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Temperature mode of a room at integrated regulation of split systems

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Abstract. The article considers simplified mathematical formulation and problem solution of internal air temperature changing. The room is equipped with automated local heating and cooling systems under variable thermal influences. At the same time, the study does not pay attention to the influence of the background general exchange system of supply and exhaust mechanical ventilation. It is shown that the main differential equation connecting the most important components of the heat flow in the room for the case under consideration belongs to the class of Emden-Fowler equations. The article provides an analysis of this equation, obtains the structure and variants of its asymptotic solutions. They describe the time dependence of the room air temperature deviation from the setpoint and the expression for the time moment at which the maximum temperature deviation is observed, with an abrupt change in the heat flow and regulation of the equipment of local heating and cooling system according to the integral law. Calculations were carried out to confirm the obtained dependencies using a numerical solution of the original differential equation by the Runge-Kutta method, as well as by comparing them with the results of field measurements in one residential building in Moscow. It is noted that the structure of the analytical solution and the type of dimensionless complexes constructed by reducing the equation to a dimensionless form directly follow from the properties of the Emden-Fowler differential equations. The obtained ratios are proposed to be used for an approximate assessment of the non-stationary thermal regime of an air-conditioned room served by local heating-cooling systems controlled by the integral law. Moreover, these ratios can be used for determination of the necessary characteristics of the regulator, including on the basis of multivariate calculations with a change in the parameters of the problem.

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1. Introduction

This paper studies the pattern of change in the temperature of the indoor air in a room equipped with automated local heating and cooling systems under variable disturbing thermal effects in the absence of the influence of a background unregulated positive pressure ventilation.

The necessary combination of indoor microclimate parameters should be maintained primarily based on the needs of ensuring a comfortable and safe living environment and implementing technological processes carried out within the premises. In order to stabilize the hydrothermal regime in actual practice, certain automatic control systems are mainly used for climate equipment, while taking into account the inherent heat stability of the premises. At the same time, the nearly always present variable nature of heat losses and heat gains in the building makes the problem under consideration materially unsteady.

It should be noted that the behavior of the internal temperature when air conditioning the premises, depending on the nature of thermal disturbances and the method of regulating the heat and cold supply, has already attracted the attention of a number of researchers. For instance, even educational and reference literature materials contain some of the most simplified solutions to the problem under consideration. Nevertheless, more serious works have recently emerged, including [1–3], which are devoted mainly to the variable thermal regime during the operation of external networks of the heat supply system. However, the approaches contained in them may be considered rather complex, yet for the same reason the results obtained there are very complex as well, resulting in their little use for engineering calculations. Another modern trend in this area is the increasing use of computer modelling when studying and analyzing unsteady regimes. From this point of view, some foreign works may present certain interest, in which approaches of this type have become the most characteristic in recent years, among which [4, 5] are worth noting. In addition, in the field under consideration, there is a number of solutions associated with specific objects operating in limited areas under specific conditions, including underground structures in harsh climatic conditions [6] or for external enclosing structures with heat-conducting inclusions [7]. An earlier publication by the author [8] devoted to the development of a very simple solution for the propagation of a temperature wave in a thick-walled cylinder may also be noted. Moreover, the authors of some studies in this area tend to solve the inverse problem - the determination of the thermophysical parameters of a material based on the study of temperature fluctuations [9] or thermography methods [10]. The works [11–16] consider modeling of processes in the premises as a whole. Thus, the authors of [11] give a fairly complete consideration of the thermal and humidity regime of the premises, but it applies only to a specific building and is mainly experimental in nature, and [12] presents a very detailed multi-parameter simulation-type computational model, applicable to the cold period of the year with the heating on only. Publications [13–16] may also be considered as comprehensive, in particular, the articles [13–14] are devoted to using the principles of fuzzy logic when organizing indoor climate management, and [15–16] mostly employ the methods of automatic control theory, however, the results obtained therein are still unsuitable for use in engineering practice due to their complexity. Finally, there are also studies based on the general principles of building engineering systems management and implementation of power-saving engineering solutions that are possible under given conditions, for example, [17–19], and precisely because of their general nature, they also lack specific dependencies that are of interest to us

That being said, it should be noted that recent engineering practice indicates a gradual expansion of the use of local heating and cooling systems in the premises, in particular, of the “chiller system” type to maintain the air temperature and fence surfaces at the required level during the warm season. Their operation involves full recirculation of the heated or cooled airflow and therefore does not affect the overall air balance in the premises. This way, the mechanical supply and exhaust general ventilation mainly functions as a sanitary and hygienic system ensuring the necessary purity of the internal air; heating or cooling the inflow, when provided, plays a secondary role, making it possible to reduce the load on local systems in the conditions under consideration as much as possible. The same is true for the installation of radiant heating and cooling systems, especially those heating or cooling ceilings.

In [20], the author obtained a sufficiently general analytical solution to the problem of changing temperature of the indoor air in the premises serviced by an air conditioning system using the law of astatic control for climate control equipment [21]. The corresponding dependence may be represented as a sufficiently better convergent series in powers of a certain independent variable, for which a dimensionless group, including the characteristic parameter of the corresponding differential equation and the period of time that has elapsed since the occurrence of thermal exposure is used. Nevertheless, there is merit in considering the possibility of simplification and further generalization of this solution while maintaining its physical validity [21]. This will provide additional opportunities for its analysis and comparison with existing analogues, and at the same time identify the limits when such a simplification is acceptable. Therefore, we can find the scope of these simplified options for engineering calculations with no losses in accuracy of calculations. In addition, it should be taken into account that for the time being local heating and cooling systems often are the only systems which are regulated, bearing the main burden of assimilating heat gains and compensating for heat losses, while the general exchange inflow plays only the background role at a minimum level. Thus, the possibility of adapting the solution obtained to the specified conditions must be assessed.

The relevance of the proposed study then lies in the feasibility of exploring relatively simple and at the same time physically reasonable analytical dependencies for the behavior of the internal temperature in the premises with climate control equipment such as split systems, regulated under the law of astatic control. We need to take into account the values of surplus heat, the characteristics of the enclosing structures and the controller, and the relationship of all listed parameters and the largest temperature deviation from a given level. The formulas obtained should be written in a fairly simple form, suitable for using in engineering practice, but at the same time, be of satisfactory accuracy and allow for multivariate evaluation calculations, to allow for the analysis of the thermal regime of the premises and the synthesis of the appropriate climate control equipment automation system. In this case, this will allow application of the results achieved for a quite wide range of objects of a similar class [21].

Thus, the aim of the paper is to construct simplified analytical methods for calculating the change in the temperature of the indoor air served by local automated split heating and cooling systems in the absence of a background unregulated inflow. The following items may be considered as study objectives:

- compilation of the basic differential equation describing the general balance of convective heat in the air conditioned premises, taking into account the simultaneous operation of automated climate control systems;
- analysis of this equation, bringing it to the dimensionless form and constructing its possible analytical solutions, including asymptotic solutions, in the form of end formulas with an abrupt change in the values of surplus heat and using the law of astatic control for regulating heating and cooling systems;
- confirmation of the dependencies obtained by comparing their variants with the exact solution in the form of an infinite series, obtained by the author in [20], and with the data of experimental measurements for the typical representative premises;
- identifying the limits of applicability of various options for asymptotical solutions based on a comparison of the discrepancies given by them with the typical error of engineering calculations and their initial data.

2. Methods

Let us consider the regime where abrupt heat surpluses in the amount of Q_{in}, *W*, are compensated by a local cooling system (for example, of a split-system type), which assimilates the amount of heat Q_c, W, at each moment of time. In this case, the system is regulated as per the law of astatic control, depending on the current deviation of the indoor air temperature t_{in} from the specified initial value (setpoint) $t_{in.0}$: $\theta_{in} = t_{in} - t_{in.0}$,°C. With that in mind, we consider that the background general ventilation does not function or its influence may be neglected. In most cases, this is achievable, since the air exchange in split systems in the premises is usually kept as low as possible to a minimum for reasons of energy saving, which is determined by the sanitary standard for the supply of outdoor air based on the number of people present in the premises. In this case, the general equation for the heat balance of the premises after replacing $z = \sqrt{\tau}$ may be written as [20]:

$$\frac{\mathrm{d}^2\theta_{in}}{\mathrm{d}z^2} + Cz\theta_{in} = 0 \ . \tag{1}$$

Here, $C = \dfrac{4K_c}{B}$ – is the characteristic parameter of the equation, $c^{-3/2}$, where K_c is the equivalent transmission ratio of the automated system, W/(K·s), over the channel "$t_{in} \rightarrow$ derivative of Q_c"; parameter B, W·s$^{1/2}$/K, may be calculated using the formula:

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

Here, λ, c and ρ are the thermal conductivity, W/(m K), specific heat capacity, J/(kg K), and the density of the material of the layer of the *i*-th solid fence facing the inside of the premises, for example, external and internal walls and partitions, as well as interflooring, respectively; A_m is the area of each of the enclosing structures listed, m^2. Thus, in this case, the expression for C differs from that obtained in [20], since another way of compensating for surplus heat in the premises is considered.

The differential equation (1) is nonlinear equation of the second order and is classified as an Emden-Fowler equation, the general form of which may be written as follows [22–24]:

$$\frac{\mathrm{d}^2 y}{\mathrm{d}z^2} + Cz^n y^m = 0 \ . \tag{3}$$

Similar equations arise in a number of physical and economic applications. Thus, (1) is a special case of (3) with m = 1, n = 1. However, the challenge lies in the fact that with m = 1, there is no end analytical solution (2) in elementary functions [22–24].

The equation (1) can be obtained from the common equation of convective heat balance for indoor air within the framework of its single-are model can be shown here as follows [24]:

$$Q_{in} + G_s c_a (t_s - t_{in})/3.6 - Q_c - B\sqrt{\tau}\frac{\mathrm{d}t_{in}}{\mathrm{d}\tau} = 0\,, \tag{4}$$

where G_s is mass flow rate, kg/h, of the supply air which is usually considered as equal to the value of the exhaust 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 (4) 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 (4) that $t_s = \mathrm{const}$.

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:

$$\frac{\mathrm{d}Q_c}{\mathrm{d}\tau} = K_c (t_{in} - t_{in.0})\,. \tag{5}$$

Using the concept of θ_{in}, and differentiating (4) term by term by τ for the possibility of substituting expressions (5) there, we can write (4) in the canonical form (1) considering G_s = 0 because when using split systems, general air exchange is minimized and for it we can assume t_s = const.

Let us consider a slightly different approach to solution (1) compared to that adopted in [20], namely, we will perform certain transformations to initially reduce it to the dimensionless form and single out its singularities to simplify further integration. Bearing in mind some general properties of the Emden-Fowler equations [22], [23], [24],in particular, the presence of a critical point with z = 0, let us initially present the solution in the form of the product $\theta_{in} = zf(z)$, and after its substituting into (1), obtain an equation for the function selected $f(z)$:

$$\frac{\mathrm{d}^2 f}{\mathrm{d}z^2} + \frac{2}{z}\frac{\mathrm{d}f}{\mathrm{d}z} + Czf = 0\ . \tag{6}$$

If we now make a substitution of variables in the form of $x = Cz^3$, where x will obviously already be a dimensionless quantity, we find the end equation for $f(x)$:

$$9x\frac{\mathrm{d}^2 f}{\mathrm{d}x^2} + 12\frac{\mathrm{d}f}{\mathrm{d}x} + f = 0\,. \tag{7}$$

With the inverse transformation, we obtain $z = \left(\frac{x}{C}\right)^{1/3}$. It is easy to see that (7) no longer contains the parameter C, meaning that the assumption made concerning the form of representation of the independent variable is correct. As the initial conditions with τ = 0 for the original equation (1), we obviously need to take θ_{in} = 0 and $\mathrm{d}\theta_{in}/\mathrm{d}z = 2Q_{in}/B$, from which it turns out for (7) that $f(0)$ = 1, since the singularity with z = 0 was singled by representing $\theta_{in} = zf(z)$, and similar to $\mathrm{d}f/\mathrm{d}x = 1/12$.

We can note that with small x the first term in (7) may be neglected, and it becomes an equation of the first order with separable variables, the solution to which has the form of $f = \exp(-x/12)$ and, thus, the asymptotic approximation of the solution for the initial moments of time takes the form of:

$$\theta_{in} = \frac{2Q_{in}}{B}\left(\frac{x}{C}\right)^{1/3}\exp(-x/12) = \frac{1.26Q_{in}}{\sqrt[3]{K_c B^2}}x^{1/3}\exp(-x/12). \tag{8}$$

It is easy to see that the first two expansion terms of the function f in a Taylor series coincide with those for the exact solution obtained in [20] by the method of undetermined coefficients

$$\exp(-x/12) = 1 - x/12 + \ldots \; . \tag{9}$$

Now, assigning a certain value $\frac{\mathrm{d}^2 f}{\mathrm{d}x^2} = a$, we write down the differential equation from (7) for the following approximation:

$$\frac{\mathrm{d}f}{\mathrm{d}x} + \frac{f}{12} + \frac{3ax}{4} = 0 . \tag{10}$$

This is a linear inhomogeneous equation of the 1st order, which may be integrated completely in elementary functions, from which, taking into account the initial condition, we obtain:

$$f = (1 - 108a)\exp(-x/12) + 108a\left(1 - \frac{x}{12}\right). \tag{11}$$

In particular, with a = 1/252 we find:

$$f = \frac{4}{7}\exp(-x/12) + \frac{3}{7}\left(1 - \frac{x}{12}\right) . \tag{12}$$

This value of a corresponds to $\frac{\mathrm{d}^2 f}{\mathrm{d}x^2}$ for the first approximation approximately at x = 6.72. It can be demonstrated that the first three terms in the expansion of this function already coincide with the exact solution:

$$\exp(-x/12) = 1 - x/12 + x^2/504 - \ldots \; . \tag{13}$$

However, calculations show that the best agreement with the exact solution with x < 10 is achieved for a = 1/312. If we calculate the x derivative of the product $x^{1/3} f$, where f is taken in accordance with (12), we find:

$$\frac{\mathrm{d}}{\mathrm{d}x}\left(x^{1/3} f\right) = \frac{1}{21x^{2/3}}\left[(4 - x)\exp(-x/12) + 3 - x\right]. \tag{14}$$

Equating (14) to zero, we find that at the maximum point x = 3.43, which coincides with the value of 3.48 for the exact solution if the error is no more than 1.5 percent [20]. In this case, the largest value of the complex itself $x^{1/3} f$ is 10/9, just as in [20].

3. Results and Discussion

For illustrative purposes, Figure 1 shows the graphs of the expressions obtained for the function of f – exponential approximation (thick solid line) and dependence (12) (dotted line). For comparison, the solid thin line shows the behavior of the direct computational solution of the original differential equation (7), found using a computer program with Runge-Kutta method of the 4th order. It coincides with the exact solution in the form of an infinite series given in [20] up to the line thickness.

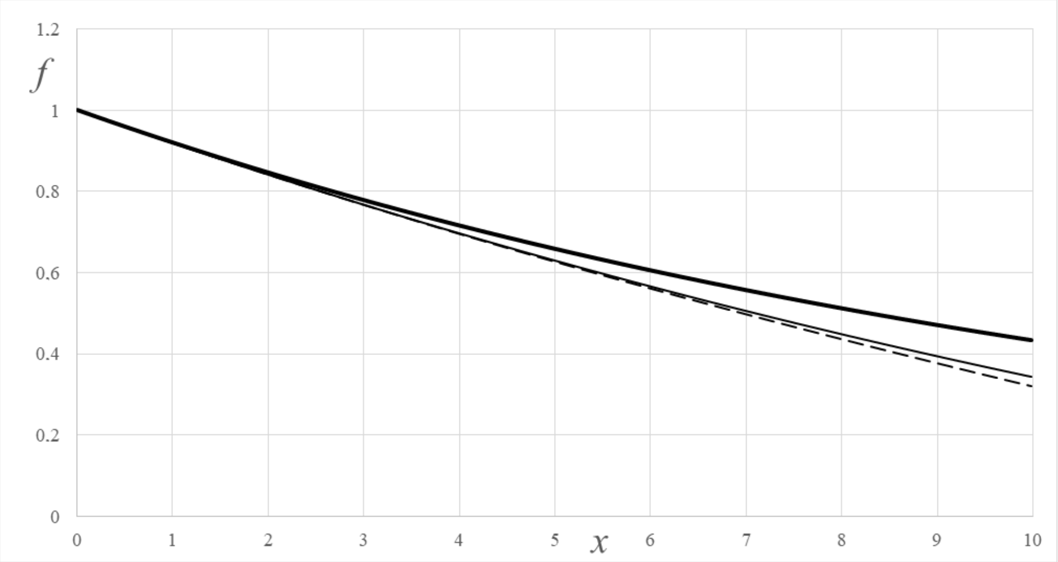

Figure 1. Graphs of the function *f* for various options of its representation.

Similarly, Figure 2 shows the graphs of the product $x^{1/3} f$,i.e., the desired dimensionless temperature. It can be seen that the accuracy of the expression (12) is good and, taking into account its simplicity and physical validity, it can be recommended for use in calculations.

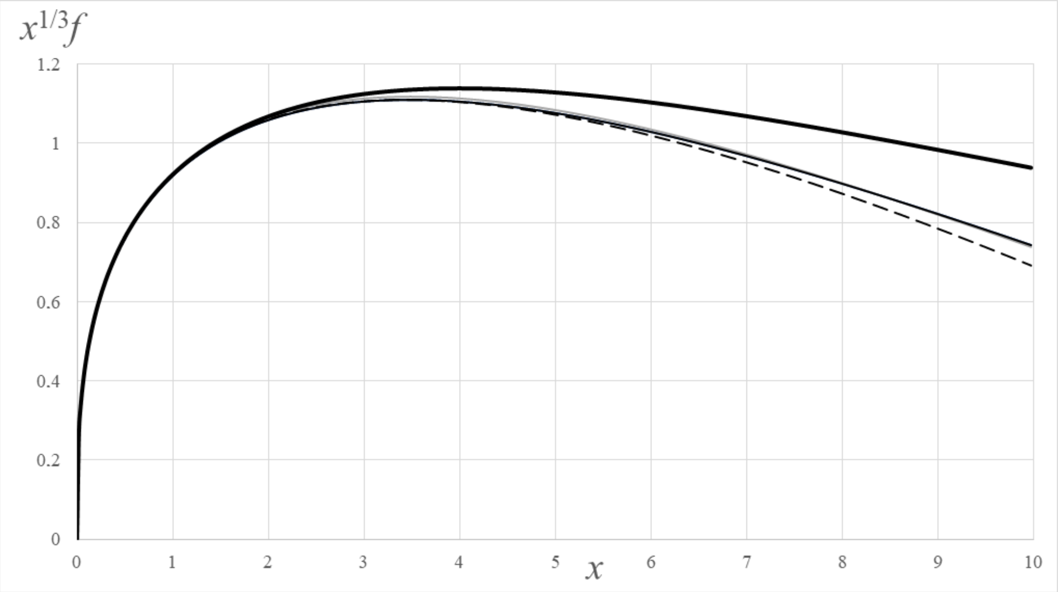

Figure 2. Graphs of dependence of the product $x^{1/3} f$ for various options of its representation.

Additional confirmation and justification of the presented mathematical model can be obtained experimentally. Figure 3 shows a comparison of the theoretical dependence for $x^{1/3} f$ (solid line) using (12) and the data obtained from direct temperature measurements in the premises served by a split cooling system. In reality, its control was implemented in a positional way, but due to the high switching rate, it approached astatic control. The abrupt rise in heat input was simulated by turning on a convective electric heater with Q_{in} = 500 W, and the value of B was determined taking into account the actual thermal parameters of building materials in the enclosures and the geometric dimensions of the premises [25]. Their area was 14 m^2, height from floor to ceiling was 3 m, depth from the outer wall could be considered equal to 6 m, internal structures had a total area of 64 m^2. They were made of reinforced concrete with a density of 1,200 kg/m^3. One must bear in mind that under the conditions considered, this entire area should be taken into account, since during the experiment the temperature wave propagates in one direction only. The outer wall was made of lightweight concrete with a density of 500 kg/m^3 and an area of 7 m^2 including a window with an area of 1.8 m^2. Then we obtain B = 2,4000 W·s $^{1/2}$/K from the expression (2).

To measure t_{in} a Testo 0560 1110 thermometer with a division value of 0.1° was used, which was installed in the center of the room at a height of 1 m from the floor. When processing the results in order to reduce the temperature to the dimensionless form in accordance with (6), its value was divided by the $\frac{1.26Q_{in}}{\sqrt[3]{K_c B^2}}$, and the parameter x was determined by the expression $Cz^3 = \frac{4K_c}{B}\tau^{3/2}$. as noted earlier. The best agreement between the theoretical and experimental dependences is observed at K_c = 2 W/(K·s), which is shown in Figure 3. Thus, when combining theoretical and experimental methods, it is possible to identify the mathematical model and determine the actual values of its certain parameters, while establishing their values antecedently may prove to be difficult.

The dashed dot shows 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 [26] (dotted line), after normalization by the magnitude of the maximum temperature deviation. It can be seen that in the begin after the appearance of the thermal disturbance, experimental measurements give a similar nature of dependence, which further confirms the theoretical provisions of the proposed work. In the future, the discrepancy begins to increase, since in [26] the regulator, in addition to the integral, also had a proportional component, so the temperature there fades faster.

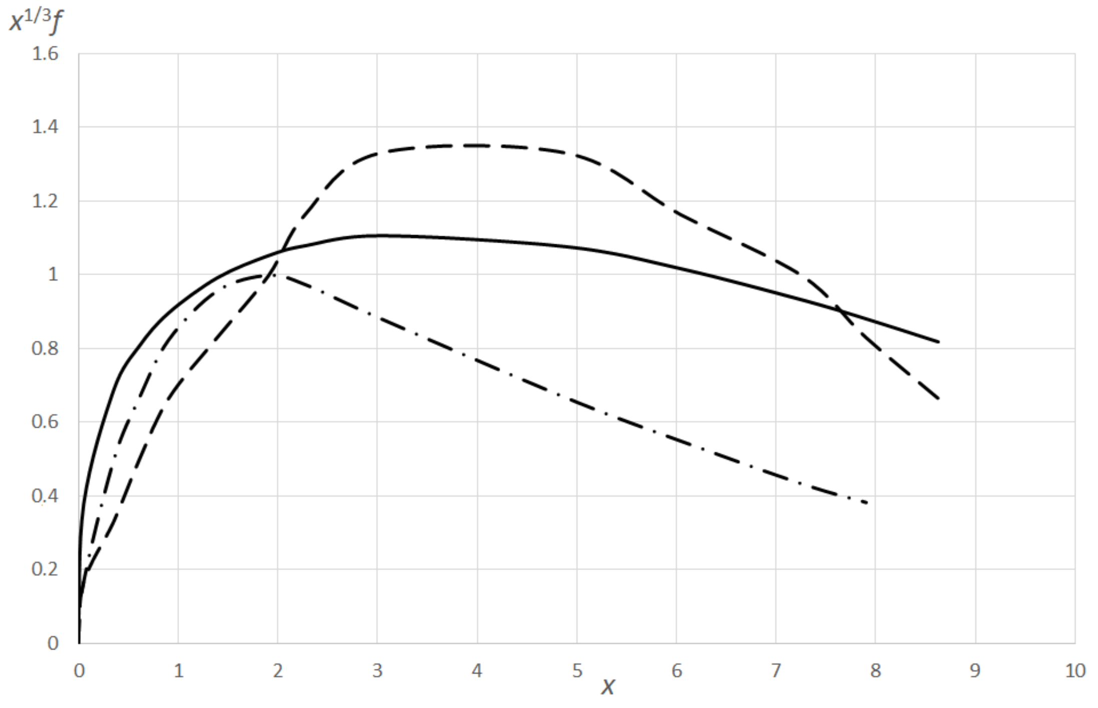

Figure 3. Dependence of θ_{in} on time for the calculated premises with automatic keeping of t_{in} (solid line, taking into account expression (12); dotted line, experiment; dashed dot, measurement data [26])).

Note that, for the general nature of dependence (10) shown in Figure 3, one can find a significant similarity between the relationships that are given, e.g. by the authors of [12] and [13] under similar initial conditions, and the observed temperature deviations from the initial values generally correspond to the level noted in [27] for similar regimes. Finally, the general concept of the approach considered and certain elements of the mathematical arrangement and solution to the problem under study are consistent with the results presented in publications [11], [14] and a number of other materials, allowing us to assume that the results of the proposed study are quite reliable and justified.

The results achieved additionally confirm the reliability of the previously found accurate analytical solution in the form of an infinite series [20]. It takes into account the thermal stability of enclosing structures during the propagation of a temperature wave in their material and the characteristics of the controller, as well as demonstrates its applicability when installing split systems and lacking background general ventilation.

4. Conclusion

1. The premises under consideration had automated climate control equipment such as split systems when regulated under the law of astatic control in the absence of a background unregulated inflow. It was proved that for such premises, the asymptotic variants of the analytical solution obtained in the paper describing the behavior of the value t_{in} describe the actual process of heating or cooling quite well, at least for not too large values of τ under the conditions of an abrupt thermal disturbance.

2. It was established that the core of the identified dependence for t_{in} may be expressed in an explicit form through the product of power and exponential functions from a dimensionless group, including the value $\tau^{3/2}$. This allows analyzing the problem quite easily and further synthesizing the automation system for climate control equipment by selecting numerical coefficients in the solution.

3. It was noted that the form of representation of the analytical solution in the dimensionless form and the composition of the dimensionless group used as an independent variable are naturally obtained in accordance with the properties of the original differential Emden–Fowler equation.

4. Experiments using full-scale measurements for the typical representative premises confirmed that the discrepancy between the actual and calculated values of t_{in} lies within the measurement accuracy limits and the typical error of engineering calculation.

5. We propose to use the variants of dependencies found in this paper for an approximate analytical assessment of the behavior of the internal temperature in the air conditioned premises in unsteady conditions with local heating and cooling systems equipped with an integrative controller to check the comfort and safety of the living environment, the possibilities of implementing the technological process, to identify the required parameters of the regulator, including on the basis of multivariate calculations with a change in the initial data.

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