Experimental Investigation of Sources of Influence of Exhaust Gas Recirculation on the Spark Ignition Combustion

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ABSTRACT

Modern internal combustion engines (ICE) are designed so that they emit pollutants under limited value and low CO_2 emissions, while delivering the required performance. Due to the requirements for higher efficiency and lower emissions, modern spark ignition (SI) engines are optimized for maximum efficiency which is usually close to the limit of the knock ignition. To enable further developments, knock avoiding strategies have to be further investigated. One of the methods to suppress knock is to lower the temperature of the end-gas, because engine knock occurs when the end-gas auto ignites. Cooled exhaust gas recirculation (EGR) has proven to be a promising way to reduce end-gas temperature. Furthermore, EGR described in this study, is shown as one of the most effective techniques for reducing engine out emission of nitrogen oxides (NO_X) in SI engines.

In this research the influence of the main sources of knock suppression by using EGR in the SI engine are characterised. The factors that influence knock suppression are flame propagation speed, chemical influence on auto ignition (CO_2, H_2O, dilution etc.) and end-gas temperature influence. The study is performed by comparison of the SI operation at specific load with and without EGR. Further on, experimental study is carried out for three cases. First case is normal operating condition of engine without the application of EGR. In the second case cooled EGR is applied and finally in the third case engine is operated with application of the EGR and higher intake air temperature obtained with air heater. Higher intake temperature is applied in order to equalize the maximum temperature of the end-gas obtained in first case (the case without EGR) in order to determine the temperature influence on knock suppression. Since for the given cases the amount of fuel supplied to the engine is supposed to be constant, the constant fresh intake charge is maintained by optimization of the intake pressure.

Results presented in this study characterize how end-gas temperature influences the knock occurrence when cold EGR is applied. Furthermore, it describes one of the ways to reduce the knock occurrence in the engine cylinder by reducing end-gas temperatures.

All tests are performed on an in-house developed experimental setup that uses a modified single cylinder diesel engine HATZ 1D81Z that was converted for running in SI operation.

KEYWORDS

SI Engine, EGR, CoV, Heat release, Experimental engine, Knock

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NOMENCLATURE

ICE	Internal Combustion Engine	ST	Spark Timing
SI	Spark Ignition	AH	Air Heater
EGR	Exhaust Gas Recirculation	IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closure	CoV_{IMEP}	Coefficient of Variation of IMEP
°CA	Degrees of Crank Angle	CO_2	Carbon Dioxide
ATDC	After Top Dead Centre	NO_X	Nitrogen Oxides
CA50	Combustion phasing, 50% fuel mass burned	IR	Infrared
MAPO	Maximum Amplitude of Pressure Oscillation		

INTRODUCTION

Environmental awareness turns focus for development of motor vehicles to prevention of future global pollution. Greenhouse gases are proven to be one of the main reasons for global warming [1,2]. The internal combustion (IC) engines are today designed to deliver required performance with limited emission of pollutants and CO_2. That implies that the efficiency of IC engines is becoming one of the primary development targets because CO_2 emission is shown to be proportional to fuel consumption [3]. In order to achieve higher efficiency in modern spark ignition (SI) engines several technologies are used: turbocharging, combustion chamber optimization, direct injection etc. One of the main technologies, turbocharging, reduces the ratio of friction losses by the increase of load, but at the same time it increases the tendency of the engine to knock. Therefore, one of main obstacles in furtherer development of SI engines and in further increase of the engine load is the occurrence of knock. Literature showed that exhaust gas recirculation (EGR) which was first implemented for reduction of NO_X emissions might be a good method to suppress knock [4,5]. As shown by some researchers the EGR influences the knock suppression by three main factors: influence on flame speed, chemical influence on auto ignition and influence on end-gas temperature [6].
The second influence that EGR has on SI combustion is the influence on NO_X formation. EGR is one of the most effective techniques for reducing engine out NO_X emission in SI engines [6]. Suppression of NO_X formation is achieved by dilution, lower in-cylinder temperature and reducing oxygen concentration [7]. When NO_X is concerned it has to be mentioned that it has been observed that NO_X on the intake that comes with EGR can increases engine knock tendency [8]. This influence can be considered as the chemical influence on auto ignition, but it is not the only chemical influence, as it is well known that CO2 and H2O also have chemical influence on auto ignition. Even though the use of EGR in SI has been researched [9] and there are some publications regarding its use, there is no comprehensive and systematic experimental analysis of the sources of EGR influence on SI combustion, knock occurrence and NO_X emissions. This study aims to systematically evaluate the influence of the end gas temperature which changes with the addition of EGR on tendency of engine towards knock and NO_X formation.
The study is performed by experimental tests that used a new approach where intake temperature is varied by EGR and air heater (AH). The methodology is based on SI engine with EGR system and air heater placed on the intake that can compensate the temperature change obtained by the application of cooled EGR. With air heater the temperature influence of cooling the combustion chamber by cold EGR is annulled and in another case increased to show the influence of higher temperatures on knock occurrence.

EXPERIMENTAL SETUP

The experimental work is performed on the experimental setup at Laboratory for IC Engines and Motor Vehicles of Faculty of Mechanical Engineering and Naval Architecture in Zagreb, shown in Figure 1**.**

Figure 1. Experimental setup, engine testbed

The experimental setup presented in this paper consists of AC Dyno, SI Engine upgraded with EGR, intake air heater, indicating equipment, emission sampling, air flow and fuel consumption measurement. Additionally temperature and pressure are measured on the intake, exhaust, engine head and oil.

Acquisition of data from SI engine in various operating conditions was enabled by the use of in house built acquisition system which is prepared so that it stores the data of boundary conditions and emissions.

The IC engine used in this experimental setup is a HATZ 1D81Z, originally a single cylinder diesel engine. It has two valves per cylinder and combustion chamber of a toroidal type. In order to run as SI engine the modifications were made to the piston, ignition and fuel injection systems. Also for the purposes of engine control the measurement of piston position was added. Main characteristics of the modified experimental engine are listed in Table 1.

Table 1. *Characteristics of experimental engine*

Manufacturer	**Hatz, 1D81Z**
Engine type	1 cylinder, 4 stroke
Bore, mm	100
Stroke, mm	85
Combustion chamber	Toroidal
Displacement, ccm	667,59
Compression ratio	12
Intake Valve Timing	Open@340°, Close @ 590°
Exhaust Valve Timing	Open@154°, Close @ 380°

Modification of the piston included machining of the piston top that resulted in lowered engine compression ratio (CR) required for the engine to operate with gasoline in SI combustion mode. CR was lowered from 20.5 to 12. Further on, for the measurement of piston position two hall sensors, one on crankshaft and one on camshaft were implemented. Based on the piston position that was measured the engine controls in terms of fuel injection and ignition timing were operated. For the purpose of control, the signals of fuel injection and ignition were monitored by the engine indicating equipment. The operation of the injection and ignition system was enabled by the in-house programme prepared in LabVIEW. *BOSCH ZS-K-1X1PME* ignition coil with spark plug *NGK IRIDIUM CR7EIX* was placed centrally on top of the cylinder where diesel injector was removed. Gasoline port fuel injector (*BOSCH EV-6-E*) was placed on custom made intake manifold with constant fuel pressure of 3 bar and with fuel mass consumption measured by *OHAUS Explorer* mass scale. Additionally, intake manifold contains ports for temperature and CO_2 emission sample measurement.

Besides the equipment for upgrading the engine, additional testbed equipment was used on experimental setup to set up and measure boundary conditions. Intake air mass was measured by the *TSI 2017L* laminar mass flow meter. In order to be able to control the intake and in-cylinder temperature after the measurement of air flow the intake line was heated by 18 kW *OSRAM SYLVANIA SureHeat* Air Heater.

Further on, for the exhaust gas emissions two different analysers were used. Acquisition of the emission of CO and CO_2 is done by *Bosch ETT 8.55 EU* analyser, while Total Hydrocarbons (THC) are measured by *Environnement GRAPHITE 52M* heated FID analyser.

EGR flow was controlled by the EGR valve (*Valeo 170A9*) with water cooled heat exchanger. The amount of EGR is controlled by opening of electric valve allowing exhaust gas to recirculate from exhaust to the intake manifold. The amount of recirculated EGR is taken as the same proportion as the ratio of the measured CO_2 at the intake in relation to the measured CO_2 in the exhaust. This is done by using two CO_2 analysers. The first one is already described Bosch analyser, placed on the exhaust manifold. For measurement of the intake CO_2 *Environnement MIR 2M* infrared (IR) analyser was used. The amount of the recirculated exhaust gas is calculated by the equation:

$$EGR(\%) = \frac{[CO_2]_{Intake\ gas} - [CO_2]_{Ambient}}{[CO_2]_{Exhaust\ gas} - [CO_2]_{Ambient}} \tag{1}$$

where $[CO_2]_{Ambient}$ is the concentration of CO_2 in ambient air, $[CO_2]_{Intake\ gas}$ is the measured concentration of CO_2 in the intake manifold and $[CO_2]_{Exhaust\ gas}$ is the CO_2 measured in the exhaust gases [9].

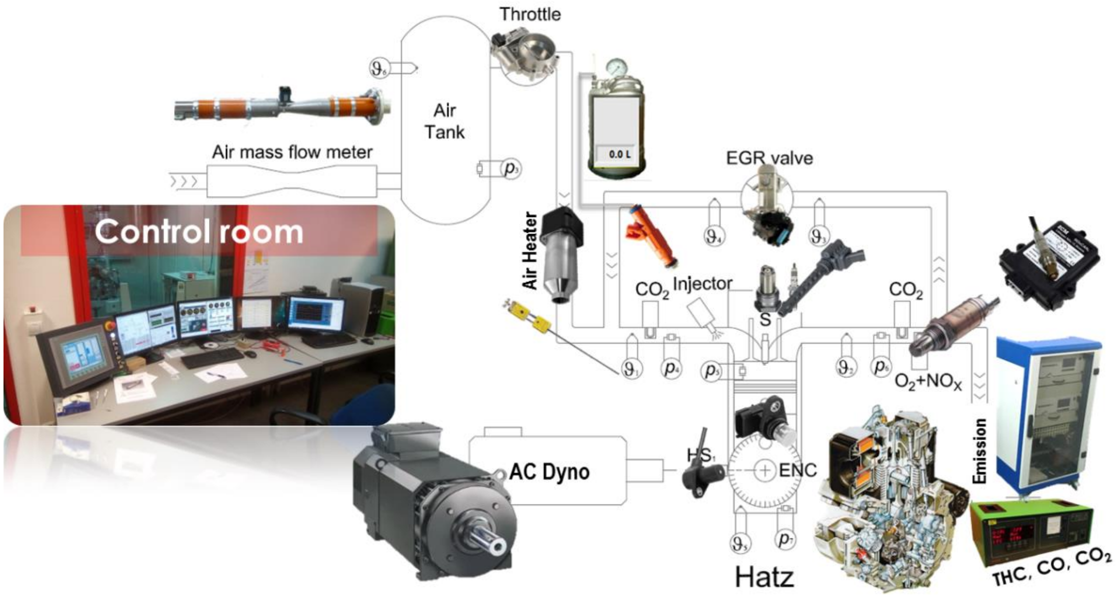

Figure 2. *Experimental SI Engine testbed scheme*

Fast crank-angle based signals typical for combustion engines are acquired and processed by engine indicating equipment. For the hardware *AVL IndiSmart 612* and for the software

Indicom were used [10,11]. Main data that was acquired with this equipment was combustion data from the engine, e.g. in-cylinder pressure, intake pressure and crank position. Further on software calculated and enabled real time monitoring of the indicated mean effective pressure (IMEP), coefficient of variation of IMEP (CoV_{IMEP}) and combustion phasing (CA50). Also during measurements monitoring of the knock occurrence, spark timing (ST), dwell time, injection timing, spray amount and intake pressure at intake valve closure (IVC) was possible.

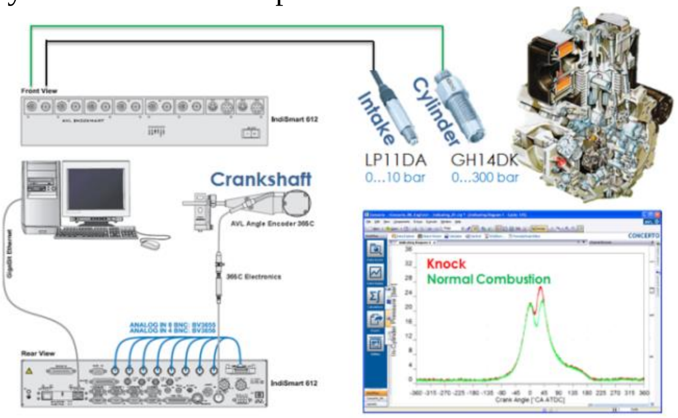

Figure 3. *Engine indicating system*

For measurement of intake pressure the low pressure *AVL LP11DA* sensor was used, while for high in-cylinder pressure the sensor was *AVL GH14DK*. For the measurement of the high pressure part of the cycles data was acquired with the resolution of 0.1 °CA, while the rest of the cycle was recorded with the resolution of 0.5 °CA, all for 300 consecutive cycles.

EXPERIMETAL WORK, RESULTS AND DISCUSSION

Experimental work is performed at 1600 rpm with and without EGR dilution, and in order to have comparable results some adjustments in terms of boundary conditions were made. As the dilution was applied (introduction of EGR), it replaced part of the fresh intake air/fuel mixture and therefore reduced the energy supplied to the cylinder. That effect was compensated by the increase of intake pressure which enabled the same air/fuel mixture flow as when there was no EGR in the system. The air to fuel mixture was set to stoichiometric in all conditions which resulted in the fuel flow that on average gave the energy of the fuel of 1432 J/cycle. The additional increase of the intake pressure was made in order to compensate the loss of intake charge as a consequence of reduction in density for the cases with the increased intake temperature. Intake pressure for the operating points used in this study is shown in Figure 4.

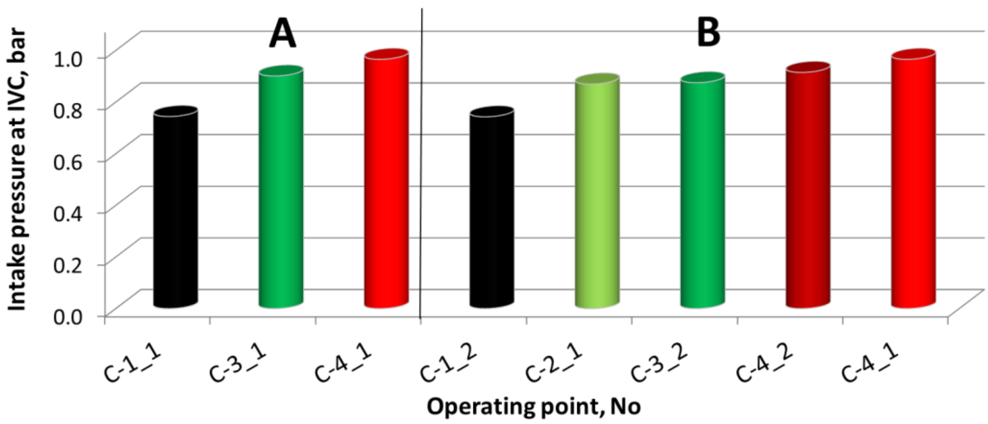

Figure 4. Intake pressure at intake valve closure (IVC) for the selected operating points

The experimental research resulted in 140 acquired data points (Figure 6). From that set of data, several operating points were selected for further analysis. Selected operating points were labelled for different measured cases (C-1_1, etc.) and were grouped in two groups A and B. The labels of cases have two numbers. The firs number represents case series, while the second number represents the case number within that case series.
The first case series was a series of normal operating conditions of the engine without the application of EGR and with intake temperature of 22 °C. The combustion phasing CA50 for the operating points of the first case series was around 16 and 21 °CA ATDC (Degrees of Crank Angle, After Top Dead Centre) and caused by spark advance of 14 and 10 °CA BTDC respectively. Intake pressure at IVC was kept constant at 0.75 bar (Figure 4). The second case series included dilution of intake charge with 15% of cooled EGR. The EGR was cooled with integrated EGR water cooler to 18 °C (C-2_1) and resulted with cooler intake charge than in first case series. In the third case series (C-3_1 and C-3_2) 15% EGR dilution was implemented but with intake air temperature increased to compensate for the effect of cold EGR and to maintain the temperature of the intake mixture the same as in first case series (22 °C). The combustion phasing CA50 for the operating points of the third case series was 12.5 and 20.7 °CA ATDC caused by spark advance of 44 and 31 °CA BTDC respectively. Finally for the selected operating points C-4-1 and C-4_2 of the fourth case series, intake temperature was heated to higher temperatures (Figure 5) in order to research the higher EGR temperature influence. Combustion phasing in the fourth case series was 20.5 and 20.9 °CA ATDC with same spark advance of 20 °CA BTDC. The group of data A contains operating points with the same peak end-gas temperature and different CA50, while group B represent operating points with the same combustion phasing CA50 and different peak end-gas temperature.

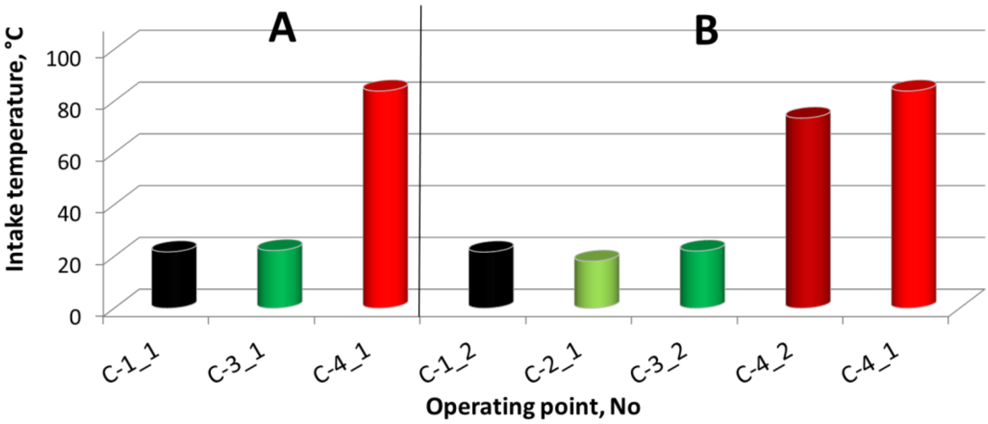

Figure 5. Intake mixture temperature for the selected operating points

The main results that were observed during the measurement and post processing were end-gas temperature during combustion and the occurrence of knock. The end-gas temperature during combustion was calculated during post processing in order to study its effect on the auto ignition of the end gas which is the source of knock.

As noted before, the main study of the paper is to show the EGR influence on knock suppression caused by temperature change, also called temperature effect of the EGR on auto ignition. The results showed that for the fixed CA50 the end-gas temperature change as the EGR is introduced to the system. Figure 6 shows the results of the peak end-gas temperature in dependence of CA50 for all measured operating points, with highlighted selected operating points from four cases described earlier. It can be noticed that as EGR is introduced to the system peak end-gas temperature decreases for the same spark timing. The increase of peak end- gas temperature at same CA50 can be obtained by heating of the intake charge, while for different CA50 it is possible to have same peak end-gas temperature by different intake temperature. As mentioned before the results are grouped in two groups. Group B represent operating points that have the same combustion phasing (CA50), but different intake temperature and different amount of EGR (0 and 15%). As a result of these conditions they have different end-gas temperatures. Group A represent operating points with similar peak end-gas temperature, but with different combustion phasing, amount of EGR dilution and intake temperature.

In order to select operating points for the groups A and B knock analysis was also done. For the group A average maximum amplitude of pressure oscillation (MAPO) index for 300 cycles was taken into consideration (Figure 7) while for the group B, because of very low average MAPO number and the inability to show the difference by using this value, the number of cycles with MAPO>0.5 bar were counted in order to show the knock tendency (Figure 8).

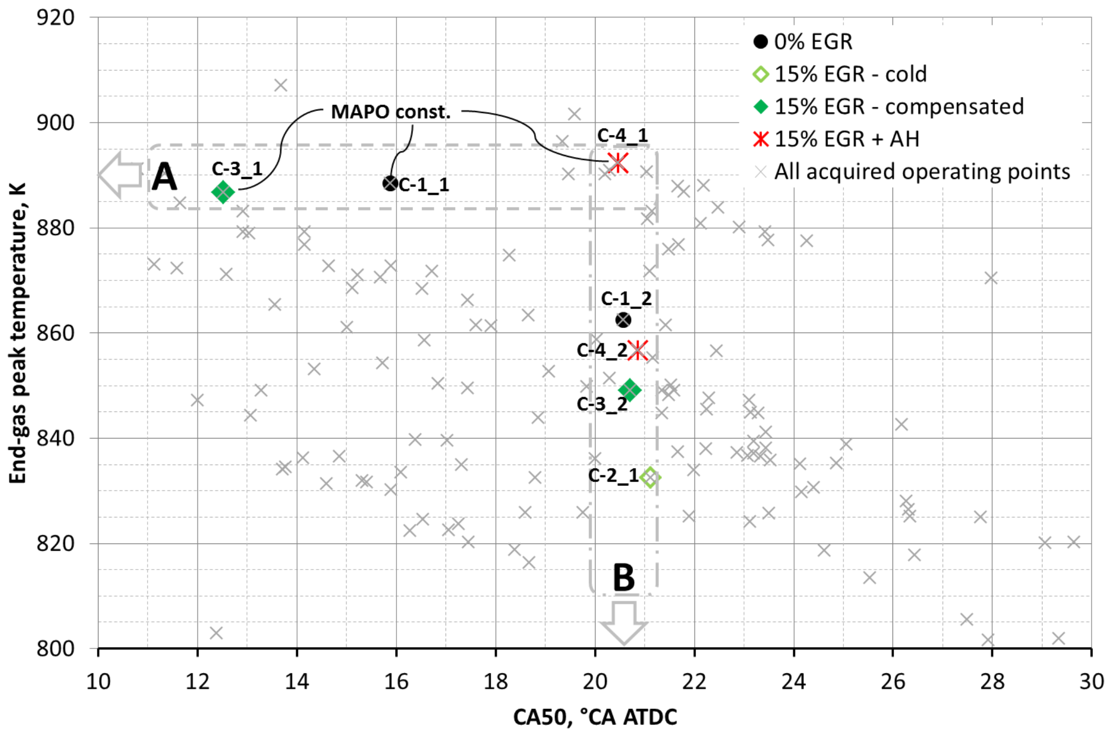

Figure 6. End-gas peak temperature versus combustion phasing
for four selected cases and two groups of data

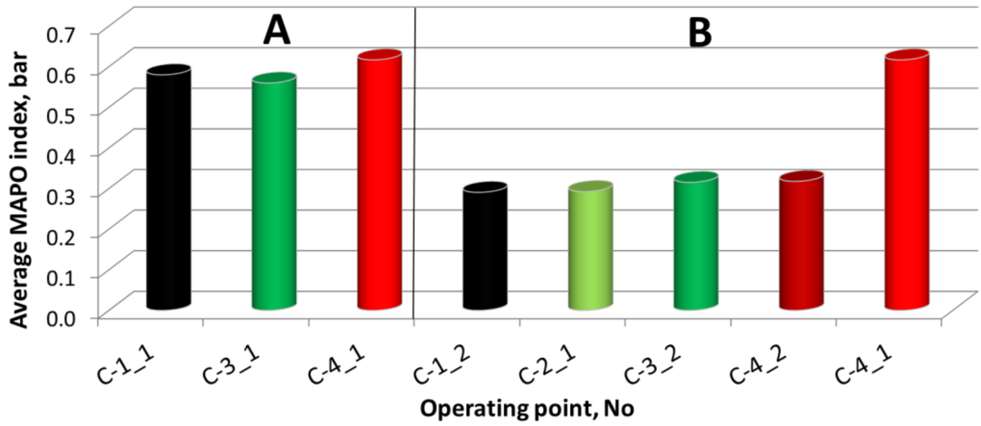

Figure 7. Average MAPO index from 300 cycles for the selected operating points

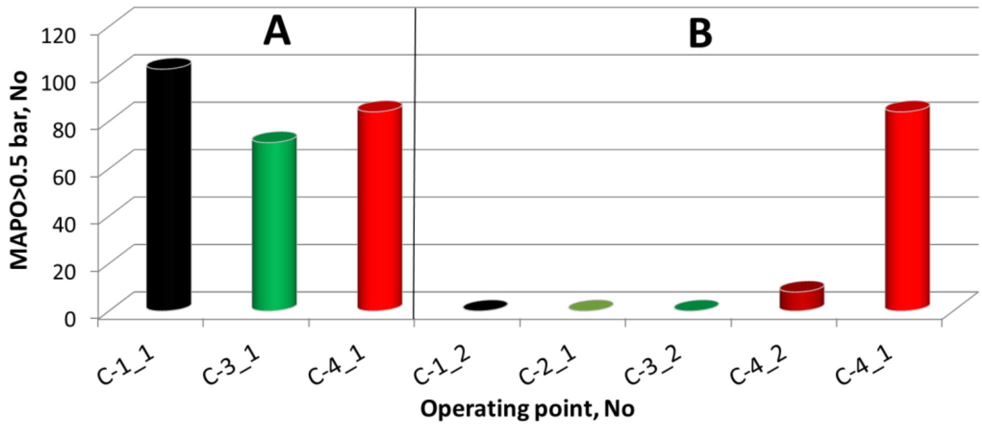

Figure 8. Number of cycles with MAPO>0.5 bar for the selected operating points

The operating points in group A (points with same end-gas temperature) all had almost the same average MAPO index, as shown of Figure 7. This shows that operating points with same end-gas temperature show the same tendency to knock regardless of EGR dilution implemented. Therefore it can be taken to consideration that the main influence on knock suppression by using EGR dilution was the influence on the end-gas temperature. Additionally, it can be seen that cold EGR dilution (C-3_1) enabled more favourable advanced combustion phasing since the EGR reduced the increase of end-gas temperature and therefore enabled the earlier combustion with same increase of end-gas temperature. Heating of the intake (C-4_1) cancelled that effect and even made it less favourable and required retarding of the combustion to the later phasing. The influence of the three different cases inside group A on the end-gas temperature profile and on the rate of heat release can be observed in Figure 9.

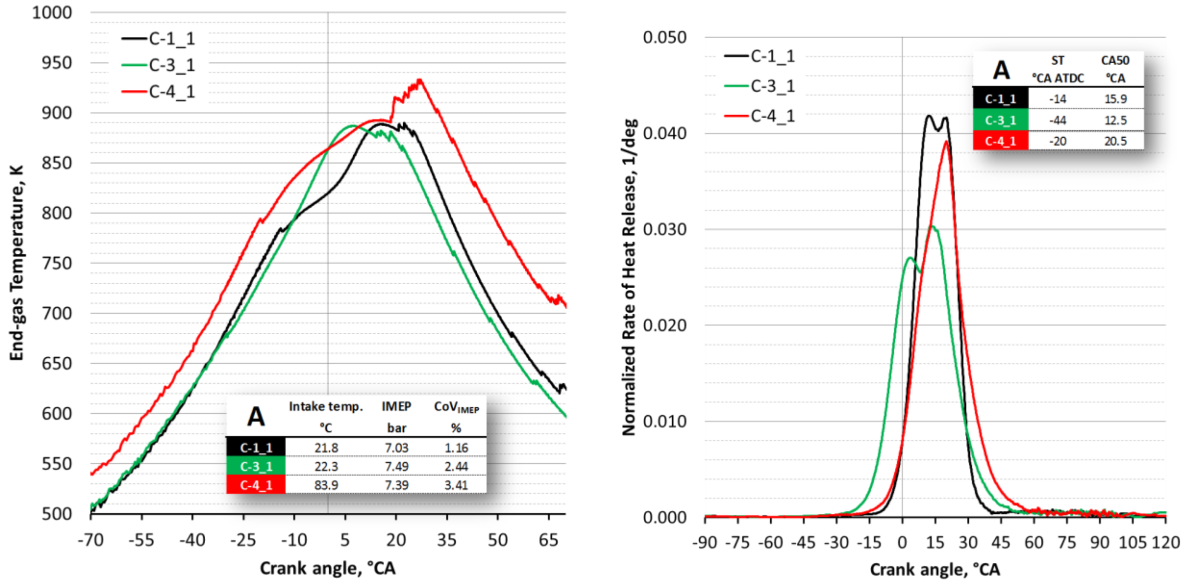

Figure 9. Group of operating points A
Left: End-gas temperature versus °CA, Right: Normalized Rate of Heat Release versus °CA

End-gas temperature shows the anomalies caused by knock combustion cycles. Maximum temperature that was considered in the comparison was the maximum temperature that occurred before the occurrence of knock. On the other hand, on normalized rate of heat release graph it can be observed that case three had earlier combustion phasing (CA50 = 12.5 °CA ATDC) and longer burn duration than case one with 0% EGR (CA50 = 15.9 °CA ATDC). Case three with heated EGR showed retarded combustion phasing (CA50 = 20.5 °CA ATDC) than the case one but higher rate of heat release than case three with compensated EGR. Earlier combustion phasing was achieved up to the limit of knock with spark advance of -44 °CA ATDC for case three compared to the case one with the spark timing of 44 °CA BTDC. With heating implemented in case four spark timing was retarded to -20 °CA ATDC compared to the cases one and three. Since EGR case three with lower temperature (C-3_1) was able to achieve advanced combustion phasing with the same knock intensity, it resulted with higher IMEP than cases one and four (table on Figure 8). As the fuel level is the same in all cases, the effect of EGR to knock suppression enabled the combustion to be more efficient.
The results of the Group B, where the points with same combustion phasing were compared (average for group operating points CA50 = 20.7 °CA ATDC) did not show significant difference in results of average MAPO index and the average MAPO index was very low. But, as each point had 300 consecutive cycles, the MAPO index analysis was made for individual cycles. Cycles with MAPO index higher than 0.5 bar were counted and shown on Figure 8.
Here it can be observed that heated case four (operating point C-4_2) was also at the limit of knock combustion. This indicates that higher intake temperature (Figure 5) influences the end-gas temperature and lowers the knock resistance in comparison to operating points with lower intake temperature (C-1_2, C-2_1 and C-3_2) where knock is supressed.
Influence on the end-gas temperature and heat release of the three different cases inside group B can be observed on the Figure 10.

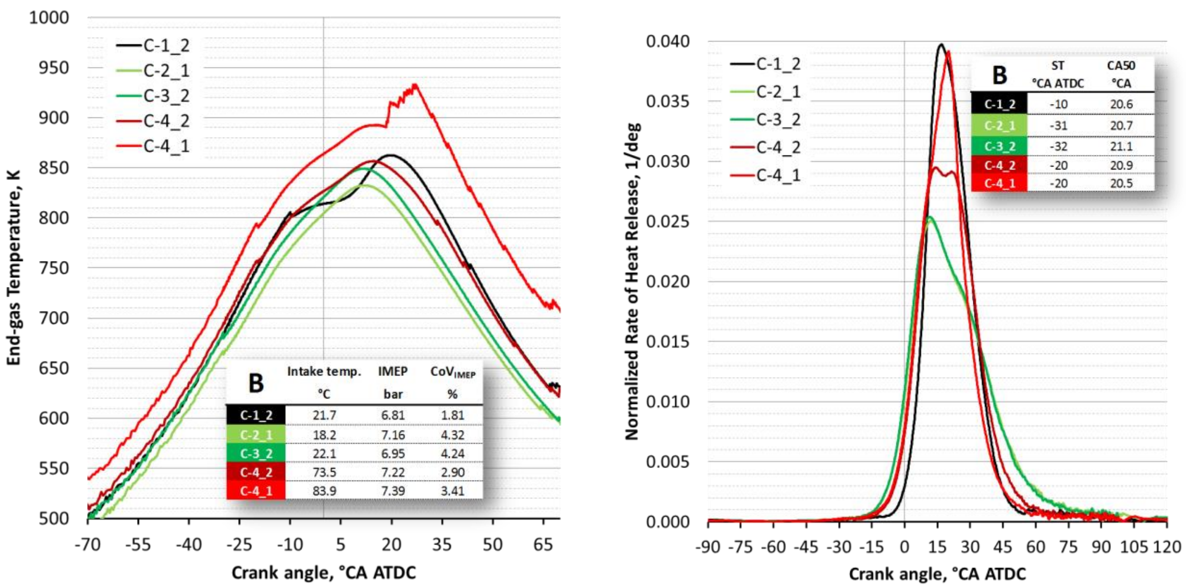

Figure 10. Group of operating points B
Left: End-gas temperature versus °CA, Right: Normalized Rate of Heat Release versus °CA

It shows that operating points with higher knock occurrence tendency (C-4_2 and C-4_1) were longer at elevated temperature of the end-gas. This higher end-gas temperature than resulted in knock occurrence because end-gas needs to have certain amount of time at high temperature so chemical reactions can occur. It can be observed that operating point C-4_2 has that longer period of high temperature than operating point C-1_2 with no EGR resulting in more knock cycles as shown by the MAPO count analysis in Figure 8.

For the emission of NO_X it can be seen that application of EGR dilution significantly reduces NO_X emission as shown on Figure 11. Even though for case three intake temperature was constant it can be seen that operating point C-3_1 have higher amount of NO_X than operating points C-2_1 and C-3_2 because of earlier combustion phasing and spark advance of the operating point C-3_1. Additional heating of the intake mixture increases amount of NO_X but it still stays significantly lower than in case one with no EGR dilution. That implies that amount of NO_X is not only influenced by the temperature of the mixture but also by the availability of the oxygen which is in cases with EGR much lower. Therefore all EGR cases have significantly lower NO_X emission.

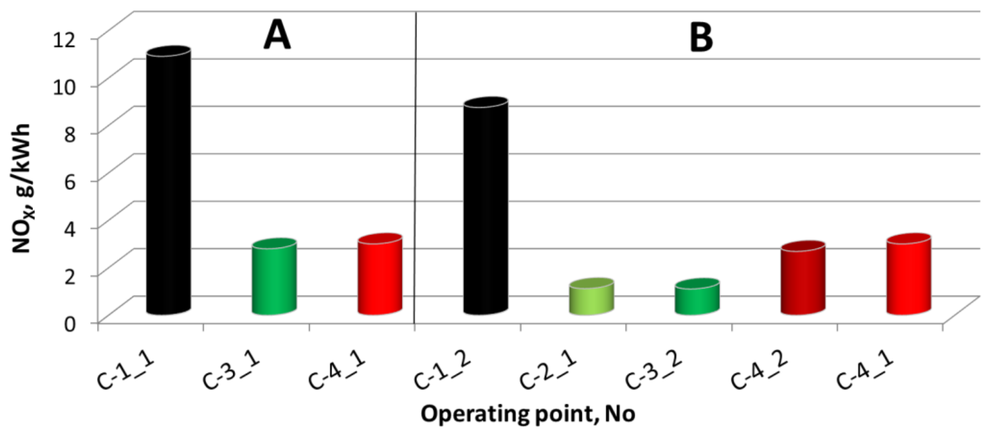

Figure 11. Emission of Nitrogen Oxides versus CA50
for the selected operating points

Research in literature [8,12] describe that certain amount of NO_X in the intake can increase engine knock tendency. The amount of recirculated NO_X in the experimental results is shown on the Figure 12. Although in the experiments there was some NO_X on the intake the influence of knock suppression obtained by influence of EGR on end-gas temperature was in these experiments greater than the influence of NO_X on the increase of knock tendency and therefore the overall result was that the cases with cold EGR had lower knock intensity.

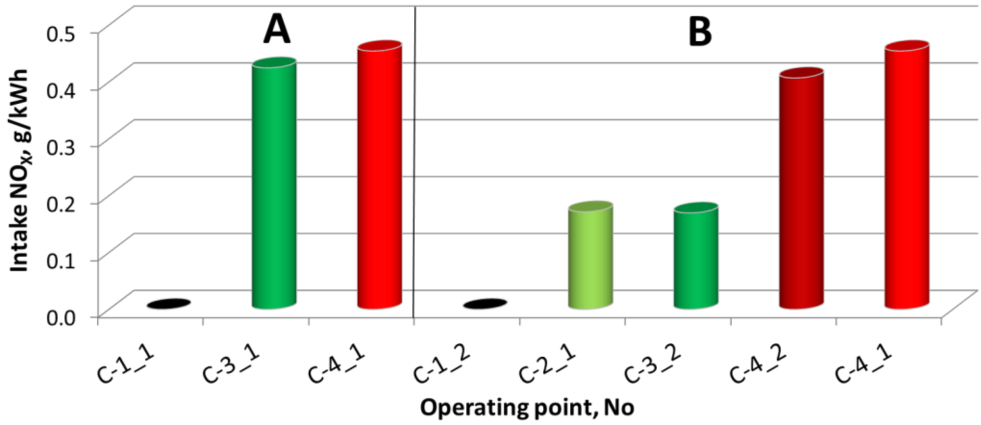

Figure 12. Emission of Nitrogen Oxides recirculated with EGR versus CA50 for the selected operating points

CONCLUSION

In this research, the use of EGR strategy was experimentally investigated in spark-ignition engine. Engine performance and emissions were studied and following conclusions have been made:

- With the use of EGR abnormal combustion was effectively suppressed.
- The operating points with EGR and with lower intake temperature suppress knock by lowering the end-gas temperature.
- If the temperature of the end-gas is constant the tendency to knock is at the similar level regardless of other operating conditions for the same level of EGR.
- The temperature effect of EGR dilution has significant influence on combustion and knock tendency in comparison to other influences of EGR, e.g. flame speed and chemical effects.
- Application of EGR dilution reduces NO_X emission significantly.

According to the results of this study, the impact of the exhaust gas recirculation on charge temperature and consequently on tendency of engine to knock and has high influence on lowering of fuel consumption. Also the charge temperature effect of EGR has significant impact on NO_X formation. Therefore the use of EGR has significant impact on the sustainability of the SI engines.

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