

Cold Start on Diesel Engine: Is Low Compression Ratio Compatible with Cold Start Requirements?

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ABSTRACT

Future emission standards for Diesel engine will require a drastic reduction of engine-out NO_x emissions with very low level of particulate matter (PM), HC and CO, and keeping under control fuel consumption and combustion noise. One of the most promising way to reach this challenge is to reduce compression ratio (CR). A stringent limitation of reducing Diesel CR is currently cold start requirements. Indeed, reduction of ambient temperature leads to penalties in fuel vaporization and auto ignition capabilities, even more at very low temperature (-20 °C and below).

In this paper, we present the work operated on an HSDI Common rail Diesel 4-cyl engine in three area : engine tests till very low temperature (down to -25 °C) ; in cylinder imaging (videoscope) and CFD code development for cold start operation.

First, combustion chamber is adapted in order to reach low compression ratio (CR 13.7:1). Then, tests at low CR are compared with conventional CR on the same engine basis and results are discussed. After that, a complete recalibration of injection settings (fuel quantity and timing, injection pressure...) is operated. It allows to reduce significantly start delay with low CR, and reference start delay with conventional CR becomes reachable. Moreover, effects of combustion chamber design such as spray position vs glow plug are studied and show a great potential regarding behaviour in cold conditions notably in order to reduce again start delay in such conditions.

In addition to engine tests, CFD combustion calculations are performed during starting operation in ambient then cold conditions. Results are in accordance with optical observations and cylinder pressure data recorded on the engine. Correlations between tests and calculations give consistency explanations justifying cold start mechanisms and first order phenomena concerning vaporization and combustion.

INTRODUCTION

Diesel engine on passengers cars have known a great success during the last decade, due to dramatic progresses in technologies, such as injection system development, turbocharging, EGR capabilities, and combustion system evolution in accordance with these optimizations. The key issue for combustion system in warm stationary operation is to perform the best trade off between efficiency, raw emissions and noise levels at part load, and specific power and torque at full load. Consequently, volumetric Compression Ratio (CR) is currently becoming lower, in order to reach lower values of NO_x / PM release, taking into account HC and CO limits and keeping under control noise and fuel consumption [1]. In addition, at full load, higher specific power and torque may be obtained as for a same maximum peak cylinder. Then, whereas compression ratio values in the current application are between 17:1 and 19:1, lower compression ratio between 14:1 and 16:1 are currently studied for future applications. The main limitation for adopting low CR for mass production is currently cold operation, specially in extreme low temperature starting and running (-20 °C and lower).

As a result, the study described in this paper focuses on the consequences of a reduced CR from 17:1 to 13.7:1 on starting operations, and the optimization of the injection settings in order to allow starting with a delay in CR 13.7:1 near the start delay in CR 17:1. The mix of three tools - experimentation, calculation and visualization - is helpful in order to achieve the challenge consisting in reducing compression ratio of future Diesel engine. Engine experimentation brings a guideline of calibration of low CR engines. In cylinder visualizations highlight some consequences of vaporization penalties such as wall wetting. CFD calculations in cold start conditions, which are in accordance with engine experimentation and visualizations, bring consistent explanations of the vaporization and auto ignition process.

ENGINE EXPERIMENTAL SET UP

TEST ENGINE AND COMBUSTION SYSTEM DESIGN

The engine tests are performed on a 4 cylinder PSA engine DW10 ATED 4 with specifications mentioned in Table 1.

	Nominal compression ratio	Reduced compression ratio
Engine reference	PSA DW10 TED4	
Engine type	4 Cylinder, turbocharged, intercooler, direct injection	
Cylinder Head	4 valve / cylinder	
Bore x Stroke	85 mm x 88 mm	
Displacement	1998 cc	
Fuel injection system	Common Rail Bosch CR11 6 holes, spray angle 144° Static hydraulic flow 354 cm ³ / 30s/10MPa	
Volumetric compression ratio	17:1	13.7:1
Bowl width	nominal	nominal - 4,3 mm
Bowl depth	nominal	nominal - 1,7mm
Glow plug	Metallic self regulating type, BERU ref. 0 100 226 344 11V	

Table 1 - Cold start engine specifications

Instrumentation taking place in the combustion chamber makes slightly decrease the compression ratio (CR) compared with the production engine (17:1 instead of 18:1). This CR 17:1 configuration is called "nominal configuration" in the following parts. BERU metallic glow plugs are used for the whole tests of this study.

To reduce the compression ratio to 13.7 :1, piston bowl volume is enlarged, keeping constant the squish area (same distance piston - cylinder head). Particularly, injectors and nozzle are the same, so as the fuel spray takes exactly the same position as in the original engine configuration. To increase the volume of the bowl and decrease the compression ratio, a homothetic law drawing is adopted, in order to keep the general shape of the nominal bowl. Consequently, width and depth are larger (see Figure 1).

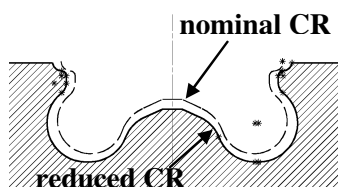


Figure 1 - Combustion system (bowl technical definition) in nominal CR (dotted line) and reduced CR (continuous line)

To check the accuracy of the injection system at very low temperature, specific tests have been operated first, on the injection system alone. While the engine is driven by a dyno, the complete injection system : HP pump, rail, tube and injectors (removed from the engine) is controlled by the ECU at various cartography breakpoints, representing a large range of injection pressure and injected quantity.

Injected quantities are measured and compared with the setpoints. A quite good correlation between targets and measurements is observed for main injection quantities (Figure 2); for example at -20°C, the gap between target

and measurement is less than 10%. For "pilot" injection (less than 20 mm³/str during starter operation), correlation is less acceptable, with a lack of injection quantity until about 25% compared to the setpoint. Moreover, no significant gap is observed between the injectors.

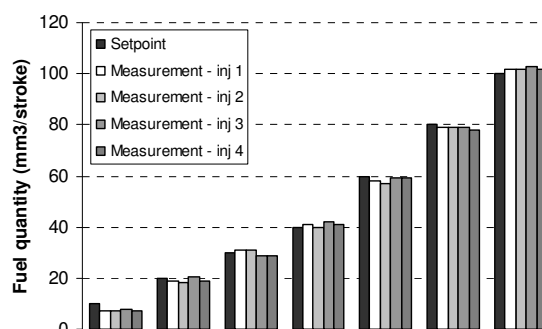


Figure 2 - Comparison in fuel injected quantity per stroke at -20°C, between setpoint and measurement for each injector, injection pressure 30 MPa

All the tests are performed with a gas oil able to support very low temperature (Futura City Diesel DIC -29/34 from Fortum Oil) and a 5W30 lubricant (Total Activa 9000).

TEST CELL AND MEASURING EQUIPMENT

The engine is installed in a cold and dehumidified room with ambient temperature that can be controlled from -30°C to +25°C. During engine running operation, ECU data acquisitions are performed by ETAS Inca system. Moreover, specific engine data are recorded such as : air, lubricant and water temperatures, temperature in the exhaust manifold close to ports, lambda sensor in the exhaust line, voltage and intensity of starter and glow plugs. Opacity is given by AVL 439 system in the exhaust line. In addition, an optical shaft encoder determines the engine crank angle inside the cycle (TDC notably). Each cylinder is instrumented with a quartz crystal pressure transducer Kistler 605BB to measure the gas pressure into the combustion chamber. This transducer is implemented with a canal machined in the cylinder head, but without pressure wave penalty for tested engine speeds. Moreover, injection pressure is measured upstream the injector. This allows to check the pressure waves at the nozzle opening and so to confirm that injection occurs as usual.

SOAKING AND STARTING PROCEDURES

The own electric starter of the engine – powered by the 12V battery – is used to crank the engine. Two starters of specific power may be used, depending on ambient temperature and required speed, mentioned in Table 2. No significant difference in starter speed was highlighted between nominal and low CR, even if acyclism is supposed to be lower with low CR . Engine is declutched during starting and warm up. Before a test, the engine

and the fuel tank are soaked in the cold test cell for about 8 hours under the required temperature. Because of higher thermal inertia, the battery is reloaded after each starting tests for a whole day in order to ensure maximum and repetitive cranking power. So three batteries are used alternatively for these tests.

Glow plugs of the production engine are used during all the tests. The preglowing duration is from 3s to 16s, respectively between ambient temperature from +20°C to -30°C. Starter is switched on at the end of this preglowing duration. In the paragraphs below, the start delay is evaluated between starter launching and ECU "startbit" signal appearance, meaning that idle speed has just been reached.

Ambient Temperature	Starter speed (rpm)	Preglowing duration (s)
20 °C	250	3
10 °C	230	3
0 °C	210	6
-10 °C	190	10
-20 °C	170	13
-25 °C	160	16

Table 2 - Starter speed and preglowing duration depending on ambient temperature

Lastly, all the tests are performed at least twice in order to check the repeatability of the results.

NUMERICAL SET UP

INTRODUCTION

The aim of the numerical part of the study is to bring information on the physical mechanisms occurring during the start of a Diesel engine with low compression ratio in low temperature conditions (-20 °C). Accurate prediction of auto ignition in the engine cycle or prediction of the engine start ability is not expected. A lot of parametric variations are performed and in addition, some Chemkin computations are done in order to evaluate the required conditions.

DESCRIPTION OF THE 3D CFD CODE

The three-dimensional simulations were performed using IFP-C3D code [2]. This one solves the unsteady equations of a chemically reactive mixture of gases, coupled with the equations for a multi-component vaporising fuel spray. This 3D CFD parallel (Open MP) code solves the Navier-Stokes equations using an ALE-extended (Arbitrary Lagrangian Eulerian) finite volume method on unstructured and conformal hexahedral meshes. The code uses the well known time splitting decomposition. The temporal integration scheme is largely implicit.

For turbulent combustion, the 3-Zone Extended Coherent Flame Model (ECFM3Z) is used [3]. The auto-ignition prediction and the influence of chemistry on reaction rates are provided by the TKI auto-ignition model [4], shortly described below. To obtain a realistic

spray modeling during the injector needle motion, the Transient Injection Conditions (TIC) model [5] is used.

SPRAY MODELING

The spray of the injector is discretized in parcels (set of droplets of same characteristics). The radius of the injected droplets is equal to the effective radius of injector hole. Turbulence model is k-ε and the Wave/FIPA model [6] is used to model the evaporation and the break-up. It may be specified that all the parameters of the injection model were fixed in order to get a good agreement with experimental measurements and visualizations performed in "normal" running conditions. For the present study (Cold start, low temperature) the parameters of this model were not modified due to a lack of experimental visualization for the present running conditions.

LIQUID FUEL FILM MODEL

The liquid fuel film on walls is modeled using an in-house lagrangian model based on the O'Rourke and Amsden [7], [8] model for the formation, the evaporation and the splashing. This model was improved using the work of Xu and Han [9], [10]. The modeling of liquid film dynamics takes into account the spray impact, the surface motions and the gas evolutions in the vicinity of the film.

The interaction with the spray induces, of course, mass evolution but changes in terms of energy and tangential momentum too. Moreover, submodels were introduced in order to take into account droplet rebound and splashing. The interaction between the liquid film and the walls depends on heat fluxes and adherence conditions. Fluid motion close to the film acts on it through heat fluxes and shear constraints. Finally the evaporation induce a decrease of film mass.

IGNITION AND COMBUSTION MODELING

For compression ignited internal combustion engines, the combustion modeling relies on the combination of two models. The ECFM-3Z model describes the turbulent mixing in each mesh cell and provides the local thermodynamical conditions and species concentrations to the TKI auto-ignition model. This one represents the chemical kinetics related to the combustion process. After the start of combustion, the representation of the turbulent mixing by the ECFM3Z model allows to define the local conditions in order to well represent the combustion progress.

The ECFM3Z model

A complete description of the ECFM3Z model is available in [3] for compression ignition operating conditions. This model allows to correctly represent the local mixing process occurring in the combustion chamber before the auto-ignition and during and after the combustion

The TKI Model

Auto-ignition is computed thanks to the TKI (Tabulated Kinetics of Ignition) model (more extensively described in [4]). It relies on tabulated auto-ignition quantities issued from detailed chemistry calculations performed with the Senkin code (part of the Chemkin package). Databases have been built for a wide range of thermodynamic conditions representative of cool flame ignition as well as main ignition using a detailed mechanism (416 species, 1992 reactions) for different n-heptane/iso-octane mixtures issued from the DCPR laboratory (Département de Chimie Physique des Réactions [11]). The 3D model variables are calculated once and for all for a large range of initial thermodynamic conditions and are stored in lookup tables. The fresh (or unburnt) gas quantities that are used as input parameters of the auto-ignition lookup tables are the temperature, the pressure, the Fuel-Air equivalence ratio, the rate of dilution and the octane number.

COMPUTATIONS ASSUMPTIONS

Several assumptions were fixed in order to decrease the computation time. The aim is to reduce the computational domain keeping a realistic level of precision.

Computational domain

Neglecting the valves and their recesses and taking into account that the multi hole injector is centered (the glow plug consideration is exposed lower) allows to assume that the combustion chamber is not really axi-symmetric but possesses periodic characteristics. In most of cases this assumption allows to consider a computational domain representing only an angular sector of the chamber corresponding to the area concerning only one spray of the injector.

Here, for cold start consideration the glow plug have to be taken into account. Making the assumption that the influence of the plug is confined to its direct neighborhood (heating due to shining is neglected), added to the fact that the aim of the present study is to understand the mechanisms inducing the first combustion allow to still consider a periodic combustion chamber: in fact the modeled chamber will possess several glow plugs. In order to better represent the location of the plug versus the sprays, an angular sector corresponding to the area concerning 2 sprays is considered. As the injector possesses 6 holes, the angle of the combustion chamber sector is equal to 120° . The computational mesh is shown in Figure 3. It can be noticed that the plug is represented by a specific wall with a given temperature (about 1000°C) that can be varied. The meshes were realized using Icem-CFD Hexa. Their size varies between about 52000 (start of computation) and 41000 hexas (during injection and start of combustion). The changes in mesh size are performed using remapping algorithm.

Blow-by and compression ratio

The piston shape was specifically designed in order to obtain a combustion chamber with a nominal compression ratio of 13.7:1. In order to "take into account" the important blow-by existing during the first engine cycle in cold start conditions and as no blow-by model was available at the beginning of this study, the nominal compression ratio used for computations was reduced to 12.5:1 (equivalent value deduced from 1D computations) by adding a crown at the outlying of the computational domain in order to reproduce experimental in-cylinder pressure trace.

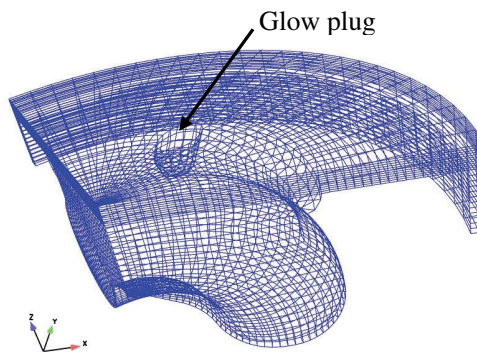


Figure 3 - Mesh of the computational domain

This mean to mimic blow-by allows to keep the original distance between cylinder head and piston and the additional fresh air in the chamber (in the crown) is not located in the area of spray and will not influence the local fuel air equivalence ratio in the bowl and close to the plug. But this methodology, even if it allows to well represent the loss of in-cylinder pressure at the Top Dead Centre due to blow-by may not allow to well represent the mean in-cylinder temperature. Indeed, the air mass heated during compression is larger in computations than in experiments. Moreover, the heat fluxes between cold walls and fluid is larger due to the increase of cylinder liner area in the crown. This is a first step for modelling and a blow-by model is under development in order to increase the accuracy of the representation.

Fluid dynamics and injection

The intensity of fluid motion in the cylinder is supposed low. The filling ratio is supposed to be equal to 1. However, in the reality the pressure drop due to the inertia of the air in all the intake manifold may induce a lower filling ratio (around 0.95). The computation starts at the Bottom Dead Center (BDC). The initial pressure is fixed at 0,1 MPa and the initial swirl motion is set equal to 0.2 for an engine speed of 170 RPM. Specifications of the injector are the same as for engine tests (Bosch CRI1 with spray angle equal to about 144° and 6 holes). The instantaneous flow rate is fitted on experimental data. The injected fuel is a single component model possessing almost the same characteristics as experimental fuel. Computations were performed with

same injected mass as experiments (small differences in terms of injected volume of fuel). Even if the injected energy is not really the same as experiments (very small difference in terms of Fuel/Air equivalence ratio) it allows to well represent the first steps of cold start and the different phenomena involved in such conditions.

Initial and boundary conditions

As said above, the computation starts at BDC. At this time, the chamber is filled with pure fresh air at the external temperature (-20°C). The piston, cylinder head and cylinder liner temperatures are similar. Only the walls of glow plug is heated at a temperature of about 1000°C .

The engine is supposed to be cranked by the starter at 170 rpm. During the modelled first cycle, this engine speed is supposed to be constant even if a combustion occurs. It may be specified that here the first engine cycle is modelled, the walls and the fluids (air and fuel) are supposed to be at external nominal temperature and the chamber is filled with pure air. In the reality the first combustion occurs in general during the second or third engine cycle where the walls may have been heated (first compression and friction) and it may be some fuel in the chamber from the previous unburnt cycles. Indeed as it is presented below, during the first cycle a lot of fuel is trapped in the liquid film on the piston walls. This fuel is not available during the first cycle but the evolution of the liquid film during the exhaust phase of the first cycle is unknown. As a consequence, a future work will consist in the computation of the two first cycles in order to give indications on the initial conditions of the second engine cycle, concerning trapped fuel mass (liquid and gas) remaining from previous cycle.

OPTICAL SET UP

In order to better understand phenomena as fuel wall wetting during cranking operation in low compression ratio (CR 13.7:1), a camera "videoscope" (AVL Pulnix TMC-9700) is used. The frequency of shots acquisition is 10 Hz ; thus this type of camera enables only one shot per engine cycle at starter speed. Connected with the optical shaft encoder mentioned above, timing of the shot can be defined in the cycle. Moreover, as the camera is non intensified, a stroboscope (supplier : EG&G) has to be used as a source of light to illuminate the combustion chamber with the good timing.

For these optical tests, a specific cylinder head was installed on the engine. This is the same as a the production DW10 cylinder head – especially : injector and swirl - but :

- two entrances have been machined one under the other, in one cylinder (cylinder n°3), in order to receive as preferred the endoscope or the source of light. In the tests presented in this paper, the source of light is located in the top entrance while the endoscope is located in the bottom one ;

- as this cylinder head is a prototype one, no glow plug is mounted in any cylinder. So combustion is degraded in cold conditions.

In this configuration, the piston re-entrant of the cylinder n°3 has been machined to facilitate optical entries when piston is near TDC. In order to keep constant the clearance volume value of this cylinder, bowl volume was slightly reduced by an homothetic law. As a result, all cylinders have the same compression ratio 13.7:1. This set up is illustrated in Figure 4.

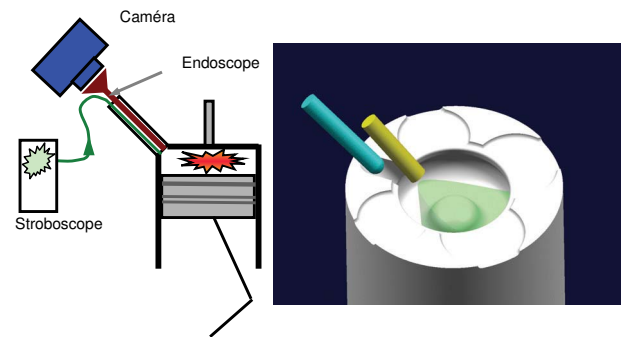


Figure 4 - Optical set up with adapted cylinder head

KEY ISSUES IN COLD OPERATION FOR LOW COMPRESSION ENGINE

VAPORIZATION AND ESTIMATION OF CYLINDER GAS TEMPERATURE

Compared with standard compression ratio, the main problem with low compression ratio is the reduction of pressure (see Figure 5) and consequently the reduction of cylinder temperature during the compression stroke and then at top dead center (TDC). This reduction of temperature causes penalties in vaporization process. So it is relevant to estimate TDC gas temperature in low CR 13.7:1 and compare it at the TDC gas temperature in nominal CR 17:1.

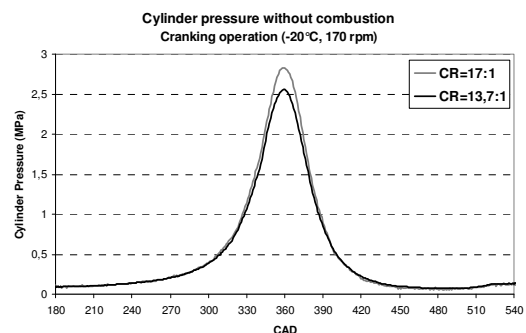


Figure 5 - Cylinder pressure during compression for compression ratios 17:1 and 13.7:1 (at -20°C). The first complete cycle during cranking operation (with injection and without combustion) is represented.

To estimate the TDC temperature (with only air charge), a simplified DW10 engine model is built with AMESim – IFP Engine 0-D code developed by IMAGINE and IFP. The first cycle during cranking phase is operated at 170

rpm for an ambient temperature of -20°C. This model takes into account the main phenomena involved in cold start operation to estimate approximately temperature of the compressed gas : wall heat losses, reduction of air filling due to the unadapted distribution diagram at very low engine speed, and blow by leakage [12].

Wall heat losses

To estimate the heat losses during pure compression (so without combustion) Q_{losses} , Woschni correlation has been used and is summarized by Equations (1) to (3) below :

$$A_1 = C_1 \cdot V_p \quad (1)$$

$$h_{conv} = 130 \cdot \frac{(p \cdot A_1)^{0,8}}{T^{0,53} \cdot B^{0,2}} \quad (2)$$

$$Q_{losses} = \int \sum_{cycle\ surfaces} h_{conv} \cdot S_i \cdot (T - T_i) \cdot d\theta \quad (3)$$

with : V_p : Average speed of the piston
 C_1 : Woschni heat exchange coefficient
 T, p : Respectively gas temperature and gas pressure
 B : Cylinder bore
 S_i, T_i : Respectively surface and temperature of various surfaces of the combustion chamber

For the first compression cycle during starter operation, cylinder and piston walls are approximately at the soaking temperature $T_i = -20^\circ\text{C}$. Because the starter speed is much lower (~170 rpm) than during "conventional" combustion engine running, coefficient C_1 has to be fitted.

Specifically to estimate C_1 coefficient, engine tests were performed at ambient temperature + 20°C on a single cylinder based on PSA DW10 ATED 4 geometry equipped with a specific measurement of blow by, at very low engine speed. In the same time, CFD calculations of the cooling circuit were performed on the same engine basis and allowed to know the temperature of the various walls of the combustion chamber for the same operating point. These tests and these calculations, performed at ambient temperature, are not described in details in this paper. By using equations (1) to (3) with the estimation of temperature walls (by CFD calculation), air filling, and blow by (by the engine tests) in this single cylinder engine operation, Woschni coefficient C_1 was estimated to around 5 in ambient conditions. As this C_1 coefficient is governed above all by the gas movement in the proximity of cylinder walls – and then by piston speed and swirl – and *not* by the gas temperature, the same value of $C_1 = 5$ was supposed to be approximately the same in warm and cold conditions at the same engine speed and it was retained for the cold starting phase on the multi cylinder tests.

Air filling and effective compression ratio

The simplified DW10 engine model developed with AMESim – IFP Engine code was used to estimate air filling during starter operation. Indeed, due to very low engine speed, air filling is penalised by the leakage due to the Late Intake Valve Closing (IVC) : cylinder air is leaving the cylinder during the beginning of the compression (see Figure 6).

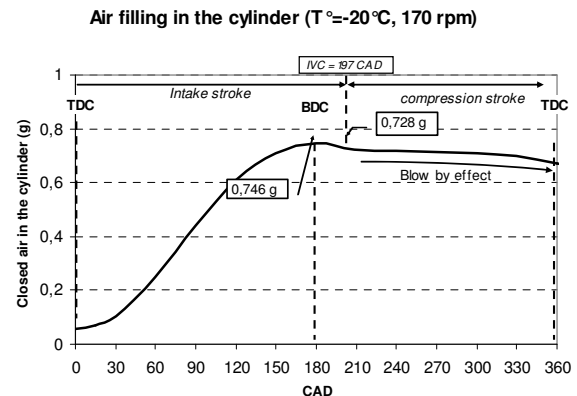


Figure 6 - Air filling profile during intake and compression for CR 13.7:1. Ambient temperature is -20°C

As a result, for -20°C operation point at 170 rpm, maximum air quantity is 746 mg at the end of the compression stroke but only 728 mg at the IVC. This causes a decrease of effective compression ratio and then in temperature at TDC when combustion is supposed to occur.

Blow-by

With air filling and wall heat losses previously estimated, blow by is calculated in cold condition during starting operation. Blow by is implemented in AMESim – IFP Engine 0-D code as a leakage surface. This surface is fitted in order to equalize the calculated pressure by AMESim and the measured pressure in the engine tests during starting operation (170 rpm). As a result, blow-by process is estimated to assume a lack in cylinder pressure of almost 0,35 MPa and a lack in cylinder temperature of 24K at TDC for the engine test at -20°C.

Then, cylinder peak temperature calculation gives about 555 K for the engine test with CR 13.7:1 at 170 rpm and ambient temperature -20°C. With the same methodology and the same operating point, the value for the CR 17:1 is near 587 K. This temperature reduction of around 30 to 35 K causes penalties for complying a complete fuel vaporization by heat transfer between air charge and fuel droplets ; as a result, the global vaporized fuel rate is decreasing, and conditions for successful combustion are all the more degraded. Moreover, the reduction of temperature and pressure at TDC with reduced CR tends to increase ignition delay [12], so as to be not compatible with starting operation and to require further