TRANSIENT PROCESSES IN OVENS WITH RECIRCULATION OF COMBUSTION PRODUCTS

A. V. Volodarskii and A. A. Mikhelev

UDC 664.655

The authors analyze transient changes in the gas composition in an oven heating system with recirculation of combustion products. The results apply also to desiccators and other devices where materials are heated in a recirculation mode.

The following designations are made:

$$K_1 = \frac{v_G}{v_G+v_A};\ \tau_t = \frac{V}{v_G+v_A}. \tag{1}$$

Here K_1 represents the concentration of combustion products v_G in the gas mixture entering the system per unit time through leakage (air v_A) and from the oven (combustion products v_G) and τ_t is the time in which the gas–air mixture entering the gas ducts of the oven at a volume flow rate $v_G + v_A$ will fill these ducts after the combustion products have been completely removed. In ovens with recirculation of the heating gases, unlike in ovens with a complete removal of combustion products, at the inlet to the working space the entering gases almost completely mix again with gases already in the heating system. Assuming that, approximately, a gas–air mixture enters and leaves the gas ducts of an oven at a volume flow rate $v_G + v_A$, we write the following equation for the balance of combustion products in the heating system of an oven:

$$(v_G+v_A)K_1 d\tau - (v_G+v_A)K d\tau = V dK. \tag{2}$$

The first term on the left-hand side represents the volume of gases entering the oven ducts during the time $d\tau$, the second term represents the volume of gases leaving it within the same time, and the right-hand side of Eq. (2) represents the change in volume of combustion products with a concentration K inside gas ducts whose volume is V.

Let $K = K_0$ for $\tau = 0$ [1]. Then, from (1) and (2) we have

$$\tau = \tau_t \ln \frac{K_1 - K_0}{K_1 - K}. \tag{3}$$

We consider special cases of Eq. (2).

1. Let $v_A = 0$ and $K_0 = 0$. Assuming $K_1 - K = 0.01$, we have from (3) and (1)

$$\frac{\tau}{\tau_t} = 2.3 \lg \frac{1}{0.01} = 4.6,$$

i.e., the transient time from the initial state $K_0 = 0$ to the quasisteady state, where K is almost equal (within 0.01) to the steady-state concentration $K = K_1 = 1$, is 4.6 times longer in a recirculation oven than in ovens with complete removal of combustion products.

2. The duration of the transient is determined for a bakery oven. In the most widely used types of ovens it is: $\tau_t \approx 170$ sec with a complete removal of combustion products, $\tau \approx 780$ sec with recirculation of combustion products.

A relation $\tau = \tau(K_1 - K; K_1 - K_C)$ is derived, suitable for calculating and measuring the length of transient processes.

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HEAT TRANSFER BETWEEN A TWO-DIMENSIONAL INCIDENT JET AND A FLAT SURFACE WITHIN THE DEVELOPED-FLOW ZONE

M. I. Davidzon

UDC 536.244

When a jet impinges perpendicularly on a flat plate, the spread of fluid over that plate surface can be approximately divided into three zones to be calculated separately: 1) acceleration flow, 2) transition flow), and 3) developed (self-adjoint) flow [1].

The author analyzes local heat transfer between a flat turbulent jet and a plate within the third of these zones. The dimensionless heat transfer coefficients are calculated by solving the momentum equation for a jet at a wall, on the assumption that the boundary layers are conservative and on the basis of the analogy between heat transfer and momentum transfer [2]. As the model serves here a zero-gradient turbulent stream flowing along an impermeable plate. The velocity at the outer edge of the boundary layer at the end of the acceleration zone is proposed as the velocity scale.

A criterial relation

$$\mathrm{St}_0 = 0.194\,\mathrm{Re}_0^{-0.2}\,\mathrm{Pr}^{-0.6}\,(\bar{h})^{-0.4}\,(\bar{x})^{-0.57} \tag{1}$$

is derived and compared with the test data in [3] by Gardon and Aquirat. Here $\mathrm{St}_0 = \alpha / c_p \rho U_0$ is the dimensionless heat transfer coefficient (Stanton number), U_0 denotes the stream velocity at the nozzle throat, $\mathrm{Re}_0 = U_0 H_0 / \nu$, H_0 is the nozzle caliber (smaller dimension), $\bar{h} = h/H_0$, h denotes the distance from nozzle throat to plate, $\bar{x} = x/H_0$, and x is the distance from the critical point.

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TEMPERATURE FIELD OF A TWO-COMPONENT POROUS BODY THROUGH WHICH A FLUID FLOWS BETWEEN IMPERMEABLE WALLS*

M. Ya. Antimirov and A. A. Panferova

UDC 536.2:622.276.4

Two problems are solved here pertaining to the heating of a porous body by a hot stream of liquid or gas, where the temperatures of both are assumed different. Into account is also taken the thermal contact between the porous body and semiinfinite impermeable walls. The thermal flux from the liquid or the gas to the porous body is considered proportional to the temperature between both. Longitudinal and radial flow are analyzed. It is assumed, furthermore, that the thickness of the porous body is sufficiently small and that, within each vertical section, the temperatures of the skeleton material and of the fluid are not functions of coordinate z. Into a porous body occupying the region $(-\infty < \bar{y} < +\infty,\ 0 < \bar{x} < +\infty,\ -h \le \bar{z} \le 0)$ is squeezed an incompressible fluid at a constant volume rate Q per unit length of the y-axis, in the linear case, through the plane $\bar{x} = 0$ in the x direction. The impermeable walls are located within the region $(0 \le \bar{x},\ z < +\infty)$ and $(0 \le \bar{x} < +\infty,\ -\infty < \bar{z} \le -h)$.

The solution to the problem is obtained in terms of simple formulas entirely suitable for numerical calculations. In the special case of the porous body removed, these formulas define the temperature field of a thick-walled channel carrying a gas stream:

$$\frac{T_G(x,\ t)-T_0}{T_e-T_0}=\exp(-\delta x)\int_0^\infty\left\{\frac{l}{\sqrt{\pi\left(t-\frac{x}{\beta_2}\right)}}\exp\left[-\frac{l_1^2\tau_1^2}{4\left(t-\frac{x}{\beta_2}\right)}\right]+\operatorname{erfc}\left(\frac{l\tau_1}{2\sqrt{t-\frac{x}{\beta_2}}}\right)\right\}\exp(-\tau_1)\,I_0(2\sqrt{\delta x\tau_1})\,d\tau_1\eta\left(t-\frac{x}{\beta_2}\right), \tag{1}$$

$$\frac{T_W(x,\ z,\ t)-T_0}{T_e-T_0}=\exp(-\delta x)\int_0^\infty\exp(-\tau_1)\operatorname{erfc}\left(\frac{l\tau_1+z}{2\sqrt{t-\frac{x}{\beta_2}}}\right)I_0(2\sqrt{\delta x\tau_1})\,d\tau_1\eta\left(t-\frac{x}{\beta_2}\right). \tag{2}$$

Here $T_G(x, t)$ and $T_W(x, z, t)$ denote the temperature of the gas and of the pipe wall respectively, T_0 denotes the initial temperature, T_e denotes the inlet temperature of the gas, x and z are space coordinates, t is time, δ, l, β_2, l_1 are dimensionless parameters involving the thermophysical properties of gas and pipe. Formula (1) is used here for calculating the gas temperature at instants of time $\bar{t} = 30$ sec and $\bar{t} = 300$ sec. Analogous formulas are derived for the radial flow of liquid or gas.

HYDRODYNAMICS AND HEAT TRANSFER IN FILM CONDENSATION OF PROPANE VAPOR IN A VERTICAL HEAT EXCHANGER†

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UDC 536.42.34

The authors analyze the results of an experimental study concerning the hydrodynamics and the heat transfer in the condensation of saturated propane vapor flowing through tubes of a vertical heat exchanger. The tubes were made of grade 1Kh18N10T steel, d = 8 mm in diameter and l = 1 m long. The tests were performed at thermal flux densities q ranging from 5 to 38 kW/m^2 and at temperature differences from 2.5 to 18°C under 5-9 bars pressure. The mean-over-the-surface heat transfer coefficients were determined while the Reynolds number Re of the vapor stream was varied from 3000 to 50,000 and the Reynolds number Re_c of the condensate film was varied from 100 to 1300.

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An analysis of the test data has revealed two fundamental hydrodynamic modes of film flow:

flow mode I characterized by the simultaneous action of gravity and interphase friction forces;

flow mode II dominated by the interphase friction forces.

The boundary relation which defines the regions where these modes prevail is:

$$\mathrm{Re}_{c,cr} = C\left(\mathrm{Fr}_V \frac{\rho_V}{\rho_c}\right)^{-2.2} \text{ for } 0.8 < \mathrm{Fr}_V \frac{\rho_V}{\rho_c} < 2.2, \tag{1}$$

with $C \simeq 500$.

When $\mathrm{Re}_c < \mathrm{Re}_{c,cr}$ (mode I), the effect of velocity on the heat transfer rate can be determined from the equations:

$$\overline{\alpha}_c/\overline{\alpha}_0 = 1.28\pi_u^{0.03} \text{ for } 2.5 < \pi_u < 20, \tag{2}$$

$$\overline{\alpha}_K/\overline{\alpha}_0 = 1.15\pi_u^{0.065} \text{ for } \pi_u > 20. \tag{3}$$

The effect of vapor velocity on the heat transfer coefficient within this range, according to these equations, agrees qualitatively with the results of the numerical Nusselt solution for laminar film condensation, although the actual heat transfer rate is much lower (by 30-50%) than the theoretical one, but the discrepancy widens with increasing velocity.

For the $\mathrm{Re}_c > \mathrm{Re}_{c,cr}$ range (mode II), the test data were generalized by the following equation:

$$\overline{\alpha}_c/\overline{\alpha}_0 = A\pi_u^m, \tag{4}$$

where $A = 0.136$ and $m = 24.8/\mathrm{Nu}_0^{0.9}$ for $50 < \pi_u < 75$, $A = 0.183$ and $m = 16.1/\mathrm{Nu}_0^{0.8}$ for $75 < \pi_u < 100$.

Within this range the test data agree sufficiently well with the relation proposed by Kruzhilin and Boiko for turbulent film flow under predominating interphase friction forces.

NOTATION

$\mathrm{Re}_c = qL/r\mu_c g$	is the Reynolds number for the film;
$\mathrm{Fr}_V = U_V^2/gd$	is the Froude number for the vapor stream;
$\mathrm{Nu}_0 = \alpha_0 d/\lambda_c$	is the Nusselt number for stationary vapor;
$\pi_u = \mathrm{Fr}_V \rho_V/\rho_c$;	
q	is the thermal flux density;
U_V	is the mean velocity of vapor stream;
α_c	is the heat transfer coefficient for vapor in motion;
α_0	is the heat transfer coefficient for vapor at standstill;
L, d	are the length and diameter of tube;
μ_c	is the dynamic viscosity of condensate;
λ_c	is the thermal conductivity of condensate;
ρ_V	is the density of vapor;
ρ_c	is the density of condensate.

ENGINEERING METHODS OF CALCULATING THE VARIATIONS IN STEAM TEMPERATURE IN THE BOILER-TURBINE BYPASS

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UDC 621.165.001.24:536.5

In the solution of various problems arising in the analysis of steam turbine heating during transient modes of operation, it is necessary to take into account that the temperature of steam entering the turbine lags the temperature variation past the boiler, as a result of heat dissipation and accumulation in the boiler-turbine bypass. For a qualitative estimate of this effect in modern aggregates of typical configurations and with typical flow rate characteristics (of fresh or superheated steam), the bypass is represented by a first-order inertial element with a time constant equal to the ratio of heat capacities of metal mass to steam per unit time: $T_c = c_M G_M / c_c D_c$. The amplitude-frequency characteristics of temperature $A(\omega) = (1 + \omega^2 T_c^2)^{-1/2}$ constructed on this basis agree fairly well with those calculated by more accurate but not less explicit formulas. Tests and calculations have shown that the effect of the bypass can be appreciable, especially at low steam rates.

When calculating the temperature of steam entering a turbine and the turbine heatup during start, faults, loading, etc., one must consider the wide variations in the flow rate as well as the complex trend of the steam temperature past the boiler and, in many cases, the nonuniform geometry with nonuniform initial and boundary conditions along the bypass. Several algorithms are shown here by which changes in the steam temperature θ and in the metal temperature t along the bypass can be calculated on the basis of recurrence formulas, using a digital computer for a numerical solution of the differential equation of heat balance for steam and metal. The stability criteria for such a solution are established with respect to the width of computation steps along the time τ and length z coordinates. The effect of a nonuniform temperature distribution across the wall thickness of an equivalent pipe is accounted for by introducing the appropriate coefficient $\psi = (1 + 0.35\ \mathrm{Bi})^{-1}$.

For a bypass with wet or saturated steam, especially in atomic power plants with water-cooled reactors, the change in steam quality under constant boundary conditions α = const and θ = const is described by the expression

$$x(z, \tau) - x(0, \tau) = -\frac{\alpha F \psi}{D_c r} (\theta - t_{\mathrm{init}}) \exp\left(-\frac{\alpha F \psi}{c_M G_M} \tau\right),$$

where F denotes the area of the heat transfer surface along the bypass over a distance z and r is the specific heat of evaporation. More complex problems as, for instance, with a variable aggregate state of the heat carrier α = var and θ = var, etc., are solved on a digital computer by the numerical method of finite differences.

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A METHOD OF CALCULATING THE TEMPERATURE OF A SHELL AROUND A ROD WITH INTERNAL HEAT GENERATION DURING A SHARP DROP IN THE RATE OF HEAT TRANSFER TO THE COOLING FLUID

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UDC 536.2.01

The temperature field of a rod with uniform internal heat generation is considered when the rod is inside a thin metallic shell. The gap between rod and shell is filled with gas. It is assumed here that steady-state cooling of the rod occurs during surface boiling. It is also assumed that, after a sharp decrease in the coolant supply, the heat transfer coefficient becomes smaller and the heat generation rate begins to drop exponentially. The heat transmission process is described by the equation of heat conduction for the rod and by the equation of heat balance for the shell. The problem is solved with the aid of a Laplace transformation; the original function is defined by a contour integral.

On the basis of the formulas derived here, the variations in the mean temperature of a shell (stainless steel 0.6 mm thick) and of a ceramic rod (8.9 mm in diameter) are calculated for an initial steady-state heat generation of $q = 6.16 \cdot 10^8$ kcal/m$^3\cdot$h. It is assumed, moreover, that the heat generation rate drops to $0.3q_0$ at time $t = 0$ and then decreases according to the relation $q = 0.3q_0e^{-0.08t}$. The results of calculations are shown in Fig. 1 for two widths of the helium-filled gap. For comparison, a curve is also shown representing an instantaneous shutoff of heat generation. According to the graph, the process continues for 1 min and has the characteristics of a thermal shock. The rise in the shell temperature, following a sharp reduction in the heat transfer rate from 100,000 to 2000 kcal/m$^2\cdot$h$\cdot$°K, is due to the rather large quantity of heat stored in the rod, which raises the temperature of the latter when the thermal resistance at the shell boundary increases.

This illustrative example shows that, as the heat transfer rate decreases, the shell temperature may for a short time exceed the allowable level.

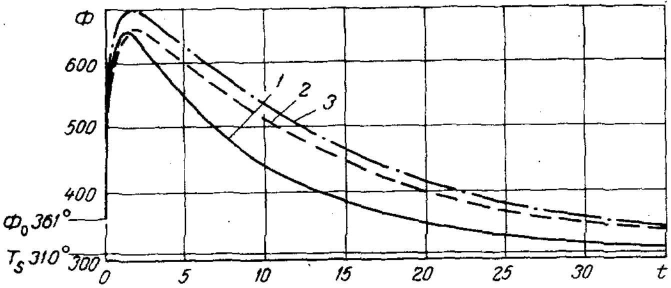

Fig. 1. Mean shell temperature (°C) as a function of time (sec) during the transient: gap width 0.096 mm (1), 0.048 mm (2); after a step decrease of the heat generation rate (3).

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*Deceased.

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SIZE DISTRIBUTION OF VAPOR BUBBLES DURING EFFERVESCENCE OF A SUPERHEATED LIQUID*

Yu. V. Mironov

UDC 536.24

The author analyzes the size distribution of vapor bubbles during effervescence of a superheated liquid. If neither bubble merger nor bubble fragmentation occurs, then the distribution density function ν is found by solving the nonlinear Cauchy problem for the differential equation

$$\frac{\partial \nu}{\partial t}+\frac{\partial}{\partial r}(v\nu)=0,$$

where t denotes time and v denotes the rate of increase of radius r.

If the nucleation rate of critical-size bubbles is determined only by the superheat of the liquid and if nonlinearities can be disregarded as negligible, then an analytical solution is given for the initial stage of the effervescence process.

SHIFT OF THE INTERPHASE BOUNDARY DURING DESUBLIMATION OF WATER VAPOR UNDER BOUNDARY CONDITIONS OF THE THIRD KIND†

V. K. Safonov and A. Z. Volynets

UDC 536.422.4

The desublimation of water vapor is analyzed under boundary conditions of the third kind, i.e., with the thermal resistances between coolant and heat dissipating surface taken into account. The coolant temperature and the vapor pressure in the apparatus are assumed constant.

The solution to the problem, namely the shift of the interphase boundary is expressed as

$$\xi(\tau)=\sqrt{4\beta^2 a\tau+\delta^2}-\delta, \tag{1}$$

with $\xi(\tau)$ denoting the thickness of the ice layer, τ denoting time, and a denoting the thermal diffusivity. Parameters β and δ are defined as follows:

$$\beta=\sqrt{\frac{c(T_s-T_f)}{2r}} \quad \text{and} \quad \delta=\lambda\left(\frac{1}{\alpha}+\sum_{i=1}^{n}R_i\right).$$

with the specific heat of ice c, the thermal conductivity of ice λ, the heat of phase transformation r, the ice temperature at the interphase boundary T_s, the coolant temperature T_f, the heat transfer coefficient α, and the thermal resistance R.

Formula (1) was checked out experimentally at a constant coolant temperature (T_f = 213°K) and various vapor pressures (0.1-4.5 mm Hg). The equivalent ice thickness (dimension δ) was varied from 0.001 to 0.01 m.

The test points did not deviate from values based on formula (1) by more than 15%.

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† Original article submitted October 6, 1971; abstract submitted April 24, 1972.

FORWARD PROBLEM AND REVERSE PROBLEM IN DESORPTION DYNAMICS WITH A VARIABLE INTRAGRAIN DIFFUSIVITY

L. K. Tsabek and I. V. Rozen

UDC 541.183

The forward problem and the reverse problem in desorption dynamics and kinetics are solved with the aid of the two-dimensional Laplace integral transformation, taking into account the linear kinetics of sorption (desorption) events as well as the equations of material balance, during diffusion, both for symmetric sorbent grains and for a cylindrical column. Desorption diffusivity is represented as a power function of the grain radius. Such a model yields a correct description of kinetic and dynamic desorption test curves with unusually long "tails" corresponding to long time periods. The equation of sorption kinetics is analyzed next. The dynamic curves of desorption fronts at the outlet end (forward problem), in the case of a column with fixed length c(L, t), and the concentration distribution along a column at a fixed instant of time c(z, T) are represented in terms of Hermite polynomials $H_n(x)$ and series in incomplete gamma functions I(b, x):

$$c_f(L, t) = \left\{1 - \operatorname{erf}(y) + 2\sqrt{2\mu_2}\sum_{n=3}^{\infty} A_n \exp(-y^2) H_{n-1}(y)\right\}\left\{1 + \operatorname{erf}(y_0) - 2\sqrt{2\mu_2}\sum_{n=3}^{\infty} A_n (-1)^n \exp(-y_0^2) H_{n-1}(y_0)\right\}^{-1};$$

$$y = (t - \alpha_1)(2\mu_2)^{-1/2}, \quad y_0 = \alpha_1 (2\mu_2)^{-1/2};$$

$$c_f(L, t) = 1 - \sum_{n=0}^{\infty} N_n \sum_{k=0}^{n} \frac{(-1)^k n!\, \Gamma(m + n + 1)}{k!\,(n - k)!} \mathrm{I}\left(m + k + 1, \frac{t}{\tau}\right);$$

$$N_n = \sum_{i=0}^{n} \frac{(-1)^i \alpha_i}{\tau^i i!\,(n - i)!\,\Gamma(m + i + 1)};$$

$$A_n = \sum_{k=0}^{\left[\frac{n}{2}\right]} \frac{(-1)^k 2^{-\left(\frac{n}{2} + k + \frac{1}{2}\right)}}{\sqrt{\pi}\, k!\,(n - 2k)!} \mu_{n-2k}\, \mu_2^{\left(k - \frac{n}{2} - \frac{1}{2}\right)};$$

τ_0, τ are parameters;

$$c_f(z, T) = 1 - \sum_{n=0}^{\infty} N_n^0 \sum_{k=0}^{n} \frac{(-1)^k n!\, \Gamma(i + k + 1)}{k!\,(n - k)} \mathrm{I}\left(i + k + 1 \frac{z}{\tau_0}\right);$$

$$N_n^0 = \sum_{j=0}^{n} \frac{(-1)^j m_j}{\tau_0^j j!\,(n - j)!\,\Gamma(i + j + 1)}.$$

The solution to the reverse problem is found with the aid of derived expressions for the initial moments (α_n, m_n) and the central moments (μ_n). As an example, the method shown here is applied to plotting dynamic curves of sorption and desorption fronts for benzene vapor and fine-porous silica gel. It is shown, furthermore, that sorption curves must be described in terms of Hermite series, desorption—elution in terms of generalized Laguerre polynomials, and desorption front curves in terms of series in referred incomplete gamma functions.

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TRAVEL OF A THERMAL PERTURBATION THROUGH A POROUS UNDEFORMABLE MEDIUM

L. K. Tsabek

UDC 541.183

The travel of a δ- and a step perturbation through a porous undeformable body made up of symmetrically shaped grains is analyzed in the forward problem and in the reverse problem formulation. Analytical expressions are derived in the form of Hermite-polynomial series and probability-integral series, with the aid of which one can describe the temperature field along the filtration channel at a fixed instant of time and also the temperature field at the exit of such a channel when its length is fixed. Two cases are considered here, both of interest in practical applications where the thermal diffusivity of the solid granular phase is either much higher (system of Eqs. (1)-(3)) or much lower than that of the filtering liquid or gaseous phase (system of Eqs. (4)-(7)):

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial z} + \delta\gamma_1(T - T^0) + h(T - T_0) = \chi\frac{\partial^2 T}{\partial z_2}; \tag{1}$$

$$\frac{\partial T^0}{\partial t} = \gamma(T - T^0), \qquad \frac{\partial T_0}{\partial t} = \gamma_2(T - T_0) - \gamma_3 T_0; \tag{2}$$

$$T|_{z=0} = T_0^0\delta(t), \quad T|_{z=0} = T_0^0\eta(t), \quad \frac{\partial T}{\partial z} = T_0^0 l_*(t), \quad T_0^0 = \text{const}; \tag{3}$$

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial z} + \delta\chi_i\left(\frac{1+\nu}{r}\frac{\partial T^*}{\partial r}\right)\bigg|_{r=a} + h(T - T_0) = \chi\frac{\partial^2 T}{\partial z^2}; \tag{4}$$

$$\frac{\partial T^*}{\partial t} = \chi_i\left(\frac{\partial^2 T^*}{\partial r^2} + \frac{\nu}{r}\frac{\partial T^*}{\partial r}\right) + \gamma_1(T^* - T^0) = 0; \tag{5}$$

$$\frac{\partial T^0}{\partial t} = \gamma(T^0 - T^*), \qquad \frac{\partial T_0}{\partial t} = \gamma_2(T - T_0) - \gamma_3 T_0; \tag{6}$$

$$\chi_i\frac{\partial T^*}{\partial r}\bigg|_{r=a} = \alpha(T - T^*)|_{r=a}, \qquad \frac{\partial T^*}{\partial r}\bigg|_{r=0} = 0. \tag{7}$$

Equations (1) and (4) are equations of heat balance in the filtering gas or liquid respectively, the first Eq. (2) describes the heat balance in the solid phase of porous grains, the Eq. (2) and the second Eq. (3) describe the heat balance in the walls of the cylindrical filtration channel. Equation (5) describes the heat balance in the liquid inside a porous grain, the first Eq. (6) describes the heat balance in the solid phase of grains (T, T^0, T^*, and T_0 denote respectively the temperature of the gas, the solid phase of a porous grain, the liquid inside a porous grain, and the wall of a cylindrical filtration channel). From the solution to Eqs. (1)-(7), obtained with the aid of a two-dimensional Laplace transformation, follow analytical expressions for the initial and four central moments. A method is also shown by which the heat transfer coefficients and the thermal diffusivities can be determined from a system of algebraic equations set up earlier (reverse problem).

The dependence of the lengthwise diffusivity on the filtration velocity is also analyzed.

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VISCOSITY OF GASEOUS ALKENES*

A. G. Zhdanov and V. E. Lyusternik

UDC 532.13

The viscosity of gaseous methane homologs was studied earlier in [1] systematically over a wide temperature range, the results indicating an interesting relation between the viscosity and the molecular structure of the substance. In view of this, it would certainly be worthwhile to examine the viscosity of the closest analog of paraffins, namely the homologous ethylene (olefine) series. The members of this series are of great practical importance, inasmuch as they provide the basic substances needed for the synthesis of polymer compounds.

A complete lack of test data pertaining to the viscosity of heavy alkenes has made it necessary in our study to examine rather high-molecular compounds of this series, namely: n-hexene-1, n-heptene-1, n-octene-1, n-decene-1, and n-dodecene-1.

The measurements were made by the absolute method with a capillary viscometer under a pressure of 0.8 bar and over a wide temperature range [1].

The total of all possible measurement errors did not exceed 1.8%.

For this study the authors used high-purity materials with an initial concentration not below 99.5%, on the average. Structural changes in the substances, occurring after high temperature tests, were examined by repetitive chromatographic analysis and changes in the viscometer geometry were examined by control tests with standard specimens.

All reliable test data pertaining to the viscosity of ethylene homologs were evaluated together and a regular trend was detected: a decrease in viscosity with increasing molecular weight (M).

The following semiempirical formula, of the same type as the Sutherland equation, is proposed for describing available test data on the viscosity of gaseous alkenes:

$$\eta = \frac{(11.24 + 0.0219M)\, T^{3/2}}{T + 84.14 + 6.05M + 0.0055MT} .$$

Over the entire test range of temperatures, it describes the viscosity of any homolog in the series with a standard deviation of 2%.

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VISCOSITY OF GASEOUS ALKYNES†

A. G. Zhdanov and V. E. Lyusternik

UDC 523.13

The systematic study concerning the viscosity trends among the various hydrocarbon homologs is concluded by this report on the acetylene series. The viscosity of gaseous substances in this series was measured only on acetylene itself and its nearest homolog propylene. Earlier the authors had studied n-hexyne-1, n-heptyne-1, n-octyne-1, and n-decyne-1 under a pressure of 0.9 bar and over a wide

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temperature range within the respective thermal stability limits. The measurements were made by the absolute method with a capillary viscometer constituting a closed circulation system.

The total of all possible measurement errors did not exceed 1.8%.

It was practically impossible to test the homologs heavier than decyne because of their thermal instability in the gaseous state.

The purity of the hydrocarbons studied here was checked before and after the tests; it did not fall below 99.5%.

An analysis of reliable existing data on the viscosity of alkynes has yielded the following formula for this viscosity as a function of both the temperature and the molecular weight:

$$\eta = \frac{(9.99 + 0.0028M)\,T^{3/2}}{T + 71.74 + 4.46M + 0.0019MT}, \tag{1}$$

a formula analogous to those for alkanes and alkenes. The coefficients in Eq. (1) were fitted numerically so as to hold the standard deviation over the entire temperature range within 2% for any of the homologs.

VISCOSITY OF GASEOUS OCTANE AND CYCLOHEXANE ISOMERS

A. G. Zhdanov and V. E. Lyusternik

UDC 532.13

The authors studied the effect of the molecular structure on the physical properties of substances and, particularly, on their viscosity, thus making it feasible to calculate these properties on the basis of the known chemical structure.

This problem is of most immediate interest in the case of various hydrocarbon isomers, the number of which increases as the number of carbon atoms increases.

As test objects the authors selected chemically pure 2,2,4-trimethylpentane and cyclohexane. With a maximum theoretical error of ±1.8%, the viscosity of their vapors was measured over a wide range of temperatures from 75 to 415°C by the absolute method with a capillary viscometer operating as a closed circulation system.

As a result, the existing test data could be refined and the temperature range of measurements could be extended.

The temperature-dependence of viscosity is, in the case of iso-octane and cyclohexane, described by the Sutherland formula with a less than ±0.5% error, which agrees within ±2.0-2.5% with all data available in the technical literature.

The effective diameters of branched and cyclic molecules were calculated from viscosity test values, whereupon these molecules were compared with those of normal hydrocarbons.

In their preceding studies the authors have demonstrated the existence of a regular relation between the gasokinetic parameters and the molecular structures of normal hydrocarbons in the various homologous series. Such a comparison is, therefore, of definite interest.

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BLOCKAGE OF WATER BY ICE DURING FREEZING IN PORES*

B. V. Zheleznyi

UDC 541.183:536.421.4

Considered are the maximum possible stresses in ice during fast crystallization of subcooled water in capillaries of two very simple shapes: conical and contained within the contact zone between two identical spheres. The adhesion force between ice and capillary walls is assumed to govern the mechanical equilibrium, inasmuch as the strength of ice is relatively high. Under this assumption, differential equations are set up which describe the extremal stress distribution in ice. Their solution indicates that, under the given conditions, water can be blocked by ice and both states can coexist for a longer time at below-zero temperature and with an interphase boundary between them.

The maximum absolute water volume (η) and the maximum relative water volume (γ) which can be blocked in capillaries of either shape are calculated for various fill levels and for various shape factors. Both γ and η as functions of the relative moisture content φ (which, according to the Kelvin equation, determines the fill level in capillaries) are calculated for water blockage under an ice cap between two touching spheres, as shown in Fig. 1 (φ expressed in fractions of the sphere volume). The extreme values of γ and η at $\varphi = 1$ for this case are $7 \cdot 10^{-6}$ and $7 \cdot 10^{-7}$ respectively.

Characteristic are, above all, the extremely small values of γ and η and, secondly, the fast increase in η with higher φ levels. The small values of γ and η indicate that the blockage has no appreciable effect at all on the phase balance during the freezing of water in porous and disperse materials, this balance being governed by physically and chemically bonded water. The fast increase in η at higher φ levels confirms this, inasmuch as the trend of η (absolute quantity of unfrozen water) versus moisture content φ is, according to experimental evidence, different from the $\eta(\varphi)$ shown in Fig. 1.

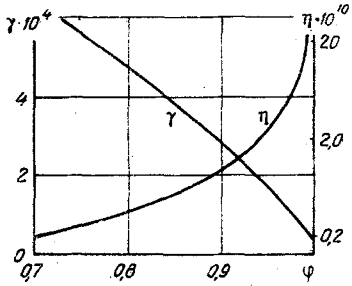

Fig. 1. Ratio of blocked water volume to ice cap volume (γ) and to sphere volume (η) around the contact point between two spheres. The η-scale is logarithmic.

SOLUTION OF STEADY-STATE HEAT CONDUCTION PROBLEMS WITH RANDOM BOUNDARY CONDITIONS†

B. S. Gonchar and A. M. Aizen

UDC 536.2

The use of "direct" methods for solving heat conduction problems with random boundary conditions of the first kind is considered in the one-dimensional case. This makes it possible to determine the probability distribution density of temperature u and thermal flux q as functions of the relatively random distributed boundary conditions, if the probability distribution density of the latter is given.

Let u_1 be a random variable with the probability distribution density $f_1(u_1)$ and let u_2 be a deterministic quantity. We then have the probability distribution density of u and q expressed as follows:

$$f(z) = \frac{1}{|\theta_{1z}|} f_1\left(\frac{z - \theta_{2z}u_2}{\theta_{1z}}\right), \tag{1}$$

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where z = u or q, and functions θ_{1z}, θ_{2z} depend on the choice of coordinates. Subscripts 1 and 2 refer to the coefficients of u_1 and u_2 respectively.

Of special interest is the case where u_1 and u_2 are random variables with probability distribution densities $f_1(u_1)$ and $f_2(u_2)$. The probability distribution density of temperature or thermal flux is determined according to the formula

$$f(z) = \frac{1}{|\theta_{2z}|} \int_{-\infty}^{\infty} f_1(u_1) f_2\left(\frac{z - \theta_{1z}u_1}{\theta_{2z}}\right) du_1. \tag{2}$$

With the values of f(z) obtained by formula (1) or (2) one can determine the mathematical expectancy, the dispersion, and the higher-order moments of random distributed temperature and thermal flux, which completely characterize the heat transfer process with random boundary conditions.

ONE METHOD OF SOLVING HEAT CONDUCTION PROBLEMS WHICH PERTAIN TO THE HEATING OF BULKY BODIES IN A MOVING BED

M. K. Kleiner

UDC 536.24

Heat conduction problems in the case of bulky bodies in a moving bed belong in the category of problems with complex boundary conditions: in addition to the usual boundary conditions of the third kind, there is also given an equation relating the temperature of the gases and the temperature of the heated bodies. in the general case:

$$W(X)\frac{dt_G(X)}{dX} = -Q(X) + \frac{dW}{dX}[t_G^0 - t_G(X)] \mp \mathrm{Bi}[t_G(X) - t(1, X)] \mp \mathrm{Bi}_1[t_G(X) - t_{\mathrm{amb}}], \tag{1}$$

where the given differentiable function W(X) depends on the water equivalents of the gases and of the heated bodies, and function Q(X) depends on the change in the heat generation in the gaseous phase. The meaning of all other symbols is as in [1].

The gist of the proposed method, generally similar to that shown in [2], is to first solve the third boundary-value problem for an arbitrary $t_G(X)$ and then, by inserting this solution into (1), to find the function $t_G(X)$ which will satisfy the resulting integrodifferential equation.

A numerical implementation of this method has revealed certain computational difficulties within the range of small W(X) values, i.e., near the point where the free term and the kernel of the integral equation have their singularities. If W(X) is a linear function [1]

$$W(X) = \frac{m_0}{2\nu + 2} \pm \beta X, \quad \beta = \frac{a_1 R}{\lambda f}, \quad m_0 = \frac{a_0}{Gc} \tag{2}$$

and $\beta < 1$ is sufficiently small, therefore, then the problem can be solved by the perturbation method as follows:

$$t(\rho, X) = \sum_{k=}^{\infty} \beta^{\kappa} t_k(\rho, X), \tag{3}$$

$$t_G(X) = \sum_{k=0}^{\infty} \beta^{\kappa} t_{G,k}(X). \tag{4}$$

With (3) and (4), the original problem can be reduced to a sytsem of recurrence equations for $t_k(\rho, X)$ and $t_{G,k}(X)$ at W(X) = const. Each of these equations is most conveniently solved by the method of integrodifferential equations shown earlier, and the resulting integral equations by the method in [3], the gist

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of the latter being as follows. Let $\Gamma(X, s)$ be the resolvent of the kernel $K(X, s)$. Then their integral operators $\Gamma^*(X, s)$ and $K^*(X, s)$ (also the two functions $\Gamma(X, s)$ and $K(X, s)$) are related as follows:

$$K^*(X, s) = \sum_{p=1}^{m} \varsigma_p C^*(y_p), \quad \Gamma^*(X, s) = \sum_{p=1}^{m} b_p C^*(z_p), \tag{5}$$

where ζ_p, y_p are known parameters and b_p, z_p are expressed in terms of ζ_p, y_p as follows:

$$1 + \sum_{n=1}^{m} \frac{\varsigma_n}{y_n - z} = 0, \quad 1 + \sum_{p=1}^{m} \frac{b_p}{y_n - z_p} = 0 \ (n = 1, 2, 3, \ldots, m). \tag{6}$$

The use of both methods covers the entire practical range of W(X) values.

These methods yield a series of approximate solutions, provided the third boundary-value problem is first solved by any one approximate method. The approximate solutions have checked very well against the exact solutions.

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TEMPERATURE FIELD OF A FLAT WALL WITH A PERIODICALLY VARYING HEAT TRANSFER COEFFICIENT

V. F. Stepanchuk and M. L. Guris

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When a vapor–water mixture flows through insulated pipes of a steam generator, the heat transfer coefficient may become periodically variable. This fluctuation is indicated by different vapor content levels in successive steam packets. The time-variation of the heat transfer coefficient can be expressed in terms of the following law:

$$\alpha_2 = m + ne^{i\omega\tau}.$$

Here ω denotes the radian frequency and m, n are constants ($n < m$).

The solution to the transient heat conduction problem for a flat wall at a constant temperature difference between the ambient phases and with a periodically variable heat transfer coefficient was found to be

$$T = T_1 \frac{\dfrac{\delta - x}{\lambda} + \dfrac{1}{m}}{\Sigma R} + e^{i\omega\tau}\left(A_1 \operatorname{sh} \sqrt{\frac{i\omega}{a}}\, x + B_1 \operatorname{ch} \sqrt{\frac{i\omega}{a}}\, x\right)$$
$$+ e^{2i\omega\tau}\left(A_2 \operatorname{sh} \sqrt{\frac{2i\omega}{a}}\, x + B_2 \operatorname{ch} \sqrt{\frac{2i\omega}{a}}\, x\right) + \ldots + e^{ki\omega\tau}\left(A_k \operatorname{sh} \sqrt{\frac{ki\omega}{a}}\, x + B_k \operatorname{ch} \sqrt{\frac{ki\omega}{a}}\, x\right),$$

with δ denoting the wall thickness, λ denoting the thermal conductivity of the wall, a denoting the thermal diffusivity, T_1 denoting the temperature of the heating medium, and coefficients n, m determined from the boundary conditions.

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The obtained solution indicates that the temperature field of a flat wall with a periodically variable heat transfer coefficient is quite complex, and its components can be represented in terms of harmonic series.

The magnitude of temperature fluctuations with time and the local temperature gradients in the wall can be quite appreciable and are determined by the ratio n:m, the radian frequency, the wall thickness, and the thermophysical properties of the wall material.

The performance of steam generator piping was calculated by this method.

The results indicate wide temperature fluctuations and large temperature gradients over a wide range of radian frequencies.

EFFECT OF MODIFICATIONS IN THE GEOMETRY OF BONDED SURFACES ON THE THERMAL RESISTANCE OF SUCH JOINTS*

V. M. Popov

UDC 621.792.053:536.24

A study was made to determine how the thermal resistance of the bonding interlayer is affected by the geometry of metallic surfaces joined together and by the viscosity of the adhesive. It has been established experimentally that the thermal resistance of such an interlayer of low-viscosity adhesive increases with surface roughness and waviness, also with macro-deflections from the plane of the joint.

The thermal resistance of an adhesive joint as a function of the interlayer thickness is, in the case of low-viscosity adhesives, accurately enough described by the formulas given here.

As the viscosity of the adhesive increases, one notes that the thermal resistance of the joint increases at a fixed interlayer thickness. This trend is explained by the slower filling of surface cavities due to the higher viscosity of the adhesive.

An effective method is proposed for reducing the thermal resistance of adhesive joints by adding a disperse filler with high thermal conductivity.

COMPUTER(DIGITAL)-AIDED DESIGN OF THERMISTOR CIRCUITS WITH VARIABLE CONDITIONS OF HEAT TRANSFER†

N. M. Nedosekina and V. A. Palagin

UDC 621.316.825

The transient equation for an $R-R_T$ circuit is nonlinear. It has not been possible to arrive at a general analytical solution to this equation, because the thermistor parameters k and c_V as well as the

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power drain are functions of the temperature and depend not only on the type of thermistor but also on the conditions of heat transfer. As a component of an electric circuit, a thermistor can be used for probing various physical properties of the ambient medium functionally related to the temperature. Such parameters are the concentration, the velocity, and the temperature of the ambient medium as well as their combination. These parameters may, moreover, vary in diverse pulse modes, which complicates appreciably the calculation of transients in an $R-R_T$ circuit. If those parameters of the medium which are related to the conditions of heat transfer vary (and thus at the same time the basic thermistor parameters vary), then the only way to analyze transients in thermistor circuits is with the aid of analog or digital computers.

The transients are analyzed here by the Runge–Kutta method. An algorithm is developed for solving the transient equation for an $R-R_T$ circuit. A program is designed for use on a PROMIN'-M digital computer. The transients are calculated for various modes of pulse variation in the conditions of heat transfer.

The results can be used in the design of thermistor circuits operating in continuous-run and pulse systems for automatically controlling and regulating the parameters of the medium.

MONODISPERSE ATOMIZATION OF A LIQUID BY A ROTATING DISK

V. F. Dunskii and N. V. Nikitin

UDC 66.069.8

Researchers have often been interested in the ability of rotating-disk atomizers to break up a liquid into a monodispersion, i.e., into identical droplets at a low flow rate of that liquid. A generalization of the research data has made it possible to develop an approximate but valid calculation procedure.

The basic formulas are given here:

First mode of atomization. The range of parameter values corresponding to the first mode of atomization is defined by the inequality

$$0 < \frac{0.86\, \omega^{0.48}\, \nu^{0.12}}{R^{0.30}} \left(\frac{\rho Q}{\sigma}\right)^{0.62} \leqslant 1. \tag{1}$$

The diameter of main droplets is

$$d = \frac{2.9}{\omega} \left(\frac{\sigma}{R\rho}\right)^{1/2}. \tag{2}$$

The weight fraction of satellite droplets is

$$E = 86\, \frac{Q^{0.62}\, \rho^{0.62}\, \omega^{0.48}\, \nu^{0.12}}{\sigma^{0.62}\, R^{0.30}}\ (\%). \tag{3}$$

The diameter of the median (with respect to mass) satellite droplets is

$$d_m = \frac{1.15}{\omega} \left(\frac{\sigma}{R\rho}\right)^{1/2}. \tag{4}$$

Second mode of atomization. The range of parameter values corresponding to the first and the second mode, i.e., to monodisperse atomization is defined by the inequality

$$0 < \frac{\rho^{1.77}\, Q^2\, \omega^{1.2}\, \nu^{0.33}}{\sigma^{1.77}\, R^{1.37}} < 4.5. \tag{5}$$

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The diameter of the median (with respect to mass) droplets is, in the case of low-viscosity liquids

$$d_m = 1.48 \left(\frac{Q\sigma}{\rho_* \omega^3 R^{5/2}} \right)^{2/7}, \tag{6}$$

and in the case of high-viscosity liquids

$$d_m = \frac{2.12\ \sigma^{0.27} Q^{0.306} \nu^{0.014}}{\rho^{0.27} R^{0.725} \omega^{0.84}}. \tag{7}$$

The size distribution of satellite droplets in the first mode and of all droplets in the second mode is a normal one with the dispersion $\beta^2 \approx 0.17\ \mathrm{dm}$.

The trajectory of droplets in the first mode and of low-viscosity liquid droplets in the second mode is defined by the formulas

$$\tau = A \ln \frac{u_0}{u} \left(\frac{1 + bu^{2/3}}{1 + bu_0^{2/3}} \right)^{3/}, \tag{8}$$

$$x = \frac{3a}{b} \left(u_0^{1/3} - u^{1/3} - \frac{1}{\sqrt{b}} (\mathrm{arctg}\ u_0^{1/3} \sqrt{b} - \mathrm{arctg}\ u^{1/3} \sqrt{b}) \right), \tag{9}$$

$$z = gA\,[\tau - A\,(1 - e^{-\tau/A})]. \tag{10}$$

NOTATION

R	is the radius of disk;
ω	is the angular velocity of disk;
Q	is the flow rate of liquid;
ρ	is the density of liquid;
ν	is the kinematic viscosity of liquid;
σ	is the surface tension of liquid;
τ	is the time;
u_0	is the absolute velocity of droplet after separation from the disk edge, equal to the peripheral disk velocity;
u	is the horizontal component of droplet velocity at point (x, z) on trajectory;
$a = d^2\rho/18\,\nu_A \rho_A$;	
$b = 0.167\,(d/\nu_A)^{2/3}$;	
ρ_A	is the density of air;
ν_A	is the kinematic viscosity of air.

TURBULENT FLOW OF ANNULAR FILMS ALONG A VERTICAL SURFACE

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M. N. Chepurnoi, and V. É. Shnaider

UDC 532.542

The authors consider the drain of turbulent liquid films along the inside surface of a vertical circular pipe. In analyzing this flow, they assume a film to be sufficiently thin as compared to the pipe radius. A film flows under the force of gravity and under a constant pressure gradient.

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