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Mathematical model of the liquefied methane phase transition in the cryogenic tank of a vehicle

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In order to increase the efficiency of using vehicles (VEH) in mining and quarrying conditions, it is necessary to improve the components of gas equipment (cryogenic tank, gas nozzles, fuel supply cryogenic tubes, etc.) for supplying liquefied natural gas to the engine, as well as storage of liquid methane in a cryogenic tank with a long service life. For this, it is necessary to consider the process of heat and mass transfer of liquefied natural gas in a two-phase liquid-gas medium, taking into account the phase transition in the closed volume of the cryogenic tank under consideration.

The article presents a model of unsteady heat and mass transfer of a two-phase liquefied methane medium in a developed two-tank cryogenic tank using a Cartesian coordinate system with fractional control volumes in space.

The experimental data confirm the efficiency of using a cryogenic tank on the VEH platform, in which the run on liquefied methane compared to standard fuels is tripled, the shelf life of liquefied gas in the proposed cryogenic tank is 2-2.5 times longer than in the standard one.

***Key words*:** two-tank cryogenic reservoir; thermal conductivity; heat capacity; liquefied methane; thermal insulation layer; external tank; internal tank; temperature; pressure; time

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Introduction. Loading and transportation of rock mass in quarry conditions is carried out by heavy vehicles, which consume a large amount of diesel fuel and pollute the environment with a high content of suspended particles formed from rock mining. In 2017, the Government of the Russian Federation adopted the state program “Expanding the Use of Natural Gas as a Motor Fuel in Transport and Special Purpose Equipment”, the implementation of which is envisaged until 2025. In this regard, the use of liquefied natural gas in special cryogenic fuel is the most promising system on board a heavy vehicle. This will reduce the emission of harmful substances by 3-4 times, increase the energy efficiency of mining equipment due to the use of a cryogenic tank filled with liquid methane on board, and reduce the number of refuelings by three times compared with compressed natural gas, for which it is necessary to use a large number of gas cylinders. From an economic and environmental point of view, the conversion of the entire fleet of mining equipment will not only increase production efficiency due to the low cost of gas and improvement of the environmental situation, but will also solve the global problem of reducing the greenhouse effect due to the efficient combustion of methane in heat engines.

Natural gas as an energy carrier is 3-5 times cheaper than petroleum fuels, which ultimately makes it very competitive [1-3, 5-7]. In vehicles operating in mining and quarrying conditions, when performing mining operations, the natural gas began to be actively used as an alternative, environmentally friendly and cheapest type of fuel. For the extraction of mineral raw materials the vehicles adapted to difficult geological conditions use compressed natural gas (CNG) as a motor fuel and gas equipment of the 4th generation, which involves the use of a large number of cylinders with compressed methane. This approach is associated with the inconvenience of placing equipment on the carrier platform, and an increased risk of fire and explosive situation in mountain and quarry conditions. A possible solution to these problems is to use liquefied natural gas. However, cryogenic fuel, in turn, has a number of shortcomings: low density, the use of ultra-low temperatures,

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high volatility, and, if improperly handled, also a high explosion hazard. This can explain the low rate of development of gasification of gas engine vehicles using liquefied natural gas and its practical implementation in various sectors of the economy [9, 10].

When modeling the thermodynamic process of a gas engine, it is necessary to take into account the theoretical foundations and features of the properties of fuel vapor and air. It is proved that under conditions of extremely low temperature, a decrease in the rate of combustion kinetics (rather than mixing) is the main controlled factor. The question arises – is it logical to use the characteristic equations for a more accurate description of the unsteady and nonequilibrium heat release process? In this case, the question remains open, which was the main factor for conducting the research. Theoretical and practical studies were carried out [16, 19, 20, 23], where equations were considered that describe the state of the working mixture mainly for traditional types of fuel – gasoline, diesel, etc. For alternative types of fuels – compressed natural gas (CNG – Compressed Natural Gas) and liquefied petroleum gas (LPG – Liquified Petroleum Gas) there are no such studies. The considered semi-theoretical forecasting method is based on three approximate theoretical equations of the flame propagation velocity: thermodynamics, the combustion process of I.I.Wiebe and the turbulent velocity of flame propagation.

Using a cryogenic fuel system in a motor vehicle, methane must be fed to actuators (a unified candle-nozzle) directly into the combustion chamber in a gaseous form, using an adapted gas equipment of the 4th, 5th or 6th generation with heating elements. It is known that methane in terms of mass heat of combustion is not inferior to traditional types of fuels; in the case of creating gas mixtures based on methane, it can exceed the parameters of heat energy release, which is important when creating an energy-efficient gas-powered vehicle.

In order to increase the energy efficiency of a gas engine, it is necessary to optimize the main parameters of individual engine mechanisms and systems using a unified electronic control system (Fig.1), which will generally improve the performance of the vehicle.

The most important direction is considered in the article – the influence of individual structural elements of the fuel supply system on the process of supplying and burning a gas-air mixture directly to the combustion chamber [8, 18].

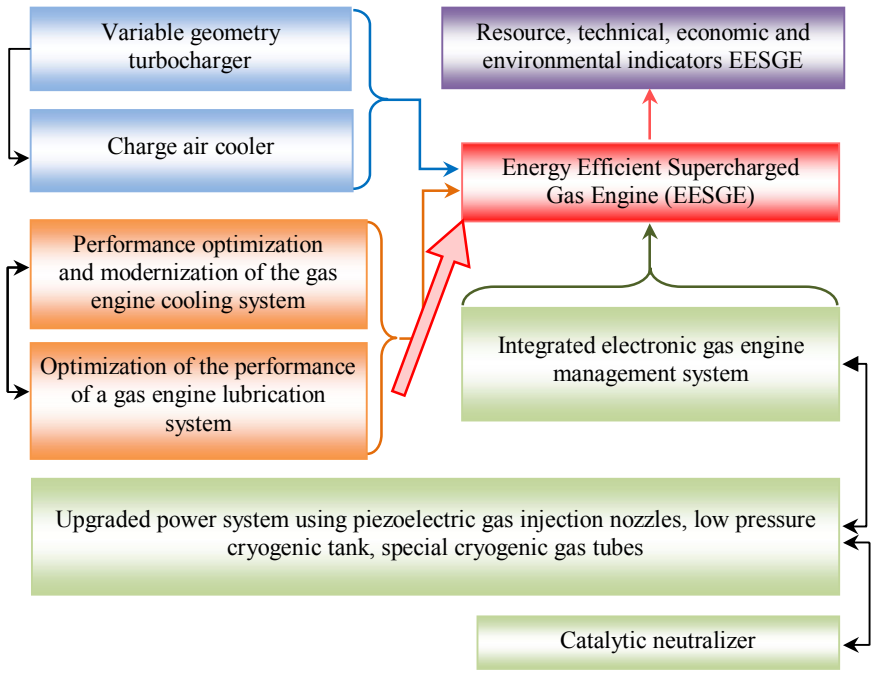


Fig.1. Energy efficiency scheme of gas engine technology with integrated optimization of mechanisms and systems of a gas engine

There are two large manufacturers of gas internal combustion engine in the Russian market of gas engine equipment, these are KamAZ with a series of KamAZ-820 engines and the GAZ YaMZ Group – the YaMZ-534 CNG engine.

To accomplish the task of improving the operational performance of a car equipped with a gas engine operating on a methane cycle with an ignition system from an electric discharge, it is necessary to consider in more detail the process of supplying liquefied methane through a fuel supply system, also, to increase the drainage period of a specially

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designed cryogenic tank (CT), to theoretically determine the main indicators of unsteady heat and mass transfer of the interphase transition of methane from liquid to gas.

We will consider in detail the process of filling CT with liquefied methane, and also, without deepening into the fuel supply process, we will determine the peak pressure in the system (CT and pipes), fragments of the mathematical model of heat and mass transfer processes and the theoretical determination of the peak pressure values in the CT and pipeline when a fresh LNG stream is supplied, as well as time after refilling the tank with a new portion of fuel. The scientific methods [4, 8, 17, 21, 24] have been laid and adapted to the basis of this mathematical model.

In the process of studying the convective heat transfer of cryogenic methane in a special two-reservoir design bureau, it is necessary to calculate the temperature and pressure gradient under the conditions of a phase transition at which the velocity vector of the interfacial transition from a liquid medium to a vapor is determined. The calculation is carried out under conditions of artificial heating and cooling with the most approximate operating conditions of the design bureau. The investigated object is heated and then cooled in a special chamber several times with different indicators of the liquefied methane filling volume. The main objective of the study is to analyze the process of temperature change during the phase transition of liquefied methane, taking into account the thermal insulation layer of CT. Based on the results of the studies, the dependences of the temperature field gradient and heat capacity of CT with thermal insulation are obtained.

Research Methodology. A mathematical model is proposed that describes the phase transition of liquefied methane in a specially developed two-tank cryogenic reservoir in order to determine the total heat capacity in the boundary conditions of the studied object.

Heat flow study. Heat flow $Q(\tau)$, coming to the outer surface of the wall of CT, created artificially by a heat source simulating solar radiation (hereinafter – the heater), which is associated with the difference in volumetric temperatures of the thermal insulation layer of the external and internal reservoir of CT $\sum T_{1\,\text{res}}(\tau_1)$, $\sum T_{2\,\text{res}}(\tau_2)$, and the artificial heater T_{heat},

$$Q(\tau) = k_{\text{т}}\left[\left(T_{1\,\text{res}}(\tau_1) + T_{2\,\text{res}}(\tau_2)\right) - T_{\text{heat}}\right], \tag{1}$$

where $k_{\text{т}}$ – thermal conductivity of a two-layer heat-insulating shell of a wall of CT.

We will write equation (1) taking into account the distribution of the heat flux between the walls of two tanks of a solid fuel CT, as well as between their two-layer insulating shells. When determining the heat flux in a thermally insulating CT, taking into account its design features, we will use the dimensionless number Dn, the expression will have the following form:

$$Q(\tau) = \left(C_1(T_{1\,\text{res}})\frac{dT_{1\,\text{res}}(\tau_1)}{d\tau_1} + C_{1\,\text{ven}}(T_{1\,\text{ven}})\frac{dT_{1\,\text{ven}}(\tau_1)}{d\tau_1}\right)Dn_1 +$$

$$+\left(C_2(T_{2\,\text{res}})\frac{dT_{2\,\text{res}}(\tau_2)}{d\tau_2} + C_{2\,\text{ven}}(T_{2\,\text{ven}})\frac{dT_{2\,\text{ven}}(\tau_2)}{d\tau_2}\right)Dn_2, \tag{2}$$

where C_i – the total heat capacity of the corresponding tank with a thermal insulation shell of the i-th layer (1 – external, 2 – internal tank); T_i – volumetric average temperature in the corresponding tank, taking into account the i-th thermal insulation layer (IL).

Knowing the composition, properties, and also the heat capacity of the object under study, we determine the total heat capacity of the CT according to the following expression:

$$C(T_{\text{CT}}) = \left(\frac{k_{\text{т}}\left[\sum T_{1\,\text{res}}(\tau_1) - T_{\text{heat}}\right] - C_{1\,\text{ven}}(T_{1\,\text{ven}})\dfrac{dT_{1\,\text{ven}}(\tau_1)}{d\tau_1}}{\dfrac{dT_{1\,\text{ven}}(\tau_1)}{d\tau_1}}\right) +$$

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$$+\left(\frac{k_{\text{т}}[\sum T_{2\,\text{res}}(\tau_2)-T_{\text{heat}}]-C_{2\,\text{ven}}(T_{2\,\text{ven}})\dfrac{dT_{2\,\text{ven}}(\tau_2)}{d\tau_2}}{\dfrac{dT_{2\,\text{ven}}(\tau_2)}{d\tau_2}}\right). \quad (3)$$

Having analyzed expression (3), it is necessary to measure the volumetric average temperature $T_{CT}(\tau)$ and two-layer thermal insulation shell $T_{iIns}(\tau_i)$. Based on the measurement results, time dependencies of two indicators $T_{\text{CT}}(\tau)$ и $T_{iIns}(\tau_i)$ are built, then, by comparing the optimum of the obtained results, the time dependence of the temperature at any selected point in the control region, likewise we build the connection with all the other temperature values of the investigated CT regions. For a more accurate distribution of the temperature field in CT, we first select the cylindrical surface of the external reservoir N 1 (ER N 1), and then the cylindrical surface of the internal reservoir N 2 (ER N 2), taking into account the second-order condition, the influence of temperature jumps on a two-layer shell at the beginning of ER N 1, then ER N 2.

The gradient of the temperature field of the cylindrical surface of ER N 2 $T_{\text{ER2}}(r, z, \tau)$ can be determined using various methods [9, 10, 17]. In this case, there is no need to consider one of the variants of the previously used methods, which are presented in detail in [11]. The analysis of the existing models in previous studies showed that there are certain boundary conditions that were not taken into account and reflected [14–15] and do not provide a complete solution of the thermal conductivity indicators of a complex system for recycling biphasic liquefied methane in the design of a CT. Therefore, to transform the selected model and for the calculation accuracy, this heat and mass transfer process will be described by the heat equation taking into account non-stationary:

$$\frac{\partial^2 T_{\text{ER2}}}{\partial r^2}+\frac{1}{r}\frac{\partial T_{\text{ER2}}}{\partial r}+\frac{\partial^2 T_{\text{ER2}}}{\partial r^2}=\frac{1}{a}\frac{\partial T_{\text{Er2}}}{\partial r}; \quad (4)$$

$$\frac{\partial^2 T_{\text{ER1}}}{\partial r^2}+\frac{1}{r}\frac{\partial T_{\text{ER1}}}{\partial r}+\frac{\partial^2 T_{\text{ER1}}}{\partial r^2}=\frac{1}{a}\frac{\partial T_{\text{ER1}}}{\partial r}. \quad (5)$$

The solution of this problem requires the installation of the following initial and boundary conditions: when heated $T_{\text{heat}} > T_{\text{ER}_i}$, where T_{ER_i} – temperature field of internal and external reservoirs (ER N 1 and ER N 2), respectively,

$$\sum T_{\text{CT}}(r, z, \tau)=\sum T_{\text{ER}_i}+\sum(T_{\text{heat}}-T_{\text{ER}_i})\delta_{\text{ER2}}(r, z, \tau); \quad (6)$$

due to the cooling $T_{\text{ER}_i} > T_{\text{heat}}$

$$\sum T_{\text{CT}}(r, z, \tau)=\sum T_{\text{ER}_i}-\sum(T_{\text{heat}}-T_{\text{E R}_i})\delta_{\text{ER2}}(r, z, \tau), \quad (7)$$

where $\delta_{\text{ER2}}(r, z, \tau)$ – relative excess temperature of ER N 2 of cryogenic reservoir; δ_{back} determined by the method [4].

The differential heat equation of ER N 1 and ER N 2, taking into account the thermal insulation layer and the indicated initial and boundary conditions, will have the following form:

$$\frac{\partial^2 T_{\text{INS1}}}{\partial z^2}+\frac{\partial^2 T_{\text{INS1}}}{\partial r^2}+\frac{1}{r}\frac{\partial T_{\text{INS1}}}{\partial r}=\frac{1}{a}\frac{\partial T_{\text{INS1}}}{\partial \tau}; \quad (8)$$

$$\frac{\partial^2 T_{\text{INS2}}}{\partial z^2}+\frac{\partial^2 T_{\text{INS2}}}{\partial r^2}+\frac{1}{r}\frac{\partial T_{\text{INS2}}}{\partial r}=\frac{1}{a}\frac{\partial T_{\text{INS2}}}{\partial \tau}. \quad (9)$$

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Analysis of the calculation results showed that each obtained average temperature in the cross section of the calculated area of the insulating layer is connected by the temperature of ER N 1 and ER N 2 of CT respectively, with the following expression.

$$\sum T_{\mathrm{INS}_i}(\tau) = T_{\mathrm{CT}} + 0.532(\sum T_{\mathrm{ER}_i} - T_{\mathrm{CT}}). \quad (10)$$

The average temperature change rate in the cross section of the calculated area of the thermal insulation layer of each tank is determined from the expression

$$\frac{dT_{\mathrm{INS}_i}(\tau)}{d\tau} = 0.468\frac{dT_{\mathrm{CT}}(\tau)}{d\tau}. \quad (11)$$

An analysis of the temperature field results obtained by expression (11) showed that for a complete understanding of the heat and mass transfer process, it is necessary to determine the total heat capacity index of a CT with an insulating shell, which has the following form:

$$\sum C(T_{\mathrm{CT}}) = \frac{k_m[T_{\mathrm{CT}} - \sum T_{\mathrm{INS}_i}]}{\frac{dT_{\mathrm{CT}}}{d\tau}} - 0.468\sum C_{\mathrm{INS}_i}(\sum T_{\mathrm{INS}_i}). \quad (12)$$

When determining the heat capacity of a two-tank CT, it is necessary to consider the mathematical model of the process of unsteady heat and mass transfer of a two-phase medium of liquefied methane inside the volume.

Investigation of the indicators of the process of unsteady heat and mass transfer of a two-phase medium of liquefied methane in CT. In the developed mathematical model, it is necessary to take into account the phases under study: we denote the liquid phase by the coefficient $k_{\mathrm{liq}} = 1$ and gaseous $k_{\mathrm{gas}} = 2$ with a variable heterogeneous surface at the interface between liquid and vapor methane in the LM-GM system. It is also necessary to take into account the solid wall of the tank, accepted as $k_{\mathrm{sol}} = 3$ as a closed volume [6]. Heat and mass transfer process studied in the volume of the internal reservoir with the boundary conditions of LM-GM is considered for each region separately for vapor and liquid methane. To determine the rate of phase transition of liquefied methane, expressing through the temperature gradient in a given CV along the entire interface, for the first time we represent in a modernized form, the model in the form of a two- or three-dimensional Cartesian coordinate system (CCS) with fractional control volumes in space:

$$\frac{\partial \rho_k}{\partial \tau}(x_k, y_k, z_k); \quad \rho_k \vec{V}_k(x_k, y_k, z_k); \quad \lambda_k T_k(x_k, y_k, z_k). \quad (13)$$

In function (13) of the heat and mass transfer process, we consider its main values in space along the axes of the Cartesian coordinate system, taking into account the vector differential operator of the changing density indices and the vector rate of the phase transition of liquefied methane located in CT.

When considering the process of heat and mass transfer of liquefied methane in a closed volume, the proposed mathematical model is based on the equation of conservation of phase mass

$$\frac{\partial \rho_{\mathrm{liq}}}{\partial \tau} + \nabla(\rho_{\mathrm{liq}}\vec{v}_{\mathrm{liq}}) = -\sum_{i=1}^{n_i} m_i, \quad (14)$$

where $\rho_{\mathrm{liq}} = \varepsilon\rho_{\mathrm{liq}}^0$ – reduced density of the studied k-th phase of natural gas; ε – volume fraction of the phase in gaseous form; n – gas mixture consisting of the i-th gas component (CH_2, C_2H_6, C_3H_8, C_4H_{10}, C_5H_{12}, H_2, CO_2, N_2 и т.д.); m_i – bulk density of the mass flow of the i-th gas component in methane, passing from liquid to gaseous phase.

The work presents fragments of a simplified mathematical model of the process of heat and mass transfer of the phase transition of liquefied methane in the closed volume of the developed

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cryogenic tank to determine the fraction of methane released from the liquid phase into the gaseous one. Determining the phase transition rate of liquefied methane in a cryogenic tank makes it possible to develop a gas recirculation technology with the aim of extending the shelf life of liquefied methane in a drainless mode. Consider the process of converting the velocity vector and the gas mass flux density in a given volume.

The modernized equation will have the following form:

$$\frac{\partial \rho_k}{\partial \tau} + \nabla(\rho_k \vec{V}_k) = 0, \tag{15}$$

$$\frac{\partial(\rho_k V_{k,x})}{\partial \tau} + \nabla(\rho_k \vec{V}_k V_{k,x} - \rho_k \nu_{k,eff} \nabla V_{k,x}) = -\frac{\partial \rho_k}{\partial_x} + g_x \rho_k, \tag{16}$$

$$\frac{\partial(\rho_k V_{k,y})}{\partial \tau} + \nabla(\rho_k \vec{V}_k V_{k,y} - \rho_k \nu_{k,eff} \nabla V_{k,y}) = -\frac{\partial \rho_k}{\partial_y} + g_y \rho_k, \tag{17}$$

$$\frac{\partial(\rho_k h_k)}{\partial \tau} + \nabla(\rho_k \vec{V}_k h_k - \lambda_{k,eff} \nabla T_k) = -\frac{\partial \rho_k}{\partial_\tau} + Q_{r,k}, \tag{18}$$

$$\lambda_{k,eff} = \lambda_k + \rho_k C_{p,k} \frac{\nu_{k,t}}{p_{\Gamma_t}}, \text{ taking } k = 1,2,$$

where $\lambda_{k,eff}$ – coefficient of effective thermal conductivity of methane in the studied k-th phase; ν_k – mass-average methane velocity in the studied k-th phase; ρ_k – methane density in the studied k-th phase; $\vec{V}_k$ – methane velocity vector during phase transition; $C_{p,k}$ – heat capacity of methane at constant pressure in the studied k-th phase; T_k – methane temperature in the studied k-th phase; $Q_{r,k}$ – heat flow in the amount of CT; h_k – length of pipe at the outlet to the fuel system; τ –filling liquid methane in CT cooling time.

The wall temperature of the cryogenic tank under boundary conditions was calculated using the heat equation

$$\rho_3 C_3 \frac{\partial T_3}{\partial \tau} - \nabla(\lambda_3 \nabla T_3) = Q_{V,3} + Q_{r,3}. \tag{19}$$

It is also necessary to consider the conditions for the phase transition of liquefied methane at the boundary of the conversion from boiling to liquid, taking into account the determination of the critical temperature of methane $T_{\text{cr. min}}$.

Under conditions of increased pressures in the investigated volumes of the design bureau, it is necessary to know the maximum permissible critical temperatures, for which the modernized Greaves - Todos equation is used for a one-component natural gas. The critical gas temperature in the cryogenic fuel system is determined by the expression:

$$T_{\text{cr. min}} = \sum_{i=1}^{n} \left(T_k / \left[1 + (1/x_i) \sum_{j \neq i}^{j=1} A_{ij} x_j \right] \right), \tag{20}$$

where x_i – mole fraction of i-th gas; x_j – mole fraction of j-th gas; A_{ij} – constant gas ratio.

The value A_{ij} in equation (20) is determined by the method presented in the source [5] as a function $\tau = T_{mj} / T_{mi}$ (ratio of normal boiling points) under the condition that $\tau > 1$. If in the cryogenic fuel system there is a mixture of gases based on methane, then expression (20) will have the following form:

$$T_{\text{cr. min}} = T_{G1} / [1 + (x_2 / x_1) A_{12}] + T_{G2} / [1 + (x_1 / x_2) A_{21}]. \tag{21}$$

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The given correlation is also applicable to cryogenic fuel systems with the content of various aromatic hydrocarbons and temperature correction of pure aromatic components according to the rule [10]:

$$T_{\text{cr.cor}} = \begin{Bmatrix} T_{\text{cr}} - 8 \text{ (если } x_{\text{arom}} > 0.6) \\ T_{\text{cr}} - 22 \text{ (если } x_{\text{arom}} < 0.6) \end{Bmatrix}. \quad (22)$$

When using a methane number as the main indicator of gas fuel in a liquefied form, it is necessary to take into account the composition of mixtures in volume fractions, for example, for using a methane-butane mixture in a fuel system, for each gas we take its constant coefficient obtained empirically, $A_{12} = 0.86$ and $A_{21} = 0.60$ (1 – methane, 2 – butane).

This calculation is necessary when using gas mixtures in liquefied methane to increase the shelf life of natural gas in a cryogenic tank in a drainless mode.

To describe the heat transfer process in a CT filled with liquid methane, with allowance for the phase transition rate, we omit equations (20)-(22) and perform the calculation using equations (13)-(18) for further modeling of the heat transfer process in a CT with a thermo-insulating shell.

It follows from equations (18) and (19) that some terms were introduced for the accuracy of calculation and further modeling of the convective heat and mass transfer process under boundary conditions between the wall in the upper part of ER N 2 and the liquid methane phase in the lower part along the entire interface.

Using some assumptions in the equation, taking into account the energy conservation law for k_i-th phase, the expression will have the following form:

$$\rho_k \frac{\partial h_k}{\partial \tau} = \nabla \left(\frac{\lambda_{k,eff}}{C_{pk}} \nabla h_k \right) + Q_{r,k}. \quad (23)$$

To calculate the change in pressure in space $dP_0 / d\tau$ we will write the expression

$$\frac{\partial \rho_k}{\partial \tau} = \left(\frac{\partial \rho_k}{\partial p} \right)_{\text{T}} \frac{\partial \rho_0}{\partial \tau} + \left(\frac{\partial \rho_k}{\partial T} \right)_p \frac{\partial T}{\partial \tau} = \rho_k \alpha_k \frac{\partial p_0}{\partial \tau} - \rho_k \beta_k \frac{\partial T}{\partial \tau}, \quad (24)$$

where $\alpha_k = \frac{1}{\rho_k} \left(\frac{\partial \rho_k}{\partial T} \right)_{\text{T}}$ – coefficient taking into account the isothermal compressibility of natural gas; $\beta_k = \frac{1}{\rho_k} \left(\frac{\partial \rho_k}{\partial T} \right)_p$ – coefficient taking into account the thermal expansion of natural gas.

Integrating expression (24), we obtain an equation corresponding to the state of the gas occupied by the kth phase in the calculated volume of the superimposed grid in the CCS. Having summed all the equations that correspond to each phase, we obtain an equation for calculating the time derivative depending on the average pressure:

$$\frac{d\rho_0}{d\tau} = \frac{\sum_k \int v_k \rho \beta_{tk} \frac{\partial T}{\partial \tau} dV}{\sum_k \int v_k \rho \alpha_{tk} dV}. \quad (25)$$

The computational grid with the designation of the main nodes and faces of the selected control volume (CV) in the cylindrical part of CT is shown in Fig.2

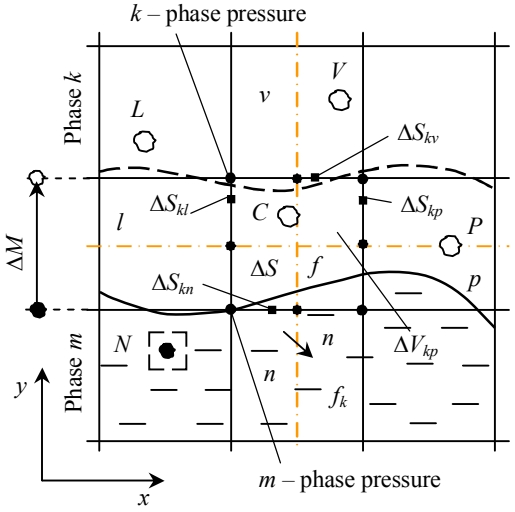


Fig.2. Simulation Grid of phase transition of liquefied methane in the system "Liquid – gas" in two-dimensional (OX, OY) in Cartesian coordinate system

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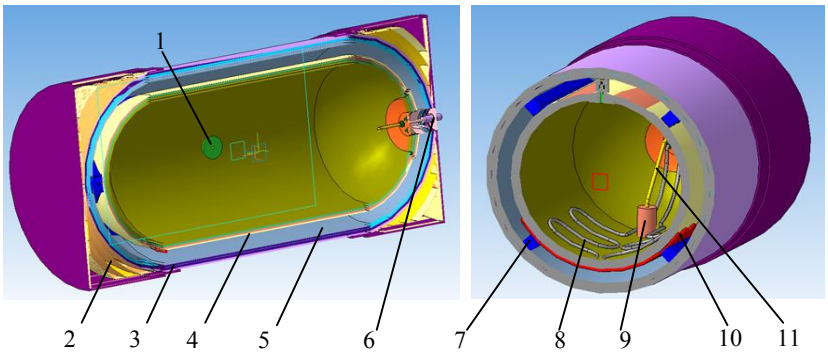


Fig.3. General view of the cryogenic tank for vehicles

1 – gas discharge valve; 2 – multilayer thermal insulation; 3 – external container with two-layer insulation; 4 – internal container with two-layer insulation; 5 – inter-tank gas space; 6 – central line for supplying compressed gas; 7 – inter-capacity spacers; 8 – gas supply pipe with cascade throttling; 9 – gas feeder; 10 – dropping tube with cascade throttling; 11 – feeder tube

[12], the letter f denotes the phase boundary. One should enter the following calculated parameters to describe fractional CV: ΔS_{kj} – face area j of CV, occupied in phase $k(j = \{p, l, v, n\})$, consisting of faces ΔS_{kp} – right one, ΔS_{kl} – left one, ΔS_{kv} – upper one, ΔS_{kn} – lower one; ΔV_{kp} – volume of methane occupying the cylindrical part of the tank, taking into account the corresponding k – phase; ΔS_{ki} – estimated area of the i-th segment, which belongs to the corresponding CV.

The nodal points C, P, L, V, N, adopted for modeling the state of the gas in the tank, were placed at the center of mass convergence of the CV occupied or filled with the corresponding phase; therefore, some points in the nodal part (L, P, C) or (C, N, V) cannot be on one straight line.

We divide the grid into the corresponding areas, which will be taken into account to determine the mass flow of liquefied methane through a conventionally accepted face at the interface.

The mass conservation equation (14), having modernized into equation (15) and integrated, will be presented in the following form:

$$\frac{\rho_p - \rho_p^0}{\Delta\tau}\Delta V_c + \sum F_j = 0, \qquad (26)$$

where $F_j = \vec{n}_j \rho_j \vec{V}_j \Delta S_j$ – mass gas flow through small segments of the CV; $\vec{n}_j$ – unit vector assigned to the interphase boundary section of the CV ($j = \{p, l, v, n, f\}$); $\Sigma F_j = F_p + F_l + F_v + F_n + F_f$ – the sum of the mass flows of gas at certain boundaries of the cryogenic tank in a cross section: p – right one, l – left one, v – upper one, n – lower one, f – central parts.

A detailed mathematical model for the description of heat and mass transfer in a cryogenic tank is presented in [18], where the boundary conditions of the CV are considered, the pressure fields are calculated at the main nodal points of the phase boundary of the calculated CV grid. The resulting system of algebraic equations as a result of discretization is solved using the method of longitudinal-transverse sweep of heat and mass transfer in the rectangle CV. To realize the universality of the mathematical model, using the FlowVision program, it was possible to simulate the process of formation of isothermal zones and fields of liquefied methane phase transition velocity vectors in a cryogenic tank.

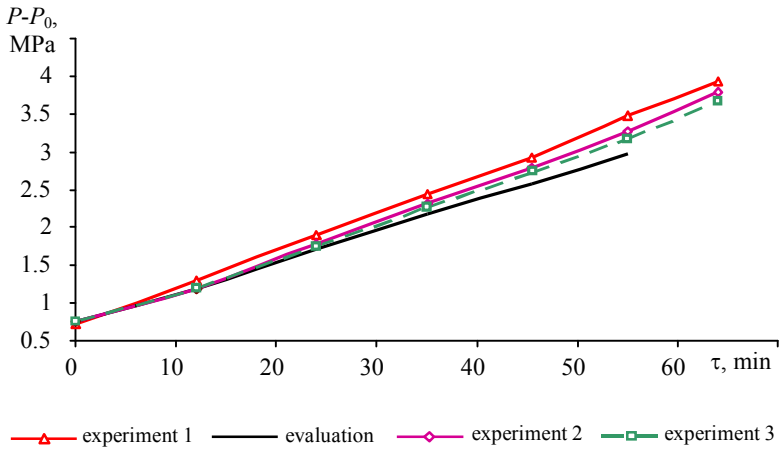


Fig.4. The dependence of the saturated gas pressure on the heating time during the phase transition of liquefied methane for experiments 1-3 with 50 % filling of the tank

Results discussion. The results of mathematical modeling of the phase transition of liquefied methane in the cylindrical part of the object under study made it possible to develop a method for controlling the temperature field in a CT in order to increase the shelf life of liquid methane in a drainless mode.

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The thermophysical properties of liquefied natural gas were used according to the obtained experimental data [28]. Based on the results obtained [13], a prototype of a two-tank cryogenic tank with a thermal insulation shell was developed (Fig.3).

ling the temperature field in CT in order to increase the shelf life of liquid methane in a drainless mode. The thermophysical properties of liquefied natural gas were used according to the obtained experimental data [28]. Based on the results obtained [13], a prototype of a two-tank cryogenic tank with a thermal insulation shell was developed (Fig.3).

Let us calculate the convective heat and mass transfer during drainless storage (DlS) of liquefied methane in a two-tank cryogenic tank. The research indicators from the initial state of heat influx in the gas volume depending on the duration of the heating time are clearly illustrated in Fig.4.

Experiments 1 and 2 are carried out under almost the same conditions when filling liquefied methane in a cryogenic tank and exposure to solar heat flux. Some discrepancy between the experimental data is caused by the difference in the initial conditions of thermal effects on the fueling system after the valve is closed.

With prolonged exposure to heat fluxes. emanating from the engine compartment of the vehicle and solar radiation to the walls of with a heat-insulating shell, in experiment 4 there is a high convergence of experimental data and calculated indicators of drainage-free storage of liquefied methane.

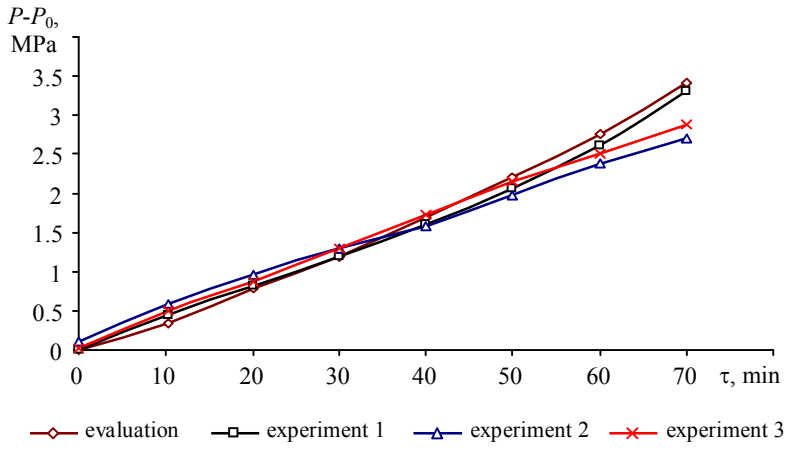


Fig.5. Dependence of methane pressure in CT on time with DlS

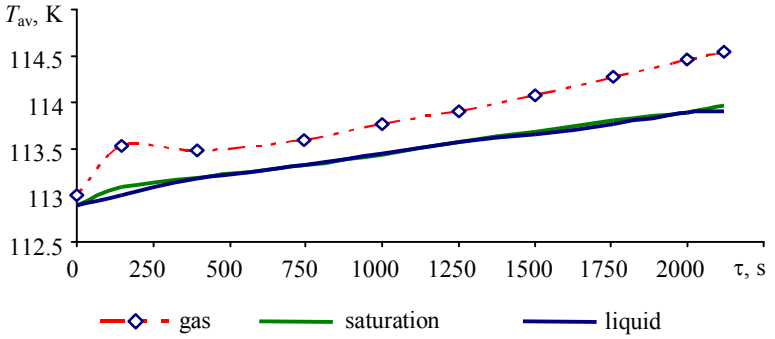


Fig.6. The dependence of the experimental results of average temperatures on the storage time of liquefied methane

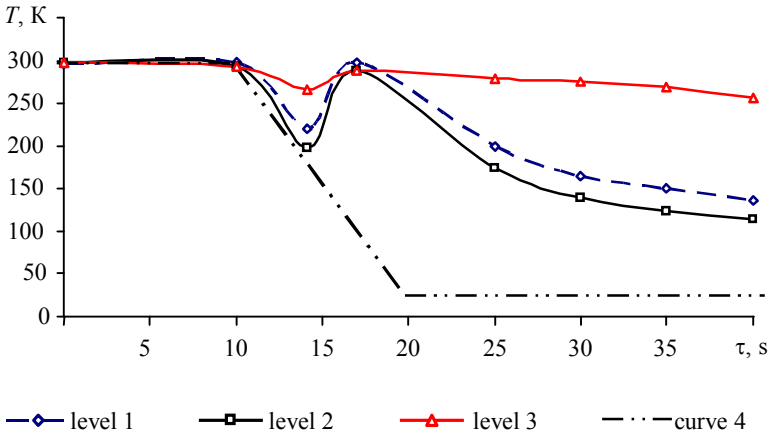


Fig.7. Dependence of experimental temperature results on time emanating from the axis of symmetry of the tank at the initial stage of the cooling process

The characteristic fields of velocity and temperature in liquid and gas are presented in Fig.5.

The rate of change of the temperature field with time in the liquid phase of methane located in CT is shown in Fig.6

Temporary results of temperatures obtained in the tank emanating from the axis of symmetry of the tank at the initial stage of the cooling process are shown in Fig.7. The curve lines corresponding to levels 1-3 refer to points located at distances below the tank axis of symmetry: –0.10, –0.2, and –0.3 m, curve 4 shows the change in temperature of the supplied methane through the inlet pipe.

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By analyzing the general structure of moving flows in the tank under steady-state conditions ($\tau > 600$ s), two zones can be distinguished in the studied region of the temperature field. The first is the boundary layer formed on the inner wall of the tank, the second is the core formed in the central part of the tank.

In the work, FlowVision software was used for mathematical modeling of the studied processes in the fuel system of mining gas engine equipment. Data processing was performed in the Microsoft Office Excel application package. The validity, reliability of scientific provisions, conclusions and recommendations are confirmed by a significant amount of laboratory tests, bench and operational tests, as well as the high convergence of the theoretical and experimental results.

Conclusion. The article discusses the heat and mass transfer process of the phase transition of liquefied methane by determining the total heat capacity of the CT in the heating - cooling mode. The measurements were carried out in the temperature range from 52 to 310 K in the mode of artificial heating – cooling. The measurements were carried out in a two-tank cryogenic tank for liquefied methane. Detailed research results are presented in published works [13, 21]. The results of the experimental data presented in Figs.4-7 confirm the efficiency of using liquefied natural gas charged into a special cryogenic tank designed for heavy vehicles operating in mountain and quarry conditions. As a result, an improvement was achieved in the following indicators: the mileage of gas engine technology increased by 3 times in comparison with operation with standard types of fuels; the shelf life of liquefied methane in CT in the drainless mode increases by 2-2.5 times.

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