the gas stream have an appreciable effect on the regenerator characteristics. If the gas pressure and the gas flow rate are in phase opposition, for example, then at the "hotter" end of the regenerator the gas leaving will be at a higher temperature than the gas entering through the same section.

NOTATION

x	is the space coordinate along the regenerator;
t	is the time;
p	is the gas pressure;
T	is the gas temperature;
q_m	is the mass flow rate through a given regenerator section;
T_w	is the temperature of regenerator head;
γ	is the adiabatic exponent;
c_p	is the specific heat at constant pressure;
R	is the constant;
A	is the passage section in the regenerator;
K	is the heat transfer coefficient;
A_φ	is the heat transfer surface of a regenerator head;
V	is the volume of gas in the regenerator;
V_w	is the volume of a regenerator head;
ρ_w	is the density of a regenerator head;
c_w	is the specific heat of a regenerator head;
τ	is the period;
$\omega=2\pi/\tau$	is the angular frequency.

SOLUTION OF CERTAIN REVERSE PROBLEMS IN IGNITION THEORY

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Solving the problem of heterogeneous ignition at the interface between two half-spaces filled with fuel and oxidizer respectively [1] and solving the problem of ignition of a reactive body in the shape of a plate [2] or of a solid of revolution [3] near the stagnation point in a stream of hot oxidizer reduces to solving nonlinear integral equations of the Volterra kind:

$$\theta_w = \frac{1}{\sqrt{\pi}A}\int_0^\tau \frac{\varphi_w[\theta_w(t)]\,dt}{\sqrt{\tau - t}}, \tag{1}$$

with $A = 1 + K_\varepsilon$ in the first problem and $A = K_\varepsilon$ in the other problems, with φ_W denoting the intensity of nonlinear heat sources at the surface due to heat generated by heterogeneous chemical reactions, and with the other symbols here the same as in [2].

The solution to the problem of homogeneous ignition of a reactive condensate by a constant thermal flux [3] can also be reduced to an equation of the Volterra kind (1).

The reverse problem of ignition corresponding to these several cases here is formulated as follows: to find the intensity of nonlinear surface sources φ_W which will make the dimensionless temperature $\theta_W(\tau)$ rise according to any stipulated law.

It is quite evident that Eq. (1) with respect to φ_W represents the Abel equation in [4], whose solution yields

$$\varphi_w = \frac{A}{\sqrt{\pi}} \cdot \frac{d}{d\tau}\left(\int_0^\tau \frac{\theta_w(t)\,dt}{\sqrt{\tau - t}}\right). \tag{2}$$

If the right-hand side of Eq. (2) is considered known at $0 \leq \tau \leq \tau_*$ from tests, then, assuming the rate of the chemical reaction to follow the Arrhenius Law and using two values for the right-hand side, one obtains a system of two equations for determining E and qk_0. Considering then that one first-order heterogeneous reaction occurs and that little reagent has burned out at time $0 < t < t_*$, one finds:

$$E = \frac{RT_{w1}T_{w2}}{T_{w2} - T_{w1}} \ln \frac{\rho_{w1} f_{2w}}{\rho_{w2} f_{1w}}, \quad qk_0 c_e \rho_{w1} = f_{1w} \exp\left[\frac{T_{w2}}{T_{w2} - T_{w1}} \ln \frac{f_{2w}\rho_{w1}}{f_{1w}\rho_{w2}}\right]. \tag{3}$$

Analogous formulas for E and qk_0 are obtained also for a homogeneous ignition of the reagent.

The effective values of kinetic constants can be found also by minimizing the functional

$$I = \int_0^{t_*} [f(t) - qU]^2\,dt, \quad U = k_0 \rho c \exp - \frac{E}{RT}. \tag{4}$$

for $\nu = 1$ with the proper choice of E and qk_0.

NOTATION

τ_* is the dimensionless ignition time;
t_* is the real ignition time;
R is the universal gas constant;
E is the energy;
q is the thermal effect of a reaction;
k_0 is the pre-exponent;
c_0 is the initial concentration of deficient active gaseous component;
φ_W is the right-hand side of Eq. (2) in dimensional form;
ρ is the density;

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ν is the order of reaction;

$T_{w1} = T_w(t_1)$, $T_{w2} = T_w(t_2)$, $0 < t_1 < t_2 < t_*$.

Subscript

w refers to the boundary between media.

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