

A Combination of Swirl Ratio and Injection Strategy to Increase Engine Efficiency

Pablo Olmeda, Jaime Martin, Antonio Garcia, and David Villalta
 Universitat Politecnica de Valencia

Alok Warey and Vicent Domenech
 General Motors Global R & D

ABSTRACT

Growing awareness about CO₂ emissions and their environmental implications are leading to an increase in the importance of thermal efficiency as criteria to design internal combustion engines (ICE). Heat transfer to the combustion chamber walls contributes to a decrease in the indicated efficiency. A strategy explored in this study to mitigate this efficiency loss is to promote low swirl conditions in the combustion chamber by using low swirl ratios. A decrease in swirl ratio leads to a reduction in heat transfer, but unfortunately, it can also lead to worsening of combustion development and a decrease in the gross indicated efficiency. Moreover, pumping work plays also an important role due to the effect of reduced intake restriction to generate the swirl motion. Current research evaluates the effect of a dedicated injection strategy to enhance combustion process when low swirl is used. For this purpose, a combination of theoretical (0D and 1D models) and experimental tools were used. In particular, experiments were conducted in a single-cylinder direct-injection light-duty diesel engine. The analysis also included theoretical calculations to estimate pumping losses. Results show that an increase in swirl ratio leads to an increase in the gross indicated efficiency (balancing heat transfer losses and combustion improvement) but the higher pumping losses negate this positive benefit. In the lowest swirl ratio case, a suitable injection strategy based on multiple injections, together with an increase in the injection pressure, can provide similar gross indicated efficiency as high swirl ratio case while avoiding high pumping losses.

CITATION: Olmeda, P., Martin, J., Garcia, A., Villalta, D. et al., "A Combination of Swirl Ratio and Injection Strategy to Increase Engine Efficiency," *SAE Int. J. Engines* 10(3):2017, doi:10.4271/2017-01-0722.

INTRODUCTION

Reduction of GHG emissions has become a major concern for the researchers and manufacturers on the automotive industry [1]. CO₂ emissions, as the main contributor to greenhouse effect, can be mitigated by diminishing fuel consumption. Thus, currently, different strategies are proposed to get this goal; thermal management improvement [2,3], indicated cycle optimization [4,5], in-cylinder heat transfer (HT) reduction [6,7,8], engine friction and auxiliaries losses reduction [9,10] or new combustion modes [11,12,13] among others. In the present work, the research effort has been focused on obtaining an engine setting combination, which leads to a heat transfer reduction with a combustion process improvement and therefore a better engine thermal efficiency.

In this framework, it is well-known that heat transfer to the combustion chamber walls contributes to decrease the indicated efficiency. One of the most investigated strategies of in-cylinder gas flow affecting heat transfer is swirl ratio variation. In this sense, it has been showed in the literature [14] that to promote low swirl conditions in the combustion chamber by using low or negligible swirl ratios mitigates the indicated efficiency loss. Indeed, low swirl ratio implies lower gas velocity and hence a decrease in the heat

transfer coefficient. [14]. In spite of its benefits, to decrease the in-cylinder gas motion also implies an important drawback. A weak air motion in the chamber worsens the air-fuel mixing process and promotes slower burning rates [15]. Therefore, low swirl conducts to a contradictory trade-off scenario where it benefits the indicated efficiency by means of a decrease in HT and worsens it since a poor combustion process is developed. In addition, as it is well known [16], the injection pattern governs the combustion process. Higher injection pressures produce an increase in the total air entrained into the fuel, which results in an improvement in the mixing process. A higher momentum of the spray due to an increase in injection pressure produces an increase in the movement of in-cylinder air. In the same way, post injection is demonstrated as a key strategy to influence mixing controlled combustion enhancing fuel oxidation and reducing soot emissions by adding extra momentum flux during this period [17].

In addition, it is necessary to mention that swirl variation has not only impact on indicated efficiency but also on brake efficiency through pumping work. Normally, a throttle plate upstream of the ports provides variable swirl [18]. Thus, when this plate is closed and the closed, throttles the airflow modifying the flow streams high swirl level

is obtained, by contrast when the plate is fully opened the lowest swirl ratio is achieved. This plate has a considerable impact on the air management process since it produces a drop pressure in the port if high swirl level is desired. This fact implies a higher boost pressure to get the same trapped mass at IVC compared with a low swirl case, therefore the turbine section must be reduced to reach the required intake pressure and consequently higher backpressure is also attained.

Considering above explanations, low swirl implies a trade-off between some important advantages (improvement on heat transfer and pumping work) and combustion process worsening. Thus, the main objective of the current research is to evaluate the effect of a dedicated injection strategy to enhance combustion process when low swirl is used with the aim of getting a suitable combination of low heat transfer and combustion process improvement

For this purpose, a combination of theoretical (0D and 1D models) and experimental tools were used. In particular, some preliminary experiments were performed in a dedicated cold test rig to characterize swirl of a single-cylinder direct-injection light-duty diesel engine. Once this work was made, swirl ratio sweeps were conducted in the single-cylinder engine. The engine efficiency analysis also included theoretical calculations to estimate pumping losses.

EXPERIMENTAL FACILITIES

Steady Cold Flow Test Rig

A cold flow test rig was used to characterize the flow in the cylinder head. The main objective was to determine experimentally how the swirl coefficient changed when the helical and tangential valves are varied. In [figure 1](#), it is shown both valves together with their corresponding pins to vary their position. Manually actuation of these valves was used. Originally, in the multi-cylinder engine base, the swirl ratio was only modified with a helical valve. The swirl ratio range was enclosed from 1.4 to 3. For thus, in the single-cylinder engine used in the current study, the tangential valve was added. This air management system modification allows to increase the range in terms of swirl ratio compared to the engine base design, from 0 to 5.2.

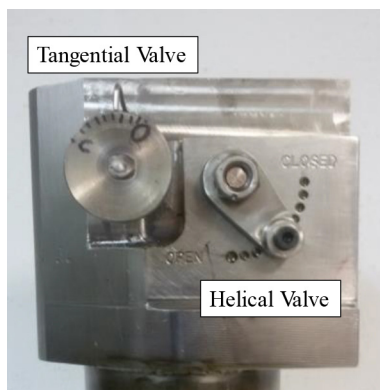


Figure 1. Picture of the helical and tangential dedicated valves to vary the swirl coefficient.

A schematic of the test rig is shown in [figure 2](#). The flow test rig consists of two different tanks, one with 200 l capacity and the other with 80 l, a compressor and an anemometer. It is equipped with a

system that can generate pulsating flow by using the main elements of the engine valve timing system. The impulse swirl meter uses a rotating honeycomb type matrix, which is restrained from rotation by a strain-gauged load cell to totally arrest the angular swirl component thereby measuring the resultant angular impulse as a torque. The aspiration tests were performed with this flow rig, for simulating the real flow operating conditions. The raw data were processed using the methodology specified by the manufacturer. The mass flow rate range is 0-0.23 kg/s, the maximum rated pressure is 1.8 bar and the maximum Mach number that can be measured is 0.18.

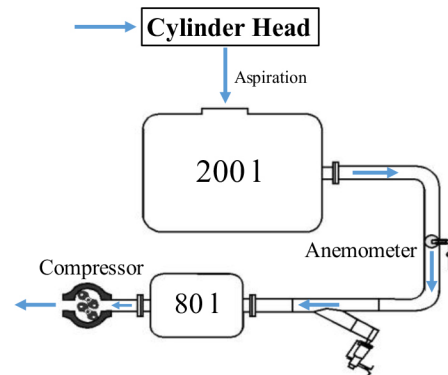


Figure 2. Schematic of the cold flow test rig.

Single-Cylinder Light-Duty Diesel Engine

The study was mainly carried out on a single-cylinder DI Diesel engine. It is derived from a production GM 1.9L Diesel engine equipped with a Common-Rail fuel injection system. The engine has four valves per cylinder, centrally located injector, and a re-entrant type combustion chamber. For the purposes of this study, the engine was equipped with dedicated swirl flaps that can modify the swirl number from 0 up to almost 5.

Engine data and injection system characteristics are depicted in [table 1](#). This engine was set up to meet EURO IV emissions regulations.

Table 1. Engine and injection system specifications

Engine Type	DI, 1-cylinder, 4-stroke
Displaced volume [mm ³]	477
Stroke [mm]	90.4
Bore [mm]	82
Combustion Chamber	Re-entrant type
Compression ratio	17.1:1
Max. Power [kW]	27.5 @ 4000rpm
Max. Torque [Nm/min ⁻¹]	80 / 2000-2750
Injection System	Bosch Common Rail (solenoid)
Max. Rail Pressure [bar]	1600
Nozzle hole diameter [mm]	0.141
Injector Nozzle Holes	7
Hydraulic flow rate [cm ³ / (30s) at 100 bar]	440

Test Cell

As figure 3 shows, the single-cylinder engine is installed in a fully instrumented test cell, with all the auxiliary facilities required for operation and control.

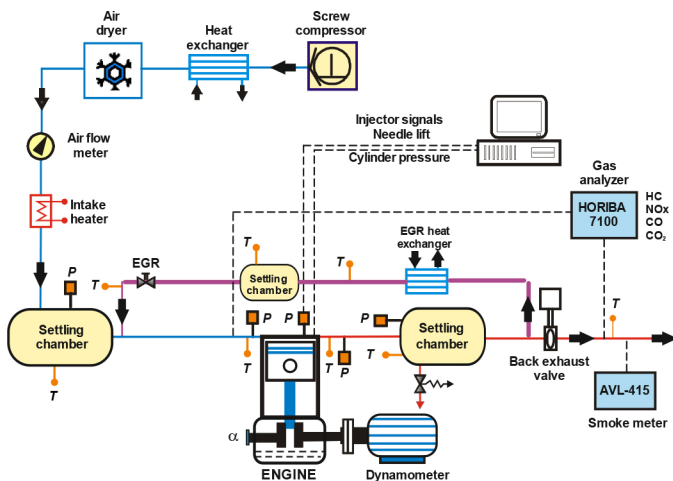


Figure 3. Complete test cell set-up.

To achieve stable intake airflow conditions, an externally driven screw compressor supplied the required boost pressure before passing through an air dryer. The compressor can develop up to 3 bar inlet pressure. The air pressure was adjusted in the intake settling chamber, while intake temperature was controlled in the intake manifold after mixing with EGR. The exhaust backpressure produced by the turbine in the real engine was replicated by means of a valve placed in the exhaust system, controlling the pressure in the exhaust settling chamber. Then, the exact EGR rate was controlled by means of a valve between the EGR settling chamber and the intake pipe, so the required exhaust gas mass flow was introduced into the intake runner depending on the desired EGR rate. The temperature regulation was performed upon the EGR-fresh air mixture using a temperature sensor in the intake manifold. The EGR coolant circuit and the oil coolant circuit were separated from the engine coolant circuit. In this way, the heat rejection to oil, coolant, intercooler and EGR heat exchangers could be analyzed independently. The intercooler was cooled by water unlike the original engine in which an air-water cooler system is used.

The exhaust gases were analyzed with a five gas Horiba MEXA 7100D analyzer bench. In order to increase the robustness of these measurements, the different pollutant volume fractions were sampled and averaged over a 60 second time period after attaining steady state operation.

Smoke emissions were measured with an AVL 415 variable sampling smoke meter, providing results directly in FSN. The FSN values used in this research are the average of three consecutive measurements at the same operating condition. These measurements were transformed into mg/m^3 by means of the correlation (1) proposed in the user manual of the device:

$$[\text{mg}/\text{m}^3] = \frac{1}{0.405} \cdot 4.95 \cdot \text{FSN} \cdot e^{0.38 \cdot \text{FSN}} \quad (1)$$

The installation also included complete instrumentation to measure different fluid temperatures and mass flows. A combination of thermocouples and resistance temperature detectors (PT100) were selected (taking into account the expected temperature variation) to measure liquid and gas temperatures. The in-cylinder pressure was measured with a Kistler 6125C10 glow-plug piezoelectric transducer and Kistler 4603B10 charge amplifier. A crank angle increment of 0.5° was used for the in-cylinder pressure acquisition [19], which was performed using DRIVEN [20]. The list of the relevant instrumentation is shown in Table 2.

The mean variables were acquired at a low sample frequency of 100 Hz using SAMARUC, a CMT-developed test system that collects the signals of different sensors and controls the dynamometer [21].

Table 2. Test cell instrumentation.

Variables Measured	Sensors	Range
Temperature of liquids (coolant, oil, cooling water at exchangers...)	Thermoresistors (PT100)	-30 - 350 °C
Temperature of gases (inlet and exhaust lines, EGR...)	K-type Thermocouples	-200 - 1250 °C
Fuel mass flow	AVL 7351	0-34.72 g/s
Air mass flow	Sensyflow P	10-400 kg/h
Coolant Flow	DN25 Flow meter	8.8-350 l/min
Torque	Dynamometer	0-400 Nm
In-cylinder pressure	Kistler 6125C10	0-300 bar.

THEORETICAL TOOLS

0-D Models

Two different 0-D single-zone thermodynamic models (CALMEC and siCiclo) were used. Main assumptions for both are the same:

- Chamber pressure and temperature are assumed to be spatially uniform. In combustion calculations, both fluid and flame velocities are lower than the speed of sound.
- Three species (air, fuel vapor and stoichiometric combustion products) are considered. The flame is located in the stoichiometric fuel-air ratio region of conventional diesel combustion and thus this hypothesis is assumed [22].
- Ideal gas law is used to calculate mean gas temperature. Perfect gas behavior is assumed.
- A filling and emptying model is used to calculate the trapped mass [23]. A simple filling and emptying model has been used to estimate the residual mass and the short-circuit mass. When they are known, the trapped mass at IVC has been calculated using next expression.

$$m_{\text{IVC}} = m_a + m_{\text{EGR}} + m_{\text{res}} - m_{\text{sc}}$$

where m_a and m_{EGR} are the air and EGR masses, m_{res} is the residual mass of the previous cycle and m_{sc} is the short-circuit mass.

- Specific heat of the gas depends on both temperature and composition. This assumption is consistent with the second and third hypothesis [24].
- Blow-by model is based on the evolution of the gas in an isentropic nozzle [23]. The instantaneous mass flow of blow-by to the crankcase is estimated by means of the equation of an adiabatic nozzle as:

$$m_{bb} = c_{bb} A_{ref} p \sqrt{\frac{x}{R_c T}}$$

where $x = \frac{2\gamma}{(\gamma-1)} \left[\left(\frac{p_{crk}}{p} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{crk}}{p} \right)^{\frac{\gamma+1}{\gamma}} \right]$. The discharge coefficient

of the nozzle, c_{bb} , was adjusted with experimental measurements so that the cumulative blow-by coincided with the actual flow, A_{ref} is the reference section ($3.5 \times 10^{-6} D$), p_{crk} is the crankcase pressure and R_c and T are compression ratio and temperature, respectively.

- Chamber volume deformation is calculated by means of a simple deformation model [25]. The instantaneous volume is obtained as the addition of three terms: the combustion chamber volume, the volume displaced by the piston (depending on crank angle) and the mechanical deformations produced by the gas pressure and the inertial efforts of the masses with reciprocating motion. A simple deformation model, described in [22], is used to calculate the last term.
- Heat transfer to the chamber walls is calculated with a modified Woschni-like model [26].

$$h = C D^{-0.2} p^{0.8} T^{-0.53} \left[C_{w1} c_m + C_{w2} c_u + C_2 \frac{V_d T_{IVC}}{V_{IVC} p_{IVC}} (p - p_0) \right]^{0.8}$$

where C and C_2 are constants whose values are 0.12 and 0.001, c_m is the mean piston speed, c_u is the instantaneous tangential velocity of the gas in the chamber that was adjusted using CFD calculations, p_0 is the pressure during motoring conditions assuming a polytropic evolution, and finally C_{w1} and C_{w2} are constants, whose values are adjusted for each engine by means of a combination of experimental and modelling methodology.

The combustion analysis was performed by means of an in-house tool named CALMEC, which is comprehensively described in [27]. This thermodynamic tool uses the in-cylinder pressure signal as the main input, which is measured from 100 consecutive engine cycles in order to reduce the cycle-to-cycle experimental uncertainties. The raw pressure signal was smoothed by the a low-pass filter and averaged in order to obtained a representative in-cylinder pressure trace, which was used to perform the analysis. Then the first law of thermodynamics was applied between IVC (Intake Valve Close) and EVO (Exhaust Valve Open), considering the combustion chamber as an open system because of blow-by and fuel injection. The main result of the thermodynamic tool used was the rate of heat release (RoHR) along with some representative parameters such as the start of combustion (defined as the crank angle degree in which the

cumulated heat release reaches 2% of the total fuel energy) and the CA50 (defined as the crank angle at 50% mass burned, among other parameters).

Siciclo [22] is a predictive thermodynamic model, in which the mass and energy conservation equations are solved in order to obtain the instantaneous gas state in the combustion chamber. This tool calculates the in-cylinder pressure trace using the experimental RoHR as main input. The model includes some specific sub-models to reproduce the blow-by leakage, the chamber deformation due to pressure and inertial efforts, the heat transfer to the chamber walls and the fuel injection. This code was used in this research to predict engine performance and fuel consumption and also to obtain boundary conditions.

1-D Model

A 1-D model developed in the GT-Suite commercial software was used to calculate pumping work when the swirl ratio is varied in order to impose them in the single-cylinder tests. The model was adjusted with multi-cylinder engine tests previously measured in [6]. In this sense, instantaneous measured and modeled in-cylinder pressure, intake and exhaust pressures, along with air mass flow measurement were compared to validate the model performance at four different operating points with four different swirl ratio levels. Inputs to the 1-D model were experimental wall temperatures, injection strategy settings and fuel mass, as well as RoHR obtained with CALMEC. Experimental gas temperatures, pressures, and air mass flow were used as targets during model tuning.

METHODOLOGY

Engine Operating Conditions and Testing Procedure

The engine was operated at two different engine loads maintaining constant engine speed at 2000 rpm. In particular, low load (5 bar BMEP) and high load (20 bar BMEP) cases were selected to evaluate the potential of combining different injection strategies and swirl ratios to maximize the engine efficiency. In this sense, different injection strategies (SoIs and injection pressure), swirl number, intake and exhaust pressures were tested.

Thus, for each load, two steps were carried out. In a first step, a parametrical swirl variation maintaining CA50 constant. In a second step, the injection pressure, SOE's, Energizing Time's, and number of injections of the lowest pumping work case (low SR condition) were modified to get similar RoHR as the case with the highest Gross Indicated Efficiency, high SR condition (not only CA50 but also CA10, CA25, CA75 and CA90). Thus, the main objective of this second step was to maximize the brake efficiency (indicated efficiency vs pumping work) for each load.

Thus, starting with the first step, to isolate the effects of swirl ratio variation, all engine parameters except the swirl were kept constant. Therefore, tests were performed with the same pressure, temperature and trapped mass at the IVC, as well as the same CA50 and fuel mass. Engine settings for this first step in each load are shown in table 3.