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# Compressed Bio Gas (CBG) in Diesel Engine

*Hasan Köten*

## Abstract

In this chapter, as an alternative to conventional engines, compressed biogas (CBG)-fueled compression ignition (CI) engine was evaluated. Biogas fuel is considerably economic due to the local product fuel compared to that of CNG and LPG fuels for many countries. In addition, due to the higher octane rate, biogas is considerably adaptable for the diesel engines. In this issue, CBG diesel-fueled engine was investigated using full geometry 3D computational fluid dynamics (CFD) simulations including intake and exhaust ports were used in optimization process to get the optimum design parameters of the CBG-diesel engine. Usage of CBG fuel in the optimized CBG engine without any constructive change in cylinder block will considerably decrease the cost. During the engine design, one-dimensional (1D) and three-dimensional (3D) CFD codes and multi-objective optimization code were employed by coupling codes. CBG and diesel fuels were defined as leading reactants using user-defined code in dual-fuel diesel engine modeling. CBG and diesel mass flow rates, start of pilot diesel fuel injection, compression ratio, valve timing, and engine speed were defined as input variables in different engine loads and evaluated about 20,000 cases to define the proper operating conditions. CBG-diesel engine and significantly lower  $\text{NO}_x$  emissions were emitted under dual-fuel operation for all cases compared to single-fuel mode at all engine load conditions. Moreover, CBG-diesel engine provided superior performance in reductions of  $\text{NO}_x$  and particulate matter (PM) emissions.

**Keywords:** CBG, biogas, CFD, emission, diesel engine

## 1. Introduction

This chapter is intended to give an overview of the CBG-fueled diesel engine performance and emission characteristics. The optimum design parameters of the CBG-diesel dual-fueled engine were studied using CFD techniques and experimental work. Also the motivation for IC engine research is presented, and the combustion process for the SI and CI engine was shortly overviewed. In addition to that, new alternative combustion concepts for CI engines were discussed, and the research background and objective of the present work were presented. CBG fuel air mixture is compressed in a PPCI mode in achieving its simultaneous ignition by pilot diesel fuel inside the combustion chamber to get the best performance and emission results.

Part load, especially direct injection systems used to perform partially premixed charge, allows for optimized fuel consumption and a low level of emissions. During

like this process, the engine has quite more homogeneous air fuel mixture and low in-cylinder temperature which caused lower  $\text{NO}_x$  emissions. Also, the use of a pilot injection has become an effective way for reducing combustion noise.

In PPCI, combustion concepts have been recently developed with the purpose to solve the problem of the high emission levels of conventional direct injection diesel engines. A good example is the PPCI combustion, a strategy in which early fuel injections are used, causing a burning process in which more air fuel is burned in premixed conditions, which affects combustion performance and exhaust emissions.

Experimental studies due to the extreme conditions inside a typical IC engine such as high combustion temperatures and pressures, precipitation of PM, other combustion products, etc. are sometimes limited in approaching exhaust emission problem. However, CFD software offers the opportunity to carry out and optimize repetitive parameter studies with clearly defined boundary conditions in order to investigate various configurations.

In this book, effects of dual-fuel combustion characteristics were investigated on the combustion performance and the reduction of exhaust emissions for a CI engine fueled with CBG-diesel dual fuel. Different approaches for alternative diesel combustion systems are also investigated by CFD and optimization software. This combustion system is investigated in homogenous CBG fuel air mixture with early and late pilot diesel injection strategy.

## **2. Usage of CBG fuel in diesel engine**

The intention of this investigation is to find out the effects of CBG-diesel dual-fuel combustion characteristics on the CI engine performance. The rate of heat release (ROHR) and other performance parameters were investigated in different modes of combustions. Moreover, combustion performance and indicated mean effective pressure (IMEP), exhaust gas temperature, and also the concentrations of PM,  $\text{NO}_x$ , HC, CO, and  $\text{CO}_2$  exhaust emissions were also investigated under various engine operating conditions to compare the exhaust emission and engine performance of single-fuel and CBG-diesel fuel modes experimentally and numerically. Within this framework, the combustion processes and performance of a commercial four-cylinder, turbocharged compression ignition engine are analyzed and improved the exhaust emission values of the engine by proposing some modifications for advance mode of combustion system by using CFD and multi-objective optimization codes.

In accordance with this purpose, first;

- Overall thermodynamic cycle simulation for one cylinder,
- In-cylinder fluid motion,
- Including inlet, exhaust manifold and valves are analyzed in 3D.

Therefore,

- To perform a detailed analysis in-cylinder cold flow,
- Fuel spray atomization,
- The combustion and emissions are investigated numerically and experimentally.

Second part of this work included optimization for an advance combustion mode needed for the following parameters:

- Valve timing.
- Compression ratio.
- Pilot diesel fuel injection timing was optimized.

Lastly, selected cases which were optimized by CFD and multi-objective optimization code analyzed and compared with existing experimental single-fuel and CBG-diesel dual-fuel diesel engine combustion performance and exhaust emissions.

In this project, methodology was designed to accomplish the objectives described in objective parameters.

The first task was to carry out an overall and critical research of available literature in the dual-fuel diesel engine field. This review was done to fully understand the progress of dual-fuel combustion process in this particular field of research; also this is aided with the identification of issues/areas of further research.

The survey considered published books, journals, and papers. It was broadened to consider information published on the Ohio State University database and on the Center for Automotive Research Laboratories.

Full geometry model: After the study of the commercial CFD software documentation, some applications were carried out to aid with the meshing of the geometry.

Simulation: A preliminary simulation in commercial CFD software was carried out in order to build confidence levels, since combustion of spray droplets involves complex models both of pilot diesel fuel and CBG fuel injectors. The main simulations were divided into four main configurations as follows:

- Cold flow (no liquid fuel)
- Air/liquid spray mixture
- Combustion simulation using turbulence-controlled eddy breakup
- Dual-fuel combustion simulation using non-premixed and partially premixed model

For each model, these five turbulence models were investigated ( $k-\epsilon$ /high Reynolds,  $k-\epsilon$ /RNG,  $k-\epsilon$ /Chen,  $k-\epsilon$ /Speziale/high Reynolds, and  $k-\epsilon$ /Standart/High Reynolds models). The method adopted for this simulation generally follows the steps outlined in commercial CFD software studies.

The analysis of the results was based on the post-processed data from all analyses carried out. In combustion modeling, two leading reactants CBG and diesel fuel are defined by using user-defined code. The predicted results by commercial CFD code were compared with each other. Detailed specifications of engine were summarized in **Table 1**. CBG fuel properties and operating conditions are given in **Tables 2** and **3**.

## 2.1 Computational grid

The engine that is modeled is a commercial four-cylinder 1.5 l light-duty diesel real engine. The geometrical specifications of the engine, as well as the engine's original valve timings, are summarized before chapter. The computational grid is

Engine parameters	Value
Type	4 Cylinder-four stroke
Bore × stroke	76 × 80.5 mm
Connecting rod length	131.2 mm
Displacement	1461 cm <sup>3</sup>
Compression ratio	18.25:1
Max. lift (exhaust)	10.1 mm
Max. lift (intake)	9.7 mm
Operating speed	2000 rpm
Maximum power	48 kW at 4000 rpm
Maximum torque	160 Nm at 2000 rpm

**Table 1.**  
*Engine specifications.*

Properties	Value
Chemical formula	Mixture
CH <sub>4</sub>	65–70% by volume
CO <sub>2</sub>	25–35% by volume
H <sub>2</sub>	1–2% by volume
Self-ignition temperature	630–810°C
Lower heating value	26 MJ/kg
F/A ratio	0.058
Octane number	135
Density	0.79 kg/m <sup>3</sup>

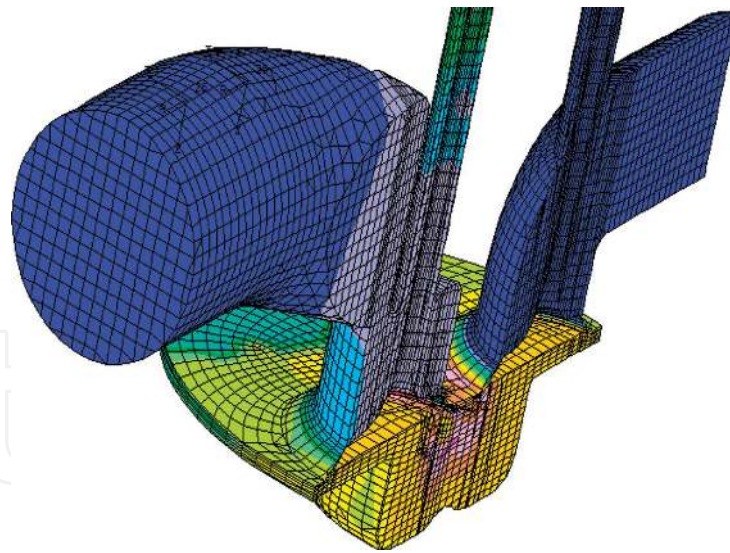
**Table 2.**  
*Properties of CBG fuel.*

RPM	2000 rev/min
Test fuels	Diesel and CBG
Single fuel	Diesel
Dual fuel	Diesel + CBG
Gas injection pressure	0.3 MPa
Gas injection type	Port injection
Injected gas mass	30 mg/cycle
Pilot diesel injection pressure	13 MPa
Start of injection	120, 18 OCA bTDC

**Table 3.**  
*Operating conditions.*

given in **Figure 1**. The mesh domain has about 700,000 elements at TDC. A finer grid could include the top-land crevice. In addition, a crevice model could be introduced in commercial CFD software, in order to simulate flow in the crevices and blow-by. Nevertheless, even though the low-temperature regions are not captured well, commercial CFD software can still provide reasonable predictions for the bulk temperature in the cylinder and the overall temperature and composition distributions.





**Figure 1.**  
*Example of volume mesh.*

In the CFD simulations before the experimental work, combustion chamber including intake and exhaust ports and valves was modeled in the development software. Mesh elements reached 1,700,000 at the BDC. In order to initialize the run, the pressure and the temperature in the cylinder at the start of the calculation were adjusted. Heat transfer and other physical models were selected according to real engine operating conditions. Complete combustion products were also defined using the user-defined code.

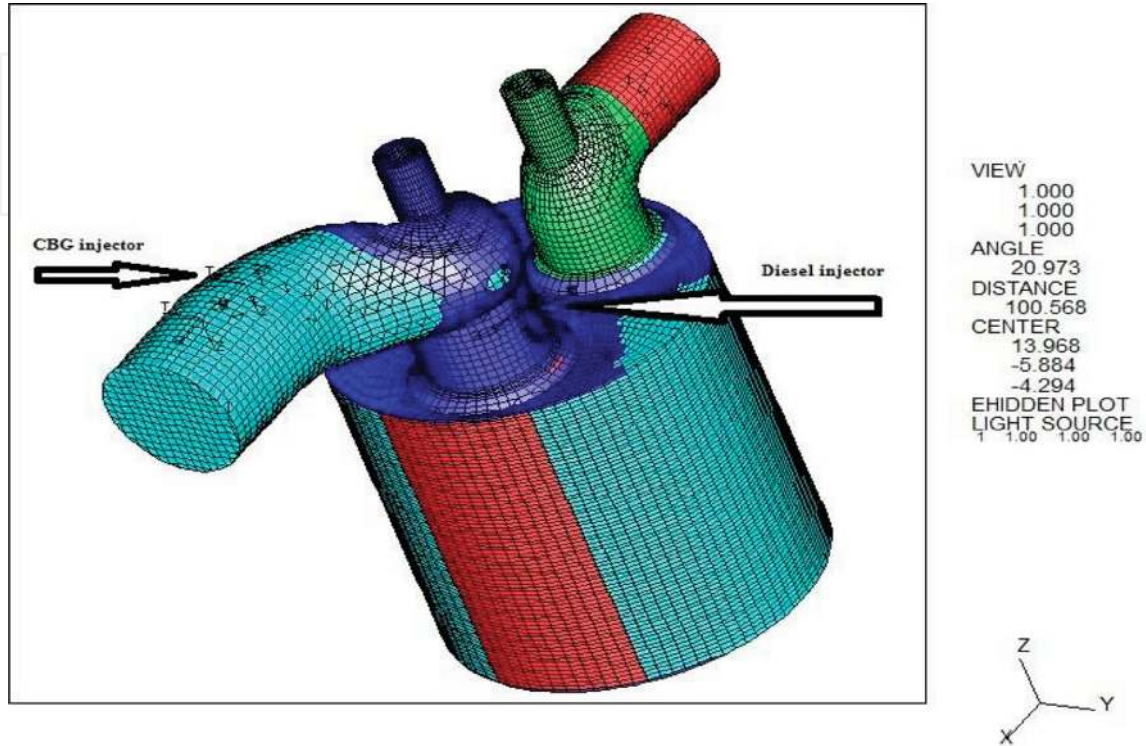
### 3. Results and discussions

#### 3.1 Dual-fuel engine spray modeling

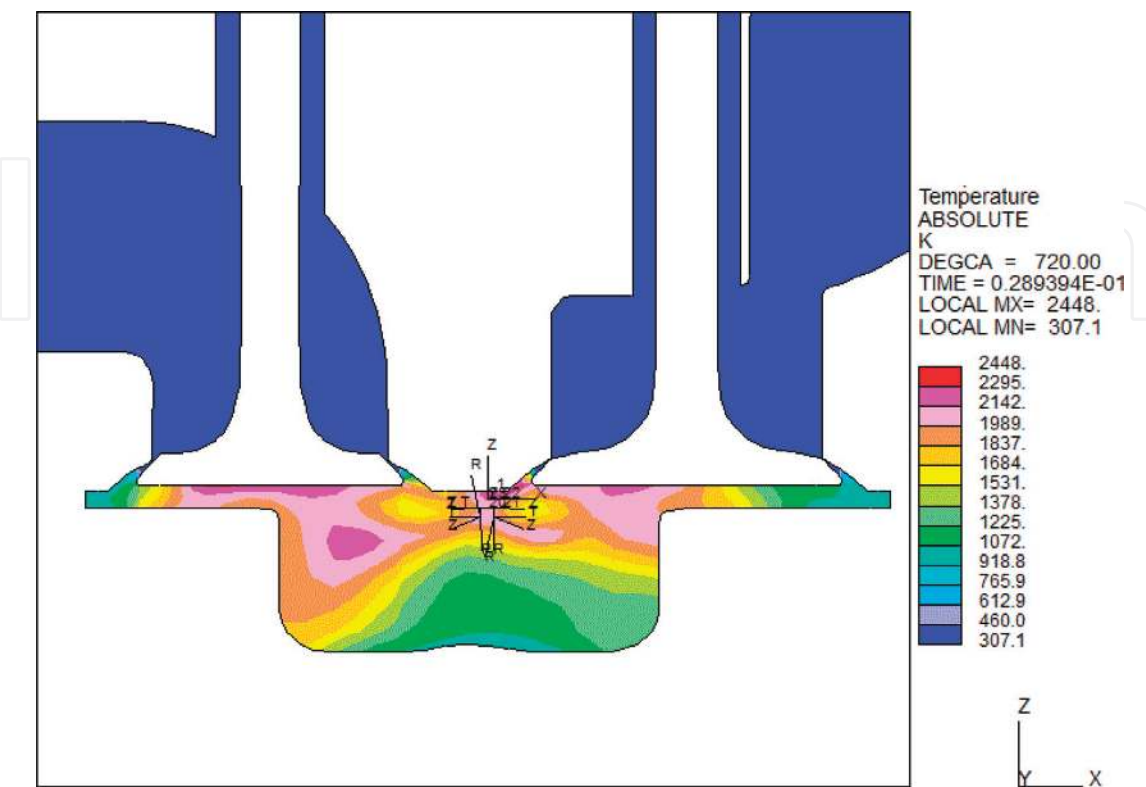
Recent researches show that it is possible to decrease the emissions considerably by modifying the geometrical sub-systems of the engine that affect the turbulence generation and spray formation. It is important to define the proper turbulence model in diesel CFD studies in terms of the exact representation of the combustion phenomenon. During these studies, a lot of turbulence models have also been analyzed to select the proper turbulence model for diesel combustion. In an attempt to improve the predictive ability of the standard  $k-\epsilon$  models, a number of alternatives have been offered. Among them the RNG  $k-\epsilon$  model [18, 19], anisotropic  $k-\epsilon$  model of Speziale [15], Morel and Mansour version of the  $k-\epsilon$  model [10], Chen's  $k-\epsilon$  model [3], and the  $k-\omega$  model of Wilcox are well-known [17]. The RNG  $k-\epsilon$  model turbulence has been used in order to predict the compressed turbulence in IC engines.

In this investigation, combustion is modeled via a new combustion model (ECFM-3Z) developed at IFP and 1D thermodynamic model. Wiebe function for 1D approximation and ECFM for 3D CFD solution were used to carry out combustion modeling. ECFM-3Z is the member of the coherent flame model (CFM) family, and it is extended to nonhomogeneous turbulent premixed and unpremixed (diffusion) regions. In dual-fuel mode, extra definitions analyze the conventional diesel combustion and partially premixed compression ignition (PPCI) cases. In a diesel combustion, NO<sub>x</sub> formation is an important challenge instead of other emissions such as smoke which is gas and carbon mixtures. Conglomeration of carbon particles calls as PM, and dust airborne particles call as a particulate matter (PM). They

are produced during incomplete combustion process. Real engine geometry was remodeled to find out dual-fuel flow structure inside the combustion chamber. In dual-fuel engine cases, air and CBG fuel mixture was ingested into the combustion chamber, and it was ignited with pilot diesel fuel at the end of compression stroke. First injector was located on the intake port as a main fuel CBG using cylindrical



**Figure 2.**  
 CBG-fueled diesel engine mesh structure.



**Figure 3.**  
 3D NOx emission contours for SF case3 at TDC in +Y direction.

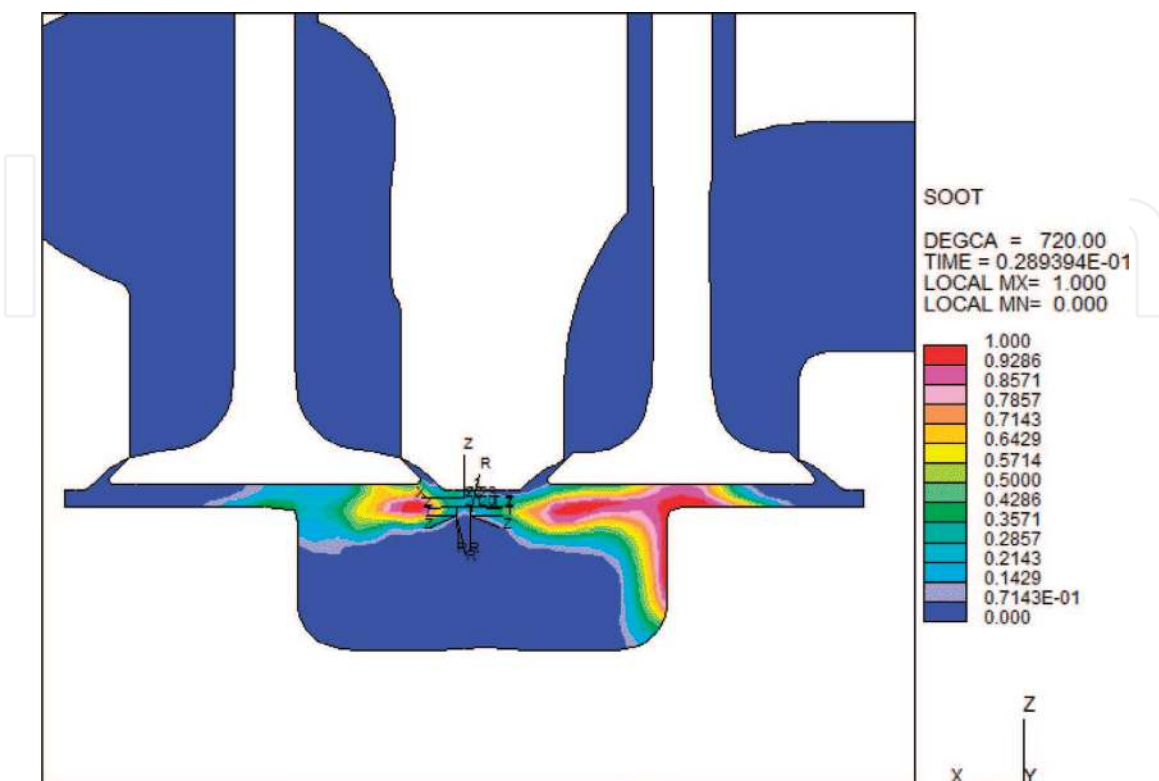
coordinate system as shown in **Figure 2**. Second injector for pilot diesel fuel was retained on the cylinder head. Injector hole diameter, cone angle, hole number, start of CBG fuel injection, and duration were entered on the CFD code.

Because of the shifting of combustion event to earlier side, this causes the increase of negative operating conditions for a conventional diesel engine. These trends are regarded as typical problems of injection strategies and injection rates that lower the thermal efficiency and increase the incomplete combustion products such as the HC and CO emissions [6–9, 11–14, 16, 20].

The effects of the engine load and dual-fuel combustion mode on the NO<sub>x</sub> emissions with different engine configurations were shown in **Figure 3**. NO<sub>x</sub> emissions showed a strong dependence on the type of combustion at constant injection

Engine type	Single-cylinder direct injection diesel engine
Engine speed	2000 RPM
Valves per cylinder	2
Bore	86 mm
Stroke	76 mm
Injection system	Common-rail
Number of nozzle holes	4
Nozzle diameter	0.170 mm
Valve overlapping	39 CAD and 19 CAD
Compression ratio	17.6
Start of injection	18 CAD bTDC

**Table 4.**  
 Specification of modified dual-fuel combustion chamber.

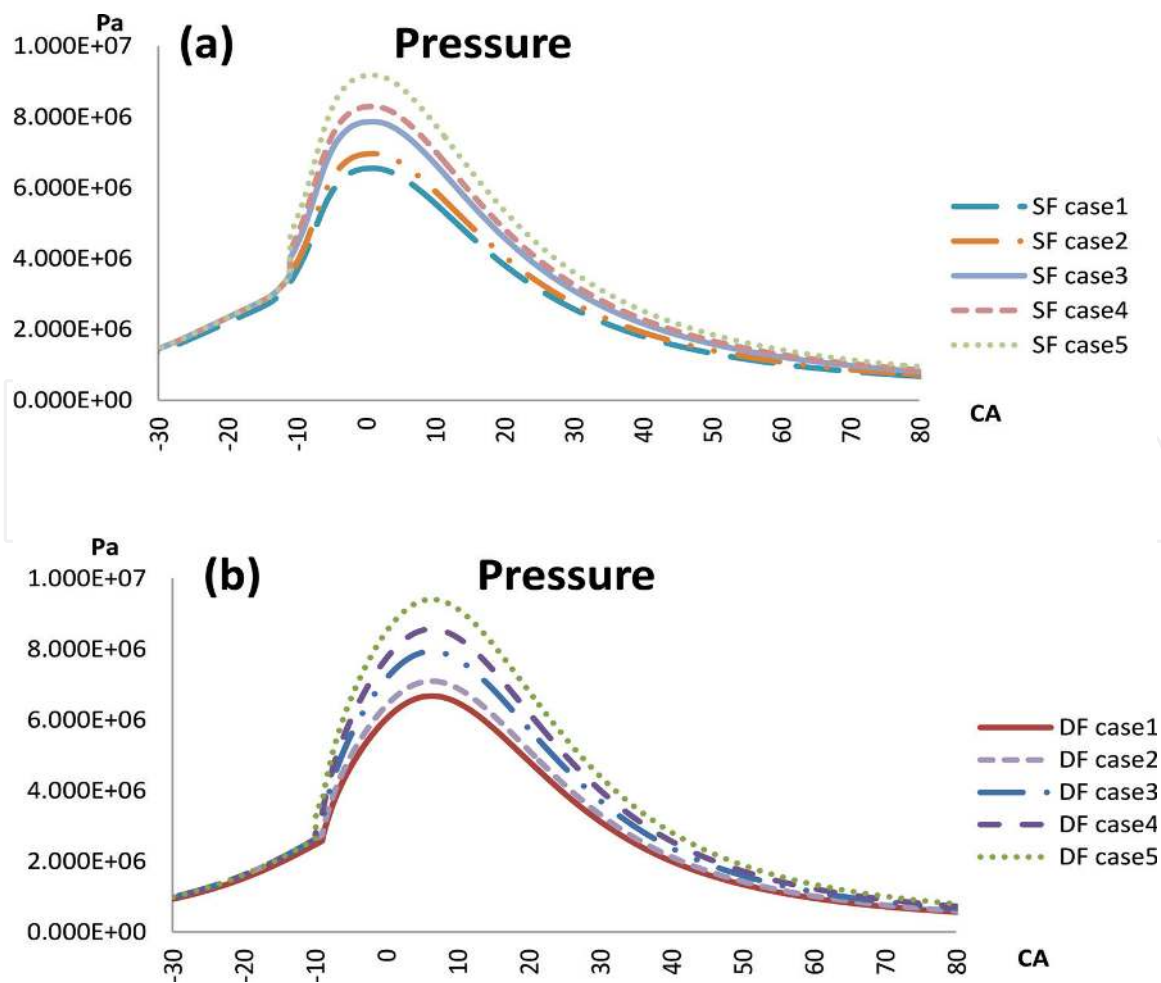


**Figure 4.**  
 Soot emission contours for SF case3 at TDC.

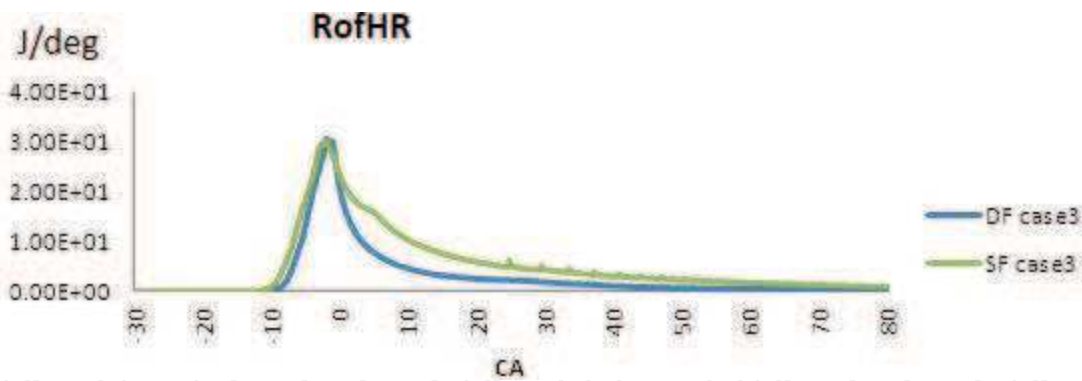


timing. The peaks of the NO<sub>x</sub> emissions occurred on the single-fuel cases at the same operating conditions. When the CBG fuel was increased, the NO<sub>x</sub> formations reached undetectable levels. This is due to the prolonged the ignition delays and premixed fuel/air mixture. In the dual-fuel modes, lower NO<sub>x</sub> formations were obtained compared to that of the conventional cases. The modified dual-fuel combustion chamber parameters are listed in **Table 4**.

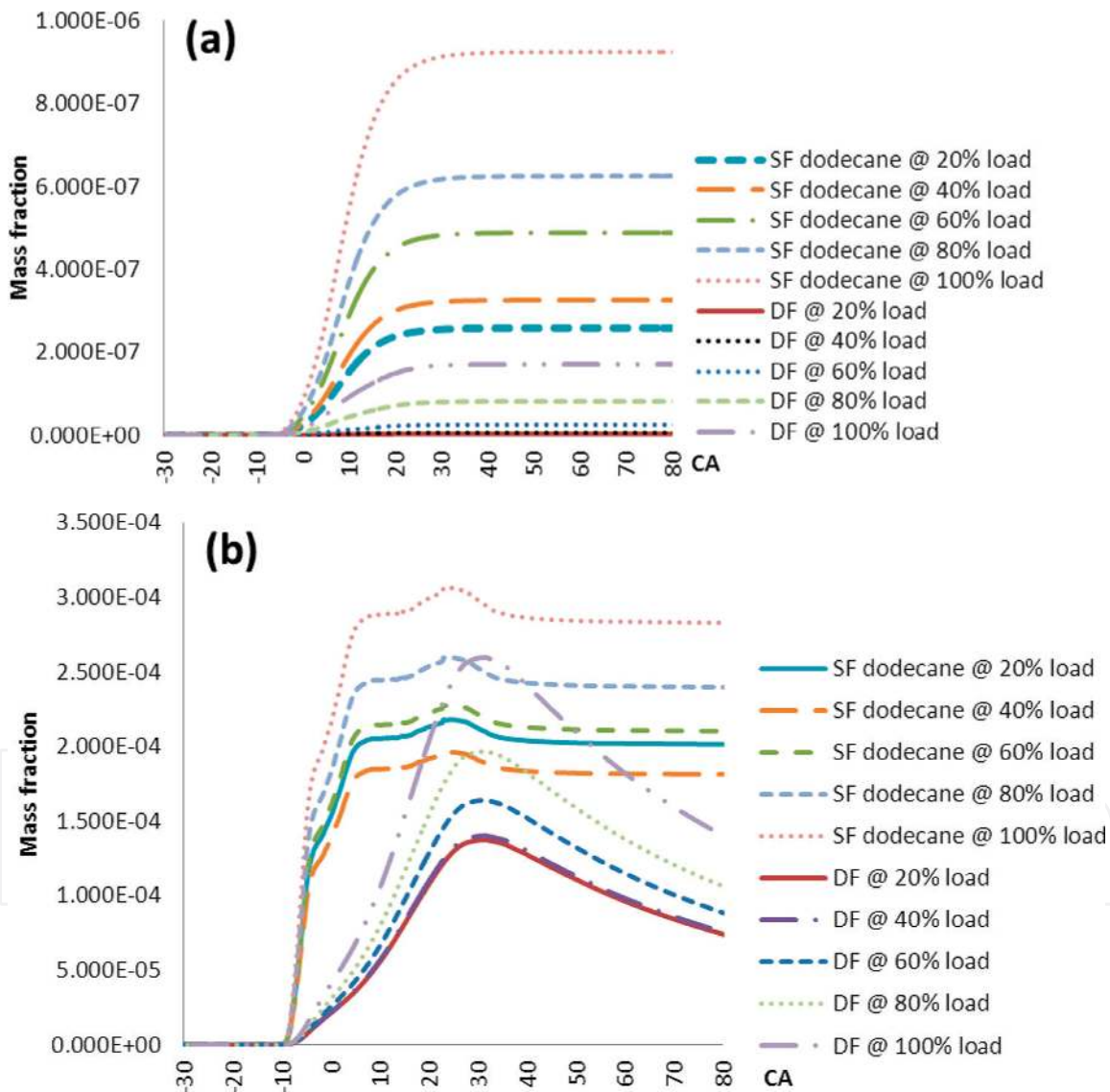
1D, 3D, and multi-objective optimization codes were employed for single-diesel fuel (dodecane) and dual-fuel (CBG-diesel) cases. Case1, case2, case3, case4, and case5 were investigated at 20, 40, 60, 80, and 100% engine loads, respectively for both single fuel and dual fuel. Soot emissions and combustion characteristics of engine can be seen in **Figures 4** and **5**. The combustion pressures and rates of heat release (ROHR) for the single-fuel mode with diesel fuel in a constant engine speed of 2000 (rev/min) were provided in **Figure 6**. The figures showed similar patterns for combustion pressure and ROHR at different engine loads. The combustion pressures and ROHRs increased for both fuels, since engine load increased at constant engine speed. At low engine load (20%), the peak pressure and also heat release were slightly lower than other cases as depicted in **Figure 6a**. The lower diesel fuel consumption (2.14 kg/h) resulted in the decrease on the combustion performance. In the 60% load, shown in **Figure 6a**, the pressure is  $P_{max} = 8.4$  MPa, and peak heat release was obtained compared to CBG-diesel case,  $P_{max} = 8.3$  MPa. Simultaneously, a greater indicated mean effective pressure (IMEP) was resulted for the conventional diesel-injected fuel mass reached 5.3 kg/h. In **Figure 7**, NO<sub>x</sub> and soot emissions are given in detail.



**Figure 5.** Combustion characteristics at different engine loads. (a) Single-fuel (dodecane) cases and (b) dual-fuel (CBG-dodecane) cases.



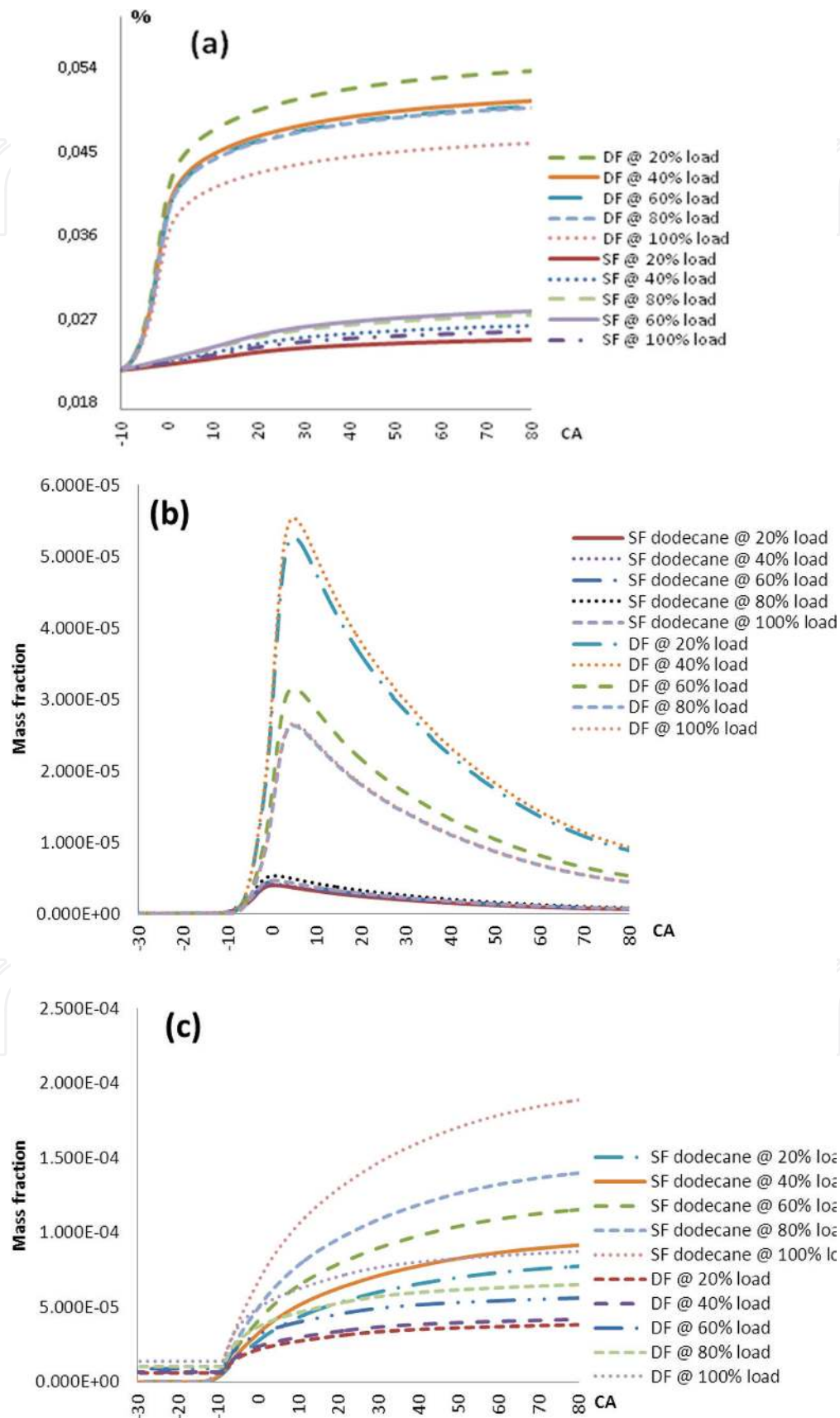
**Figure 6.**  
 Effect of fuel types on rate of heat release inside cylinder at 60% engine load.



**Figure 7.**  
 NOx and soot emissions for single- and dual-fuel cases versus CA. (a) NOx emissions and (b) soot emissions.

In terms of the ignition delay, conventional diesel combustion has shorter time due to the air fuel mixture process. Ignition ability in a diesel engine is mainly relying on caffeine and physical fuel properties such as structure of fuel composition, density, bulk module, cetane number, oxygen content, and aromatic content of the fuel. Meanwhile, the oxygen amount of the air fuel mixture plays an important role in short ignition delays. Engine parameters such as SOI need to adjust for different

operating conditions. Additionally, the diesel fuel used in the works has a long carbon chain, and it has important role for the short ignition delay. CO<sub>2</sub>, HC, and CO concentrations were shown in **Figure 8a–c** for single- and dual-fuel cases at various engine loads [5].



**Figure 8.** Exhaust emissions for single- and dual-fuel cases with different engine loads. (a) Unburned HC; (b) CO; (c) CO<sub>2</sub>.

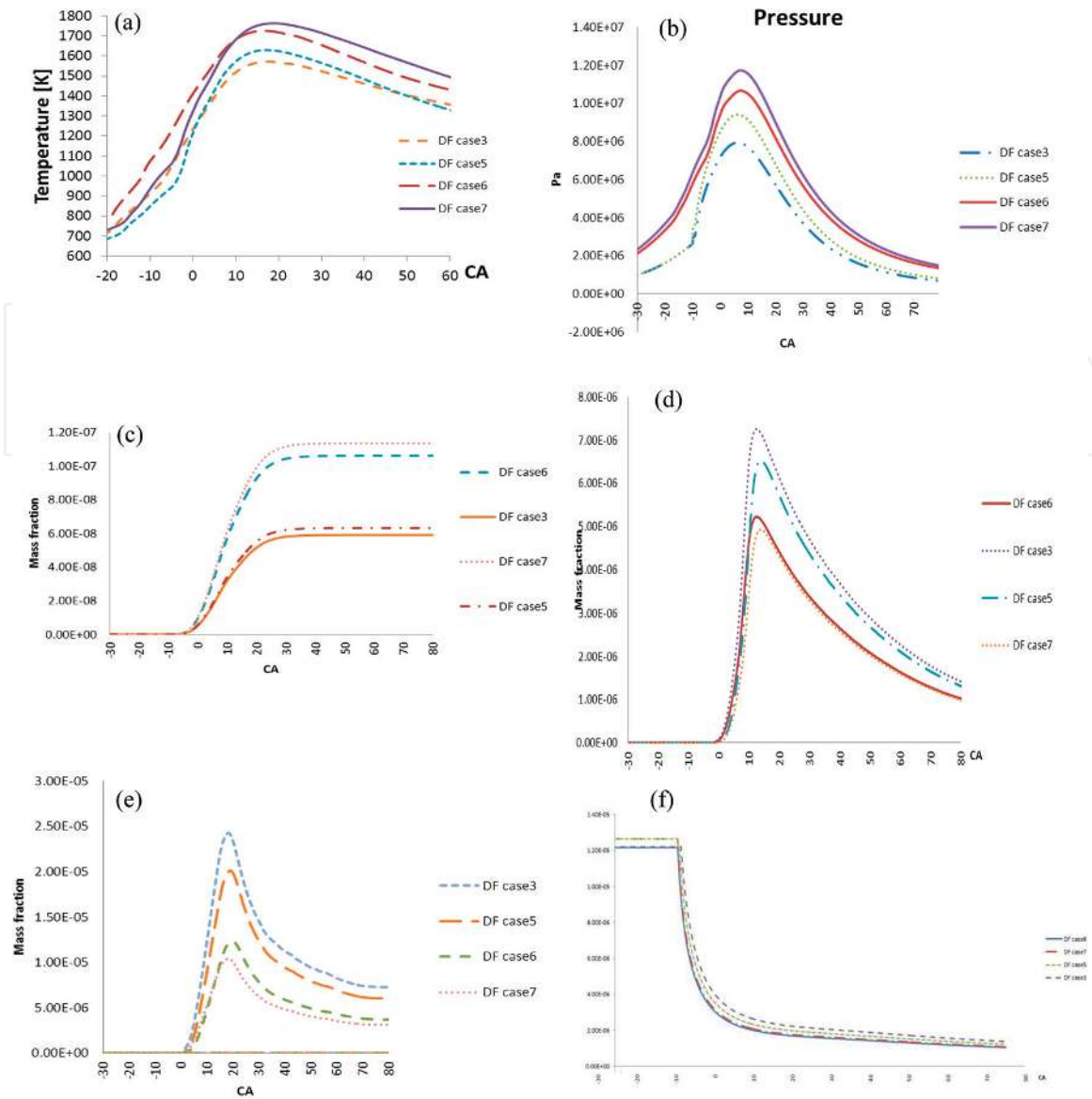


### 3.2 Optimization for CBG combustion

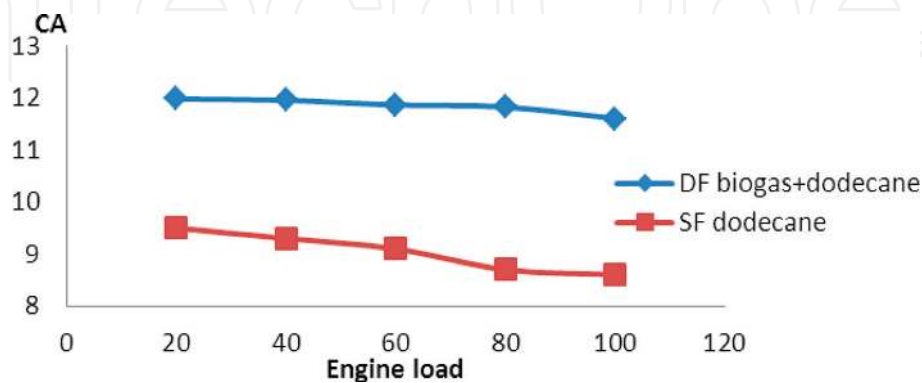
In final simulation, compression ratio of simulated engine was reduced from 18.25:1 to 17.6:1 by widened engine bore diameter to keep more heat inside the cylinder due to the lower heating value of CBG fuel. According to optimization results, larger and smaller valves overlapped engines more suitable for CBG-diesel dual-fuel combustion. Because of the surface to volume ratio effect on combustion temperature, heavy-duty dual-fueled CI engines have better results on combustion performance and unburned HC emissions than light-duty dual-fueled CI engines. Real engine geometry cases have low thermal efficiency due to the valve overlap characteristics of conventional diesel engines. Valve overlap process facilitates scavenging between the intake and exhaust valves. However, in dual-fuel combustion, valve overlapping caused an increase in unburned HC emissions due to leaving of unburned CBG-diesel air fuel mixture from cylinder. At the same time, low valve overlap for dual-fuel CI engine caused incomplete combustion inside the combustion chamber due to the insufficient scavenging process. Valve overlap values also were optimized in final CFD simulation. Because of compression ratio effects on temperature and pressure during the compression phase, the engine compression ratio has an influence on the autoignition phase of the combustion: a reduction prolongs the air/fuel mixing process before combustion. In optimization study, compression ratio was limited in 19:1 due to the knock phenomenon during the compression stroke of CBG-air mixture. Higher compression ratio resulted in lower power due to the autoignition of air fuel mixture. Different works [1, 2, 4] studied on experimental single-cylinder engines showed this significant advantage. Another optimization parameter is SOI for modified dual-fuel engine geometry. In dual fuel-modified engine geometry cases, SOI was reduced to about 18° CA bTDC by optimization study due to the late ignition delay of CBG-air fuel mixture. Single-fuel cases have low ignition delay compared to that of the CBG-diesel dual-fuel combustion as seen in ignition delay figure. In single-fuel cases, diesel fuel has higher cetane number, and this allowed faster combustion than dual-fuel engine cases. Optimized dual-fuel engine cases resulted in better combustion performance by changing SOI, compression ratio, modified engine size, and valve overlap values.

The peaks of pressure and temperature values occurred in DF case6 and DF case7 cases which have lower valve overlap, 19° CA. Therefore, these two cases have indicated that the high temperature reaction (HTR) occurs at around 1200–1300 K. Calculated peak gas temperature for reduced valve overlap cases as shown in **Figure 9a** was 1790 K such as conventional single-fuel diesel combustion; also these cases have lower CO formation and slightly higher NO<sub>x</sub> formation but quite under acceptable emission standards.

As the valve overlap reduced, the peaks of heat release in-cylinder pressure and temperature rapidly increased, and the initiating timings of the reaction were also fastened. In real engine geometry cases, the ignition delay was very long, and ignition had begun very late after pilot started at 12° bTDC. After SOI started at 18° bTDC and valve overlap reduced to 19° CA in optimization study, this led to significant development in engine performance and better combustion control during combustion for CBG-diesel dual-fuel cases. In addition to engine performance development, CO emissions were decreased to very low levels by means of exact combustion. NO<sub>x</sub> emissions resulted in higher DF case6 and DF case7 than other DF cases, but these NO<sub>x</sub> emission values are very low in regard to international emission standards. Similarly, PM emissions resulted in better optimized DF case6 and DF case7 than other DF cases and kept in a reduction trend. Furthermore, it can be said that CBG fuel was burned effectively in regard to other cases (**Figure 9f**) especially for DF case7 which has 19° CA valve overlap value. Besides valve overlap



**Figure 9.** Effects of valve timing on the dualfuel combustion performance. (a) In-cylinder calculated temperature; (b) In-cylinder calculated pressure; (c) NOx emissions; (d) CO emissions; (e) soot emissions; and (f) total fuel mass.



**Figure 10.** Effect of single and dual fuel combustion mode on the ignition delay.

value, optimization results showed that larger CI engines which have big surface to volume ratio have better combustion performance according to light-duty CI engines. It can be concluded that CBG-diesel dual-fuel process with these optimization parameters is more proper for heavy-duty CI engines (Figure 10).



## 4. Conclusion

In this chapter, the engine performance and emission results were studied and compared for the conventional diesel and CBG-diesel dual-fuel operations. CBG and diesel fuels were defined as leading reactants by writing user-defined code. In this work, conventional diesel combustion and dual-fuel pilot diesel combustion were examined. Obtained differences in the results between SF and DF are the result of fuel mixture ratios in the calculation, and this affects the efficiency of the engine. Combustion time is calculated by the software according to chemical compounds and gradients. Fuel ratio can be seen in **Table 5**. International emission standards were taken into consideration in the studies for the wide automotive market, and further studies can be evaluated the next regulations. Although CO<sub>2</sub> is an inert gas in the mixture of air fuel, it is expected that CO<sub>2</sub> ratio affects the emissions. However this is due to the mixture of biogas formation. Higher cetane number of diesel and the faster injection timing shortened the ignition delay, and this reduction is related to a decrease in fuel-rich zone throughout the combustion process.

Due to the volumetric efficiency, in the dual-fuel case concentrations, CO emissions were considerably higher than others under all test conditions. In the dual-fuel cases, CBG gas fuel is replaced by air which causes more CO emissions. The concentrations of CO<sub>2</sub> emissions for dual-fuel cases are obtained under those regarding single-fuel diesel combustion modes. In terms of the ignition delays, conventional diesel combustion exhibited better performance with respect to CBG-diesel cases because of the overall specific heat capacity and oxygen rate. Also, exhaust gas temperature has lower value in dual-fuel cases. BSFC and PM results have better value in the CBG-diesel dual-fuel cases. More oxygen rate in single-fuel cases allowed more CO emissions to oxidize into CO<sub>2</sub> and resulted in higher concentrations of CO<sub>2</sub> emissions.

Case #	CBG rate (kg/h)	Diesel fuel (kg/h)	Engine load (%)	SOI CA
SF case1	—	2.12	20	-12
SF case2	—	3.13	40	-12
SF case3	—	5.22	60	-12
SF case4	—	8.54	80	-12
SF case5	—	11.44	100	-12
DF case1	2.27	1.62	20	-12
DF case2	2.33	2.63	40	-12
DF case3	2.61	4.37	60	-12
DF case4	2.76	6.48	80	-12
DF case5	3.25	7.88	100	-12

**Table 5.**  
*Case studies.*

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