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A numerical and experimental study of dual fuel diesel engine for different injection timings

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A B S T R A C T

The dual-fuel technology has the potential to offer significant improvements in the CO_2 emissions from light-duty compression ignition engines, and in this sense, such a concept represents a viable solution to reduce emissions from diesel engines by using natural gas as an alternative fuel. In light duty, high-speed engines, where the combustion event can be temporally shorter, the injection timing plays an important role on the performance and emissions. Following a methodology that has been proposed in previous authors' papers, this article summarizes the results of a combined numerical and experimental study on the effect of injection timing on performance and pollutant fractions of a common rail diesel engine supplied with natural gas and diesel oil. The study of dual-fuel engine carried out in this paper aims at the evaluation of the CFD potential to predict the main features of this particular technology. The experimental investigations allow the validation of the CFD modeling and, at the same time, highlight the major aspects that arise from the actual engine operation with different diesel injection advances. The fluid-dynamic calculations are extremely useful to put into evidence the key phenomena that take place during the dual-fuel operation, say the typical flame propagation throughout the premixed methane–air medium that is activated by the early self-ignition of the diesel fuel. While previous papers have discussed the effect of the NG/diesel fuel ratio, this paper focuses the attention on different conditions that have been induced by varying the injection timing at fixed brake torque and diesel amount. Actually, the start of injection timing can considerably influence the combustion development and therefore THC and NOx fractions production. Calculations making use of the fluent code have been compared with the experimental data and a comparison between full diesel and dual fuel operation has been made, in terms of performance and pollutant levels.

1. Introduction and state of art

Dual-fuel engine (diesel/NG) represents a possible solution to reduce emissions from diesel engine by using a natural gas mixture as an alternative fuel. The dual-fuel engines are supplied with both natural gas and diesel fuel simultaneously, but the most of fuel burned is natural gas. Diesel fuel acts essentially as a "spread spark plug" as it self-ignites after compression and then aids the ignition of the natural gas – air mixture, whose equivalence ratio is close to the lower flammability limits.

The idea of the dual fuel mode is to exploit the favorable characteristics of natural gas by keeping the diesel engine compression ratio and its efficiency level. The NG could be responsible not only for lower NO_x emissions but also for a reduction of CO_2 formation, since it is the fossil fuel with the lowest C/H ratio.

An improvement of the dual fuel engine performance can be obtained especially if the engine is equipped with an exhaust gas recirculation (EGR) system for aiding the development of a smoother process; actually if considering the high octane number of NG [1,2], in high pressure gradients due to a quick combustion with a high risk of knock due to end gas, autoignition should be avoided in some operating conditions. In addition, at low loads, the recirculation of part of the hot exhaust gas allows an improvement of the combustion due both to the presence of radicals and to the temperature increase [3].

Several authors are investigating on dual fuel mode in a diesel engine, by assuming natural gas as main fuel and varying the chief engine parameters to define the best operating condition in terms of diesel injection law [4], diesel/NG ratio and many others. Some authors consider also other gaseous fuels as a primary premixed fuel, as described in reference 5.

Combined with an experimental activity, the numerical simulation can be helpful in understanding the complicated combustion phenomenon in a hybrid engine such as the dual-fuel one. In

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reference 6, the combustion process within the diesel and diesel/gas dual-fuel engine is investigated by means of a coupled 3D-CFD/chemical kinetics framework. In this study, methane and n-heptane are used as representatives of the natural gas and diesel fuels. The KIVA-3V code, with modified combustion and heat transfer models, incorporates a chemical kinetics mechanism for n-heptane and methane oxidation chemistry. A chemical kinetics mechanism of 42 species and 57 reactions is used for predicting the n-heptane oxidation chemistry. The results show that Zheng and Yao's n-heptane mechanism, which had been previously validated in their work, is able to model the diesel and dual-fuel combustion, where fuel-rich zones are present. Based on constant total mixture input energy in dual-fuel combustion, increasing pilot fuel amount leads to shorter ignition delay and sharper pressure peaks. On the other hand, the NO_x and CO emissions tend to increase with higher rates of pilot fuel injection.

The paper [7] presents numerical investigations of the combustion characteristics and maximum possible natural gas substitution on a large stationary diesel engine dual fueled with natural gas operation at 600 rpm. The performance effect in the dual fuel diesel engines is investigated by using the coupled 1D/3D computer simulation of GT-Power and Kiva-3 computer codes for combustion optimization and emissions reduction. In the work of Prakash et al. [8], a correlation for the ignition delay in a biogas–diesel dual fuel engine has been proposed on the basis of experimental results. The Hardenberg & Hase correlation for ignition delay in diesel engines has been modified for the dual fuel situation by accounting for the effect of changes in both the final compression temperature and oxygen concentration in the charge. The proposed correlation shows reasonable agreement with experimental results for a biogas–diesel dual fuel engine.

Indeed, in order to improve the DF engines performance, many parameters should be controlled. The ignition delay of the pilot fuel is greater in DF compared to a standard diesel engine [9,10] due to the presence of methane, which results in a reduction of oxygen concentration and in a lower temperature and pressure at the end of compression. Abd et al. [11] analyzed performance and emissions of the DF engine, for different pilot injection advances and total equivalence ratios. For a fixed equivalence ratio, increasing the injection advance induces a reduction in THC due to a better penetration of the pilot jet inside the intake charge so that a greater amount of air/gas mixture is incorporated in the spray, before the ignition takes place. In reference 12, it is highlighted that for the same reasons, the pressure gradient increases.

Badr et al. [13] have found four operating regions separated by three limit values of the total equivalence ratio, at natural gas flow rate increasing, and thus total equivalent ratio and engine load increasing.

In previous works [14,15], the authors have described some CFD results obtained by the code KIVA 3V by varying the diesel/NG ratio and in Fig. 1 the NO behaviors, for several values of NG ratio, are shown. They have outlined the critical situation of the case with 90% NG/total fuel ratio when the methane combustion induces higher temperature peaks because of a sudden and simultaneous combustion of the whole reacting mixture. As a consequence, the NO distribution will present the highest concentration in this case.

For this reason, after a preliminary analysis on the NG ratio influence, in the present paper the attention is focused on a single natural gas ratio (NG mass/total fuel mass) of about 80%. In particular, the diesel fuel amount is fixed at the minimum value for a stable combustion, while only the injection advance has been varied to research the best condition of combustion with reduced THC without increasing the nitric oxide concentration.

The global engine efficiency has been also monitored in order to prevent an excessive reduction compared to full diesel operation.

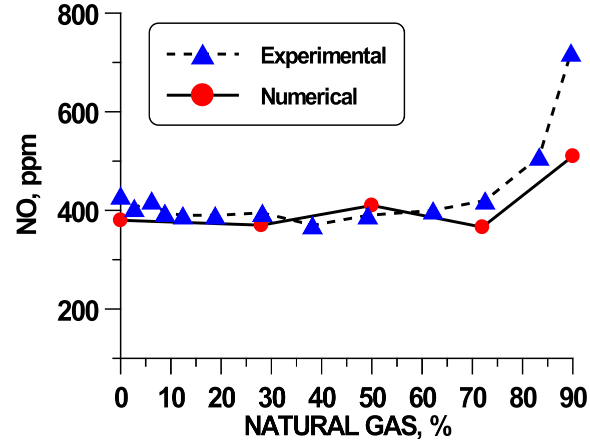

Fig. 1. NO trend [ppm] for different NG ratios at TDC [14,15].

2. Experimental activity

The experimental activity, performed on a light duty direct injection turbocharged diesel engine (Table 1) at steady state test bench, has been focused on the study of the combustion development, both in full diesel and *DF* mode (*D/NG*) at different diesel pilot injection timings, by varying it from extremely advanced to highly delayed values with respect to the optimal one adopted under full diesel condition. In this way, the experimental results cover a wide range of combustion developments and emission levels, so representing a useful reference target for the CFD based activities.

The engine was operated in full diesel and dual fuel mode by setting the electrical signal (of about 16 A) of the main injection equal to that of the pilot (Fig. 2) and by varying only the duration. In particular, for the cases studied, the injection duration of the main was changed to reduce the diesel injection amount in DF mode, while that of the pilot was kept unchanged.

In Table 2, the natural gas composition and its main characteristics are reported. In the cases investigated during the test, the amount of diesel oil was fixed at a minimum value (less than 20%) to maintain the same torque and speed (100 Nm and 2000 rpm) by varying only the injection timing. Consequently, the natural gas amount was regulated automatically and its measured value was almost constant, of nearly 80% of the total fuel mass. The load level of 100 Nm was chosen after several preliminary test cases at variable loads. The choice of this level, which is slightly lower than the maximum one, was due to the need of studying the phenomenon of this hybrid combustion under reliable conditions. Actually, at higher loads, an increased probability of knocking phenomena exist or, on the contrary, at very lower loads larger difficulties arise to characterize the excessively lean combustion.

Initially, some test cases have been performed in full diesel mode, by varying the injection advance of pilot and main jets fixed the dwell angle (i.e., the angular shift between pilot and main injection starts) approximately equal to 20°. The objective of these experiments was to establish a set of reference data, in order to compare the

Table 1
Engine main characteristics for experimental tests.

Engine	1.9 Multijet
Type	4 stroke
Number of cylinder valves/cyl.	4
	2
Bore (mm)	82
Stroke (mm)	90.4
Compression ratio	18:1

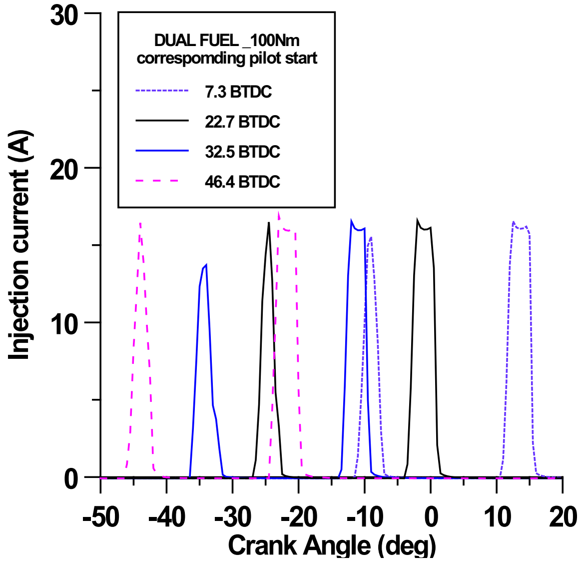

Fig. 2. Diesel injector command signal for some injection laws.

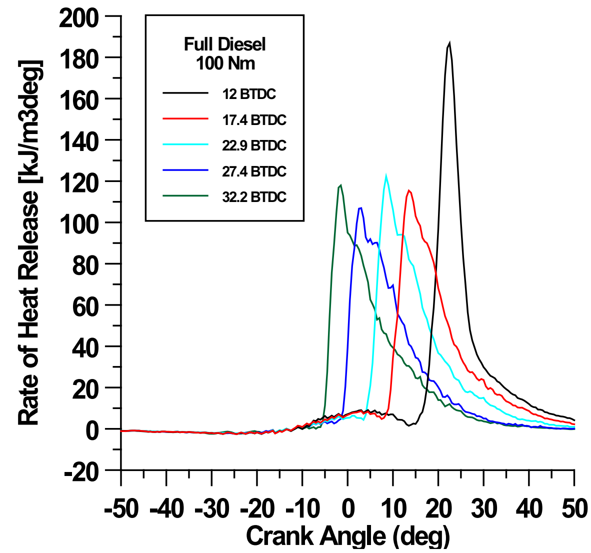

Fig. 4. Experimental rate of heat release at different injection timings (full diesel).

performance and emission trends in dual fuel mode with those deriving from a standard engine operation. In Fig. 3, the experimental in-cylinder pressure values are shown for the full diesel mode.

By varying the injection start, the engine response changes: the most anticipated values involve an early ignition and a much more rapid combustion, with a steep growth of the pressure and higher peaks, because the combustion of the pilot fuel takes place much before TDC (as demonstrated also in Fig. 4), in order to allow a quick ignition of the fuel from the main injection at the favorable conditions of temperature and pressure. The rate of heat release presents a higher premixing peak for delayed start of injection, due to the larger quantity of diesel fuel that burns compared with the anticipated cases. In particular, the effect of the pilot is already extinguished when the main injection starts as in the case more delayed by 12° BTDC. The NO_x trend is strictly in agreement with the pressure cycles. The most anticipated cycles reach higher peaks of pressure and temperatures, so that the NO_x are consequently higher. As usual in a diesel engine, the levels of burned and unburned species are very low, thanks to the favorable effect of the high air/fuel ratio, which was of nearly 22 in these experimental tests. In particular, the CO contents in the exhausts increase if delaying the fuel injection, due to the lower oxidation kinetics, while the THC level appears to be practically independent of the injection timing.

Once the reference data in full diesel mode has been defined, a set of dual fuel experiments was carried out. In this case, a wider range of injection timings was investigated, from 42.4° to 7.3° BTDC, in order to achieve more extended information about the most appropriate choice of the injection start. The pressure cycles in Fig. 5 confirm that the highest peak corresponds to the most anticipated injection (42.4°BTDC), while smoother pressure profiles are obtained by the delayed injections, like in the full diesel mode. In Fig. 2, the injector current profiles are represented for four test cases included between the extreme injection advances. It is worth noting that the occurrence of actual injector opening is delayed by about 4° with respect to the electrical signal. In Fig. 6, the rate of heat release is plotted for the same four cases: increasing the injection advance induces an anticipated ignition of the air/NG mixture. In the Figs. 7 and 8, the CO and NO_x emissions are shown and compared with the full diesel results. In dual fuel mode, the NO_x is slightly lower for the same injection advances while the CO values increase considerably, as expected. Finally, the considerable rise of the THC amount (Fig. 9) in dual fuel mode is noticeable, not comparable with that of the full diesel case whose values are below 120 ppm for all injection advances as demonstrated by Fig. 10.

In Fig. 11 the efficiency for the different engine modes is plotted versus injection advance: in the full diesel case, the highest value is achieved in the 27.4–22.8° BTDC cases that represent the transition from a typical diesel engine to a lean combustion behavior;

Table 2
NG composition and its main characteristics.

Fuel properties	Basic data
αst	15.73
ρ(kg/Stdm^3)	0.83
LHV(MJ/kg)	45.83
CH_4 (%vol)	85.44
Ethane (%vol)	8.38
Propane (%vol)	1.71
Buthane (%vol)	0.47

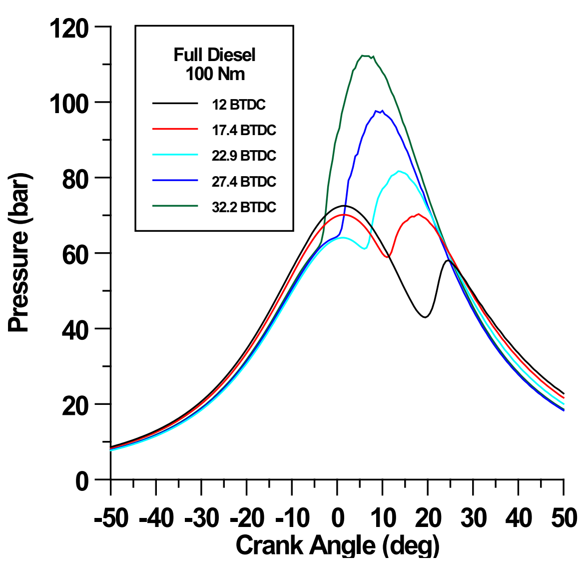

Fig. 3. Experimental pressure cycles at different injection timings (full diesel).

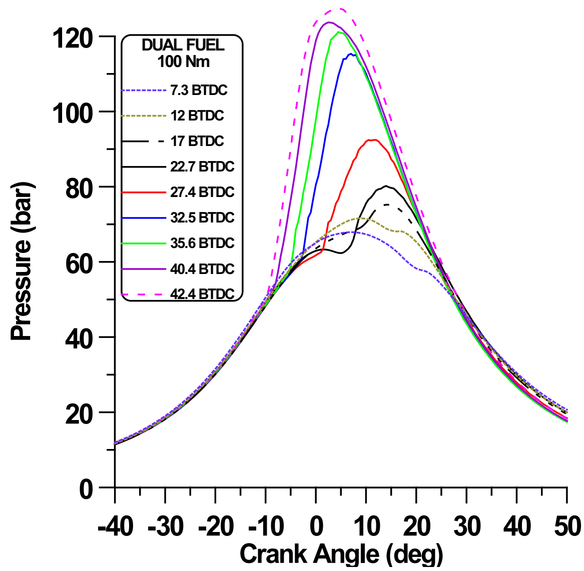

Fig. 5. Experimental pressure cycles at different injection timings in dual fuel mode.

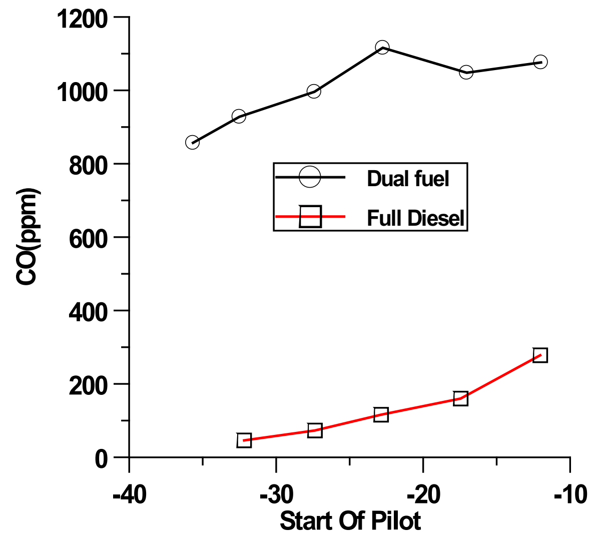

Fig. 7. Experimental CO values at different injection timings in dual-fuel mode compared with full diesel ones.

in the dual fuel case, the trend is similar but with lower values, as expected, and the highest value is reached by an anticipated case (SOP = 32.5° BTDC). The more relevant efficiency drop at the highest injection delays may be explained by the concurrent effect of an inefficient combustion, with high CO and THC levels, and the less favorable thermodynamic conditions due to the combustion development during the expansion stroke.

Based on the above considerations, the most interesting reduction in nitric oxides that would occur at the lowest injection advances cannot be exploited but a compromise must be accepted between a limited NOx reduction and an acceptable efficiency level. Therefore, among all test cases performed, particular attention was paid to four conditions that were complying with this compromise, ranging from SOP = 17° BTDC to 32.5° BTDC. In Table 3, the main parameters of interest for the dual-fuel mode are reported for such cases.

If comparing the full diesel and dual fuel results for the same injection timings (Table 4), it is worth noting that despite the efficiency reduction, not only the CO_2 contents in the exhausts but also the CO_2 emission index decrease in the DF case, so confirming the validity of this fuel supply system.

A direct comparison of the full diesel and dual full operation for two different injection timings is reported in Figs. 12 and 13. Actually, the overall heat release laws (Fig. 12) appear to be, in practice, coincident and the same result is detectable in terms of pressure cycles (Fig. 13). Although such indications are encouraging for a reliable adoption of the dual fuel concept, further results cannot be explained by these diagrams: in particular, the considerable increase in unburned species, like shown in Figs. 7 and 9, is worthy of a deeper analysis. In addition, the exact role played by the pilot and main injection for ignition of the NG–air mixture cannot be identified by the heat release laws in Fig. 13. Therefore, a CFD simulation has been carried out based on input data from the experimental tests, in order to achieve a more thorough characterization of the combustion phenomena in dual fuel mode.

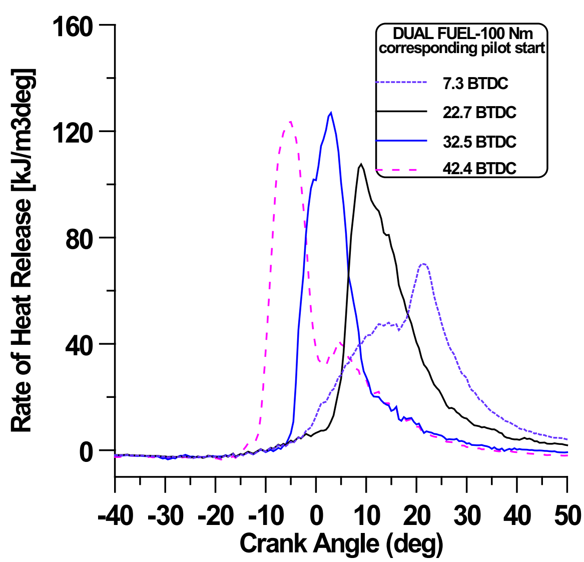

Fig. 6. Experimental rate of rate of heat release of some test cases in dual fuel mode.

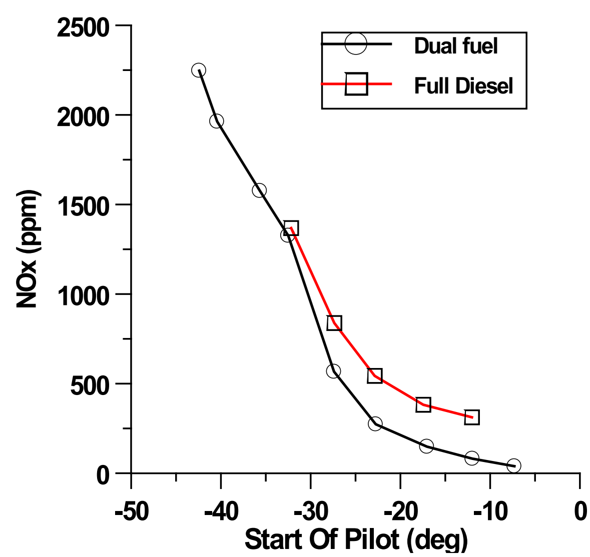

Fig. 8. Experimental NOx values in dual-fuel mode compared with full diesel ones.

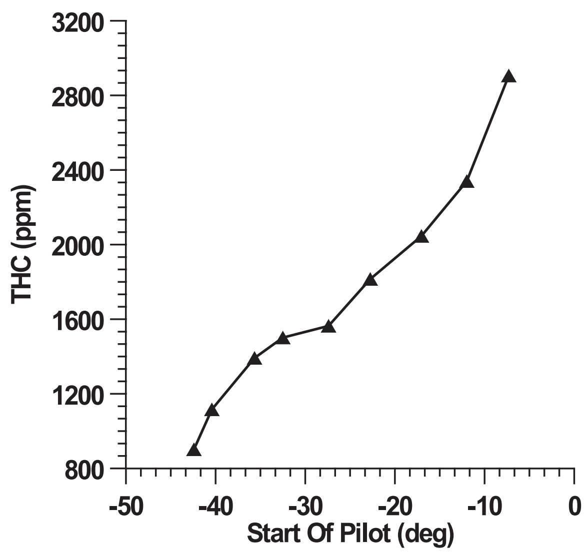

Fig. 9. Experimental THC values in dual-fuel mode.

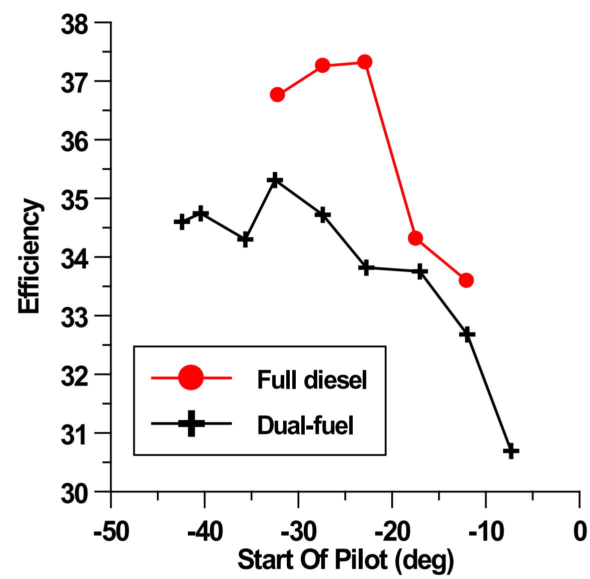

Fig. 11. Global efficiency in the dual-fuel and full diesel cases by varying injection advance.

3. Numerical activity

The computational activities address the simulation of the in-cylinder phenomena and the CFD results have been obtained by the Ansys – Fluent code. Once obtained, a satisfactory validation of the computational tool, the main differences that occur by varying dual-fuel injection rates and timings, especially with high NG ratios (over 80%), may be highlighted. As stated before, the role of the pilot and main injection of the diesel fuel on the ignition of the methane–air mixture should be worthy of an accurate assessment. Under this aspect, a CFD based simulation can be useful to provide a more thorough interpretation of the in-cylinder processes and, therefore, to pursue optimized strategies to reduce emissions and to avoid undesirable phenomena such as a sharp combustion development, especially at low load [8,16].

Fig. 14 represents the full computational domain that includes the inlet and exhaust ducts and valves geometry. The cylinder and duct block-unstructured mesh was obtained by using the ANSYS meshing tool. The size of the cylinder mesh is of about 600,000 cells at BDC and 300,000 calls at TDC, while the bowl's one is of 84,000 cells. Since the external ducts are included in the computational domain together with the valve lift laws, the CFD based solution is extended to the open valve periods, and the computation of multiple engine cycles is needed to reach the numerical convergence to the periodic regime: in this way the liquid spray–air interaction is studied within a flow field generated by the previous NG/air exchange process. Similarly, the progress in a homogeneous formation of the NG/air mixture throughout the cylinder is accurately predicted.

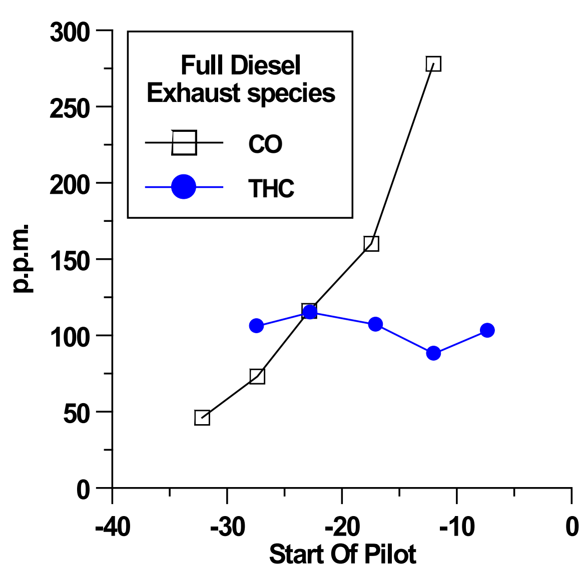

Fig. 10. Experimental CO and THC values at different injection timings in full diesel mode.

A combustion model to solve the natural gas ignition is added to the diesel fuel injection and oxidation models. In particular, the atomization model has been tested in a previous authors' work [17], and therefore, the Wave model was adopted. The Hardenberg & Hase (H&H) correlation [18] was used to reproduce the combustion delay for the diesel fuel. The well known formulation, like that implemented in the Fluent solver, has been modified in terms of pre-exponential parameter C_1 (whose standard value is of 0.36 as shown in Table 5) to 1.44 in the most advanced case and up to 100 for the delayed case. Actually, the H&H correlation was conceived for standard, single-injection diesel fuel injections: the higher values required for the C_1 constant may be explained by the strongly modified conditions due to both lower equivalence ratios and split injection of the diesel fuel.

The combustion model relies on the Finite Rate/Eddy Dissipation scheme with four reactions; the first one is the one step reaction of diesel oil oxidation and the other three for the methane oxidation [19] are shown below:

$$\begin{aligned}
&1)\, CH_4 + 3/2O_2 \Rightarrow CO + 2H_2O \\
&R_1 = 10^{13.354-0.004628P}[CH_4]^{1.3-0.01148P}[O_2]^{0.01426}[CO]^{0.1987}e^{-\frac{21932+269.4P}{T}} \\
&2)\, CO + 1/2O_2 \Rightarrow CO_2 \\
&R_2 = 10^{14.338-0.1091P}[CO]^{1.359-0.0109P}[H_2O]^{0.0912+0.0909P} \\
&\quad [O_2]^{0.891+0.0127P}e^{-\frac{22398+75.1P}{T}} \\
&3)\, CO_2 \Rightarrow CO + 1/2O_2 \\
&R_3 = 10^{15.8144+0.07163P}[CO_2]e^{-\frac{64925.8-334.31P}{T}}
\end{aligned} \qquad (1)$$

Table 3
Mixture conditions for four test cases examined.

Test case SOP	Air flow rate (kg/h)	Diesel mass flow rate (kg/h)	NG mass flow rate (kg/h)	NG/tot fuel %	OER = α_{st}/α	NG/air ER
17.0 BTDC	101	0.91	3.99	81.43	0.751	0.620
22.7 BTDC	102	0.92	4.02	81.38	0.751	0.620
27.4 BTDC	99	0.89	3.93	81.53	0.755	0.625
32.5 BTDC	98	0.83	3.92	82.53	0.755	0.632

As said before, starting from the measured electrical command signal of the injector opening (Fig. 2), the effective injection start in the model was assumed to be 4° delayed for both pilot and main phases.

Table 4
Engine efficiency and CO_2 emissions.

Test case	SOP CAD BTDC	CO_2 %	Brake efficiency	$EICO_2$ g/kWh
Full diesel	17	6.0	34.3	482.8
	22.8	6.1	37.3	465.6
	27.4	6.0	37.3	464.7
	32.2	6.2	36.8	480.3
Dual-fuel	17	5.0	33.8	376.1
80% NG	22.7	5.0	33.8	374.7
	27.4	5.2	34.7	375.0
	32.5	5.4	35.3	383.2

Two test cases, chosen between the four experimental cases of Table 3, were simulated to better understand the combustion phenomenon in the dual fuel mode: with SOP (Start of Pilot) equal 22.7° BTDC and 32.5° BTDC. Table 6 reports the main input and target data that have been derived from the experimental tests.

The most challenging problem for a correct CFD analysis was to establish an appropriate criterion for the ignition start in both cases. Actually, the Hardenberg and Hase correlation applies to the diesel fuel ignition but the methane itself is subject to delay, with respect to the achievement of the self-ignition temperature, which is of nearly 873 K. Actually, the computed air temperatures (Table 6) at the start of pilot injection indicate that the two computational cases are characterized by rather different phenomena. In other words, the ignition and flame propagation through the methane–air mixture may be differently affected by the diesel fuel injection timings in the two cases. In any case, a different adjustment to the combustion model constants was needed, depending on the features of the

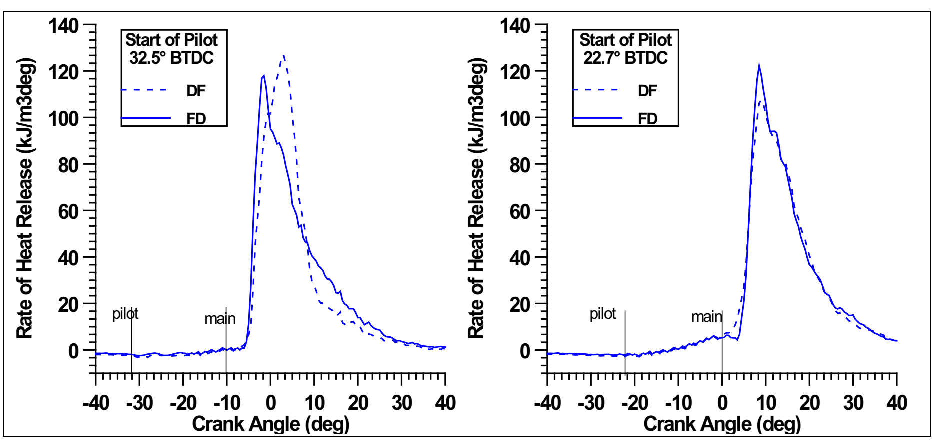

Fig. 12. Comparison between full diesel and dual fuel mode in terms of rate of heat release for two test cases (32.5° BTCD and 22.7° BTDC).

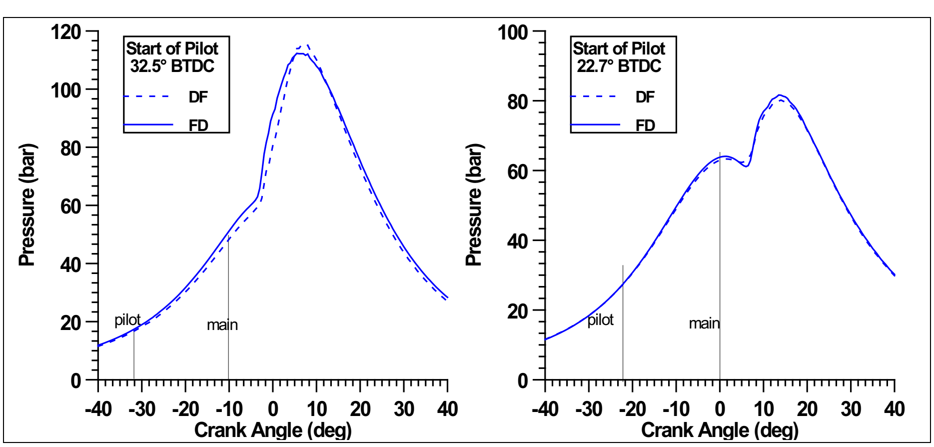

Fig. 13. Comparison between full diesel and dual fuel mode in terms of in cylinder pressure for two test cases (32.5° BTCD and 22.7° BTDC).

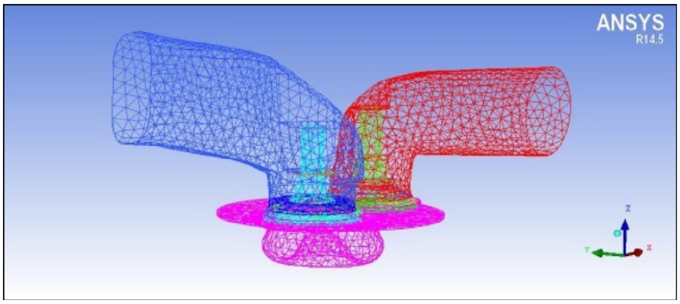

Fig. 14. Computational domain and unstructured mesh.

process development. In particular, the kinetic mechanism in equations (1) was left unchanged but the mixing rate constants of the eddy dissipation mechanism were properly calibrated.

The pressure cycles in Fig. 15 represent a comparison between numerical and experimental results for the two cases examined. A quite good fitting of experimental curve is realized by CFD computations for the SOP = 32.5°BTDC. The other case (SOP = 22.7°) presented more difficulties to approach the experimental behavior, because the higher temperature close to TDC leads the methane combustion to be independent of main diesel fuel injection, as shown in Fig. 16: the methane oxidation develops at TDC reaching relatively low values in a smooth mode. Conversely, in the SOP = 32.5° case, the oxidation rates of the two fuels proceed almost simultaneously and the methane curve rises sharply during the main injection, having reached the optimal conditions (T, p) for combustion and for the presence of several flame cores in the bowl.

The temperature distributions in the chamber (Figs. 17 and 18) confirm a different combustion development in the two cases. The one with a more anticipated injection (SOP = 32.5°) reaches an almost uniform temperature distribution at already 10° ATDC and a complete homogeneity at 20°. The other case shows evidence of a more irregular combustion development, since at 20° ATDC remarkable temperature gradients are still present. A non-complete combustion in the test case with SOP = 22.7° is so expected as demonstrated in the Fig. 19 where CO and methane amounts at the end of expansion are still present. The residual presence of methane is evident also in the distribution of Fig. 20 at 10° ATDC.

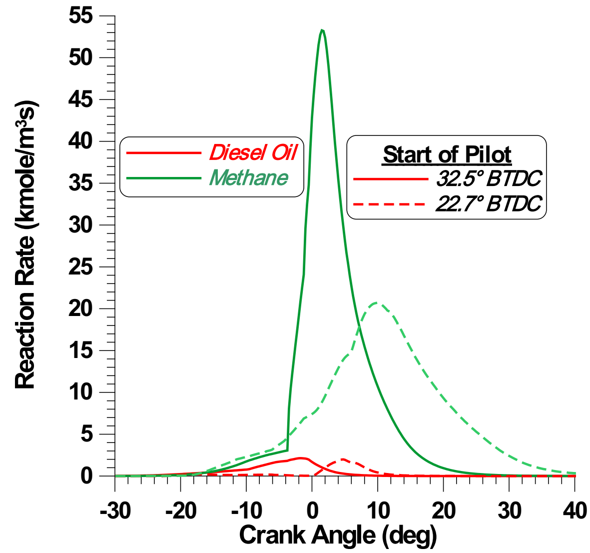

Fig. 16. Reaction rates of diesel oil and methane fuel for the two test cases.

Table 5
H&H correlation and its default values of the variables.

$$\tau_{id} = \left(\frac{C_1 + 0.22\bar{S}_p}{6N}\right)\exp\left[E_a\left(\frac{1}{RT} - \frac{1}{17{,}190}\right) + \left(\frac{21.2}{p-12.4}\right)^{ep}\right]$$

Variable	Ehh	CN	C1	ep
Default	618,840	25	0.36	0.63

Table 6
Boundary conditions for the numerical test cases.

Start of pilot	22.7 BTDC	32.5 BTDC
Boost pressure	1.35 bar	1.31 bar
Inlet air temp.	301.6 K	301.9 K
Discharge pressure	1.39 bar	1.36 bar
Diesel fuel mass	3.8 mg/cycle	3.5 mg/cycle
CH_4 /total fuel ratio	81.4 %	82.6 %
Air mass	425 mg/cycle	407 mg/cycle
Temperature at SOP	909.7 K	847.9 K

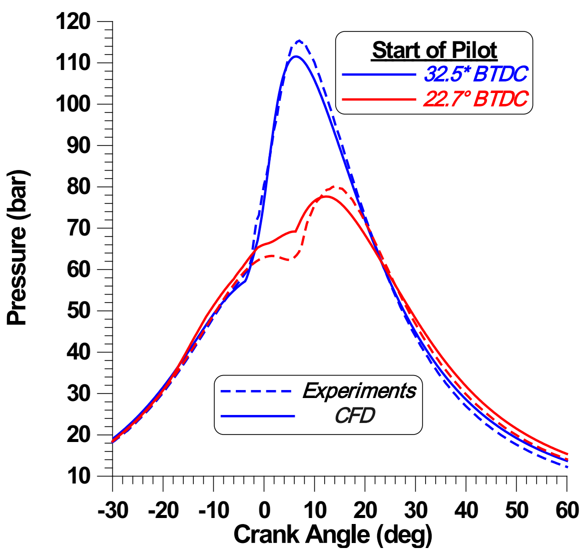

Fig. 15. Pressure cycles for the two cases examined.

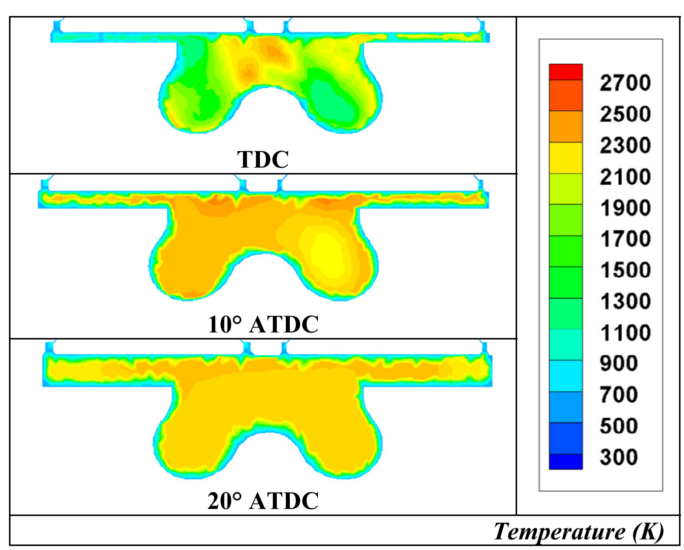

Fig. 17. Temperature distributions in the bowl at three different crank angles (SOP = 32.5°).

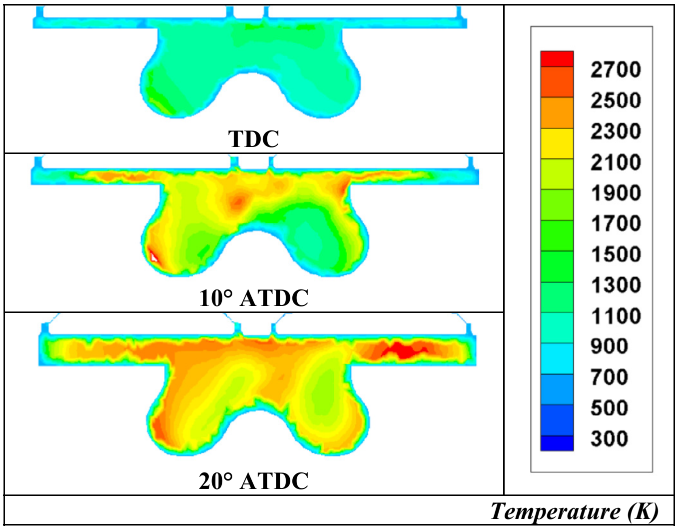

Fig. 18. Temperature distributions in the bowl at three different crank angles (SOP = 22.7°).

On the contrary, for the case with SOP = 32.5° (Fig. 21), lower values are expected for partially burned and unburned species in agreement with the methane distribution in Fig. 22 that disappears after only 10 ATDC, as also the diesel oil (Fig. 23).

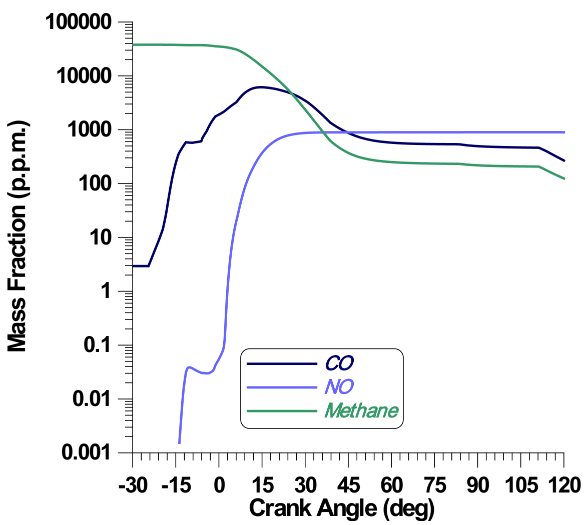

Fig. 19. Mass fraction of CO, NO and CH_4 species (case-SOP = 22.7°BTDC).

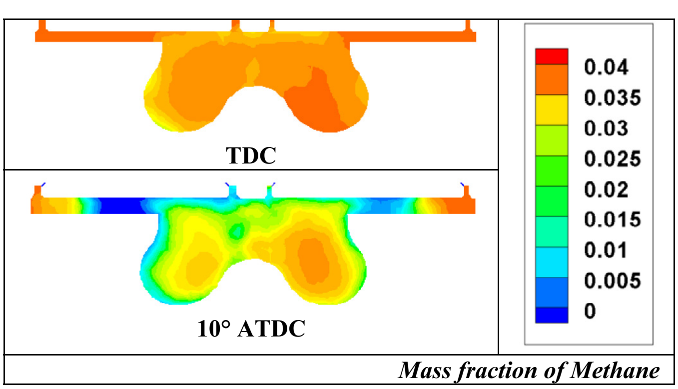

Fig. 20. Methane distributions in the bowl (SOP = 22.7°).

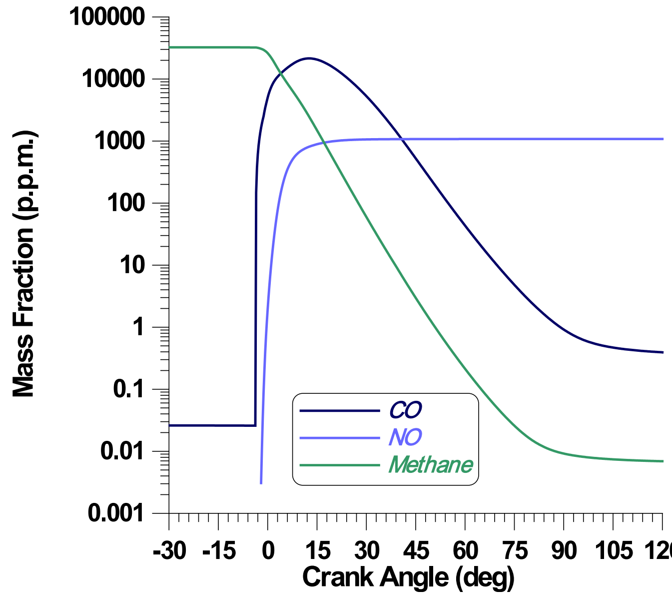

Fig. 21. Mass fraction of CO, NO and CH_4 species (case-SOP = 32.5°BTDC).

As a concluding remark, the CFD based analysis confirms that the dual fuel operation with the more anticipated injection time appears to be more efficient in terms of reduction of unburned species and, therefore, of combustion efficiency. In this sense, the computed results are in a fairly qualitative agreement with the experimental ones in Figs 7, 9 and 11.

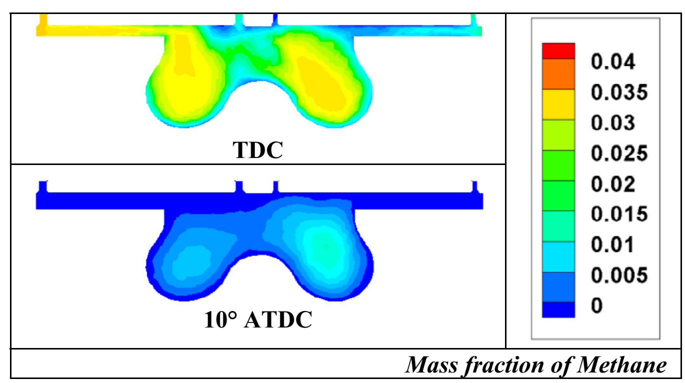

Fig. 22. Methane distributions in the bowl at two different crank angles (SOP = 32.5°).

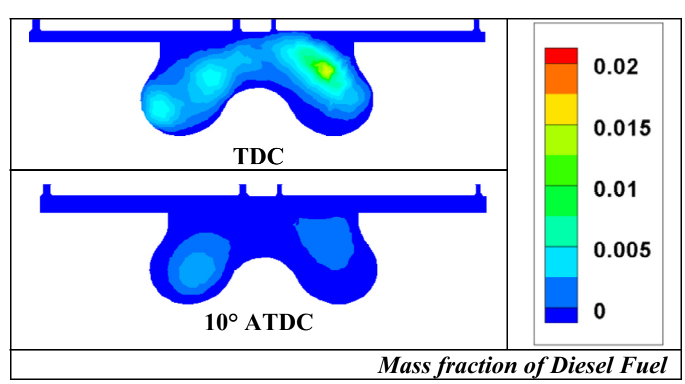

Fig. 23. Diesel oil distributions in the bowl (SOP = 32.5°).

4. Conclusions

The authors have discussed in this paper the effect of the diesel oil injection timing on the performance and emissions of a dual fuel compression ignition engine. The experimental results put into evidence a strict sensitivity of the engine response to the start of the injection and this confirms that proper strategies should be determined for optimizing the dual-fuel operation in terms of both energetic and environmental behavior.

The role of the CFD analysis consisted, in this phase, in a trial of explaining the results of complex combustion phenomena that take place within the in-cylinder processes. The latter is characterized by the simultaneous occurrence of two reacting mixtures, one of the discontinuous and the other one of the homogeneous type, each of them presenting different ignition delays. Consequently, the classical correlations available in literature, such as the one from Hardenberg and Hase, are not appropriate since they cannot take into account the simultaneous presence of different fuels. Actually, some improvements in the predictivity level have been obtained by a proper adjustment of the constants in the above correlation and in the mixing rate mechanism. This allowed a fair description of the combustion development in two cases characterized by anticipated and delayed diesel fuel injections. The second case exhibits, in particular, a much smoother combustion development, in accordance with the trend to a lean combustion in diesel engines, but it also presents an increase in unburned species. Anticipating the diesel fuel injection results in better combustion efficiency but also in a NOx increase together with a sharper in-cylinder pressure rise. The authors' work has so demonstrated that an accurate optimization is needed for the best compromise between a homogenous and lean combustion and the emission and efficiency control.

On the other hand, the above results confirm that a more careful and deeper sensitivity analysis of the combustion models to the characteristic constants used should be done. As a matter of fact, one of the most critical points is undoubtedly the ignition delay phase description for both fuels. To overcome this problem, more complex kinetic schemes coupled with a Flamelet Model should be introduced to properly describe the early phase and the development of the combustion process. The accurate modeling of these phenomena will make possible, on one hand, the reliable interpretation of the experimental conditions, while, on the other one, it will suggest to the experimentalists the optimal set up of the control variables for the engine at the test bench.

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The CFD computations are licensed by *ANSYS-FLUENT*.

Nomenclature

ATDC	After TDC
BTDC	Before TDC
CFD	Computational fluid dynamics
D	Diesel oil
DF	Dual fuel
EGR	Exhaust gas recirculation
ER	Equivalence ratio
FD	Full diesel
NG	Natural gas
OER	Overall equivalence ratio
R	Reaction rate
SOP	Start of pilot
TDC	Top dead centre
THC	Total unburned hydrocarbons

Greek symbols

α_{st}	Stoichiometric fuel/air ratio
α	Overall fuel/air ratio

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