Switching strategy between HP- and LPEGR systems for reduced fuel consumption and emissions

José Manuel Luján[a], Carlos Guardiola[a], Benjamín Pla[1 a], Alberto Reig[a]

[a]*CMT Motores Térmicos, Universidad Politécnica de Valencia, Valencia, Spain.*

Abstract

Exhaust gas recirculation (EGR) plays a major role in current Diesel internal combustion engines as a cost-effective solution to reduce NO_x emissions. EGR systems will suffer a significant evolution with the introduction of NO_x after-treatment and the proliferation of more complex EGR architectures such as low pressure EGR or dual EGR. In this paper the combination of high pressure EGR (HPEGR) with low pressure EGR (LPEGR) is presented as a method to minimise fuel consumption with reduced NO_x emissions. Particularly, the paper proposes to switch between HPEGR and LPEGR architectures depending on the engine operating conditions in order to exploit the potential of both systems. In this sense, given a driving cycle, in the case at hand the NEDC, the proposed strategy seeks the EGR layout to use at each instant of the cycle to minimise the fuel consumption such that NO_x emissions are kept below a certain limit. The experimental results obtained show that combining both EGR systems sequentially along the NEDC allows to reduce noticeably the NO_x emissions of the HPEGR system with a small impact on the fuel consumption.

Keywords: Diesel engine emissions, Low Pressure EGR, NEDC

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1. Introduction

The evolution of automotive internal combustion engines (ICE) is driven by the needs of achieving simultaneously stricter emission regulations and increased efficiency and drivabilty demands. Despite significant improvements have been introduced in production engines during the last years there are still important challenges to address in order to satisfy emission regulations and customer demands. Particularly, in the case of Diesel engines, NO_x emissions is one of the most challenging issues to address in the near future. During the last decade high pressure exhaust gas recirculation (HPEGR) has become an essential system in modern Diesel engines as a cost-effective solution to meet stringent NO_x regulations. In fact, the need for higher EGR rates has led to an evolution of the EGR and turbocharging

[1]email: benplamo@mot.upv.es; phone number: +34 963 877 650
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systems. Traditional pneumatic valves are being replaced by solenoid and DC motor valves, and EGR coolers are a widespread solution nowadays. Nevertheless, while high EGR rates ($\geq 50\%$ at some operating conditions) are required and its impact on engine efficiency and other pollutant emissions as particulate matter (PM) becomes important, different approaches to the standard HPEGR should be considered.

One possible approach to NO_x control is after-treatment systems such as lean NO_x traps (LNT) and selective catalytic reduction (SCR) systems, which are a widespread solutions in heavy duty Diesel engines and are spreading in the light duty sector. Although after-treatment entails a penalty in fuel consumption due to the increase in back-pressure and the needs of regeneration strategies, they may impose a lower fuel penalty than other strategies for controlling the raw pollutant production.

Another possibility is to consider Low Pressure EGR, hereinafter LPEGR. The traditional HPEGR consists in guiding a fraction of the exhaust gas from the exhaust manifold (upstream the turbine) to the intake manifold (downstream the compressor). In this configuration the exhaust gas is reintroduced into the cylinders at high temperature despite using EGR coolers due to the high temperatures at the turbine inlet. In addition, the introduction of the EGR close to the cylinders usually leads to a poor homogeneity in the EGR distribution amongst cylinders [1]. Both effects impact negatively the engine efficiency and emissions, especially if Low Temperature Combustion (LTC) concepts are applied [2]. LPEGR takes the exhaust gas downstream of the after-treatment system and drives it to the compressor inlet [3]. While the HPEGR routing has been traditionally preferred because of compressor wheel reliability, the widespread use of Diesel Particulate Filters (DPF) in current engines allows the use of LPEGR configuration [4]. LPEGR becomes a suitable alternative to HPEGR since it can provide high EGR rates without a significant increase in intake temperature and minimising cylinder-to-cylinder charge dispersion [5], amongst other benefits related to the turbocharger operation [6, 7, 8]. In general, due to the better EGR distribution [9] and lower temperature [10, 6], the use of LPEGR route involves a reduction in NO_x and PM. Nevertheless, the HPEGR route has a faster settling time than that of the LPEGR route due to the length of the EGR path [11, 12], produces lower HC emissions and shows a higher efficiency, especially at cold conditions, due to the increase in the intake temperature [13].

Today there is no general consensus about which is the most cost-effective solution for NO_x control, so HPEGR, LPEGR and aftertreatment will coexist during the next years [14]. Particularly, different authors propose the dual-loop EGR system as a possible method to combine the advantages of the HP- and LPEGR routes [15, 16, 17]. Those works are focused on controlling simultaneously both EGR circuits to reach the intake conditions which lead to the desired fuel consumption and emissions. Considering two different EGR systems working simultaneously makes the air loop control more complex and important problems concerning the gas fraction estimation and control must be addressed [18, 19]. A simpler approach to take advantage of the benefits o both LP- and HPEGR systems is to choose which

Stroke (S)	88 mm
Bore (D)	85 mm
Number of cylinders (z)	4
Displacement	2000 cm^3
EGR	HP
Turbocharging system	VGT
Valves by cylinder	4
Maximum power	125 kW@4000 rpm
Compression ratio	17:1

Table 1. Basic engine features

one to use depending on the engine operating conditions but without combination of both routes at the same time. The present paper follows the second approach since the EGR loops will not operate simultaneously, *i.e.* the EGR is carried out alternatively with the HP- or the LPEGR system. Then, the problem addressed may be stated as find the optimal sequence of switches between the HP- and LPEGR circuits to minimise the fuel consumption during the NEDC given some constraints on the NO_x emissions. The NEDC has been chosen because is the driving cycle used to assess the pollutant emissions with current regulations in EU.

The paper is organised as follows: Section 2 describes the experimental facility and the tests carried out. In section 3 the analysis of the tests with the HP- and LPEGR architectures allows to identify the operating conditions where any of the two considered EGR loops shows a better potential. Section 4 provides a mathematical formulation and solution of the addressed problem, *i.e.* choose at each instant of a driving cycle, namely NEDC, the EGR loop which minimises the fuel consumption taking into account a maximum NO_x limit. The experimental results confirming the suitability of the proposed strategy are presented in section 5. To conclude, the most important contributions of the paper are outlined in section 6.

2. Experimental set up

The study of the effects of the LP- and HPEGR architectures on engine performance and emissions has been performed experimentally with a state-of-art 2.0 litre HSDI Diesel engine meeting EURO V. The engine, whose main characteristics appear in table 2, was originally equipped with variable geometry turbine (VGT), intercooler, Diesel particulate filter (DPF) and a cooled HPEGR loop.

As shown in figure 1 the engine was upgraded with a LPEGR circuit. An open code ECU was used to modify the engine calibration. Also, the engine was fully instrumented to measure temperatures and pressures in different locations of the intake and exhaust lines. The engine was installed in a test cell equipped with a variable frequency fast response dynamometer able to carry out engine in the loop tests.

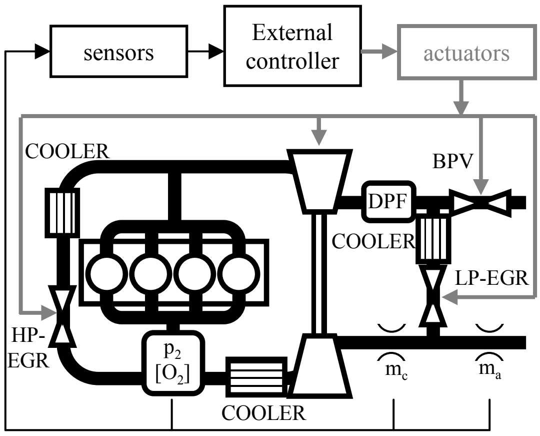

Figure 1. Experimental set up.

Regarding pollutant emissions, a Horiba Mexa 7100 DEGR was used to measure the intake CO_2 and exhaust gas concentrations (NO_x, HC, CO, CO_2 and O_2). Both intake and exhaust CO_2 have been measured by a non-dispersive infrared analyser (NDIR). A heated chemiluminescent detector (HCLD) allows measuring NO_x emissions. The HC analyser consists of a heated flame ionisation detector (HFID). Pollutant measurements were taken before the after-treatment system.

To address the effects of the EGR architecture on engine fuel consumption and pollutant emissions the engine behaviour in the NEDC has been analysed. The reason for such selection is that on the one hand, this cycle represents the operating conditions where the emission limits should be met. On the other hand, analysing the engine behaviour with both the LP- and HPEGR architectures provides insight on the conditions where they show their best potential. The NEDC tests have been conducted according to the methodology described in [20].

The control strategy of the engine is based on the use of look-up tables, where the desired operating parameters (set points) are mapped with the engine speed and fuel demand. Particularly, in the case of the air loop control, due to the coupling between its main systems, *i.e.* EGR circuit and VGT [11], the standard strategy is to apply a divide and conquer approach. At low load and speed, where EGR may be applied, the EGR valve is used to follow an air mass flow set point, while the turbine is controlled according to a position set point. In this sense, in the operating area where EGR is applied, the intake pressure is not controlled in closed loop. Whereas, when the engine operates at high loads or high speeds, *i.e.*, far from the homologation region, the standard approach is to close the EGR valve and control the intake pressure in closed loop with the turbine. Then, the air mass flow and the intake pressure are not controlled simultaneously in closed loop in any circumstance, avoiding control problems due to the coupling between EGR and turbocharging systems. Despite the coupling issues in the air loop are strongly reduced when the LPEGR loop is used (the mass flow through the turbine is not reduced when increasing the EGR), the standard control strategy with two operating regions has been used with the LPEGR configuration. Moreover, the standard engine calibration has been maintained independently of the EGR layout used. Then, the standard engine set points, which are optimised for the HPEGR layout, have been also applied when the LPEGR circuit is employed. Of course, in a final application, the engine calibration is to be adapted to the EGR architecture used, however this has been avoided in the present study because of two main reasons:

- Sharing the same calibration makes the EGR architecture the only difference between the tested configurations, in such a way that differences in engine performance can not be attributed to other reasons such as differences in the injection parameters or other control variables.
- The complexity of carrying out a complete calibration of the engine exceeds the scope of the present work.

The results obtained in such tests were used to design the optimal LP-HPEGR sequence to minimise the fuel consumption given some limits in the NO_x emissions, and finally the proposed strategy is validated experimentally.

3. Insight into the effects of HP and LP EGR

The evolution of some of the most important parameters during engine operation in the NEDC are shown in figure 2. As far as the engine calibration has been kept constant with both EGR architectures, the evolution in the VGT is the same in both tests and it has important consequences in the intake pressure. In fact, the HPEGR architecture prevents the recirculated gas from being expanded in the turbine, while with the LPEGR loop all the exhaust gases flow though the turbine, and this increase in the turbine power leads to a higher compressor mass flow and higher intake pressure as shown in the upper plot of figure 2.

Again, as the engine control parameters have not been modified, the set points for the air mass flow with both EGR configurations are roughly the same. Since the engine speed evolution is imposed by the driving cycle, only small differences in the air mass flow set point may appear due to differences in the fuelling rate required by both systems. In this sense, the air mass flow with both EGR loops is similar as shown in the second plot of figure 2. To keep the actual air mass flow near the set point, the opening of the EGR valve is continuously modified along the cycle. However, at some parts of the cycle, the LPEGR shows higher air mass flows than demanded. At those conditions, even with fully open EGR valve, the air mass flow exceeds the set point. The reason for such a deviation from the set point is twofold. On the one hand, the mass admitted by the cylinders with the LPEGR is higher due to the higher intake density, product of the higher energy in the turbine, but also of the lower intake temperature. On the other hand, the limited pressure ratio in the LPEGR circuit prevents from reaching the high LPEGR flows necessary to reduce the air mass flow to the set point level.

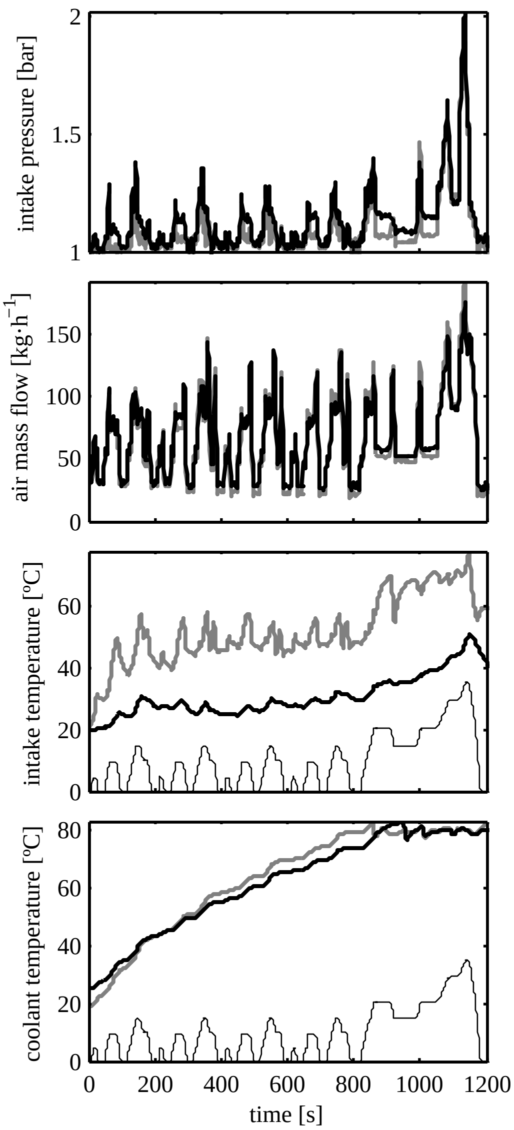

Figure 2. Evolution of the air mass flow, intake pressure, intake temperature and coolant temperature during the NEDC with LPEGR (black line) and HPEGR (grey line).

Intake temperature is also a parameter that plays a major role in the combustion process. The third plot in figure 2 shows an important reduction in the temperature of the intake gasses when the LPEGR architecture is used. The lower intake temperature and higher intake pressure leads to a noticeable increase in intake density, which involves an increase in the mass admitted by the engine cylinders. As far as the air mass flow is similar with both architectures, the increase in intake density involves a higher amount of recirculated gas in the case of using the LPEGR circuit. Changes in intake gas temperature, pressure and composition will lead to noticeable variations in the engine NO_x emissions and fuel consumption depending on the EGR architecture.

As a thermal engine, the ICE is strongly affected by temperatures, particularly, since the NEDC establishes the engine starting from some initial temperature between 20^{o}C and 30^{o}C, the ICE behaviour evolves during the NEDC as its temperature increases. The engine warmup impacts the fuel consumption due to changes in the heat transfer affecting the engine thermal efficiency and variations in the lubricant viscosity which affect friction losses. A faster engine warmup usually involves benefits from the point of view of fuel consumption, while its effects on emissions is not so easy to address. Generally, higher temperatures involve higher NO_x emissions, while other pollutants such as unburned hydrocarbons are usually reduced. The interested reader will find a deep analysis of the effects of engine temperatures on fuel consumption and emissions in [21, 22]. Regarding the effects of the EGR architecture on the engine warmup, figure 2 shows the evolution of the coolant temperature, as a representative temperature of the thermal state of the engine, along the test cycle with the addressed EGR loops. It is observed that despite a slightly lower temperature at the beginning of the cycle, the engine warm up is faster with the HPEGR, reaching the steady state temperature, namely 78^{o}C around 100 seconds faster than with the LPEGR. This is due to the fact that the gas recirculated with the HPEGR system, taken from the exhaust manifold and then with high temperature, contributes to the engine warm up by heating the coolant in the HPEGR heat exchanger and also by allowing a higher intake temperature that impacts the heat transfer in the combustion process. On the contrary, with the LPEGR, the recirculated gases are taken from the end of the exhaust line, specifically from the particulate filter outlet, with lower temperature and then with lower impact on the engine warm up.

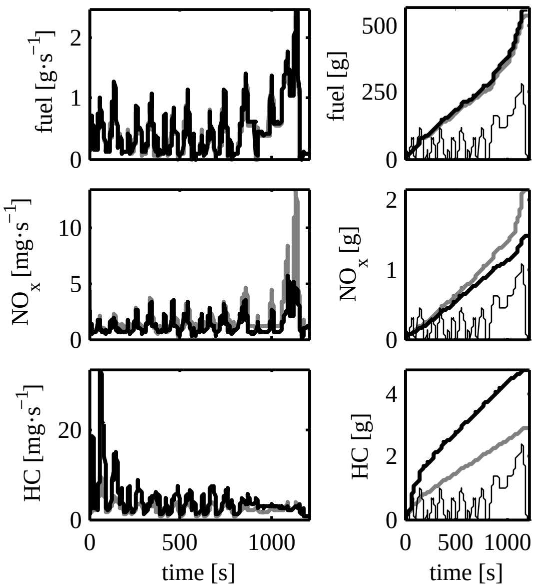

Figure 3. Evolution of the accumulated fuel consumption and NO_x emissions during the NEDC with LPEGR (black line) and HPEGR (grey line).

Differences in the variables discussed in the previous paragraphs, which are exclusively due to the EGR architecture employed, lead to important variations in the engine fuel consumption and emissions. Figure 3 shows the instantaneous and accumulated values for fuel consumption and NO_x emissions which are the main engine outputs considered in the present paper but also for the unburned hydrocarbon emissions (HC). Excepting some peaks during accelerations that can be attributed to measuring problems due to the poor dynamic response of the fuel measuring device (fuel balance), LPEGR test shows a higher fuel consumption. At the end of the complete cycle, the tests with LPEGR needs a 5.2% increase in fuel compared to the corresponding test with HPEGR. This increase in fuel consumption is due to two main causes:

- The higher mass flow through intake and exhaust systems leads to higher pumping losses [5]. It should be taken into account that the recirculated flow with LPEGR passes through elements such as the after-treatment system or the intercooler, however it is not the case with HPEGR. In addition, when HPEGR is used, communicating the intake and exhaust manifolds reduces the pressure difference and therefore pumping losses.
- The lower intake temperature and the higher amount of burnt gases recirculated both contribute to a later combustion, with lower temperatures but also with lower indicated efficiency. The effect of the EGR and the intake temperature on the engine efficiency are well-known and extensively addressed in the literature, particularly a detailed analysis of the effects of the intake temperature and composition in the combustion process may be found in [6, 10, 23, 24].

Regarding the last point, the higher EGR rates and lower intake temperatures reached with the LPEGR are also responsible for the NO_x reduction and the HC increase. Figure 3 shows lower NO_x emissions with the LPEGR during the whole test, independently of the operating conditions. As a consequence, at the end of the cycle a noticeable reduction of 30% in the NO_x emissions can be observed with the LPEGR architecture. On the contrary, the LPEGR produces an increase of 60% in the HC emissions respect to HPEGR. Figure 3 shows that this increase in HC emissions is produced mainly during the first part of the cycle, after the cold starting, which is consistent with the negative impact of low intake temperatures and high EGR rates on HC emissions [21]. For further discussion on the effects of the EGR layout on fuel consumption and emissions the reader is referred to [5, 13, 25].

From figure 3 it is clear that HPEGR will prevail in any EGR circuit switching strategy aimed to minimise fuel consumption, and that such strategy will impact positively HC emissions. Conversely, the weight of LPEGR strategies focused on the NO_x reduction are also apparent. The tradeoff obtained with both EGR architectures regarding fuel consumption and NO_x emissions show that a proper strategy to combine HP- and LPEGR systems is needed to obtain an equilibrium between pollutant emissions and fuel consumption. The next section is aimed to propose such a strategy.

4. Optimal HP and LP EGR sequence

Consider the previous tests, where the sequence of engine speed and torque is exactly the same in both cases. Given that both tests share the same engine calibration, the only difference between both tests is the EGR architecture used and the control parameters associated to the differences in fuel injection needed to follow the engine torque profile. Therefore, for this particular cycle, the fuel consumption (m_f) can be represented by the following linear system:

$$m_f\left(u\left(t\right),t\right) = a\left(t\right)u\left(t\right) + b\left(t\right) \tag{1}$$

where t represents the time spent since the engine start. a and b are time-varying parameters determining the system response and the input variable u is defined as:

$$u = \begin{cases} 1 & \text{then use LPEGR} \\ 0 & \text{then use HPEGR} \end{cases} \tag{2}$$

Note that the dependence of m_f on engine speed (n) and torque (M) is implicit since their evolution with time is predefined by the NEDC.

Regarding NO_x emission, a similar structure than that of equation 1 can be used, thus:

$$NO_x\left(u\left(t\right),t\right) = c\left(t\right)u\left(t\right) + d\left(t\right) \tag{3}$$

where parameters c and d define the time-varying linear response of NO_x to the EGR architecture used. Note that despite neither the fuel consumption nor the NO_x emissions show a linear response with the percentage of EGR done with HP- or LPEGR, as far as in the present paper only the extreme cases are considered (fully HPEGR or fully LPEGR) the linear approach becomes valid.

Regarding the parameters a, b, c and d, they can be obtained experimentally from the tests analysed in section 3 as:

$$\begin{aligned} a\left(t\right) &= m_f^{LPEGR}\left(t\right) - m_f^{HPEGR}\left(t\right) \\ b\left(t\right) &= m_f^{HPEGR}\left(t\right) \\ c\left(t\right) &= NO_x^{LPEGR}\left(t\right) - NO_x^{HPEGR}\left(t\right) \\ d\left(t\right) &= NO_x^{HPEGR}\left(t\right) \end{aligned} \tag{4}$$

where the superscript indicates the EGR loop employed.

Note the non-causality of the previous representation since the model provides the fuel consumption given the EGR architecture used and the torque and speed profiles (time evolution in the NEDC), while the physical process is exactly the opposite, *i.e.* given a certain amount of fuel injected, the engine will produce some torque and the balance between the engine torque and the road load will determine the resulting engine speed. However, since the model relies on experimental information on the particular cycle to study (NEDC), the physical causality can be inverted to simplify the model.

It should be also noticed that the model assumes quasi-steady behaviour [26, 27, 28, 29]. From a modelling perspective it can be stated that the proposed model has no states. Indeed, changes in the control input (the EGR architecture) involve progressive changes in engine variables during a transient whose duration can be not negligible, while the model proposed does not consider those transients. This simplification jeopardises the applicability of the proposed model as will be discussed later.

In any case, assuming the model suitability, the control problem may be stated as find the control policy during the NEDC that minimises the cost:

$$J = \int_0^{t_{end}} m_f\left(u\left(t\right), t\right) \mathrm{dt} \tag{5}$$

where n and M follow the trajectories predefined by the NEDC. The problem is constrained since there are restrictions concerning the maximum amount of pollutants emitted during the complete driving cycle. In the present paper, only NO_x emissions will be consider since there is not widespread after-treatment for NO_x while oxidation catalyst (DOC) and particle filters (DPF) are usually able to reduce the rest of pollutants up to regulation limits. In addition, strategies aimed to reduce the fuel consumption generally lead generally to a reduction of other pollutants such as HC. In this sense the constraint on pollutant emissions can be expressed as:

$$\int_0^{t_{end}} NO_x\left(u\left(t\right), t\right) \mathrm{dt} \leq NO_x^{lim} \tag{6}$$

where NO_x^{lim} represents the maximum allowed emissions of NO_x. In any case, constraints on the emissions of other pollutants may be considered just by adding the corresponding equations.

The problem presented by expressions (5) and (6) is in general difficult to solve due to the complex relation between the inputs (the decision variable u) and the outputs (the fuel consumption and NO_x emissions). However, as far as the model proposed neglects the system dynamics by assuming quasi-steady behaviour, the optimisation problem can be transformed into a static optimisation problem that can be addressed by the method of Lagrange multipliers. Then the integral problem represented by equations (5) and (6) can be replaced by a set of optimisation problems in which the following cost function (F) is to be minimised at each time step:

$$F_1(t,\lambda) = m_f(u(t),t) + \lambda NO_x(u(t),t) \tag{7}$$

The optimisation process consists in choosing, at any time step of the driving cycle, the control action u, *i.e.* the EGR loop, which minimises the cost function F_1. As only two discrete values are allowed for u, the problem is solved just considering the value providing the minimum cost at the considered time, which defines the optimal control policy $u^*(t,\lambda)$. From equation 7, it follows that the higher the value of λ, the higher the weight of the NO_x emissions on the cost function. Then, given the tradeoff between fuel consumption and NO_x, as λ increases the optimal NO_x emissions will decrease progressively at the expense of some fuel penalty. For that reason, the optimisation problem is reduced to find the value of λ which leads to:

$$\int_0^{t_{end}} NO_x(u^*(t,\lambda),t)\,\mathrm{dt} = NO_x^{lim} \tag{8}$$

Nevertheless, a cost function as simple as that proposed in (7) will result in a highly oscillating optimal control signal (u) that will not produce the desirable results when applying the control policy to the real engine. It should be recalled that, every time there is a switch between EGR configurations, the actual engine suffers a transient that the model is not able to take into account. In this sense, figure 4 shows the evolution of the key engine parameters during the switching from HPEGR to LPEGR at idle conditions. After a transient of 2.2 seconds the target air mass flow is reached. Note that the steady state air mass flow with both systems is exactly the same since the engine control establishes exactly the same air mass flow set point. Regarding the exhaust pressure, the steady state value is reached in less than 1 second. Closing the HPEGR valve has a direct impact on exhaust pressure while the effect on intake pressure is slower (3.4 seconds) due to the turbocharger dynamics. Regarding the temperature evolution in the intake manifold, it can be observed how slowly decreases due to the replacement of the hot HPEGR gas by LPEGR gas coming from the intercooler. On the contrary, the temperature at the compressor inlet smoothly increases due to the arrival of exhaust gases coming from the DPF outlet through the LPEGR circuit. It should be noted that the slow response of both temperatures is to a great extent due to the thermal inertia of the temperature sensors, in the case at hand k-type thermocouples, so the response time in the order of 20-30 seconds shown in the figure should exceed the characteristic time of the real process. Taking into account the evolution of intake and exhaust pressures in figure 4, it can be noticed that the pumping losses are increased with the use of the LPEGR, this increase in addition to the higher EGR rate involves a penalty in fuel consumption that can be observed in the right-upper part of figure 4. Note that there is not an appreciable delay in the increase in fuel consumption, in fact it is almost instantaneous after the LPEGR circuit is activated. Taking into account the engine displacement (2 litres) and the volume of the intake line (less than 8 litres) the gases from the LPEGR circuit will need around 4 engine cycles to arrive to the cylinders. Considering an engine speed of 750 rpm at idle, that means that in 2.7 ms the cooled gases from the LPEGR circuit will arrive to the cylinders involving an increase in the fuel needed to keep the engine speed and torque. Taking into account that the characteristic time of the fuel consumption response to a EGR switch is much faster than the characteristic time of the pedal evolution during the NEDC, a quasi-static behaviour for the fuel consumption can be considered. On the contrary, figure 4 shows that the response of the pollutant emissions to the EGR switching is not so fast. In addition, the response of the HC and NO_x emissions show important non-linearities such as minimum phase behaviour. Despite the slow response time of the exhaust gas analysers has an important impact on the time needed to achieve the steady state conditions after the EGR switch it should be admitted that the quasi-static hypothesis is far from the reality in the case of pollutant emissions. Consequently, the more switches between EGR systems, the stronger the impact of model uncertainties, especially emission models, on the optimisation.

To deal with this issue the following cost function is proposed:

$$F_2\left(t,\lambda_1,\lambda_2\right) = m_f\left(u\left(t\right)\right) + \lambda_1 NO_x\left(u\left(t\right)\right) + \lambda_2 abs\{\delta u\left(t\right)\} \tag{9}$$

where λ_2 is a second Lagrange multiplier that penalises the changes in the control signal (δu). Take into account that other restrictions, *e.g.* add limits on other pollutants, can be added by introducing new Lagrange multipliers.

With the cost function defined in expression (9) a penalty is associated to changes in the EGR configuration used, so optimal control policies minimising the cost function F_2 will take into account the number of switches between EGR architectures and the impact of the model uncertainties on the optimal solution due to the quasi-static approach will be reduced.

5. Results

Taking into account the model and the optimisation strategy described in the previous section, figure 5 shows the results obtained with λ_1 and λ_2 parameters ranging from 0 to 1.

In particular figure 5 shows the trade off between fuel consumption and NO_x obtained. The colour scale shows the number of switches along the cycle between EGR configurations, ranging from 0 in black to 165 in white. While the square and the circle show the experimental results obtained with the HPEGR and LPEGR architectures respectively, the solution can be moved along a near straight line linking both points with a relatively low number of switches between EGR architectures (dark points). However, the Pareto front (area where a reduction in NO_x involves unavoid-

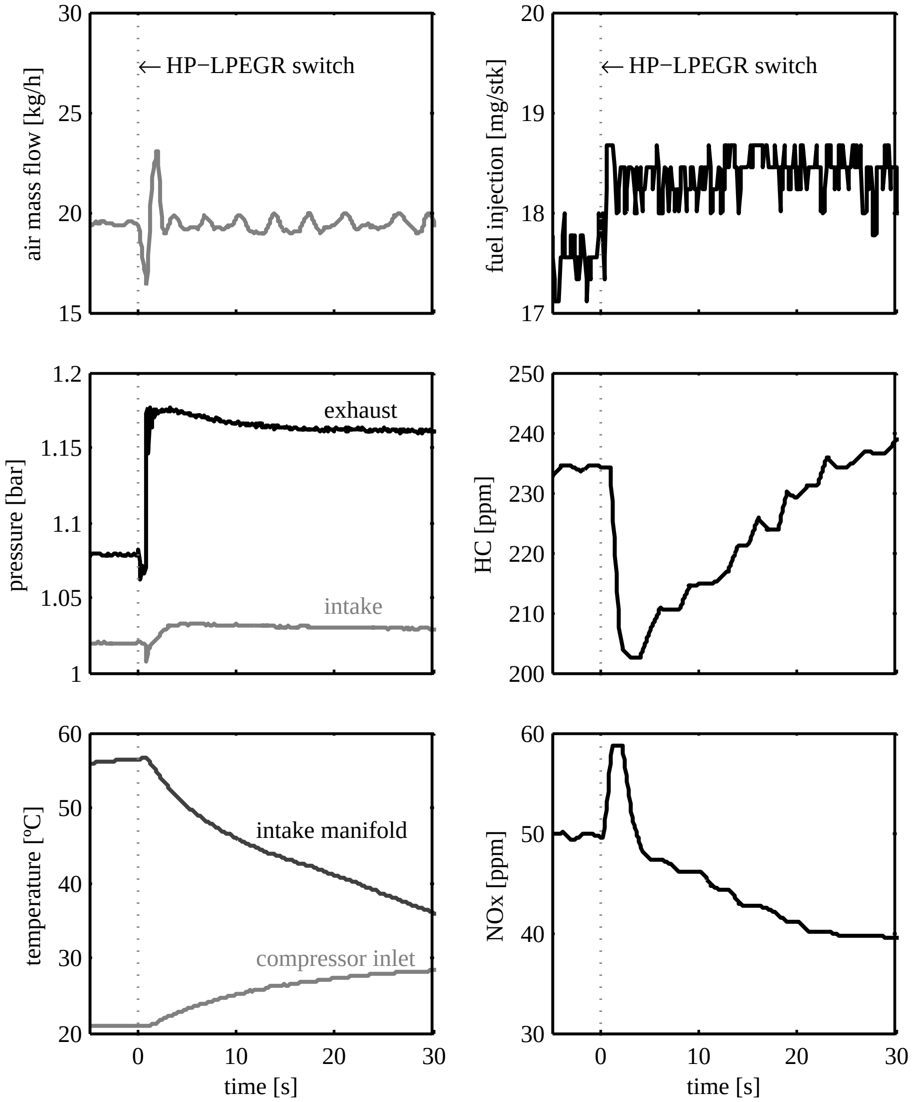

Figure 4. Evolution of the engine operating conditions (left) and engine performance (right) when switching from HPEGR to LPEGR at idle conditions.

ably an increase in fuel consumption) is defined by points with high number of switches between EGR configurations (light grey points).

There is also a quite clear stratification of the points in the NO_x fuel diagram according to the number of switches. Of course, that number of switches depend on the weight of δu on the cost function (9), and the higher the value of λ_2 the lower the number of changes in the EGR configuration.

To analyse in depth the effects of the Lagrange parameters on the optimal EGR sequence, figure 6 illustrates the obtained results for different values of λ_1 and λ_2. For a given row in figure 6, the evolution of the optimal EGR circuit to use during the NEDC, the total fuel consumed, the total NO_x emitted and the number of switches depending on the

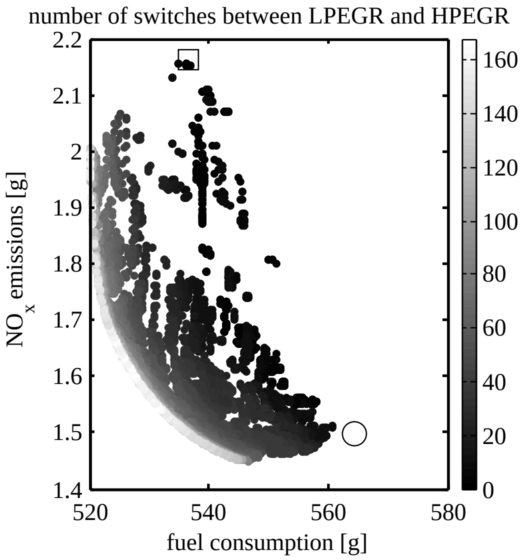

Figure 5. Tradeoff between fuel consumption and NO_x emissions calculated for $\lambda_1 \in [0, 1]$ and $\lambda_2 \in [0, 1]$. The colorscale represents the number of switches between HP and LPEGR during the cycle. The circle and the square show the experimental results obtained with LPEGR and HPEGR respectively.

parameter λ_1 are shown.

The results in the upper row are obtained with a λ_2=1, results in the lowest row are obtained with λ_2=0, while rows in between contain results for intermediate values of λ_2. Regarding the evolution of the optimal EGR architecture to use (left plots) the areas in grey represent LPEGR, while HPEGR is represented by areas in white. The following conclusions can be extracted from figure 6 analysis:

- For a given value of λ_2, increasing λ_1 the weight of NO_x emissions on the cost function F_2 rises, which involves a reduction in the NO_x emissions at the expense of an increase in the fuel consumption, which points out the trade-off between both parameters.
- As λ_1 increases, the optimal solution tends progressively to LPEGR because its lower NO_x emissions, however HPEGR prevails in solutions where fuel consumption is the prime objective.
- Optimal trajectories obtained with low λ_2 values show frequent changes between LPEGR and HPEGR architectures, which prevents the quasi-steady approach from providing good estimations of engine fuel consumption and NO_x emissions.
- The value of λ_1 has a negligible impact on the number of switches, that are almost exclusively affected by λ_2.
- As expected, increasing the value of λ_2 has a positive effect in the number of switches between EGR architectures. Nevertheless, increasing λ_2 involves a penalty in the theoretical minimum of fuel consumption and NO_x

emissions since the control policy is somehow constrained by the maximum number of switches.

- Solutions with affordable number of switches for the quasi-steady hypothesis ($\lambda_2 = 1$), show that the LPEGR architecture has higher potential at the last part of the cycle, while the HPEGR provides maximum benefits during the first phases of the NEDC. This is due to the fact that the benefits in fuel consumption of the HPEGR architecture are more important during the cold starting and the warm up, where the increase in temperature provided by the HPEGR contributes to a better combustion. On the contrary, in the last phase of the NEDC, where higher vehicle speeds are reached, the weight of NO_x emissions is more important as can be checked in figure 7. Then is in this last part of the cycle where the potential of LPEGR to reduce NO_x should be exploited. In addition, since the engine is warmed up, at these conditions, the penalty of LPEGR on fuel is not as important as during the cold start and warm up.

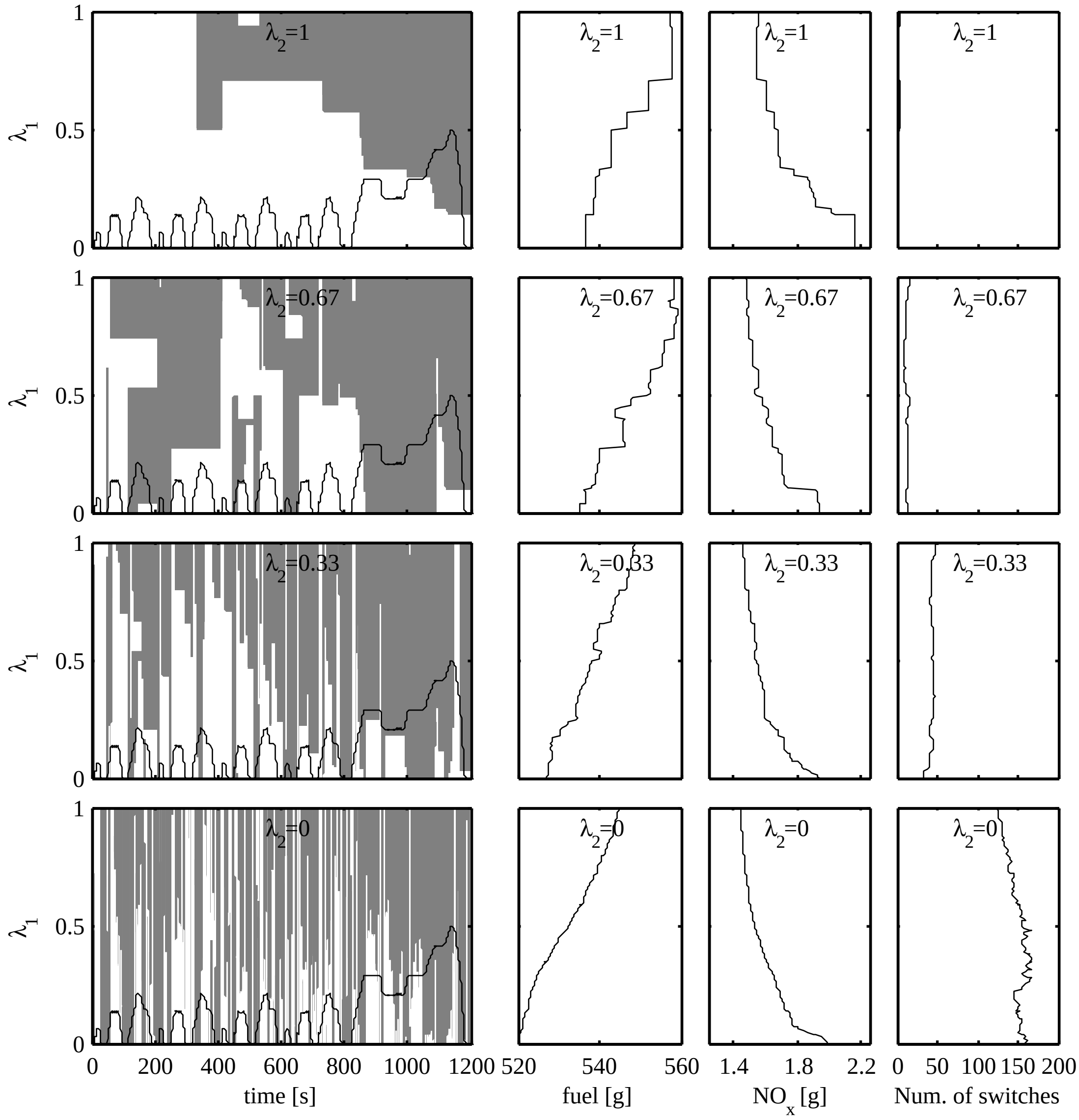

Figure 6. Evolution of the optimal EGR architecture employed, total fuel consumption, emissions and number of switches between HPEGR and LPEGR during the cycle depending on λ_1 and λ_2. Areas in grey represent LPEGR while HPEGR is shown in white

The last conclusion is probably the most interesting from the point of view of proposing a strategy to minimise the fuel consumption given a NO_x limit, since it can be deduced that the optimal control strategy would be to start with HPEGR and to keep this configuration up to a defined time where the LPEGR architecture should be used to meet the NO_x limit. In this sense, such a time is shown in the upper left plot of figure 6, where it is observed that the lower the NO_x limit, the earlier the LPEGR circuit should be used.

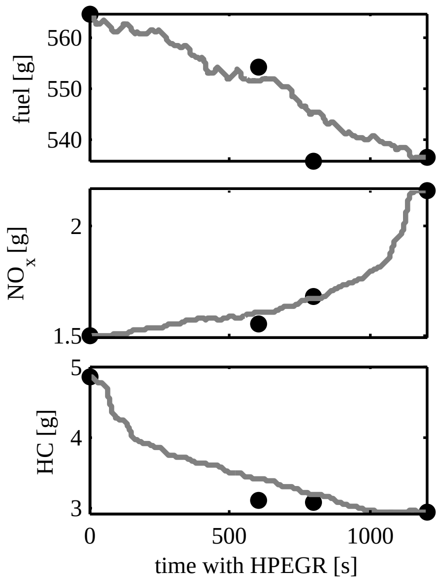

Figure 7. Final fuel consumption and NO_x emissions depending on the time spent with HPEGR. Black dots represent experimental measurements while the grey line shows the model results.

With the aim to validate the conclusions obtained with the optimisation, two additional tests have been carried out with different HP- LPEGR switching times, namely 600 s and 800 s. The obtained results in terms of fuel consumption, NO_x and HC emissions are compared with the optimisation results in figure 7. It can be observed a relatively good fitting between experimental and modelling results confirming that the optimal switching time depending on NO_x limitations obtained by modelling (grey line) can be used to calibrate the proposed control strategy. In addition, it is shown that optimising the fuel consumption also leads to minimise the HC emissions. Finally, the last conclusion of figure 7 is that the quasi-steady approach of the model is suitable for the optimisation when a small number of switches is allowed. Admittedly, from a calibration point of view is more convenient to define the switch between HPEGR and LPEGR in terms of coolant temperature instead of time; for the considered engine the 600 to 800 s interval corresponds to 70 to 79°C. Otherwise, in addition to regulation issues, unexpected results for different driving cycles than the NEDC can be obtained, *e.g.* in the case of cold conditions such as those reached in northern countries where 600 or 800 s may be insufficient to warm up the engine. At those conditions, a time based strategy for the EGR switching will lead to the use of LPEGR with a cold engine, then producing excessive HC emissions and fuel

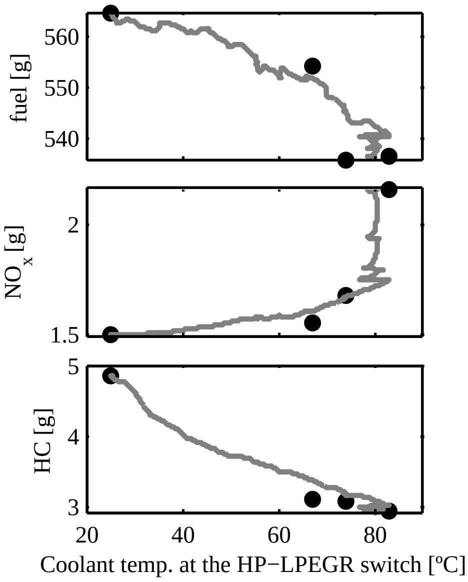

Figure 8. Final fuel consumption and NO_x emissions depending on the coolant temperature at the HP- LPEGR switch. Black dots represent experimental measurements while the grey line shows the model results.

consumption or even promoting missfiring. In this line, figure 8 shows the fuel consumption and NO_x emissions for different coolant temperatures at the HP- LPEGR switch.

6. Conclusions

The presented paper proposes the combination of HP- and LPEGR systems to minimise fuel consumption with reduced NO_x emissions. Particularly, a methodology to find the optimal switching strategy amongst EGR architectures during the NEDC has been developed. The proposed strategy is based on the optimal control theory so the optimal control policy depends on the definition of the cost function, which contains three main variables to minimise, namely the fuel consumption, the NO_x emissions and the number of switches between EGR architectures. Particularly, the number of switches between EGR architectures is a limiting factor that should be taken into account because of the quasi-steady hypothesis used to develop the optimisation model.

The analysis of the optimal control strategy shows that the optimal control policy to apply is to start with HPEGR and keep this configuration up to a defined time where the LPEGR architecture should be used to meet the NOx limit. This result is consistent with the fact that at cold conditions, the higher intake temperature and lower EGR rate produced with HPEGR involves noticeable benefits in terms of fuel consumption that are progressively diluted as the engine warms up, and on the other hand, at the end of the cycle, the NO_x emissions become more important, so the LPEGR becomes a suitable method to reduce NO_x emissions with a lower fuel consumption penalty.

The experimental results validate the strategy developed and show that combining both EGR systems sequentially along the NEDC allows to reduce noticeably the NO_x emissions of the HPEGR system with a reduced impact on the fuel consumption.

Finally, an additional contribution of the paper is that the optimisation strategy presented can be easily extended to take into account other restrictions (*e.g.* other pollutants).

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