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Diagnostics and Regulation of Fuel Equipment of Diesels on Stands with Injection to Medium with Counter-Pressure

Ildar Gabitov, Samat Insafuddinov, Denis Kharisov, Filyus Safin andrei Negovora,
Nail Yunusbaev, Alfir Akhmetov, Timur Farhutdinov and Albert Sharafeev
Federal State Budgetary Educational Establishment of Higher Education,
Bashkir State Agrarian University, 50-Letia Octyabrya Str., 34, Ufa, Russia

Abstract: The purpose of the study is to substantiate the possible design of a counter-pressure device for fuel injection, intended for modernization of adjusting stands of diesel fuel injection equipment. The factors influencing the fuel supply parameters are analyzed and, taking them into account, the design and method for calculating the counter-pressure device for injection and regulation with its use of fuel injection equipment are developed. The distinctive feature of the proposed device is that the injection of fuel comes into the chamber where the counter-pressure is created by the injected fuel itself and the adjusting parameters of the fuel supply are calculated from the pressure in the injection chamber using a computer and the electronic unit operating under the proposed program.

Key words: Injection, indication, gas pressure, compression, expansion, program

INTRODUCTION

The power, economic and environmental performance of the diesel engines is determined primarily by the differential characteristic (the fuel supply law) of their Fuel Injection Equipment (FIE), estimated by the angles of the start of the High-Pressure Pump (HPP) and injection timing, cyclic fuel supply, pressure and duration of injection, etc.

When adjusting fuel systems, the fuel supply law is corrected by affecting the start of delivery, the injection start pressure and the cyclic delivery.

The fuel injection start pressure depends on (among other things) the counter-pressure of the injection and is simply controlled by the tightening of the needle spring of the atomizer. There are no difficulties when adjusting the timing of the beginning of the injection by the HPP and advancing the injection. They appear in the regulation cyclic fuel supply, especially in systems with centrifugal regulators.

On the adjusting stands, the cyclic supply is usually estimated experimentally by the measuring method (Marcic, 2006; Anonymous, 2002, 2006). Other methods can also, be used, for example, based on the dynamic effect of the leaking out stream of fuel on the membrane, the injection of fuel into a closed space, the calculation of the pressure in the fuel injection line at the injector and the electric charge that occurs in the fuel pipe due to the movement of fuel, etc. It can be estimated by calculation methods (Anonymous, 1974). Calculation methods are based on determining the cyclic supply rate by integrating the fuel supply law (Safin *et al*., 2014):

$$g_\mu = \int_{\varphi1}^{\varphi2} f(\varphi).d\varphi \quad (1)$$

Where:

g_μ = The value of the cyclic fuel supply
φ_1 И φ_2 = The angles of the beginning and end of the injection
$f(\varphi)$ = The fuel supply law

In semi-calculating methods, the fuel supply law is usually determined from the volumetric rate of fuel outflow from nozzles of the nozzle atomizer (Didmanidze, 2017):

$$V_t = \frac{\mu_p.f_p.\sqrt{2.(p\phi\text{-}p\Pi)/\rho_T}}{6.n_H} \quad (2)$$

Where:

$\mu_p f_p$ = The effective cross-section of the atomizer
ρ_T = The density of fuel
$p\phi$ = The injection pressure (under the spray cone)
p_n = The counter-pressure to the injection (pressure of the medium into which the fuel is injected)
n_H = The frequency of rotation of a cam shaft of the High Pressure Fuel Pump (HPFP)

Corresponding Author: Ildar Gabitov, Federal State Budgetary Educational Establishment of Higher Education, Bashkir State Agrarian University, 50-Letia Octyabrya Str., 34, Ufa, Russia

To measure the pressure under the needle cone in the formula, custom-made nozzles are usually used. Investigation wiht this method of pin nebulizers is not carried out due to the difficulty of measuring the pressure under the needle cone and the dependence of the effective cross section of the nozzle opening from the raising of the injector's needle.

Calculations using this formula can also, be carried out from the pressure in the discharge pipe at the injector nozzle (Safin *et al*., 2017). However, in this case, there are significant errors due to the fact that the nature of the pressure change at the nozzle differs from that under the needle cone of the atomizer in addition to the complexity of the measurements and the great complexity of processing their results.

From these expressions it follows that the value of the cyclic supply also, depends on the counter-pressure to the injection which is determined by the indicator diagram of the engine.

The disadvantage of modern adjustment stands is that fuel injection is performed without taking this circumstance into account in a medium with atmospheric pressure. Because of this discrepancy when installing the FIE on the engine, its cyclic fuel supply is reduced. That is why FIE is adjusted on the stand for supply, increased by 10-25%. However, it does not completely solve the problem. This is explained by the fact that due to the hydraulic non-identity of the FIE sections, the considered reduction in the cyclic supply rate is not the same in the engine cylinders and causes a significant increase in the unevenness of the fuel supply and as a consequence, makes it difficult to achieve the proper technical and economic performance of the engine (Didmanidze, 2017). This problem can be solved only by creating counter-pressure to fuel injection on the adjustment stands which changes in a manner analogous to the pressure of gases in the engine cylinder.

The firm "R. Bosch" proposed a technique that provides creating counter-pressure to the injection while adjusting FIE (Bosch, 1964). However, the design of the counter-pressure device used for this is not yet fully justified.

The purpose of the study is to study the effect of counter-pressure to injection on cyclic fuel supply and, on the basis of the data obtained, refine the design of the device for creating counter-pressure to injection and develop a methodology for adjusting fuel equipment on stands upgraded using this device.

MATERIALS AND METHODS

An experimental device for studying the effect of counter-pressure to injection was developed using a hydromechanical accumulator (Gabitor *et al*., 2008; Bosch, 1964) which is a closed space in which the injection counter-pressure is created by the injected fuel itself.

The counter-pressure fuel injection in it is estimated by the ratio proposed by us called "accumulator stiffness":

$$C_{ak} = \frac{\Delta P}{\Delta V} \quad (3)$$

Where:

Δp = The pressure increase

ΔV = The amount of fuel delivered during fuel injection

The stiffness of the accumulator should be such that the pressure in it varies like the gas pressure in the engine cylinder (Anonymous, 1984, 1985, 2014). The accumulator can be mechanical two-spring and combined spring-hydraulic (Table 1).

With mechanical accumulators, the pressure analogous to the pressure in the cylinder in the first

Table 1: Possible types of combined hydromechanical fuel accumulators and their stiffness

Scheme		
Of the springs work	Of the accumulator	Stiffness C_{ak}, Pa/mm^3
Gradual-parallel work of mechanic springs		By the moment of gap X_1 selection $\frac{1}{\beta \cdot \left(V_{aK} + \frac{\pi^2 \cdot d_\pi^4 \cdot \Delta p_1}{16 \cdot c_{M1}}\right) + \frac{\pi^2 \cdot d_\pi^4}{16 \cdot c_{M1}}}$ By the end of the course after the gapX_1 selection $\frac{1}{\left(\beta \cdot \left(V_{aK} + \frac{\pi^2 \cdot d_\pi^4 \cdot \Delta p_2}{16 \cdot (c_{M1} + c_{M2})}\right) + \frac{\pi^2 \cdot d_\pi^4}{16 \cdot (c_{M1} + c_{M2})}\right)}$

Table 1: Continue

Scheme: Of the springs work	Scheme: Of the accumulator	Stiffness C_{ak}, (Pa/mm^3)
Gradual work of mechanic springs		By the moment of gap X_1 selection $\dfrac{1}{\beta\cdot\left(V_{aK}+\dfrac{\pi^2\cdot d_\pi^4\cdot\Delta p_1}{16\cdot c_{M1}}\right)+\dfrac{\pi^2\cdot d_\pi^4}{16\cdot c_{M1}}}$
		By the end of the course after the gap X_1 selection $\dfrac{1}{\beta\cdot\left(V_{aK}+\dfrac{\pi^2\cdot d_\pi^4\cdot\Delta p_2}{16\cdot c_{M2}}\right)+\dfrac{\pi^2\cdot d_\pi^4}{16\cdot c_{M2}}}$
Parallel work of hydraulic and mechanic springs		$\dfrac{1}{\beta\cdot\left(V_{aK}+\dfrac{\pi^2\cdot d_\pi^4\cdot\Delta ?}{16\cdot c_{M}}\right)+\dfrac{\pi^2\cdot d_\pi^4}{16\cdot c_{M}}}$

The table shows: 1 and 2) The first and sec springs; 3) Piston of mechanical accumulator; 4) Injection chamber; 5) Support washer; 6, 7 and 11) The adjusting screws of the respective springs; 8 and 9) Aadjusting washers and spring 1clips; 10) One coil accumulator spring; V_{ak}) Volume (darkened) of the injection chamber (hydraulic accumulator); d_n is the piston diameter of the mechanical accumulator; c_M, c_{M1} and c_{M2}- the rigidity of one, the first and sec springs; β is the coefficient of the fuel compressibility; X_1-the course of the spring 1 before the start of the spring 2; g_μ-the volume of fuel injected into the accu, ulator chamber; Δp_1 and Δp_2-increase of pressure at work of the first spring and at joint work of the first and sec springs

period of the combustion process is created by the first spring and in the sec-by the sec. The amount of fuel in the accumulator should be negligible here.

RESULTS AND DISCUSSION

The results of the research conducted by us showed that the volume of fuel in the injection chamber (V_{ak}) can not be neglected because, firstly, it is significant initially (due to the gaps between the battery case and a number of its parts-the nozzle atomizer, piston, pressure sensor, etc.) and, secondly, increases as fuel is supplied the injection process (moving the piston).

This space is filled with fuel, because of the compressibility of the fuel in it, this will accumulate part of the injected fuel.

Thus, it turns out that the fuel in the injection chamber serves as a hydraulic accumulator and the piston loaded with a coil spring (springs)-as a mechanical one. Hydraulic accumulator in accordance with the principle of its operation of course, can be considered as a “hydraulic spring”. In view of this circumstance, the proposed batteries should be considered a combined hydromechanical one.

The stiffnesses of the proposed combined hydromechanical accumulators can be determined by the expressions given in the table.

Thee first (according to Table 1) version of the accumulator is preferred for use as an experimental versionbecause the device is simpler, the sec spring is less rigid and the adjustment of the first spring is more accurate.

The diagram of the counter-pressure device for injection with such an accumulator is shown in Fig. 1. Here, the hydraulic “spring” (accumulator) with the first mechanical spring creates pressure similar to the pressure in the cylinder during the compression of gases and from the sec-the combustion of the fuel. The initial pressure corresponding to the gas pressure at the time of the injection p_0 is ensured by constant drain of the fuel through the throttle 11 (Fig. 1) mounted on the drain line.

The required stiffnesses of the individual elements of the hydromechanical accumulator are determined by the engine’s indicator diagram (dotted lines of Fig. 2) and the FIE fuel supply law.

To simplify the calculations, the indicator diagram can be represented in the form of linearized sections 0-1 and 1-2 (solid lines) corresponding to the processes of gas compression and combustion of fuel and the fuel supply law is considered linear (that is, assume that the fuel is injected at a constant speed) . These simplifications do not substantially contradict the calculated and experimental data (Safin *et al.*, 2014).

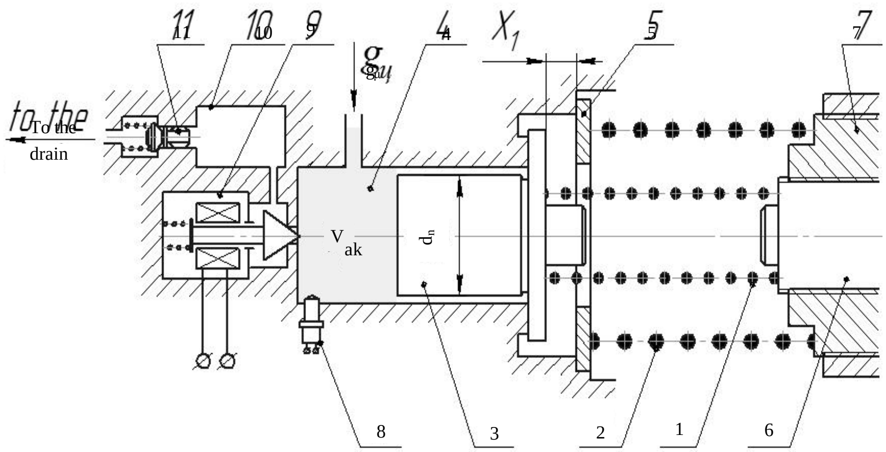

Fig. 1: Injection counter-pressure device 8-strain gauge pressure sensor of Мд series; 9-discharge solenoid valve; 10-hdroaccumulator of residual pressure; throttle; the remaining symbols are the same as in the scheme of Table 1

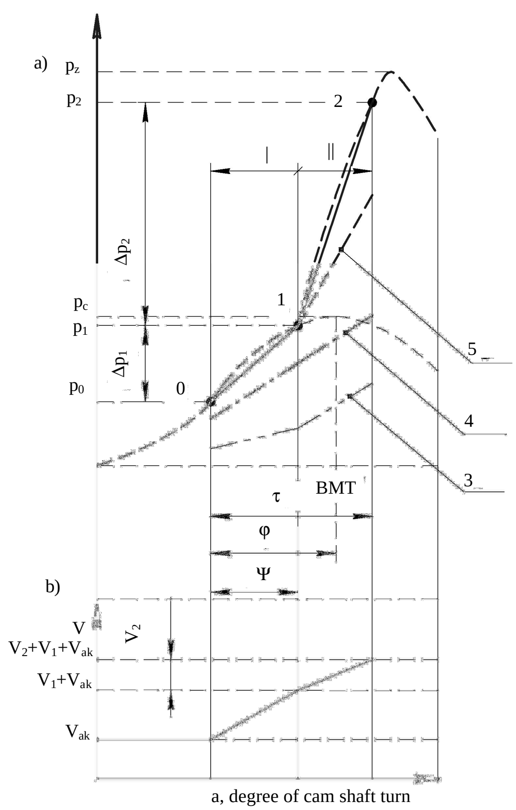

Fig. 2: a) Actual engine indicator diagram (dashed lines) and its linearized sections (solid lines) and b) Graphs of the volume change of the counter-pressure chamber V

P_c and p_z are the pressures of the gases in the cylinder at the end of their compression and the maximum; p_0 and p_2-are the pressures in the injection chamber at the moments of the beginning and end of the fuel injection; p_1 is the pressure of the start of a perceptible increase in pressure in the cylinder after autoignition of the fuel; Δp_1 and Δp_2 are the gas pressure increases corresponding to the first and second periods of the combustion process; V_1 and V_2 are the increase in the volume of the accumulator due to the fuel supplied by the injection of fuel in the first and second periods of the combustion process; φ and τ and Ψ-lead time and duration of injection and the period of fuel self-ignition delay; BMT-is the top dead point of the piston of the engine; 3-5 are the pressures developed by the "hydraulic" and the first and second mechanical springs.

The volume of fuel injected into the accumulator chamber of the injection counter-pressure device under consideration (cyclic supply rate g_μ) is distributed as follows:

$$g\mu = g\mu 1 + g\mu 2 \quad (4)$$

Where $g_{\mu 1}$ and $g_{\mu 2}$ are respectively the supplies in the first (in zone 0-1) and in the second (in zone 1-2) periods of the combustion process.

The value of cyclic supply rate $g_{\mu 1}$, determined by the dynamic factor of fuel cycle in turn determines the stroke of the piston X_1 (before its contact with the support washer 5) and $g_{\mu 2}$- its subsequent course.

Cyclical fuel supply can also, be recorded through the rigidity of the springs operating in specified periods:

$$g\mu = \frac{\Delta P_1}{C_{ak1}} + \frac{\Delta P_2}{C_{ak2}} \quad (5)$$

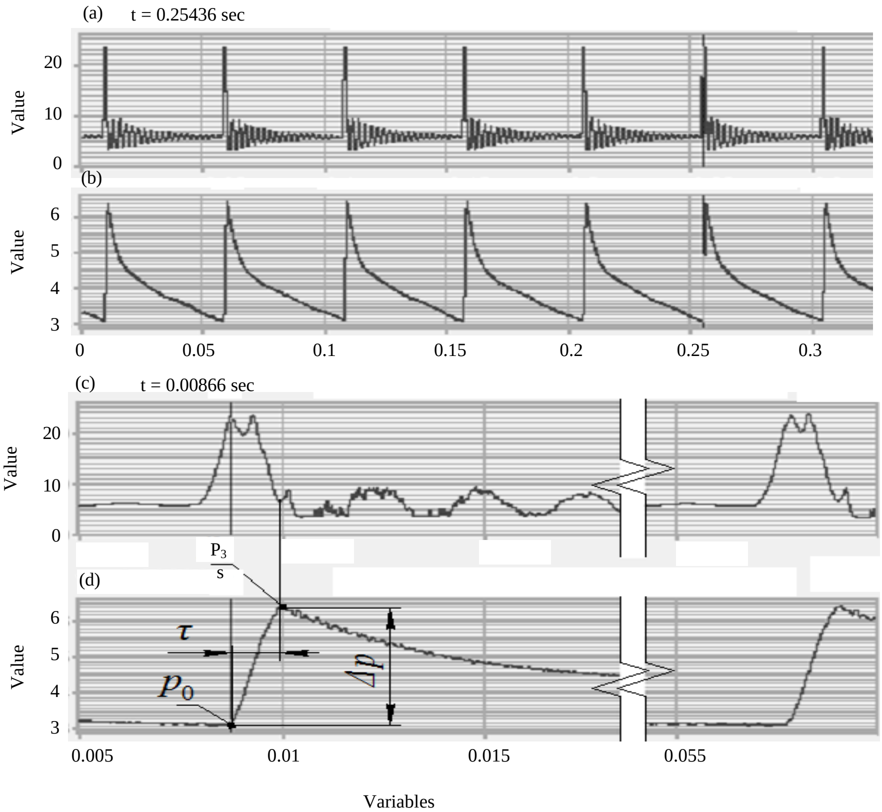

Fig. 3: Oscillograms of pressure at the nozzle; a) In the counter-pressure chamber. Pressure at the nozzle 2 = 23.563 MPa; b) The correspond-ing extended axes. Pressure at the chamer 2 = 2.999 MPa; c) Pressure at the nozzlen 2 = 23.419 Mpa and d) In the test mode p0 and p3 are the initial and final counter-pressure to injection; Δp is the counter-pressure rise value; τ- is the duration of injection. Pressure at the chamer =3.003 MPa

Where C_{ak1} and C_{ak2} are the stiffness of the accumulator in the first and sec periods of the combustion process (before selection of the gap X_1 and after its selection).

For linearized sections of the indicator diagram (Fig. 3a), the cyclic supply can be determined from the pressures at points 0, 1 and 2 by calculating them (using curves 3-5) or experimentally.

Naturally when creating the counter-pressure injection, adjustment of the FIE on the stand should be made at the rated nominal (rather than increased) cyclic supply.

This method was used to calculate the device for FIE of the Д-144 tractor diesel. At this engine, the calculated cyclic supply rate g_μ = 51 mm^3/cycle, the duration of injection is 25°cam shaft turn (c.sh.t.) and the autoignition delay is 17°C.sh.t. During the autoignition ignition delay period (in the first period), $g_{\mu 1}$ = 34.68 mm^3/cycle (68% of the entire cyclic supply, i.e., the fuel cycle dynamics factor is equal to σ = 0.68) enters the injection chamber and in the sec-the remaining part $g_{\mu 2}$ = 16.32 mm^3/cycle.

The following was determined by calculations: p_0 = 1.989 MPa; p_1 = 3.987 MPa; p_2 = 7.095 MPa; p_c = 4.127 MPa; p_z = 7.495 MPa.

The diameter of the piston of the battery is assumed d_n = 4 mm. At the same time, the initial internal volume of the accumulator, determined from design considerations, was 4.2 cm^3.

In accordance with these data and the expressions in Table 1, the first spring was made with a stiffness c_{M1} = 11,568 N/mm and the sec one was c_{M2} = 86.072 N/mm. Using the program "Kompas-Spring" of the Joint-stock company ASCON for these rigidity, cylindrical compression springs were selected, the dimensions of which are indicated in Table 2.

During the experiments, an accumulator with the specified parameters was used. The value of the cyclic supply during the experiments was controlled by the pressure in the accumulator injection chamber using a computer and an electronic unit working according to the calculation program proposed by us. The results of the experiments using the proposed accumulator are shown in Fig. 3 and 4.

Table 2: Estimated springs of the accumulator for the counter-pressure device of injection

Parameters	Springs First	Springs Second
Material	Б-2-1.0	Б-2-2.75
External diameter (mm)	5.00	11.00
Coil diameter (mm)	1.00	2.75
Number of working turns	13.00	11.50
Full number of turns	14.50	13.00
Work course (mm)	2.65	0.45
Spring length in free condition (mm)	19.31	35.52
Spring length at pre-deformation (mm)	17.19	34.94
Spring length at working deformation (mm)	14.54	34.49
Spring weight (kg)	0.001	0.016
Spring rigidity (N/mm)	11.79	86.90

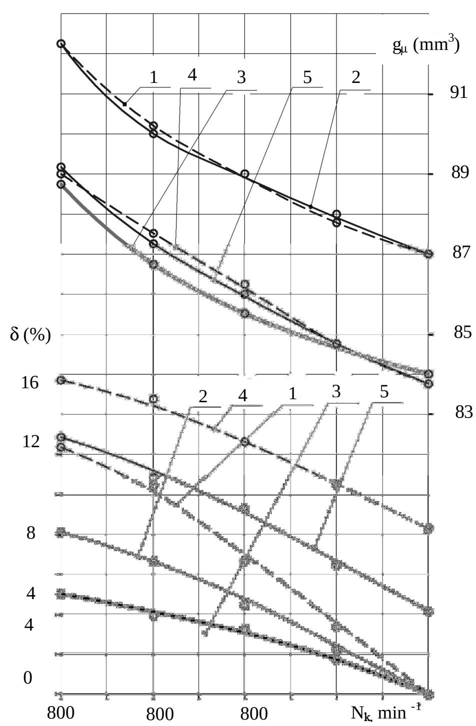

Fig. 4: The speed characteristics of g_μ and δ_c are shown in

The experiments showed that with the proposed device with counter-pressure to the injection with the accumulator with such springs in the injection chamber was actually provided with linearized sections of the indicator diagram.

From the oscillograms of Fig. 3, it follows that after the injection, the pressure in the accumulator gradually decreased (because of the received constant throttling drain of fuel from the accumulator). Acceleration of the unloading of the accumulator could be achieved by the increase in the diameter of the flow section of the throttle 11 (Fig. 2).

The pressures in the injection chamber at the beginning and end of the injection were p_0 = 3.0 MPa and p_3 = 6.4 MPa, i.e., their difference was Δp = 3.4 MPa. The injection advance was φ = 19 deg.p.c., injection time τ = 22 deg.p.c. and the pressure at the injector nozzle at the beginning of the injection is 23.8 MPa and at the end-7.4 MPa. The maximum injection pressure was 24.0 MPa and the residual pressure in the fuel line was 5.0 MPa.

Figure 4 dependences of the average cyclic supply and fuel supply irregularity (according to the Bashkir State Agrarian University method) on the speed of the camshaft of the injection pump FHPP 4NUTNI-T-1111007: 1-5-when adjusting the FIE in accordance with the existing procedure, the same with the pre-adjusted nozzles with counter-pressure with the adjustment of the FIE as a whole with counter-pressure to injection and verification after adjustments 1 and 2 with the introduction of counter-pressure and the corresponding average cyclic supplies.

When adjusting the nozzles on the stand with the counter-pressure, the fuel supply irregularity decreased by 8 and 11%, respectively in the nominal and maximum torque modes.

Motor tests of the engine 4Ч11/12.5 showed that the specific fuel consumption decreased by an average of 2.6% (by 5.46 g/kWh).

CONCLUSION

The relatively simple design of the counter-pressure device for fuel injection is achieved when using combined hydromechanical accumulators in particular with one mechanical spring (according to the third scheme of Table 1).

The device with such an accumulator almost completely approximates the operating conditions of the FIE on the stand to those in the engine cylinder. It can be integrated into any adjustment stand.

It is expedient to determine the optimal dimensions and parameters of the main accumulator cells by its stiffness, calculated using the engine indicator diagram and the fuel supply law of its fuel system. For the engine д 245.12 C-692:

The spring must be made with the stiffness c = 21.47 N/mm and have a height in the free and in pre-compressed states of 80 and 63 mm, respectively; the hydraulic accumulator should be made in volume of 2.266 mm^3, the diameter and the course of the accumulator-hydraulic measuring plug must be d_n = 0.008 m, h = 0.0022 m.

The adjustment of the fuel equipment on the upgraded stand with this device should be carried out with a preliminary adjustment of the nozzle. With this adjustment, the inter-sectional irregularity of the fuel supply is reduced to 11% and the effective power and economic performance of the engine increase by 7 and 2.6% (6 kW and 5.46 g/kWh), respectively.

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