Effects of EGR and its temperature on spark-ignition engine using ethanol

Luana Cristina Xavier Camargos
Antonio Luiz Pacifico
Polytechnic School of the University of São Paulo

Clayton Barcelos Zabeu
Institute Maua of Technology

ABSTRACT

The need of reducing greenhouse gas emissions through the automotive industry has demanding the increase in efficiency of combustion engines. The utilization of renewable fuels - as the use of biofuels – has also an impact on the reduction of net CO_2 emitted to the atmosphere. There are several possibilities to achieve this increase in efficiency in internal combustion engines – one of them being the exhaust gas recirculation (EGR). In order to better understanding of the impacts caused by the insertion of an EGR system and recirculated gas temperature variation on the combustion, efficiency, and emissions of a PFI engine, tests were conducted in a 3-cylinder, flex fuel, 1,2L displaced volume engine, and equipped with an open ECU. Through the analysis of results of fuel consumption, emissions and indicated quantities on the three cylinders it was possible to determine the impact on the specific fuel consumption and raw emissions. Such tests were conducted on two operational points of the engine. The hydrated ethanol (E95h) was used as fuel. The results indicated a reduction until of 7% in specific fuel consumption in specific EGR rates. There was also significant reduction in NOx, of up to 90%. However, depending on the rate, some instability in combustion and an increase in unburned hydrocarbons (UHC) can be observed.

Keywords: Exhaust gas recirculation. Pollutant emissions. Spark ignition engines. Efficiency increase. Ethanol. EGR temperature.

INTRODUCTION

The exhaust gas recirculation (EGR) technology consists of extracting part of the burnt gases in the exhaust system and inserting them back into the engine intake system. This technique was initially used in diesel engines with the objective of reducing nitrogen oxide (NOx) emissions.

NOx formation is causally related to the maximum temperature inside the combustion chamber due to the Zeldovich formation mechanism, and above 1800 K the formation of these oxides increases considerably (TURNS, 2012) [1]. The insertion of an inert gas in the combustion chamber decreases the peak combustion temperature, as well as reducing the partial pressure of oxygen, causing reduction in the formation of nitrogen oxides.

In addition to the benefit of reducing NOx emissions, in spark ignition (SI) engines, the EGR makes possible to increase the efficiency in partial loads. Figure 1 illustrates the mechanism behind this efficiency gain: at part load, the throttle restricts the air flow into the cylinders and regulates the final amount of air (and fuel) trapped in them needed to achieve the desired engine power. By adding the entrance of a diluent – EGR – the throttle must be set in a less restrictive condition, imposing a higher-pressure level in the intake manifold to provide that approximately the same mass of fresh mixture. This less restrictive condition reflects in lower pumping work.

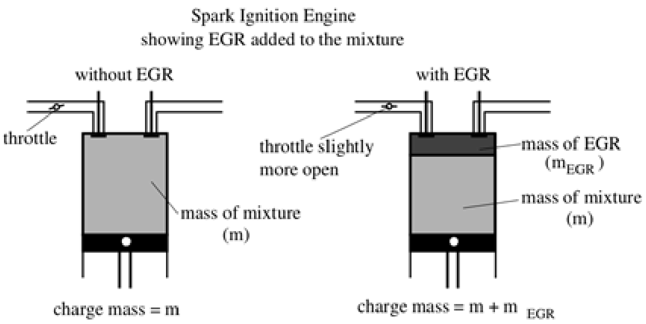

Figure 1 - Comparison between SI without and with EGR. Adapted from (ABD-ALLA, 2001) [2]

As the EGR rate increases, all stages of the combustion process lengthen, reaching the point where there is not enough time for combustion to complete inside the cylinder, or there may be cycles where there are misfires and the mixture does not ignite. There is a trade-off between the maximum EGR rate to be used and the benefits that will be achieved.

A measure widely used to limit the stable operation of the engine is the coefficient of variation of the indicated mean effective pressure - COV_{IMEP} - where the standard deviation of the indicated mean effective pressure (IMEP) is divided by the mean of the IMEP, usually expressed as a percentage, as shown in the equation (1)

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP} \times 100 \qquad (1)$$

The COV coefficient defines the cyclic variability in the indicated work, demonstrating the instability and repeatability of the cycle against the average of engine operation, and is a very important parameter in the study of engines. According to (HEYWOOD, 2018) [3] the COV_{IMEP} value between 2 to 5% guarantees stability in the vehicle's drivability, depending on the engine's operating conditions.

As shown in Figure 2 above 10% of EGR there is significantly increase of COV. The maximum rate of EGR for SI engines using gasoline is known to be between 10-15%, but with the use of ethanol it is not possible to find much data in literature. However, (Splitter, J. P, 2016) [4] did a study and proved that due to the higher flame speed of ethanol, it has a greater tolerance for receiving a greater amount of diluent.

In addition, EGR temperature has significant effects on emissions and combustion, which can be studied at partial loads without causing problems such as knock.

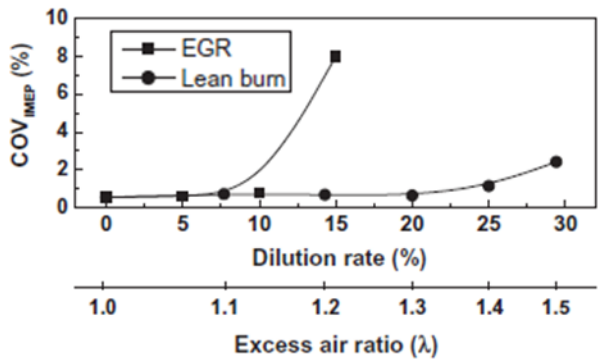

Figure 2 - Comparation COV_{IMEP} EGR and lean burn operation with low calorific gas 1800rpm@60kW (LEE, PARK, et al., 2014) [5]

According to (WEI, Haiqiao et al, 2012) [6] a higher EGR temperature should result in a shorter combustion duration, greater thermal efficiency, and lower pumping work due to the decrease in specific mass, when compared to cold EGR.

Figure 3 shows there is a significant difference in the specific fuel consumption due to temperature variation in a SI gasoline engine. On the left, at the point with the highest EGR rate, 19% lower brake specific fuel consumption (BSFC) is observed for the hot EGR compared to the cold one. On the other hand, the figure on the right shows that cold EGR promotes a slightly lower emission of NOx, which is not so relevant compared to the other gains described with the hot EGR.

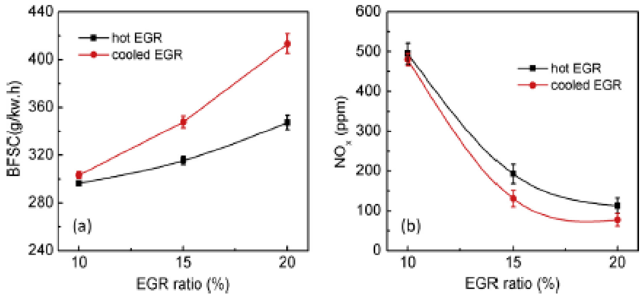

Figure 3 - Comparation BSFC and NOx with hot and cooled EGR with gasoline 2000@581kPa BMEP (XIE, Fangxi *et al*, 2017) [7]

In this context, the objective of this work is to present and discuss these benefits, based in experimental data using ethanol as engine fuel and to verify the effect of EGR temperature on efficiency and emissions. In addition, the influence of EGR temperature on combustion characteristics was also explored.

MATERIAL AND METHODS

The work was carried out using a commercially available flex-fuel, 1.2L, 3-cylinder naturally-aspirated port fuel injection (PFI) engine (see Table 1), at the Division of Engines and Vehicles of Institute Maua of Technology, which has been modified and received an external EGR circuit. This engine counts on dual variable valve timing (VVT) system, allowing it to operate in Miller condition (when both intake and exhaust valves open and close lately) or in Otto condition (where intake valves open and exhaust valve close around top dead center).

Table 1 - Engine specs

Displaced volume [cm³]	1199
Cylinder [mm]	75
Stroke [mm]	90.5
Compression ratio	12.5:1
Number of valves per cylinder	4
Power (E95h[1]) [kW]	66.2 @5750 rpm
Torque (E95h) [N.m]	127.5 @2750 rpm

To obtain a homogeneous mixture of air and EGR in the three cylinders, a mixer already used in previous works by (CAMARGOS, Luana C. X.; AVEDISSIAN, Nicholas Q.; DOS SANTOS, Arthur H, 2017) [8] and an additional swirler was installed after the throttle. Figure 4 shows the EGR mixer assembly that was used and Figure 5 shows the assembled system and the exhaust gas extraction point (before the catalyst) for insertion into the intake after the throttle valve.

[1] E95h – Hydrous ethanol – 95% ethanol

Figure 4 - EGR mixer

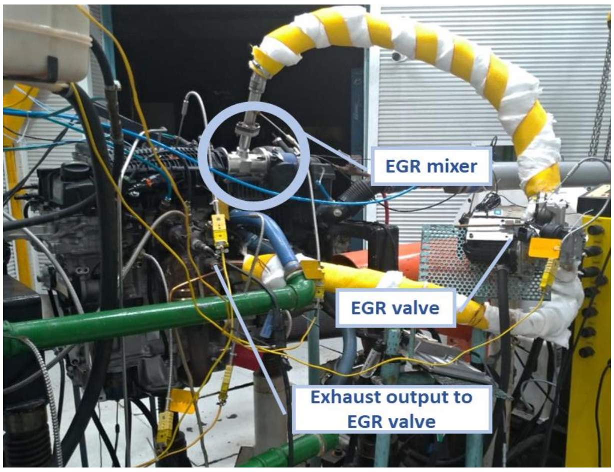

Figure 5 - EGR system

In order to carry out the tests and control the engine on the test bench, the automation system of A&D Technology was used in conjunction with ETAS INCA® to have access to a development ECU (Engine Control Unit) that is completely open for engine calibration. In addition, to record the in-cylinder pressures, the AVL Indimodul data acquisition system and AVL IndiCom® software were used in conjunction with instrumented spark plugs (model AVL-Z131_Y5S) with piezoelectric pressure transducers and an AVL-365 encoder to assess the crankshaft's angular position, which allowed the calculation of indicated mean effective pressures and of the heat release rate, among other indicated results.

The exhaust emission measurements and CO2 concentration at the intake plenum were performed using a Horiba Mexa 200 bench, making it possible to measure the following gases at the exhaust outlet: NOx, THC, CO, O_2 and CO_2.

The engine load/speed points were 2750/25 Nm (2.6 BMEP) and 4000/40 Nm (4.2 BMEP). At the first operational point, the engine calibration originally forces Miller cycle mode, with late opening and closing of intake and exhaust valves. So, to evaluate the use of EGR in an Otto-like, the position of the VVT was modified for the engine to operate in the Otto cycle. At the second operational point, engine operates originally in the Otto cycle.

In all tests the engine was warmed up until the coolant temperature reached 90 °C, the ignition advance was always adjusted to reach a value of CA50 (crank angle corresponding to burnt mass fraction of 50%) in the better BSFC, and when necessary, the fuel injection was individually adjusted to obtain this value in the three cylinders.

The EGR rate was calculated according to the following equation (2) by (LIU, ZHANG, *et al.*, 2017) [9]:

$$\mathrm{EGR} = \frac{[CO_2]_{admissão} - [CO_2]_{ambiente}}{[CO_2]_{escape} - [CO_2]_{ambiente}} \tag{2}$$

The tests carried out in this work were executed in the following steps: in a fixed EGR rate, CA50 was varied looking for the highest torque. Subsequently sweeps of the EGR rate and the of the EGR temperature were performed.

EGR temperature at mixer inlet has been controlled by adjustments on the cooling water flow supplied to the heat exchanger integrated into de EGR valve.

The temperature, pressure, speed, and fuel consumption data were recorded at a rate of 5Hz for 5 minutes. These frequency and time recording conditions were statistically evaluated and found to be sufficient to evaluate the engine behavior. The indicated data were recorded during 300 consecutive cycles at each engine operating condition.

RESULTS AND DISCUSSION

According to (HEYWOOD, 2018) [3] the angle value at which the MBT is obtained, or where the optimal angle occurs at which 50% of the energy is released (CA50) is around 8°, for operating conditions without diluent. However, when EGR is added to the pressure curve, the energy release rate, flame laminar velocity, flame front propagation and combustion temperature change.

The tests conducted in this work showed a different CA50 when the EGR is added, to 2750/25 Nm the MBT was in CA50 equal 6°CA with 16% EGR and to 4000/40 Nm was 5° CA with 25% EGR.

The dispersion between the initially symmetrical firing profiles (10-90% and IMEP) increases asymmetrically with more dilute conditions. This asymmetry will be substantially increased with longer duration of the initial phase of 0-10%. As a mixture becomes more dilute, there is an increasing spread in the burn duration distribution from 0-10% and an extension of the burn duration from 0-10% (average), which increases combustion asymmetry (AYALA e HEYWOOD, 2007) [10].

Consequently, CA50 does not follow anymore a linear behavior when combustion variability develops, particularly under diluent conditions (AYALA e HEYWOOD, 2007) [10], which can lead to an optimal CA50 with EGR other than 8°.

So, in all tests the CA50 used was the MBT for each operation point, defined in a fixed EGR rate.

EGR VARIATION

Figure 6 shows the relationship between BSFC (normalized to 0% EGR condition) and EGR rate, at 2750@25Nm operational point. One can see there is a decrease of approximately 7% in specific consumption between 18 and 20% of EGR. The points represent the tested and recorded data, the dotted line refers to the trend line to represent the behavior of consumption with increasing EGR rate.

Figure 7 shows the same relationship, but now at 4000@40 Nm operational point. The same trend can be observed with a maximum reduction of almost 4.5% in BSFC.

During these initial tests, the EGR cooler was not active, therefore the air-EGR mixture temperature at intake manifold was just a consequence of EGR rate and temperature.

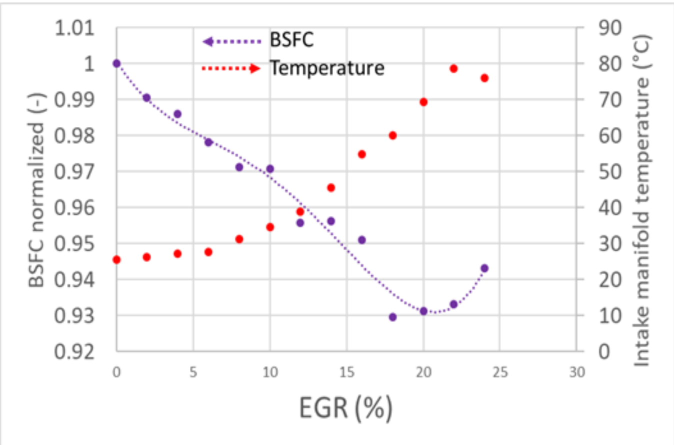

Figure 6 - BSFC and intake manifold temperature 2750@25Nm, all points at fixed spark timing

In both operational points a significant reduction in NOx and an increase in THC emissions were observed. So, results from 2750@25 Nm were presented from now.

In Figure 8 it possible to see the nitrogen oxides decreased up to 97.5% at the maximum EGR point, confirming that exhaust gas recirculation is one of the most efficient ways to reduce NOx emissions as reported in the literature.

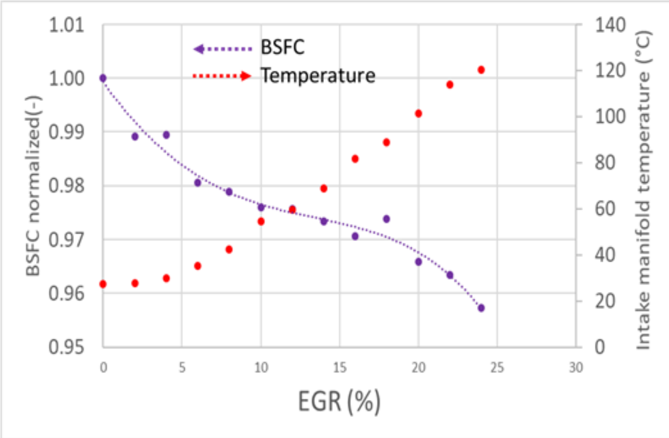

Figure 7 - BSFC and intake manifold temperature 4000@40Nm, all points at fixed spark timing.

Still in the same graph, it can be observed that the THC had an increase of up to 60% when we compare the point of 0% with 24% of EGR, where the engine already presented instability. In the range from the first point with EGR up to a rate of 20%, the THC variation is not significant, however at the point of 22 and 24% EGR there is a significant increase, as when comparing the point without EGR and with a rate of 2%.

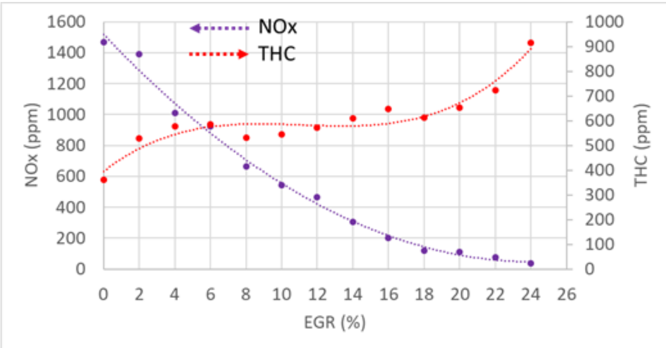

Figure 8 - NOx and THC emissions 2750@25Nm

In the same line, the combustion parameters in both operation points present the same behavior.

Table 2 indicates the values of average COV_{IMEP}, IMEP–low (the indicated mean effective pressure of pumping work) and the duration of the fast part of the combustion (CA10-CA90) to 2750@25Nm. The average COV_{IMEP} increased with the addition of EGR, reaching up to 2.59%. The addition of EGR was only possible up to 24% due to the equalization of the exhaust pressure with the intake, although at this rate it was already observed that the engine operation was already unstable.

The average effective pressure indicated low, which indicates the pumping work performed by the engine, went from -0.639 bar to -0.492 bar, indicating a decrease in the pumping work, justifying the gain in the specific consumption obtained. The points with an EGR rate of 22% and 24%, despite having a lower IMEP low when compared to the point of 20%, there was a worsening in the specific consumption, as this gain was already being negatively compensated by the deterioration in combustion.

The duration of the fast-burning phase increases as the EGR rate increases, showing the consequence of the presence of an inert gas on the laminar speed of the flame, making it slower.

Table 2 - COV_{IMEP}, IMEP low and combustion duration 2750@25Nm

EGR rate	COV_{IMEP} average	IMEP - *low*	CA10-CA90
%	%	bar	°CA
0	1.29%	-0.639	19.01
2	1.13%	-0.627	19.3
4	1.29%	-0.619	20.28
6	1.32%	-0.615	20.85
8	1.27%	-0.605	21.61
10	1.35%	-0.594	21.42
12	1.52%	-0.586	23.32
14	1.60%	-0.569	24.7
16	2.14%	-0.543	26.44
18	1.97%	-0.549	25.79
20	2.35%	-0.514	28.82
22	2.73%	-0.491	30.5
24	2.59%	-0.492	29.63

EGR TEMPERATURE VARIATION

The sensibility study of EGR temperature effects on the engine efficiency has been conducted with engine operating in both operating points. In 2750rpm/25Nm with 16% of EGR and 4000rpm/40Nm with 25%. The tested EGR temperature range was limited by the achievable conditions allowed by experimental setup.

To maintain the brake mean effective pressure at 2.6 bar (corresponding to 25 Nm of torque) and 4.2 bar (corresponding to 40 Nm of torque), the increase in mixture temperature in the intake manifold caused a reduction in the restriction imposed by the throttle, and the relative pressure at intake manifold how shows Figure 9. This increase in the average intake manifold pressure level caused a gradual reduction in the pumping work with the increase in temperature.

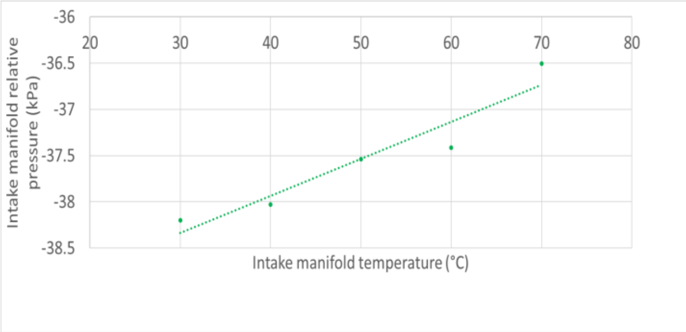

Figure 9 - Pressure restriction measured at intake manifold as function of mixture temperature 4000@40Nm

This reduction in pumping work is consistent with the increase in the intake manifold pressure. Its results in a lower BSFC how can see in Figure 10 and Figure 11.

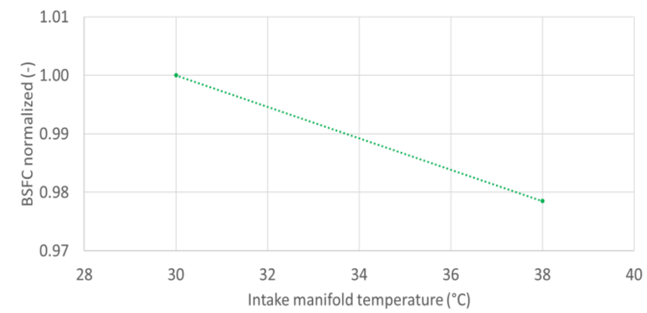

Figure 10 - BSFC with EGR temperature variation 2750@25Nm

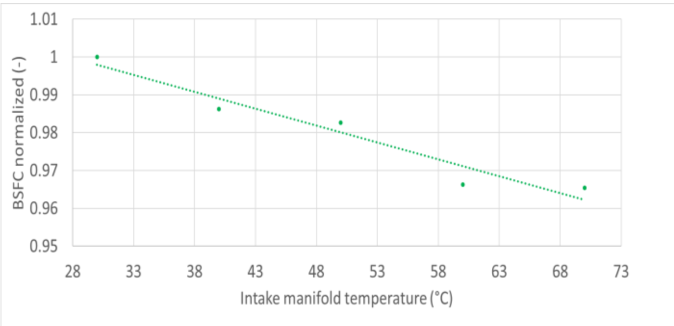

Figure 11 - BSFC with EGR temperature variation 4000@40Nm

The variation tests for the point of 2750@25Nm proved to be limited with the assembled configuration, despite that the results obtained still followed the coherence with what is reported in the literature and the point 4000@40Nm.

The maximum temperature obtained in 2750@25Nm was 38 °C. As can be seen in Figure 10, there was a 2% reduction in specific consumption with the increase in temperature. In the 4000@40Nm the maximum temperature was 70°C and we observed a 3.5% reduction in BSFC.

For both operations points the results are the same, so will be presented combustion parameters of 4000rpm/40Nm because was possible to analyze a bigger temperature range.

For the temperature levels tested, combustion happen faster when the mixture temperature was increased, despite of the combustion phasing did not show the same behavior. The combustion phasing and duration – calculated by the heat release analysis – are shown in Figure 12. The trendlines indicates that crank angle locations corresponding to 5 and 10% burnt mass fractions are happening earlier in the cycle and there is a reduction of 7° CA in combustion duration with mixture temperature increase.

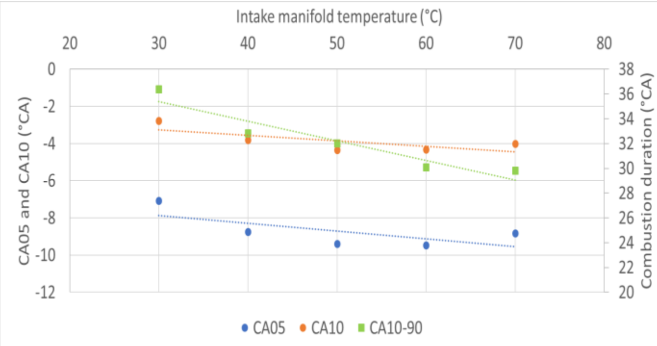

Figure 12 - Combustion phasing and duration as a function of intake manifold temperature

CONCLUSIONS

This work studied the influence of the use of EGR system and the temperature of recirculated gasses on the combustion efficiency and emissions of a PFI engine.

Results show that EGR is promisingly efficient for reducing NOx emissions when applied to SI engines. In addition, it is observed that ethanol is more tolerant to dilution in relation to what is reported in the literature for gasoline, reaching rates of 24% EGR.

The efficiency gain until 7% obtained with EGR when engine operates in Otto cycle is significant given the large number of engines in the market that do not count on dual VVT and operate in this mode. This technology route is a more economically viable compared to VVT and Miller cycle.

Regarding the data obtained with the temperature variation of the EGR, it can be concluded that there was a decrease in the pumping work due to the lower specific mass of the hotter intake gases and the combustion was accelerated. For the temperature variation range obtained with the current EGR system, a lower brake specific fuel consumption values with increase of temperature was observed – to 4000@40 Nm less 3.5% between 60 and 70 °C and to 2750@25Nm 2.0% decreasing in 38 °C.

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