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INVESTIGATION ON THE PROCESS OF INJECTION OF COMMERCIAL GRADE AND BLENDED FUEL IN A DIESEL ENGINE

Purpose. A computational valuation of the parameters of the process of commercial grade and blended fuel flow in the injector nozzle of a locomotive diesel engine, and its impact on spraying conditions in the combustion chamber.

Methodology. The scientific investigation is based on using the technique of a comparative numerical experiment. Modern numerical methods in computational fluid dynamics are used for simulating fuel flow and spraying processes in the injector nozzle and combustion chamber.

Findings. It was found that when working with commercial grade fuel with maximum pressure in the area of the injector well of 85 MPa, the fuel flow velocity in the fuel injection nozzle hole reaches 434 m/s, whereas when working with fuel blended with alcohol the velocity decreases to 429 m/s (at a 25 % alcohol concentration). Due to the lower pressure of saturated vapours of the blended fuel, as compared to that of commercial grade fuel, the fuel-air mixing conditions degrade at the operating duty being investigated.

Originality. The investigation helped to study the impact of blended fuel composition on changes in the processes of its flow in the injector nozzle, and on the injection into the combustion chamber and the fuel-air mixing conditions. The study results helped to develop recommendations on ensuring effective ICE operation with blended fuel.

Practical value. A change in the conditions of blended fuel spraying and fuel-air mixing should be taken into account when choosing effective fuel injection advance angles and fueling principles in order to ensure high ICE economic and ecological performance.

Keywords: *diesel engine, injector nozzle, fuel flow, cavitation, combustion chamber, environmental indicators*

Introduction. Increased demands on the environmental indicators of internal combustion engines, primarily diesel ones, requires the use of a package of measures to improve their environmental and economic indicators [1–5]. The use of heavy diesel fuels for diesel locomotive engines causes a high level of toxicity of exhaust gases in combination with a significant mass emission of solid particles [2, 4]. Because of their developed surface, solid particles are known to be the carriers of carcinogenic and mutagenic substances, which have an adverse impact on the environment and human body [4].

The use of alcohol-containing mixed fuels makes it possible to partially replace fossil fuels and at the same time, after adjusting the diesel engine management system and optimising the engine operating cycle parameters, reduce the mass emission of solid particles with exhaust gases.

The presence of sulfur in a diesel fuel enables additional lubrication of rubbing, loaded elements of the diesel engine fuel equipment. However, in the process of diesel fuel combustion, sulfur is involved in the formation of solid particles [6]. The addition of alcohol to diesel fuel also reduces the sulfur content, but at the same time worsens the lubrication conditions of rubbing surfaces. To ensure the reliable operation of a diesel engine on alcohol-containing fuels, it is necessary to add special additives that improve the lubricity of a mixed fuel [7, 8].

Literature review. Predicting and investigating the conditions of occurrence of hydrodynamic cavitation in the fuel injection equipment of modern diesel engines is an important research-and-engineering task [8, 9].

The investigation and refinement of engine cycles in modern diesel engines requires the refinement of fuel injection equipment. Changing the configuration of the combustion chamber in the piston and the injection pressure, and reducing the filminess fraction calls for additional research in processes occurring in fuel injection equipment, primarily, in the injector nozzle.

Experimental research in the processes of fuel injection and spraying enables assessing the impact of design and operating duty factors on these processes with high accuracy, but such research is very resource-intensive [9, 10].

Coupling numerical and experimental research during engine development yields acceptable results within a short time and with minimum input requirements [10, 11].

As it is known, fuel flow in the injector nozzle holes at certain operating duties can cause hydrodynamic cavitation resulting in excessive wear of nozzle holes with an adverse impact on engine performance [11].

Hydrodynamic cavitation occurs due to a local pressure drop lower than the pressure of fuel saturated vapours (due to an increasing flow velocity) [12]. This involves fuel flashing and the formation of cavitation bubbles. Following their nucleation, the bubbles pass through growth, compression and collapse stages. The compression of a cavitation bubble reduces its volume multiply and in very short time, followed by a sharp pressure and temperature increase inside the bubble [12]. The bubble collapse process liberates substantial energy in the form of a micro spray, resulting in damage (cavitation erosion) during the interaction of the micro spray with the injector nozzle walls [11, 12].

Refining the procedure of numerical research in fuel injection and spraying processes, and evaluating the erosion wear of nozzle holes under hydrodynamic cavitation impact is a promising research area.

The fuel injection intensity and spraying quality in a diesel engine combustion chamber has a decisive impact on the conditions of air-fuel mixture formation, combustion and creation of harmful substances in an ICE cylinder [8].

A numerical simulation of the fuel flow processes in the injector nozzle makes it possible to evaluate the influence of structural and operating factors on the processes in the injector nozzle and nozzle holes [9], predict the conditions for the occurrence of hydrodynamic cavitation and the associated erosion of the surfaces of the injector nozzle needle and nozzle holes [10]. During operation, such erosion affects the quality of operating process management and the environmental indicators of a diesel engine.

Numerical simulation of the process of atomizing fuel in the combustion chamber enables investigating the conditions of mixture formation and developing recommendations for optimising the processes of pre-flame preparation of fuel in the combustion chamber of a diesel engine [11].

Unsolved aspects of the problem. As clear from the results of the literature review, the study on the processes of fuel injection and atomization, improvement of methods for the numerical simulation of these processes, and development of recommendations for optimizing the processes of injection, mixture formation, and combustion of fuel is an important and crucial task.

Purpose. The purpose of this study is to calculate the parameters of the process of diesel fuel and mixed fuel flow in the injector nozzle of a diesel locomotive engine and its effect on the spray pattern in the combustion chamber.

To achieve this purpose, the study addressed the following tasks:

- to conduct a literature review in modern methods of computational research in the processes of fuel injection and atomization;
- to form a data set with the thermal characteristics of a mixed fuel, depending on the concentration of the alcohol component;
- to conduct a computational study on the diesel fuel and mixed fuel flow in the injector nozzle in the mode under investigation;
- to conduct a computational study on the process of atomizing the diesel fuel and mixed fuel in the combustion chamber in the mode under investigation;
- to draw conclusions and make recommendations for improving the conditions of mixture formation for the case when the engine is running on mixed fuel and adjusting the diesel engine management system.

The objects of the study are the processes of diesel fuel and mixed fuel injection and atomization in the injector nozzle and combustion chamber of the 16 ChN26/27 diesel locomotive engine when operating in the Ne = 2940 kW rated power mode with the crankshaft rotational speed of $n = 1000\ \text{min}^{-1}$.

The engine under investigation has a Hesselman-type combustion chamber with implementation of the volumetric mixture formation process. The injector nozzle is centrally located in the combustion chamber, and has eight 0.42 mm diameter holes.

The thermo-physical properties of the fuels considered in this work are presented in Table 1 below.

As evident from Table 1, adding ethanol to diesel fuel changes all the thermophysical properties of a mixed fuel.

Table 1

Thermo-physical properties of diesel fuel and mixed fuels

Fuel	ρ	*M*	*Cp*	η	λ	*Ps**
	kg/m^3	kg/kmol	J/(kg · K)	MPa · s	Wt/(m · K)	KPa
Diesel 100 %	829	200	2051.53	2.77	0.116	17.9
E5-D	827	192.3	2019.95	2.58	0.156	17.3
E15-D	823	176.9	1956.8	2.26	0.235	16.1
E25-D	819	161.5	1893.65	1.96	0.314	14.9
Ethanol 100 %	789	46.07	1420	1.2	0.91	5.95

* – at 40 °C

Thus, adding 25 % of ethanol decreases density ρ only by 1.2 %, dynamic viscosity η by 29 % and saturated vapours pressure by 16 %. Adding 25 % of ethanol reduces heat capacity *Cp* insignificantly by 8 %, though thermal conduction λ increases by 2.7 times. As expected, the molar mass *M* of the mixed fuel also decreases by 19 %.

The advance angle of fuel injection to the top dead center (TDC) is 20 degrees of crankshaft rotation and the fuel injection duration is 20 degrees of crankshaft rotation.

Methods. The geometry of a fragment of the injector nozzle with a needle, the computational domain and computational grid describing the configuration of the injector nozzle of the 16 ChN 26/27 diesel engine are shown in Fig. 1.

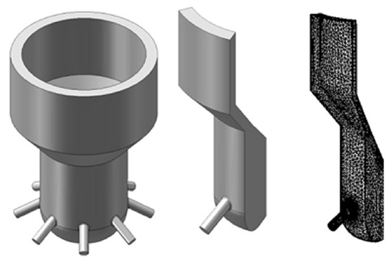
The geometry of the injector nozzle with a needle (Fig. 1, *a*) is formed for the case when the needle is fully lifted. In the following calculations, a computational domain (sector of 45 degrees) is used. The domain contains a volume describing the annular gap between the needle and injector body, a nozzle seat, and a section describing the nozzle holes (Fig. 1, *b*). The computational grid contains 450 000 computational cells and has a local concentration near solid walls – five layers of computational cells with a minimum height of 0.01 mm to correctly simulate the wall effects of the computational cells (Fig. 1, *c*).

Table 2 is a brief specification of diesel engine 16ChN26/27 considered in the paper.

This paper considers both diesel fuel and mixed fuel (with added alcohol, and liquid and fuel vapours) as working fluids [11]. In order to describe the turbulent flows in both the injector and combustion chamber, the *k*-epsilon turbulence model is used [12].

The standard *k*-ε model equations have the form

$$\frac{\partial}{\partial t}(\rho\cdot k)+\frac{\partial}{\partial x_i}(\rho\cdot k\cdot\overline{u}_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+$$
$$+G_k+G_b-\rho\cdot\varepsilon-Y_M+S_k;$$



a *b* *c*

Fig. 1. Configuration of a fragment of the injector nozzle of diesel engine 16ChN26/27: nozzle geometry (a), computational domain (nozzle fragment) (b) and computational grid (c)

Table 2

Brief specification of diesel engine 16ChN26/27

No	Parameter	Value
1	Rated power, kW	2.940
2	Rotational speed corresponding to rated power conditions, min^{-1}	1.000
3	Cylinder diameter, mm	260
4	Piston stroke, mm	270
5	Rated power, kW	2.940

$$\frac{\partial}{\partial t}(\rho\cdot\varepsilon)+\frac{\partial}{\partial x_i}(\rho\cdot\varepsilon\cdot\overline{u}_i)=$$
$$=-\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right]+G_{1\varepsilon}\frac{\varepsilon}{k}(G_k+G_{3\varepsilon}\cdot G_b)-C_{2\varepsilon}\cdot\rho\frac{\varepsilon^2}{k}+S_\varepsilon,$$

where k is specific kinetic turbulent energy; ε is rate of viscous dissipation of turbulent energy; ρ is gas density; μ_t is turbulent dynamic viscosity; $\overline{u}_i$ is averaged velocity; σ_k is a dimensionless empirical constant; G_k is turbulent kinetic energy formed by mean velocity gradients; G_b is displacement force kinetic energy; $C_{3\varepsilon}$, $C_{2\varepsilon}$ are constants; Y_M is the contribution of the variable expansion during compression turbulence to the total dissipation rate; S_k is invariant of the strain tensor.

In order to describe the process of the phase transition from a liquid to a gaseous state, the hydrodynamic cavitation model is used [12]. In order to describe the two-phase flow process (fuel and vapours), the mixture model is used [12].

Basic model equations

$$X=\frac{p-p_v}{(1/2\cdot\rho\cdot U^2)},$$

where X is the cavitation number; p is reference pressure for the flow; p_v is vapour pressure for the liquid; ρ is liquid density; U is flow velocity.

The Rayleigh-Plesset model equation provides the basis for the rate equation that controls vapour generation and condensation. The Rayleigh-Plesset equation describes the growth of a gas bubble as

$$R_B\frac{d^2R_B}{dt^2}+\frac{3}{2}\cdot\left(\frac{dR_B}{dt}\right)^2+\frac{2\cdot\sigma}{\rho_f\cdot R_B}=\frac{p_v-p}{\rho_f},$$

where R_B represents the bubble radius; p_v is pressure in the bubble (assumed to be the vapour pressure at the liquid temperature); p is pressure in the liquid surrounding the bubble; ρ_f is liquid density; σ is the surface tension coefficient between the liquid and vapour.

The mixture model [12] is used for describing the two-phase stream flow process (diesel fuel and vapours).

Basic model equations are

$$D_{\alpha\beta}=C_D\cdot\rho_{\alpha\beta}\cdot A_{\alpha\beta}\cdot\left|U_\beta-U_\alpha\right|\cdot(U_\beta-U_\alpha),$$

where $D_{\alpha\beta}$ is total drag exerted by phase β on phase α per unit volume.

The mixture density $\rho_{\alpha\beta}$ is given by

$$\rho_{\alpha\beta}=f_\alpha\cdot\rho_\alpha+f_\beta\cdot\rho_\beta,$$

where f_α, f_β are volume particles of phases α and β.

The interfacial area per unit volume $A_{\alpha\beta}$ is given as

$$A_{\alpha\beta}=\frac{f_\alpha\cdot f_\beta}{d_{\alpha\beta}},$$

where $d_{\alpha\beta}$ is a user-specified mixture length scale.

The fuel injection process is simulated for the nominal mode of engine operation with a maximum injection pressure of 85 MPa and an injection rate of 1.4 g/cycle.

As the boundary conditions (Fig. 2), we used the fuel pressure (Table 3) at the injector nozzle inlet (85 MPa), and at the injector nozzle outlet, the conditions for efflux into the combustion chamber (pressure and temperature in the combustion chamber, which were described based on the simulation results of the operating cycle on diesel fuel [8]).

Results. The results of the numerical simulation of the process of diesel fuel and mixed fuel flow in the injector nozzle are shown in Figs. 3–6.

The results of calculation of the process of diesel fuel flow in the injector nozzle are shown in Fig. 3.

The fuel pressure distribution pattern in the vertical plane section is shown in Fig. 3, *a*. During fuel injection, in the place where the nozzle hole is coupled with the nozzle seat, one can observe a local pressure decrease to 0.01 MPa and a flow rate increase up to 434 m/s (Fig. 3, *b*). At the same time, in this domain, one can observe a phase transition from a liquid to a gaseous state (hydrodynamic cavitation) (Fig. 3, *c*).

In moving away from the given domain, the vapour fuels condense and the flow changes from a two-phase flow to a one-phase one. These are the areas (with a sharp change in the geometry of the injector nozzle flow part) where cavitation erosion loci appear later. They have an adverse impact on the effectiveness of the fuel injection process and the engine cycle indicators, and they degrade the diesel engine environmental performance.

The emergence of hydrodynamic cavitation is associated with a sharp change in the flow characteristics and a local pressure decrease below the saturated fuel vapour pressure. This leads to an off-design increase in the length of fuel sprays, and subsequently, at a certain operating time, to the erosive wear of the injector nozzle holes with deterioration of the fuel injection rate and the degree of fuel atomization by the injector nozzle in the combustion chamber.

The results of calculation of the mixed fuel (95 % of diesel fuel and 5 % of alcohol) flow in the injector nozzle are presented in Fig. 4.

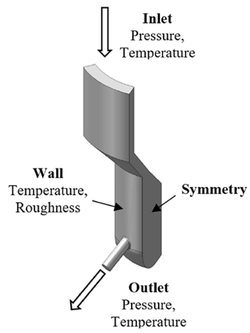
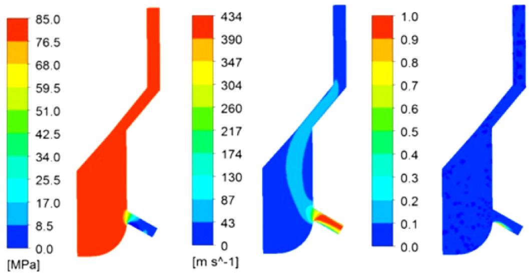


Fig. 2. Schematic presentation for specifying the boundary conditions to describe the fuel flow process in an injector nozzle fragment

Table 3

Boundary conditions

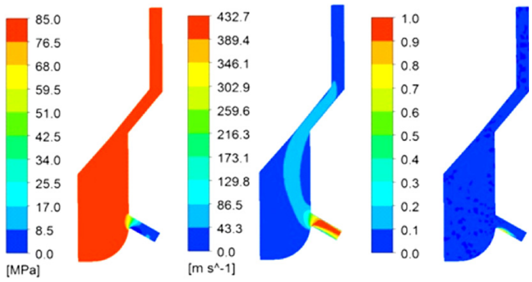
Boundary	Boundary type	Value
Inlet	Injection pressure, MPa	85
	Fuel temperature, °C	90
Outlet	Pressure in the combustion chamber, MPa	5
	Temperature in the combustion chamber, °C	540
Wall	Injector spray nozzle wall temperatures, °C	180
	Wall roughness, mm	20



a *b* *c*

Fig. 3. Results of numerical simulation of the diesel fuel flow process in the injector nozzle flow path:

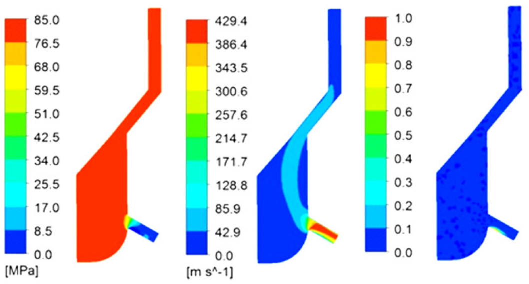
a – fuel pressure distribution; b – flow rate distribution; c – vapour phase volume fraction distribution



a *b* *c*

Fig. 4. Results of numerical simulation of the process of mixed fuel (*95 % of diesel fuel and 5 % of alcohol*) *flow in the injector nozzle flow path*):

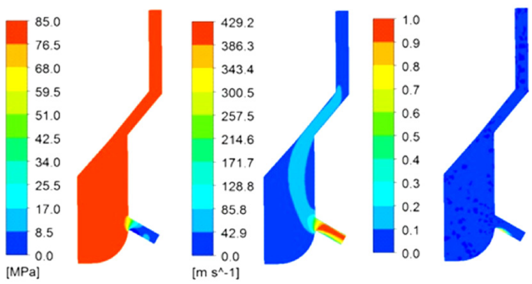
a – fuel pressure distribution; b – flow rate distribution; c – vapour phase volume fraction distribution



a *b* *c*

Fig. 5. Results of numerical simulation of mixed fuel (*85 % of diesel fuel and 15% of alcohol*) *flow in the injector nozzle flow path*):

a – fuel pressure distribution; b – flow rate distribution; c – vapour phase volume fraction distribution



a *b* *c*

Fig. 6. Results of numerical simulation of mixed fuel (*75 % of diesel fuel and 25 % of alcohol*) *flow in the injector nozzle flow path*:

a – fuel pressure distribution; b – flow rate distribution; c – vapour phase volume fraction distribution

The changing viscosity, density and pressure of saturated mixed fuel vapours inevitably affect the flow process in the flow part of the injector nozzle. With growing alcohol concentration in the mixed fuel, this impact increases.

The fuel pressure distribution pattern in the vertical plane section is shown in Fig. 4, *a*. During fuel injection, in the place where the nozzle hole is coupled with the nozzle seat, one can observe a local pressure decrease to 0.015 MPa and a flow rate increase up to 433.2 m/s (Fig. 4, *b*). At the same time, in this domain, one can observe a phase transition from the liquid state to the gaseous state (hydrodynamic cavitation) (Fig. 4, *c*). The lower saturated vapour pressure and mixed fuel viscosity (Table 1) affect both the mixed fuel flow and the conditions for the occurrence of hydrodynamic cavitation.

That is, for the variants of mixed fuels being considered the cavitation number X will be greater, on the average, by 0.85–3.5 % than that for standard (diesel) fuel. This reduces the probability of occurrence of hydrodynamic cavitation in the flow part of the fuel injector nozzle.

The results of calculation of mixed fuel (85 % of diesel fuel and 15 % of alcohol) flow in the injector nozzle are presented in Fig. 5.

The fuel pressure distribution pattern in the vertical plane section is shown in Fig. 5, *a*. During fuel injection, in the place where the nozzle hole is coupled with the injector nozzle seat, one can observe a local pressure decrease to 0.012 MPa and a flow rate increase up to 429.4 m/s (Fig. 5, *b*). At the same time, in this domain, one can observe a phase transition from the liquid state to the gaseous state (hydrodynamic cavitation) (Fig. 5, *c*).

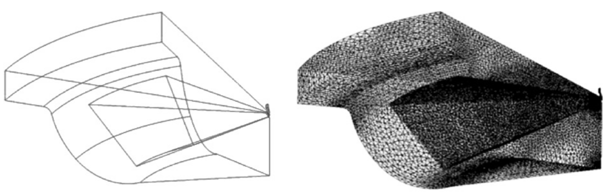
The results of calculation of mixed fuel (75 % of diesel fuel and 25 % of alcohol) flow in the injector nozzle are presented in Fig. 6.

The fuel pressure distribution pattern in the vertical plane section is shown in Fig. 6, *a*. During fuel injection, in the place where the nozzle hole is coupled with the injector nozzle seat, one can observe a local pressure decrease to 0.011 MPa and a flow rate increase up to 429.2 m/s (Fig. 6, *b*). At the same time, in this domain, one can observe a phase transition from the liquid state to the gaseous state (hydrodynamic cavitation) (Fig. 6, *c*). At the same time, the lower vapour pressure of the mixed fuel with an alcohol content of 25 % reduces the hydrodynamic cavitation intensity (Fig. 6, *c*).

Additional computational studies were conducted for all-round investigation of the impact of the thermophysical properties of the fuel on the process of fuel spray spread in the combustion chamber. To this end, numerical methods were used to simulate the process of injection of standard and mixed fuels into the combustion chamber.

Such studies enable to investigate the conditions of propagation of the fuel spray within the combustion chamber volume, fuel spray blow off during its interaction with the air vortex, the conditions of evaporation of fuel droplets, the processes of preflame, and ignition and formation of toxic and carcinogenic substances.

The study on the process of fuel injection into the model combustion chamber (combustion chamber sector of 45°) is shown in Fig. 7.



a *b*

Fig. 7. Sector of combustion chamber in diesel engine 16ChN26/27: computational domain (*a*) *and computational grid* (*b*)

To reduce the dimensionality of the problem being considered, a fragment (sector) of the combustion chamber was considered in the paper. Such an approach is common when investigating fuel injection processes and simulating processes in diesel engines with a central-installed injector and a symmetrical combustion chamber in the piston.

The computational domain includes an injector nozzle sector (45°) with one spray hole (Fig. 7, *a*) and a computational grid describing the configuration of the combustion chamber sector (3.5 million design cells), Fig. 7, *b*).

To increase numerical simulation accuracy, the computational grid in the paper was adapted in the direction of propagation of the fuel spray in the combustion chamber, Fig. 7, *b*. A computation domain fragment was selected, in which the sizes of the computational meshes were reduced by a factor of five to increase the accuracy of simulating complex phase transition processes (fuel droplets evaporation) on the fuel spray top.

Identifying the core factors that influence the process of fuel spray propagation in the combustion chamber and its disintegration (formation of droplets and their subsequent decay) is practical for the case of injection into a static charge, especially when investigating mixed fuels with drastically differing thermophysical properties.

In the paper, fuel injection was simulated for the rated power mode with a maximum injection pressure of 85 MPa into the static charge (without taking into account the movement of air in the combustion chamber during engine operation).

The results of numerical simulation of the process of atomizing diesel fuel and mixed fuel in the combustion chamber sector are presented in Figs. 8, 9.

Fig. 8 shows the fuel spray velocity distribution and the fuel mass fraction distribution pattern in the combustion chamber sector volume.

When diesel fuel is injected into the combustion chamber, its maximum flow velocity reaches 457.2 m/s where the fuel jet leaves the injector nozzle hole. As the fuel spray approaches the combustion chamber walls, its velocity changes from 300 to 45 m/s (Fig. 8, *a*). During injection of the mixed fuel with a 5 % content of alcohol, the fuel jet maximum velocity reaches 455.9 m/s (Fig. 8, *b*). During injection of the mixed fuel with a 15 % content of alcohol, the fuel jet maximum velocity reaches 455 m/s (Fig. 8, *c*). During injection of the mixed fuel with a 25 % content of alcohol, the fuel jet maximum velocity reaches 449.6 m/s (Fig. 8, *d*). The fuel spray structure has a clearly expressed zonal pattern: the fuel spray core, transition section, and section of fine-dispersed droplets and fuel vapour can be observed in it (Figs. 8, *e–h*).

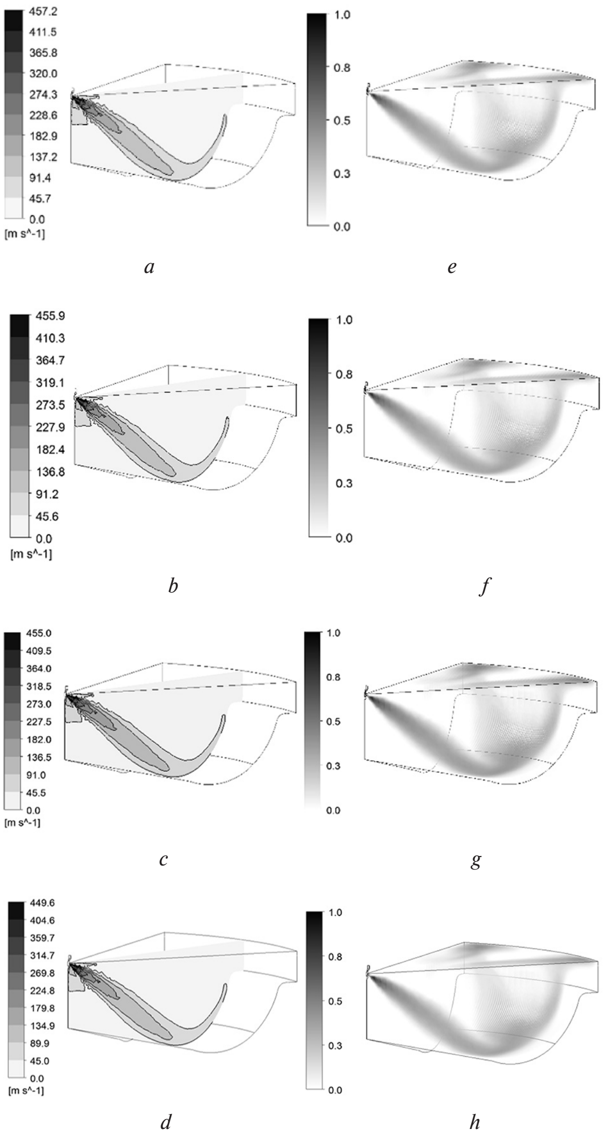


Fig. 8. Distribution of the flow rate and mass fraction of diesel fuel and mixed fuels during flow in the injector nozzle:

a, b – diesel fuel; c, d – 5 % alcohol content; e, f – 15 % alcohol content; g, h – 25 % alcohol content

The fuel dispersion distribution in the meridional plane of the combustion chamber sector is shown in Fig. 9.

During diesel fuel atomizing, the maximum droplet size reaches 90 microns and, on average, varies from 50 to 80 microns (Fig. 9, *a*). During the atomizing of mixed fuel with a 5 % content of alcohol (Fig. 9, *b*), the droplet size varies, on average, from 35 to 75 microns. For a mixed fuel with a 15 % (Fig. 9, *c*) and 25 % content of alcohol (Fig. 9, *d*), the maximum droplet size reaches 85 microns, and the average droplet size varies from 30 to 70 microns.

The decreasing size of droplets with an increasing alcohol concentration in the mixed fuel is attributed to a decreasing viscosity, density and pressure of saturated vapours of the mixed fuel (Table 1).

The decreasing size of fuel droplets increases the rate of fuel evaporation and reduces the ignition delay period, making the diesel engine process more controllable because fuel combustion follows the fuel feed law.

Hence, numerical methods were used to study the processes of injection and spraying of standard and mixed fuels for the D80 diesel engine in a locomotive.

It was shown that changing the fuel thermophysical properties (primarily, viscosity and pressure of saturated vapours) affects the conditions of possible occurrence of hydrodynamic cavitation in the injector nozzle and the conditions of propagation of the fuel spray in the combustion chamber.

The approach considered in the paper reveals the core factors that affect the fuel spraying conditions in the combustion chamber when working with standard and mixed fuels. The practical effect of the study consists in an in-depth analysis and development of scientific and practical recommendations for increasing the effectiveness of the process of injection and spraying of standard and mixed fuels in a locomotive diesel engine. Accounting for these factors in the future will help choose efficient design and operating duty factors for fuel injection equipment to ensure effective diesel engine performance with standard and mixed fuels.

Conclusions. According to the results of the comparative computational study, the following can be noted:

- based on the literature review, it was established that the use of mixed fuels produced from renewable raw materials reduces both the level of consumption of fossil fuels and the level of toxicity of diesel engine exhaust gases;
- increasing the concentration of alcohol in the mixed fuel for diesel engines to over 15 % requires using additives that compensate for reduced fuel lubricity;

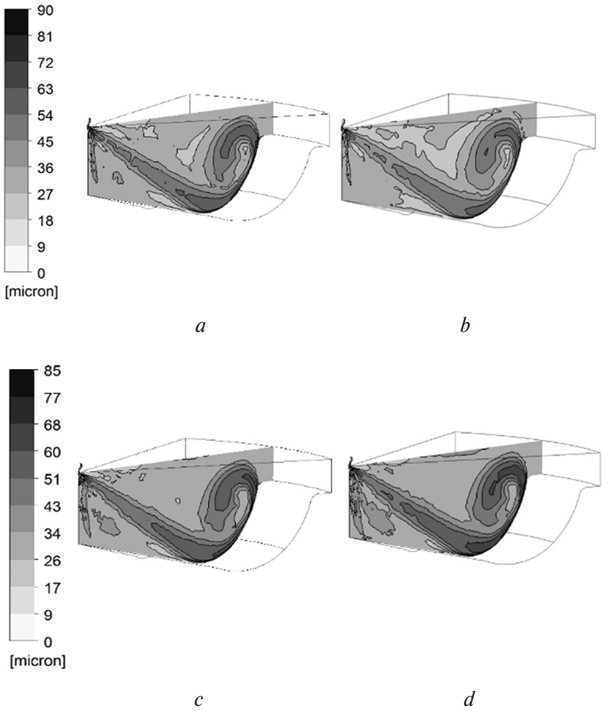


Fig. 9. Distribution of diesel fuel and mixed fuel dispersion during atomization in the combustion chamber sector:

a – diesel fuel; b – 5 % content of alcohol; c – 15 % content of alcohol; d – 25 % content of alcohol

- changing the thermophysical properties of the mixed fuel, primarily, reducing the saturated vapour pressure and the fuel density and viscosity minimizes the factors that cause the occurrence of hydrodynamic cavitation in the injector nozzle (for the cavitation number X, the considered alcohol concentrations in the mixed fuel of 5 to 25 % increase X from 0.85 to 3.5 %);
- using mixed fuel with a 15 % content of alcohol or more increases the length of fuel sprays and affects the conditions of mixture formation, requiring additional computational and experimental studies to optimize the fuel supply process (especially for diesel locomotive engines with a volumetric method of mixture formation);
- a joint study on the processes in both the injector and combustion chamber during fuel injection facilitates a comprehensive study of these processes and the development of recommendations for improving the conditions of pre-flame preparation of both diesel and mixed fuels.

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Дослідження процесу впорскування штатного й сумішевого палива в дизельному двигуні

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Мета. Розрахункова оцінка параметрів процесу течії штатного й сумішевого палив у розпилювачі форсунки тепловозного дизельного двигуна та їх вплив на умови розпилювання в камері згоряння.

Методика. Наукове дослідження засноване на використанні методики порівняльного чисельного експерименту. Сучасні чисельні методи обчислювальної аерогідродинаміки використовуються для моделювання процесів течії й розпилювання палива в розпилювачі форсунки та камері згоряння.

Результати. Встановлено, що при роботі на штатному паливі з максимальним тиском в області колодязя розпилювача 85 МПа швидкість течії палива в сопловому отворі досягає 434 м/с, а при роботі на паливі з додаванням спирту – знижується до 429 м/с (для концентрації 25 % спирту). Через більш низький тиск насиченої пари сумішевого палива, у порівнянні зі штатним, умови сумішоутворення на досліджуваному режимі погіршуються.

Наукова новизна. Дослідження дозволило вивчити вплив складу сумішевого палива на зміну процесів її течії

в розпилювачі форсунки, упорскування в камеру згоряння та умови сумішоутворення. Отримані результати дозволили сформувати рекомендації із забезпечення ефективної роботи двигунів внутрішнього згоряння (ДВЗ) на сумішевому паливі.

Практична значимість. Зміну умов розпилювання сумішевого палива й сумішоутворення надалі необхідно враховувати при виборі раціональних кутів випередження упорскування й закону паливоподачі для забезпечення високих економічних й екологічних показників ДВЗ.

Ключові слова: *дизельний двигун, розпилювач форсунки, течія палива, кавітація, камера згоряння, екологічні показники*

Исследование процесса впрыска штатного и смесевого топлива в дизельном двигателе

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Цель. Расчетная оценка параметров процесса течения штатного и смесевого топлив в распылителе форсунки тепловозного дизельного двигателя и их влияния на условия распыливания в камере сгорания.

Методика. Научное исследование основано на использовании методики сравнительного численного эксперимента. Современные численные методы вычислительной аэрогидродинамики используются для моделирования процессов течения и распыливания топлива в распылителе форсунки и камере сгорания.

Результаты. Установлено, что при работе на штатном топливе с максимальным давлением в области колодца распылителя 85 МПа скорость течения топлива в сопловом отверстии достигает 434 м/с, а при работе на топливе с добавлением спирта – снижается до 429 м/с (для концентрации 25 % спирта). Из-за более низкого давления насыщенных паров смесевого топлива, по сравнению со штатным, условия смесеобразования на исследуемом режиме ухудшаются.

Научная новизна. Исследование позволило изучить влияние состава смесевого топлива на изменение процессов его течения в распылителе форсунки, впрыска в камеру сгорания и условия смесеобразования. Полученные результаты позволили сформировать рекомендации по обеспечению эффективной работы двигателей внутреннего сгорания (ДВС) на смесевом топливе.

Практическая значимость. Изменение условий распыливания смесевого топлива и смесеобразования в дальнейшем необходимо учитывать при выборе рациональных углов опережения впрыска и закона топливоподачи для обеспечения высоких экономических и экологических показателей ДВС.

Ключевые слова: *дизельный двигатель, распылитель форсунки, течение топлива, кавитация, камера сгорания, экологические показатели*

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