Journal of Engineering Physics and Thermophysics, Vol. 79, No. 3, 2006

A NEW APPROACH TO HEAT AND MOISTURE REGENERATION IN THE VENTILATION SYSTEM OF ROOMS. II. PROTOTYPE OF THE REAL DEVICE

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UDC 697.978

This paper presents the results of investigations conducted on the real prototype of a regenerator whose operating conditions depended on the weather conditions during the testing (winter of 2004). It has been shown that the heat and moisture recovery coefficients can be purposefully and independently regulated over a wide range by selecting the quantity of the adsorbent and heat-accumulating medium. The use of the proposed device in the ventilation system of a standard two-room apartment under the conditions of Novosibirsk can lead to a 44% reduction of heating costs annually.

Introduction. In the first part of this work [1], for simultaneous solution of the heat regeneration and anti-icing problems in the ventilation system, as well as for maintaining comfortable humidity in the ventilated room, a new heat- and mass-transfer device was proposed. In this device, ahead of the heat-accumulating medium acting as a heat regenerator an adsorbent layer is located. The adsorbent layer permits solving the problem of controlling the humidity of air flows. To analyze the optimum operating conditions of the heat-accumulating medium, we investigated the heat exchange as the air was cyclically passing through the stationary layer of the heat-accumulating medium in the laboratory prototype of the regenerator. In [1], we investigated the influence of the duration of contact of air with the heat-accumulating packing and the periodicity of flow reversal on the degree of regeneration of heat. The results obtained have made it possible to choose the material for the heat-accumulating packing and the size of its granules.

The moisture exchange between the stationary adsorbent layer and the air flow passing through it in the cyclic regime was investigated experimentally [2]. In the first (and any odd) half-cycle, humid air was supplied onto the sorbent, which, passing through the adsorbent layer, was completely or partly dehumidified. In the second (and any even) half-cycle, from the other end of the layer relatively dry air was supplied, which, passing though the layer, became humid. It turned out that the use of commercial adsorbents — dehumidifiers — KSM, IK-011-1, and aluminum oxide as moisture buffers permits effective control of the humidity of the outward air flow at the stage of both water adsorption (dehumidification of the air from the room) and desorption (humidification of the air from the outside). The IK-011-1 composite adsorbent is the most effective for both dehumidification and humidification [3]. It was shown that the regulation of the degree of modification of the sorbent and the size of its granules, as well as of the air feed rate (contact time) makes it possible to purposefully obtain the required degree of moisture exchange.

Based on these results, we further investigated the simultaneous heat and moisture exchange in the course of cyclic passage of air through the sequential stationary layers of the heat-accumulating medium and the adsorbent. The experiments were conducted on a real prototype of the device: air was sequentially forced by the fans out of the room into the street and back. The operating conditions of the device depended on particular weather conditions during the testing (winter of 2004).

Experimental. The regenerator was installed in a polyethylene tube of diameter 210 mm and length 800 mm insulated on the outside. Inside the tube were placed cassettes with the adsorbent and the heat-accumulating material. The adsorbent of volume about 3 liters (IK-011-1, granules of D_{ad} = 1.8 and 4.5 mm and length L_{ad} = 5–7 mm) was

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Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 79, No. 3, pp. 151–157, May–June, 2006. Original article submitted May 13, 2005.

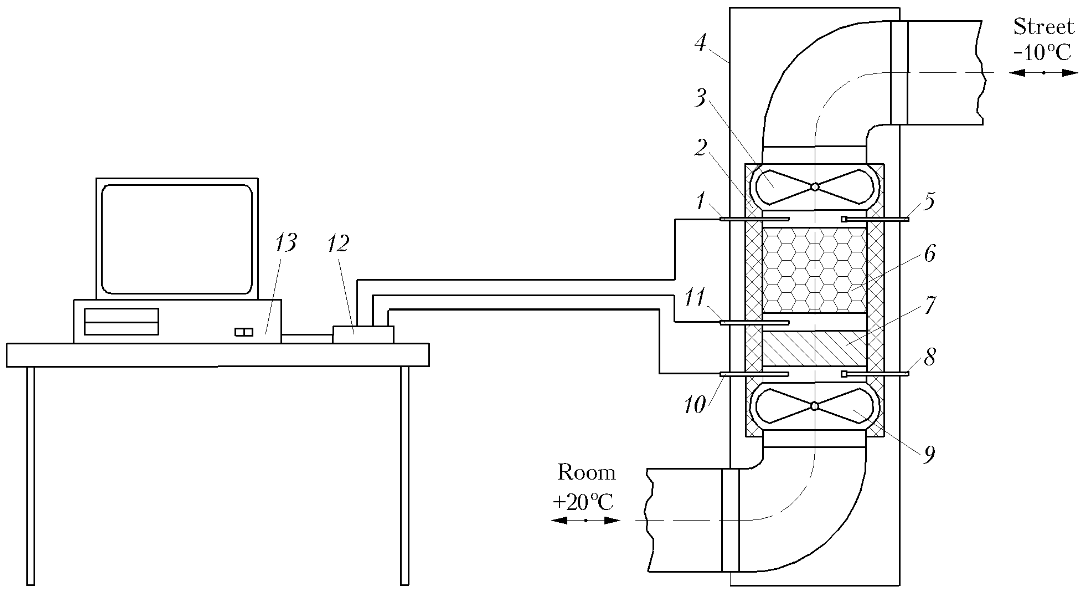

Fig. 1. Schematic representation of the experimental facility of the prototype of the heat and moisture regenerator: 1) thermocouple at the cold end of the prototype; 2) heat insulation; 3) intake fan; 4) case; 5) humidity detector at the cold end of the prototype; 6) cassette with the heat-accumulating packing; 7) cassette with the adsorbent; 8) humidity detector at the warm end of the prototype; 9) exhaust fan; 10) thermocouple at the warm end of the prototype; 11) thermocouple between cassettes; 12) analog-to-digital converter; 13) computer.

fed into a cassette of thickness 160 mm, which was set so that it was the first cassette on the side of warm-air intake (Fig. 1). Then a cassette with glass balls of diameter D_b = 3.2 mm, acting as a heat-accumulating medium, was placed. The total length of the balls was 166 mm. After each cassette, chromel-copel thermocouples were set. The absolute air humidity was measured by IVA-6 detectors on the side of the room and the street. In the real prototype, warm and cool air was supplied by fans of the type Vent 200L positioned at the inlet and outlet of the device. In this case, as opposed to the laboratory prototype [1], only the volume air feed rate (in the range from 24 to 32 m^3/h) could be controlled.

Results and Discussion. In the prototype of the real device, moist and warm air from the room (exhaust cycle) was blown by the fan through the layer of the relatively dry adsorbent, which absorbed the moisture contained in the air. Then the dry and warm air arrived at the heat-exchange packing and heated it, cooling thereby to the open air temperature (Fig. 2). The air temperature at the outlet from the device was first equal to the open air temperature and then, because of the finite heat capacity of the heat-exchange packing, began to increase gradually. When the outlet temperature changed by a given value (ΔT = 2.5, 5.0, 7.5 or 10.0°C), the direction of the air flow was reversed. The cold and dry air from the outside (intake cycle) was heated in the layer of the heat-accumulating packing and then humidified in the adsorbent layer and entered the room with a temperature and humidity close to the values of the air that had entered the regenerator in the exhaust cycle. Gradually, the values of T and φ of the intake air decreased and when the temperature was changed by ΔT, the direction of the air flow was reversed and the cycle repeated.

It turned out that in the presence of the adsorbent the half-cycle duration $\Delta\tau$ almost doubled compared to the respective values of $\Delta\tau$ in the regenerator charged only with glass balls. This growth is likely to be due to (a) the high heat capacity of the adsorbent acting as an additional heat-accumulating medium, which is confirmed by the rather marked change in the temperature at the cold end of the sorbent (Fig. 2, curve 2) from −6 to +12°C, i.e., by 18°C (for balls by 30°C); consequently, this part of the adsorbent exchanges a considerable quantity of heat with the air passing through it; (b) the heat released by sorption, since the time between reversals of direction is much longer in the case of using an adsorbent in the form of small granules (Fig. 3). Since the adsorbent mass in the cassette is

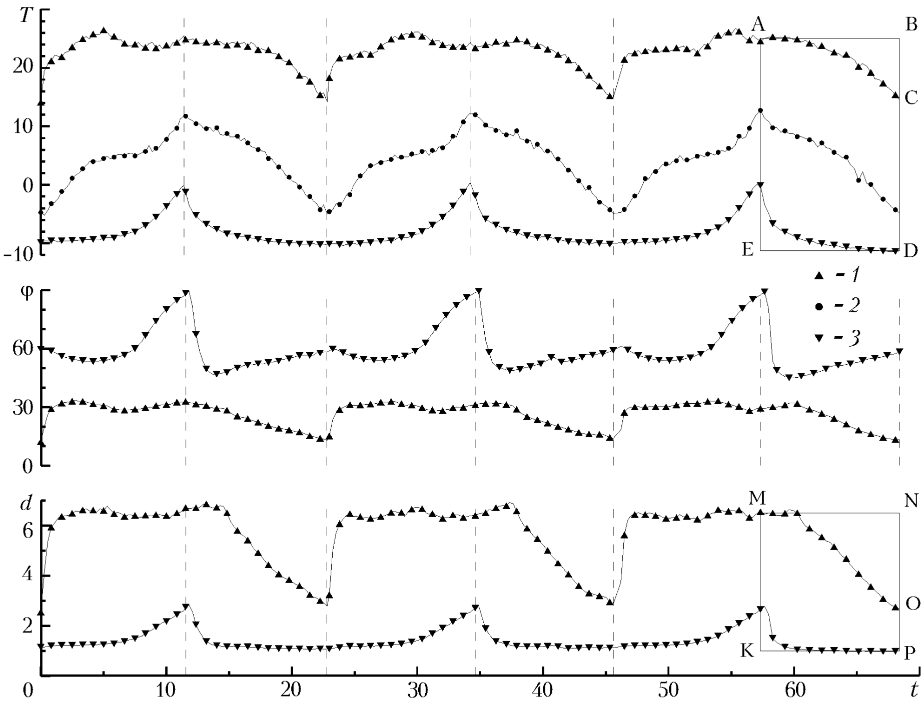

Fig. 2. Change in the temperature T, relative humidity φ, and absolute humidity d in the course of operation of the heat and moisture regenerator 1) at the warm end; 2) between adsorbent and packing; 3) at the cold end. V = 29.4 m^3/h; ΔT = 10.0°C; adsorbent D_{ad}/L_{ad} = 1.8/5–7 mm; glass balls D_b = 3.2 mm. T, °C; φ, %; d, g/m^3; t, min.

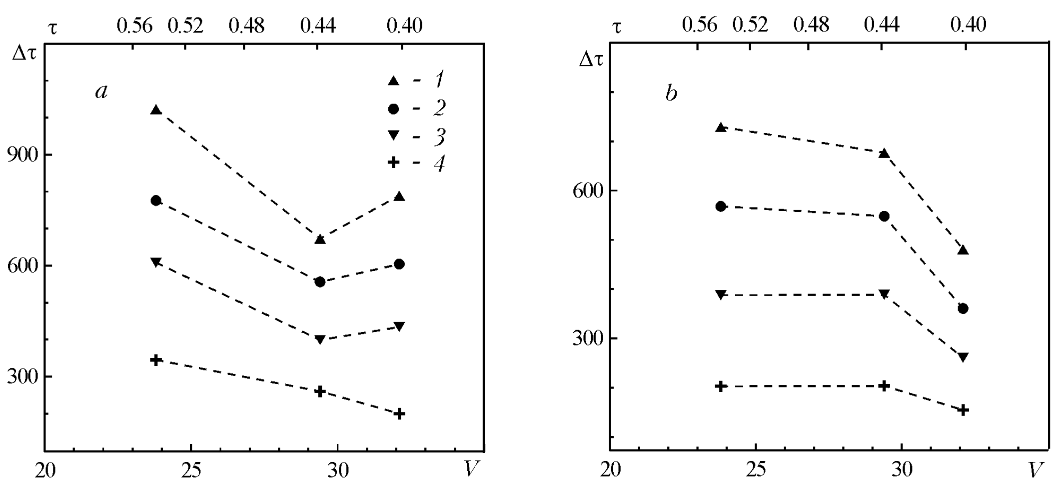

Fig. 3. Time of reversal $\Delta\tau$ of flows versus the air flow rate V for the IK-011-1 adsorbent with granules of D_{ad}/L_{ad} = 1.8/5–7 mm (a), D_{ad}/L_{ad} = 4.5/5–7 mm (b) and glass balls of D_b = 3.2 mm; 1) ΔT = 10; 2) 7.5; 3) 5; 4) 2.5°C. $\Delta\tau$, sec; V, m^3/h.

practically independent of the granule size, this effect is likely to be due to the higher degree of moisture exchange of smaller-sized granules [2]. As a result, under water absorption an additional quantity of heat stored in the layer and used for air heating and water evaporation at the intake stage is released.

At ΔT = 2.5°C, the time between reversals of the flow τ was 200–350 sec and decreased monotonically with increasing air feed rate V (Fig. 3). At a greater temperature difference, the value of τ depended on the air feed rate in

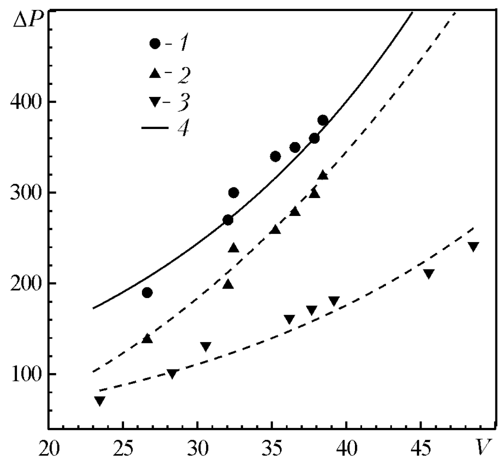

Fig. 4. Hydrodynamic resistance ΔP of the layer depending on the air flow rate V: 1) glass balls, D_b = 3.2 mm, $L_{b.l}$ = 166 mm; 2) IK-011-1 adsorbent, granule of D_{ad}/L_{ad} = 1.8/5–7 mm, $L_{ad.l}$ = 160 mm; 3) IK-011-1 adsorbent, granule of D_{ad}/L_{ad} = 4.5/5–7 mm, $L_{ad.1}$ = 160 mm; 4) calculation for balls by the Ergan formula [4]. ΔP, Pa; V, m^3/h.

a more complicated way. For instance, with the use of the IK-011-1 adsorbent in the form of granules of diameter D_a = 1.8 mm, the $\tau(V)$ curve was characterized by a minimum (Fig. 3a), and for larger granules (D_{ad} = 4.5 mm) the time between reversals was practically independent of the flow rate at $V < 29.4$ m^3/h (Fig. 3b). Such behavior is likely to be a consequence of the related heat and mass transfer in the adsorbent and the packing of glass balls (D_b = 3.2 mm). Indirect evidence of this is the considerable increase in the time between reversals for the adsorbent with small granules, which is especially noticeable at large values of ΔT and a feed rate of 24 and 32 m^3/h. At the same time, at a feed rate of 29.4 m^3/h the times practically equalize, which can result from the complex interaction of the temperature fronts and the concentration of water. Understanding of such an interaction will make it possible to optimize the process of heat and mass transfer in the layer and increase the time between reversals of the flow.

It has been established that the use of a smaller adsorbent makes it possible, on average, to reverse flows less frequently, which is important from the practical point of view. However, in choosing the size of adsorbent granules, one should take into consideration the substantial increase in the hydrodynamical resistance of the layer ΔP with decreasing size of granules (Fig. 4) and the corresponding increase in the expenditure of energy for pumping air through the device:

$$W = \frac{V\Delta P}{\eta}. \tag{1}$$

Assuming that η = 0.7, we obtain that at V = 30 m^3/h, 6 W are expended in air pumping at ΔP = 500 Pa, which is much less than the power needed to heat the adsorbent and the medium at the exhaust stage, which can be estimated as $c_a' V \Delta T_{max}$ = 330 W at ΔT_{max} = 30°C (here ΔT_{max} is the maximum temperature difference of air on the regenerator, i.e., the current difference of temperatures in the room and outside).

Since, under the conditions of the working cycle, the adsorbent temperature varies over a relatively narrow range, it may be expected that commercial dehumidifiers (single-component ones, of the silica gel type, and those modified by hygroscopic salts) will bear a large number of sorption–desorption cycles in such a device. The advantage of modified sorbents (e.g., IK-011-1) is their large adsorption capacity [3] and, accordingly, a smaller fill, which permits smaller dimensions of the device.

The regeneration coefficients of heat θ and moisture β were calculated as the area ratios S_{ACDE}/S_{ABDE} and S_{MOPK}/S_{MNPK}, respectively (see Fig. 2). Since the difference in the temperature in the room and out of doors varies in the course of the experiments, it is convenient to represent the heat-regeneration coefficient depending on the dimensionless temperature difference $\Delta\tilde{T} = \Delta T/\Delta T_{max}$ and the moisture-regeneration coefficient as depending on the di-

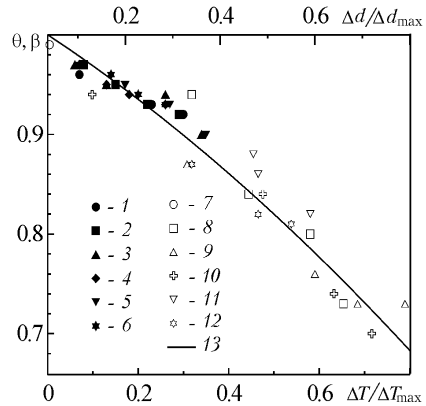

Fig. 5. Heat-recovery coefficients θ (1–6) and moisture-recovery coefficients β (7–12) versus the temperature ratio $\Delta T/\Delta T_{max}$ and absolute humidity $\Delta d/\Delta d_{max}$: 1), 4), 7), 10) V = 32.1; 2), 5), 8), 11) 29.4; 3), 6), 9), 12) 23.8 m^3/h; 13) calculation by formula (2): [1–3, 7–9) IK-011-1 adsorbent, granules of D_{ad}/L_{ad} = 1.8/5–7 mm; 4–6, 10–12) IK-011-1 adsorbent, granules of D_{ad}/L_{ad} = 4.5/5–7 mm]. $L_{ad.l}$ = 160 mm, glass balls of D_b = 3.2 mm, $L_{b.l}$ = 166 mm.

mensionless absolute air humidity difference $\Delta\tilde{d} = \Delta d/\Delta d_{max}$ (Fig. 5). It turned out that all experimental data on the heat- and moisture-regeneration coefficients can be described with an accuracy sufficient for engineering practice by the general relation

$$\theta\,(\beta) = 1 - 0.3\cdot\Delta\tilde{T}\,(\Delta\tilde{d}) - 0.12\cdot\Delta\tilde{T}\,(\Delta\tilde{d})^2\,, \tag{2}$$

that holds for the coefficient θ in the range of change in $\Delta\tilde{T}$ from 0 to 0.35, and for β in the 0–0.8 range of change in $\Delta\tilde{d}$ (Fig. 5). Thus, both regeneration coefficients can be changed purposefully by varying the values of $\Delta\tilde{T}$ and $\Delta\tilde{d}$, which is easy to do in practice by changing the half-cycle time. Testing of the proposed device has shown that the regeneration coefficients have high values: $\theta > 0.9$ at $\Delta\tilde{T}$ = 0–0.25 and $\beta > 0.7$ at $\Delta\tilde{D}$ = 0–0.7.

The fact that the dependences $\theta(\Delta\tilde{T})$ and $\beta(\Delta\tilde{d})$ are described by Eq. (2) with the same numerical coefficients may point to a close relation between the heat- and mass-transfer processes. At the same time, the values of these coefficients apparently depend on the particular design of the regenerator, as well as on the nature and quantity of the adsorbent and the heat-accumulating medium.

Thus, varying the nature and size of granules of the adsorbent and the heat-accumulating packing, as well as their quantity, one can purposefully and relatively independently vary θ and β over a wide range, up to unity. The solution of the above two problems provides both a high degree of regeneration of ventilation heat and regulated return of moisture into the room, and, consequently, maintaining of thermal comfort in it. The choice of the coefficients θ and β for a concrete room depends on the general thermal and moisture balance in it (in particular, on how much heat and moisture is regenerated inside it) and the required temperature and humidity conditions and is a subject of special analysis.

The use of the adsorbent in the proposed device also permits solving the problem of icing of the heat-exchange surfaces at the regenerator outlet at a low open-air temperature. Indeed, even at $\Delta T = 10.0^oC$ the values of the relative humidity φ at the outlet from the device do not exceed 90% (Fig. 2), which permits the prevention of icing. In practice, this gives to the proposed device an important advantage compared to the existing designs of heat regenerators and recuperators in the ventilation system and permits using it under cold climatic conditions. The developed and investigated prototype can be used as a basis in the ventilation system of apartments and offices. The estimate of the efficiency of using this device in a standard two-room apparent in the climatic zone of Western Siberia is given in the Appendix (in prices as of the end of 2004). It points to the possibility of an annual savings of 1425 roubles (with allowance for the cost of electric energy for air pumping), which amounts to 44% of the cost of heating such an apartment.

Conclusions. Full-scale tests on the new heat- and mass-transfer device carried out at a low temperature of open air have shown that the heat- and moisture-regeneration coefficients can be varied purposefully and relatively independently over a wide range by varying the half-cycle time and the nature and size of granules of the adsorbent and the heat-accumulating packing, as well as their quantity. The use of the adsorbent provides, along with a high degree of regeneration of ventilation heat, regulated return of moisture into the room and prevents icing at the outlet from the regenerator at a low temperature of open air. The time between reversals of the air-flow direction depends on the required multiplicity of air exchange and the quantity of the adsorbent and the heat-accumulating packing, as well as on the given degree of heat and moisture regeneration.

The use of the proposed device for heat recovery in the ventilation system of a standard two-room apartment in Novosibirsk can lead to about a 44% reduction of costs for heating, which amounts to 1425 roubles a year.

Part of this work was supported by the Program "Leading Scientific Schools" (grant No. NSh-523.2003.1) and a Program of the Siberian Branch of the Russian Academy of Sciences, "Energy Saving."

APPENDIX

The use of the heat of ventilation releases for heating the intake air in the ventilation system can considerably reduce thermal energy expenditures [5–7]. Let us calculate the economic efficiency of using the proposed heat- and mass-transfer device for a standard two-room apartment in Novosibirsk with a living area of 32.5 m^2 at the following initial data:

Duration of the heating season	227 days
Temperature inside the apartment	20^{o}C
Average temperature of open air in the heating season	–9.1^{o}C
Cost of thermal energy with allowance for VAT (end of 2004)	
1 Gcal	432 roubles
1 GJ	103.1 roubles
Cost of electric energy with allowance for VAT (end of 2004)	
1 kW/h	0.91 roubles
1 GJ	252 roubles
Total area of enclosure	28.5 m^2
Ratio of the area of the wall without window openings to the total area of enclosure	0.7
Ratio of the area of window openings of the wall to the total area of enclosure	0.3
Thermal resistance of the wall	3.7 (m^2·^{o}C)/W
Thermal resistance of the window	0.55 (m^2·^{o}C)/W
Living area of the two-room apartment	32.5 m^2
Air density	1.26 kg/m^3
Mass heat capacity of air	1000 J/(kg·K)
Resistance of the regenerative heat exchanger	500 Pa
Efficiency of the fan	0.5

We obtain that, during the heating period, the effective heat-transfer coefficient of the enclosure is

$$K = \frac{F_{\mathrm{w}}/F}{R_{\mathrm{w}}} + \frac{F_{\mathrm{wind}}/F}{R_{\mathrm{wind}}} = \frac{0.7}{3.7} + \frac{0.3}{0.55} = 0.734 \;\; \mathrm{W/(m^{2}\cdot{}^{o}C)} \; ;$$

the heat loss through the enclosure is

$$Q_{\mathrm{enc}} = KF\,(t_{\mathrm{r}} - t_{\mathrm{h.p}})\; n\; 24\cdot 3600 = 0.734\;(20 + 9.1)\cdot 28.5\cdot 227\cdot 24\cdot 3600 = 11.9 \text{ GJ} \; ;$$

the air flow rate in the ventilation system (3 m^3/h per m^2) is

$$V = 3S/3600 = 3{\cdot}32.5/3600 = 0.0271\ \text{m}^3/\text{sec}\ ;$$

the heat consumption for heating air in the ventilation system is

$$Q_a = \rho V c_p\,(t_r - t_{h.p})\ n\ 24{\cdot}3600 = 1.26{\cdot}0.0271{\cdot}1000\ (20 + 9.1){\cdot}227{\cdot}24{\cdot}3600 = 19.3\ \text{GJ}\ ;$$

the electric power of the fan drive (with regard for two antiphase devices) is

$$N = 2V\Delta P/\eta = 2{\cdot}0.0271{\cdot}500/0.5 = 54.2\ \text{W}\ ;$$

the total consumption of electric energy for the fan drive is

$$Q_{dr} = Nn\ 24{\cdot}3600 = 54.2{\cdot}227{\cdot}24{\cdot}3600 = 1.06\ \text{GJ}\ ;$$

the cost of total losses through enclosures is

$$Z_{enc} = c_{th} Q_{enc} = 103.1{\cdot}11.9 = 1227\ \text{roubles}\ ;$$

the cost of thermal energy expended in heating the air in the ventilation system is

$$Z_a = c_{th} Q_a = 103.1{\cdot}19.3 = 1990\ \text{roubles}\ ;$$

the cost of electric energy for the fan drive is

$$Z_{dr} = c_e Q_{dr} = 252{\cdot}1.06 = 268\ \text{roubles}\ ;$$

the total cost of heating in the absence of the regenerative heat exchanger is

$$Z = Z_{enc} + Z_a = 1227 + 1990 = 3217\ \text{roubles}\ ;$$

the total cost of heating in the presence of the regenerative heat exchanger (at a heat-recovery coefficient $\theta = 0.85$) is

$$Z_{r.h} = Z_{enc} + 0.15 Z_a + Z_{dr} = 1227 + 0.15{\cdot}1990 + 268 = 1794\ \text{roubles}\ .$$

Thus, the savings will equal

$$E = Z - Z_{r.h} = 3217 - 1794 = 1423\ \text{roubles}\ .$$

This calculation shows that the total costs for heating and ventilation during the entire heating period with the use of the proposed heat- and mass-transfer device is reduced by about 44%. It may be expected that in offices, where a more intensive ventilation is needed, savings will be much higher.

NOTATION

c'_a, volume specific heat capacity of air, J/(m^3·K); c_{th}, cost of thermal energy, roubles; c_e, cost of electric energy, roubles; c_p, mass heat capacity of air, J/(kg·K); D_{ad}, diameter of adsorbent granules, mm; D_b, diameter of balls, mm; d, absolute humidity, g/m^3; Δd, change in absolute air humidity, g/m^3; Δd_{max}, maximum difference of absolute air humidity on the regenerator, g/m^3; E, cash savings, roubles; F, total area of enclosure, m^2; F_w, area of the wall, m^2; F_{wind}, area of window openings, m^2; K, heat-transfer coefficient of enclosure, W/(m^2·^{o}C); L_{ad}, length of adsorbent granules, mm; $L_{ad.l}$, length of the adsorbent layer, mm; $L_{b.l}$, length of the ball layer, mm; N, power of the fan drive, W; n, duration of the heating period, day; ΔP, hydrodynamic resistance of the layer, Pa; Q_a, consumption of heat for heating air, J; Q_{enc}, heat loss through enclosure, J; Q_{dr}, electric energy consumption for the fan drive, J; R_w, thermal resistance of the wall, (m^2·^{o}C)/W; R_{wind}, thermal resistance of the window, (m^2·^{o}C)/W; S, living area of the two-room apartment, m^2; T, temperature, ^oC; ΔT, change in the air temperature, ^oC; ΔT_{max}, maximum air temperature difference on the regenerator, ^oC; t, time of the process, min; t_r, temperature inside the room, ^oC; $t_{h.p}$, average temperature of open air over the heating period, ^oC; V, air flow rate, m^3/h; W, expenditure of energy for pumping air through the device, W; Z, costs, roubles; Z_a, costs of air heating in the ventilation system, roubles; Z_{enc}, costs of total losses through enclosures, roubles; Z_{dr}, fan drive costs, roubles; $Z_{r.h}$, heating costs in the presence of the regenerative exchanger, roubles; β, moisture-recovery coefficient; η, efficiency of the fan; θ, heat-recovery coefficient; ρ, air density, kg/m^3; τ, contact time, sec; $\Delta\tau$, time of reversal of the half-cycle, sec; φ, relative humidity, %. Subscripts: max, maximum; ad, adsorbent; a, air; r, room; enc, enclosure; wind, window; h.p, heating period; dr, drive; r.h, regenerative heat exchanger; ad.l, adsorbent layer; w, wall; b.l, ball layer; th, thermal; b, ball; e, electric.

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