

A Laboratory Investigation into the Fuel Atomization Process in a Diesel Engine for Different Configurations of the Injector Nozzles and Flow Conditions

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Abstract: This paper reports a laboratory investigation of the fuel injection process in a diesel engine. The atomization process of the considered fuel (a hydrocarbon liquid) and the ensuing mixing with air is studied experimentally under high-pressure conditions. Different types of injector nozzles are examined, including (two) new configurations, which are compared in terms of performances to a standard injector manufactured by the Bosch company. For the two alternate configurations, the intake edges of one atomizing hole (hole No. 1) are located in the sack volume while for the other (hole No. 2) they are located on the locking cone of the needle valve. The injection process, the fuel atomization fineness and fuel supply speed characteristics are studied as functions of high-pressure fuel pump camshaft speed and rotation angle. The results obtained show that a decrease in the high-pressure fuel pump camshaft speed can produce fuel redistribution depending on the injector operation. In general, however, the hole No. 1 can ensure fuel flow with higher speed with respect to the hole No. 2 for all the operation modes of the injector. Based on such an analysis, we conclude that the use of certain injectors can enable a fine tuning of the propagation process of fuel sprays into various areas of the diesel engine combustion chamber.

Keywords: Injection; atomization; injector nozzle; channel; injection characteristic; flow coefficient

1 Introduction

Operational characteristics of diesel engines depend much on the quality of fuel atomization and combustion processes. Fuel supply equipment, which ensures fuel injection and atomization, plays an important role in air-fuel mixing and combustion. Combustion rate is largely determined by the dynamic and evaporation of fuel drops, as well as the formation of a mixture of the gas phase of the fuel with air [1–3].

Experience shows that for every model of diesel engine, its specific optimal combinations of injection and atomization characteristics with ones of the air charge and combustion chamber exist, which ensure the best operation parameters of the diesel engine [4–7].

The latest publications show that the further considerable decrease of exhaust gases (EG) toxic emissions of diesel engines using hydrocarbon fuels is associated with the increase of the injection



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pressure p_{in} up to 300 MPa and injection characteristic shape control [8–12]. At the same time, it is evident that one should envisage the optimization of so high injection pressures with the combustion chamber shape and air charge characteristics.

Every engine has its specific optimal value of p_{in} from the viewpoint of fuel spray dynamics. When injection pressure is increased higher than the optimal value, fuel may get on the combustion chamber walls. As a result, there will be a film-volumetric air-fuel mixing rather than pure volumetric one which is worse concerning toxic emissions of EG.

At the same time, the level of pressure p_{in} , which is optimal concerning atomized fuel spray dynamics, does not always ensure the required characteristics of fuel atomization fineness. Here, one should, first of all, take into consideration the period of fuel injection into a diesel engine combustion chamber during the time of injection termination, i.e., fuel supply with decreasing speed. In this case, there is a definite opinion that the better the fuel atomization, the better the diesel engine performance.

So the relevance of searching the ways of improving the quality of fuel supply, atomization and air-fuel mixing with limitation of maximal injection pressure is evident.

Considerable reserves of improvement of the abovementioned processes are behind the improvement of the design of the injectors by optimizing the ratio of the length of the atomization channel (hole) to its diameter. Atomization fineness depends much on the location of the entry edges of the injector nozzle, injector holes, needle valve, i.e., position of the needle valve during the injection process.

2 Research Objects

The objects of comparative tests were two experimental injector nozzles Nos. 1 and 2 designed according to the schemes shown in Figs. 1a [13,14] and 1b [15,16] respectively, and series-produced one by the Bosch company.

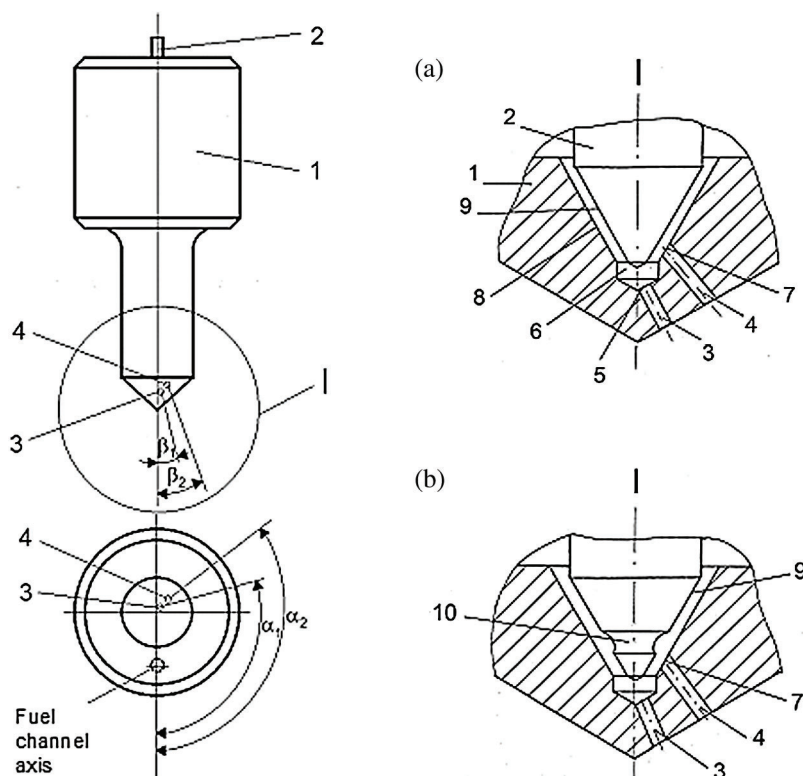


Figure 1: Schemes of experimental injector nozzles

All three injector nozzles have two injector holes (Fig. 1). The following symbols are entered in Fig. 1: a—injector nozzles No. 1; b—injector nozzles No. 2; 1—body; 2—needle valve; 3, 4—atomizing holes; 5, 7—entry edges of atomizing holes; 6—sack volume; 8, 9—locking cones of the body and needle valve; 10—circular groove on the needle.

The difference between the abovementioned injector nozzle designs consists of the location of the holes 3 and 4 at different levels. The entry edges 5 of the hole 3 are in the sack volume and entry edges 7 of the hole 4 are on the locking cone. At the same time, in contrast to the injector nozzle No. 1, the design of the injector nozzle No. 2 includes a circular groove 10 on the needle valve 9 (Fig. 1b). The Bosch needle valve has an additional cone with the basal diameter d_{ac} and angle β_{ac} .

The geometrical parameters of the injector nozzles studied are presented in Tab. 1.

Table 1: Geometrical parameters of the injector nozzles studied

No	Parameter	Injector nozzle Nos. 1 and 2	Injector nozzle Bosch
1.	Diameter of the needle valve locking cone d_x (mm)	2.8	2.6
2.	Diameter of the additional cone base d_{ac} (mm)	—	1.3
3.	Angle of the additional cone of the needle valve β_{ac} (°)	—	120
4.	Diameter of the channel under the needle valve d_s (mm)	1.2	1.5
5.	Diameter of the injector hole d_c (mm)		0.45
6.	Diameter of the injector hole with entry edges in the sack volume d_{cs} (mm)	0.43	—
7.	Diameter of the injector hole with entry edges on the locking cone d_{cc} (mm)	0.43	—
8.	Angle α_1 formed by projections on the plane of the injector nozzle connector of the planes passing through the 1 st atomizing hole and fuel channel (Fig. 1) (°)	138	134
9.	Angle α_2 formed by projections on the plane of the injector nozzle connector of the planes passing through the 2 nd injector hole and fuel channel (Fig. 1) (°)	135	137
10.	Angle β_1 characterizing deviation of the axis of the 1 st atomizing hole from the injector axis (Fig. 1) (°)	19	21
11.	Angle β_2 characterizing deviation of the axis of the 2 nd atomizing hole from the injector axis (Fig. 1) (°)	30	30

3 Test Equipment End Research Methods

Fuel injection intensity is defined by a differential injection characteristic which is understood as a dependence of mass fuel supply speed ΔG on time τ or rotation angle φ_c of the camshaft of the HP fuel pump pumping fuel into the fuel line.

Injection characteristics and fuel atomization fineness were determined on a mechanical slot type stroboscope shown in Fig. 2: 1—cycles counter; 2—tachometer; 3—lamp; 4—disc; 5—photo element; 6—HP fuel pump camshaft speed sensor; 7—electric motor; 8—gearbox; 9—coupling; 10—HP fuel pump; 11—fuel feed pump 12—fuel filter; 13, 14, 17, 19, 20, 35—fuel lines; 15—fuel tank; 16—bed plate; 18—fuel discharge reservoir; 21—measuring reservoir; 22—shutoff device; 23—foam suppressor;

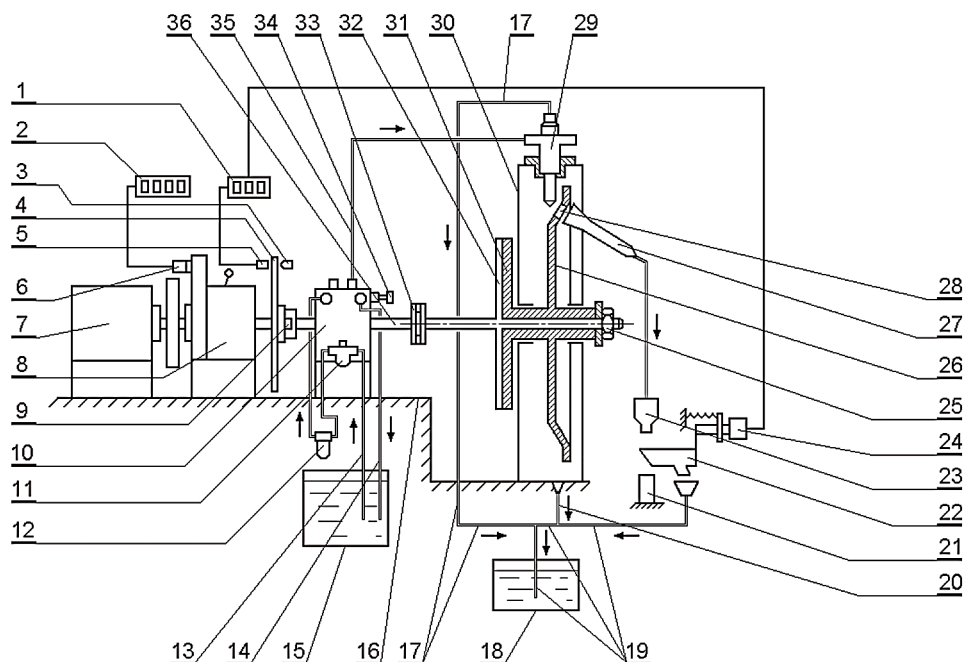


Figure 2: Layout of the test setup for fuel injection processes study

24—electromagnet; 25—stroboscope shaft fastening nut; 26—stroboscope disc; 27—receiver tube; 28—measuring slot; 29—injector; 30—stroboscope body; 31, 32—discs; 33—coupling; 34—fuel rate control rack of the HP fuel pump; 36—toe of the HP fuel pump camshaft.

This method is well known and is widely used in scientific research and the high school educational process [17,18]. The size of the slot 28 of the stroboscope disc 26 was 1° of rotation of the camshaft of the HP fuel pump 10 which enabled to estimate the fuel consumption during this period. Differential characteristic of fuel injection by the injector 29 was obtained by varying position of the slot 28 of the stroboscope disc 26 by φ_c of the shaft 36 of the HP fuel pump.

A part of the fuel mixture supplied by the injector is shut off by the slot 28 in the rotating disc 26. A shutoff portion of the fuel mixture injected gets into the receiver tube 27, foam suppressor 23 and shutoff device 22. The mixture which was shut off runs by the fuel line 20 connected with the fuel line 19 into a special tank 18. Another part of the injected mixture which had not got into the grade slot 28 also flows from the stroboscope body to the tank 18 by the fuel line 20 connected with the fuel line 19.

The number of cycles of the HP fuel pump 10 operations during which the shutoff device 22 is drawn aside by the electromagnet 24 is set with the aid of the cycles counter 1 mounted on the control board. In this way, the portions of the fuel mixture which had passed through the grade slot 28 get into the measuring reservoir 21 rather than into the tank 18.

The measuring reservoir 21 was weighted on the analytical balance ZMP, model WA-21 (not shown in Fig. 2), before and after its mounting under the cutoff device 22. The difference in weight corresponded to the weight of the mixture which had passed through the grade slot 28 during the number of cycles that was set. The error of measuring the weight on this balance did not exceed 0.1%.

The fuel atomization fineness was estimated by a medium-volume diameter of the fuel drop using a well-known method [19,20].

Hydraulic characteristics of the atomizing holes of the injector nozzle were determined at the stationary flow of fuel on the experimental setup presented in Fig. 3 [21]: 1—electric motor driving the fuel supply pump; 2—electric motor driving the HP fuel pump; 3—fuel supply pump; 4—HP fuel pump; 5—block of fuel fine filters; 6—pointer pressure gauges; 7—high volume common rail; 8—small common rail; 9—bypassed fuel volume control valve; 10—throttling valve of injection pressure fine adjustment; 11—injector body with nozzle and needle valve lift indicator; 12—chamber; 13—fuel draw backpressure control valve; 14—throttle valve with electric drive; 15—measuring reservoir; 16—fuel tank; 17—heat exchanger for cooling fuel.

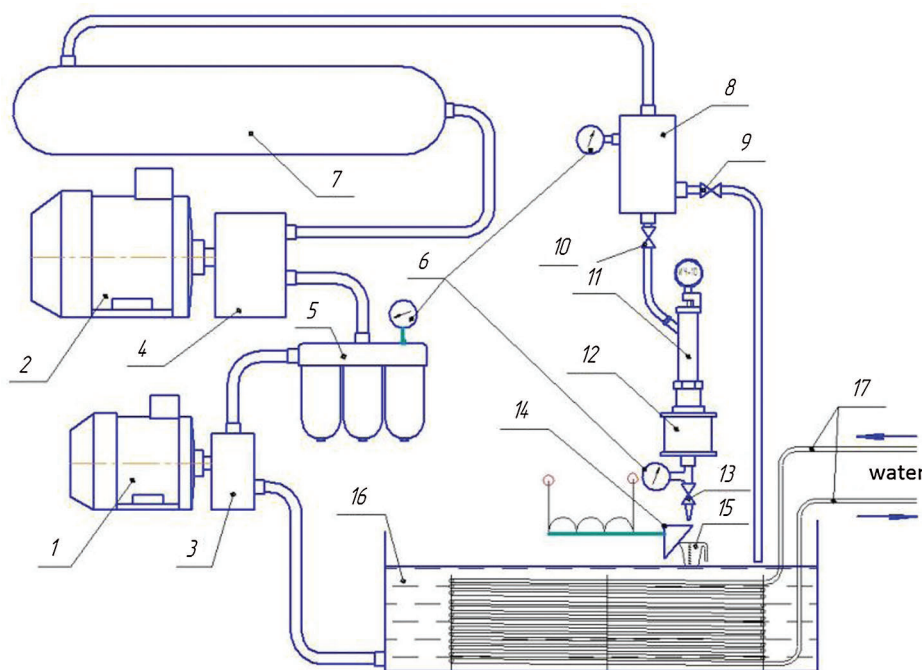


Figure 3: Layout of the experimental setup for obtaining hydraulic characteristics of injectors

The setup works in the following way. Fuel from the tank 16 is supplied by the fuel supply pump 3 to the HP fuel pump 4 via the cascade of fine filters 5. To smooth fuel pressure pulsations, the fuel from the HP fuel pump having a high pressure is supplied via the large volume common rail 7 to the small volume common rail 8 on which the control valves 9 and 10 are mounted. The required injection pressure is attained by variation of the passage area of the bypass valve 9 by changing the amount of fuel drained from the common rail 8. The fuel is supplied through the valve 10 to the injector body with the injector nozzle investigated. Under pressure, the needle valve of the injector nozzle lifts and the fuel enters the chamber 12. The needle valve lift is set by the rest (not shown in Fig. 3). The pressure in the chamber 12 is controlled by the valve 13. The backpressure value is controlled by the needle gauge. The fuel from the chamber 12 and common rail 8 is drained back to the tank. A heat exchanger is mounted in the fuel tank. The heat exchanger is cooled by the flowing water 17 which returns to the waterworks.

An automatic system of measuring the amount of fuel passing through the injector nozzle in unit time is mounted on the stand. When the timer is started (not shown in Fig. 3), the throttle 14 is moved by the electromagnet to the left end position ensuring the entry of the fuel to the measuring reservoir 15. After termination of the timer work, the electromagnet of the throttle is turned off and it returns to the right end position redirecting the flow of fuel from chamber 12 to the fuel tank 16.

Experimental research of the atomized fuel spray propagation was carried out on the setup with static environmental parameters (Fig. 4). The following symbols are entered in Fig. 4: 1—bedplate; 2—chamber body; 3—upper body cover; 4—lower body cover; 5—pressing ring; 6—threaded pressing ring; 7—plate; 8—body cover gasket; 9—front glass; 10—rear glass; 11—plate gasket; 12—bedplate; 13—laptop; 14—USB cable (mini USB); 15—photo camera with tripod; 16—halogen lamp; 17—connector.

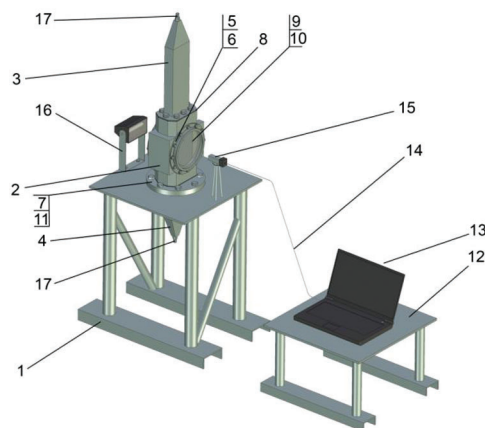


Figure 4: Setup for registration of injected fuel sprays

The setup (Fig. 4) has a closed volume 2 with two flat and parallel transparent walls 9, 10. The working body of the modeling setup is nitrogen at temperature 20°C and pressure 20 MPa. The nitrogen gas cylinder is connected to the coupling 17. In this way, the density of the medium into which the fuel was injected corresponded approximately to the air charge density in the diesel engine combustion chamber at the compression endpoint.

In the chamber body 2, the injector nozzle studied is mounted by connecting the high-pressure fuel line with the testing setup (not shown in Fig. 4) on which the fuel equipment for supplying the fuel into the injector is mounted (not shown in Fig. 4).

For studying the propagation of the atomized fuel sprays, a high-speed digital camera with the shooting speed up to 20000 frames/s was used.

4 Results of Investigation of Fuel Injection Characteristics and Fuel Atomization Fineness

The work program included obtaining injection characteristics and speed characteristics of fuel supply and also envisaged estimation of fuel distribution by atomizing holes depending on the injector nozzle operation mode.

The first stage of testing the experimental injector nozzles (Fig. 1) envisaged checking the possibility of redistribution of masses of fuel by the atomizing holes 3 and 4.

Hydraulic characteristics of the experimental injector nozzle No. 1 and injector nozzle Bosch obtained at the test setup (Fig. 3) are presented in Fig. 5. As seen, the effective area of passage $\mu_c f_c$ of the injector nozzle No. 1 is a bit lower than that of the Bosch injector nozzle at approximately the same lift of the needle valve.

Fuel supply speed characteristics through the injector nozzles studied (Fig. 6) were obtained on the stand shown in Fig. 2 using a commonly used method. The injector nozzles Nos. 1 and 2 were mounted in turn into the injector 29 which was supplied with the fuel by the Bosch model PE80D421 HP pump 10 via the HP fuel line 35 having 450 mm in length and internal diameter 1.6 mm.

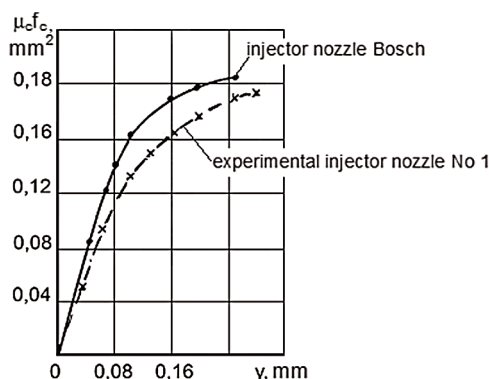


Figure 5: Hydraulic characteristics of the injector nozzles

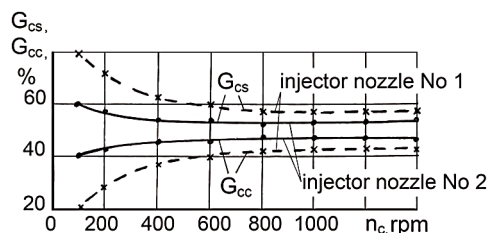


Figure 6: Fuel supply speed characteristics

The tests were carried out at a fixed position of the control element 34 of the HP fuel pump 10 ensuring constant fuel rate $G_f = 47$ mg at HP fuel pump camshaft speed $n_c = 1400$ rpm. At $n_c = 1000$ rpm and $n_c = 600$ rpm, the injector nozzle No. 1 provides $G_f = 46.0$ mg and $G_f = 45.9$ mg respectively.

In Fig. 6, fuel supply speed characteristics, G_{cs} and G_{cc} , vs. HP fuel pump shaft speeds are presented, where G_{cs} is the amount of fuel supplied through the atomizing holes 3 (Fig. 1) with entering edges in the sack volume, and G_{cc} is the amount of fuel supplied through the atomizing holes 4 (Fig. 1) with entering edges on the locking cone. The total fuel rate G_f is the sum of G_{cs} and G_{cc} .

A specific feature of the speed characteristics obtained is that with decreasing the HP fuel pump shaft speed (decreasing the dynamics of changing pressure variation in the fuel line and injector channels during fuel injection), one can see fuel redistribution by the atomizing holes. So, at the speed $n_c = 1400$ rpm, for the injector nozzle No. 1, 55% of G_f is supplied through the hole 3 (Fig. 1a), as shown in Fig. 6. When the n_c value decreases to 600 rpm, a portion of fuel supplied through this hole increases to 60%. If n_c is further decreased, the dependences $G_{cs} = f(n_c)$ and $G_{cc} = f(n_c)$ change more.

The experimental dependencies $G_{cs} = f(n_c)$ and $G_{cc} = f(n_c)$ confirm the assumptions used in the design of the injector nozzles Nos. 1 and 2 and the way of fuel distribution by the combustion chamber areas due to its supply through the channels with different geometrical parameters and locations on the injector nozzle tip (position I in Fig. 1). In particular, a higher share of fuel rate is supplied through the holes 3 (Fig. 1) in all operation modes of the fuel supply system, therefore it is reasonable to direct this hole to the most remote (from the injector nozzle tip) wall of the diesel engine combustion chamber.

A specific feature of the shape of the fuel supply speed characteristic of the injector nozzle No. 1 is the fact that in engine start mode ($n_c = 100$ rpm), the basic mass of fuel rate is injected through the hole with its intake edges in the sack volume (3, Fig. 1a). In this way, if this hole is directed not only to the most remote but also to the most heated area of the combustion chamber (in the engine start mode), there will be conditions

not only for a more efficient use of air in the combustion chamber during the working cycle but also for improving ignition of the air-fuel mixture in the diesel engine start mode.

The results obtained (Fig. 6) are explained by different hydraulic characteristics of the atomizing holes. At maximal needle valve lifts, the flow coefficients of the atomizing holes 3 (μ_{cs}) (Fig. 1) and 4 (μ_{cc}) differ approximately by 10% and, at partial needle valve lifts, they differ by 2 ... 3 times. Fuel injection in the engine start mode takes place at $y < 0.15$ mm [22], therefore, in this mode, the main portion of the fuel is supplied through the holes of the first group.

As follows from comparison of the speed characteristics (Fig. 6) of the injector nozzle No. 2 with those of the injector nozzle No. 1, a groove 10 (Fig. 1) made on the needle valve changes considerably their shape, i.e., they become more linear, which is explained by the modification of the coefficients μ_{cs} and μ_{cc} .

The results of studies of the fuel supply system equipped with the injector nozzle No. 1 are presented in Figs. 7 and 8. In these figures, ΔG_s is the mass fuel supply speed through the atomizing holes 3 (Fig. 1a) and ΔG_c is that through the hole 4. The characteristics were obtained on the experimental setup shown in Fig. 2.

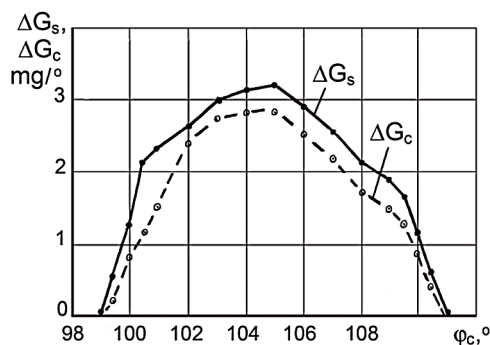


Figure 7: Injection characteristics: $n_c = 1400$ rpm, $G_f = 48$ mg

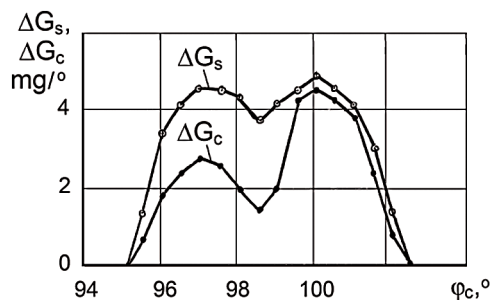


Figure 8: Injection characteristics: $n_c = 600$ rpm, $G_f = 46$ mg

Analysis of the injection characteristics measured for each of the atomizing holes of the experimental injector nozzle demonstrates a considerable difference in dynamics of fuel entry into the combustion chamber and in mass fuel supplies (Figs. 7 and 8). At that, the main difference was observed at partial values of y when the coefficients μ_{cs} and μ_{cc} differ considerably.

In this way, in the mode $n_c = 1400$ rpm (Fig. 7), at the injection start, when the needle valve moves from the seat to the rest, 1, 5 ... 2 times more fuel is supplied through the hole with μ_{cs} than through the hole with μ_{cc} . At that, a high average speed of fuel supply rise to the combustion chamber is observed (approximately 1.5 times) at the injection start (during the first 1.5 ... 2° of the HP fuel pump shaft rotation). Later the speed

of fuel supply rise from the hole with μ_{cc} is higher than that from the hole with μ_{cs} . As a result, the mass fuel supplies become equal and at $\varphi_c > 102^\circ$, i.e., when the needle valve reaches the rest, the difference between ΔG_s and ΔG_c does not exceed 10%. During the period of moving of the needle valve to the seat, also more fuel is supplied through the hole with μ_{cs} than through the hole with μ_{cc} . At that, a bit higher speed of decreasing ΔG_s is observed.

The dependencies mentioned are typical also for other modes studied of the FSS equipped with the injector nozzle No. 1.

It's important to note that at $n_c = 600$ rpm, a considerable difference in the values of ΔG_s и ΔG_c in the middle of the injection process was registered (Fig. 8). In this way, for the hole with μ_{cc} , the characteristic has a clearly seen "dip" in fuel supply at $\varphi_c = 98.5^\circ$, which was not observed in such an extent for the hole with μ_{cs} . This can be explained by the following. In this FSS operation mode at $\varphi_c = 98 \dots 99^\circ$, the needle valve moves from the rest to the seat which results in the variation of the flow rate of the hole with μ_{cc} to a greater extent than the hole with μ_{cs} .

Examination of the fuel atomization fineness was carried out at the stand presented in Fig. 2 when the injector 29 (Fig. 2) was equipped alternatively with the injector nozzle Bosch No. 8 and injector nozzle No. 1. The tests were performed at a fixed position of the rack of the HP fuel pump and envisaged obtaining of the injection characteristics $\Delta G = f(\varphi_c)$ and atomization fineness. The atomization fineness was estimated by the average volumetric drop diameter d_{30} .

The results of comparative tests of fuel atomization fineness ensured by the injector nozzle No 1 and injector nozzle Bosch are presented in Fig. 9. Testing conditions are the following, $n_c = 1000$ rpm and $G_f = 48$ mg.

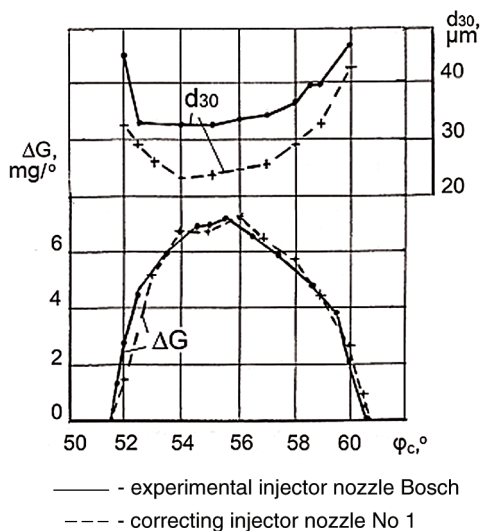


Figure 9: Injection characteristics: $n_c = 1000$ rpm, $G_f = 46$ mg

The tests demonstrated that the injector nozzle No. 1 ensured more tiny atomization at all the instants of the injection process. In the mode $n_c = 1000$ rpm during injection of the main portion of fuel (when the needle valve was placed on the rest), d_{30} was 25 ... 30 μm , which was by 25 ... 30% lower than in case of the injector nozzle Bosch.

Fig. 10 shows the results of fuel injection through the injector nozzle No. 1 and injector nozzle Bosch in the mode $n_c = 100$ rpm, $G_f = 98$ mg.

It is seen that the injector nozzle design (injector nozzle Bosch or injector nozzle No. 1) practically has little influence on the general shape of the injecting characteristic, i.e., on the behavior of the mass supply per 1° of the pump rotation, as well as on the maximal and minimal values of ΔG . At the same time, the rate of the fuel supply rise at the fuel injection start ($\varphi_c = 351 \dots 349^\circ$) when testing the injector nozzle No. 1 was lower. During the tests, the fuel consumption was measured through two holes simultaneously. In reality, the injector nozzle No. 1 makes it possible to ensure a considerable redistribution of fuel supplied through each hole which follows in particular from Fig. 6.

It should be noted that in the engine start modes ($n_c = 100$ rpm, $G_f = 98$ mg), the needle valve of the injector nozzle performs high-frequency oscillations with frequency attaining $1000 \dots 1500$ Hz. This phenomenon was studied in papers [22,23]. The resolution power of the stroboscope plant (Fig. 2) does not allow us to estimate the variation of flow ΔG stipulated only by the high-frequency needle valve oscillations. Therefore, the injection characteristic presented in Fig. 10 is the result of the joint influence of the volumetric balance at the injector which depends on fuel supply by the HP fuel pump and high-frequency oscillations of the needle valve.

Comparison of the fuel atomization fineness by the injector nozzles No. 1 and Bosch (Fig. 10) shows that the injector nozzle No. 1 improves the fuel atomization fineness in the first half of the injection process (at φ_c from 351 to 347°) when the most considerable changes of the ΔG are observed, which are stipulated by the wave propagation processes in the high-pressure fuel line, as well as in the needle valve spring.

So, the injector nozzle No. 1 compared to the injector nozzle Bosch, ensures the improvement of fuel atomization fineness the FSS operation mode studied at $n_c = 1000$ rpm and in the main phase of the diesel engine start process.

5 Results of the Study of the Atomized Fuel Spray Propagation

For registration of the sprays of the injected fuel, an experimental setup shown in Fig. 2 was used. The components of the FSS mounted on it is described in the previous section of the paper. Injector 29 (Fig. 2) with the injector nozzle No. 1 was mounted into the body of the special chamber 2 (Fig. 4).

The injector nozzle No. 1 has directions of the sprays ($\alpha_1, \alpha_2, \beta_1$ и β_2) which do not differ considerably from the Bosch injector nozzle (Tab. 1). The design of both the injector nozzles makes it possible to trace the movement of the front edge of each spray of the atomized fuel. But it is difficult to estimate correctly the width of the spray and its cone. For this, larger changes in the difference of the angles β_1 и β_2 are required. These angles define the direction of the sprays relative to the injector axis. In this way, the Bosch injector nozzle ensures practically one spray formed by the merging of two flows.

The following parameters of the sprays were examined: L_{sps} , C_{sps} are distance traveled and the front edge speed for the atomizing hole 3 respectively (Fig. 1a), and L_{spc} , C_{spc} are distance traveled and the front edge speed for the atomizing hole 4 respectively (Fig. 1a). At that, the speeds C_{sps} and C_{spc} were determined using the numerical differentiation of the travel curve.

The results of the experimental studies of the parameters of the fuel sprays running from the injector nozzle No. 1 are presented in Figs. 11 and 12.

It is seen from the results obtained that the length of the fuel spray injected from the atomizing hole with μ_{cs} , is larger in all modes of the FSS operation studied. In that way, during the period $\tau = 1.0$ ms and the mode $n = 1400$ rpm, $G_f = 48$ mg (Fig. 11), the front edge of the hole with μ_{cs} passes the distance of 64 mm and for the fuel spray from the atomizing hole with μ_{cc} , this value is 58 mm, i.e., by 10% lower. At the same time, at $n_c = 600$ rpm, the lengths of the fuel sprays studied differ by 19% (Fig. 12).

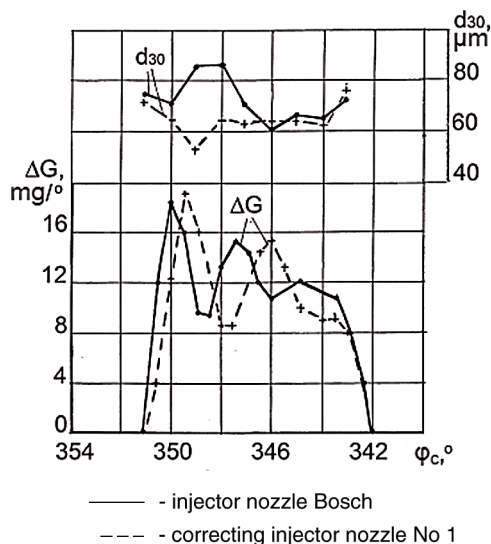


Figure 10: Injection characteristics: $n_c = 100$ rpm, $G_f = 98$ mg

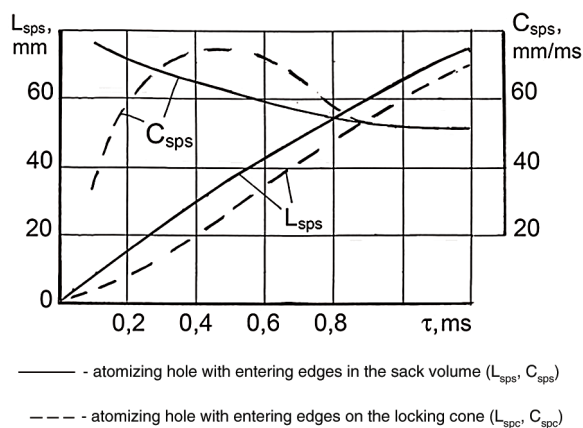


Figure 11: Parameters of the fuel spray: $n = 1400$ rpm, $G_f = 48$ mg

The results presented (Figs. 11 and 12), as well as the analysis of video registration, show that both the fuel sprays start their movement practically at the same time but with considerably different speeds. So, in the mode $n = 1400$ rpm, $G_f = 48$ mg, at the injection start ($\tau = 0.1$ ms), the average speed C_{sps} exceeds the C_{spc} value approximately 2.3 times. Further, the speeds become equal and at $\tau > 0.8$ ms, the front edges of the fuel sprays injected from both the atomizing holes move with practically equal speeds. A similar picture took place also in other operation modes of the FSS equipped with the injector nozzle No. 1.

Dependencies presented in Figs. 11 and 12 are defined by the design of the injector nozzle and correspond to the specific features of the shape of the fuel injection characteristics. As mentioned above, at the injection process start (at $\tau = 0 \dots 0.3$ ms), the mass fuel supplies through the atomizing hole with the entering edges in the sack volume exceed the mass fuel supplies through the atomizing hole with the entering edges on the locking cone (Figs. 7 and 8). Further, in the time interval $\tau = 0.3 \dots 0.8$ ms, $C_{spc} > C_{sps}$ (Fig. 11). The difference reaches 20 ... 25%. This is explained by the higher rate of fuel supply rise through the atomizing hole with μ_{cs} at $\varphi_c = 100.5 \dots 102^\circ$ (Fig. 7) which was observed when obtaining the injection characteristics at the stroboscope plant (Fig. 2).

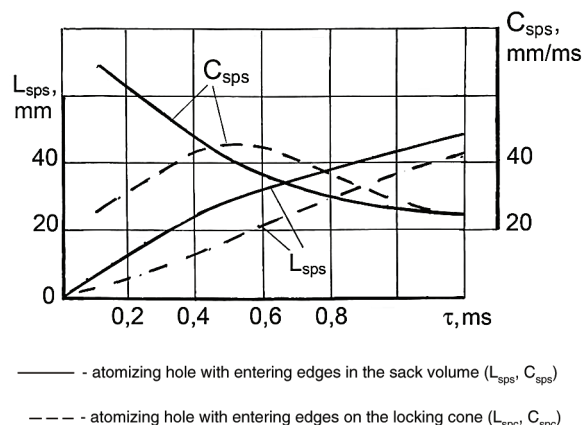


Figure 12: Parameters of the fuel spray: $n = 600$ rpm, $G_f = 46$ mg

The results obtained are a special case of a specific design in a particular operating mode of the FSS. At the same time, the direct link of the injection characteristics with the length of the injected spray and its speed is evident. So, these results agree well with the known provisions and correlations of fuel supply and atomization in diesel engines [19,20].

It should be also mentioned that the derating of the injection process by the injection pressure (decreasing n_c from 1400 to 600 rpm) results in increasing the difference of the length of the sprays supplied from the holes with μ_{cs} and μ_{cc} . This is explained by the following. When the injection pressure is decreased, the time of the needle valve movement from the seat to the rest increases, i.e., the time influence of the y on the μ_c and on the fuel flow at the injection start is higher. Dynamics of fuel supply and injection rates also predetermine the maximal values of the L_{sp} .

In this way, experimental injector nozzles Nos. 1 and 2 (Fig. 1) enable to carry out a relative correction of propagation of the injected fuel sprays directed to the different areas of the diesel engine combustion chamber. This enables to control the processes individually by several areas of the diesel engine combustion chamber and thus improve its fuel efficiency and ecological parameters.

6 Conclusions

1. A possibility of the mass fuel rate redistribution between the atomizing holes of one injector nozzle (hole No. 1 in the sack volume and hole No. 2 on the locking cone) whose effective flow passage depends on the location of their intake edges was confirmed.
2. Dynamics of pressure variation in the fuel line and injector channels during fuel injection have an influence on the fuel redistribution by the atomizing holes of the injector nozzle. In this way, in the mode $n_c = 1400$ rpm, 55% of the total mass of the fuel G_f is supplied through the hole No. 1. As the n_c value decreases to 600 rpm, the share of the fuel supplied through this hole increases to 60%.
3. Analysis of the video registration shows that the fuel sprays (from the hole No. 1 in the sack volume and the hole No. 2 on the locking cone) start their movement practically at the same time but with considerably different speeds. So, at the fuel supply start, the average spray speed from the hole No. 1 exceeds the spray speed from the hole No. 2 more than twice. Further, the speeds of the sprays become equal and by the end of the injection process, the front edges of the fuel sprays injected from both the atomizing holes move with practically equal speeds.
4. Experimental injector nozzles with the holes Nos. 1 and 2 enable to perform the relative correction of propagation of the atomized fuel sprays directed to the different areas of the diesel engine combustion

chamber. This makes it possible to control the processes separately by several areas of the diesel engine combustion chamber and thus improve its fuel efficiency and ecological parameters.

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