COMBUSTION OPTIMIZATION IN A MODERN DIESEL ENGINE BY MEANS OF PRE-INJECTION STRATEGY

ОПТИМИЗАЦИЯ ПРОЦЕССА СГОРАНИЯ В СОВРЕМЕННОГО ДИЗЕЛЬНОГО ДВИГАТЕЛЯ ЧЕРЕЗ СТРАТЕГИИ ДЛЯ ПРЕДВАРИТЕЛЬНОГО ВПРЫСКА

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Abstract: *The article presents a numerical study of pre-injection strategy in order to reduce the rate of heat release and pressure rise in a modern direct injection diesel engine, developed for passenger car. A model of the engine was built in advanced simulation code AVL BOOST. In order to determine the injection rate a supplementary model of the solenoid injector was built in AVL HYDSIM. A study of rate of heat release and pressure rise into combustion chamber was conducted at single operating point. The engine effective power was taken into consideration as well. The results revealed that pre-injection strategy is a promising approach for reducing the rate of heat released and the engine noise at low speed and load. However, a precise control of pre-injected mass and injection timing has to be realised by engine control system.*

***KEYWORDS**: DIESEL ENGINES, DIRECT INJECTION, ENGINE SIMULATION, RATE OF HEAT RELEASE, PRE-INJECTION*

1. Introduction

One of the most significant disadvantages of direct injection (DI) diesel engines is the high level of noise as a result of high rate of heat release (ROHR) during the premixed combustion period [1]. That high combustion speed is determined by injected fuel mass during the ignition delay period in DI diesel engines [1]. The more fuel injected during the ignition delay, the stronger the initial pressure rise in the combustion chamber, and the better boundary conditions for NOx formation occurs. For those reasons it is important to reduce the ROHR during the premixed combustion period [2, 3]. There are a few methods of reducing the ROHR such as control of injection rate at the beginning of injection, dual-stage injection, reduction of ignition delay period by in-cylinder turbulent flow, pre-injection strategy and etc. [1, 2].

However, number of research [4-8] revealed that pre-injection strategy is the most effective approach. In fact, this approach is realized by one or few injections of small fuel mass shortly before the main injection. This small quantity of fuel is called pre-injection. It is easy to realize that strategy by means of the modern common rail fuel systems Fig.1 [2]. A typical injection process in common rail systems consists of a pre-injection, main injection and post-injection. Mainly, the aim of pre-injection is to reduce the rate of heat release as it reduces the engine noise. However, the pre-injection can be used for control the NO_x formation by mean of temperature reduction in combustion chamber. Usually, the pilot injection quantity is not more than 5% of the injected fuel. Common rail systems that use solenoid injectors can realize up to five separate injections which means that not more than two pre-injection can be produced. In order to further reduction of ROHR at the beginning of the combustion process, the main injection can be realized with different shape such as boot type or ramp type (Fig.1). The most commonly used is rectangular type of main injection due to the simplification of injector design. The early post-injection is used in order to soot formation control as the late post-injection is realized at that engine operating mode in which the regeneration of the particle filter occurs. Piezo-controlled injectors can produce more than five injections due to fast response time of the injector control valve.

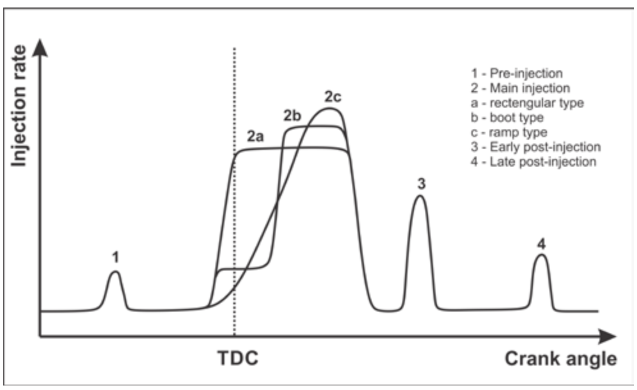

Figure 1. Schematic diagram on multiple injection in modern Common rail DI Diesel engines

The pre-injected fuel mass evaporates before the main injection and pre-combustion reactions start. It increases the temperature in local volumes in the combustion chamber therefore the ignition delay period is reduced significantly. It means that the fuel injected at begin of main injection is involved immediately in combustion process after very short ignition delay period.

2. Pre-injection strategy.

The single pre-injection strategy is most commonly used because it is less complicate to realize. Both common rail injector (servo and piezo-controlled) can inject a small fuel mass before main injection. The effects of this single pre-injection on the ROHR can be seen in Fig.2.

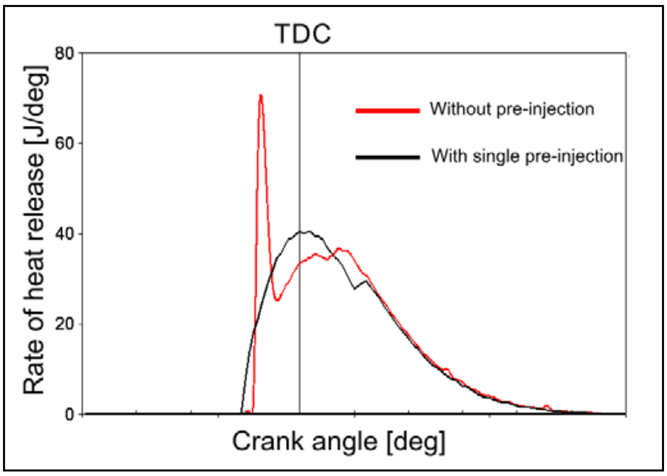

***Figure 2.** Effect of pre-injection on the ROHR.*

The diagram reveals that ROHR increases much slower at the beginning of combustion and the maximum ROHR is reduced almost twice. As a result, the maximum pressure rise in the combustion chamber decreases from 0,85MPa/deg to 0,46MPa/deg. In order to provide the same output power and BMEP it is

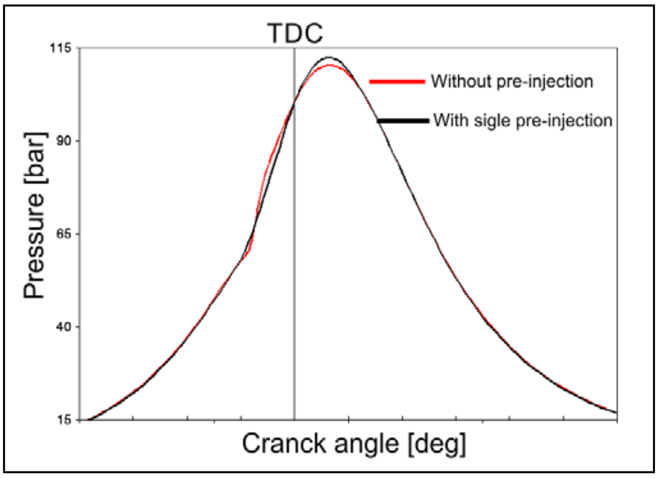

Figure 3. *Effect of pre-injection on pressure variation into the cylinder.*

necessary to reduce the injection timing (the start of injection) once the pre-injection is realized. It is a result of the shorter ignition delay period. The main injection timing has to be reduced from 15deg BTDC to 10deg BTDC.

Several parameters have to be taken in consideration: the mass of pre-injected fuel, the timing of pre-injection and the rotation angle between pilot and main injection. The various research on pre-injection has been reported [2, 7, 8]. Based on previous study it is important to note that the quantity of pre-injected fuel should be chosen within the range of 2% and 5% of overall injected mass [7]. The start of pilot and main injection should be chosen in way that start of main injection coincides with the start of combustion.

The basic research shows that a single pre-injection is an efficient method of reducing the maximum pressure rise in combustion chamber respectively the noise produced by diesel engines. For that reason the main aim of this paper is to study the influence of multiple (dual) pre-injection on the ROHR and pressure rise in a DI diesel engine.

*3. **Simulation model.***

A simulation model was built in advanced simulation package AVL. It consist an engine model built in AVL Boost and an injector model built in AVL HYDSIM. This model provides opportunities to study the combustion process by means of rate of injection, previously defined by injector simulation. The engine performance, pollutant emissions, fuel consumption and other parameters can be estimated as well.

The engine under study is 2.0liter four cylinders direct injection diesel engine, developed for passenger car. The maximum output power is 101kW at 4000rpm as the maximum torque is 320Nm at 2000rpm. The engine is equipped with variable geometry turbocharger. The boost pressure is limited to 1.4bar. The common rail fuel system of the engine is delivered by Delphi. The maximum injection pressure is 1600 bar. The engine is also equipped with EGR system and post treatment system including catalytic converter and DPF. The cylinder is equipped with four valves per cylinder. The main geometrical parameters of the engine are listed in Table 1.

Table 1

Type of engine	HDI
Number of cylinders	4
Total volume	2L
Cylinder bore	85 mm
Cylinder stroke	88 mm
Compression ratio	17,6
Valves per cylinder	4

3.1. Engine model

In order to build a realistic engine model it was necessary to input geometrical parameters of intake and exhaust system as well as valves diameters and valves lift curves. The parameters (diameters and length of the pipes) of intake and exhaust system were measured on a real engine mounted on a test bed. The valves lift curves were calculated as the cams lifts had been measured by means of special equipment and the kinematic scheme of valve train mechanism was used. The valves seats diameters were taken from the technical documentation.

The engine model was built in advanced simulation code AVL Boost by means of elements available into (Fig. 4). The main elements which were used are: cylinder, plenum, pipe, intercooler, turbocomressor and general engine element (E1).

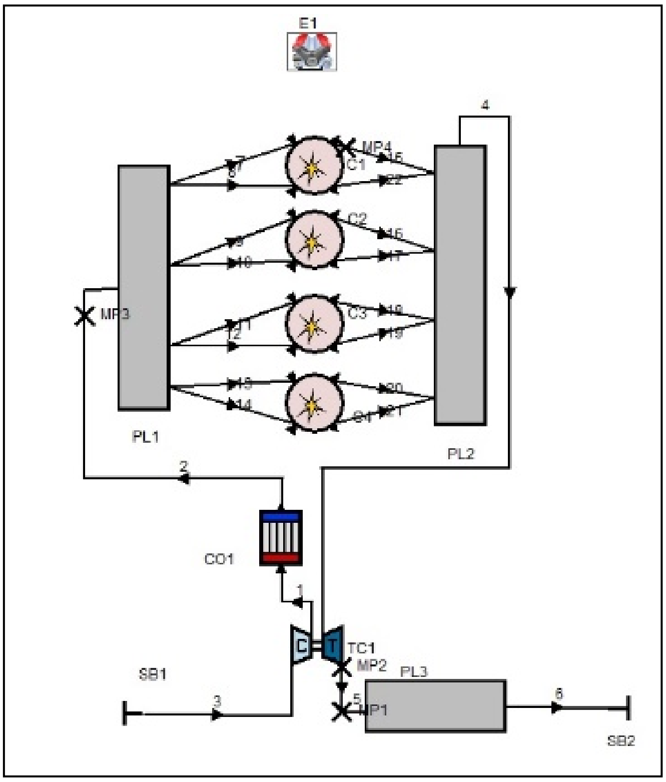

Figure 4. *Engine simulation model, built in advanced simulation software AVL Boost*

The main control parameters such as engine speed, number of cylinders, type of engine and simulating duration was input into the element E1. The parameters needed for estimation the mechanical losses into the engine are also defined in this element.

The element "Cylinder" was used to define the geometrical parameters of the cylinder, injection and combustion model as well as heat transfer to cylinder wall. Mixing-controlled combustion model, developed by AVL was used in this simulation. For this model it was necessary to define the injection parameters such as rate of injection, injected mass, nozzle geometrical parameters, injection pressure and injection timing. Air-excess ratio was used as a control parameter of the injection process. The most commonly used model of Woschni was chosen for heat transfer estimation.

The intake system includes: air filter, compressor, intercooler, plenum and pipes. The elements by means of which exhaust system was developed are: plenum, turbine, silencer and pipes. The compressor and turbine turns with same speed due to the connection shaft.

3.2. Injector model

In order to estimate the rate of injection an injector model was built in simulation software AVL HYDSIM. Engine fuel system was developed by Delphi. The system includes: high pressure pump driven by engine camshaft, pressure control valve (IMV), a common rail and solenoid injector for each cylinder. High pressure pump is developed to operate up to 1650bar. A transfer pump is mounted into the high pressure pump due to provide fuel supply pressure of 6bar. The pressure is electronically controlled by a solenoid valve on the low pressure side of the high pressure pump. The system operates with great flow at the return fuel line due to necessity of intensive fuel cooling. The injectors can realize up to five injections for a cycle.

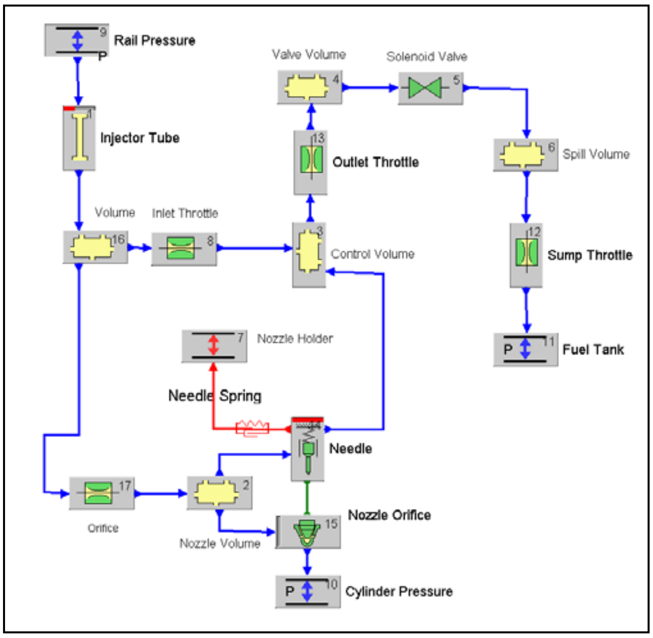

Figure 5. *Injector simulation model, built in AVL HYDSIM.*

Injector model includes various elements such as mechanical components (needle and spring), hydraulics elements (volumes, tubes, throttles and nozzle orifices), electronic component (solenoid valve) and constant pressure elements (rail, cylinder and fuel tank). Some geometrical parameters such as needle mass and size, needle spring stiffness, injector tube length and diameter as well as the volumes were measured as an injector was disassembled. Other parameters such as throttles sections and nozzle orifices were taken by Delphi technical documentation. The injector model is presented in Fig. 5.

The main injector parameters are listed in Table 2.

Table 2

Type of injector	Delphi DFI 1.3
Nominal voltage	Battery voltage
Initial current	22 to 26A
Operating pressure	100 to 1650bar
Number of nozzle holes	6
Diameter of nozzle holes	0.15mm
Spray angle	50°
Minimum injected mass	0.5mg

By means of the injector model, injection rate was estimated as injected mass was separated up to three – two pre-injections and a main injection (depending on injection strategy). The amount of fuel injected by each pre-injection was 0.5mg or 1mg. In this article the timing interval between each injection was not studied. An injection simulation was conducted at engine speed corresponding to maximum engine torque – 2000rpm. Single engine operating points was studied. The overall injected fuel mass was 25mg. At studied operating point the reference engine output power was 27,5kW. The rail pressure was 800bar due to provide the duration of the main injection counted to 19deg by crank angle rotation. In order to realize the targeting pre-injected mass of 0,5mg and total injected mass of 25mg the injector solenoid valve was opened respectively for a period of 0.29ms and 0.83ms. The variation of injection rate estimated by the model in case of double pre-injection is shown in Fig. 6.

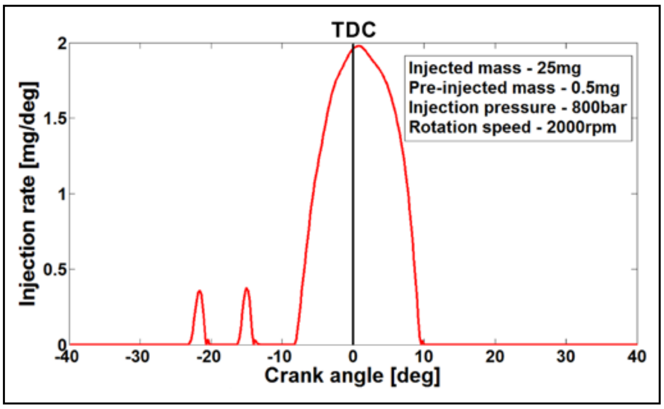

Figure 6. *Fuel injection rate with two pre-injection, estimated by injector model*

4. *Results and discussion.*

The numerical study was conducted at an engine operating point, defined by engine speed of 2000rpm and air-excess ratio of 2.

Firstly, a study of pre-injection strategy was conducted. For that the ROHR and in-cylinder pressure rise were estimated as three injection strategies was applied – without pre-injection, single pre-injection and double pre-injection. In order to achieve comparative results some constrains was defined:

- The overall mass of injected fuel was 25mg;
- The start of main injection was constant – 8deg BTDC;
- The overall mass of pre-injected fuel was 1mg;
- In double pre-injection case, the pre-injected mass was separate by two equal parts;
- The start of first pre-injection was constant – 20deg BTDC.

The influence of injection strategy on ROHR is presented in Fig. 7. It was observed a significant reduction of ROHR in case of pre-injection. The maximum ROHR was reduced from 89,1J/deg without pre-injection to 67,4J/deg in case of pre-injection. The function variation revealed that pre-injection strategy eliminated the pre-mixed combustion period. It provides slightly increasing of ROHR up to maximum value.

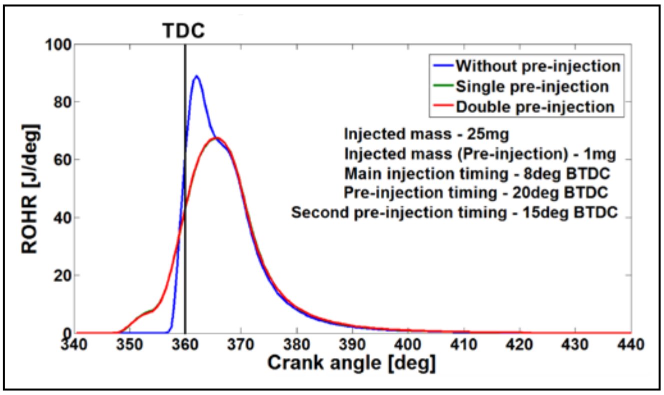

Figure 7. *ROHR variation as function of injection strategy*

Obviously, the reduction of ROHR decreases the maximum pressure rise into the cylinder – Fig.8. Pre-injection reduced maximum pressure rise from 0,65MPa/deg to 0,37MPa/deg. This significant reduction accounts to 43% lower maximum pressure rise. Although the combustion noise was not studied, we expected much lower noise level.

The influence of double pre-injection strategy on combustion parameters was insignificant in comparison with single pre-injection. Practically, the ROHR and pressure rise curves were absolutely the same and were not influenced by pre-injection separation. That interesting fact can be explained with equal pre-injected fuel mass.

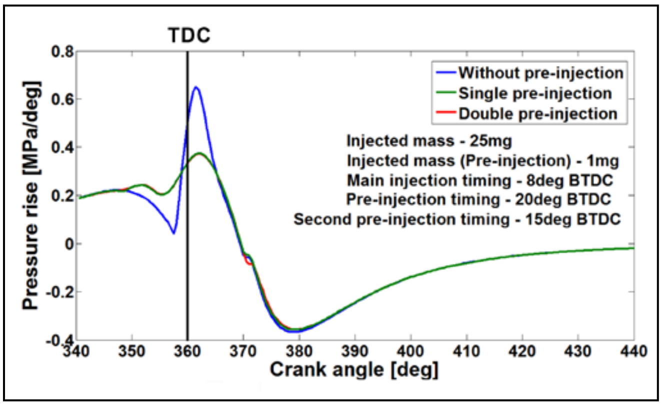

Figure 8. *Pressure rise variation as function of injection strategy.*

The single pre-injection reduced engine output power by 0,3%. In order to achieve the same effective power, respectively the same engine efficiency injection timing of both pre and main injection were retarded by 2deg. It was applied to single pre-injection mode. In result of that, the maximum pressure rise was additionally reduced to 0,29MPa/deg or the final reduction by 55% was observed compared to the maximum pressure rise achieved without pre-injection. The effect of late injection on the variation of ROHR and pressure rise is shown in Fig. 9 and 10.

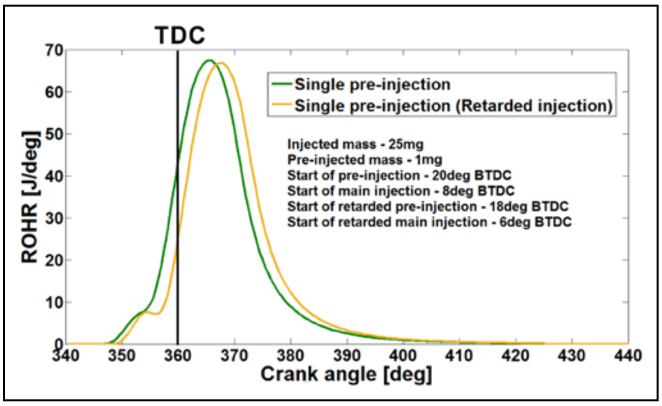

Figure 9. *ROHR variation as function of injection timing*

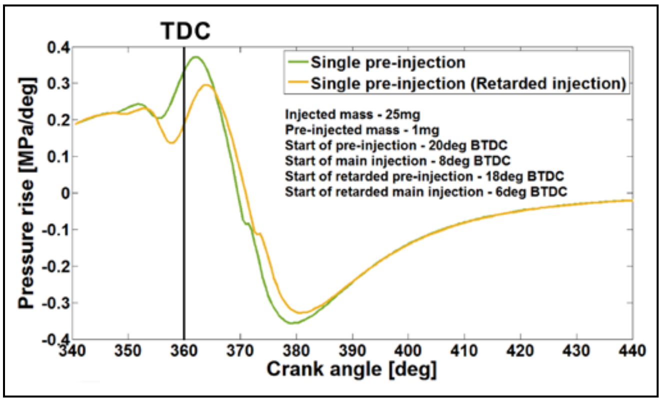

Figure 10. *Pressure rise variation as function of injection timing*

Finally, a study of double pre-injection strategy was conducted as the pre-injected fuel was increased up to 2mg. The injected fuel was separate equally between first and second pre-injection. The pre-injection timing was the same as that in the first study. Increasing of pre-injected fuel reduced the engine effective power by 0,7% and the effect on ROHR and pressure rise was not significant. The maximum ROHR was 65,6 J/deg and maximum pressure rise was 0,36MPa/deg. In order to eliminate engine efficiency reduction the double pre-injection process was retarded by 3deg as the timing interval between each of injection was the same. The pressure rise variation is shown in Fig. 11.

The combination of double pre-injection with high pre-injected fuel mass and retarded injection timing reduced maximum pressure rise up to 0,25MPa/deg. It was observed that pressure rise curve had two maximums with the same values. This fact revealed that further increasing of pre-injected fuel mass would increase maximum pressure rise due to increasing the first maximum of the curve. The timing retard in case of double pre-injection has not reduced significantly the maximum of ROHR.

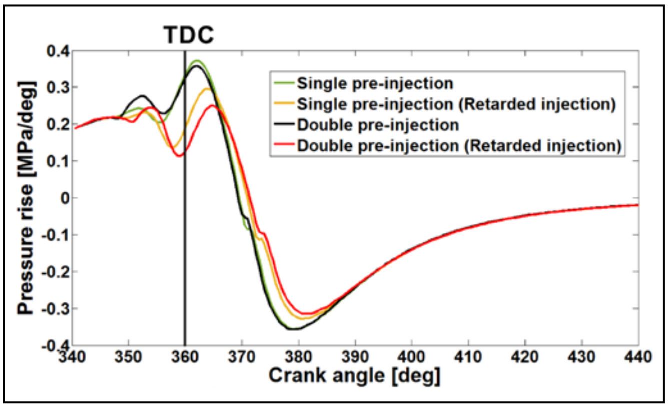

Figure 11. *Influence of double pre-injection strategy to pressure*

5. Conclusion

Pre-injection strategy can be successfully applied as combustion control strategy due to reducing of maximum pressure rise at low speed and engine load. The study revealed that single pre-injection strategy can reduce pressure rise by 43%. By applying this strategy it is necessary to reduce injection timing in order to obtain the same engine power. Additionally, it can reduce pressure rise up to 55%.

Double pre-injection is more effective. To achieve significant impact on in-cylinder pressure rise it is necessary to increase pre-injected mass and to retard the injection up to 3deg by crank angle rotation for studied operating point. In that case pressure rise reduction by 61,5% was observed.

This study optimized the pre-injection parameters at single engine operating point. The results revealed that pre-injection strategy have to be precisely optimized at whole operating maps taking into consideration the injection system parameters.

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