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Thermal mode of a room with integrated regulation of cooling systems

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Abstract. An air-conditioned room with automatic regulation of the climate systems serving it using complex control algorithms is one of the most difficult objects for calculating non-stationary thermal regime, so to date, this method is insufficiently studied. At the same time, such objects are typical when organizing the internal microclimate of civil buildings. In this paper, we consider the mathematical formulation and solution of the problem of changing the temperature of internal air in a room equipped with automated local cooling systems and a background supply ventilation system under variable thermal influences. The main equations connecting the most important components of the heat flow in the room are analyzed. The dependence on time for the deviation of the room air temperature from the setpoint is presented, and the expression for the moment of time at which the maximum temperature deviation is observed, with a jump-like change in the heat flow from heat sources in the case of regulating the equipment of local cooling systems according to the integral law. Calculations were performed to confirm the analytical solution obtained using a finite-difference approximation of the differential equations of heat balance and heat transfer on the example of one of the currently existing residential buildings in the climatic conditions of Moscow, taking into account the structural characteristics of the building and the thermal properties of its enclosing structures. It is noted that the greatest deviation of the temperature from the setpoint (dynamic control error) in the first approximation is inversely proportional to the cubic root of the transmission coefficient of the regulator, as well as the moment of time for which this deviation is observed. The obtained relations are proposed to be used for an analytical assessment of the non-stationary thermal regime of an air-conditioned room served by local cooling systems equipped with an automation system with an integral law of regulation, to check the conditions of human comfort and safety, as well as to determine the required parameters of the regulator.

1. Introduction

The proposed publication studies the change in the indoor air temperature in a room equipped with automated local cooling systems and a background supply ventilation system under the conditions of variable thermal effects.

The necessity to maintain a set of indoor microclimate parameters in the range determined by human comfort conditions or ensuring technological process is primarily associated with health and safety requirements. In practice, hygrothermal regime stabilization is carried out primarily through the automatic regulation and control by climate systems, taking into account the heat resistance of buildings themselves. At the same time, since the thermal effects on the premises and their heat loss are found to be variable in almost all cases, this task is of a significantly dynamic nature.

The problem of calculating the temperature in air-conditioned rooms with the emergence of thermal disturbances has long been studied. Barring some simplified analytical solutions to this problem, given in the reference and educational literature and mainly related to insulated enclosure structures in the building cooling mode a during an emergency shutdown of heat supply or with periodic thermal disturbances without taking into account the overall heat equilibrium of the room, we can note more complex works recently

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appeared, for example [1], however, their results often prove difficult to use in engineering calculations because of their considerable complexity. At the same time, using numerical methods for studying dynamic modes and their modeling is becoming increasingly widespread. In this regard, the following publications [2–3], are of particular interest and approaches of this kind are now especially incident to foreign studies, of which we can name such fairly distinctive works as [4–6]. However, various solution options are related to specific types of facilities used in limited areas and operating in special conditions, for example, for underground pipelines [7], in the presence of changes in phase [8] or for ventilated facades integrated with natural ventilation systems [9]. We should also mention one of the previous publications of the author [10], where a reasonably simple solution for the temperature-wave propagation in a thick-walled cylinder was obtained. In addition, the authors of certain works in this area are rather focused on solving the inverse problem – determining the thermal and physical characteristics of a material based on a study of temperature fluctuations [11–13], or using thermographic methods [14]. There are works [15–20] devoted to modeling processes in the room as a whole. A sufficiently accurate analytical solution for calculating temperature fluctuations is given in [15], however, it applies only to enclosure structures, and [16] considers a remarkably detailed simulation-type multiparameter numerical model, but it refers to the cold season with heating. Publications [17–20] also possess a certain integrity, in particular, [17–18] relate to the use of fuzzy logic principles in the organization of microclimate management, and [19–20] are more focused on the methods of automatic control theory, but their results are quite complicated for using them in engineering practice. Finally, there are also works related to the general principles of building engineering systems management and energy saving in these conditions, for example, [21–23], but due to their generality, they also do not contain specific dependencies of interest to us.

The author in [24] managed to obtain an analytical solution to the problem of lowering the temperature in the room during an emergency shutdown of heat supply using the heat equilibrium differential equation for the room as a whole. However, it remains advisable to conduct studies for other cases where such an equation lends itself to integration, especially if regular air conditioning systems equipped with the necessary equipment for their automatic regulation are functioning in the room, which allows maintaining internal meteorological parameters in the required range.

Although proportional control units are the most simply arranged, they are known to have a drawback, which may be significant under certain circumstances, namely, a non-zero steady-state error of maintaining the controlled parameter. Therefore, preference in a number of situations is given to control algorithms with an integral component, where the control action is proportional to the sum of the accumulated deviation from the onset of thermal disturbance and the control process is carried out until this deviation is completely eliminated.

At the same time, a situation where stabilization of the indoor air and fences temperature occurs through the local cooling systems, for example, such as “chiller-fan coil” or split systems, employing complete recirculation of the air used and thus not participating in the general air balance of the room, or even with the help of panel-radiant cooling systems, the most common of which are cooled ceilings, is quite common. There, a mechanical supply system plays mainly a sanitary and hygienic role, providing the necessary cleanliness of the internal air, and in cases where it ensures cooling of the supply, it is done primarily so as not to increase the heat load on local cooling systems due to the inflow of high temperature air into the room. In this case, it will be shown that, if switching to the regulation of the inflow temperature and in the absence of local cooling systems, the corresponding solution will be a special case of a more general one, directly considered in this paper.

Thus, the relevance of the proposed study lies in the need to search for sufficiently accurate and physically justified dependences on the time of the change in the temperature of the indoor air in a room cooled with local automated systems at the same time acceptable for engineering use, taking into account the amount of heat input, room heat resistance and control unit characteristics, as well as the relationship of these characteristics with the maximum temperature deviation from a given value. The results obtained may be applicable to a very wide range of facilities of this type.

The aim of the paper is to develop methods for calculating the dynamic thermal regime of a room where local automated cooling systems are operated in the presence of a background unregulated inflow. The objectives of the study are:

- compilation of the basic system of differential equations describing the heat equilibrium, heat transmission and heat exchange in a room air-conditioned with automated climate systems;
- construction of an analytical solution of this equation with abrupt thermal disturbance and regulation of cooling systems according to the integral law;
- obtaining analytical dependences for the maximum deviation of the room temperature from the set point and for the point in time when this deviation is observed, and their confirmation based on a comparison of theoretical results with the data of programmed generation.

2. Methods

The equation of convective heat balance for indoor air within the framework of its single-are model can be shown here as follows:

$$Q_{in} + G_s c_a \left(t_s - t_{in}\right) / 3.6 - Q_c - B\sqrt{\tau}\,\frac{dt_{in}}{d\tau} = 0\,, \tag{1}$$

Where Q_{in} is the sensible heat inflow in the room air from sources, W; τ is time interval, s, from the moment of the beginning of variable thermal effect; G_s is mass flow rate, kg/h, which is usually considered as equal to the value of the flow rate G_{ex} due to the almost instantaneous stationary state of the air equilibrium of the room compared to the heat equilibrium; c_a is specific heat of air equal to 1.005 kJ/(kg K); t_{in} is indoor air temperature, °C; t_s is inflow temperature, °C. Equation (1), compared with that presented in [24], contains an additional term of Q_c, representing the value of the regulated heat flow, W, from local cooling systems, which is designed to compensate for heat input. For the same reason, it is now assumed in (1) that t_s = const.

As in [19], parameter B in expression (1) is calculated by the formula:

$$B = \sum\left[A_m\sqrt{\lambda c\rho}\right]. \tag{2}$$

Here λ, c and ρ are the thermal conductivity, W/(m K), specific heat, J/(kg K), and the density of the material of the layer of the i-th massive fence facing the inside of the room, respectively, for example, of external and internal walls and partitions, as well as floors; A_m is area of each of the enclosure structures listed, m^2.

If the value of t_{in} is automatically supported by a control unit implementing a continuous integral law with the necessary change in the value of Q_c, the additional constraint equation the most conveniently written in this form:

$$\frac{dQ_c}{d\tau} = K_c\left(t_{in} - t_{in.0}\right). \tag{3}$$

Here, K_c is the equivalent transfer ratio of the automated system, W/(K s), over the channel $t_{in} \rightarrow$ derivative of Q_c". Using the concept of excess temperature $\theta_{in} = t_{in} - t_{in.0}$, where $t_{in.0}$ is a controlled level of t_{in}, or the so-called control point, and differentiating (1) term by term by τ for the possibility of substituting expressions (3) there, we can write (1) in the canonical form:

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

The parameter of C, s$^{-1/2}$, can be determined by the formula:

$$C = \frac{G_s c_a}{3.6B}. \tag{5}$$

It is easy to see that differential equation (4) refers to linear homogeneous equations of the second order, albeit with variable coefficients, as a result of which the simplest solution methods like operation variants cannot be directly applied to it. However, if we use substitution $z = \sqrt{\tau}$, it is reduced to a somewhat simpler form in which one of the terms containing the first derivative is gone:

$$\frac{d^2\theta_{in}}{dz^2} + 2C\frac{d\theta_{in}}{dz} + 4zD\theta_{in} = 0\,, \tag{6}$$

where $D = K_c/B$, s$^{-3/2}$.

Generally speaking, this equation may be solved analytically, and it is the most easily found in the form of a series expansion in powers of z by the method of undetermined coefficients. Moreover, as initial conditions at τ = 0, we have θ_0 = 0 and, obviously, $d\theta_{in}/dz = 2Q_{in}/B$, since at the initial moment of time for a given law of control in (1), only the first and last terms will be non-zero. In this case, the desired function is recorded in the following form:

$$\theta(z) = \frac{2Q_{in}z}{B}\begin{bmatrix} 1 - Cz + \frac{2}{3}C^2z^2 - \frac{1}{3}\left(C^3 + D\right)z^3 + \frac{1}{15}C\left(2C^3 + D\right)z^4 - \\ -\frac{1}{45}C^2\left(2C^3 + 3D\right)z^5 + \frac{1}{315}\left(4C^6 + 16C^3D + 10D^2\right)z^6 - ... \end{bmatrix}. \quad (7)$$

Note that the parameter of D, which directly includes the quantity K_c, appears only with members of the series, starting with z^4, i.e. τ^2. It is easy to verify that for not very large τ and z values, respectively, the series convergence is rather fast.

At the same time, it can be obtained that in the limiting case of D = 0, i.e. if the automatic control system does not function, we have the following instead of (6):

$$\frac{d^2\theta_{in}}{dz^2} + 2C\frac{d\theta_{in}}{dz} = 0. \quad (8)$$

By substitution $X = \frac{d\theta_{in}}{dz}$ this equation is reduced to an equation of the first order with separable variables, from which we obtain the following taking into account the initial conditions and the expression for parameter of C:

$$\theta_{in} = \frac{3.6Q_{in}}{G_s c_a}\left(1 - exp\left[-\frac{2G_s c_a}{B}\sqrt{\tau}\right]\right). \quad (9)$$

It is easy to verify that the first three terms in the decomposition of solution (9) into the Taylor series in in the vicinity of the point τ = 0 coincide with those for (7) therefore, (9) is indeed an asymptotic approximation for (7) at $D \to 0$. Moreover, from the same decomposition it follows that for small τ, dependence (9) will be equivalent to the following:

$$\theta_{in} = \frac{2Q_{in}}{B}\sqrt{\tau}. \quad (10)$$

As in other cases considered earlier, this coincides with the formula for the natural change in temperature in the room with no automatic control. At the same time, at $\tau \to \infty$ the expression (9) gives us $\theta_{in} \to \frac{3.6Q_{in}}{G_s c_a}$, which corresponds to the steady state during the operation of a mechanical ventilation system with a constant inflow temperature.

When the automatic control system is in operation, the value of θ_{in} reaches its peak value, and then approaches zero again, as it should. The moment where $\theta_{in} = \theta_{max}$, can be determined by calculating the derivative of $d\theta_{in}/dz$ and equate it to zero. From (7) we obtain the following decomposition:

$$\frac{d\theta(z)}{dz} = \frac{2Q_{in}}{B}\left[1 - 2Cz + 2C^2z^2 - \frac{4}{3}\left(C^3 + D\right)z^3 + \frac{1}{3}C\left(2C^3 + D\right)z^4 - \frac{2}{15}C^2\left(2C^3 + 3D\right)z^5 + ...\right]. \quad (11)$$

It is easy to see that this series also converges rather quickly. Since obviously $\frac{2Q_{in}}{B} \neq 0$, for $d\theta_{in}/dz = 0$ expressions in brackets must be equal to zero.

3. Results and Discussion

For clarity, we give an example of calculating the value of t_{in} by formula (7) in the room with the parameters of B = 12000 W c$^{1/2}$/K; C = 0.01 s$^{-1/2}$ (corresponding to G_s = 430 kg/h) and K_c = 0.25 W/(K s), from which $D = K_c/B$ = 2.083 10^{-5}, s$^{-3/2}$. When calculating, the initial level of $t_{in.0}$ was taken equal to +20 °C, a jump in heat gain (design cooling load) Q_{in} = 500 W, and the value of B was determined taking into account the actual thermal and technical characteristics of building materials in the building and the geometry of the room. The dimensions of the room and its layout with some of the symbols used are shown in Fig. 1. The area is 14 m^2, inner structures made of reinforced concrete with density 1200 kg/m^3 and a total area of 64 m^2, which accounted for half, since it is assumed that the neighboring rooms are in similar conditions, and the influence of temperature is on both sides, the outer wall of lightweight concrete with a density of 500 kg/m^3 and a 7 m^2 with window with an area of 1.8 m^2.

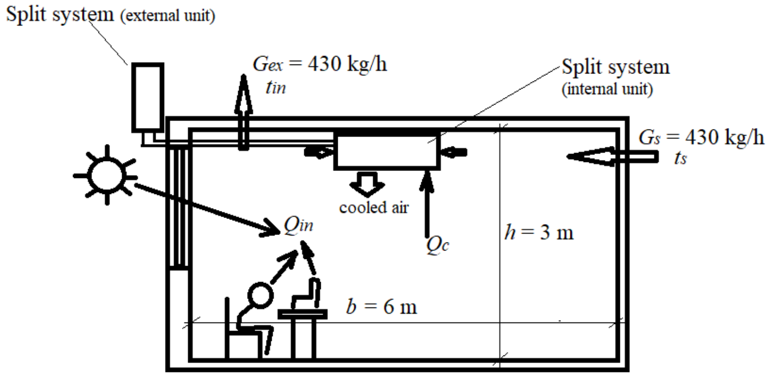

Figure 1. The scheme design of the room.

The corresponding graph is shown in Fig. 2 by a solid line. For comparison, the dotted line shows the results of numerical simulation for the same conditions using the computer program developed by the author and based on a direct solution of the system of differential equations of dynamic heat transmission in fences and heat exchange on their surfaces [24].

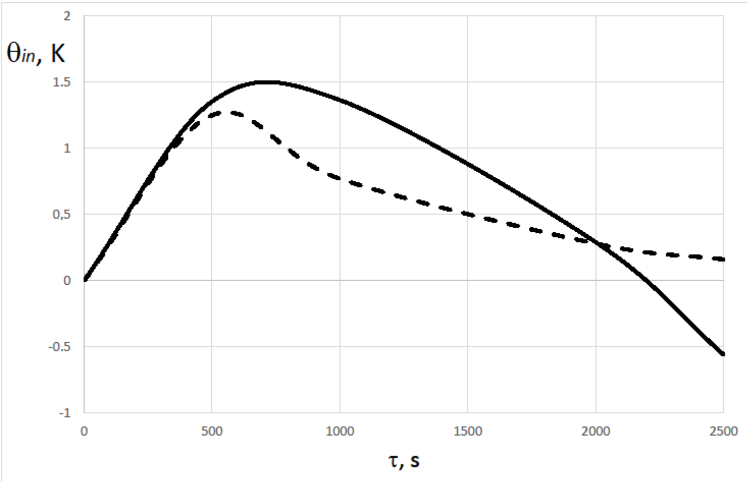

Figure 2. Dependence of θ_{in} Time dependence for the design room (solid line – according to the formula (7), dotted line – numerical calculation).

It is easy to notice that the coincidence of the data of numerical calculation and expression (11) is quite satisfactory, especially at the initial moments of time, at least from the viewpoint of the maximum deviation t_{in} and the time when it takes place. In the future, the calculation of (11) begins to give somewhat overestimated, and later, on the contrary, underestimated results, especially for bigger τ, when the curve under the conditions when using integral control should asymptotically tend to zero, which is reflected in the results of numerical calculation. It seems that the restrictions used in writing the original equation (1) are already beginning to affect here, namely, the assumption that the temperature wave does not have time to penetrate to the outer surface of the fence or to its axis of symmetry. However, since we are primarily interested in the value and moment of the greatest temperature deviation, the solution can be considered satisfactory.

The reliability of the obtained result was confirmed by a full-scale experiment for the same room for which the graphs in Fig. 2 were obtained. In this case, the jump in heat gain was simulated by switching on a convective electric heater of the appropriate power. It should only be borne in mind that in this case, in contrast to the comparison with the software calculation, which is focused on the simultaneous flow of similar processes in all neighboring areas, to construct the theoretical curve for (7), you need to take twice the value B = 24000 W c$^{1/2}$/K, since the temperature wave in the fences will propagate only in one direction. To compensate for heat surpluses, an internal split-system unit was used, which is regulated positionally, but due to the high frequency of switching on and off, this regulation is close to integral. The value of K_c in this case was 0.5 W/(K s), so the parameter D remains equal to 2.083 10^{-5}, s$^{-3/2}$. Fig. 3 shows the solid line graph of the internal temperature deviation calculated from (7), the dotted line shows the experimental data. It is easy to see that given the measurement accuracy of 0.1 degrees, the match is quite good.

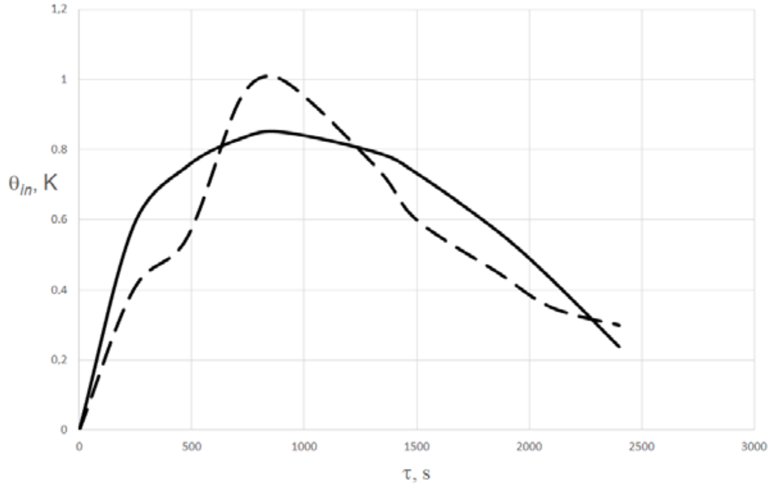

Figure 3. Dependence of θ_{in} Time dependence for the design room (solid line – according to the formula (7), dotted line – experiment).

It should be borne in mind that, based on the structure of equation (11), which includes two parameters of C and D, fairly simple formulas for θ_{max} and the corresponding level of z in this mode, are not obtained so easily. However, if we hold only the first terms containing C and D in (11), we can find the approximate dependence for z in the following form following linearization:

$$z = z_0 \left(1 - \frac{1}{1 + N z_0} \right), \tag{12}$$

where $z_0 = \dfrac{0.956}{\sqrt[3]{D}}$ is the value of z at $\theta = \theta_{max}$ and C = 0, obtained by dimensional analysis [25]; $N = 1.82 \dfrac{\sqrt[3]{D^2}}{C}$ is the correction related to the emergence of unregulated inflows. Let us note that when substituting z_0 in (7) we can obtain that the value of θ_{max} must also be inversely proportional to the parameter of D raised to the power of 1/3. In the conditions of our example, we find z_0 = 34.7 s$^{1/2}$ and N = 0.14, therefore, z = 28.7 s$^{1/2}$, or τ = 824 s, which closely enough coincides with that shown in Fig. 2. Obviously, for G_s and, consequently non-zero C, the both values of θ_{max} and z will be lower than in the absence of background air exchange.

At the same time, it is easy to verify that, for C = 0, only terms with powers of z equal to 1, 4, 7, etc. remain in the decomposition (7) and thus, such a solution will be another limiting variant (7), being fair if there is no unregulated temperature inflow. In physical terms, this will mean replacing the regulated local cooling system with the direct maintenance of the internal microclimate by regulating the temperature of the influx of the central air conditioning system. From expression (12) it follows that in this case z_0 and, evidently, θ_{max} increase, i.e. the addition of unregulated flow increases the room's own heat resistance. It can also be noted that the general nature of dependence (7), shown in Fig. 2, is similar to the results obtained, in particular, by the authors of [16–17] under similar initial conditions, and the concept of the approach considered in the present paper as a whole, the obtained temperature deviations are of the same order as those given in publications [26–27] in similar modes, and a number of elements of the mathematical formulation and solution of the problem under study are close to the data contained in [3], [15], [18] and some other works, therefore, the results of this study may be considered fairly reliable and reasonable.

4. Conclusions

1. It is shown that the analytical solution obtained in the paper describing the change in the value of tin in a room employing a background unregulated inflow and automated local cooling systems, provided that they are regulated according to the integral law, describes quite well the real process of heating or cooling for an abrupt change in heat supply, in any case, at the values of τ which are not too large.

2. It is noted that, as in other regulation modes, the dependence for tin has an asymptotically exponential character, however, unlike solutions showing the behavior of the indoor temperature for lengthy moments of time, the argument of the exponential function now contains the value $\sqrt{\tau}$ in explicit form, but not that of τ.

3. It is found that the largest deviation of tin from the set point (dynamic control error) is proportional to the magnitude of the thermal disturbance and, as a first approximation, inversely proportional to the regulator transfer ratio raised to the power of 1/3, as the point in time for which this deviation is observed.

4. It is proved that, other factors being equal, this deviation while maintaining tin due to the regulation of local cooling systems, will be lower than with a direct change in the temperature of the inflow in the central air conditioning system, since unregulated inflow increases the room's own heat resistance.

5. It is proposed to apply the ratios obtained in the paper for the analytical assessment of the dynamic thermal regime of an air-conditioned room employing local cooling systems equipped with an automation system with an integral law of control, in order to verify the conditions of human comfort and safety, as well as to determine the required parameters of the control unit.

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