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Effect of swirl ratio on NG/diesel dual-fuel combustion at low to high engine load conditions

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HIGHLIGHTS

- The effect of swirl ratio on the combustion of dual-fuel IC engine is investigated.
- Swirl motion enhances mixture preparation, diffusion, and NG flame propagation.
- The benefit of swirl motion is offset by higher heat loss at very high swirl ratio.
- An optimum swirl ratio of 1.5 is found for all examined load-speed conditions.

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ABSTRACT

Recent regulation on pollutant and greenhouse gas (GHG) emissions has exerted great pressure on diesel engine industries which generate significant amount of GHG and pollutants. The concept of lean burn pilot ignited natural gas/diesel dual-fuel (NDDF) combustion is considered as one of the most suitable engine platforms to meet emissions and fuel economy regulations over a short to medium term. However, a major challenge is the slightly lower fuel efficiency and high level of methane (CH_4) and carbon monoxide (CO) emissions, especially under low to medium engine load conditions. This paper numerically investigates the influence of swirl ratio on the combustion performance and emissions of a NDDF engine under low to high load conditions. The results at low load-low speed condition and retarded injection timing of 14 crank angle degrees before top dead center (BTDC) suggest that increasing swirl ratio from 0.5 to 1.5 significantly improves fuel efficiency and CH_4 and CO emissions. However, under the same engine load-speed condition but at advanced injection timing of 30 crank angle degrees BTDC, increasing swirl ratio deteriorates the fuel efficiency and CH_4 and CO emissions. Under a medium load-high speed condition, swirl ratio significantly improves diffusion combustion and turbulent flame propagation of natural gas. The results show that OH radical propagates more rapidly in the azimuthal direction when increasing the swirl ratio from 0.5 to 1.5. Further increase in the swirl ratio causes the peak pressure to exceed the limit (160 bar). At high load-low speed condition, increasing the swirl ratio significantly improves diesel diffusion and flame propagation of natural gas, which consequently enhances fuel efficiency. Under this engine load-speed condition, OH radical distribution shows that the combustion progresses rapidly within each jet in both the azimuthal and radial directions. Considering fuel efficiency and emissions, a swirl ratio of 1.5 is found to be the optimum. Overall, it is concluded that swirl motion may provide better mixture preparation, diesel diffusion, and natural gas flame propagation. However, this benefit may not persist under very high swirl ratio (swirl ratio > 1.5) due to higher heat losses.

1. Introduction

Compression ignition is the most common combustion strategy in heavy-duty diesel engine market due to its higher fuel efficiency. However, diesel engines face great challenges for meeting stringent emissions regulations beside the fact that their fuel efficiency is high. NO_x and soot emission limits imposed by the emissions regulations for diesel engines have become more and more stringent over the years, which consequently fuelled further motivations to develop advanced combustion strategies to achieve at the same time high efficiency and low emissions. The advanced low temperature combustion (LTC) strategies offer attractive combustion and emission characteristics and have

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Abbreviations

AMR	Adaptive Mesh Refinement
ATDC	After Top Dead Center
BMEP	Break Mean effective Pressure
BTDC	Before Top Dead Center
CAD	Crank Angle Degree
CFD	Computational Fluid Dynamic
CO	Carbon Monoxide
DIT	Diesel Injection Timing
EGR	Exhaust Gas Recirculation
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
HCCI	Homogeneous Charge Compression Ignition
HRR	Heat Release Rate
IMEP	Indicated Mean Effective Pressure
IFE	Indicated Fuel Efficiency
IVC	Intake Valve Closing
IVO	Intake Valve Opening
LTC	Low Temperature Combustion
NDDF	Natural Gas/Diesel Dual-Fuel
NG	Natural Gas
NO_x	Nitrogen Oxides
NTC	No Time Counter
RANS	Reynolds-Averaged Navier-Stokes
RCCI	Reactivity Controlled Compression Ignition
RNG	Re-Normalization Group
rpm	Revolution Per Minute
SOC	Start of Combustion
SR	Swirl Ratio
TDC	Top Dead Center
UHC	Unburned Hydrocarbon

the potential to meet the emissions and fuel economy regulations of the future. Two key features of the LTC strategies are low combustion temperature and long ignition delay time [1]. The low combustion temperature inhibits NO_x formation and the long ignition delay provides sufficient time for air-fuel mixing, reduces fuel-rich zones, and prevents soot formation. Moreover, the LTC strategies have been shown to be well-suited for alternative fuels (e.g. natural gas [2,3]). Compression ignition LTC modes have been demonstrated to result in high fuel efficiency through a combination of lean operation, optimal combustion phasing near top dead center (TDC), short combustion duration, and reduced heat transfer [4]. LTC strategies can be governed by different mechanisms depending on the timing of the fuel injection event [5]. When the injection timing is early in the cycle (or port fuel injection), ignition timing is kinetically controlled, which is often termed as homogeneous charge compression ignition (HCCI) combustion. Practical operation of HCCI combustion is still challenging due to the lack of control of the combustion phasing or duration. Dual-fuel combustion can achieve control of the combustion phasing and duration by adjusting the global fuel reactivity and cylinder stratification of the mixture reactivity using the differences in fuel's physical and chemical properties [4]. For dual-fuel combustion, a low reactivity fuel is premixed with the intake air and a second fuel with higher reactivity (usually diesel) is directly injected. The charge distribution of dual-fuel combustion is more heterogeneous than HCCI, as it consists of lean and rich regions at the time of ignition. The combustion phasing and duration are controlled by the ratio of the two fuels and injection strategy of the high reactivity fuel. Because of the higher ignition temperature, natural gas is an ideal candidate for the role of the low reactivity fuel in dual-fuel combustion.

The main constituent of natural gas, which is CH4, has lower carbon content and offers a significant reduction of CO_2 emissions (if burnt completely) compared to diesel fuel. Compared to diesel, a 20–30% CO_2 emissions reduction is achievable, and this number can be further improved if natural gas comes from a renewable source such as biogas [6]. Moreover, one of the major targets for developing NDDF combustion is to minimize the injected pilot fuel quantity in order to reduce diesel fuel dependency. In recent years, natural gas has drawn substantial interest as a low reactivity fuel in dual-fuel combustion and some original engine manufacturers have commercialized dual-fuel engine based on premixed natural gas [7]. NDDF combustion tends to retain most positive features of conventional diesel engines [8]. In addition, NDDF mode has drawn much attention due to other advantages. For instance, dual-fuel combustion can be achieved without major engine hardware modifications, which not only reduces the engine development costs but also means that the engine can revert to fully diesel combustion if needed to [9]. However, higher CH4 and CO emissions and lower fuel efficiency (particularly at low engine loads) compared to fully diesel combustion strategy are the main drawbacks of NDDF engines [10]. Many studies were conducted to improve efficiency and CH4 and CO emissions of NDDF combustion. Most of these studies focused on advancing diesel injection timing (DIT) [11,12], adopting hot exhaust gas recirculation (EGR) [13], or increasing diesel injection pressure [14]. A much smaller subset was focused on the effect of swirl ratio on NDDF combustion. For instance, Jha et al. [15] numerically investigated the effect of different initial swirl ratios (0.05–1.5) on NDDF combustion under low engine load conditions (BMEP = 3.3 bar and rpm = 1500). They found that increasing swirl ratio from 0.05 to 1.2 decreased the unburned hydrocarbon (UHC) and CO emissions by 60% and 50%, respectively. However, the reductions in HC and CO emissions were accompanied by 26% increase in NO_x emissions. Carlucci et al. [16] examined the effect of three different intake ports including swirl, tumble, and swirl/tumble on CH4/diesel dual-fuel combustion under low load (IMEP = 4 bar and rpm = 1500) and high load (IMEP = 8 bar and rpm = 2000) conditions. They noted that the charge bulk flow induced by the swirl intake port helped improve the charge mixing of the diesel spray and the propagation of the turbulent flame front. This consequently led to the oxidation of air-natural gas mixtures located farther away from the pilot ignition nuclei, which resulted in lower unburned hydrocarbon emissions. In agreement with the findings reported in [15], they found that NO_x emissions slightly increased when swirl intake port was used.

Introducing a swirl flow into the combustion chamber increases the burning rate and extends the flammability limits. A swirl with an appropriate strength improves fuel efficiency and smoke emissions due to increased rates of fuel-air mixing [17]. Disadvantages of swirl ratio include higher heat loss, higher peak pressure (especially at high engine load conditions), and higher pressure rise rate (PRR) which may lead to greater engine noise [18]. Although NO_x emissions were observed to increase with increasing swirl, it can be overcomed by slightly retarding the injection timing and also by adopting EGR. The overall benefits of swirl flow may be achieved if an optimum swirl intensity is attained, where this will depend on the engine operating conditions.

From the above discussion, a limited number of studies attempted to examine the effect of the initial swirl ratio on NDDF combustion performance. However, to achieve an optimum swirl ratio, an extensive study should be performed under different engine load-speed conditions. In the present study, a computational study based on CONVERGE 2.4 software was conducted under three engine load/speed conditions, a low load-low speed (BMEP = 4.05 bar and rpm = 910), a high load-low speed (BMEP = 17.6 bar and rpm = 1120), and a medium load-high speed (BMEP = 11.24 bar and rpm = 1750) in order to identify an optimum swirl ratio. In addition to the pressure, heat release rate (HRR) and engine out emissions, the heat losses, OH radical distribution, and charge temperature were all analyzed to investigate how significant the