

Potential of a Variable Compression Ratio Gasoline SI Engine with Very High Expansion Ratio and Variable Valve Actuation

Paul Ferrey, Yves Miehe, Cyrille Constensou, and Vincent Collee
MCE-5 Development

ABSTRACT

Combustion simulations and single cylinder engine tests show a clear potential when coupling the Variable Compression Ratio (VCR) engine with the Variable Valve Actuation (VVA) technologies.

Simulations demonstrate the thermodynamic benefit from increasing the geometric Compression Ratio ($CR > 18:1$) in combination with VVA compared to VVT strategies, thanks to the use of Atkinson / Miller Cycles. 3D combustion simulations of high compression ratio combustion chamber geometries used with Early or Late Intake Valve Closing strategies have been carried out with IFP-C3D™. They show an indicated efficiency increase up to 12 or 13% between compression ratio 10:1 and 18-20:1 at low loads ($BMEP < 8$ bar).

Single cylinder engine tests have been performed with specific combustion chambers up to $CR 23:1$ and have confirmed the simulation results.

0D GT-POWER™ simulations have been correlated to the engine tests and used to extrapolate them to a 3 cylinders 1.1L TGD I VCR-VVA engine, for both EIVC and LIVC intake valve lift strategies.

Finally, simulations and tests show correlated thermodynamic benefits of the association of VCR and VVA, improving fuel consumption on driving cycles between 8 and 11% compared to a 10.5:1 fixed compression ratio engine equipped with 2 VVT. Combining the VCR and VVA optimizes each potential far more than when separated (alone). Furthermore, the fuel consumption benefit is robust through driving cycles and downsizing level.

KEYWORDS: VCR, VVA, Compression ratio, Expansion Ratio, Miller, Atkinson, EIVC, LIVC, Gasoline

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INTRODUCTION

The purpose of this work is to establish to what extent Variable Compression Ratio engines take advantages of Miller/EIVC or Atkinson/LIVC intake valves lifts achievable strategies with VVA devices. Miller and Atkinson cycles decrease the in-cylinder temperature and pressure at the end of the compression stroke thanks to lower effective compression ratio and in-cylinder temperature at the intake valve closing. Thus, knock tendency is reduced and the geometric compression ratio can be increased. In addition, with a VCR engine, the geometric compression ratio doesn't have to be compromised (if 20 bar BMEP and 80 kW/L, the CR should be lower than 11:1 on a FCR TGD I engine, even with Miller/Atkinson strategies) and can therefore be increased at low and medium loads to very high values.

Historically, the optimal compression ratio of a SI gasoline engines at low loads has been estimated to be close to 14 or 15:1. Two stage VCR FEV results [3] and AVL test bench results [4] on MCE-5 VCRi tend to confirm this optimal compression ratio with traditional technologies and strategies. But recent developments have concluded that the optimal compression ratio could be higher.

Akihisa & Sawada at Toyota [8] have found an advantage to increase the compression ratio when associated with an Atkinson cycle and a Late Intake Valve Closing (LIVC) valve lift strategy, thus separating the effective compression ratio from the expansion ratio. They have found, on engine test bench, a 13% thermal efficiency increase between 13:1 (Prius 1.5L base engine) and 25:1 compression ratios, thanks to modified piston

and enhanced squish effect. According to the authors, such a combustion chamber design promotes a fast combustion together with low wall heat transfer losses.

Yamakawa & co at Mazda [6] have found that the combustion chamber design has a strong influence on the thermal efficiency benefit from high compression ratios. Combustion chamber allowing a rapid initiation of combustion with a center bowl was found to be more efficient than those with convex piston.

The aim of this study, carried out in five steps, is to analyze the thermodynamic potential of coupling VCR with VVA and the potential of high compression ratios ($CR > 15:1$) combustion chambers. The first part shortly explains the Miller/Atkinson cycle and its advantage to reduce the effective compression ratio and increase the geometric compression ratio. The second part is devoted to 3D combustion simulation results on high compression ratios combustion chambers ($>15:1$) with VVA. EIVC and LIVC valve lift strategies are evaluated with different speeds, loads, turbulence levels in order to select the piston geometries for engine tests. The third part describes the single cylinder engine tests with the EIVC and LIVC intake valve lifts and high compression ratios combustion chambers defined in the second part. In order to extrapolate the test bench results to a complete BSFC map, the fourth part presents the 0D/1D model of a VCR-VVA engine with a combustion model correlated with the third part's single cylinder engine tests. This model simulates the fuel consumption benefit from a 3 cylinders 1.1L TGD1 (80 kW/L) VCR with VVA EIVC or LIVC engine on the speed/load map. The last part shows the VCR-VVA fuel consumption simulations on standard driving cycles.

MILLER/ATKINSON CYCLE WITH HIGH GEOMETRIC CR ASSOCIATION

Ideal Cycles

If only VVT valve lift strategies are used as in the ideal Otto cycle, the expansion and effective compression ratios are the same (Figure 1). However, the effective compression ratio is limited for a Spark Ignition gasoline engine, due to the knock phenomenon. As a consequence, the expansion ratio is limited to the maximum allowable compression ratio.

Considering the example of the Miller cycle (see ideal cycle in Figure 1) and an intake valve lift duration managed by a VVA, the effective compression ratio is decreased. The Miller cycle dissociates the effective compression ratio from the expansion ratio, giving the possibility to increase the geometric compression ratio and the expansion ratio while keeping a reasonably effective compression ratio. As a result of the expansion ratio increase, the thermodynamic efficiency is increased.

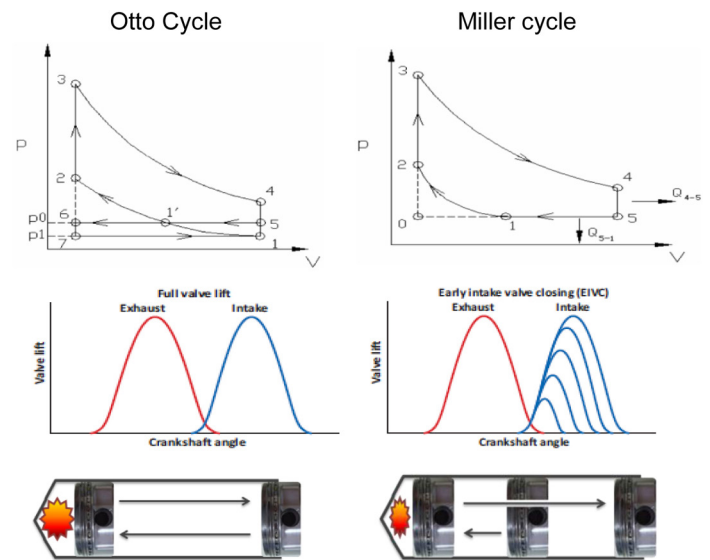


Figure 1. Otto and Miller Ideal cycles description

"Real" Cycles (GTPOWER™ Simulations)

As shown in Figure 2, the Miller cycle reduces the in-cylinder temperature and pressure at the beginning of compression. It is a combined effect of reduced in-cylinder temperature and pressure at the intake valve closing event and reduced effective compression ratio. Therefore, the in-cylinder pressure and temperature at the end of the compression stroke are reduced, reducing the knocking tendency of the engine.

Thanks to the reduced knocking tendency with Miller cycle, the geometric and then the expansion ratio may be increased up to 14:1 at 2000 rpm and 12 bar BMEP with MFB50 < 20 °CA (see Figure 2). A 7% BSFC reduction is obtained with Miller cycle (VVA) and CR14:1 compared to Otto cycle (VVT) with CR 10:1 on such a point.

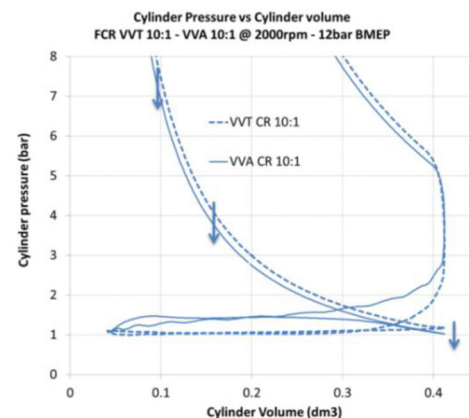


Figure 2. Miller and Otto cycle comparison at 2000rpm - 12 bar BMEP- GTPOWER™ simulations

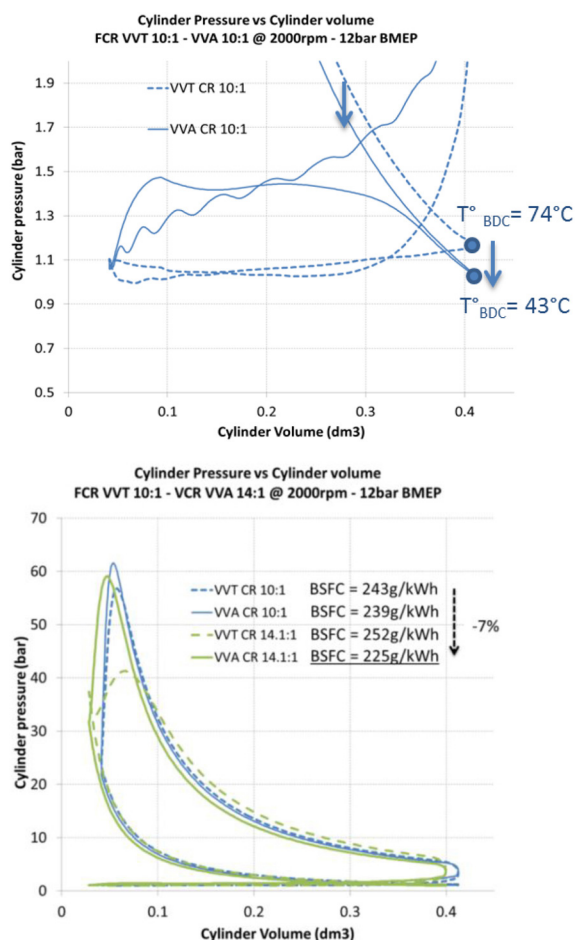


Figure 2. (cont.) Miller and Otto cycle comparison at 2000rpm - 12 bar BMEP- GTPOWER™ simulations

HIGH COMPRESSION RATIOS COMBUSTION CHAMBER 3D SIMULATIONS

3D simulations have been carried out with the "IFP Energies Nouvelles" software IFP-C3D™ [10] provided by LMS (IFP rev 220 - AMESim™ rev 11) with the most widely used gasoline combustion model, ECFM, developed by IFP. The purposes of 3D simulations were to:

- Help to specify combustion chamber geometries (cylinder head and piston crown modification), intake port geometries for aerodynamics and intake valve strategies for single cylinder engine testing.
- Improve the GTPOWER™ 0D combustion model at high compression ratios (>15:1) with VVA strategies (specific aerodynamic and turbulence) and different combustion chamber geometries. The 0D model is necessary to reduce the computation time on the complete BSFC maps.

3D Modeling and Correlations with VVT Engine Tests

Main Hypothesis

The main assessments are the use of an homogeneous mixture (error about 1% on ISFC) and an average turbulence modeling (RANS).

Model

Model description:

	Model
Air	Compressible / Ideal gas
Fuel	gfl1" : gasoline SP95 TAE5970 - PCI = 42543 kJ/kg
Transported species	gfl1, O2, N2, CO2, H2O, CO, O, N, NO, H, H2, OH
Turbulence model	RANS (k-ε RNG)
Combustion model	ECFM [10] (Metaghalshi & Keck correlation)
Heat transfer Model	" Angelberger" (Kay & Crawford) [12]
Ignition	AKTIM [10]

Half a chamber is modeled as the geometry is symmetrical, as presented hereafter in Figure 3.

The mesh size has been defined to minimize the mesh dependency of the 3D combustion IFP-C3D™ results.

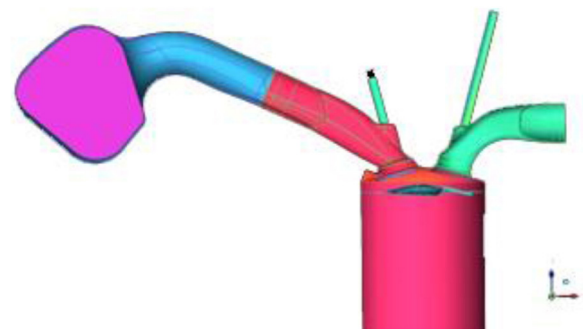


Figure 3. geometry of MCE-5 combustion chamber

Correlation with 4 Cylinders 1.5L MCE-5 VCR-VVT TGD Engine

MCE-5 VCRi 1.5L engine description [4]:

First the initial 3D model is correlated to the 4 cylinders 1.5L MCE-5 VCRi GDI engine tests results detailed in [4].

In order to make the comparison, the volumetric efficiency and the fuel mass trapped into the cylinder are fitted after the exhaust/intake stroke simulation to match the experiment value [4]. An example of correlation at 2000 rpm - 2 bar BMEP at a CR of 15:1 is presented in Figure 4. Simulated pressure gradient and combustion speed seem to be well correlated to

experiment if the BMEP is lower than 15 bar. But the peak cylinder pressure is a bit overestimated by simulation by 4 to 5 %.

Cylinder number	4
Bore x stroke (mm)	75 x 84
Cylinder capacity (cm ³)	371
Equivalent connected rod length (mm)	181.5
4 valves cylinder head 15° half angle pentroof	intake: D = 26.3 mm exhaust: D = 23.8 mm
2 VVT (max lift position relative to TDC)	intake: [84;148] exhaust: [-132;-68]
Valve lifts	intake : 190 °CA @ 1 mm exhaust : 180 °CA @ 1 mm
Compression Ratio	continuously controlled from 6:1 to 15:1
Injection	direct (side mounted)

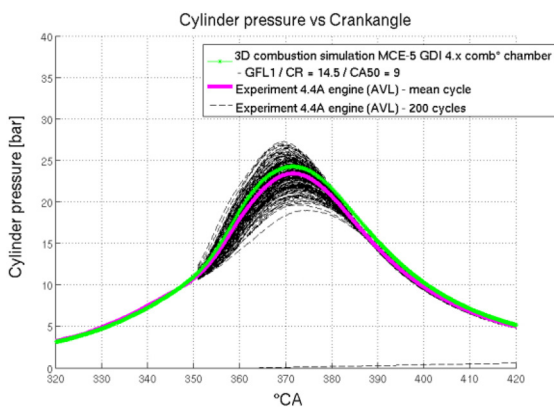


Figure 4. cylinder pressure comparison between IFP-C3D simulation (green) and AVL test bench results [4] (pink and black) - 2000 rpm - 2 bar BMEP - CR 15:1

The cylinder pressure overestimation leads to IMEP overestimation and ISFC underestimation by about 5% compared to the average experimental one [4] (see table below). These simulated ISFC are corrected with a Willan's line to match the experimental IMEP and compare ISFC. The maximum error on the ISFC experiment value is estimated to be lower than 2 or 3%, considering the precise fitting of the TDC value (0.2 °CA precision).

Table 1. IFP-C3D ISFC correlation with AVL engine tests results [4]

Speed rpm	BMEP bar	MFB50 °CA	CR :1	Corrected ISFC @ exp. IMEP		
				Exp. g/kWh	Sim. g/kWh	Diff. (%)
1000	1.0	9°	14.0	372	348	-6%
1500	2.0	10°	14.5	270	261	-4%
1500	4.0	11°	13.9	237	225	-5%
2000	2.0	9°	14.9	265	250	-6%
2000	4.0	12°	14.3	233	220	-5%

A comparison at 2000 rpm - 4 bar BMEP in Figure 5 shows that the ISFC decrease with compression ratio increase is overestimated by the simulation (Blue) compared to AVL test bench results [4] (Red). Therefore, the difference of ISFC between test and simulation increases when compression ratio

increases. A good precision could be obtained at low CR values (CR 10:1 for instance) but not at high CR values (CR > 12:1).

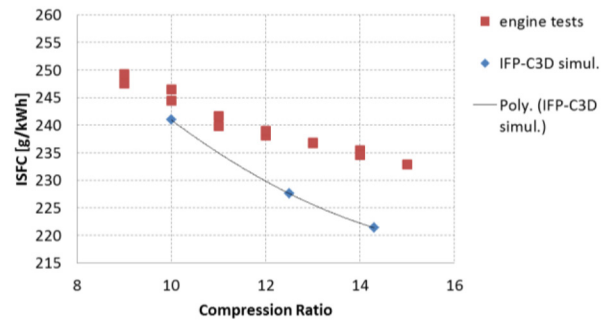


Figure 5. ISFC vs Compression Ratio @ 2000 rpm - 4 bar BMEP - comparison between IFP-C3D™ simulation with engine tests [4]

The first explanation could be an error on the wall heat flux modeling. Indeed, according to Boust [11], the wall flux could be underestimated with the Angelberger [12] wall law modeling used by IFP-C3D™. But the comparison of cylinder pressure in a logP-logV diagram at 2000rpm-2 bar in Figure 6 seems to validate the wall heat flux modeling (log(P) function of log(V) slopes), even if a more precise analysis is required to validate a wall heat flux modeling. This validation is out of the scope of this paper.

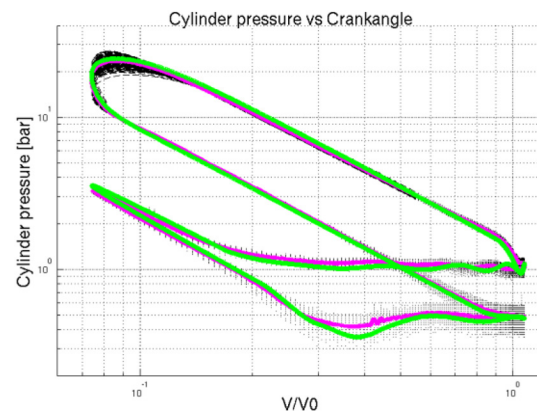


Figure 6. cylinder pressure comparison between IFP-C3D™ simulation and AVL test bench results [4] - log P log V - 2000 rpm - 2 bar BMEP - CR 15:1

The flame quenching model, the blow-by flow and the crevice volume are the only parameters identified that could explain the differences between simulation and experiment. The blow-by flow rate, which has been measured, can be neglected (<1% in mass flow rate) and the flame quenching is supposed to be well modeled and correlated in IFP-C3D™.

The last possible explanation is a crevice volume effect which is a common issue on high CR engines. The crevice volume is the sum of the "dead" volume between piston and cylinder liner above the first ring plus the volume between cylinder head, cylinder liner and cylinder head gasket. This crevice volume must be taken into consideration because of its impact on combustion efficiency: the unburned mixture trapped in the

crevice volume decreases the combustion efficiency (see Figure 27). And the higher the compression ratio is, the higher the combustion loss in the crevice volume will be. Therefore, the pressure cylinder differences and the behavior of ISFC on CR variations would be coherent with an effect of crevice volume. A crevice model, similar to [7], has been developed in IFP-C3D™. This additional model could overcome a part of the simulation error but the simulation results won't be presented here as it would need additional correlations.

Piston and Combustion Chamber Designs

Various designs have been considered, especially for the piston. Parametric models have been built in the following configurations:

- GDI cylinder head modified for squish pistons
- Squish pistons with spherical or flat bowl (similar to Toyota [5]) and convex pistons, simple or similar to Mazda piston with bowl [6]

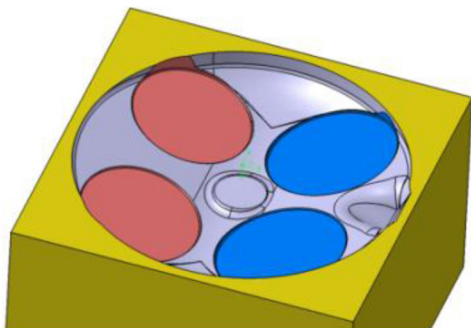


Figure 7. cylinder head geometry for 3D simulations

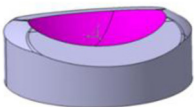
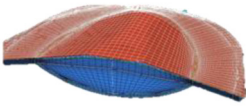
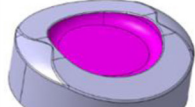
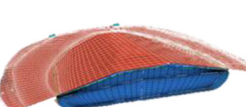
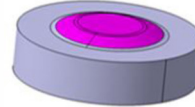
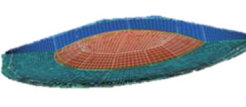
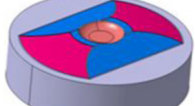
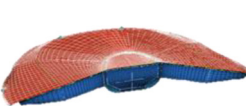
	Piston	Mesh @ TDC max CR
A		
B		
C		
D		

Figure 8. tested pistons and corresponding meshes at TDC

Design of Experiment

The design of experiment presented in Table 2 covers a large variety of engine speeds, loads, aerodynamic conditions and combustion chamber geometries. Each experiment is optimized to minimize ISFC with a spark advance variation.

Table 2. Design of experiment for 3D combustion simulations

	Input data	min	Max	Tested config.
Operating points	Engine Speed (rpm)	1500	2000	1500 / 2000
	Injected Fuel mass (mg/stk)	7	16	7 / 11 / 16
	Tumble	0	1.5	0 / 0.5 / 1.5
	Turbulence level	weak	strong	Weak/medium/strong
Valvetrain	Strategy/equipment			VVT only, EIVC+VVA, LIVC+VVA
Geom. with squish	half pentroof angle (°)	10	15	10 / 15
	Bowl diam. (mm)	35	55	35/45/55
	Sphere bowl radius (mm) [bowl depth (mm)]	20	50	20/35/50 [6-20]
	Squish height (mm)	1	4	1/2/4

Due to the high number of geometries of squish pistons (A and B), the geometries and meshes have been automatically created with ICEM-CFD™, with the Figure 9 parameters.

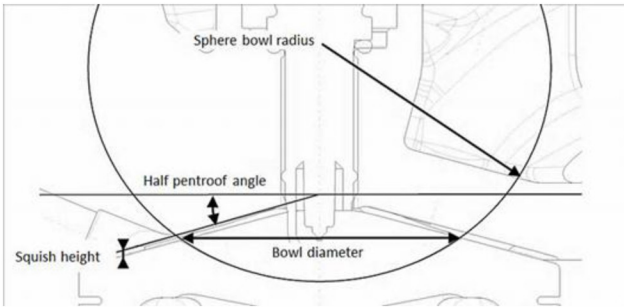


Figure 9. Squish piston combustion chamber geometries parameters to be varied in the DOE

A sufficient number of cells in the squish zone has been introduced in the vertical and radial directions to model the stretched shape of the front flame (see Figure 10 below).

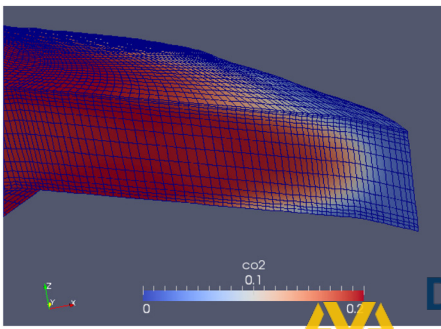


Figure 10. mesh in the squish area

Manual geometries and meshes have been constructed for convex pistons (C and D). These geometries are less sensitive to the mesh size than squish piston combustion chamber.

Simulation Initialization

The change of the exhaust and intake strokes with the CR is neglected (hypothesis). The change in the low pressure loop is indeed very low when compression ratio changes as shown in the Figure 11. The trapped air mass is also almost constant when CR changes in EIVC intake valve opening strategies. In addition, aerodynamics is supposed to remain unchanged when CR varies.

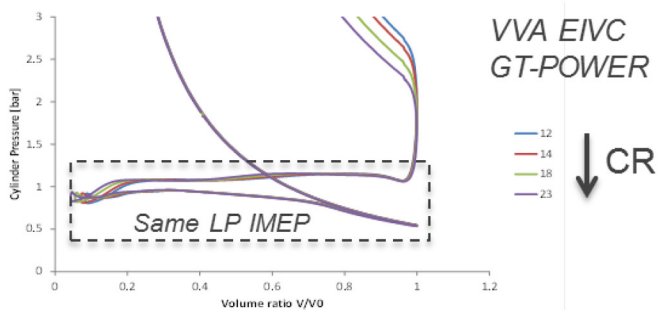


Figure 11. Low Pressure loop PV diagram in VVA-EIVC strategy for different geometric CR

Therefore, the compression-combustion-expansion simulations are initialized at the intake valve closing angle with the same exhaust-intake strokes simulation whatever the CR (see example result in Figure 12). Several exhaust-intake strokes simulations are made for each valve opening strategy and each engine speed / load operating point.

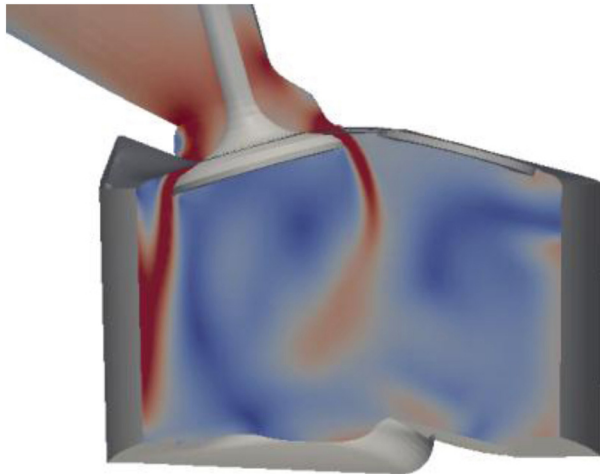


Figure 12. 3D simulation IFP-C3D™ example of intake stroke simulation - velocity magnitude

Simulation Results and Analysis

The IMEP, combustion duration MFB10-90, combustion efficiency and wall thermal losses (only of the compression-combustion-expansion part of the engine cycle) are calculated at their optimal spark advance/MFB50 (for a maximum IMEP

criterion). The ISFC @ constant IMEP is deduced with Willan slopes, dependent to the compression ratio (law extracted from 3D and 0D simulations), according to the equation:

$$ISFC(IMEP_{ref}) = ISFC(IMEP) \frac{IMEP}{IMEP_{ref}} + \frac{0.15 \times 755.25 \times CR^{-0.203} (IMEP_{ref} - IMEP)}{cyl_c \cdot IMEP_{ref}} \quad (1)$$

where cyl_c is the cylinder capacity (0.371) in L and CR the compression ratio.

The increase in thermal efficiency with CR compared to a CR 10:1 has been investigated for relevant operating points with different valve lift profile strategies (VVT strategy, VVA strategy with EIVC and LIVC) at the optimal MFB50. Several combustion chamber geometries have been examined.

With EIVC strategies (see example of the best geometries configurations in Figure 13), the benefit on thermal efficiency compared to CR 10:1 remains between 8 and 12% for CR above CR 18:1. The optimal compression ratio is close to 20:1.

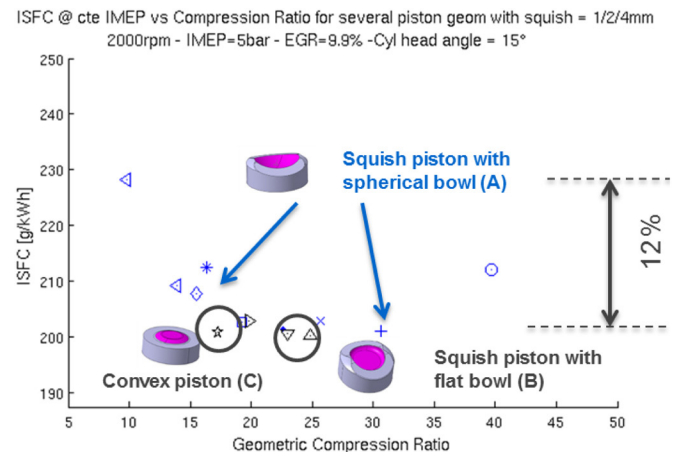


Figure 13. ISFC at 2000 rpm - 5 bar IMEP - EIVC valve lift

In case of squish bowl combustion chamber (A and B), the limitation of the thermal efficiency at high CR seems to be linked to the combustion efficiency decrease when CR increases. The wall heat transfer does not vary substantially when CR varies (see Figure 14). The slow combustion duration with such EIVC intake valve lift, which is a well-known issue [1,2], may increase the combustion losses at the high compression ratio. In other words, a lower ISFC is expected with a quicker combustion (higher turbulence for instance).

Note that the squish piston geometries have lower thermal losses but higher combustion losses compared to convex piston geometries (the black star symbol in Figure 14 corresponds the convex piston and blue symbols to bowl pistons).