Improving the Efficiency of the Supply- and-Exhaust Ventilation System

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ABSTRACT

The article considers a schematic diagram of the supply-and-exhaust ventilation system. To increase its efficiency, a regenerative heat exchanger is proposed, which is a heat utilizer in a cold season and, in a warm season, it is a device for evaporative air cooling. The paper presents the results of theoretical studies of hydrodynamics, heat and mass transfer of the device. In addition, an experiment was performed to analyze these processes and experimental data were obtained. Analytical and empirical formulas are determined for the engineering methodology for calculating the heat exchanger on the basis of the two types of research.

INTRODUCTION

A significant amount of energy is currently being spent to create specific conditions for people in industrial and administrative premises. For these purposes, a supply-and-exhaust ventilation system is often used, a schematic diagram of which is shown in Figure 1 [1].

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We consider the main drawback of the presented system to be a large expenditure of energy (heat) for heating the supply air. This issue is particularly critical when the outside temperature decreases $t_{oa} < -16 \, ^\circ C$ (the estimated temperature for the city of Voronezh is considered as an example). If $t_{oa} > 18 \, ^\circ C$, the indoor temperature becomes inconsistent with sanitary norms and rules. To reduce these drawbacks, we propose to include heat exchanger 3 in the circuit. In a cold season, it will act as an exhausted air heat utilizer and preheat the supply air by 5-7 °C, which will reduce the expenditure of energy consumed by the heater 2 by an amount from 20 to 25%. When $t_{oa} > 18 \, ^\circ C$ due to the organization in this heat exchanger direct or indirect evaporation cooling of the air supplied to the room, this device will work as a cooler. The second version of cooling consists of two stages: 1) evaporation in the exhaust air flow; 2) cooling the supply air with a constant moisture content. Devices operating on such a principle, which essentially uses the thermodynamic irregularity of the surrounding air, are considered to be devices using a renewable energy source.

**DESCRIPTION OF THE DESIGN OF THE HEAT EXCHANGER**

Earlier theoretical and experimental studies [3-5] showed the high practical significance of the use of a regenerative heat exchanger with a "boiling" centrifugally circulating layer. Such an exchanger is applicable to the system of supply-and-exhaust ventilation. The movement of dispersed material here occurs under the action of two flows of air: main and auxiliary [6]. The design of the proposed device is shown in Figure 2.
Figure 2. Design of the heat exchanger
1 – outer shell; 2 – inner shell; 3 – partition; 4 – overflow window; 5 – gas distribution grid; 6 – nozzle.

The heat exchanger consists of a body formed by two cylindrical shells having a common axis of symmetry. The material of the outer shell is transparent to visualize the processes happening inside. The two partitions 3 divide the inner space of the device into two chambers designed for different air flows: supply and exhaust air. Using the evaporation process in this heat exchanger allows us to call these chambers "wet" and "dry." The blades located at a certain angle to the air flow and the grid superimposed on them form the so-called gas distribution grid of this device. It allows us to orient the movement of the supply and exhaust air flows along with a layer of dispersed material located on the grid at the initial time. The nozzle 6 is located at the entrance to the "wet" chamber to moisten the attachment.

The temperature of the wetted particles of the dispersed material, which are in the "boiling" state, starts to decrease due to evaporation of moisture from their surface and becomes equal to the temperature of the wet thermometer. The dried particles are moved to the "dry" camera and reduce the temperature of the main air flow intended for supply to the consumer. The temperature of the particles of the dispersed material is increased to the temperature of the particles at the entrance to the “wet” chamber at the initial moment of time. Then the movement of the attachment is repeated, thus organizing a closed circular movement. In a cold
season, the water is not supplied to the nozzle and the heat exchanger works as a heat utilizor of the exhaust air.

Heat exchangers of this type are not yet widely used in the housing and utilities sector and in industrial enterprises due to the lack of engineering calculation methods based on a complete study of the heat and mass transfer process in the circulating “boiling” layer. We will model it in relation to the structure under consideration.

MODELING HEAT AND MASS TRANSFER IN A REGENERATIVE HEAT EXCHANGER

Using the heat balance equation (1) for a particle in a “wet” chamber, we find the time in which water is completely evaporated from one particle of the dispersed material. At the same time, it is assumed that a decrease in the volume of the water from a particle of the material by an amount \(dv\) happens at time \(d\tau\).

\[
q_f p d\tau = -\rho_f \left[ c_f \left( t_{sat} - t_f \right) + r_v \right] dv,
\]

where \(q\) - the heat flux density \(W/m^2\); \(f_p\) - the surface area of a wetted particle, \(m^2\); \(\tau\) - the time, \(s\); \(\rho_f\) - the water density, \(kg/m^3\); \(c_f\) - the heat capacity of water, \(J/(kg\cdot K)\); \(t_{sat}\) - the saturation temperature, \(K\); \(t_f\) - the water temperature, \(K\); \(r_v\) - the latent heat of vaporization, \(J/kg\); \(v\) - the volume.

Taking into account that \(dv = f_p dr\), we get from (1)

\[
\tau = \int_{r+\delta}^{r} \frac{\rho_f \left[ c_f \left( t_{sat} - t_f \right) + r_v \right]}{q} dr,
\]

where \(\delta\) - the thickness of the water film on the surface of the particle, \(m\); \(r\) - the particle radius, \(m\);

To determine the heat flux density, it should be taken into account that thermal energy is supplied to the particle through the convection process and, therefore, the application of Newton-Richman law is possible. Therefore

\[
q = \alpha (t_a - t_{sat}),
\]

where \(\alpha\) - the interphase heat transfer coefficient between the air flow and the particle of the dispersed material in the “wet” chamber, \(W/(m^2\cdot K)\); \(t_a\) - the air temperature in the auxiliary flow.
The thickness of the water film on the surface of the particle of the dispersed material is very small. In this regard, its influence on the process of heat transfer is insignificant. Given this assumption, we can determine the heat transfer coefficient, using the following criterial equation [7]

\[ Nu = 0.51 \text{Re}^{0.65}, \]  
(4)

where \( Nu = \frac{ad}{\lambda_a} \) - Nusselt number; \( \text{Re} = \frac{w_d}{v_a} \) - Reynolds number.

In the formulas to determine the criteria, the following notations are used: \( \lambda_a \) - the air thermal conductivity, W/(m·K); \( v_a \) - the kinematic viscosity coefficient, m²/s; \( d_e \) - the equivalent particle diameter, m; \( w_a \) - the air velocity, m/s.

From the joint solution (2), (3) and (4) and integration, we obtain

\[ \tau = \frac{\rho_f \left[ c_f \left( t_{sat} - t_f \right) + r_a \right] }{0.42 \lambda_a w_a^{0.65} v_a^{-0.65} (t_a - t_{sat})} \left[ (r + \delta)^{0.65} - r^{0.65} \right]. \]  
(5)

To calculate the temperature fields in the "dry" chamber of the air cooler, a system of differential equations was solved, which included the heat balance equation and the Newton-Richman equation. This system was recorded for the elemental volume of the "boiling" circulating layer. For the air temperature at the height of the attachment layer, the following relationship is obtained

\[ t_a = t_p + \left( t_{p} + t_{p}' \right) \exp \left\{ -\frac{\alpha (1 - \varepsilon) f_p}{c_a w_a \rho_a} y \right\}, \]  
(6)

where \( t_p \) – the temperature of particles of the dispersed material, K; \( t_{p}', t_{p}' \) - the temperature of air flow and particles at the entrance to the "dry" chamber, K; \( \varepsilon \) - the porosity of the "boiling" layer; \( f_p \) - the specific surface of the attachment layer m²/m³; \( \rho_a \) - the air density, kg/m³; \( c_a \) - the heat capacity of the air, J/(kg·K); \( y \) - the coordinate.

The independence of the temperature of particles from the y coordinate is due to their active mixing. With this in mind, the dependence for the temperature of the particles of the dispersed material is obtained as follows

\[ t_p = t_{p}' + \left( t_{a} - t_{p}' \right) \exp \left\{ -\frac{c_a w_a \rho_a x}{c_p w_p (1 - \varepsilon) \rho_p h} \left[ 1 - \exp \left( -\frac{\alpha f_p h (1 - \varepsilon)}{c_a \rho_a} \right) \right] \right\}, \]  
(7)
where $\rho_p$ – the material checker density, kg/m$^3$; $c_p$ - the heat capacity of the material checker, J/(kg·K); $w_p$ - the material checker movement velocity, m/s; $h$ - the height of the material checker layer, m; $x$ – the coordinate, m.

**EXPERIMENTAL STUDY OF A HEAT EXCHANGER**

To examine the heat exchanger for compliance with the requirements of the specified technical parameters for the system of supply and exhaust ventilation, an empirical study of an experimental sample, the design of which is described above, was performed. Here we give only the specific geometric and thermal parameters and the method of organizing the physical process. Thus, the diameters of the outer and inner shells were 200 and 300 mm, respectively, and their height was 500 mm. The angle of installation of the blades could vary from 20 to 40°. The air supply to each of the chambers of the device was carried out by high-pressure fans VC 10-28 no. 3, and its consumption was measured by calculating and using anemometers TTM-2/4-06. Thermocouples TP-0188 were used to measure the air temperature at various points of the device. As a dispersed material of the attachment we used particles from various materials: zinc-aluminum alloy ($\rho_p = 2850$ kg/m$^3$, $d_e = 2.6; 2.9; 4.6$ and $5.0$ mm), sand ($\rho_p = 2650$ kg/m$^3$, $d_e = 2.7$ and $3.2$ mm). To study the required number of operating modes of the heat exchanger, both types of attachments were used, and their mass was in the range from 0.5 to 3.5 kg. The flow rate of water through the mechanical nozzle ranged from $4 \cdot 10^{-4}$ to $24 \cdot 10^{-4}$ kg/s. The loss of air pressure in the chambers of the device was measured with high-precision micro manometers MMN-240.

The following experiment algorithm was used. The specified mass of particles of the dispersed material was poured into both chambers. Then VC 10-28 was activated. Using a frequency converter, the air consumption in the chambers was adjusted to the values corresponding to the steady displacement of the “boiling” layer along the grid. After that, the water mechanical nozzle was connected, which wetted the particles in the “wet” chamber of the device. After reaching the pseudo-stationary state of the entire system, various thermal and physical characteristics were measured: the temperatures, the air and water consumption, and the pressure. For this, the SCADA system was used along with the hardware and software of the OVEN firm. Thus, a study of more than fifty modes of operation of the heat exchanger was carried out.

According to the measurement results, the efficiency coefficient was calculated

$$\eta = \frac{t'_w - \bar{t}_w}{t'_a - t_{wet}} \cdot 100,$$  

(8)
where \( \bar{t}_a \) – the average integral temperature of the coolant at the outlet of the “dry” chamber, K; \( t_{wet} \) - the temperature of the “wet” thermometer, K.

Analyzing the results of the experiment, we determine the main parameters, the change of which leads to a change in the thermal efficiency of the heat exchanger. These parameters are \( w_a \) and \( M \), present in (9). As a result of the approximation of the obtained data from the experiment by means of MathCad, the following equation was obtained

\[
\eta = 4.55 w_a^{1.01} M^{0.04},
\]

where \( M \) – the dispersed material mass, kg.

The dependence of the efficiency coefficient on the mass of the attachment and the velocity of the main air flow is shown in Figure 3. The standard deviation is about 3%.

\[ \eta, \%
\]
\[ 80 \quad 60 \quad 40 \quad 20 \]
\[ 1.5 \quad 2.0 \quad 2.5 \quad 3.0 \quad 3.5 \quad 4.0 \]
\[ M, \text{kg} \]

Figure 3. Dependence of the efficiency coefficient on the mass of the attachment and the velocity of the main air flow

- \( w_a = 5.13 \) m/s, \( w_a = 3.06 \) m/s; — calculation by (9).
Experimental data on the pressure drop of the heat exchanger were also subjected to statistical processing, which resulted in the following relationship

$$
\Delta P = 2.58w_w^{2.15}M^{0.73}.
$$  \tag{10}

Figure 4 shows the dependence of the hydraulic resistance for some fixed values of the mass of the dispersed material, depending on the air velocity. Here, the error comparing to the calculated data was 4%. Also, the pressure difference between the “wet” chamber and the calculated value by (10) was determined. This value was 20%.

![Figure 4. Dependence of the pressure drop of the air cooler chamber](image)

- $M = 3 \text{ kg}$, $M = 2.5 \text{ kg}$, $M = 2 \text{ kg}$; — calculation by (10).

**CONCLUSIONS**

As a result of the conducted research, it was shown that it is expedient to include the regenerative heat exchanger with the attachment in the form of a “boiling” centrifugally circulating layer of the dispersed material into the scheme of supply and exhaust ventilation of various premises. In a cold season, such a heat exchanger will act as an exhaust air heat utilizer, and in a warm season, it will cool the supply air due to the evaporation of water.

Theoretical and experimental studies have shown the efficiency of the device and also allowed us to obtain several analytical and empirical relationships that can create a scientific basis for its engineering calculation.
REFERENCES