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THE USE OF EGR AND THE INFLUENCE OF ITS TEMPERATURE ON SPARK-IGNITION ETHANOL ENGINES

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***Abstract**: The use of EGR is currently well known as a promising method for modifying the combustion process to reduce NO_x emissions. However, many variables still need further exploration to determine their degree of influence on this technique. In this sense, this work experimentally explores the influence of the exhaust gas recirculation temperature on the combustion & gas exchange processes, efficiency, and emissions of a PFI engine. The tests were conducted with a 3-cylinder engine, of 1,2L of volumetric displacement using ethanol as fuel. The results indicated significant reduction in NO_x and efficiency increase in Otto cycle at some EGR rates. The increasing EGR temperature shows a reduction in pumping work but demonstrates an efficiency decrease in the positive work of the cycle.*

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

1. INTRODUCTION

The exhaust gas recirculation (EGR) technology consists of extracting part of the burnt gases from 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 (NO_x) emissions.

NO_x formation is causally related to the maximum temperature inside the combustion chamber due to the Zeldovich formation mechanism, above 1800 K the formation of these oxides increases considerably (Turns, 2012). The insertion of an inert gas into the combustion chamber decreases the peak combustion temperature, as well as reduces the partial pressure of oxygen, causing reduction in the formation of nitrogen oxides.

In addition to the benefit of reducing NO_x emissions, in spark ignition (SI) engines, the EGR makes possible to increase the efficiency at part 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 throttle condition results in lower pumping work.

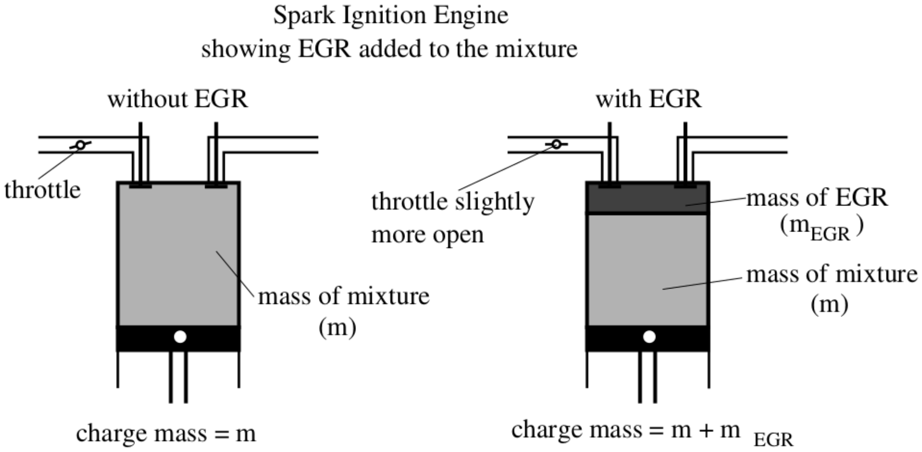

Figure 1 – Comparison between SI without and with EGR.

Adapted from (Abd-Alla, 2001)

The increasing of diluent rate in the cylinder can also cause instability in combustion as shown in Figure 2. There is a trade-off between the maximum EGR rate to be used and the benefits that will be achieved. This maximum rate 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 still now. However, (Splitter, 2016) 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.

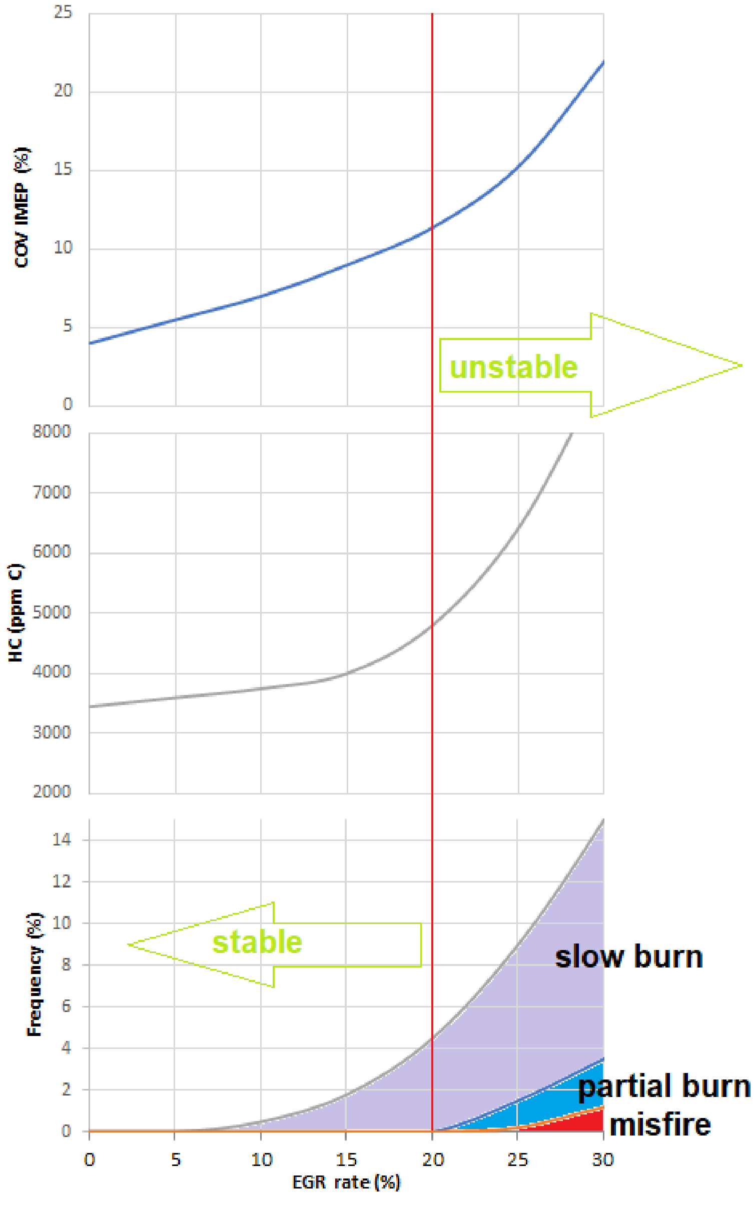

Figure 2 – Coefficient of variation (COV) in IMEP, hydrocarbon (HC) emission and burn behavior.
Adapted from (Heywood, 2018)

According to (Wei *et al.*, 2012) 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.

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. In addition, the influence of EGR temperature on combustion characteristics was also explored.

2. METHODOLOGY

The work was carried out using a commercially available flex-fuel, 1.2L, 3-cylinder naturally-aspirated port fuel injection (PFI) engine, at the Division of Motors and Vehicles of Instituto Mauá de Tecnologia, 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).

To obtain a homogeneous mixture of air and EGR in the three cylinders, a mixer already used in previous works (Camargos et al, 2017) and an additional swirler was installed after the throttle. Figure 3 shows the EGR mixer assembly that was used and Figure 4 shows the assembled system and the exhaust gas extraction point (before the catalyst) for insertion into the intake after the throttle valve.

Figure 3 – EGR mixer.

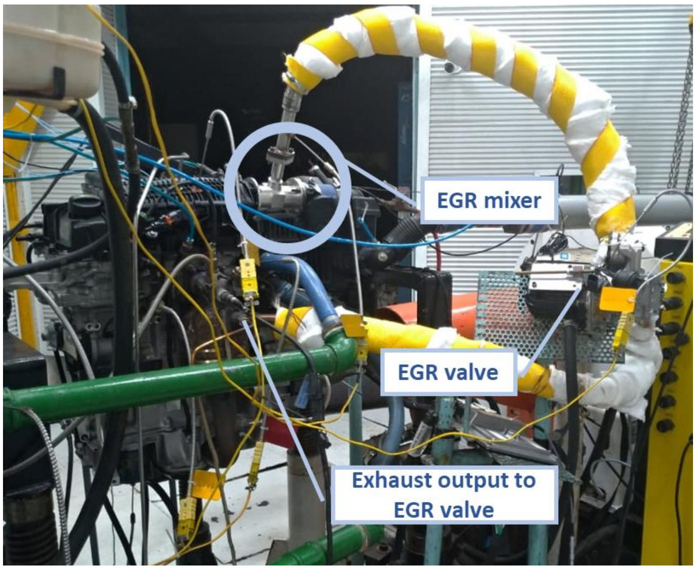

Figure 4 – EGR system.

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 an ECU (Engine Control Unit) original that is completely open for development use. In addition, to record the in-cylinder pressures, the Indimodul data acquisition system and IndiCom® software were used in conjunction with instrumented spark plugs (model AVL-Z131_Y5S) with piezoelectric pressure transducers and an AVL-365 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 CO_2 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: NO_x, THC, CO, O_2, CO_2.

The engine load/speed points were defined based on the 5 most significant points of accumulated fuel consumption in the Brazilian emissions and efficiency test, which uses the American Federal Procedure Tests (FTP-75) (Zabeu *et al.*, 2017). Out of these five points, three were chosen: 1750 rpm/15Nm, 1750/25 Nm and 2750 rpm/25Nm, the first the most representative fuel consumption point, the second is the intermediate and the last the less representative. Due the engine instability on the first point it was excluded and the tests had been made only on the last two points.

The two points on this engine originally work in the Miller cycle, with late opening and closing of intake and exhaust valves. In addition to evaluating the use of EGR in this cycle, the position of the VVT was modified for the engine to operate in the Otto cycle and to verify the differences in gains that would be possible to obtain.

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%) of 8° after the top dead center, and when necessary, the fuel injection was individually adjusted to obtain this value in the three cylinders.

Initially, the intake manifold temperature was maintained at 25 ± 3 °C and the EGR concentration was varied. The EGR rate was calculated according to the following (Baert et al, 1999),

$$EGR = \frac{[CO_2]_{intake} - [CO_2]_{environment}}{[CO_2]_{exhaust}} \quad (1)$$

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 10 minutes. These frequency and time recording conditions were statistically evaluated and found sufficient to evaluate the engine behavior. The indicated data were recorded during 6000 consecutive cycles at each engine operating condition.

3. RESULTS

The tests are in the beginning of the campaign, and only part of all results are presented here, namely those focused on the operation point corresponding to 2750 rpm and 25 Nm.

It is observed both for the engine in the Miller and Otto conditions that there was a significant decrease in the concentration of nitrogen oxides emitted as the EGR rate increases. Compared to values observed at 0% EGR rate, maximum reduction for the Miller cycle was approximately 85% and for Otto around 95%, as it can be seen in Figure 5.

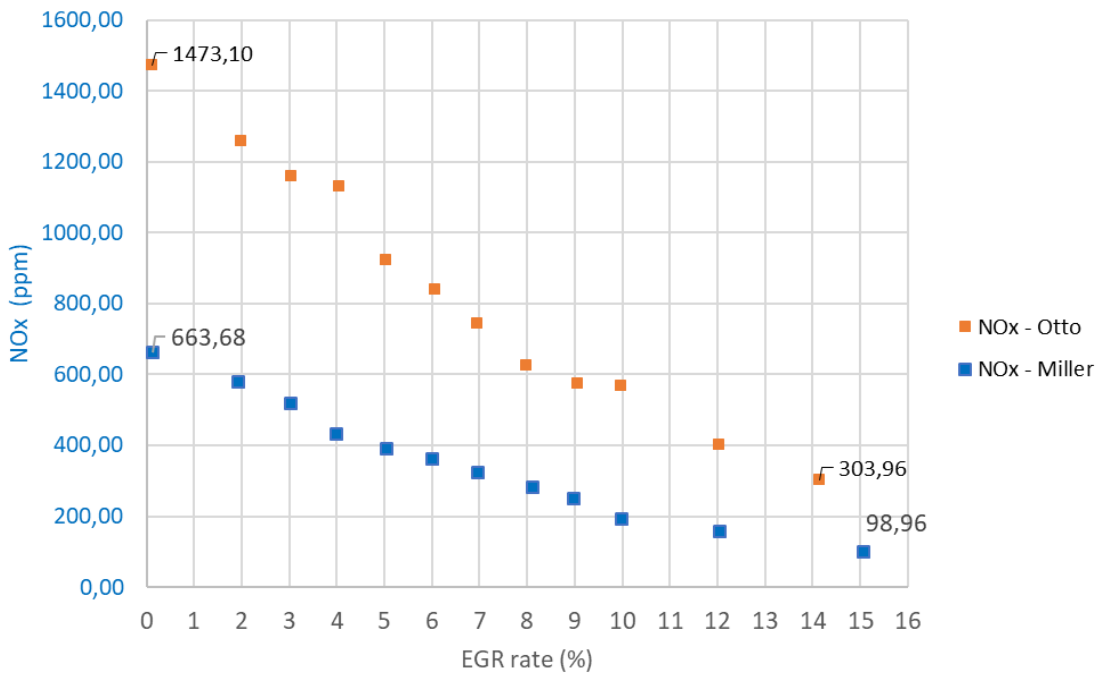

Figure 5 – NO_x for 2750 rpm/25 Nm at Otto and Miller cycles.

When engine operates in Miller regime, there is already a considerable amount of residual gases from the previous cycle - known as internal EGR - which is responsible for reducing the formation of NO_x. It was possible to observe that the engine became unstable with rates of approximately 17% EGR (data not recorded due high variability). Also, it was not possible to obtain high rates in Miller mode as it was reached when engine operated in the Otto cycle (~ 22% still stable).

For the engine operating according to Miller cycle, there were no efficiency gains with the use of external EGR. Figure 6 shows that the increase in the EGR rate does not significantly reduce the values of brake specific fuel consumption (BSFC): the gains obtained in reducing the pumping work have been offset by worsening in the combustion stability due to the presence of the gases that already existed and what was added. After 10% of EGR, the combustion becomes unstable and the BSFC gets worse. The unburned hydrocarbon values started growing at EGR levels around 6%, indicating a deterioration in combustion efficiency.

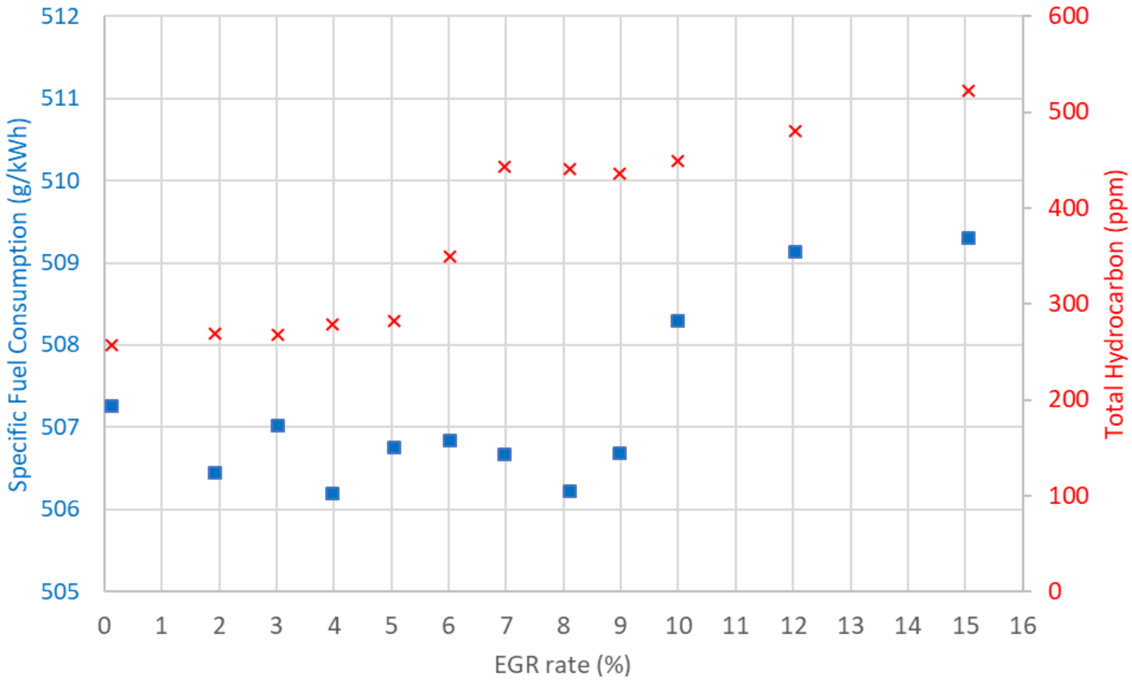

Figure 6 – BSFC and THC at 2750 rpm and 25 Nm for Miller cycle.

The Figure 7 shows the results when the engine operates according to Otto cycle. It is possible to obtain a decrease in the BSFC of about 2% at an EGR rate of approximately 16%. In addition to the gain in specific fuel consumption up to a rate of 22% EGR, the engine was not unstable and the total unburned hydrocarbons (THC) did not significantly raise up.

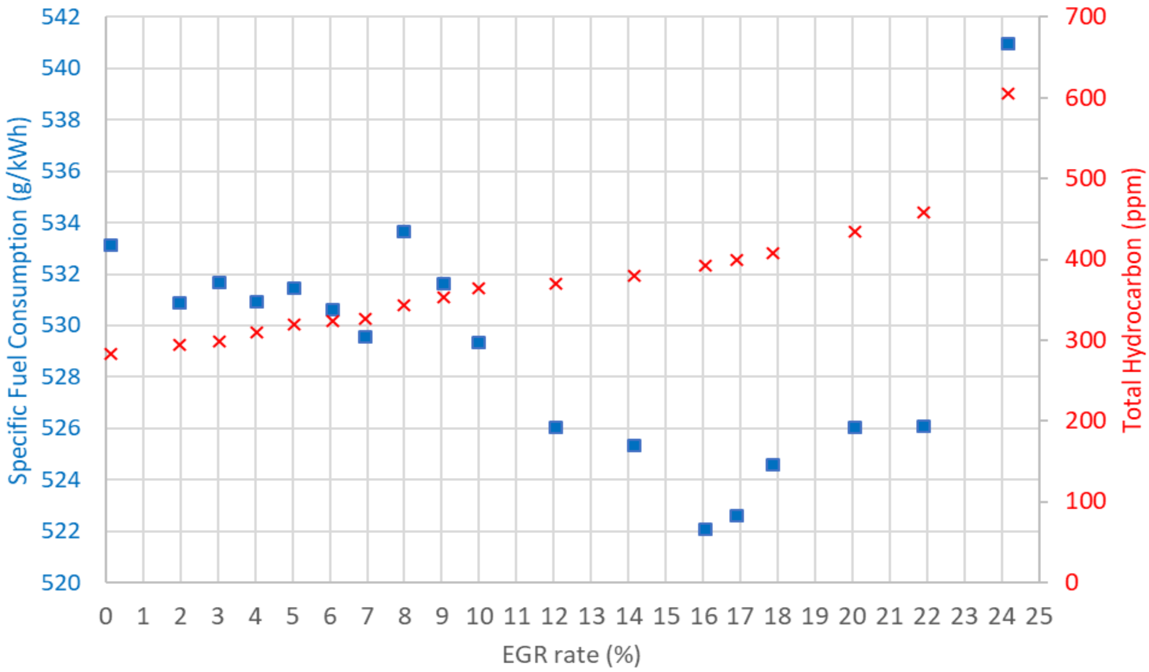

Figure 7 – BSFC and THC at 2750 rpm and 25 Nm for Otto cycle.

Although no significant efficiency gains have been obtained when engine operated in Miller cycle, EGR can still be used in this case to decrease NO_x emissions, without impairing the engine's operation.

The sensibility study of EGR temperature effects on the engine efficiency has been conducted with engine operating in Otto cycle and at the condition of minimum specific fuel consumption (2750 rpm, 25Nm) – 16% of EGR according the previous tests (Figure 7). The tested EGR temperature range was 20 °C to 55 ° C.

To maintain the brake mean effective pressure at 2.62 bar (corresponding to 25 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 raised from -53.25 to -48.57 kPa, as indicated in Figure 8. This increase in the average intake manifold pressure level caused a gradual reduction in the pumping work with the increase in temperature as it can be seen in the low pressure region of the indicated diagram depicted in Figure 9. This reduction in pumping work is consistent with the increase in the intake manifold pressure as shown in Figure 10, which presents a pressure 8.9% higher, when compared to the lower and the higher temperature in the intake manifold.

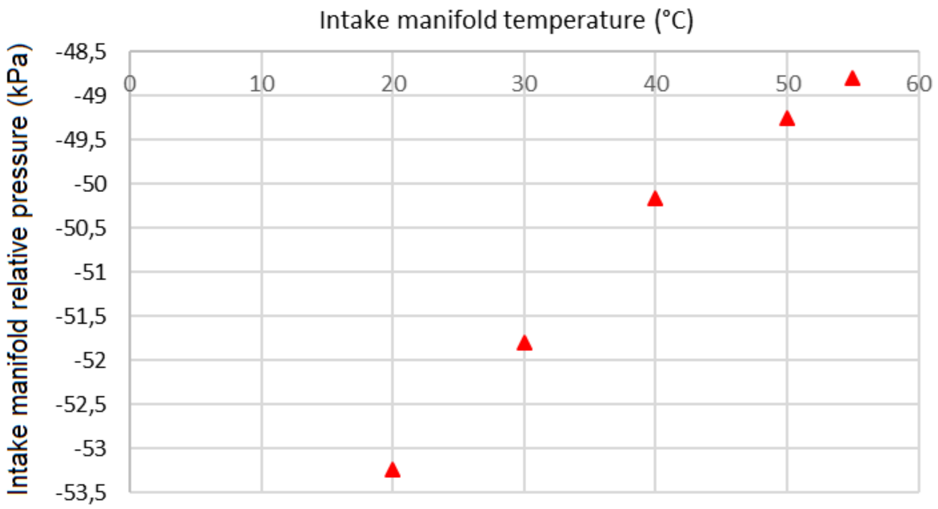

Figure 8 – Pressure restriction measured at intake manifold as function of mixture temperature.

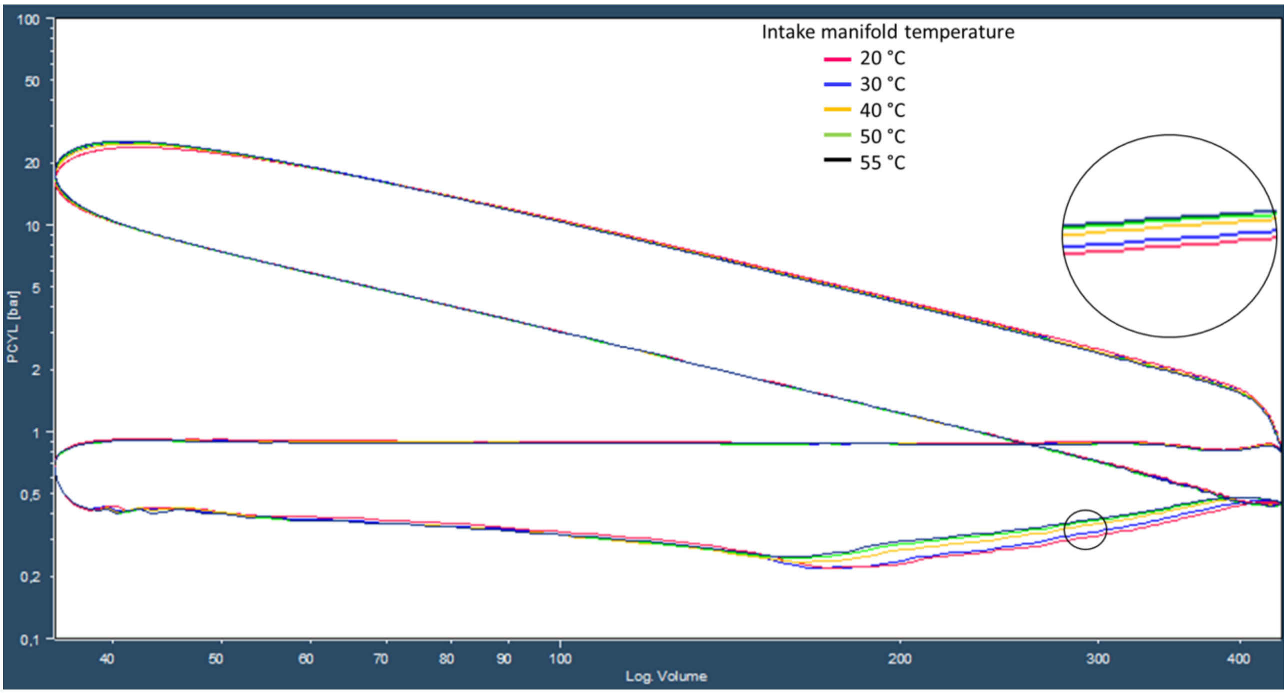

Figure 9 – Indicated diagram: average curves at 2750 rpm and 25 Nm for Otto cycle.
Print screen from IndiCom® software

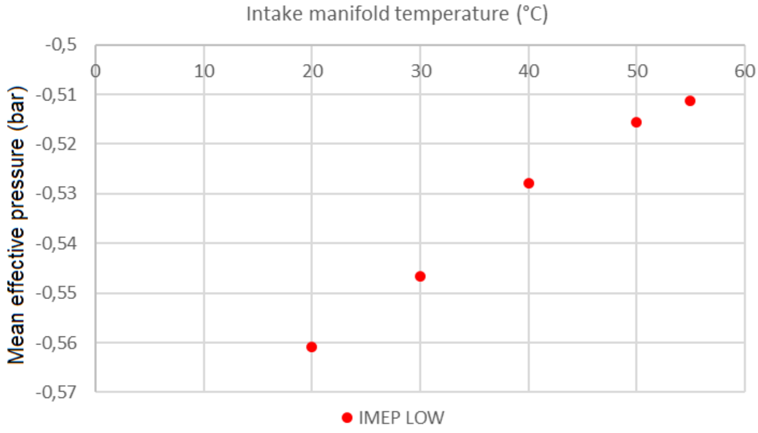

Figure 10 – IMEP LOW as a function of intake manifold temperature.

For the temperature levels tested, combustion seemed to happen a little earlier and a little faster when the mixture temperature was increased. The combustion phasing and duration – calculated by the heat release analysis – are shown in Figure 11. The trendlines indicate that crank angle locations corresponding to 5%, 10%, 50 & 90% burnt mass fractions are happening earlier in the cycle and there is a reduction of 0.02 CA in combustion duration for each °C in mixture temperature increase.

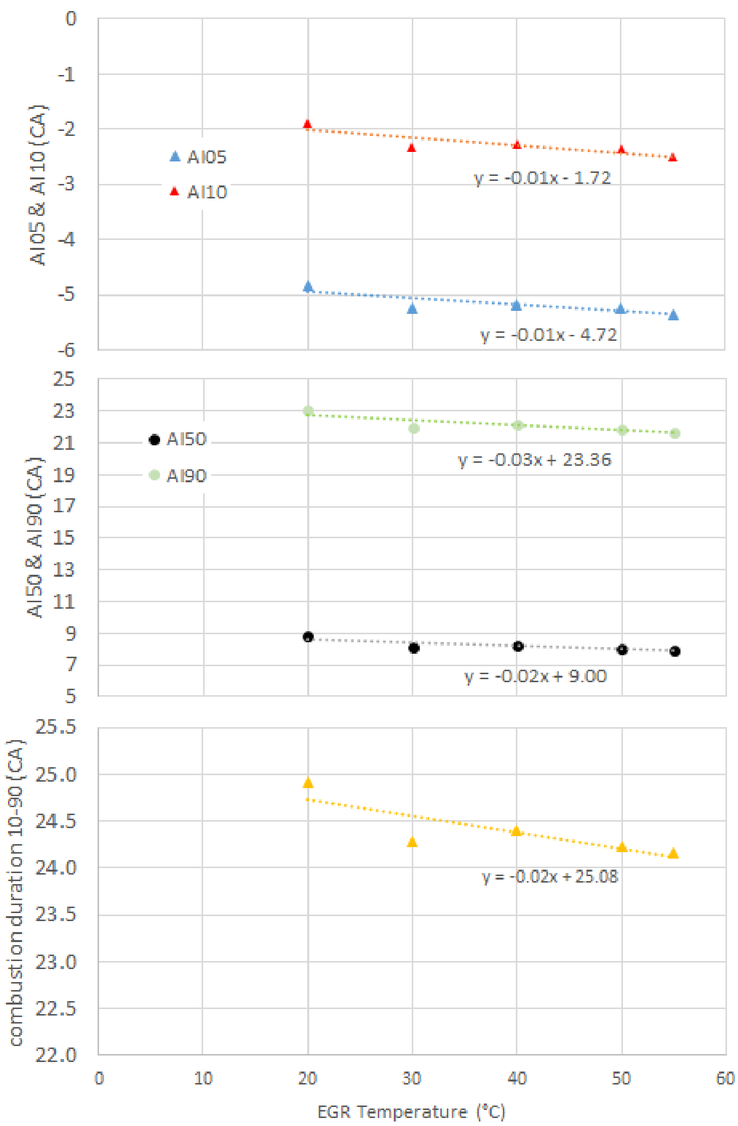

Figure 11 – Combustion phasing and duration as a function of intake manifold temperature.

The combustion stability, shown by the coefficient of variation of the IMEP, also remained practically the same for all evaluated conditions, as indicated in Table 1.

Table 1 - Average IMEP and COV at 2750 rpm and 25 Nm for Otto cycle.

	IMEP_20°C	IMEP_30°C	IMEP_40°C	IMEP_50°C	IMEP_55°C
MIN (bar)	3.02	3.23	3.14	3.24	3.28
MEAN (bar)	3.53	3.53	3.53	3.53	3.53
MAX (bar)	3.70	3.69	3.68	3.67	3.68
CoV%	1.63	1.45	1.46	1.44	1.42

As a consequence of keeping the same BMEP (and IMEP, if one considers engine friction remained constant) for all tested cases, the IMEP HIGH suffered the same absolute reduction of IMEP LOW. All MEP values are shown in Table 2 and plotted in Figure 12.

Table 2 - Average IMEP, IMEP LOW, IMEP HIGH and BMEP at 2750 rpm and 25 Nm for Otto cycle.

	20°C	30°C	40°C	50°C	55°C
IMEP HIGH (bar)	4.09	4.08	4.06	4.05	4.04
IMEP LOW (bar)	-0.56	-0.55	-0.53	-0.52	-0.51
IMEP (bar)	3.53	3.53	3.53	3.53	3.53
BMEP (bar)	2.62	2.62	2.62	2.62	2.62
variation IMEP HIGH (@20°C) (bar)	0.000	-0.014	-0.033	-0.046	-0.049
% variation IMEP HIGH (@20°C)	0.0%	-0.3%	-0.8%	-1.1%	-1.2%
variation IMEP LOW (@20°C) (bar)	0.000	0.013	0.034	0.047	0.050
% variation IMEP LOW (@20°C)	0.0%	-2.3%	-6.1%	-8.4%	-8.9%

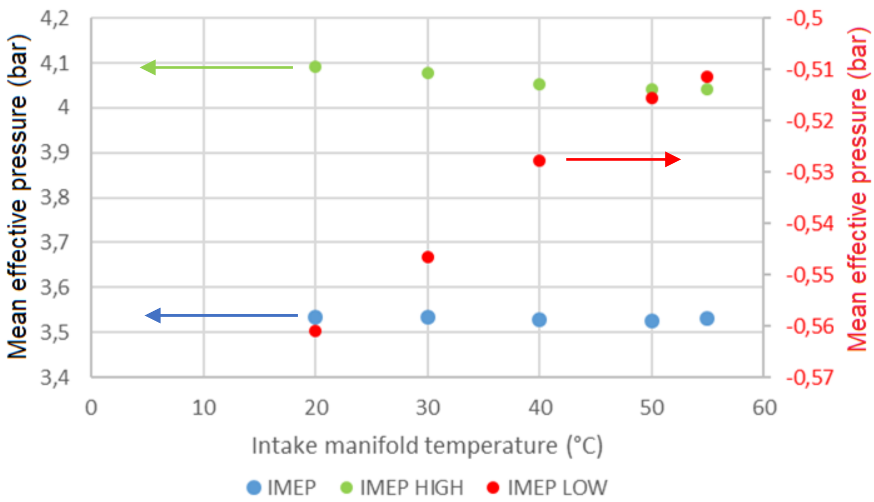

Figure 12 - IMEP, IMEP HIGH and IMEP LOW as a function of intake manifold temperature.

The maximum observed reduction of 50 mbar in the pumping work at 55°C – corresponding to -8.9% variation – in comparison to the 20°C case was accompanied by the approximately same reduction of 49.8 mbar in gross mean indicated pressure (IMEP HIGH). This reduction corresponds to -1.2% in IMEP HIGH which would lead to a reduction of the same order in fuel consumption. However the specific consumption of the engine varied according to the plot in Figure 13 (normalized to the point at 20°C), indicating a maximum reduction of 0.26% at 40°C instead. In spite of having a decreasing trend with the increase of mixture temperature as expected, the measured fuel specific consumption value was higher than expected and could be explained by the measurement uncertainty of the fuel consumption meter which is around 1%.

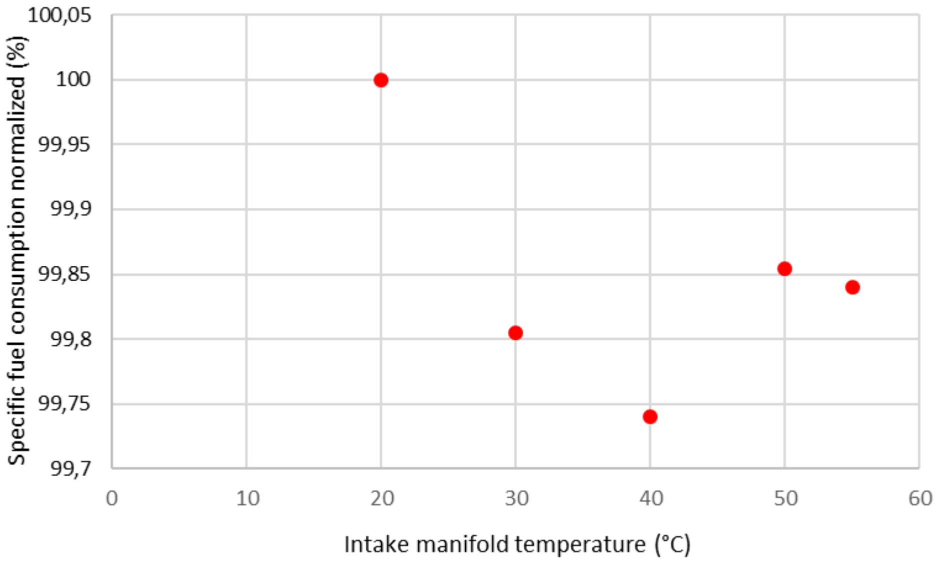

Figure 13 – Specific fuel consumption normalized as a function of intake manifold temperature.

The results show that despite of the gains in the pumping work and observed trends of faster combustion with higher mixture temperatures, the overall cycle did not promoted differences in efficiency that could be captured by fuel consumption measurement device. The work is still in progress and it is desired to reach higher temperatures (~ 90 °C), with the installation of a heater in the EGR line, which will allow analysis at higher mixture temperatures. Improved ways of measuring fuel consumption are planed as well.

4. CONCLUSIONS

Results from previous works show that EGR is promisingly efficient for reducing NO_x 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 22% EGR.

The efficiency gains of 2% 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, but despite this gain, the Miller cycle is still more efficient.

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 a trend to accelerated the combustion. However, for the temperature variation range obtained with the current EGR system, the observed brake specific fuel consumption values have fallen inside fuel measurement device uncertainty range not allowing a concrete conclusion. Additional studies need to be conducted, coupled with simulations and experiments, to confirm whether it would be possible to achieve higher efficiency levels by use of even hotter EGR gases mixed with intake air.

5. ACKNOWLEDMENTS

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