Influence of increased exhaust gas recirculation ratio on the thermodynamic processes in CI DI engine

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Abstract. The aim of the conducted research was the thermodynamic identification of combustion process involving up to 50–60% recirculated exhaust gasses in compression ignition engine. The values of the respective indicators obtained for the high share of exhaust gasses in the cylinder were compared to the values obtained in the engine working without EGR. The research was conducted on the single-cylinder AVL 5804 engine equipped with recirculated gas cooling system. The conditions of combustion process were determined using indicators of engine operation and measurements of fast-varying cylinder pressure. The evaluation of the influence of different share of exhaust gases in the cylinder on the combustion course and heat release was conducted. As a result of the conducted research the possibility of utilizing the high share of exhaust gasses (50–60%) with simultaneous ecological benefits, not only in relation to nitrogen oxides, was demonstrated.

1. Introduction and analysis of the literature

Requirements for meeting emission standards give rise to the need to seek new structural solutions in the construction of modern combustion engines and engine exhaust aftertreatment systems. Structural methods to meet the stringent energy and emissions requirements (with regard to the emission of CO₂) by propulsion units are known as the concept of downsizing [1] or hybridization of propulsion systems [2]. Among these solutions the most important is the role of the modern combustion and exhaust gas recirculation systems [3–5]. Exhaust gas recirculation, which is supplying of exhaust gases into the combustion chamber, is one of the methods to reduce nitrogen oxides emissions, especially for the medium loads. In certain scopes of engine operation it also allows to reduce fuel consumption without significant deterioration of the operating indicators. The efficiency of the exhaust gas recirculation systems, however, depends to a large extent on the way they are accomplished. Supplying a given amount of exhaust gases into the operating cylinder can be achieved in two ways:

• using the internal recirculation (IGR—internal gas residual), which involves retaining in the cylinder a part of the exhaust gases from the previous cycle by appropriate control of variable valve timing;
• by the use of an external recirculation system (EGR—exhaust gas recirculation), returning the part of the exhaust gases leaving the cylinder (cylinders) from the previous engine cycle.

Despite numerous possibilities of EGR systems application, still a major concern is the method of their accomplishment and the control over the amount of recirculated exhaust gases. Figure 1 shows the division of the cylinder exhaust gas supply systems into internal systems of exhaust gas recirculation...
and external systems supplying the exhaust gases into the cylinder. The former can be accomplished in two ways: by variable valve timing and by variable valve lift. Both methods of regulation involve retaining in the cylinder a part of the exhaust gas from the previous engine operation cycle. The external exhaust gas recirculation systems can be divided into high-pressure (1, figure 1), low-pressure (2) and mixed systems. The external exhaust gas recirculation can take place before the exhaust aftertreatment system—by the so-called “dirty EGR” (2a on figure 1) and after the aftertreatment system—by the so-called “clean EGR” (2b on figure 1).

![Figure 1. Systematics of the cylinder exhaust gas supply systems [3, 4, 6].](image)

From the analysis of literature [7–10], it ensures that one of the most beneficial solutions for controlling thermodynamic processes in CI engines is low-temperature combustion (LTC), which causes the reduction of emissions of nitrogen oxides and particulate matter (soot). Such a process is accomplished with participation of significant share of recirculated exhaust gases reducing the maximum combustion temperature. This process can be supported by fuel dose splitting and a reduction of the compression ratio, although in the LTC combustion system it is also possible to use single doses of injected fuel. Obtaining the LTC type combustion requires the appropriate control of the injection time in the close to the piston top dead centre (TDC) while applying a large quantity of recirculated gases (50–70%) [11, 12]. Such a large share of the exhaust gases in a charge, what also mean significant share of chemical inert molecules instead active molecules of oxygen, leads to increased self-ignition delays (due to the longer time of mixing), resulting in an increased homogeneity of the mixture.

An important aspect of the exhaust gas recirculation systems use is their impact on the volumetric efficiency of fresh air in the cylinders. As it was indicated in [13], that the introduction of 25% share of EGR reduces the value of fresh air volumetric efficiency by about 14%. Within certain scope of partial loads, an increase in thermal efficiency and reduction of specific fuel consumption were observed. The discussed decrease in the volumetric efficiency of fresh air can be compensated by changing the supercharging parameters, as a result of which there will be a further increase in thermal efficiency. The vast majority of published research findings on the EGR systems include works with the maximum participation of the exhaust gases of 25–30%, which raised the need to complement our knowledge on the matter in terms of a larger share of the recirculated exhaust gases.

The studies conducted up to now by [14, 15] as well as by Knight [16] indicate only the influence of EGR amount on the exhaust gas emission. However, the influence of the recirculated exhaust gases on the thermodynamic parameters of the combustion engine in selected operating points is omitted. In the cited works the research involved one selected point of engine operation \((n = 1500 \text{ rpm}, P_{\text{inj}} = 85 \text{ MPa})\) taking into account, however, its variable supercharging. It was proved that for large EGR share of 50–60% it is possible to obtain reduced emissions of NO\(_x\) and soot; those conclusions confirmed the expectations of the authors of these works.

The thermodynamic analysis of the combustion process was conducted only in few studies limited to the systems operating with the share of the recirculated exhaust gases of about 36% [17]. The authors of this study conducted the research on the thermodynamic characteristics of the combustion process in the CI engine with significant share of recirculated exhaust gases of up to 60% [18].
2. The research problem
The basic research problem formulated by the authors was the explanation of the significance of the influence of the recirculated exhaust gases amount on the change in the basic parameters and indicators of the thermodynamic engine cycle. It was decided to demonstrate the extent to which the increase in the share of exhaust gases in the closed cylinder affects the variation of the compression-ignition engine operation on the basis of fluctuations in the charge pressure during combustion, the variability of mean indicated pressure and temperature of the charge and exhaust gas. Also the changes in the heat release ratio and the angular position of the stipulated centre of combustion (CoC, position of the point of 50% heat release) were assessed. It was also expected that the possibility and the scope of reduction in the nitrogen oxides and particulate matter (soot) emissions for the low temperature combustion will be explained.

3. Research methodology

3.1. Research method
In order to explain the issues and research problems defined in section 2 it was decided that the method of experimental test bench would be utilized. To this purpose, the engine test bench with electrical dynamometer brake was appropriately adapted to enable maintaining the constant engine speed. On the test bench was installed a tested object which was a direct injection compression-ignition engine.

3.2. Research object
The study was carried out on a single-cylinder AVL 5804 test engine equipped with the system of cooling recirculated exhaust gases. Engine technical specification is shown in table 1. The asynchronous electric dynamometer brake AMK ASYN of DW13-170-4-AOW type was used. The EGR system was equipped with two solenoid valves controlled by the pulse-width modulation (PWM), which enabled repeatable setting of the valves position on the air inlet and exhaust gas outlet. The system of two valves made it possible to obtain the exhaust recirculation ratio of up to 60%. The amount of recirculated exhaust gases was determined by volume on basis of carbon dioxide volumetric concentration measurement in the air in the intake channel and in the engine outlet according to the equation:

\[ \text{EGR} = \left( \frac{\text{CO}_2\text{-ex} - \text{CO}_2\text{-in}}{\text{CO}_2\text{-ex}} \right) \cdot 100\% \]  

where: \( \text{CO}_2\text{-ex} \) – the percentage of carbon dioxide in exhaust gases, \( \text{CO}_2\text{-in} \) – percentage of \( \text{CO}_2 \) in the engine outlet channel.

| Parameter                  | Value                                      |
|----------------------------|--------------------------------------------|
| Engine capacity            | 510.7 cm\(^3\)                            |
| Piston stroke              | 90 mm                                      |
| Cylinder diameter          | 85 mm                                      |
| Compression ratio          | 16.2                                       |
| Injector type              | piezoelectric, common rail, 8-hole, \( d = 0.117 \text{ mm} \) |
| Cylinder head gasket thickness | 2 mm (required to obtain expected compression ratio) |

3.3. Measuring equipment
The test stand (figure 2) was equipped with control and measuring apparatus including: (a) a system for fast-varying processes measurement AVL IndiCom 621 enabling measurement of pressure in a cylinder \( P_{\text{cyl}} \) with the use of pressure sensor AVL GH14D with the measuring range 0–250 bar and sensitivity of 18.84 pC/bar, (b) system for data acquisition AVL Concerto, (c) system for control of the fuel injection process enabling the control of the injection time and injection angle with resolution of \( \Delta \alpha = 0.5 \text{ CA} \) and fuel pressure up to 200 MPa developed by Mechatronika Poland, (d) the system controlling...
settings of the solenoid valves for exhaust gas recirculation with the use of PWM signals with resolution of 1% developed by Mechatronika, e) the system of measuring gaseous components—Horiba Mexa 7100D for type approval testing, measuring concentration of CO, THC (HFID), NOₓ, CO₂, (f) the system measuring particle matter (soot)—Micro Soot Sensor by AVL.

**Figure 2.** Diagram of the test bench with testing apparatus and characteristic parameters of the process.

### 3.4. The methodology and scope of the research

The tests were conducted for two measurement points for which, at varying amount of recirculated exhaust gases, were defined: (1) indicators of the combustion engine operation, (2) thermodynamic indicators of the combustion process, (3) emissions of toxic components of exhaust gases. The indicators of combustion engine operation included (figure 3): (1) maximum pressure in the cylinder—\( P_{\text{mx}} \) and the phase (angle of CA) of its occurrence \( A P_{\text{mx}} \), (2) mean indicated pressure—designated as \( p_i \); (3) the maximum pressure rise after self-ignition \( R_{\text{mx}} \) and the phase (angle of CA) of its occurrence \( A R_{\text{mx}} \).

There were also determined: (1) exhaust gas temperature—\( T_{\text{exh}} \), (2) exhaust gas temperature at the valve of recirculation before the inlet channel—\( T_{\text{EGR}} \), (3) temperature of the charge at the inlet (air and exhaust gas mixture)—\( T_{\text{EGR+air}} \).

Thermodynamic indicators of engine operation included: (1) the start of combustion (SOC)—defined as the beginning of the positive heat release value during compression, (2) angle of combustion of 5% of a fuel dose—MBF5 (Mass Burnt Fuel), (3) angle of occurrence of the centre of combustion (CoC) which is combustion of 50% of fuel dose—MBF50, (4) angle of the end of combustion (EoC) defined as combustion of 90% of fuel dose—MBF90.

Specific emission of fuel components (in g kWh⁻¹) was defined, with the use of engine operation indicators, on the basis of concentration measurements of CO, HC, NOₓ in ppm and carbon mass C in mg m⁻³ as:

\[
e_{\text{C}} = C \ p_{\text{exh}} (G_e + G_{\text{air}}) \ p_i^{-1} 
\]

\[
e_{\text{CO}} = 0.000966 \ \text{CO} \ (G_e + G_{\text{air}}) \ p_i^{-1} 
\]

\[
e_{\text{HC}} = 0.001578 \ \text{HC} \ (G_e + G_{\text{air}}) \ p_i^{-1} 
\]

\[
e_{\text{NO}} = 0.001519 \ \text{NO}_x \ (G_e + G_{\text{air}}) \ p_i^{-1} 
\]
Selection of indicators analysed during tests on diesel fuel combustion with significant share of exhaust gases.

Using the results of indicator tests determined were also the rate of heat release – $\frac{dQ}{d\alpha}$ and accumulated heat $Q$ as:

$$
\frac{dQ}{d\alpha} = \frac{\kappa}{\kappa-1}\left(P_{\alpha}+P_{\alpha+1}\right)\left(V_{\alpha+1}-V_{\alpha}\right)+\frac{1}{\kappa-1}\left(V_{\alpha}+V_{\alpha+1}\right)\left(P_{\alpha+1}-P_{\alpha}\right)
$$

where the indexes $\alpha$ and $\alpha + 1$ are the CA current and the subsequent values of pressure in cylinder ($P$) or their corresponding volume of cylinder ($V$).

The tests were carried out for two values of engine speed: 1000 rpm and 1500 rpm for different diesel fuel doses: $q_o = 12$ and $q_o = 23$ mg inj$^{-1}$. The tests were carried out so as to maintain relatively constant angle of combustion of 50% of fuel dose – the angle is called the centre of combustion (CoC). In the analysed operating conditions this angle amounted to, respectively: 9 and 11º of CA after TDC. Without taking into account this condition, the fuel burning would be much delayed and the correct operation of the engine would be impossible. Conditions of conducting measurements are presented in table 2.

| No. | $n$ (rpm) | $q_o$ (mg inj$^{-1}$) | CoC (deg aTDC) | $P_{ni}$ (MPa) | $P_{in}$ (hPa) | Share of exhaust gases (%) |
|-----|-----------|----------------------|----------------|----------------|----------------|---------------------------|
| 1   | 1000      | 12                   | 9              | 44             | 1024           | 0; 9; 19; 30; 45; 51       |
| 2   | 1500      | 23                   | 9              | 1120           | 1219           | 0; 9; 21; 30; 42; 50; 60   |

4. Estimation of indicators of engine operation for different shares of exhaust gases

4.1. General indicators

Before proceeding to data analysis, the characteristics of pressure in cylinder were smoothed with the use of low-pass filter of 4 kHz. Supplying exhaust gases for the subsequent cycle of engine operation has dominating influence on the composition of the exhaust gases and indicators of engine operation. The analysis of the tests results was conducted in terms of general, thermodynamic and ecological indicators.

The dependency determining the influence of the amount of recirculated exhaust gases in the engine cylinder on the general indicators of its operation is shown in figure 4. For smaller fuel dose ($q_o = 12$ mg inj$^{-1}$) the highest value of the maximum combustion pressure $P_{mx}$ occurs at 20% share of exhaust gases.
gases and amounts to 54.6 bar. Introduction of 10% share of EGR caused a decrease of $P_{\text{max}}$ to 53.2 bar (a change by approx. 3%), without any influence on the value of the mean indicated pressure—$p_i$. Its higher value increases from 2.9 bar to 3.7 bar along with the increasing share of recirculated exhaust gases (up to the value of 50%). It means a change of the $p_i$ value by more than 25%. It can be concluded, therefore, that an increase in the share of exhaust gases to 50% causes an increase in the mean indicated pressure by about 25%.

![Figure 4](image)

Figure 4. The main indicators of engine operation in relation to the share of recirculated exhaust gases in the engine: (a) at $n = 1000$ rpm and $q_o = 12$ mg inj$^{-1}$, (b) at $n = 1500$ rpm and $q_o = 23$ mg inj$^{-1}$.

At the same time, along with the increase of the EGR share, the pressure increase rate decreases which results in the decrease of the maximum pressure to the value of 53.4 bar for the share of recirculated exhaust gases ranging from 20% to 50%. Simultaneously the changes of the occurrence angle of the maximum pressure $AP_{\text{max}}$ and the angle of the maximum pressure increase rate $AR_{\text{max}}$ were observed for the crankshaft speed 1.5 deg.

Use of a larger dose and higher engine speed result in a change in the relationship between these parameters. Along with the increase of the EGR share, the value of the maximum combustion pressure $P_{\text{max}}$ and the value of the pressure increment $R_{\text{max}}$ decrease. The highest value of the mean indicated pressure of 6.7 bar and the occurrence angle of the maximum pressure $AP_{\text{max}}$ are observed for 30% share of EGR. A change in the share of recirculated exhaust gases does not affect the angle of the maximum increase rate occurrence of the pressure $AR_{\text{max}}$ and amounts to 10.5 deg CA.

The results of the analysis concerning the impact of the recirculated exhaust gases share on the excess air ratio and exhaust gas temperature are shown in figure 5. With the increasing share of exhaust gases in the cylinder, the excess air ratio decreases irrespective of the applied fuel dose. During combustion of a large fuel dose the global excess air ratio achieved is below 1.0. These are conditions rendering combustion of diesel oil difficult.

During a smaller fuel dose combustion, the exhaust gases temperature decreases and the temperature of recirculated exhaust gases increases along with the increase of their share in the cylinder charge. This increase is associated with an increased flow rate of the exhaust gases. An increased share of exhaust gases with higher temperature results in an increase of the mixture of air and exhaust gases temperature ($T_{\text{EGR+air}}$) irrespective of the applied fuel dose. The highest exhaust gases temperature occurs at 30% share of EGR. For this point of operation was obtained the largest value of the mean indicated pressure, which is shown in figure 5b.
4.2. Thermodynamic indicators

The research on combustion process with significant share of recirculated exhaust gases was conducted mainly based on the analysis of thermodynamic parameters of this process. The changes in those parameters during engine operation for two different fuel doses are presented in figure 6 where the angle dependencies of the start (MBF5), centre (MBF50) and end of combustion (MBF90) and the start of ignition (SOC) are shown. Independent of the fuel dose amount, an increase in the share of recirculated exhaust gases results in the delayed start of combustion (SOC). The value (angle) of 50% combustion of fuel dose (CoC) was maintained on the assumed level according to the fuel dose: 10ºCA (for reduced dose) and 12ºCA for combustion of a larger dose. The variation for both strategies was below 1ºCA which should be considered an appropriate value. Maintaining such characteristics of the combustion process for an increased share of the recirculated exhaust gases requires acceleration of the start of combustion, which causes elongation of the whole process. Such dependencies were confirmed for both fuel doses.

During the tests the analysis of the cylinder pressure characteristics was conducted, which is presented in figure 7. The use of recirculated exhaust gases for engine speed of \( n = 1000 \) rpm causes initially (for
10% EGR) significant delays in pressure increase after self-ignition. Further increase of the exhaust gases share in the cylinder only slightly deviates from the characteristics of pressure at the lack of exhaust gases. The analysis of this operation indicates point insignificant decrease in the maximum pressure in the cylinder for increased exhaust gases share (which was also confirmed in figure 4a). The second measurement at the engine speed of \( n = 1500 \) rpm, depending on the exhaust gases share in the cylinder, was characterised by a higher variability compared to the first measuring point. Here are observed both, changes in the pressure value before combustion and maximum pressure changes. An increase in the exhaust gases share resulted in prolonging the start of combustion, which corresponds to the almost flat cylinder pressure characteristics for the range from 4–8º CA after TDC. The maximum value change of \( P_{\text{cyl}} \) at EGR = 0% and 60% achieved the value of 7 bar.

![Figure 7.](image)

**Figure 7.** Analysis of the maximum cylinder pressure changes for engine speed of: (a) \( n = 1000 \) rpm, (b) \( n = 1500 \) rpm.

Introduction of exhaust gases recirculation resulted in an increase in the heat release rate in the cylinder (figure 8). Increase in the share of exhaust gases during small fuel dose combustion leads to insignificant limitation in the heat release rate for small and medium exhaust gases share. During fuel combustion with maximum exhaust gases share the limitation reaches the value of 20%. The occurrence angle of the maximum heat release rate does not undergo significant changes and amounts to 8.5–10º CA after TDC.

![Figure 8.](image)

**Figure 8.** Analysis of the heat release rate values for the speeds of: (a) \( n = 1000 \) rpm, (b) \( n = 1500 \) rpm.

Combustion of a large fuel dose with a recirculated exhaust gases share leads to significant changes in the heat release rate. The changes in the heat release rate obtain values exceeding 50%. It means a two-fold increase in the values of changes compared to the small fuel dose combustion.
5. Summary
The conducted research confirmed that the variable amount of exhaust gases in the cylinder results in changes in the engine operation parameters with significant variation of the trends for particular components. It forces the necessity to find the optimum solution for a particular application.
A noticeable drop in the maximum pressure in cylinder for an increased exhaust gases share, especially during larger fuel dose combustion, does not result in a decrease in the mean indicated pressure. The $P_{mx}$ parameter achieves maximum values at 20% share of exhaust gases, and the mean indicated pressure value for this range stays at the constant level compared to other measuring points. Only for the 30–45% share of exhaust gases an increase in this indicator might be observed. The exhaust gases share amount has no essential influence on acceleration or the time delay of the maximum pressure in cylinder occurrence in relation to the crankshaft angle (for the assumed value of CoC).
On the ecological analysis basis it might be concluded that combustion of fuel with a large share or recirculated exhaust gases is efficient only when the excess air ratio at large exhaust gases share (over 40%) is much higher than 1.0. Such a situation was obtained only for a small fuel dose combustion. During a large fuel dose combustion the increase in the engine supercharging value with air would be necessary.

Symbols and abbreviations
- $AP_{mx}$: crankshaft angle of maximum cylinder pressure
- $AR_{mx}$: crankshaft angle of maximum rise of cylinder pressure
- CoC: centre of combustion (MBF50)
- CoV: coefficient of variation
- $dQ$: heat release rate
- $dQ_{mx}$: maximum of heat release rate
- EGR: exhaust gas recirculation
- ID: ignition delay
- IGR: internal gas residual
- $n$: engine speed
- $p_i$: indicated mean effective pressure
- LTC: low temperature combustion
- MBF: mass burnt fuel
- $P_{cyl}$: cylinder pressure
- $P_{mx}$: maximum cylinder pressure
- $P_{inj}$: injection pressure
- $R_{mx}$: maximum rise of cylinder pressure
- SOC: start of combustion
- $T_r$: throttle
- $t$: time characteristic for the combustion phase
- $T_{EGR+air}$: mixture temperature (air and EGR)
- $T_{EGR}$: temperature of recirculated gases
- $T_{exh}$: exhaust gases temperature
- $q_o$: fuel dose
- $Q$: heat release
- $\lambda$: air excess ratio

References
[1] Pielecha I, Cieślik W, Borowski P, Czajka J and Bueschke W 2014 Comb. Eng. 159 p 12–25
[2] Pielecha I, Cieślik W, Borowski P, Czajka J and Bueschke W 2014 Comb. Eng. 158 p 23–35
[3] Berg T, Thiele O, Seefeldt S and Vanhaelst R 2013 Mag. MTZ 6 p 472–7
Acknowledgements
The research presented in this paper was performed within the European Research Project PowerFul FP7, grant agreement No. 234032, POWERtrain for Future Light-duty vehicles.