International Journal for Quality Research 16(3) 803–816
ISSN 1800-6450

Andrey Balakin
Alexey Roslyakov
Yury Klochkov[1]

DIESEL ENGINE QUALITY AND RELIABILITY GROWTH BY SUPPLYING A HOMOGENEOUS MIXTURE OF THE AIR AND ETHANOL THROUGH THE INLET HEADER

Article info:
Received 22.12.2021.
Accepted 23.05.2022.

UDC – 665.75
DOI – 10.24874/IJQR16.03-10

Abstract: *The aim of this research is the study of negative physical processes in diesel engines, which are induced by the low value of cetane number of ethanol that characterizes the efficiency of the ignition process. The authors present the calculation assessment of thermodynamic state of methanol and diesel fuel mix combustion products in various proportions and calculations of the thermal state of injector elements by analytical and computational methods. Analytical dependencies are taken from traditional branches of science such as heat engine theory, heat transmission. As a technical solution to compensate the low value of the cetane number the authors use diesel fuel as an ignition dose and ethanol as the main fuel. The paper shows that having other conditions equal the power of a diesel engine slightly increases due to raise of water vapor in combustion products. The authors present the methods and calculation of fuel heating in the boundary layer of the injector nozzle channel during the injection cycle of the D240 diesel engine. With the decrease of the fuel priming in the boundary layer the temperature rises, and that is the reason of the gumming which leads to the needle jamming.*

Keywords: *Diesel fuel; Ethanol; Ignition dose of diesel*

1. Introduction

Over the last years energy issues have been significantly influenced all over the world. At the same time alcohol fuels which have an almost sustainable raw material resources base for their production are recognized as a substitute for traditional fuels of petroleum origin. Ethanol is the most competitive fuel among alcohol fuels. This is mostly because ethanol is two hundred times less toxic than methanol.

When converting diesel engines to ethanol, it becomes necessary to perform a number of research projects. It is necessary to ensure reliable, appropriate and stable ignition of fuel and reliable operation of pumps and injectors by modernizing the two-fuel supply system and optimizing its operation.

On the one hand when ethanol is used as an oxygenated fuel, problems arise due to poor miscibility with petroleum and many alternative fuels. These problems can be solved by using anhydrous ethanol. However, it takes energy inputs and costs during anhydrous ethanol production. There are various requirements to the ethanol composition in different countries. For

[1] Corresponding author: Andrey Balakin
Email: y.kloch@gmail.com

example, in Canada the water ratio in ethanol is admissible until 0,1%, in Poland it is until 0,4 %, and in the USA until 7,9 %. The water ratio in diesel fuel is not permissible.

Nowadays high pressure fuel pumps with plunger pairs and injectors with precision pairs are used in fuel supply systems of diesel generators of locomotives. Injector needle is installed in the body with a gap of 1,5...2 microns. Diesel fuel is a lubricant that reduces the friction of precision pairs. For example, water contamination into the fuel due to water vapor condensation from the atmosphere or when using water-fuel emulsions as fuel leads to jamming of precision pairs and failure of pumps and injectors.

On the other hand, a water vapor increase in the combustion products leads to an increase of the diesel engine cycle at the expansion stroke. Thermodynamic analysis of the heat engine cycle shows that the work during the extension of combustion products in the cylinder is proportional to the gas constant of the fluid. The physical meaning of the gas constant is that when it cools down by one degree, a kilogram of the fluid performs an operation that is numerically equal to the value of the gas constant. Consequently, with an increase in the water vapor proportion with a gas constant 55% more than the average for the combustion products mixture, the extension increases in accordance with an increase in the proportion of water vapor in the combustion products.

The article considers a structural version when ethanol is fed into the engine through the inlet header and then with air into cylinders in order to exclude water contamination into precision pairs.

Only diesel fuel is used at starting the engine and at idle rounds. With an increase of the mode, a small consumption of at least 15% of diesel fuel is supplied as an ignition dose, and ethanol is supplied as the main fuel (Kavtaradze**,** 2008; Kavtaradze, et al., 2009; Gaivoronskiy et al., 2007; Bromberg et al., 2006).

2. Methods and materials

The calculation of a three-dimensional model of the flow section of the inlet header of a PD1M diesel engine installed on a TEM 18 locomotive was carried out by the FLUENT computational gas dynamics complex in order to find the velocity fields and ethanol concentration in the air entering the cylinders. The paper shows the results of calculating the parameters when the engine is running at nominal rating.

The calculation was carried out by the finite element method with a division of the calculated volume into 1.16 million elements.

When designing the computational domain and setting the boundary conditions for the designed compressor station, the following assumptions were made:

- fluid – incompressible ideal gas;
- the boundary layer on the walls of the model fits into the dimensions of the wall elements;
- a uniform field of velocities and pressures is set at the inlet to the collector;
- calculations are carried out using the basic implicit solver PressureBasedImplicit (PBI) and in the second-order of calculation accuracy.
- k-εRealizable turbulence model is applied.

The research of temperature fields in the injector are carried out by the analytical method. (Stechkin, 2001; Roslyakov, 2004; Lykov, 1978), which were numerously verified during the creation and further development of heat engines by comparing the calculation results and the results of experimental tests.

The reference literature lists the physical properties of diesel fuel and ethanol, for example (Abbasov, 2020; Fuel Technologies, 2007; Markov et al., 2015; IHS Global Methanol Market Overview, 2011; China Issues Methanol Vehicle Promotion Policy, 2019; Wuebben, 2016; Gaivoronskiy et al., 2007; Yuen et al, 2010; Westet al., 2007; Ramadan et al, 2003; Nichols, 2003; Cohn et al., 2005; Devyanin et al., 2014; Bromberg et al., 2006; Sendilvelan et al., 2017; Klein, 2020; Bilgn et al., 2002).

A propellant mixture is used due to the fact that the low cetane number and underperformance of the ignition process are among the significant disadvantages of alcohol fuel, and the cetane number is known to be a measure of the flammability of the fuel when adapting diesel engines to alternative fuels. Diesel fuel with a high cetane number is used as the ignition dose and alcohol is used as the main fuel.

3. Ethanol in diesel fuel influence on adiabatic expansion work and thermal properties of combustion products

With the known values of the main indicators of the initial fuels, including diesel fuel and ethanol, the values of the mixture indicators are defined by the sum of the products of the component fraction by its indicator.

An exception to this approach is the calculation of the kinematic viscosity of the mixture. To calculate the kinematic viscosity of fuel mixtures, it is recommended to use the following equation (Markov et al., 2015; Gaivoronskiy et al., 2007; Cooney et al., 1981):

$$lg\nu_{\text{см}} = r_1 lg\nu_1 + r_2 lg\nu_2,$$

where ν1 and ν2 – kinematic viscosity of 1 and 2 components;
r1 and r2 – mole fraction of components.

When fuel is burned, combustion products are produced in diesel cylinders. Theoretically required amount of combustion air for 1 kg of diesel fuel is (Markov et al., 2015)

$$l_0 = \frac{1}{0{,}23}\left(\frac{8}{3}\,C + 8\,H - O\right) = \frac{1}{0{,}23}\left(\frac{8}{3}\,0{,}87 + 8\cdot 0{,}126 - 0{,}004\right) = 14{,}45 \text{ kg of air/kg of fuel}$$

Table 1 shows the main physical properties of a propellant mixture consisting of diesel fuel and ethanol.

Table 1. The main physical properties of a propellant mixture consisting of diesel fuel and ethanol

Coefficient		Fuel						
		Diesel fuel (DF)	Ethanol (E)	96 % DF + 4 % E	80 % DF + 20 % E	50 % DF + 50 % E	30 % DF + 70 % E	15 % DF + 85 % E
Density by 20 °C, kg/m^3		830,0	789,3	828,4	821,86	809,65	801,51	795,41
Kinematic viscosity by 20 °C, mm^2/s		3,8	1,0	3,5	3,24	2,4	1,84	1,42
Lower heating value, kJ/kg		42500	26800	41800	39360	34650	31510	29155
The amount of air required for combustion of 1 kg fuel, kg		14,31	9,01	14,09	13,25	11,66	10,6	9,805
Mass fraction, %:	C	87,0	52,2	85,6	80,04	69,6	62,64	57,42
	H	12,6	13,1	12,6	12,7	12,85	12,95	13,025
	O	0,4	34,7	1,8	7,26	17,55	24,41	29,555
	S	0,200	—	0,192	0,16	0,1	0,06	0,03

By the complete combustion of one kilo of diesel fuel with an excess air factor α = 1 we get 3,19 kg of CO_2, 1,134 kg H_2O and 11,09 kg of N_2. It is obvious that N_2 has the largest share in combustion products, where the gas constant R = 296,9 kJ/kg, followed by CO_2 with the gas constant R = 189 kJ/kg. H_2O with a gas constant R = 461.9 kJ/kg in a mixture of combustion products has the smallest proportion.

The combustion of ethanol proceeds in accordance with the equation:

$$C_2H_5OH + 3O_2 = 2CO_2 + 3H_2O$$

During the combustion of 1 molecule of ethanol, 3 molecules of oxygen are consumed as an oxidizing agent and 2 molecules of carbon dioxide and 3 molecules of water vapor are appeared at the output. Equal molecular weights are maintained 50 + 96 = 92 + 54 = 146.

Let us consider the option when we have 15% diesel fuel and 85% ethanol in the propellant mixture, which means for 1 kg of propellant mixture we have 850 grams of ethanol. Ethanol has a molecular weight of 46 grams per mole. Consequently, we have 18,48 moles of ethanol in 850 grams. According to the equation (1), water vapor is released 3 times more during combustion, so we have water vapor of 55,4 moles or taking into account the molecular weight we have 0,998 kg. Besides, we have to bear in mind that the calorific value of ethanol is less than that of diesel fuel and it has to be burn more. It is also necessary to take into account that the amount of air for combustion of 1 kg of fuel for ethanol is 9 kg instead of 14,7 kg for diesel fuel. To get the same excess air ratio, it is necessary to supply more propellant mixture. Besides, ethanol in the state of delivery contains water impurities, the amount of which is allowed in accordance with the technical requirements for the production and supply of ethanol. As it was shown above, in the USA this amount is up to 7,9%.

The adiabatic gas expansion Lad, which is also called the available heat drop H, can be determined from a joint consideration of the generalized Bernoulli equation for the adiabatic process of gas expansion and the corresponding energy conservation equation (Kavtaradze et al., 2009). The adiabatic gas expansion Lad or the available heat drop H is defined according to the equation:

$$L_{ad}^{*} = H^{*} = \frac{k_{cm}}{k_{cm}-1} R_{\mathrm{cm}} T_0^{*} \left(1 - \frac{1}{\pi^{\frac{k_{cm}-1}{k_{cm}}}}\right)$$

where$k_{\mathrm{cm}}R_{\mathrm{cm}}$ - adiabatic coefficients and gas constant of a mixture in an adiabatic expansion process that proceeds without losses.

The adiabatic coefficient of the fluid depends on the number of atoms in the gas molecule, so for monatomic gases according to the laws of statistical physics it is equal to - 1,666, for two atomic gases - 1,407 and for three atomic gases - 1,286 (Stechkin, 2001).

The adiabatic coefficient k = 1,4. is applied for the air, which consists mainly of two atomic nitrogen N_2 and O_2. In exhaust gases with excess air ratios greater than unity,residual oxygen, nitrogen and combustion products are present: carbon dioxide (CO_2) with k = 1,305 and water vapor H_2O also with k = 1,305.

A number of factors affecting the expansion process, including heat exchange with the cylinder wall, were not considered from a theoretical point of view to simplify the task and highlight the main goal, which means evaluating the influence of an increase in the proportion of water vapor in combustion products on the expansion work.

During the combustion of hydrocarbon fuel, the adiabatic coefficient of combustion products is k = 1,33 and the value of the gas constant is R = 287 J/(kg*K). This value of the gas constant corresponds to a molecular weight of 30 kg/kmol.

According tothe analysis of dependence (2) with an increase in the proportion of water vapor, the adiabatic coefficient of combustion products can change in the limit from k = 1,33 to k = 1,3 and as a result of this change the engine power will increase by 2%.

Table 2 and Fig.1 show the share of the components of hydrocarbon fuel combustion products depending on the excess air ratio.

Table 2. The share of the components of hydrocarbon fuel combustion products depending on the excess air ratio

Mode	CO_2	H_2O	N_2	O_2
α = 1	0,098	0,185	0,717	0
α = 2	0,051	0,113	0,739	0,098

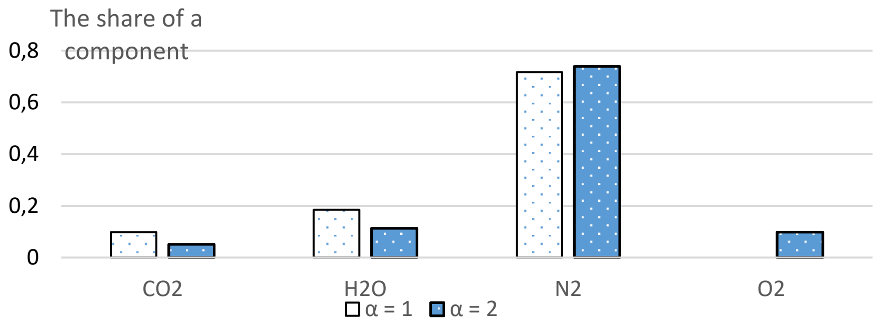


Figure 1. The share of the components of hydrocarbon fuel combustion products depending on the excess air ratio

The table and figure show that with an increase of the excess air ratio, the proportion of triatomic components (CO_2 and H_2O) increases, while the proportion of diatomic components (N_2 and O_2) decreases.

Table 3 and Figure 2 show the data of the influence of the water vapor proportion increase in combustion products on the gas constant.

Table 3. The influence of the water vapor proportion increase in combustion products on the gas constant

Indicator	The increase of the proportion of water vapor in combustion products						
	Without H_2O	+ 2 % H_2O	+ 4 % H_2O	+ 6 % H_2O	+ 8 % H_2O	+ 10 % H_2O	+ 12 % H_2O
R, kJ/kg	297	300,3	303,6	306,9	310,2	313,5	316,8

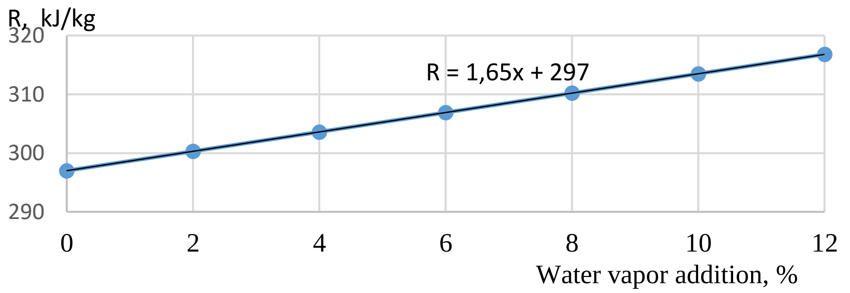


Figure 2. The influence of the water vapor proportion increase in combustion products on the gas constant

According to Table 3 and Figure 2 and the regression equation R=1,65*x+297, we can see that an increase in the proportion of water vapor in combustion products by 12% leads to an increase in the gas constant of combustion products from 297 kJ/kg to 316,8 kJ/kg, which is the increase by 6.7%. The expansion of the fluid will increase by the same amount, with the raise of the water vapor proportion.

4. Ethanol supply system through inlet header

In majority of cases ethanol introduction is performed by changing the existing designs of diesel engines. Currently, many diesel engines have reached their assigned resource and need general maintenance and enhancement. The simplest and most cost-effective enhancement method is to introduce ethanol feed directly into the inlet header. Figure 3 shows a model of such an inlet header with an ethanol supply pipe and a mixture supply pipe nipple to the 6^{th} cylinder of the PD1M engine. The model is made in the NX graphic editor.

The mixture parameters were calculated in a quasi-static version using the FLUENT computational fluid dynamics system. Besides, the following initial data were set: excess air coefficient $\alpha = 2$, air consumption $G_a = 0{,}267$ kg/s, ethanol consumption $G_e = 0{,}0148$kg/s. The values of other parameters are given in section 1.

Figure 4 shows the fields of ethanol concentrations when fed into the first (a) and sixth (b) cylinders.

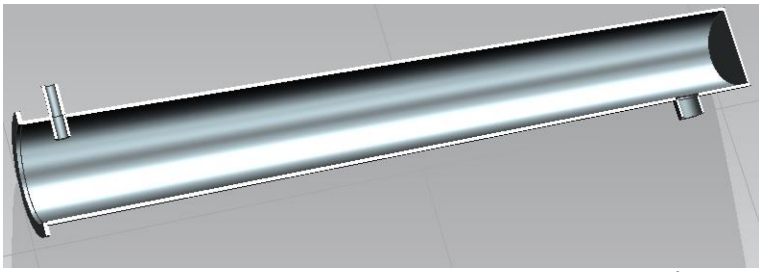
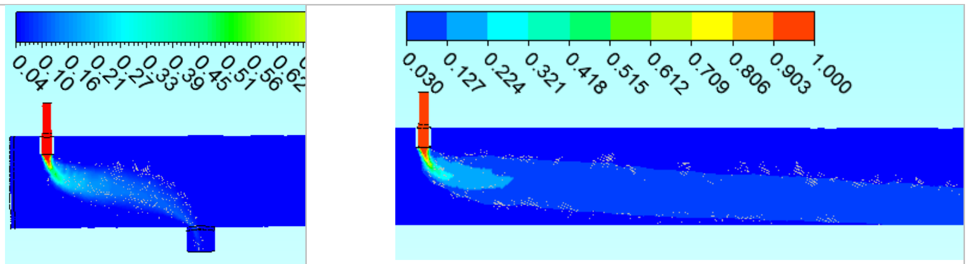


Figure 3. Inlet header with a mixture supply pipe nipple to the 6^{th} cylinder



a	b

Figure 4. Fields of ethanol concentrations when fed into the first (a) and sixth (b) cylinders

Figure 5 shows the field of ethanol concentrations in the supply pipe nipple at the first cylinder.

The difference of ethanol concentration along the channel section reaches 6…7 %.

To reduce the difference in ethanol concentration at the inlet to the cylinders a mixing device was engineered, the diagram of which is shown in Figure 6.

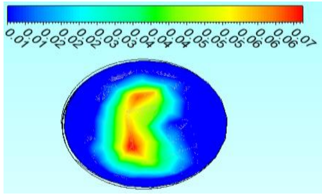


Figure 5. Fields of ethanol concentrations when fed into the first (a) and sixth (b) cylinder

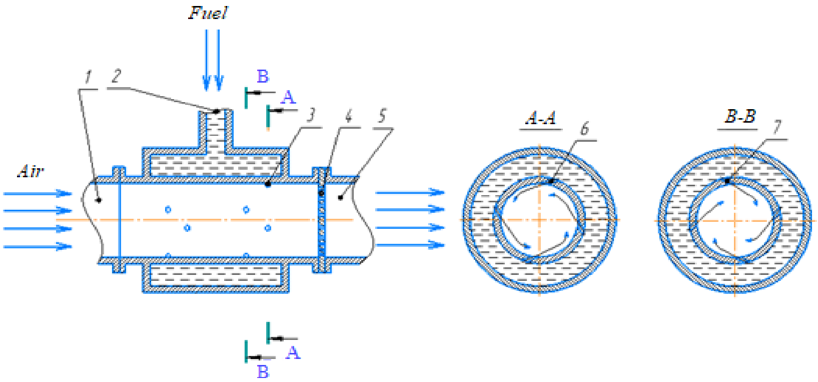


Figure 6. Mixing device diagram

The air enters the mixing device through a pipeline. 1. Fuel enters the annulus through the choke 2. Out of the annulus the fuel goes into the mixing chamber through the holes 3. Besides, there is an even number of rows of inlet holes 3. Moreover, on one of the rows the holes 6 are made at an angle of 15…45 degrees from the tangential direction to the internal diameter of the pipeline, for example, to the right, and holes 7 in the next row after the first row are made at an angle of 15…45 degrees in the other direction. A flow rotating in opposite directions is being created in the mixing cavity. Having mixed the fuel with air, the mixture passes through the drilled diffuser grid 4 into the pipeline 5 and further into the inlet header. Such a flow with increased turbulence forms the process of mixing and getting a homogeneous mixture with a uniform concentration of fuel in the air. Then the fuel mixture is fed into the cylinders of diesel engine through the air supply system. The device mixing ethanol with air before feeding it into the engine ensures a uniform concentration of ethanol in the air mixture and as a result a uniform heat load on the diesel engine cylinders.

Figure 7 shows a model of a device for mixing ethanol with air assembled with an inlet header and a mixture supply pipe nipple to cylinder 1 of the PD1M engine.

Figure 8 shows the field of ethanol concentrations in the inlet header and in the mixture supply pipe nipple to the first cylinder.

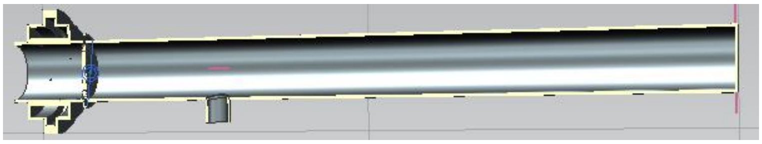
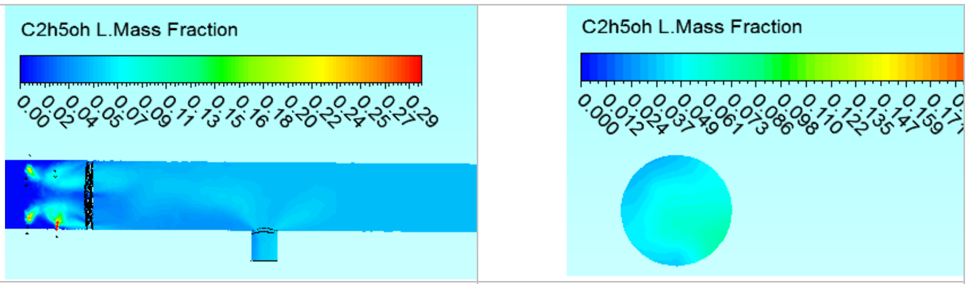


Figure 7. A model of a device for mixing ethanol with air assembled with an inlet header.



a b

Figure 8. Ethanol concentrations field in the inlet header a) and in pipe nipple b) mixture supply to the first cylinder

The difference in ethanol concentration over the section of the inlet channel decreased to 1…1,5 %. The mixing device allows us to get a homogeneous mixture of fuel with air and to increase the reliability of the diesel engine as a whole.

5. Warming up diesel fuel in the d-240 diesel nozzle while operating on diesel fuel and ethanol

One of the main problems of adapting a diesel engine to a dual-fuel one is the discrepancy between the standard fuel equipment and the new conditions (Kavtaradze**,** 2008; Kavtaradze et al., 2009; Gaivoronskiy et al., 2007; Bromberg et al., 2006):

- high irregularity of cycle feeds when operating at minimum stable idle speed;
- for mass-produced fuel equipment, operating in modes from idle and up to a power of 30-40%, the instability of the fuel supply processes from cycle to cycle (especially at a reduced speed of rotation) is typical, which leads to subharmonic vibrations of engine torque and raised irregularity of speed rotation, this circumstance does not allow reducing the minimum ignition supply;
- nonconformity of the standard mass-produced fuel supply system with two extremely opposite requirements: ensuring the parameters of typical operating modes and modes of light feeds and rotating speeds;
- it is difficult to ensure the operation efficiency of the injection nozzle due to the extensive completion phase of fuel supply at a reduced injection pressure coking of injector nozzles take place, and the needle may hang up.

The minimum amount of ignition diesel fuel is defined by the energy required for ignition and complete combustion of the mixture. Depending on the perfection degree of the gas-diesel system the consumption of diesel fuel is in average 15 %...30 %. With a decrease of the ignition dose of diesel fuel less than 15% the service life of the injector nozzles shortens. During engine operation at advanced modes with the ignition dose of diesel fuel and with the parallel supply of ethanol to cylinders the temperature of the injector nozzles rises up. The temperature increases in the nozzles outlet and on the needle surface and as a result the intensity of the carbon-forming processes rises up as well. These processes proceed according to the mechanism of liquid-phase oxidation of hydrocarbon molecules at a diesel fuel temperature of 130 °C and above and with the dissolved oxygen in the fuel (Yanovsky et al., 2002).

Currently, there is a tendency towards a decrease in the number of pilot and model experiments in the total volume of scientific and engineering works due to a heavy increase of the cost of energy resources. One of the ways to maintain the research and technology development in such conditions is to increase the volume of computational and experimental research.

This paper mainly covers the question of the influence of the diesel fuel ignition dose value on the heating depth of the D-240 diesel engine nozzle walls when operating in dual fuel modes. The processes of liquid-phase oxidation of hydrocarbon molecules largely depend on the temperature in the boundary layer.

The analysis is based on the following facts: the standard version of the D-240 diesel fuel system works for a safe operation resource without significant remarks when using only diesel fuel due to the long-term refinement for reliability and resource.

The thermal state of the injector elements including the injector and the needle is influenced by two competing processes: heat from the combustion zone rises the injector temperature and the heat that is transferred to the water cooling system of the lid and convective heat transfer to the diesel fuel stream decreases the temperature, so some balance appears, that forms a balanced thermal state with a temperature of less than 130 °C. With the mode rise the crankshaft rotation speed increases and as a consequence the frequency of the thermal effect on the injector increases on the one hand and the diesel fuel consumption on the other hand. There is a balance in the thermal state of the injector. When switching to a dual fuel cycle with an increase of the operating mode, the consumption of the ignition dose of diesel fuel remains unchanged. In this case the balance between the heating and cooling processes is disturbed and as a consequence the temperature of the injector increases.

So, when the mode is increased from 600 rpm to 2200 rpm operating only on diesel fuel, the so-called “heat load” which depends on the ratio of thermal effects frequency or flash frequency rate to the consumption of diesel fuel per time unit increases by 27%. When switching to a dual fuel cycle with the operating mode increase from 600 rpm to 2200 rpm, having the flow rate of diesel fuel ignition dose remained unchanged, the nominal heat load increases by 154%.

A pintle-type injector FD22 is used in the D240 engine. A spray nozzle is installed on the lower end of its body using a special nut. A spray needle is pressed against the taper seat of spray nozzle by a spring that transmits load to a beam. Fig. 9 shows a typical layout of a diesel injector nozzle, which has a body 1 and a needle 2.

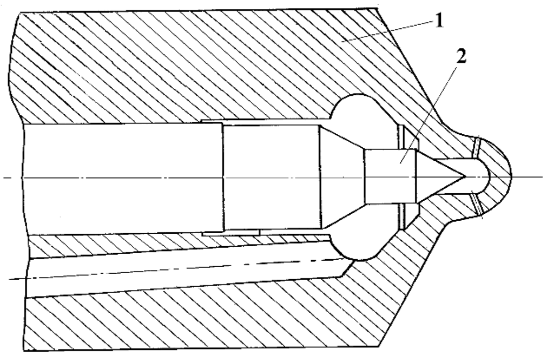


Figure 9. A typical layout of a diesel injector spray nozzle
1-body, 2-needle

The authors proposed the methods and calculated the fuel heating depth in the boundary layer of the injector spray nozzle channel for the time period between diesel fuel injections. The physical conditions realized in the process of fuel heating in the sprayers of D-240 diesel engine also take place in the sprayers of other diesel engines.

One of the main factors affecting the heating of the fuel in the cross section of the channel under dynamic conditions is the heating time. We have to mention that there is one fuel injection cycle per 2 revolutions of the diesel crankshaft.

In calculations we used the thermal diffusivity of the fuel a = 0,57*10^{-6} m^2/sec. The estimation of the depth of heating of the boundary layer is carried out at three values of the nozzle wall temperature, which can take place under different operating modes, t_w = 400, 300 and 200 ºC.

A preliminary analysis of the diesel engine fuel system operation showed that the heat flow from the cylinder cavity has a component passing through the spray nozzle body into the fuel flow.

With a short duration of the heating cycle the fuel flow in the nozzle channel is heated not enough and as a consequence the channel curve radius can be neglected. In such a formulation of the problem the fuel flow is considered as a semi-infinite body.

When it concerns a semi-infinite body, where the temperature field changes only in one direction, the Fourier-Kirchhoff equation is used (Roslyakov, 2004;Lykov, 1978)

$$\frac{\partial\theta}{\partial\tau} = a\frac{\partial^2\theta}{\partial X^2}$$

where θ =t_s– t - temperature difference;
t_s – wall temperature;
t – current temperature;
t_o – free fluid temperature;
X – normal coordinate to the wall;
a - thermal diffusivity.

The function δ(τ) called the penetration depth (thermal layer depth) isconsidered. For all X>δ(τ) the temperature of the medium is equal to the initial temperature t_o.The depth of the thermal layer at a constant wall temperature is defined by the dependence δ(τ) = $6\sqrt{a\tau}$.

To simplify the transformations when integrating the Fourier-Kirchhoff equation the so-called error function is introduced.

$$erfZ = \frac{2}{\sqrt{\pi}}\int_o^Z \exp\left(-t^2\right)dt$$

The value of the function changes from erf (Z) = 0withZ = 0 toerf (Z) = 1 withZ = 3.

Based on this a relative coordinate was introduced.

$$Z = \frac{X}{2\sqrt{a\tau}}$$

By integrating the Fourier-Kirchhoff equation we get the following expression for the temperature distribution in the heated layer (Roslyakov, 2004):

$$\frac{\theta}{\theta_o} = erf(z) \text{ or } \frac{t_s - t}{t_s - t_o} = erf(z)$$

θ_o - a maximum temperature difference θ_o =t_s– t_o

The calculation results of the cycle duration ($\tau_{prog.}$) and the depth of the thermal layer in the fuel flow depending on the engine crankshaft speed (*n*) are given in Table 4.

The results of calculating the fuel temperature over the channel section in the direction from the wall to the channel axis using dependencies (3) and (4) with the tabulated erf (z) function are shown in Fig.10 and Fig.11. The authors show the graphs of the fuel temperature distribution over the channel section at different operating modes of the diesel engine.

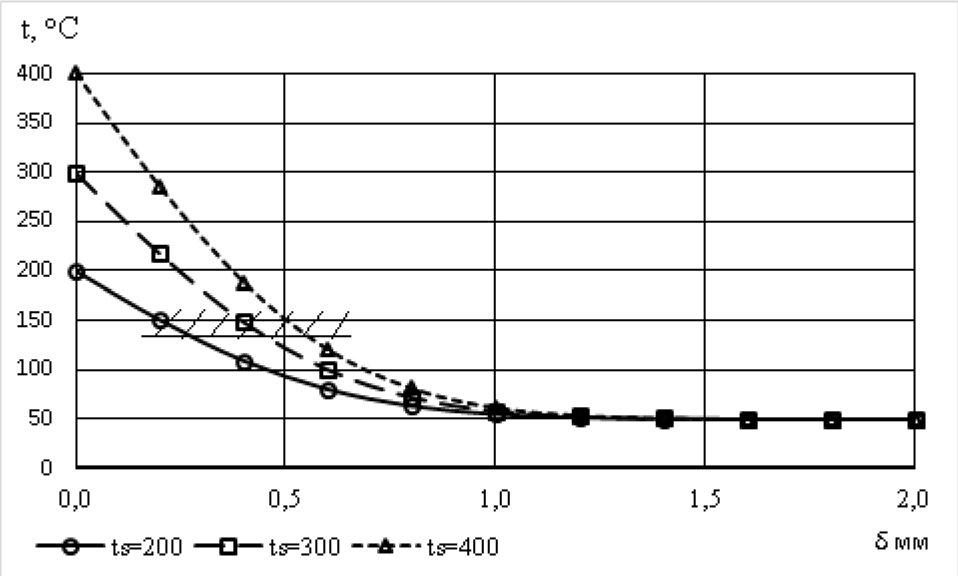
Table 4. Dependence of the cycle duration and fuel heating depth on the crankshaft speed

Crankshaft revolutions, n, rpm	600	1000	1400	1800	2200
Number of revolutions, n, sek -1	10	16,67	23,33	30	36,67
Cycle time, s	0,100	0,060	0,043	0,033	0,027
Injection pulse width, s	0,00556	0,00333	0,00238	0,00185	0,00152
Heating period, (τ), c	0,195	0,117	0,0836	0,065	0,0532
Heating depth, mm	0,002	0,001549	0,00131	0,001155	0,001045

These curves are universal for assessing the thermal state of the nozzle injectorin general.

It should be mentioned that the processes of fuel oxidation by oxygen and as a consequence the carbon-producingprocesses begin to proceed intensively at fuel temperatures of 130 ºC and above.

According to Fig. 10 and Fig. 11 it can be seen that with a decrease in engine speed from n= 2200 rpmton= 600 rpm at a nozzle wall temperature of 300 ºC, the thickness of the heated layer up to a fuel temperature of 135ºC and more increases from 0,24 mm to 0,45 mm, which is almost doubled.When all other conditions are equal, an intensification of the process of gumming and carbon producing in the nozzle channels takes place in the same proportion.



- the limit of acceptable fuel temperatures

Figure 10. Distribution of fuel temperature in the channel depending on the wall temperature of the nozzle at a crankshaft speed of the D-240 diesel equal to 600 rpm

In addition, as it was mentioned above, when switching to a dual fuel cycle with an increase in the operating mode from 600 rpm to 2200 rpm, if the flow rate of the ignition dose of diesel fuel remains unchanged, the conditional heat load increases by 154% and the wall and fluid temperature increases in the boundary layer.

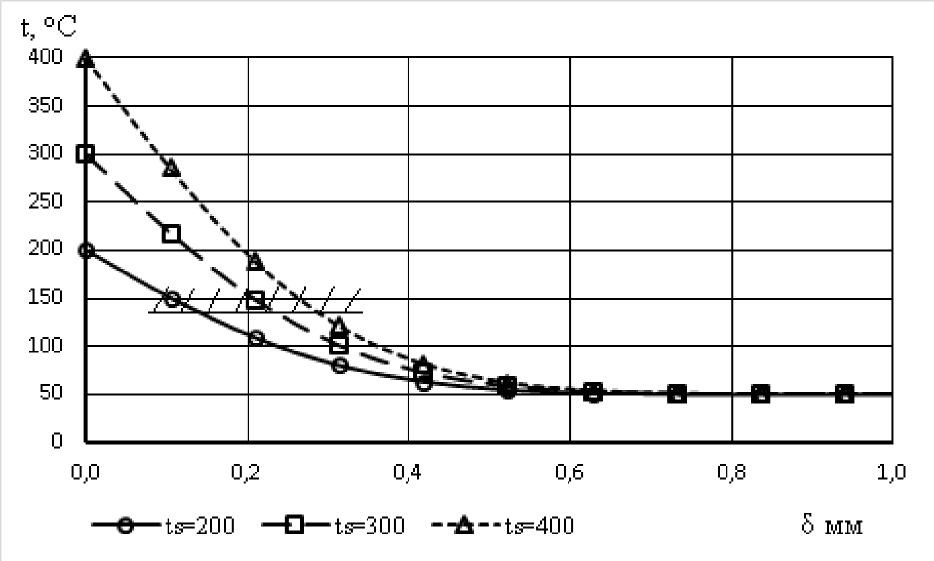


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Figure 11. Distribution of fuel temperature in the channel depending on the wall temperature of the nozzle at a crankshaft speed of the D-240 diesel equal to 2200 rpm

6. Discussion

The physical and chemical properties of a diesel fuel and ethanol mixture change significantly with a change in the component composition. This change must be taken into account when designing the fuel delivery system.The lower level of ethanol heat is less than the combustion efficiency of diesel fuel. However, the power of the diesel engine with ethanol supply can be maintained at the same level. In order to do this it is necessary to select the optimal composition of the mixed fuel.It is necessary to get the maximum possible amount of water vapor in the combustion products.This follows from the thermodynamic features of combustion products, which perform the functions of a fluidin the diesel cylinder. The process during expansion of combustion products in the diesel cylinder is proportional to the gas constant of the fluid.The physical meaning of the gas constant is when it cools down by one degree, a kilogram of the fluid does work that is numerically equal to the value of the gas constant.Consequently, with an increase of water vapor share having gas constant 55% greater than the average for the combustion productsmixture, the operation of the cycle on the expansion stroke increases.This approach to the selection of the mixed fuelcomposition is compensated by a decrease in the power of the diesel engine due to the lower heat of ethanol combustion.

Besides, taking into account that ethanol has a lower calorific value than diesel fuel it is necessary to evaluate the efficiency of a diesel engine cycle not by specific fuel consumption, but by net efficiency.

The calculation results show that when adding ethanol to the cylinders of a diesel engine, it leads to an increase in the water vapor share in the combustion products up to 12%.This leads to an increase in the gas constant of the combustion products from 297 kJ/kg to 316,8 kJ/kg and the expansion work of the fluid increases by 6.7%.

What concerns the diesel fuel heating in the D-240 diesel nozzle when operating on diesel fuel and ethanol, the authors have got the results that explain the reason for the failure of the injectors when the proportion of the ignition dose of diesel fuel decreases to less than 15% and the injector temperature increases to 130 ºC and above.Gums and carbon deposits appear in the nozzle chamber.The most unfavorable zone in terms ofgums and other carbon deposits is the zone between the nozzle body and the needle. The gap between the body and the needle, which is approximately 0,15mm, is comparable to the fuel heating depth. The fuel in this gap heats up to the temperature of the nozzle body and is a source of gum deposits, which lead to the jamming of the needle.

7. Conclusion

So, according to calculation studies theauthorshavethe results that letthem to explain some of the unfavorable features of the processes occurring in a diesel engine, when a significant part of diesel fuel is replaced with ethanol:

- to get the same excess air ratio, it is necessary to supply extra mixed propellant;
- ethanol during delivery contains water impurities, the amount of which is allowed in accordance with the technical requirements for the production and supply of ethanol, including up to 8% in the USA;
- to exclude the probability of nozzles jamming because of water for supplying a diesel engine, ethanol must be supplied through an isolated system;
- with a decrease of the ignition dose share of diesel fuel to values less than 15%, an additional cooling of the nozzle body is required to ensure a temperature of less than 130 ºC.

Funding: The research is partially funded by the Ministry of Science and Higher Education of the Russian Federation under the strategic academic leadership program 'Priority 2030' (Agreement 075-15-2021-1333 dated 30.09.2021)

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Andrey Balakin
Samara State Transport University,
Samara, Russia
lok@samgups.ru
ORCID 0000-0002-1321-6156

Alexey Roslyakov
Samara State Transport University,
Samara, Russia
roslykov_ad@mail.ru
ORCID 0000-0002-5086-6027

Yury Klochkov
Peter the Great St.Petersburg Polytechnic University,
St.Petersburg, Russia
y.kloch@gmail.com
ORCID 0000-0002-7913-8285