ABSTRACTS OF PAPERS DEPOSITED AT VINITI*

CONVECTIVE HEAT EXCHANGE BETWEEN COAXIAL ROTATING CYLINDERS

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UDC 532.54

The results of an experimental study of the heat exchange between horizontal, coaxial, rotating cylinders during laminar flow with macrovortices showed that with rotation of only the inner cylinder and a temperature difference of up to 220°C between the surfaces of the cylinders in a range of Taylor numbers of $2 \cdot 10^3 - 10^5$ the effect of a negative radial temperature gradient on the coefficient of convective heat exchange is taken into account by calculating the physical parameters for the arithmetic mean temperature of the gap.

With simultaneous rotation of the coaxial cylinders an increase in the rotation rate of the outer cylinder promotes stabilization of the flow and therefore a decrease in the heat exchange coefficient. The results of experiments conducted in a range of Taylor numbers of $0\text{-}10^5$ and ratios of the angular velocities of the cylinders of 2.5-25/36 are generalized by empirical equations.

A study of the effect of a negative radial temperature gradient on the stability of the Couette flow of a strongly viscous liquid, for which we used solutions of glycerin in water at concentrations of 77.4 and 94.1%, showed that whereas with the flow of a gas the negative radial temperature gradients promote an increase in the stability threshold, in the given case they decrease the stability of the flow by creating large viscosity gradients. The change in the stability threshold can be determined from the empirical equation proposed.

A visual study of the secondary flows confirmed the stabilizing effect of the rotation of the outer cylinder and the possibility of the development of instability under the effect of asymmetrical disturbances during the rotation of the cylinders in opposite directions.

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HEAT EXCHANGE AND CRITICAL HEAT LOADS DURING BOILING IN A HORIZONTAL SLOT IMMERSED IN A FREE VOLUME OF LIQUID

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UDC 536.423.1

The work is devoted to an experimental study of the heat exchange and critical heat loads during boiling in a plane-parallel slot gap with one-sided heating immersed in a free volume of liquid.

The experiments were conducted on an instrument of the usual type. The working section consisted of a glass-textolite backing with a nichrome plate, the heater, cemented on. A movable plate was fastened

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on top, forming the slot gap. The heating of the nichrome plate was produced by the direct passage of an alternating current. The heater temperature was determined by copper–constantan thermocouples and the critical heat load was determined at the moment of the automatic breaking of the power circuit upon a sharp increase in the temperature of the working section.

The studies were conducted with the boiling of water and ethyl alcohol under conditions of atmospheric pressure. The load was varied in the range of 10^4–10^6 W/m^2.

It is noted that at small heights of the slot gap one records the mechanism of a process differing considerably from the mechanism of boiling in a free volume. With slot sizes close in absolute value to the separation diameter d_0 of a bubble,one observes an increase in the heat exchange intensity and a decrease in the critical heat loads.

Based on the limiting relationships, we propose as the dimensionless geometrical parameter of the slot the value

$$\Gamma = \frac{\delta}{D_0}.$$

A generalization of the experimental data on the heat exchange and critical heat loads is presented in the form

$$\mathrm{Nu}_* = G_1 \mathrm{Re}_*^n \mathrm{Pr}^m \frac{1}{\mathrm{th}\,(G_2\Gamma)}, \tag{1}$$

$$q_{\mathrm{cr.b}} = q_{\mathrm{cr.v}}\,\mathrm{th}\,(G_3\Gamma), \tag{2}$$

where Nu_*, Re_*, and the constants G_1, n, and m correspond to any criterial relationships adopted for the calculation of heat transfer during boiling.

From an analysis of the experimental data $G_2 = 0.48$ and $G_3 = 1.2$.

The study conducted indicates the existence of definite properties of the boiling process in a plane horizontal slot immersed in a free volume of liquid.

Equations (1) and (2) can be recommended in a first approximation for engineering calculations.

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THERMODYNAMIC CHARACTERISTICS OF A VORTEX TUBE WITH SUPPLEMENTARY FLOW

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532.501.312

The relatively low energy efficiency of vortex tubes restricts the reasonable areas of their application in engineering. Because of this, questions of the study of the possibility of increasing their adiabatic efficiency take on important meaning. The question of the improvement of processes of energy transport in vortex tubes through the maximum approximation of the working constructions to the physical model is examined in this report. This is achieved through the introduction of a supplementary mass of gas compressed to a slight excess pressure (P = 0.02-0.15 bar) from the side of the hot end into the axial zone which has a reduced pressure.

The hypothesis of vortex interaction was laid at the basis of the gas-thermodynamic analysis of the suggested possibility for increasing the efficiency. Equations for the determination of the relative flow rate of the cold stream, its relative temperature, and the adiabatic efficiency of the tube were obtained in a form convenient for computer calculations.

As the determining parameters we chose both geometrical (the relative radius of the diaphragm opening and the relative area of the nozzle) and operating (the pressure at the entrance to the vortex tube, the degree of expansion of the gas in the tube) parameters.

Numerical calculations are made with the following limits of variation of the determining parameters:

1. Total pressure at the entrance to the vortex tube

$$P_1^* = 2.94-5.88 \text{ bar}$$

2. Relative radius of diaphragm opening

$$\bar{r}_d = 0.4-0.9.$$

3. Relative area of nozzle entrances

$$\bar{F}_n = 0.02-0.12.$$

The characteristics of a vortex tube with supplementary flow are presented in the form of graphs. Their analysis indicates the sufficiently great possibilities of the proposed method of increasing the efficiency of vortex tubes.

One important advantage of vortex tubes with supplementary flow is the possibility of cooling a large mass of gas by means of a small amount of compressed gas.

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THEORY AND CALCULATION OF A FILTER BED

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Consider the one-dimensional isothermal flow of a dusting gas mixture through a granular filter baffle.

Following equations are taken together:

1) Filtration of the flow

$$\frac{m}{g}\frac{d}{dt}\left(\frac{v_x}{m}\right)+\frac{1}{\gamma}\frac{\partial p}{\partial x}+\frac{v_x}{f}=0, \tag{1}$$

2) continuity

$$\frac{\partial(m\gamma)}{\partial t}+\frac{\partial(\gamma v_x)}{\partial x}=0, \tag{2}$$

3) state

$$\frac{p}{\gamma}=RT, \tag{3}$$

4) conservation of dust matter

$$\gamma_d\frac{\partial m}{\partial t}=-\frac{\partial}{\partial x}(cv_x), \tag{4}$$

5) dust removal kinetics

$$-\frac{\partial(v_x c)}{\partial x}=bv_x c-a\gamma_n m, \tag{5}$$

6) the f dependence of φ

$$\varphi=\frac{\mu}{\gamma}f, \tag{6}$$

where

$$\varphi = Amd_e^2, \quad \frac{1}{d_e} = \frac{3}{2}\,\frac{\Delta g_1}{d_1} + \sum_{i=1}^{n} \alpha_i \lg \frac{d_{i+1}}{d_i}, \quad \alpha_i = \frac{\Delta g_i}{d_{i+1}-d_i},$$

which gives an expression for the performance of the granular filter which relates the basic parameters of the gas flow (p, v_x), the dust (γ_d, a, b, c_0), and the filter (A, m_0, d_e, $\partial p/\partial x$);

$$\eta = 1 - \frac{\mu v_{x0} \sum_{n=1}^{\infty} \frac{(bx)^{n-1}}{(n-1)!}\, \tau_d \exp[-(bx+at)]}{Ad_e^2\left[m_0 - \frac{c_0 v_{x0}}{\gamma_d} T \exp(-bx)\, \frac{\partial p}{\partial x}\right]}, \tag{7}$$

where

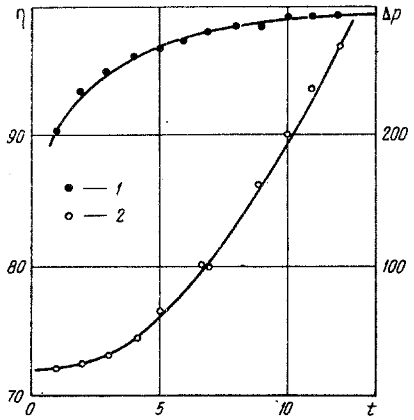

Fig. 1. Performance of layer filter 1 and pressure drop 2 as functions of time (the solid line represents the theoretical result; η in % and Δp in mm of water, with t in min.

$$\tau_d = \tau_{n-1} - \frac{(at)^{n-2}}{(n-2)!}, \quad \tau_1 = \exp at, \quad T = \int_0^t B \exp(-at)\, dt,$$

$$B = \sum_{n=1}^{\infty} \tau_d \frac{b^{n-1}x^{n-2}}{(n-2)!}\left(1 - \frac{bx}{n-1}\right).$$

1. For the steady state

$$p = p_0\left(1 - \frac{x}{L}\right) + K_1\left[\int_0^x\int_0^x \exp(-bx)\,dx\,dx - \int_0^L\int_0^L B\exp(-bx)\,dx\,dx\right]\frac{x}{L} + p_k\frac{x}{L}, \tag{8}$$

where $K_1 = \mu c_0 v_{x0}/\gamma_n A m_0 d_e^2$ and the boundary conditions are $p = p_0$ at $x = 0$ and $p = p_k$ at $x = L$.

2. For a thin filtering layer under the same boundary conditions:

$$p = \sqrt{p_0^2 + p_k^2 - p_0^2\frac{x}{L}}. \tag{9}$$

3. For the nonstationary filtration mode:

$$p = p_0 \exp\left(-\int_0^t \frac{c_0 v_{x0} B \exp[-(bx+at)]}{m_0\gamma_d - c_0 v_{x0} T \exp(-bx)}\, dt\right). \tag{10}$$

The paper also deals with the definition of the flow rate of filtering material.

Expressions are derived for the filter, which have been checked by experiment; the calculated performance and pressure difference given by (7) and (10) agree well with experiment (Fig. 1).

NOTATION

x, current layer thickness; v_{x0} and v_x, the initial and current filtration rates, respectively; p, gas flow pressure; p_0, initial pressure; γ, gas density; T^0, absolute temperature of gas flow; γ_d, density of dust; R, gas constant; c_0, initial dust concentration; μ, dynamic viscosity; m_0 and m, initial and current porosities; φ, permeability function; A, a, and b, experimental coefficients; d_e, effective filling diameter; d_1, largest diameter of last fraction; d_i and d_{i+1}, extreme diameters of a given fraction; Δg_1, weight proportion of last fraction as in relation to total weight of specimen, Δg_i, weight of fraction i as a proportion of total weight; a_i, slope of mechanical-analysis curve; L, layer thickness; η, filter performance; f, filtration function; t, filtration time; and g, acceleration due to gravity.

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State Basic Chemistry Research and Design Institute, Khar'kov.
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APPROXIMATE MATRIX CALCULATION OF THE TEMPERATURE FIELD OF A PLANE LAYERED SYSTEM

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UDC 536.24

The author solves the one-dimensional problem of the temperature field distribution in a plane n-layered wall heated on one side by a heat flux $q(\tau)$ that varies with time τ according to an arbitrary predetermined law. The temperature distribution in each layer (sublayer) of the n-layered wall is approximated by the linear relation

$$t_k(\tau) = P_k(\tau) + \frac{x - \delta_k}{h_k}\,\theta_k, \tag{1}$$

in which $P_k(\tau)$ is the average temperature of the layer, δ_k is the coordinate of the median plane of the k-th layer, h_k is half the thickness of the layer, x is the coordinate, and θ_k is the temperature differential between the boundaries of the k-th layer.

Using the equality of the temperatures and heat fluxes at the layer interfaces and the uniform initial temperature distribution in the wall, we obtain a vector system for the determination of the temperature differentials in each layer:

$$\dot{\mathbf{s}} + A\mathbf{s} = -q(\tau)\,\mathbf{e}, \tag{2}$$

where $\mathbf{s}$ and $\mathbf{e}$ are column vectors [$\mathbf{s} = (s_1, s_2, \ldots, s_n)$; $\mathbf{e} = (1, 0, 0, \ldots, 0)$] and A is the tridiagonal "oscillation" matrix

$$A = \begin{Vmatrix} 2k_1 & -k_2 & 0 & 0 & 0 \ldots\ldots 0 \\ -k_1 & 2k_2 & -k_3 & 0 & 0 \ldots\ldots 0 \\ 0 & -k_2 & 2k_3 & -k_4 & 0 \ldots\ldots 0 \\ 0 & 0 & \ddots & & \vdots \\ \vdots & & & \ddots & \vdots \\ 0 & 0 \ldots\ldots & -k_{n-2} & 2k_{n-1} & -k_n \\ 0 & 0 \ldots\ldots & 0 & -k_{n-1} & 2k_n \end{Vmatrix},$$

whose elements are determined by the thermophysical properties and thickness of each layer of the wall:

$$k_k = \frac{3}{2} \cdot \frac{\lambda_k}{c_k \gamma_k h_k^2}.$$

To solve Eq. (2) we must determine the eigenvalues μ and orthonormal vectors $\mathbf{U}$ and $\mathbf{V}$ corresponding to those eigenvalues.

The solution of Eq. (2) has the form

$$f_k(\tau) = -V_k^1 \int_0^t q(\tau) \exp[-\mu_k (t - \tau)]\, d\tau. \tag{3}$$

Knowing the function $q(\tau)$, we use Eq. (3) to find $f_k(\tau_1)$ for a specific time τ_1, and then the functions $s_k(\tau_1)$ and

$$\theta_k(\tau_1) = \frac{3}{2} \frac{1}{c_k \gamma_k h_k} s_k(\tau_1).$$

The average temperature $P_k(\tau_1)$ in each layer of the wall is determined from the equal-temperature conditions at the interfaces and the balance equation

$$2 \sum_{k=1}^{n} c_k \gamma_k h_k P_k = \int_0^{\tau} q(\tau)\, d\tau.$$

A computer flow chart for the calculations is given, the convergence problem is analyzed, and examples are given illustrating the numerical implementation of the given technique for several laws of time variation of the heat flux.

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AN APPARATUS FOR PORE-RADIUS DISTRIBUTION DETERMINATION

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UDC 541.182.02

The distribution is measured by a method based on the weight change when one liquid displaces another in the pores. A strain-gauge balance is used to measure the weight change, which has some advantages over earlier hydrostatic microbalances: one can balance out the initial mass, which means that one can measure large specimens of any shape without loss of sensitivity even though the mass changes are small. Also, the design and use are simple, and the course of the process can be followed directly from the recorder chart. This has been used in measurement of specimens made by sintering various fractions of metal powders.

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EFFECTS OF MATERIAL PROPERTIES ON DISLOCATION ETCHING

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UDC 536.21

A (111) surface of silicon has been found to show a marked change in the shape of the dislocation etch pits, and this occurs in a metal-silicon system when the thermal expansion coefficients of the metal and silicon differ by more than a factor 1.1-1.3. Dislocation pits as obtuse-angled triangles extending along the short side of the crystal lie mainly in the center of the specimen. The shape of the pits does not alter when the difference in the above characteristics is less, and the pits are then regular triangles.

The effects are due to thermal stresses in the silicon; these alter the stress energy of the dislocations and affect the chemical etching at the point of emergence. The minimum stress required to produce an appreciable pit shape change is about 1.5 kgf/mm^2.

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EFFECTS OF SOME FACTORS ON SUBSTRATE TEMPERATURE

G. B. Dinzburg

UDC 536.241

A substrate is carefully cleaned in alcohol and then in chromic acid and placed under a cover in an oven for heating to the set temperature. The half-ring oven has 8 places, three of which take glass, copper, and duralumin substrates. A metal film is deposited on each under vacuum, the metal evaporating from a tungsten crucible of conical shape containing a known amount of metal.

This metal is heated to its boiling point to produce uniform atomic motion; some of the evaporating atoms fall on the substrate, and set up a mixture temperature θ °C.

The substrate temperature is substantially affected by the thermal radiation from the tungsten crucible and the heat radiated from the evaporator, which does not act exactly as a black body. The heat radiated from the substrate and film reduces the final substrate temperature somewhat.

The vacuum of 10^{-6} mm Hg was provided by a VN-46M rotary point and TSVL-100 diffusion pump.

The substrate temperature was monitored with sealed-in thermocouples. The substrate masses were measured with analytical balances, while the film thickness was measured with a thickness gauge.

The deviation of the measurements from the theoretical values range from 0.5 to 1° with a 10-second deposition, the exact value being dependent on the substrate material and evaporating metal.

There was a certain effect on the substrate temperature from the mounting; the temperature rose by 0.4-0.5° in 10 sec on deposition on substrates having good thermal conductivity. The difference in final temperature between the theoretical calculations and the measurements enables one to detect defects in the film and to determine the crystallite size, as well as to determine the film thickness and various other quantities.

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TRANSFER PROCESSES IN MULTIPLE VALVES

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UDC 536.75

In the present article equations are derived which permit one to estimate the conducting properties of a valve aggregate of the parallel or series type from the properties of its component elements.

It is assumed that in the stationary state the transfer processes in each valve a are described by the equations

$$ {}_aJ_i = \sum_{j=1}^{n} {}_aL_{ij}\, {}_aX_j, $$

where J are flows, X are forces, and L are phenomenological coefficients, with the matrices $({}_aL)$ being symmetrical and corresponding to positive definite quadratic forms. The forces have the form $X_j = \Delta F$, where F_j is some intensive parameter not undergoing sudden changes at the boundary of separation between two media.

In the case of penetrable valves it is shown that an aggregate of m elements connected in parallel is equivalent to a new valve with a matrix of coefficients

$$(M) = \sum_{a=1}^{m} ({}_aL).$$

When these elements are connected in series an aggregate is obtained which is characterized by a matrix of phenomenological coefficients

$$(P) = \left(\sum_{a=1}^{m} ({}_aL)^{-1}\right)^{-1}.$$

If semipenetrable elements are connected in parallel then the matter is reduced to the previous case by replacing all the matrices $({}_aL)$ by matrices $({}_aL')$ of identical order. The form of the matrices $({}_aL')$ corresponds to some conditionally introduced valve which is penetrable for any flow which penetrates at least one of the m real valves taken. In this case the aggregate is characterized by the matrix of coefficients

$$(M') = \sum_{a=1}^{m} ({}_aL').$$

When semipenetrable elements are connected in series the aggregate is described by the matrix

$$(P') = \left(\sum_{a=1}^{m} ({}_aC)^{-1}\right)^{-1}.$$

The elements of the matrix $({}_aC)$ are determined by the equation

$${}_aC_{jk} = {}_aL_{jk} - \frac{1}{{}_a\Delta} \sum_{i=s+1}^{n_a} {}_aL_{ji}\, {}_a\Delta_{ik}.$$

Here it is assumed that all the valves are penetrable for the flows $J_1, J_2, \ldots, J_s$, while n_a is the total number of flows passing through valve a. ${}_a\Delta$ denotes the angular minor of the matrix $({}_aL)$, obtained through deletion of the first s rows and columns in it. The determinant ${}_a\Delta_{ik}$ is obtained from ${}_a\Delta$ by replacing all the elements in the i-th column by the rule ${}_aL_{ji} \to {}_aL_{jk}$.

All the matrices (M), (P), (M'), and (P') are symmetrical and correspond to positive definite quadratic forms.

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