Study of unsteady thermal conductivity of a package consisting of one layer of equal-size balls

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Abstract. The study of unsteady heat transfer of a package consisting of the equal-size balls was performed at boundary conditions of the first kind. The package was located on a horizontal surface and was limited by the walls of a cylindrical thermostat 129 mm in diameter. The experiments were carried out using glass balls with a diameter of 8.85 mm, 16.0 mm, as well as polyoxymethylene balls with a diameter of 9.0 mm. The package was filled with water. The height of the water layer was equal to the diameter of the balls. In the experiments, the time was determined during which the ball package was heated or cooled to the temperature of the thermostat. It is shown that the time of temperature equalization between the package and the thermostat depends not only on the material and size of the balls but also on the direction of the heat flux. It is shown that the heating time of one layer of equal-size balls is much shorter than the cooling time.

1. Introduction
Spherical layers are widely used in geology, oil and gas production, energy, various industrial systems, such as gas-cooled nuclear reactors, drying processes, catalytic reactors, which include studies of the chemical-technological processes of the catalytic conversion of constituent gases or liquids on solid surfaces of catalysts, as well as processes involving transpiration cooling, and in highly efficient cryogenic isolation. Since most of these processes are related to energy transfer, the correct knowledge of the thermal properties of materials is necessary for the proper design of such thermal systems. Effective thermal conductivity is the general heat transfer characteristic of a packed layer of balls and should be taken into account when analyzing and designing various systems and devices. This information is of particular importance in the development and operation of high-temperature gas-cooled reactors with a granular layer [1, 2]. Besides, a thorough knowledge of the porous structure in the granular layer is important for modeling the effective thermal conductivity in the forced flow of fluid through a ball bed, where there is a problem of correctly determining the molecular component of the effective thermal conductivity of the bed [3-5]. In the framework of the “homogeneous approximation”, the classical Fourier law uses the effective thermal conductivity to describe heat transfer in the granular layer $\lambda^{\text{ef}}$. Its value is determined by two factors: 1) the geometry of the granular layer, which determines the spatial configuration of the heat flux field in the pore space; 2) convective heat transfer caused by the deviation of the local velocity of the mobile phase from the average value and additional mixing of the liquid. Traditionally, $\lambda^{\text{ef}}$ is written as:

$$\lambda^{\text{ef}} = \lambda^{\text{ef}}_s + \lambda^{\text{ef}}_k,$$

(1)
where \( \lambda^e_k \) is the effective thermal conductivity in the absence of filtration, and the convective component \( \lambda^c_k \) is written as

\[
\lambda^c_k = \rho \ \varepsilon \ \mathcal{C}_f.
\]

Here \( \rho \) and \( \mathcal{C}_f \) are the density and specific heat of the mobile phase, \( \varepsilon = b_U d \) is the mixing coefficient, \( U \) is the filtration rate, \( d \) is the diameter of the ball, \( b_U \) is the numerical constant, which for most practical tasks can be taken as 0.1 [6].

The widespread use of various composite and polymeric materials leads to the need to evaluate the effective thermal conductivity of these compositions based on polymer binders (silicone, polyurethane, epoxy resin) with powder heat-conducting dielectric fillers (quartz, corundum, aluminum nitride, silicon carbide, and their pair combinations). The possibility of using linear models to approximate estimates of the effective thermal conductivity of ternary mixtures for a wide range of powder filler contents was shown in [7]. In [8], models of effective thermal conductivity and filtration are considered, for example, in a catalytic medium formed by weakly or unconsolidated particles with two-scale porosity, whose inner space consists of pores communicating with channels. In terms of heat transfer, a porous medium can be represented as a medium with some effective thermal conductivity, depending on the structure of its internal space. In this case, the liquid or gas in the pore space can be considered motionless. In this case, the catalytic medium can be considered as a two-phase medium for which one can find the effective thermal conductivity. Further, the thermal problem can be considered in a homogeneous medium with the corresponding effective thermal conductivity. The correctness of this approach is violated if the liquid or gas undergoes natural convection, which must be taken into account when constructing the effective thermal conductivity coefficient.

In the general case, heating or cooling of bodies is accompanied by a continuous change in temperature inside these bodies and on their surfaces and, due to its physical nature, is associated with a change in heat content. Since the rate of change in heat content is directly proportional to the ability of the material to conduct heat, i.e., the heat conductivity coefficient \( \lambda \) and is inversely proportional to its accumulating capacity, i.e. volumetric heat capacity \( \mathcal{C}_p \), the rate of heating or cooling of the body under unsteady conditions is generally determined by the thermal diffusivity: \( a = \lambda/(C_p) \).

The study of unsteady heat conduction patterns in a monolayer package of balls of the same diameter filled with water without forced convection is the objective of the present research.

2. Experimental part

The experiments were carried out using the setup shown in Figure 1. To create packages \( l \), balls of the equal diameter were used, which were laid in one layer on a 0.5 mm thick fluoroplastic plate. The plate was mounted horizontally with an accuracy of 0.2 mm/m, which corresponds to 0.03 mm per plate diameter. A layer of balls was placed inside a stainless steel thermostat 2 with an inner cylinder diameter of 129.4 mm. The balls were poured with distilled water 3 0.1 mm above the filling from the balls. Water was poured from the measuring vessel; the volume of water being poured was controlled with an accuracy of 0.2 ml. The density of water was controlled by hydrometer with an accuracy of 1 g/ml. To measure the surface temperature of the balls package, a NEC TH7100 thermal imager 4 was used. To increase the accuracy of temperature measurement by reducing external thermal radiation, the heat-reflecting screens 5 made of polished aluminum with central holes of different diameters were installed between the thermal imager lens and the package.

Before filling with water, the package of the balls was photographed and the geometric characteristics of the package were determined from the photographs. The mass of the balls \( m \) used in the package was determined on Vibra 6200 electronic balance with an accuracy of 10 mg. The total volume of balls \( V \) was measured in a measuring cup with an accuracy of 0.2 ml.

The heat capacity of water \( C_w(T) \) was taken from the reference Tables [10], and the heat capacity of the balls \( \mathcal{C}_b \) was determined with an accuracy of ~ 2.5% relative to the heat capacity of water. For this, all balls prepared for the package were heated in a water bath to a temperature of \( T_1 \) of about 60°C, and then cooled in a Dewar vessel in the water with a temperature of \( T_2 \) of about 3°C. The mass of
water in the Dewar vessel was approximately two times the mass of balls $M_b$. Water was cooled by adding small pieces of ice to a level of less than 3°C. The total weight of the Dewar vessel with water was controlled to determine the mass of water $M_w$ for cooling. After placing the heated balls in water, weighing was also carried out to determine the amount of water $M_c$, which was attached to the balls during the transfer from the water bath. The heat capacity was calculated from the heat balance (3) when the equilibrium temperature $T_3$ was reached, assuming the heat capacity of the balls to be constant within the temperature range from 3 to 60°C.

$$C_b \cdot (T_1 - T_3) \cdot M_b = \left[ C_w \cdot (T_2 - T_1) \cdot T_2 \right] \cdot M_w - \left[ C_w \cdot (T_3 - T_1) \cdot T_3 \right] \cdot M_c.$$

(3)

Figure 1. Schematic diagram of the experimental setup: 1 – a single layer of balls, 2 – water thermostat, 3 – distilled water, 4 – IR camera, 5 – heat-reflecting screens.

To determine the temperature with a thermal imager, for each type of surface to be measured, the surface blackness coefficient $\varepsilon$ was set in the hardware. The value was determined from calibration experiments in which thermal imaging of surfaces with a known temperature of 0°C and 41°C was carried out with a set value of the apparatus value $\varepsilon$ equal to unity. When processing thermograms with Replay Thermal Image ver. 1.21 (TH71RX) from NEC San-ei with an accuracy of 0.01, the $\varepsilon$ value was determined at which the hardware temperature value corresponded to the measured temperature. Figure 2 shows the experimental data for the hardware determination of the surface blackness coefficient $\varepsilon$ obtained from the processing of thermal imaging. During the experiments, the following values of $\varepsilon$ were used: 0.96 - for water, 0.89 - for glass balls, and 0.86 - for balls made of polyoxymethylene.

The data obtained during the processing of thermograms with a change in temperature as a function of time $t$ in the center of the package were compared with the dependence $f(t)$ to determine the characteristic time $\tau$ (determination accuracy $\tau$ was 2%). Here $T_0$ is package temperature, $T_1$ is thermostat temperature. The countdown of time $t$ begins after a coolant with temperature $T_1$ is supplied to the thermostat (taking into account the time of heating the thermostat walls for 10-20 seconds):

$$f(t) = T_0 + (T_1 - T_0) \cdot \text{th} \left( \frac{t}{\tau} \right).$$

(4)
Figure 2. Experimental data for the hardware determination of the surface blackness coefficient $\varepsilon$ for the NEC TH7100 thermal imager: 1 – water, 2 – polyoxymethylene, 3 – glass S-FSL5.

Three types of packages “PB-1” (balls made of glass S-FSL5 with a diameter of 16.00 mm), “PB-2” (balls made of glass S-FSL5 with a diameter of 8.85 mm), and “PB-3” (balls made of polyoxymethylene (POM) with a diameter of 9.00 mm). Parameters and thermophysical properties of packages include the following: $D$ is the diameter of balls, $N$ is the number of balls in the package, $V$ is the volume of all balls, $M$ is the mass of balls, $\rho$ is the density of balls, $C$ is the specific heat capacity of balls, $C \cdot M$ is the heat capacity of balls (measured), $\lambda$ is the thermal conductivity [9], $h_{\text{water}}$ is the height of the water layer above the fluoroplastic plate, $M_{\text{water}}$ is the mass of water [10], $V_{\text{packed}}$ is the packing volume, $M_{\text{packed}}$ is the packing mass (ball mass plus water mass), $\varepsilon$ is the ratio of the ball volume to the package volume, $C \cdot M_{\text{packed}}$ is the heat capacity of the package, $\lambda_{\text{packed}}$ is the effective thermal conductivity of the package [2], $S_h$ is the area of the heat-exchange section of the thermostat with the package, shown in Table 1.

After the liquid was supplied to the thermostat, the constant temperature was established on the surface of the thermostat's inner cylinder for 20-30 seconds. During the experiments, the wall temperature was maintained constant with an accuracy of $\pm 0.2^\circ$C.

Table 1. Parameters and thermophysical properties of the package at a temperature of 22°C.

|     | "PB-1" | "PB-2" | "PB-3" |
|-----|--------|--------|--------|
| $D$ mm | 16.0   | 8.85   | 9.0    |
| $N$   | 52     | 172    | 165    |
| $V$ sm$^3$ | 111.5 | 62.4   | 63.0   |
| $M$ g    | 243.90 | 137.58 | 89.18  |
| $\rho$ kg/m$^3$ | 2187  | 2204   | 1416   |
| $C$ J/(kg·K) | 855   | 878    | 1466   |
| $C \cdot M$ J/K | 208.53 | 120.80 | 130.74 |
| $\lambda$ W/(m·K) | 0.740  | 0.740  | 0.310  |
| $h_{\text{water}}$ mm | 16.1   | 8.95   | 9.1    |
| $M_{\text{water}}$ g | 100.01 | 55.17  | 56.58  |
| $V_{\text{packed}}$ sm$^3$ | 211.7  | 117.8  | 119.7  |
| $M_{\text{packed}}$ g | 343.91 | 192.75 | 145.76 |
| $\varepsilon$ - | 0.527  | 0.530  | 0.526  |
| $C \cdot M_{\text{packed}}$ J/K | 627.27 | 351.78 | 367.64 |
| $\lambda_{\text{packed}}$ W/(m·K) | 0.669  | 0.666  | 0.447  |
| $S_h$ m$^2$ | $6.55 \cdot 10^{-3}$ | $3.64 \cdot 10^{-3}$ | $3.70 \cdot 10^{-3}$ |
3. Results and Discussion
The heat-conducting characteristics of one layer of equal-size balls in a stationary liquid were studied in the heating and cooling mode of the layer with a stepwise change in temperature on the external perimeter of the package.

The study showed different types of peculiarities in package heating. The heating time very much depends on the direction of the heat flux on the thermostat wall, namely: when heating, the heating time is much shorter than when cooling. Figure 3 shows the temperature change in the center of the package versus time (icons), and the curves show the dependence on the characteristic time $\tau$. Data are provided for cases where the temperature difference for heating and cooling is the same. It can be seen from the figure that during heating, the characteristic time $\tau$ is 145-400 sec and depends on the thermal conductivity of the package $\lambda$. ($1 - \lambda = 0.31 \text{ W/(m·K)},\ 3 - \lambda = 0.74 \text{ W/(m·K)},\ 5 - \lambda = 0.74 \text{ W/(m·K)}$). During cooling, the $\tau$ varies from 600 to 1600 sec and depends on the diameter of the ball (curves 4 and 6) and is much weaker depending on the thermal conductivity of the balls (curves 2 and 6). Thus, the strong influence of the heat flux direction on the time of alignment of the package temperature with the temperature of the thermostat was revealed. The cooling time of the package significantly exceeds the time during which the package warms up, provided that the initial value of the temperature difference between the package and the thermostat is the same. One can see that this difference is several times (3-7 times).

Similar differences and effects in the intensity of heat transfer during cooling and heating in pipes were indicated in forced convection [11] that was associated with the temperature dependence of the physical properties of the liquid. A change in temperature over the cross-section leads to a change in viscosity. At that, the greater the temperature difference, the stronger is the change of viscosity and other physical parameters (heat capacity, thermal conductivity) over the cross-section of the pipe. A change in viscosity leads to a change in the velocity profile, which affects the intensity of heat transfer. Depending on the direction of the heat flux, the change in the velocity profile is different. When a liquid is cooled, its temperature near the wall is lower and its viscosity is higher than in the center of the pipe. When heating the liquid, on the contrary, the flow rate of the liquid near the wall is higher and lower in the center of the pipe. In this particular case, the fluid is stationary and fills the entire pore space between the balls.

![Figure 3](image-url)

**Figure 3.** Temperature change in the center of the package filled with balls and water: 1 – "PB3" (heating), 2 – "PB3" (cooling), 3 – "PB1" (heating), 4 – "PB1" (cooling), 5 – "PB2" (heating), 6 – "PB2" (cooling).
On the other hand, experiments with the dry package without liquid during heating and cooling showed that the measured times $\tau$ for heating and cooling the packs of balls coincide within the experimental error, and it was also found that the direction of the heat flux does not affect the rate of temperature change. Figure 4 shows data on changes in the temperature of the package in the air. In this case, the time is determined for the ball located near the center of the package. The position of such a ball is shown for each case on frames of the thermogram (at $t=1050$ s). The deviation from the package center for "PB3" is 4.02 mm (points 1, 2 in Figure 4), for "PB2" - 10.91 mm (points 3, 4 in Figure 4), "PB1" - 4.77 mm (points 5, 6 in Figure 4).

Figure 4. Change in the surface temperature of the ball closest to the package center. Thermograms indicating the position of the measured ball: 1 – "PB3" (heating), 2 – "PB3" (cooling), 3 – "PB1" (heating), 4 – "PB1" (cooling), 5 – "PB2" (heating), 6 – "PB2" (cooling).

It should be noted that similar experiments with a 9.1 mm and 16.1 mm thick water layer without backfill showed that for water, the times $\tau$ during heating and cooling differ, but not as much as for packages, the difference in cooling and heating times does not exceed 25%. The data are shown in Figure 5. For levels of 9.1 and 16.1 mm, no significant differences were noticed; the differences do not exceed the measurement error.
Figure 5. Temperature change in the center of water layer with the height of $h$ 9.1 mm and 16.1 mm: 1 – $h = 9$ mm (heating), 2 – $h = 9$ mm (cooling), 3 – $h = 16$ mm (heating), 4 – $h = 16$ mm (cooling).

Conclusions
The study showed that in the case where the liquid medium between the balls is stationary, the heat transfer from the wall to the package is characterized by certain peculiarities depending on the direction of the heat flux. The main assumption is associated with the influence of convective heat transfer in the liquid phase of the package. Moreover, the structure of inter-ball volumes filled with a liquid significantly affects the development of convective heat transfer in a liquid, since the influence of the direction of the heat flux increases significantly in the package. To fully understand the features of heat transfer in the packing of a monolayer of balls in a stationary fluid, it is necessary to compare the experimental results with the numerical simulations of heat transfer in the appropriate conditions. In particular, it is very important to determine the effect of convective flows in the air on the package heating rate.

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References
[1] De Beer M, Du Toit C G, Rousseau P G 2017 Nuclear Engineering and Design 314(1) 67
[2] Van Antwerpen W, Du Toit C G and Rousseau P G 2010 Nuclear Engineering and Design 240(7) 1803
[3] Dekhtyar R A, Sikovsky D Ph, Gorine A V, Mukhin V A 2002 High Temperature 40(5) 693
[4] Avdeev A A, Zudin Yu B, Balunov B F and Rybin R A 2009 High Temperature 47(5) 692
[5] Avdeev A A, Balunov B F, Zudin Yu B 2006 Proceedings of the Russian National Conference on Heat Transfer-4 6 33
[6] Wakao N, Kaguei S 1982 N-Y: Gordon and Breach Science Publishers 364
[7] Mikheev V A, Sulaberidze V Sh, Mushenko V D 2017 Izv. Universities. Instrumentation 60(3) 275
[8] Lyubchik L M, Bed' E V 2013 Visnik NTU 977(3) 141
[9] Warlimont H, Martiennssen W 2018 Handbook of Materials Data 2nd Edition (Springer) p 1140.
[10] Rivkin S L and Aleksandrov A A 1980 Thermophysical Properties of Water and Steam (M: Energy Press) p 424
[11] Mikheev M A, Mikheeva I M 1977 Heat Transfer Fundamentals, 2nd Ed. (M.: Energy) p 344