Experimental study of heat exchange processes with variable operating modes of air heater-fan system

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Abstract. The variable modes of operation in the model heater-fan-room system were studied experimentally and analyzed. The relative excess heat is proposed as a dimensionless parameter in the analysis and rationing of experimental data. The analysis of its evolution was carried out with an abrupt change in the performance of the air heater and at a fixed fan performance; with an abrupt change in the performance of the fan and a fixed performance of the air heater. The regularities and main qualitative types of evolution of relative excess heat in the elements of the model system were revealed.

1. Introduction
Improving the microclimate support systems, managing them and optimizing their work are among the key tasks arising from the operation of buildings and structures for various purposes [1, 2]. The main task of climate systems is stabilization of the microclimate parameters in a room within the specified limits. The complexity of its solution is due to the fact that the corresponding climate systems usually have to operate in a non-stationary mode [3-5]. The emerging instability is a consequence of the constantly changing external and internal thermal perturbations that determine the unsteady operation of the elements of climate systems. The quality of the microclimate parameters and stability of the work of the climate system itself are determined by its reaction to the emerging disturbances, which are commonly called transient. Since any climate system consists of heat and mass transfer apparatus and blowers [6, 7], the stability of the system is determined by its response to the resulting disturbances. This reaction is called transient [8, 9]. The purpose of this work is to study experimentally the evolution of transient heat exchange processes in the elements of the model climate system: air heater-fan-room [10].

2. Experimental installation and measurement techniques
To carry out experimental studies, a special stand, equipped with all the necessary measuring instruments, was built (Fig. 1). The stand consisted of a centrifugal duct fan VR60-30 and an electric air heater EA40-20/12. Air heater performance was controlled by a triac regulator of electric power $AW$, and a fan performance was controlled by a frequency converter of electric current $AF$. The choice of the location of the temperature, speed, and relative humidity sensors at the control points was determined by the position at which the recorded parameter was equal to the value averaged over the channel cross-sectional area. The flow condition parameters were recorded by the PLC150 programmable logic controller in the control sections of the air heater – fan – room channel before and after each element under study equidistantly after one second.
The change in the thermal performance of the air heater was carried out by a triac controller of electric power $AW$ (TTC-25) from 0.0 (0.0 kW) to 1.0 (12 kW) with a step of 0.1 (1.2 kW). The change in fan performance was set by the $AF$ frequency converter (SINUS M) from 0.3 (21 Hz) to -1.0 (70 Hz) with a step of 0.1 (7 Hz). The experiments were carried out in series; their number was determined by a combination of possible modes of mutual operation of the air heater and fan. The transients under study in the first series of experiments (Fig. 2a) were triggered by a stepwise change in the thermal performance of the air heater by the control signal $AW$ at a constant fan capacity. For various transients, the fan performance was set before the experiment in fractions from the maximum frequency of the electric current from 0.3 (21 Hz) to -1.0 (70 Hz) with a step of 0.1 (7 Hz). The initial capacity of the heater was set in fractions of the maximum power of the heater from 0.0 (0.0 kW) to 0.9 (10.8 kW) in increments of 0.1 (1.2 kW) depending on the magnitude of the surge. The magnitude of the surge could vary from 0.1 to 0.5 in increments of 0.1 (1.2 kW). The studied transients in the second series of experiments (Fig. 2b) were triggered by an abrupt change in the fan performance by the control signal $AF$ at a constant heat output of the heater.

For various types of transients, the heat output of the heater was set before the experiment in fractions of the maximum heat capacity from 0.0 (0.0 kW) to -1.0 (12 kW) in increments of 0.1 (1.2 kW). The initial fan performance was set in fractions of the maximum frequency of the electric current from 0.3 (21 Hz) to 0.9 (63 Hz) with a step of 0.1 (7 Hz) depending on the magnitude of the surge. The value of the surge could vary from 0.1 to 0.5 in increments of 0.1 (7 Hz). The transition process in the first series of experiments was triggered by an abrupt change in the thermal performance of the air heater (Fig. 2a), in the second, an abrupt change in the performance of the fan (Fig. 2b). The transient was monitored until the temperature of the air flow stabilized for the objects under study. The
experiments were carried out around the clock in the cold season; the amplitude of outdoor air fluctuations did not exceed 20 degrees. At maximum fan performance, the flow rate was 7.6 m/s, with a minimum of 1.5 m/s. The area of the room in which the stand was located was 50 m², height 3 m. The external fencing consisted of 640 mm thick brickwork. In the system under study, the temperature was recorded at the inlet to the heater and at the outlet ($t_1$ and $t_2$, respectively), at the outlet of the fan and at the inlet of the room ($t_3$) and at the outlet of the room ($t_4$). Simultaneously with the temperature, the flow velocity in the channel ($u$), the pressure drop across the fan ($\Delta P$) and the relative humidity of air in the channel ($\phi$) were measured.

3. Measurement results
As a parameter of similarity, describing the evolution of transient processes in the system, it is proposed to use the relative excess heat

$$ Q' = \frac{Q(\tau) - Q(0)}{Q(t_0) - Q(0)}, $$

where $Q(\tau)$, $Q(0)$ and $Q(\tau_*)$ are the heat released / absorbed by the element at the current $\tau$, initial $\tau = 0$, and final $\tau_*$ time points. Heat $Q(\tau)$ was calculated as the product of the temperature difference at the outlet and input of the element, the mass flow rate and heat capacity of air.

Figures 2a and 2b show the evolution of the relative excess heat of the stream passing through the air heater $Q'_h$, fan $Q'_f$ and room $Q'_r$. The amount of apparent heat released at the current time by the air heater and fan is set according to the relations: $Q'_h = c_a G(t_2 - t_1)$, $Q'_f = c_a G(t_3 - t_2)$, where $c_a$ is the heat capacity of the air, it was assumed to be 1005 J/kg·K. Finally, the amount of heat accumulated and assimilated by the room was determined by the formula: $Q'_r = c_a G(t_4 - t_3)$. The mass flow rate of air in the processing of experimental data was determined by the relationship: $G = \rho(t_3) u F$, where $F$ is the duct section area at the installation site of the speed and temperature sensor $t_3$, $\rho(t_3)$ is the air density determined from the temperature $t_3$.

The experiments carried out allow us to distinguish three basic states of the system elements: active, reactive, and passive. The active state is characteristic of the control element: in the first case it is an air heater (Fig. 2a), in the second, it is a fan (Fig. 2b). The reactive state for the element that reacts to external flow disturbance is for the fan (Fig. 2b) and the air heater (Fig. 2a). A systematic analysis of the active, reactive, and passive states was carried out; the qualitative laws of their evolution were established.

4. Results and discussion
The results were quite typical and qualitatively similar to those presented in the previous section. To demonstrate the revealed regularities, this section considers the evolution of relative excess heat, on the one hand, with two fixed values surges in the heat output of the heater 0.2, but with different fan performance (0.9, 0.7, 0.6, 0.5, 0.4, 0.3), and on the other, with two fixed values of fan performance of 0.4, but with different values of the surge in the heat output of the heater (0.1, 0.2, 0.3, 0.4, 0.5).

Figure 3 shows the evolution of the relative excess heat of the air heater, which is in the active state, with different constant operating modes of the fan (0.9, 0.7, 0.6, 0.5, 0.4, 0.3) and with two variants of the heater’ heat output surge 0.2. The change has a characteristic appearance, the corresponding curves grow monotonously when heated (upper curves) or decrease when cooled (lower curves) the air. In this case, those and other experimental data are almost symmetrical about the abscissa axis (time). The shape of the curves depends weakly on the magnitude of the surge and is determined only by the performance of the fan. For example, at a fan operation mode of 0.9, when the heat capacity of a heater
is surged to 0.2 (Fig. 3), the shape of the curves looks practically the same. The convexity of the curves increases with increasing fan performance.

**Figure 3.** Changes $Q'_h$ vs. time during heating and cooling the flow for different modes of the fan and surge to 0.2 thermal capacity of the heater.

The revealed regularity is confirmed by a comparison of evolution at various surges in the thermal power of the heater, but with a fixed fan performance. Evolution $Q'_h$ for the two variants of the fixed fan performance 0.4 is shown as an example in Fig. 4 with the following surges in the heater' heat output: (0.1, 0.2, 0.3, 0.4 and 0.5). Thus, for an air heater that is in the active state, evolution $Q'_h$ does not depend on the magnitude of the surge in the heat output of the air heater and is determined only by the performance of the fan (compare Fig. 4a and 4b). The greater the convexity of the curve, the higher the fan performance. Evolution curves of released -absorbed (assimilated) heat for a fan in the reactive $Q'_f$ and passive $Q'_r$ state behave similarly.

**Figure 4.** Changes $Q'_h$ vs. time during heating and cooling of the flow for different surges of the heater' heat output and fan operation modes 0.4.

**Conclusions**

The results of studying the work of the model air heater-fan-room system given in this article allow formulating fairly general conclusions about the patterns of operation of such systems. The considered system, although it is a model, is quite representative for practical purposes, since it used typical channel ventilation equipment manufactured by both Russian and foreign manufacturers.

The presence of transients in the elements of climate systems with variable air flow (in air conditioning systems, ventilation, heating with variable coolant flow, etc.) is typical and requires a universal criterion for their assessment, classification and regulation. In this paper, such a criterion is proposed to choose the relative excess heat. This made it possible not only to generalize and normalize the experimental data obtained, but it also made it possible for the first time to identify patterns of evolution of relative excess heat and the qualitative form of functional connections between the initial and final parameters of the heat flow. The performed qualitative identification of experimental data
allows us to describe functionally the evolution of the relative excess heat for a fairly broad class of systems. It is shown that the response to a possible change in the performance of climate system elements is predictable. Thus, the results of the experiments performed allow us to create real tools for regulating and controlling the indoor climate. The subsequent functional and parametric identification of transient heat exchange processes between the flow and the elements of the system according to the experimental data will make it possible to determine the current parameters of the flow state at known initial values. Knowing the functional dependence of the relative excess heat on time, temperature and flow rate, it becomes possible to solve the direct problem, i.e. predict the flow response to a disturbance, both for individual elements of the climate system and the system as a whole, and for the room.

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