Numerical simulation of the energy-saving device for ventilation with periodic veering of an air flow

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Abstract. Two-dimensional mathematical model of energy-saving device for ventilation with cyclic veering of an air flow is developed. Results of calculations by two-dimensional model are compared with those obtained by one-dimensional model and with experimental data. It is shown that one-dimensional model [4] basically correctly describes the heat exchange in terms of temperature, averaged over cross-section. Parametrical researches are fulfilled by a method of numerical modelling to optimise the operating of the regenerative heat exchanger. Influence of parameters of the device on its energy efficiency is revealed.

1. Introduction
During the winter period the ventilation in premises leads to heat losses which can reach 50% from the total heat supply of a building. One of perspective ways of energy saving is utilization of air going outside for heating the incoming air by means of heat accumulated in the special device [1−4]. So, for example, the experimental research of the heat exchanger for ventilation with cyclic reversible air flow was executed in [3]. The lead or glass balls were used as elements of heat-exchange body in this study. In works [4, 5] the results of experimental and numerical research of the regenerative heat exchanger with periodic veering of an air flow are presented. In this device the matrix made of polypropylene with longitudinal air channels, is used as a heat-exchange body. The possibility of applying the developed one-dimensional mathematical model of the regenerative heat exchanger for design and operating regime optimization is also shown. In the given work the results of numerical simulating of energy-

Figure 1. Multi-channel matrix (a) and velocity of air flow (b) depending on time.
saving device for ventilation which was applied in experiments [4] are presented. The device represents the cylindrical tank in which the reversible fan and a heat-exchange matrix with the longitudinal rectangular thin-walled channels are located (see Fig.1 a). The left end of a multi-channel matrix contacts with the external air having temperature $T_{\text{cold}}$, and the right end is allocated indoors where temperature $T_{\text{hot}} > T_{\text{cold}}$ is maintained. The fan moves air or in a premise, or outside, changing a flow direction after each half period, which duration is $\tau$. Switching of a flow direction to the opposite one occurs during time $\tau$. (Fig.1 b). During one half-cycle cold external air receives heat from a multi-channel matrix (an inflow phase). Then during a following half-cycle the matrix absorbs a part of heat of air exiting outside (an exhaust phase). The described device allows a decrease essentially in a heat expense on support of the necessary temperature in a ventilated premise as the outdoor air arriving to a premise appears already partially heated up. One-dimensional model [4] describes the heat transfer in terms of temperature of a matrix and temperatures of air, averaged over cross-section. For closure of the equations in model [4] the heat exchange coefficient $\alpha$ was set as for thermally stabilized flow by means of Nusselt number in the form of $\text{Nu} = \alpha d l / \lambda \approx 4$). In our work, unlike [4], the two-dimensional model taking into account the non-uniformity of temperature of air on cross-section of the channel and temperature and pressure on cross-section of partitions is developed.

2. Theoretical model

The channel we consider as the cylinder of radius $R$ whose sectional area is the same as at the real rectangular channel. The thickness of wall $D$ was set so that the relation of the square of cross-section of a wall to the square of cross-section of the channel matched to a real multi-channel matrix. Two-dimensional temperature fields of air flow $T_g(x,r,t)$ and channel wall $T_s(x,r,t)$ are calculated by the equations of non-stationary heat transfer for air and wall material:

$$\frac{\partial T_g}{\partial t} + u(r,t) \frac{\partial T_g}{\partial x} = \frac{a_x}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_g}{\partial r} \right), \quad \text{at } 0 < r < R,$$

(1)

$$\frac{\partial T_s}{\partial t} = a_r \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_s}{\partial r} \right) + \frac{\partial^2 T_s}{\partial x^2} \right), \quad \text{at } R < r < R + D,$$

(2)

On an axis of the channel and on an external surface of a wall the symmetry conditions were set

$$\frac{\partial T_g}{\partial r} \bigg|_{r=0} = 0, \quad \frac{\partial T_g}{\partial r} \bigg|_{r=R+D} = 0 .$$

(3)

On an interior surface of a wall the conditions of a continuity of temperature and a heat flux were set

$$T_s \bigg|_{r=R+D} = T_g \bigg|_{r=R+D}, \quad \lambda_s \frac{\partial T_s}{\partial r} \bigg|_{r=R} - \lambda_g \frac{\partial T_g}{\partial r} \bigg|_{r=R} = 0 .$$

(4)

Here $r$ is radial coordinate, $x$ is coordinate along a channel axis (cold end of the channel corresponds to $x = 0$), $a$ is thermal diffusivity, $\lambda$ is a thermal conduction, $u$ is velocity of air. The subscript "g" corresponds to gas, subscript "s" corresponds to a solid body.

Thus, in our model there is no necessity to set the heat exchange coefficient $\alpha$.

Velocity of a laminar flow of air in the channel is defined by relationship $u = u_0 (1 - r^2 / R^2)$, where $u_0$ is velocity on an axis of the channel. Thus, the velocity of air on an axis is twice more than velocity $U$ averaged over the channel cross-section. The temperature of air entering into the channel was set according to a cycle phase, i.e. in an inflow phase $T_g(0,t) = T_{\text{cold}}$, in an exhaust phase $T_g(L,t) = T_{\text{hot}}$.

The system of equations (1), (2) has been solved by a finite difference method on section $0 < x < L$ (here $L$ is length of the channel) under the conditions of change in the air flow direction with temporary period $\tau_0 = 2 \tau$. The equations (1), (2) have been converted to a dimensionless form. Using distance scale $l = 1$ mm, time scale $t_n = l / a_g$ and scale of temperature $\Delta T = T_{\text{hot}} - T_{\text{cold}}$, dimensionless variables $(T - T_{\text{cold}}) / \Delta T$, $x/l$, $r/l$, $t_n$ were introduced, keeping for all variables former designations. Then the equations were written in difference form under the implicit scheme and have been solved by a sweep method with usage of boundary conditions (3), (4). The split method to coordinates was applied to a
solution of the equation (2). While calculating wall temperature distribution along coordinate $x$ the conditions $T_{a}|_{x=0}=T_{cold}, T_{a}|_{x=L}=T_{hot}$ were maintained on a face surfaces of the channel ends.

3. Calculation results

Calculations are fulfilled for the device used in experiments [4] where the matrix with length $L=180$ mm had rectangular channels sizes of $3.25\times1.5$ mm and a thickness of a partition between them of 0.5 mm. In this case equivalent radius of cylindrical channel $R=1.25$ mm, a thickness of wall $D=0.3$ mm. Proceeding from two-dimensional field of temperature, air and wall temperatures, averaged over the cross-section, were calculated as follows:

$$T_{g,av}(x,t)=\frac{1}{\pi R^2} \int_{0}^{\pi R^2} T_{g}(x,r,t)2\pi dr, \quad T_{s,av}(x,t)=\frac{1}{\pi(R+D)^2-\pi R^2} \int_{R}^{R+D} T_{s}(x,r,t)2\pi dr$$ (5)

These variables averaged over the cross-section, were compared with experimental data and calculations by one-dimensional model [4]. Using variables (5), the local coefficient of heat exchange between a wall and air $\alpha(x,t)=q(T_{a,av}-T_{g,av})$ and the local Nusselt number $Nu(x,t)=2R\alpha/\lambda_g$ were calculated. Here $q(x,t)=\dot{\lambda} \frac{\partial T}{\partial r}\bigg|_{r=R}$ is density of a heat flux on a channel wall.

![Figure 2. Temperature of air averaged over cross-section depending on time in various cross-sections of the channel. Distance from an input (mm) 0 (1), 120 (2), 180 (3). Continuous curves – calculation, points are experiment [4].](image)

![Figure 3. Temperature of wall averaged over cross-section depending on time in various cross-sections of the channel. Distance from an input (mm) 0 (1), 120 (2), 180 (3). Continuous curves – calculation, points – experiment [4].](image)
In Figures 2–5 the results of calculations of the periodic regime are shown at $\tau = 7$ s, $\tau = 41$ s (i.e. period $\tau_0 = 82$ s). Dependences $T_{g_{av}}(t)$ for various sections of the channel in comparison with experimental data [4] are shown in Fig. 2. It is obvious from a figure that the temperature pulses with the period of the reversible fan $\tau_0$ in all sections. On the channel ends (curves 1 and 3) at the instants of veering of air flow the temperature jumps occur, but in a median part of the channel (a curve 2) such jumps are absent. Temperature at the cold end of the channel (the curve 1) grows monotonously in a phase of exhaust, and drops only at a decrease in rotating of the fan. Accordingly, at the hot end of the channel (a curve 3) in a phase of inflow the temperature drops monotonously. In Fig. 3 the same dependences for temperature of multi-channel matrix, averaged over cross-section, are shown. Here all curves qualitatively have the same appearance as in Fig. 2, but are more smoothed. Thus, the temperature jumps on curves 1 and 3 are absent due to higher thermal inertance of a wall in comparison with air. Calculated time dependences for variables (5) quite well agree with experimental data [4]. Calculations have shown that the one-dimensional model [4] basically correctly describes heat exchange in terms of temperature, averaged over the cross-section.

![Figure 4](image-url)  
*Figure 4.* Temperature depending on the radial coordinate at $x = 0$ (a) and at $x = L$ (b). Dashed lines show the channel walls. Curves 1-4 show various instants in the beginning of inflow phase with a time interval $t/\tau = 0.016$ (a) and in the beginning of exhaust phase with a time interval $t/\tau = 0.032$ (b).

Figure 4 shows temperature profiles on radial coordinate on the channel ends in various instants of the beginning of a phase of inflow and an exhaust phase. It can be seen from the figure that immediately after changing the direction of the air flow, the temperature is significantly inhomogeneous along the radius. Then, during a short interval of a time the wall temperature becomes almost uniform for the entire channel. Almost along the entire channel (except for relatively short sections near to the channel ends) the self-similar profile of temperature of air forms quickly.
Figure 5. Distribution $\text{Nu}(x)$ in the end of inflow phase (1) and in the end of exhaust phase (2).

Figure 5 shows the distributions $\text{Nu}(x)$ for the instants of the termination of inflow phase and exhaust phase. In figure it is obvious that Nusselt number remains constant and equal approximately 5.58 along a considerable part of the channel, except for short sections on the ends of the channel about 30 mm in length. On these sections an intergrowth of a thermal boundary layer through channel cross-section occurs, therefore there $\text{Nu}$ is essentially higher.

Efficiency energy-saving device can be defined, comparing direct-flow ventilation and system of ventilation with periodic veering of an air flow. In case of direct-flow ventilation any mass of air $m$, flowing into a premise, heats up from temperature $T_{\text{cold}}$ to temperature $T_{\text{hot}}$ and gains heat $Q_0 = mc_i\Delta T$.

Then, at an exhaust process, this air returns to the atmosphere, carrying away with itself all gained heat. Thus, the heat equal $Q_0$ is thrown out to the atmosphere. If energy-saving device is applied the exhaust air has average temperature $T = \frac{1}{\tau} \int_0^\tau T_{\text{av}}(0,t)dt$. This value is less, than $T_{\text{hot}}$ since part of heat is transferred to the channel walls. Thus heat $Q = mc_i(T_{\text{hot}} - T_{\text{cold}}) < Q_0$ is thrown out to the atmosphere. Thus, heat loss decreases by $\Delta Q = Q_0 - Q$, and energy efficiency of device is equal to $E_f = \Delta Q / Q_0$. In the accepted dimensionless variables this criterion looks like

$$E_f = 1 - \frac{1}{\tau} \int_0^\tau T_{\text{av}}(0,t)dt$$  \hspace{1cm} (6)$$

The energy efficiency $E_f$ depending on duration of a half-cycle $\tau$ for various values of average velocity of air $U$ is shown in Fig. 6. As it can be seen from the figure, with growth of $\tau$ (at constant $\tau_c = 7$ s) the value of $E_f$ decreases monotonously. With growth of an average air velocity $U$ the value of $E_f$ decreases also, and for great $U$ values sharper falling of $E_f$ occurs with the growth of $\tau$. 

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**Figure 5.** Distribution $\text{Nu}(x)$ in the end of inflow phase (1) and in the end of exhaust phase (2).
Conclusion
Two-dimensional mathematical model of energy-saving device for ventilation with cyclic veering of an air flow is developed. In calculations it is shown that one-dimensional model [4] basically correctly describes heat exchange in terms of temperature averaged over the cross-section. The criterion of efficiency of energy-saving device for system of ventilation with periodic veering of an air flow is formulated. This criterion characterizes a decrease in heat losses at the expense of usage of heat of exhaust air. Parametric researches are executed by a method of numerical simulating to optimize operating of regenerative heat exchanger. It is shown, that in the investigated range of parameters the increase in duration of a cycle and velocity of air flow leads to a decrease in efficiency of the device.

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Figure 6. Efficiency of energy-saving device depending on duration of a half-cycle for various values of an average velocity of air flow (m/s): 0.5 (1), 1. (2), 1.5 (3).

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure6}
\caption{Efficiency of energy-saving device depending on duration of a half-cycle for various values of an average velocity of air flow (m/s): 0.5 (1), 1. (2), 1.5 (3).}
\end{figure}