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 with 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
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 the heat exchange processes of the car with the environment. The energy balance of such a system in a steady regime is described by the following equation:

\[ \dot{Q}_{in} - \dot{Q}_{out} = \dot{Q}_{accum} \]

where \( \dot{Q}_{in} \) and \( \dot{Q}_{out} \) are the thermal energy that comes in and that given by the heat exchange processes of the car with the environment, respectively, and \( \dot{Q}_{accum} \) is the thermal energy accumulation.

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 the requirements, 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 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 air;
- difference between the internal partition wall and the average air temperature in the car does not exceed 3 K, since the temperature difference between the external environment and the air in the car practically coincides with the temperature of 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 \( \tau \), 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:

\[
Q_{TEH}(\tau) = Q_{hs} + Q_{pp} + Q_{cl} + Q_{hl},
\]

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; \( Q_{hl} \) — 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:

\[
Q_{hs} = C_{hs} \frac{dt}{d\tau},
\]

\[
Q_{pp} = c_w G_w (t_{in} - t_{out}),
\]

\[
Q_{cl} = c_w G_w (t_{in} - t_{out}),
\]

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:

\[
t_{out} = t_a + (t_0 - t_a) e^{-\alpha \tau},
\]

where \( t_0 \) — coolant temperature at the inlet to the heating pipes; \( t_a \) — room air temperature; \( \tau \) — length of the heating pipes; \( \alpha \) — coefficient determined by the expression:

\[
\alpha = -\frac{4 k_{pp} D_o}{c_w \rho_w D_f \omega_w},
\]

where \( k_{pp} \) — coefficient of heat transfer of heating pipes; \( D_o \), \( D_f \) — outside and inside diameter of the heating pipes; \( \rho_w \) — coolant density; \( \omega_w \) — coolant speed in the heating pipes.

To analyse the heating regime of the car heating system, the equation \( (1) \div (3) \) 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 \( \tau \), at any stage and has the form:

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

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 \( \tau \), is uniquely associated with the change in its enthalpy and is determined by the relation:

\[
Q_a(\tau) = (I(t_a^* \phi_a^*) - I(t_a \phi_a)) \rho_a V_o(\tau),
\]

where \( I(t_a^* \phi_a^*) \) — 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 \( I d V \) — wet air diagram is used; \( t_a^* \) — air temperature heated by the calorifer is determined by the expression:

\[
t_a^* = \frac{c_w G_w (t_{in} - t_{out})}{c_a \rho_a V_o} + t_o.
\]
The heat brought by the heated outside air can be determined by another less precise expression:
\[ Q_a (\tau) = c_a \rho_a V_a (t_a - t_o) (\tau), \]  
(10)

The heat flows \( Q_{pp}(\tau), Q_{inf}(\tau), Q(\tau) \), entering the car at the time \( \tau \), are absorbed by three components:
\[ (Q_t + Q_a + Q_{pp})(\tau) = Q_{los} + Q_{inf} + Q_{car} , \]  
(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:
\[ Q_{los} = k_a F_o (t_o - t_e) , \]  
(12)
\[ Q_{inf} = c_o \rho_o V_{inf} (t_o - t_e) , \]  
(13)
\[ Q_{car} = C_{car} \frac{dt}{d\tau} , \]  
(14)

where \( k_a \) – coefficient of heat transfer through outer lining; \( F_o \) – area of outer lining; \( t_o \) – 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 \( \tau \), has the form:
\[ t_a (\tau) = t(0) + \left( \frac{(Q_{pp} + Q_e + Q_t) - (Q_{los} + Q_{inf})}{C_{car}} \right) \frac{dt}{d\tau} . \]  
(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:
\[ c_a; c_w; \rho_a; \rho_w; k_a; F_o; \phi_a; C_{car}; C_{inf}; G_w = \text{const} \]  
(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:
\[ \frac{dt_a}{d\tau} = \frac{Q_{pp} + (Q_t + Q_a - Q_{inf}) - Q_{los}}{C_{car}} , \]  
(17)

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

So, this is the equation of two variables:
\[ t_a = t_a (\tau) \text{ and } t_{in} = t_b (\tau) \]

The equation describing the temperature of the coolant in the boiler (7) takes the form:
\[ \frac{dt_b}{d\tau} = \frac{Q_{TEH} - (Q_{pp} + Q_{el})}{C_{hs}} , \]  
(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:
\[
\begin{cases}
\frac{dt_a}{d\tau} = \frac{Q_{pp} + (Q_t + Q_a - Q_{inf}) - Q_{los}}{C_{car}} ; \\
\frac{dt_b}{d\tau} = \frac{Q_{TEH} - (Q_{pp} + Q_{el})}{C_{hs}} .
\end{cases}
\]  
(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:
\[
\begin{pmatrix}
(C_w G_w - e^{al} \cdot C_w G_w) t_b + \\
(C_w G_w e^{al} - C_w G_w - k_o F_o) \times
\end{pmatrix}
\]  
(20)

\[ t_a + (Q_t + Q_a - Q_{inf}) + k_a F_o \frac{dt}{d\tau} = e_t t_b + \alpha_1 t_a + \theta_1 \]

where \( e_t, \alpha_2 \) – indicators of car thermal inertia at the considered stage; \( \theta_1 \) – index of car heat entropy at the considered stage:
\[ e_t = \frac{C_w G_w (1 - e^{al})}{C_{car}} , \]  
(21)

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\[ D = p^2 - 4g. \]  

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

\[ t_b(\tau) = K_1 e^{K_1 \tau} + K_2 e^{K_2 \tau} + \frac{f}{g}, \]

\[ t_a(\tau) = K_1 R_1 e^{R_1 \tau} + K_2 R_2 e^{R_2 \tau} - A_2 t_b - C_2 \]

where \( R_1, R_2 \) are the roots of the characteristic equation:

\[ R_1 = -\frac{p - \sqrt{D}}{2}, \]

\[ R_2 = -\frac{p + \sqrt{D}}{2}. \]

Discriminator of the characteristic equation:

\[ K_1 = \begin{pmatrix} \frac{\theta_1 + \theta_2 + \frac{f}{g}}{g} \\ -(R_2 - \theta_2) \left( t_b(0) - \frac{f}{g} \right) \end{pmatrix} \]

\[ K_2 = t_b(0) - \frac{f}{g} - K_1. \]

Thus, the initial system of equations has the form:

\[
\begin{aligned}
\frac{dt_b}{d\tau} &= \varepsilon_1 t_b + \omega_1 t_a + \theta_1; \\
\frac{dt_a}{d\tau} &= \varepsilon_2 t_a + \omega_2 t_a + \theta_2,
\end{aligned}
\
\]

That is, linear equations with constant coefficients.

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

\[ t_b'' + p \cdot t_b' + g \cdot t_b = f, \]

where

\[ p = -(\omega_1 + \varepsilon_2), \]

\[ g = \omega_1 \varepsilon_2 - \omega_1 \omega_2, \]

\[ f = \omega_2 \theta_1 - \omega_1 \theta_2. \]
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**

| Parameter                                                                 | Value       |
|---------------------------------------------------------------------------|-------------|
| Body area, m²                                                               | 330         |
| Coefficient of heat transfer of partitions, taking into account the speed of movement | 1.3         |
| Infiltration volume, m³/h                                                  | 200         |
| Area of heating pipes, taking into account the coefficient of finning, m²·kₙₙ | 19.5–2.9    |
| Coolant circulation velocity, m/s                                          | 0.04        |
| Number of passengers, people                                              | 52          |
| Car heat capacity, Cₖₑₗ, kW                                               | 3056        |
| Heat capacity of the heating system, Cₘₗₖ, kW                             | 1000        |
| Coefficient of heat transfer of heating pipes, W/m²·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 a 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
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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 досліджувались 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 виникала потреба й у створенні нового математичного апарату, який давав би змогу враховувати ці особливості та їх впливи 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
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загальних методом моделювання була створена система диференціальних рівнянь, які описують нестационарні теплові процеси при опаленні пасажирського вагона; для розв'язання складеної системи рівнянь використо- вувався метод елементарних балансів. Результати. Розроблено розрахунковий алгоритм та створено комп’ютерну програму для моделювання переходних теплових процесів у пасажирському вагоні локомотивної тяги, що дозволяє враховувати різні конструктивні рішення системи життєобезпечення пасажирських вагонів та здійснювати моделювання нестационарних теплових процесів на будь-якому етапі рейсу. Наукова новизна. Вперше розроблено математичну модель теплових процесів в вагоні з системою опалення, що дозволяє, на відміну від існуючих моделей, досліджувати нестационарність теплотехнічного стану в салоні вагона за різних умов експлуатації та порівнювати роботу різних систем життєобезпечення з точки зору їх конструктивних рішень. Практична значимість. Розроблено математична модель нестационарного теплового режиму пасажирського вагона з системою опалення для оцінки ефективності нестационарних переходних термічних станів у приміщеннях пасажирського вагону з урахуванням особливостей конструкції системи опалення та нормативних вимог. Це дозволяє здійснювати розробку та реалізацію оптимальних тех- 2424-8479

МАТЕМАТИЧЕСКАЯ МОДЕЛЬ НЕСТАЦИОНАРНОГО ТЕПЛООБМЕНА ПАССАЖИРСКИХ ВАГОНОВ С СИСТЕМОЙ ОТОПЛЕНИЯ

Цель. Существующие математические модели нестационарных тепловых процессов в пассажирском ва- гоне 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. Во связи с появлением более эффектив- ных технических решений системы жизнеобеспечения возникла необходимость в создании нового матема- тического аппарата, который давал бы возможность учесть эти особенности и их влияние на ход нестаци- онарных тепловых процессов в течение всего времени рейса. Цель данной работы – создание математической модели теплового режима пассажирского вагона с системой отопления, учитывающей нестационарность тепловых процессов. Методика. Для реализации поставленной задачи методом моделирования была созда- на система дифференциальных уравнений, которые описывают нестационарные тепловые процессы при отоплении пассажирского вагона; для решения составленной системы уравнений использовался метод еле- ментарных 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 осу- ществлять моделирование нестационарных тепловых процессов на любом этапе рейса. Научная новизна. Впервые разработана математическая модель тепловых процессов в вагоне с системой отопления, позволя- ющая, 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 термических состояний в помещениях пассажирского вагона с учетом особенностей конструкции системы отопления и нормативных требований. Это позволяет осу- ществлять разработку и реализацию оптимальных технических характеристик приборов отопления и по- строения алгоритма управления их работой в соответствии с условиями эксплуатации, в том числе с учетом

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

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теплової інерції вагона при переходних режимах роботи системи опалення путем математичного моделювання.

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

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