ABSTRACTS OF PAPERS DEPOSITED AT VINITI*

HEAT-TRANSFER PROCESS LOOPS IN INTERMEDIATE-CARRIER SYSTEMS

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UDC 536.27

A heat-transfer system with an intermediate heat carrier (the first loop in a nuclear power system, fire tubes in indirectly heated systems), is considered as a device with a heat-transfer loop; in such a loop one has a closed sequence of processes, which bring the parameters of the intermediate carrier back to the initial values and which are intended to transmit heat from the primary carrier to the secondary one. An essential difference between such a cycle and a thermodynamic cycle is that there is no conversion of heat into external mechanical work. The heat-transfer cycle consists of processes of heating, cooling, and displacement of the intermediate carrier. The cycle will be ideal if the heat losses are small, as are the pressure and mass losses. Then the transport processes will be adiabatic for the intermediate carrier, while the heat-transfer processes will be isobaric; the looping number and change in the amount of heat are

$$\mu_{\mathrm{I}} \equiv G_{\mathrm{i}}/G_{\mathrm{i}} = c_1 W_{\mathrm{i}}/c_{\mathrm{i}} W_1 = \mathrm{const}; \qquad \mu_{\mathrm{II}} \equiv G_{\mathrm{i}}/G_2 = \mathrm{const}; \quad \oint T dS = 0. \tag{1}$$

There is no meaning in representing an ideal cycle in T — S coordinates, since the area in such coordinates will be zero; in t — αF coordinates (or t — kF for recuperators) the areas bounded above and below by the temperature curves correspond to the amount of heat transferred. Figure 1 shows the variations in the heat-carrier temperatures (in the general case, in the enthalpies) and in the temperature differences between them. The efficiencies σ_{I} and σ_{II} of the components and of the system as a whole σ_{c} are defined by the ratios of the corresponding areas. For instance, for the system

$$\sigma_{\mathrm{c}} \equiv \frac{Q}{W_{\mathrm{min.c}}\, \Delta t_{\mathrm{max.c}}} \simeq \frac{2 f_{abcd}}{f_{1'32'4}} \cdot \frac{W_{\mathrm{i}}}{W_{\mathrm{min.c}}}; \quad \Delta t_{\mathrm{max.c}} \equiv t_1' - t_2' = \Delta t_{\mathrm{maxI}} + \Delta t_{\mathrm{maxII}} - \delta t_{\mathrm{i}} \tag{2}$$

$$\frac{1}{\sigma_{\mathrm{c}}} = \frac{1}{\sigma_{\mathrm{I}}} \frac{W_{\mathrm{min.c}}}{W_{\mathrm{min.I}}} + \frac{W_{\mathrm{min.c}}}{W_{\mathrm{min.II}}} \cdot \frac{1}{\sigma_{\mathrm{II}}} - \frac{W_{\mathrm{min.c}}}{W_{\mathrm{i}}}. \tag{3}$$

Formula (3) is suggested as a general relationship for the efficiency of such a loop; one can obtain as particular cases expressions given in [1] for particular ratios of the water numbers of the carriers.

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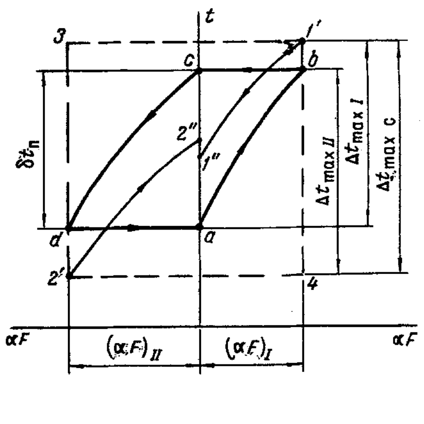

Fig. 1. Heat-transfer cycle with an intermediate heat carrier in a system with two countercurrent regenerative chambers (heat carriers: abcd intermediate; 1'11" primary; 2'2" secondary).

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For instance, with $W_1 > W_2 > W_i$ it follows from (3) that $1/\sigma_c = (1/\sigma_I + 1/\sigma_{II} - 1) W_2/W_i$. The area $f_{1'32'4}$ corresponds to the standard heat-transfer cycle, i.e., $\sigma = 1$, and the efficiency is the higher the more the actual cycle fills the standard one (the closer δt_i approaches $\Delta T_{max.c}$) for a given $W_i/W_{min.c}$. Concepts on such cycles are also useful under real conditions of use, as in many closed-loop systems. Combined analysis of the thermodynamic cycle and the heat-transfer cycle may here be useful.

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Original article submitted October 31, 1971.

MODIFIED HIRSCHFELDER EQUATION OF STATE FOR GASES

A. M. Shelomentsev and N. K. Bolotin

UDC 536.71

Hirschfelder's well-known equation of state for the gaseous phase of individual substances is placed at the basis of a generalized equation of state for nonpolar substances.

The modified equation finally takes on the following form:

I. Any T_R, $\rho_R \le 1$.

$$P_R = T_R[-W_1(T_R)\rho_R^2 - W_2(T_R)\rho_R^3 + g(\rho_R)],$$

$$W_1(T_R) = \frac{5.5}{T_R} + \frac{m - 5.5}{T_R^2},$$

$$W_2(T_R) = 0.5\,[2m - 175.7 + 191.8\beta - 72.48\beta^2 + 8.885\beta^3]\left(1 - \frac{1}{T_R^2}\right),$$

$$g(\rho_R) = \frac{(1+m)^3\rho_R}{m(3m-1) - (3m^2 - 6m - 1)\rho_R + m(m-3)\rho_R^2},$$

$$m = 43.164 - 272.73\left(1.03 - 0.157\beta - \frac{0.871}{\beta}\right) + 706.63\left(1.03 - 0.157\beta - \frac{0.871}{\beta}\right)^2 - 705.41\left(1 - 0.157\beta - \frac{0.871}{\beta}\right)^3.$$

II. $T_R \ge 1$, $\rho_R \ge 1$.

$$P_R = T_R\left[-W_1(T_R)\rho_R^2 - W_2(T_R)\rho_R^3 + 1 + m\rho_R^2 + \frac{S(\rho_R - 1)^5}{\rho_R} + D(\rho_R, T_R)\right],$$

$$S = -8.44 - 4.50m - 0.363m^2,$$

$$D(\rho_R, T_R) = (\rho_R - 1)^3 (T_R - 1)\left[\frac{1}{\rho_R}\left(\frac{88.5 - 3.12m}{T_R} - 44.4 + 5.22m\right) - \frac{47.8 - 4.06m}{T_R} + 23.7 - 3.26m\right],$$

$$\rho_R = \frac{\rho R_\mu T_c\left(1.03 - 0.157\beta - \frac{0.871}{\beta}\right)}{P_c M}.$$

The correlation factor β is calculated from the equation

$$\beta = \frac{T_{BR}\log(0.98162P_c)}{1 - T_{BR}}.$$

where T_{BR} is the reduced boiling temperature at atmospheric pressure; in the equations $P_R = P/P_c$, $T_R = T/T_c$, $T_{BR} = T_B/T_c$; T, °K is the temperature; P, bars is the pressure, ρ, kg/m³ is the density; M, kg/mole is the mass of a kilomole; $R_\mu = 0.083144$ m³·bar/kmole·deg.

An advantage of the equation obtained over the original equation is that it has a higher accuracy and requires only the normal boiling temperature and two critical parameters, the temperature and pressure, for its use.

A check against the P — V — T data for 11 substances showed that the average error in determining the density in the range of reduced temperatures and pressures of 0.5-10 and 0.002-40, respectively, was 0.5-2% with a maximum of 2-7%. The proposed equation can be recommended for the calculation of the thermodynamic properties of the gaseous phase of little-studied substances.

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CALCULATION OF HEAT EXCHANGE IN TURBULENTLY FLOWING LIQUID FILMS

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UDC 532.59:536.242

The main hydrodynamic parameters of turbulent film flow in vertical tubes were studied analytically. A power law of velocity distribution was used. The equations obtained for the average flow velocity, film thickness, and coefficient of hydraulic friction agree with the experimental data of L. Ya. Zhivaikin and B. V. Volgin with an accuracy of 10%.

An analysis of the heat exchange according to well-known schemes is presented on the basis of the hydrodynamic flow characteristics obtained. Using the Reynolds analogy and the semiempirical transport theory of L. G. Loitsyanskii, in particular, the corresponding criterial equations are derived:

$$Nu = 0.0284 Re^{0.75} Pr^{0.5} ,$$

$$Nu = 0.0172 Re^{0.823} Pr^{0.4} .$$

A comparison of these functions (curves 5 and 6 in Fig. 1) with the experimental data of E. G. Vorontsov, I. M. Fedotkin, W. H. MacAdams, and W. Wilke showed that the semiempirical transport theory describes the heat exchange in turbulently flowing films somewhat better than the Reynolds analogy.

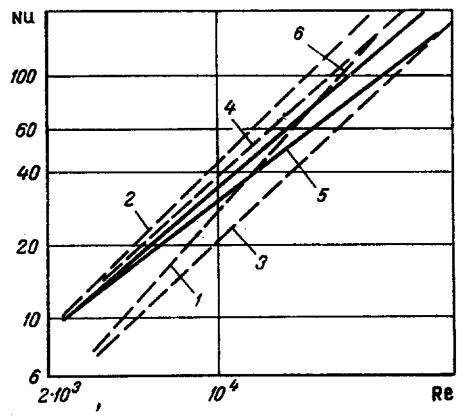

Fig. 1. Comparison of functions Nu = f(Re) at Pr = 1 according to Eqs. (1) and (2) (curves 5 and 6, respectively) with data: 1) of [1]; 2) of [2]; 3) of [3]; 4) of [4].

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Dep. 2295-74, April 17, 1974.
Vinnitsa Polytechnic Institute.
Kiev Technological Institute of the Food Industry.
Original article submitted June 14, 1973.

HEAT EXCHANGE AND FRICTION IN A FLAT TUBE WITH ONE-SIDED HEAT SUPPLY

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UDC 536.244

Turbulent flow of a compressible gas in the initial section of a flat tube with nonsymmetrical heat supply is examined. A system of two-dimensional boundary-layer equations with initial uniform velocity and temperature profiles and linear variation in the heat flux or temperature at one of the walls of the tube is solved by the grid method. The temperature variation in the physical properties of the gas in a direction perpendicular to the flow is taken into account. The data of Reichardt and Goldman are used for the characteristics of turbulent transfer. The system of differential equations is approximated with the help of an implicit six-point scheme. The algebraic equations obtained are solved by the trial run method in conjunction with the iteration method.

The results of the numerical solution show that the nonsymmetrical heat supply affects not only the temperature profile, but also the velocity profile. For example, the velocity maximum is displaced toward the hotter wall. This is caused by a drop in the density near the wall and a corresponding increase in velocity. The Nusselt number with turbulent flow and one-sided heat supply is 10-12% lower than the Nusselt number with symmetrical heat supply, which is explained by the relative growth of the thermal boundary layer. The level of the decrease in the Nusselt number is in good agreement with the results of experimental studies of heat exchange in flat channels. The effect of nonsymmetrical heat supply on the coefficient of resistance is negligible.

The results of numerical calculations with one-sided heat supply and with temperature factors of $T_{wa}/T_g = 3$-4.5 are satisfactorily generalized by known criterial functions obtained for the initial section of a round tube, by B. S. Petukhov's equation, in particular. An equation obtained by the author and which describes the results of numerical calculations with an accuracy of ±6% is presented for calculations with large values of $T_{wa}/T_g = 10$-28.

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Original article submitted July 4, 1973.

HYDRODYNAMIC EFFECTS IN SURFACE HEAT RELEASE FOR DRY METAL—POLYMER FRICTION

V. V. Kharitonov

UDC 539.538.62

The paper deals with the heat propagation for a system of bodies in dynamic contact, with heat released within the surface layer due to the formation and disruption of a certain structure there, the layer having a depth from some microns to 1 mm or more, the loading conditions and materials governing the exact depth. The solution for the nonstationary case gives temperature-distribution curves having a characteristic maximum at a certain distance from the contact surface, which agrees well with experimental evidence.

It is assumed that the contacting materials are very different in thermophysical properties, and also that the properties of the surface layer producing the heat are different from those of the main undeformed material. The friction problem for two bodies is thus reduced to that of heat propagation in a system of three bodies with a distributed bulk source in the middle body, and the properties of this third body are determined from the conditions for numerical agreement between the calculated and measured values.

The solution shows that the surface layer of metal in friction against a polymer should have very much elevated thermal conductivity relative to the base metal, while the heat transfer in it is similar to that in flow of a liquid metal in a narrow slot of width of the order of the thickness of the deformed zone, i.e., the metal behaves more or less as a liquid under these conditions, its hydrodynamic behavior then

defines the heat transport and momentum transfer through the surface layer. This enables one to use the theory of similarity for transport processes.

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Gomel Branch, Belorussian Polytechnic Institute.
Original article submitted October 2, 1972.

THERMODYNAMIC EQUILIBRIUM IN THE Si—Cl—H SYSTEM IN AUTOEPITAXIAL GROWTH OF SILICON FILMS

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UDC 532.696

An analytical discussion is used to examine the effects of convective mass transfer on the equilibrium composition at the surface of the substrate in an epitaxial deposition system.

The phenomenological model completely corresponds to that of [3], but some additional assumptions are made.

The transport equations for the mass of component i are put in the following dimensionless form:

$$u\frac{\partial \tilde{c}_i}{\partial x_i}=\frac{\partial^2 \tilde{c}_i}{\partial r^2}+\frac{1}{r}\frac{\partial \tilde{c}_i}{\partial r},\quad i=2\text{–}6. \tag{1}$$

The boundary conditions and solution are analogous to those for heat transfer [2]. The solution of [1] is used in a flux-balance equation for chlorine at the surface:

$$(\Sigma q_{\mathrm{Cl}_i})_n=0. \tag{2}$$

Algebraic transformations give a transcendental equation for the z parameter [1], which has been solved numerically by computer.

Results show that the equilibrium partial pressures of the components at the surface should be calculated not from the initial mixture composition but from the values defined from the condition of [2], namely, the conservation of the chlorine flux at this surface.

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Original article submitted January 15, 1973.

THERMAL CONDUCTIVITY MEASUREMENT ON HEAVY WATER WITH A RELATIVE NULL METHOD EMPLOYING A TRANSIENTLY HEATED WIRE

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and I. V. Radchenko

UDC 536.22

A method has been described [1] for measuring the thermal-conductivity ratio for two liquids (one of them a standard) with very high accuracy if the difference in thermal conductivity is not very large; this has been applied to the thermal conductivity of D_2O in the range 0-40°C. The theory is presented briefly. The apparatus consists of a bridge containing two cells, which themselves contain the test and standard liquids, the bridge being fed from an audio oscillator, and working into a recording section consisting of a precision amplifier and a loop oscilloscope. Glass-coated microwire is used in the heater and resistance thermometer. The thermal conductivity is defined by

$$\lambda/\lambda_{st} = \gamma/\gamma_0, \tag{1}$$

where λ and λ_{st} are the thermal conductivities of the test and standard liquids, while γ is the ratio of the bridge arms when the bridge is insensitive to resistance change in the cell wires, with γ_0 the same during calibration measurements with the standard liquid in both cells.

A detailed description is given of the method of measuring γ. This quantity is calculated on measurements on the rate of bridge unbalance for certain specified values of γ_i. The accuracy of the method and measurements are evaluated. The heavy water had 99.8 atom % D_2O. The standard liquid was double-distilled ordinary water. The results are represented in the form

$$\lambda_{D_2O}/\lambda_{H_2O} = a + bt + ct^2, \tag{2}$$

where $a = 0.996$; $b = -0.65 \cdot 10^{-3}\ K^{-1}$; $c = 0.13 \cdot 10^{-6}\ K^{-2}$; the coefficient of variation is estimated as about 0.3%. A comparison is made with published values.

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Original article submitted July 9, 1971.

EFFECTS OF TANK GAS STIRRING CONDITIONS ON HEAT TRANSFER TO MELTING ICE

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536.421

Experiments have been performed on the effects of bubbling conditions on the heat transfer in the melting of ice cylinders of diameter 65 mm and height 70-80 mm in water and 20% NaCl solution. In the first series, the air was supplied through holes at the bottom directly under the cylinder at a distance of 120 mm. The heat-transfer coefficient α is related to the air-flow rate Q. Table 1 compares for the bubbling and static states for Q of 40 liter/min.

The figures show that the bubbling has more effect on the heat-transfer coefficient when the latter is small for the liquid at rest; this is particularly so when pure water is compared with the NaCl solution. Bubbling thus eliminates differences in heat-transfer coefficients observed for a liquid at rest. This is to be expected, since the bubbling largely eliminates the hydrodynamic features of the heat transfer in the stagnant state.

In the second series, α was measured for five forms of hole disposition. In the first form, the air was supplied from a circle of the same diameter as the cylinder. In the second and third, the air was

*Deceased.

TABLE 1

Liquid temp., °C	NaCl, wt. %	Heat-transfer factor W/(m²·deg)		Increase in transfer factor produced by gas flow (x)
		gas flow	no flow	
10	0	5000	221	22,6
20	0	5140	358	14,4
20	20	4850	625	7,7
10	20	4740	500	9,5

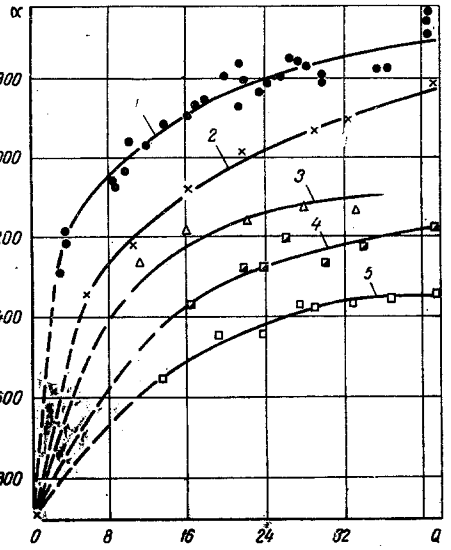

Fig. 1. Effects of air-inlet disposition on the heat-transfer factor α [W/(m^2·deg) for ice melting in water at 20°C. Q in liter/min.

supplied through a ring having an internal diameter the same as the cylinder diameter and outside diameters of 110 or 130 mm. In the fourth, the holes were distributed throughout the cross section of the tank (220 × 225 mm), except for a circle under the cylinder, while in the fifth the holes were placed at the corners of the tank. Figure 1 shows the results. The numbers on the curves correspond to the numbers of the styles. It is clear that the heat transfer is not dependent on the number of holes per unit area, and also the initial speed of the air does not influence the heat transfer.

The values of α are largest when the air is supplied under the bottom of the cylinder. It then flushes the bottom and sides of the cylinder. As the air is distributed over a larger volume, there is a tendency for α to fall; even larger falls occur when the air is supplied at points distant from the melting surface.

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STUDY OF THE CONVECTIVE DRYING OF CORDS

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UDC 66.047:677.4.921.36

In the drying of cords made of chemical filaments impregnated with latex compounds there are two sections of time stabilization of the temperature on the temperature curve $t(\tau)$: near the temperature of a wet thermometer and near the boiling temperature of water. The corresponding critical moisture contents are estimated as $\overline{U}_{1cr} \simeq 0.56\ \overline{U}_0$ (according to A. V. Lykov) and $\overline{U}_{2cr} \simeq 0.08$.

It is shown from a study of the migration of the compound along large specimens that at first the drying occurs from the surface, while after the material reaches a temperature of 100°C the evaporation becomes volumetric. The phase-transition criterion increases sharply from 0 to 1, accordingly.

It is shown that for surface evaporation one can neglect the thermal resistance of the cylinder when $\mathrm{Bi}[\mathrm{Rb}/(\mathrm{Rb}+1)] \leq 0.5$, and when $\mathrm{Bi} \leq 0.5$ for uniformly volumetric evaporation.

A zonal method of calculating the moisture content and temperature of the cords for the entire drying process is given. The drying rate is determined on the basis of a well-known piecewise-linear approximation. The temperature is calculated on the basis of the solution obtained for the basic equation of drying kinetics

$$t = t_{\mathrm{m}} - \frac{C_3}{C_2 - K} \exp(-K\tau) - \left(t_{\mathrm{m}} - t_{\mathrm{s.z}} - \frac{C_3}{C_2 - K}\right) \exp(-C_2\tau).$$

Here Bi and Rb are the Biot and Rebinder numbers; t_m is the temperature of the medium; K is the drying coefficient; $C_2 = \alpha F/M_0 c$; $C_3 = (r/c)N_{s.z}$, where $t_{s.z}$ and $N_{s.z}$ are the temperature of the material and the drying rate at the start of the zones with decreasing velocity.

Functions of the type $Nu = ARe^m$ are given for calculating the heat-transfer coefficients during evaporation and during "pure" heating for longitudinal and transverse blowing of air over the cords.

The time error in the calculation of the $\bar{U}(\tau)$ and $\bar{t}(\tau)$ curves is about 25%.

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METHOD OF ENGINEERING CALCULATION OF HEAT BALANCES FOR DRIERS AND OF FINDING THE MOISTURE LOSS DURING THE DRYING PROCESS

V. A. Kurochkin

UDC 66.047

Extensive use is made of Id diagrams in the practice of calculating drying processes. A new method is proposed here for the engineering calculation of balances for drying installations which does not yield to the Id diagram in the simplicity of the calculations but considerably increases their accuracy and speed. The method was worked out on geometrical concepts concerning the drying process in the Id diagram.

The equation of the method based on Fig. 1 is the following:

$$\operatorname{tg}\theta = \frac{AC}{BC} = \frac{AE + EC}{BC} = \frac{M_g \Delta I}{M_d \Delta d} + \operatorname{tg}\alpha. \quad (1)$$

We take

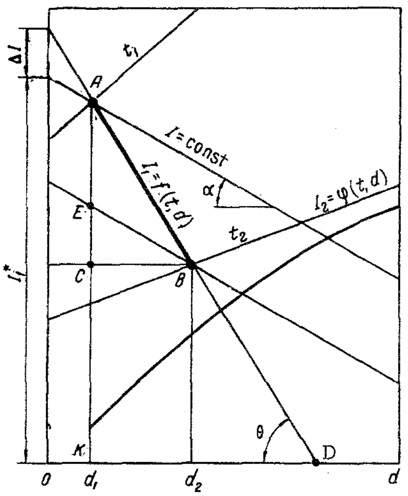
$$\operatorname{tg}\theta = q(u_1 - u_2)^{-1}; \; M_g = M_d = 1; \; \operatorname{tg}\alpha = 0, \quad (2)$$

where $q = c_{dr}(\vartheta_2 - \vartheta_1) + c_{mo}(u_2\vartheta_2 - u_1\vartheta_1) \pm G^{-1}\Sigma Q$ are the components of the heat balance per kilogram of absolutely dry material.

Simple equations for practical application are found with the use of the linear properties of the Id diagram in a constant mass of dry material during the drying process. For example, the variation in moisture saturation of the heat-transfer agent can be found from

$$\Delta d = 1000 c_g^* (t_1 - t_2) R_\theta^{-1}, \quad (3)$$

where $R_\theta = r + c_{va}t_2 + \tan\theta$; $c_g^* = c_g + 0.001 c_{va} d_1$.



From this the specific flow rate of the heat-transfer agent per kilogram of evaporated moisture is

$$l = 1000(\Delta d)^{-1} = R_{\theta}\,[c_g^*(t_1 - t_2)]^{-1}. \tag{4}$$

The differential equation of the method is convenient for the analysis of the drying kinetics. In this case the local $\tan\theta$ is found from

$$\operatorname{tg}\theta = \frac{dq}{du} = (c_{\mathrm{dr}} + c_{\mathrm{mo}}u)\frac{d\vartheta}{du} - c_{\mathrm{mo}}\vartheta \pm \frac{dq'}{du}. \tag{5}$$

On the example of an analysis of the drying process in a tunnel drier it is shown that the intermediate values of the parameters of the heat-transfer agent are represented in the diagram in the form of an S-shaped curve. In an experimental plan the use of the method allows one to determine the moisture loss in any zone of an industrial drier analytically using the differential equation of drying kinetics, which permits the solution of problems of drying technology. In this case the accuracy of the calculations depends only on the care in conducting the experiment and on the methods of solving the differential equation of drying kinetics.

Thus, a method is proposed for the engineering calculation of the heat and material balances for driers which permits the calculation of drying processes with acceptable accuracy and speed. The joint use of the equation of drying kinetics and the proposed method allows one to calculate the course of the drying process and to determine the moisture loss in any zone of the drier.

NOTATION

I, d are the heat content and moisture content of heat-transfer agent in J/kg and kg/kg, respectively;
G is the capacity of drier, kg/h;
$t_1, t_2, \vartheta_1, \vartheta_2$ are the initial and final temperatures of heat-transfer agent and material being dried, °C;
r is the specific heat of evaporation of moisture, J/kg;
$c_g, c_{va}, c_{dr}, c_{mo}$ are the specific heat capacities of dry gases, moisture vapor, dry material, and moisture, respectively, J/kg·deg;
u_1, u_2 are the initial and final moisture contents of material being dried, kg/kg.

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Original article submitted October 9, 1972.

EFFECT OF THE POROSITY OF A SOLID STREAM OF GRANULAR CATALYZERS ON THE FRICTIONAL RESISTANCE

Zh. F. Galimov

UDC 547.315.2.07:66.094.185

The large pressure losses during the pneumatic transport of granular catalyzers in a solid stream are caused by the friction of the granules against the surface of the tube. The dependence of the frictional force on the porosity of the moving stream can be established if one considers a layer of the catalyzer in a vertical tube as a particular case of the state of a granular soil in a closed contour.

According to the well-known Coulomb law in soil mechanics

$$\mathscr{F} = \varphi_e \mathscr{F}_N.$$

The lateral pressure of the layer against the wall of the tube is proportional to the hydrostatic pressure,

$$\mathscr{F}_N = \beta\rho_b H.$$

With allowance for the elementary forces of lateral pressure along the height of the layer the total frictional force is

$$F=\int_0^H \varphi_e\beta\rho_b H\pi D dH,$$

and in calculating the friction per unit surface it is

$$\mathcal{F}=\frac{1}{2}\,\frac{\varphi_e\beta\rho_e\pi DH^2}{\pi DH}=\frac{\varphi_e\beta}{2}\rho_b H.$$

In the general case the force of friction is determined by the ratio of the forces of sliding and rolling friction, the individual coefficients of which differ considerably. The angle of internal friction, which characterizes the mobility of the layer, also has a power-law dependence on the porosity of the layer. The value of the effective coefficient of friction and the hydraulic equivalent can therefore be represented in the form of the equations

$$\varphi_e=a\left(\frac{\varepsilon}{\varepsilon_0}\right)^{n'} \text{and}\ \beta=b\left(\frac{\varepsilon}{\varepsilon_0}\right)^{n''}.$$

Keeping in mind that $\rho_b=\rho_a(i-\varepsilon)$ and taking into account the relationship of the coefficients φ_e and β, one can write

$$\mathcal{F}=\mu H\rho_a(1-\varepsilon)\left(\frac{\varepsilon}{\varepsilon_0}\right)^n.$$

A special method was worked out for the experimental determination of the coefficient μ and the exponent n. The studies showed that the force of friction has an important dependence on the porosity only in the range of from 1 to 1.20 of the relative looseness $\varepsilon/\varepsilon_0$ of the layer. The exponent $n \gg 3$ and depends on the height of the layer of granules.

NOTATION

β	is the hydraulic equivalent of lateral pressure of layer;
ρ_b and ρ_a	are the bulk and apparent densities of the catalyzer material;
H	is the height of catalyzer layer in tube;
φ_e	is the effective coefficient of friction;
D	is the diameter of tube;
ε_0 and ε	are the porosities of layer in the states of rest and motion;
μ	is the modified coefficient of friction.

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Original article submitted May 31, 1973.

COEFFICIENT OF RESISTANCE OF THE CHARGE IN A HOPPER

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UDC 541.182

In the first part of the work the coefficient of resistance of the charge in a hopper is studied experimentally. The experimental points are analyzed in accordance with the equation [1]

$$\xi=\tau\Big/\frac{\gamma V^2}{2g},\ \text{where}\ \tau=\frac{\Delta p}{H}\,\frac{d_c}{4}. \tag{1}$$

Here Δp includes, first, the resistance of the layer of charge, and, second, the resistance caused by the fact that the area of the opening through which the gas escapes from the hopper is less than the cross-sectional area of the hopper.

The experiments were conducted on a flat hopper [2] with a slot opening crossing the entire bottom. Polystyrene in ball form, scattered in narrow fractions [3], was used as the material of the charge. An

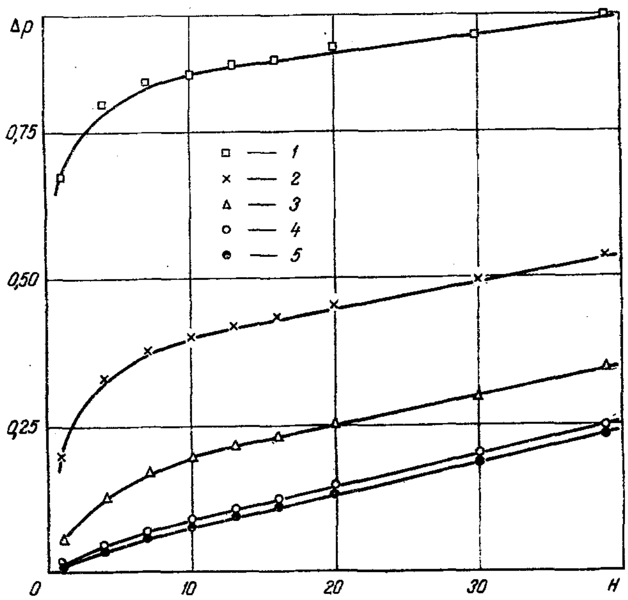

Fig. 1. Pressure curves for the filtering gas in a layer of free-flowing material: d = 3.2 mm; Re = 915; 1) $\bar{f}$ = 17.1; 2) 6.84; 3) 3.42; 4) 1.71; 5) 1.37; Δp, mm; H, cm.

analysis of the experimental data for a charge with a height of $H_* = 0.4$ m led to the dependence

$$\xi = 50/\mathrm{Re} + 0.3 + 7\left(\frac{1}{\delta} - \frac{1}{\delta_f}\right) + 0.146\bar{f}. \tag{2}$$

It was established by the experiments (Fig. 1) that the entire region of variation in the pressure of the gas filtering in the layer of the charge can be divided into two sections, a linear section Δp_l and a nonlinear section Δp_n, and that the height h of the nonlinear section does not depend on $\bar{f}$. It was also noted that Δp_n does not depend on the height of the charge when V = const and H > h.

On the basis of these experimental data and Eq. (2) the function $\xi = \xi(\mathrm{Re}, \bar{f}, \delta)$ was established for a layer of free-flowing material of arbitrary height,

$$\xi = \xi_f\left[1 + T(\bar{f} - 1)\frac{h}{H}\right],$$

$$T = 1 + 48/\delta_f;\ \xi_f = 50/\mathrm{Re} + 0.446;\ h = h_*/\xi_f;\ h_* = 0.0584\,\mathrm{m}. \tag{3}$$

When $\bar{f} = 1$ Eq. (3) is rewritten in the form $\xi = \xi_f$, which corresponds to well-known equations for the coefficient of hydraulic resistance in a layer of free-flowing material. The coefficients entering into the expression for ε_f lie within the field of scatter indicated for them in [4].

The problem of the mechanism of the resistance of a charge consisting of particles of spherical shape is examined in the second part of the work. A model of the flow over a spherical particle which is in a layer of the charge is proposed, making use of the constancy of the average porosity ε of a free-flowing material [5] and the linearity of the dependence of the gas pressure on the height of the layer through which it filters. On the basis of this model equations are written for the determination of the forces of pressure and friction with allowance for the possible separation of the flow from the surface of a particle.

The force of the total resistance acting on a particle in the charge was determined experimentally by the weight method. The experiment was performed on a charge consisting of particles of spherical shape whose number was determined from the volume of the charge, and equivalent diameter d of the balls, and the porosity ε. The result obtained makes it possible to confirm that in the range 10 < Re < 4000 of Reynolds numbers studied the flow over spherical particles in a charge occurs with almost no separation. In this connection one can conclude that the principal element in the mechanism of resistance in layers of ball charges is the resistance of friction.

NOTATION

ξ is the coefficient of resistance of the charge — hopper system;
ξ_f is the coefficient of resistance of a layer of free-flowing material;
ε is the porosity;
d_c is the diameter of an elementary channel in a layer of free-flowing material;
H is the height of layer of free-flowing material;
Δp_n is the excess pressure in the layer at the height $H = h$;
Δp_l is the pressure increment on a linear section as a function of $H - h$;
Re is the Reynolds number determined from diameter of a spherical particle;
$\bar{f}$ is the ratio of cross-sectional area of hopper to area of discharge opening;
δ is the reduced size of discharge opening.

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STATISTICAL METHOD OF DESCRIBING THE PROCESS OF DUST COLLECTION IN A WET DUST COLLECTOR WITH A DISK LIQUID ATOMIZER

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UDC 532.529.5

It is known that the highest dust-collection efficiency is achieved in those devices in which the process of interaction of the drops with the dust is successfully accomplished under conditions where they are comparable in size and the relative velocity of their motion is high [1].

The results of a study of an improved construction for a wet dust collector proposed earlier [2], which has a crossed system of motion of the drops and dust, and a probability model of the process are presented in the present report. The apparatus was tested in the following range of variation of the factors: number of disks of dust collector 1-3; linear velocity of separation of drops from disk 30-60 m/sec; degree of dispersion of dust (median diameter) 10, 13, 18 μ (quartz), 25 μ (dolomite); dust concentration 0.5-5 g/m^3; specific flow rate of water 0.01-0.1 liter/m^3; air temperature 20°C; flow rate of air 1000-4000 m^3/h.

In the study we used the method of statistical modeling [3], in accordance with which the experiment was planned and the mathematical dependence of the dust collection efficiency on the basic factors was found. The data were analyzed on a Mir-1 electronic computer and it was possible to establish that the cleaning efficiency of a wet dust collector with a disk atomizer can be determined from the equation

$$\eta = 1 - \exp\left\{-\frac{0.062\,[21.21 + (x_2 - 0.824)^2]\,[0.56 + (x_3 - 0.874)^{0.064}]\,x_4^{0.087}}{x_1^{0.46}}\right\}.$$

It was established by a correlation analysis that the factors are arranged in the following order with respect to the degree of influence on the cleaning efficiency at the optimum water flow rate: height of the spray jet, degree of dispersion of the drops, degree of dispersion of the dust. The cleaning efficiency in an apparatus with three disks 0.4 m in diameter for quartz dust with a median diameter of 10 and 18 μ was

93.8 and 94.5%, respectively, with a hydraulic resistance not exceeding 450 N/m^2, which enables one to recommend this apparatus for the cleaning of ventilation exhausts.

NOTATION

η	is the degree of cleaning;
x_1	is the relative weight content of fractions of $\leq 10\ \mu$ in the dust studied;
x_2	is the number of disks of atomizer;
x_3	is the relative calculating content of drops with a diameter of $\leq 35\ \mu$;
x_4	is the initial dust concentration.

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AXIALLY SYMMETRICAL TEMPERATURE FIELDS IN LUNAR SOIL

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UDC 536.24.02

The axially symmetrical problem of the thermal conductivity in a semi-infinite medium under conditions imitating the heat exchange on the moon's surface is solved numerically. Zones of non-one-dimensionality of the temperature field as a consequence of thermal insulation of part of the surface are studied. The size of the thermally insulated region and the coefficient of thermal conductivity are varied.

The axially symmetrical problem of the thermal conductivity in a semi-infinite medium being analyzed is described by the equation

$$c\rho \frac{\partial T}{\partial \tau} = \lambda\left(\frac{\partial^2 T}{\partial Z^2} + \frac{1}{R}\frac{\partial T}{\partial R} + \frac{\partial^2 T}{\partial R^2}\right),\ \tau \in [\tau_0,\ \infty),\ Z \in [0,\ \infty),\ R \in (0,\ \infty)$$

and the boundary conditions

$$\left.\frac{\partial T}{\partial R}\right|_{R=0^+} = 0,\ \lambda \frac{\partial T}{\partial Z}(\tau,\ R,\ 0) = \begin{cases} 0, & 0 \leqslant R < L, \\ \varepsilon\sigma T^4 - A_s q_s(\tau), & R \geqslant L. \end{cases}$$

The initial distribution $T(\tau_0,\ R,\ Z)$, assumed to be one-dimensional ($\partial T/\partial R = 0$), is obtained by solving the problem of the establishment of a temperature field which is periodic in time under the effect of the solar radiation $q_S(\tau)$ and the self-radiation of the surface. The solar radiation $q_S(\tau)$ is given by the equation

$$q_s(\tau) = q_s^0 \sin^+(2\pi\tau/\tau_*).$$

The main purpose of solving the problem is to clarify the pattern of spread of the heat flux under the insulated part of the surface ($0 \leq R \leq L$) and to determine the zones in which the solution of the problem can be considered as one-dimensional with the assigned accuracy. The problem was solved numerically by the method of elementary balances (explicit scheme) [1]. In the basic calculating version the thermophysical properties of the lunar soil are taken in accordance with the results of [2]:

$$c = 840\ \mathrm{J/kg{\cdot}deg},\ \rho = 1500\ \mathrm{kg/m^3},\ \lambda = 4\cdot 10^{-6}\ \mathrm{kW/m{\cdot}deg}.$$

Values of λ equal to $4\cdot 10^{-5}$ and $4\cdot 10^{-4}$ were also examined. The parameter L was varied in the range of 1-10 m. The results of the calculations are presented in the form of graphs determining the zone of non-one-dimensionality of the temperature field as a function of the parameters λ, L, and the allowable error.

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Central Aero-Hydrodynamics Institute.
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MOTION OF A ROTATING SPHERICAL PARTICLE IN AN OFF-SETTING FLOW

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UDC 532.582.7:532.135

Motion of a rotating spherical particle through an off-setting flow is considered. The fluid velocity far away from the particle can be represented as follows:

$$\mathbf{V}_\infty = \dot{\gamma} x \mathbf{e}_y + V\mathbf{e}_x, \quad V \gg \dot{\gamma} R, \ \omega R.$$

In a number of articles the solution of this problem has been found in the Stokes approximation, that is, if the interaction between uniform and gradient flows is ignored. In this paper an attempt is made to take into account this interaction in the first approximation with respect to the number $\mathrm{Re} = VR/\nu$ starting with the equations of the Oseen type:

$$\rho V \frac{\partial \mathbf{v}}{\partial x} = -\nabla p + \mu \Delta \mathbf{v}, \quad \nabla \cdot \mathbf{v} = 0,$$

$$\mathbf{v}|_{r=R} = \omega (x\mathbf{e}_y - y\mathbf{e}_x), \quad \mathbf{v}|_{r\to\infty} \to \mathbf{V}_\infty. \tag{1}$$

The solution of the problem (1) is obtained by expanding in powers of the Reynolds number with the terms of the order $O(\mathrm{Re}^2)$ ignored. Using this solution one can compute the force as well as the torque acting on the particle:

$$\mathbf{F} = \mathbf{F}_1 + \mathbf{F}_2, \quad \mathbf{M} = 8\pi\mu(\dot{\gamma}/2 - \omega) R^3 \mathbf{e}_z,$$

$$\mathbf{F}_1 = 6\pi\mu R V\left(1 + \frac{3}{8}\mathrm{Re}\right)\mathbf{e}_x, \quad \mathbf{F}_2 = 2\pi\mu \,\mathrm{Re}\left(\dot{\gamma} - \frac{1}{3}\omega\right) R^2 \mathbf{e}_y.$$

Thus, interaction between uniform and gradient flows results in adding a lateral force F_1 to the resistance force F_2 (in the Oseen approximation) and the torque M. One should mention here that the torque M contains no correction O(Re). This also follows from M being an axial vector and V a polar one.

NOTATION

- v is the velocity;
- p is the pressure;
- ρ is the density;
- μ is the dynamic-viscosity coefficient;
- R is the particle radius;
- ω is the angular velocity of rotating particle;
- γ is the off-setting velocity.

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TRANSFORMATION OF SERIES AND SEQUENCES IN THE THEORY OF HEAT CONDUCTION

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1. Finite and infinite series constitute the most widely used form of solutions of the equations of mathematical physics. In many cases, however, the convergence of the series is slow; the problem which then arises is how to speed up the process of the numerical calculation of solutions.

2. The first approach to this problem consists in transforming the series themselves in order to improve their convergence in some continuous space — time region. In the theory of heat and mass transfer successful use has been made, in particular, of an imaginary transformation of the Jacobi theta function. Using this, it is possible to represent the solutions of some problems in two forms, one of which converges rapidly for small values of the Fourier criterion Fo, and the other converges rapidly for large values of Fo. An effective transformation of the series can be carried out, however, only in individual quasi-one-dimensional problems. Even in the simplest case of a plate with boundary conditions of the third kind, the Jacobi transformation leads to cumbersome formulas.

3. Since the procedures for the transformation of series are not standardized, so that it is impossible to use electronic computers for the purpose, it becomes desirable to improve the convergence by another approach which is not sensitive to the individual properties of the functions and reduces to a limited number of stereotyped operations. This approach is based on the fact that in the numerical realization of the solutions of physical problems we usually consider not a space — time continuum but a multidimensional vector of solutions, i.e., a discrete set of values of the function at the nodes of the coordinate network.

The value of the function at a node is equal to the limit of a sequence of partial sums S_n. Making use of additional information on the manner in which they approach the limit, we can make the convergence considerably stronger. Since the operations are performed on sequences of numbers, not of functions, the form of the general term of the series has no effect at all on the formulation of the computational procedure. It can in fact be standardized and is suitable for both manual and machine calculations.

A powerful and general method for producing such information and accelerating the convergence of numerical sequences is found in Shanks's nonlinear transformations. First-order transformations convert the sequence S_n into T_n according to the formula

$$T_n = e_n(S_n) = \frac{S_n S_{n+2} - S_{n+1}^2}{S_n - 2S_{n+1} - S_{n+2}}.$$

To the sequence T_n, in turn, we can apply the Shanks transformation $p_n = e(T_n) = e^2(S_n)$, making the convergence even stronger, and so on.

4. As an example, we calculate to five significant figures the temperature on the surface $r = R$ and along the axis $r = 0$ of a cylinder subjected to inductive heating, with a boundary condition of the second kind, where the specific power is

$$t(r/R,\ \mathrm{Fo}) \frac{\lambda}{pR} = 2\mathrm{Fo} + \frac{1}{2}(r/R)^2 - f - \left(\frac{1}{4} - \bar{f}\right) + u\left(\frac{r}{R},\ \mathrm{Fo}\right),$$

where f is a given function of the coordinate and the electromagnetic parameter; $\bar{f}$ is the mean value

TABLE 1

n	$u(1;\ 0{,}01)\ \lambda/pR$			$u(0;\ 0{,}01)\ \lambda/pR$		
	$-S_n$	$-e(S_n)$	$-e^2(S_n)$	S_n	$e(S_n)$	$e^2(S_n)$
1	0,0654659			0,162526		
2	0,0699156	0,0709195		0,147698	0,14977	
3	0,0705167	0,0706377	0,0706376	0,150106	0,14972	0,14972
4	0,0706174	0,0706316	0,0706377	0,149645	0,14972	0,14972
5	0,07063421	0,0706374		0,149730	0,14972	
6	0,0706369			0,149715		

obtained by averaging over a cross section; λ is the thermal conductivity; μ_i are eigenvalues;

$$u = \sum_{i=1}^{\infty} A_i J\left(\mu_i \frac{r}{R}\right) \exp(\mu_i \mathrm{Fo}).$$

For Fo = 0.01 and r = 0, obtaining a solution with five accurate figures would require the calculation of 10 partial sums, i.e., the solution of 10 transcendental characteristic equations, followed by the calculation of the values of the eigenfunctions, exponential factors, and coefficients A_i. The use of Shanks transformations, each of which requires only eight arithmetic operations, made it possible to obtain the answers by calculating only five or six partial sums. A very significant reduction in the amount of calculation work is achieved by the transformation of numerical sequences in the solution of problems with inhomogeneous media in two or three dimensions.

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CALCULATION OF TEMPERATURE DISTRIBUTIONS IN THIN-WALLED CYLINDRICAL STRUCTURES

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UDC 536.2

Unconditionally stable difference methods for solving one- and two-dimensional heat-conduction problems require the solution of large numbers of linear algebraic equations. In the one-dimensional case when the systems have a diagonal form the most satisfactory method of solving them is the pivotal method. In determining two-dimensional temperature distributions in the cross sections of thin-walled cylindrical structures we obtain a set of one-dimensional problems connected with one another at isolated points [1, 2]. We solve the system of algebraic equations obtained by a modified pivotal method, taking account of the special form of the matrices.

We introduce two types of rod systems: systems which do not contain closed contours, and systems which do. We solve problems of the first type by an algorithm which reduces to a counter pivot method [3] for two-rod systems. A problem of the second kind is solved by the method of sections, introducing unknown temperatures or heat fluxes into the sections. Their values are chosen so as to satisfy the continuity of temperature and heat flux across a section. An algorithm is proposed which takes account of contact resistances at rod junctions.

As an example we determine the temperature distribution in the reinforced cladding of an aircraft, taking account of external radiation and contact resistance.

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TEMPERATURE DISTRIBUTION ALONG A FINITE RADIATING ROD

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An approximate analytic expression is obtained for calculating the steady-state temperature distribution along a finite thin rod radiating into a medium with a temperature of absolute zero in the absence of internal heat sources and incident radiation for a specified temperature T_0 at one end of the rod and no heat transfer at the other.

The steady-state heat-conduction equation in dimensionless parameters for a finite rod has the form

$$\frac{d^2\theta}{dX^2} = N\theta^4. \tag{1}$$

The temperature distribution along the rod can be obtained by numerical integration [1, 2], and for values from 0 to 10 can be calculated by an approximate formula recommended in [3].

Equation (1) can be used to find the temperature distribution along a semibounded rod by expressing the running coordinate x in terms of the length of the finite rod ($X = x/l$). An analytic solution of this problem was derived in [2], and in the standard notation the temperature of a semibounded rod at the cross section $x = l$ is

$$\theta_{ls} = \frac{1}{\left(1+\sqrt{\frac{9}{10}N}\right)^{2/3}}. \tag{2}$$

A comparison of the temperature at the end of a finite rod θ_l obtained by numerical integration with the temperature of a semibounded rod at the section $x = l$ for the same values of N shows that for $N \geq 20$ the ratio θ_l/θ_{ls} can be considered constant and equal to 1.231 to four significant figures, and for the whole range considered $N = 0\text{–}10^{21}$ the temperature at the end of a finite rod is given by the relation

$$\theta_l = \frac{1.231}{\left(1+\sqrt{\frac{9}{10}N+0.1338}\right)^{2/3}}. \tag{3}$$

For any cross section x along the rod the thermal-conductivity parameter of the part of the rod in the segment $l - x$ is given by $N(1-x)^2\theta^3$, and by analogy with (3) can be written as

$$\frac{\theta_l}{\theta} = \frac{1.231}{\left(1+\sqrt{\frac{9}{10}N(1-X)^2\theta^3+0.1338}\right)^{2/3}}. \tag{4}$$

Hence, by taking account of (3) we obtain an equation for the temperature distribution along a finite rod for practically all values of the parameter N encountered in engineering practice:

$$\theta = \left[\frac{1+\sqrt{\frac{9}{10}N+0.1338}+0.3658\sqrt{\left(1+\sqrt{\frac{9}{10}N+0.1338}\right)^2+5.826N(1-X)^2}}{\left(1+\sqrt{\frac{9}{10}N+0.1338}\right)^2-\frac{9}{10}N(1-X)^2}\right]^{2/3}. \tag{5}$$

The difference between the analytic relations (3) and (5) and the results of the numerical integration presented in the paper is no more than 2%.

NOTATION

T	is the temperature;
σ	is the Stefan–Boltzmann constant;
ε	is the emissivity;
λ	is the thermal conductivity;
u	is the perimeter of the rod;
f	is the cross section of the rod;
x	is the coordinate along the rod;
l	is the length of the rod;
$\theta = T/T_0$	is the dimensionless temperature;
$X = x/l$	is the dimensionless length.

Subscripts:

0	in reference to parameters at the beginning of the rod;
l	in reference to parameters at the end of the rod;
s	in reference to parameters of a semibounded rod;
$N = \sigma\varepsilon u T_0^3 l^2/\lambda f$	is a dimensionless thermal-conductivity parameter.

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APPROXIMATE SOLUTION OF THE ONE-DIMENSIONAL NONLINEAR HEAT-CONDUCTION EQUATION

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UDC 536.2.023

An approximate solution of the heat-conduction equation which takes account of the temperature dependence of the diffusivity is presented. Certain of the solutions obtained are compared with exact solutions of nonlinear equations given in [1]. The problem is solved by applying the comparison theorem [2] in the following way. The constant minimum and maximum temperatures, for example, for a semibounded region the initial and then the limiting value, are substituted in succession into the equation. In each case this yields an equation with constant coefficients which is readily solved. On the basis of the comparison theorem and from physical considerations it can be expected that the curve sought for the temperature distribution along x described by the initial nonlinear equation will lie between the curves for the temperature distributions found for the two extreme values of the diffusivity. Calculations show that if the approximate value of the temperature is taken as the geometric (and sometimes the arithmetic) mean of the two temperatures found, the values obtained are in good agreement with the exact values except for very small t or very large x. Problems involving various forms of the temperature dependence of the diffusivity can be solved in this way. In addition, the method can be used to solve the nonlinear heat-conduction equation taking account of convective heat transfer.

The solutions of the problems indicated are obtained for both semiinfinite and finite regions. It is shown that in many cases the Galerkin method also gives good results.

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