Thermal and aerodynamic efficiency of a stationary switching regenerative heat exchanger

Nikolay Monarkin¹*, Sergey Lukin¹, Yu M Anurov², B A Tihomirov², G A Agasiants², S M Galileev² and T R Akhmetov³

¹Vologda State University, Institute of civil engineering, 160000 Vologda Lenina st. 15, Russia
²Peter the Great St. Petersburg Polytechnic University, St. Petersburg, Russian Federation
³Kazan State Power Engineering University, Kazan, Russian Federation

* Corresponding author: nikolay-monarkin@yandex.ru

Abstract. The article is devoted to comparison of thermal and aerodynamic efficiency of two configurations of the regenerative nozzle used in the regenerative heat exchanger for room ventilation. The impact of nozzle channel diameter, nozzle length and stage time on the thermal efficiency of the nozzle, its aerodynamic resistance and the required fan power is considered.

1. Introduction

In civilian buildings (public, administrative, residential, etc.) there may be problems concerning the sustainable air exchange [1-9]. This is caused, for example, by the exception of natural influx when using modern hermetic windows, low functionality of existing ventilation systems due to its wear and clogging. One of solutions of such problems is usage of stationary switching regenerative heat exchangers (SSRHE). SSRHE is a compact decentralized ventilation device that allows utilizing the heat of exhaust air in the cold season [10-13]. For effective ventilation of one or several adjacent rooms, SSRHE is installed in a pair in antiphase mode. A reversing fan is used to induce air movement in the heat exchanger. The heat exchange element is a regenerative nozzle, which is a cylindrical beam of channels of small cross-sectional dimensions. The principle of operation of the SSRHE consists in alternately changing modes of external air inflow and exhaust of the internal one after a certain period of time. In the exhaust mode, the internal warm air of the room passes through the nozzle and heats it in order to transfer some thermal energy to the outside air after switching over to the inflow. Thus, the device alternates the stages of accumulation and regeneration of thermal energy.

The purpose of this study is to solve the optimization problem. For a given cross-sectional area (a given nozzle diameter) and a given air flow rate V, it is required to determine the diameter of a single channel d, the nozzle length L, the duration of stages for regeneration and accumulation of thermal energy at which the nozzle will work efficiently enough and with the least aerodynamic resistance.

2. Methods

The paper [14] shows the scheme of the SSRHE regenerative nozzle (Figures 1 and 2) and proposes a mathematical model (1)-(3) of heat exchange processes in a single channel of nozzle. The solution of mathematical model (1)-(3) allows one to find the temperature fields of nozzle and air depending on the
coordinate \( z \) and time \( \tau \): \( T_n(z, \tau) \), \( T_a(z, \tau) \). The proposed mathematical model is implemented using the Matlab software package.

\[
\frac{c_n \cdot \rho_n \cdot S_n \cdot \frac{\partial T_n(z, \tau)}{\partial \tau}}{G \cdot c_a \cdot \frac{\partial T_a(z, \tau)}{\partial x} + P \cdot \alpha \cdot \left( T_n(z, \tau) - T_a(z, \tau) \right) + c_a \cdot \rho_a \cdot S_a \cdot \frac{\partial T_a(z, \tau)}{\partial \tau}} = 0, \quad 0 < \tau < n \cdot \tau_0, \quad 0 < z < L. \quad (1)
\]

Equations (1) and (2) describe the temperature fields of the nozzle and air, respectively. Expressions (3) are the boundary conditions of the problem that are specified.

In model (1)-(3): \( G \) is mass flow rate of air in the channel, kg/s; \( c_n, c_a \) are mass heat capacity of nozzle and air, respectively, J/(kg \cdot K); \( \rho_n, \rho_a \) is density of nozzle and air, respectively, kg/m\(^3\); \( P \) is channel perimeter, m; \( S_n, S_a \) is cross-sectional area of nozzle and air channel, respectively, m\(^2\); \( \alpha \) is heat transfer coefficient, W/(m\(^2\) \cdot K); \( \tau_0 = \tau_{acc} + \tau_{reg} \) is duration of the regenerator cycle, s; \( \tau_{acc} \) is duration of the accumulation period; \( \tau_{reg} \) is duration of the regeneration period.

To assess the efficiency of the nozzle, one can use the following factors for the stages of accumulation and regeneration:

\[
E_{acc} = \frac{\delta T_1}{\Delta T_{max \ reg}} \cdot \frac{\delta T_2}{\Delta T_{max}} \quad (4)
\]

where \( \Delta T_{max} = (T_{in} - T_{out}) \) is maximum available nozzle air cooling or heating from the internal air temperature \( T_{in} \) to the external one \( T_{out}, \) °C;

\( \delta T_1 \) is cooling the internal air in the nozzle at the end of the accumulation stage, °C;

\( \delta T_2 \) is heating the outside air in the nozzle at the end of the regeneration stage, °C.
The coefficients $E_{\text{acc}}$ and $E_{\text{reg}}$ cannot be greater than 1, but, for example, when the air flow is large, or the heat exchange area is small, or the duration of accumulation and regeneration stages $\tau_{\text{acc}}, \tau_{\text{reg}}$ are large, these coefficients can be significantly less than one. For example, if $E_{\text{acc}}=0.5$, this means that at the end of the accumulation stage, the internal air gives the regenerator only half of its heat from the maximum possible. The operation of the regenerator can be considered sufficiently effective if $E_{\text{acc}} \geq 0.9$, and $E_{\text{reg}} \geq 0.9$, that is, more than 90% of the heat of the internal air is utilized in the regenerator during the whole process.

The diameter of the regenerator nozzle is limited, based on design considerations. However, the length of the available prototype has a margin to increase, if we start from the average thickness of the walls of buildings in Russia. Keeping the nozzle diameter constant will cause a decrease in the number of channels in the nozzle as the diameter of a single channel increases. As the channel diameter $d$ increases, the total heat exchange area in the regenerator nozzle decreases, and the coefficients $E_{\text{acc}}$ and $E_{\text{reg}}$ for the same duration $\tau_{\text{acc}}, \tau_{\text{reg}}$ will begin to decrease at some point in time. To prevent this, one can simultaneously reduce the duration of $\tau_{\text{acc}}, \tau_{\text{reg}}$. Then, each channel diameter $d$ will correspond to its time $\tau_{\text{acc}}=\tau_{\text{reg}}$, which preserves the efficiency of the regenerator.

3. Results

Table 1 shows a comparison of two nozzle configurations that differ in size of a single channel $d$, while the wall thickness between the channels is kept constant. The comparison was made at changing the nozzle length and the time of one stage of accumulation (regeneration) of thermal energy.

| Nozzle length, m | Stage time, s | Efficiency factors for different $d$ |
|------------------|--------------|-------------------------------------|
|                  |              | $d_1=0.0016$ | $d_2=0.004$ |
|                  |              | $E_{\text{acc}}$ | $E_{\text{reg}}$ | $E_{\text{acc}}$ | $E_{\text{reg}}$ |
| L1=0.2           | 20           | 0.8777        | 0.9502         | 0.7937         | 0.8457         |
| L1=0.2           | 40           | 0.8804        | 0.9350         | 0.7916         | 0.8290         |
| L1=0.2           | 60           | 0.8845        | 0.9200         | 0.7904         | 0.8120         |
| L2=0.64          | 20           | 0.8935        | 0.9840         | 0.8692         | 0.9486         |
| L2=0.64          | 40           | 0.8940        | 0.9788         | 0.8698         | 0.9415         |
| L2=0.64          | 60           | 0.8948        | 0.9730         | 0.8705         | 0.9347         |

Using the data from Table 1, the graphs were plotted, shown in Figure 3.

![Figure 3](image)
According to table 1 and figure 3 we can draw the following conclusions:

1. With an increase in the length of channel (nozzle) and maintaining the diameter of channel, the efficiency factors $E_{\text{acc}}$ and $E_{\text{reg}}$ increase. At the same time, it is seen from Figure 1 that the graphs corresponding to a larger channel length are located predominantly in the upper part of the coordinate plane, which allows one to conclude that an increase in the channel length has a favorable effect on the thermal efficiency of the nozzle.

2. An increase in channel diameter leads to a decrease in efficiency factors $E_{\text{acc}}$ and $E_{\text{reg}}$ regardless of the stage time and channel length.

3. An increase in time of accumulation (regeneration) stage causes a decrease in regenerative capacity of nozzle ($E_{\text{reg}}$) for both diameters and channel lengths. In this case, accumulation capacity ($E_{\text{acc}}$) of nozzle with an increase in stage time increases in all cases, moreover, when length is $L = 0.2$ m, diameter $d_2 = 0.004$ m.

It should be noted that in the future it is more convenient to operate with coefficients $E_{\text{acc}}$ and $E_{\text{reg}}$ per cycle, and they will be higher than the coefficients determined at the end of the accumulation and regeneration stages. In general, the average $E_{\text{acc}}$ and $E_{\text{reg}}$ should be equal to each other, and this will simply be coefficient of thermal efficiency of regenerator $E$. Then the amount of heat recovered by the regenerator can be determined as follows:

$$Q = E \cdot c \cdot V (t_{\text{in}} - t_{\text{out}}), \text{W}$$

(5)

where $V$ is volume air consumption, m$^3$/s; $c$ is air capacity, kJ/(kg·°C); $t_{\text{in}}$ and $t_{\text{out}}$ are indoor and outdoor air temperatures, °C.

For the regenerator, an important indicator is power consumption of fan, which can be estimated by expression:

$$N = \frac{V \cdot \Delta p_{\text{sum}}}{\eta} \text{W},$$

(6)

where $\eta$ is fan efficiency; $\Delta p_{\text{sum}}$ is total pressure loss in regenerator, Pa.

The main element that creates resistance to air passage in regenerator is nozzle. Since the regenerator is of switching type, the air inlet into the channels of the nozzle is carried out from different sides, depending on the stage. Therefore, we can assume that the total pressure loss in the nozzle ($\Delta p_{\text{sum}}$) consists of loss at the entrance to channel ($\Delta p_{\text{in}}$) and the friction loss in channel ($\Delta p_{\text{fr}}$):

$$\Delta p_{\text{sum}} = \Delta p_{\text{in}} + \Delta p_{\text{fr}}, \text{Pa}. \quad (7)$$

Since the nozzle channels are identical and parallel to each other, to determine the total pressure loss in the nozzle, it is sufficient to determine the loss in a single channel.

Loss of pressure on the friction in the channel can be estimated by the expression:

$$\Delta p_{\text{fr}} = \xi \cdot \frac{L}{d} \cdot \frac{\rho \cdot w^2}{2}, \text{Pa.} \quad (8)$$

Where for laminar flow mode the coefficient $\xi$ is determined by the formula:

$$\xi = \frac{64}{Re} \frac{\mu}{w \cdot d \cdot \rho} \frac{w}{d}, \text{Pa.} \quad (9)$$

If we substitute (9) into the expression for $\Delta p$, we get:

$$\Delta p_{\text{fr}} = \frac{64 \mu}{w \cdot d \cdot \rho} \cdot \frac{L}{d} \cdot \frac{\rho \cdot w^2}{2} = \frac{32 \cdot L \cdot \mu \cdot w}{d^2}, \text{Pa}, \quad (10)$$

where the air velocity in the channel is determined for the average air density at a temperature $T_{av} = (T_{\text{in}} + T_{\text{out}})/2$.

We assume that the speed of movement of air in a room is sufficiently small, so the pressure loss when air enters the channel will be equal to one speed (dynamic) pressure. That is, the coefficient of local resistance of the input will be equal to 1 [15]. Then

$$\Delta p_{\text{sum}} = \Delta p_{\text{in}} + \Delta p_{\text{fr}} = \frac{\rho \cdot w^2}{2} + \frac{32 L \cdot \mu \cdot w}{d^2}, \text{Pa.} \quad (11)$$
Table 2. Comparison of pressure loss and fan power for various lengths and channel diameters.

| Nozzle length, m | Pressure losses $\Delta p_{\text{sum}}$, Pa | Fan power N, W |
|------------------|--------------------------|----------------|
|                  | For the channel diameter, m |                  |
|                  | d1=0.0016 | d2=0.004 | d1=0.0016 | d2=0.004 |
| 0.2              | 27.2      | 3.0      | 0.261     | 0.028     |
| 0.64             | 86.6      | 9.3      | 0.829     | 0.089     |

The calculation of pressure losses and fan power was performed for air consumption $V = 0.00861 \, \text{m}^3/\text{s}$; air speeds in a single channel were: $w_1=0.62; w_2=0.41$.

From table 2 it follows:
1) A 2.5 times increase in diameter reduces the pressure loss and the required fan power by about 9.5 times.
2) Increasing the channel length from 0.2 to 0.64 m (for 3.2 times) with a constant diameter increases the pressure loss and the required fan power by about 3.2 times.

Table 3. Pressure losses in a regenerative nozzle

| Nozzle length, m | Pressure losses, Pa |
|------------------|---------------------|
|                  | At the inlet $\Delta p_{\text{in}}$ | Per friction $\Delta p_{\text{fr}}$ | Total $\Delta p_{\text{sum}}$ |
|                  | For the channel diameter, m |                  |                  |
|                  | d1=0.0016 | d2=0.004 | d1=0.0016 | d2=0.004 | d1=0.0016 | d2=0.004 |
| 0.2              | 0.242      | 0.107      | 27.0      | 2.9      | 27.2      | 3.0      |
| 0.64             | 0.242      | 0.107      | 86.4      | 9.2      | 86.6      | 9.3      |

According to Table 3, it can be concluded that the pressure loss at the air inlet to the channels of the nozzle is extremely small compared to the friction loss. Therefore, they can be neglected in the future.

In this case, from formula (10) it can be seen that $\Delta p_{\text{fr}} \sim 1/d^2$, whereas the heat transfer coefficient $\alpha \sim 1/d$. That is, as the diameter of the channel decreases, pressure losses grow much faster than the heat transfer coefficient increases.

4. Conclusions

As a result, we can draw the following conclusion. An increase in diameter of a single channel leads to a decrease in the heat transfer coefficient, to a decrease in air velocity (at the same flow rate) and to a sharp decrease in pressure loss. Increasing the length of the regenerator nozzle increases the pressure loss almost in direct proportion, but significantly increases the thermal efficiency coefficients $E_{\text{acc}}$ and $E_{\text{reg}}$. Therefore, to achieve maximum thermal efficiency of the regenerator, it is advisable to increase the nozzle length, while increasing the channel diameter can be used as a way to reduce the aerodynamic resistance of the nozzle and, as a result, reduce the required fan power.

In general, optimization of regenerator parameters should be carried out according to economic or energy criteria. The economic criterion is saving money on heating costs in the presence of a regenerator minus the cost of electricity consumed by fan. Energy criterion is saving heat energy for heating less energy costs for the production and transportation of electricity consumed by fan. The optimal parameters of the regenerator must meet the maximum energy or economic criterion.

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