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Investigation of combustion and emissions of natural gas-diesel dual-fuel PPCI engine

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ABSTRACT

Premixed combustion concepts like HCCI and PPCI have attracted much attention, since these concepts offer possibilities to reduce engine out emissions to a low level, while still achieving good efficiency. Recently researchers have shown that premixed combustion concepts with two fuels with a large difference in auto-ignition properties can generate a spatial gradient of fuel-air mixtures and reactivity offers improved fuel efficiency and lower NOx and smoke exhaust emissions when compared to conventional diesel diffusion combustion due to longer ignition delay. The increased availability of natural gas in the world has renewed interest in order to realize fuel cost savings and reduce pollutant emissions due low carbon content. Thus, premixed combustion using natural gas as a less reactive fuel and diesel fuel with a higher reactivity fuel warrants consideration. This study presents results from an experimental study of Natural Gas-Diesel premixed operation on a single cylinder diesel engine. After that, a computational study using the CFD software has been conducted to investigate the in-cylinder phenomena in different situations. The results reveal both low NOx and smoke emissions can be achieved in dual fuel PPCI concept in compared to conventional diesel but at the expense of increasing of unburned hydrocarbon emissions.

1) Introduction

The use of alternative gaseous fuels in diesel engines continues to be strong due to environmental concerns and their relatively increased availability at low prices. Natural gas satisfies previous requirements. In the last decade, diesel engine market has seen a substantial growth in the application of natural gas technologies due to mentioned benefits. Dual fuel engine use diesel fuel as the pilot fuel and gaseous fuel (natural gas in the present work) as the main fuel. It can substantially reduce the NOx emissions while produces almost zero smoke and PM; which is extremely difficult to achieve in DI diesel engines. It can also contribute to the reduction of carbon dioxide (CO2) emissions, due to the low carbon-to-hydrogen ratio. Moreover, it mixes uniformly with air, resulting in efficient combustion to such an extent that it can yield a high thermal efficiency comparable to the diesel version at higher loads [1-3].

Advanced combustion concepts using premixed compression ignition have recently shown potential to meet ultra-low emissions in combination with high thermal efficiencies. Low-temperature combustion strategies include homogeneous charge compression ignition (HCCI) and premixed charge compression ignition (PPCI) have the ability to yield low NOx and soot emissions, while maintaining high fuel efficiency [4-6]. In these ways, by premixing a significant portion of the fuel and relying on long ignition delay (ID) times, high-temperature flame fronts can be avoided, which inhibits NOx formation and reduces heat transfer losses, and the resulting higher ratio of specific heats can further increase efficiency. In addition, rich regions that contribute to high levels of PM formation are avoided with long ID times to promote mixing.

According to latest investigation, dual fuel PPCI was introduced due to mitigate difficulties in controlling the heat release rate and the lack of a combustion phasing control mechanism in premixed charge combustion like HCCI[7-11]. In this strategy two fuels with different auto-ignition characteristics are blended inside the combustion chamber. Combustion phasing is controlled by the relative ratios of these two fuels and the combustion duration is controlled by spatial stratification between them. Due to the high octane number of natural gas and higher auto ignition temperature, it was hypothesized a good fuel for dual fuel PPCI, in which a large reactivity gradient between the two fuels is beneficial in controlling the combustion phasing [12].

The goal of this study is to investigate an early diesel injection strategy on the performance and emissions of natural gas-diesel PPCI combustion. This concept examined here utilizes port-induction natural gas and direct injection of diesel. The primarily focus will be on experimental rather than computational investigations for this paper. Simulation-based studies will be presented in due to investigation of in-cylinder fluid flow, spray dynamics, and fuel oxidation in different cases.

2) Experimental Setup

The engine tests were conducted on four stroke single cylinder water cooled direct injection compression ignition engine with a displacement volume of 662 cc, compression ratio of 17.5:1, developing 5.2 kW at 1500 rpm. Figure 1 shows the overall view of setup. The specifications and the engine operating conditions are listed in Table 1.

Figure 1: Overall view of Experimental Setup

Table 1: Engine specification

Characteristics	Values (Unit)
Type	Single cylinder, water cooled
Fuel	Natural gas- Diesel Oil
Bore×Stroke	78 mm × 110 mm
Engine Speed	1500 RPM
CR	17.5
Number of nozzle hole	6

The setup consists of a diesel engine connected to eddy current type dynamometer for loading and used with necessary instruments for combustion pressure and crank-angle measurements. Sensors are used for interfacing airflow, fuel flow, temperatures and load measurement. The setup has stand-alone panel box consisting of air box, two fuel tanks for duel fuel test, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and engine indicator. The governor of the engine was used to control the engine speed. The engine was provided with a hemispherical combustion chamber with overhead valves operated through push rods. Cooling of the engine was accomplished by circulating water through the jackets on the engine block and cylinder head. A piezoelectric pressure transducer was mounted with the cylinder head surface to measure the cylinder pressure. Exhaust gas opacity was measured using the Hartridge smoke opacity meter. The exhaust gas composition was measured using an exhaust gas analyzer (Make: MRU, Model: Delta 1600S). In order to reduce error in measurement of emissions five readings were recorded and averaged out readings are only presented.

3) Computational Approach

The numerical model for diesel engine with the specifications and operating conditions on Table 1 is carried out using FIRE code. Figure 2 shows the 60° sector computational mesh of combustion chamber in three dimensional at TDC. Since a 6-hole nozzle is used, only a 60° sector has been modeled. This takes advantage of the symmetry of the chamber geometric setup, which significantly reduces computational runtime. Number of cells in the mesh is 27,153 cells at TDC. This fine mesh size will be able to provide good spatial resolution for the distribution of most variables within the combustion chamber. Calculations are carried out on the closed system from IVC at -147.5°CA ATDC to EVO at 145.5°CA ATDC.

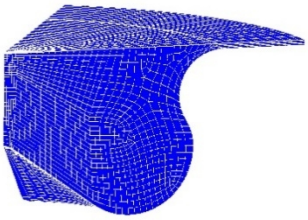


Figure 2: Computational grid at TDC

At the applied code, the compressible, turbulent, three dimensional transient conservation equations are solved for reacting multi-component gas mixtures with the flow dynamics of an evaporating liquid spray by Amsden et al. [13]. The turbulent flows within the combustion chamber are simulated using the RNG k-ε turbulence model which is presented by Han and Reitz [14], modified for variable-density engine flows. The spray module is based on a statistical method referred to as the discrete droplet method (DDM). This operates by solving ordinary differential equations for the trajectory, momentum, heat and mass transfer of single droplets, each being a member of a group of identical non-interacting droplets termed a parcel. Thus one member of the group represents the behavior of the complete parcel.

The Kelvin-Helmholtz Rayleigh-Taylor (KH-RT) model was selected to represent spray breakup [15]. In this model Kelvin-Helmholtz (KH) surface waves and Rayleigh-Taylor (RT) disturbances should be in continuous competition of breaking up the droplets.

The Multi component model [16] was applied for treating the heat-up and evaporation of the droplets.

A multi-dimensional CFD model coupled with chemical kinetics (AVL FIRE-CHEMKIN) was developed in the present study which accounts for the effect of fluid flow, spray dynamics, and fuel oxidation. The CFD code feeds the species to CHEMKIN program as well as the thermodynamic information for each cell. The CHEMKIN simulation provides information on the new species and energy release after solving the chemistry. For the modeling of the diesel fuel oxidation, an internal chemical mechanism was used by AVL-FIRE software. In the present paper, premixed fuel (natural gas) was ignited by pilot diesel fuel. The GRI-Mech 3.0 chemical kinetic mechanism, which consists of 53 species and 325 reactions, was used to describe the premixed natural gas oxidation [17] where the natural gas was represented by methane. The CFD code solves the average transport equations of total mixture mass, momentum and enthalpy.

NOx formation model is derived by systematic reduction of multi-step chemistry, which is based on the partial equilibrium assumption of the considered elementary reactions using the extended Zeldovich mechanism [18] describing the thermal nitrous oxide formation.

The overall soot formation rate is modeled as the difference between soot formation and soot oxidation. Soot formation is based on Hiroyasu model and the soot oxidation rate is adopted from Nagle and Strickland-Constable [19].

All above equations are taken into account simultaneously to predict spray distribution and combustion progress in the turbulent flow field, wall impingement and diesel combustion rate using two stage pressure correction algorithms.

4) Model validation

In order to ensure the accuracy of the simulation, the calculation results were validated with experimental data of diesel and natural gas-diesel dual fuel mode at 100% load. Figure 3 shows the comparisons between the measure and predicted cylinder pressure. This figure shows that the cylinder pressure curves predicted by the present numerical simulation are in good agreement with the experimental results. It can be seen that, the peak pressure and its magnitude are well predicted by the present comprehensive CFD-chemistry model. In general, the simulations are able to capture the cylinder pressure. It is due to time step and computational grid independency of obtained results. Start of diesel fuel injection was set on 10 CA BTDC for both condition and CNG flow rate was 1 kg/hr in dual fuel mode. This figure shows that the calculated heat release rate is in good agreement with their experimental counterparts, indicating good reliability of the mechanisms and models used in the present numerical code to simulate diesel and dual-fuel combustion and performance.

5) Comparison of Conventional Diesel and dual fuel PPCI Combustion

To gauge the emissions and performance benefits of dual fuel PPCI, comparisons are made to conventional diesel combustion experimentally. The CFD modeling is used to explain the differences in-cylinder phenomena and its effect on emissions and performance of dual fuel PPCI and conventional diesel combustion.

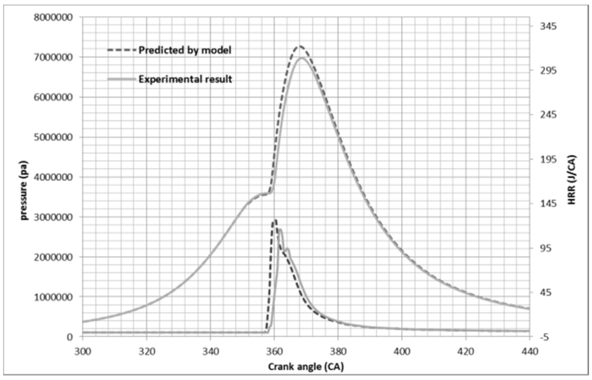


Figure 3a. Numerical and experimental results of diesel mode

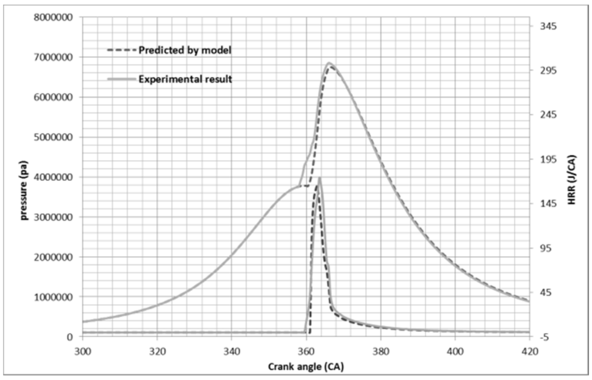


Figure 3b. Numerical and experimental results of dual fuel mode

As mentioned before, dual fuel PPCI strategy can be reached using two fuels with different properties at advanced high reactivity fuel injection in premixed low reactivity fuel. Therefore, the fuel with high octane number and low boiling point is inducted into intake port, while diesel is injected into cylinder, so the stratified distribution of fuel reactivity is formed which leads to stratified combustion when be ignited by compression. The operating conditions for both natural gas-diesel dual fuel PPCI and diesel combustion are shown in Table 2. As the start of injection of diesel fuel was advanced before 27 CA BTDC in 1 kg/h CNG flow rate, engine knock was observed. Therefore it is appropriated SOI for dual fuel PPCI mode.

Table 2: Operation condition

	Dual fuel PPCI	Conv. diesel
Engine speed (rpm)	1500	1500
CR	17.5	17.5
CNG flow rate (kg/hr)	1	0
Diesel injection timing	27 CA BTDC	10 CA BTDC

According to test results, Figures 4 and 5 show a comparison of the cylinder pressure and heat release rate (HRR) for conventional diesel and PPCI dual fuel engine. The comparison of exhaust emissions is shown in Table 3.

The in-cylinder pressure with dual fuel operating mode becomes higher than conventional diesel engine the notably at the rapid premixed combustion period. This is the consequence of a higher heat released under dual fuel mode compared to conventional diesel at that period.

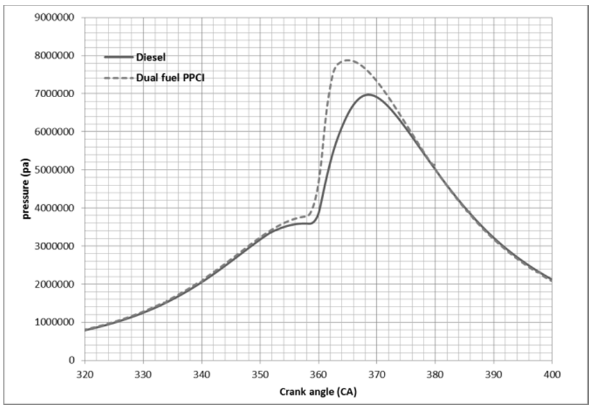


Figure 4. Cylinder pressure for conventional diesel and dual fuel PPCI combustion

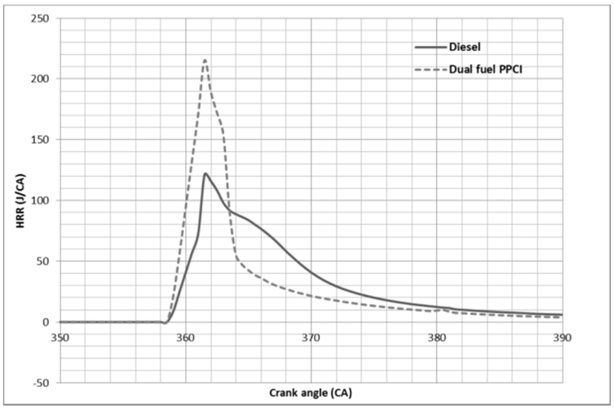


Figure 5. Heat release rate for conventional diesel and dual fuel PPCI combustion

Table 3: Comparing the performance and emissions of dual fuel PPCI and conventional diesel combustion

	Dual fuel PPCI	Conv. diesel
NOx (ppm)	932	1650
HC (ppm)	88	80
CO (%)	0.19	0.13
Smoke (HSU)	48	93

A typical combustion process of dual-fuel engine can be identified to take place in three stages: the first stage is premixed combustion of pilot diesel and a small part of entrained gaseous fuel; the second stage is the premixed combustion of gaseous fuel where the majority of the gaseous fuel is burned; the third stage, which is diminishing and less well-defined, is the diffusion combustion of residual pilot diesel fuel and gaseous fuel. Hence, the in-cylinder pressure peak becomes higher for dual-fuel mode compared to conventional diesel as a consequence of the above-mentioned improvement of the gaseous fuel combustion. It is seen from HRR diagram that ignition delay was significantly increased in dual fuel mode. This increase in ignition delay period is mainly due to low reactivity of natural gas fuel and lowering of compression temperature. However, a part of the air was replaced by natural gas and it reduced the oxygen concentration in the fuel mixture. The rapid heat release is caused by the premixed burn of natural gas, controlled by chemical kinetics, compared to the diesel fuel only case where the burn rate is controlled by the rate of fuel injection.

As widely recognized, the formation of nitrogen oxides is favored by high oxygen concentration and higher charge temperature. NOx concentration of dual fuel mode is lower in comparison to the conventional diesel one. The higher oxygen concentration of conventional diesel leads to more significant level of NOx emissions.

Hydrocarbon emissions were found to be higher for the dual fuel combinations compared to neat diesel engine due to the CNG charge, which causes lean, homogeneous, low-temperature combustion, resulting in less complete combustion. This is because small amount of pilot fuel cannot propagate fast and far enough to ignite the whole premixed fuel mixture. Under dual fuel operation, the filling of the crevice volumes with unburned mixture of air and gaseous fuel during compression and combustion is an important source dominating the formation of HC emissions.

The exhaust emissions of carbon monoxide were higher for CNG-diesel dual fuel operations compared to conventional diesel combustion. In dual fuel mode, CNG is induced into intake manifold and form a premixed mixture at intake stroke and this leads to lower local fuel-rich region. The reduced in-cylinder temperature in this mode together with lower oxygen concentration causes the incomplete combustion which results in more carbon monoxide and unburned hydrocarbon.

Diesel-CNG dual fuel operation shows lower smoke opacity compared to conventional diesel engine. The results show clearly that dual fuel mode is a very efficient technique to reduce soot emissions. Accordingly, the use of natural gas allows a drastic reduction of soot emissions. This is to be expected since natural gas, where methane represents the main component, has very small tendency to produce soot.

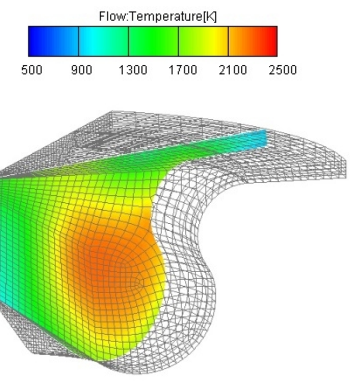
To investigate the in-cylinder flow field, Figure 6 shows contours of predicted temperatures for the dual fuel PPCI and diesel combustion in a vertical cut-plane with temperature contours at 370 CA. The significantly heterogeneous nature of the combustion process in the conventional diesel case is evident when compared to the RCCI case. Since the dual fuel PPCI strategy allows sufficient time for mixing prior to ignition, the peak local equivalence ratio during the combustion process is lower and so the lean operation results in a lower peak temperature for dual fuel PPCI.

6) Conclusions

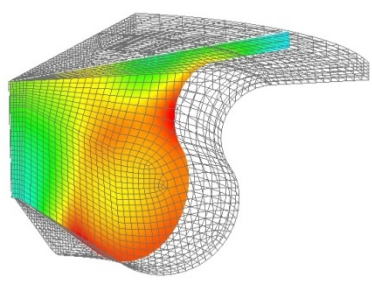
In this study, the effect of dual fuel PPCI strategy on combustion characteristics and pollutants emissions of an existing diesel engine using natural gas as primary fuel and neat diesel as pilot fuel has been examined experimentally. Also, modeling of dual fuel PPCI combustion was successfully conducted using natural gas and diesel fuel in a computational study. Model was validated with experimental data of natural gas/diesel dual fuel engine. The following conclusions can be drawn from this study:

- Compared to conventional diesel combustion, dual fuel PPCI shows higher heat release rate in the premixed-combustion phase. Consequently, the in-cylinder pressure peak for dual fuel mode becomes higher than the corresponding one for conventional diesel.
- Results show that the use of natural gas for dual fuel mode is a very efficient technique to reduce soot emissions. Dual fuel PPCI strategy indicates extremely low engine out NOx and smoke emissions. This is due to homogenous combustion with lower peak local equivalence ratio during the combustion and natural gas fuel properties.
- Hydrocarbon emissions are significantly higher for dual fuel mode. CO emissions are also higher for dual fuel mode. Dual fuel PPCI combustion shows that incomplete combustion losses (in this study, are shown with the percent of UHC and CO) increased.

Therefore, dual fuel PPCI strategy can be applied on existing DI diesel engines for controlling soot and NOx emissions which are noticeable challenges of diesel engines.



Dual fuel Combustion, 370 CA



diesel Combustion, 370 CA

Figure 6. Temperature contours at 370 CA.

List of Symbols

PPCI	Partially premixed compression ignition
HCCI	Homogenous charge compression ignition
CR	Compression ratio
CA	Crank angle
BTDC	Before top dead center
IVC	Inlet valve close
EVO	Exhaust valve open
HRR	heat release rate

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بررسی احتراق و آلایندگی در مفهوم احتراق پیش آمیخته جزیی در یک موتور دوگانه سوز با استفاده از سوختهای گاز طبیعی و دیزلی

حسن خاتم نژاد[1*]، شهرام خلیل آریا[2]، صمد جعفرمدار[3]، ناگاراج بناپورمت[4]، سید مصطفی میرسلیم[5]

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چکیده

مفاهیم احتراق پیش آمیخته نظیر احتراق اشتعال تراکمی همگن و احتراق پیش آمیخته جزیی همگن به سبب کاهش آلاینده های خروجی تا یک میزان کم در کنار بازدهی مناسب، توجه زیادی را به خود معطوف نمودند. تحقیقات اخیر نشان می دهد، مفاهیم احتراق پیش آمیخته با استفاده از دو سوخت با اختلاف زیاد در ویژگی های خود اشتعالی سبب تولید تغییرات واکنش پذیری مخلوط در محفظه احتراق و افزایش زمان تاخیر در اشتعال شده و منجر به کاهش همزمان دوده و اکسید ازت در مقایسه با مفهوم دیزلی مرسوم می شود. سوخت گاز طبیعی بدلیل افزایش میزان تولید، هزینه پایینتر و آلایندگی کمتر می تواند با توجه به واکنش پذیری پایین به عنوان یک سوخت مناسب به همراه سوخت دیزل با واکنش پذیری بالا در یک مفهوم احتراق پیش آمیخته جزیی استفاده شود. در این تحقیق نتایج بررسی تجربی یک احتراق پیش آمیخته با استفاده از سوخت گاز طبیعی و دیزل ارائه شده و پس از آن بررسی جریان داخل محفظه احتراق با استفاده از تحلیل سه بعدی به روش CFD انجام می شود. نتایج نشان دهنده کاهش همزمان آلاینده های دوده و اکسید ازت و افزایش هیدروکربنهای نسوخته در حالت پیش آمیخته جزیی دوگانه سوز در مقایسه با مفهوم دیزلی مرسوم می باشد.

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