Reduction of Pollutants Emissions on SI Engines - Accomplishments With Efficiency Increase

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This paper presents an experimental study aiming to identify the means to minimize the reduction of the overall performance of a gasoline engine when employing the Exhaust-Gas Recirculation (EGR) technique that reduces NOx emissions. The increase of the compression ratio and turbocharging was evaluated as a mean to recover the original performance. The formation of pollutants and the engine performance were verified at full and partial loads. The results show that the combination of exhaust gas recirculation with turbocharger or through an increase of the compression ratio enhance the relation between the engine performance and the emission of NO. However, the turbocharger seemed to be more sensitive to the negative effects of the EGR technology.
***Keywords**: EGR, NOx, pollutant emissions control*

Introduction

The methods and techniques to reduce emission of pollutants from internal combustion engines usually decrease its performance. Considering the impossibility of a short term modification in the current standards of energy consumption, the most effective way for reducing environmental impacts relies on increasing the efficiency of the thermal engines. In other words, research should be carried out on development of more efficient engines or to apply means, for the current level of technology, to minimize entropy generation. Specifically, for internal combustion engines, a reasonable solution is the reduction on pollutant formation by controlling some combustion parameters in such way that engine performance is kept unaltered. An effective way for reducing nitrous oxide (NO_x) emissions may be accomplished by changing the engine combustion process through the recycling of exhausted gases. This process is accomplished by adding combustion products to the fresh fuel-air mixture during the intake process. This technology is known as *Exhaust Gas Recirculation* (EGR) and has been applied in both spark ignition engines and compression ignition engines. The presence of inert molecules reduces the temperature and the combustion pressure inhibiting the formation of NO, by the thermal mechanism, as well as increases the detonation tolerance, (Heyhood, 1998). This method, however, while effective in reducing NO_x emissions, may lead to considerable losses in engine performance. Several authors, Sato et al (1997), Sousa (2000), Kohketsu (1997), Han, S., and Cheng, W.K.(1998) amongst others, have discussed the advantages and disadvantages of the EGR technology. They have, also, proposed ways to minimize the drawbacks when applying EGR technology in different types of diesel and gas engines.

Bortolet et al. (1999), presented a fuzzy modeling method to an engine air inlet that operates on EGR. Abd-Alla and co-workers (2001), investigated the effects of diluent admissions and intake air temperature in EGR on the emissions of an indirect injection dual fuel engine. They found that diluent addition decreased NOx emissions. Even larger reduction was observed when carbon dioxide was added to the inlet gaseous fuel air charge. Abd-Alla (2002), presented a review on exhaust gas recirculation applied to internal combustion engines. The aim of the work was to review the potential of EGR to reduce exhaust emissions, mostly NO_x, and to delimit the application range of the technology. Zheng et al. (2004), reviewed the advanced and novel concepts in diesel engine exhaust gas recirculation. They claimed that EGR is effective to reduce nitrogen oxides from diesel engines while increasing particulate matter. Power losses are significant when EGR is further increased. Different ways to implement EGR are outlined in the paper. In addition, new concepts regarding EGR stream treatment and EGR hydrogen reforming are proposed. More recently, Lü and co-workers (2005) conducted a fundamental study on the control of the HCCI combustion and emissions by fuel design concepts combined with controllable EGR. Cooling EGR prolonged the time for combustion. EGR had little effects on CO and UHC emissions on HCCI engines.

Despite the number of works on EGR technology, there is still a room for further investigation in SI engines when combining EGR, for pollutant reduction, and turbo charging for performance recovery. The lack of information on this specific subject has, therefore, motivated this work.

Inert gases have a combined effect of reducing NO_x and increase the knocking tolerance, (Heyhood, 1998). Therefore, it is possible to increase the compression ratio as well as to apply turbocharging without reaching self-ignition levels that may jeopardize engine integrity. It is well known that increasing compression ratio or turbo charging the engine favor the NO formation because combustion peak temperature is augmented. Conversely, inert gases mixed with the intake charge, inhibit the formation of NO_x, during the combustion process. In this work, we searched for the best combination on EGR application and engine performance recover.

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The results are presented and discussed only for a 3000 rpm, since the trends remain for a broad range of engine speeds.

Experimental Apparatus and Methodology

The experimental apparatus comprised a hydraulic dynamometer with the auxiliary instruments allowing a complete monitoring of the main engine parameters, such as, torque, power, fuel consumption, air consumption, temperature and related pressures. A gas analyzer was used for measuring the concentrations of CO_2, CO, O_2, NO_x and unburned hydrocarbon, in the combustion products. This analyzer also provided the air-fuel ratio based on the concentration of some specific gases in the exhaust system.

The dynamic pressure inside the cylinder was also measured. The sensor was installed in the cylinder head as a mean to track knocking occurrence. The primary element was a piezoelectric sensor with an operational band from 0 to 250 bar. This element was associated to a system set for acquisition and data treatment.. The signal obtained by the sensor was amplified and processed by a dynamic signal analyzer. All this system was in compliance with the ISO TAG4/WG3 (1999) and the Vianna et al. (1999), procedures for data acquisition. For frequencies, the maximum uncertainty was 1.96% of the measured value up to 1.2 kHz and of 3.2% for frequencies ranging between 1.2 and 1.6 kHz.

The indication of the top dead center (TDC) was done through an optical sensor and a perforated disk installed at the extremity of the dynamometer shaft. The deviation of this measurement, in relation to the geometric measuring, presented an error of 0.3^o regardless of the speed, as observed by Oliveira et al. (1996).

It was used, for the investigation, a 4-cylinder engine, 1927 cm^3 displacement, 8.2:1 compression ratio, single-point fuel injection. Several experiments were conducted in order to set reference parameters, necessary for further comparisons. Volumetric fractions of CO_2, CO, O_2, NO_x, and unburned hydrocarbons were also obtained as a reference. Commercial gasoline with 23% of anhydrous alcohol was used and the tests were conducted at 2500, 3000 and 4000 rpm. The dynamic pressure inside the cylinder was measured for speeds. Test planning is shown in Table 1. Test 1 was set as reference.

The investigation started with the EGR valve installed between the intake and exhaust manifolds (Test 2). This valve was specially designed to vary the amount of exhaust gases that were added to the air-fuel mixture in the intake manifold. For each engine speed, the amount of exhaust gases added to the fresh mixture was varied. The runs were then conducted, at full load, by measuring all the relevant parameters.

Test 3 combined EGR application along with turbo charging but keeping compression ratio the same as in Tests 1 and 2. This set up was replicated in Test 4 though at partial loads, 50 and 70%.

The compression ratio was finally altered to 8.9:1 (Tests 5 and 6). EGR was applied in Test 6 and set off in Test 5. Compression ratio was increases using a cylinder head with a smaller combustion chamber, but with the same geometry.

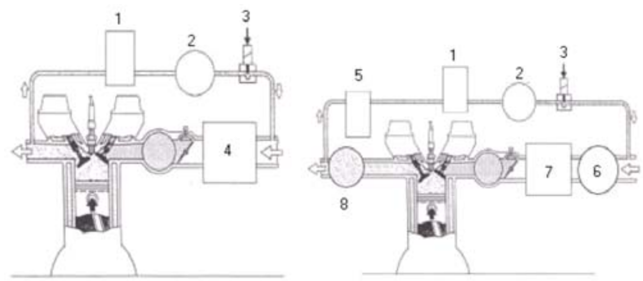
When the compression ratio was 8.2:1, the engine operated with turbocharger. In this case, the EGR system was installed before the compressor inlet.

In Fig. 1 the EGR assembling is shown where it can be seen the details and the accessories for a naturally aspirated and supercharged engine configurations.

In order to ensure consistency of the data, stoichiometric air-fuel ratio was set, to all speeds of the engine. In two configurations, for naturally aspirated engine, the ignition angle was adjusted to the maximum torque in the dynamometer, for each speed. When operating with turbocharger, the optimization of the ignition timing, based on the torque, was limited by the presence of detonation, checked through the dynamic pressure curves.

Table 1. Test Planning.

Test	Compression Ratio	EGR	Turbocharge	Load
1	8.2	uninstalled	uninstalled	Full
2	8.2	*installed*	uninstalled	Full
3	8.2	*installed*	*installed*	Full
4	8.2	*installed*	*installed*	50 and 70%
5	8.9	uninstalled	uninstalled	Full
6	8.9	*installed*	uninstalled	Full



(a) (b)

Figure 1. Exhaust gases recirculation system assembly for naturally aspirated (a) and turbocharged (b) – System components: 1 humidity separator; 2 booster; 3 EGR valve; 4 single point injection; 5 heat exchanger; 6 compressor; 7 single point injection and equalization box; 8 turbine.

The tests were divided in three stages. The first stage conducted in a naturally aspirated engine, for which the compression ratios were set to both 8.12:1 and 8.9:1. Tests were also conducted with turbocharging, Test 3 and 4. These tests, under full load, allowed the assessment of the effect of the exhaust-gas recirculation in the global performance of the engine, for every planned configuration. These experiments also allowed a better understanding of the effects of the EGR technology on the operation of the engine. As a first conclusion, the results showed that the engine presented better overall performance including reduction of the emissions when operating under supercharging. The second stage was conducted with the turbocharger, still under full load, and aimed to infer whether the positive effects of the recirculation remained when some operational parameters were optimized. In the third and last stage, experiments were conducted under partial loads, keeping the supercharging and the optimized configuration.

The following procedures were adopted for each of the three stages:

Procedure 1: this procedure was employed for all three engine configurations running at 2500, 3000 and 4000 rpm. For each of these configurations, the spark timing was adjusted for the highest torque without the presence of knocking. The stoichiometric air-fuel ratio was set the same as the engine without recirculation.

For every speed, the recirculation ratio of the exhaust gases was increased, without altering the load or the ignition timing. This would result in a fall of the speed, and, as the fuel flow remained the same, the air-fuel mixture kept enriching as the recirculation increased. This procedure allowed the assessment of the effects of applying EGR technology, for the different operating regimes. Also, after the experiments, it was clear which configuration was more sensitive to the negative effects of the EGR. The results clearly indicated that turbocharging is the configuration that associates the best performance to the highest reduction on the emissions.

Procedure 2: based on the results achieved in procedure 1, the engine with supercharging was tested for the same speeds, however, the air-fuel ratio was kept stoichiometric and the ignition timing adjusted for the maximum torque, for each recirculation ratio.

Procedure 3: again with turbocharging and the engine optimized to full load, the investigation was also conducted with 75% and 50% of the full load, but varying the recirculation ratio.

The degree of recirculation (percentage) was defined by relating the amount of CO_2 in the intake manifold with that in the exhaust pipe, through the Eq. (1).

$$EGR(\%) = \frac{CO_{2adm} - CO_{2amb}}{CO_{2ex} - CO_{2amb}} \cdot 100\% , \qquad (1)$$

where the carbon dioxide concentrations used in the expression are such that CO_{2amb} was measured in the environment, CO_{2adm} was checked in the intake manifold and CO_{2ex} was measured in the exhaust gases.

Results and Discussion

The experimental results obtained in Procedure 1 allowed the investigation of, both, engine performance and emissions levels. In addition, it was possible to infer the effects of the recirculation in the progress of the flame front, for the engine with compression ratios of 8.2:1 and 8.9:1. When applying turbo charge the compression ratio was fixed at 8.2:1.

The tests were conducted under full load, varying the degree of recirculation (EGR) as well as engine speed. Here, only the results at 3000 rpm are shown and discussed since for the remaining speeds the trends were the same.

Figures 2, 3 and 4 show the dynamic pressure against crank angle for different degree of gas recirculation as well as ignition timing, for three configurations, Test 1, 6 and 3, respectively.

It can be observed in Fig. 2 that, keeping the amount of fuel and the spark timing, while increasing the presence of inert gases in the combustion chamber, the flame front was decelerated resulting a displacement of the pressure curve in relation to the Top Dead Center (TDC). The maximum pressure falls from 38 MPa, without recirculation, to 30 MPa with 8.11% of EGR. One of the consequences is the reduction of the net work of the cycle, considering that the curves, in the compression stage, are coincident until very close to the TDC. The practical results were, then, lower torque and power, which, in turn, decreases the engine global performance.

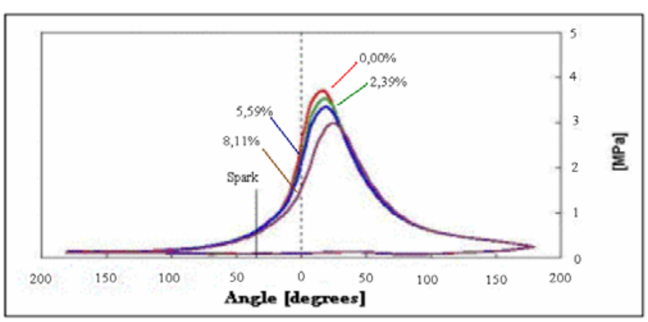


Figure 2. Pressure curves in the interior of the combustion chamber as a result of the crankshaft angle and the percentage of recirculation for a 3000 rpm for the naturally aspirated engine with a compression ratio of 8.2:1 and ignition angle of 35°.

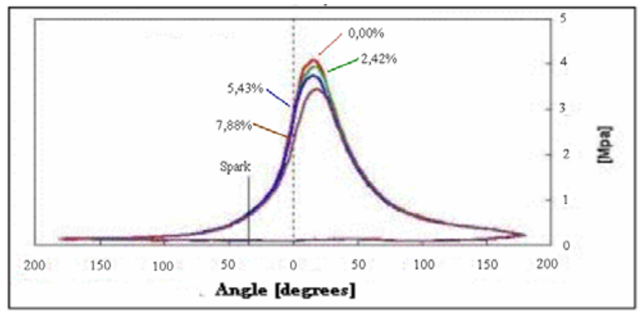


Figure 3. Pressure curves in the interior of the combustion chamber as a result of the crankshaft angle and the percentage of recirculation at 3000 rpm for a naturally aspirated engine with compression ratio of 8.9:1 and ignition angle of 37°.

Figure 3 presents the results in which the engine compression ratio was increased to 8.9:1 and the spark timing adjusted to maximum torque without the presence of detonation. Two cases can be seen, first with the EGR valve deactivated and the second where EGR was operational - Test 5 and 6, respectively. Due to the increase in the compression ratio, it was observed an increase in the maximum pressure up to 42 MPa, no recirculation, and as high as 35 MPa with 7.88% in gas recirculation. The trends here, regarding engine performance were about the same as discussed in Fig. 2, Test 2.

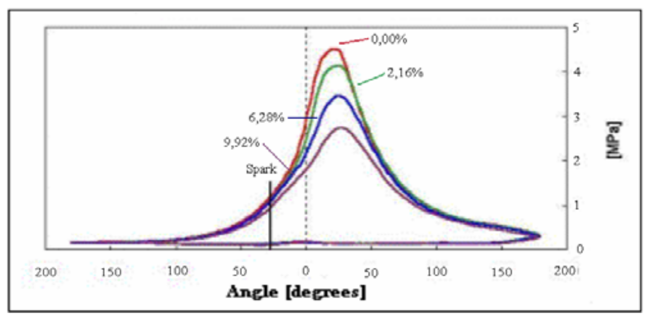


Figure 4. Pressure curves in the interior of the combustion chamber as a result of the crankshaft angle and the percentage of recirculation at 3000 rpm for the turbocharged engine, with a compression ratio of 8.2:1 and ignition angle of 30°.

Figure 4 shows that, some recirculation resulted in a drastic reduction of the maximum pressure of the cycle. However, with the same degree of recirculation the value of the maximum pressure of the cycles is quite similar, in all three configurations.

Also, in Fig. 2, 3 and 4 it possible to see the effect of the EGR on the combustion ratio as well as on the maximum cycle pressures. The reduction in the peak of the pressure is 20, 15 and 40% for the engine operating with compression ratio of 8.12:1, 8.9:1 and supercharged, respectively. Under such conditions, neither the spark timing nor the air-fuel ratio were adjusted. The results show that with turbocharger the engine was more sensitive to the increase in gas recirculation compared to that operating in a naturally aspirated mode. That resulted in a great loss of the net work of the cycle, which strongly affected the effective power of the engine. This was all confirmed by the dynamometric tests.

Figure 5 shows the results of the dynamometer tests, under full load, for a fixed ignition angle, while varying gas recirculation through the EGR valve. It is possible to notice that for a naturally aspirated engine, with compression ratios of 8.2:1 and 8.9:1 did not show great variations of power output with the increase of the degree of exhaust gas recirculation However, for the supercharged engine, recirculation progressively reduces the power. This loss, when the EGR was set beyond 8%, went down to the point where the engine operated as it were naturally aspirated, It can be seen, in Fig. 5 that the error bars are mixed with the experimental points.

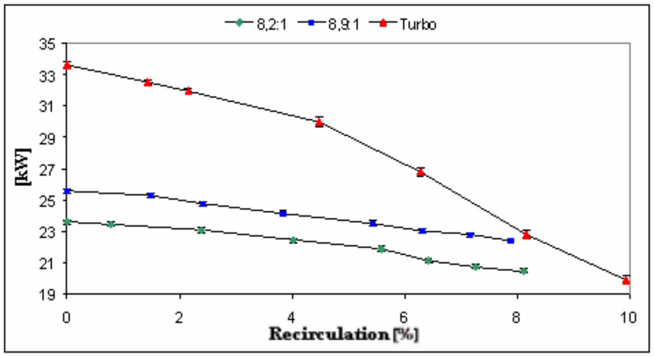


Figure 5. Power as result of the recirculation of the exhaust gases for the engine with compression ratios of 8,2:1; 8,9:1 and turbocharged at 3000 rpm.

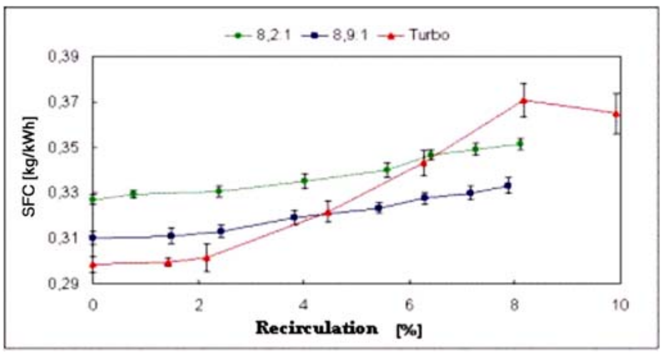


Figure 6. Specific Fuel Consumption as a result of the recirculation of the exhaust gases for the engine with compression ratios of 8.2:1; 8.9:1 and turbocharged at 3000 rpm.

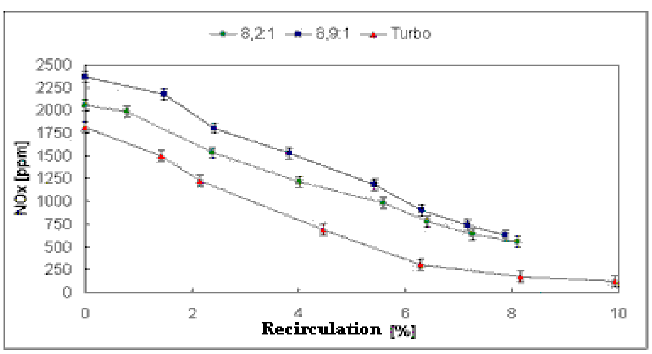


Figure 7. NOx emissions as a result of the recirculation of the exhaust gases for the engine under full load with compression ratios 8.2:1; 8.9:1 and turbocharged at 3000 rpm.

The global performance of the engine, supercharged, and consequently, its specific fuel consumption (SFC) are strongly affected by the recirculation. Figure 6 presents the results of the experiments with all three configurations. The curves show a small variation in the SFC, for naturally aspirated engine, which was not observed when operating with turbocharger.

The NO_x, CO and HC volume fractions were estmated in all the tests reported. The efficiency of the EGR in inhibiting the formation of NO and, consequently, reducing the emission of NO_x with the engine under full load can be observed in Fig. 7. Under these circumstances, the recirculation allowed the increase in the compression ratio, while keeping lower emissions of NO_x. Higher levels of NO_x emissions were not observed after increasing the compression ratio, which, in turn, enhanced engine performance. Figure 7 also indicates that the efficiency of the recirculation, for the turbocharged engine, is much higher than that observed for the naturally aspirated engine. The trends are the same, regardless of the compression ratio, even though the engine performance showed a negative sensibility to the increase of the recirculation.

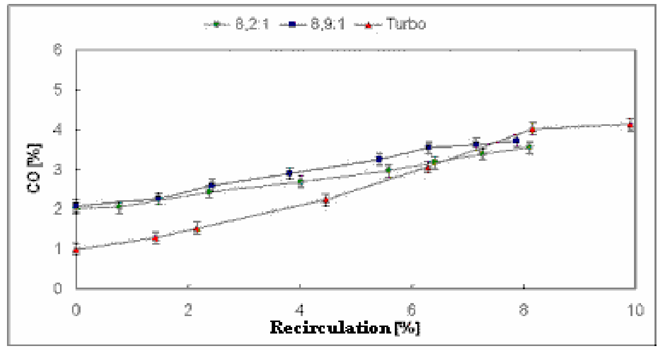


Figure 8. CO emissions as a result of the recirculation of the exhaust gases for the engine under full load with a compression ratio 8,2:1; 8,9:1 and turbocharged at 3000 rpm.

As regarded to the emissions of other pollutant gases, the EGR seemed to perform unfavorably. Figure 8 and 9 show that the emissions of CO and HC increased, for 8% recirculation, about 75 and 67%, respectively,. Since EGR has a direct interference in the combustion process this might explain the increment in the emission of HC. The higher emissions of CO are due to the enrichment of the mixture, as pointed out by Sousa (2000).

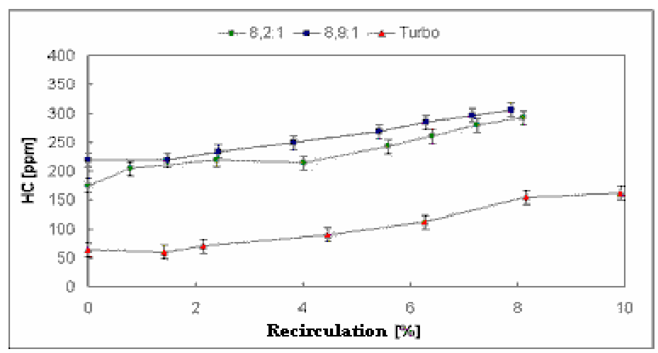


Figure 9. Emissions of HC as a result of the recirculaion of the exhaust gases for the engine under full load with a compression ratio 8,2:1; 8,9:1 and turbocharged at 3000 rpm.

These preliminary results indicate that, as regarded to emissions and engine performance, operation with supercharger seem more appropriate than a naturally aspirated configuration. Therefore, Procedure 2 was carried out only for the engine with turbocharger. In this procedure the tests were done by adjusting the ignition angle for the maximum torque while keeping stoichiometry for the air-fuel ratio. The results of this procedure are presented in the following.

In Fig. 10 it is possible to observe the influence of the EGR on the dynamic pressure of the combustion chamber, Test 3. It can be seen that, by adjusting the spark angle and the speed, the loss of output power with 0 and 4.2% of recirculation is much lower than the one found for the same conditions of Procedure 1, fixed spark angle. This loss of output power is observed more clearly in Fig. 11, 3.5% reduction in the power for a recirculation ratio of 4.2%.

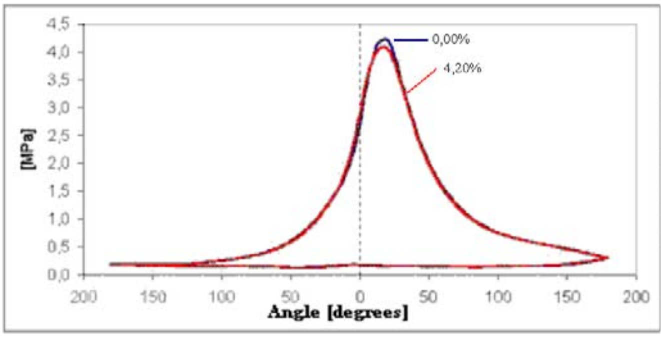


Figure 10. Dynamic pressure curves for two recirculation ratios of the turbocharged engines, under full load, with the adjustment of the spark angle for maximum torque, at 3000 rpm.

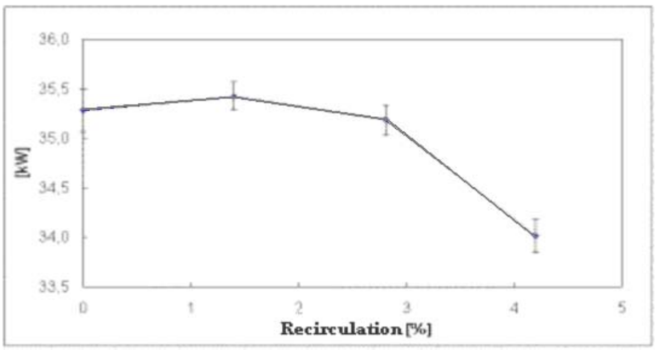


Figure 11. Power curve as a result of the recirculation of the turbocharged engine, under full load, with the adjustment of the spark angle for maximum torque, at 3000 rpm.

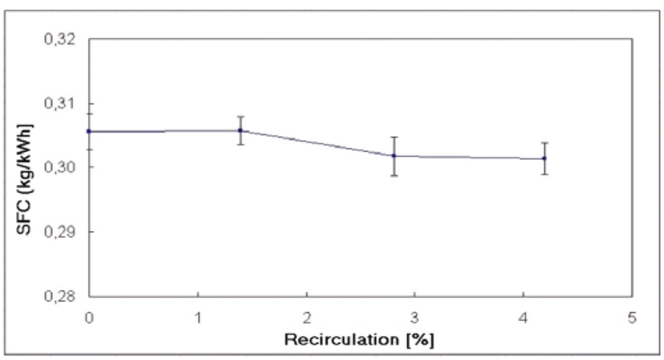


Figure 12. Specific Fuel Consumption (SFC) as a result of the recirculation of the turbocharged engine, under full load, with the djustment of the spark angle for the maximum torque, at 3000 rpm.

The loss in power output, showed in Fig. 11, was followed by a slight reduction in the specific fuel consumption (SFC), as Fig. 12 shows. Even though the maximum recirculation ratio achieved was not as high as that when the spark angle was fixed (Procedure 1), a 0.1% reduction in the specific fuel consumption was observed in relation to the original engine, for the same degree of recirculation.

Figures 13, 14 and 15 present the emission of pollutant, with the ignition timing adjusted for maximum torque under full load with turbocharging. These results show that, under these conditions, the turbocharger worked as a power output recover without increasing emissions. Although the emissions of NO_x was reduced only 22%, with 4% recirculation, compared to a reduction of 54% achieved with Procedure 1, HC emissions were kept relatively stable. As regarded to CO emissions, the adjustment of the spark angle along with constant stoichiometric air-fuel ratio was extremely effective. It is important to mention that, under constant regime, the temperature of the exaust gases is quite high, and therefore, emission values of CO and HC, might be masked.

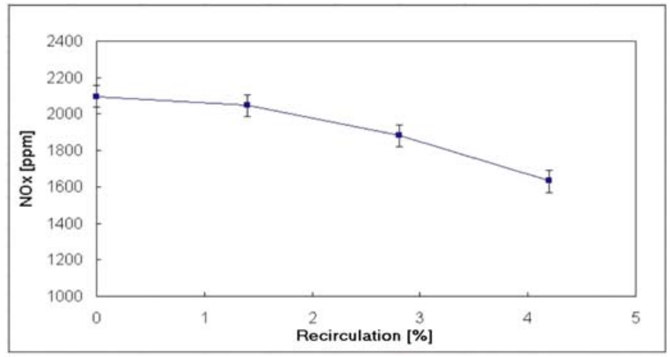


Figure 13. NOx emission curves as a result of the recirculation of the turbocharged engine, under full load, with the adjustment of the spark angle to full torque, at 3000 rpm.

The results under partial loads, following Procedure 3, are presented in Fig. 16 and 17. It is possible to observe that recirculation had positive and negative effects under partial loads.

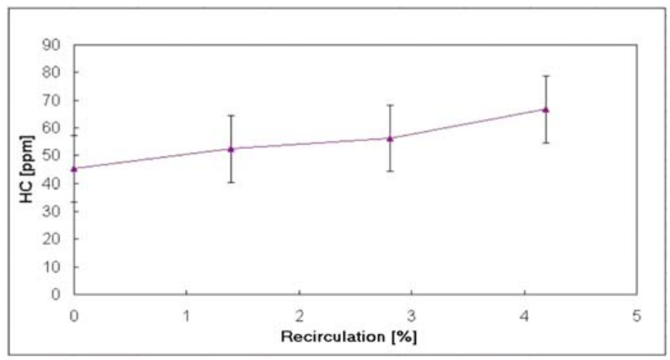


Figure 14. CO Emission curves as a result of the recirculation of the turbocharged engine, under full load, with the adjustment of the spark angle to full torque, at 3000 rpm.

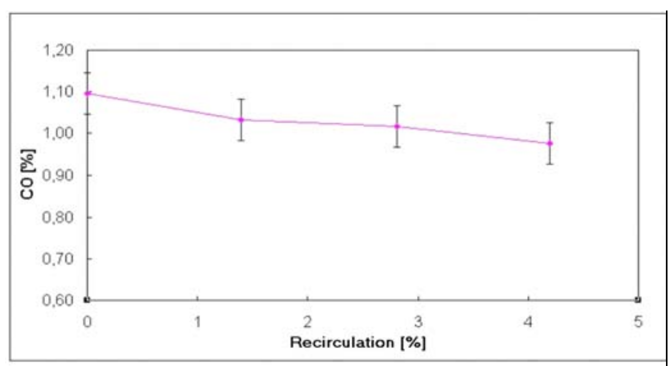


Figure 15. HC Emission curves as a result of the recirculation of the turbocharged engine, under full load, with the adjustment of the spark angle to full torque, at 3000 rpm.

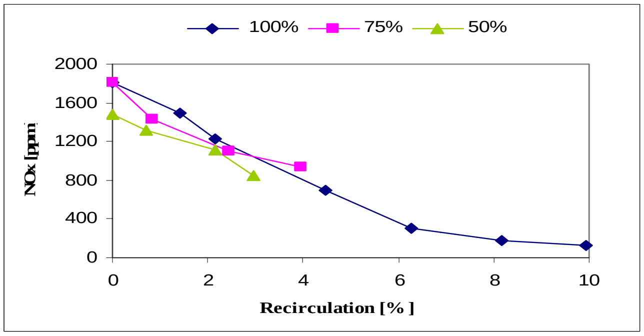


Figure 16. NOx Emission curves as a result of the recirculation of the turbocharged engine, under 50%, 70% and 100% of the full load, with the adjustment of the spark to full torque, at 3000 rpm.

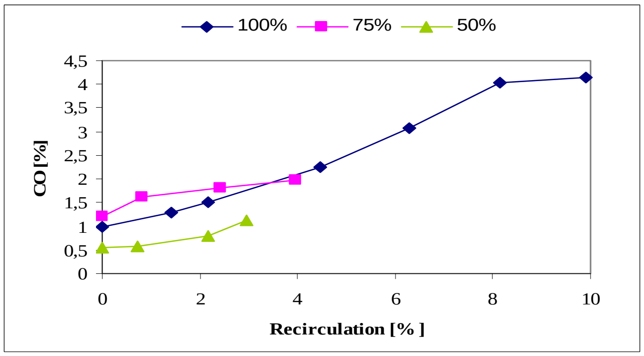


Figure 17. CO Emission curves as a result of the recirculation of the turbocharged engine, under 50%, 70% and 100% of the full load, with the adjustment of the spark angle to full torque, at 3000 rpm.

Conclusions

The experimental investigation conducted in this work indicated that increasing the compression ratio of a SI engine that employs EGR technology is an effective way to correct the loss in the performance. Some degree of recirculation was enough to keep emissions down to an acceptable level. Increasing the compression ratio from 8.2:1 to 8.9:1, with 6% recirculation ratio leaded to a 50% reduction in the emission of NO_x, combined to a 10% increase in power output of the engine.

With supercharging, the engine operated quite sensitive to recirculation. High degree of recirculation suppressed the presence of the turbocharger. Gains in engine performance with a substantial reduction in the emissions of NO_x were accomplished only with proper selection of degree of the recirculation, proper spark timing and a specific value of air-fuel ratio. Comparison between the engine in its standard configuration and after applying turbocharging and recirculation ratio of 4.2% showed reduction of 20%, 61% and 52% in the emissions of NO_x, HC and CO respectively. The increase in power was 33% with 3% reduction in the specific consumption at 3000 rpm.

The advantages and disadvantages of the use of EGR technology in engine with supercharging, under full load, were the same as under partial loads.

This work presented the results only for 3000 rpm, however, similar trends were achieved for other speeds.

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