A RAPID THERMAL METHOD OF WATER CONTENT DETERMINATION

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UDC 53.093.083

A structure is given for a primary transducer for water content determination of granular materials; a thermal technique is used, with the idealized scheme as follows.

Consider a layer of moist granular material of thickness R and area S in contact with a thin metal plate (heater); the system is adiabatically isolated. At the initial instant $\tau = 0$, the plate begins to be heated by a current of constant power W. The temperature $T_1(\tau)$ as a function of time is to be derived subject to the following assumptions: the temperature T_1 may be considered as localized by virtue of the small thickness of the plate and the high thermal conductivity, while the temperature T of the material (neglecting edge effects) may be considered as dependent only on the x coordinate $(0 \leq x \leq R)$, which is perpendicular to the surface of the plate, together with the time $T = T(x, \tau)$, with diffusion and evaporation neglected.

The solution in dimensionless form is as follows:

$$\Theta(\mathrm{Fo}) = \frac{\mathrm{Po}'}{(1+\varepsilon)^2} + \frac{\mathrm{Po}'\mathrm{Fo}}{1+\varepsilon} - \sum_{i=1}^{\infty} \frac{2\mathrm{Po}'}{\mu_i^2(\varepsilon + \sec \mu_i^2)} \exp(-\mu_i^2 \mathrm{Fo}), \tag{1}$$

where $\Theta = [T_1(\tau) - T_0]/T_0$ is the relative excess temperature of the plate, $\varepsilon = C_p/C_M$ is the ratio of the total thermal capacities of the plate and material, $\mathrm{Po}' = WR/\lambda T_0 S$, λ is the thermal conductivity of the material, μ_i are the roots of the equation $\varepsilon \mu_i + \tan \mu_i = 0$, and Fo is taken for the material.

Equation (1) becomes asymptotically linear (for large Fo):

$$\Theta_{as}(\mathrm{Fo}) = \frac{\mathrm{Po}'}{(1+\varepsilon)^2} + \frac{\mathrm{Po}'\mathrm{Fo}}{1+\varepsilon}. \tag{2}$$

An important point is that when (2) is written in terms of real time, $T_{1\,as}(\tau) = b + k\tau$ is dependent only on the specific heat C_M of the material, which itself has a simple relation to the water content. Therefore, the slope of the asymptote to the $k = W/(C_M + C_p)$ curve is a parameter providing information on the water content. It is shown that the relative error $\Delta = (\Theta(\mathrm{Fo}) - \Theta_{as}(\mathrm{Fo}))/\Theta(\mathrm{Fo})$ becomes less than 1% for $\mathrm{Fo} \geq 1/2$, which corresponds to $\tau = 9$ sec for the lower bound to the thermal diffusivities of moist granular materials and the value $R = 5 \cdot 10^{-3}$ m usual in experiments, which shows that the method is a rapid one. Experimental results are also reported.

It is found that these agree well with the calculations; however, as is usual in water-content measurement, one should use an experimental calibration curve that relates the plate heating rate to the water content for each particular material.

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DRYING OF A POWDER IN A UHF FIELD

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UDC 66.047.354

The cavities of UHF ovens have the electromagnetic energy completely absorbed by the working material; however, such ovens have the disadvantage that the heating is unevenly distributed, since the material is in a standing wave. Local overheating can be avoided with granular materials by fluidizing the latter or by vibration. Both techniques are very effective in equalizing the heating, but the latter is better combined with UHF heating. In fact, the heat release per unit volume in UHF treatment is proportional to the water content in the volume accessible to the electric field, i.e., the rate of heat production for a given water content is proportional to the bed density, and the latter varies with the frequency of the vibrator when the bed is fluidized by vibration, while remaining on average higher than the density of a fluidized bed of the same porosity. This is the particular advantage of vibrational fluidization in UHF heating.

The uneven distribution of the UHF energy in the cavity was determined calorimetrically from the heating of 1 cm^3 of distilled water, which was placed in a glass vessel with double walls, the air being removed from between them. The water temperature was measured with a copper–Constantan thermocouple directly after switching off the UHF source. The heating time was controlled by a time switch. The maximum deviation of the energy density from the mean was 50-60%. No irregularities in the field distribution over the cavity were observed.

The drying tests were done with a typical porous material, namely, MSM silica gel. Pulsed drying was used, with the water content measured by weighing. The vibrational system had the following parameters: amplitude, 1 mm; frequency, 100 Hz. The temperature distribution in the material was monitored with a set of three thermocouples at three different positions. The UHF heating with the bed immobile produced a maximum deviation of 20-25°C from the mean temperature of 65°C. Vibrational fluidization reduced the temperature variations throughout the volume to 1-2°C with the same energy input, so vibration minimizes the temperature differences in UHF drying.

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PHYSICAL SIMULATION OF FREEZING IN SAND-LOADED WAGONS

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UDC 624.138.35

Results are presented on the freezing of sand in railroad wagons of the commoner type (4-axle 63-ton wagons). The results have been obtained by physical simulation of thermal processes in granular media [1]. The dimensions of the models and natural objects were in the ratio 1 : 10. The boundary temperatures were produced by a multichannel temperature regulator [3]. The mode of freezing was determined from the temperature distribution [2]. The transducers were 48 copper–Constantan thermocouples disposed at various points within the sand.

The measurements were made with constant values for the initial water content and temperature (5, 7, or 10% water and 2°C) and with constant boundary conditions of the first kind (−10, −20, −30, or −40°C). The sands were taken from three quarries in Western Siberia and Central Yakutia.

The results indicate the mode of motion of the phase-transition front, the temperature distribution in the frozen zone with respect to the central axis of symmetry, and the time for total freezing τ_t. The most important conclusions are as follows.

The depth of freezing along the central vertical axis Z may be determined along with τ_t by solving Stefan's problem for a line and a semiinfinite medium. The discrepancy between the analytical calculation and the measurements did not exceed 15%.

The freezing rates along the central symmetry axes at first fall, but then rise again on account of the overall cooling effects of all the surfaces. The rise in the rate begins at $0.9\,\tau_t$ on the vertical central symmetry axis.

The temperature distribution over each of the central symmetry axes is nearly linear; the linearity persists on the z axis throughout the freezing, while on the y axis it persists up to $0.8-0.9\,\tau_t$, and on the x axis up to $0.6-0.7\tau_t$. The latter points are very important, since they provide a basis for solving problems of this type by approximate analytical methods.

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Original article submitted December 19, 1974.

CHARACTERISTICS OF TOTAL-PRESSURE AND STATIC-PRESSURE PROBES FOR M = 4.5 IN WET STEAM AT PRESSURES BELOW THE TRIPLE-POINT VALUE

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UDC 53.082.32

Some results are presented from measurements on total-pressure and setting-pressure probes used in a two-phase supersonic flow of water vapor at low pressures where the condensed phase is in the solid state.

Ice grows on the probes under these conditions; special measures are needed to prevent the probes from becoming blocked. In particular, the ice may be removed from a probe by displacing it to a region of relatively high temperature and pressure.

Figure 1 shows the effects of the length of the cylindrical part of the probe and the cone angle of the end on the static-pressure reading. The errors are large for $\bar{l} = 5$, but the effects of $\bar{l}$ and of φ decrease as $\bar{l}$ increases. At $\bar{l} = 15$, probes with substantially different tips indicate the same pressure. The shape of the curves indicates that any further increase in $\bar{l}$ should not result in any substantial change in reading.

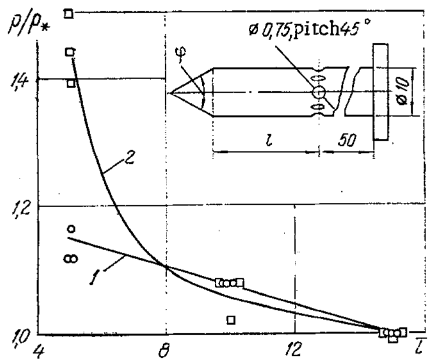

Fig. 1. Reading of static-pressure probe in relation to length $\bar{l}$ for $p_0 = 196{,}000$ Pa, $T_0 = 433$°K, M = 4.5, $p_* = 123.7$ Pa: 1) $\varphi = 5°$; 2) $\varphi = 80°$. Probe dimensions in mm.

The static-pressure probe (with parameters $\bar{l}=5$, $\varphi=20°$) and the total-pressure probe (outside diameter, 4 mm; internal diameter, 3 mm) were used in sensitivity tests, with the angle of inclination varying from 0 to 7° at $p_0=196{,}000$ Pa, $T_0=445$°K, M=4.3; within that range of angles, the readings of the total-pressure probe varied by not more than 2.5%. The static-pressure probe gives a reading independent of angle up to 5°.

NOTATION

l, d, $\bar{l}=l/d$, length, diameter, and relative length of cylindrical part of probe; φ, cone angle of probe; M, Mach number; p, static pressure indicated by probes; p_*, static pressure indicated by probes of length $\bar{l}=15$; p_0, T_0, pressure and temperature at the inlet to the nozzle.

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ACCELERATED HEAT TRANSFER IN LOW-PRESSURE BOILING

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UDC 536.423.1

Heat transfer may be accelerated and boiling stabilized at low pressures by depositing a layer of porous nickel about 0.5 mm thick having a pore diameter of 8-10 μm on the heat-transfer surface (a copper disk of diameter 30 mm). The method has been shown to be effective for liquids boiling at atmospheric pressure and above; the heat-transfer coefficient is increased by a factor 2-4 [1, 2], and the first critical heat-flux density is increased by a factor 2-3 [1].

To provide for comparability of the results for any temperature difference ΔT, complete curves were recorded for bubble boiling of n-octane at pressures of 0.014 bar (Fig. 1) and 0.037 bar for smooth and polished surfaces.

The boiling on a polished surface was unstable and explosive with considerable superheating; in the region of unstable bubble boiling, the $q=f(\Delta T)$ relation had a vertical section. The difference between the critical heat-flux densities q_{max} for the smooth surface at the above pressures did not exceed the error of experiment.

The porous surface had stable vaporization centers; these not only stabilized the process, but also reduced ΔT, while the heat-transfer coefficient was increased by almost a factor 10 for certain ΔT. At 0.014 bar, the porous coating reduced q_{max} by a factor 2, while at 0.037 bar the value was almost unaltered. The latter probably indicates that the pore size needs to be selected to suit each particular pressure and liquid.

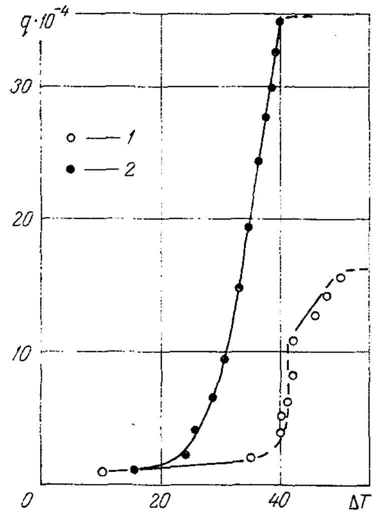

Fig. 1. Heat flux density $q(\cdot 10^{-4}$ W/m$^2)$ as a function of temperature difference ΔT (°C) at a pressure of 0.014 bar for: 1) polished surface; 2) porous surface.

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Original article submitted June 25, 1975.

DEFINITION OF THE BEST MODE OF COOLING IN CONTINUOUS CASTING FROM A MATHEMATICAL MODEL

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UDC 669.18-412:612.746.6

The heat-transfer rate in the secondary-cooling zone is important in producing high-grade continuous castings; the modes of cooling in this zone should produce a uniform temperature distribution around the perimeter of the solidified crust and monotonic temperature reduction over the height of the casting to a level not less than 800-900°C (in order to avoid loss of plasticity).

These requirements have been incorporated in calculations on the economical use of water in this zone for a continuous casting of cross section 0.24×1.71 m^2 for steel type 17G2SF, for which purpose a solidification model similar to that previously described was used.

An M-222 computer was used with a finite-difference method for one quarter of the cross section (by virtue of the symmetry) to solve the two-dimensional differential equation for nonstationary thermal conduction, which incorporated the latent heat of crystallization released in the liquidus–solidus temperature range.

When the casting emerges from the crystallizer, there is an uneven temperature distribution around the perimeter; the subsequent cooling in the secondary zone is considered for boundary conditions of the first kind in the form

$$t_{i,j} = \frac{t^0_{i,j}}{(h - h_c + 1)^{n_{i,j}}},$$

where

$$n_{i,j} = \frac{\ln \dfrac{t^0_{i,j}}{800}}{\ln (H - h_c + 1)};$$

with h the current height of the casting as reckoned from the meniscus in the crystallizer; h_c and H are the height of the casting in the crystallizer and at the end of the secondary zone, respectively; j = 0 and i = 1, 2,..., 59 for the wide face, and i = 0 and j = 1, 2,..., 10 for the narrow face.

This boundary condition amounts to meeting the specification that the nonuniformity in the temperature distribution around the perimeter should be reduced while simultaneously providing a monotonic temperature reduction over the height of the casting to the 800°C level at the end of the secondary-cooling zone.

The program provides for printing out the temperature distribution over the cross section by steps along the height, as well as the heat-flux densities at the perimeter, the local heat-transfer coefficients, the mean integral heat-transfer coefficients over the surfaces of the wide and narrow faces for each of the secondary-cooling sections, and the specific and total water consumption rates in the section. The relationship between the mean heat-transfer coefficient and specific water consumption is related to the form of cooling (nozzles ranged along a rod or in a circular pipe), the values being taken from published data. The calculated flow rates are compared with those measured under industrial conditions. Recommendations are made on the economical use of water in such sections.

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CHARGE DISTRIBUTION TRANSVERSE TO THE FLOW DIRECTION IN A TURBULENT LIQUID

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UDC 66.015.23

Mass transfer is considered for a turbulent flow of a charged liquid along an infinite plane y = 0, which has charge sources; the equation of continuity for the diffusion current is used with Poisson's equation for the field set up by the charges in the liquid to derive an equation for the steady state that incorporates the turbulence and the conductivity of the liquid, this giving the charge distribution perpendicular to the flow:

$$\frac{d}{dy}\left(\hat{D}\frac{dg}{dy}\right)=\frac{g}{\tau}.$$

Here $\hat{D}=D+D_{tur}$ (D is the molecular diffusion coefficient, while D_{tur} is the turbulent diffusion coefficient), g is the charge concentration in the liquid, y is the coordinate in the direction perpendicular to the flow; $\tau=\varepsilon\varepsilon_0/\sigma$, where ε is the relative dielectric constant of the liquid, ε_0 is the dielectric constant, and σ is the specific conductivity of the liquid.

This equation may be solved by regions; the entire region transverse to the flow is divided into three zones: 1) $0 \le y \le \delta$; 2) $\delta \le y \le \delta_0$; 3) $\delta_0 \le y < \infty$, where δ is the thickness of the diffusion sublayer and δ_0 is the thickness of the Prandtl viscous layer.

The molecular-diffusion coefficient in the first zone exceeds the turbulent-diffusion coefficient there, and the latter may be neglected. In the second and third zones, one can neglect D by comparison with D_{tur}. The equation has an exact solution for each of the three zones. Solutions are linked up at the zone boundaries. The charge distribution is dependent on the conductivity of the liquid and on the hydrodynamic characteristics of the flow. The following two limiting cases are considered.

1) $\delta \ll (D\tau)^{1/2}$, namely, low specific conductivity and strong turbulence. This case is characteristic of a charged insulator flowing in a tube during electrification.

2) $\delta \gg (D\tau)^{1/2}$, namely, high conductivity and weak turbulence, which is characteristic of a charged electrolyte.

This charge distribution has been used to determine the diffusion current to the wall and the thickness of the Nernst diffusion layer δ_N. In the first limiting case, with $\delta \ll (D\tau)^{1/2}$, it is found that $\delta_N = (4/3)\delta$, and this result agrees exactly with that obtained previously for mass transport in a neutral liquid in turbulent flow. In the second limiting case, with $\delta \gg (D\tau)^{1/2}$, the result is $\delta_N = (D\tau)^{1/2}$, which agrees with the result for an electrolyte at rest with respect to a metallic plane. These formulas allow one to calculate the mass transport in turbulent flow of a charged liquid, and they are also suitable for calculating electrification currents for liquid insulators during transport in tubes.

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Original article submitted April 22, 1974.

TEMPERATURE DISTRIBUTION PRODUCED BY AN ANNULAR SOURCE AND TEMPERATURE

M. A. Galakhov and S. V. Meshkov

UDC 536.24.01

1. Consider a thermally conducting space in a cylindrical coordinate system (x, r, φ); in the space there is a cavity consisting of an unbounded circular cylinder of radius R having its axis of rotation coincident with the x axis. Consider unit heat flux from the cavity to the space. The flux is localized in an infinitely narrow ring x = 0. The boundary condition is

$$2\pi Rk\frac{\partial T}{\partial r}=-\delta(x),$$

where k is the thermal conductivity and $\delta(x)$ is a δ function. The Green's function takes the form

$$G(r,\ x,\ \xi)=\frac{1}{2\pi^2Rk}\int\limits_0^\infty\frac{K_0(\lambda r)\cos[(x-\xi)\lambda]}{\lambda K_1(\lambda R)}d\lambda, \tag{1}$$

where K_0 and K_1 are Bessel functions of imaginary argument.

2. The heat flux in a ball bearing is usually discussed in terms of the average with respect to the angle φ [1, 2]. The distribution of the flux in x is considered uniform. The exact solution for this case is

$$T(R,\ x)=\frac{Q}{4\pi^2ka}\left[J\left(\frac{x+a}{R}\right)-J\left(\frac{x-a}{R}\right)\right]. \tag{2}$$

The values of $J(\alpha)$ for $0<\alpha<0{,}1$ are as follows:

α	0,01	0,02	0,03	0,04	0,05	0,06	0,07	0,08	0,09
$J(\alpha)$	0,06	0,11	0,15	0,19	0,23	0,26	0,30	0,33	0,36

If $T(R,\infty)=T_\infty\neq 0$, then T_∞ must be added to (2); unfortunately, T(R, x) decreases only slowly as x increases, so one can reasonably introduce the temperature T_1 measured at point x_1. Then we have

$$T_{\max}=T_1+\frac{Q}{4\pi^2ka}\left\{2J\left(\frac{a}{R}\right)-\left[J\left(\frac{x_1+a}{R}\right)-J\left(\frac{x_1-a}{R}\right)\right]\right\}.$$

The correction to a cruder model for the two-dimensional case is typically 20%.

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Dep. 3784-75, November 10, 1975.
Original article submitted February 20, 1975.

TEMPERATURE DISTRIBUTION IN A HOLLOW CYLINDER DUE TO A SURFACE RING SOURCE OF FINITE WIDTH

I. A. Zernov and V. F. Martyushov UDC 536.21

Temperature distributions have been calculated for solid and hollow unbounded cylinders with ring heat sources of finite width and uniformly distributed output. Newton's law applies to the heat transfer at the outer surface. The thermophysical parameters are taken as constant. The exact solution to the initial equation has been derived by Laplace transformation in conjunction with finite Hankel transformations and by varying the constants. The general solution yields the following solution for the quasistationary case:

$$(\theta_L)_{z\leqslant 0} = \sum_{i=1}^{\infty} \varphi(r) \frac{\exp\left[\left(-\frac{\mathrm{Pe}}{2}+k\right)z\right]-\exp\left[\left(-\frac{\mathrm{Pe}}{2}+k\right)(z-l)\right]}{-\frac{\mathrm{Pe}}{2}k+k^2};$$

$$(\theta_L)_{0\leqslant z\leqslant l} = \sum_{i=1}^{\infty} \varphi(r)\left\{\frac{2}{\mu_i^2}-\frac{\exp\left[\left(-\frac{\mathrm{Pe}}{2}-k\right)z\right]}{\frac{\mathrm{Pe}}{2}k+k^2}-\frac{\exp\left[\left(-\frac{\mathrm{Pe}}{2}+k\right)(z-l)\right]}{-\frac{\mathrm{Pe}}{2}k+k^2}\right\};$$

$$(\theta_L)_{z\geqslant l} = \sum_{i=1}^{\infty} \varphi(r) \frac{\exp\left[\left(-\frac{\mathrm{Pe}}{2}-k\right)(z-l)\right]-\exp\left[\left(-\frac{\mathrm{Pe}}{2}-k\right)z\right]}{\frac{\mathrm{Pe}}{2}k+k^2},$$

where θ_L is the dimensionless temperature, $\varphi(r)$ is the radial temperature distribution, $k=\sqrt{(\mathrm{Pe}/2)^2+\mu_i^2}$; $\mathrm{Pe}=vR_0/a$; $z=z_1/R_0$; $l=L/R_0$; and L is the source width.

The maximum temperature in the quasistationary case is found in the heater zone $0\leq z\leq l$ and is

$$(\theta_L)_{\max} = \sum_{i=1}^{\infty} \varphi(r)\frac{2}{\mu_i^2}\left(1-\exp\frac{-\mu_i^2 l}{2k}\right).$$

Formulas are given for the coordinates of the peak temperature, and also ones for calculating the temperature distribution in stationary heating. The calculated temperatures agree satisfactorily with measurements. The comparison is made graphically.

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Original article submitted August 12, 1974.

HEAT TRANSFER WITH PERIODIC VARIATIONS IN HEAT-TRANSFER COEFFICIENT AND ENVIRONMENTAL TEMPERATURE FOR A SEMIINFINITE BODY

A. G. Gindoyan and M. A. Pak UDC 536.21

Heat transfer is considered for a semiinfinite solid and a medium at temperature T(t), the heat-transfer coefficient being $\alpha(t)$. The process is described by a one-dimensional boundary-value problem of the third kind:

$$\frac{\partial U}{\partial t} = a\cdot\frac{\partial^2 U}{\partial x^2},$$

$$\lambda\left.\frac{\partial U}{\partial x}\right|_{x=0} + \alpha(t)\left[T(t)-U|_{x=0}\right] = 0.$$

It is assumed that the temperature of the medium and the heat-transfer coefficient vary periodically with the same period, both being piecewise-smooth functions of time.

The steady-state solution is derived, which is independent of the initial conditions; the method of solution is as follows. Laplace transformation yields an integral equation, and the asymptotic solution is derived as a Fourier series.

The basic technique is asymptotic estimation of the contour integrals. It is found that the steady-state temperature distribution can be found as the sum of a Fourier series. The Fourier coefficients for this distribution constitute an infinite system of linear algebraic equations:

$$Y_\mu \sqrt{\frac{|\mu|\omega}{a}} \exp\left\{i\frac{\pi}{4}\operatorname{sgn}\mu\right\} + \sum_{\nu=-\infty}^{+\infty} K_{\mu-\nu} Y_\nu = \varphi_\mu,$$

$$-\infty < \mu < +\infty,$$

where μ is the number of the equation, Y_ν is the ν-th Fourier coefficient in the temperature distribution for $\exp\{i\nu\omega t\}$, ω is the frequency of the temperature variation in the medium, and K_ν and φ_ν are the ν-th Fourier coefficients for $\alpha(t)/\lambda$ and $\alpha(t)T(t)/\lambda$, respectively.

If the heat-transfer coefficient is constant, the infinite system of algebraic equations splits up and gives a known solution for this case [1]. In general, the infinite system must be solved approximately.

An example is considered where $\alpha(t)$ and $T(t)$ are step functions, which take two values and have common points of discontinuity. The physical criteria that control the temperature distribution are derived, and also the functional relationship for the minimum, maximum, and integral means over a period in terms of these criteria.

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Original article submitted May 4, 1975.

A NEW METHOD OF CALCULATING NONSTATIONARY THERMAL CONDUCTION IN BODIES OF COMPLICATED SHAPE

V. D. Polugaevskii

UDC 536.21

Nonstationary thermal conduction in real two-dimensional or three-dimensional bodies of complex shape may be calculated by replacing the latter by bodies of simple shape, such as rectangles having the same area and perimeter as the two-dimensional body, or a finite cylinder having the same volume and surface area as the real body.

This replacement also extends considerably the range of useful application of the theory of regular thermal conditions: measurement of the form factor for a complicated body is replaced by the use of simple analytical expressions for the form factor for a rectangle or finite cylinder.

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Original article submitted August 5, 1975.

NONSTATIONARY HEAT TRANSFER BETWEEN AN UNDERGROUND PIPELINE AND THE ENVIRONMENT

B. L. Krivoshein, V. P. Radchenko, and V. M. Azapkin

UDC 536.242:681.3

General-purpose pipelines may be subject to almost instantaneous temperature change in the fluid; the durations of the corresponding thermal transients and the fluctuations in the heat flux from the tube to the soil may be determined by considering a step change in the temperature.

The equation of nonstationary thermal conduction has been integrated numerically for a semiinfinite body of soil around a pipeline subject to boundary conditions of the third kind; exact determination of the heat flux from the pipeline to the soil requires conformal transformation of the region to a rectangle by means of a bipolar coordinate system [1].

A finite-difference approximation over the internal nodes of a square grid have been used to integrate the equation of thermal conduction for the soil in the new region; the system of finite-difference equations has been solved by Seidel's iteration method with a BÉSM-6 computer.

Results are presented on the temperature distribution in the soil, the heat flux from pipeline to the environment, and the effect of heat transfer at the soil–atmosphere boundary.

The calculations indicate measures for ensuring reliable operation when pipeline heaters are switched off, or gas coolers are tripped during emergency change in the degree of compression, together with a means of measuring gas temperature during gas pipeline operation.

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Dep. 331-76, December 22, 1975.
Original article submitted April 1, 1975.