

# THERMODYNAMIC ASPECTS OF COMBUSTION IN GASOLINE ENGINES FITTED WITH A MULTIPLE FUEL INJECTION

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## **Abstract**

*The multiple fuel injection is being applied not only in diesel engines but in direct injection gasoline engines as well. The injection duration in gasoline engines is similar (pressure values at present approx. 20 MPa) to the injection duration of diesel fuel. The methodology and results of the tests related to the fuel dose division and injection strategy on the thermodynamic indexes during the combustion process have been presented in the paper. The tests were performed for several ways of fuel dose division at injection pressures of 5, 10 and 20 MPa (modifying also the time of the injection).*

*The paper presents: state of research on the effects of multiple injection in gasoline engines, research equipment and methodology when applying the Rapid Compression Machine, estimation of the thermodynamic indexes, injection strategies, influence of injection pressure on the thermodynamic indexes, single fuel dose, two-phase fuel injection, three-phase fuel injection, changes in the combustion pressure, changes of the average thermodynamic temperature of the charge in the cylinder, heat release rate, heat release and relative value of the heat used for strategies, the influence of the fuel injection pressure on the course of the combustion in a Rapid Compression Machine including the course of the heat release*

**Keywords:** *direct injection, combustion engines fuel dose, rapid compression machine*

## **1. Introduction**

Spark ignition engines used in modern vehicles are not as uniform in terms of injection and combustion as diesel engines. In combustion engines both indirect injection (to the intake manifold, low-pressure) and direct injection systems (to the cylinder, middle-high pressure) are used. Designs that combine these two systems simultaneously are very rare and did not become overly popular (engine by Lexus 2GR-FSE uses both direct and indirect fuel injection to the cylinder). Despite a variety of gasoline direct injection solutions fully satisfactory results in terms of engine operating parameters and exhaust emissions have not yet been obtained.

Direct injection of fuel into the cylinder should enable a homogenous combustion that is in fact realized in direct injection systems, yet charge losses occur before its delivery to the closed space of the cylinder. This system should also enable a formation of non-homogenous stratified charge that allows burning lean mixtures (currently the main trend in the global research and development). In this respect gasoline engines are becoming more similar to diesel engines.

The use of high-pressure gasoline direct injection system that can divide the fuel dose into smaller portions is currently the main trend in the research of these systems. This allows a free qualitative and quantitative control of the preparation of the combustible mixture and influences the character of its later combustion.

## **2. State of research on the effects of multiple injection in gasoline engines**

The development of the first generation of direct injection systems in gasoline engines focused mainly on the wall guided fuel atomisation, next – on the charge-guided ones. The current generation of the direct injection systems (spray-guided) has shown a potential that allows a reduction in the fuel consumption through extending of the engine work area through charge stratification, with a simultaneous reduction of the exhaust emissions [12, 14]. Van Der Wege and others [14] pointed to the possibility of obtaining a stratified charge for high engine loads (effective pressure exceeding 0.5 MPa and engine speed above 4000 rpm) through the application of multiple injection. Also, the possibility was indicated of an improvement of the quality of the obtained homogenous mixture under high loads through an application of a wide-angle spray cone generated from an injector centrally located in the combustion chamber.

The first generation of engines fitted with direct injection systems (wall/charge-guided) enabled a reduction of the fuel consumption as compared to indirect injection systems (injection to the intake duct) thanks to the reduction of the pumping losses, advantageous conditions of operation on lean mixtures and lower heat losses at a lower charge temperature. The first generation also enabled the application of higher compression ratios thanks to the use of the cooling effect generated by the vaporization of the injected fuel [1, 2, 12]. In the case of spray-guided system of mixture formation the heat losses during combustion are lower, which leads to a further reduction of the fuel consumption. This solution is currently dominating in direct injection systems.

The advantage of this system of combustion is a central location of the injector near the spark plug, which allows a reduction of the delay in pressure growth after the ignition of fuel. Hence, we can more efficiently control the course of the combustion [7], the piston has a much larger area of heat transfer [12, 14] and as a result lower air swirl in the cylinder is necessary [13]. Contemporary research of combustion in transparent engines indicates that the fuel penetration at the end of the compression stroke is not sufficient to reach the piston area. This is confirmed by Raimann's investigations [11] carried out on outward-opening piezoelectric injectors. The fact that the fuel reaches the piston surface causes increased emission of hydrocarbons and an increased opacity [3].

Frohlich and Borgmann [6] have shown that vehicles operated with piezoelectric outward-opening injectors have better fuel economy by 20% in relation to the vehicles fitted with conventional engines (IDI engines, medium unit power output 60 kW/dm<sup>3</sup>). Wirth [18] in the investigations concerning the use of multihole injectors obtained 15.5% reduction in the fuel consumption as opposed to the PFI engine. He used the NEDC test as a basis for his comparison.

Apart from very few exceptions, in the literature there is no thermodynamic analysis of the engine work cycle using multiple injection and outward-opening injectors with an injector centrally located in the combustion chamber. Only the injection strategies are fragmentarily presented [5, 9, 15, 16] without their in-depth analysis. And also the elements of evaluation of the possibility of application of the strategy of multiple injection in the heating of the catalytic converter could be found here [4, 9].

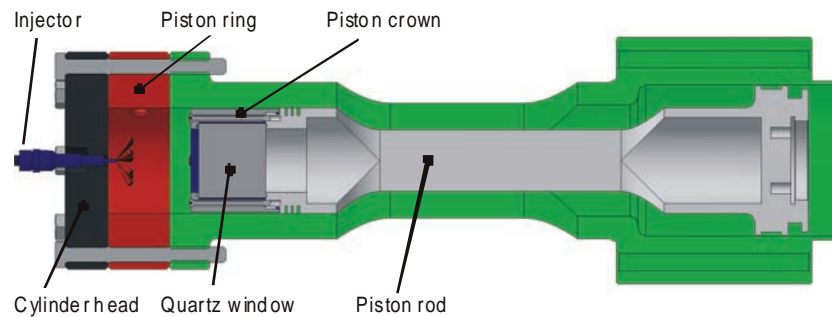
The relevant literature does not provide a full description of the phenomena related to the physical aspects of fuel atomization in high-pressure gasoline injection systems nor the qualitative and quantitative analysis of its combustion. For this reason investigations have been initiated in order to search for qualitative and quantitative dependencies of the combustion process indexes from strategy of fuel dose division (considering the proportions of the division and the durations of individual doses).

## **3. Research equipment and methodology when applying the Rapid Compression Machine**

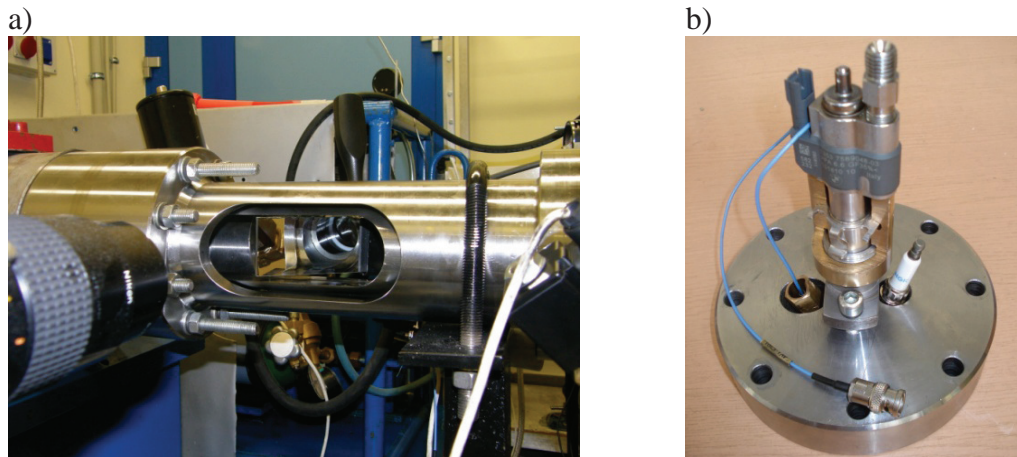
The investigations of the onset of the injection and combustion through optical methods are currently carried out in many research and development centres on real engines [8, 17] as well as in Rapid Compression Machines (RCM). The RCM utilized in Poznan University of Technology

allows basic testing of a single work cycle of a combustion engine, particularly with respect to the fuel injection processes, charge motion, ignition and combustion. The machine is composed of a cylinder in which a piston is placed in such a way as to enable optical access to the combustion chamber from the direction of the piston crown (Fig. 1, 2); the option of optical access from the direction of the cylinder head is also convenient: this solution has been described in detail in [16, 19] for the other case of research. The optical access through the piston crown allows using the whole area of the cylinder head to reflect the location of all the injectors and the spark plug.

Instead of a traditional crankshaft system a pneumatic system forcing the piston motion has been applied. For the control of the piston motion (compression stroke) the system uses air of adjustable pressure of up to 8 MPa fed to the area under the piston from an external air accumulator. The piston motion in the RCM fairly well reflects the operation of a real engine and corresponding thermodynamic indexes of the work cycle.



*Fig. 1. Schematics of a Rapid Compression Machine with optical access to the combustion chamber*



*Fig. 2. Rapid Compression Machine: a) optical access to the combustion chamber, b) cylinder head with the fuel injection system and combustion pressure sensor*

The operation of the RCM is controlled by a sequencer that generates individual signals to the actuators (electromagnetic valves). The system allows control in 16 channels with a resolution of  $\pm 1$  nanosecond. The RCM can be fuelled with positive ignition fuels (gasoline, ethanol, methanol and the mixtures thereof) and diesel oil (or alternative fuels like fatty acid methyl esters, eq. popular B100 or the mixtures thereof). Gasoline is fed from a standalone Common Rail fuel system of own design of operating injection pressure in the range of 5-30 MPa [17, 19]. A high-pressure pump (also used in BMW-engines) has been used in the system including outward-opening injectors [12]. The minimum injection time is 200  $\mu\text{s}$ , and the dwell-time is 150  $\mu\text{s}$  (which corresponds to 0.006 CA at engine speed of 4000 rpm).

The pressure measurements were performed with the use of a piezoelectric sensor AVL GM11D (sensitivity 2.52 pC/bar), the piston travel was measured with the use of contactless

potentiometer displacement sensor LSR 150 ST R5k by Megatron. The indicator measurements were archived through AVL IndiModul (with the use of charge amplifiers MicroIFEM), then processed in AVL Concerto V4.3.

#### 4. Estimation of the thermodynamic indexes

The basic thermodynamic quantities of the cycle in the RCM were determined based on the pressure course in the cylinder and the piston travel. Using Concerto V4.3 by AVL an analysis of the course of heat release was performed.

The amount of heat released in the work cycle can be described with a formula:

$$Q_i = \int p dV + dU_s, \quad (1)$$

where  $dU_s$  denotes internal energy:

$$U_s [t, T_g(t)] = M(t) \cdot c_{v,m} [t, T_g(t)] \cdot T_g(t), \quad (2)$$

and the instant amount of released heat is:

$$\Delta Q_i(t) = [U_s(t + \Delta t) - U_s(t)] + \frac{p(t) + p(t + \Delta t)}{2} [V(t + \Delta t) - V(t)], \quad (3)$$

The joint amount of heat taken by the working medium from the onset of the calculations (time  $t$ ) amounts to:

$$Q_i(t) = [U_s(t) - U_s(t_a)] + \sum_{t_a}^t \left\{ \frac{p(t) + p(t + \Delta t)}{2} [V(t + \Delta t) - V(t)] \right\}. \quad (4)$$

Using the Wiebe function, whose values were determined based on the generally applied formula:

$$\bar{x}(t) = 1 - \exp \left\{ a \cdot \left[ \frac{t - t_{SOC}}{t_{EOC} - t_{SOC}} \right]^{m_s + 1} \right\} [-], \quad (5)$$

for which  $a = -6.908$  and  $m = 0.3$  was assumed (for spark ignition engines), the temperature of the gases was determined derived from the Clapeyron equation expressed as follows:

$$T_g(t) = \frac{P(t) \cdot V(t)}{\beta_x(t) \cdot M(t) \cdot \bar{R}} \quad [K]. \quad (6)$$

in which  $\beta_x(t) = 1 + \frac{\beta_t - 1}{1 + \gamma} \cdot \bar{x}(t)$  is the current coefficient of molar transformation during the combustion.

The amount of heat supplied with the fuel was assumed as:

$$Q = m_F \cdot H_F, \quad (7)$$

where  $m_F$  denotes the mass of the injected fuel and  $H_F$  its calorific value.

The value of heat  $Q_i(t)$  taken by the working medium calculated from equation (4) was compared to the total energy supplied to the cylinder – the values of the relative heat use  $X_i(t)$  were obtained from:

$$X_i(t) = \frac{Q_i(t)}{m_F \cdot H_F}. \quad (8)$$

The results of the calculations with the use of formulas (1)-(8) have been presented in the further part of the paper.

## 5. The test plan

The investigations of the combustion with a division of the fuel dose were performed according to a plan shown in Tab. 1 and Fig. 5. The measuring points were used without a division of the fuel dose and with a division of the fuel dose into two parts at the same time maintaining the total fuel dose and the air excess coefficient  $\lambda = 2$ . The tests were conducted at the injection pressure of 5, 10 and 20 MPa; at higher pressure the injection time was reduced respectively due to a higher fuel outflow velocity from the injector (the flow characteristics of the injector was examined earlier). The fuel ignition took place in the moment of the onset of the main injection (or the undivided fuel dose).

Tab. 1: Strategies of the fuel dose division

Strategy	P <sub>inj</sub> [MPa]					Ignition			
		q <sub>1</sub> [ms]	t <sub>1</sub> [ms]	q <sub>2</sub> [ms]	t <sub>2</sub> [ms]	q <sub>3</sub> [ms]	t <sub>3</sub> [ms]	q <sub>4</sub> [ms]	
1.	5					1.3			
2.	10					0.75			
3.	20					0.55			
4.	5			0.3	3	0.8			
5.	10			0.3	3	0.4			
6.	20			0.3	3	0.3			
7.	5	0.3	3	0.3	3	0.4			
8.	5	0.3	1	0.3	3	0.4			
9.	5			0.3	3	0.4	1	0.3	

q<sub>1</sub>, q<sub>2</sub>, q<sub>3</sub>, q<sub>4</sub> – time of injection;  
t<sub>1</sub>, t<sub>2</sub>, t<sub>3</sub> – dwell-time.

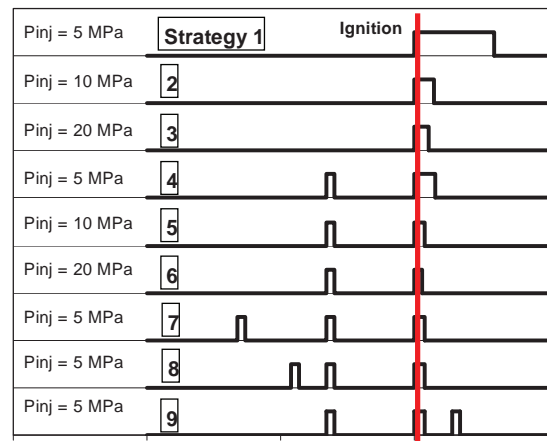


Fig. 5. The test plan and proportions of the fuel doses ( $\lambda = 2$ )

## 6. Injection strategies

The investigations into the selection of the strategy of fuel injection were performed for the air excess coefficient in the cylinder  $\lambda = 2$  (lean mixture). The strategies of the injection were selected as follows (comp. Tab. 1, Fig. 5): combustion without division of the fuel dose (strategies 1-3) at different fuel injection pressures (5; 10; 20 MPa – higher injection pressure causes respective reduction of the injection duration); the combustion with the fuel dose division into two parts (strategies 4-6) with the use of different fuel injection pressures (5; 10; 20 MPa) and the division of the fuel dose into three parts for the pressure of 5 MPa (higher injection pressure and minimum injection duration prevented the obtaining of the sufficiently small fuel doses, strategies 7-9). In the tests the value of the fuel dose was 36 mg (for all injection strategies). The ignition was realized at a certain point of the engine RCM operation, which allowed performing the comparative analysis of the variability of the thermodynamic indexes.

## 7. Discussion of the results of multiple gasoline injection

### 7.1. Influence of injection pressure on the thermodynamic indexes

#### Single fuel dose

Variant investigations were performed without dividing of the fuel dose at different injection pressures. Variable injection pressure at the same fuel dose (36 mg) forced the application of different injection durations. At variable pressure of the value of 5; 10 and 20 MPa the injection durations were: 1.3; 0.75 and 0.55 ms respectively.

The courses of the combustion pressure have been presented in Fig. 6. Due to a variable fuel atomization, the forced ignition (in the same moment) of the fuel results in different combustion courses. The courses were related to the time axis and not to the crankshaft angle because the RCM is not equipped with any crankshaft system typical for the conventional IC-engine.

The maximum cylinder pressures are similar (between the injection pressure 5 MPa and 10 MPa the maximum difference of  $P_{cyl-max}$  is 0.6 bar). As no significant differences were recorded in the pressure course the outstanding thermodynamic parameters of the cycles were determined: the time of the occurrence of the maximum combustion pressure ( $P_{max}$ ), the rate of pressure increment after the injection ( $dP/dt$ ) and the times of its occurrence  $t_{-P_{max}}$  and  $t_{-dP-dt}$  respectively (Fig. 7).

The time after which the maximum combustion pressure occurs (taken from the onset of the injection) is slightly reduced as the injection pressure grows. The maximum difference between the moments of occurrence of the maximum combustion pressure at the injection pressure of 5 MPa and 20 MPa amounts to 1.2 ms. This denotes an acceleration of the course of the combustion. This opinion is confirmed by the rate of pressure increment after the ignition ( $dP/dt$ ). In the first case it is 7.55 bar/ms and at the injection pressure of 20 MPa it grows to the value of 11.67 bar/ms. This indicates the increase in the combustion rate, which is caused by the quality of the fuel atomization. The moment of occurrence of the maximum pressure  $(dP/dt)_{max}$  is also reduced and amounts to 1.4 ms (between the extremes).

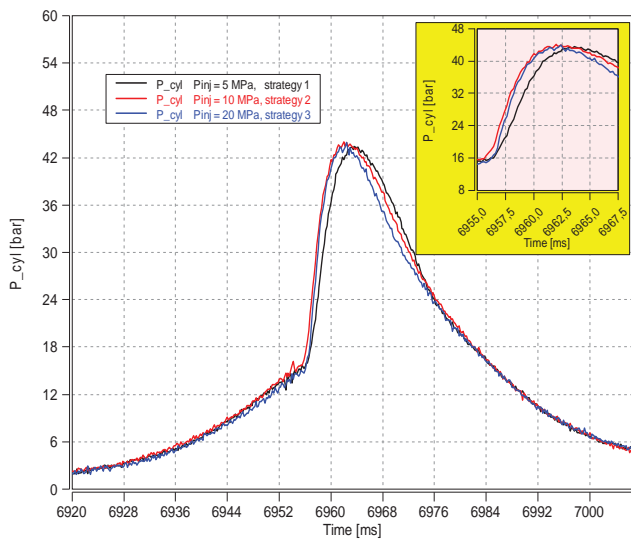


Fig. 6. The influence of the fuel injection pressure on the course of the combustion in a Rapid Compression Machine including the course of the heat release (corner)

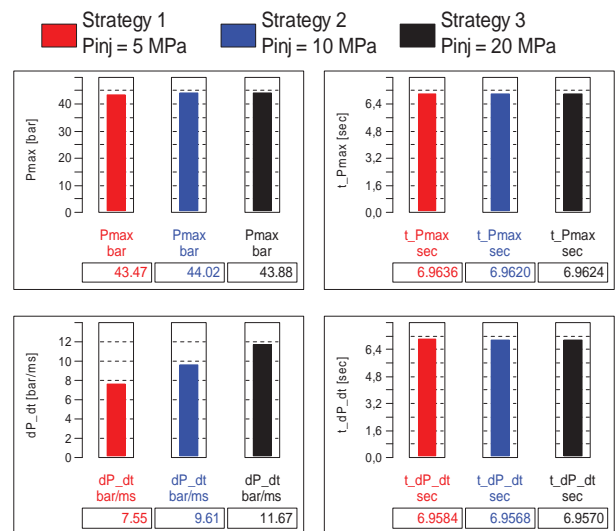


Fig. 7. The values of the indexes of the course of the combustion with no fuel dose division

A low pressure of the injected fuel results in certain thermodynamic consequences. The lack of fuel dose division results in that the maximum combustion temperature is obtained at the lowest fuel injection pressure (Fig. 8). Such a course of combustion is caused by poor fuel atomization and its high concentration around the spark plug. During the analysis it should be remembered that the values of the average temperature of the charge, not the local flame temperature are presented in Fig. 8. The determining of the flame temperature is possible with the use of optical methods.

The changes in the injection pressure ( $P_{inj} = 5, 10$  and  $20$  MPa) result in small changes in the maximum temperatures of the charge (average in the cylinder). The analysis of the released heat (Fig. 9) indicates similar course of combustion (Fig. 6) for different fuel injection pressures. This is confirmed by both the course of pressure in the cylinder and the rate of heat release. Similar results were obtained for the used heat  $Q_i$  of the value of 190 kJ/ms (the maximum values were

obtained for the fuel injection pressure  $P_{inj} = 10$  MPa). The maximum value of the rate of heat release for strategies 1, 2 and 3 amounts to 195, 199 and 188 kJ/ms respectively.

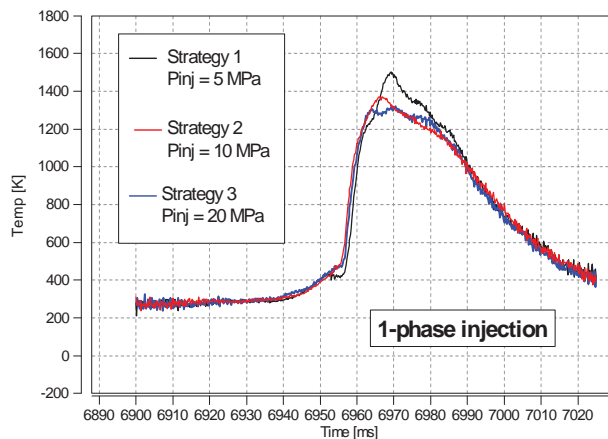


Fig. 8. Changes of the average thermodynamic temperature of the charge in the cylinder (injection strategies 1, 2, 3)

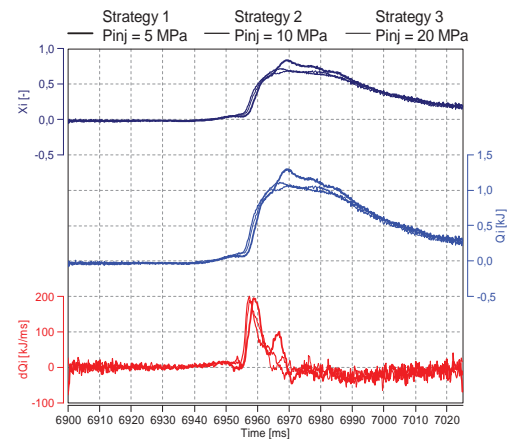


Fig. 9. Heat release rate, heat release and relative value of the heat used for strategies 1, 2, 3

### Two-phase fuel injection

The injection with two phases is realized in strategies 4, 5 and 6 (Tab. 1 and Fig. 5). The tests were made for the same value of excess air coefficient and the same constant cumulative fuel dose at different variant of fuel injection pressure. In these cases greater differences in the course of the combustion pressure (Fig. 10) and significant differences in the values describing this pressure ( $P_{max}$ ,  $dp/dt$  – Fig. 11) have been observed. The increase in the injection pressure results in the increase in the maximum combustion pressure. The time corresponding to this value is identical for all three strategies (Fig. 11b). Higher injection pressure ( $P_{inj} = 20$  MPa, strategy 6) results in a better fuel atomization and increased combustion pressure increment after the ignition. In this case this value amounts to 11.3 MPa/ms and at the lowest injection pressure it drops to the value of 6.5 MPa; this denotes a change by over 70%. This should result in a much shorter combustion, which was also confirmed during the optical investigations (Fig. 19, 20).

A growth in the injection pressure results in better fuel penetration in the combustion chamber and adds to the growth of the combustion temperature (average maximum combustion temperature amounts to 1400 K), yet the high rate of changes of this temperature is observed in the final stage of the combustion. The average thermodynamic temperature of the cycle changes rapidly, which could confirm a quick end of the process of combustion (Fig. 12). This is also confirmed by the course of the combustion pressure presented earlier. Strategy 5 results in higher temperature values in the cylinder as compared to strategy 4 (the lowest injection pressure – 5 MPa).

Advantageous changes on the course of temperature results in similar changes in the rate and amount of heat released. The maximum heat release rate of 230 kJ/ms was obtained for strategy 6 (Fig. 13). In the other cases this value was approximately 130 kJ/ms. The relative amount of heat used ( $X_i$ ) reaches the

### Three-phase fuel injection

The three-phase injection was realized in different strategies at the injection pressure  $P_{inj} = 5$  MPa. Only up to that value of pressure the fuel dose which does not exceed the value  $q_o = 36$  mg/injection could be obtained, at the injection duration not less than  $\Delta t = 0.3$  ms (design limitations). The increase in the injection pressure would have to result in much shorter injector opening times. Strategies 7-9 (Fig. 5) caused great differences in the value of the observed combustion pressure (Fig. 14). Injection advance of the first fuel dose (strategy 7) caused a better preparation of the air fuel mixture which resulted in high values of maximum combustion pressure ( $P_{max} = 42.6$  MPa).

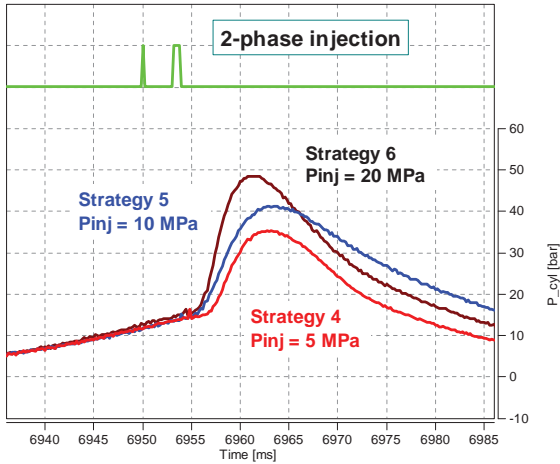


Fig. 10. Changes in the combustion pressure in the case of two phase fuel injection: fuel injection and injection duration were varied

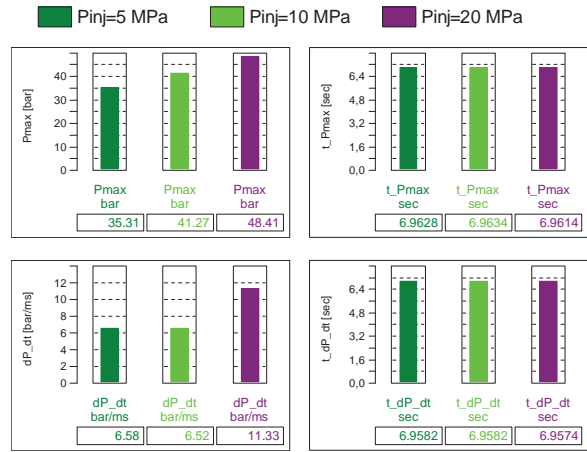


Fig. 11. Values of the indexes of the course of the combustion in the case of division of the fuel into two doses

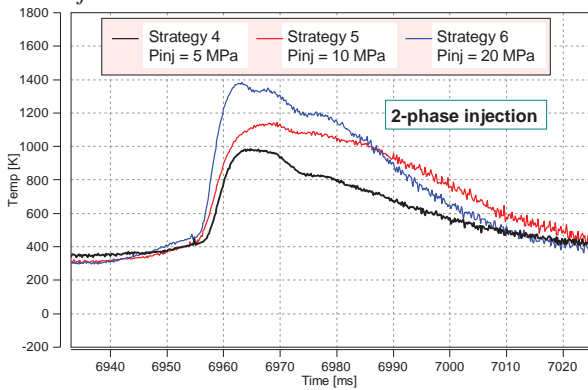


Fig. 12. Change in the temperature of the charge in the cylinder in the case of division of the fuel into two doses

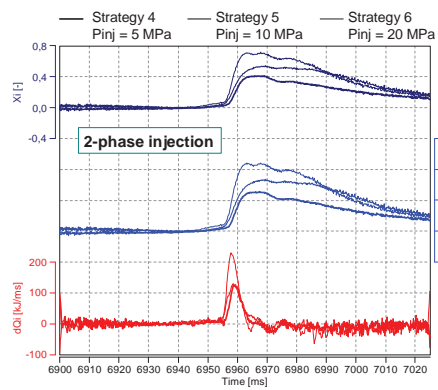


Fig. 13. Heat release ratio, heat release and relative value of the heat used for strategies 4, 5, 6

The post-injection of fuel after 1 ms from the end of injection of the second dose (strategy 9) results in the maximum combustion pressure reaching the level of  $P_{max} = 40$  bar. It is a higher value than in the case when the combustion takes place after the injection of the third dose (strategy 8), Fig. 15. From the above it results that an advantageous solution could be an injection of an additional fuel dose after the onset of combustion.

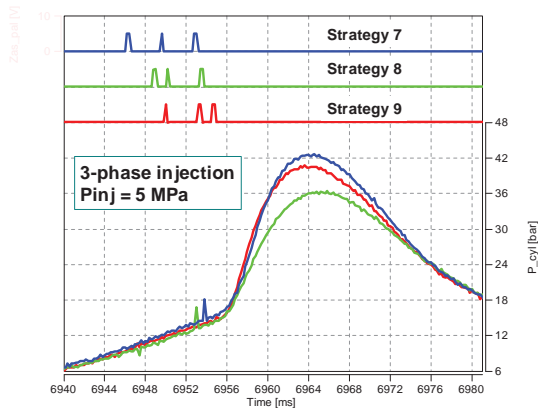


Fig. 14. Changes in the combustion pressure in the case of three-phase fuel injection: the moments of fuel injection of individual doses were varied at the same injection duration

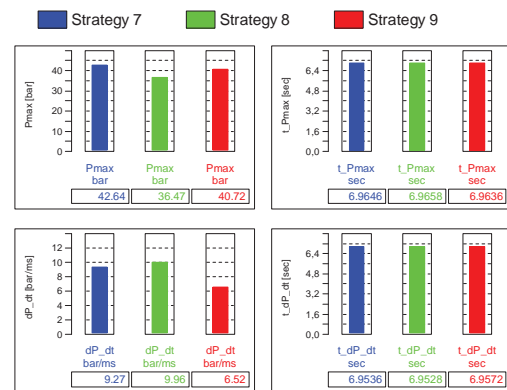


Fig. 15. Values of the indexes of the course of the combustion in the case the injected fuel division into three doses



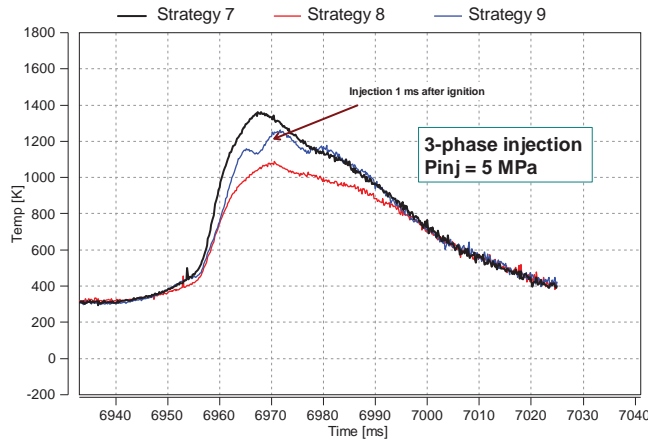


Fig. 16. Change in the temperature of the charge in the cylinder in the case of division of the fuel into three doses

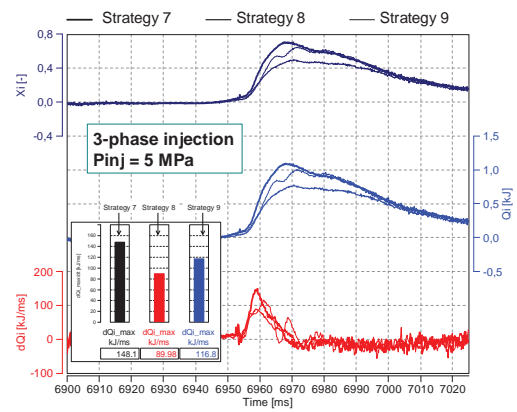


Fig. 17. Heat release ratio, heat release and relative value of the heat used for strategies 7, 8, 9

The authors did not carry out the measurement of the exhaust emissions, hence the emission level of e.g. PM is unknown after the selection of such an injection strategy, yet because of the injection of the fuel into the flame we can expect a growth in the emission of PM and hydrocarbons. From the course of the temperature (Fig. 16) it results that its values for strategy 9 are not the maximum ones. The increment of temperature is observable though after injecting of an additional fuel dose. The maximum values of the temperature (approximately 1400 K) were observed for strategy 6 (early injection of the first fuel dose, the final dose is the ignition one).

An additional fuel injection after the ignition results in lower values of the rate of heat release. A value of 116 kJ/ms was obtained, which confirms a rather slow combustion rate and expresses itself through a shift of its maximum towards later combustion (Fig. 17). Yet the value of the released heat ( $Q_i$ ) for strategy 9 shows tendencies to merge between strategies 7 and 8. This denotes advantageous thermodynamic consequences that result from the post injection during gasoline combustion. The maximum relative values of the heat transferred reach the level of approximately 0.7-0.75.

The thermodynamic analysis is insufficient for the extended qualitative and quantitative analysis of the course of the combustion. Further works were conducted with the use of optical methods, which allows determining the local values of the flame front temperature and also the quantitative evaluation of the direction and rate of flame development in the combustion chamber.

## 9. Conclusion

The obtained results lead to a conclusion that the rate of pressure increment after the ignition is higher if there are more fuel doses injected before the ignition of the main injection; it also grows along the growth of the pressure of the injected fuel. The rate of heat release is proportional to the cylinder pressure increment and depends on the same relations.

The combustion time strictly depends on the injection strategy: the growth of the number of fuel doses reduces the combustion time and an increase in the fuel injection pressure (better fuel atomization) allows a further reduction of the time of the flame occurrence.

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## **Definitions, acronyms, abbreviations**

ASOC	- After start of combustion
$dP/dt$	- Combustion pressure increment after the ignition
$dQ_i$	- Heat release rate
DI	- Direct Injection
EOC	- End of combustion
NEDC	- New European Driving Cycle
$P, P_{cyl}$	- Cylinder pressure
PFI	- Port Fuel Injection
$P_{inj}$	- Fuel injection pressure
$P_{max}$	- Maximum cylinder pressure
PM	- Particulate Matter
$q$	- Fuel dose
RCM	- Rapid Compression Machine
$Q_i$	- Heat release
SOC	- Start of combustion
T	- Temperature
t	- Time
$U_s$	- Internal energy
$X_i$	- Relative of heat use
$\lambda$	- Air excess coefficient