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HEAT TRANSFER OF A VERTICAL BUNDLE OF HEAT-RELEASING RODS IN THE ABSENCE OF CIRCULATION OF THE HEAT CARRIER

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A method is presented for approximate calculation of conductive–radiative heat transfer in bundles of heat-releasing rods, and an empirical estimate is given of the effective thermal conductivity.

It has recently become necessary to develop methods of calculating heat transfer in bundles of heat-releasing rods in the absence of circulation of the heat carrier (coolant). This problem has arisen in connection with the storage and transport of spent fuel assemblies.

Below we examine a system of vertically positioned fuel rods placed in a shell. The greatest difficulty in calculating the temperature regime is presented by allowing for the effect of natural convection. This problem can be solved only by using experimental data. In the limiting case of the absence of natural convection, if we ignore end effects, we come to a two-dimensional problem of conductive–radiative heat transfer. Its solution in an exact formulation presents serious problems in connection with the exceptional awkwardness of the calculations.

To approximately solve the above problem, we will assume that the thermal conductivity of the rods is great enough so that we can assume a constant temperature about the perimeter. We will also assume that the temperature of all of the rods in one row (Fig. 1) is the same, which allows us to examine heat flow only from one row to another. These simplifications made it possible, without serious complications, to superimpose the conductive and radiative heat flows.

First we will examine the radiative heat flow. If we move in the direction of the heat flow, we find that the rods of each row i interact mainly with the rods of the row $i + 1$. The rods of row i also interact with the rods of row $i + 2$ although with considerably lower values of reciprocal surface. These three rows constitute an open system of three gray bodies. The radiative heat-transfer problem in such a system is fairly complex. We will therefore introduce a simplification which consists mainly of the assumption that the row $i + 1$ can be approximately regarded as a shield between the i-th and $i + 2$nd rows. Supposing that the shield reduces heat flux by a factor of two, instead of examining the interaction between the i-th and $i + 2$nd rows we will examine the interaction between the i-th and $i + 1$st rows, but we will have increased the reciprocal surface $H_{i,i+2}$ by a factor of two and added it to the reciprocal surface $H_{i,i+1}$. This permits us to reduce the problem to examination of the interaction of two gray bodies. Using the familiar relations from [1] to solve this problem, we obtain the following formula for the radiant heat flow between the rows for the condition of equality of the emissivities of the radiating surfaces:

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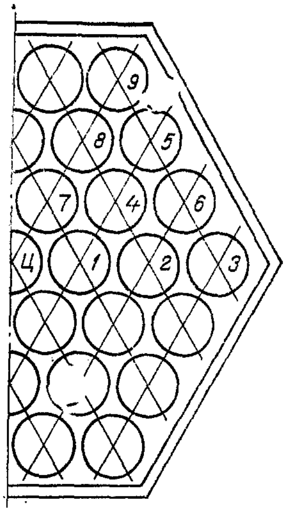

Fig. 1. Cross section of a fuel assembly, model No. 1.

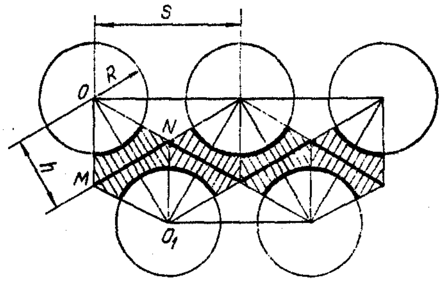

Fig. 2. Diagram of conductive heat transfer from row to row.

$$Q^{p}_{i,i+1} = \frac{\sigma_0 \varepsilon^2 H^{\mathrm{mm}}_{i,i+1} (T_i^4 - T_{i+1}^4)}{1 - (1-\varepsilon)^2 \varphi_{i,i+1} \varphi_{i+1,i}}, \quad (1)$$

where

$$\varphi_{i,i+1} = \frac{H_{i,i+1}}{S_i}; \quad \varphi_{i+1,i} = \frac{H_{i,i+1}}{S_{i+1}};$$

$$H^{\mathrm{mm}}_{i,i+1} = H_{i,i+1} + 2H_{i,i+2}.$$

We will use the following approach to analyze the conductive component of the heat flow. Heat is transmitted from row to row through a system of "bridges" (Fig. 2). If we know the thermal resistance of each bridge, then, assuming a constant temperature for the rods located in one row, we have the following for the thermal resistance between the rows

$$R_{\mathrm{b}} = \frac{r_{\mathrm{b}}}{N}, \quad (2)$$

where N is the number of bridges. To determine r_{m} we will use a method which can be called the method of temperature-field construction. The essence of the method is that a system of lines of heat flow for some simpler problem is situated as a whole or in part in the region where it is necessary to find a solution. We then find its integral characteristics.

For the case being examined, it is convenient to use the solution to the problem of a pipe embedded in a semiinfinite mass [2]. This solution basically also gives us the solution to the problem of the temperature field between two parallel pipes having different temperatures. It is described by the equation

$$\frac{t - t_2}{t_1 - t_2} = \frac{\ln \dfrac{x^2 + (y + \sqrt{h^2 - R^2})^2}{x^2 + (y - \sqrt{h^2 - R^2})^2}}{2 \ln \left(\dfrac{h}{R} + \sqrt{\left(\dfrac{h}{R}\right)^2 - 1} \right)}. \quad (3)$$

Figure 2 explains the quantities in Eq. (3).

If we limit the heat-transfer section to the segment (—S/2; S/2), then for 1 lin. m of length

$$Q_s^{\mathrm{t}} = \frac{4\lambda \Delta t \,\mathrm{arc\,tg}\, \dfrac{S}{2\sqrt{h^2 - R^2}}}{\ln \left(\dfrac{h}{R} + \sqrt{\left(\dfrac{h}{R}\right)^2 - 1} \right)}, \quad (4)$$

which corresponds to heat transfer from part of the perimeter. The technique used here obviously gives a more accurate result, the closer the ratio h/R is to unity (however, it must be remembered that when h/R = 1 Eqs. (3) and (4) lose meaning).

This adequate accuracy of Eq. (3) is evidenced by the fact that even with h/R = 4 and S/2R = 1.4 the error of Eq. (3) is no greater than 12%.

No problems are caused by the use of Eq. (4) to evaluate the thermal resistance between the outermost row of rods and the shell. In the transition from one row to another, it is evident from Fig. 1 that the thermal resistance of each bridge can be evaluated as double the thermal resistance calculated from Eq. (4), where S = MN.

Finally, the heat flow between rows is expressed by the formula

$$Q_{i,i+1} = \frac{N}{r_b}(T_i - T_{i+1}) + \frac{\sigma_0 \varepsilon^2 H^{mm}_{i,i+1}(T_i^4 - T_{i+1}^4)}{1-(1-\varepsilon)^2 \varphi_{i,i+1}\varphi_{i+1,i}}. \quad (5)$$

Equation (5) can be simplified considerably if we consider that within the temperature range of interest the heat conductivity of air can, with an accuracy sufficient for practical purposes, be represented in the form

$$\lambda = aT, \quad (6)$$

where a is constant. Then in calculating heat transfer between rows i and i + 1 we write the thermal conductivity

$$\lambda = a\,\frac{T_i + T_{i+1}}{2}. \quad (7)$$

Equation (5) takes the form

$$Q_{i,i+1} = A_i(T_i^2 - T_{i+1}^2) + B_i(T_i^4 - T_{i+1}^4), \quad (8)$$

where A_i and B_i are numerical coefficients.

If one of the temperatures is known, then we have on equation which is biquadratic relative to the order. In the case where approximation (6) is not accurate enough, the solution of Eq. (8) gives a first approximation for the sought temperature. If the temperature of the shell is given, successive solution of equations of type (8) gives the temperatures of all of the rows. The resulting solution is valid when convection is absent.

The boundaries of applicability of the approach used above can be determined only with the use of empirical data. Any nonunidimensional stratification is unstable [3] and, thus, causes convective flows. However, the theory of stability says nothing about the intensity of these motions. Since temperature is measured in experiments, then the presence of convection can be judged from distortion of the temperature field. Thus, the boundary between the radiative—convective and radiative—convective—conductive regimes was determined from deformation of the temperature field.

Tests were conducted on two models. Model No. 1 was a vertical bundle consisting of 37 stainless tubes 10 × 1 in diameter. The tubes were arranged in hexagonal fashion with a relative spacing $\sigma = 1.4$. The bundle was enclosed in a hexagonal shell with a dimension of 89 mm "under the key." The bundle was heated by passing an electric current directly through the walls of the tubes. The length of the heat-releasing part was 1270 mm. The central tube was not heated. The model had a removable jacket with a length L = 1300 mm. The jacket nearly coincided with the heat-releasing length of the heated tubes. The pumping of water through the jacket ensured isothermal conditions on the surface of the shell. The temperature of the coolant water was measured with laboratory thermometers with graduations of 0.1°C. Water flow rate was determined by the volumetric method.

Eleven thermocouples were embedded in the model over its height on one of the generatrices of the body to determine its temperature.

Air was used as the heat carrier in the tests. The set-up allowed us to increase air pressure in the model above the atmospheric level by supplying compressed air from a receiver or to obtain a negative pressure by means of a fore pump.

To measure the temperature distribution across the working section, we picked out a characteristic sector consisting of 10 tubes (see Fig. 1). Forty Chromel—Alumel thermocouples with ϕ 0.2 mm were installed at four levels over the height of the bundle. The heads of the thermocouples were welded to the inside surfaces of the walls of the heat-releasing tubes in the bundle. The ends of the thermocouples were brought outside through a top cut in the tubes. This cut was tightly sealed about the thermocouple to prevent air from circulating inside the tubes. The beginning of a steady-state thermal regime was determined

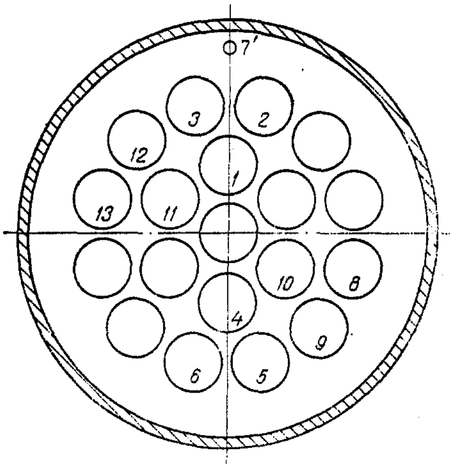

Fig. 3. Cross section of heat-releasing assembly, model No. 2.

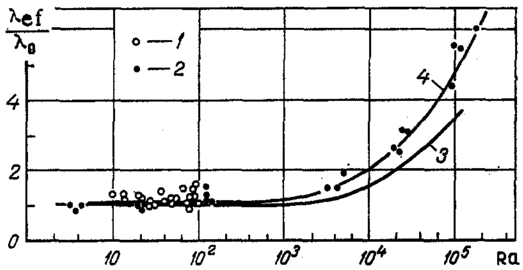

Fig. 4. Effective thermal conductivity with free air movement in a rod bundle: 1) empirical points, model No. 1; 2) same, model No. 2; 3) [5]; 4) curve averaging our data.

from the readings of 24 thermocouples connected to an ÉPP-09 potentiometer. The readings of the thermocouples during this regime were measured with an R-330 potentiometer.

Model No. 2 was a vertical bundles made up of 19 tubes with a diameter of 13.5 × 1. The tubes were arranged in a mixed pattern (Fig. 3). Six of the tubes were installed around a central (unheated) equilateral hexagon with a spacing of 16 mm. The remaining 12 tubes were positioned uniformly in a circle 62 mm in diameter. The bundle was placed in a cylindrical shell 98 mm in diameter. The tubes were heated by the direct passage of an electrical current. The length of the heat-releasing section was 2845 mm. The tubes were fixed in place by spacing grates with a low hydraulic resistance. The grates themselves were spaced 600 mm apart. The shell was cooled with water. Forty-four thermocouples were installed on the tubes at eight levels over their height. A hole was drilled in the tube wall at the site of thermcouple installation and the thermocouple was led inside the tube. The head of each thermocouple was left outside and welded to the outer surface of the tube. The rest of the hole was then sealed. To check the uniformity of the temperature distribution across the tubes, some thermocouples were oriented toward the inside of the tube and some toward the outside. The air temperature in the gap between the body of the bundle and the outside row of tubes was measured with seven thermocouples. The change in air pressure inside the bundle was accomplished in the same manner as on model No. 1.

The tests were begun by studying the temperature field with negative pressure. The idea behind these experiments was as follows. The thermal conductivity of air under these conditions did not depend on pressure up until the attainment of negative pressures of the order of 1 mm Hg. On the other hand, the Rayleigh number, characterizing free convection, changed in proportion to the square of the pressure. Thus, with pressures on the order of 2-3 mm Hg in the test section, we had radiative-conductive heat transfer, which allowed us to check the calculating method. Comparison of the experimental values of mean temperature among the rows with calculated values for tests with negative pressure showed good agreement (deviation no greater than 6-7%).

In further analyzing the experimental data obtained, we should note that the effect of free motion on heat transfer may be manifest in two ways. One of the mechanisms of this effect is the formation of a circulation loop covering the test section. As the pressure increases, the temperatures of the different rows approach one another and increase linearly over the height. This development has to do with the development of a lifting motion in the central part of the bundle. End effects result in a temperature maximum in the upper part of the heated tubes. Besides this, small-scale motion may develop. The effect of the latter is similar to an increase in thermal conductivity and is described by the so-called effective thermal conductivity [4].

The literature [5, 6] contains relations for calculating the effective thermal conductivity for the simplest geometric configurations — a two-dimensional gap, a cylinderical configuration, etc. These relations are usually represented in the form

$$\frac{\lambda_{ef}}{\lambda} = f(\mathrm{Ra}) = f\left(\frac{g\beta\delta^3\Delta t}{\nu a}\right). \tag{9}$$

There are no recommendations for more complicated geometric configurations. The completed tests showed that it is possible to obtain such recommendations for heat transfer between rows of tubes and from a row of tubes to the shell. The premise here is as follows. At pressures close to atmospheric, the temperature maximum over the height of the heat-releasing tubes is not pronounced and is located far from the upper chamber. This allows us to suggest that heat is moved out of this zone mainly in the radial direction. Using Eq. (1), we can pick out the radiative component from the total heat flow. Let there be two regimes with similar heat releases: one for the case of negative pressure, the other for a pressure $p \geqslant 0.1$ MPa. By determining the gradients between rows, we arrive at the following proportion:

$$\frac{(Q-Q^{\mathrm{p}})_{\mathrm{p}}}{(Q-Q^{\mathrm{p}})_0} = \frac{\lambda_{ef}\Delta t_{\mathrm{p}}}{\lambda_0 \Delta t_0} K_t, \tag{10}$$

where the subscript 0 pertains to the negative-pressure regime and the index p pertains to the regime with the other pressure; K_t is a correction characterizing the difference in mean temperatures in the gap in question for the cases being compared. It follows from (10) that:

$$\frac{\lambda_{ef}}{\lambda} = \frac{(Q-Q^{\mathrm{p}})_{\mathrm{p}}\,\Delta t_0}{(Q-Q^{\mathrm{p}})_0\,\Delta t_{\mathrm{p}} K_t}. \tag{11}$$

The method associated with the use of Eq. (11) is accurate enough if only Q^{p} is significantly less than Q. This condition was satisfied in our experiments, since the emissivity of the surface of the tubes and shell was not great — $\varepsilon = 0.35$

The ratio λ_{ef}/λ_0 was determined for those regimes where it was more or less possible to clearly distinguish the zone of radiative heat transfer. We took the minimum gap between the tubes or between the tubes and the shell as the characteristic dimension in the Rayleigh number when we generalized the data obtained here. The resulting values of λ_{ef}/λ_0 are shown in Fig. 4. Also shown is curve 3 from [5]. The data obtained here lies above this curve, which should be expected from the selection of a linear dimension in the Rayleigh number. Using a certain mean characteristic gap δ_{av} instead of the quantity δ_{min}, we could obtain agreement with curve 3 — which generalizes data for planar and cylindrical gaps. However, the use of the minimum gap value is convenient from a practical point of view. The data obtained here makes it possible to construct a theoretical model to analytically determine the temperature field in bundles of heat-releasing rods under conditions whereby the effect of natural convection is substantial.

NOTATION

Q, heat flux; λ, thermal conductivity; t, T, temperature, δ, gap size; σ, relative spacing; r, thermal resistance; H, reciprocal surface of two radiating bodies; ε, emissivity of the surface; φ, radiation factor. Indices: i, number of row of rods in bundle; b, bridge; ef, effective; av, mean value; min, minimum.

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EXPERIMENTAL STUDY OF THE PROCESS OF THE FILLING OF A CONTAINER WITH COMPRESSED AIR

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UDC 533.17

Certain laws governing the process of the filling of a container with compressed air are determined experimentally.

In engineering practice it is sometimes necessary to fill containers with compressed air. This is usually done from stationary high-pressure accumulators or by means of a compressor. Complex gasdynamic and thermal processes take place in the container being filled, and these processes are accompanied by an increase in the temperature of the gas. The amount of this heating depends on the heat exchange between the gas and the wall of the container, as well as on many other factors.

As the test container 3 (Fig. 1) we used a standard cylinder having a length of 1.33 m. The outside diameter of the cylindrical part was 0.226 m, while the wall thickness was 0.008 m. The cylinder was made of steel 5. The working gas was air, which was admitted through a headpiece 4 from an air ramp under a constant pressure of 8-15 MPa. The axis of the headpiece coincided with the axis of the cylinder. We used two types of headpieces. The first took the form of sonic nozzles with a minimum diameter d_0 equal to 2, 3, and 5 mm. The gas passed through these nozzles into the container in the form of an axisymmetric jet. The second type was also a sonic nozzle, but here a flat, circular screen was positioned normally with respect to its axis. The flow was deflected in its interaction with the screen and flowed into the container in the form of a V-shaped stream. The test unit permitted the container to be oriented different ways: horizontally, vertically, etc.

We measured the following parameters during the tests: the temperature of the gas and the temperature of the outer surface of the wall at different stations along the cylinder, the gas pressure in the container and the ramp, and the total pressure and temperature of the gas ahead of the headpiece. The temperature was measured with Chromel–Alumel thermocouples with wires 0.3 mm in diameter. Pressure in the container was measured with an MD 400T potentiometric transducer. The recording element in the measurement system was an N-700 light-ray oscillograph equipped with M001-1 galvanometers. The transducers for measuring gas temperature and pressure in the container were installed at three stations located distances from the headpiece edge x = 0.15, 0.65, and 1.15 m, respectively. The transducers for measuring the temperature of the outside surface was placed at six stations x = 0.11, 0.37, 0.49, 0.62, 0.84, and 1.18 m.

The experiment was conducted in the following sequence. After the recording equipment was turned on, we injected air into the container through electric valve 1. When the pressure in the container reached the pressure value in the ramp, the injection main was cut off by the same valve. The parameters of the gas in the container were subsequently measured

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