ABSTRACTS

EFFECTS OF VIBRATION ON SOIL FREEZING AROUND A BOREHOLE

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UDC 536.421.4

The freezing of soil by an oscillating flow of cold gas is considered; it is found that an oscillatory gas flow increases the heat-transfer coefficient to the borehole wall. A study has been made of the effects of the heat-transfer coefficient on the movement of the freezing boundary. One-dimensional equations are used for the heat flux in the frozen and unfrozen parts of the soil. Stefan's condition is used at the freezing boundary. The solution is found by successive approximation. In the semiinfinite unfrozen region, the solution may be derived by introducing a thermal-influence radius, beyond which there are no thermal perturbations, and at which the temperature and heat flux are continuous.

A system of two ordinary differential equations is used to determine the position of the freezing boundary and the above radius as functions of time. A Nairi computer has been used to solve the system numerically via a standard program for specified values of the parameters. The results are presented for various values of the heat-transfer coefficient and show that there is a substantial increase in the rate of advance of the boundary as the heat-transfer increases.

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HEAT TRANSFER FOR A LIQUID BOILING IN A BED OF GRANULAR MATERIAL

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An experimental study is reported for heat transfer and critical heat loadings for a liquid boiling in a layer of granular material.

The heating was provided by a wire, while the granular material was quartz sand, the heat carrier being at atmospheric pressure 5°C below the boiling point. The measurements were made with a deep layer of water and capillary water influx.

It is found that three different heat-transfer mechanisms are involved in the boiling in such beds, and each involves a critical heat loading:

1) transport by a process equivalent to thermal conduction;

2) transport in a fluidized mode;

3) transport with channeling.

The second and third conditions can be realized in accordance with the depth of the wire in the bed.

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The established hydrodynamic theory of boiling crises can be modified to give the following relationship for the wire:

$$\left(\frac{q_{cr}}{r\rho''}\right)^2 \rho'' > \left[\sqrt{g\sigma(\rho' - \rho'')} - \Pi_{so}\right] K_1^2, \quad (1)$$

and then the relationships are as follows for the critical heat loadings:

$$q_{cr} \geqslant \sqrt{q_{cr_0}^2 - K_1^2 r\rho'' g H_{la}(\rho_s - \rho')(1-\varepsilon)} \quad (2)$$

in the fluidized state and

$$q_{cr} \geqslant \sqrt{q_{cr_0} + \left\{K_1^2 r\rho'' \frac{128\nu'' d_n\left[\frac{L_n^2}{8m^2} + \frac{H_{la}\cdot L_n}{m}\right]}{D_g^Y \cdot \bar{\varphi}}\right\}^2 - K_1^2 r\rho'' \frac{128\nu'' d_n\left[\frac{L_n^2}{8m^2} + \frac{H_{la} L_n}{m}\right]}{D_g^Y \cdot \bar{\varphi}}} \quad (3)$$

in the channeling state.

The results are in satisfactory agreement with experiment. Heat-transfer data are presented together with empirical formulas. Photographs of the different states are presented.

NOTATION

q_{cr}	is the critical heat load;
$r, \sigma, \rho', \rho'', \nu''$	are the heat of evaporation, surface tension, vapor density, liquid density, and kinematic viscosity of vapor;
$k_1 = 0.13-0.22$	is the coefficient of proportionality;
q_{cr_0}	is the critical heat loading on boiling in the free liquid at the same pressure and temperature;
$H_{la}, \varepsilon, \rho_s$	are the depth of layer, porosity, and density of the solid component;
d_n, L_n, m	are the diameter and length of heater and number of channels;
$D_g, \bar{\varphi}$	are the diameter of channel and volume of vapor in channel.

It was assumed that $\bar{\varphi} = 0.8$ in the calculations; it was shown that D_g may be taken as equal to the diameter of a bubble on breaking away, as in the experiments, and a formula is suggested for determining m.

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CONTINUOUS PLASTIC-VISCOSITY MEASUREMENT ON A LIQUID FROM THE DAMPED RECIPROCATING MOTION OF A PLATE

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The authors consider the scope for using damped oscillation of a system with lumped parameters to monitor the plastic viscosity of a liquid obeying Bingham's law.

The equations for the boundary layer in such a medium are used to derive theoretically the relationship between the plastic voscosity and the damping parameter for a plate freely vibrating as a whole, which is linked to a spring and is perpendicular to the steady flow of viscoplastic liquid.

It is assumed that the vibration amplitude is small ($\xi_0 \ll L$) and that $2mU\alpha/S \gg \tau_T$ (which is equivalent to the condition $\sqrt{\rho U^3 \eta / 2L} \gg \tau_T$), in which case the relationship takes the form

$$\alpha \simeq K\sqrt{\rho\eta}, \quad K = \frac{1}{\rho_0 d}\sqrt{2U/15L}.$$

This relationship enables one to monitor the plastic viscosity continuously.

NOTATION

ξ_0	is the amplitude of vibration;
L, d	are the length and thickness of plate;
m	is the mass;
U	is the velocity of incident flow;
α	is the damping factor for the vibrational system;
S	is the area of one face of plate;
ρ, η, τ_T	are the density, plastic viscosity, and yield point of liquid;
K	is the coefficient of proportionality between damping parameter and square root of the product of plastic viscosity and density;
ρ_0	is the density of plate material.

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TEMPERATURE DISTRIBUTION IN A FLAT CONDUCTOR WITH TEMPERATURE-DEPENDENT RESISTIVITY CARRYING AN ALTERNATING CURRENT

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UDC 538.56+536.212.2

The steady-state temperature distribution is described together with that of a monochromatic electric field in a planar conductor $0 \le x \le 2$ by the following system of differential equations:

$$t'' = -\sigma (u^2 + v^2), \quad u'' = -n^2\sigma v, \quad v'' = n^2\sigma u;$$
$$u(0) = 1, \quad t(0) = v(0) = t'(1) = u'(1) = v'(1) = 0 \tag{1}$$

$(0 \le x \le 1)$, where x is coordinate, t is temperature reckoned from the temperature at the surface of the wire, $e \equiv u + iv$ is the complex electric field amplitude, $\sigma = \sigma(t) = (1 + kt)^{-1}$ is the conductivity referred respectively to the half-thickness of a conducting planar layer a, to the combination $(ae_0)^2\sigma_0/2\lambda$, and to the amplitude of the electric field at the surface of the conductor e_0, where the conductivity is $\sigma_0 \equiv \sigma(0)$; $n \equiv a\sqrt{\sigma_0\mu\omega}$ (the frequency factor) and $k \equiv \sigma_0(ae_0)^2\alpha/2\lambda$ (the nonlinearity parameter) are the definitive criteria in the problem; λ, μ, and α are the thermal conductivity, magnetic permeability and temperature coefficient of the specific resistance $\rho(t) = \sigma^{-1}(t)$ of the conductor; and ω is the circular frequency of the field.

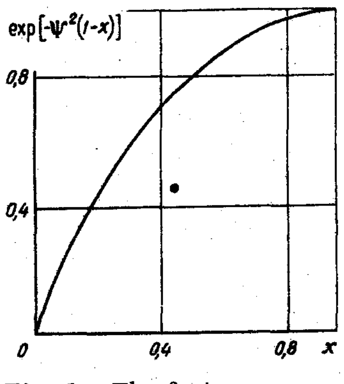

Fig. 1. The factor $\exp[-\Psi^2(1-x)]$ characterizing the coordinate dependence of $t\sqrt{(\pi/2)}k$ for large values of k.

We are interested in the asymptotic behavior of t for large k; if $k = \infty$, the solutions to (1) are almost everywhere the constants $t = v = 0$, $u = 1$ so (1) readily shows that $v \sim t = O(k^{-1/2})$ and $u = 1 - O(k^{-1})$ for k large (weak dependence on field and temperature, strong current skin effect). The first equation in (1) takes the asymptotic form $t'' = -(kt + 1)^{-1}$ if one substitutes the principal term for the square of the field modulus and integrates twice in quadratures:

$$t' = \sqrt{\frac{2}{k} \ln \frac{\rho[t(1)]}{\rho(t)}},$$
$$t = \frac{1}{k}\left\{\rho[t(1)] \exp\left[-\Psi^2\left(\frac{1-x}{\rho[t(1)]}\sqrt{\frac{2k}{\pi}}\right)\right] - 1\right\}, \tag{2}$$

where $\Psi(\xi)$ is a function reciprocal to the probability integral $\Phi(\eta)$ and the constant of integration $\rho[t(1)]$ is defined by $\rho[t(1)]\Phi\{\sqrt{\ln\rho[t(1)]}\} = \sqrt{2k/\pi}$; in this approximation, t is not dependent on n.

If $kt(1) \gg 1$ (or else $\sqrt{k} \gg \sqrt{\pi/2})\Phi\{\sqrt{\ln\rho[t(1)]}\} \simeq 1)$, and (2) becomes

$$t' \simeq \sqrt{\frac{2}{k} \ln \frac{\sqrt{2k/\pi}}{\rho(t)}}, \qquad (3)$$

$$t \simeq \sqrt{\frac{2}{\pi k}} \exp[-\Psi^2 (1-x)] - \frac{1}{k} \quad \left(x > x_0 \equiv \frac{2}{\sqrt{k \ln (k/2\pi)}} \ll 1\right)$$

(Fig. 1), where the residue is negligibly small.

We integrate equations (1) twice with respect to v" and u" to get revised asymptotic estimates incorporating the role of n for the temperature and field components for k large:

$$t \sim k^{-1/2}, \quad \sigma \sim k^{-1/2}; \quad 1 - |e| \sim 1 - u \sim n^4 k^{-1}, \ \arg e \simeq v \sim -n^2 k^{-1/2} \ (x > x_0). \qquad (4)$$

This relationship can readily be transferred to the case of any reasonably rapidly decreasing $\sigma(t)$ while retaining the above general features. In particular, in an analogous asymptotic case we have in place of (3) the integral relations

$$t' = \sqrt{2 \int_t^{t(1)} \sigma(t)\, dt}, \qquad (1-x)\sqrt{2} = \int_t^{t(1)} \frac{dt}{\sqrt{\int_t^{t(1)} \sigma(t)\, dt}}, \qquad (5)$$

where the constant of integration of t of (1) is defined by $\int_0^{t(1)} \frac{dt}{\sqrt{\int_t^{t(1)} \sigma(t)\, dt}} = \sqrt{2}$.

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UPWARD FLOW OF A LIQUID FILM IN RESPONSE TO A SPIRAL GAS FLOW

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Results are given from an experimental study of the film thickness for a two-phase spiral flow in a short tube. The film thickness was measured with a contact needle. The main tests were done in a tube with d = 30 mm, and $l/d = 5$. The spiral gas flow was provided by 6 tangential slots of height equal to the diameter and with a ratio of the total slot area to the area of tube cross-section n = 1.0 (spirality parameter of the flow). The liquid was admitted via a ring slot in the wall of width 2 mm at a distance 0.5 d above the tangential channels. The tests were done with Re_{fi} for the film of 70-1100 and Re_g for the gas of $3.3\text{-}8.0 \cdot 10^4$. Measurements were made of the minimum film thickness representing a continuous layer of liquid directly at the wall, the average thickness δ_{av}, and the maximum thickness δ_{max}.

It was found that the surface of the film is always perturbed by irregular waves of different types, which are elongated formations, with δ_{max}/δ_{av} of 4 or larger.

It was also found that δ_{min} may be of the order of 0.01 mm, i.e., the main body of the liquid travels in the waves. The results are described to ±12% by

$$\frac{\delta_{min}}{d} = 2.48 \cdot 10^3 \cdot Re_g^{-1.3}. \qquad (1)$$

The results were examined as regards the dependence of δ_{av} and δ_{max} on Re_{fi}, which reveal three hydrodynamic states of film flow. The first state C is observed up to a film Re_{fi} of about 300, while the second D occurs over the range 300 to 600, and the third E above 600. The results are represented to ±15% and 17% respectively by

$$\frac{\delta_{av}}{d} = A\, Re_g^{-a}\, Re_{fi}^{b} \left(\frac{x}{d}\right)^c, \qquad (2)$$

TABLE 1

Mode	A	a	b	c	B	m	h	k
C	$2{,}033 \cdot 10^4$	1,61	0,323	0,2	46,25	1,4	1,195	0,335
D	10,5	1,3	1,1	0,3	10,5	1,1	0,918	0,26
E	$1{,}097 \cdot 10^3$	1,3	0,391	0,2	$7{,}715 \cdot 10^2$	1,1	0,26	0,125

$$\frac{\delta_{max}}{d} = B\,\mathrm{Re}_g^{-m}\,\mathrm{Re}_{fi}^h\left(\frac{x}{d}\right)^k. \quad (3)$$

Table 1 gives the results.

State C corresponds to laminar flow of the liquid and is characterized by relatively stable wave production. In that state one can get Taylor–Goettler vortices. In state D, the film is turbulent, while in state E there is considerable removal of liquid from the surface of the film as droplets in the gas flow, so δ_{av} and δ_{max} vary little with the Re_{fi} for the films.

A study has also been made of the effects of viscosity and tube diameter. Values have been drawn up for δ_{av} in axial and spiral flows.

NOTATION

x	is the longitudinal coordinate;
d, l	are the diameter and length of tube;
$\mathrm{Re}_g = ud/\nu$	is the Reynolds number for the gas;
$\mathrm{Re}_{fi} = W/\pi d\nu_w$	is the Reynolds number for the liquid film;
u	is the mean axial velocity of gas;
W	is the liquid volume flow rate;
ν, ν_w	are the kinematic velocity of gas and water.

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TRANSITION FUNCTIONS AND THE SOLUTION OF DIRECT AND INVERSE PROBLEMS IN THERMOELASTICITY FOR A PLATE

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The thermoelastic stresses in a body are usually derived from the temperature distribution; the method is presented for solving the thermoelastic problem for a plate with symmetrical boundary conditions of the third kind using transition functions. These simplify the procedure for deriving the stresses without deriving the temperature distribution in an explicit form. The stress is given by

$$\sigma_{xx}(z_k,\ \mathrm{Fo}_N) = \sum_{h=1}^{N} \mathrm{Bi}\,(\mathrm{Fo}_h)\,\frac{\lambda}{R}\,[\upsilon_c - \upsilon\,(R,\ \mathrm{Fo}_h)]\,\Delta\sigma_{xx2}^{\mathrm{II2}}\,(z_k,\ \mathrm{Fo}_{N-h+1}).$$

The surface temperature is ϑ(R, Fo_N) is found from

$$\upsilon(R,\ \mathrm{Fo}_N) = \frac{1}{1+A_N}\left\{A_N\upsilon_c + \sum_{h=1}^{N-1} \mathrm{Bi}\,(\mathrm{Fo}_h)\,\frac{\lambda}{R}\,[\upsilon_c - \upsilon\,(R,\ \mathrm{Fo}_h)]\,\Delta\upsilon_2^{\mathrm{II2}}\,(R,\ \mathrm{Fo}_{N-h+1})\right\},$$

where $A_N = \mathrm{Bi}(\mathrm{Fo}_N)(\lambda/R)v_2^{\mathrm{II2}}(R,\ \mathrm{Fo}_1)$; $v_2^{\mathrm{II2}}(R,\ \mathrm{Fo}_N)$ is the transition function for the thermal-conduction case, and $\sigma_{xx2}^{\mathrm{II2}}(z_k,\ \mathrm{Fo}_N)$ is the same for the thermoelastic problem, while $v = t - t_0$ and t_0 is the initial temperature of the plate.

In engineering, one often has to maintain some specified stresses at particular points in the body; this inverse problem can be solved for a plate with symmetrical boundary conditions by determining the heat-flux density from

$$q(\mathrm{Fo}_N) = \frac{1}{\sigma_{xx2}^{\mathrm{II2}}(z_k, \mathrm{Fo}_1)} \left[\sigma_{xx}(z_k, \mathrm{Fo}_N) - \sum_{h=1}^{N-1} q(\mathrm{Fo}_h)\, \Delta\sigma_{xx2}^{\mathrm{II2}}(z_k, \mathrm{Fo}_{N-h+1}) \right].$$

From the heat-flux density one can readily determine the temperature at the surface, and then the heat-transfer coefficient for a given external temperature (or else the external temperature for a given variation in the heat transfer).

These methods are used to consider the optimum heating of a flange in a horizontal port in steam turbines. The heating occurs in two stages. In the first stage one has to consider the thermoelastic problem up to the point where the limiting permissible stress is attained. For this purpose one specifies a law for the steam temperature variation with a maximum rate of change determined by the engineering possibilities. In the second stage one solves the inverse thermoelastic problem, which is controlled by the limiting permissible stress and one thereby obtains the temperature of the medium with a known law for the heat-transfer coefficient.

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CALCULATION OF PIPELINE STRESS AND TEMPERATURE BY INITIAL-CONDITION RELAXATION AND TIME STEPPING

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A method is given for calculating the transient temperature and stress distributions in a pipeline for arbitrary variations in the surrounding temperature and the heat-transfer coefficient at the internal surface. The distribution of T is determined for instants τ_ν ($\nu = 1, 2, \ldots$), and this is a function of the boundary conditions for τ_ν in the period from $\tau_{\nu-1}$ to τ_ν together with the initial conditions at $\tau_{\nu-1}$. The instants τ_ν are chosen in such a way that the time interval $\Delta\tau_\nu = \tau_\nu - \tau_{\nu-1}$ results in negligible changes in the heat-transfer coefficient and the thermophysical properties, while being sufficient for one to ease the boundary conditions on the basis $t_{\nu-1} = \bar{t}_{\nu-1}$ (here $\bar{t}_{\nu-1}$ is the mean-integral wall temperature).

Each of the intervals $\Delta\tau_\nu$ is itself divided into intervals τ_k ($k = 0, 1, \ldots, m$), which correspond to the vertices on a bent line, which represents the temperature of the environment as a function of time. The final solution for time τ_ν for a given $\bar{t}_{\nu-1}$ can be obtained by linear superposition of the solutions for stepwise and linear variation in the surrounding temperature.

The minimum permissible value for $\Delta\tau_\nu$ corresponds to Fo = 0.3 (Fo is the Fourier criterion). The results from this method agree satisfactorily with those from computer treatment.

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Abstract submitted March 12, 1973.

DERIVATION OF HEAT- AND MASS-TRANSFER COEFFICIENTS FOR HEAT EXCHANGERS FROM FREQUENCY CHARACTERISTICS

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UDC 536.24.08

Refrigerating and heating equipment employs systems with extensive surfaces with fins of various shapes, tubes of various cross-sections, and tube bundles. Calculations on such equipment require a knowledge of the heat- and mass-transfer coefficients in terms of the thermal and hydraulic characteristics. There is therefore considerable interest in devising methods of calculating mean values, and any such technique should be suitable not only for laboratory work but also for actual industrial conditions.

A method is presented for calculating $\overline{\alpha}$ and $\overline{\beta}$ as follows:

1. Equations are derived for the transient processes in a heat transfer apparatus as an object with distributed parameters and various perturbing and control channels. For this purpose one proceeds as follows:

a) one writes out the equations for heat and material balance in the equipment in differential form with partial derivatives;

b) the appropriate transformations are performed to linearize the relationships (step a) for small deviations, with zero initial conditions; a Laplace transformation is applied with respect to the appropriate coordinates (time and a coordinate characterizing the distribution of the parameters in the heat exchanger);

c) the system of equations is solved (step b) in operator form by appropriate methods;

d) an inverse Laplace transformation is applied with respect to the appropriate coordinates. The theory of operational calculus is used to obtain the transfer functions for the heat exchanger that define the transient process in respect of various perturbations.

2. One uses the equation from a step **d** above to derive the real and imaginary frequency characteristics.

3. If the input perturbation is not a stepwise one (as occurs under actual industrial use of such equipment), one determines the real and imaginary parts of the frequency characteristics from the curve representing the perturbation as a function of time.

4. Generalized real and imaginary frequency characteristics are derived as in point 2 above for a given perturbation (point 3).

5. The transient response of the exchange to a given perturbation is used with the same approach (point 3 above) to calculate the real and imaginary frequency characteristics of the object that can be determined by experiment.

6. The real and imaginary generalized frequency characteristics of point 2 above are compared with the real and imaginary frequency characteristics of point 5 for a given frequency ω_i:

$$R_{ea}(\omega_i) = P(\omega_i)\, P_G(\omega_i) - Q(\omega_i)\, Q_G(\omega_i) = R_{ee}(\omega_i); \tag{1}$$

$$Q_a(\omega_i) = P(\omega_i)\, Q_G(\omega_i) + Q(\omega_i)\, P_G(\omega_i) = I_{me}(\omega_i), \tag{2}$$

where $P(\omega_i)$ and $Q(\omega_i)$ are the real and imaginary frequency characteristics of point 2, while $P_G(\omega_i)$ and $Q_G(\omega_i)$ are the real and imaginary frequency characteristics of the perturbation..

Then from (1) and (2) one can determine $\overline{\alpha}$ and $\overline{\beta}$; in (1) and (2) one can use frequencies from $\omega_0 = 0$ to ω_{co} (the cutoff frequency). However, one should avoid resonance frequencies commonly found in such systems.

This method allows one to determine the transfer coefficients for laboratory and industrial equipment, which greatly facilitates design.

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