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Application of an EGR system in a direct injection diesel engine to reduce NO_X emissions

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Abstract. This work presents the application of an exhaust gas recirculation (EGR) system in a direct injection diesel engine operating with diesel oil containing 7% biodiesel (B7). EGR rates of up to 10% were applied with the primary aim to reduce oxides of nitrogen (NO_X) emissions. The experiments were conducted in a 44 kW diesel power generator to evaluate engine performance and emissions for different load settings. The use of EGR caused a peak pressure reduction during the combustion process and a decrease in thermal efficiency, mainly at high engine loads. A reduction of NO_X emissions of up to 26% was achieved, though penalizing carbon monoxide (CO) and total hydrocarbons (THC) emissions.

1. Introduction

In recent years, the environmental sustainability issue has gained higher relevance due to phenomena such as climate changes or rate of air pollution. At present, the intensive use of petroleum-derived fuels is the major responsible for the environmental threat in terms of climatic alterations and degradation of air quality. The control of emissions has become the driving force in the development of internal combustion engines, through the use of modern mechanical components, alternative fuels and electronic systems [1].

Differently to spark ignition engines, diesel emissions include visible smoke and particulates, which has made it an easy target of political campaigns [2]. For diesel engine exhaust, the emission of oxides of nitrogen (NO_X) and particulate matter (PM) is a major pollution problem. Two examples of technology, aimed to reduce the emissions levels, are the EGR (Exhaust Gas Recirculation) and the SCR (Selective Catalytic Reduction). The EGR is a method by which a portion of the engine exhaust is returned to the combustion chamber via the intake system [2,3]. This technique allows to extract heat from the combustion process, thus lowering its temperature and reducing NO_X emissions [4].

Zheng et al. [5] explain that diesel exhaust contains carbon dioxide (CO_2), water vapor (H_2O), nitrogen (N_2) and oxygen (O_2) in thermodynamically significant quantities, and carbon monoxide (CO), hydrocarbons (HC), NO_X and soot in thermodynamically insignificant but environmentally harmful quantities. In modern diesel engines, the combination of CO_2, H_2, N_2 and O_2 normally comprises more than 99% of the exhaust, while the combination of CO, HC, NO_X and soot accounts for less than 1% in quantity. Thus, the challenge is to minimize the pollutants by manipulating the thermodynamic properties and the oxygen concentration of the cylinder charge whilst keeping minimum degradations in power and efficiency, which is the principal reason to apply diesel EGR.

Hountalas et al. [6] used a 3D multi-zone model in order to examine the effect of EGR temperature on performance and emissions of a turbocharged direct-injection (DI) diesel engine. EGR cooling affects brake engine efficiency, peak combustion pressure and soot emissions negatively. In particular, EGR cooling is necessary to prevent soot emissions from rising to unacceptable levels. The need for EGR cooling is more evident at high EGR rates and low engine speeds.

Maiboom et al. [7] and Millo et al. [8] explain that the possibility to reduce NO_X emissions, with the increase of EGR rate, is the result of various thermal and chemical effects. Hussain et al. [10] studied the effect of EGR on performance and emissions of a diesel engine. The reduced oxygen and lower flame temperatures affect performance and emissions of diesel engine in different ways. Thermal efficiency is slightly increased and specific consumption is decreased at lower loads with the use of EGR. Moreover, when EGR is applied, NO_X emission decreases significantly. It can be observed that 15% EGR rate is found to be effective to reduce NO_X emission substantially without deteriorating engine performance in terms of thermal efficiency, brake specific fuel consumption (BSFC), and emissions. The increase of CO, HC, and PM emissions can be reduced by using exhaust after-treatment techniques, such as diesel oxidation catalysts and soot traps.

This work analyzes the application of an EGR system to a 49 kW diesel power generator operating with diesel oil containing 7% biodiesel (B7). The EGR effect in performance, emissions and combustion characteristics are presented next.

2. Methodology

Experiments were carried out in a diesel power generator equipped with a naturally aspirated, four-stroke and four-cylinder diesel engine. Table 1 shows the main engine characteristics.

Table 1. MWM diesel engine specifications.

Manufacturer	MWM
Model	D229-4
N° of cylinders	4
N° of strokes	4
Type of injection	Direct
Bore × stroke	102 mm × 120 mm
Total displacement	3.922 liters
Firing order	1-3-4-2
Maximum power at 1800 rpm	44 kW
Aspiration	Natural
Compression ratio	17:1
Type of cooling	Water

For recirculation of the exhaust gas, an appropriate plumbing was done. No insulation on the pipe line was provided, therefore allowing the recirculated exhaust gases to partially cool down. The quantity of EGR was regulated by an electric valve, installed in the EGR loop. An electronic system was developed to control the valve position and the EGR rate (%).

The EGR rate is defined as the mass percent of the recirculated exhaust (M_{EGR}) in the total intake mixture (M_{in}). Several authors report in their works [5,7,8,9] that the EGR percentage can be also calculated as the recirculated CO_2 fraction. Indeed, the fresh air intake contains negligible amounts of CO_2, while the recycled portion carries a substantial amount of CO_2 that increases with EGR flow rate and engine load. Notably, CO_2 is merely a combustion product. Thus, it is intuitive and practical to measure EGR ratio by comparing the CO_2 concentrations between the engine exhaust (CO_{2_exh}) and intake (CO_{2_EGR}), as shown by Equation (1). Two taps were built in order to acquire the CO_2 percentages in the intake and in the exhaust manifolds and calculate the EGR ratios.

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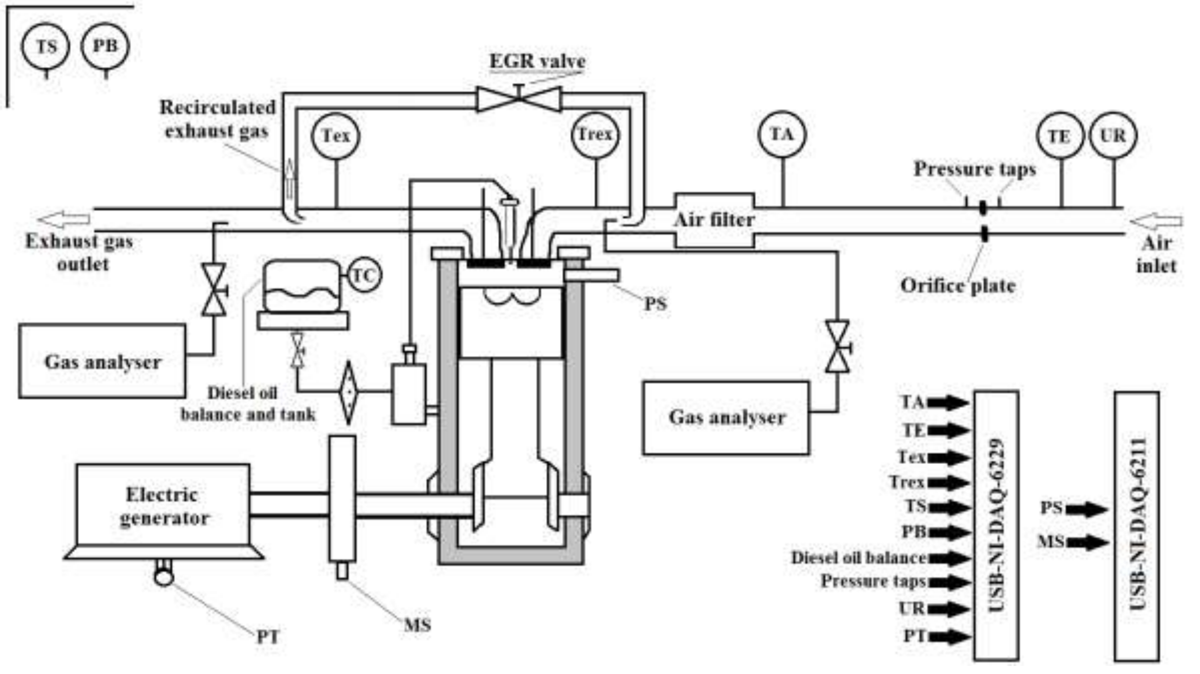
$$EGR\ (\%) = \frac{(CO_{2\_EGR} - CO_{2\_atm})}{(CO_{2\_exh} - CO_{2\_atm})} \tag{1}$$

A data acquisition system was used to assess engine performance. The system consists of sensors, transducers, signal conditioning circuits, two data acquisition boards and a program developed in LabVIEW 7.1 platform. A schematics of the experimental apparatus is shown by Figure 1. Fuel consumption measurement was done using a balance with 5 g of resolution. A piezoelectric pressure transducer with ± 0.5% of resolution was used to measure the pressure inside the first cylinder of the engine. Temperatures at several locations were measured, such as fuel tank, ambient air, exhaust gas, cooling water, and EGR system. The intake air mass flow rate was measured through an orifice plate, with uncertainty of ±2.3%. Non-dispersive infrared analyzers were used to measure CO (resolution of ± 1 ppm) and CO_2 (resolution of ± 0.01%) emissions. Total HC emissions were measured using a heated flame ionization detector (resolution of ± 1 ppm) and NOx emissions were measured by a heated chemiluminescent analyser (resolution of ± 1 ppm).



TA - Intake air temperature **TC** - Diesel oil temperature **TE** - Inlet air temperature **TS** - Test room temperature	**Tex** - Exhaust gas temperature **Trex** - Recirculated exhaust gas/air mixture temperature **PB** - Test room barometric pressure	**PS** - Cylinder pressure sensor **MS** - Magnetic sensor **PT** - Power trasducer **UR** - Relative humidity

Figure 1. Schematics of the experimental apparatus.

The fuel used was N. 2 diesel oil containing 7% of biodiesel. The engine load was varied from 0 kW to 30 kW in intervals of 5 kW, and three tests for each EGR ratio and load were carried out. The ISO 3046-1:2002 standard was used to correct the load power and fuel consumption to standard conditions. The engine was operated with and without EGR, and the results compared. Table 2 reports the EGR ratios and nomenclature used to present the results.

Table 2. EGR ratios and relative nomenclature.

EGR ratio	Nomenclature
0.0% EGR (No EGR)	EGR0
2.5% EGR	EGR2.5
5.0% EGR	EGR5
7.5% EGR	EGR7.5
10.0% EGR	EGR10

3. Results and discussions

The engine was operated at 1800 rpm with different loads and EGR rates (from 0% to 10%) to investigate the effect of EGR on engine performance and emissions. The performance and emission data was analyzed and presented graphically for gas pressure inside the cylinder, heat release rate, air mass flow rate, thermal efficiency, CO_2, CO, THC and NO_X specific emissions.

Engine performance analysis

The in-cylinder pressure and the corresponding heat release rate at the load 30 of kW are shown in Figures 2 and 3, respectively. The peak pressures are found to have slightly decreased for all the EGR rates in comparison with operation with no EGR (Figure 2). The reductions found were of 0.5% for EGR2.5, 0.6% for EGR5, 2.5% for EGR7.5 and 1.1% for EGR10. This pressure decrease with the use of EGR causes decreased combustion temperature, if ideal gas behavior is considered, leading to a reduction in NO_X formation. As a consequence of decreased combustion temperature, the peak heat release rate is also reduced with the use of EGR (Figure 3). The higher reductions of the peak heat release rate were observed for operation with 7.5% and 10% EGR.

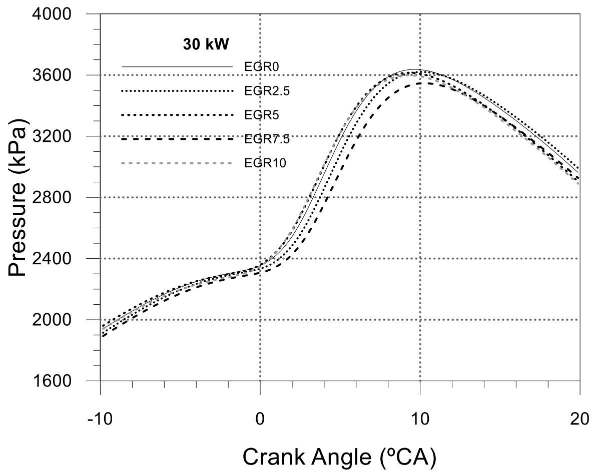


Figure 2. In-cylinder pressure at 30 kW for different EGR rates.

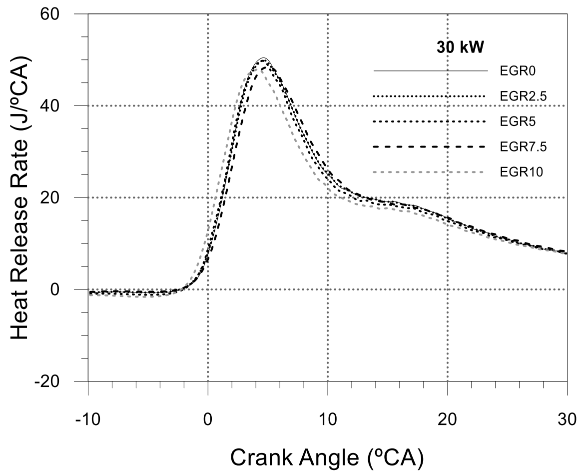


Figure 3. Heat release rate curve at 30 kW for different EGR rates.

Figure 4 shows the intake air mass flow rate for different EGR operations. Since part of the intake air is replaced by the exhaust gas, increasing EGR rate reduces the intake air mass flow rate. Compared with the EGR0 case, the average reductions found were of 3% for EGR2.5, 4% for EGR5, 7% for EGR7.5, and 9% for EGR10. The engine volumetric efficiency is also affected at the same proportions, as less air amount is admitted into the engine cylinders with increasing EGR rates.

Figure 5 presents the variation of engine thermal efficiency, normalized to the standard atmospheric conditions, for different loads and EGR rates. The use of EGR causes a negative effect on engine thermal efficiency, being in agreement with the behaviour reported by [9]. The decrease of the brake thermal efficiency is mainly due to the reduction of air/fuel ratio (Figure 6). Here, the brake thermal efficiency was not affected by a supposed increase of in-cylinder mean gas temperature and heat losses [6], as no specific trend was observed for the latter (Table 3). In the load range investigated, the brake thermal efficiency is not significantly affected by EGR only in the extremes, at 5 kW and 30 kW, where the values are close to the case of 0% of EGR, but they remained within the uncertainty ranges. For all intermediate loads the brake thermal efficiency decreases with the use of EGR. The reductions of brake thermal efficiency found for operation with EGR2.5 were of 2.2%, at 10 kW, 3.0%, at 20 kW, and 1.0%, at 30 kW. With EGR5, the brake thermal efficiency dropped by 1.0%, 3.4% and 2.2% for the same loads, respectively, and, with EGR7.5, the reductions were of 1.3%, 2.7% and 1.1%. With EGR10 the brake thermal efficiency were reduced by 3.3%, 4.3% and 5.7% for 10 kW, 20 kW and 25 kW, respectively. So, in general, it is not attractive to use high EGR rates at high loads gas because the combustion tends to deteriorate, leading to reduced engine thermal efficiency.

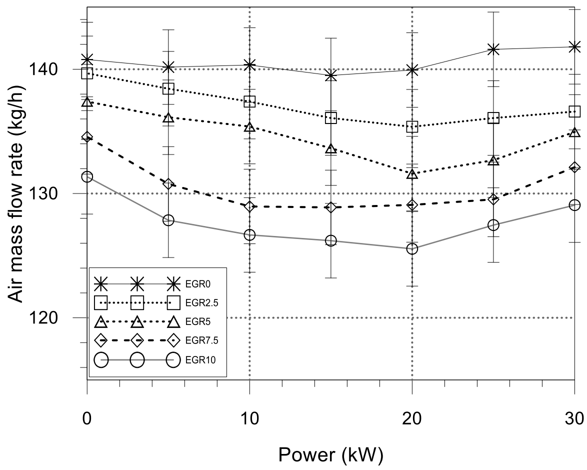


Figure 4. Air mass flow rate for different EGR rates.

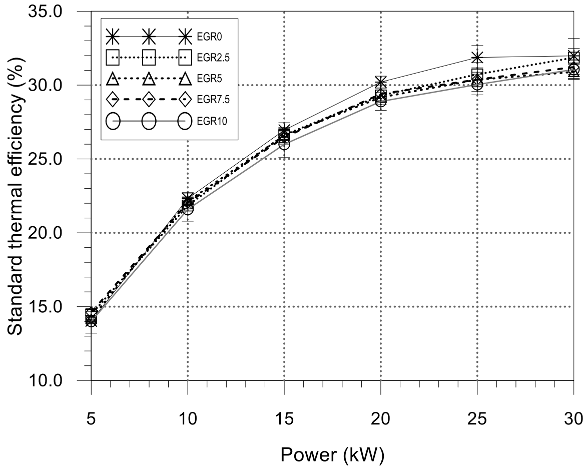


Figure 5. Standard thermal efficiency for different EGR rates.

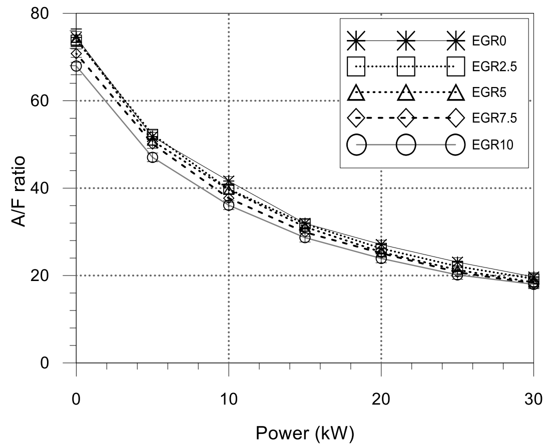


Figure 6. Air/Fuel ratio for different EGR rates.

Table 3. Heat release mean values at 30 kW load.

EGR rate (%)	Heat Release (kJ)	Variation (%)
0.0	- 401.80	0.0
2.5	- 409.59	+ 1.9
5.0	- 389.99	- 2.9
7.5	- 411.92	+ 2.5
10.0	- 392.40	- 2.3

Engine specific emission analysis

The effects of EGR on CO_2, CO and THC emissions are shown in Figures 7 to 9. It is noticed that CO_2, CO and THC emissions increase with the increase of EGR rate. For CO_2 (Figure 7), this trend is explained by the fact that the fresh intake air contains negligible amounts of CO_2, while the EGR fraction carries a substantial amount of CO_2, which is increased with EGR flow rate and engine load [5,10]. This increment becomes more relevant for higher EGR rates and engine loads. For operation with EGR10, the increase in CO_2 observed were of 6.1%, at 15 kW, 8.5%, at 25 kW, and 11.2%, at 30 kW. The highest CO_2 specific emission values are found at lower engine loads; then these emissions reduce until 20 to 25 kW, and finally increased again for higher loads. At low engine loads of 10 kW and lower, the EGR rate did not significantly affect CO_2 specific emissions, showing variations lower than 1.2%.

The increase of EGR rate reduces the oxygen availability in the combustion chamber and decelerates the reaction rates of the air-fuel mixture, thus producing lower temperatures. In such case, the flame propagation may not be sustained with relatively leaner mixtures and this heterogeneous mixture does not burn completely, resulting in higher CO (Figure 8) and THC (Figure 9) specific emissions. The largest CO increments were obtained for engine loads below 15 kW and above 25 kW. In particular, at 5 kW and 10 kW, the average increases were of 3.9% and 6.1%, respectively. At part loads, from 15 kW to 25kW, the CO specific emissions did not change significantly. With regard to THC specific emissions, the largest variations in comparison with EGR0 were found for EGR2.5 operation, showing increments from 7.1%, at 5 kW, to 27.6%, at 30 kW. The average variations of THC emissions for the other EGR rates in comparison with EGR0, for all engine loads, were decrease of 2.2% for EGR5, decrease of 1.4% for EGR7.5, and increase of 4.0% for EGR10.

Figure 10 shows that NO_X specific emissions is reduced with increasing EGR rate for all engine load range investigated. This behaviour is justified by the reduction of oxygen mass fraction due to the displacement of some of the oxygen in the fresh intake air charge by inert gases. This causes a reduction in the local flame temperature because of the spatial broadening of the flame due to the reduction in the oxygen molar fraction. Also, there is the thermal effect due to the increase in the average specific heat capacity of the gases in the combustion zone, since the recirculated exhaust gas contains CO_2 and H_2O with higher specific heat than that of air. Finally, there is a reduction in the combustion temperature due to endothermic chemical reactions, such as the CO_2 and H_2O dissociation [7,8]. In comparison with EGR0 operation, the average reductions for all engine loads were of -6.2% for EGR2.5, -13.1% for EGR5, -22.3% for EGR7.5, and -26.3% for EGR10.

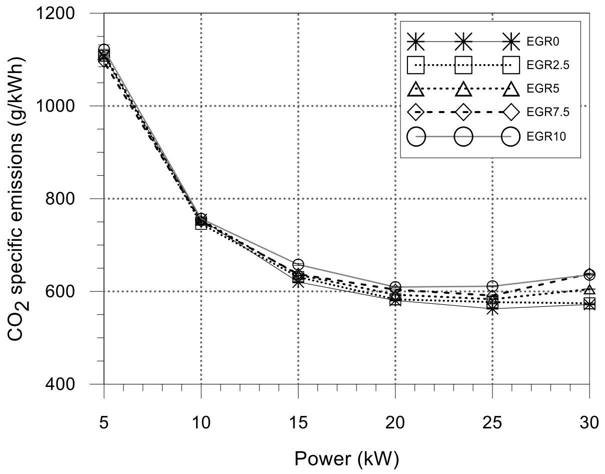


Figure 7. CO_2 specific emissions for different EGR rates.

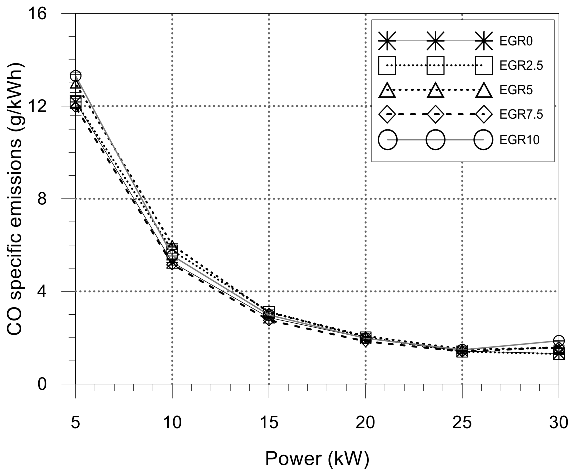


Figure 8. CO specific emissions for different EGR rates.

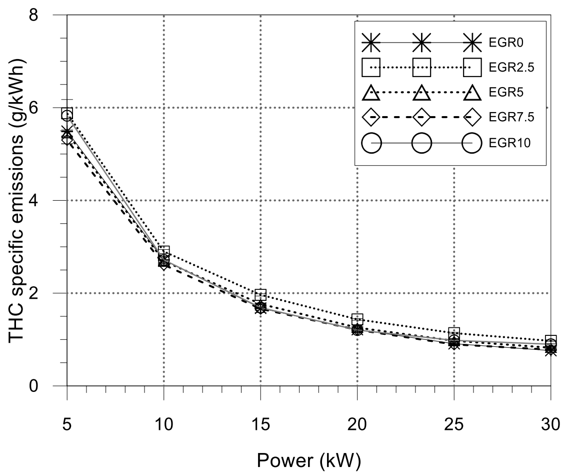


Figure 9. THC specific emissions for different EGR rates.

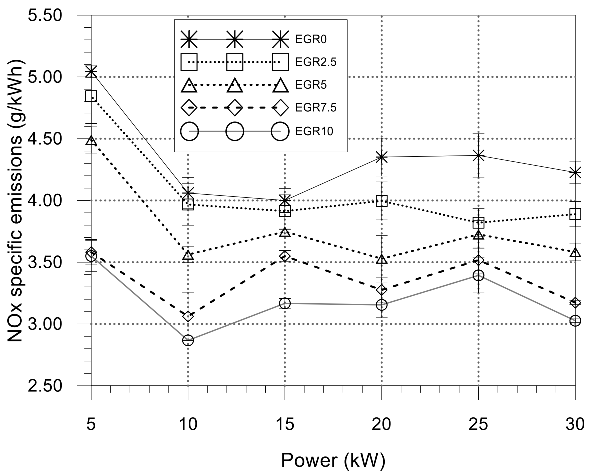


Figure 10. NO_X specific emissions for different EGR rates.

4. Conclusions

An experimental investigation was conducted to study the effect of EGR on the performance and emissions of a stationary diesel engine, showing the feasibility of using an EGR system for the abatement of NO_X emissions. Increasing EGR rates caused a decrease in the in-cylinder peak pressure, in comparison with operation without the EGR system. The largest peak pressure reduction was found for the highest engine load and highest EGR rate tested. With increasing EGR rate the intake air mass flow rate was reduced, leading to a reduction on oxygen availability inside the cylinder and, consequently, decreased air/fuel ratio. As a result, THC and CO emissions were increased with increasing EGR rate. The brake thermal efficiency was reduced with the increase of EGR rate, mainly at high engine loads. Increasing EGR rate and engine load caused increased CO_2 specific emissions. With the increase of EGR rate and engine load, were reduced by up to 26%. It can be concluded that EGR can be applied to reduce NO_X emissions without major penalties on engine efficiency, while the increase on CO and THC can be adequately controlled using exhaust after-treatment techniques, such as diesel oxidation catalysts and soot traps.

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