

LP EGR and IGR Compromise on a GDI Engine at Middle Load

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ABSTRACT

Burned gas recirculation is emerging as a promising technology to reduce fuel consumption without compromising performance in turbocharged spark ignited engines. This recirculation can be done internally through Internal Gas Residual (IGR) using Variable Valve Timing (VVT) or externally through classical Exhaust Gas Recirculation circuit (EGR). Both have a large impact on combustion. The purpose of the paper is to give clues to get the best compromise at moderate load between these two technologies in terms of fuel consumption.

This experimental work was performed on a Gasoline Direct Injection (GDI) engine, 2.0L displacement, dual independent VVT, equipped with a Low Pressure, cooled and catalyzed EGR loop (LP EGR). The load region covers 6 to 10 bar Indicated Mean Effective Pressure (IMEP). EGR rates obtained vary between 0 and 15%. IGR variation is obtained by using the VVT in order to vary the valve overlap. IGR rates vary from 4 to 8%. They are estimated using the thermodynamic properties of intake and exhaust gases.

In-depth analysis of IGR and LP EGR effects is performed using an original method of split of losses. This method helps identifying what are the main causes of efficiency losses in the thermodynamic cycle. It has proven to be very useful to weigh pros and cons of IGR and LP EGR.

Results show firstly that LP EGR has a positive effect on Indicated Specific Fuel Consumption (ISFC) once knock is present, as it pushes back the knock limit, therefore improving the combustion phasing. For example at 2000 rev/min 10 bar IMEP, up to 4% gains on ISFC can be achieved with EGR (depending on VVT positions). Secondly, fuel consumption is lowered by 12 g/kW.h by optimizing VVT positions at 2000 rev/min 10 bar. This corresponds to the benefit of reducing the IGR rate from 6 to 4%. A further gain of 3 g/kW.h is obtained adding 10% EGR. Thirdly, the increase of IGR rate from 5 to 8% at 1500 rev/min and 8 bar IMEP increases the ISFC by 8%. This is mainly due to the negative effect of IGR on combustion phasing and duration. IGR increases the temperature at the end of compression stroke, therefore increasing the knock sensitivity.

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INTRODUCTION

For spark ignition engines, a good way to reduce consumption and also maintain the full load performance is to apply the downsizing concept, which mainly consists in reducing the engine displacement and adding a turbocharger [1, 2, 3]. Burned gas recirculation is emerging as a promising technology to address nowadays challenges. Indeed, it can provide benefits on indicated efficiency [4, 5, 6], or increase full load performance [7, 8] and limit pre-combustion occurrence [9], or is a mean to extend the Controlled Auto Ignition (CAI) region [10, 11], or else is a way to change the Particulate Matter (PM) distribution [12]. Practically, two types of recirculation can be used, the first one consists of a readmission of exhaust gases using a conventional EGR

circuit. Several EGR loop architectures are available for the GDI engine: HP or LP, cooled or not and even catalyzed or not [4, 13]. This study uses a LP EGR circuit, catalyzed and cooled at middle loads. A LP EGR circuit was chosen to also enable the use of EGR at low engine speed and high load when intake pressure is greater than exhaust pressure. EGR gases are cooled in order to increase the difference with IGR (which are hot gases) and to increase the temperature effect to decrease knock sensitivity. The EGR circuit is catalyzed in order to limit its fouling. The second recirculation type: IGR, consists in trapping a fraction of the burnt gases in the combustion chamber. Dual independent VVT are used to optimize the gas transfer. There is no unique optimal position, it has to be adjusted with the engine speed and load.

Here, VVT is used to vary the valve overlap in order to obtain different IGR rates.

The dilution of fresh air with EGR implies an increase of specific heat Capacity (C_p), which advantageously reduces in-cylinder temperature and then decreases knock sensitivity. Fuel enrichment at high loads can also be discarded. EGR, especially at low load improves the pumping losses due to decreasing throttling need [4, 5, 14, 15]. However, this effect competes with a longer combustion duration. Concerning the dilution with IGR, competitive effects can also be observed between dilution and thermal load.

The aim of this study is to understand, at mid-load, how EGR and IGR influence the indicated efficiency.

EXPERIMENTAL SET-UP

Engine Configuration

This study was performed on a GDI engine, four cylinders, 2.0L displacement equipped with LP, catalyzed and cooled EGR and dual independent VVT. Main characteristics of the engine are given in Table 1. Figure 1 gives an overview of the EGR loop installed at the test bench. A Three-Way Catalyst (TWC) system is placed downstream the turbine. It enables to avoid HydroCarbon (HC) condensation in the coolers (EGR and intercooler). The EGR cooler is tilted in order to make the condensed water flow in the exhaust line and not in the EGR circuit. As TWC is potentially debasing in long time operation of the engine, metallic parts of the TWC can be present in the exhaust line. If they go into the EGR circuit, this may cause serious damages to the compressor blades. A dedicated attention was paid to the deterioration of the compressor blades. After

operating the engine for about 100 hours with cooled external EGR flowing through the compressor at an average EGR rate of around 5%, no significant damage was found on the compressor. To control IGR rate, the engine is equipped with dual independent VVT: one at the intake and one at the exhaust. During this study, both the intake and exhaust cam profiles remained unchanged (i.e. lift amplitude). Both intake and exhaust valve duration are 190°CA at 0.7 mm lift (the 0.7 mm lift corresponds to the engine manufacturer reference lift). Each one of the cam phaser can be controlled independently.

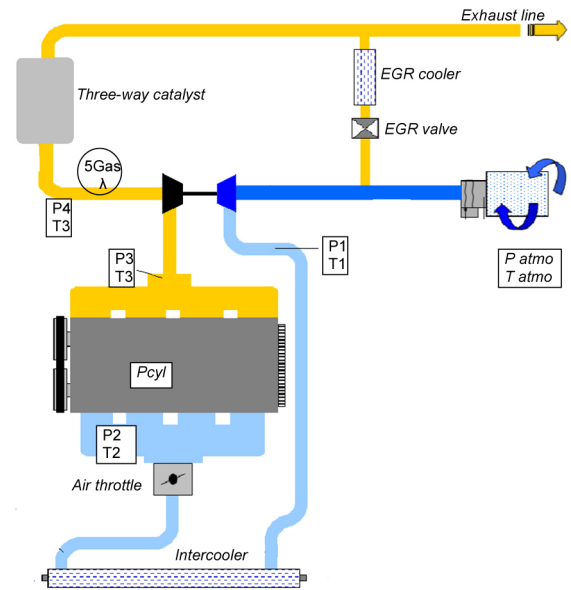


Figure 1. Test facilities: air loop and EGR loop

Table 1. Engine parameters and settings

Engine Type	L-4
Displaced volume	1998 cc
Bore	82.7 mm
Stroke	93 mm
Connecting Rod	144 mm
Compression ratio	10.7:1
Number of Valves	16
Exhaust Valve Opening	From 39° to -1° BTDC
Exhaust Valve Closing	From -29 to 11° BTDC
Inlet Valve Opening	From 30° to -10° BTDC
Inlet Valve Closing	From -20 to 20° BTDC
Fuel	97.5 RON with 5% ethanol
Fuel pressure	From 50 to 120 bar depending on the operating point
Injector	Centrally mounted Siemens DKDI XL1 Swirl
Boosting system	HTT C224 – T231, single scroll
Water / Oil temperature	90°C

Test Bed Configuration

The engine configuration is described in Figure 1. A conventional gas analyzer is used at the exhaust. Several thermocouples and pressure sensors are positioned as shown in Figure 1. Intake (P2), in-cylinder (Pcyl) and exhaust (P3) are rapid pressure sensors.

The original Engine Control Unit (ECU) is replaced by a D2T rapid prototyping system. In addition to the on-line adiabatic combustion analysis, the high frequency acquisition data are saved and used for an off-line combustion analysis which considers wall heat exchanges. This tool was internally developed and provides Heat Release Rate (HRR), Burnt Mass Fraction (BMF) and gas temperature. Wall heat losses are estimated using a Woschni model.

TEST METHODOLOGY

All tests are performed under steady state operation with a warm engine, i.e. 90°C for engine water and oil. Figure 2 illustrates the valve lifts for the earliest and the latest Exhaust Valve Closing (EVC) and Inlet Valve Opening (IVO) values. Maximum dephasing amplitude for both VVT is 40°C. The maximum valve overlap corresponds to a late EVC (11°C at 0.7mm lift) and an early IVO (30°C at 0.7mm lift) i.e. a 41°C valve overlap. Higher valve overlap value cannot be reached otherwise the valves would impact the piston¹. The maximum negative overlap corresponds to an early EVC (-29°C) and a late IVO (-10°C) i.e. a -39°C valve overlap. Negative valve overlap means that the exhaust valves are closed before the intake valves open.

Spark advance is controlled to phase the combustion such that 50% Burnt Mass Fraction (BMF) is at 7°C ATDC at low load or at the knock limit at higher loads. The knock limit is determined online, by two complementary ways. First, by listening the combustion noise and the characteristic metallic sound of knock. Second, by the in-cylinder pressure monitoring to detect high frequencies.

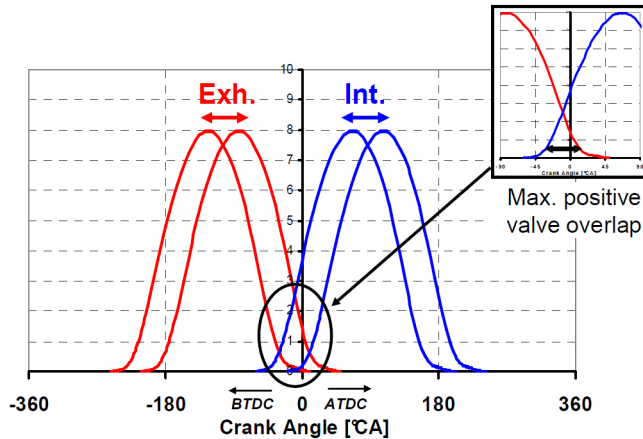


Figure 2. Extreme valve lift variations performed by VVTs

ANALYSIS TOOLS

IGR Calculation

Different methods can lead to an estimation of IGR [16, 17]. The present study focuses on an off-line IGR estimation method [16]. The mass fraction of residual burnt gas (IGR) at IVC: Y_{IGR}^{IVC} is computed using an iterative procedure which stops when convergence is reached. First, in case of a negative valve overlap, the residual gas temperature at IVO T_{IGR}^{IVO} is calculated assuming an adiabatic expansion of the

residual burnt gases at intake valve opening as shown in equation (1):

$$\frac{T_{IGR}^{IVO}}{T_{IGR}^{EVC}} = \frac{p^{EVC}}{p^{IVO}} \frac{Y}{Y-1} \quad (1)$$

p^{IVO} , p^{EVC} and T_{IGR}^{EVC} are provided by the engine bench and are assumed to be equal to the intake pressure at IVO, exhaust pressure at EVC and exhaust mean temperature respectively. In case of a positive valve overlap, the residual gas temperature at IVO is taken equal to T_{IGR}^{EVC} , the exhaust mean temperature. T^{IVC} is then computed following equation (2):

$$h(T^{IVC}, Y_i^{IVC}) = Y_{IGR}^{IVC^{n-1}} * h(T_{IGR}^{IVO}, Y_i^{IVC^{n-1}}) + \dots + (1 - Y_{IGR}^{IVC^{n-1}}) * h(T_{intake}^{IVO}, Y_i^{intake}) \quad (2)$$

where h is the sensible enthalpy, $Y_{IGR}^{IVC^{n-1}}$ is the residual gas mass fraction at iteration $n - 1$ and subscript i denotes either air, fuel or burnt gases. The intake temperature T_{intake} and composition Y_i^{intake} are given by experiments, while compositions at IVC Y_i^{IVC} and at IVO Y_i^{IVO} are calculated using the intake composition and residuals mass fraction at iteration $n - 1$. The enclosed mass at IVC m_{IVC} is calculated with equation (3):

$$m_{IVC} = \frac{p^{IVC} * V^{IVC}}{r^{IVC} * T^{IVC}} \quad (3)$$

where V^{IVC} is the cylinder volume at IVC and where the mixture constant at IVC r^{IVC} is estimated from the composition at IVC at iteration $n - 1$ through the equation (4):

$$r^{IVC} = Y_{air}^{IVC} * r_{air} + Y_{fuel}^{IVC} * r_{fuel} + Y_{IGR}^{IVC} * r_{IGR} \quad (4)$$

Finally, the residual gas mass fraction at iteration n , $Y_{IGR}^{IVC^n}$, is given by the perfect gas law at EVC as written in equation (5):

$$Y_{IGR}^{IVC^n} * m_{IVC} = \frac{p^{EVC} * V^{EVC}}{r_{IGR} * T^{EVC}} \quad (5)$$

where V^{EVC} is the cylinder volume at EVC and r_{IGR} is estimated using the residual composition at iteration $n - 1$.

¹Moreover, later IVC (i.e. the use of Atkinson cycle) are not studied here.

Refined Analysis of Engine Thermal Efficiency : Internal Combustion Engine Thermal Efficiency Analysis (ICETEA)

Improvement of engine efficiency is a top priority. It requires the ability to precisely analyze the efficiency of the thermodynamic cycle in order to be able to compare different operating points and/or different engines. The goal is to understand and to quantify the main elements that influence the indicated efficiency in order to find ways of improvement. As already describe in [18], ICETEA is the IFP Energies nouvelles' (IFPEN) tool that performs such an analysis inspired on the method of the split of losses [19]. The method is briefly explained in this section.

The Beau de Rochas cycle is, by definition, the cycle with maximum efficiency. The real thermodynamic cycle can be derived from the ideal Beau de Rochas cycle through the addition of several type of losses (wall heat transfers, pumping losses,...). Each type of loss represents a decrease in efficiency. The ideal cycle efficiency might then be expressed as the real cycle efficiency plus the sum of all these losses: equation (6).

$$\eta_{ideal} = \eta_{real} + \sum_i loss(i) \quad (6)$$

The starting point of the analysis is the real cycle and its indicated efficiency. The indicated efficiency is calculated from the Heat Release Rate (HRR). The combustion analysis that provides the HRR of the real operating point also estimates the coefficient for wall heat losses (coefficient C2 from the Woschni model) so that wall heat losses might be computed. The energy balance of the real cycle is an output of this tool, i.e. the division of the energy between indicated work, heat transfers, pumping losses, unburned and exhaust gases. Once the real cycle has been modeled, ICETEA works step by step, each step representing a type of loss in order to progressively approach the ideal cycle. At each step, it is possible to compute a new thermodynamic cycle that includes all previous hypotheses so that the indicated efficiency and the energy balance can be calculated. Once all the losses have been removed from the real cycle, the ideal (Beau de Rochas) cycle is rebuilt. Based on the work of [19], 7 steps representing various types of losses are defined (cf. Figure 3):

1. Pumping losses: these losses are the result of the intake and exhaust processes. To remove these losses, only the High Pressure (HP) part of the cycle is computed, i.e. from the start of the compression stroke at the Bottom Dead Center (BDC) to the end of the expansion stroke at the next BDC.
2. Wall heat losses: these losses represent the energy lost to the walls. This effect is eliminated by computing an adiabatic cycle.
3. Real working gas: the thermodynamic properties of the mixture are not constant during the cycle (they depend on the

temperature and the composition of the mixture). This loss is avoided by the computation of a cycle in which the properties of the mixture depend only of the composition at the reference temperature (293 K).

4. Combustion phasing: for several reasons, the combustion phasing may not be optimum. The best phasing of the combustion is computed by translating the HRR law.

5. Combustion velocity: real combustion is not instantaneous. Combustion duration is defined as the difference between 90% BMF and 10% BMF. These losses are removed by the computation of a cycle in which the combustion is instantaneous at Top Dead Center (TDC).

6. Incomplete combustion (also called "combustion efficiency"): this includes all losses related to a final burnt mass fraction lower than 1. In the corresponding ideal cycle, all the fuel is burnt.

7. Real mixture: the working gas is a mixture of air, fuel and burned gases. This loss is removed by computing a cycle for air only. This cycle corresponds to the Beau de Rochas cycle.

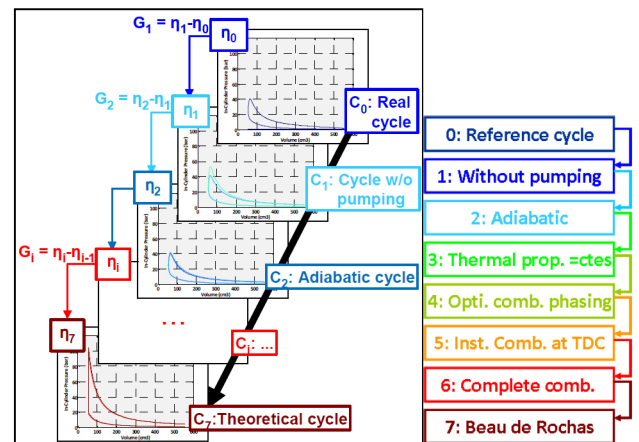


Figure 3. List of the steps of the split of losses method

The limit of this approach is that the order in which these losses are computed may have an impact on their value. However, the global analysis would not be modified.

The two main outputs of this tool are:

1. The energy balance of each cycle, i.e the division of the energy between indicated work, heat transfers, pumping, unburned and exhaust gases;
2. The evolution of the indicated efficiency from step to step.

The analysis of this form gives information on the main sources of losses of the combustion process. In the present study, this tool is used to analyze in detail where the gain in the indicated efficiency can be found when comparing