# THERMAL CONVECTION IN SECTIONS OF MULTILAYER CRYOGENIC HEAT INSULATION

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An experimental investigation has been made of heat transfer during natural convection in multilayer heat insulating structures consisting of anisotropic permeable porous materials.

Numerous papers have dealt with investigations of natural convection in permeable porous media. One of the most important applications of these investigations is to thermal insulation, and in particular to the thermal insulation of vessels containing cryogenic liquids.

In cryogenic engineering widespread use is made of various types of multilayer thermal insulations consisting of assemblies of perforated sheets (screens) separated by layers of permeable materials [1, 2]. Such structures, which have significantly anisotropic properties, are also quite valuable in astronautical engineering.

For efficient operation, the multilayer thermal insulations should be evacuated, so that they are also termed vacuum-screen thermal insulations (VSTI). During use on the ground of astronautical systems the VSTIs are filled as a rule with gas having a pressure close to atmospheric pressure. In this case thermal convection occurs in the thermal insulation of fuel tanks filled with cryogenic liquids. The contribution of the convective component to the overall transfer of heat through an element of the gas-filled insulation is determined in many respects by the permeability of the heat-insulating material. It will be obvious that for multilayer structures the permeability may depend on the direction.

A two-dimensional mathematical model was developed in [3] for natural convection in anisotropic porous media using the Darcy–Boussinesq approximation, and numerical investigations were carried out of the flow and heat transfer in vertical layers (thermal insulation elements). Calculated results were obtained for a wide range of values of the Rayleigh number for filtration, $Ra^* = Gr \cdot Pr \cdot Da$, and of the coefficient of anisotropy $K_x/K_y$, defined as the ratio of the permeability coefficients along the two coordinate axes. The parametric investigations which were carried out showed that the presence of a considerable anisotropy of the permeabilities leads to previously unknown features in the structure of the flow and heat transfer.

The ability to form an opinion on the reliability of the results which have been obtained, and hence on the validity of the model proposed in [3], can only be based on experimental data. Up to now such data based on a high degree of anisotropy of the permeabilities ($K_x/K_y > 10^2$) have been lacking. Such values of the coefficient of anisotropy are characteristic of multilayer gas-filled thermal insulations, which can be regarded as porous materials. Because of this, an experimental investigation of convection in this type of insulation would be of considerable value for proving out the calculation procedures.

This paper presents the results of an experimental determination of the mean heat transfer in vertical elements of a VSTI, and a comparison is then made of these results with the corresponding results of calculations carried out by the procedures of [3]. Experimental data are also provided on the permeabilities of the heat insulations. The values of the coefficients of permeability obtained along and transverse to the screens ($K_x$ and $K_y$, respectively) are used for determining the values of the Rayleigh number and the coefficient of anisotropy corresponding to the conditions of the experiments which have been carried out.

1. The mathematical model of [3], which is used for obtaining the calculated results, consists of the following equations:

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$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0, \qquad (1)$$

$$\frac{K_y}{K_x}u=-\frac{\partial p}{\partial x}+\mathrm{Ra}^*\theta, \qquad (2)$$

$$v=-\frac{\partial p}{\partial y}, \qquad (3)$$

$$u\frac{\partial \theta}{\partial x}+v\frac{\partial \theta}{\partial y}=\frac{\partial^2 \theta}{\partial x^2}+\frac{\partial^2 \theta}{\partial y^2}. \qquad (4)$$

Equations (1)-(4) rewritten in dimensionless form were solved for a vertical plane layer with a relative extension L/H on the side surfaces on which temperatures of $T_1$ and $T_2$ were maintained, while the end surfaces were thermally insulated. As the scales for length, temperature, velocity, and pressure, use was made, respectively, of: H (the thickness of the layer), $\Delta T = T_2 - T_1$, $a^*/H = \lambda^*/\rho_0 c_p H$, and $\mu a^*/K_y$. The x-axis is directed along the layer, and the y-axis transverse to it.

Within the framework of the steady-state model being considered the similarity criteria are L/H, $K_x/K_y$, and $\mathrm{Ra}^* = g\beta H K_y \rho_0{}^2 \Delta T/(\mu\lambda^*)$ (generally accepted symbols and scales have been used in the equations and dimensionless groups). The mean rate of heat transfer through the layer is of the greatest practical interest; this is determined by the Nusselt number $\mathrm{Nu} = \lambda_{ef}/\lambda^*$, which represents the ratio of the effective thermal conductivity taking convection into account to the thermal conductivity of the porous medium filled with a stationary gas (or liquid). The general dimensionless equation for the mean rate of heat transfer has the form

$$\mathrm{Nu} = f(\mathrm{Ra}^*,\ K_x/K_y,\ L/H). \qquad (5)$$

2. The experiments were carried out with the thermal insulating material type ÉSTI-2V, which is widely used in astronautical engineering. The screens had perforations of diameter 2 mm with a pitch of 10 mm.

The basis for the experimental determination of the permeability was the linearized law of Darcy, according to which the rate of filtration is proportional to the pressure gradient: $V = -(K/\mu)\mathrm{grad}\ p$. For the investigations, planar samples (packets) were used with various numbers of screens (from 10 to 60) and with stacking densities of 10-20 layers/cm. The screens had the form of rectangles of size 500 mm × 500 mm. A uniform laminar flow of air was provided in the samples in two directions: along and transverse to the screens, which made it possible to determine the values of the corresponding coefficients of permeability $K_x$ and $K_y$ which appear in the convection equations (1)-(4). The overall error in the measurements did not exceed 20%.

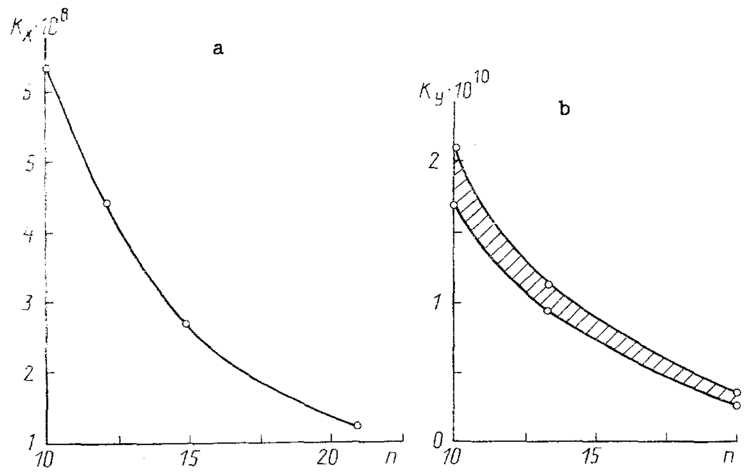


Fig. 1. Effect of the screen packing density on the longitudinal (a) and transverse (b) permeabilities of the VSTIs. $K_x$, $K_y$ are given in $m^2$, and n in number of screens/cm.

The dependence of the coefficient of permeability, along the screens $K_x$, on the packing density of the screens is shown in Fig. 1a. In contrast to $K_x$, the results of the measurements of the coefficient of permeability transverse to the screens, $K_y$, show a considerable degree of spread, so that in Fig. 1b the results are shown in the form of a zone (band) bounded by the minimum and maximum values of $K_y$. From the relationships shown in Fig. 1 it can be seen that the permeabilities of the VSTIs along the screens are two to three orders of magnitude larger than transverse to the screens.

An important characteristic of a thermal insulation is its effective thermal conductivity $\lambda_{ef}$. The mean rate of heat transfer through the VSTI is primarily determined by its value:

$$q = \lambda_{ef} \frac{T_2 - T_1}{\delta}. \qquad (6)$$

As can be seen from Eq. (6), in order to determine $\lambda_{ef}$ it is necessary to measure the boundary temperatures $T_1$ and $T_2$ and the specific heat flux q through a heat insulating layer of thickness $\delta$.

The VSTI being investigated was formed into a model consisting of a vertical cylinder of diameter 150 mm by being wound into a spiral coil. As a result, a cylindrical stratification was obtained in which the screens are arranged along the generatrix. The two-dimensional mathematical model of convection in such configurations is described by the same Eqs. (1)-(4) as for a planar anisotroic layer. The results of the calculations for the planar elements can therefore be validly compared with the experimental data for the cylindrical stratification. In the experiments the boundary conditions for the temperatures were similar to those described in the mathematical model. The cylindrical stratification had impermeable boundaries, which was correspondingly taken into account in the boundary conditions for the filtration rates.

The experiments were carried out for VSTIs filled with air, nitrogen, and helium. In order to avoid condensation of the gases the cylinder walls were not cooled below the temperatures at which the phase changes began. Cooling was accomplished by pumping cold gaseous helium through the model. The mean rate of heat transfer through the heat-insulating layer was characterized in accordance with Eq. (6) by the specific heat flux q. Under steady-state thermal conditions the value of q was determined by measuring the enthalpies of the helium pumped through the model:

$$q = \frac{G}{S}(i_{h2} - i_{h1}) = \frac{Gc_p}{S}(T_{h2} - T_{h1}). \qquad (7)$$

Here G is the mass flow rate of helium; $T_{h1}$, $T_{h2}$ are the helium temperatures at the inlet and outlet of the working section of the model, respectively; S is the arithmetic mean area of the outside and inside surfaces of the heat insulation.

In dimensionless form the mean rate of heat transfer is determined by the ratio of $\lambda_{ef}$ to the thermal conductivity of the VSTI filled with stationary gas $\lambda^*$. Evaluations carried out according to formulas for composite materials [4] showed that to an error not exceeding 5% it can be assumed that the coefficient $\lambda^*$ is equal to the thermal conductivity of the gas filling the VSTI.

The experiments were carried out for cylindrical stratifications having a ratio of the height to the thickness L/H equal to 10. The packing density was about 10 screens per centimeter with a total number of screens equal to 50. As can be seen from Fig. 1, this packing density corresponds to the following values of the coefficients of permeability and their ratio: $K_x = 6.3 \times 10^{-8}$ m², $K_y = 1.9 \times 10^{-10}$ m², $K_x/K_y = 3.3 \times 10^2$. The convection intensity in the experiments corresponded to Rayleigh numbers in the range $0 < Ra^* \leqslant 2.0$.

The results of the experimental investigations of the mean rates of heat transfer are shown in Fig. 2 in the form of a relationship between the Nusselt number $Nu = \lambda_{ef}/\lambda^*$ and the filtration Rayleigh number. The values of the effective thermal conductivity appearing in the relationship for the Nusselt number were determined with an error not exceeding 20%.

The experiments which have been carried out have confirmed the conclusion reached in [3] on the basis of numerical investigations that in the presence of marked anisotropy of

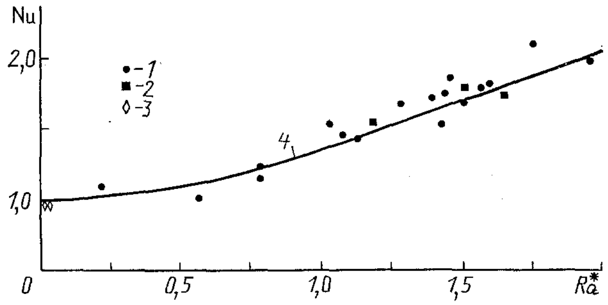


Fig. 2. Comparison of the experimental results with the calculated values of the mean heat transfer rates: 1) air; 2) nitrogen; 3) helium; 4) calculated by the procedure of [3].

the permeabilities, such as is characteristic of VSTIs, convection can be quite intensive, and can lead to a significant contribution to the overall heat transfer. Thus, for an insulation filled with air (Fig. 2, points 1) the transfer of heat at Ra* = 2.0 is increased by approximately a factor of two as a result of convection. If the air is replaced by nitrogen (points 2) the mean rate of heat transfer is practically unchanged, while on filling the VSTI with helium (points 3) the contribution of the convective component becomes negligibly small compared to the thermal conductivity of the gas.

For the same values as in the experiments of the dimensionless parameters Ra*, $K_x/K_y$, and L/H, which according to (5) govern convection in the VSTI being investigated, calculations were made of the mean rates of heat transfer according to the procedures of [3]. The calculated dependence of the Nusselt number on the Rayleigh number is also shown in Fig. 2. From the comparison of the results of the calculation (curve 4) with the experimental data it follows that the mathematical model gives a generally valid description of the real processes occurring in gas-filled VSTIs.

Thus, a result of the check which has been made of the calculation procedure of [3] it is possible to draw conclusions which are of practical importance. In particular, the procedure for calculating convection in anisotropic porous media based on the solution of Eqs. (1)-(4) for vertical plane layers with impermeable boundaries makes it possible to determine with sufficient accuracy for engineering calculations the mean rate of heat transfer in elements of multilayer gas-filled heat insulation.

## NOTATION

$K_x$, $K_y$, coefficients of permeability in the directions of the x and y axes; L, H, length and height of layer; $T_1$, $T_2$, boundary temperatures; $\Delta T = T_2 - T_1$; $\rho_0$, $c_p$, $\mu$, $\beta$, density, specific heat, viscosity, and coefficient of thermal expansion of gas; $\lambda^*$, thermal conductivity of porous medium filled with stationary gas; $a^* = \lambda^*/\rho_0 c_p$, temperature conductivity; g, acceleration of free fall; $\lambda_{ef}$, effective thermal conductivity; q, specific heat flux; $\delta$, thickness of insulation; u, v, components of filtration velocity; p, pressure; $\theta = (T - T_1)/(T_2 - T_1)$, dimensionless temperature; x, y, Cartesian coordinates; $Ra^* = g\beta H K_y \rho_0{}^2 c_p \Delta T/(\mu\lambda^*)$, Rayleigh number for filtration; $Gr = g\beta H^3 \rho_0{}^2 \Delta T/\mu^2$, Grashof number; $Pr = \mu/(\rho_0 a^*)$ Prandtl number; $Da = K_y/H^2$, Darcy number; $Nu = gH/(\lambda^* \Delta T)$, Nusselt number.

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