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An effort to enhance hydrogen energy share in a compression ignition engine under dual-fuel mode using low temperature combustion strategies

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HIGHLIGHTS

- H_2 energy share increased from 18% with DDM to 36% with WDM (water injection).
- H_2 energy share improved marginally with retarded injection timing mode (RDM).
- Energy efficiency increased with increasing amount of H_2 in dual-fuel engine.
- NO_x emission decreased with water injection and retarded pilot fuel injection.
- HC, CO and smoke emissions increased slightly with low temperature combustion.

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ABSTRACT

A limited hydrogen (H_2) energy share due to knocking is the major hurdle for effective utilization of H_2 in compression ignition (CI) engines under dual-fuel operation. The present study aims at improvement of H_2 energy share in a 7.4 kW direct injection CI engine under dual-fuel mode with two low temperature combustion (LTC) strategies; (i) retarded pilot fuel injection timing and (ii) water injection. Experiments were carried out under conventional strategies of diesel dual-fuel mode (DDM) and B20 dual-fuel mode (BDM); and LTC strategies of retarded injection timing dual-fuel mode (RDM) and water injected dual-fuel mode (WDM). The results explored that the H_2 energy share increased significantly from 18% with conventional DDM to 24, and 36% with RDM, and WDM respectively. The energy efficiency increased with increasing H_2 energy share under dual-fuel operation; however, for a particular energy share of 18% H_2, it decreased from 34.8% with DDM to 33.7% with BDM, 32.7% with WDM and 29.9% with RDM. At 18% H_2 energy share, oxides of nitrogen emission decreased by 37% with RDM and 32% with WDM as compared to conventional DDM due to reduction of in-cylinder temperature, while it increased slightly about 5% with BDM. It is emerged from the study that water injection technique is the viable option among all other strategies to enhance the H_2 energy share in the engine with a slight penalty of increase in smoke, hydrocarbon, and carbon monoxide emissions.

1. Introduction

Abundant availability and cost advantage of natural gas (NG) are the predominant motivating factors for the original engine manufacturers (OEMs) to shift transportation vehicles including railroad and marine to NG fueled vehicles. Commercial dual-fuel (diesel-NG) engines from different OEMs such as Cummins (Models: QSK50, KTA38-G2A, and KTA50-G3) [1], Caterpillar (Models: C-10 DFNG, C-12 DFNG, C-15 DFNG, M 46 DF, and M 43 C) [2], Wartsila (Models: 34DF and 50DF) [3], and MAN Diesel (Models: 32/40DF and 51/60DF) [4] are booming up in the real world market. Dual-fuel compression ignition (CI) engines offer a greater benefit of fuel flexibility i.e., the engines could run automatically either under dual-fuel mode (if gaseous fuel NG/hydrogen (H_2)/Biogas/producer gas, etc. is available) or under traditional single fuel mode in absence of the gaseous fuel. The other benefit is that the existing CI engines could easily be converted to dual-fuel engines with a simple retrofit gas injection system. In this direction, GE Transportation is the first OEM, which has developed a retrofit kit for inducting NG into existing diesel locomotive engines for dual-fuel operation [5]. However, in the current

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Nomenclature

Abbreviations

θ	degree crank angle
$\dot{m}$	mass flow rate of fluid/species, kg/s
a	molar ratio of fuel to air
A	area of cylinder, m^2
ATDC	after top dead centre
b	molar ratio of fuels
B	cylinder bore, m
B20	diesel (80% volume) biodiesel (20% volume) blend
B_c	clearance between piston and cylinder liner, m
BDM	B20 dual-fuel mode
BP	brake power, kW
c	molar ratio of water to fuel
C	carbon atom
CA	crank angle, degrees
CI	compression ignition
CO	carbon monoxide, g/kW h
CO_2	carbon dioxide
COV	coefficient of variation,%
C_p	specific heat at constant pressure, kJ/kg-K
C_v	specific heat at constant volume, kJ/kg-K
DDM	diesel dual-fuel mode
E	Internal energy, kW
ECU	electronic control unit
GE, AVL, MAN diesel	name of the companies
GHG	greenhouse gas
h	Convective heat transfer coefficient, kW/m^2-K
H	hydrogen atom
H_2	hydrogen gas
H_2O	water
HC	hydrocarbon emission, g/kW h
h_{tl}	crevice top land height, m
IMEP	indicated mean effective pressure, N/m^2
LHV	Lower heating value, kJ/kg
M	molecular weight
N	nitrogen atom
n	number of moles
N_2	nitrogen
NG	natural gas
NO	nitric oxide
NO_x	oxides of nitrogen, g/kW h
n_{TP}	total number of moles of products
$\varnothing$	equivalence ratio
O	oxygen atom
O_2	oxygen
OEM	original engine manufacturer
p	in-cylinder pressure, N/m^2
Q	heat energy/heat transfer/heat release, kW
RDM	retard injection dual-fuel mode
RPR	rate of pressure rise, bar/degree CA
R_u	universal gas constant, J/mol-K
SOI	start of injection, degree CA
SWC	specific water consumption, g/kW h
T	temperature, K
TDC	top dead centre
V	cylinder instantaneous volume, m^3
$V_{crevice}$	crevice volume, m^3
v_{gas}	velocity of gases inside the cylinder, m/s
W	work energy, kW
WDM	water injection dual-fuel mode
WTF	water fuel ratio
x	mole fraction
γ	specific heat ratio

scenario of depletion of fossil fuels (coal, petroleum products, and NG) and stringent emission norms, H_2 is considered as a promising candidate for future energy systems. Moreover, H_2 is suggested as a transportation fuel and clean energy source for long term needs to mitigate carbon dioxide (CO_2) emission for tackling global warming and climate changes [6]. The studies by Chandran Govindaraju and Tang revealed the fact that the CO_2 emission annual growth rate in India and China was about 5.7% in a span of eight years [7]. Pongthanaisawan and Sorapipatana found a drastic increase in greenhouse gas (GHG) emissions from 27.5 to 53.9 Mt of CO_2-eq over a period of 17 years from Thailand's transport sector [8]. Utilization of H_2 in CI engines under dual-fuel is one of the options to reduce the GHG emissions drastically. For instance, the experimental investigations carried out by Subramanian and Chintala indicated 46% reduction in CO_2 emission with the use of a small amount of H_2 energy share of 20% in a H_2 dual-fuel engine at full load condition [9]. However, a large amount of H_2 substitution in CI engines is a major challenge to the researchers due to knocking problem. The utilization of various amounts of H_2 energy shares under dual-fuel mode was reported in the literature (Table 1). It is observed from the table that the researchers could only achieve a small amount of H_2 substitution (volume/energy basis) in CI engines, in order to avoid knocking combustion, power drop and efficiency losses. However, some researchers achieved the maximum H_2 energy share about 38% with a penalty of energy efficiency [13]. In order to address the issue of limited H_2 substitution in CI engines, several strategies may be adopted. For example, low temperature combustion (LTC) strategies such as retardation of pilot fuel injection timing, reduction of compression ratio, and addition of diluents (nitrogen, helium, CO_2, exhaust gas and water). In this context, the present investigation is aimed at enhancement of H_2 energy share in a CI engine with two different LTC strategies; (i) pilot fuel injection timing retardation, (ii) use of water as an additive. The relevant literature on these strategies is briefly illustrated below.

Table 1
Literature review on percentage of H_2 fuel substitution in CI engine.

S. no.	Reference	CI engine specifications	H_2 injection type	Amount of H_2 fuel (volume/energy) substitution
1	Saravanan and Nagarajan [10]	Make: Kirloskar, AV1; P = 3.7 kW @ 1500 rpm; CR = 16.5:1	Port injection	7.5 lpm at rated load
2	Saravanan et al. [11]	– do –	Port injection	12.9% energy share at 100% load
3	Saravanan and Nagarajan [12]	– do –	Manifold injection	5.5 lpm at full load
4	Mathur et al. [13]	P = 4 kW @ 1500 rpm; CR = 17:1	Manifold injection	38% energy share at full load
5	Bose and Maji [14]	Make: Kirloskar; P = 5.2 kW @ 1500 rpm; CR = 17.5:1	Manifold injection	0.15 kg/h at 80% load
6	Wu H-W and Wu Z-Y [15]	Make: Kubota RK-125; P = 9.2 kW @ 2400 rpm; CR = 18:1	Manifold injection	20% energy share at 60% load