ISSN 2307–3489 (Print), ISSN 2307–6666 (Online)

Наука та прогрес транспорту. Вісник Дніпропетровського національного університету залізничного транспорту, 2018, № 1 (73)

РУХОМИЙ СКЛАД ЗАЛІЗНИЦЬ І ТЯГА ПОЇЗДІВ

UDC 629.45.048.7

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MATHEMATICAL MODEL OF UNSTEADY HEAT TRANSFER OF PASSENGER CAR WITH HEATING SYSTEM

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Purpose. The existing mathematical models of unsteady heat processes in a passenger car do not fully reflect the thermal processes, occurring in the car wits a heating system. In addition, unsteady heat processes are often studied in steady regime, when the heat fluxes and the parameters of the thermal circuit are constant and do not depend on time. In connection with the emergence of more effective technical solutions to the life support system there is a need for creating a new mathematical apparatus, which would allow taking into account these features and their influence on the course of unsteady heat processes throughout the travel time. The purpose of this work is to create a mathematical model of the heat regime of a passenger car with a heating system that takes into account the unsteady heat processes. **Methodology**. To achieve this task the author composed a system of differential equations, describing unsteady heat processes during the heating of a passenger car. For the solution of the composed system of equations, the author used the method of elementary balances. **Findings.** The paper presents the developed numerical algorithm and computer program for simulation of transitional heat processes in a locomotive traction passenger car, which allows taking into account the various constructive solutions of the life support system of passenger cars and to simulate unsteady heat processes at any stage of the trip. **Originality.** For the first time the author developed a mathematical model of heat processes in a car with a heating system, that unlike existing models, allows to investigate the unsteady heat engineering performance in the cabin of the car under different operating conditions and compare the work of various life support systems from the point of view their constructive solutions. **Practical value.** The work presented the developed mathematical model of the unsteady heat regime of the passenger car with a heating system to estimate the efficiency of unsteady, transitional temperature states in passenger cars, taking into account the design features of the heating system and the regulatory requirements. This allows the development and implementation of optimal technical characteristics of heating appliances and the construction of an algorithm for controlling their operation in accordance with operating conditions, taking into account the thermal inertia of the car in the transitional modes of heating, on the basis of mathematical modeling.

Keywords: mathematical modelling; passenger car; unsteady heat processes; heating system

Introduction

At present, mathematical modelling is widely used to assess the effectiveness of various constructive solutions. A peculiarity of mathematical modelling is the large amount of computational work; therefore, recently, in terms of accessibility and improvement of the capabilities of computer technology, numerical experiment has become widespread. Mathematical modelling with the use of adequate mathematical models has much in common with the field experiment. This way of research allows simulating the processes that arise in the actual operation of separate equipment and life support systems in general, as well as the impact of various factors thereon. The basis of mathematical modelling is the method of differential balance equations [11].

The mathematical modelling of heat processes in passenger cars with heating systems is usually realized in steady regime, when the heat fluxes and parameters of the thermal circuit are constant, do

doi: 10.15802/stp2018/123409

not depend on time. The steady regime refers to the situation in the car, when there is balance between the thermal energy that comes in and that given by fencing structures into the environment. The energy balance of such a system in a steady regime is studied quite well [4-8, 12]. But, any heat exchange is dynamic, and it is not enough to describe a single steady regime. The worse situation is with the analysis of the thermal condition of heated cars in unsteady conditions, in particular, when the heating of high-voltage heating systems is switched off in motion and in parking places with further heating, resulting in transitional heating regimes.

The mathematical model of the car heat condition, when the car is heated with an air heating system, is considered in the works [14, 16, 17], the model, when a car air conditioner operates in the heat pump mode, is considered in the works [3, 13]. The works [2, 15] present a mathematical modelling of unsteady heat exchange processes of a passenger car with air conditioning systems. These mathematical models are similar to each other, and allow the insertion and extraction of additional elements into the calculation scheme rather easily, but the air flow is used for heating and cooling as a coolant. The presented mathematical models do not fully reflect the thermal processes occurring in the car when using a water heating system, where the intermediate coolant is water. There are no such indicators as heat energy accumulation [1].

The works devoted to passenger car heating systems indicate that a car needs the heating system with capacity of 48 kW in the winter period. These requirements were substantiated in the 70-80s of the last century for cars with an effective thermal conductivity of about 1.7 W/ (m^2 · K) with the environment temperature of –40°C in winter, with a number of passengers from 32 to 60 people (considered by Kitayev B. N. [6, 7], Kuzmin L. D. [11], Zharikov V. A. [4, 5], Sidorov Yu. P. [10] and other researchers).

Already at the beginning of the 21st century, the car builders of PJSC «Kryukov Railway Car Building Works» reached an effective conductivity in a passenger car of about (0.8 ÷ 1.0) W / (m^2 · K), and the installation of double-glazed windows significantly increased its tightness.

Thus, taking into account the car structural changes and the new trends, a more thorough analysis of the car heat regime is required, taking into account the unsteadiness of the process, when heating the passenger car.

Purpose

The purpose of this study was to create a mathematical model of the unsteady heat regime of a passenger car with a water heating system to evaluate the role of unsteady, transitional temperature states of the passenger car, the selection of optimal technical characteristics of heating devices, and constructing an algorithm for their operation control in accordance with operating conditions, and particularly taking into account the manifestation of the car thermal inertia during the transitional operating modes of the heating system.

Methodology

To achieve the set task the author composed a system of differential equations, describing unsteady heat processes during the heating of a passenger car. For the solution of the composed system of equations, the author used the method of elementary balances.

When studying the transitional regimes in the process of cooling and subsequent heating of a passenger car during the operation, the conditions are taken into account when the heat from the TEHs (tubular electric heaters) is perceived by the intermediate coolant and then transmitted to the car. The same is when the car is cooled from the initial temperature to the critical, at which the next heating process begins. The dynamic equation of the temperature process in this case should be solved in two stages: in relation to the intermediate coolant, and from the coolant to the air in the car and then to the outside air.

During formation of the car heat model, physically grounded and experimentally confirmed features of the car heat condition were taken into account, namely:

– temperature of the interior partitions of the car practically coincides with the temperature of the car air;

– difference between the internal partition wall with the temperature t_p and the average air temperature t_a in the car does not exceed 3 K, since the temperature difference between the external environment with the temperature t_e and the car air with the temperature t_c is mainly damped on its thermal insulation;

– temperature of the air removed from the car through deflectors is equal to the air temperature in the car t_a.

– due to increasing the coefficient of heat transfer by convection on the outer surfaces of the partitions, depending on the car speed from 0 to 80 km/h, the heat transfer of the car body increases by 10%, at speeds from 80 to 160 km/h, the coefficient increases by 1%;

– air infiltration volume, depending on the car speed up to 120 km/h, can reach 325 m^3/h;

The physical essence of these equations is reduced to the following.

The heat flow is evolved from TEHs $Q_{TEH}(\tau)$ in the time interval τ, is transmitted to the intermediate coolant and the metal structure of the heating system. Since the heating devices can not physically transfer the entire heat flow $Q_{TEH}(\tau)$ evolved from TEHs, part of this heat is accumulated in the coolant and the metal structure of the heating system Q_{hs}.

In accordance with the energy conservation law (heat balance), the heat flow $Q_{TEH}(\tau)$ is consumed on four main components:

$$Q_{TEH}(\tau) = Q_{hs} + Q_{pp} + Q_{cl} + Q_{bl}, \qquad (1)$$

where Q_{hs} – heat accumulated by heating system; Q_{pp} – heat consumed by heating pipes; Q_{cl} – heat consumed by the coolant to heat the outdoor air; Q_{bl} – heat consumed for water heating for hot water supply, as the boiler does not affect the microclimate in the car and has a slight consumption of heat, this parameter will not be taken into account further.

The listed components are determined by the relationships:

$$Q_{hs} = C_{hs}\frac{dt}{d\tau}, \qquad (2)$$

$$Q_{pp} = c_w G_w (t_{in} - \mathrm{t}_{\mathrm{out}}), \qquad (3)$$

$$Q_{cl} = c_w G_w (t_{in} - t_{out}), \qquad (4)$$

where C_{hs} – total heat capacity of the water and the metal structure of the heating system; t_{in} – coolant temperature at the inlet to the heating pipes and heater; t_{out} – coolant temperature at the outlet from the heating pipes is determined by the formula:

$$t_{\mathrm{out}} = t_{\mathrm{a}} + (t_0 - t_{\mathrm{a}})\mathrm{e}^{-\mathrm{al}}, \qquad (5)$$

where t_0 – coolant temperature at the inlet to the heating pipes; t_a, – room air temperature; l – length of the heating pipes; a – coefficient determined by the expression:

$$a = -\frac{4k_{pp}D_o}{c_w\rho_w D_i^2\omega_w}, \qquad (6)$$

where k_{pp} – coefficient of heat transfer of heating pipes; D_o, D_i – outside and inside diameter of the heating pipes; ρ_w – coolant density; ω_w – coolant speed in the heating pipes.

To analyse the heating regime of the car heating system, the equation (1) ÷ (3) must be supplemented by another equation that is used to calculate the heating and cooling of the heating system coolant from a given initial temperature $t(0)$ to a certain final temperature t_b for a short period of time τ, at any stage and has the form:

$$t_b(\tau) = t(0) + \frac{Q_{TEH} - (Q_{pp} + Q_{cl})}{C_{hs}}\tau. \qquad (7)$$

The amount of heat entering the car from $Q_{pp}(\tau)$, as can be seen from equation (3), depends on the coolant temperature, the heating pipes area, the heat transfer coefficient k_{pp}, the rate of coolant circulation in the heating pipes. The heat $Q_a(\tau)$, evolved by the air flow V_O at the time τ, is uniquely associated with the change in its enthalpy and is determined by the relation:

$$Q_a(\tau) = \left(I(t_a^c\phi_a^c) - I(t_a\phi_a)\right)\rho_a V_O(\tau), \qquad (8)$$

where $I(t_a^c\phi_a^c)$ – specific enthalpy, temperature and relative humidity of air entering the car after heating in calorifer; $I(t_a\phi_a)$ – specific enthalpy, temperature and relative humidity of air in the car; V_O – volume of outside air supplied by the ventilation system; to determine the specific enthalpy of air I d – wet air diagram is used; t_a^c –air temperature heated by the calorifer is determined by the expression:

$$t_a^c = \frac{c_w G_w (t_{in} - t_{out})}{c_a\rho_a V_O} + t_o. \qquad (9)$$

doi: 10.15802/stp2018/123409

The heat brought by the heated outside air can be determined by another less precise expression:

$$Q_a(\tau) = c_a \rho_a V_O (t_a^c - t_a)(\tau)\,, \qquad (10)$$

The heat flows $Q_{pp}(\tau)$, $Q_a(\tau)$, $Q_l(\tau)$, entering the car at the time τ, are absorbed by three components:

$$(Q_l + Q_a + Q_{pp})(\tau) = Q_{los} + Q_{\inf} + Q_{car}\,, \qquad (11)$$

where Q_{los} – heat lost by partitions, as well as windows; Q_{inf} – heat consumed for heating of the cold air, which penetrates through the body imperfections and is characterized by $V_{inf}(S)$, that is, by the volume of infiltrated air, depending on the speed of movement; Q_{car} – heat consumed for heating the internal air and equipment of the car.

The listed components are determined by the relationships:

$$Q_{los} = k_o F_o (t_a - t_e)\,, \qquad (12)$$

$$Q_{\inf} = c_a \rho_a V_{inf} (t_a - t_e)\,, \qquad (13)$$

$$Q_{car} = C_{car} \frac{dt}{d\tau}\,, \qquad (14)$$

where k_o – coefficient of heat transfer through outer lining; F_o – area of outer lining; t_a – room air temperature; t_e – environment temperature; c_a – heat capacity of air; V_{inf} – volume of air entered into the car as a result of infiltration; C_{car} – total heat capacity of all internal partitions, wooden lining of external car frame and half heat capacity of the heat-shielding layer.

For an integrated analysis of the car heat regime we need one more equation to calculate the heating and cooling of the car air temperature from a given initial temperature $t(0)$ to a certain final temperature t_a for a short period of time τ, has the form:

$$t_a(\tau) = t(0) + \begin{pmatrix} (Q_{pp} + Q_c + Q_l) - \\ -(Q_{los} + Q_{\inf}) \end{pmatrix} / \, C_{car}\tau\,. \qquad (15)$$

It is advisable to highlight several of the most characteristic stages of the car heat regime, within each of which almost constant values of the output parameters are kept:

$$c_a;\, c_w;\, \rho_a;\, \rho_w;\, k_o;\, F_o;\, \phi_a;\, C_{car};\, C_{hs};\, G_w = const \qquad (16)$$

With restrictions (16), equation (7) (15) has at its separate stage its own, individual analytical solution of the form:

The equation describing the change in air temperature (15) takes the form of:

$$\frac{dt_a}{d\tau} = \frac{Q_{pp} + (Q_l + Q_a - Q_{inf}) - Q_{los}}{C_{car}}\,, \qquad (17)$$

where the step part depends on both t_a, and $t_{in} = t_b$.

So, this is the equation of two variables:

$$t_{\mathrm{a}} = t_{\mathrm{a}}(\tau) \text{ and } \mathrm{t_{in}} = \mathrm{t_b}(\tau)$$

The equation describing the temperature of the coolant in the boiler (7) takes the form:

$$\frac{dt_b}{d\tau} = \frac{Q_{TEH} - (Q_{pp} + Q_{cl})}{C_{hs}}\,, \qquad (18)$$

where the right side also depends on t_{a} and t_{b}.

Consequently, we have a system of two differential equations with two variables:

$$\begin{cases} \dfrac{dt_a}{d\tau} = \dfrac{Q_{pp} + (Q_l + Q_a - Q_{inf}) - Q_{los}}{C_{car}}; \\[2ex] \dfrac{dt_b}{d\tau} = \dfrac{Q_{TEH} - (Q_{pp} + Q_{cl})}{C_{hs}}. \end{cases} \qquad (19)$$

The grouping of the right-hand sides of the equations with respect to the variables t_a, and t_b and after transformations has the form:

$$\begin{pmatrix} (C_w G_w - \mathrm{e}^{\text{-al}} \cdot C_w G_w) t_b + \\ (C_w G_w \mathrm{e}^{\text{-al}} - C_w G_w - k_o F_o) \times \\ \mathrm{t_a} + (Q_l + Q_a - Q_{inf}) + \\ k_o F_o t_o \end{pmatrix} / \, C_{car} = \qquad (20)$$

$$= \varepsilon_1 t_b + \mathrm{o}_1 \mathrm{t_a} + \theta_1$$

where $\varepsilon_1, \mathrm{o}_2$ – indicators of car thermal inertia at the considered stage; θ_1 – index of car heat entropy at the considered stage:

$$\varepsilon_1 = \frac{C_w G_w (1 - \mathrm{e}^{\text{-al}})}{C_{car}}\,, \qquad (21)$$

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$$o_1 = \frac{C_w G_w \cdot (e^{-al} - 1) - \ k_o F_o}{C_{car}}, \qquad (22)$$

$$\theta_1 \frac{(Q_l + Q_a - Q_{inf}) + k_o F_o t_e}{C_{car}}, \qquad (23)$$

$$\frac{(e^{-al} - 1) C_w G_w t_b + (1 - e^{-al}) C_w G_w t_a}{C_{hs}} + \frac{Q_{TEH} - Q_{cl}}{C_{hs}} = \varepsilon_2 t_b + o_2 t_a \ + \theta_2, \qquad (24)$$

where ε_2, o_2 – indicators of thermal inertia of the heating system at the considered stage; θ_2 –indicator of heat entropy of the heating system at the considered stage.

$$\varepsilon_2 = \frac{(e^{-al} - 1) C_w G_w}{C_{hs}}, \qquad (25)$$

$$o_2 = \frac{(1 - e^{-al}) C_w G_w}{C_{hs}}, \qquad (26)$$

$$\theta_2 = \frac{Q_{TEH} - Q_{cl}}{C_{hs}}. \qquad (27)$$

Thus, the initial system of equations has the form:

$$\begin{cases} \dfrac{dt_a}{d\tau} = \varepsilon_1 t_b + o_1 t_a + \theta_1; \\ \\ \dfrac{dt_b}{d\tau} = \varepsilon_2 t_b + o_2 t_a + \theta_2, \end{cases} \qquad (28)$$

That is, linear equations with constant coefficients.

The linear non-homogeneous second-order equation with third-order coefficients has the form:

$$t_b'' + p \cdot t_b' + g \cdot t_b = f, \qquad (29)$$

where

$$p = -(o_1 + \varepsilon_2), \qquad (30)$$

$$g = o_1 \varepsilon_2 - \varepsilon_1 o_2, \qquad (31)$$

$$f = o_2 \theta_1 - o_1 \theta_2. \qquad (32)$$

Discriminator of the characteristic equation:

$$D = p^2 - 4g. \qquad (33)$$

The solution of homogeneous equations for the boiler and the car room temperature has the form:

$$t_b(\tau) = K_1 e^{R_1 \tau} + K_2 e^{R_1 \tau} + \frac{f}{g}, \qquad (34)$$

$$t_a(\tau) = \frac{K_1 R_1 e^{R_1 \tau} + K_2 R_2 e^{R_2 \tau} - A_2 t_b - C_2}{B_2}. \qquad (35)$$

where R_1, R_2 are the roots of the characteristic equation:

$$R_1 = \frac{-p - \sqrt{D}}{2}, \qquad (36)$$

$$R_2 = \frac{-p + \sqrt{D}}{2}, \qquad (37)$$

$$K_1 = \left[\begin{matrix} o_2 t_a(0) + \theta_2 + \varepsilon_1 \cdot \dfrac{f}{g} - \\ -(R_2 - \varepsilon_2) \cdot \left(t_b(0) - \dfrac{f}{g} \right) \end{matrix} \right] / \left(R_1 - R_2 \right), \qquad (38)$$

$$K_2 = t_b(0) - \frac{f}{g} - K_1. \qquad (39)$$

The expressions (34), (35) allow us to estimate not only the temperature of the coolant in the combined electric-coal boiler and the air inside the car, but also to carry out a comprehensive analysis of the heat processes while heating the passenger car, taking into account the structural changes and the unsteadiness of the processes, and evaluate the efficiency of the system «heating system – passenger car». To do this, the initial temperatures $t_b(0)$, $t(0)$ of the boiler and inside the car at this stage and the value of the output parameters should be known.

Approbation of the mathematical model when compared with experimental data. The mathematical model described above allowed constructing a calculated model of the car temperature condition, using a water heating system with natural circulation and a discrete two stage high-power heat supply (2 groups of 24 kW).

For simplicity, the infiltration volume was taken from the average speed of movement. The heat capacity of the internal equipment and the heating system is taken in the water equivalent. Heat-efficiency

ISSN 2307–3489 (Print), ISSN 2307–6666 (Online)

Наука та прогрес транспорту. Вісник Дніпропетровського національного університету залізничного транспорту, 2018, № 1 (73)

of the combined water-heating boiler is 24+24 kW. The work of the ventilation system was not taken into account; it was not switched on during the experiment.

The dimensions and physical parameters of the elements used to construct the calculation model are given.

The model approbation used the experimental data obtained by the author. The experiment was conducted during train movement, the car number 26487, manufactured at «KVZ» in 1985, overhaul reconditioning on 10.12.2014. Measurement of temperatures was carried out by stationary means, the temperature of the car air was measured by two thermometers located on the boiler and no-boiler side of the car, the temperature of the coolant in the boiler was measured by a regular remote thermometer with a remote sensor.

As can be seen from the data given in Fig. 1, the modelling results quite well coincide with the results of the experiment, that is, the constructed model can be considered rather accurate and used for theoretical studies.

Input data for the calculation model

Body area, m^2	330
Coefficient of heat transfer of partitions, taking into account the speed of movement	1.3
Infiltration volume, m^3/h	200
Area of heating pipes, taking into account the coefficient of finning, $m^2 \cdot k_{fn}$	19.5·2.9
Coolant circulation velocity, m/s	0.04
Number of passengers, people	52
Car heat capacity, C_{car}, kW	3056
Heat capacity of the heating system, C_{hs}, kW	1000
Coefficient of heat transfer of heating pipes, $W/m^2 \cdot K$	10.8

Findings

The computational algorithm allowed developing a computer program for conducting a complex analysis of heat processes during the heating of a passenger car, taking into account structural changes and unsteadiness of processes, estimation of the efficiency of the system «heating system – passenger car».

Originality and practical value

The mathematical model of the unsteady heat regime of the passenger car with a water heating system was developed for evaluation of the role of unsteady, transitional temperature states of a passenger car, taking into account the features that are determined by existing requirements. This allowed the selection of optimal technical characteristics of heating devices and the construction of an algorithm for controlling their operation, in accordance with the operating conditions, including in view of the car thermal inertia at the transitional operating modes of the heating system. For the mathematical modelling of unsteady heat regime of a passenger car with a water heating system, the method of elementary balance was applied. The model makes it possible to simulate the operation of the heating system, to conduct a comprehensive analysis of the thermal processes in the passenger car heating, taking into account the structural changes and unsteadiness of the processes and evaluate the efficiency of their work.

Conclusion

The paper presented a mathematical model of unsteady heat exchange processes in passenger cars when using the heating systems. There were analysed the existing mathematical models, which do not fully reflect the thermal processes occurring in the car using a water heating system, where the intermediate coolant is water. The system of differential equations that characterize the unstable processes of heat transfer in a passenger car allowed developing a computational algorithm. The computer program was developed for the complex analysis of thermal processes during passenger car

doi: 10.15802/stp2018/123409

ISSN 2307–3489 (Print), ISSN 2307–6666 (Online)

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heating, taking into account structural changes and unsteadiness of processes, estimation of operation efficiency, by means of a mathematical experiment.

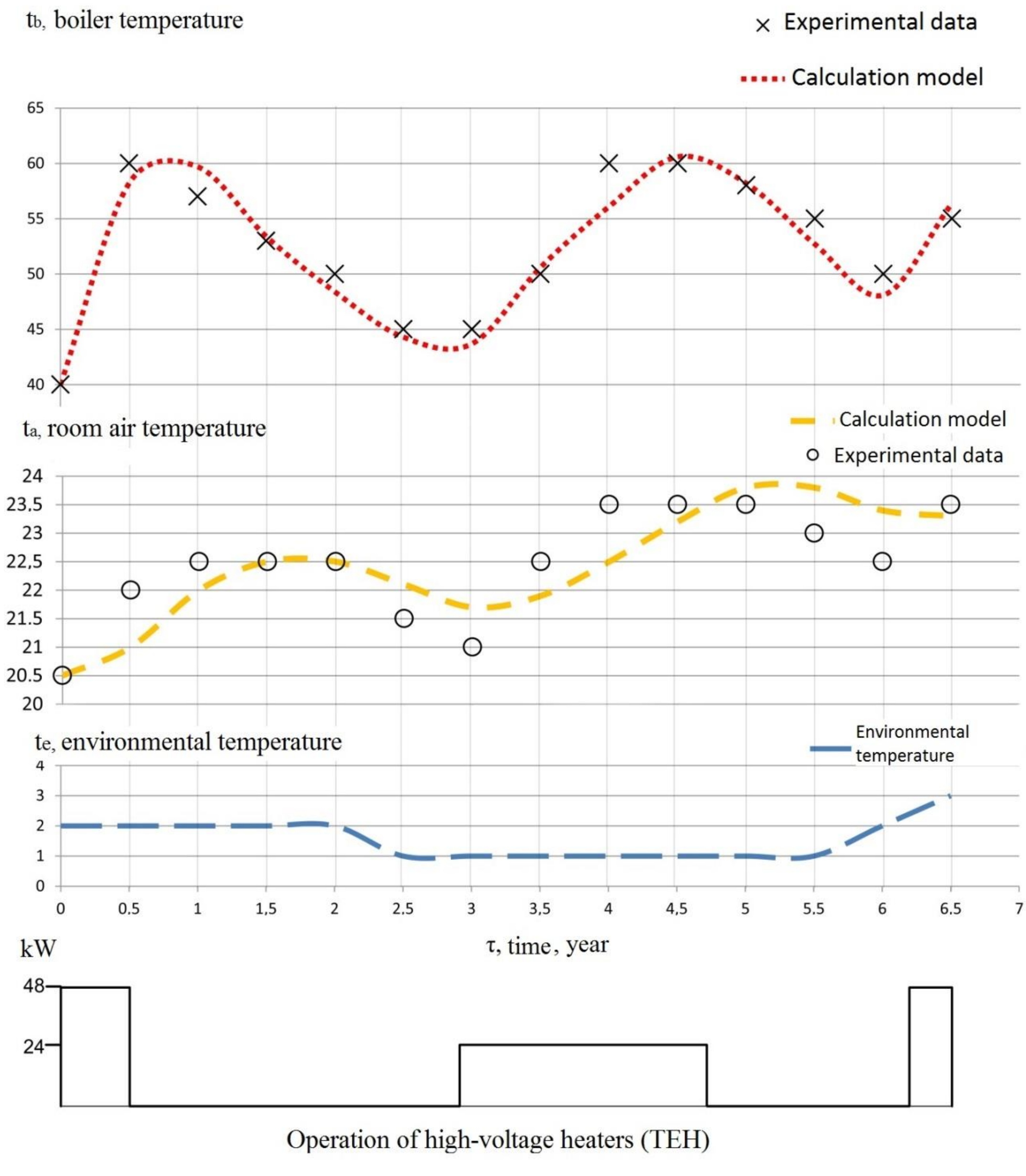

Fig. 1. Experimental temperature graphs and calculation model

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МАТЕМАТИЧНА МОДЕЛЬ НЕСТАЦІОНАРНОГО ТЕПЛООБМІНУ ПАСАЖИРСЬКОГО ВАГОНА З СИСТЕМОЮ ОПАЛЕННЯ

Мета. Існуючі математичні моделі нестаціонарних теплових процесів у пасажирському вагоні не в повній мірі відображають процеси, що відбуваються при використанні системи опалення. Крім того, нестаціонарні теплові процеси найчастіше досліджувалися в стаціонарному режимі, коли потоки та параметри теплового контуру постійні та не залежать від часу. У зв’язку з появою більш ефективних технічних рішень системи життєзабезпечення виникла потреба й у створенні нового математичного апарату, який давав би змогу врахувати ці особливості та їх вплив на перебіг нестаціонарних теплових процесів протягом усього рейсу. Мета даної роботи – створення математичної моделі теплового режиму пасажирського вагона з системою опалення, що враховує нестаціонарність теплових процесів. **Методика.** Для реалізації поставленої

doi: 10.15802/stp2018/123409

ISSN 2307–3489 (Print), ISSN 2307–6666 (Online)

Наука та прогрес транспорту. Вісник Дніпропетровського
національного університету залізничного транспорту, 2018, № 1 (73)

задачі методом моделювання була створена система диференціальних рівнянь, які описують нестаціонарні теплові процеси при опаленні пасажирського вагона; для розв'язання складеної системи рівнянь використовувався метод елементарних балансів. **Результати.** Розроблено розрахунковий алгоритм та створено комп'ютерну програму для моделювання перехідних теплових процесів у пасажирському вагоні локомотивної тяги, що дозволяє враховувати різні конструктивні рішення системи життєзабезпечення пасажирських вагонів та здійснювати моделювання нестаціонарних теплових процесів на будь-якому етапі рейсу. **Наукова новизна.** Вперше розроблено математичну модель теплових процесів у вагоні з системою опалення, що дозволяє, на відміну від існуючих моделей, досліджувати нестаціонарність теплотехнічного стану в салоні вагона за різних умов експлуатації та порівнювати роботу різних систем життєзабезпечення з точки зору їх конструктивних рішень. **Практична значимість.** Розроблена математична модель нестаціонарного теплового режиму пасажирського вагона з системою опалення для оцінки ефективності нестаціонарних перехідних температурних станів у приміщеннях пасажирського вагону з урахуванням особливостей конструкції системи опалення та нормативних вимог. Це дозволяє здійснювати розробку й реалізацію оптимальних технічних характеристик приладів опалення та побудову алгоритму керування їх роботою відповідно до умов експлуатації, у тому числі з урахуванням теплової інерції вагону при перехідних режимах роботи системи опалення шляхом математичного моделювання.

Ключові слова: математичне моделювання; пасажирський вагон; нестаціонарні теплові процеси; система опалення

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МАТЕМАТИЧЕСКАЯ МОДЕЛЬ НЕСТАЦИОНАРНОГО ТЕПЛООБМЕНА ПАССАЖИРСКИХ ВАГОНОВ С СИСТЕМОЙ ОТОПЛЕНИЯ

Цель. Существующие математические модели нестационарных тепловых процессов в пассажирском вагоне не в полной мере отражают процессы, протекающие при использовании системы отопления. Кроме того, нестационарные тепловые процессы чаще всего исследовались в стационарном режиме, когда потоки и параметры теплового контура постоянные, не зависящие от времени. В связи с появлением более эффективных технических решений системы жизнеобеспечения возникла необходимость в создании нового математического аппарата, который давал бы возможность учесть эти особенности и их влияние на ход нестационарных тепловых процессов в течение всего времени рейса. Цель данной работы – создание математической модели теплового режима пассажирского вагона с системой отопления, учитывающую нестационарность тепловых процессов. **Методика.** Для реализации поставленной задачи методом моделирования была создана система дифференциальных уравнений, которые описывают нестационарные тепловые процессы при отоплении пассажирского вагона; для решения составленной системы уравнений использовался метод элементарных балансов. **Результаты.** Разработан расчетный алгоритм и создана компьютерная программа для моделирования переходных тепловых процессов в пассажирском вагоне локомотивной тяги, позволяющая учитывать различные конструктивные решения системы жизнеобеспечения пассажирских вагонов и осуществлять моделирование нестационарных тепловых процессов на любом этапе рейса. **Научная новизна.** Впервые разработана математическая модель тепловых процессов в вагоне с системой отопления, позволяющая, в отличие от существующих моделей, исследовать нестационарность теплотехнического состояния в салоне вагона при различных условиях эксплуатации и сравнивать работу различных систем жизнеобеспечения с точки зрения их конструктивных решений. **Практическая значимость.** Разработана математическая модель нестационарного теплового режима пассажирского вагона с системой отопления для оценки эффективности нестационарных переходных температурных состояний в помещениях пассажирского вагона с учетом особенностей конструкции системы отопления и нормативных требований. Это позволяет осуществлять разработку и реализацию оптимальных технических характеристик приборов отопления и построения алгоритма управления их работой в соответствии с условиями эксплуатации, в том числе с учетом

doi: 10.15802/stp2018/123409

тепловой инерции вагона при переходных режимах работы системы отопления путем математического моделирования.

Ключевые слова: математическое моделирование; пассажирский вагон; нестационарные тепловые процессы; система отопления

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Prof. V. L. Horobets, D. Sc. (Tech.), (Ukraine); Prof. V. O. Habrynets, D. Sc. (Tech.), (Ukraine) recommended this article to be published

Received: Oct. 31, 2017
Accessed: Jan. 05, 2017

doi: 10.15802/stp2018/123409