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Research Article

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Influence of Injector Nozzle-Hole Configuration on the Performance, Combustion, and Emission Characteristics of a Jatropha Biodiesel–Hydrogen Dual-Fuel Diesel Engine

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ABSTRACT

The present study experimentally investigates the influence of injector nozzle-hole configuration on the performance, combustion, and emission characteristics of a direct-injection compression-ignition engine operated with Jatropha biodiesel-hydrogen dual fuel. Experiments were conducted on a single-cylinder, four-stroke diesel engine at 3000 rpm and 20–100% of rated load. Three injector configurations, namely N3, N4, and N5, representing three-, four-, and five-hole nozzles, respectively, were investigated. The individual orifice diameter was varied to maintain approximately matched total geometric orifice area, with a maximum difference of about 1.3%. Diesel (D100) and a 15% Jatropha biodiesel blend (JB15) were used as reference fuels, while JB15 with 8 L/min hydrogen was employed for the principal dual-fuel investigation. The results showed that hydrogen enrichment improved engine performance and reduced carbon-containing emissions. At full load, the JB15+8H₂-N4 configuration achieved the highest brake thermal efficiency of 31.8% and the lowest brake-specific fuel consumption of 0.274 kg/kWh. Compared with JB15-N4, hydrogen addition improved BTE by 5.6% and reduced BSFC, CO, HC, and smoke opacity by 5.8%, 14.4%, 14.0%, and 17.1%, respectively. However, NO_x increased by 10.7% due to enhanced combustion intensity and higher combustion temperatures. The N4 configuration also produced the highest peak cylinder pressure and heat-release rate. Overall, N4 provided the most favourable balance between performance and carbon-related emissions under JB15-hydrogen operation, although further optimization is required to mitigate the associated NO_x increase.

Keywords: *Jatropha biodiesel; Hydrogen dual-fuel; Injector nozzle configuration; Nozzle-hole number; Emission characteristics.*

1. INTRODUCTION

The increasing demand for transportation energy, depletion of conventional petroleum resources, and growing concern over greenhouse-gas and pollutant emissions have intensified research into cleaner and more sustainable fuels for compression-ignition (CI) engines. Diesel engines continue to be widely used in transportation, agricultural, power-generation, and industrial applications because of their high thermal efficiency, durability, and reliability. However, conventional diesel combustion produces nitrogen oxides (NO_x), particulate matter (PM), smoke, carbon monoxide (CO), and unburned hydrocarbons (HC), while continued dependence on petroleum-derived fuels raises concerns regarding energy security and environmental sustainability. Consequently, increasing attention has been directed toward renewable fuels, alternative gaseous fuels, and advanced fuel-injection strategies that can reduce emissions while maintaining the performance advantages of CI engines [1]–[3].

Jatropha curcas biodiesel has attracted considerable interest as a renewable liquid fuel because Jatropha oil is non-edible and can therefore reduce direct competition with food resources. Biodiesel can also be used in existing diesel-engine technology with relatively limited modifications. However, compared with petroleum diesel, Jatropha biodiesel generally exhibits higher viscosity and density, different volatility characteristics, and a lower heating value. These properties influence fuel injection, spray atomization, evaporation, fuel-air mixing, ignition, and combustion development. Previous investigations have reported acceptable engine performance with Jatropha biodiesel and its blends, although changes in combustion characteristics and, under certain conditions, increased NO_x emissions have also been observed [4]–[13]. Therefore, improving the utilization of Jatropha biodiesel requires not only suitable blend selection but also effective control of fuel-injection and combustion processes.

Hydrogen has emerged as a promising supplementary fuel for CI engines because of its high mass-specific heating value, wide flammability range, high flame-propagation velocity, and absence of carbon in the fuel molecule. In a hydrogen dual-fuel CI engine, hydrogen is generally introduced through the intake system to form a premixed hydrogen-air mixture, while a small quantity of diesel or biodiesel is directly injected into the cylinder to initiate combustion. Hydrogen supplementation can improve combustion and thermal efficiency and can reduce carbon-containing emissions under suitable operating conditions [14]–[18]. Nevertheless, excessive hydrogen addition may increase combustion temperature and consequently promote NO_x formation. The effectiveness of

hydrogen dual-fuel operation is therefore strongly dependent on hydrogen flow rate, pilot-fuel properties, injection timing, injection pressure, engine load, and combustion phasing.

The combination of Jatropha biodiesel and hydrogen provides an attractive pathway for reducing the dependence on petroleum-derived fuel while retaining the compression-ignition characteristics of the engine. In this configuration, Jatropha biodiesel acts as the directly injected liquid fuel, whereas hydrogen serves as the supplementary gaseous fuel. Previous studies have demonstrated that Jatropha biodiesel-hydrogen dual-fuel combustion is sensitive to injection pressure, injection timing, hydrogen substitution, and engine operating conditions [15]-[21]. Kumar et al. reported that injection pressure and injection timing substantially influence the combustion, performance, and emission characteristics of a Jatropha biodiesel-hydrogen dual-fuel engine [15]. Other studies using different biodiesel feedstocks have similarly demonstrated that hydrogen enrichment and injection-related parameters interact strongly and should be considered together when optimizing dual-fuel engine operation [16]-[21]. These findings indicate that the liquid-fuel injection system plays a critical role in determining the effectiveness of hydrogen supplementation.

Among the parameters governing fuel injection, injector nozzle geometry is particularly important because it controls the distribution and spatial dispersion of the liquid fuel inside the combustion chamber. Nozzle-hole number, orifice diameter, spray angle, and injector opening pressure influence spray momentum, penetration, atomization, evaporation, and fuel-air mixing. Increasing the number of nozzle holes generally produces a greater number of smaller spray plumes and can improve fuel distribution within the combustion chamber. However, an excessive number of spray plumes may reduce penetration or increase plume interaction and wall impingement, depending on the combustion-chamber geometry and operating conditions. Therefore, nozzle geometry must be considered in relation to fuel properties and engine operating conditions rather than as an isolated injector-design parameter [22]-[28].

Previous experimental and numerical studies have established that nozzle-hole number can significantly affect diesel-engine combustion and emissions. Kang et al. investigated different combinations of nozzle-hole number and orifice diameter and reported that changes in nozzle configuration influence injection characteristics, fuel distribution, spray interaction, and combustion development [29]. Other investigations involving diesel and biodiesel blends have reported changes in brake thermal efficiency, brake-specific fuel consumption, cylinder pressure, heat-release characteristics, and exhaust emissions with variations in nozzle-hole number [30]. Studies of spray penetration, flame development, and injector internal flow have further demonstrated that nozzle configuration can modify the spatial and temporal characteristics of fuel delivery [31]-[34]. These findings establish nozzle-hole configuration as an important injector-design parameter for controlling the combustion behavior of CI engines.

The importance of nozzle-hole configuration may become more pronounced under biodiesel-hydrogen dual-fuel operation because the liquid pilot fuel must initiate combustion within a premixed hydrogen-air environment. Palani and Subramanian investigated three-, four-, and five-hole injectors in a dual-fuel diesel engine operating with diesel, a 20% *Prosopis juliflora* biodiesel blend, and hydrogen supplementation [35]. Their results demonstrated that nozzle-hole configuration influenced engine performance, combustion characteristics, and emissions, with the four-hole configuration providing favorable overall performance among the investigated configurations. The study confirmed that injector geometry can significantly influence the interaction between biodiesel spray development and hydrogen-assisted combustion. However, the biodiesel feedstock used in that investigation was *Prosopis juliflora*, rather than Jatropha biodiesel.

Despite the reported benefits of hydrogen supplementation and the demonstrated influence of nozzle-hole configuration on biodiesel-hydrogen combustion, relatively limited experimental information is available on the effect of injector nozzle-hole configuration specifically under Jatropha biodiesel-hydrogen dual-fuel operation. Previous studies involving biodiesel-hydrogen dual-fuel operation have primarily examined parameters such as injection pressure, injection timing, hydrogen substitution, fuel composition, and operating conditions [15]-[21], whereas the effect of nozzle-hole configuration has received comparatively limited systematic investigation. Furthermore, results obtained using other biodiesel feedstocks cannot be directly generalized to Jatropha biodiesel because differences in density, viscosity, volatility, oxygen content, cetane characteristics, and heating value can modify spray development and combustion behavior. Therefore, a controlled experimental comparison of different nozzle-hole configurations under Jatropha biodiesel-hydrogen dual-fuel operation is required.

Accordingly, this study experimentally investigates the influence of injector nozzle-hole configuration on the performance, combustion, and emission characteristics of a single-cylinder DI CI engine operated with JB15-hydrogen dual fuel. Three injector configurations, namely N3, N4, and N5, corresponding to three-, four-, and five-hole nozzles, respectively, are evaluated at a constant hydrogen flow rate of 8 L/min. The individual orifice diameter is adjusted among the three configurations to maintain approximately matched total geometric orifice

area, with the maximum difference limited to approximately 1.3%. Engine speed, injector opening pressure, injection timing, nominal spray angle, liquid-fuel composition, hydrogen flow rate, and load are maintained consistently across the nozzle configurations. D100 and JB15 are used as reference conditions, while the principal comparison focuses on N3, N4, and N5 under JB15-hydrogen dual-fuel operation. Brake power, brake-specific fuel consumption, brake thermal efficiency, exhaust gas temperature, cylinder pressure, heat-release characteristics, ignition delay, and exhaust emissions are evaluated to determine the influence of injector configuration on dual-fuel combustion and overall engine performance. The study therefore provides a controlled experimental assessment of injector nozzle-hole configuration under Jatropha biodiesel-hydrogen operation while minimizing the influence of differences in total geometric orifice area.

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Test Engine and Experimental Setup

The experiments were conducted on a single-cylinder, four-stroke, air-cooled, vertical compression-ignition (CI) engine equipped with a direct-injection (DI) fuel system. The engine has a displacement of 406 cm³, with a bore of 86 mm and a stroke of 70 mm. The engine was rated at 7.5 kW at the specified rated speed. All experiments were conducted at a constant engine speed of 3000 rpm, while the dynamometer was used to vary the engine load.

The primary objective of the experimental investigation was to evaluate the influence of injector nozzle-hole configuration on the performance, combustion, and emission characteristics of a Jatropha biodiesel-hydrogen dual-fuel engine. Three multi-hole injector configurations, designated N3, N4, and N5, corresponding to three-, four-, and five-hole nozzles, respectively, were investigated under a constant hydrogen flow rate of 8 L/min. Conventional diesel (D100) and Jatropha biodiesel blend (JB15) were used as reference fuels to establish the baseline engine characteristics, whereas the principal comparative analysis was performed for JB15-hydrogen dual-fuel operation with the N3, N4, and N5 injector configurations. The engine specifications used in the experiments are presented in Table 1.

Table 1: Specifications of the Test Engine

Parameter	Specification
Engine type	Single-cylinder, four-stroke CI engine
Cooling system	Air cooled
Cylinder orientation	Vertical
Displacement	406 cm ³
Bore	86 mm
Stroke	70 mm
Compression ratio	19:1
Rated power	7.5 kW
Operating speed	3000 rpm
Fuel injection system	Direct injection
Injector nozzle configurations	N3 (3-hole), N4 (4-hole), N5 (5-hole)
Reference liquid fuel	Diesel
Biodiesel blend	JB15
Hydrogen flow rate	8 L/min
Loading device	Eddy Current Dynamometer

The experimental setup consisted of the test engine, eddy-current dynamometer, liquid-fuel supply system, hydrogen induction system, air-flow measurement arrangement, combustion data-acquisition system, and exhaust emission measurement equipment. The dynamometer was used to apply controlled loads while maintaining the engine speed at the specified operating condition. Fuel consumption was measured using a calibrated fuel-flow measuring device, while hydrogen was supplied to the intake system through a regulated gas-supply arrangement and calibrated flow-control device.

In-cylinder pressure was measured using a pressure transducer coupled with a crank-angle encoder. The pressure and crank-angle signals were synchronously acquired through a data-acquisition system and subsequently processed to determine the combustion characteristics, including peak cylinder pressure and heat-release rate. The

exhaust system was equipped with a calibrated gas analyzer for measuring gaseous emissions and a smoke meter for determining smoke opacity.

A schematic representation of the experimental setup is shown in Figure 1, while a photograph of the actual engine test bench used for the experiments is presented in Figure 2.

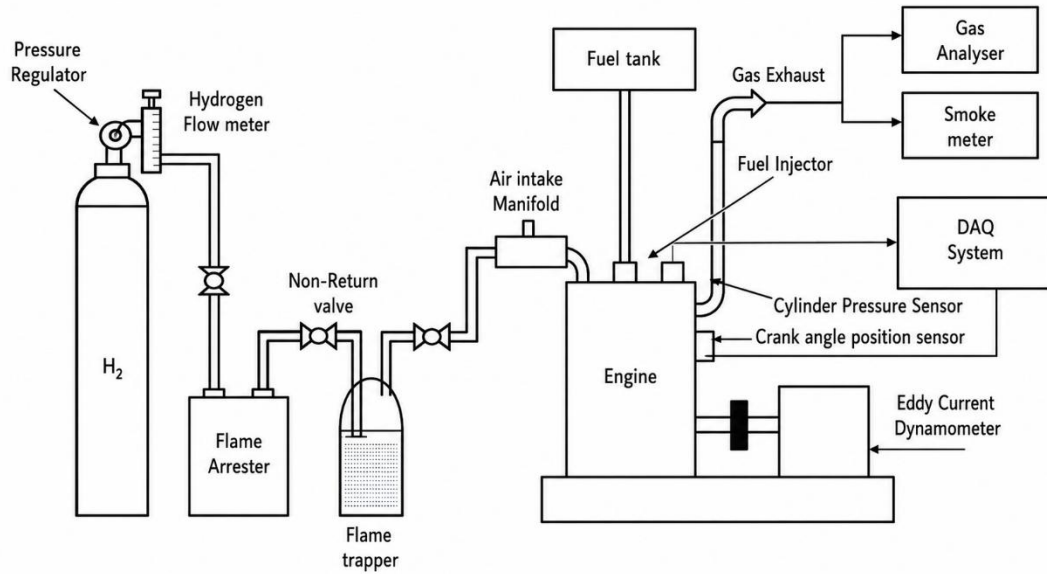


Figure 1: Schematic representation of the experimental setup



Figure 2: Photograph of the engine test bench

2.2 Injector Nozzle Configurations

Injector nozzle configuration, characterized by the number of nozzle holes and the corresponding individual orifice diameter, was selected as the principal experimental variable. Three multi-hole injector configurations,

designated N3, N4, and N5, corresponding to three-, four-, and five-hole nozzles, respectively, were investigated. The injector body, injector opening pressure, injection timing, nominal spray angle, fuel composition, hydrogen flow rate, engine speed, and load conditions were maintained unchanged among the three configurations. The individual orifice diameter was varied to maintain approximately comparable total geometric orifice area, thereby minimizing differences in overall geometric flow area among the tested nozzles.

The injector opening pressure was maintained at 220 bar, and the injection timing was fixed at 23° crank angle before top dead centre (BTDC). A nominal spray angle of approximately 120° was maintained for all configurations. The individual orifice diameters of N3, N4, and N5 were 0.26, 0.225, and 0.20 mm, respectively. The corresponding total geometric orifice areas were 0.1592, 0.1590, and 0.1571 mm², respectively, representing a maximum difference of approximately 1.3%. The nozzle specifications are presented in Table 2.

Table 2: Specifications of Injector Nozzle Configurations

Parameter	N3	N4	N5
No. of nozzle holes	3	4	5
Orifice diameter, mm	0.26	0.225	0.20
Total geometric orifice area, mm ²	0.159	0.159	0.157
Injector opening pressure, bar	220	220	220
Injection timing, °BTDC	23	23	23
Nominal spray angle, °	120	120	120
Injector type	Multi-hole DI	Multi-hole DI	Multi-hole DI

Changing the nozzle-hole configuration alters the number, size, and spatial distribution of the fuel spray plumes and consequently affects spray penetration, atomization, evaporation, and fuel-air mixing. The N3 configuration produces fewer and relatively larger spray plumes, whereas N4 and N5 generate a greater number of smaller plumes. These differences can modify the spatial distribution of the liquid fuel within the combustion chamber and consequently influence the interaction between the biodiesel spray and the premixed hydrogen-air mixture.

For the primary comparative experiments, JB15 was used as the directly injected liquid fuel and hydrogen was supplied at a constant flow rate of 8 L/min. Engine speed, load, injector opening pressure, injection timing, fuel composition, hydrogen flow rate, and nominal spray angle were kept constant for the N3, N4, and N5 configurations. Thus, the experimental design maintained approximately comparable total geometric orifice area while systematically varying the nozzle-hole configuration and corresponding individual orifice diameter.

2.3 Fuel Selection and Preparation

Commercial diesel was used as the reference fuel, while Jatropha biodiesel was selected as the renewable fuel component. Jatropha biodiesel was produced from Jatropha curcas seed oil through a transesterification process. The triglycerides present in the vegetable oil were converted into fatty acid methyl esters (FAME) using methanol and an alkaline catalyst. Following completion of the reaction, the mixture was allowed to settle, enabling separation of the glycerol-rich phase from the biodiesel phase. The biodiesel was subsequently washed to remove residual catalyst, glycerol, soap, and other impurities and was then dried before blending.

A 15% Jatropha biodiesel and 85% diesel blend by volume, designated as JB15, was prepared and used as the liquid fuel for the biodiesel-based experiments. Commercial diesel (D100) was used as the reference fuel, while JB15 was used to establish the effect of biodiesel blending relative to diesel operation. The subsequent dual-fuel experiments were performed using JB15 as the directly injected liquid fuel with hydrogen supplied at a constant flow rate of 8 L/min. Accordingly, D100 and JB15 served as baseline/reference conditions, whereas JB15 + 8 L/min H₂ constituted the primary dual-fuel condition used to evaluate the influence of injector nozzle-hole configuration.

The fuel conditions considered in the investigation are summarized in Table 3.

Table 3: Reference and primary experimental fuel conditions

Fuel code	Fuel composition	Operating condition
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D100	100% diesel	Reference baseline
JB15	15% Jatropha biodiesel + 85% diesel	Biodiesel baseline
JB15+H ₂	JB15 + 8 L/min H ₂	Primary dual-fuel investigation

2.4 Hydrogen Supply and Induction

Hydrogen was supplied from a high-pressure cylinder through a pressure regulator, non-return valve, and calibrated flow-control device. The regulated hydrogen was introduced upstream of the intake valve, allowing it to premix with the incoming air before entering the combustion chamber.

The hydrogen flow rate was maintained at 8 L/min throughout the dual-fuel experiments. Maintaining a constant hydrogen flow rate enabled the effect of injector nozzle-hole configuration to be investigated without introducing variations in hydrogen substitution. The hydrogen supply system was inspected for leakage before each test, and the flow rate was verified using the calibrated flow meter.

Hydrogen addition can significantly modify the combustion process because of its high diffusivity, wide flammability range, high flame-propagation rate, and high mass-specific heating value. Therefore, the interaction between hydrogen enrichment and the spray characteristics produced by the different nozzle configurations was considered in the analysis.

2.5 Fuel Properties

The important physical and chemical properties of the test fuels are presented in Table 4. The fuel properties adopted for the calculations are presented in Table 4 based on the supplier specifications. These properties influence fuel injection, spray atomization, evaporation, ignition, combustion, and emission formation.

Table 4: Properties of Test Fuels

Property	Diesel	JB15	Hydrogen
Density, kg/m ³	840	842	0.0899 at 20 °C and 1 atm
Kinematic viscosity at 40 °C, cSt	2.80	2.95	-
Lower heating value, MJ/kg	43.0	42.2	120.5
Flash point, °C	65	74	-
Cetane number	50	48	-
Flammability range, vol.% in air	-	-	4-75

The JB15 blend exhibits a slightly higher viscosity and lower heating value than diesel. The presence of biodiesel also provides chemically bound oxygen, which can promote oxidation during combustion. In contrast, hydrogen has a considerably higher heating value on a mass basis and a high flame propagation velocity. These differences in fuel properties can significantly affect the combustion process and emission formation when hydrogen is combined with Jatropha biodiesel.

2.6 Experimental Procedure

Before each test, the engine was operated until stable thermal conditions were achieved. All experiments were conducted at a constant engine speed of 3000 rpm, and the dynamometer was used to apply the required load. Tests were conducted at 20%, 40%, 60%, 80%, and 100% of rated brake power, corresponding to approximately 1.5, 3.0, 4.5, 6.0, and 7.5 kW, respectively.

At each operating point, sufficient time was allowed for the engine to reach steady-state conditions before measurements were recorded. Engine speed, torque, liquid-fuel consumption, exhaust-gas temperature, cylinder pressure, and exhaust emissions were measured at each load condition.

The experimental programme was conducted in two stages. In the first stage, D100 and JB15 were tested to establish the reference engine performance and emission characteristics under the given injector configurations. These tests provided baseline values for comparison with the biodiesel-hydrogen dual-fuel operation.

In the second and principal stage, JB15 was used as the directly injected liquid fuel and hydrogen was introduced into the intake manifold at a constant flow rate of 8 L/min. The hydrogen flow rate was maintained at 8 L/min under the flow-meter reference conditions. The corresponding hydrogen mass flow rate was calculated from the measured volumetric flow rate and hydrogen density at the specified reference conditions. The JB15-

hydrogen dual-fuel experiments were conducted separately with the N3, N4, and N5 injector configurations. Engine speed, injection pressure, injection timing, hydrogen flow rate, spray angle, and load conditions were maintained constant for all three nozzle configurations.

Therefore, the primary experimental comparison was between N3, N4, and N5 under identical JB15-hydrogen dual-fuel operating conditions. The D100 and JB15 results were used as reference cases to establish the baseline performance and emission characteristics of diesel and biodiesel-blend operation.

The same experimental procedure and operating conditions were maintained for all nozzle configurations so that differences observed among N3, N4, and N5 could primarily be attributed to the nozzle-hole configuration and corresponding individual orifice diameter under approximately comparable total geometric orifice area.

2.7 Engine Performance Analysis

The principal performance parameters evaluated were brake power (BP), brake-specific fuel consumption (BSFC), brake thermal efficiency (BTE), and exhaust gas temperature (EGT). Engine torque was measured using the dynamometer, while engine speed was monitored using a calibrated speed sensor.

The brake power was calculated from the measured torque and engine speed using

$$BP = \frac{2\pi NT}{60\,000} \quad (1)$$

where BP is the brake power in kW, N is the engine speed in rpm, and T is the brake torque in N·m.

The brake-specific fuel consumption was calculated as

$$BSFC = \frac{\dot{m}_f}{BP} \quad (2)$$

where $\dot{m}_f$ is the liquid-fuel mass flow rate in kg/h and BP is the brake power in kW.

For single-fuel operation, the brake thermal efficiency was calculated as

$$BTE = \frac{BP}{\dot{m}_f LHV_f} \times 100 \quad (3)$$

where LHV_f is the lower heating value of the corresponding liquid fuel.

For dual-fuel operation, the energy contribution of both JB15 and hydrogen was considered. The total-fuel brake thermal efficiency was calculated using

$$BTE_{dual} = \frac{BP}{\dot{m}_f LHV_f + \dot{m}_{H_2} LHV_{H_2}} \times 100 \quad (4)$$

where $\dot{m}_{H_2}$ is the mass flow rate of hydrogen and LHV_{H_2} is the lower heating value of hydrogen. This definition accounts for the total chemical energy supplied to the engine and provides an appropriate basis for comparing single-fuel and dual-fuel operation.

2.8 Combustion Analysis

The combustion characteristics were determined from the variation of in-cylinder pressure with crank angle. A pressure transducer was used to measure the instantaneous cylinder pressure, while a crank-angle encoder provided synchronization between the pressure signal and crank position.

The pressure data were recorded over consecutive engine cycles and averaged to minimize cycle-to-cycle variations. The resulting pressure-crank-angle data were used to determine peak cylinder pressure (PCP), ignition delay, maximum heat-release rate, crank angle corresponding to peak pressure, and combustion duration. Ignition delay was determined as the crank-angle interval between the start of fuel injection and the start of combustion, with the latter identified from the heat-release-rate curve using the adopted threshold criterion.

The net heat-release rate was calculated using the first-law analysis of the closed portion of the engine cycle:

$$\frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \quad (5)$$

where $dQ_{net}/d\theta$ is the net heat-release rate with respect to crank angle, P is the instantaneous cylinder pressure, V is the instantaneous cylinder volume, θ is the crank angle, and γ is the ratio of specific heats.

The combustion analysis was used to evaluate the influence of nozzle-hole configuration on fuel-air mixing, ignition development, premixed combustion, diffusion-controlled combustion, and heat-release characteristics. The pressure and heat-release results were analyzed together with the corresponding performance and emission parameters.

2.9 Exhaust Emission Measurement

Exhaust emissions were measured under steady-state operating conditions using a calibrated exhaust-gas analyzer. The gaseous emissions considered were carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x), and carbon dioxide (CO₂). Smoke opacity was measured using a calibrated smoke meter.

Before each measurement, the gas analyzer was allowed to reach its operating condition and was calibrated according to the manufacturer's recommended procedure. The sampling probe was positioned in the exhaust line to obtain a representative exhaust-gas sample. Emission values were recorded after the analyzer readings became stable.

Variation in nozzle-hole configuration can modify spray penetration, atomization, fuel-air mixing, local equivalence ratio, and combustion temperature, thereby affecting gaseous emissions and smoke formation. Hydrogen enrichment may promote oxidation of carbon-containing species because of its high reactivity and flame-propagation characteristics; however, the associated changes in combustion temperature can also influence NO_x formation. Accordingly, CO, HC, NO_x, CO₂, and smoke were considered collectively to assess the environmental performance of the different injector configurations.

2.10 Uncertainty Analysis

An uncertainty analysis was performed to assess the reliability of the measured and calculated results. The major sources of experimental uncertainty included engine speed, torque, fuel-flow measurement, air-flow measurement, exhaust temperature, cylinder pressure, crank angle, hydrogen flow rate, exhaust-gas analysis, and smoke measurement.

For independent measurement variables, the combined uncertainty was estimated using the root-sum-square method:

$$U_R = \sqrt{U_1^2 + U_2^2 + \dots + U_n^2} \quad (6)$$

where U_R represents the combined uncertainty and $U_1, U_2, \dots, U_n$ represent the individual measurement uncertainties.

The uncertainties associated with the principal measurement parameters are presented in Table 5.

Table 5: Uncertainty of Measured Parameters

Measured parameter	Uncertainty
Engine speed	±1 rpm
Torque	±0.5%
Fuel-flow measurement	±1.0%
Air-flow measurement	±1.5%
Hydrogen-flow measurement	±1.0%
Exhaust temperature	±2 °C
Cylinder pressure	±0.5% of full scale
NO _x analyzer	±2%
CO analyzer	±2%
HC analyzer	±1.5%
Smoke meter	±1% FS
Crank-angle encoder	±0.2° CA

All measuring instruments were calibrated before the experiments. Each experimental condition was repeated three times, and the average value was used for analysis. The resulting uncertainty was considered acceptable for comparative engine testing.

2.11 Experimental Test Matrix

The experimental test matrix was designed to evaluate the effect of injector nozzle-hole configuration under diesel, Jatropha biodiesel, and Jatropha biodiesel-hydrogen dual-fuel conditions. The principal experimental parameters are summarized in Table 6.

Table 6: Experimental Test Matrix

Test condition	Liquid fuel	H ₂	Injector	Speed	Load
D100	Diesel	-	N3, N4, N5	3000 rpm	20%, 40%, 60%, 80%, 100% of rated load
JB15	JB15	-	N3, N4, N5	3000 rpm	20%, 40%, 60%, 80%, 100% of rated load
JB15-H ₂ -N3	JB15	8 L/min	N3	3000 rpm	20%, 40%, 60%, 80%, 100% of rated load
JB15-H ₂ -N4	JB15	8 L/min	N4	3000 rpm	20%, 40%, 60%, 80%, 100% of rated load
JB15-H ₂ -N5	JB15	8 L/min	N5	3000 rpm	20%, 40%, 60%, 80%, 100% of rated load

The experimental matrix was structured such that D100 and JB15 represented the reference operating conditions, whereas the principal investigation focused on JB15-H₂ dual-fuel operation. Under the primary dual-fuel condition, the injector configuration was systematically varied among N3, N4, and N5 while engine speed, load, injector opening pressure, injection timing, nominal spray angle, liquid-fuel composition, and hydrogen flow rate were maintained constant. The total geometric orifice area was approximately matched among the three configurations, with a maximum difference of approximately 1.3%, while the individual orifice diameter was adjusted according to the number of nozzle holes. Thus, the primary experimental variable was the injector nozzle-hole configuration, defined by the number of holes and corresponding orifice diameter at approximately matched total geometric orifice area. This experimental design enabled a controlled comparison of the influence of injector configuration on engine performance, combustion characteristics, and exhaust emissions under *Jatropha* biodiesel-hydrogen dual-fuel operation.

3. RESULTS AND DISCUSSION

The experimental investigation was carried out to assess the effects of injector nozzle-hole configuration and hydrogen enrichment on the performance, combustion, and emission characteristics of the direct-injection compression-ignition engine. Three injector configurations, namely N3, N4, and N5, representing three-, four-, and five-hole nozzles, respectively, were evaluated. The corresponding orifice diameters were adjusted to maintain approximately the same total geometric orifice area, thereby limiting the influence of differences in overall geometric flow area. The engine was operated at 3000 rpm over a load range of 20-100% of the rated brake power. Diesel (D100) and JB15 were considered reference operating conditions, while JB15 with 8 L/min hydrogen (JB15+8H₂) constituted the principal dual-fuel condition.

The measured parameters included brake thermal efficiency (BTE), brake-specific fuel consumption (BSFC), exhaust gas temperature (EGT), carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x), smoke opacity, cylinder pressure, and heat-release rate (HRR). The results are discussed with particular emphasis on the interaction between fuel properties, hydrogen enrichment, and injector nozzle-hole configuration.

3.1 Brake Thermal Efficiency

The variation of brake thermal efficiency (BTE) with engine load for the investigated fuel and nozzle configurations is presented in Figure 3. BTE increased progressively with increasing engine load for all test conditions. At low load, a relatively large fraction of the fuel energy is consumed in overcoming frictional, pumping, and other fixed losses. As the load increases, the useful brake power increases more rapidly than these approximately load-independent losses, resulting in an improvement in brake thermal efficiency.

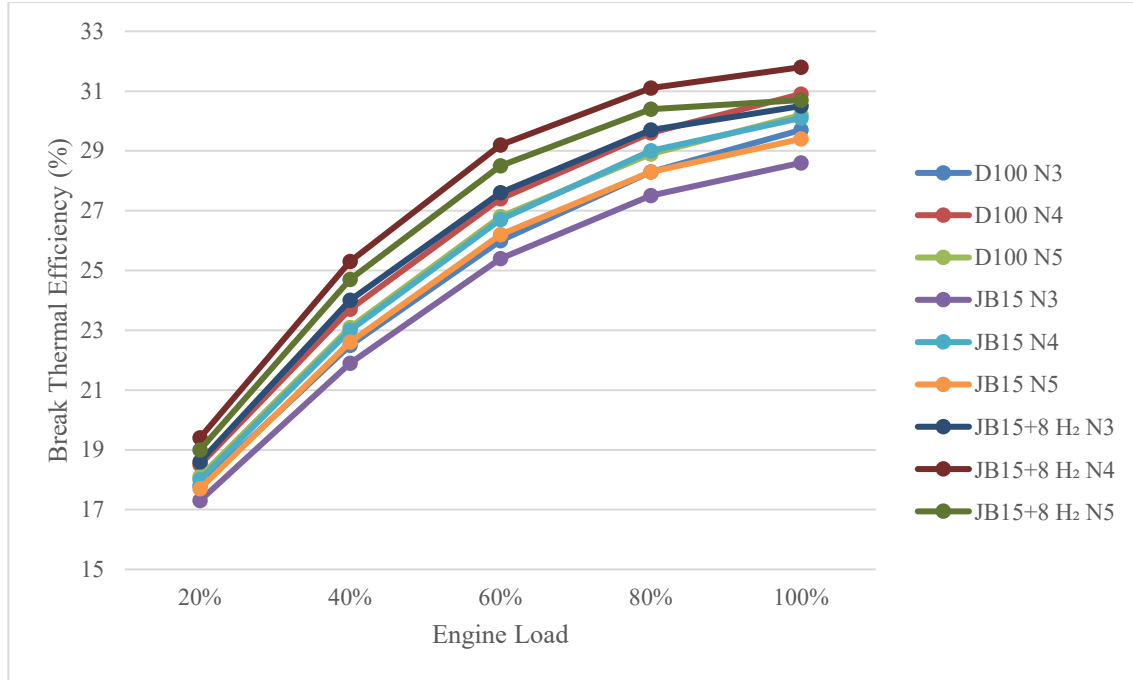


Figure 3: Variation of brake thermal efficiency with engine load for different fuel and nozzle configurations

For a given fuel condition, the nozzle-hole configuration produced measurable differences in BTE. The N4 configuration generally exhibited the highest BTE, indicating a favourable balance between spray penetration, atomization, fuel distribution, and utilization of the air available within the combustion chamber. The N3 nozzle produces fewer and relatively larger spray plumes, which may result in more concentrated fuel distribution. In contrast, the N5 configuration divides the injected fuel among a greater number of smaller plumes, potentially reducing individual spray momentum and penetration. The N4 configuration therefore provides an intermediate spray structure that promotes favourable interaction between the injected liquid fuel and the surrounding charge.

Hydrogen enrichment further improved BTE for all three nozzle configurations. At full load, the BTE values for JB15 with N3, N4, and N5 were 28.6%, 30.1%, and 29.4%, respectively. With the addition of 8 L/min hydrogen, the corresponding BTE values increased to 30.5%, 31.8%, and 30.7%, respectively. The maximum BTE of 31.8% was obtained with the JB15+8H₂-N4 configuration.

The improvement in BTE with hydrogen can be attributed to its high diffusivity, rapid flame propagation, and wide flammability range. Hydrogen introduced through the intake system forms a relatively homogeneous premixed charge, while the directly injected JB15 acts as the ignition source. Once ignition occurs, the premixed hydrogen-air mixture can burn rapidly, improving the utilization of the chemical energy supplied to the engine. The higher BTE obtained with N4 suggests that its spray characteristics provide favourable conditions for initiating and sustaining hydrogen-assisted combustion.

Overall, the results indicate that both hydrogen enrichment and nozzle-hole configuration influence the conversion of fuel energy into useful brake work. Among the tested configurations, N4 provided the most favourable BTE under JB15-hydrogen dual-fuel operation.

3.2 Brake-Specific Fuel Consumption

The variation of brake-specific fuel consumption (BSFC) with engine load is presented in Figure 4. BSFC decreased substantially with increasing load for all fuel and nozzle configurations. At low load, the fuel required to overcome frictional and pumping losses constitutes a relatively high proportion of the total fuel consumption. With increasing load, the useful brake power increases and the relative contribution of these losses decreases, resulting in a reduction in BSFC.

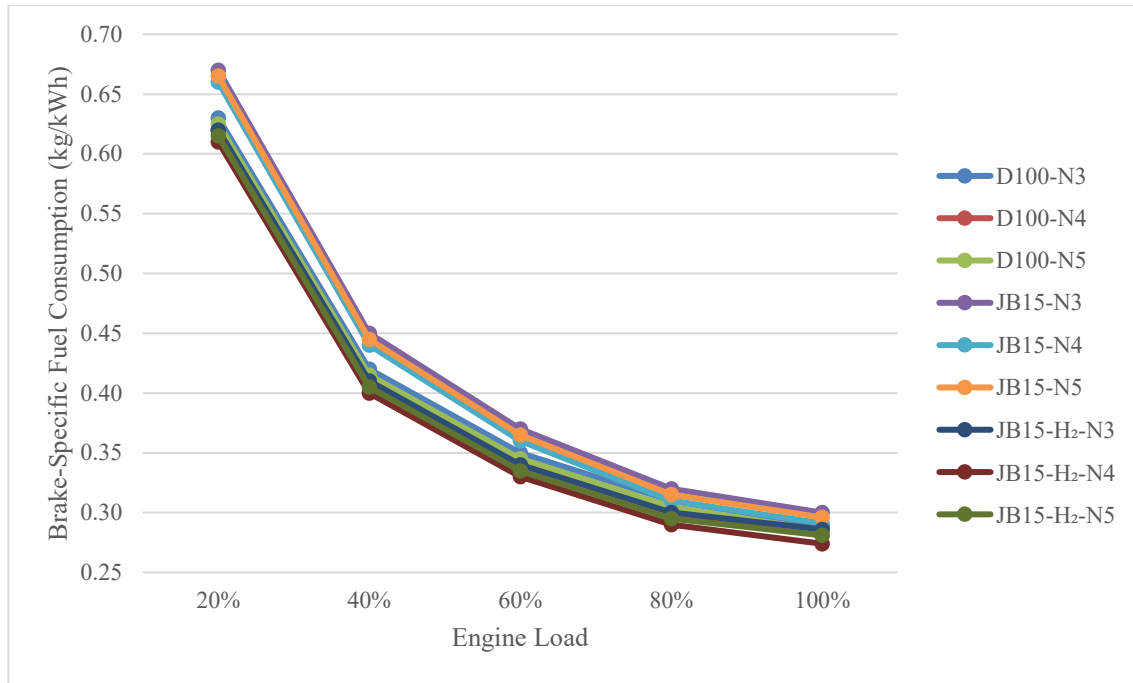


Figure 4: Variation of brake-specific fuel consumption with engine load for different fuel and nozzle configurations

The JB15 configurations generally exhibited higher BSFC than the corresponding D100 configurations. This behaviour is primarily associated with the lower heating value of JB15 compared with diesel. Consequently, a greater mass of JB15 is required to supply an equivalent amount of chemical energy to the engine. The slightly higher viscosity and density of the biodiesel blend may also influence spray atomization and fuel-air mixing.

Hydrogen addition reduced the required liquid-fuel consumption for a given brake power because part of the total chemical energy supplied to the engine was provided by hydrogen. The reduction in BSFC was particularly evident at higher loads, where the contribution of hydrogen-assisted combustion became more significant.

At full load, the BSFC values for D100-N3, D100-N4, and D100-N5 were 0.290, 0.281, and 0.285 kg/kWh, respectively. The corresponding values for JB15-N3, JB15-N4, and JB15-N5 were 0.300, 0.291, and 0.296 kg/kWh. With hydrogen enrichment, the BSFC decreased to 0.286, 0.274, and 0.281 kg/kWh for N3, N4, and N5, respectively.

The minimum BSFC of 0.274 kg/kWh was obtained for JB15+8H₂-N4. Compared with JB15-N4, this represents a reduction of approximately 5.8%. The reduction is consistent with the increase in BTE observed for the same operating condition. The N4 configuration again provided the lowest BSFC among the three nozzle configurations, indicating that its spray distribution and subsequent combustion characteristics resulted in more effective utilization of the supplied fuel energy.

Thus, the combined effect of biodiesel properties, hydrogen supplementation, and injector configuration determines the fuel conversion characteristics. The N4 nozzle provided the most favourable BSFC under hydrogen-enriched JB15 operation.

3.3 Exhaust Gas Temperature

The variation of exhaust gas temperature (EGT) with engine load is shown in Figure 5. EGT increased progressively with increasing engine load for all fuel and nozzle configurations. The increase is associated primarily with the higher quantity of fuel supplied at increased loads and the corresponding increase in the total heat released during combustion.

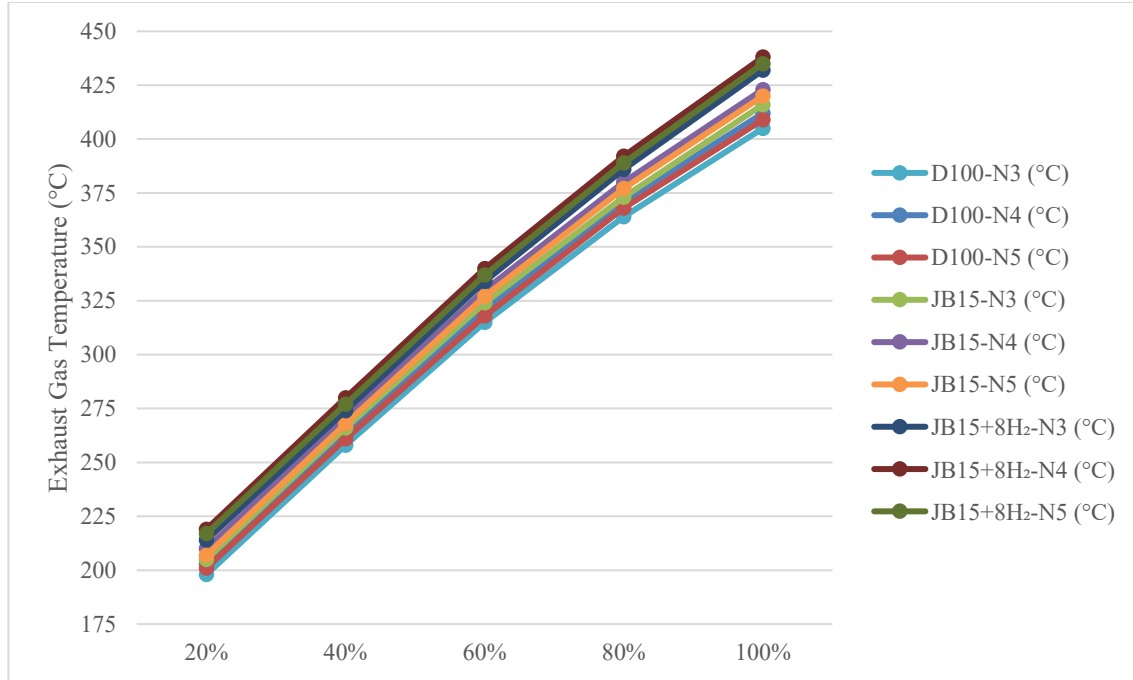


Figure 5: Variation of exhaust gas temperature with engine load

At full load, the EGT values for D100-N3, D100-N4, and D100-N5 were 405, 412, and 409 °C, respectively. The corresponding values for JB15 were 416, 423, and 420 °C. With hydrogen enrichment, the EGT increased further to 432, 438, and 435 °C for N3, N4, and N5, respectively.

For all three fuel conditions, N4 produced the highest EGT, followed by N5 and N3. The higher EGT observed with N4 is consistent with its higher BTE, peak cylinder pressure, and heat-release rate. Improved spray distribution can promote more effective fuel-air mixing and consequently enhance the intensity of combustion. The resulting higher combustion temperature can increase the amount of sensible heat remaining in the exhaust gases.

The increase in EGT with JB15 compared with D100 can be influenced by the different physical and chemical properties of the biodiesel blend, including its oxygen content, viscosity, volatility, and heating value. Hydrogen addition produced the highest EGT among the investigated fuel conditions. Its rapid combustion and high flame propagation rate can increase the rate and intensity of heat release, leading to higher combustion temperatures and consequently higher exhaust temperatures.

At full load, the observed EGT trend was therefore $D100 < JB15 < JB15+8H_2$, while the nozzle-dependent trend was generally $N3 < N5 < N4$. The results demonstrate that the nozzle configuration influences the thermal characteristics of combustion, with N4 providing the most intense combustion among the tested configurations.

3.4 Carbon Monoxide Emissions

The variation of carbon monoxide (CO) emissions with engine load is presented in Figure 6. CO is primarily formed under conditions of incomplete oxidation of carbon-containing fuel species. At low load, relatively low combustion temperatures and less favourable oxidation conditions can result in higher CO formation. As the load increases, the higher in-cylinder temperature and increased combustion intensity generally promote more complete oxidation of intermediate carbon species, resulting in lower CO emissions.

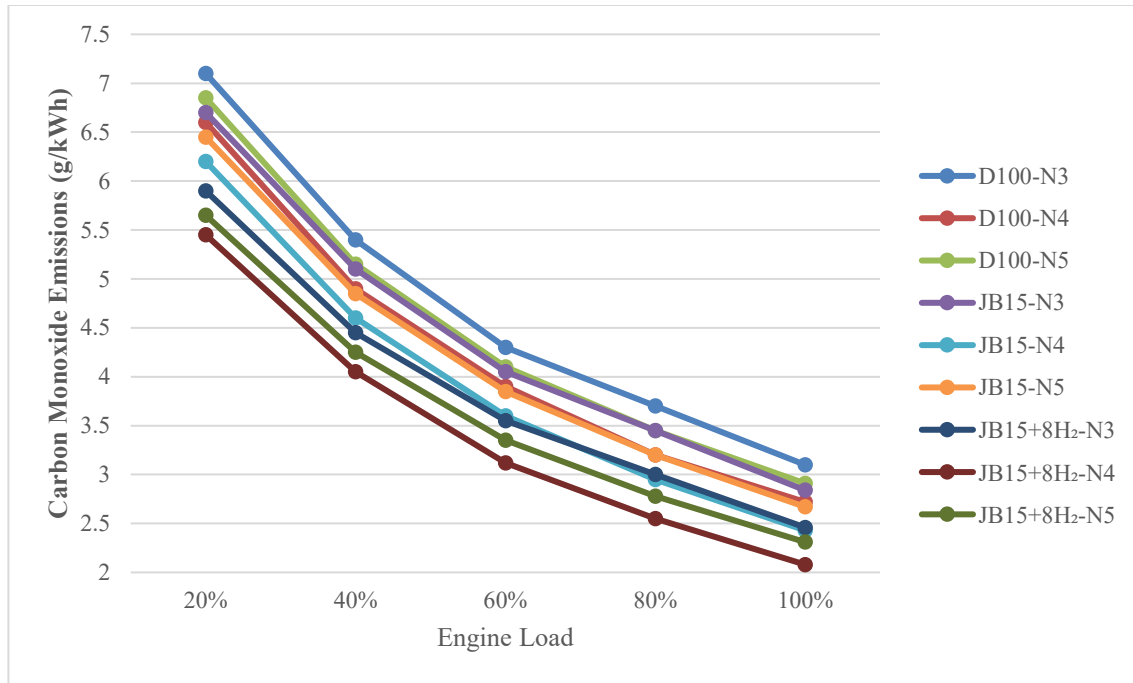


Figure 6: Variation of carbon monoxide emissions with engine load

The injector configuration also influenced CO formation. The N4 configuration generally produced lower CO emissions than N3 and N5. This behaviour can be associated with its favourable spray penetration and spatial distribution, which promote more effective interaction between the injected liquid fuel and the available air. The N3 configuration produces fewer, relatively larger spray plumes, whereas the N5 configuration produces a greater number of smaller plumes with lower individual momentum. The N4 configuration therefore provides an intermediate spray structure that appears favourable for oxidation.

Hydrogen enrichment further reduced CO emissions for all nozzle configurations. Hydrogen contains no carbon and therefore does not directly contribute to carbon monoxide formation. In addition, its high diffusivity and rapid flame propagation can enhance the oxidation of carbon-containing species originating from the JB15 spray.

At full load, the CO emission for JB15-N4 decreased from 2.43 g/kWh to 2.08 g/kWh with the addition of 8 L/min hydrogen, corresponding to a reduction of approximately 14.4%. The reduction demonstrates the positive influence of hydrogen-assisted combustion on carbon oxidation.

The N5 configuration also exhibited lower CO emissions than N3, although its performance remained slightly inferior to N4. Overall, the results indicate that improved spray distribution in combination with hydrogen enrichment can promote more complete combustion and consequently reduce CO emissions.

3.5 Unburned Hydrocarbon Emissions

The variation of unburned hydrocarbon (HC) emissions with engine load is presented in Figure 7. HC emissions generally decreased as the engine load increased. At low load, lower combustion temperatures, weaker oxidation conditions, and incomplete utilization of the injected fuel can increase the quantity of unburned and partially oxidized hydrocarbons in the exhaust. With increasing load, the higher combustion temperature and increased oxidation intensity generally reduce HC emissions.

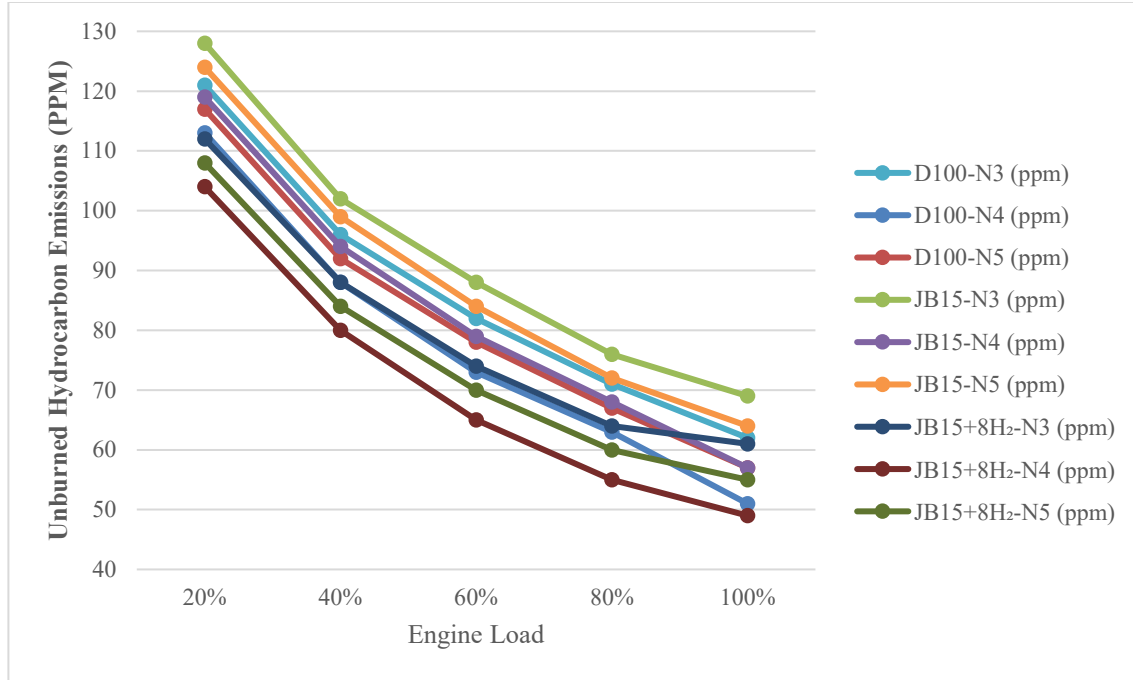


Figure 7: Variation of unburned hydrocarbon emissions with engine load

The nozzle-hole configuration produced a clear influence on HC emissions. For D100, the HC emissions decreased from 121, 113, and 117 ppm at 20% load to 62, 51, and 57 ppm at full load for N3, N4, and N5, respectively. At full load, the corresponding values for JB15 were 69, 57, and 64 ppm. These results show that N4 consistently produced the lowest HC emissions among the investigated configurations.

The lower HC emissions with N4 can be attributed to improved spray distribution and more favourable fuel-air interaction. The N3 nozzle produces relatively concentrated fuel jets, which may result in localized fuel-rich regions. Although the N5 nozzle provides greater spatial distribution of the fuel, excessive division of the injected quantity among several spray plumes can reduce individual spray momentum and modify penetration characteristics. N4 provides an intermediate configuration that appears to offer a favourable balance between penetration and dispersion.

Hydrogen enrichment further reduced HC emissions. At full load, HC emissions decreased to 61, 49, and 55 ppm for N3, N4, and N5, respectively, under JB15+8H₂ operation. For N4, the reduction from 57 ppm for JB15 to 49 ppm with hydrogen corresponds to approximately 14.0%.

The reduction in HC with hydrogen can be attributed to its high diffusivity, rapid flame propagation, and enhanced oxidation of fuel-rich regions. The overall nozzle-dependent trend was N4 < N5 < N3, demonstrating that the N4 configuration provided the most favourable HC emission characteristics under the investigated conditions.

3.6 Nitrogen Oxide Emissions

The variation of nitrogen oxide (NO_x) emissions with engine load is shown in Figure 8. In contrast to CO and HC, NO_x emissions increased significantly with increasing engine load. NO_x formation in CI engines is strongly influenced by combustion temperature, oxygen availability, combustion duration, and the residence time of the combustion products at elevated temperatures.

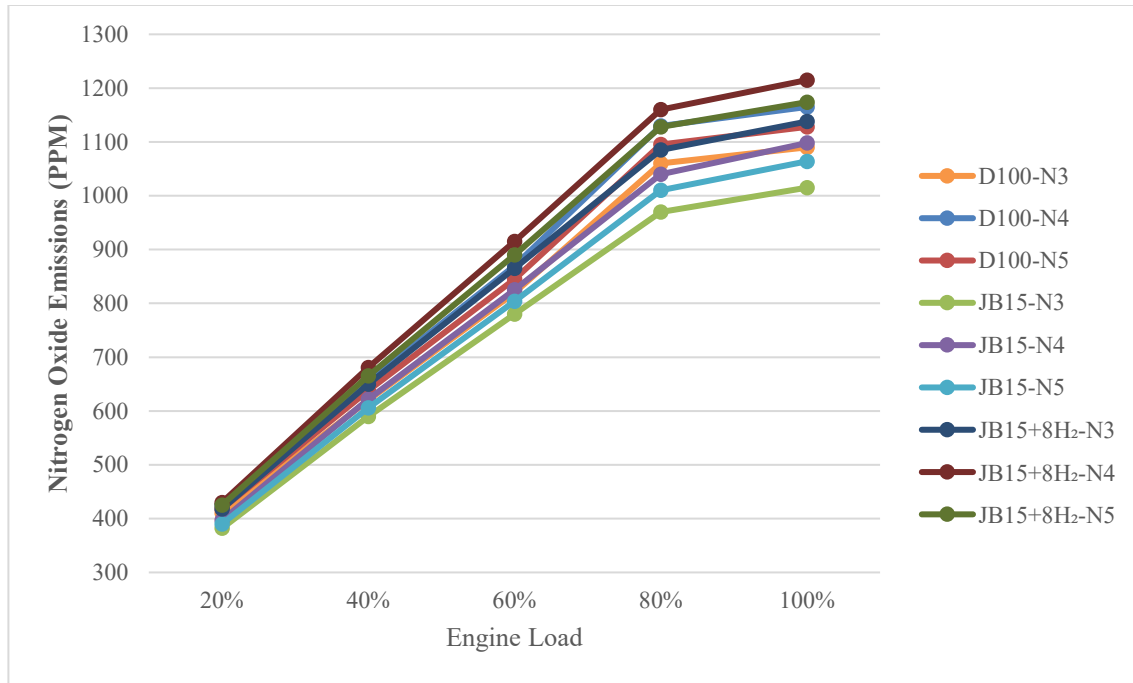


Figure 8: Variation of nitrogen oxide emissions with engine load

The nozzle-hole configuration produced a measurable influence on NO_x formation. Among the investigated configurations, N4 generally produced the highest NO_x emissions, whereas N3 and N5 exhibited comparatively lower values. At full load, the NO_x emissions for D100-N4, JB15-N4, and JB15+8H₂-N4 were approximately 1165, 1098, and 1215 ppm, respectively.

The higher NO_x emission associated with N4 can be related to its favourable spray distribution and improved fuel-air mixing. More effective mixing can increase the proportion of fuel participating in relatively rapid combustion and can produce higher local combustion temperatures. Consequently, although N4 improves BTE and reduces CO, HC, and smoke, the associated increase in combustion intensity can promote thermal NO_x formation.

The JB15 condition generally produced lower NO_x emissions than D100, particularly at medium and high loads. This behaviour can be influenced by the lower heating value and different combustion characteristics of the biodiesel blend. The oxygen content of biodiesel may promote oxidation, while its physical properties and ignition characteristics can modify combustion phasing and temperature development.

Hydrogen enrichment increased NO_x emissions relative to JB15 operation. Hydrogen has a high flame propagation rate and can enhance the rate of premixed combustion after ignition of the liquid pilot fuel. The resulting increase in combustion intensity and local temperature can favour thermal NO_x formation.

At full load, the NO_x emission for N4 increased from approximately 1098 ppm for JB15 to 1215 ppm with 8 L/min hydrogen, representing an increase of approximately 10.7%. Therefore, the improvement in thermal efficiency and reduction in carbon-containing emissions obtained with hydrogen are accompanied by an increase in NO_x.

These results demonstrate an important performance-emission trade-off. The N4 configuration provides favourable combustion and performance characteristics but also produces relatively higher NO_x because of its more effective combustion process. Further optimization of hydrogen flow rate, injection timing, or other combustion-control parameters may therefore be required to reduce NO_x while retaining the performance advantages of hydrogen enrichment.

3.7 Smoke Opacity

The variation of smoke opacity with engine load for the investigated fuel and nozzle configurations is presented in Figure 9. Smoke opacity generally increased with increasing engine load up to approximately 80% load, followed by a slight reduction at full load. The increase in smoke at higher loads is associated with the increased quantity of fuel supplied, locally fuel-rich regions, diffusion-controlled combustion, and enhanced soot

formation. The slight reduction at full load may result from changes in combustion temperature, spray-air interaction, and enhanced oxidation of the accumulated soot.

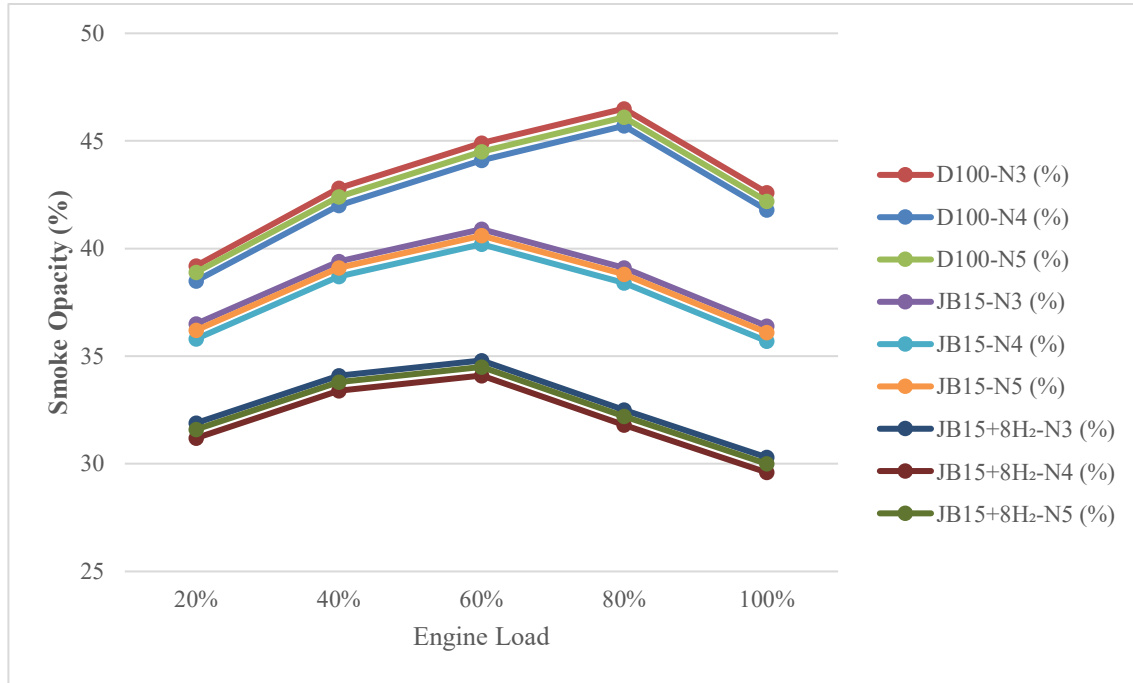


Figure 9: Variation of smoke opacity with engine load for different fuel and nozzle configurations

The fuel type had a pronounced effect on smoke formation. D100 generally produced higher smoke opacity than JB15, whereas the addition of hydrogen resulted in a further reduction. The lower smoke formation associated with JB15 can be attributed partly to the chemically bound oxygen in biodiesel, which facilitates oxidation of soot precursors. Hydrogen provides an additional benefit because it contains no carbon and therefore does not directly participate in soot formation. Its high diffusivity also promotes more homogeneous mixing with the intake air.

The injector configuration produced a comparatively smaller but consistent influence on smoke opacity. N4 generally exhibited the lowest smoke opacity, followed by N5 and N3. At full load, the smoke opacity for JB15 decreased from 36.4% for N3 to 35.7% for N4. Under hydrogen-enriched operation, the corresponding values for N3, N4, and N5 were 30.3%, 29.6%, and 30.0%, respectively.

For N4, hydrogen addition reduced smoke opacity from 35.7% for JB15 to 29.6% for JB15+8H₂, corresponding to a reduction of approximately 17.1%. This substantial reduction demonstrates the combined effect of the carbon-free hydrogen fuel and improved oxidation conditions.

The overall fuel-dependent trend was D100 > JB15 > JB15+8H₂, while the nozzle-dependent trend under the investigated conditions was generally N3 > N5 > N4. The results indicate that hydrogen enrichment is particularly effective in suppressing smoke, while the N4 nozzle provides the most favourable spray configuration for minimizing soot formation.

3.8 Cylinder Pressure

The full-load cylinder-pressure characteristics for the investigated configurations are presented in Figure 10. Cylinder pressure provides an important indication of combustion intensity and combustion phasing. The pressure development inside the cylinder is governed by the rate of fuel-air mixing, ignition, premixed combustion, diffusion-controlled combustion, and the timing of heat release.

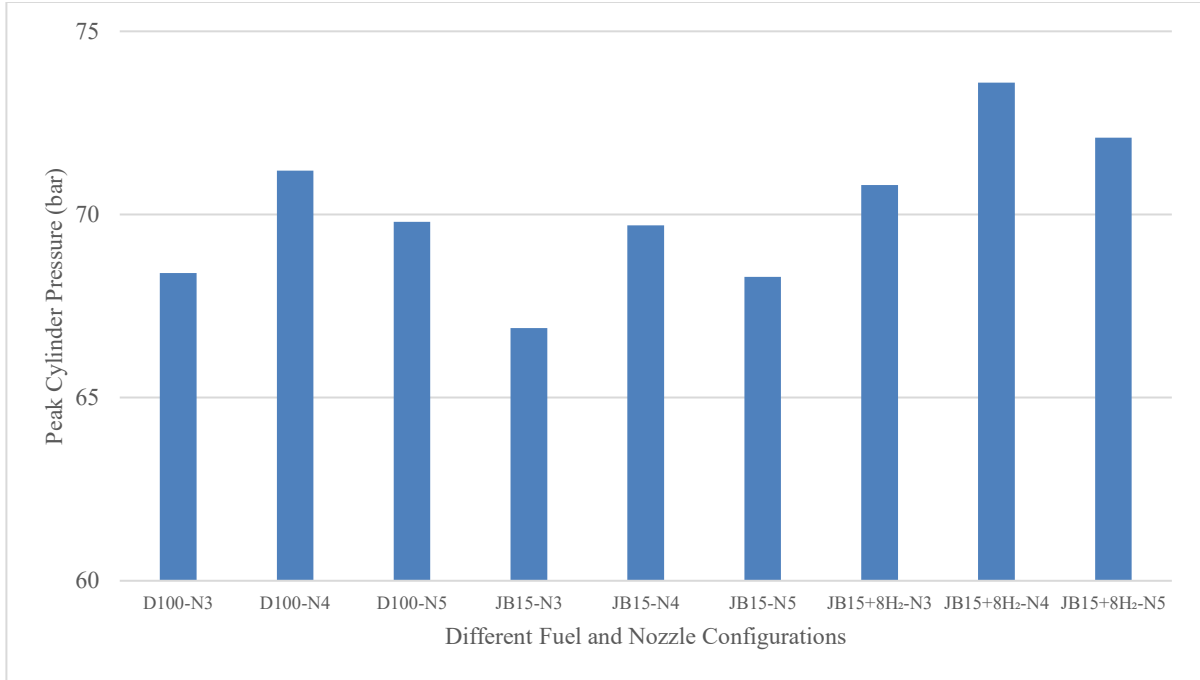


Figure 10: Full-load peak cylinder pressure for different fuel and nozzle configurations

Hydrogen enrichment increased the peak cylinder pressure for all three nozzle configurations. At full load, the maximum peak cylinder pressure of 73.6 bar was obtained for JB15+8H₂-N4, compared with 69.7 bar for JB15-N4. The addition of hydrogen therefore increased the peak cylinder pressure for N4 by approximately 5.6%.

The increase in peak pressure can be attributed to the rapid combustion of the premixed hydrogen-air charge following ignition by the directly injected JB15. Hydrogen possesses a high flame propagation rate and can substantially increase the rate of heat release during the early stages of combustion. Consequently, a greater fraction of the released chemical energy is converted into cylinder pressure.

The N4 configuration produced the highest peak cylinder pressure among the investigated nozzle configurations. This behaviour is consistent with the higher BTE, EGT, and HRR observed for N4. The favourable balance between spray penetration and dispersion appears to promote more effective utilization of the available air and consequently more intense combustion.

The hydrogen-assisted increase in peak cylinder pressure indicates an increase in combustion intensity. While this can contribute positively to thermal efficiency, excessive pressure rise can potentially increase mechanical and thermal loading on engine components. Therefore, the observed pressure characteristics should be considered together with NO_x emissions and heat-release behaviour when evaluating the suitability of the injector configurations.

3.9 Heat-Release Rate

The peak heat-release rate (HRR) at full load is compared in Figure 11. HRR provides an indication of the rate and distribution of chemical energy release during the combustion process and is therefore useful for evaluating the effects of injector configuration and hydrogen enrichment on combustion development.

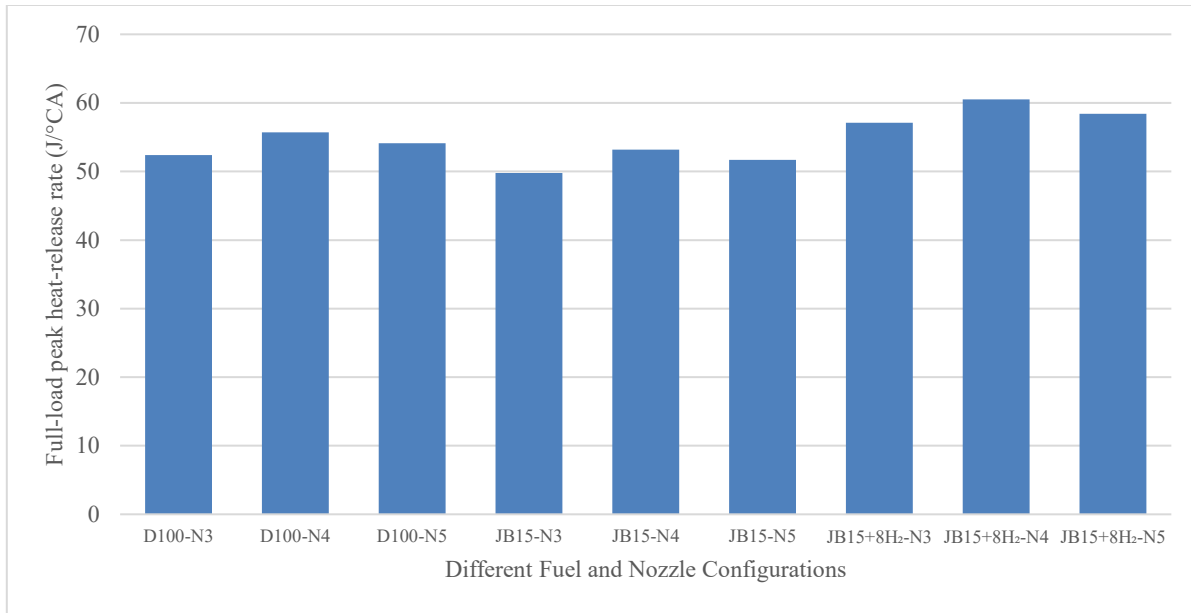


Figure 11: Full-load peak heat-release rate for different fuel and nozzle configurations

Hydrogen addition significantly increased the peak HRR for all three nozzle configurations. For the N4 nozzle, the peak HRR increased from 53.2 J/°CA for JB15-N4 to 60.5 J/°CA for JB15+8H₂-N4, corresponding to an increase of approximately 13.7%.

The increase in HRR can be attributed to the rapid burning characteristics of hydrogen and its high diffusivity within the intake charge. The premixed hydrogen-air mixture is capable of undergoing rapid combustion after ignition of the directly injected JB15. This increases the rate of energy release during the premixed combustion phase.

Among the investigated configurations, N4 produced the highest peak HRR. For N4, the location of the peak HRR shifted from approximately 3.9°CA ATDC for JB15-N4 to 2.9°CA ATDC for JB15+8H₂-N4. The earlier occurrence and higher magnitude of the HRR peak indicate faster combustion and a greater contribution from premixed combustion following hydrogen enrichment.

The higher HRR obtained with N4 is consistent with its higher peak cylinder pressure and BTE. However, the increased rate of heat release also contributes to higher combustion temperatures, which helps explain the corresponding increase in NO_x emissions. Thus, the HRR results reinforce the observed trade-off between improved combustion efficiency and increased NO_x formation.

3.10 Comparative Assessment of Nozzle-Hole Configuration

A comparative assessment of the three nozzle-hole configurations under JB15+8H₂ operation is presented in Figure 12. The results demonstrate that injector nozzle-hole configuration has a significant influence on the overall performance and emission characteristics of the dual-fuel engine.

Among the investigated configurations, N4 provided the most favourable overall performance. It achieved the highest BTE of 31.8% and the lowest BSFC of 0.274 kg/kWh at full load. The N4 configuration also produced the lowest CO, HC, and smoke emissions among the tested nozzle configurations. These results indicate that N4 provides a favourable balance between fuel penetration, atomization, spray dispersion, and utilization of the combustion-chamber air.

The N3 configuration produces fewer but relatively larger spray plumes. Such a configuration can maintain higher individual spray momentum but may produce less uniform spatial fuel distribution. Conversely, N5 generates a greater number of smaller spray plumes. Although this can improve spatial coverage, the reduction in individual plume momentum may alter penetration and local fuel-air mixing. Under the present experimental conditions, N4 provides an intermediate arrangement that appears to achieve the most favourable balance.

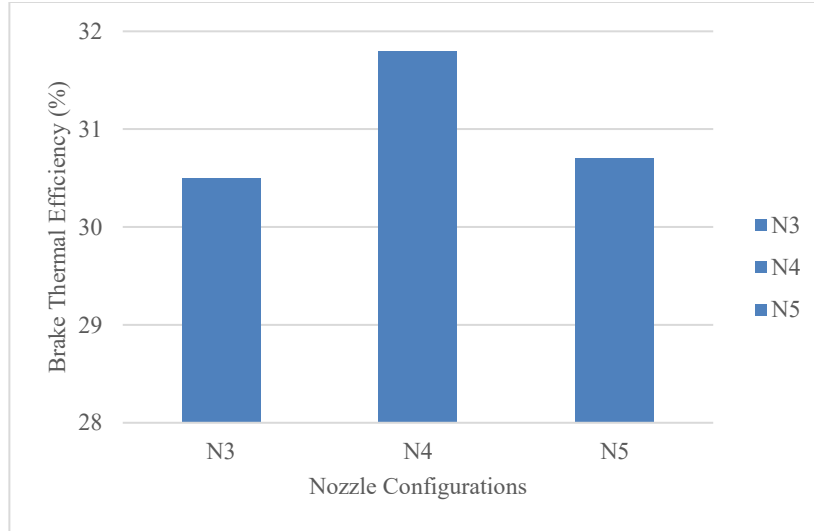


Figure 12: Comparison of brake thermal efficiency for N3, N4 and N5 nozzles under JB15+8H₂ operation

The improved performance of N4 was accompanied by an increase in NO_x. The N4 configuration produced the highest NO_x emission under JB15+8H₂ operation, approximately 1215 ppm at full load. It also produced the highest peak cylinder pressure and peak HRR. These observations indicate that the superior combustion quality associated with N4 is accompanied by greater combustion intensity and higher-temperature regions that favour NO_x formation.

Therefore, the nozzle configuration that provides the highest thermal efficiency does not necessarily provide the lowest emissions for all pollutants. In particular, the reduction in CO, HC, and smoke obtained with N4 occurs simultaneously with an increase in NO_x. Injector optimization for hydrogen-assisted biodiesel operation should consequently be based on a combined assessment of performance, combustion, and emissions rather than on a single parameter.

Overall, N4 provided the most favourable compromise among the investigated configurations under the specified JB15+8H₂ operating conditions.

3.11 Effect of Hydrogen Addition

The effect of hydrogen enrichment was isolated by comparing JB15-N4 and JB15+8H₂-N4 at full load, as presented in Figure 13 and summarized in Table 7. The N4 configuration was selected for this comparison because it provided the most favourable overall performance among the three injector configurations.

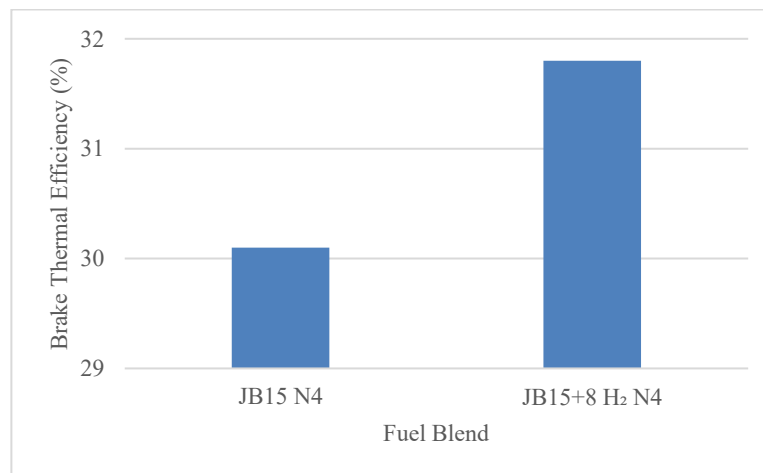


Figure 13. Effect of hydrogen addition on brake thermal efficiency with the N4 nozzle

Hydrogen addition increased BTE from 30.1% to 31.8%, corresponding to an improvement of approximately 5.6%. Simultaneously, BSFC decreased from 0.291 to 0.274 kg/kWh, representing a reduction of approximately 5.8%. These changes indicate improved conversion of the supplied chemical energy into useful brake power.

The addition of hydrogen also produced substantial reductions in carbon-containing emissions. CO decreased from 2.43 to 2.08 g/kWh, corresponding to a reduction of approximately 14.4%. HC decreased from 57 to 49 ppm, representing a reduction of approximately 14.0%. Smoke opacity decreased from 35.7% to 29.6%, corresponding to a reduction of approximately 17.1%.

The reductions in CO, HC, and smoke can be attributed to the carbon-free nature of hydrogen, its high diffusivity, rapid flame propagation, and enhancement of the oxidation of fuel-rich regions. Hydrogen supplementation therefore improves the combustion quality of JB15 operation and substantially reduces carbonaceous emissions.

However, hydrogen addition increased NO_x from 1098 to 1215 ppm, corresponding to an increase of approximately 10.7%. The increase is associated with the faster combustion and higher local combustion temperatures promoted by hydrogen. The increase in peak cylinder pressure from 69.7 to 73.6 bar and the increase in peak HRR from 53.2 to 60.5 J/°CA further confirm the intensification of the combustion process. The respective increases were approximately 5.6% and 13.7%.

Table 7. Effect of hydrogen addition with the N4 nozzle at full load

Parameter	JB15-N4	JB15+8H ₂ -N4	Change
BTE (%)	30.1	31.8	+5.6%
BSFC (kg/kWh)	0.291	0.274	-5.8%
CO (g/kWh)	2.43	2.08	-14.4%
HC (ppm)	57	49	-14.0%
NO _x (ppm)	1098	1215	+10.7%
Smoke (%)	35.7	29.6	-17.1%
Peak pressure (bar)	69.7	73.6	+5.6%
Peak HRR (J/°CA)	53.2	60.5	+13.7%

The hydrogen-addition results demonstrate a clear improvement in combustion efficiency and carbon-related emissions, accompanied by a NO_x penalty. The simultaneous increase in BTE and reduction in BSFC indicate that hydrogen can effectively supplement JB15 as a gaseous secondary fuel. The reductions in CO, HC, and smoke further demonstrate the potential of hydrogen to improve oxidation and reduce carbonaceous emissions.

Among the investigated configurations, JB15+8H₂-N4 provided the most favourable overall combination of performance and emission characteristics. It achieved the highest BTE, lowest BSFC, and lowest CO, HC, and smoke emissions, although it also produced the highest NO_x emission and the highest peak combustion parameters. These results indicate that the N4 configuration provides an effective injector arrangement for hydrogen-assisted JB15 operation under the present experimental conditions.

The observed NO_x increase indicates that further optimization is required if simultaneous reductions in NO_x and carbonaceous emissions are desired. Parameters such as hydrogen flow rate, injection timing, injection pressure, and combustion phasing could be optimized in future investigations to reduce the temperature-driven NO_x penalty while retaining the performance and emission benefits obtained from hydrogen enrichment.

4. CONCLUSION:

An experimental investigation was conducted on a direct-injection diesel engine to evaluate the effect of injector nozzle-hole configuration on the performance, combustion, and emission characteristics of diesel, JB15, and JB15-hydrogen dual-fuel operation. Three injector configurations, namely 3-hole (N3), 4-hole (N4), and 5-hole (N5) nozzles, were investigated at 3000 rpm and different engine loads. The individual orifice diameter was varied to maintain approximately matched total geometric orifice area, with a maximum difference of about 1.3%. The primary objective was to identify the nozzle configuration providing the most favourable combination of engine performance, combustion behaviour, and emissions under JB15-hydrogen dual-fuel operation.

Among the tested configurations, the 4-hole N4 nozzle provided the overall best performance. At full load, the N4 nozzle achieved a brake thermal efficiency of 31.8% and a minimum brake-specific fuel consumption of 0.274 kg/kWh with JB15+8 L/min hydrogen. Compared with JB15-N4, hydrogen addition improved BTE by

approximately 5.6% and reduced BSFC by 5.8%. The improved performance can be attributed to the favourable spray penetration and fuel distribution of the N4 configuration combined with the high diffusivity and rapid flame propagation characteristics of hydrogen.

Hydrogen addition also improved the carbon-related emission characteristics. For the N4 nozzle at full load, CO, HC, and smoke opacity decreased by approximately 14.4%, 14.0%, and 17.1%, respectively, compared with JB15 operation. These reductions were associated with improved fuel-air mixing, enhanced oxidation, and the carbon-free nature of hydrogen. The N4 configuration generally produced lower CO, HC, and smoke emissions than the N3 and N5 configurations, indicating a favourable balance between spray dispersion and combustion-zone air utilization.

In contrast, hydrogen enrichment increased NO_x emissions. For the N4 nozzle, NO_x increased from approximately 1098 ppm for JB15 to 1215 ppm with 8 L/min hydrogen, representing an increase of about 10.7%. The increase in NO_x was accompanied by increases in peak cylinder pressure and heat-release rate. Peak pressure increased from 69.7 to 73.6 bar, while peak HRR increased from 53.2 to 60.5 J/°CA, corresponding to increases of approximately 5.6% and 13.7%, respectively. These results indicate that hydrogen promotes faster and more intense combustion, which improves efficiency but can increase combustion temperature and consequently thermal NO_x formation.

Overall, the JB15+8H₂-N4 configuration provided the most favourable compromise between engine performance, combustion characteristics, and carbon-related emissions under the investigated conditions. The results demonstrate that injector nozzle-hole configuration plays an important role in controlling spray distribution, fuel-air mixing, combustion intensity, and emission formation in hydrogen-assisted biodiesel operation. However, the associated increase in NO_x highlights the need for further optimization of hydrogen flow rate, injection timing, injection pressure, and combustion phasing.

Future investigations should therefore focus on optimizing injector and hydrogen-supply parameters simultaneously and on developing suitable NO_x-control strategies to retain the efficiency and emission benefits of hydrogen-assisted *Jatropha* biodiesel combustion while minimizing the NO_x penalty.

CRedit authorship contribution statement

Swapnil Thikane: Conceptualization, Methodology, Investigation, Formal analysis, Data curation, Writing - original draft, Writing - review & editing, Visualization, and Project administration.

Declaration of competing interests:

The author declares that there are no known competing financial interests or personal relationships that could have appeared to influence the work reported in this article.

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Declaration of generative AI use:

During the preparation of this work, the author used ChatGPT (OpenAI) for language refinement and editing. The author reviewed the output and takes full responsibility for the final content.

Data Availability Statement

The author confirms that all data supporting the findings of this study are included within this article. No additional datasets were generated or analyzed during the current study.

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