

## A NEW APPROACH TO HEAT AND MOISTURE REGENERATION IN THE VENTILATION SYSTEM OF ROOMS. I. LABORATORY PROTOTYPE OF THE REGENERATOR

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*We propose a new heat-and-mass transfer device that permits simultaneous heat and moisture regeneration in the ventilation system, provides anti-icing at the heat-exchanger outlet, and maintains comfortable humidity in a room. In this device, ahead of the heat-accumulating medium acting as a heat regenerator, a fine-grained adsorbent that controls the humidity of air flows is located. In the first part of the work, we have investigated the cyclic process of heat exchange (in the absence of adsorbent) and conducted its optimization for attaining maximum regeneration of heat. It has been shown that the regeneration efficiency can exceed 90%.*

A large part of the territory of Russia and practically the whole of Siberia are situated in the zone of cold climate, which requires large expenditures for heating living accommodations and industrial premises. One part of these expenditures is connected with heat losses through outer enclosures (walls and windows), and the other part is the cost of energy expended in heating the free air in the ventilation system. Since recently the standards for the values of thermal resistance of outer enclosures have been sharply raised, the first part of the heat losses has markedly decreased. As a result, the ratio between different components of the heat balance has changed and the quantity of heat needed to heat the ventilation air now includes more than half of the total thermal energy supplied for heating rooms [1, 2]. All of this makes the problem of reducing expenditures of energy for heating the intake air in the ventilation system extremely urgent.

The simplest solution of this problem is connected with the use of the warm air flowing out of the room to heat the inflowing cold air. To this end, recuperative and regenerative heat exchangers are usually used [2–5]. However, on the way toward practical realization of this idea, serious obstacles arise:

(a) the presence of a large quantity of moisture in the air flowing out of the room, which may lead to icing in the cold part of the heat exchanger and worsen or even block its operation;

(b) disturbance of the moisture balance in the room, since, with outflowing air, more moisture goes out into the street than comes in with fresh air from the outside (naturally, the air humidity in the room where such devices operate will decrease and go beyond the comfort zone).

In the present work, to solve these problems — heat regeneration, providing of anti-icing, and maintaining of comfortable humidity — we propose a radically new regenerator, in which, ahead of the fine-grained heat-accumulating medium acting as a heat regenerator, a fine-grained adsorbent layer controlling the humidity of air flows is located. The air from the room first passes through the sorbent layer absorbing the moisture contained in it so that the humidity at the outlet becomes lower than the condensation limit at the outdoor temperature. Then it gets into the heat-accumulating medium, where its temperature drops practically to the free air temperature. As the layer is heated, the temperature of the outflowing air increases and when the predetermined temperature difference at the outlet of the device and outside is reached, the air flow reverses direction. The free air is first heated in the heat-accumulating layer and then is saturated with moisture in the sorbent layer to a moisture close to the initial humidity in the room. In so doing, the adsorbent is used as a moisture buffer. Thus, for effective operation of such a cyclic device, simultaneous

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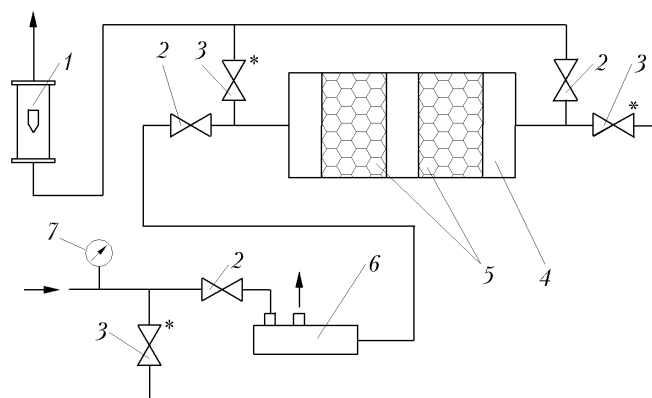


Fig. 1. Hydraulic scheme of the facility: 1) rotameter; 2) valves that are open under cooling conditions and closed under heating conditions; 3) valves that are open under heating conditions and closed under cooling conditions; 4) heat exchanger; 5) heat-accumulating packing; 6) Ranque tube; 7) manometer.

regulation and optimization of the heat and moisture exchange in the stationary layer of the heat-accumulating medium and adsorbent are required.

The moisture exchange between the gas flow and the stationary sorbent layer has been the subject of many papers, whose results have been systematized, e.g., in [6–8]. Aristov et al. [9] made an experimental study of the moisture exchange between the stationary adsorbent layer and the flow through it under cyclic conditions, i.e., they simulated the moisture exchange in the above device. On this basis, the problem of optimization of the adsorbent and the moisture-exchange conditions was solved.

Many investigations were devoted to the solution of the problem of heat exchange between a gas and a stationary layer of a granular material (see, e.g., [10–20]). Among them there are two main groups of works considering the following:

- (a) grained media consisting of elements having a high heat conductivity or sizes so small that their thermal resistance can be neglected;
- (b) grained media in which it is necessary to take into account the thermal resistance of elements.

Since the regenerative heat exchanger considered by us consists of small-sized elements, the system can be classified with the first group, in which the following two cases are distinguished:

- 1) liquid- and solid-phase temperatures are equalized instantaneously, and one can introduce effective characteristics of such a composite medium (effective heat capacity, heat conductivity, etc.); in this case, the change in temperatures is satisfactorily described from the viewpoint of the homogeneous model [13];
- 2) liquid and particle temperatures are different, and the temperature on the particle radius at each instant of time remains practically unchanged (for the air flow passing through a granular layer of small-sized particles, where the values of the coefficient of heat transfer from the grain surface are low and the external thermal resistance is dominant, exactly such a condition is fulfilled); in [11], for this case the temperature change with time to complete heating (cooling) of the layer was calculated.

Cyclic operating conditions are usually investigated in regenerative heat-exchangers for high-temperature heating of air, in which large heat-accumulating elements are used (when it is necessary to take into account the thermal resistance of elements) [21, 22]. Theoretical studies of such processes as applied to real devices are difficult because of the necessity to take into account the heat conductivity of added elements, the three-dimensionality of temperature fields (due to thermal losses on the side walls), the longitudinal effective transfer, etc. Hence, as before, the dominant role in developing regenerative heat exchangers is played by experiments. Of great practical interest is the choice of the material of the heat-accumulating fill and the size of its granules, as well as the investigation of the influence of the air–fill contact time on the periodicity of reversal. The aim of the present work is to study the process of heat exchange (in the absence of adsorbent) and optimize it to attain the maximum heat-regeneration coefficient.

**Experimental.** The laboratory prototype of the regenerative device was installed in a polyethylene tube of diameter 210 mm and length 800 mm. Cassettes with the heat-accumulating material, for which we used lead balls

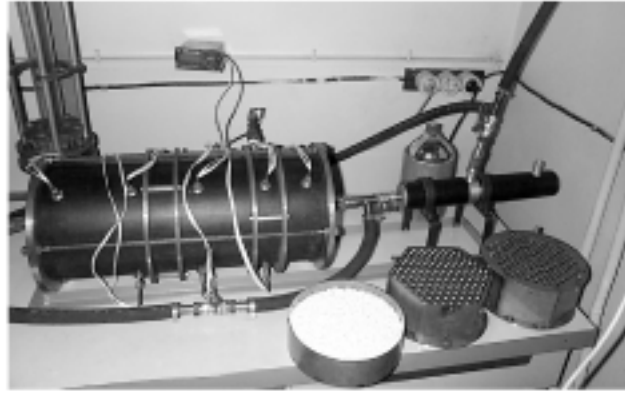


Fig. 2. Laboratory facility for investigating the heat-regeneration process (without heat insulation).

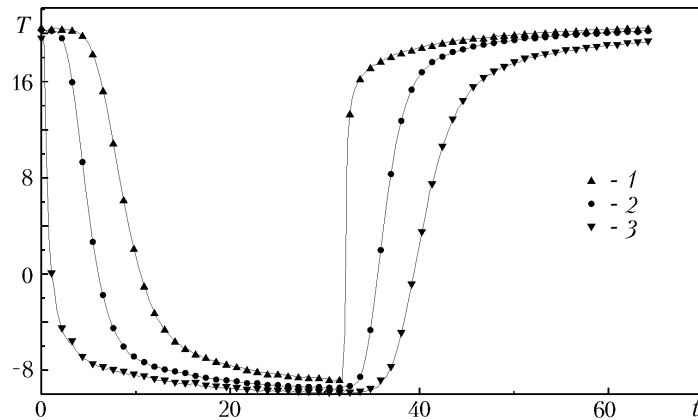


Fig. 3. Typical cooling and heating curves for the heat-accumulating medium: 1)  $T_w$ ; 2) temperature between cassettes; 3)  $T_c$ . Lead balls of  $D = 4.5$  mm;  $V = 30.3$  m<sup>3</sup>/h.  $T$ , °C;  $t$ , min.

( $D = 2.9, 3.5,$  and  $4.5$  mm), glass balls ( $D = 3.2$  mm), and cast iron grit of irregular shape (average size from 0.8 to 1.4 mm) were placed inside the tube. The total length of the layer of balls was 83 or 166 mm. In the laboratory prototype, as an air source, compressed air from the laboratory network, into which a dosed quantity of water vapors was fed by means of an evaporator was used (Fig. 1). To produce a cold air flow ( $T = -5... -20^\circ\text{C}$ ), a Ranque tube was used. Such a scheme permitted independent control of the temperature, moisture, and volume air velocity at the inlet to the device. After each cassette, chromel-copel thermocouples were located. The exterior view of the facility is shown in Fig. 2.

We used the following two experimental procedures:

1. Cold air at a temperature  $T_c = -10^\circ\text{C}$  was supplied to the cold inlet of the regenerator and after each cassette the air temperature depending on time (cooling curve) was measured. When equilibrium was established, air at a temperature  $T_w = 20.5^\circ\text{C}$  was supplied to the warm inlet and the heating curve was recorded in a similar manner (Fig. 3). When equilibrium was attained, switching to periodic heat regeneration conditions was realized in the following way: air at a temperature  $T_c = -10^\circ\text{C}$  was supplied to the cold inlet of the regenerator. Upon passing through the balls, the air was heated to  $T_w = 20.5^\circ\text{C}$  and then, as the balls were cooled, the air temperature at the warm end of the regenerator began to decrease gradually. When it decreased by a given value of  $\Delta T = 2.5, 5.0, 7.5,$  or  $10.0^\circ\text{C}$  (after time  $\Delta\tau_1$ ), the direction of the flow was reversed and air at  $T_w = 20.5^\circ\text{C}$  was supplied to the warm inlet and blown during the same time  $\Delta\tau_1$ . Then the flow was reversed again and air at  $T_c = -8^\circ\text{C}$  was supplied to the cold inlet of the regenerator. After time  $\Delta\tau_2 < \Delta\tau_1$ , when the temperature at the warm end of the regenerator decreased by  $\Delta T = 2.5, 5.0, 7.5,$  or  $10^\circ\text{C}$ , the flow direction was reversed again, and so on (Fig. 4a).

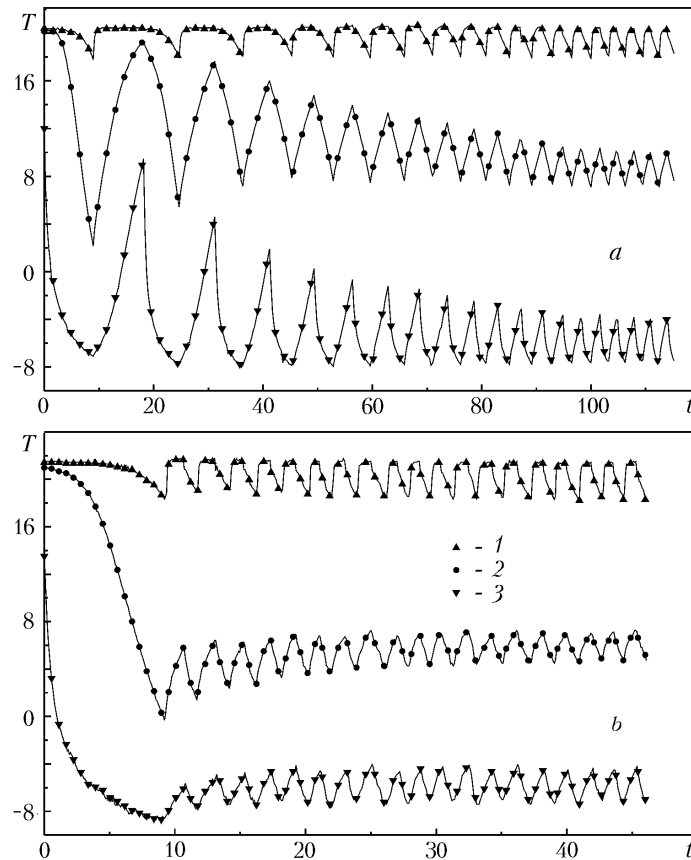


Fig. 4. Establishment of the periodic regime for lead balls of  $D = 4.5$  mm by procedure 1 — a (the first half-cycle at  $\Delta T = 2.5^\circ\text{C}$ , the second one — according to the time of the first half-cycle) and by procedure 2 — b (the first and second half-cycles at  $\Delta T = 2.5^\circ\text{C}$ ): 1)  $T_w$ ; 2) temperature between cassettes; 3)  $T_c$ ;  $V = 18.6$  m<sup>3</sup>/h.  $T$ , °C;  $t$ , min.

2. Unlike procedure 1, the flows were reversed when the temperature difference at both the warm and cold ends of the refrigerator reached the given value of  $\Delta T = 2.5, 5.0, 7.5,$  or  $10^\circ\text{C}$  (Fig. 4b).

To reduce heat losses in the cooling process, the cassettes were heat-insulated from the case by means of isolone ( $\lambda = 0.34$  W/(m·K)), and the case was heat-insulated from the environment by a layer of 50-mm-thick synthepone ( $\lambda = 0.04$  W/(m·K)).

**Results and Discussion.** *Cooling and heating dynamics of the heat-accumulating medium.* The typical curves of cooling and subsequent heating of the layer of lead balls are given in Fig. 3. When cold air is supplied, the inlet temperature rapidly drops to  $-10^\circ\text{C}$ . However, because of the nonzero heat capacity of the conveying pipes, this drop is described not by a staircase curve but by a smooth one, which makes it impossible to use for analysis of the analytical equations proposed in [12]. The temperature drop in the middle of the ball layer and at its ends occurs with a delay. This delay is the greater, the larger the volume heat capacity of the layer, in particular, for the fill of glass balls it is greater than for the fill of lead balls of the same volume. Indeed, the volume heat capacity of the glass is 1960 kJ/(m<sup>3</sup>·kg), whereas in lead it is about one and a half times smaller — 1450 kJ/(m<sup>3</sup>·kg) (Table 1). The thermal front propagating in the layer turns out to be rather sharp. For instance, when the inlet temperature decreases from 20 to  $17^\circ\text{C}$ , the temperature in the middle of the layer drops to  $-2^\circ\text{C}$ , i.e., by  $22^\circ\text{C}$ .

*Dynamics of switching to the periodic regime.* The typical evolution of temperature in going to the periodic regime is given in Fig. 4. It turned out that for both methods of reversing air flows the time between sequential reversals  $\Delta\tau_i$  ( $i = 1, 2$ ) decreases until it reaches a constant value of  $\Delta\tau$ , after which the process proceeds in the steady-state regime. The efficiency of heat recovery  $\theta$  in this regime (intake cycle) was calculated as

TABLE 1. Mass and Volume Heat Capacities of Some Metal and Ceramic Materials and Water

Material	$\rho$ , kg/m <sup>3</sup>	$c$ , J/(kg·K)	$c'$ , kJ/(m <sup>3</sup> ·K)
Al	2700	903	2430
Fe	7870	448	3530
Pb	11 340	128	1450
Au	19 340	132	2550
Al <sub>2</sub> O <sub>3</sub>	3960	770	3050
SiO <sub>2</sub>	2650	740	1960
H <sub>2</sub> O	1000	4200	4200

TABLE 2. Value of the Reversal Time  $\Delta\tau$  (sec) at  $\Delta = 2.5^\circ\text{C}$  (Procedure 1) for Various Packings Depending on the Air Flow Rate (experimental data, packing length of 166 mm)

Packing	$D$ , mm	$V$ , m <sup>3</sup> /h					
		5.0	10.7	14.6	18.6	24.9	31.3
Lead balls	4.5	190	145	120	85	73	63
Lead balls	3.5	375	248	230	210	170	138
Glass balls	3.2	735	612	490	390	320	210
Cast iron grit	0.9—1.4	—	—	—	755	—	470

TABLE 3. Value of the Reversal Time  $\Delta\tau$  (sec) (Procedure 2) for Packings from Lead Balls Depending on the Air Flow Rate (experimental data, packing length of 166 mm)

$V$ , m <sup>3</sup> /h	$D = 2.0$ mm				$D = 4.5$ mm			
	$\Delta T$ , °C							
	2.5	5.0	7.5	10.0	2.5	5.0	7.5	10.0
5.0	55	280	690	1340	62	225	635	1275
10.7	55	235	530	840	60	210	490	773
14.6	55	225	500	735	55	190	415	615
18.6	55	215	415	570	50	145	305	465
24.9	55	205	320	445	43	130	225	350
31.3	55	160	255	350	43	115	210	268

$$\theta = \frac{T_{w2} - T_{c2}}{T_{w1} - T_{c2}} \quad (1)$$

Note that in reversing air flows by procedure 1 the temperature difference at the cold end turned out to be greater than at the warm end (Fig. 4a). This corresponds to a "slip" of part of the warm air and a decrease in the degree of heat regeneration. For instance, for the lead balls at  $\Delta T = 2.5^\circ\text{C}$  the coefficient of heat recovery by procedure 2 turned out to be equal to 0.91, but not 0.84 as in the case of procedure 1. If, in procedure 2, the temperature difference is increased to  $4.5^\circ\text{C}$ , which just corresponds to the heat-recovery coefficient of 0.84, then the reversal time will increase to 140 sec, i.e., it will increase by 40% compared to procedure 1. Thus, the use of procedure 2 for reversing air flows (Fig. 4b) makes it possible to increase either the heat-recovery coefficient or the half-cycle time; therefore, all further experiments were performed by it. The experimental data obtained by procedure 1 are presented in Table 2 and by procedure 2 — in Table 3.

*Heat regeneration in the steady-state regime.* The experiments have shown that in the steady-state regime the time between reversals of air flows  $\Delta\tau$  depends on the air flow rate (air-heat-accumulating medium contact time  $\tau$ ), the temperature difference  $\Delta T$ , the material, and the size of particles of the heat-accumulating medium.

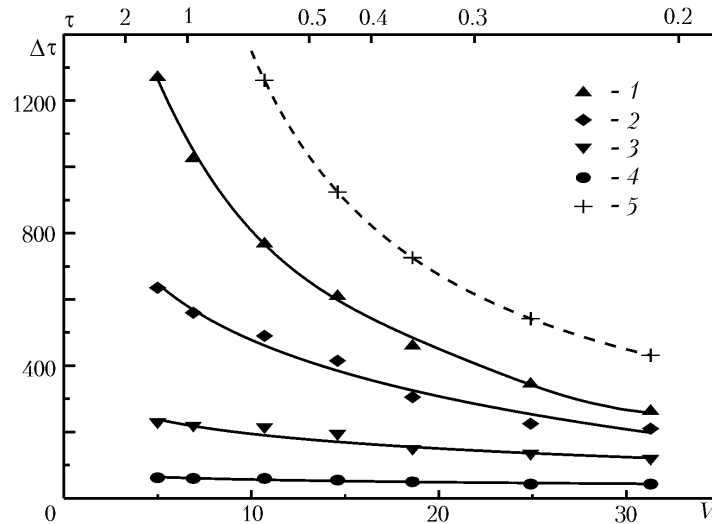


Fig. 5. Reversal time  $\Delta\tau$  of flows versus the flow rate  $V$  air: 1)  $\Delta T = 10$ ; 2) 7.5; 3) 5; 4) 2.5°C; 5) calculation by formula (2). Heat-accumulating medium — lead balls of diameter  $D = 4.5$  mm, packing length — 166 mm.  $\Delta\tau$ , sec;  $V$ , m<sup>3</sup>/h.

TABLE 4. Pressure Drop (Pa) Along the Granular Layer for Lead Balls (experimental data, packing length of 166 mm)

$D$ , mm	$V$ , m <sup>3</sup> /h					
	5.0	10.7	14.6	18.6	24.9	31.3
2.0	80	160	220	280	360	440
4.5	10	30	50	70	100	130

The dependence of the flow reversal time on the flow rate is given in Fig. 5. For comparison, Fig. 5 also shows the theoretical dependence calculated on the assumption of a staircase thermal front:

$$\Delta\tau = \frac{cm}{c'_a V} \sim \frac{1}{V}. \quad (2)$$

For the lead balls of mass 32.1 kg used by us in the experiments,  $\Delta\tau(c) = 13,500/V$  and for the glass balls of mass 7.6 kg  $\Delta\tau = 16,900/V$  (the inlet volume air flow velocity  $V$  is given in m<sup>3</sup>/h). While for an ideal staircase front  $\Delta\tau$  is independent of  $\Delta T$ , in real experiments  $\Delta\tau$  rapidly increases with increasing  $\Delta T$  (see Fig. 5), which is particularly manifest at low velocities. For instance, for  $V = 5$  m<sup>3</sup>/h at  $\Delta T = 2.5^\circ\text{C}$   $\Delta\tau = 62$  sec, and at  $\Delta T = 10^\circ\text{C}$  it is almost 20 times larger:  $\Delta\tau = 1275$  sec. At  $V = 25$  m<sup>3</sup>/h this difference is much smaller:  $\Delta\tau = 43$  sec at  $\Delta T = 25^\circ\text{C}$  and  $\Delta\tau = 350$  sec at  $\Delta T = 10^\circ\text{C}$ . At  $\Delta T = 2.5^\circ\text{C}$  the value of  $\Delta\tau$  is practically independent of  $V$ , which evidently points to the fact that the tail of the thermal front is very sloping. With increasing  $\Delta T$ , when a considerable part of the fill is involved in the heat-exchange process, the  $\Delta\tau(V)$  curve approaches the hyperbolic curve predicted by relation (2).

Comparison of the heating/cooling curves of the heat-accumulating fill, as well as of the reversal times  $\Delta\tau$  for lead balls of  $D = 3.5$  mm and glass balls of  $D = 3.2$  mm (i.e., close in size) shows that  $\Delta\tau$  varies approximately as the volume heat capacity of the material  $c'$ , i.e., as the product of the mass heat capacity by the density. Consequently, as a heat-accumulating packing, it is reasonable to use materials (see Table 1) having the maximum value of  $c'$ . The best characteristics are displayed by water having a high mass heat capacity. From the practical point of view, of the greatest interest are metal oxides. For instance, the volume heat capacity for ceramics based on aluminum oxide is almost the same as for iron at a much lower cost. Mineral compounds — clays, gravel, etc. have approximately the same characteristics. Therefore, in practical devices it is probably reasonable to make fills from oxide materials of the type of ceramic waste, slags, gravel screenings, etc. or metallurgical production waste, e.g., cast iron grit.

Another important characteristic in cyclic heat-storing systems is the heat conductivity of the heat-accumulating materials. Below are its values for some promising materials (in W/(m·K)):

Fused quartz	1.34	Calcite	4.2
Glass	1.09	Aluminum	210
Corundum	27.6	Iron	82.3
Quartz	13.9	Copper	403
Magnesium oxide	59.2	Silver	410
Silicon	162	Water	0.59

It turned out that although in ceramics the heat conductivity  $\lambda$  is much lower than in a metal (they differ by 1–2 orders of magnitude), this practically does not influence the rate of heat exchange between the material of the granular fill and the air flow. This is explained by the fact that under the conditions of the process realized in the regenerator the heat transfer from the air to the solid body (and back) is limited by the heat transfer on the packing surface. Indeed, under the conditions realized in the regenerator the heat-transfer coefficient  $\alpha$  varies from 20 to 100 W/(m<sup>2</sup>·K), and its related thermal resistance is equal to  $1/\alpha = (1-5) \cdot 10^{-2}$  (m<sup>2</sup>·K)/W. The thermal resistance inside a single particle of the fill (tentatively in the form of a sphere of radius  $R$ ) can be estimated as  $R/\lambda$ . For balls of radius 2 mm it is equal to  $2.5 \cdot 10^{-5}$  (m<sup>2</sup>·K)/W for iron and  $2 \cdot 10^{-3}$  (m<sup>2</sup>·K)/W for glass, which is much less than the resistance on the outer surface of the particle. Thus, even in the case of the ceramic particle the temperature gradient on its radius can be neglected. Therefore, the heat conductivity of the fill material imposes no serious restrictions on the process of its heat exchange with the air flow up to the size of fill particles of 6–10 mm. Moreover, the small longitudinal heat-conductivity coefficient of the layer of ceramic particles makes the thermal front narrower, which increases the degree of heat regeneration or the time between reversals.

It has been found that the time between reversals of flows somewhat increases with decreasing sizes of the balls, which is seen from Table 2 with the example of lead balls. This is likely to be due to the fact that in the layer of smaller balls a narrower thermal front is formed. In our opinion, in the granular variant of the packing the size of its particles should be chosen in the range from 1.5–2 to 4–6 mm. For larger particles their low natural heat conductivity begins to manifest itself and, moreover, the thermal front width increases. Smaller particles create a high hydrodynamical resistance (Table 4), which increases energy expenditures for air pumping and impairs the device's parameters.

Under the conditions of our experiments, the heat-recovery coefficient  $\theta$  linearly depends on the relative temperature difference  $\Delta T/\Delta T_{\max}$  (upon attaining a certain value of  $\Delta T/\Delta T_{\max}$  the flows were reversed):

$$\theta = 1 - a \frac{\Delta T}{\Delta T_{\max}} = 1 - 0.3 \frac{\Delta T}{\Delta T_{\max}}.$$

## CONCLUSIONS

1. A new method for controlling the heat and moisture exchange in the ventilation system in apartments and offices that permits solving problems of heat regeneration, anti-icing, and maintaining comfortable humidity has been proposed. To this end, we suggest using a new regenerator, in which an adsorbent layer is placed ahead of the heat-accumulating medium acting as a heat regenerator.

2. It has been shown that the use of lead and glass balls of size 1.5–6 mm, as well as of cast iron grit as a heat-accumulating medium makes it possible to effectively store the heat of the air flowing out of the room (exhaust cycle) and transfer it to the air flowing in from the outside (intake cycle). The heat-recovery efficiency depends on how the process of reversing air flows is organized and can exceed 0.9, i.e., over 90% of the heat of outflowing air returns into the room, which saves much energy expended in heating cold air.

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## NOTATION

$a$ , coefficient, 1/°C;  $c$ , mass specific heat capacity, J/(kg·K);  $c'$ , volume specific heat capacity; J/(m<sup>3</sup>·K);  $c'_a$ , volume specific heat capacity of air, J/(m<sup>3</sup>·K);  $D$ , diameter, mm;  $m$ , mass of packing, kg;  $R$ , ball radius, mm;  $T$ , tem-

perature, °C;  $T_w$ , temperature at the warm end of the regenerator, °C;  $T_{w1}$  and  $T_{w2}$ , initial and final temperatures of heated air at the warm end of the regenerator, °C;  $T_c$ , temperature at the cold end of the regenerator, °C;  $T_{c2}$ , final temperature of air at the cold end of the regenerator, °C;  $\Delta T$ , change in the air temperature at the warm or cold end of the regenerator, °C;  $\Delta T_{\max}$ , maximal temperature drop of air on the regenerator, °C;  $t$ , time of the process, min;  $V$ , air flow rate, m<sup>3</sup>/h;  $\theta$ , heat-recovery coefficient;  $\lambda$ , heat-conductivity coefficient, W/(m·K);  $\rho$ , density of material, kg/m<sup>3</sup>;  $\tau$ , contact times, sec;  $\Delta\tau$ , reversal time, sec;  $\Delta\tau_1$  and  $\Delta\tau_2$ , reversal time for the first and second half-cycles, sec. Subscripts: max, maximum; a, air; w and c, warm and cold ends of the regenerators; 1 and 2, beginning and end of the process.

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