



Experimental and RSM-based Analysis of Hydrogen-enriched Microalgae Biodiesel Combustion in a CI Engine

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ABSTRACT

The study examined the efficiency and exhaust features of a compression ignition (CI) engine powered by a blends of biodiesel and diesel enriched with H₂ at distinct engine speeds, blending rates and hydrogen flow rates. For the experimental tests, a mono-cylinder water-cooled Kirloskar CI engine with a nominal power of 8.5 kW was used to measure brake power, brake specific consumption and torque, and emissions, including NO_x, CO₂, and CO. The biodiesel blends (SDMB 10 and SDMB 20) have been optimised by a response surface methodology with central compositing to maximise braking power and minimise fuel consumption and emissions. The hydrogen supplementation has significantly improved the combustion characteristics, increased braking power and torque while reducing brake-specific fuel consumption (BSFC) and CO emissions, especially at higher hydrogen flow rates (6 L per minute). NO_x emissions showed a slight increase with hydrogen enrichment, while CO₂ control remained difficult. The multi-mode optimization has demonstrated optimum operating conditions, reaching high values of power (7.74 kW), torque (26.82 Nm), fuel efficiency (243.16 g BSCF) and emissions, emphasizing the balance between performance and environmental performance. This study confirms the potential of blending microalgae-enriched biodiesel with H₂ to refine the performance of CI engines with acceptable emission profiles.

Keywords: Microalgae biodiesel; Hydrogen enrichment; Dual-fuel CI engine; Injection parameters; RSM optimization.

1. INTRODUCTION

The combination of limited hydrocarbon resources and rising energy demand has augmented efforts to build cleaner, cleaner energy substitute. Compression ignition (CI) engines, which are widely used in the industrial and mobility sectors owing to their raised efficiency, contribute significantly to environmental pollution because of their reliance on fossil fuels. Replacing diesel with alternative fuels is a viable approach to reducing exhaust gases and reliance on conventional energy sources (Prasad *et al.* 2020; Oyekunle *et al.* 2023). Various renewable fuels, including biodiesel, natural gas, hydrogen, and ethanol, have been explored for CI engines because of their favourable combustion characteristics and partial compatibility with the existing infrastructure (El-Sheekh *et al.* 2022). Among these, hydrogen and biodiesel are particularly promising alternatives. Hydrogen combusts cleanly, producing only water, and its high flame speed and energy density enhance the combustion efficiency while reducing carbon-based emissions (Singh and Kumar, 2022). Although biodiesel is renewable and biodegradable, it generally exhibits reduced brake thermal energy (BTE) and increased brake-specific fuel consumption (BSFC) than petroleum diesel (Asif and Muneer, 2007). To mitigate these limitations, dual-fuel

strategies that combine biofuels with gaseous fuels, such as hydrogen, have been employed. The fast flame speed and clean-burning nature of hydrogen enhance biodiesel combustion, offsetting performance losses and further lowering emissions. However, the sustainability of biodiesel is strongly dependent on the feedstock. First- and second-generation sources (cooking oils, non-cooking crops, animal fats, and waste oils) face challenges, including conflict with food availability, upland land and water requirements, and narrow lipid yields (Dahlseide *et al.* 2024). Third-generation biodiesel derived from microalgae has attracted significant attention because of its high productivity, non-reliance on arable land, and ability to grow in saline or wastewater environments. Species such as *Scenedesmus dimorphous* exhibit high lipid accumulation, rapid growth, and adaptability to diverse conditions, making them highly suitable for sustainable biofuel production. Microalgae-derived biodiesel also demonstrates favourable fuel parameter, such as acceptable viscosity, good cetane index, and oxidative stability, ensuring its compatibility with CI engines (Rajak *et al.* 2020; Vinodraj *et al.* 2025). Despite its slightly lower calorific value than diesel, it enables cleaner combustion and reduces CO, HC, and smoke emissions, while requiring minimal engine modifications. Prior studies have evaluated the efficiency of microalgal biodiesel in combustion engines under

various operating conditions. (Uzar, 2020; Mata *et al.* 2010) reported that a B20 blend of *Chlorella vulgaris* biodiesel reduced CO and HC emissions, though it slightly decreased BTE and increased BSFC. (Vinodraj *et al.* 2025) examined *Chlorella prototheorids* biodiesel blends and confirmed their potential as renewable fuels. Wang *et al.* (Nguyen *et al.* 2024) and (Al-Iwayzy and Yusaf, 2017) demonstrated that addition of hydrogen to *Botryococcus braunii* biodiesel mixture improved combustion stability, reduced noise, and lowered emissions at a H₂ output rate of 5 litres per minute (LPM). Similarly, (Haik *et al.* 2011; Pata, 2021) highlighted the effects of operational variables on algae oil and its esters, showing properties comparable to diesel. (Rodriguez *et al.* 2017; Sekar *et al.* 2024) reported reduced CO₂ and particulate emissions with algae biodiesel blends, although NO_x increased slightly. Recently, (Sekar *et al.* 2024; Behera *et al.* 2019) studied biodiesel produced from *Chlorella vulgaris* enhanced by biogas and nanoparticles, achieving enhanced performance and declined HC, CO₂, and CO exhaust, albeit with slightly higher NO_x. Modelling and optimization methods have been employed to lower costs and shorten experimental time, with artificial intelligence techniques like neural networks and fuzzy modelling widely applied to model engine efficiency and exhaust. (Koç and Şener, 2021). Predictions derived from statistical optimization methods like Response Surface Methodology (RSM) closely match experimental data and enable reliable forecasting of engine efficiency and emissions for biodiesel-fuelled engines. (Dey *et al.* 2021) optimized a mono-cylinder CI engine fueled by ethanol blends and palm/diesel using RSM and ANN, achieving the best performance with a 20% palm biodiesel, 75% diesel, and 5% ethanol blend at rated load. (Krishnamoorthi *et al.* 2019) optimized diesel engine efficiency using ANN and RSM for blends of vegetable oil, diethyl ether, and diesel fuel at various load levels and compression ratios, with prediction errors under 5%. (Dey *et al.* 2024) utilized fuzzy-based Taguchi methods to improve the efficiency of CI engines and decrease emissions by varying biodiesel blend ratios and engine loads. (Shirmeshan *et al.* 2016) examined biodiesel-ethanol fuel blends using RSM and further fine-tuned the parameters with genetic algorithms. (Rith *et al.* 2019) applied RSM combined with the desirability function approach to optimize a dual gas-diesel engine, minimizing specific fuel consumption (SFC) and exhaust emissions at 3000 rpm, achieving an optimal desirability score of 0.829 with an input parameter of 11° BTDC.

This study addresses a gap in existing research by investigating the combined use of third-generation microalgal biodiesel and hydrogen as dual fuels in a CI engine. While biodiesel and hydrogen have been studied separately as alternative fuels, limited work has explored their synergistic application, particularly using *Scenedesmus dimorphus* methyl ester (SDMB). The engine was evaluated under full-load conditions at varying speeds, with hydrogen supplied at flow rates of 3

and 6 LPM through the intake manifold. Response Surface Methodology was employed to optimize key engine parameters, including the SDMB blend ratio, hydrogen flow rate, and engine speed, with the objective of maximizing performance and reducing emissions. The novelty of this work lies in the integration of third-generation microalgal biodiesel with hydrogen in a dual-fuel CI engine, combined with a systematic RSM-based optimization to identify optimal operating conditions. Although there is a slight rise in NO_x emissions due to increased in-cylinder temperatures, this method offers a carefully optimized solution for achieving cleaner, more efficient, and sustainable engine operation.

2. EXPERIMENTAL SETUP, PROCEDURE, MODELLING AND OPTIMIZATION

Fig. 1 shows the experimental setup, with (a) a system overview and (b) a visual view of the Kirloskar single-cylinder diesel engine. A mono-cylinder, water-cooled Kirloskar compression ignition engine, with a rated output of 8.5 kW and peak torque of 90 Nm at 3000 rpm, was utilized for the investigation, as detailed in Table 1. The engine operated at a fuel injection pressure of 220 bar and had an overall weight of approximately 300 kg. An eddy current engine tester was coupled with the engine to accurately measure the torque and brake power (BP). The hydrogen supply to the intake manifold was regulated using a mass flow transmitter to ensure precise control of the enrichment levels. The exhaust gases were routed to an AVL gas analyzer for emission characterization, including CO, HC, NO_x, and CO₂ levels. A data acquisition system (DAQ) was incorporated into the experimental arrangement to continuously gather and log engine performance and combustion metrics. This configuration facilitated a robust assessment of the influence of biodiesel-hydrogen fuel blends on the engine's operational efficiency and emission profiles.

Table 1. Engine specification

Engine Name	Kirloskar Diesel Engine
Cylinder count	One
Method of cooling	Water cooling
Turning moment	90 Nm @ 3000 rpm
Fuel IP	220 bar
Weight	300 kg
Torque Measurement	Eddy current engine tester
H ₂ gas regulation	H ₂ mass flow transmitter
Power of the engine	8.5 kW
Data collection	Data acquisition system
Emission measurement	AVL gas analyser

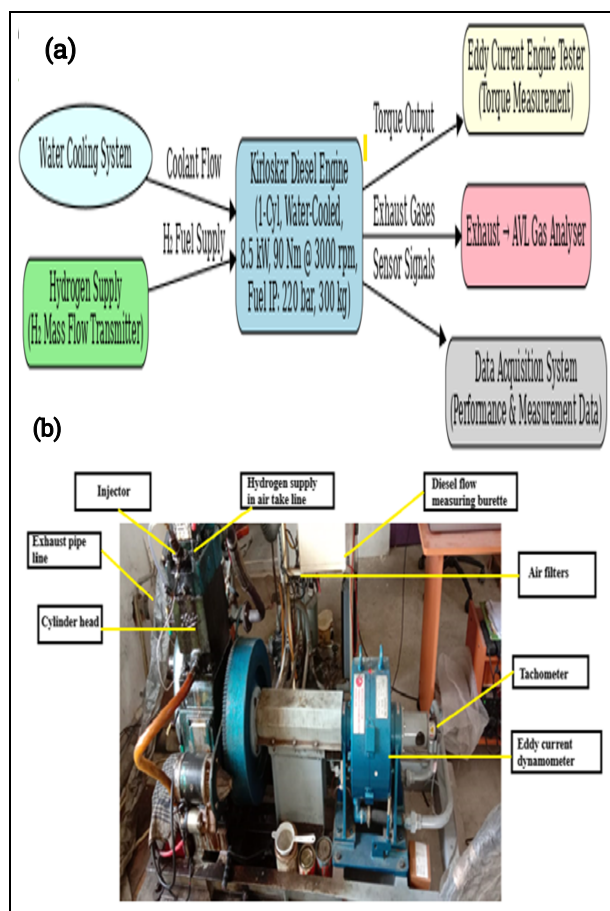


Fig. 1: Experimental setup: (a) Schematic diagram showing integration of hydrogen supply, cooling system, dynamometer, gas analyzer, and data acquisition system and (b) Photograph of the Kirloskar single-cylinder diesel engine

2.1 Microalgae Feedstock and Fuel Preparation

Scenedesmus dimorphus was selected as the biodiesel feedstock because of its high lipid content and adaptability to non-arable land with minimal nutrient requirements. The harvested biomass was filtered, oven-dried at 60 °C, and analyzed for lipid content. Lipid extraction was performed using an n-hexane–ethanol solvent mixture, and the extracted oil was characterized for fatty acid composition, confirming the predominance of triglycerides suitable for biodiesel production. The biodiesel SDMB was synthesized *via* base-catalyzed transesterification using potassium hydroxide and methanol (6:1 molar ratio) of 1 wt% at 60 °C for 2 hours under continuous stirring, followed by separation, washing, and drying, as illustrated in the process flowchart (Fig. 2).

Two volumetric blends of biodiesel and diesel were formulated: SDMB10 comprising 10% biodiesel and 90% diesel, and SDMB20 consisting of 20% biodiesel and 80% diesel. Hydrogen gas, exceeding 99.9% purity, was drawn from a high-pressure cylinder

and its flow was precisely modulated using a calibrated mass flow controller at rates of 3 and 6 LPM. Hydrogen was introduced into the air induction system to guarantee uniform dispersion with the intake air stream. The physicochemical parameters of the diesel and biodiesel mixtures were ascertained in compliance with ASTM protocols, and the findings are summarized in Table 2.

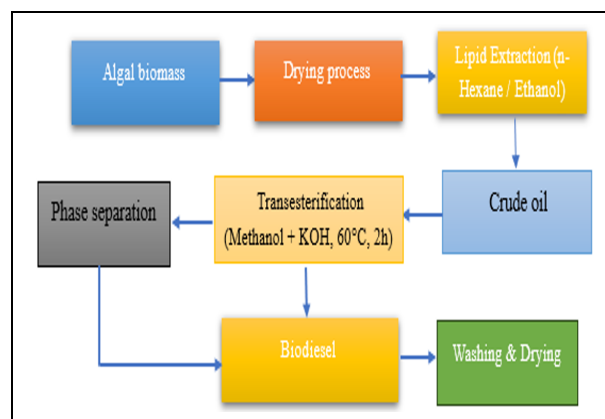


Fig. 2: Biodiesel production from microalgae: drying, extraction, transesterification, separation and purification

Table 2. Fuel parameters

Parameters	Diesel	SDMB	SDMB 10%	SDMB 20%	ASTM Standards
Cetane Number	90	130	132	137	ASTMD613
Density kg/m ³	832	835	841	845	
Flash Point °C	89	125	129	132	ASTMD93
KV mm ² / s	2.7	3.5	3.9	4.1	ASTMD445
Calorific Value (MJ /kg)	48	32.5	37.5	35.5	ASTMD40

2.2 Modelling and Optimization

Response Surface Methodology was utilized to refine both performance and emission properties of a compression ignition engine operating on a blend of microalgae biodiesel, diesel, and hydrogen enrichment. The aim centered on achieving peak brake thermal efficiency while reducing fuel consumption and limiting pollutants including nitrogen oxides, carbon monoxide, hydrocarbons, carbon dioxide, and smoke. The investigative framework was constructed utilizing Central Composite Design, enabling accurate capture of linear and second-degree nonlinear influences. The interplay between process variables and resulting engine responses is expressed through a quadratic polynomial equation.

$$Y = \beta_0 + \sum_{i=1}^k \beta_i X_i + \sum_{i=1}^k \beta_{ii} X_i^2 + \sum_{i < j}^k \beta_{ij} X_i X_j + \epsilon, \quad \dots (1)$$

where Y denotes the response (BTE, BSFC, or emission parameter), X_i represents the coded input variables (biodiesel blend ratio, hydrogen enrichment level, engine load, and injection pressure), β stands for regression

coefficients, and ε is the error term. The suitability of the model was confirmed by conducting Analysis of Variance (ANOVA), scrutinizing the significance levels (p-values), along with the coefficients of determination (R^2 and adjusted R^2) to ensure the model's reliability. Statistically significant terms were retained to improve model accuracy. Fig. 3 shows the methodology flowchart for engine efficiency and exhaust optimization.

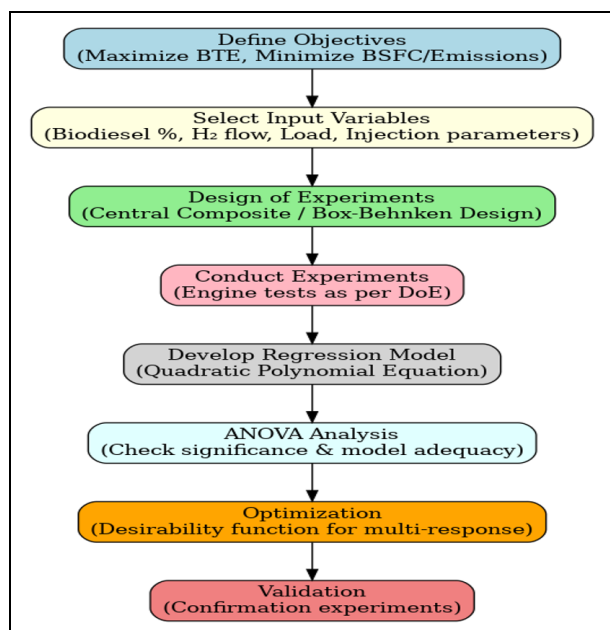


Fig. 3: Methodology flowchart for engine efficiency and emission optimization

3. RESULTS AND DISCUSSIONS

3.1 CI Engine Performance Graph

This study measured brake power, brake torque (BT), and BSFC to assess the performance of a CI engine. To evaluate the environmental impact, emissions of CO, CO₂, and NO_x were analyzed. Experiments were conducted at multiple engine speeds under full load conditions, both with and without hydrogen supplementation.

3.1.1 Brake Power of CI Engine

Fig. 4 presents the connection between BP and engine speed (rpm) for the different fuels tested with and without hydrogen addition. The lower volatility and delayed ignition of biodiesel lead to less efficient combustion, thereby reducing its performance. Introduction of hydrogen into the intake air enhances combustion quality and leads to noticeable improvements in brake power. Across all fuel types, BP increased consistently as engine speed raised from 1500 rpm to 3000 rpm. Neat diesel delivered the highest BP at raised speeds, while *Scenedesmus dimorphous* biodiesel also demonstrated improved performance with

increasing speed, consistent with the findings of earlier studies (Kumar *et al.* 2003). Hydrogen supplementation addresses the inherent limitations of biodiesel, such as its poor volatility and delayed ignition delay. This leads to improved air–fuel mixing and faster combustion, thereby enhancing the overall combustion efficiency across all biodiesel blends. At 1500 rpm, SDMB 10% and SDMB 20% exhibit 5–10% lower brake power than diesel, but 3 LPM hydrogen elevates power by 3–6%, and 6 LPM boosts it by 6–10%. Similarly, at 2000 rpm, biodiesel blends show 4–9% lower power, yet 3 LPM hydrogen raises it by 5–8%, and 6 LPM enhances it by 10–12%. This trend continues at 2500 rpm (7–10% improvement with 3 LPM; 12–15% with 6 LPM), peaking at 3000 rpm, where 6 LPM hydrogen supplementation delivered an 18% power increase, nearly matching diesel output. This improvement occurs because hydrogen burns quickly, has a wider flammability range, and mixes superior with air, leading to quicker and more complete combustion. Higher cylinder pressure and fewer carbon deposits help maintain performance, balancing the lower energy content of biodiesel and allowing it to perform almost like diesel under the right conditions.

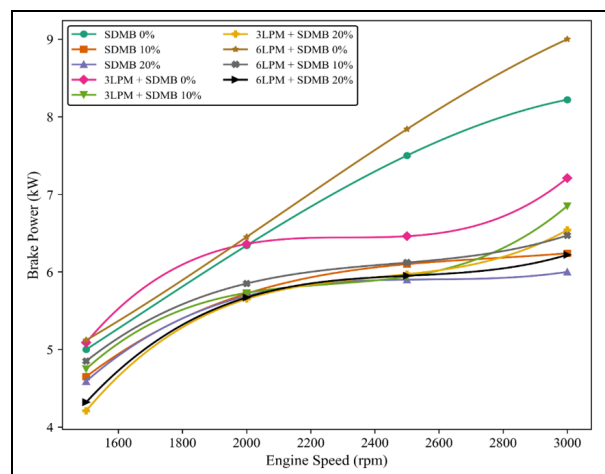


Fig. 4: Concentration of brake power with engine speed

3.1.2 Brake Torque of CI Engine

Fig. 5 illustrates the concentration of brake torque (Nm) with respect to engine speed (rpm) under different experimental conditions. Engine torque, the rotational force generated by the engine, is measured in Newton-meters (Nm) and typically decreases at elevated engine speeds owing to raised friction and reduced volumetric efficiency. Generally, brake torque increased from 1500 rpm to 2000 rpm, reaching its peak around 2000 rpm, before gradually declining as engine speed rises further, likely owing to limitations in air intake and fuel combustion efficiency. Pure diesel produced the highest BT, whereas SDMB 20% recorded the lowest without hydrogen addition. The decrease in BT at improved engine rpm is primarily linked to a reduction in oxygen intake and the lessened heating value of the fuel.

However, the results for SDMB 10% were found to be very similar to those of SDMB 0%. The raised energy concentration of hydrogen unit per mass compensates for the reduced heating value of biodiesel blends, resulting in improved power output. The mixture of *Botryococcus braunii* microalgae biodiesel and hydrogen showed that hydrogen-enriched biodiesel blends substantially increase engine torque.

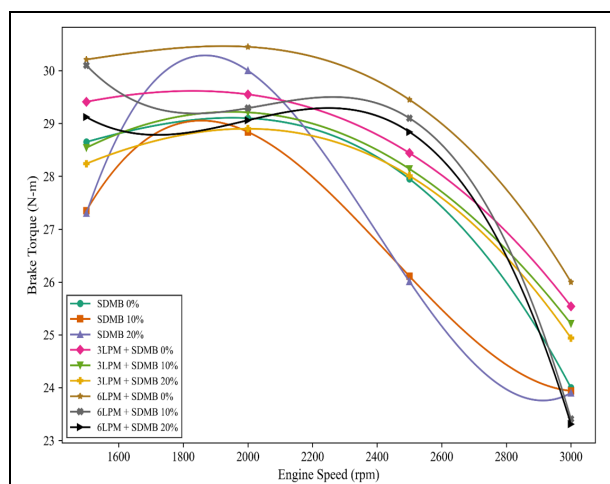


Fig. 5: Variations of brake torque (Nm) with engine speed (rpm)

SDMB 0% (pure diesel) produced the highest torque of about 30.2 Nm at 2000 rpm, followed closely by 6LPM + SDMB 0% (30.2 Nm) and 6LPM + SDMB 10% (29.0 Nm). This suggests that adding hydrogen enhanced the combustion process. Biodiesel-only mixture had lessened torque, with SDMB 20% reaching approximately 28.0 Nm owing to its lower energy content and higher thickness, which affect fuel flow and burning. At higher speeds (3000 rpm), hydrogen-enriched blends performed better, with 6LPM + SDMB 0% maintaining approximately 24.5 Nm, whereas 3LPM + SDMB 10% had approximately 23.2 Nm. A higher hydrogen supply (6LPM) yielded better results than 3LPM, proving that hydrogen helps burn fuel more efficiently and compensates for the lower energy in biodiesel.

3.1.3 BSFC of CI Engine

Fig. 6 illustrates the concentration of BSFC (g/kWh) with engine speed (rpm) for different fuel blends, including diesel (SDMB 0%), biodiesel (SDMB 10%, SDMB 20%), and hydrogen-enriched biodiesel (3LPM and 6LPM hydrogen addition). The amount of fuel an engine consumes to produce one unit of power is quantified by BSFC, making it a key indicator for assessing fuel efficiency. As the engine rpm increases, further fuel is required owing to friction, heat loss, and incomplete combustion, which increases the BSFC and lowers the overall thermal efficiency (Ge *et al.* 2022).

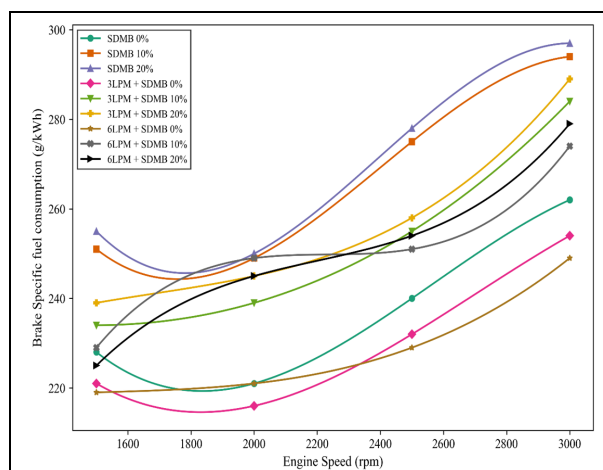


Fig. 6: Concentration of BSFC (g/kWh) with engine speed (rpm)

At lower engine speeds (1500–2000 rpm), BSFC remained relatively low, indicating better fuel efficiency. However, at 3000 rpm, BSFC increased across all blends, confirming that higher speeds require more fuel to sustain the power output. Among the tested blends, SDMB 20% exhibited the highest BSFC (290 g/kWh at 3000 rpm) owing to its lessened heating value and elevated viscosity, which reduced the combustion efficiency. In contrast, hydrogen-enriched blends (6LPM + SDMB 0% and 6LPM + SDMB 10%) demonstrated lower BSFC, with 6LPM + SDMB 0% recording approximately 250 g/kWh at 3000 rpm, reflecting improved combustion characteristics. The 6LPM hydrogen addition reduced BSFC by approximately 13.8% compared to SDMB 20%, indicating superior combustion efficiency. The higher hydrogen flow rate (6LPM) consistently outperformed 3LPM, confirming that increased hydrogen enrichment enhances combustion efficiency by accelerating flame propagation and optimizing fuel utilization.

3.1.4 CO of CI Engine

Fig. 7 shows the concentration of CO exhaust (ppm) with varying engine speeds (rpm). The results indicate that CO emissions are higher at lower engine speeds (1500–2000 rpm) and decrease with increasing speed (2500–3000 rpm) because of the improved combustion efficiency. Carbon monoxide is formed due to a lack of oxygen, causing incomplete fuel combustion and results in partially burned carbon. Among the evaluated mixtures, SDMB 20% showed increased CO emissions (1.9 ppm at 1500 rpm and 1.8 ppm at 2000 rpm), attributed to its elevated viscosity and lessened volatility, which resulted in incomplete burning. Diesel (SDMB 0%) also showed relatively high CO emissions (1.5 ppm at 1500 rpm), reflecting its higher carbon content and suboptimal air-fuel mixing at lower speeds.

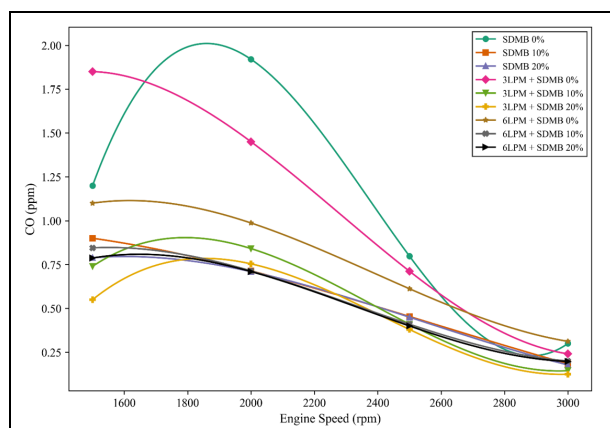


Fig. 7: Concentration of carbon monoxide (ppm) with engine speed (rpm)

In contrast, hydrogen-enriched blends significantly reduced CO emissions, with 6LPM + SDMB 0% recording 0.4 ppm at 1500 rpm and 0.3 ppm at 2000 rpm. This reduction occurs because hydrogen provides an additional lean combustion effect, enhances turbulence in the combustion chamber, and reduces carbon content in the fuel, leading to lower CO formation (Mohan *et al.* 2023). As engine speed increased to the range of 2500–3000 rpm, CO emissions declined for all fuel blends due to the rise in in-cylinder temperatures and enhanced mixing of air and fuel, which led to more effective combustion. At 3000 rpm, CO exhaust was lowest for 6LPM + SDMB 0% (0.1 ppm) and 6LPM + SDMB 10% (0.15 ppm), whereas SDMB 20% still exhibited relatively higher CO levels (0.4 ppm). The higher hydrogen flow rate (6LPM) consistently outperformed 3LPM across all speeds, indicating that increased hydrogen addition enhances oxidation reactions, reduced incomplete combustion, and minimized CO emissions.

3.1.5 CO₂ of CI Engine

Fig. 8 presents CO₂ emissions (%) for multiple fuel blends across changing engine speeds. The findings point out that CO₂ exhaust increases with rising engine speed owing to the improved combustion efficiency at higher temperatures. Carbon dioxide is produced when fuel burns completely, with the carbon in the fuel combining with oxygen. Better combustion, more oxygen, and higher temperatures lead to the formation of more CO₂ (Tayari and Abedi, 2019). At 1500 rpm, CO₂ emissions remained lower (8.5–9.0%) for all blends owing to incomplete combustion at low temperatures. At 3000 rpm, CO₂ emissions peaked (12.0% for SDMB 0%), demonstrating higher oxidation rates. Biodiesel blends exhibit slightly lower CO₂ emissions (11.5% for SDMB 20%) owing to their inherent oxygen concentration, which boosts combustion but also reduces the carbon fraction of the fuel. Hydrogen-enriched blends, particularly 6LPM + SDMB 0%, show CO₂ levels close to diesel (11.8%), indicating that hydrogen

addition improves combustion through enhanced air-fuel mixing and increased reaction kinetics. The results confirmed that hydrogen promotes complete oxidation, increasing CO₂ formation while reducing unburned carbon emissions such as CO.

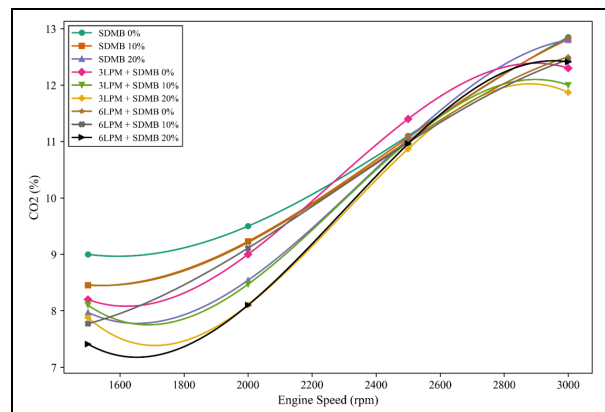


Fig. 8: Variation of carbon dioxide (%) with engine speed (rpm)

3.1.6 NO_x of CI Engine

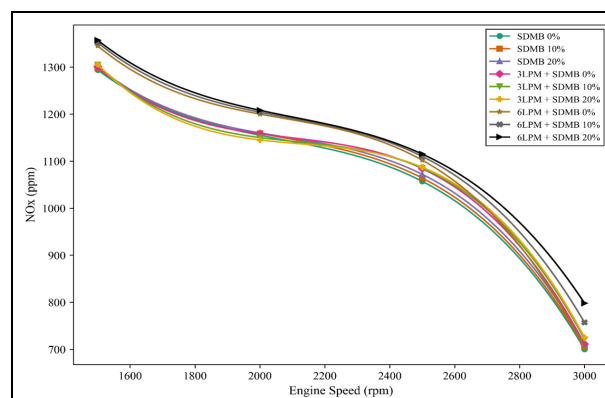


Fig. 9: Variations of oxides of nitrogen (ppm) with engine speed (rpm)

Fig. 9 shows the variation in NO_x exhaust (ppm) relative to engine speed for different fuel blends, including SDMB, diesel, and hydrogen-enriched biodiesel. NO_x emissions primarily originate from elevated combustion temperatures and surplus oxygen, which facilitate the chemical interaction between nitrogen and oxygen present in the air (Jayabal, 2024). At 1500 rpm, NO_x emissions were the highest, reaching 1400 ppm for SDMB 0%, owing to the prolonged combustion duration at lower speeds, leading to higher in-cylinder temperatures. Biodiesel blends (SDMB 10% and SDMB 20%) showed slightly lower NO_x emissions (1350 ppm and 1320 ppm, respectively) owing to their increased oxygen concentration, which promoted better burning but also resulted in lower peak flame temperatures. At 2500 rpm, NO_x emissions stabilized at approximately 1050–1100 ppm, as improved turbulence enhanced mixing but slightly reduced the residence time

for NO_x formation. At 3000 rpm, NO_x emissions dropped significantly, with SDMB 0% recording 750 ppm, as shorter combustion durations and reduced in-cylinder temperatures limited NO_x formation. Hydrogen enrichment at 6 LPM increased NO_x exhaust owing to its elevated combustion temperature and enhanced reaction kinetics, with 6LPM + SDMB 0% recording 1380 ppm at 1500 rpm. However, at higher speeds, NO_x levels decrease because of the faster flame propagation and shorter residence time for nitrogen oxidation reactions. These results indicate that while the addition of hydrogen enhances combustion efficiency, it also necessitates the implementation of NO_x reduction techniques, such as exhaust gas recirculation or water injection to effectively control emission.

3.2 Response Analysis

A second-order response surface model was developed using 36 experimental runs to evaluate the exhaust and efficiency features of the CI engine under varying biodiesel mixture, speed, and hydrogen enrichment. The design matrix consisted of 14 factorial points, 10 axial (star) points, and 12 runs at the centre level, which are utilized to estimate the pure error component and evaluate the reproducibility of the experimental results.

3.3 Model Validation

The quadratic model was chosen as the optimal fit due to its status as the highest order polynomial, indicating no correlation between additional terms and the model itself. The formulated equations, expressed in coded variables namely, Speed (rpm), Blend (%), and Hydrogen flow rate (LPM), represent the brake power, brake thermal efficiency, brake specific fuel consumption, carbon monoxide, carbon dioxide, and nitrogen oxides according to Equations (1) through (6), respectively.

$$\text{Brake power (kW)} = -0.22 + (0.000006 \text{ speed}) - (0.00322 \text{ blend}) - (0.0647 \text{ hydrogen flow rate}) - (0.000000 \text{ speed} \times \text{speed}) + (0.000037 \text{ blend} \times \text{blend}) + (0.00221 \text{ hydrogen flow rate}) - (0.000000 \text{ speed} \times \text{blend}) + (0.000000 \text{ speed}) \times (\text{hydrogen flow rate}) - (0.000095 \text{ blend} (\%) \times (\text{hydrogen flow rate})). \quad \dots (1)$$

$$\text{Brake Torque (Nm)} = 12.55 + (0.000022 \text{ speed}) - 0.00825 \text{ blend} + (0.257 \text{ hydrogen flow rate}) - (0.000000 \text{ speed} \times \text{speed}) + (0.000033 \text{ blend} \times \text{blend}) - 0.00348 \text{ hydrogen flow rate} + (\text{hydrogen flow rate} - 0.000000 \text{ speed} \times \text{blend}) - (0.000000 \text{ speed} \times \text{hydrogen flow rate}) - (0.000228 \text{ hydrogen flow rate}). \quad \dots (2)$$

$$\text{BSFC (g/kWh)} = 2784.4 - (0.000086 \text{ speed} + 0.2953 \text{ blend}) - (1.414 \text{ hydrogen flow rate}) + (0.000000 \text{ speed} \times \text{speed}) - (0.001092 \text{ blend} \times \text{blend}) + (0.0473 \text{ hydrogen flow rate} \times \text{hydrogen flow rate}) + (0.000000 \text{ speed} \times \text{blend}) - (0.000000 \text{ speed} \times \text{hydrogen flow rate}) - (0.00306 \text{ blend} \times \text{hydrogen flow rate}). \quad \dots (3)$$

$$\text{Carbon monoxide (ppm)} = 1.593 + (0.000001 \text{ speed}) - (0.01202 \text{ blend}) - (0.0192 \text{ hydrogen flow rate}) - (0.000000 \text{ speed} \times \text{speed}) + (0.000018 \text{ blend} \times \text{blend}) - (0.000270 \text{ hydrogen flow rate} \times \text{hydrogen flow rate}) + (0.000000 \text{ speed} \times \text{blend}) + (0.000000 \text{ speed} \times \text{hydrogen flow rate}) + (0.000082 \text{ blend} \times \text{hydrogen flow rate}). \quad \dots (4)$$

$$\text{Carbon dioxide (\%)} = 8.43 - (0.000001 \text{ speed} - 0.00809 \text{ blend}) - (0.1173 \text{ hydrogen flow rate}) + (0.000000 \text{ speed} \times \text{speed}) - (0.000002 \text{ blend} \times \text{blend}) + (0.00327 \text{ hydrogen flow rate} \times \text{hydrogen flow rate}) + (0.000000 \text{ speed} \times \text{blend}) + (0.000000 \text{ speed} \times \text{hydrogen flow rate}) - (0.000015 \text{ blend} \times \text{hydrogen flow rate}). \quad \dots (5)$$

$$\text{Nitrogen oxide (ppm)} = 898 + (0.000779 \text{ speed}) - (0.222 \text{ blend}) - (2.07 \text{ hydrogen flow rate}) - (0.000000 \text{ speed} \times \text{speed}) - (0.00012 \text{ blend} \times \text{blend}) + (0.190 \text{ hydrogen flow rate} \times \text{hydrogen flow rate}) + (0.000000 \text{ speed} \times \text{blend}) - (0.000000 \text{ speed} \times \text{hydrogen flow rate}) + (0.0065 \text{ blend} \times \text{hydrogen flow rate}). \quad \dots (6)$$

The suitability and adequacy of the quadratic model, along with the impact of significant individual variables and their correlation with the selected responses, were statistically evaluated using ANOVA. A summary of these findings is provided in Table 3.

Table 3. Range of used models and their ANOVA-derived p-values

Models	p-value					
	BP (kW)	BT (Nm)	BSFC (g.kWh)	CO (ppm)	CO ₂ (%)	NO _x (ppm)
Model	0.000	0.000	0.000	0.000	0.000	0.000
Linear	0.001	0.000	0.000	0.000	0.099	0.003
Speed (rpm)	0.000	0.000	0.000	0.338	0.633	0.000
Blend (%)	0.508	0.308	0.000	0.000	0.079	0.697
Hydrogen Flow Rate (LPM)	0.267	0.014	0.060	0.426	0.059	0.882
Square	0.004	0.000	0.000	0.015	0.023	0.000
Speed (rpm) × speed (rpm)	0.013	0.000	0.000	0.026	0.010	0.000
Blend (%) × Blend (%)	0.009	0.197	0.000	0.014	0.908	0.942
Hydrogen Flow Rate (LPM) × Hydrogen Flow Rate (LPM)	0.181	0.267	0.029	0.756	0.068	0.353
2- Way Interaction	0.021	0.431	0.008	0.024	0.320	0.876
Speed (rpm) × Blend (%)	0.004	0.879	0.006	0.007	0.081	0.526
Speed (rpm) × Hydrogen Flow Rate (LPM)	0.363	0.239	0.645	0.562	0.549	0.900
(Blend %) × Hydrogen Flow Rate (LPM)	0.360	0.253	0.027	0.148	0.895	0.617

3.3.1 Performance Model – BP, BT and BSFC

Table 4. Accuracy metrics for various response variables

Response	Precision Index Values			
	R ²	Adj-R ²	Pred R ²	Adeq Precision
BP (kW)	0.9096	0.8784	0.8099	36.76
BT (Nm)	0.9200	0.8923	0.8475	70.30
BSFC (g/kWh)	0.9672	0.9559	0.9371	47.04
CO (ppm)	0.8487	0.7963	0.6808	19.68
CO ₂ (%)	0.9640	0.9516	0.9392	39.39
NO _x (ppm)	0.9648	0.9574	0.9453	45.93

The ANOVA results confirmed the significance of the BP, BT, and BSFC models ($p = 0.000$). For BP, engine speed was the most dominant factor ($p < 0.001$), with quadratic and interaction effects such as Speed² ($p = 0.013$) and Speed $\times$ Blend ($p = 0.004$) contributing

meaningfully (Table 3). The model demonstrated strong adequacy with $R^2 = 0.9096$, Adjusted $R^2 = 0.874$, Predicted $R^2 = 0.8099$ and adequate precision = 36.76. The Actual vs. Predicted plot (Fig. 10 a) shows that most points closely aligned with the 45° line, with a maximum deviation of $\pm 3\%$. For BT, speed remained the most influential factor ($p < 0.001$), followed by hydrogen enrichment ($p = 0.014$), with significant quadratic terms (Speed², $p < 0.001$). The model adequacy as shown in the Table 4 indices were $R^2 = 0.9200$, Adjusted $R^2 = 0.8923$, Predicted $R^2 = 0.8475$ and adequate precision = 70.30, with Actual vs. Predicted points (Fig. 10 b) deviating within $\pm 4\%$. For BSFC, speed ($p < 0.001$) and blend ratio ($p < 0.001$) were dominant, with significant quadratic effects (Speed², Blend², $p < 0.001$). The model showed strong adequacy where $R^2 = 0.9672$, Adjusted $R^2 = 0.9559$, Predicted $R^2 = 0.9371$ and adequate precision = 47.04 and the Actual vs. Predicted plot (Fig. 10 c) indicated near-perfect alignment within $\pm 3\%$ deviation, confirming robustness and predictive accuracy across operating conditions.

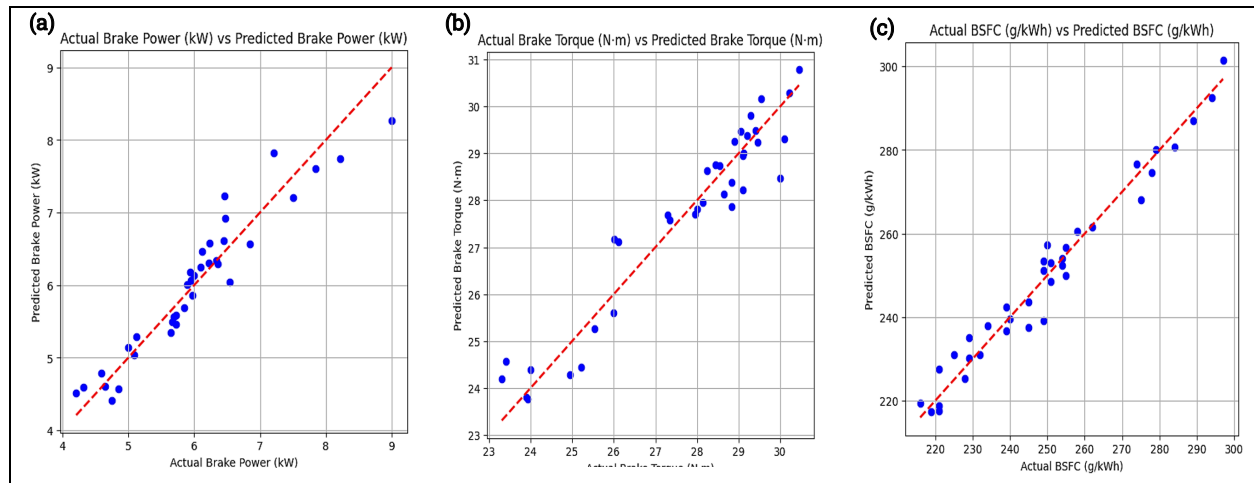


Fig. 10: (a), (b) and (c) Actual vs Prediction graph for BP, BT and BSFC

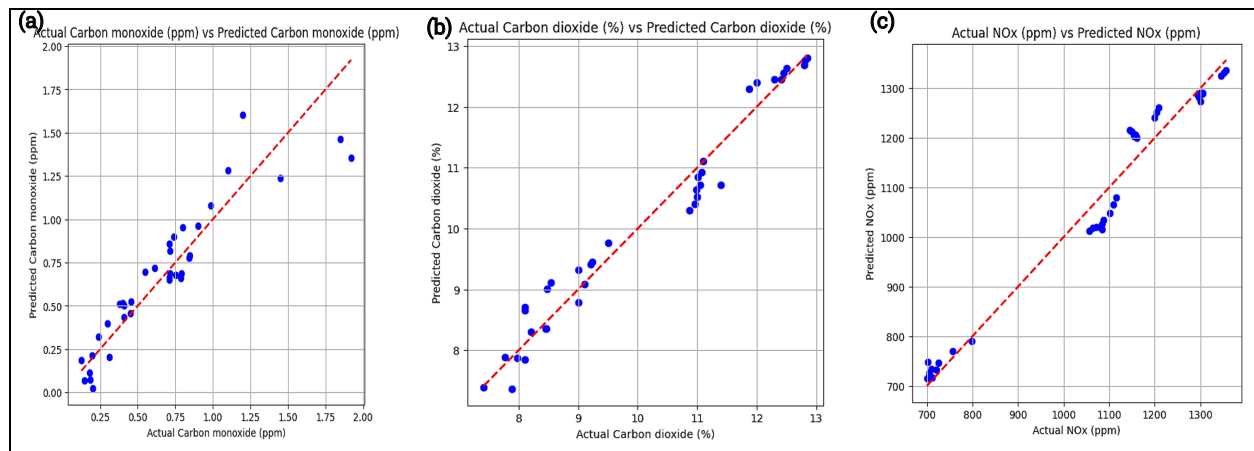


Fig. 11: (a), (b) and (c) Actual vs Prediction graph for CO, CO₂ and NO_x

The ANOVA results confirmed the significance of the emission models for CO, CO₂, and NO_x ($p = 0.000$). For CO, blend ratio was the most influential and significant factor ($p < 0.001$) (Table 3) with quadratic effects of Speed² ($p = 0.026$) and Blend² ($p = 0.014$). The model showed moderate adequacy with, $R^2 = 0.8487$, Adjusted $R^2 = 0.7963$, Predicted $R^2 = 0.6808$, and adequate precision = 19.68. The Actual vs. Predicted plot (Fig. 11a) showed most points along the 45° line, with minor scatter at higher CO levels. For CO₂, speed had limited linear influence ($p = 0.633$), while quadratic terms such as Speed² ($p = 0.010$) were significant and blend ratio showed borderline significance ($p = 0.079$).

The model exhibited strong adequacy $R^2 = 0.9640$, Adjusted $R^2 = 0.9516$, Predicted $R^2 = 0.9392$, and adequate precision = 39.39 with Actual vs. Predicted points (Fig. 11b) closely aligned along the 45° line within $\pm 3\%$ deviation. For NO_x, speed was the dominant contributor ($p < 0.001$), with significant quadratic effects (Speed², $p < 0.001$) and minor contributions from blend and hydrogen enrichment in interaction terms. The model precision was excellent with $R^2 = 0.9648$, Adjusted $R^2 = 0.9574$, Predicted $R^2 = 0.9453$, and adequate precision = 45.93. The Actual vs. Predicted plot (Fig. 11c) showed very close clustering along the 45° line, with deviations within $\pm 3\%$, demonstrating high predictive accuracy across operating conditions.

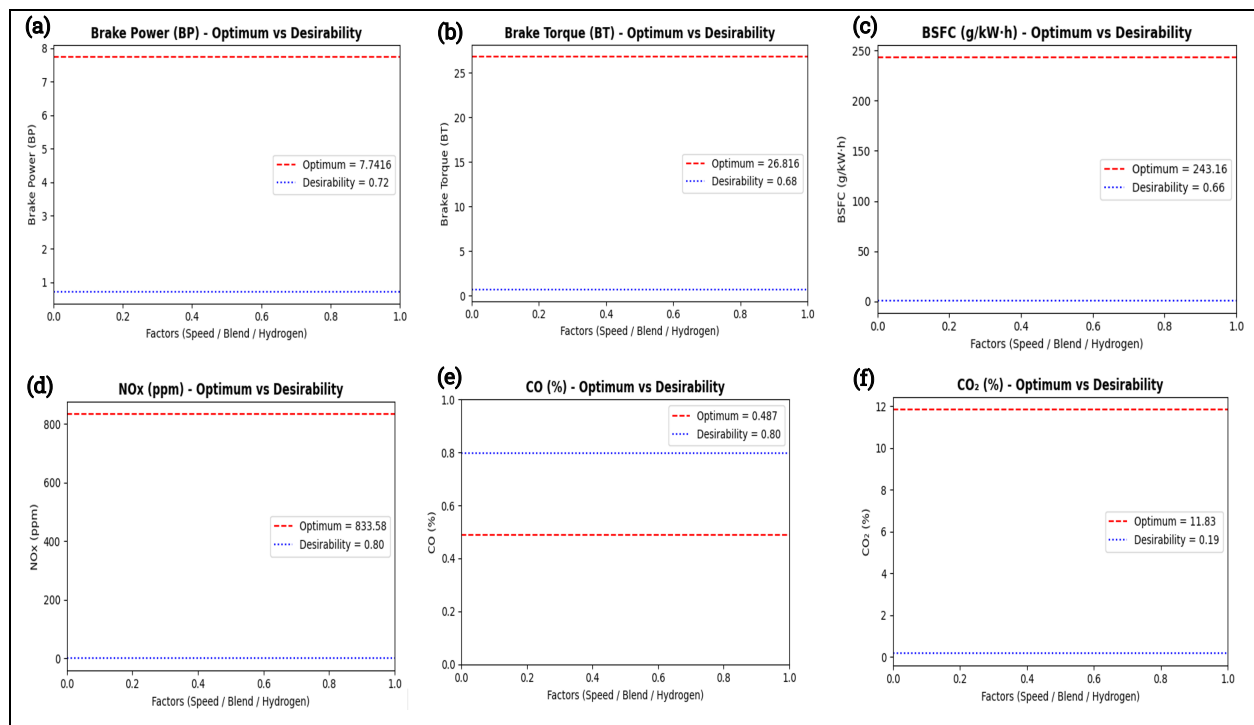


Fig. 12: Optimization and desirability plots for engine emission and performance parameters

3.4 Refinement of Parameters and Confirmation of Experimental Results

Fig. 12 (a to f) show the optimization and desirability plots of various emission and performance parameters for a CI engine under different operating conditions involving speed, blend percentage, and hydrogen enrichment. The engine achieved an optimum BP of 7.7416 kW and BT of 26.816 Nm, with corresponding desirability values of 0.72 and 0.68, respectively, indicating strong alignment with target performance goals and demonstrating significant influence of blend and hydrogen enrichment on power and torque output. The BSFC was optimized at 243.16 g/kWh with a desirability of 0.66, reflecting fuel consumption close to optimal and highlighting improved

efficiency potential under tested conditions. Emission analysis showed NO_x emissions optimized at 833.38 ppm with a desirability of 0.80, indicating a moderate balance between performance and nitrogen oxide formation. CO emissions were minimized at 0.487% with desirability also at 0.80, while CO₂ emissions had an optimum of 11.83% but a low desirability of 0.19, suggesting challenges in controlling CO₂ without affecting engine performance adversely. Overall, the multi-response optimization approach effectively balances diesel engine performance and emission reduction, with high desirability in power, torque, fuel burn rate, and NO_x release, while CO₂ reduction remains challenging and may require additional strategies such as after-treatment technologies or alternative fuel blends.

4. CONCLUSION

The experimental investigation and multi-response optimization of a Kirloskar mono-cylinder CI engine fueled with hydrogen-enriched microalgae biodiesel demonstrated notable improvements in engine performance and emissions. Optimal operating conditions yielded a brake power of 7.74 kW and a brake torque of 26.82 Nm, indicating enhanced combustion due to hydrogen supplementation, along with improved fuel efficiency reflected by a reduced brake specific fuel consumption of 243.16 g/kWh. Emission optimization resulted in lower NO_x levels of 833 ppm and minimized CO emissions of 0.487%, while CO₂ reduction remained limited at an optimum value of 11.83%, highlighting the need for additional mitigation strategies. Response Surface Methodology using Central Composite Design produced robust quadratic models with R² values exceeding 0.85, enabling effective optimization of engine speed, blend ratio, and hydrogen flow rate. Overall, the results validate hydrogen-enriched microalgae biodiesel as a promising sustainable fuel for CI engine operation, offering enhanced performance and reduced carbon monoxide emissions, while further efforts are required to comprehensively address nitrogen oxide and carbon dioxide emissions.

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CONFLICTS OF INTEREST

The authors declare that there is no conflict of interest.

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