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Temperature mode of a room at proportional-integrated regulation of climate systems

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Abstract. The complex including a room serviced by automated microclimate control systems, heat and mass exchange equipment of such systems and their technical means of automation is very complicated for mathematical description of transient processes under conditions of thermal disturbances when solving problems of ensuring comfort of internal meteorological parameters and synthesis of automatic control systems. The nonlinearity of the process of temperature wave propagation in the arrays of enclosing structures leads to a significant error in the case of representation of the room as a linear inertial link. Therefore, the correct mathematical description of transient processes in this case leads to nonlinear differential equations of the Emden–Fowler type, the solution of which in the considered conditions is not expressed in elementary functions and requires numerical methods. One of the most complicated variants in this case is the use of combined proportional-integral law of air conditioning system control in the absence of local heating-cooling systems. Then the solution can be obtained in a parametric form containing a generalized dimensionless parameter of the automated climate control system, including the proportional and integral components transfer factors and the air throughput of the system. The paper shows that in this case, with the growth of the integral component, the dynamic error, i.e., the largest deviation of the room temperature from the setpoint, and the control time are reduced; at the same time, for small moments of time, all the calculated curves describing the temperature behavior asymptotically coincide with the initial heating curve. After reducing the solution to a dimensionless form, the universal dependencies for the indoor temperature behavior depending on the characteristics of the room and regulator parameters, suitable for use in the engineering practice, are obtained. The calculations performed for a typical representative room, in comparison with the relevant experimental data, confirm the reliability of the obtained results and their applicability for mass design.

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1. Introduction

As an object of the research, the proposed work considers the behavior of the indoor air temperature in a room, in which a comfortable set of internal meteorological parameters is maintained by central air systems of microclimate provision equipped with automation devices using a combined proportional-integral (PI-) control algorithm.

It should be noted that the development of scientific research in the field of calculations of the transient thermal regime of rooms, analysis of the operation of the equipment of microclimate systems servicing them and their automation systems, as well as methods of their selection, was traditionally carried out in several parallel directions. One of them concerned the processes in the indoor air and in the enclosing structures under different nature of variable thermal disturbances, both periodic and one-time. In this direction, the developed engineering and analytical theories included, in particular, the theory of thermal stability, the method of “response factor”, and other solutions, including those considering cooling of the buildings during their emergency cutoff from heat supply systems or, on the contrary, initial heating up of premises as a result of a heat supply jump, e.g., [1–3]. This could involve climate systems but mainly in terms of their role in the overall heat balance of the facility as a source of relevant additional heat inputs or their removal. In addition, in some cases, the issues of optimization of thermal protection properties of enclosures and operation modes of heating and heat supply systems could be resolved using economic methods, but the results obtained in this case are not usually generalized to other objects [4]. With the development of computer engineering, numerical methods are certainly added to engineering-analytical methods, although it should be noted that works with their application are still more common abroad. Here, we can note, in particular, publications [5–7]: their advantage can be considered to be the inclusion of experimental data in addition to numerical calculations, which allows identifying the theoretical model.

The processes directly in the heat and mass exchange equipment for inflow treatment or in heat supply systems were usually considered separately, using specific methods including heat balance and heat transfer equations, in particular, and, in [1, 2], including in a dimensionless form, although it is quite obvious that such equipment constitutes a single system with the room and technical means of automation. Finally, the issues of regulation of climatic systems were usually considered from the standpoint of the theory of automatic control, which operates with the concepts of system links represented as a “black box”, actually disregarding the physical essence of the processes occurring in them. To simplify the equations and to obtain solutions in an engineering form, these links are usually considered as linear [1, 2, 6, 7], which, as will be seen from the following description, gives a significant error for transient processes in the rooms. This kind of separate consideration leads to a considerable number of publications devoted to the study of processes in specific individual objects, for example, in ventilated facades with account of the operation of the natural ventilation system only [8], or even the paper of the author of the proposed work [9], in which a very simple dependence for the penetration velocity of a heat wave in a thick-walled cylindrical structure was found. The same features are characteristic for the works, the authors of which are engaged in solving inverse problems of searching for initial data and characteristics, for example, thermophysical properties of building materials, according to the results of temperature behavior measured in one way or another, in particular [10, 11].

In some cases, the use of numerical methods makes it possible to build a comprehensive model taking into account all the above-mentioned systems and factors, but due to their specificity such approaches cannot provide universal engineering methods suitable for tentative estimates and for introduction into normative and reference documents, since they require individual calculations on the computer for each particular object. In addition, this may require knowledge of a large number of parameters of the room and engineering systems, which should be used as input data for the relevant program, e.g., [12]. Still less suitable for the engineering practice, due to the lack of specific sufficiently simple analytical or graphical dependencies, are the methods that have been recently gaining ground, based on the so-called fuzzy logic [13, 14] and other cross-system concepts [15, 16], as well as works devoted to the general principles of energy saving in buildings through various solutions to reduce energy consumption, including automation of climate systems [17–19]. Thus, to date, there are practically no models and solutions that allow joint consideration of nonstationary processes in the rooms serviced by automated microclimate systems, with account of their physical essence, interconnection and mutual influence and available for application in mass design.

In previous works, the author made attempts of such a comprehensive consideration for the simplest variants of laws of control: proportional (P-) [20] and integral (I-) [21, 22]. However, in practice, especially for air conditioning systems, more complex algorithms are used for more accurate maintenance of internal meteorological parameters and ensuring the required comfort in the serviced zone of the room. One of the most common of them is a combination of the said laws, namely PI-control combining some of their advantages, in particular, the relative speed of P-controllers and the zero static error of integrators. Therefore, it seems relevant to continue the research of transient modes in indoor and climatic systems for this case as well.

Thus, the object of the research is to build a mathematical model of nonstationary thermal processes in a complex including a room and heat and mass exchange equipment of automated central air conditioning systems servicing it, controlled by a PI-algorithm, and to develop a methodology for evaluating such a regime, suitable for use in engineering practice and taking into account all the main parameters of the system.

Thus, the tasks of the work are as follows:

– to formulate a system of differential equations describing transient processes in a room under a jump-like thermal disturbance and its assimilation by central air climatic systems equipped with a PI-controller;

- to obtain the solution of this system describing the behavior of indoor air temperature in the specified conditions, recording it in a dimensionless form and its analysis under different initial data, including the change of the ratio between P- and I-components of the law of control;
- to search for possible asymptotics of the detected solution at limiting values of the room and regulator parameters;
- to compare the results with the available experimental data, including those obtained by other authors, and to construct correlations linking the system parameters and indoor temperature which would be suitable for application in mass design.

2. *Methods*

As in the works [20–22], using the concept of excess temperature $\theta_{in} = t_{in} - t_{in.0}$, where $t_{in.0}$ is a controlled level of inside air temperature t_{in}, or the so-called input reference, for a sudden change in thermal disturbance and its compensation due to the operation of a central air conditioning system that simultaneously performs ventilation functions, the equation of the convective heat balance for the room as a whole can be written as follows:

$$Q_{in} + G_s c_a \left(\theta_s - \theta_{in}\right)/3.6 - B\sqrt{\tau}\frac{\mathrm{d}\theta_{in}}{\mathrm{d}\tau} = 0, \tag{1}$$

where Q_{in} is the inflow of explicit convective heat into the room air from sources, W; τ is the time interval, s, from the moment of occurrence of a thermal disturbance; G_s is the mass flow rate of the intake air, kg/h, which is equal to the flow rate of the exhaust G_{ex}, since the air balance of the room, unlike from thermal, almost happens instantly-stationary; c_a is the specific mass heat capacity of air equal to 1.005 kJ/(kg·K); $\theta_s = t_s - t_{in.0}$ is the excessive inflow rate, K.

Similarly [20–22], parameter B, W·s$^{1/2}$/K, in equation (1) is determined by the formula:

$$B = \sum \left\lfloor A_m \sqrt{\lambda c \rho} \right\rfloor_i, \tag{2}$$

where λ, c and ρ are, respectively, the thermal conductivity, W/(m·K), the specific heat capacity, J/(kg·K), and the density of the material layer of the i-th massive enclosing structure (external and internal walls, ceilings, partitions) facing inside the room; A_m is the area of each of the listed fences, m^2.

If the value of θ_{in} is automatically maintained by a regulator implementing a continuous PI-law, the complementary constraint equation is easiest to write in this form:

$$\frac{\mathrm{d}\left(\theta_s - \theta_{in}\right)}{\mathrm{d}\tau} = -K_i\theta_{in} - K_p\frac{\mathrm{d}\theta_{in}}{\mathrm{d}\tau}, \tag{3}$$

where K_i is equivalent coefficient of transfer of integral component of the automated system s^{-1}, by the channel "$\theta_{in} \rightarrow$ derivative of the difference between $\theta_s - \theta_{in}$", K_p is the same for proportional component by the channel "$\theta_{in} \rightarrow$ difference between $\theta_s - \theta_{in}$" (dimensionless). Differentiating term-by-term (1) by τ in order to be able to substitute expression (3) there, we bring (1) to the canonical form:

$$\frac{\mathrm{d}^2\theta_{in}}{\mathrm{d}\tau^2} + \left(\frac{1}{2\tau} + \frac{C}{\sqrt{\tau}}\right)\frac{\mathrm{d}\theta_{in}}{\mathrm{d}\tau} + \frac{D}{\sqrt{\tau}}\theta_{in} = 0. \tag{4}$$

The parameters C, $s^{-1/2}$, and D, $s^{-3/2}$ can be determined by the formulas:

$$C = \frac{G_s c_a K_p}{3.6B}; \quad D = \frac{G_s c_a K_i}{3.6B}. \tag{5}$$

Thus, formally (4) coincides with the equation obtained previously by the author in [21], but the coefficients C and D have a different physical meaning and are expressed differently in terms of the original data. After substitution, the relation $z=\sqrt{\tau}$ (4) can be reduced to a somewhat simpler form, in which the first of the terms including the first derivative disappears:

$$\frac{\mathrm{d}^2\theta_{in}}{\mathrm{d}z^2}+2C\frac{\mathrm{d}\theta_{in}}{\mathrm{d}z}+4zD\theta_{in}=0. \tag{6}$$

It is easy to see that for $C=0,$ i.e., with the exclusion of the proportional component of the regulator, (6) will be an Emden–Fowler equation [23, 24]. However, in the general case under consideration, at $C\neq 0,$ the methods used in these works are no longer suitable, although some general properties of the equation and, as a consequence, its solutions associated with nonlinearity due to the presence of a multiplier z at the third term are still preserved. Therefore, we transform the resulting equation further in order to highlight the singularity at $z=0,$ namely, we present the solution as a product of $\theta_{in}=zf(z)$ and then we make another replacement of the independent variable, that is, we assume $x=4Dz^3$ – a dimensionless time argument, from which, respectively, $z=\left(\frac{x}{4D}\right)^{1/3}$. Then from (6), we get:

$$9x\frac{\mathrm{d}^2 f}{\mathrm{d}x^2}+\left[12+6C\left(\frac{x}{4D}\right)^{1/3}\right]\frac{\mathrm{d}f}{\mathrm{d}x}+\left[1+\frac{2C}{(4Dx)^{1/3}}\right]f=0. \tag{7}$$

If we now assume that $K=\frac{2C}{(4D)^{1/3}}=K_p\sqrt[3]{\frac{2}{K_i}\left(\frac{G_s c_a}{3.6B}\right)^2}$ is a generalized dimensionless parameter of the automated climate system, we can finally obtain:

$$9x\frac{\mathrm{d}^2 f}{\mathrm{d}x^2}+\left[12+3Kx^{1/3}\right]\frac{\mathrm{d}f}{\mathrm{d}x}+\left[1+\frac{K}{x^{1/3}}\right]f=0. \tag{8}$$

Obviously, $f(0)=1,\ \mathrm{d}f(0)/\mathrm{d}x=1/12$ should be taken as the initial conditions for (8).

It is clear that the solution of equation (8) gives a universal dependence that is applicable for any objects and conditions. Therefore, in a dimensional form, the ratio for the current ambient air temperature will look like this:

$$\theta_{in}=\frac{2Q_{in}\sqrt{\tau}}{B}f(x). \tag{9}$$

At the same time, if we consider (8) for little periods of time, we can notice that the first term can already be neglected, and we get a 1st-order equation with separable variables. In this case, its solution will be represented by $f(x)=\exp(-f_1(x)),$ where the auxiliary function f_1 is defined as an integral:

$$f_1(x)=\frac{1}{12}\int_0^x\frac{1+Kx^{-1/3}}{1+Kx^{1/3}/4}\,\mathrm{d}x, \tag{10}$$

or asymptotically for small K and x:

$$f_1(x)=\frac{1}{12}\left[\left(1-\frac{K^2}{4}\right)x+\frac{3}{2}Kx^{2/3}\right]. \tag{11}$$

3. Results and Discussion

Fig. 1 shows the type of functions $f\left(x\right)$ for various values of parameter K from 0 to 4, obtained by numerical solution (9) using a computer program by the Runge–Kutta method of the 4th order. According to (5) and (6), we have here $x = \frac{4G_{in}c_aK_i}{3.6B}\tau^{3/2}$.

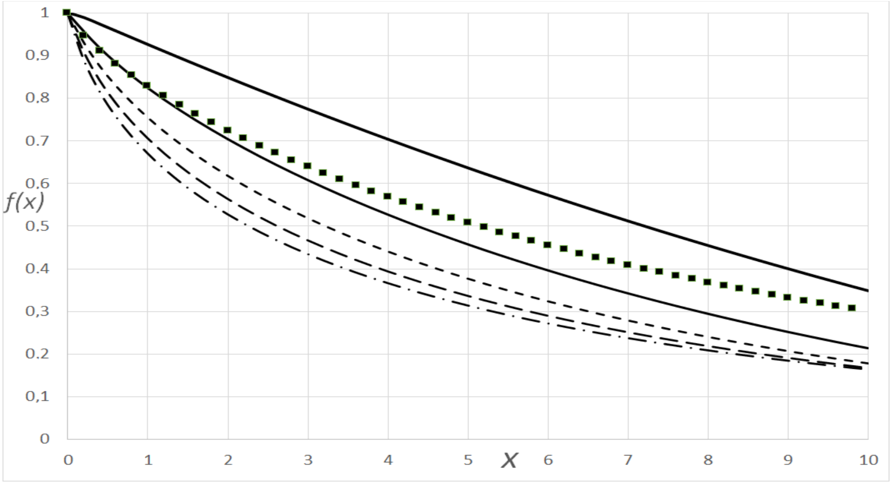

Figure 1. Dimensionless dependence $f(x)$ for K = 0, 1, 2, 3, and 4 (lines from top to bottom), for K = 1 with the use of (11) (square markers).

By virtue of the above, the constructed graphs can serve as a universal nomogram, which can be used in engineering calculations in combination with formula (9). For comparison, square markers show a curve using expression (11) at K = 1.

The nature of the change of θ_{in} for one of the rooms in a public building, previously considered by the author in [20, 21], is shown for clarity in Fig. 2. At the same time, it was assumed that Q_{in} = 500 W, B = 12,000 W·s$^{1/2}$/K in accordance with the structural solutions of the room and the thermal properties of its enclosing structures, the inflow flow rate G_s = 430 kg/h and K_p = 0.84, from where C = 0.0084 s$^{-1/2}$, and the coefficients K_i were considered equal to 0 (that is, for the limiting case of purely proportional regulation), 8.4·10^{-6}, 8.4·10^{-5}, and 8.4·10^{-4}, whence, respectively, D = 0, 8.4·10^{-8}, 8.4·10^{-7}, and 8.4·10^{-6} s$^{-3/2}$. Then, the parameter K by its definition will acquire the values ∞, 2.4, 1.11, and 0.52. With the exception of the first one, they lie within the limits for which the graphs in Fig. 1 are calculated. As for K_i = 0, it requires separate consideration, since there is an analytical solution obtained by the author in [20], although the original equation (6) can be solved numerically and directly at D = 0, and from (5), the corresponding result is obtained by the limiting transition $K \to \infty$.

From Fig. 2, it is easy to see that, as in the case of control objects represented as a linear inertial link, the appearance of an integral component of the regulator immediately leads to the fact that the residual temperature deviation, or static control error, becomes zero, and the maximum value of θ_{in} gradually decreases with increasing K_i, as does the control time. At the same time, at small τ, all curves coincide asymptotically, which confirms the conclusions about the proportionality of θ_{in} and $\sqrt{\tau}$ in the mode of initial heating or cooling of the room.

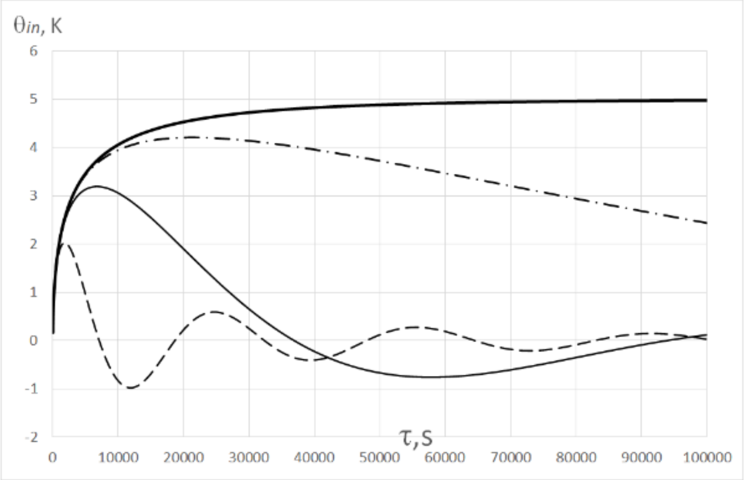

Figure 2. Dependence of θ_{in} on time for a specific room at K_i = 0, 8.4·10^{-6}, 8.4·10^{-5}, and 8.4·10^{-4} (from top to bottom).

In Fig. 3, the dotted line shows the results of experimental measurements in the representative room shown above at Q_{in} = 500 W. The measurements were carried out similarly to those presented in [21] using Testo 0560 1110 Mini penetration thermometer with a division of 0.1°, which was installed in the center of the room at a height of 1 m from the floor. In fact, the regulation of the air conditioning system was carried out positionally (“on/off”), but due to the high frequency of switching, it was approaching continuous. The solid line represents the data of direct numerical integration of the original equation (6) using a computer program by the 4th-order Runge–Kutta method. The best match within the accuracy of the measuring device is observed at K_p = 0.0167 and K_i = 0.0125 s^{-1}, which corresponds to the values of the parameters C = 8.33·10^{-5} $s^{-1/2}$ and D = 6.25·10^{-5} $s^{-3/2}$ with the value of B = 24000 W·$s^{1/2}$/K, that is, in 2 times higher than was accepted for calculations in Fig. 2, since a separate room was viewed here and, thus, the temperature wave propagated only in one direction.

It should only be kept in mind that for small τ, the coincidence of theoretical and experimental dependence turns out to be somewhat worse, which can be explained by the influence of the thermal inertia of the indoor air volume, which was not directly taken into account when writing the initial equation (1), as a result of which the measured temperatures are obtained up to a certain point lower than according to (6). The same applies to the rightmost area of the graph, after the temperature has passed the maximum. This issue has already been analyzed to some extent by the author in [20, 21], where it was noted that, apparently, another limitation used in conclusion (1) is already beginning to affect here, namely the assumption that the temperature wave does not have time to penetrate to the outside the surface of the fence or up to its axis of symmetry. Nevertheless, since from the point of view of using the results in engineering practice, we are primarily interested in the value and moment of the greatest temperature deviation, the coincidence can be considered satisfactory.

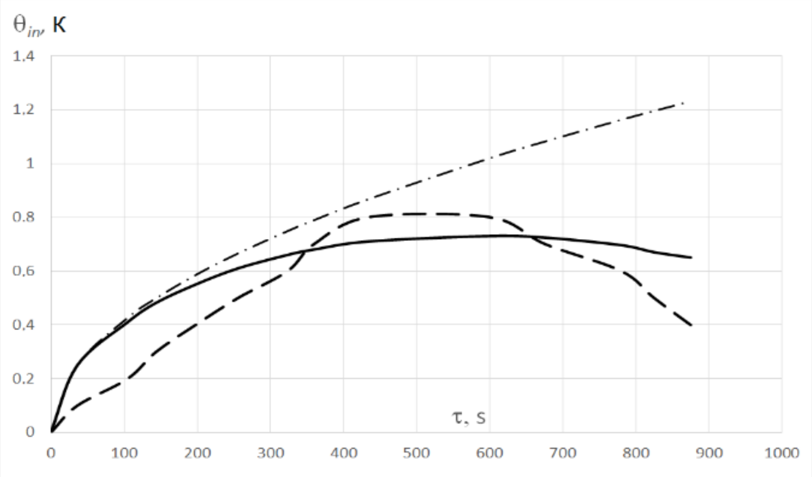

Figure 3. Dependence of θ_{in} on time for a specific room in a full-scale experiment (solid line – calculation by (6), dotted line – experiment, dashed dot – initial heating curve).

Thus, comparison with experimental data can be used to identify a mathematical model, that is, to identify the actual parameters of the control system, which would be difficult to determine directly. For comparison, the dashed dot line depicts the process of initial heating of a given room in the case of Q_{in} = 500 W and the absence of compensation for heat gain by expression (9) at $f\left(x\right)$ = 1 [20, 21]. It is not difficult to notice that both the theoretical solution (6) and the experimental results, although to a somewhat lesser extent, asymptotically coincide with the initial heating curve at small $\tau,$ which further confirms their correctness.

In addition, Fig. 4 shows a comparison of the calculation results for (6) for the same room but with slightly different initial parameters, namely K_p = 0.071 and K_i = 0.025 s^{-1}, which corresponds to the values of parameters C = 3.55·10^{-4} s$^{-1/2}$ and D = 1.25·10^{-4} s$^{-3/2}$ also at a value of B = 24000 W·s$^{1/2}$/K, with the data of full-scale measurements of the non-stationary thermal regime of the room, which is equipped with the automated air heating system under similar control conditions given in [25] (dotted line), after normalization by the magnitude of the maximum temperature deviation.

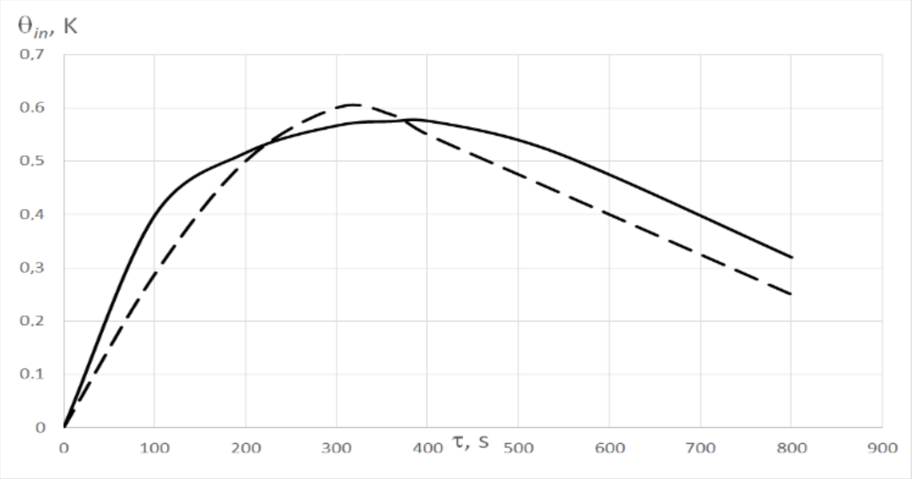

Figure 4. Dependence of θ_{in} on time for the room under study (solid line – according to equation (6), dotted line – measurement data [25]).

It can be seen that in almost the entire time interval under consideration since the appearance of the thermal disturbance, experimental measurements give a similar nature of dependence, which further confirms the theoretical provisions of the proposed work, and the discrepancies have the same appearance as in Fig. 3 and can be explained in a similar way.

4. Conclusion

- It has been shown that the mathematical description of transient processes at a jump-like thermal disturbance in the room serviced by central air conditioning systems, formulated in the previous works of the author in the form of the differential equation of the Emden–Fowler type, is a limiting case arising at exclusion of the P-component of the regulator.
- It is noted that the solution of the obtained system of equations for the PI-law of control still retains some properties characteristic of the Emden–Fowler type equations, namely, it has a specific feature at $\tau = 0$ and admits a representation in the form of a product of $zf\left(z\right),$ where $z = \sqrt{\tau}.$
- It has been proved that the presence of the P-component of the regulator can be taken into account by introducing a generalized dimensionless parameter of the automated climatic system $K,$ including the P- and I-components transfer factors and the air throughput of the system.
- It has been shown that the reduction of the solution to a dimensionless form allows constructing universal correlations for the behavior of the indoor temperature depending on the characteristics of the room and regulator parameters, suitable for use in engineering calculations, its argument including the product of the transfer factor of the I-component and $\tau^{3/2}$ value.
- It has been demonstrated that the dynamic error, i.e., the largest deviation of the room temperature from the setpoint, and the control time decrease with the growth of the I-component;

at the same time, for small moments of time, all the calculated curves describing the temperature behavior asymptotically coincide with the initial heating curve.

- It is noted that the change in the behavior of the solution at the appearance and strengthening of the integral component of the regulator qualitatively coincides with the known regularities in the representation of the room as a linear inertial link.
- It has been confirmed experimentally using field measurements in a typical representative room that the discrepancy between the actual and theoretical values of θ_{in} is within the measurement accuracy and the usual error of engineering calculation, at least in the zone of maximum temperature deviation from the setpoint, which is of the greatest interest for engineering applications.

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