The Non-Stationary Process of Heat-Mass Exchange of Liquefied Methane in a Cryogenic Fuel Tank of Automotive and Tractor Equipment

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Abstract. This science article presents the competitive advantages of liquefied natural gas as a motor fuel used in internal combustion engines. Block diagram of interrelation of the influence of construction of aggregates, mechanisms and systems with the purpose of increasing the energy-efficient work of a gas engine is presented. Special attention in article is devoted to the research of critical parameters of pressure and temperature of liquefied methane in the mixture, with associated gases that are in a small fraction, as well as the determination of the heat capacity in the free-cooling mode of the fuel supply system of the gas engine and the cryogenic tank in the filled state in a special thermal insulation. Determination of the critical pressure and temperature parameters is considered under the condition that methane flows in a special round tube to supply gas in a two-phase form to the engine for further conversion of thermal energy to mechanical.

1. Introduction

Natural gas as an energy carrier is 3-5 times cheaper than fuel obtained from oil, which makes it very competitive[1, 2, 3, 4, 5]. However, cryogenic fuel has the following drawbacks: sufficiently low density, ultra-low temperatures in the cryogenic state, high volatility, and with mismanaging - explosion hazard, which initially causes certain caution when posing the question of their use and doubt in the feasibility of widespread introduction of this type of energy carriers, on energy, economic and environmental benefits. This can largely explain the long-term inhibition of investment and the slow pace of work on the practical solution of problems and the introduction of LNG (Liquified Natural Gas) in industry and agriculture [6, 7].

When modeling the working process of gas internal combustion engines, one must rely on knowledge of the properties of air, fuel vapors, etc. so the authors By Gordon L. Dugger, Robert C. Weast and Sheldon Heimel [24] pointed out that, under the conditions of limiting temperature, the decrease in the rate of combustion kinetics, rather than mixing, can be a controlling 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 is the
main factor for carrying out research work in this field. The authors of John D. Maples, James S. Moore, et al. [25, 26] consider the equation of the state of working bodies mainly for traditional fuels (gasoline, diesel, etc.), as for alternative fuels: compressed natural gas (CNG - Compressed Natural Gas) and LPG (Liquified Petroleum Gas), there are no such studies in this source. The considered semi-theoretical method of prediction, based on three approximate theoretical equations of the flame velocity: equation, equation and equation.

In contrast to standard fuels (gasoline, diesel), methane to exclude the two-phase state must be supplied to the combustion chamber of the engine in gaseous form, using special gas-cylinder equipment (GCE) of the 4th, 5th or 6th generation with improved environmental performance.

At the same time methane by mass heat of combustion exceeds kerosene by 14%, which is important for optimization of combustion parameters of in-cylinder processes in piston engines.

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**Figure 1.** Block diagram of interrelation of the influence of construction of aggregates, mechanisms and systems with the purpose of increasing the energy-efficient work of a gas engine.

On the Figure 1 is a diagram of the interrelationships of aggregates and systems. Without optimization, it will be difficult to achieve an increase in the operational and energy-efficient characteristics of a piston engine. In this article, we will only consider a separate direction - the effect of the design of the power system on the processes of supply and combustion of the methane-air mixture in the combustion chamber [13, 16].

In Russia at the moment, there are two manufacturers of gas engines: KamAZ with the KamAZ-820 family of engines and URAL with the YaMZ-530 CNG family, and both plants used similar advanced engineering designs [17].

For realization of a technical task of improvement of operational indicators of the car completed with purely gas engine of the ignition working on a methane cycle with system from electric discharge it is necessary to consider more deeply process of supply of the liquefied methane on fuel system and also for increase in term of non-drainage storage of specially designed cryogenic tank (CT) to theoretically define key indicators of a non-stationary heat-mass exchange of interphase transition of methane from liquid state in gas.

We will consider process of filling with the liquefied methane in the cryogenic tank (CT) in more detail.
Further we will consider fragments of mathematical model of heat-mass exchange and theoretical determination of values of peak pressure in CB and the pipeline when giving a fresh stream of LNG and also time after replenishment of a tank in the new portion of fuel, in a basis of this mathematical model have put and A.A. Grebennikov, G.G. Yankov, S.S. Kutateladze, I.V. Barmina and also S. of Prakash, S. Patankar, W adapted techniques of the following authors. Lee, T. Rogers and P. Petersen.

Mathematical modeling of process of cooling of the fuel tank with cryogenic liquid (the liquefied methane) when filling, represents, both a three-dimensional, and two-dimensional task presented in the Cartesian system of coordinates taking into account a non-stationary heat-mass exchange of the expiration of a two-phase stream. For the solution of this regional task it is necessary to consider heatphysical properties of methane and its indicators at a turbulent flow, feature of a design of a cryogenic tank and special insulating materials which name and properties we won't present in article, for realization of an objective works of authors on given at the direction were studied [8, 9, 10, 11, 12].

Main investigation phases, following:
1. Determination of peak pressure of the liquefied methane and the present associated gases of so-called mix of gases of a methane row in the studied tank as the projected product;
2. Measurements of thermal capacity in the mode of free heating and cooling of fuel system with the liquefied methane.

In a basis of this modeling lies the equations reflecting interphase exchange in the studied gas chemically uniform mix respectively of energy, an impulse and weight. In this case the main equations in the Cartesian system of coordinates of separately taken cylindrical part (r, z) are presented in the following form:

\[
\frac{\partial \rho_g^0}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r} \left( \rho_g^0 rv_{g,r} \right) + \frac{\partial}{\partial z} \left( \rho_g^0 V_{g,z} \right) = 0, \tag{1}
\]

\[
C^* \frac{\partial T}{\partial r} + \frac{1}{r} \frac{\partial}{\partial r} \left[ r \left( \rho_g^0 c_{pg} V_{g,r} T - \lambda^* \frac{\partial T}{\partial r} \right) \right] + \frac{\partial}{\partial z} \left( \rho_g c_{pg} V_{g,z} - \lambda^* \frac{\partial T}{\partial z} \right) = 0. \tag{2}
\]

where: \( V_{g,r}, V_{g,z} \) – components of a vector average (according to Reynolds) speeds, \( C^* \) – volume thermal capacity; \( \lambda^* \) – the auxiliary value accepting values \( \lambda^* = \lambda_s \) in a firm wall \( (\epsilon = 0) \) and \( \lambda^* = \lambda^c + \frac{(\nu/Pr_l)\rho_g^0 c_{pg}}{\rho} \) in gas \( (\epsilon = 1) \); \( \rho \) – methane density; \( T \) – gas temperature; \( \tau \) – time; \( r \) – tank radius.

This equation is presented in a form that corresponds to the energy equation for methane with \( \epsilon = 1 \) and, accordingly, the heat conductivity equation for the wall of the investigated part \( \epsilon = 0 \). Provided that the velocity \( \vec{V}_g \) in the contact zone with the wall assumes a zero value, since we artificially introduce the boundary conditions \( \vec{V}_g = 0 \) for the calculated reference volume on the outer part of the container surface. This approach is well familiar to specialists in works [14], the convenience of this approach consists in stitching several indicators at the same time such as temperature field and thermal stream on boundary zones of contact "gas - a firm wall" the studied volume of capacity or a tube with the liquefied methane. Further, the mathematical models of heat-mass exchange processes in CT [5,6] mainly based on the equations of thermodynamic phenomena of irreversible processes in a methane mixture:

\[
\rho \frac{di}{dt} = \lambda \Delta^2 T + div \left[ i_i - i_i \rho D \left( \Delta m + k_T \frac{\Delta P}{\rho} \right) \right], \tag{3}
\]

where \( \rho \) – is the density of the gas mixture; \( i \) – is the enthalpy of the gas mixture; \( \lambda \) – is the thermal conductivity coefficient of the system under consideration; \( D \) – is the coefficient of mutual diffusion of the gas mixture; \( i_i \) – enthalpy of methane; \( i_e \) – is the enthalpy of ethane; \( i_i \) – enthalpy of the i-th component of the gas mixture; \( m_i \) – relative concentration of the i-th component by mass; \( k_T, k_p \) – thermal diffusion and barodiffusion ratio, \( p, T \) – pressure and temperature of the mixture, \( t \) – time.
The peak pressure in the gas space of the CT and the pipeline can be determined from the following expression:

\[ P_{pk} = P_G \frac{\Delta m_{CH_4} R_C H_4 T_g}{V_G}. \]  

(4)

where \( P_G \) – initial pressure in the gaseous medium of the CT and the pipeline, \( \text{Pa} \); \( \Delta m \) – mass fraction of methane converted to gaseous state, kg; \( R_{CH_4} \) – gas constant of methane; \( T_g \) – temperature of methane in gas medium, °C; \( V_g \) – volume of gaseous medium, m³.

The simulation of the process is also based on the determination of the mass flow of the drained gas in a system whose expression has the following form:

\[ \frac{d}{d\tau} \left[ \frac{V_G P_G}{R_{CH_4} T_g} \right] = m_i - m_{dr}. \]  

(5)

where: \( m_i \) – mass flow of evaporating liquid for 1 s, kg/s; \( m_{dr} \) – mass flow of the gas drained in 1 s, kg/s.

Therefore, it is necessary to find the mass flow of methane drained in the system, which will be determined from the expression:

\[ m_{dr} = \frac{\pi d_{tr}^2}{4} \sqrt{\frac{p_{TR}^2 - p_i^2}{\left( \frac{\lambda l_{tr}}{d_{tr}} + \sum \xi_{i} \right) R_{TR}}} . \]  

(6)

where: \( T_{TR} \) – tube wall temperature, K; \( \lambda \) – the Darcy coefficient; \( \xi_{i} \) – coefficients of local resistance in the fuel supply system; \( P \) – pressure in the ejection process, Pa; \( d_{tr} \) – tube diameter, m; \( l_{tr} \) – tube length, m.

To determine the peak pressure, the system needs to find the amount of heat supplied to the volumes in question (CT and the tube), therefore, the change in the amount of heat in the liquid phase of methane as the temperature varies with time will look like this:

\[ Q = m_{zm} \cdot c_{rz} \cdot T_{ZM}, \]  

(7)

where \( m_{zm} \) – mass of liquid methane, kg; \( c_{rz} \) – heat capacity of liquid methane, J / (kg · K).

Based on the analysis of the results of modeling the supply of fresh volume of liquefied methane [17] to the investigated areas (CT and pipelines), it is established that the equation for the change in the maximum pressure in the case of its occurrence will take the form of a linear function, and will practically not depend on the volume of the CT itself, to estimate the peak pressure in the tank under consideration:

\[ P_{pk} = k_{pl} \cdot k_{g} \cdot k_{kon} \cdot P_{pk_{-}t}, \]  

(8)

where: \( P_{pk} \) – peak pressure in CT; \( k_{pl} \) – he coefficient of the depth of the filling with methane; \( k_{g} \) – similarity coefficient of the geometric parameters of CT; \( k_{kon} \) – coefficient of initial concentrations; \( P_{pk_{-}t} \) – accepted peak pressure in KB for the standard.

Peak pressure based on the calculation results of the considered CT is \( P_{pk} = 2.96 \text{ MPa} \), and the corresponding coefficients influencing the changes in the maximum pressure can be determined from the following expressions:

\[ k_{rad} = 0.4595 \frac{R_{pr}}{R_{r} + 0.5405} , \]  

(9)

where \( R_{r}, R_{pr} \) – radius of the tubes projected and taken as the standard.

\[ k_{kon} = \frac{\Delta CH_{4_{pr}}}{\Delta CH_{4_{r}}}, \]  

(10)
The difference in the concentration of methane layers at the initial stages in the design CT for LNG in comparison with the reference one (theoretically chosen conditions to which the projected one should correspond):

$$k_{vt} = 1,3248 \frac{h_{pr}}{h_T} - 0,3248.$$  \hspace{1cm} (11)

where \( h_{pr}, h_T \) – filling depth of the liquefied gas at the bottom of the CT (projected and standard).

The range of changes in the determining parameters is presented in the expressions for the numerically found coefficients:

$$1 \leq \frac{\delta \lambda_{pr}}{\delta \lambda_{pr}^1} \leq 10; \frac{\lambda_{pr}}{\lambda_{pr}^1} \leq 1,333,$$  \hspace{1cm} (12)

$$0,5 \leq \frac{\Delta CH_{pr}}{\Delta CH_{pr}^1} \leq 1,5; \frac{h_{pr}}{h_T} \leq 1,105.$$  \hspace{1cm} (13)

Permissible limits of errors for an effective evaluation of the peak pressure in the test KB should not exceed 10% according to the presented method.

During projecting CT, this condition will reduce the possibility of accidents, even in the initial stages, which will allow applying a number of developed measures for the safety of the fuel supply system in operation (applicable safety measures are presented in [2, 3]).

Having determined the values of the peak pressure \( (P_{pk}) \) in the pipeline, it is expedient to determine the critical values for the temperature \( (T_{kr}) \) of the natural gas mixture (95% methane, the remaining 5% ethane, propane, butane and other associated gases), as well as analysis of the change \( (P, T) \) in the flow of the two-gas mixture [17].

The gas mixture of the methane series is divided into three components: methane; ethane, remaining n-components (propane, butane, hexane, etc.). The calculation is conducted from a condition that methane moves in the liquefied look when filling a CT at automobile gas-accumulating compressor station (AGACS).

The temperature at which any further increase in pressure can not affect the further liquefaction of the gas is called the critical temperature. Respectively, this pressure at which natural gas is liquefied is called critical pressure. Thus, it is necessary to correctly determine, in the first place, the critical temperature of a given methane gas mixture.

The critical temperature of the mixtures in the first approximation can be calculated from the formula presented below:

$$T_{kr,m} = k_1 x_1 T_{krit 1} + k_2 x_2 T_{krit 2-n}.$$  \hspace{1cm} (14)

where \( x_1, x_2 \) – ass fractions of methane, ethane and the remaining n-components (propane, butane, hexane, etc.); \( T_{krit 1}, T_{krit 2-n} \) – critical temperature of methane, ethane and the remaining n-mixtures, respectively; \( k_1, k_2 \) – correction factors, which are assumed to be 1.0 and 0.95.

Next, a detailed calculation of the entire fuel supply system from the filling of CT and the supply of liquefied methane in a two-phase flow, taking into account the present associated gases (propane, ethane, etc.) which is not given in this article is calculated in the work [6].

To start modeling the heat capacity of the thermal insulation layer, the following scientific papers were used [4-12 and 22-28].

The peculiarity of the mathematical description lies in the adaptation of the most suitable model that determines the heat capacity in the boundary conditions of the product under consideration (a two-tank cryogenic fuel tank, which is not presented in the work because patent application was filed), which in the relative approximation and further transformation of the set of equations makes it possible to use the method of setting dimensionless numbers.

The heat flux \( Q \) that flows to the CT surface from the side of the artificial heat source simulating the solar heat flow (we will call the heater) is related to the difference in the mean volume
temperatures of the thermal insulation shell of the first and second tanks \( \sum T_{ibak} \tau_1 + \sum T_{2bak} \tau_2 \), and the artificial heater \( T_{nagr} \).

\[
Q \tau = k_T \left[ T_{ibak} \tau_1 + T_{2bak} \tau_2 - T_{nagr} \right],
\]

where \( k_T \) – effective thermal conductivity of the double insulating layer of the CT wall.

Let us write equation (15) taking into account the redistribution of the heat flux between the wall surfaces (external and internal) of two tanks of a solid fuel tank and insulating layers inside and outside, respectively.

This heat flux, taking into account the dimensionless number (Dimensionless numerus in the future we will use the notation - Dn, the calculation of the dimensionless number in the article is not given) will have the following form:

\[
Q \tau = \left( C_1 T_{ibak} \right) \frac{dT_{ibak}}{d\tau_1} + C_{1T} T_{1T} \frac{dT_{1T}}{d\tau_1} +
\]

\[
\left( C_2 T_{2bak} \right) \frac{dT_{2bak}}{d\tau_2} + C_{2T} T_{2T} \frac{dT_{2T}}{d\tau_2} + Dn
\]

(16)

where \( C_{iH} \) – total heat capacity of the thermal insulation layer of the i-th layer of the corresponding reservoir (1 - external reservoir, 2 - internal reservoir); \( T_{iH} \) – average volume temperature of the i-th thermal insulation layer of the corresponding reservoir.

Taking into account the peculiarity of CT construction, as well as the analysis of temperature fields using the above equations, it is possible to present the following expression for determining the heat capacity of a general isolated shell of a CB in the following form:

\[
\sum C \ T_{kb} = k_T \left[ T_{kb} \sum T_{kb} \right] - 0.468 \sum C_{i} \sum T_{i}.
\]

(17)

2. Conclusion

According to the results of the study, it can be noted that methane in the liquefied state has a rather complex physical state in the thermodynamic phase transition from low to high temperatures and pressures. The mathematical models presented above make it possible to determine the main indicators of individual components of the gas engine feed system, in order to optimize the supply of liquefied natural gas to improve fuel-economic and environmental performance without reducing power, the main results obtained experimentally in the article are not shown because they are presented in detail in works [16, 17]. The positive side of the issue under consideration is that there are all grounds for realizing the set goal with the proper selection of materials for further experimental development of experimental samples of components of the gas engine feed system, such as: a cryogenic tank or pipes [11].

3. References

[1] Afanasiev A S 2014 Influence of modes of use of a diesel engine on smoke of the fulfilled gases Technical and technological problems of service 2 (28) 56-58

[2] Didmanidze O N 2014 Power plants of a new generation of cars Drive Technology 4 36-53

[3] Dolganov K E 1995 Power supply system and regulation for the conversion of diesel engines in gas diesel engines Engine building 2 6-10

[4] Zhou D 2006 Extended irreversible thermodynamics Institute for Computer Research 528

[5] Zaichenko V M 2014 Pyrolysis on carbon matrices Nedra Publishing House 235

[6] Isserlin A S 1980 Basics of Combustion of Gas Fuel: A Reference Guide L .: Nedra 263

[7] Kochin Y Yu 1986 Technique and experiment planning L .: LPI 70
[8] Loitsyansky L G 2003 Mechanics of fluid and gas: Proc. for universities. 7 th ed. M .: Drofa, 840
[9] Maikov I L 2011 Math modeling Hydrodynamics Chemical kinetics LAPLAMBERT Academic Publishing GmbH & Co. KG KG 215
[10] 2015 Mathematical modeling of hydrodynamics and heat transfer in moving liquids: Monograph / I V Kudinov, V A Kudinov, A V Eremin, S V Kolosenkov; under. Ed. EM. Kartashov. SPb: Lan 208
[11] Mikheev V P 1966 Gas fuel and its combustion L .: Nedra
[12] Nikolaienko A V 2006 Mathematical modeling and calculation of the working process of the gas modification of a diesel engine Improvement of operational performance of engines, tractors and cars: Sat. sci. tr. Intern. scientific-techn. Conf. SPb: SPbGAU 363-387
[13] Patankar S 1984 Numerical methods for solving problems of heat transfer and fluid dynamics: Trans. with English Moscow: Energoatomizdat
[14] Rusinov R V 1998 Engines of cars and tractors Design and calculation of engine systems SPb: SPbSTU 120
[15] Khakimov P T 2008 Influence of Burnout Characteristics on Gas Engine Operating Cycle Performance Using Electronic Control System Truck 4 27-29
[16] Khakimov R T 2010 Analysis of experimental studies of the working processes of convertible gas engines Proceedings of the St. Petersburg State Agrarian University 21 297-302
[17] Tsoy P V 2005 System methods for calculating boundary-value heat and mass transfer problems Moscow: Izd-vo MPEI 568
[18] Shashkov A G 2004 Wave phenomena of thermal conductivity: a system-structural approach M .: Editorial URSS
[19] Liss W E 1991 Natural Gas as a Stationary Engine and Vehicular Fuel URL:https://doi.org/104271/912364 (date of access 5.05.2017)
[20] Khakimov R, Shirokov S, Zykin A, Vetrova E 2017 Strategic assessment aspect of vehicles' technical condition influence upon the ecosystem in regions Transportation Research Procedia 12th International Conference «Organization and Traffic Safety Management in Large Cities» SPbOTSIC-2016 295-300
[21] 2014 Study on Performance and Exhaust Gas Characteristics of Directly Injected CNG Engine International Journal of Bio-Science and Bio-Technology vol 6 1 179-186
[22] Weaver C S 2017 Dual Fuel Natural Gas/Diesel Engines: Technology, Performance, and Emissions URL:https://doi.org/104271/940548 (date of access 5.05.2017)
[23] Dugger G L, Weast R C, Heimel S 1955 Effect of preflame reaction on flame velocity of propane-air mixtures Symposium (International) on Combustion 5 1 589-595
[24] Maples J D, Moore J S, Patterson Jr, P D, Schaper V D 2000 Alternative Fuels for U.S. Transportation Degobert P 1955 Automobiles and Pollution, SAE Publications, Warrendale, PA
[25] Pesterev A P, Vasilyeva A I, Ammosova M N, Solovev D B 2018 Unexplored soils of the Western Yakutia IOP Conference Series: Materials Science and Engineering 463 Part 1 Paper № 022001[Online]. Available: https://doi.org/10.1088/1757-899X/463/2/022001
[26] Prakash C, Patankar S V 1985 A control volume-based finite-element method for solving the Navie-Stokes equations using equal-order velocity-pressure interpolation // Numerical Heat Transfer vol 8 259