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

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Global warming concerns, and emissions regulations require to improve internal combustion engine (ICE) efficiency and its emissions level. Focusing on spark ignition engines, the main trend is to increase the compression ratio (CR) in combination with Miller cycle for the turbocharged engines. One well known drawback of the Miller 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 minimize this drawback, 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 SI engines use mainly tumble fluid motion.

To develop a new engine using this innovative in-cylinder fluid motion and demonstrate its potential, CFD calculations and tests on a single cylinder engine have been performed.

In a first phase, the 3D CFD calculations performed highlight the good adequacy of such complex aerodynamic motion when coupled with Miller cycles, especially when compared to tumble combustion system. The impact of the valve opening duration on the production of TKE is much reduced, allowing the effective use of aggressive Miller cycle. In a second phase, test bench results confirm the improved combustion speed, resulting in higher combustion efficiency. A greater capacity for dilution and an important reduction of the emitted particles are also demonstrated. The optimal configuration takes benefits from IGR and EGR, and uses lambda one operation throughout the whole engine map. Maximum power output of 99 kWi/L together with a pick indicated efficiency of 45 % have been obtained on a single cylinder engine. In addition, indicated efficiency higher than 42 % covers a significant area of the engine map.

This paper details the different steps of this development, and the results obtained under specific high efficiency strategies and on the optimized engine map.

1. Introduction

The need to reduce greenhouse gases (GHG) is pressing upon the transportation sector, as aims to limit global warming is imposed and encouraged by the European Union and the Paris treaty. Transportation is a high contributor to GHG and other pollutant emissions. The transportation sector is responsible for 20 % of GHG emissions in Europe [1], and 25 % worldwide. Among these, road transportation represents the majority of the total GHG emissions, reaching 72 % in Europe [2], and 71 % worldwide (2018 [3]). In addition to hybridization, increasing engine's efficiency allows to reduce vehicle carbon dioxide (CO_2) emissions. It is estimated that to respond to COP21 treaty, CO_2 emissions should be further reduced by 50 to 85% from 2000 levels by year 2050 [4]. More specifically, light-duty vehicle transportation target in Europe is to improve the overall GHG impact by approximately 60 % [1].

Latest developments have allowed to increase the maximum thermal efficiency of gasoline internal combustion engines from values of 37 % to 41 % in 2018 [5], this under stoichiometric conditions. For spark ignition engines, the two major developments that have helped reaching high level of efficiency are the integration of exhaust gas recirculation (EGR) and Millerisation of the intake valve lift. There are several levers by which EGR improves efficiency: reducing of the thermal and pumping losses, and increasing the auto-ignition delay and thus extending the knocking limit. However, the maximum EGR rate is limited by the increase of combustion instability, through the reduction of the combustion speed, and the delay of the end of combustion, quenching the flame progress, and increasing unburned mixture. Under certain high load operating conditions, additional EGR rate increase does not provide further optimization of the knock limit.

The integration of the Miller cycle, i.e. Early Intake Valve Closing (EIVC), results in the reduction of the effective compression ratio. Together with the increase of the geometrical compression ratio, an optimum compression, at increased geometric expansion ratio can be achieved. However, because of the EIVC, air mixture expansion under closed valve condition tends to reduce turbulence, and this is further aggravated by the subsequent compression prior to ignition. Lower turbulence will have a direct effect over air-fuel mixture, ignition delay, stability of the initial flame kernel, and flame speed; thus reducing cycle efficiency and increasing pollutants. EGR rate and Miller concepts are strategies that can be achieved at limited technological cost, a requisite that becomes mandatory for hybridized powertrain solutions, and thus their optimization is a significant technological step.

Other than GHG, transportation is also source of other pollutants, as are particles, unburned hydrocarbons (UHC) and carbon monoxide (CO). The introduction of Real driving emissions regulations (RDE) increases the level of severity and introduces a new challenge to vehicle manufacturers. Hence, the understanding of particle formation, its modeling, and reduction of sources of formation, have all been of increasing interest. Moreover, study of the lower particle diameter emissions, through the extension of the lower particle diameter measuring range, from 23 to 10 nm, could promote increased severity of the regulations. Therefore, enhancing fuel mixture and a full combustion are major requisites within engine development programs.

To respond to this concern, IFPEN proposes the development of an optimized engine architecture, that enhances air turbulence, perfectly adapted to Miller strategies and high dilution rates, and thus improves the engine efficiency over a large range of operating conditions. The works here presented will clarify the optimization works that have been realized, and the resulting outcome that is achieved. Moreover, further developments are underway, and the perspectives to which they will lead are also addressed.

2. Engine development

IFPEN proposes the optimisation of a gasoline direct injection (DI) under Miller strategy, capable of accepting an increasing level of EGR rate. The development and engine testing here presented are carried out on a single cylinder engine for proof of concept. A major concern in the development is the preservation, and further enhancement of the turbulence at the time of ignition and throughout the combustion.

Previous works by Cordier et al. [6] have highlighted the advantages of Miller and Atkinson cycles (early and late intake valve closing, before and after bottom dead centre, respectively) as it enables reduced pumping work and heat losses. They also show the advantage of combining these strategies to higher CR, in order to improve the thermal efficiency while limiting knock event at high load. However, combustion development is typically affected by the loss of turbulence. In the case of Miller cycle, the tumble motion can be significantly reduced, as illustrated in Figure 1. Works by Cordier et al. quantified a loss of 66 % of the tumble motion at 70 °CA before top dead centre (BTDC); and a reduction of the turbulent kinetic energy (TKE) of 52 % at TDC. The consequence of the lower turbulence has been estimated to increase combustion duration by more than 10 °CA at high load operation, as illustrated in Figure 2.

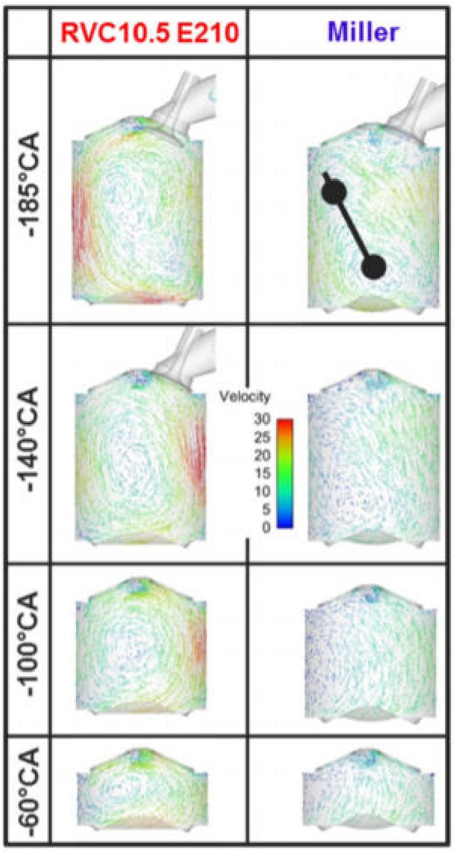


Figure 1: Numerical visualisation of the effect of Miller early intake valve closing (EIVC) on tumble motion. Courtesy of Cordier et al [6].

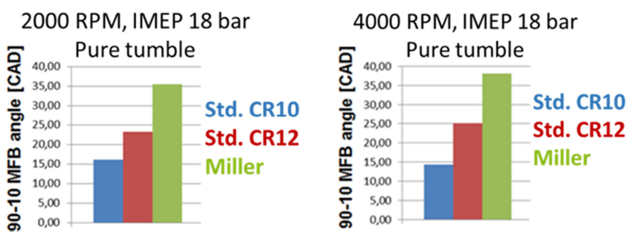


Figure 2: Combustion duration under standard (Std.) and Miller valve lift (CR 12:1). From Cordier et al [6].

Typical strategies that help increase tumble motion have a detrimental effect on the flow capacity. If the application of Miller or Atkinson cycles are to be used as means of increasing engine cycle efficiency, means of improving turbulence at limited detriment in flow capacity are to be developed.

Study of the effect of the flow velocity on the combustion development has been analysed by Laget et al. [7] by means of CFD simulation, allowing a wide variation of tumble and swirl motion on a gasoline engine. The results have indicated that increasing the in-cylinder charge motion has a positive impact on the improvement of the combustion phasing MFB50 and on the combustion efficiency. Moreover, increasing the turbulence intensity also minimizes the ignition delay, MFB10.

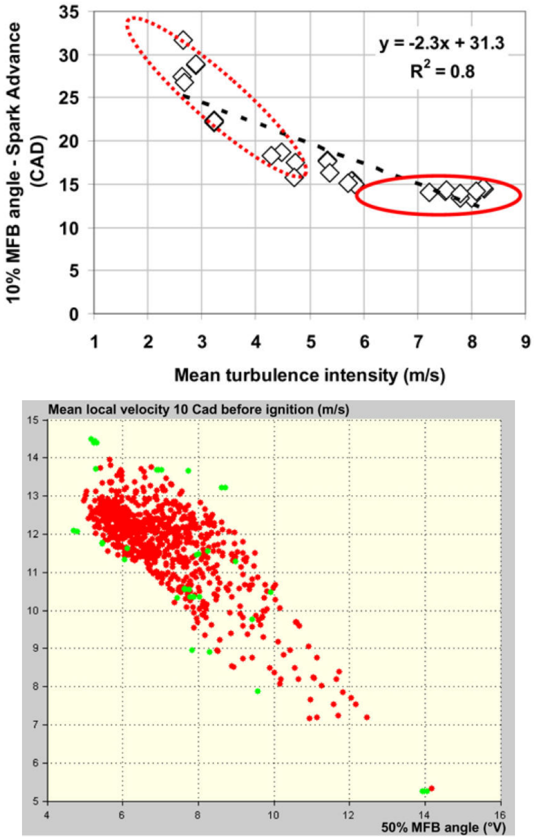


Figure 3. Impact of the mean turbulence intensity over the MFB10 (above), and of the mean local velocity over the MFB50 (bellow), 2000 rpm, 15 bar IMEP [7].

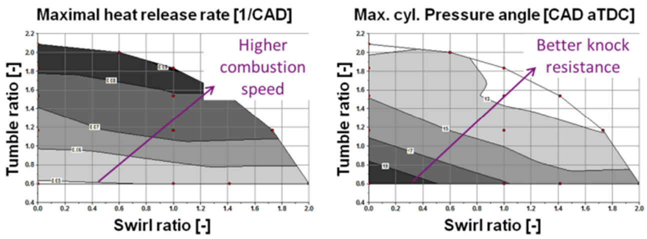


Figure 4: Maximal heat release and Maximal Pressure angle as a function of tumble and Swirl ratio.

The dependency of the maximal heat release and maximum cylinder pressure, on the tumble and swirl ratio, show that these are parameters having significant impact on the combustion. If Tumble ratio is a dominant parameter over heat release, Figure 4 indicates that swirl motion also reduces knock sensibility and thus advances combustion's maximal cylinder pressure angle. As a consequence of these findings, a combination of Swirl and Tumble motions is a promising means of combustion efficiency.

First experimental study on a single cylinder engine of the combination of Swirl and Tumble motion (Swumble™) was realised on a two valves per cylinder configuration, and have been published by Bourhis et al [8]. Base configuration at compression ratio of 10.5, is compared to Miller cycles at intake valve lifts of 140 and 100 °CA, and geometric compression ratio of 13:1. 3D CFD simulation performed on CONVERGE indicated that this engine is characterised by having marginal variations of the Turbulent Kinetic Energy (TKE) with respect to the intake valve lift duration. This has been confirmed on the experimental engine, as represented in Figure 5, where no impact on the maximum heat release (RoHR) has been observed as a consequence of the advanced intake valve closing.

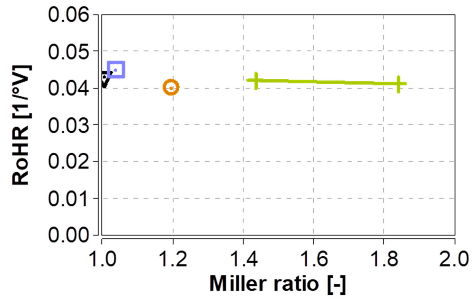


Figure 5: Maximum heat release rate as a function of the miller intensity (early intake valve closing). From study by Bourhis et al. [8].

Optimisation of intake port design for the high Swumble™ concept

These findings have been implemented on the design of a 4 valves per cylinder configuration. In order to obtain maximum efficiency increase under stoichiometric conditions, the study aims at developing an optimized in-cylinder fluid motion, mixing both tumble and swirl, on an increased compression ratio and strong Miller configuration, without compromising the flow capacity at the intake. The design of adapted intake port requires a strong collaboration between the design and 3D CFD calculation teams. At every step, results from proposed designs were analysed by means of Converge™ RANS simulation tools. 3D CFD calculations take into account the complete description of the aerodynamic motion, the analysis of the air, fuel motion during the compression stroke, and evaluation of the turbulence level at top dead centre (TDC). The results were analysed at standard intake valve lift and Miller type short valve lift in order to assess the good adequacy between this innovative fluid motion and the Miller cycle. The aim is to generate a specific Swumble™ motion that guarantees the right location of the vortex and its desired evolution during the compression stroke.

The optimisation of the intake duct resulted in high TKE values, comparable to the reference configuration, whilst respecting the high reference tumble motion under both standard and Miller valve lift, and the addition of a swirl composition. Comparison of the TKE values under reference and Swumble™ configurations are illustrated in Figure 6 at 2000 rpm, 8 and 20 bar IMEP. It illustrates that, unlike the standard tumble configuration, the Swumble™ concept shows TKE values comparable to the optimised reference level, even under Miller valve lift. At same valve lift duration the Swumble™ concept results in the increase of 19 % of TKE at 8 bar IMEP, and 58 % increase at a load of 20 bar IMEP, at the end of the compression stroke, 20 °CA before TDC. Further increasing the Miller ratio does not compromise the TKE value for the Swumble™ configuration, as illustrated in Figure 7. Given the results obtained, 3D CFD combustion calculations were carried out for the reference Tumble and the Swumble™ configurations, under variable Miller ratios, and at constant in-cylinder mass. The simulations estimate that the Swumble™ configuration can result in an efficiency increase of 5 %, at 2000 rpm and 20 bar IMEP, and at 3000 rpm and 12 bar IMEP.

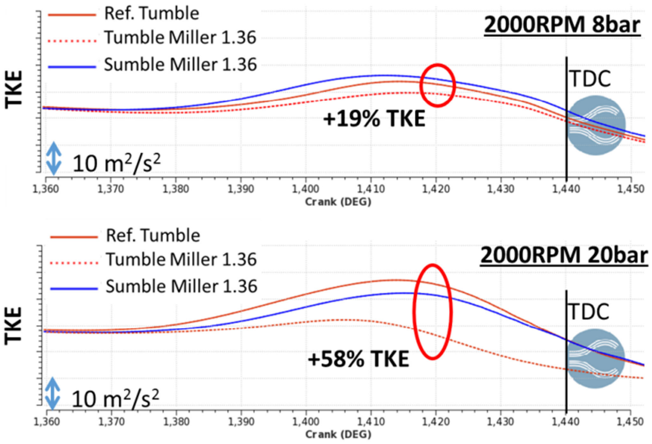


Figure 6: TKE throughout the end of compression stroke, for the tumble reference standard (red solid) and Miller (red dashed) intake valve lifts, and Swumble™ configuration Miller (blue).

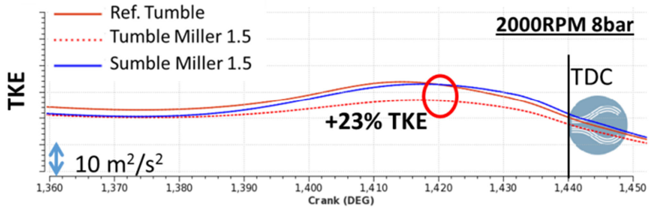


Figure 7: TKE throughout the end of compression stroke, for the tumble reference standard (red solid) and strong Miller (red dashed) intake valve lifts, and Swumble™ configuration Miller (blue).

3. Experimental results

Engine configuration

Experimental tests were performed on an IFP Energies nouvelles single cylinder, 4 valve engine, for the reference high tumble cylinder head and the optimised Swumble™ concept head. The engine presents a 0.41 l displacement, with a centrally mounted multi-hole injector and variable valve timing actuation. A cooled pressure inducer has been mounted, also in central position, to determine combustion phasing and heat release. Intake valve spread has a duration of 140°CA (@ 1 mm valve lift), resulting in an early valve closing, Miller cycle. Full engine characteristics are listed in Table 1. No modification of the injection system, spark, or piston is considered. Only injection timing is adapted accordingly for both configurations. Intake temperature and exhaust pressure are regulated as a function of the engine speed and mass flowrate, to simulate conditions of a turbocharged multi-cylinder engine. The engine is equipped with a low pressure EGR circuit, allowing the variation of the EGR rate over a wide range of operating conditions. Variable valve timing regulates the internal recirculation of the gas rate by variation of the valve overlap. Injection duration is set to ensure a stoichiometric ratio, regulated by means of an exhaust lambda sensor, and verified through exhaust gas composition analysis.

Measurements are taken for the intake and exhaust pressure and temperature, as well as standard exhaust gas analysis of CO, UHC, CH_4, NOx and smoke. Intake manifold CO_2 analysis helps estimate the EGR rate. The intake flow rate is also measured, and together with the exhaust gas analysis, verifies estimation of the fuel consumption, for maximum precision. Exhaust particle are measured with an AVL Smoke Meter, giving an indication of the soot concentration in particles. Particle number are measured using a DMS500, in a range from 5 nm to 1000 nm. The noise and error of the measurement implies that an adequate measurement confidence is assured for signals above 1e4 #/cc.

Engine displacement:	0.41 l
Volumetric compression ratio:	13.0 : 1
Bore x Stroke (mm):	75 x 93
Number of intake /exhaust valves:	2 / 2
Valve Lift duration, at 1mm valve lift (°CA) :	Intake 140 Exhaust 210
Injection system:	DI central
Injector	Bosch HDEV 6 holes
Injection pressure (bar)	200

Table 1. Single cylinder Engine characteristics.

Tests have been performed to qualify the engines behaviour with respect to main parametric variations, as are the exhaust gas recirculation (EGR) rate, valve timing, and lambda. Furthermore, additional optimisation of engine map has been realized to characterize the global engine operation for the Swumble™ concept.

Impact of the Swumble™ air motion with and without EGR

Results at 2000 rpm, and 4 bar IMEP are illustrated in Figure 8, under negative valve overlap, early intake valve opening of -10°CA, without EGR and at maximum EGR rate. The variations were carried out at optimum combustion phasing, MFB 7 °CA. The figure shows the indicated fuel consumption (ISFC), combustions cycle to cycle instability, maximum heat release rate, indicated CO emissions and total number of particles emitted, for the full size range, and accumulation mode. In the absence of EGR, the Swumble™ concept allows a reduction of the ISFC of 2 g/kWh. This is attributed to the faster combustion, as illustrated by the maximum heat release rate. The acceleration of the combustion speed is considered to be a consequence of the enhanced turbulence obtained on the Swumble™ architecture, which intensifies the flame propagation speed. The resulting improved mixture homogenization can also be inferred by the lower CO emissions. For this operating point, particle emissions have been reduced by a 9 fold, as they attain levels of 0.5e6 #/cc, compared to reference values of 4.5e6 #/cc. The reduction is to be found for all particle size range.

For the reference configuration, a maximum EGR rate of 25 % has been reached, and allows for the reduction of the ISFC by 20 g/kWh, at combustion instability of 2 %. For the Swumble™ configuration, a maximum EGR rate of 30 % was attained, at lower cycle to cycle instability. Results confirm the advantage of the enhanced flow motion as means of improving mixture formation and combustion stability, thus increasing maximum allowable dilution rate. As compared to the reference configuration, an increased maximum heat release, reduced CO emissions are present, a further ISFC reduction of 1 g/kWh can be attained. UHC emissions remain similar between configurations and are not represented. Particle emissions are slightly increased at this higher EGR rate.

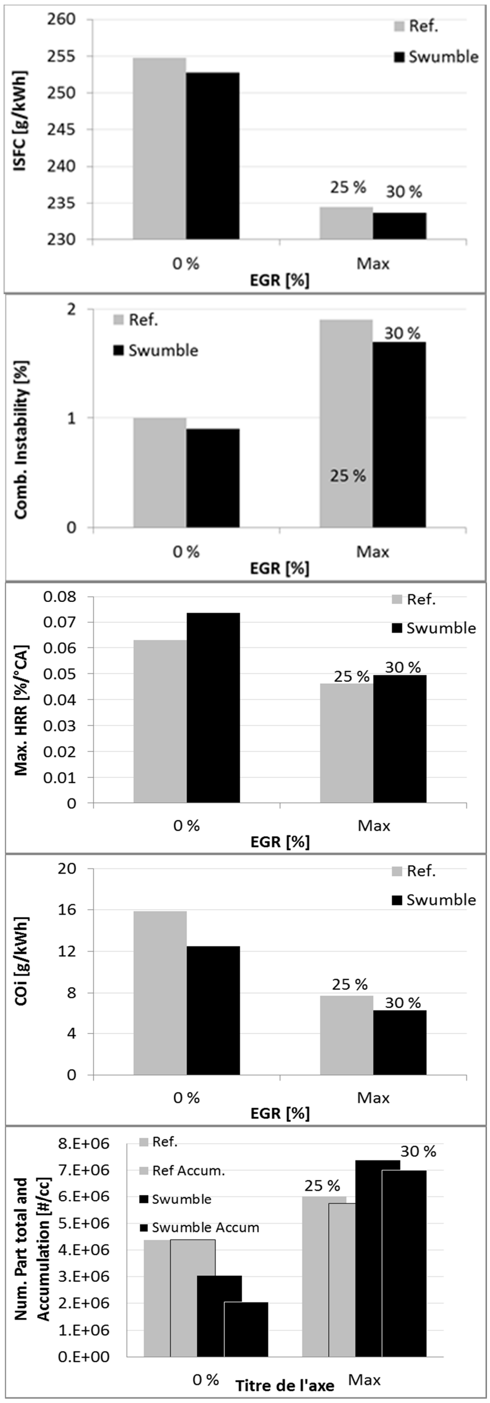


Figure 8. Results at 2000 rpm, 4 bar IMEP, without EGR and at maximum EGR rate.

Analysis of particle emissions at 20 % EGR is illustrated in **Erreur ! Source du renvoi introuvable.**. The complex aerodynamics introduced by the Swumble™ motion reduce strongly the number of particles above a diameter of 18 nm. A reduction in the number of particles is observed: the number of particles in the regulated range 23-1000 nm is reduced by 34 %, and the number of particles in the 10-1000 nm range is reduced by 24 %. If for the Swumble™ concept, the ratio of ultrafine particles bellow 20 nm increases, it is to bear in mind that exhaust gas conditioning is particularly important in this specific range of ultrafine particles mainly as no exhaust gas striper is integrated before the particle counter.

Results at 2500 rpm, 12 bar IMEP are illustrated in Figure 9, at 0 % EGR and maximum EGR of 30 %. Once again, the Swumble™ configuration contributes to a reduction of the ISFC, lessen by 4 % for all EGR rates. Efficiency is thus increased by 1.5 to 2 %. At 0 % EGR rate, the gain is partially attributed to the improved combustion phasing, as knocking auto-ignition delay is counteracted by a faster combustion. However, combustion speed and complete combustion also contribute to the reduction of the fuel consumption. At maximum EGR rate of 30 %, configurations present similar MFB50. The ISFC is reduced by 4 g/kWh, due to the faster flame development and to the lower CO emissions. UHC emissions are similar or slightly reduced and therefore not represented. For the Swumble™ configuration, the total number of particles emitted are significantly reduced over the full size range considered, and the ratio of fine particles to the total number of particles is reduced. An example of particle number distribution is presented in Figure 10 at no EGR condition. It illustrates the strong particle number reduction that can be obtained by enhancing the air flow motion with the Swumble™ concept. It should be noted that smoke emissions remain very low and comparable for both configurations, being bellow 0.05 FSN.

Cycle to cycle analysis on the EGR variation is used to determine the combustion stability throughout the development of the combustion, by means of the magnitudes MFB05, MFB10, MFB50, and MFB90. These are represented in Figure 11 for the reference and Swumble™ configurations. For the reference configuration, as the EGR level increases the combustion becomes more instable, and the instability increases as the combustion develops, thus being progressively stronger at MFB50, and MFB90. For the Swumble™ configuration, the stability is improved throughout the combustion, having a strong reduction at MFB50 and MFB90. Moreover, the impact of the EGR increase on combustion instability is attenuated, and high stability is maintained for an EGR increase from 0 to 25 %. Similar results have been found throughout experimental variations of the internal gas recirculation (valve overlapping) and air dilution, confirming the beneficial impact of enhanced turbulence on flame development, and at variable operating points.

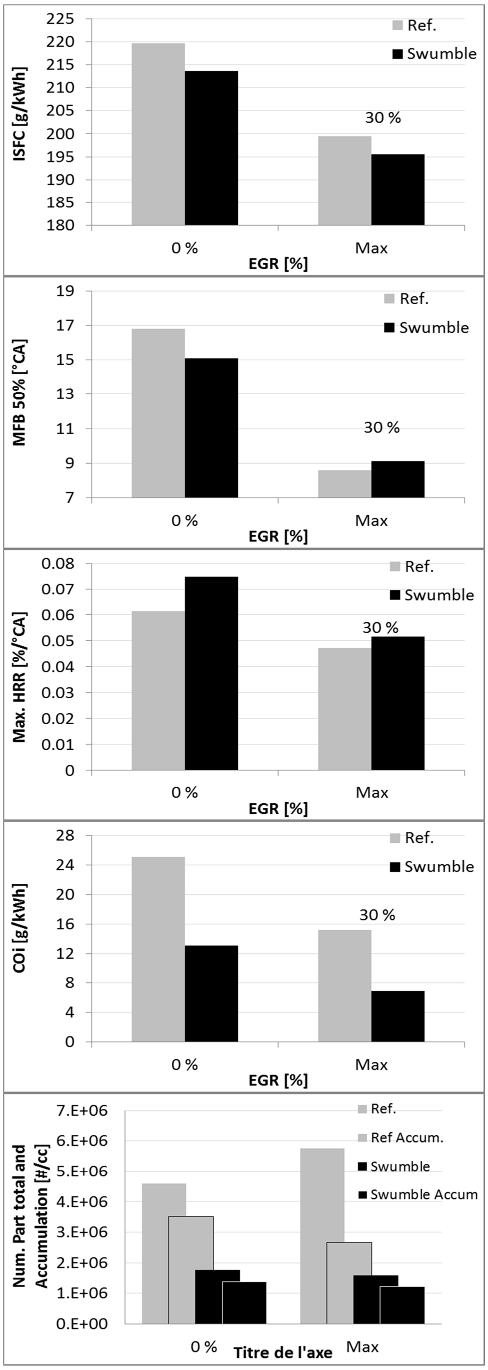


Figure 9. 2500 rpm, 12 bar IMEP, 50 % MFB combustion timing, ISFC, Maximum HRR, CO and number of particle (5-1000 nm, and accumulation mode) emissions.

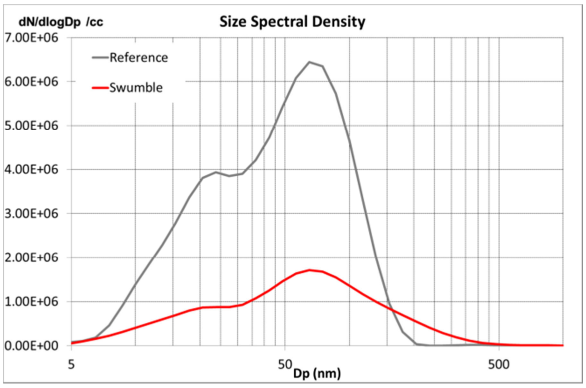


Figure 10. Particle size distribution obtained at 2500 rpm, 12 bar IMEP, negative valve overlap, for reference and Swumble™ configurations.

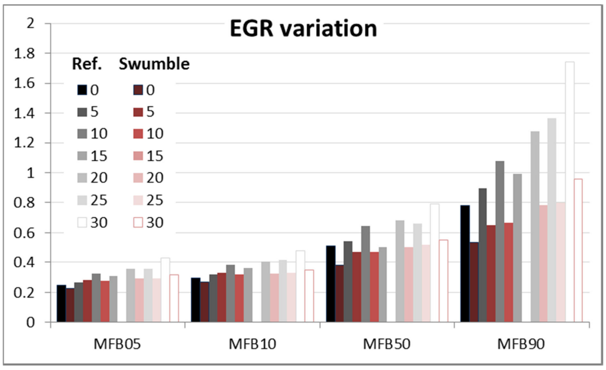


Figure 11. Cycle to cycle variation of the development of combustion, at different MFB, at 2000 rpm, 12 bar IMEP, and EGR sweep, for reference and Swumble™ configurations.

Impact of the valve timing

the effect of internal gas recirculation, by means of intake valve crossing, is also analysed. Standard valve timing for this engine corresponds to an intake valve opening IVO of -10 °CA (10 °CA after top dead centre TDC), and late exhaust valve closing EVC of -10 °CA (10 °CA before TDC), resulting in a total negative overlap of -20 °CA, and reduced exhaust gas backflow. Valve crossing is then obtained by advancing the intake valve opening, and further delaying the exhaust valve closing. The higher exhaust pressure, as compared to the intake pressure, results in part of the exhaust gas being bypassed to the cylinder and the inlet pipe at the beginning of the intake stroke. An advantage of this strategy is the necessary increase in the intake pressure, and hence reduction of the pumping mean effective pressure PMEP, in addition with the reduction of the thermal losses due to the higher intake temperatures. Results for the operating point at 2000 rpm, 4 bar IMEP are illustrated in

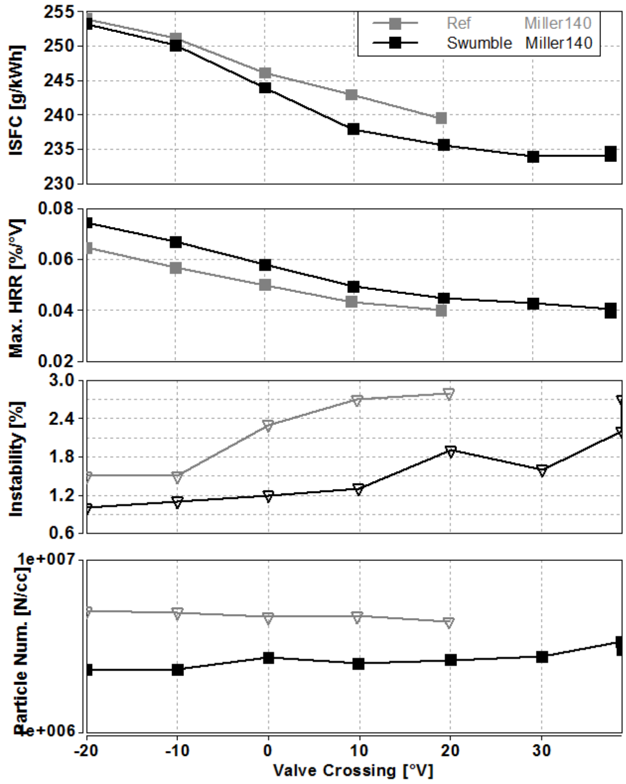


Figure 12. As the valve crossing is increased the Swumble™ configuration provide a lower ISFC, having a total maximum reduction of 2.3 %. This is due partly to the faster combustion, as shown by the increased maximum heat release, the reduced cycle to cycle variations, and to the maximum allowable valve crossing of 40 °CA obtained on the Swumble™ configuration. The reference configuration is limited to a total 20 °CA valve crossing. At higher internal exhaust gas recirculation, combustion instability increases beyond maximum allowable level. The number of particle emissions are reduced by approximately 50 % along the variation. Similar results were obtained at 3000 rpm, 7 bar IMEP, and at 2500 rpm, 12 bar IMEP, with lessen difference in maximum allowable IGR, as the delta in exhaust and intake pressure is reduced for these operating points.

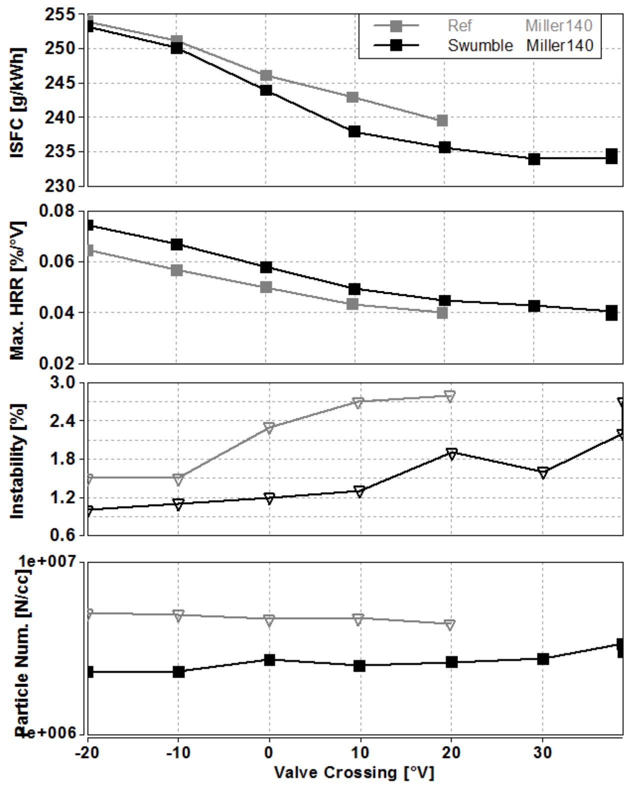


Figure 12. 2000 rpm, 4 bar IMEP. Effect of IGR, through valve crossing, on the ISFC, Max HRR, Combustion cycle to cycle variation and total particle emissions (5-1000 nm).

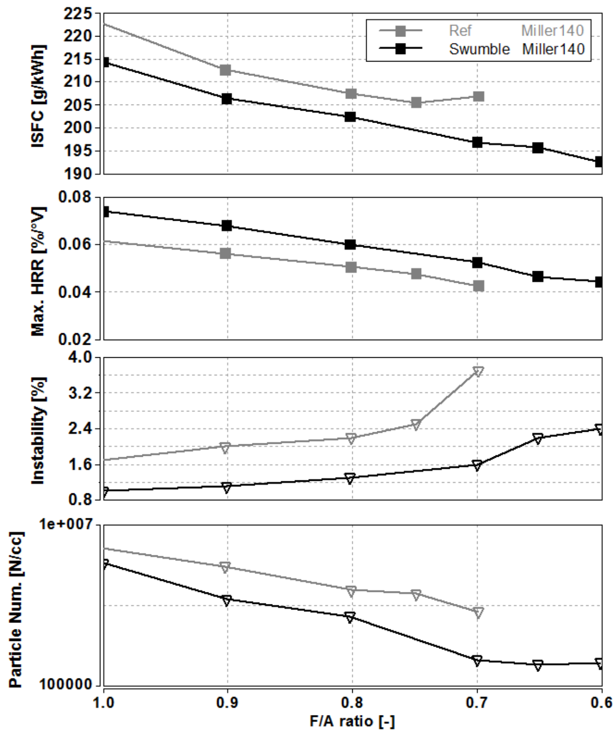


Figure 13. Effect or air dilution con ISFC, maximum HRR, total number of particles and indicated efficiency.

Air dilution

The Swumble™ configuration has also been evaluated under low F/A ratio operating conditions. Under lean operation, combustion development and flame propagation can become instable as ignition delay increases and flame speed is reduced at lean combustion conditions, and thus reducing the potential gains that can be obtained with air dilution. Fuel to air ratio variations were carried out on both architectures, to highlight the behaviour of the two combustion systems, whilst respecting maximum allowable instability conditions. The results are illustrated in Figure 13, for the 2500 rpm, 12 bar IMEP. The enhanced turbulence obtained on the Swumble™ configuration makes it possible to reduce the minimum allowable F/A ratio from 0.75, on the reference configuration, to 0.6. As a consequence, an additional ISFC reduction of 5 %, and an additional increase of indicated efficiency of more than 2 % is obtained, without compromising stability or maximum burn rate. The result on total number of particle emissions is remarkable, with a 86 % reduction obtained at high air dilution.

Cold operation, warm up strategy

Different injection strategies coupled with delayed combustion, were evaluated under cold fluid temperature to investigate the impact of the Swumble™ concept on the warm up phase, as the cold conditions and late combustion are strong contributors to particle emissions. On a first optimisation, one single injection was introduced at optimum phasing, and the spark advance was retarded to obtain a late combustion and increase of the exhaust temperature. As a consequence, post-oxidation time is reduced and can result in the increase of uncompleted combustion emission. The results are illustrated in Figure 14. As the MFB 50 is retarded, UHC emissions tend to increase and reach a maximum at MFB50 of 50 °CA. At further combustion delay, combustion development is hindered, CO emissions increase rapidly (not presented here) and particle emissions increase more significantly. With the high turbulence concept, particle emissions remain almost constant up to MFB of 50 °CA. At this combustion phasing, the referenc e configuration presents a fast increase in particle emissions, whilst the slope is lower for the Swumble™ configuration. As a consequence, for late combustion, beyond 50 °CA, the total amount of particle emissions is reduced for the Swumble™ configuration, by close to 20 %. It is possible that the swirl component deflects the fuel droplets from the cylinder walls, and that the higher flow speed and the faster combustion could favour film evaporation and particle post-oxidation time. Similar results have been observed on an optical engine, with strong tumble and Swumble™ aerodynamics, as reported by Bardi et al. [9].

At equivalent combustion phasing a mean ISFC reduction of 20 to 30 g/kWh can be obtained, and this minimizes fuel consumption increase, at maximum exhaust enthalpy potential.

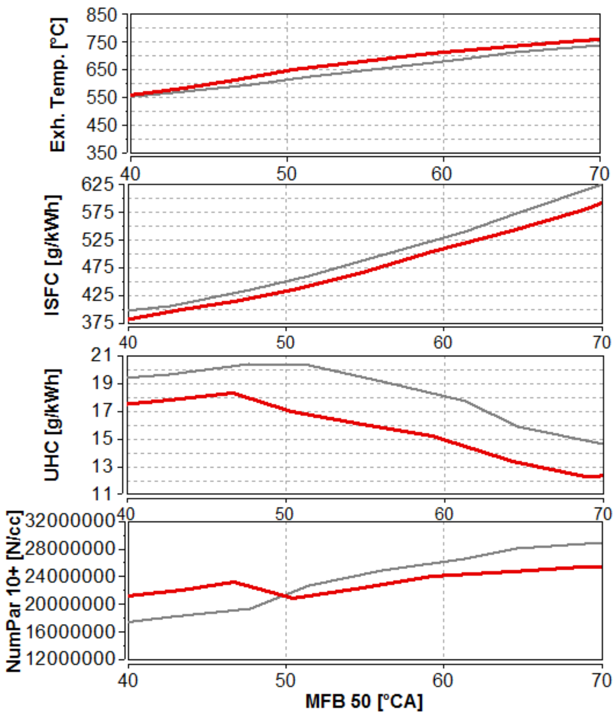


Figure 14. Effect of delayed combustion on exhaust temperature, fuel consumption, cycle to cycle instability and emissions, at 1350 rpm, 3 bar IMEP.

Further optimization of the cold operation by means of multiple injection is presented in Figure 15. Both configurations are calibrated with a triple injection and delayed combustion to maximize exhaust temperature under combustion cycle to cycle instability limits. The Swumble™ configuration attains a delayed combustion, by further 7 °CA, and consequently the exhaust temperature is further increased by 40 °C. However, as per single injection case, at multiple injection, the Swumble™ concepts presents lower ISFC at equal combustion phasing. In order to attain a delayed combustion, late injections are operated, and these being further delayed for the Swumble™ concept. The effect of the colder operating temperature and late combustion result in high UHC emissions. However, these are significantly reduced with the Swumble™ concept. At the same time, particle emissions are reduced by 45 % in the full size and accumulation mode ranges. It is possible that the higher flow speed and turbulence deviates fuel droplets from the piston and chamber surfaces, and enhances liquid film evaporation.

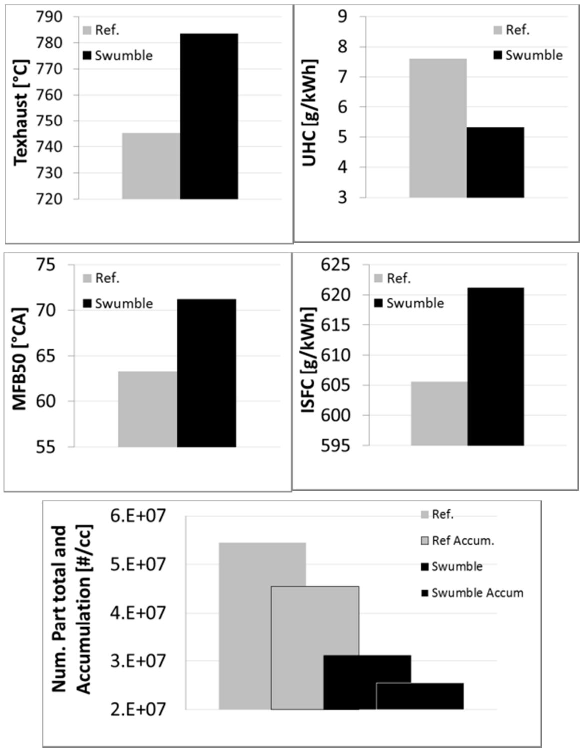


Figure 15. Optimized warm up operating point for maximal exhaust temperature, 3 injections, at 1350 rpm, 3 bar IMEP.

Engine load sweep under optimized configuration

Given the improved results obtained on the Swumble™ configuration, regarding particle emissions, fuel consumption and CO emissions, a full map optimization was carried-out. Operating points were optimized on the basis of valve timing and EGR rate, over 1000 to 5500 rpm, and full load sweep. A load sweep variation at 2000 rpm is presented in Figure 16. Overall the Swumble™ configuration presents a 5 % ISFC reduction in the region from 10 bar IMEP to full load. The maximum heat release rate has been increased over the full charge range, excepting at high load of 18 bar IMEP, despite EGR rate being 5 to 10 % higher. Another characteristic of the Swumble™ port configuration is the reduction of the end of combustion duration, between 50 and 90 % of the MFB. Similar tendencies have been found at other engine speeds, with ISFC reduction between 4 to 5 % from loads of 7 bar IMEP.

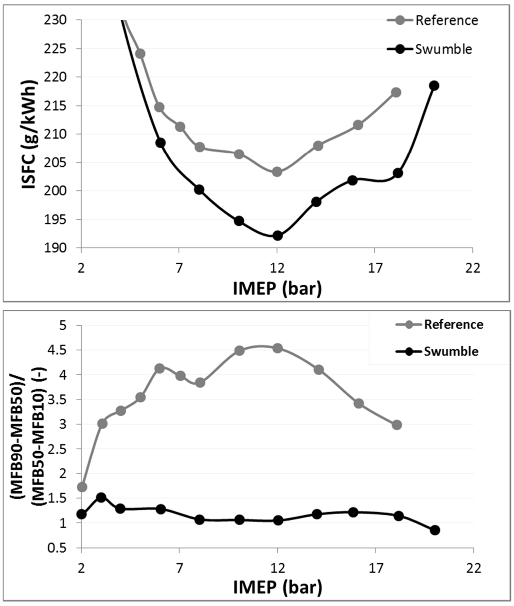
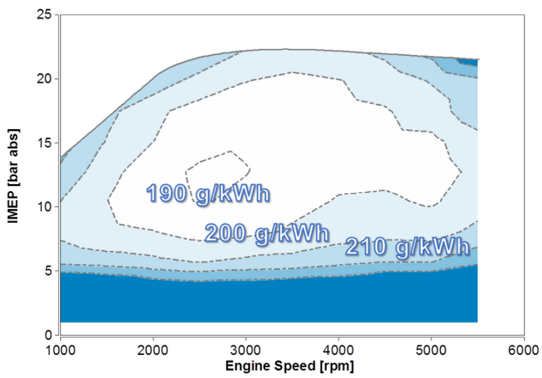


Figure 16. Charge sweep at 2000 rpm for optimized EGR and valve timing settings, over ISFC, maximal HRR and combustion speed.

<u>Engine map</u>

Results for the Swumble™ configuration over the full engine speed and load region are presented in Figure 17. The engine map has been fully optimized at lambda 1 condition. The ISFC map shows that a majority of the operating area presents a fuel consumption bellow 200 g/kWhi, and that a region at 3000 rpm, and 10 to 14 bar IMEP, presents a fuel consumption bellow 190 g/kWhi. The effective efficiency is calculated based on Friction Mean Effective Pressure published by Shibata et al. [4] to simulate multi-cylinder output. As can be seen, estimations yield a high effective efficiency of 43 %, and a maximum power of 92 kW/l. It shows a large area at high efficiency, where most of the operating points present efficiency above 40 %.



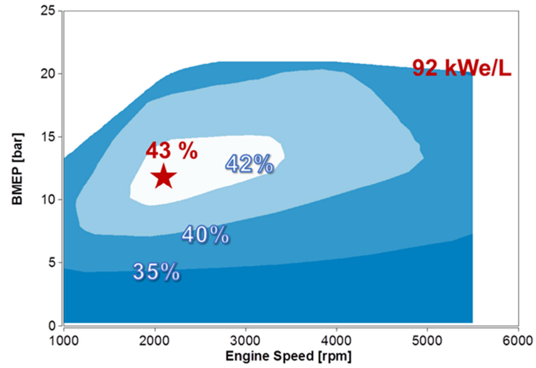


Figure 17. Swumble™ configuration's ISFC and estimated effective efficiency maps.

4. Conclusion

Experimental results on a high turbulence Swumble™ engine confirm the advantage that can be obtained over air fuel mixture improvement, high combustion speed, mitigation of knock auto-ignition, and reduced combustion cycle to cycle instability.

Inlet duct design has been optimized with joint modification and analysis of the design and CFD 3D teams. The CFD calculations estimated the improved design to ensure maintaining a high flow turbulence before combustion, even when using Miller cycle valve lift, and thus to attain 5 % ISFC improvements. This gain has been confirmed by the experimental results obtained on a single cylinder engine when comparing optimized calibration of a conventional high tumble cylinder head against the modified Swumble™ concept.

The improved turbulence has been confirmed through the higher maximum heat release rate over several operating points and conditions. The Swumble™ concept is characterised by a lower cycle to cycle instability, this from early combustion initialization until the end of combustion. Moreover improved cycle to cycle stability was found at typical instable operating conditions, as are high EGR/IGR rate and strong air dilution. As a consequence, higher EGR rates could be attained, thus further improving the ISFC. Similarly, the fuel to air ratio could be decreased from 0.7 to 0.6.

Another characteristic of the improved air flow motion is found on the reduced particle emissions, as fuel is more strongly deviated from cylinder piston, walls and valves, and film evaporation before combustion is enhanced. Particle emission reduction was present on both 5 to 1000 nm, and 23 to 1000 nm ranges. Particle emissions reduction is found to be highest at 2500 rpm, 12 bar IMEP, where it reaches a reduction of 70 to 80 %. Operating point 2000 rpm, 4 bar IMEP produced a particle reduction between 25 to 50 %. Lowest particle emission reduction was obtained under cold operating, and reached 20 % under single injection conditions, and using a multiple injection strategies number of particles emitted can be reduced by 45 %. Similarly, because of the enhanced fuel mixture, CO emissions are strongly reduced, and UHC reduction has also been obtained under severe cold operation. Last but not least, the implementation of the Swumble™ concept is straightforward, requiring very local modifications of the design of the intake ports, and will not require modifications of the industrial production tool, hence making this a "plug and play" solution.

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