Swumble In-Cylinder Fluid Motion: a Pathway to High Efficiency Gasoline SI Engines

Guillaume Bourhis, Dr. Olivier Laget, Rajesh Kumar, Xavier Gautrot
IFP Energies nouvelles, 1 et 4 avenue de Bois-Préau, 92852 Rueil-Malmaison, France, Institut Carnot IFPEN Transports Energie

Summary

Global warming concerns, and emissions regulations require to improve internal combustion engine (ICE) efficiency and its emissions levels. Focusing on spark ignition engines, main trend is to increase compression ratio (CR) combined with the use of Miller/Atkinson cycle. One drawback of Miller/Atkinson cycle is the reduction of the in-cylinder fluid motion and thus a loss of turbulent kinetic energy (TKE), resulting in a decrease of the combustion speed and then a limitation of the engine efficiency gains. To improve this weakness, IFP Energies nouvelles (IFPEN) has been working on the development of a complex in-cylinder fluid motion so-called swumble. This is the combination of tumble, cross-tumble and swirl motion whereas current engines use mainly tumble fluid motion.

To develop a new engine using this innovative in-cylinder fluid motion, CFD calculations and tests on a single cylinder engine have been performed. By extrapolation from single cylinder results, the engine outputs of a three-cylinder engine are estimated.

Simulation results highlight the good adequacy when using such complex aerodynamic motion coupled with Miller/Atkinson cycles. There is nearly no impact of the valve opening duration on the production of TKE. Moreover, test bench results confirm the nearly constant combustion speed when using different level of Miller cycle. A greater capacity for dilution and a reduction of the emitted particles are also demonstrated. The optimal configuration is a 13:1 CR with a 140 CAD intake lift duration, taking benefits from IGR and EGR and using lambda one operation throughout the whole engine map. The extrapolated maximum output torque and power are respectively 175 N.m/L and 70 kW/L with a peak brake efficiency of 42%. In addition, brake efficiency higher than 40% covers a significant area of the engine map.

This paper details the different steps of this development. Further improvements of this swumble motion approach are currently under progress to make it applicable for different architectures and engine capacities.

1 Introduction

For decades now, automotive industry is improving efficiency and emissions of internal combustion engines [1–4]. As far as the spark ignition engine is concerned, several items have been optimized: valve actuation by using VVT and VVL systems, injection from MPI to GDI, friction by using new materials, heat management by using e-pump and novel water coolant circuit. Of course, this list is not exhaustive and a lot of other devices were also improved. A major current trend is the increase of CR and the use of Miller/Atkinson cycle. This trend enables to de-correlate effective CR (which decreases) and expansion ratio (which increases), enabling gains in terms of pumping losses and knock sensitivity [5–7]. However, this Miller/Atkinson approach faces problems such as fluid motion reduction which impedes the combustion speed and becomes the limiting factor to further improve overall combustion efficiency. Consequently, main trend is to increase fluid motion thanks to high tumble ports [2; 8; 9] and the use of new production processes such as laser cladded valve seats [10].

In the current paper, main idea is to find a way to maintain constant fluid motion whatever the Miller/Atkinson cycle intensity is. So that, a new design is proposed to enhance engine efficiency of relatively low displacement engine while being compliant with Euro 6c and further [11] especially concerning particulates emissions. In terms of combustion process, only propagative combustion process in stoichiometric conditions is considered here. Advanced combustion modes such as HCCI, CAI, GCI or SACI are not considered here.

First, fluid motion is studied and optimized using CFD calculations in order to decrease the effect of Miller/Atkinson cycle on combustion speed. The corresponding fluid motion is then implemented using a dedicated cylinder head and a global campaign on a single cylinder engine at the test bench is performed. An estimation of what could be the results for a three-cylinder engine is performed in the last part of the paper.

2 CFD calculations to design a combustion system with swumble in-cylinder fluid motion

2.1 Engine main characteristics

Main target of CFD calculations is the improvement of fluid motion especially in case of increased CR and use of Miller/Atkinson cycle. It is decided to obtain this result without using any mobile device at the intake of the engine such as aerodynamic flap [12; 13].

Fig. 1: shows the different main fluid motions that can occur in an internal engine. While most SI engines present tumble motion at different level [2; 8; 9], the idea is not anymore to increase tumble motion in case of coupling with Miller/Atkinson cycle. The idea is in fact to develop an innovative fluid motion which is not anymore sensitive to Miller/Atkinson cycle i.e. not sensitive to IVC timing (EIVC or LIVC). The combinations of swirl (mainly used in Diesel engines), tumble and cross tumble are considered here for the calculations.

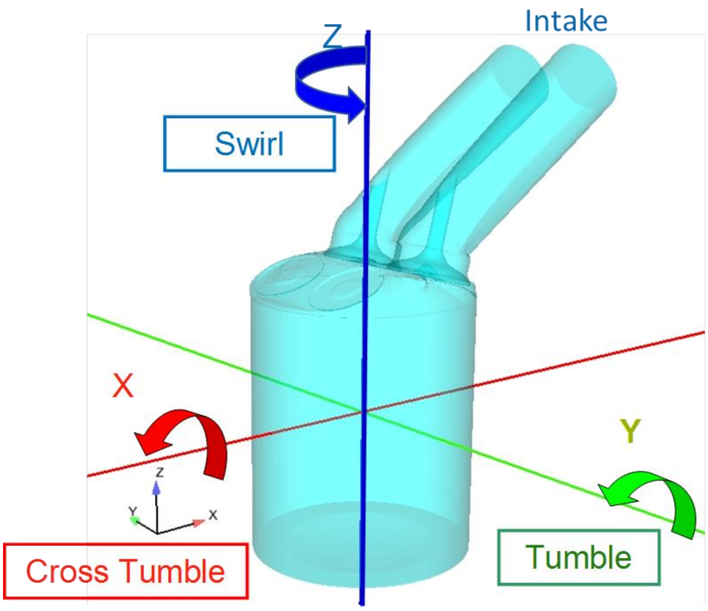

Fig. 1: Different fluid motions in an internal combustion engine

A low engine displacement is considered. The engine bore is 72.2mm. Only two valves (one intake and one exhaust) are implemented. To maximize valve sizes and especially the intake one, the spark plug can no longer be positioned at the center of the combustion chamber. As one spark plug is off-center, a second one is used in order to decrease maximum distance from the spark plug to the wall thus knock sensitivity.

2.2 3D CFD simulations

Once, complete in-house Computer Assisted Design (CAD) of the IFPEN swumble engine (cylinder head, liner and piston) was performed, 3D CFD simulations have been performed using CONVERGE version 2.3.16. Those calculations were performed in parallel with 3D CFD modelling (using the same tool) of a reference engine configuration using pure tumble motion as in-cylinder aerodynamics (Tab. 1:).

In the present paper CFD simulations aimed at representing only the exhaust, intake and compression strokes of the engine cycle. Indeed, no combustion simulation is needed to obtain fluid motion up to spark timing. For both configurations, the initial conditions in terms of in-cylinder pressure, temperature and gas composition were provided from reference engine single cylinder tests.

The turbulence is modeled owing to the k-ε RNG model [14] with default constants, and a standard law-of-the-wall is used as wall treatment. Walls are non-slip, hydraulically smooth and their temperatures are assumed to be constant. Regarding the numerical aspects, the 2nd order upwind scheme is retained for all balance equations, and the iterative method is the PISO algorithm from Issa [15]. The time step is variable in the simulation and defined as the minimum over the computational domain between a convective CFL number constrained below unity and an acoustic CFL number below 50.

3D CFD simulations allow detailed representation of the TKE on reference engine and on IFPEN concept with Miller/Atkinson cycles. Fig. 2: displays TKE on a given operating point (3 operating conditions have been computed). Whereas the TKE at spark advance is largely influenced by Miller/Atkinson cycle in reference configuration with pure tumble fluid motion (divided by 2 with Atkinson 240 CAD compared to reference configuration), TKE of swumble configuration is nearly constant in Miller/Atkinson compared to the one produced with standard valve lift configuration. Thanks to 3D CFD simulations, it was demonstrated that in the case of swumble motion there is nearly no impact of the valve opening duration on the production of TKE.

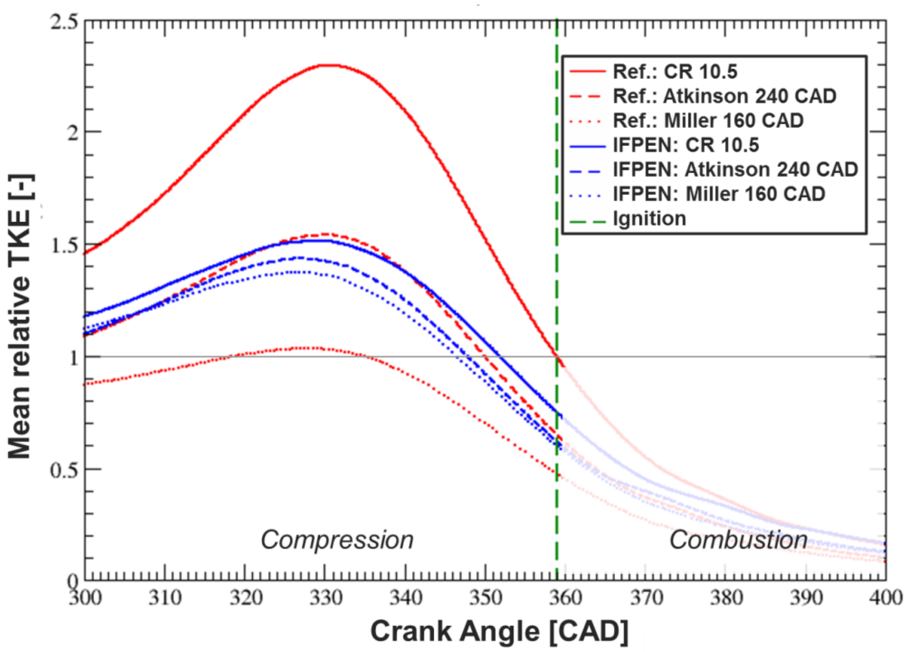

Fig. 2: Turbulent Kinetic Energy in combustion chamber during compression stroke for different engine configurations at 2000 rpm 18 bar IMEP

3 Experimental work

3.1 Engine configurations

The single cylinder engine used in this program is a prototype designed at IFPEN. It has a displacement equal to 350 cm3, a compression ratio of 10.5:1 (can be increased to 13:1 or more), and Gasoline Direct Injection (GDI) in central position. In this study a solenoid injector was used and enables up to 4 injections per cycle. Ignition energy can be varied from 50 to 200 mJ, using a high energy coil. A dedicated Low Pressure (LP) EGR circuit enables high level of EGR rates to be achieved. During this program most of the technological levers mentioned above have been used. Main characteristics of the engine are summarized in Tab. 1:, as well as the reference engine: single cylinder derived from OEM multi-cylinder stock engine.

	IFPEN Prototype Single Cylinder Engine	Reference Engine
Type	Single cylinder, 2 valves	Single cylinder, 4 valves
Capacity	350 cm3	400cm3
Bore x Stroke (S/B)	72.2 x 85.8 mm (1.19)	75 x 90.5 mm (1.21)
Compression ratio	10.5 to 13: 1	10.5 to 12: 1
Injection	GDI central	GDI central
Fluid motion	Tumble + Swirl motion	Tumble motion
Ignition	Single coil, single spark (50 to 200 mJ) 2 spark plugs	Single coil, single spark
Camshaft	VVT intake and exhaust	VVT intake and exhaust
Type of combustion	Stoichiometric (lambda 1.0), with or without EGR	Stoichiometric (lambda 1.0), without EGR
Fuel	Standard Euro 6 E10	Standard Euro 6 E10

Tab. 1: Main characteristics of the two tested engines

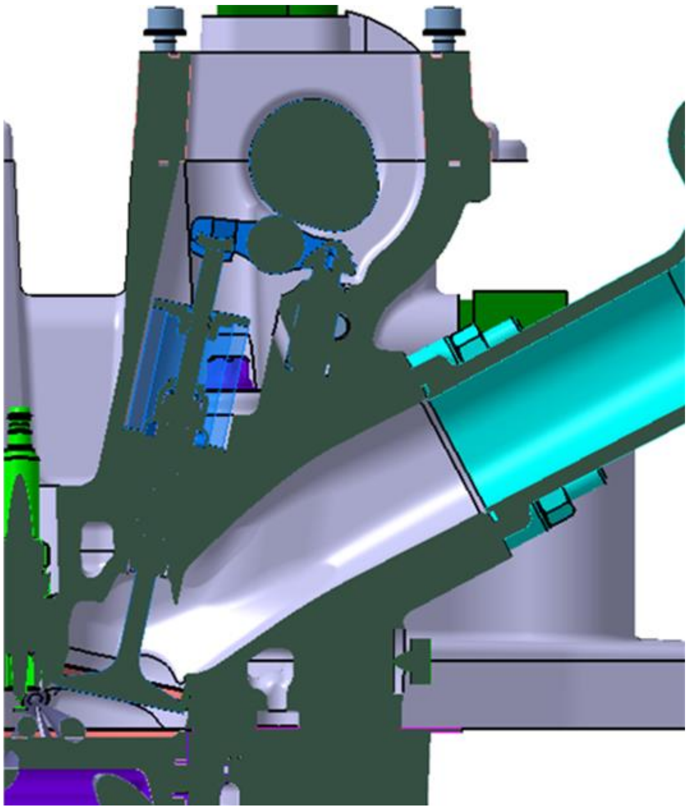

Fig. 3: Partial CAD view of combustion chamber and intake port of IFPEN swumble engine

3.2 Test bed configuration

The engine is equipped with an in-cylinder pressure sensor, an exhaust gas analyzer, and several pressure sensors and thermocouples as shown in Fig. 4:. Morphée 2 from FEV is used as the test supervisor and Osiris (FEV) for rapid acquisition. Combustion analysis with heat exchanges is performed automatically with a dedicated tool in-house developed at IFPEN.

The use of EGR on a SI single cylinder engine implies the use of a complex air loop, as shown in Fig. 4:. More details about the specificities of the IFPEN single cylinder EGR loop are given by Dauphin et al. in [16]. The air flow is controlled by a set of sonic orifices. Intake air temperature and exhaust pressure are controlled to simulate realistic turbo-charged conditions.

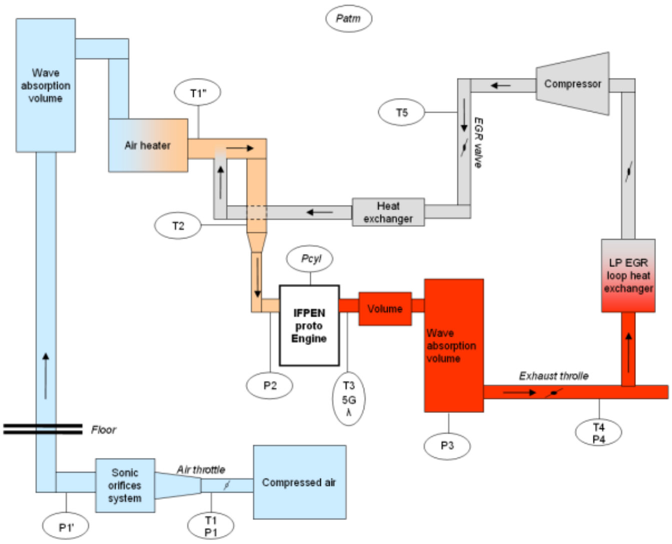

Fig. 4: Test facilities: air loop and EGR loops

3.3 Optimization of CR, intake valve opening duration and use of IGR/EGR

3.3.1 Experimental validation of swumble concept with Miller cycle

Different engine configurations are tested at the test bench using different CR values and Miller intensities, see Tab. 2:.

Miller ratio can be defined in several ways [17; 7; 6]. In the present study the following definition was chosen (eq. 1): Miller ratio is the ratio of volume expansion to effective compression volume at intake valve closure (0.3mm is the reference valve lift). This definition can be used also in case of Atkinson cycle.

$$Miller\ ratio = \frac{V_{expansion}}{V_{effective\ compression}}\ (eq.1)$$

CR [-]	Miller intensity expressed in valve lift duration (@ 1mm lift) [CAD] (Miller ratio)	Label
10.5:1	210 (1.01)	Ref.: CR 10.5
12:1	240 (1.13)	Ref.: CR 12 Atk. 240 CAD
10.5:1	190 (1.00)	IFPEN: CR 10.5
13:1	190 (1.00)	(not presented)
13:1	140 (1.14)	IFPEN: CR 13 Miller 140 CAD
13:1	100 (1.64)	IFPEN: CR 13 Miller 100 CAD

Tab. 2: Tested configurations

CFD calculations give interesting results concerning the almost constant TKE value near spark advance for different Miller or Atkinson intensities. Single cylinder operation at a test bench is a further step to validate IFPEN swumble concept in order to further improve efficiency of engine using Miller/Atkinson cycle.

To do so, a comparison of maximum RoHR (Rate of Heat Release) is done for different intake valve lift durations on the operating point 2000 rpm 2.7 bar IMEP. The combustion speed for Miller ratios between 1 and 1.64 is constant (see Fig. 5: and Fig. 6:), validating the CFD calculation previously presented (see paragraph 2.2).

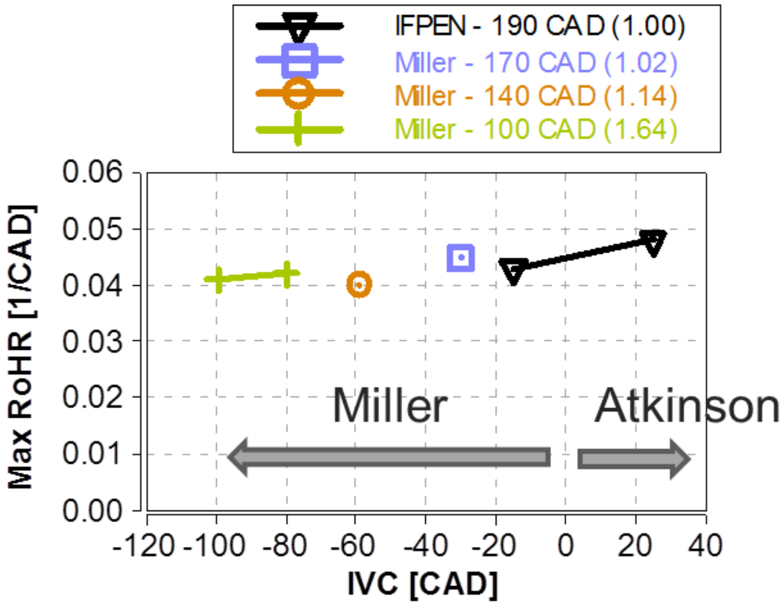

Fig. 5: Maximum RoHR at 2000 rpm 2.7 bar IMEP for IFPEN swumble engine at CR 13

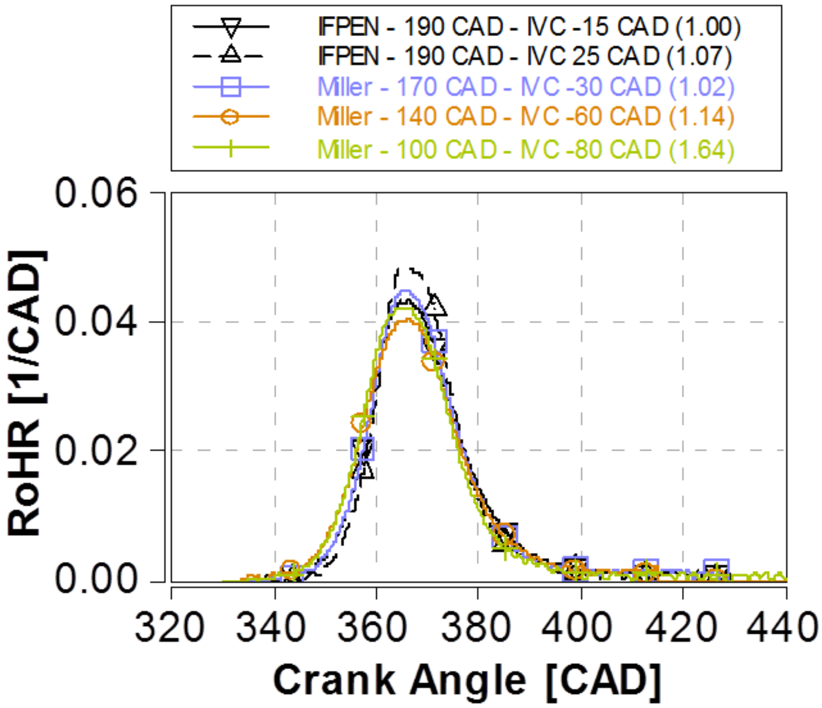

Fig. 6: RoHR at 2000 rpm 2.7 bar IMEP for IFPEN swumble engine at CR 13

3.3.2 Optimization at lambda 1, without EGR

For each engine configuration of Tab. 2:, several operating points have been optimized in terms of IGR rate thanks to intake and exhaust VVT sweeps. Fig. 7: displays the efficiency for the most relevant configurations for a load sweep at 2000 rpm.

Two configurations are compared using the reference engine: stock engine in configuration CR 10.5 and the optimized version: CR12 with Atkinson cycle: intensity 1.13. The optimized reference configuration enables to improve indicated efficiency between 2 and 4% at low and medium load (without knock limitation), see Fig. 7:. At high load, no gain is obtained. A detailed analysis is following.

With IFPEN swumble configuration, we can notice that at the same CR 10.5 value compared to the reference engine, the swumble concept enables gains between 1 to 3%. Increasing the CR up to 13 and adding Miller cycle (140 CAD - Miller ratio 1.14) enables gains up to 10% at low loads. Increasing even more Miller cycle up to Miller ratio 1.64 (100 CAD) does not enable further gains and even decreases efficiency down to 7% at full load. This is mainly due to Miller effect on volumetric efficiency. Indeed, at low and medium load, the increase of intake pressure reduces IGR rates thus increases heat losses. For the highest loads, main drawback is the volumetric efficiency decrease with Miller cycle which increases knock sensitivity through the increase of exhaust back pressure.

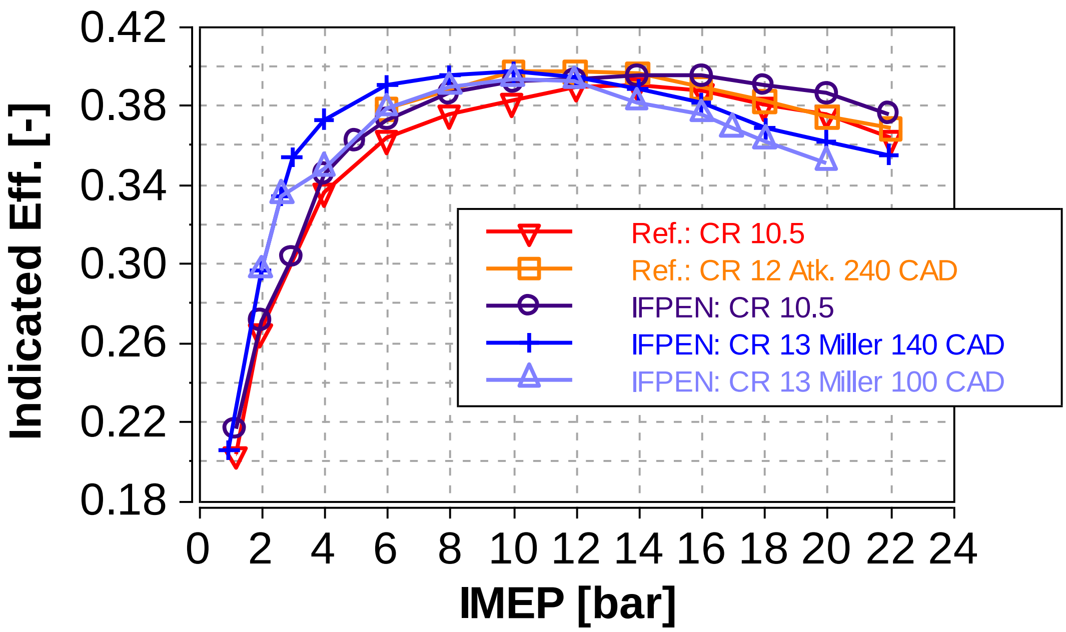

Fig. 7: Indicated efficiency for different engine configurations on a load sweep at 2000 rpm (lambda 1.0)

A detailed analysis of efficiency gaps is performed at medium load: 6 bar IMEP (2000rpm), results are displayed in Fig. 8: and Fig. 9: for 4 different configurations. At this engine load, whatever the CR is, there is no knock limitation. The comparison of the two engine configurations with CR 10.5 (left part of Fig. 9:) shows an improvement of 2.5% for indicated efficiency thanks to a decrease of heat losses ("adiabatic" 3.1%) for IFPEN swumble configuration. This is mainly due to higher IGR rate. This diluted combustion (without EGR at this step of optimization) reduces maximum in-cylinder temperature during combustion so that gamma value is less decreased during combustion. This is the second positive effect of IGR rate on efficiency ("constant gas temperature" 2.1%). However, the slight gap in intake valve durations (190 CAD vs. 210 CAD) is limiting gains thanks to pumping losses, which are favorable with reference engine ("without pumping losses" 2.1%).

The optimized reference engine (CR 12 Atkinson 240 CAD) enables a 3.2% improvement of indicated efficiency at 2000 rpm 6 bar IMEP (middle part of Fig. 9:), compared to reference configuration. This is mainly due to the increase of CR from 10.5 to 12 ("Beau de Rochas cycle" theoretical efficiency 3.3%), because gains in pumping losses thanks to Atkinson cycle (0.8%) and loss in combustion efficiency (-0.8%) cancel each other out.

The optimized IFPEN swumble configuration: CR 13 and Miller 140 CAD enables larger gains compared to CR 10.5 configuration. A 7.2% increase of indicated efficiency is obtained compared to reference engine (CR 10.5), (right part of Fig. 9:). First order effect at 6 bar IMEP is the increase of CR to 13 ("Beau de Rochas cycle" 5.1%) and the improvement of both heat losses (1.2%) and gamma value (2.2%). In this case, this is the decreased effective CR thanks to Miller cycle which improves these two items. It is worth mentioning that the increase of combustion duration is minor: corresponding to an efficiency decreased of 1.1%. This is due to both the increase of compression ratio and also the Miller cycle. Thus, confirming the good ability of swumble concept to keep combustion duration almost constant whatever the Miller intensity (see also Fig. 5:).

Thanks to this detailed analysis, we can say that despite the change in internal fluid motion (swumble vs. Tumble), there is no negative effect of swumble motion on heat losses.

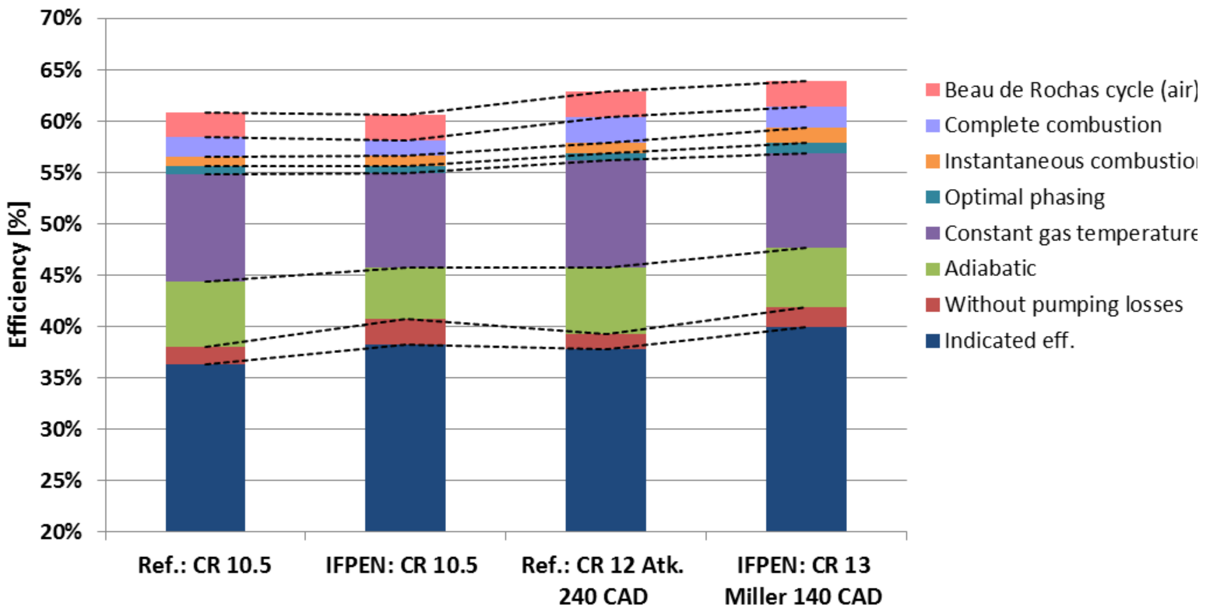

Fig. 8: Energy balances at 2000 rpm 6 bar IMEP for configurations reference CR 10.5, IFPEN Swumble CR 10.5, Ref CR 12 Atkinson 240 CAD and IFPEN Swumble CR 13 Miller 140 CAD

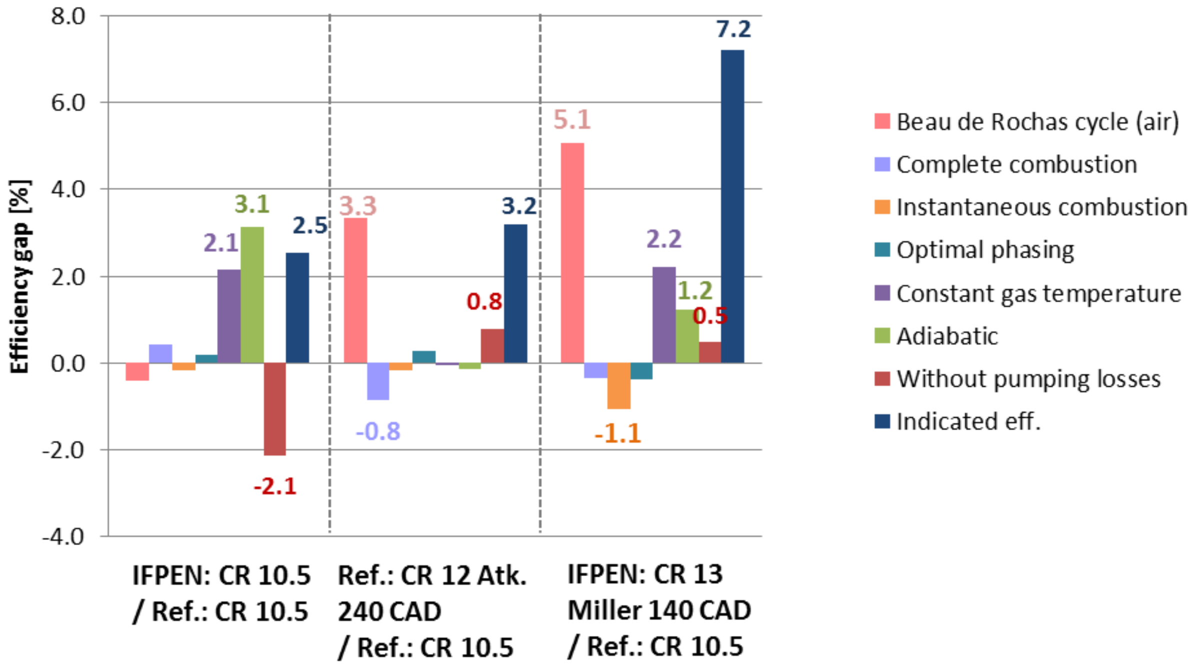

Fig. 9: Details of the energy balances at 2000 rpm 6 bar IMEP for configurations IFPEN Swumble CR 10.5 and IFPEN Swumble CR 13 Miller 140 CAD compared to reference configuration CR 10.5

3.3.3 EGR dilution tolerance of swumble concept

An important way to improve efficiency of SI engine is to dilute fresh air. In this study only lambda 1 operation is considered, so that dilution may consist in IGR (cf. Paragraph 3.3.2) or EGR [18]. In this paragraph, to better compare dilution tolerance of both engines, an EGR sweep is performed at 4000 rpm 6 bar IMEP (CA50 is kept constant, to avoid unwanted effect).

Fig. 10: shows main engine outputs of the two EGR sweeps. EGR has the same effect on both engines for indicated efficiency improvement: same slope. However, it also shows that IFPEN swumble configuration enables higher dilution rate, about +5pt at same covariance IMEP (EGR rate 20 to 25%). This higher EGR tolerance of IFPEN swumble concept enables further efficiency gains thanks to this higher dilution rate.

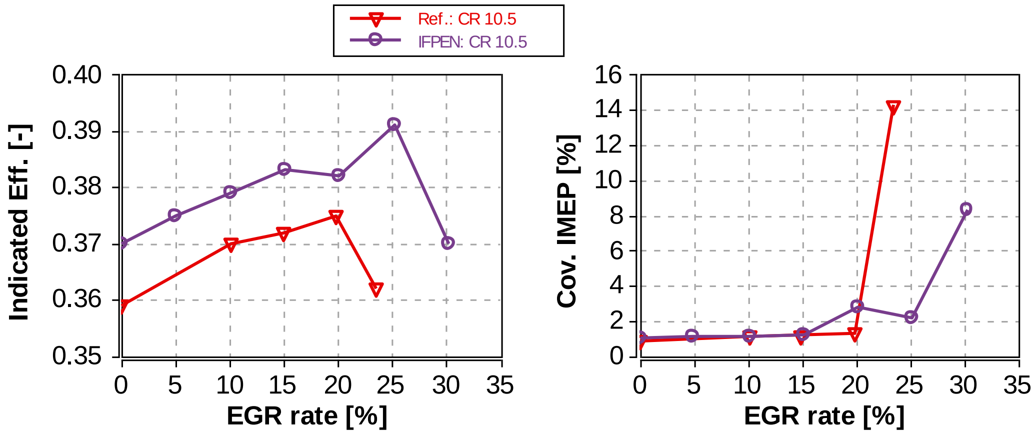

Fig. 10: Comparison of EGR dilution capacity for reference and swumble engine CR 10.5 at 4000 rpm 6 bar IMEP

3.3.4 Particulate Matters emissions

For this first engine campaign, focus is not put on PM emissions. Indeed, basic injection strategy with a stock solenoid injector at 200 bar is used. However, it is important to benchmark PM emission levels for IFPEN swumble and reference engine in configuration CR 10.5 (no Miller cycle). Fig. 11: displays PM emissions levels for different loads. Emission levels for IFPEN swumble engine are lower than the reference engine for almost every engine loads at 2000 rpm. It has to be mentioned that emission level of reference engine complies with Euro 6b. So that the IFPEN swumble engine is well positioned in terms of PM: level is below 1.10^6 #/cc. Nevertheless, to reach current standard (Euro 6d), further optimization of injection system is still possible.

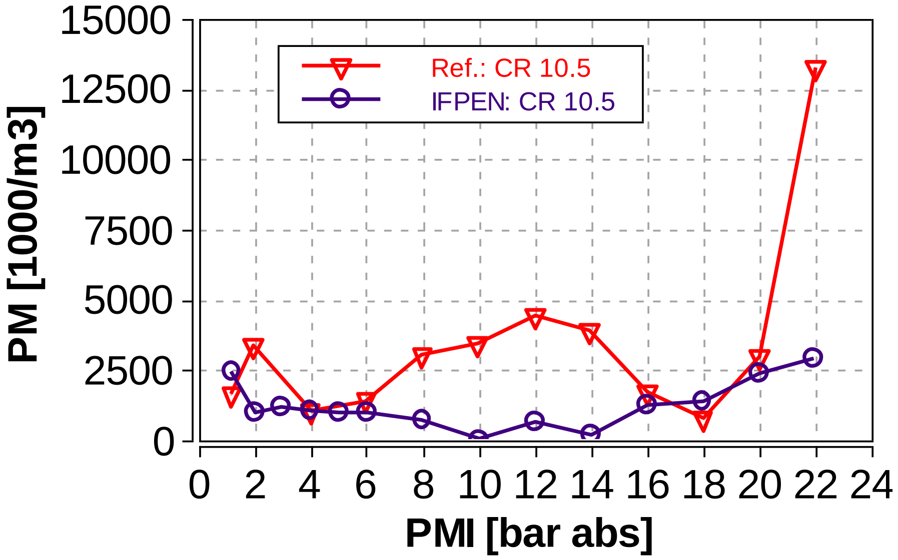

Fig. 11: Particulate measurements from PPS Pegasor (≥23nm) at different loads at 2000 rpm for reference and IFPEN swumble engine CR 10.5

3.3.5 Benefit of EGR dilution

Previously mentioned optimization of IFPEN swumble engine consisted in CR and Miller intensity adaptation and IGR optimization (using intake and exhaust VVT, cf. Paragraph 3.3.2). Thanks to results presented in Fig. 7:, optimal configuration CR 13 Miller ratio 1.14 is chosen mainly because of efficiency gains at low and mid-load compared to reference engine. However, Miller ratio 1.14 was not enough to mitigate CR increase (10.5 to 13) when considering knock sensitivity. EGR dilution is an effective mean to improve engine efficiency both at mid-load and in knock limited operation [18–20] while keeping lambda 1 operation. This way, the exhaust after-treatment system (i.e. TWC) can be maintained.

Fig. 12: displays improvements of efficiency thanks to the use of EGR dilution. High dilution rate up to 37% are possible due to the good EGR dilution tolerance of IFPEN swumble engine mentioned in paragraph 3.3.3. Efficiency gains with EGR varies between 2 and 8%, while keeping covariance IMEP below 3%. At higher engine speed: 4000 rpm, results of EGR dilution are not presented but it enables up to 4% efficiency gains. This lower efficiency gain is mainly due to the lower knock sensitivity as engine speed increases [21].

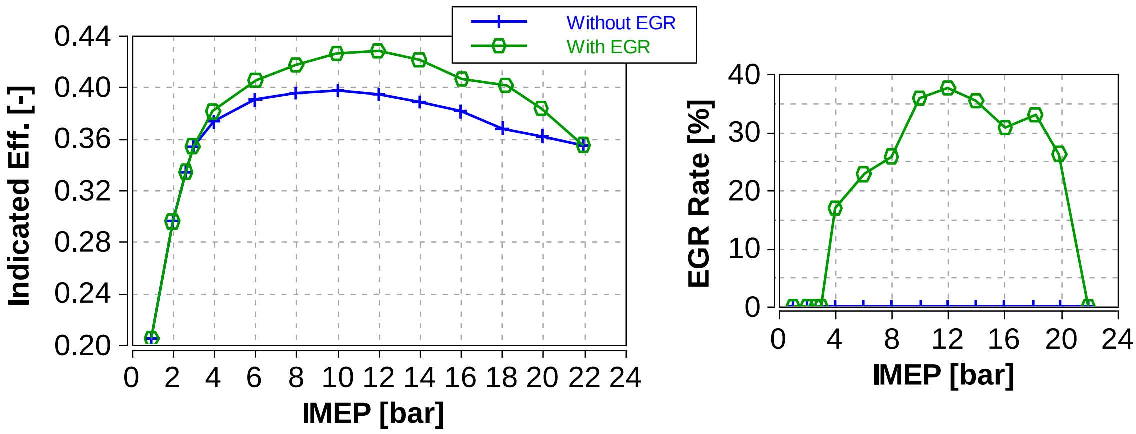

Fig. 12: Indicated efficiency improvement using EGR or not (left part) and corresponding EGR rate (right part) on IFPEN swumble engine with configuration CR 13, Miller 140 CAD (Miller ratio 1.14) at different load at 2000 rpm

The entire engine map has been optimized from EGR and IGR dilution point of view: from 1000 to 5000 RPM, in configuration CR 13 Miller 140 CAD (Miller ratio 1.14). Peak indicated efficiency of 44.2% at 3000 rpm IMEP 14bar (30% EGR) and a large zone with efficiency higher than 42% are obtained, as seen on Fig. 13:. Thanks to this high efficiency combustion chamber, at high engine speed and load, no fuel enrichment is needed so that the engine operation is fully at lambda 1.

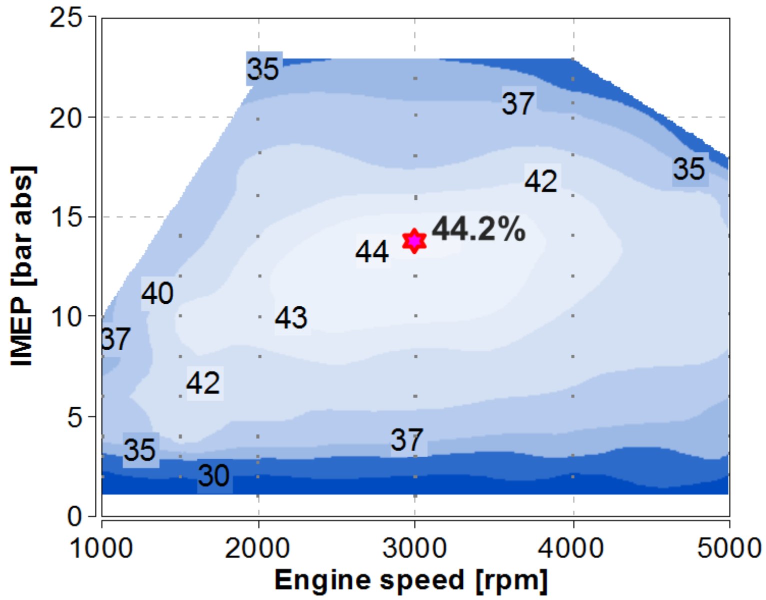

Fig. 13: Indicated efficiency map for IFPEN swumble engine, configuration CR 13 Miller 140 CAD (Miller ratio 1.14) with EGR and fully lambda 1 operation

4 Three-cylinder extrapolation

The main idea is here to extrapolate the results obtained with single cylinder IFPEN swumble engine at the test bench. Obviously several hypothesis have been done - using up-to-date data - and would need further investigation to be consolidated. However, it is important to present what could be the multi-cylinder output of the IFPEN swumble concept.

Main hypothesis concerns FMEP which are assumed to be only a function of engine speed. FMEP level is deduced from scatter band presented in [9], presenting the Honda 1L TGDI three-cylinder engine.

Results of the three-cylinder extrapolation is given in Fig. 14:. Maximum brake engine thermal efficiency is 42% with a large band over 40%. The corresponding engine displacement is 1.05L which gives maximum output torque and power of respectively 175 N.m/L and 70 kW/L. Fig. 15: from Toyota SAE paper [10] show the good position of the IFPEN swumble concept in the trade-off: efficiency vs. output power, at this prototype stage compared to stock engines.

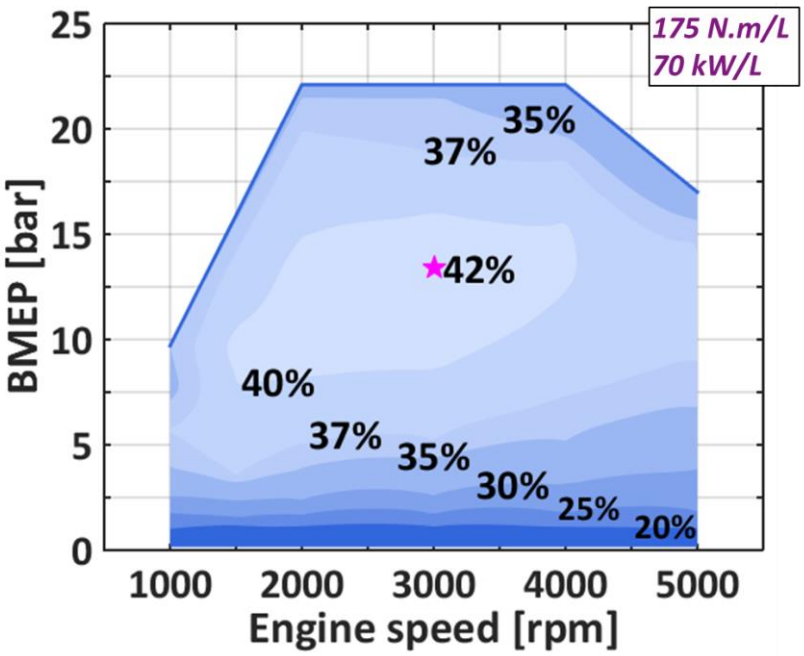

Fig. 14: three-cylinder extrapolation brake efficiency [%] of single cylinder IFPEN swumble results for CR 13, Miller ratio 1.14, with EGR dilution, fully lambda 1

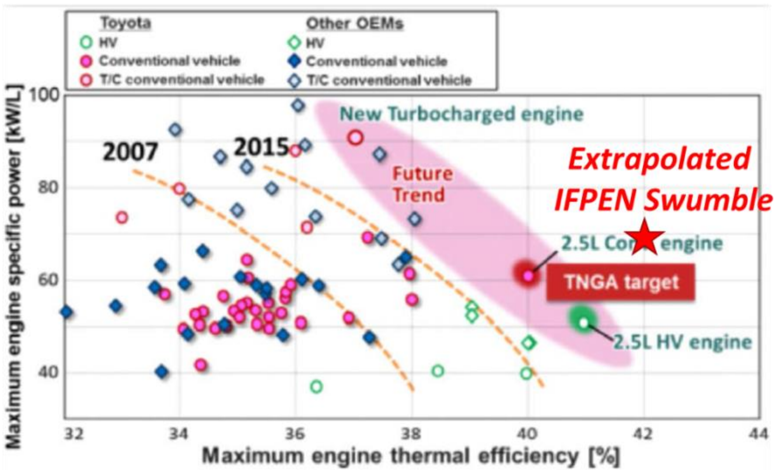

Fig. 15: Prototype IFPEN swumble three-cylinder extrapolation placed in Toyota benchmark giving maximum engine specific power depending on maximum engine thermal efficiency for stock engines, source [10]

5 Conclusions and perspectives

This paper displays different level of results concerning a new concept for high efficiency spark ignition engine. This concept is based on an innovative fluid motion which is combining tumble and swirl creating a swumble motion whereas current engines only use pure tumble motion to generate high fluid motion and turbulence in order to obtain fast combustion process. No mobile device is used at the intake of the engine to create this swumble motion. Both numerical and experimental investigations on swumble innovative fluid motion show that it does maintain good combustion speed even when using Miller or Atkinson cycle (EIVC or LIVC), thus enabling further gains using more aggressive Miller or Atkinson cycle. Moreover, this new fluid motion does not suffer from higher heat losses as compared to reference engine with tumble motion.

Design of the engine is performed to generate swumble motion. CFD calculations confirm the good behavior of the concept when considering the evolution of TKE for different IVC values as compared to the reference engine (Fig. 2:). Experimental work is then performed on a single cylinder engine at the test bench in order to, first validate the constant combustion speed as IVC varies (Fig. 5:) and then to optimize the engine configuration. The optimal configuration is the designed swumble motion associated to CR 13, Miller cycle 140 CAD corresponding to Miller ratio 1.14. Using the engine in fully lambda 1 condition with EGR dilution, enables to reach maximum indicated efficiency of 44.2%. Extrapolation of the results for a three-cylinder 1.05L delivers a maximum 42% brake thermal efficiency associated with a maximum specific power of 70kW/L and maximum torque of 175 N.m/L. Still in a prototype stage (single cylinder), the tradeoff efficiency vs. output power of such engine is well positioned compared to current state-of-the-art (Fig. 15:). IFPEN swumble concept can so far be considered as a pathway to high efficiency gasoline SI engines.

In addition to the good efficiency and output power, the IFPEN swumble concept demonstrated good PM emissions levels. Lower levels to a reference Euro 6b engine are obtained without any specific optimization of injection pattern.

IFPEN is currently developing the new swumble concept generation with increased combustion speed and at the same time the ability of high power density higher than 85 kW/L. Moreover, to overcome flow capacity limitation of the 2 valves engine, the new generation is based on standard 4 valves per cylinder architecture.

6 Acknowledgements

The authors would like to thank other members of the team who have contributed to this paper – notably, but not exclusively, Yann Rabois, Jeremy Jarry, Vincent Berquier and Bertrand Gatellier.

7 Acronyms

- Atk: Atkinson (cycle)
- CA50: Crank Angle of 50% burnt mass fraction
- CAD: Crank Angle Degree
- CAD: Computer Assisted Design
- CAI: Controlled Auto Ignition
- CFD: Computed Fluid Dynamics
- CR: Compression Ratio
- EIVC: Early Intake Valve Closure (Miller cycle)

- EGR: Exhaust Gas Recirculation
- FMEP: Friction Mean Effective Pressure
- GDI: Gasoline Direct Injection
- HCCI: Homogenous Charge Compression Ignition
- IGR: Internal Residual Gas
- IMEP: Indicated Mean Effective Pressure (also called Net IMEP)
- IVC: Intake Valve Closure
- LIVC: Late Intake Valve Closure (Atkinson cycle)
- LP: Low Pressure
- PISO: Pressure Implicit with Splitting Operator
- PM: Particulate Matters
- RoHR: Rate of Heat Released
- SACI: Spark Assisted Compressed Ignition
- TGDI: Turbocharged Gasoline Direct Injection (engine)
- TKE: Turbulent Kinetic Energy
- TWC: Three Way Catalyst
- VGT: Variable Geometry Turbine
- VVT: Variable Valve Timing
- VVL: Variable Valve Lift

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