The regenerative heat exchanger with periodic veering of the flow

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Abstract. In the given work numerical simulating of the regenerative heat exchanger with a cyclic flow veering is executed in application to the ventilation system. The heat exchanger contains the grained medium consisting of solid spherical elements, storing heat. During a half cycle the entering cold air passes through the grained medium, heating it up to the room temperature. Then the air flow direction is reversed and during another half cycle the exhaust warm air refunds heat to the grained medium. Unlike the former researches, the case of low thermal diffusivity of spherical elements, when heat transfer is determined by the properties of a grained medium, is investigated. The mathematical model describing a cyclic mode of heat exchanger operation is developed. The basic criteria determining heat transfer and their influence on heat regeneration efficiency are revealed.

1. Introduction

When air is ventilated in residential buildings during the winter period, heat losses, which can reach 50% of the total energy balance, occur. One of the most promising solutions to the problem of energy saving is utilization of the exhaust air heat to heat the incoming air. The schematic diagram of the regenerative heat exchanger in the ventilation system is quite simple. The exhaust warm air from the room passes through a heat-accumulating charge (fixed bed of granulated material), which absorbs a part of the thermal energy of air. Then the air flow direction is changed, and the incoming cold air passes through the heat-accumulating charge, receiving heat from it. Heat transfer between gas and granular layer occurs during the passage of gas through the channels formed by solid particles (elements of the charge). The characteristics of regenerators with heat exchange matrices in the form of charges of various materials are discussed in [1, 2]. The regimes associated with complete single heating (or cooling) of the heat-accumulating layer are mainly considered in literature. At that, there are large discrepancies on the heat transfer coefficient between the data of different authors, which is associated with a complex, poorly reproducible structure of the granular medium. Non-stationary heat transfer in a cyclic reversible air flow through a stationary heat-accumulating charge was experimentally studied in [2]. Lead balls with a diameter of 2–4.5 mm and glass balls with a diameter of 3.2 mm were used as the charge elements. The results of experiments were compared with calculations by the model of [3]. It is shown that good agreement between the experimental data and calculations is observed only for the large Reynolds numbers. The results of experimental and numerical study of the thermal parameters of an air-to-air regenerative heat exchanger with a periodic change in the air flow direction, where a polypropylene matrix with longitudinal air channels is used as a heat exchange body, are presented in [4]. The developed mathematical model of the regenerative
heat exchanger is described and possibility of its application for optimization of operating and design parameters of the heat exchanger is demonstrated. It is assumed in theoretical models [3, 4] that the temperature of the charge element is uniform in volume, and thermal resistance is concentrated in the gas phase. In this work, operation of a regenerative heat exchanger with cyclic flow reverse and heat-accumulating charge of spherical elements was simulated numerically. In contrast to the models of [3, 4], a charge of a material with low thermal diffusivity is considered here. In this case, heat exchange between gas and charge is controlled by distribution of heat in the solid phase.

2. Theoretical model

The problem of heat transfer between the granular charge and gas flow was solved in the following simplest one-dimensional formulation. Let us consider a granular medium with volumetric porosity \( \phi \) consisting of solid spherical granules with radius \( R \). The gas flow moves through this medium at constant velocity \( U \) and exchanges heat with the granules (figure 1). Gas temperature \( T_g(x,t) \) in elementary volume \( dV=Fdx \) (\( F \) is the cross-sectional area) is determined from the equation of thermal energy for gas

\[
\rho_g c_g \frac{\partial T_g}{\partial t} + U \frac{\partial T_g}{\partial x} \varphi \cdot dV = q(x,t) dS_g .
\]  
(1)

Here \( x \) is coordinate in the direction of gas motion, \( \rho_g \) and \( c_g \) are density and specific heat capacity of gas, \( dS_g \) is area of granule surface surrounded by gas, \( q(x,t) \) is density of heat flux to gas on the granule surface.

Thermal diffusivity of a gas \( a_{gas} \) is much greater than that of a solid \( a_s \) and the characteristic size of the gas-filled gaps \( \delta = 0.2 \ R \) is much less than radius of grains. From here follows, that the characteristic time of the stabilization of uniform temperature in gas is much less, than that in solid grain (\( \delta^2 / a_{gas} \ll R^2 / a_s \)). Thus on a scale of \( \delta \) the gas can be considered temperature-uniform, and the surface temperature of solid granules is equal to the gas temperature. The temperature distribution inside the granule determines the heat flux on its surface:

\[
\frac{\partial T_s}{\partial t} = a_s \frac{\partial^2 T_s}{\partial r^2} - \frac{\lambda_s}{\rho_s c_s} \left[ \frac{\partial T_g}{\partial r} \right]_{r=R} .
\]  
(3)

![Figure 1. Scheme of regenerative heat exchanger with periodically veering of flow. Air inward flow gains heat from the grained medium (a). The exhaust air flow refunds heat to the grained medium (b).](image-url)

Temperature distribution \( T_s(x,r,t) \) inside a spherical granule is described by equation

\[
\frac{\partial T_s}{\partial t} = a_s \frac{\partial^2 T_s}{\partial r^2} - \frac{\lambda_s}{\rho_s c_s} \left[ \frac{\partial T_g}{\partial r} \right]_{r=R} .
\]  
(4)
with boundary conditions \( \frac{\partial T_s}{\partial r} |_{r=R} = 0 \), \( T_g \) is thermal diffusivity of granule material. The system of equations (3), (4) describes heat transfer between the granular medium and gas, passing through this medium. Then, we apply the developed model to simulate operation of a device for heat recovery in a ventilation system with periodic veering of the air flow direction.

3. Problem statement

The device for heat recovery in the ventilation system is a pipe of length \( L \), filled with heat-accumulating granules of a spherical shape with radius \( R \) with volumetric porosity \( \phi \). On the left end, where the origin of \( x \) axis is placed, the pipe is connected to the atmospheric air with temperature \( T_{\text{cold}} \) and the right end of the pipe is located in a room where temperature \( T_{\text{warm}} > T_{\text{cold}} \) is maintained. At the initial time, the air temperature inside the pipe and temperature of granules are \( T_{\text{warm}} \). During time period \( \tau \) (the first half of the period), the cold atmospheric air flows into the pipe with velocity \( U \) and is heated up, receiving heat from the granules. At that, the granules, of course, cool. Then, the direction of air movement is reversed (i.e., \( U \) is replaced with \( -U \)) and during the second half period the warm air from the room flows into the atmosphere, heating the granular medium. After several such cycles, the heat transfer regime with a periodic change in temperature in each pipe cross-section is stabilized. The described device can significantly reduce energy consumption for maintaining the desired temperature \( T_{\text{warm}} \) in a ventilated room, since the atmospheric air that has passed through the device and enters the room is already heated.

Let us introduce scale distance \( l \), scale time \( a t/l^2 \), and scale temperature \( \Delta T = T_{\text{warm}} - T_{\text{cold}} \), and turn to dimensionless variables \( x/l, r/l, t/a \), keeping the same letter designations for all variables. In dimensionless variables, equations (3), (4) take form

\[
\begin{align*}
\frac{\partial T_g}{\partial t} + \frac{U}{l} \frac{\partial T_g}{\partial x} &= -\left( \frac{1 - \phi}{\phi} \right) \frac{\rho_C \lambda}{R} \frac{\partial T_s}{\partial r} - \frac{\partial T_g}{\partial r} \\
\frac{\partial T_s}{\partial r} &= \left( \frac{a}{l^2} \frac{\partial^2 T_s}{\partial r^2} \right)
\end{align*}
\]

In the phase of inflow (during odd half-periods), the air temperature was set at the left end of the pipe: \( T_g(0,t) = 0 \). In the phase of exhaust (during even half-periods), the air temperature was set at the right end of the pipe: \( T_g(L,t) = 1 \).

There are two dimensionless parameters in equation (5):

\[
F_g = \frac{3(1 - \phi)}{\phi} \frac{\rho_C \lambda}{\rho_g C_g R} \quad \text{and} \quad F_e = \frac{U}{l}.
\]

Parameter \( F_g \), containing the properties of a granular medium and gas, characterizes the intensity of the thermal effect of medium on the air flow. Parameter \( F_e \) characterizes the effect of air flow on heat transfer. In addition, in the problem under consideration there is another parameter \( \Omega = L/U \), which equals the ratio of the time of air movement through the device \( L/U \) to half-period of the flow \( \tau \). Thus, parameter \( \Omega \) is the dimensionless frequency of switching the air flow direction.

4. Numerical technique

The system of equation (5), (6) was solved by the finite-difference method. Let us introduce the grids with constant step \( \Delta x = L/N \) and \( \Delta r = R/M \) by coordinates \( x \) and \( r \), where \( N \) and \( M \) are the numbers of grid cells along coordinates \( x \) and \( r \), respectively. Let us indicate the solid phase temperature at the node with coordinates \( x_i = (i - 1)\Delta x \), \( r_j = (j - 1)\Delta r \) at time \( t_n \) as \( T_{s,i,j}^n \). Correspondingly, the air temperature at node \( x_i = (i - 1)\Delta x \) is indicated as \( T_{g,i,j}^n \). Difference equation (5) in the phase of incoming flow is
The convective term on the left side of (7) is calculated on the “old” time instant \( t_n \), and the heat flux on the surface of granules on the right side of (7) is calculated on the “new” time instant \( t_{n+1} \).

In the phase of exhaust flow, difference equation (5) is

\[
\frac{T^0_{g,j}-T^n_{g,j}}{\Delta t} + F_g \frac{T^n_{g,j} - T^n_{g,j-1}}{\Delta x} = -F_g \frac{T^0_{g,j,M+1}-T^0_{g,j,M}}{\Delta r}, \quad i = 2,3,...N+1
\]

(7)

In equations (7), (8), derivative \( \partial T/\partial x \) is written “upstream”. The numerical algorithm uses the conditions of equality of the gas and solid temperatures on the granule surface

\[ T^{n+1}_{g,j} = T^{n+1}_{s,j} \]

(9)

The time step was set maximal: \( \Delta t = \Delta x / F_g \) (i.e., the Courant criterion is equal to one). For this choice of time step, propagation of the jump-like profile without distortion is reproduced in numerical calculation. To calculate heat transfer in the solid phase, equation (6) is transformed as follows. Grid nodes along the radial coordinate divide the granule into spherical cells with volume \( V_j = 4\pi(r_j^3 - r_{j-1}^3)/3 \). Equation (6) was written in the form of heat balance for the spherical cells located between nodes \( r_j \) and \( r_{j+1} \):

\[
V_{j,1/2} \frac{T^n_{s,j}-T^n_{s,j-1}}{\Delta t} = 4\pi \left( r_j^2 \frac{T^n_{s,j+1}-T^n_{s,j}}{\Delta r} - r_{j+1/2}^2 \frac{T^n_{s,j+1}-T^n_{s,j-1}}{\Delta r} \right), \quad j = 2,3,...M
\]

(10)

Here, \( r_{j+1/2} = r_j + \Delta r / 2 \) and \( r_{j-1/2} = r_j - \Delta r / 2 \) are the boundaries of a spherical cell, \( V_{j,1/2} = (V_j + V_{j-1})/2 \) is cell volume. It is easy to show that implicit finite difference scheme (9) is conservative. Equations (10) for the solid phase were solved together with the equation for the air flow by the sweep method using boundary condition (9) and symmetry conditions at the center of a spherical granule \( \partial T/\partial r = 0 \), which has the form \( T^{n+1}_{s,1} = T^{n+1}_{s,M} \).

5. Calculation results

The calculations were carried out for a device with length \( L = 200 \text{ mm} \) (the chosen scale distance was \( l = 1 \text{ mm} \)) at air velocity \( U = 0.5 \text{ m/s} \). The granulated medium with volumetric porosity \( \varphi = 0.4 \) consisted from glass beads with radius \( R = 1.5 \text{ mm} \). The properties of grained medium are: heat capacity \( c_s = 800 \text{ J/kg K} \), density \( \rho_s = 2500 \text{ kg/m}^3 \), thermal diffusivity \( \alpha_s = 3.4 \times 10^{-7} \text{ m}^2/\text{s} \). The properties of air are: heat capacity \( c_{\text{gas}} = 1000 \text{ J/kg K} \), density \( \rho_{\text{gas}} = 1.3 \text{ kg/m}^3 \).

The process of cooling the initially heated charge by a cold air flow (initial conditions for equations (5), (6): \( T_g(x,0) = T_s(x,r,0) = 1 \)) is shown in figure 2. Temperature distributions \( T_g(x) \) and \( T_s(x) \) at different time instants are presented in the figure. It can be seen that the jump-like temperature profile propagates from left to right like a wave with gradually increasing front width. The velocity of temperature jump propagation is much less than the air velocity and it is controlled by the process of heat transfer from the solid phase to the cold air flow. Stabilization of a periodic regime of heat transfer at \( \tau = 200 \text{ s} \) is demonstrated in figure 3. The figure show the gas temperature as a function of time in different cross-sections. At the device inlet (curve 1), the dimensionless temperature is zero in the half-periods of incoming flow, and the “bursts” of temperature correspond to the half-periods of the exhaust flow. At some distance from the inlet (curves 2, 3), the temperature pulsates (it grows in the exhaust phase and falls during the inflow phase). The amplitude of these pulsations (difference between the maximum and minimum values) decreases with distance, since temperature perturbation spreads from the inlet (see figure 2) and gradually covers the entire device. After several tens of pulsations, a periodic regime of heat transfer is stabilized.
Figures 4-7 show the results of calculations of the stabilized periodic regime. An important factor determining the efficiency of device is the periodicity of a veering of an airflow. Figure 4 shows distributions of the gas temperature along the device when the inflow phase ends (curves 1, 3) and when the exhaust phase ends (curves 2, 4) for two different values of a half-period $\tau = 750$ s (curves 1, 2) and $\tau = 200$ s (curves 3, 4). It is visible in figure 4, that when inflow ends $T_g \approx 0$ in the area near the left end of the pipe. Correspondingly, when the exhaust ends $T_g \approx 1$ in the area near the right end of the pipe. Thus, in the areas near the pipe ends, the maximally possible exchange of thermal energy between the granular medium and the air flow occur. In an middle part of the device where the temperature approximately linearly depends on the $x$ coordinate, heat transfer occurs only partially. At $\tau = 200$ s the length of areas of deep heat exchange between the grained medium and air approximately is 40 mm, but at $\tau = 750$ s the length of such areas essentially more and approximately is 120 mm. Thus, efficiency of the regenerative device grows with increase in duration of a cycle.

Figure 5 shows the temperature profiles in the grains placed on the cold end of the device (i.e. $x = 0$) in different instants of time. The curves 1 and 2 correspond to instants before the termination of an exhaust phase, and the curves 3-5 correspond to instants at once after the beginning of inflow phase.
From a figure it is visible, that at veering of gas flow (i.e. in the conditions of fast change of local gas temperature) there is an essential non-uniformity of temperature in granules. In the end of an exhaust phase the local air temperature increases fast on the cold end of the device. At veering of air flow an input cold air arrives and the temperature of the surface of granules decreases by jump. The value of a heat flux on an interface is controlled by a temperature gradient in solid (see curves 3-5).

Figures 6 and 7 show the gas temperature as a function of time at the left end of device (curve 1), in the middle cross-section (curve 2) and at the right end of device (curve 3) at the same two half-periods as in figures 4 and 5. For a relatively small value of $\tau = 200$ s (see figure 6), the amplitude of temperature fluctuations in the end sections does not exceed 0.085, and in the middle cross-section the temperature pulses from 0.38 to 0.64. For a large value of $\tau = 750$ s (see figure 7), the temperature in the middle cross-section pulses from a minimum equal to zero to a maximum equal to unity. It also shows that the maximally possible depth of heat transfer from the granular medium to the air flow and back occurs almost along the entire length of the pipe.

Conclusion
The mathematical model of a regenerative device for the system of ventilation with cyclical reverse of flow direction has been developed. A granulated medium consisting of spherical elements with low thermal diffusivity is used as a heat-accumulating charge. In this case, heat exchange between the gas and solid phases is limited by distribution of heat in the solid phase. Operation of such a device has been simulated numerically. The basic criteria determining heat exchange and their influence on the efficiency of heat regeneration have been revealed.

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