Thermal buoyancy ventilation systems

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Abstract. Objectives. To increase the efficiency of natural ventilation systems, it is possible to use the thermal incentives - additional heating of the exhaust ventilation duct, which allows to expand temperature difference between ambient and exhaust air. This method enables stable air exchange in hot period, regardless of wind force. However, existing systems have several limitations to application in multi-storey residential buildings in countries with a cold climate. The main objective of the study is to develop a thermal buoyancy ventilation system, suitable for typical residential building in Russia. Methods. The study of internal natural convection was carried out using the CFD software, based on the finite volume method with unstructured mesh. Results of computer simulation were validated by field experiments. Results. The article presents the results of numerical modelling of natural convection currents arising in the heated channel of the last floor ventilation system in a multi-storey residential building. In the course of the study, there was found the dependence of outlet velocity on temperature difference $\theta, ^\circ C$ for various heating methods. Conclusions. Using of thermal buoyancy systems with vertical heating provides stable air exchange in the warm and transitional periods of the year. The obtained results illustrate the high level of influence of heating area location on free convection currents in ventilation duct.

1. Introduction

Thermal buoyancy ventilation systems are systems with additional heating of the exhaust duct, which increases the intensity of natural air exchange in the room. Heating of the ventilation duct leads to the formation of ascending convective currents in it. The movement of air is due to the presence of a density heterogeneity caused by the temperature diversity. Until now, in modern scientific researches on thermal buoyancy ventilation systems [1-3], only solar chimneys were considered. Existing systems are not suitable for multi-storey residential buildings, especially in countries with a cold climate [4]. However, this method of intensifying the work of natural ventilation has many advantages, such as: ensuring a stable air exchange during the warm period of the year, independence from the force and direction of the wind, exclusion of acoustic pollution from ventilation equipment. Thus, the aim of the study is to develop a thermal buoyancy ventilation system, suitable for use in the cold climate for typical residential buildings.

In modern multi-storey residential buildings, the apartments of the last and penultimate floors are equipped with individual ventilation ducts to avoid the flow of polluted air from the main collection channel to the upper floors [5]. The main problem of the last floor ventilation systems is the weakened

or reverse draft head [6]. To improve the quality of ventilation, designers often place axial fans in the ducts [7]. However, this solution degrades the air exchange in the apartments, because during the period of inactivity exhaust fan is creating excessive aerodynamic resistance. In view of the above, the duct of the last floor is adopted as a calculation model for the study of free convection currents in thermal buoyancy ventilation system.

2. Numerical simulation

Convective air movement in the duct of thermal buoyancy ventilation system made of sheet steel is considered. The air movement occurs in bounded region Ω_1 with a conditional temperature on heated walls (boundary B_1) equal to t_c (Figure 1). The lower part of the channel is the inlet section (B_2), through which the internal air from the room (Ω_2) with temperature t_i and pressure P_i enters the duct. The upper part of the channel is the outlet section B_3, through which the exhaust air with temperature t_o enters the atmosphere (Ω_3).
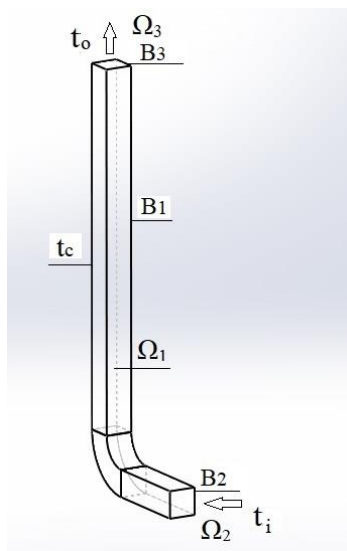
Figure 1. Calculation scheme of the mathematical model: t_c – conditional temperature on the heated walls; t_i – inlet temperature; t_o – outlet temperature
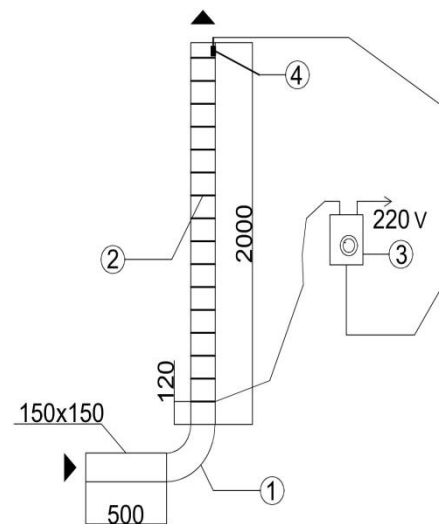
Figure 2. Scheme of the experimental model: 1 – ventilation duct; 2 – flexible tape heating element; 3 – electronic temperature controller; 4 – thermoelectric sensor

Location and size of heating area are largely determining the movement of convective air currents. Efficiency of natural ventilation systems depends on the uniformity of velocity distribution. It is therefore proposed to make a comparative analysis of several methods of thermal motivation: with bottom, horizontal and vertical heating, as well as heating of the entire channel.

In the general form, the free convection of viscous incompressible gas in bounded region Ω_1 is described by the convection equations in the Oberbeck-Boussinesq approximation [8]:

$$\begin{cases} \dfrac{\partial v}{\partial t} + (v\nabla)\cdot v = -\dfrac{1}{\rho}\cdot\nabla p + \upsilon\cdot\Delta v + g\cdot\beta\cdot T\cdot\gamma \\[2mm] \dfrac{\partial T}{\partial t} + v\cdot\nabla T = \chi\cdot\Delta T \\[2mm] div\,v = 0 \end{cases} \qquad (2.1)$$

v - air flow velocity, m/s;

p - convective additive to the hydrostatic pressure corresponding to the average temperature and density, Pa;

$\overline{\rho}$ - average density, kg/m^3;

T - temperature, measured from a certain average value, K;

g - acceleration of gravity, m/s^2;

υ и χ - coefficients of kinematic viscosity (m^2/s) and thermal diffusivity (m^2/s), assumed to be constant;

β - volume coefficient of thermal expansion, 1/K;

γ - unit vector, vertically upward.

In the Oberbeck-Boussinesq approximation, it is assumed that density heterogeneity caused by pressure heterogeneity are small, and they can be neglected. A linear approximation is used to describe the dependence of density on temperature [8]:

$$\rho = \overline{\rho} \cdot (1 - \beta \cdot \Delta T) \tag{2.2}$$

ΔT – temperature deviation from equilibrium.

In the course of the heating process, with the temperature difference between conditional temperature on the heated walls and inlet temperature $\theta = 5 \div 40\,°C$, the Rayleigh number varies from $0.04 \cdot 10^9$ до $72.07 \cdot 10^9$. According to [9], the laminar flow regime is conserved at $10^3 < Ra < 10^9$. When $10^9 < Ra < 6 \cdot 10^{10}$, transitional regime begins, that is characterized by instability of the flow. For calculation of transitional regime most commonly used k-ε model of turbulence [10]. However, in the case of simulation of natural convection under internal conditions in the presence of thermal conduction boundaries, which in this case are the boundaries of the inlet (B$_2$) and outlet section (B$_3$), for streams with $Ra < 10^{11}$, accounting for flow turbulence has virtually no effect on the final results [11]. In this case, the experimental and calculated data give good agreement at any point under consideration. Thus, in the submitted range of the Rayleigh number values, the application of the laminar approximation is justified.

The basic assumptions used in mathematical model are:

1. movement of air in the channel arises because of the difference in air densities in the room and in the considered area of ventilation duct;

2. calculated mathematical model is a model with distributed parameters. The motion and heat transfer in the ventilation duct (region Ω_1) are described by the convection equations in the Oberbeck-Boussinesq approximation;

3. air in the channel is transparent for thermal radiation of the walls;

4. pressure distribution in regions Ω_2 and Ω_3 is determined by hydrostatics;

5. calculated mathematical model is stationary;

6. air movement mode is laminar.

Boundary conditions:

1. velocity of air flow on the surface of the ventilation duct is zero (sticking condition)

$$v|_{B1} = 0 \tag{2.3}$$

2. conditional temperature on the surface of ventilation duct is constant

$$T|_{B1} = T_c(x; y; z) = const \tag{2.4}$$

3. air mobility in the regions Ω_2 and Ω_3 is zero

$$v|_{\Omega 2} = v|_{\Omega 3} = 0 \tag{2.5}$$

4. in boundary B$_2$, air has a temperature equal to the indoor air temperature

$$T|_{B2} = T_i = const \tag{2.6}$$

5. the Oberbeck-Boussinesq equation includes the value of acting pressure, which is the difference between actual and hydrostatic pressure. Thus in the inlet section B_2 and outlet section B_3 boundary condition is written in the form [12]:

$$P|_{B2} = P|_{B3} = 0 \qquad\qquad (2.7)$$

The program Ansys Fluent was used as the calculation tool of the study, which has a high level of convergence of numerical solutions and full-scale measurements of convective currents [11, 13 and 14].

3. Experimental study

The experimental model consists of a steel galvanized air duct 0.5 mm thick. Heating channel was carried out using a resistive heating tape ENGL-1, which was connected to AC network through thermostat ART-18-10H (Figure 2). The standard difference between on and off temperatures (hysteresis) of thermostat was 2 ° C. To ensure maximum contact between heating cable and ventilation duct, a self-adhesive aluminum reinforced tape was used (Figure 3).

Figure 3. Photo of the experimental model. Left: resistance heating cable ENGL-1, right: cable fixation with self-adhesive aluminum reinforced tape

To reduce heat losses, a heat-insulating layer was laid outside the ventilation duct, so that the "cold end" of resistance cable remains outside.

4. Results and discussions

As shown by the results of the conducted studies, inlet velocity depends only on the calculated temperature difference between the indoor air in the room and the conditional temperature on the internal surface of the ventilation duct, $\theta, °C$ (Figure 4).

The discrepancy of experimental and calculated results increases in cases when θ > 20 °C, especially in systems with vertical heating (with elbow and without it). This phenomenon is due to the fact that with increasing temperature of resistance cable, it is quite difficult to achieve an even heating of ventilation duct in real conditions [15].
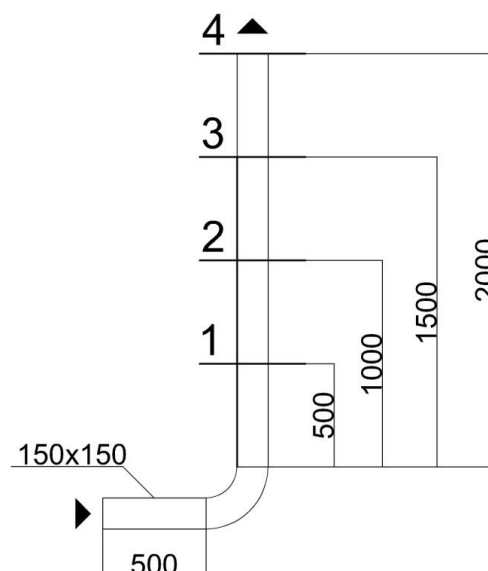
The maximum deviation of the results is 4.4 %.

Horizontal part of ventilation duct is shorter than vertical, so the actual warm-up time to the calculated temperature difference $\theta = 40°C$ was about 30-40 minutes, which is almost 4 times less than for vertical heating with elbow. The maximum deviation of the results is less than 2.9 %.
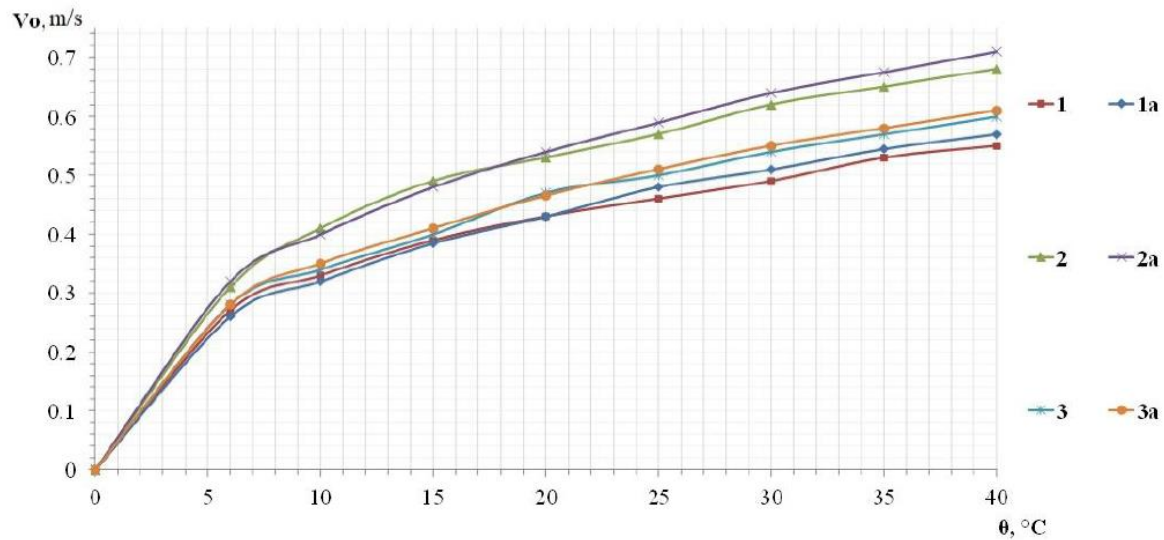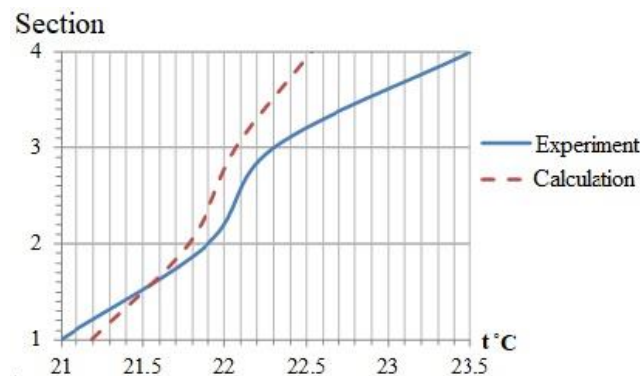
Figure 4. Comparison of numerical simulation and experiment. Graphs of axial inlet velocity (V_o, m/s), depending on calculated temperature difference (θ, ° C). 1 - experimental data for vertical heating, 1a - modeling data for vertical heating, 2 - experimental data for vertical heating with elbow, 2a - modeling data for vertical heating with elbow, 3 - experimental data for horizontal heating, 3a - experimental data for horizontal heating

The measurement of the axial air flow temperature along the height of ventilation duct was carried out according to four control sections (Figure 5).

Figure 5. Control sections

Figure 6 shows the change of axial temperature along the height of ventilation duct with vertical heating jointly by elbow. The maximum deviation of the results is less than 4.1 %.

Figure 6. Comparison of numerical simulation and experiment. Graphs of axial temperature along the height of ventilation duct. $\theta = 8 \degree$ C

Experimental study allows to make a conclusion about reliability of numerical modeling results in the considered range of main parameters values (temperature and velocity), and about the possibility of CFD modeling of free convective currents in limited areas.

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