

Clean Diesel Combustion by Means of the HCPC Concept

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Ettore Musu, Riccardo Rossi and Roberto Gentili
Università di Pisa

Rolf D. Reitz
University of Wisconsin-Madison

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ABSTRACT

Homogeneous-charge, compression-ignition (HCCI) combustion is triggered by spontaneous ignition in dilute homogeneous mixtures. The combustion rate must be reduced by suitable solutions such as high rates of Exhaust Gas Recirculation (EGR) and/or lean mixtures. HCCI is considered a very effective way to reduce engine pollutant emissions, however only a few HCCI engines have entered into production. HCCI combustion currently cannot be extended to the whole engine operating range, especially to high loads, since the use of EGR displaces air from the cylinder, limiting engine mean effective pressure, thus the engine must be able to operate also in conventional mode.

This paper concerns a study of an innovative concept to control HCCI combustion in diesel-fuelled engines. The concept consists in forming a pre-compressed homogeneous charge outside the cylinder and gradually admitting it into the cylinder during the combustion process. In this way, combustion can be controlled by the transfer flow rate and high pressure rise rates, typical of standard HCCI combustion, can be avoided. This new combustion concept was called Homogenous Charge Progressive Combustion (HCPC).

The paper illustrates a CFD analysis focused on improving efficiency and reducing pollutant emissions at medium and heavy load. Different geometries of the transfer duct were considered. Results show negligible soot emission up to equivalence ratios around 0.85, with indicated efficiency around 46 %. As well, a moderate level of external cooled EGR allows reducing NO_x emissions up to levels that are typical of low-temperature combustion.

INTRODUCTION

Homogeneous-charge, compression-ignition (HCCI) combustion takes place in homogeneous mixtures once the spontaneous combustion temperature and the latency time are exceeded. However the reactions take place more gradually than in explosive combustion, since lean mixtures are used together with high levels of charge dilution by exhaust gas. Yet, the admission of exhaust gas into the cylinder negatively impacts the mean effective pressure the engine can reach.

HCCI is considered a very beneficial solution to reduce engine pollutant emissions. So far, various experimental prototypes and a few production engines have been inspired by this concept and much research is in progress.

HCCI is feasible with fuels both for S.I. and for C.I. engines, but it still does not cover the whole engine operating range, thus the engine must still be built to operate basically as a conventional one.

In engines suitable for S.I. fuels, a premixed homogeneous charge is formed, as in normal S.I. engines, but combustion takes place due to spontaneous auto ignition, as in C.I. engines, after a critical temperature and latency time are exceeded. To control the combustion rate, charge dilution with exhaust gas, sometimes together with a lean mixture, is applied. Clear advantages in thermal efficiency due to less decrease of the specific heat ratio, as well as reduction of dissociation, and of thermal exchange with the cylinder walls occur. Also the pumping work is reduced, due to decreased intake throttling. NO_x emissions take advantage from the replacement of the high temperature flame front, typical of S.I. engines, with a combustion process that extends to the

whole combustion chamber volume and from the presence of a diluted charge.

For engines operating with diesel fuels, outside the HCCI operating range, the engine must operate with diffusive combustion as a normal diesel engine, thus usually it must keep the basic characteristics of diesel engines. The most important advantage of HCCI combustion compared to the diffusive one is a remarkable reduction of soot and NOx emissions. This occurs because HCCI combustion avoids the presence in the combustion chamber both of excessively fuel-rich zones and of local temperature peaks, which are responsible for soot and NOx formation, respectively.

Since HCCI is triggered by homogeneous charge spontaneous ignition during compression, the control of combustion timing and development is not easy (in conventional S.I. and in diesel engines the control is respectively achieved by spark timing and injection timing and rate). Therefore, high peaks in the heat release profile can occur, leading to high pressure gradients, which are dangerous for the engine structure.

A basic problem to solve is to extend HCCI combustion feasibility to all engine operating conditions (or at least to a wide operating range), including the most difficult situations (heavy loads and transients). So far many efforts to control HCCI combustion, to extend its operating range and to minimize pollutant emissions, have been made [1]. Solutions have been proposed that consist of: charge pre-heating [2, 3], control of charge turbulence and heterogeneity [4, 5], variation of compression ratio [6, 7], EGR modulation [8, 9]. Water injection has also been considered for extending HCCI towards heavy loads [10, 11]. Some studies concern the use of special fuel additives [12] or of two different fuels [13, 14, 15]. Split-PCI combustion with a two-injection strategy was used by Okude et al. [16] at full load. Recent experiments by Bessonette et al. [17] were able to extend the HCCI operating range up to 1.6 MPa BMEP, using a proper fuel blend. Boost pressure were also considered [18, 19] for high load HCCI.

Current solutions for diesel HCCI can be distinguished into three main categories according to fuel introduction technique: 1) premixed HCCI [20, 21], in which the fuel is dispersed into the air before the intake process; 2) early direct injection HCCI [11, 22], in which the fuel is injected sufficiently in advance with respect to Top Dead Center (TDC) to allow mixing fulfilment before the auto-ignition process; 3) late direct injection HCCI [23, 24], in which the fuel is injected later than in conventional diesel combustion to delay the ignition and the mixing is promoted by high levels of turbulence and swirl.

All these solutions need the presence of a diluent, normally exhaust gas, to control the combustion. As previously noticed, the admission of exhaust gas into the cylinder, at the expense of air, implies that HCCI combustion is feasible only

with modest mean effective pressures. Overcoming this obstacle by increasing the boost pressure beyond a certain level is impractical, since the mechanical efficiency would decrease excessively because of the friction increase due both to the direct effect of the increased gas pressure, and to the necessity of larger reciprocating masses to face such pressure level.

AIMS OF THE STUDY

This paper concerns an innovative method to control HCCI combustion behaviour with diesel fuels without relying on exhaust gas recirculation. Thus, HCCI combustion can be attained even with high mean effective pressures and, as needed, the amount of EGR compatible with the engine load can be used for the purpose of limiting NOx emissions.

The method consists in forming a pre-compressed homogeneous mixture outside the cylinder and gradually transferring the charge into the cylinder during the engine combustion phase, so that combustion control is delegated to the transfer process.

Since the charge is progressively admitted into the combustion chamber, this combustion method was called Homogenous Charge Progressive Combustion (HCPC). A preliminary study of this concept was presented in 2007 [25], then in 2009 [26] a CFD study demonstrated the validity of the concept. In a previous paper [27] a new HCPC engine scheme was investigated and CFD results showed around 45% indicated efficiency together with a consistent reduction of soot emissions compared to conventional diesel engines.

HCPC is based on split-cycle concept, consisting in performing the intake and compression phases outside the cylinder. Recently the split-cycle engine idea has been reintroduced mainly by the Helsinki University of Technology for a diesel engine [28-29], by the Scuderi Group for a S.I. engine [30] and by the Innogy Company for a large power-plant gas engine [31].

In the HCPC concept the compression is performed by a reciprocating external compressor which accomplishes the same compression work that in a four stroke engine is provided by the engine piston. The high-temperature compressed air is transferred to the combustor cylinder during the engine combustion phase. Contemporary with the air transfer, fuel is injected into the transfer duct, evaporates and mixes with the air, bringing about the conditions needed for homogeneous combustion. Compressor work recovery takes place during the piston expansion stroke, due to the engine work produced by the transferred air [26].

In this work a CFD analysis focused on improving efficiency and reducing pollutant emissions of the HCPC engine at medium and heavy load was performed. Starting from the

engine scheme shown by Musu et al. [27] (Fig.1) different transfer duct geometries were considered to reduce soot emissions at high loads. As well, the use of external cooled EGR was investigated in order to reduce NOx and a study on wall heat losses was performed in order to improve engine thermal efficiency.

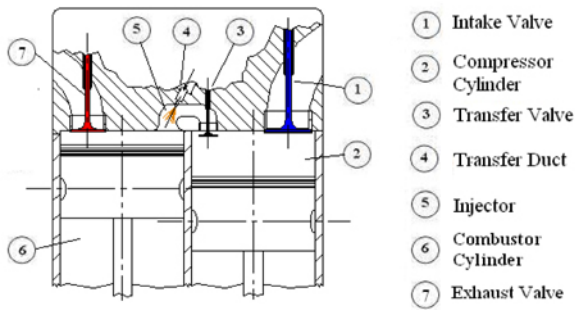


Fig. 1. HCPC engine scheme

CFD SIMULATION SETUP

The study was conducted by means of the AVL FIRE code [32] using the ECFM-3Z model [33] for the combustion. For the injection process, the Discrete Droplet Model (DDM) [34] was employed together with the KH-RT model [35] and the Dukowicz evaporation model [36]. The turbulence model based on the RANS (Reynolds Average Navier Stokes) method is the two-equation K- ζ -f model, which takes the effects of turbulence in the mean flow into account. As regards the emissions, the Zeldovich and Kennedy-Hiroyasu-Magnussen models have been used to predict NOx and soot production respectively.

TRANSFER DUCT GEOMETRY STUDY

In Fig. 2 the three transfer duct geometries considered in the study are shown. Starting from the first is the one of the previous work [27] the transfer duct volume has been reduced from 9.1 cm³ to 6.6 cm³ for geometry 2 and 4.8 cm³ for geometry 3. In geometry 3 the transfer valve has been inclined by 25° and the compressor dead volume was lowered thanks to a specifically designed piston. Decreasing the compressor dead volume the compressor expansion is performed in a shorter crank angle phase, therefore the inlet valves can be opened earlier, increasing the trapped air mass and therefore the specific power.

The simulations were made at 2000 rpm starting from the transfer valve opening (TVO) to the EVO and therefore the calculation domains do not include intake and exhaust ports (Table 1). Two cases with different fuel amounts (16 and 22 mg), were considered using a single hole injector. Injection

boundary conditions are summarized in Table 2, and other geometrical details and parameters are summarized next.

The emission results in Figs. 4 and 5 show that reducing the transfer duct volume leads to a noticeable reduction of CO and soot emission. Temperature maps in Fig. 6 show that by lowering the transfer duct volume combustion tends to propagate in the entire chamber. Figure 7 shows in fact that a significant part of the oxygen is not used with geometries 1 and 2 and is more stratified. In geometry 3, instead, the oxygen concentration is much more uniform, leading to a better soot and CO oxidation. The third geometry was therefore chosen.

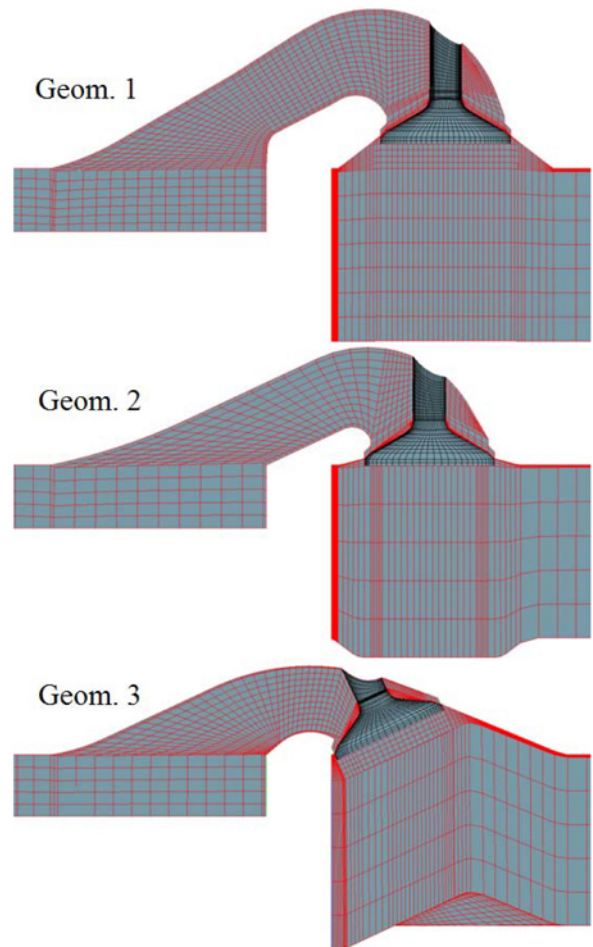


Fig. 2. Transfer duct geometries considered in the study

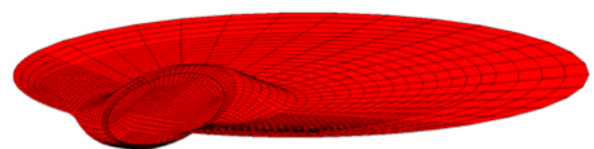


Fig. 3. compressor piston of geometry 3

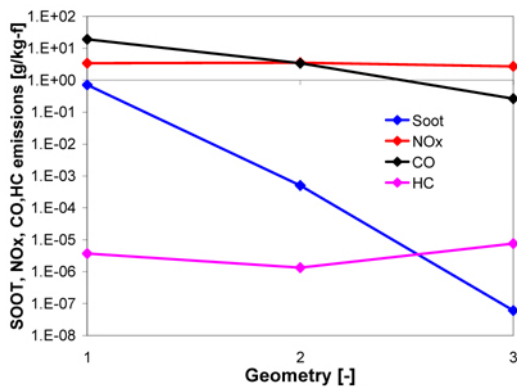


Fig. 4. Emission results for the different geometries for the case 16mg

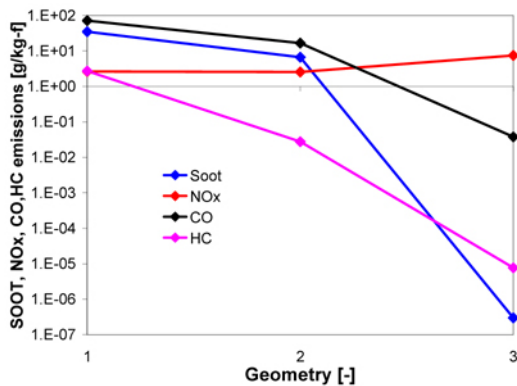


Fig. 5. Emission results for the different geometries for the case 22mg

Table 1. simulation parameters

Engine speed	2000 rpm
Combustor Initial T, P	820 K, 350 kPa
Compressor Initial T, P	520 K, 500 kPa
Simulation Start (TVO)	330° ATDC
Simulation End (EVO)	525° ATDC

Table 2. Injection boundary condition

Injection Type	Transfer Duct Injection
Fuel	C14H30
Fuel Temperature	363 K
SOI (Pilot injection)	-8° ATDC
EOI (Pilot injection)	-6° ATDC
SOI (Main injection)	0° ATDC
EOI (Pilot injection)	20° ATDC
Injector	Single-Hole
Fuel Injected	16, 22 mg
Nozzle Diameter	257 μm
Cone Angle	22°
Injection rate	Square profile

<figure 6 here>

<figure 7 here>

RESULTS AT 2000 RPM

In this paragraph results of the complete cycle simulation are presented. The fluid domain (Fig. 8) includes the compressor, with the intake valves and ducts, and the combustor with the exhaust valves and ducts. From the picture some design features of the HCPC engine can be seen. The intake and exhaust valves (2 valves of 38 mm for both the 86 mm cylinder bore) are much bigger than the ones of a similar-displacement engine, with advantages in volumetric efficiency.

Pressure is assumed to be kept constant at the inlet and the outlet of the fluid domain. The initial conditions were calculated by means of a one-dimensional model (Fig. 9) realized with the AVL BOOST code. The same code was used to define various parameters such as the intake, transfer and exhaust valve lifts and timings, the intake and exhaust duct lengths, the cylinder-compressor crank-angle phasing and the boundary conditions for the EGR study. The engine specifications obtained are summarized in Table 3 and 4.

The valve lift curves are shown in Fig. 10 (CA is always referred to the combustor TDC). In order to have wider cam phasing, an unconventional valve design with an additional cylindrical seal [26, 29] was used for all the engine valves. With this additional seal, the valve remains closed, apart from the leakage through the cylindrical seal clearance, during the first part of the valve lift (2 mm for intake and exhaust valves, 1 mm for the transfer valve), and hence valve motion begins before the effective valve opening.

3D calculations were started at Effective TVO, when the transfer valve clears its cylindrical seal and consider the

combustor and compressor complete cycle. Two complete cycles were calculated in order to reduce the dependency of the results on the initial conditions [27].

Six equivalence ratios from 0.4 to 0.96 were considered by injecting 16 to 27 mg of fuel each cycle. The initial conditions are summarized in Table 5. A pilot injection was used to achieve stable ignition timing. A 7 hole injector was used and it is located in the transfer duct, as shown in Fig. 11. The spray boundary conditions used in the simulations are summarized in Table 6. Computational grids were generated by using the ICEM-CFD software and contain about 85,000 cells at TDC (half model).

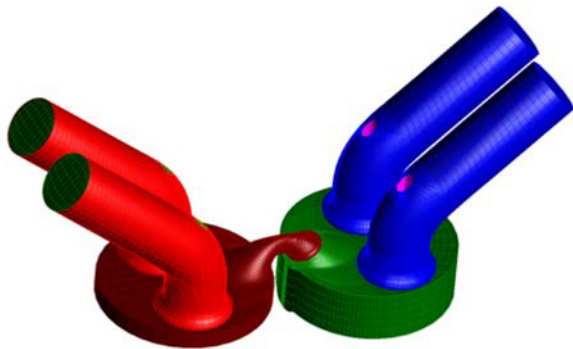


Fig. 8. HCPC fluid domain at TVO (30° BTDC)

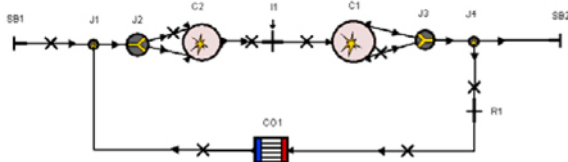


Fig. 9. One dimensional model

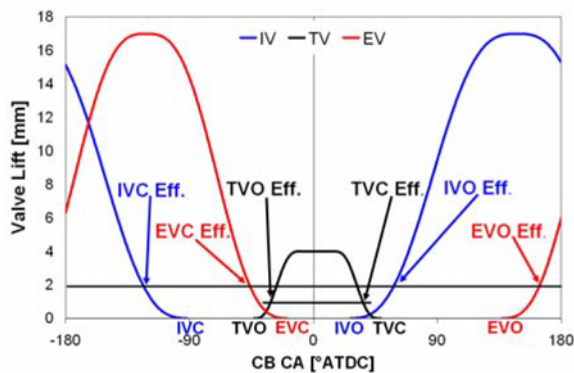


Fig. 10. Valve lift curves

Table 3. Compressor specification

Displacement	499 cm ³
Bore	86 mm
Stroke	86 mm
Squish Height	0.5 mm
Geometric Compression ratio	117:1
IVO	35° ATDC
IVO Effective	60° ATDC
IVC	-101° ATDC
IVC Effective	-126° ATDC
TVO	-30° ATDC
TVO Effective	-47° ATDC
TVC	35 ATDC
TVC Effective	52 ATDC

Table 4. Combustor specification

Displacement	598 cm ³
Bore	86 mm
Stroke	103 mm
Squish Height	0.5 mm
Geometric Compression ratio	85:1
EVO	138° ATDC
EVO Effective	165° ATDC
EVC	-36° ATDC
EVC Effective	-9° ATDC

Tab. 5. Initial and boundary condition

Engine speed	2000rpm
Inlet T, P	323 K, 100kPa
Outlet P	100kPa
Combustor Initial T, P	750-850 K, 280-350 kPa
Compressor Initial T, P	480-530 K, 480-520 kPa
Simulation Start	330° ATDC
Simulation End	1050° ATDC (2° cycle)

Table 6. Injection boundary conditions

Injection Type	Transfer Duct Injection
Fuel	C14H30
Fuel Temperature	363 K
SOI (Pilot injection)	-8° ATDC
EOI (Pilot injection)	-6° ATDC
SOI (Main injection)	0° ATDC
EOI (Pilot injection)	20° ATDC
Injector	7-Hole
Fuel Injected	12-27 mg
Nozzle Diameter	140 μm
Cone Angle	12°
Injection rate	Square profile

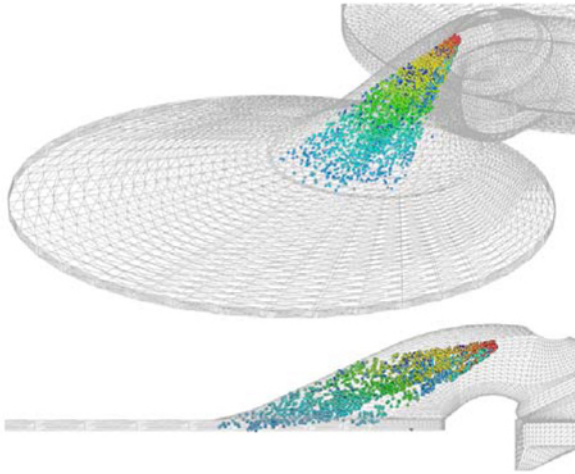


Fig. 11. Injector position

The performance indicated data were derived from the indicated work (Fig. 12) calculated with Equation 1. The combustor and compressor work were obtained integrating the CFD pressure results in the discretized form of Equations 2 and 3.

$$\text{Indicated_Work} = W_{CB} + W_{CP} \quad (1)$$

Where:

$$W_{CB} = \oint p_{CB} dV \approx \sum_{n=0}^N 0.5 \cdot [(p_{n_{CB}} + p_{n-1_{CB}}) \cdot (V_{n_{CB}} - V_{n-1_{CB}})] \quad (2)$$

$$W_{CP} = \oint p_{CP} dV \approx \sum_{n=0}^N 0.5 \cdot [(p_{n_{CP}} + p_{n-1_{CP}}) \cdot (V_{n_{CP}} - V_{n-1_{CP}})] \quad (3)$$

In Fig. 13 indicated power and efficiency vs. equivalence ratio are given. Thermal efficiency is predicted to be very promising reaching 46% between 0.55 and 0.85. The naturally aspirated HCPC unit consisting of a 600cc combustor and a 500 cc compressor produces more than 16kW at 2000 RPM leading to a PMI of 1.9 MPa referred to the combustor displacement. Further benefits in terms of efficiency can be obtained using a turbocharger. NOx emissions are in the same order of magnitude of a diesel engine and decrease with decreasing oxygen partial pressure that is inversely proportional to the equivalence ratio (Fig. 14). Soot (Fig. 14), HC and CO (Fig. 15) emission are very low up to equivalence ratio of 0.85.

Pressure traces and HRR are shown in Fig. 16. A smooth pressure rise that occurs almost linearly without high gradients with a PPRR that is around 0.3 MPa per degree can be noticed, with benefits for engine noise and vibrations.

CFD results for the equivalence ratio and temperature distributions during combustion (Fig. 17) show that diffusion flames are limited in extent and most of the combustion takes place mainly where the equivalence ratio is higher, generating volumetric heat release characteristics instead of diffusion flames. Therefore the combustion takes place progressively under almost homogeneous charge conditions while the charge is admitted into the combustor chamber. The equivalence ratio maps were obtained using the definition in Equation 4 [37].

$$\Phi = \frac{2 \cdot \sum_{i=1}^n C_i^{\#} + \frac{1}{2} \cdot \sum_{i=1}^n H_i^{\#}}{\sum_{i=1}^n O_i^{\#}} \quad (4)$$

Where $C_i^{\#}$, $H_i^{\#}$, $O_i^{\#}$ are the numbers of atoms of the i^{th} species and n is the total number of species in the calculation domain.