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A numerical investigation on combustion and emission characteristics of a dual fuel engine at part load condition

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HIGHLIGHTS

- Flame structure in dual fuel engines depends on operating conditions.
- At part load, diesel liquid drops evaporate lately and far from injector nozzles.
- At part load, UHC is remained in the most remote areas from diesel fuel injector.
- Enlargement of diesel combustion region could ignite lean methane/air mixture.

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ABSTRACT

Dual fuel engines are more attractive due to lower emission levels in comparison with conventional diesel engines particularly at full loads. But it is required to study dual fuel combustion process with more details at part loads due to the poor performance and high CO and UHC emissions at these conditions. In the present study, numerical modeling of OM-355 dual fuel (injection of diesel pilot fuel to premixed mixture of air and methane) engine has been performed by using KIVA-3V code at part and full loads. Sub-models of the code were modified to simulate the fuel spray atomization, combustion and pollutants emissions processes, accurately. Results indicate that in-cylinder pressure, heat release rate and exhaust emissions predictions are in good agreement with experiments at all loads. Results show that a lean premixed natural gas mixture is ignited slowly. The slow progress of combustion process at part load, leads the heat release to be drawn more toward the expansion stroke which causes incomplete combustion, and consequently high amounts of UHC and CO will be emitted. It is found that at part loads, areas that are influenced by diesel diffusion flames are ignited and premixed natural gas flame could not be propagated properly. Hence development of diesel diffusion flame is required to burn lean natural gas mixture. But at full load, in addition to the diesel diffusion flames, premixed natural gas flame could be propagated suitably. Also, at part load because of low gas temperature in the environment of diesel spray and low diesel fuel temperature, diesel liquid droplets evaporate lately which are far from injector nozzles. Hence, it causes diesel diffusion flame from spray of each injector nozzles to be developed distinctly. It can be deduced that the flame structure is affected by operating conditions. Finally the effect of increasing the diesel fuel quantity on improving methane combustion is studied. The studied strategy could help to improving natural gas combustion due to enlarge the size of diesel combustion region.

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1. Introduction

The compression ignition engine of the dual fuel type is a respectable alternative for conventional diesel engine. The dual fuel engine has many advantages over conventional diesel engine include economical, technical and environmental benefits. Dual fuel engine has been employed in a wide range of applications to utilize various gaseous fuel resources meanwhile exhaust gas emissions are minimized without excessive increase in the engine cost compared to that of conventional diesel engines [1]. Dual fuel engines emit low amounts of carbon dioxide due to the low C/H proportion of methane [2,3]. Also dual fuel engines have a notable potential to reduce Nitrogen Oxides (NO_x) and Particulate Matter (PM) emissions. PM emissions could be reduced with the substitu-

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Nomenclature

A	pre-exponential factor of Arrhenius equation
A'	coefficient of eddy dissipation model
a	radius of parent drops (μm)
B_0	adaptable empiric coefficients
B_1	adaptable empiric coefficients
$\overline{C}$	local average concentration (g/cm^3)
E_a	activation energy (kcal/mol)
k	turbulence kinetic energy (m^2/s^2)
$\left(\frac{O}{F}\right)_{st}$	stoichiometric oxygen fuel ratio
R	universal gas constant (kcal/mol K)
r	radius of the child drop (μm)
T	temperature (K)
t	time (m s)

Greek symbols

Ω	growth rate (1/m s)
τ	breakup time (m s)
Λ	surface wave length (μm)
ε	turbulence dissipation rate (m^2/s^3)

Abbreviations

ALE	Arbitrary Lagrangian Eulerian
ATDC	After Top Dead Center
BDC	Bottom Dead Center
CAD	Crank Angle Degree
CO	Carbon Monoxide
CFD	Computational Fluid Dynamic
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Open
HCCI	Homogeneous Charge Compression Ignition
IVC	Inlet Valve Closure
NO	Nitric Oxide
NO_2	Nitrogen Dioxide
NO_x	Nitrogen Oxides
PM	Particulate Matter
RCCI	Reactivity Charge Compression Ignition
RR	Reaction Rate
SI	Spark Ignition
SOI	Start of Injection
UHC	Unburned Hydrocarbon

tion of a large quantity of diesel fuel with natural gas and NO_x emissions of a dual fuel engine depend on the parameters of the diesel injection [3]. On the other hand, the low cost and easily converting possibility of the conventional diesel engines to dual fuel mode is the other benefit [4]. As the gaseous fuel is mixed with intake air in inlet manifold, the mixture formation is modified greatly, and then, inside the cylinder, the mixture undergoes a multi-point ignition due to combustion of a pilot diesel fuel spray. Then, flame propagation occurs throughout the premixed natural gas and air mixture. Thus, dual fuel operation with natural gas fuel can yield a high thermal efficiency, almost comparable to the same engine operating on diesel fuel at higher loads. However, dual fuel engines suffer from lower performance parameters, higher Carbon Monoxide (CO) and Unburned Hydrocarbon (UHC) emissions and consequently higher amount of chemical availability of unburned fuels at part loads. The main reason for this poor part load performance is due to the presence of very lean mixtures and poor flame propagation in the lean mixture [1,5]. The lean premixed natural gas mixtures are hard to ignite and slow to burn [1].

Karim et al. showed that with very lean mixture and small pilot quantities, the flames initiated by pilot fuel are unable to develop throughout the combustion chamber [6,7]. Tao concluded experimentally that Nitric Oxide (NO) emissions in dual fuel engines is much less than conventional diesel engines. Also, it was shown that late combustion and low temperature burning of gas result in low NO in dual fuel engines at all conditions [7,8]. Micklow and Gong studied performance and emission characteristics of a dual fuel engine at part load conditions using KIVA-3V code. Results showed that 40% of methane remained unburned near the cylinder wall [9]. Kusaka et al. used KIVA-3V code coupled with Chemkin-II for modeling of a dual fuel engine at part load conditions. They used detailed chemical kinetics including 173 reactions and 43 species. Results showed that combustion of premixed mixture of natural gas and air is very slow, so natural gas burns incompletely [10]. They also investigated dual fuel engines at 2/5 load condition by using a multi-dimensional model combined with the detailed chemical kinetics including 290 reactions and 57 species. In this study, the effect of premixed charge concentration on combustion was examined. At 2/5 load due to the low concentration of the gaseous fuel in the mixture, combustion occurs incompletely which cause to decrease the thermal efficiency and to increase the UHC. When four cylinder dual fuel engine operates with two cylinder, combustion occurs completely due to high concentration of gaseous fuel in the mixture [11]. Reitz et al. investigated combustion of a dual fuel engine by using a multi-dimensional Computational Fluid Dynamic (CFD) model with KIVA-3V code. They represented that, the characteristic-time model could predict engine combustion, performance and emission characteristics very well for cases with natural gas up to 90 percent (10 percent diesel pilot quantity). If the energy supplied by the diesel fuel is less than about 10%, a flame propagation model such a Spark Ignition (SI) engine should be used for natural gas combustion [12]. Liu et al. studied the dual fuel engine combustion with numerical and experimental methods. A 3D-CFD model based on KIVA was developed. The simulation includes a reduced detailed chemical kinetics for the diesel fuel and detailed chemical kinetics for the gaseous fuel component. They concluded that dual fuel engine combustion may be an effective approach to utilize gaseous fuel–air mixtures of low energy density and to achieve stable combustion, a minimum absolute quantity of diesel pilot is needed [13]. Hosseinzadeh et al. concluded that in dual fuel engine without hot Exhaust Gas Recirculation (EGR), because of the incomplete combustion at part load condition, chemical availability of unburned fuel is too much. In this condition, 28% of total input chemical availability is exhausted to the atmosphere [14]. Puduppakkam et al. studied the use of a five-component gasoline surrogate and a one-component diesel surrogate by using a multi-dimensional CFD model coupled with detailed chemical kinetics to simulate Homogeneous Charge Compression Ignition (HCCI) and Reactivity Charge Compression Ignition (RCCI) dual fuel engine operation. The results showed that the model predicted the combustion and emissions very well. Also the model represented that the most-reactive fuel component, n heptane (component in both diesel and gasoline surrogates) is consumed at a much faster rate than other less-reactive gasoline surrogate components such as iso-octane and toluene [15]. Maghbouli et al. investigated the combustion process in the diesel and dual fuel engine with utilizing a 3D-CFD model coupled with chemical kinetics including 57 reactions and 42 species. They showed that by shifting the diesel combustion to the dual fuel combustion mode, the ignition delay time is increased [16]. Maghbouli et al. studied the combustion process under knocking conditions in dual fuel engines. They

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