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# Experimental investigations on effect of different compression ratios on enhancement of maximum hydrogen energy share in a compression ignition engine under dual-fuel mode

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ABSTRACT

The study deals the effect of different compression ratios on maximum hydrogen energy share, thermal efficiency, and emissions in a 7.4 kW direct injection CI (compression ignition) engine under dual-fuel mode. Experimental tests were conducted on the engine with three different compression ratios (19.5:1 base ($CR_1$), 16.5:1 ($CR_2$), and 15.4:1 ($CR_3$)) using hydrogen as main fuel and diesel as pilot fuel at 100% load and constant speed of 1500 rpm. Knock limited maximum hydrogen energy share enhanced significantly from 19% with $CR_1$ to 59% and 63% with $CR_2$ and $CR_3$. The percentage reductions of $NO_x$ emission in the engine with $CR_2$ and $CR_3$ are about 43% and 48% respectively. HC (Hydrocarbon) and CO (carbon monoxide) emissions reached to zero level with the hydrogen addition at all compression ratios. The optimum compression ratio is 16.5:1 in view of higher thermal efficiency and lower emissions (HC, CO, smoke, and $NO_x$). A notable conclusion emerged from the study is that the reduction in compression ratio of the engine is a promising option for the improvement in hydrogen energy share and thermal efficiency along with benefits of lower emissions.



## 1. Introduction

$H_2$ (Hydrogen) based dual-fuel CI (compression ignition) engines offer remarkable benefits including high thermal efficiency, high combustion efficiency, high degree of constant volume combustion, less combustion irreversibility, and near zero carbon based emissions (HC (hydro carbon), CO (carbon monoxide) and smoke) [1]. However, these dual-fuel engines suffer a major obstacle of limited hydrogen energy substitution for their effective utilization in future energy systems. The maximum hydrogen energy share in a dual-fuel engine is typically restricted by onset of knocking. Knocking could be defined as abnormal combustion phenomenon which constraints the improvement in engine performance. Knocking combustion could be detected in several ways such as in-cylinder pressure based detection, cylinder block vibration measurement, acoustic wave measurement analysis, heat transfer based analysis, etc. Both RPR (rate of pressure rise) and heat release rate together can be used for an analysis of the knock tendency in a CI engine [2]. Torregrosa et al. supported the fact that the knocking combustion in a CI engine is directly proportional to its maximum rate of pressure rise [3]. A thermodynamic model was developed for knock detection in a SI (spark ignition) engine operated with different gaseous fuels [4]. If a CI engine operates with knocking, the engine gets severe damage including breakage of piston rings, piston melting, and erosion of cylinder head. So, CI engines typically operate with less hydrogen energy share for knock prevention in the engine. A suitable technology needs to be identified and assessed for substitution of high hydrogen energy share in CI engines under dual-fuel mode for knock free operation.

The details of literature review on the maximum amount of hydrogen utilized in diesel engines at different loads under dual-fuel mode are given in Table 1. It could be observed from the table that the maximum hydrogen energy share achieved with a timed manifold injection technique is in the range of 6%–16.4% (with brake mean effective pressure range: 5 bar–9.2 bar). However, this energy share could further be increased to 30% at lower load (lower brake mean effective pressure of 2.2 bar) with a port injection method. It was established that the hydrogen energy share in CI engines decreases with increase in engine load. For example, the hydrogen energy share increased from 18.8% at 100% load to 48.4% at 50% load in a 7.4 kW rated power CI engine [1]. The various problems associated to engine operation with high

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**Table 1**
Literature summary of the maximum hydrogen energy share at different BMEPs in hydrogen fueled dual-fuel engines.

| Reference | Engine details | Strategy used | Maximum amount of $H_2$ substitution |
|---|---|---|---|
| Saravanan et al. [14] | Nc = 1, CR = 16.5:1, BMEP = 5.4 bar | Timed manifold injection | 6.7% energy share |
| Edwin et al. [15] | Nc = 1, CR = 17.5:1, BMEP = 5.3 bar | Timed manifold injection | 12.7% energy share |
| Mathur et al. [10] | Nc = 1, CR = 17.5:1, BMEP = 4.9 bar | Timed manifold injection | 14.8% energy share (without power loss) |
| de Morais et al. [16] | Nc = 4, CR = 17:1, BMEP = 6.5 bar | Timed manifold injection | 20% energy share |
| Nguyen and Mikami [17] | Nc = 1, CR = 16.7:1, BMEP = 7.3 bar | Timed manifold injection | 10% volume of intake air (or) 15% energy share (approx.) |
| Bose et al. [18] | Nc = 1, CR = 17.5:1, BMEP = 6.4 bar | Timed manifold injection | $H_2$ flow rate of 0.15 kg/h (or)15.6% energy share (approx.) |
| Yadav et al. [19] | Nc = 1, CR = 17.5:1, BMEP = 5.3 bar | Timed manifold injection | 16.4% energy share at 100% load |
| Christodoulou and Megaritis [20] | Nc = 4, CR = 18.2:1, BMEP = 9.2 bar | Timed manifold injection | 8% volume of intake air (or) 12.8% energy share (approx.) |
| Dhole et al. [21] | Nc = 4, CR = 17.5:1, BMEP = 4.3 bar | Continuous manifold induction | 25% energy share |
| Wu H-W and Wu Z-Y [22] | Nc = 1, CR = 18:1, BMEP = 2.2 bar | Timed port injection | 30% energy share |
| Saravanan et al. [23] | Nc = 1, CR = 16.5:1, BMEP = 5.4 bar | Timed port injection | 10% energy share |
| Wu H-W and Wu Z-Y [24] | Nc = 1, CR = 18:1, BMEP = 4.4 bar | Timed port injection | 20% energy share |
| Adnan et al. [12] | Nc = 1, CR = 17.5:1, BMEP = 6.1 bar | Timed port injection | A constant flow rate of 5 lpm (or) 11.5% energy share (approx.) |

hydrogen energy share are high rate of pressure rise, high in-cylinder peak pressure, too advanced combustion, high in-cylinder peak temperature, autoignition of premixed hydrogen-air charge, and loss of available work [5–8]. Miyamoto et al. reported the occurrence of auto-ignition phenomenon in a dual fuel CI engine when the hydrogen fraction is higher than 8% volume [6]. This similar trend with diesel-propane fuel was reported by Polk et al. [7]. Wong and Karim revealed reasons for auto-ignition of hydrogen-air charge are due to high polytropic index of hydrogen, higher in-cylinder temperature, and increasing preignition chemical reactions [8]. Selim, in his experimental investigation, found an increase in mass of gaseous fuel (Liquefied Petroleum Gas/Methane/Natural gas) would lead to significant increase in the maximum rate of pressure rise in a single cylinder variable compression indirect injection diesel engine (Ricardo E6: 9 kW rated power) [9]. He also concluded that the gaseous fuel existing in the combustion chamber could be more favorable to auto-ignition.

A number of specific strategies including retarded injection timing of liquid fuel (pilot fuel), use of high cetane number pilot fuel, EGR (exhaust gas recirculation), water injection, and compression ratio reduction could provide some solutions to the enhancement of the hydrogen energy share in a dual-fuel engine. A very few studies are available in literature on the enhancement of the hydrogen energy share using water injection and compression ratio reduction techniques. For example, the hydrogen energy share was increased from 14.8% with conventional dual-fuel mode to 66% with dual-fuel mode using water addition [10]. Similarly, the other study indicates the improvement of the hydrogen energy share from 19% with conventional duel fuel mode to 36% with water added dual-fuel mode [11]. Adnan et al. suggested an optimum water injection timing of 20° CA (crank angle) after TDC (top dead center) for better performance of a hydrogen dual-fuel CI engine [12]. Masood et al. reported an increment in the amount of hydrogen substitution from 0.096 kg/h with 24.5:1 compression ratio to 0.138 kg/h with 16.35:1 compression ratio in a hydrogen fueled dual-fuel engine [13]. With this motivation, the present study is aimed at an enhancement of the hydrogen energy share in a direct injection CI engine under dual-fuel mode with reduction of its compression ratio. Comparative analysis has been made among the results with three different compression ratios; 19.5:1 (base), 16.5:1, and 15.4:1.

## 2. Compression ratio reduction details

Compression ratio of the engine was reduced with insertion of extra metal gaskets (MG1, MG2, and MG3) between cylinder head and cylinder barrel as shown in Fig. 1. An addition of metal gasket increases the clearance volume of the engine, resulting in reduction of the compression ratio. The thickness of three metal gaskets namely MG1, MG2, and MG3 are 0.4, 1.2, and 0.6 mm respectively. The compression ratio obtained with addition of each metal gasket could be determined using Eqs. 1–14. The detailed calculations for compression ratio reduction are given in Appendix A.

$$\text{Compression ratio (CR)} = \text{Total volume}/\text{Clearance volume} \tag{1}$$

$$CR = (V_s+V_c)/V_c \tag{2}$$

$$CR = 1 + (V_s/V_c) \tag{3}$$

$$CR_1 = 1 + (V_s/V_{c1}) \tag{4}$$

$$CR_2 = 1 + (V_s/V_{c2}) \tag{5}$$

$$CR_3 = 1 + (V_s/V_{c3}) \tag{6}$$

$$V_{c1} = V_{MG1} + V_{CG} + V_{BW} \tag{7}$$

$$V_{c2} = V_{MG1} + V_{MG2} + V_{CG} + V_{BW} \tag{8}$$

$$V_{c3} = V_{MG1} + V_{MG2} + V_{MG3} + V_{CG} + V_{BW} \tag{9}$$