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Research Article

Effect of hydrogen co-firing on combustion efficiency, steam production, and emissions of a B7 diesel water-tube boiler: An experimental and semi-empirical combustion modeling study

Somkid Warasinchai¹ and Pongsakorn Kachapongkun^{*2}

Rattanakosin College for Sustainable Energy and Environment, Rajamangala University of Technology Rattanakosin (RMUTR), Thailand

Abstract. Hydrogen co-firing provides a near-term opportunity to reduce emissions from industrial steam boilers, particularly in Thailand where B7 diesel containing 7% palm-oil fatty acid methyl ester (FAME) is the required diesel fuel. However, clear evidence of this system for this type of boiler is still insufficient. In this study, a systematic investigation was performed on a 1,000 kg/h water-tube boiler (Kawasaki 15750) at hydrogen/diesel energy proportions varying from 0 to 30% under a constant low-fire condition. Supplemental oxygen was precisely controlled to maintain an almost constant stoichiometric concentration. The results were further analyzed with a semi-empirical combustion model using extended Zeldovich NO_x kinetics. At an H₂/diesel energy ratio of 30%, steam production increased by 24.3%, from 323.8 to 402.4 kg/hr. Although diesel consumption rate is almost constant, the addition of hydrogen increased overall fuel consumption energy input by 29 %. The concentration of CO decreased by 73.4%, from 107.0 to 28.5 ppm, and CO₂ emission per unit of steam decreased by 20.1%, from 196.9 to 157.3 g/kg. Measured NO₂ increased by 70.8%, consistent with predicted NO_x ranges of 36–150 ppm at baseline and 62–256 ppm at 30% H₂, depending on the assumed NO₂/NO_x fraction. Multivariable analysis indicated that hydrogen addition had a greater impact on the emission response than oxygen enrichment. System efficiency, computed using an input-output method based on ASME PTC 4.1, decreased by 3.60 percentage points. Because the drum water inventory was not recorded at the beginning and end of each test, this decrease reflected a maximum limit on the efficiency loss. Presently, the price of hydrogen costs 753 THB/kg (23 USD/kg). So, hydrogen co-firing was still more expensive than diesel-only operation at all tested ratios. As the price of hydrogen decreases to approximately 120–230 THB/kg (3.7–7.0 USD/kg), which is within the range projected by IRENA for green hydrogen before 2030, hydrogen co-firing could become economically competitive. Therefore, under current hydrogen price, the hydrogen co-firing mainly contributes to increased steam production and lower CO and CO₂ emissions, but it does not provide fuel-cost savings and results in higher NO_x emissions.

Keywords: ASME PTC 4.1; B7 biodiesel; CO reduction; CO₂ intensity; Decarbonization; Emission index; Extended Zeldovich kinetics; Fuel-cost sensitivity; NO_x; Oxygen enrichment



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1. Introduction

In 2024, heat represented almost half of total final energy consumption and 37% of energy-related CO₂ emissions. However, renewable sources supplied only 14% of global heat demand. The renewable share of industrial heat became even lower, at 12%, and it is expected to reach only 16% by 2030 (IEA, 2025). It is also known that decarbonizing industrial heat is difficult because it is generally produced on-site and must satisfy a wide range of process temperature specifications. As a result, the replacement of existing fuels and thermal systems is often difficult. In addition, the use of electricity for industrial heating is still limited, accounting for less than 5% of global industrial heat consumption (IEA, 2026). Thailand has a significant use of fossil fuels within its energy industry (Energy Policy and Planning Office, 2023). Fossil fuels remain the main source of industrial heat in Southeast Asia (IEA, 2024). Water-tube boilers are extensively utilized to generate steam for the food-processing, textile, chemical, and palm-oil industries.

Nevertheless, their operation causes CO, NO_x, SO₂, and CO₂ emissions. In Thailand, B7 diesel containing 7% v/v palm-oil fatty acid methyl ester (FAME) is required by the Ministry of Energy (Department of Energy Business, 2020). Water-tube boilers with capacities of 0.5–5 MW are frequently employed for industrial steam generation (Department of Industrial Works, 2024; Energy Policy and Planning Office, 2021).

Presently, hydrogen can be co-fired with the existing fuel without major modifications to the boiler system. At low-to-moderate mixing ratios, this technique can use most of the current boiler equipment without requirements for burner replacement or major modifications to the control system. Technically, hydrogen has several combustion properties, contributing to its use in boilers. For example, it has a wide flammability range of 4.0–75.0 vol.% in air and a burning velocity of 2.65–3.25 m/s under normal temperature and pressure conditions. This fuel also has a wide flammability range and a higher burning velocity than standard hydrocarbon fuels (NASA, 1997). Furthermore, hydrogen contains no carbon.

* Corresponding author

Email: pongsakorn.kerd@rmutr.ac.th (P. Kachapongkun)

Thus, it cannot produce CO₂ directly during combustion. Recent reviews have examined these combustion characteristics, together with the technical modifications, economic limitations, and safety concerns of hydrogen utilization (Capurso *et al.*, 2022; Long Huynh *et al.*, 2024). Their results are useful. However, most previous investigations paid attention mainly to internal-combustion engines. Thus, quantitative evidence for hydrogen co-firing in liquid-fuel industrial boilers remains limited, particularly for diesel-biodiesel blends such as B7.

Presently, experimental information on H₂ co-firing in liquid-fuel industrial boilers remains limited. In addition, most of the previous studies have focused on H₂–natural gas mixtures in domestic appliances and industrial gas burners (Boulahlib *et al.*,

2021; Gersen *et al.*, 2020) or in industrial gas turbines (Laget *et al.*, 2023). Nevertheless, conventional liquid fuels and natural gas have different combustion chemistry. Studies combining biodiesel and H₂ have mainly focused on compression-ignition engines (Jayabal & Sivanraju, 2025; Periyathambi *et al.*, 2025). In their review of dual-fuel diesel engines, Hosseini *et al.* (2023) discovered a common emission pattern. The hydrogen addition typically reduced CO, unburnt hydrocarbons, and particle emissions, but increased NO_x because of the elevated cylinder temperatures. They also found that the H₂ addition rate when knocking occurs is related to the engine, pilot fuel, and operating conditions. Therefore, a single effective H₂ addition rate cannot be applied to all engines. The mechanisms found in

Table 1.
H₂ blending ratios reported in the recent literature on hydrogen co-firing in industrial combustion systems, set against the present study.

Ref	Study / System	Method	Primary fuel	H ₂ range	Key finding
(Boulahlib <i>et al.</i> , 2021)	Boulahlib <i>et al.</i> 2021 — domestic boiler, 15 kW (16-burner).	Experimental	CH ₄	0–45 vol.%	+1.2% combustion efficiency and –16% CO ₂ at 40% H ₂ ; optimum at 20% H ₂ with fuel-rich staging factor 2.0.
(Gersen <i>et al.</i> , 2020)	Gersen <i>et al.</i> 2020 — forced-draught industrial burner, 475 kW boiler.	Experimental + System development	Natural gas	up to ~30 vol.%	No substantial change in flame stability across 0–100 vol.% H ₂ .
(Laget <i>et al.</i> , 2023)	Laget <i>et al.</i> 2023 — industrial gas turbine, 24 MW (Siemens SGT-600)	Experimental (industrial demonstration)	Natural gas	0–25 vol.%	Stable operation across base/part load; NO _x and combustion dynamics monitored.
(Lan <i>et al.</i> , 2024)	Lan <i>et al.</i> 2024 — NG industrial heating furnace.	Experimental + CFD	Natural gas	15–60 vol.% (4 mixtures)	Progressive CO reduction with H ₂ ; OH-mediated burnout dominant.
(Wang <i>et al.</i> , 2022)	Wang <i>et al.</i> 2022 — industrial steam boiler, 108 t/h.	Experimental (case study)	Natural gas	0–90 vol.%	Trade-off between thermal efficiency and NO _x ; quantified decarbonization potential.
(Nemitallah <i>et al.</i> , 2024)	Nemitallah <i>et al.</i> 2024 — industrial water-tube boiler, 207 MW.	Experimental (retrofit)	Natural gas	0–100 vol.%	Identified hardware modifications required; NO _x rises sharply at low H ₂ and stabilizes above 30%.
(Sun <i>et al.</i> , 2024)	Sun <i>et al.</i> 2024 — gas-fired steam boiler, 2 t/h (WNS2-1.25-Q).	Numerical (kinetic modeling) + Experimental (field test)	Natural gas	0–25 vol.%	CHEMKIN PRO (PREMIX + OPPDIF) for kinetics + on-site field test; trade-off between thermal efficiency and NO _x emission as H ₂ ratio rises.
(Xin <i>et al.</i> , 2021)	Xin <i>et al.</i> 2021 — casing swirl burner gas boiler, 200 kW.	CFD	Natural gas	0–100 vol.% (6 conditions)	CFD with constant power and constant excess air; H ₂ increases peak temperature, accelerates combustion rate, reduces CO ₂ and total flue gas, and inhibits NO _x and soot under their fixed-stoichiometry constraint.
(Schwarz <i>et al.</i> , 2024)	Schwarz <i>et al.</i> 2024 — semi-industrial oxy-fuel burner, 100–140 kW.	Experimental	Natural gas (O ₂ enriched)	0–100 vol.%	Pure-H ₂ lowers flue-gas T by 126 K and raises load-cooling capacity by 16.3% at constant fuel energy; heat-transfer shifts toward more convective load.
(Glanville <i>et al.</i> , 2022)	Glanville <i>et al.</i> 2022 — partially premixed combustion equipment.	Experimental	Natural gas	0–30 vol.%	Reliable operation up to 30% H ₂ with minor performance changes; consistent NO _x rise.
(Premasathit <i>et al.</i> , 2025)	Premasathit <i>et al.</i> 2025 — industrial burner, diesel with hydrogen.	Numerical, with experiment validating the diesel-only case	Diesel	~23% of fuel heating value	Higher flame temperature at fixed heat input and lower CO ₂ . Equivalence ratio varied from 0.65 to 1.10, so part of the temperature response is nitrogen dilution rather than the fuel change.
This study	B7 diesel water-tube boiler, 1,000 kg/hr (Kawasaki 15750, low-fire).	Experimental + Semi-empirical correlation	B7 diesel	0–30% H ₂ /diesel energy	+24.3% steam at 30%; –20.1% CO ₂ /kg steam; –73.4% CO emission; +70.8% measured NO ₂ ; efficiency penalty bounded at ≤3.6 pp (ASME PTC 4.1 Direct Method).

engine research, such as accelerated radical generation, reduced ignition delay, and the formation of leaner localized flame regions, could also occur in diesel-fired boilers. However, these results cannot be directly applied to boilers since boiler flames are not premixed and normally operate at lower peak temperatures than combustion engines. Khiraiya *et al.* (2021) and Lin and Wu (2023) investigated biodiesel mixtures from 25% biodiesel to pure biodiesel in experimental boilers, but no details of hydrogen co-firing were included in their works.

Recent studies on hydrogen co-firing system applied to industrial burners and boilers have shown alterations in NO_x emissions are mainly contributed to thermal processes (Lan *et al.*, 2024; Nemitallah *et al.*, 2024; Wang *et al.*, 2022). However, the NO_x response varied throughout the investigations. In a particular investigation, the direction of the NO_x response changed with the boiler operating strategy, but another study indicated a non-linear response to the hydrogen proportion. Sun *et al.* (2024) employed a chemical-reactor-network model to establish a comparable balance between boiler efficiency and NO_x emissions in a 2 t/h gas boiler. Xin *et al.* (2021) employed computational fluid dynamics to investigate the variation in heat emission within a 200-kW burner. Schwarz *et al.* (2024) found a 16.3% increase in load-cooling capacity in oxy-fuel environments.

Apart from these works, the co-firing of hydrogen with biodiesel-based fuels in industrial boilers has gained limited attention. Particularly, few studies have systematically analyzed combustion behavior, steam production, and thermal efficiency in B7-fired industrial boilers. Table 1 provides the H₂/fuel ratios used in previous studies and compares them with those examined in the present study. The study most closely comparable to the present work was reported by Premasathit *et al.* (2025). In this study, they investigated co-firing of diesel-hydrogen in an industrial oil burner with a water-cooled chamber. Hydrogen delivered 23% of the total fuel energy, which was close to the highest energy ratio examined in the present study. Their research employed computational fluid dynamics to set up a model which was subsequently validated by the experimental measurements of the pure-diesel condition. Flame temperature, hot-gas temperature, O₂, and CO₂ were monitored. Their results clearly show that the addition of hydrogen improved the flame temperature and decreased CO₂ with constant heat input. Nevertheless, only a single hydrogen ratio was studied. CO, NO_x, and steam output were not reported.

As indicated by the literature review, three research gaps, RG1 to RG3, were identified for the boiler configuration investigated in this study. RG1: the CO and NO_x patterns reported for hydrogen–natural-gas systems cannot immediately be applied to liquid-fuel boilers. In a distillate-fuel burner, the process of combustion is greatly affected by the atomization of the spray and the evaporation of droplets, rather than just by the mixing of gases. Therefore, the initial CO concentration and the level of CO reduction resulting from hydrogen addition may vary from those observed in gas-fired systems. RG2: previous studies combining hydrogen with biodiesel fuels have mainly focused on internal-combustion engines. In fact, these engines operate at higher peak temperatures and pressures than industrial boilers. Thus, the relationship between thermal NO_x formation and radical-enhanced CO oxidation may differ under the lower-temperature and non-premixed conditions of a boiler flame. RG3: no published study has reported both standard-method boiler efficiency and flue-gas emissions over a range of H₂ ratios in a diesel-fired water-tube boiler. The present study bridges this gap by measuring boiler efficiency and flue-gas emissions at seven H₂/diesel energy ratios.

Controlled experiments were performed on a commercial 1,000 kg/h vertical water-tube boiler (Kawasaki 15750) operating at the low-fire setting and generating saturated steam at 0 bar(g). A calibrated semi-empirical combustion model was employed to enhance the interpretation of the experimental results. The study investigated the effects of H₂/diesel energy ratios from 0 to 30% on CO, NO_x, CO₂, SO₂, flue-gas temperature, and thermal efficiency. It also evaluated the variation in steam production, established polynomial correlations for specific performance parameters within the tested range, and evaluated the economic and emission effects caused by hydrogen co-firing.

Four research questions, RQ1-RQ4, were established to clarify RG3. RQ1 investigates the effect of increasing the H₂/diesel energy ratio from 0 to 30% on boiler efficiency. It also examines whether the efficiency changes consistently in one direction across the tested ratios. Boiler efficiency was calculated from the feedwater totalizer readings using the Direct Method described in ASME PTC 4.1. Two opposing mechanisms may affect boiler efficiency. On one hand, hydrogen can increase the heat-release rate near the burner. On the other hand, the combustion products contain more H₂O and less CO₂, which may reduce radiative heat transfer. These competing effects may produce a non-monotonic efficiency response at low H₂ ratios. RQ2 examines whether hydrogen addition reduces flue-gas CO and whether the observed pattern can be expressed by a simple correlation within the tested range. RQ3 determines the increase in NO_x emissions and evaluates whether the emissions remain within the limits applied to industrial boilers in Thailand. RQ4 evaluates the hydrogen prices and blending ratios at which co-firing can compete with diesel operation on a fuel-cost basis. These questions guide the analysis and discussion of the experimental results.

This study presents boiler efficiency, steam production, and flue-gas emissions measured at seven H₂/diesel energy ratios in a commercial B7 diesel-fired water-tube boiler. The experiments were conducted on a production boiler operating at an industrial plant. A semi-empirical combustion model was used to analyze the experimental results and establish screening correlations within the examined range. Unlike most previous studies, which employed numerical simulations or reported efficiency and emissions separately, the present study evaluates these parameters together. Mean values obtained at the seven H₂/diesel energy ratios were previously reported in a preliminary Thai-language paper by Warasinchai and Kachapongkun (2026). Both articles are based on the same experimental program. Nevertheless, the preliminary paper presented only experimental observations. The present study extends the earlier work by adding four components: (1) a refined semi-empirical combustion model which includes advanced Zeldovich NO_x kinetics; (2) polynomial regression correlations for engineering applications (Eqs. 4–7); (3) an economic sensitivity analysis based on multiple hydrogen-price assumptions, including the IRENA Green Hydrogen Cost Reduction projections; and (4) a quantitative analysis of life-cycle CO₂ comparison of gray, blue, and green hydrogen.

2. Materials and Methods

2.1 Boiler System and Test Configuration

Experiments were conducted on a vertical water-tube boiler (Kawasaki 15750) rated to produce 1,000 kg/h of saturated steam at 0 bar(g). A steam quality of ($x = 0.95$) was assumed. The boiler operated continuously at the low-fire

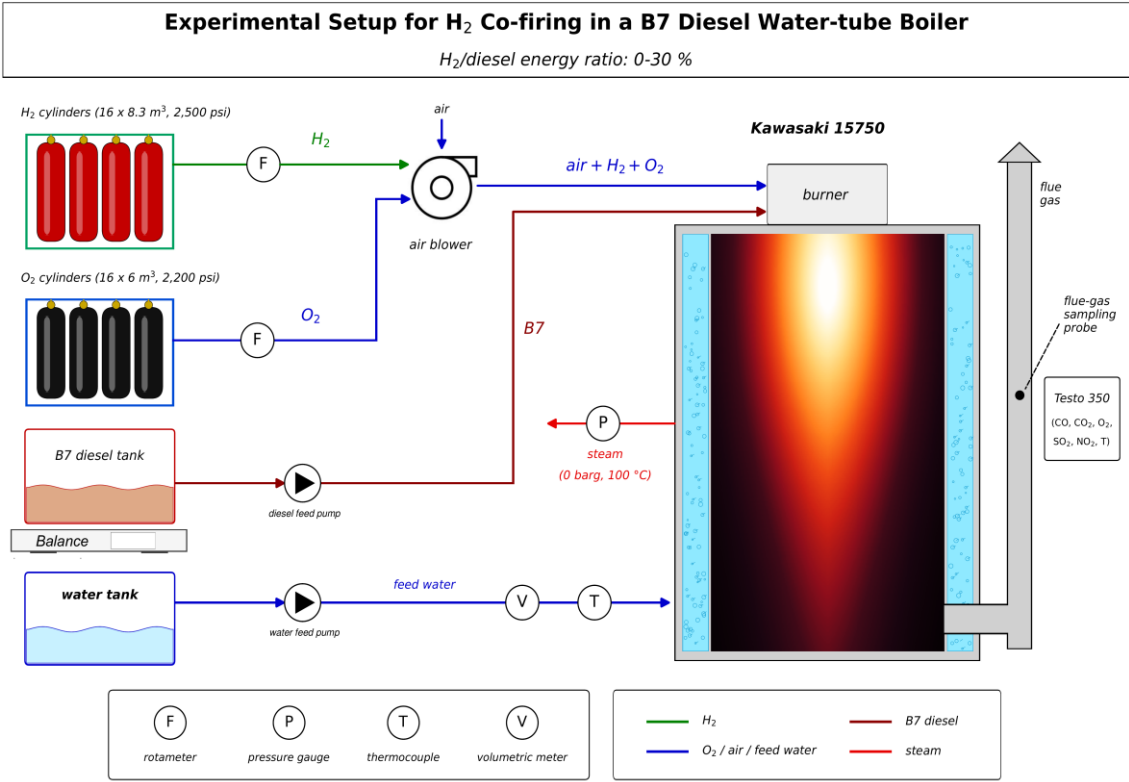


Fig. 1 Schematic of the experimental setup for H₂ co-firing in a B7 diesel water-tube boiler. B7 diesel, H₂, supplemental O₂, feedwater, steam, and flue-gas measurement locations are shown. F: flowmeter; P: pressure gauge; T: thermocouple; V: volumetric meter. The diagram is not to scale.

Table 2
Fuel properties of B7 diesel and hydrogen used in this study.

Property	B7 Diesel	Hydrogen	Unit
Lower Heating Value (LHV)	42.62	119.93	MJ/kg
Density (at 15 °C)	0.840	0.0838	kg/L / kg/m ³ (STP)
Molecular formula (surrogate)	C _{12.5} H _{22.1} O _{0.2} S _{0.002}	H ₂	—
Molecular weight (surrogate)	175.7	2.016	g/mol
Stoich. O ₂ requirement	3.27	7.94	kg O ₂ /kg fuel
Adiabatic flame temp. (λ = 1.0)	~2363	~2385	K
Laminar flame speed	~0.4	~2.9	m/s

setting of approximately 870 MJ/h without cycling or load modulation. The pressure-atomizing burner was fired with B7 diesel. At the same time, H₂ was injected upstream of the combustion-air inlet, as shown in Fig. 1.

Hydrogen and supplemental O₂, both with a purity of at least 99.5%, were supplied from high-pressure cylinders and adjusted to 5 bars. Supplemental O₂ was required because the burner operated with low excess air. Its flow was increased with the H₂ ratio to maintain the dry flue-gas O₂ concentration between 2.06% and 2.27%. The O₂ flow increased from 10 L/min at baseline to 150 L/min at 30% H₂, corresponding to 1.0–11.9% of the total oxygen supply. Because the H₂ ratio and O₂ supply increased together, their individual effects could not be completely separated.

The experiments were conducted on a commercial production boiler operating at an industrial plant in Thailand. This boiler configuration is commonly used in Thai industry (Department of Industrial Works, 2024). Therefore, the measurements can reflect the actual burner condition, fouling, and control behavior of an in-service boiler. However, the seven test conditions were established specifically for this study and did not represent the plant's normal operating schedule. The results and empirical correlations should not be applied directly to boilers with different burner designs, residence times, or heat-transfer geometries.

H₂ has a wide flammability range of 4–75 vol.% in air and a low minimum ignition energy of approximately 0.017 mJ. Thus, appropriate safety measures were required throughout the experiments. Certified flashback arresters were installed at the

H₂ and O₂ regulator outlets and injection-port inlets. The H₂ injection pressure was maintained below 0.5 bar(g). Two H₂-specific electrochemical detectors with audible alarms were used to monitor gas leakage, and continuous forced ventilation was provided. All hydrogen-handling procedures followed NFPA 2 and Thai DIW practices for compressed-gas operations.

B7 diesel consumption was measured gravimetrically using a calibrated platform scale with an accuracy of ± 0.001 kg. Steam production was estimated from the change in cumulative readings of a turbine-type feedwater meter over each measurement period. The meter accuracy was ± 0.5 kg/h. Hydrogen flow was measured using an H₂-calibrated rotameter with an accuracy of $\pm 2\%$ of full scale. The measured volumetric flow rate was converted to mass flow using the hydrogen density at the recorded ambient temperature and pressure. For each target H₂/diesel energy ratio, the rotameter set point was calculated from Eq. 2 using the measured diesel consumption and the lower heating values of both fuels. The uncertainty of the H₂ energy ratio was estimated to be approximately ± 3 percentage points. Supplemental O₂ flow was measured at the blower inlet using an air-calibrated rotameter with an accuracy of $\pm 2\%$ of full scale. The readings were corrected for the density difference between air and oxygen using a correction factor of 0.951.

The uncertainty of the H₂/diesel energy ratio, (*r*), was estimated by first-order error propagation. Considering the H₂ flow measurement ($\pm 2\%$) and the estimated uncertainty of the B7 lower heating value ($\pm 0.3\%$) as the main uncertainty sources, the relative uncertainty in (*r*) was approximately $\pm 2.1\%$. At (*r* = 30%), this value corresponds to approximately ± 0.6 percentage points. A wider operational tolerance of ± 3 percentage points was used to account for additional variation associated with manual adjustment of the hydrogen flow.

At the B7-only baseline, the Testo 350 indicated an average flue-gas H₂ concentration of approximately 26.75 ppm, although no hydrogen was supplied. The eight readings ranged from 11 to 44 ppm, with a coefficient variation of 48%. In contrast, the analyzer indicated zero H₂ at all conditions from (*r* = 20%) upward, despite hydrogen feed rates of 1.4–2.2 kg/h. These inconsistent readings indicated that the H₂ channel was unreliable under the test conditions. Therefore, the analyzer H₂ data were excluded from the quantitative analysis, and the H₂/diesel energy ratio was determined from the hydrogen flow measured by the calibrated rotameter.

2.2 Fuel Properties

The B7 diesel complied with the Thai diesel-fuel specification and consisted of 93 vol.% fossil diesel and 7 vol.% palm-oil FAME (Department of Energy Business, 2020). For the combustion model, B7 was represented by the surrogate formula C_{12.5}H_{22.1}O_{0.2}S_{0.002} with a molar mass of 175.7 g/mol. The B7 lower heating value used in the calculations was 42.62 MJ/kg, based on the 93:7 blend ratio. Converting the volume fractions to mass fractions using component densities gives 42.60 MJ/kg, a difference of only 0.04%. The hydrogen lower heating value was 119.93 MJ/kg (NASA, 1997). Table 2 summarizes the fuel properties used in this study.

2.3 Measurement System and Instrumentation

Flue-gas composition was monitored using a portable gas analyzer (Testo 350) installed at the stack. The analyzer recorded dry-basis O₂, CO, NO₂, CO₂, SO₂, residual H₂, and flue-gas temperature. Because the instrument did not measure NO directly, total NO_x could not be determined from the measured data. Total NO_x was estimated using two assumed NO₂/NO_x

fractions. The first assumption considered NO₂ to represent 5% of total NO_x, based on a general engineering approximation. Under this assumption, the estimated NO_x concentration was 20 times the measured NO₂ concentration. The second assumption used an NO₂/NO_x fraction of 1.2%, based on the average value reported by EPRI (2020) for oil-fired boilers. Under this assumption, the estimated NO_x concentration was approximately 83 times the measured NO₂ concentration. Thus, the 5% fraction provided the lower NO_x estimate. The 1.2% fraction provided the upper estimate. Neither fraction was measured for the boiler examined in this study. Therefore, the reported NO_x values should be considered semi-quantitative engineering estimates.

Previous studies have discussed the conversion between NO and NO₂ in combustion systems (Glarborg *et al.*, 2018; Pan *et al.*, 2023; Song *et al.*, 2019). However, these studies did not present an in-stack NO₂/NO_x ratio for a boiler comparable to the unit examined here. Applying the two assumed fractions to the measured NO₂ concentrations of 1.80–3.08 ppm gave estimated NO_x ranges of 36–62 ppm for (*f*=5%) and 150–256 ppm for (*f*=1.2%). Even though the selected fraction affects the estimated NO_x concentration, it does not affect the measured upward trend. The 70.8% increase from the baseline to the 30% H₂ condition is the same for both assumptions. Direct measurement of NO is required to determine the actual NO_x concentration. A kinetic calculation was also performed to estimate the NO₂/NO_x fraction. However, the predicted fraction was highly sensitive to the assumed gas-cooling history and could not be used to determine a unique conversion value. Therefore, a fraction of 5% was employed for the primary NO_x estimates, and 1.2% was used to calculate the conservative upper estimates.

Boiler efficiency was computed using the input-output method of ASME PTC 4.1 (Test Code for Steam Generating Units, Section 4), also known as the Direct Method, published by the American Society of Mechanical Engineers (ASME, 1964). Efficiency is the ratio of useful heat output to fuel heat input:

$$\eta = \frac{\dot{m}_{st}(h_f + x h_{fg} - h_{fw})}{\dot{m}_{fuel} LHV_{fuel}} \times 100\% \quad (1)$$

In Eq. 1, $\dot{m}_{st}$ is saturated steam mass flow (kg/hr), h_f is the saturated-liquid enthalpy (419 kJ/kg) and h_{fg} is the latent heat of vaporization (2,257 kJ/kg) and h_{fw} is the feedwater enthalpy (approximately 125.8 kJ/kg). The steam quality (*x*) was assumed to be 0.95 based on the range reported for saturated-steam boilers (Spirax Sarco, 2026). The effect of this assumption was included in the uncertainty analysis. Each condition was tested for 23.5–29.2 min, with seven or eight flue-gas readings recorded. After transient values were excluded, four readings were retained for calculating the mean, standard deviation, and 95% confidence interval. A sensitivity check using all reading produced similar trends. Boiler efficiency was calculated once for each condition; therefore, its reported uncertainty represents propagated instrument uncertainty rather than replicate variation. Steam production was estimated from the feedwater volume, with the blowdown valve closed and the boiler operated continuously. Changes in drum water inventory were not measured and remain a source of uncertainty in the calculated steam flow and efficiency.

Boiler efficiency was calculated using an input-output method adapted from the ASME PTC 4.1 Direct Method. The

Table 3

Test conditions and operating parameters. (*r*) is the H₂/diesel energy ratio defined in Eq. 2. The value (*n*=4) applies only to the flue-gas measurements. Efficiency uncertainty was calculated by error propagation (Table A1(b)). O₂ enrichment is the supplemental O₂ fraction of the total oxygen supply.

r (%)	Diesel Rate (kg/hr)	H₂ Flow (kg/hr)	O₂ assist (LPM)	O₂ enrichment (%)	Steam Rate (kg/hr)	Diesel Heat Input (MJ/hr)	System Eff. (%)	n (readings)
0	20.36	0.000	10	1.0	323.8	867.6	91.05 ± 0.44	4
5	20.30	0.361	20	1.9	348.0	865.3	93.42 ± 0.45	4
10	20.31	0.722	28	2.6	363.8	865.6	93.18 ± 0.46	4
15	20.52	1.094	100	8.8	368.6	874.8	89.47 ± 0.47	4
20	20.31	1.443	115	9.8	384.4	865.5	90.23 ± 0.50	4
25	20.25	1.798	142	11.7	397.1	862.9	89.68 ± 0.53	4
30	20.22	2.155	150	11.9	402.4	861.6	87.45 ± 0.55	4

useful steam-energy output was divided by the combined energy input from B7 diesel and H₂. The procedure differed from the standard code in three respects: lower heating values were used instead of higher heating values, minor heat credits were excluded, and the single-fuel method was extended to include two fuel streams. Therefore, the reported values should be described as efficiencies calculated using a method based on ASME PTC 4.1, rather than results from a test conducted fully in accordance with the code. The indirect efficiency reported by the Testo 350 was not used because the analyzer was configured for a single fuel.

The uncertainty in efficiency was estimated by first-order error propagation following ASME PTC 19.1 (ASME, 2018). The calculation included uncertainties in feedwater measurement, hydrogen flow, B7 lower heating value, and feedwater temperature. The combined instrument uncertainty ranged from ±0.4 to ±0.6 percentage points. Steam quality was treated separately because it was assumed rather than measured. Varying the steam quality from 0.92 to 0.98 changed the absolute efficiency by approximately ±2.5 percentage points but did not change the relative difference between the test conditions. The complete uncertainty analysis is provided in Table A1.

2.4 Experimental Conditions

Seven test conditions were evaluated at H₂/diesel energy ratios of 0%, 5%, 10%, 15%, 20%, 25%, and 30%. The H₂/diesel energy ratio, *r*, was calculated as follows:

$$r = \frac{\dot{m}_{H_2} LHV_{H_2}}{\dot{m}_{B7} LHV_{B7}} \quad (2)$$

Under this definition, an *r* value of 0.30 means the hydrogen energy input is 30% of the diesel energy input. Thus, hydrogen supplies 23.1% of the total fuel energy, calculated as $r/(1+r)$. The diesel feed rate remained almost constant at 20.22–20.52 kg/h (CV = 0.49%). Therefore, hydrogen was used as an additional fuel rather than used to replace the use of diesel. To avoid day-to-day variation, all seven conditions were tested on the same day (12 September 2025). The ambient temperature ranged from 31.7 to 34.0 °C. Table 3 presents the test conditions and measured values. The maximum ratio of 30% was selected according to the range of hydrogen addition reported in previous burner and boiler studies listed in Table 1. No flashback, ignition, or flame-instability events were observed

during the tests. Nevertheless, 30% represents only the highest ratio tested and should not be interpreted as the operating limit of the boiler. The results and correlations should not be extrapolated beyond the tested range.

The coefficients of variation indicate the short-term repeatability of the measurements within each test condition. However, they do not include the accuracy limits of the instruments. For NO₂, the accuracy of the NO₂ analyzer was ±5 ppm, but the measured mean concentrations were only 1.80–3.08 ppm. Therefore, the NO₂ results were used mainly to evaluate the trend rather than the exact concentration.

2.5 Semi-empirical Combustion Model

A lumped-parameter semi-empirical combustion model was developed to analyze the experimental results. The model consisted of three connected sub-models.

(a) *Adiabatic flame temperature (T_{ad})*. T_{ad} is computed from an energy balance of the combustion products at stoichiometric composition, using mean molar heat capacities evaluated at 1,200 K for CO₂ (54.3 J/mol/K), H₂O (39.3), O₂ (32.7), N₂ (31.1), and SO₂ (51.1). The excess-air ratio λ is back-calculated from the measured flue-gas O₂.

(b) *NO_x formation*. Thermal NO_x formation was calculated using the extended Zeldovich mechanism (Turns, 2012) and the calculated T_{ad}. An empirical OH-enhancement term was added to represent the effect of hydrogen addition :

$$NO_x = k_{scale} \exp\left(-\frac{38370}{T_{ad}}\right) [O_2]^{0.5} [N_2] \tau (1 + \beta r) \quad (3)$$

(*r* as the fraction of Eq. 2)

The calibration constant (*k_{scale}*) was fitted to the baseline condition (*r* = 0). The estimated reactor residence time (τ) was 0.5 s. The OH-enhancement coefficient (β = 1.74) was fitted to the seven experimental conditions by reducing the mean absolute percentage error between the calculated and measured NO_x values.

Equation 3 is a simplified form of the extended Zeldovich rate expression reported by Turns (2012). It retains the Arrhenius temperature term, but uses the oxygen dependence as [O₂]^{0.5} instead of the equilibrium oxygen-atom concentration. This simplification reduces the temperature sensitivity of the original rate expression. Therefore, *k_{scale}* and β can be considered as empirical calibration parameters rather than independent physical constants.

(c) *CO burnout and boiler efficiency.* CO burnout in the post-flame zone was modeled as a plug-flow reactor using the Dryer–Glassman rate expression (Dryer & Glassman, 1973). Boiler efficiency was calculated using the input-output method as given in Eq. 1. This calculation was conducted using an assumed steam quality of ($x=0.95$) and feedwater enthalpy calculated from the measured inlet temperature. k_{scale} and CO_{scale} coefficients were fitted to the baseline condition. The OH-enhancement coefficient (β) was varied from 0 to 4 in increments of 0.005, and the value producing the lowest mean absolute percentage error across the seven conditions was selected. Table A2(a) lists all model parameters and their sources, and Table A2(b) compares the calculated and measured values. Because the same experimental data were used for model calibration and comparison, the results indicate agreement between the calculated and measured values rather than independent model validation.

3. Results and Discussion

3.1 Combustion Emissions Overview

The experimental results are summarized in Table 4. The flue-gas measurements are reported as mean $\pm$ standard deviation from four retained readings. System efficiency is repeated from Table 3 and represents one determination for each condition. Its reported uncertainty is the propagated instrument uncertainty as given in Table A1(b), not the standard deviation of repeated measurements.

As shown in Table 4, SO_2 remained close to zero for all tested conditions. Small values were recorded at H_2 /diesel energy ratios of 5% and 10%, but they were within the variation of the measurements and did not indicate a clear pattern. Therefore, the SO_2 results should be considered qualitative and should not be used for compliance assessment. The sulfur content assigned to the B7 surrogate in Table 2 was higher than the limit specified for commercial B7 diesel. This surrogate was used only in the combustion model and was not used to estimate SO_2 emissions.

3.2 Steam Production Capacity

As shown in Fig. 2, steam output increased with hydrogen addition. At an H_2 /diesel energy ratio of 30%, steam output increased by 24.3%, from 323.8 to 402.4 kg/h. The diesel flow remained almost constant at around 20.22–20.52 kg/h (CV = 0.49%). Therefore, the increase in steam output clearly resulted

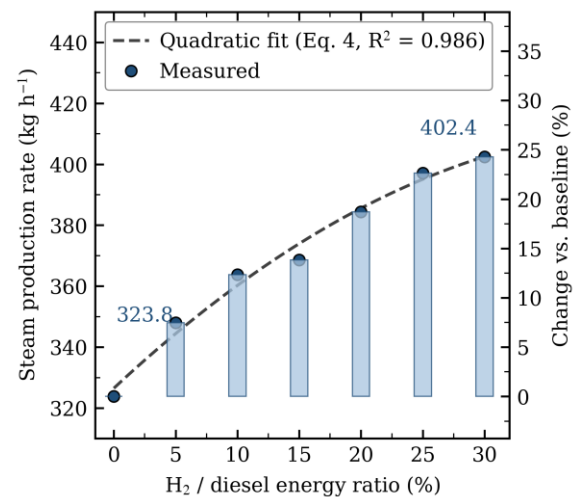


Fig. 2. Steam output and percentage change from the baseline at each H_2 /diesel energy ratio. Each point represents a single feedwater-totalizer measurement; the measurement uncertainty is reported in Table A1(b).

from the additional energy given by hydrogen and it should not be interpreted as an improvement in boiler efficiency.

Steam output ($\dot{m}_{steam}$, kg/hr) increased at each tested H_2 ratio. The relationship was fitted using a quadratic (Eq. 4, $R^2 = 0.986$), which is valid only within the examined range of $0 \leq r \leq 30$, where r is expressed as a percentage.

$$\dot{m}_{steam} = -0.042r^2 + 3.793r + 326.5 \quad (4)$$

The negative coefficient in Eq. 4 indicates that the increasing rate of steam output became slightly lower at higher H_2 ratios. Nevertheless, the cause of this change could not be determined from the present measurements.

3.3 Effect on CO Emissions

As exhibited in Fig. 3, CO decreased continuously from 107.0 ± 7.7 ppm at 0% H_2 to 28.5 ± 1.7 ppm at 30% H_2 , corresponding to a reduction of 73.4%. This reduction may result from the higher flame temperature and greater availability of OH radicals, which promote CO oxidation through the reaction $CO + OH \rightarrow CO_2 + H$. The Dryer–Glassman model (Dryer & Glassman, 1973) reproduced the decreasing trend but slightly overestimated the measured CO concentrations.

Table 4

Experimental results at each H_2 /diesel energy ratio. Flue-gas values are reported as mean $\pm$ SD from four retained readings. NO_2 was measured directly, and the NO_x range was estimated using NO_2/NO_x fractions of 5% for the lower estimate and 1.2% for the upper estimate. The correlations in Eqs. 4–7 were fitted using the unrounded condition means

r (%)	CO (ppm)	NO_2 measured (ppm) [†]	NO_x estimated (ppm)	CO_2 (%) [‡]	SO_2 (ppm)	T _{flue} (°C)	System Eff. (%)
0	107.0 ± 7.7	1.80 ± 0.14	36–150	12.87 ± 0.06	0.00 ± 0.00	193.2 ± 0.3	91.05 ± 0.44
5	70.3 ± 6.7	2.30 ± 0.08	46–192	13.34 ± 0.06	0.50 ± 0.58	194.1 ± 0.3	93.42 ± 0.45
10	58.3 ± 6.6	2.50 ± 0.00	50–208	13.52 ± 0.02	0.25 ± 0.50	193.5 ± 0.1	93.18 ± 0.46
15	46.8 ± 4.2	2.60 ± 0.08	52–217	13.70 ± 0.03	0.00 ± 0.00	194.0 ± 0.1	89.47 ± 0.47
20	41.8 ± 3.3	2.73 ± 0.05	55–227	13.79 ± 0.05	0.00 ± 0.00	194.2 ± 0.4	90.23 ± 0.50
25	38.0 ± 2.2	2.88 ± 0.05	58–240	13.96 ± 0.03	0.00 ± 0.00	196.5 ± 0.1	89.68 ± 0.53
30	28.5 ± 1.7	3.08 ± 0.05	62–256	14.07 ± 0.03	0.00 ± 0.00	197.2 ± 0.0	87.45 ± 0.55

Note: Flue-gas values are reported as mean $\pm$ SD from four retained readings. NO_2 was measured directly using the Testo 350. NO_x was estimated using assumed NO_2/NO_x fractions of 5% for the lower estimate and 1.2% for the upper estimate. The uncertainty reported for system efficiency was calculated by error propagation and is not a standard deviation

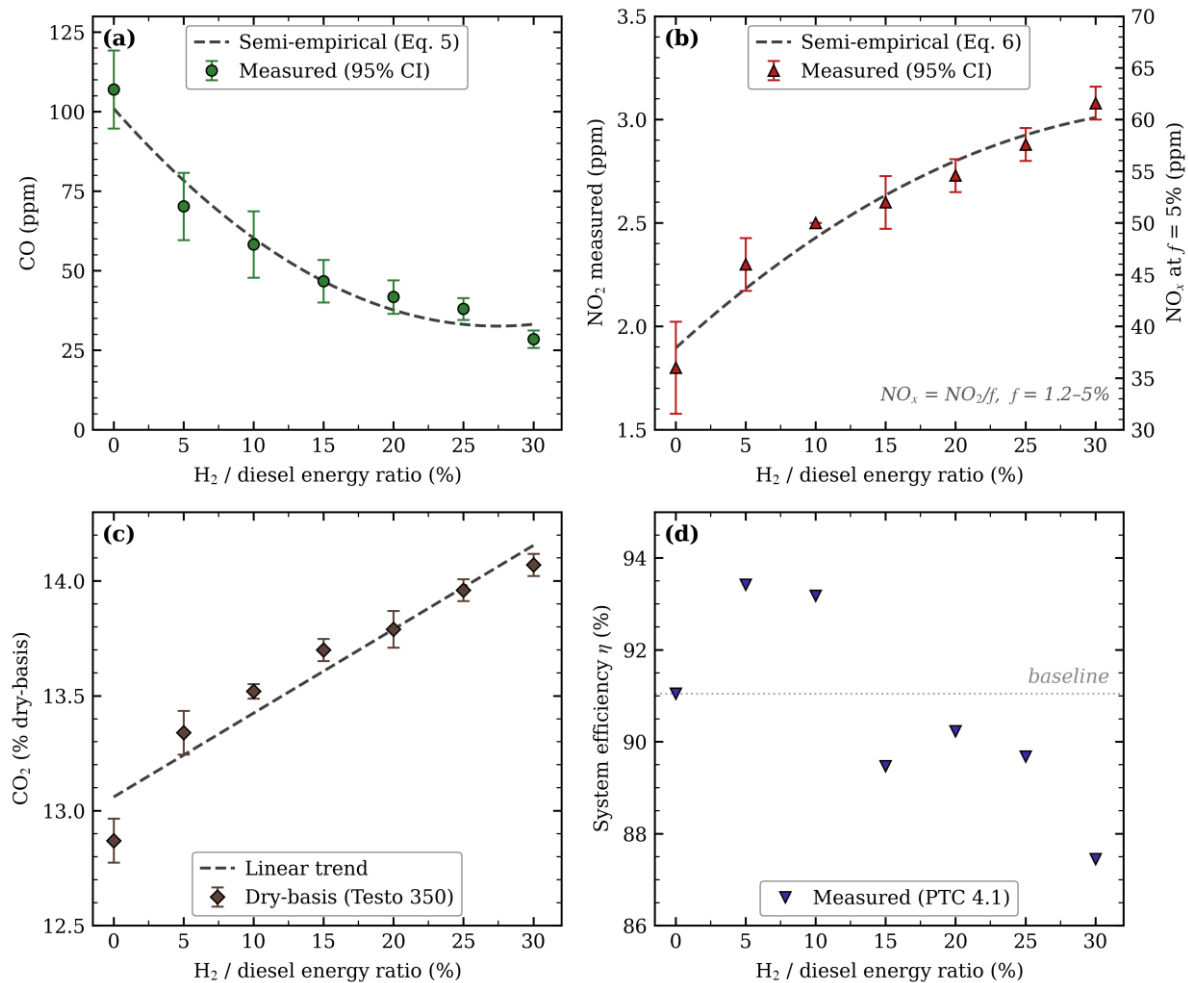


Fig. 3. Combustion parameters as a function of H₂/diesel energy ratio: (a) CO, (b) measured NO₂ with NO_x at f = 5% on the right-hand scale, (c) CO₂ dry-basis, (d) system efficiency (ASME PTC 4.1 Direct Method)

Because the model was calibrated using experimental data, this agreement does not represent independent validation.

Chemical-kinetic calculations were performed in Cantera (Goodwin *et al.*, 2025) using the CRECK C1–C16 high-temperature mechanism with NO_x chemistry (Ranzi *et al.*, 2012; Ranzi *et al.*, 2014; Song *et al.*, 2019). The equivalence ratio for each condition was determined from the measured flue-gas oxygen concentration. Two methods were used to estimate OH availability. The equilibrium OH concentration calculated using the CRECK mechanism increased by 22.9% from the baseline to 30% H₂. A one-dimensional flame calculation using GRI-Mech 3.0 (Smith *et al.*, 1999) showed an increase of 16.9% over the same range. GRI-Mech was used because the available CRECK mechanism did not include the transport data required for the flame calculation. However, the GRI-Mech calculation used methane and was included only to confirm the direction of the change. Therefore, the discussion is based mainly on the CRECK equilibrium result. The calculated increase in OH supports the proposed mechanism of faster CO oxidation through $CO + OH \rightarrow CO_2 + H$. CO (ppm) decreased by 78.5 ppm across the tested range, from 107.0 to 28.5 ppm. The data were fitted using a quadratic regression (Eq. 5, $R^2 = 0.960$). The equation is valid only within the tested range of $0 \leq r \leq 30$, where r is expressed as a percentage. At $r = 0$, the equation gives 101.0 ppm, which is 5.6% lower than the measured baseline of 107.0 ppm.

$$CO = 0.0907r^2 - 4.982r + 101.0 \quad (5)$$

The decrease in CO agrees with previous studies of hydrogen-enriched combustion. Lan *et al.* (2024) reported a CO reduction of up to 70% in a natural-gas industrial furnace at 60 vol.% hydrogen. This value is close to the 73.4% reduction observed in the present study. However, direct comparison is limited because the studies used different fuels and definitions of hydrogen ratio. Glanville *et al.* (2022) also observed decreasing CO emissions in partially premixed combustion equipment using H₂/natural-gas blends.

3.4 Effect on NO_x Emissions

Estimated NO_x rose with the H₂/diesel energy ratio, as shown in Fig. 3. Using an assumed NO₂/NO_x fraction of 5%, estimated NO_x increased from 36.0 ± 2.8 ppm at the baseline to 61.5 ± 1.0 ppm at 30% H₂. This means an increase of 70.8%, as exhibited in Fig. 3(b). The coefficient of variation decreased from 7.9% to 1.6%, implying less variation among the retained readings at the higher H₂ ratio. Nevertheless, this result does not confirm an improvement in flame stability. Clearly, the increase in NO_x is consistent with previous studies of hydrogen-enriched hydrocarbon combustion. Technically, NO_x can be formed by numerous mechanisms. For example, the Zeldovich thermal mechanism, the rapid mechanism suggested by Fenimore

(1971) and reactions with OH atoms. Nonetheless, the flue-gas data were not adequate to indicate the contribution of each mechanism. The addition of the fitted OH-enhancement coefficient ($\beta = 1.74$) reduced the mean absolute percentage error from 21.1% to 4.0%. However, β was also obtained from the same experimental data. Therefore, the model cannot show which NO_x formation mechanism was most important.

Estimated NO_x increased almost linearly with the H_2 ratio (Eq. 6), although the Zeldovich mechanism has an exponential dependence on temperature. This behavior can be explained by the small change in the calculated adiabatic flame temperature, which increased by only 14 K across the tested range. Over such a narrow temperature range, the response can appear nearly linear. Equation 3 also includes an OH-enhancement term that changes with the H_2 ratio. This term is an empirical correction rather than a separate NO_x formation mechanism.

Under the adiabatic assumption, the rate-of-production analysis showed that approximately 97–98% of NO formation occurred through the thermal Zeldovich mechanism. The N_2O -intermediate mechanism contributed about 3%, and the rapid mechanism contributed less than 0.01%. However, these values changed significantly when a different temperature history was assumed. Therefore, the results indicate only the potential importance of each mechanism and should not be considered as actual contributions.

The oxygen supply increased together with the H_2 ratio (Table 3), so their effects could not be completely separated. However, the changes in NO_x were not related to the change in oxygen enrichment. The largest NO_x increase (27.8%) occurred between 0% and 5% H_2 , whereas oxygen enrichment increased by only 0.9 percentage points. In contrast, the largest increase in oxygen enrichment produced the smallest increase in NO_x (4.0%).

Multivariable regression produced $\text{NO}_x = 40.20 + 1.008r - 0.624e$ ($R^2 = 0.938$), where r is the H_2 /diesel energy ratio (%) and e is the oxygen enrichment (%). The H_2 ratio had a statistically significant effect ($p = 0.037$), but the effect of oxygen enrichment was not. The H_2 ratio also explained more variation in NO_x than oxygen enrichment separately ($R^2 = 0.927$ and 0.792 , respectively). However, the two variables were strongly correlated ($r = 0.956$; $\text{VIF} = 11.7$), so their individual effects could not be determined precisely. The CO analysis also indicated no evident effect of oxygen enrichment. These results suggest that hydrogen was the main factor in the emission changes, although the effects of hydrogen and oxygen could not be fully separated. A separate control test with a fixed H_2 ratio and different oxygen flow rates is then required to determine their individual effects. The NO_x data were fitted using quadratic regression (Eq. 6, $R^2 = 0.960$). The equation is valid only within the tested range of $0 \leq r \leq 30$, where r is expressed as a percentage:

$$\text{NO}_x = -0.0162r^2 + 1.229r + 37.90 \text{ (ppm)} \quad (6)$$

The increase in NO_x agrees with previous studies of hydrogen-enriched combustion. Nemitallah *et al.* (2024) reported higher NO_x emissions after adding hydrogen to a natural-gas-fired water-tube boiler. However, their NO_x values were much higher and did not increase consistently with the H_2 ratio. Wang and Yang (2024) also reported that hydrogen can increase thermal NO_x formation through the Zeldovich mechanism. Nguyen *et al.* (2026) observed lower CO and higher NO_x in hydrogen-assisted diesel engines. These studies obviously indicate that adding hydrogen can reduce CO emissions but increase NO_x emissions.

Thai emission standards report NO_x as NO_2 on a dry basis at 25 °C and 7% O_2 in the flue gas (Ministry of Industry, 2006, 2026). The measured flue-gas O_2 concentration was 2.06–

2.27%. Therefore, a correction factor of 0.74 was applied. After using an NO_2/NO_x fraction of 5%, NO_x increased from 36 to 61 ppm before correction and from 27 to 46 ppm after correction. After using the more conservative fraction of 1.2%, the corresponding values were 150–256 ppm and 111–191 ppm. In fact, the standard correction assumes that excess oxygen is supplied by air. In this experiment, some oxygen was supplied as pure O_2 . Therefore, the corrected values may differ slightly from the actual values.

The corrected NO_x values remained below the nationwide limit of 200 ppm for liquid-fuel-fired heat sources (Ministry of Industry, 2006). For the Bangkok limit of 160 ppm, the results complied when the 5% conversion fraction was used. However, the values calculated using the 1.2% fraction exceeded this limit at the higher H_2 ratios. Although the test site is outside Bangkok, direct NO measurement would be needed to confirm compliance at high H_2 ratios.

Corrected CO decreased from 79 to 21 ppm, which was below the limit of 690 ppm. SO_2 remained close to the detection limit. Because the measured NO_2 concentrations were below the stated accuracy of the analyzer, the absolute NO_x values should be treated as semi-quantitative estimates. Fig. 4 shows the overall emission trend: CO decreased as the H_2 ratio increased, but NO_x increased

3.5 CO_2 Concentration and Dry-basis Measurement Effect

The measured dry-basis CO_2 concentration increased from $12.87 \pm 0.06\%$ at 0% H_2 to $14.07 \pm 0.03\%$ at 30% H_2 (Fig. 3). In contrast, the calculated dry-basis CO_2 concentration decreased from 14.22% to 13.23%, as the H_2 ratio increased. This decrease was expected because hydrogen contains no carbon. The analyzer reading was 1.35 percentage points lower than the calculated value at the baseline and 0.84 percentage points higher at 30% H_2 . The measured value was lower than the calculated value at the baseline but higher at 30% H_2 . Therefore, the difference was not consistent across the tested conditions.

The cause of this difference could not be identified from the present experiment. Therefore, the measured CO_2 concentrations were not used in the quantitative analysis. CO_2 intensity was then calculated from the measured fuel consumption and the carbon content of the representative B7 fuel. Dry flue-gas flow was obtained from the carbon and oxygen balance. Future work should check the CO_2 and O_2 channels using certified calibration gas

3.6 Boiler Thermal Efficiency and Flue-gas Temperature

As shown in Fig. 3, system efficiency increased from 91.05% at the baseline to 93.42% at 5% H_2 and then decreased to 87.45% at 30% H_2 . The obvious decrease from the baseline to 30% H_2 was 3.60 percentage points. The propagated instrument uncertainty ranged from ± 0.44 to ± 0.55 percentage points (Table A1(b)). However, this uncertainty did not include changes in drum water inventory.

A change of only 7.2 kg in drum water inventory during the experiment could account for the entire 3.60-percentage-point decrease. The incremental efficiencies at 5% and 10% H_2 also exceeded 100%, indicating that the calculated efficiency was affected by the steam-flow measurement. Therefore, the detailed changes at 5%, 15%, 20%, and 25% H_2 were not evaluated as physical effects. The 3.60-percentage-point decrease should be regarded as the maximum observed penalty rather than an exact loss of efficiency. More reliable assessment requires direct steam-flow measurement and drum-level readings at the beginning and end of each test.

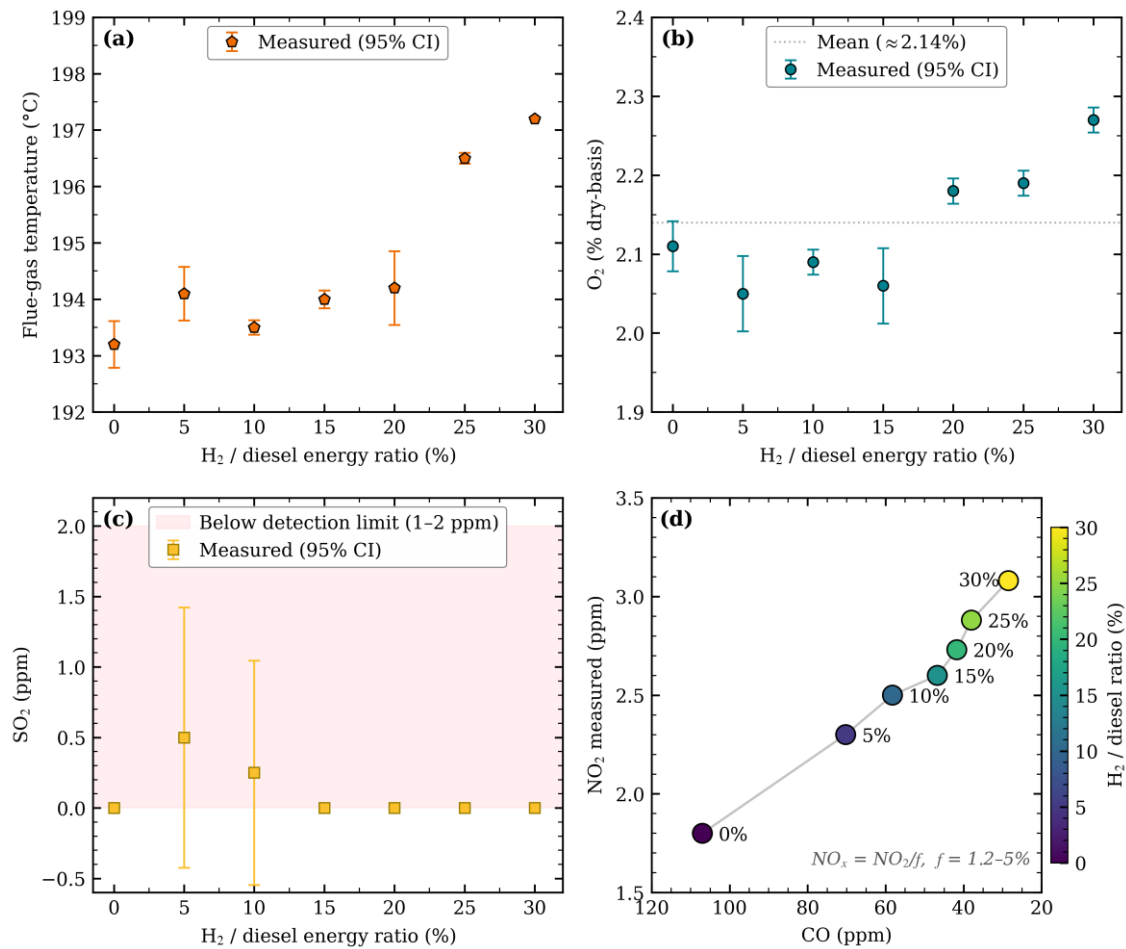


Fig. 4. Additional combustion and emission results: (a) flue-gas temperature, (b) dry-basis O₂ concentration, (c) SO₂ concentration, and (d) relationship between CO and NO₂. Error bars show 95% confidence intervals. The shaded area in panel (c) indicates the SO₂ detection range of 0–2 ppm

Fig. 4 presents the flue-gas temperature, O₂ concentration, SO₂ concentration, and the relationship between CO and NO₂. As shown in Fig. 4(a), the flue-gas temperature increased by only 4.0 °C, from 193.2 ± 0.3 °C at the baseline to 197.2 ± 0.0 °C at 30% H₂. This small increase was consistent with the calculated rise in adiabatic flame temperature from 2,202 to 2,216 K.

3.7 Normalized Emission Trends and Regression Models

Fig. 5 compares the percentage change in each parameter relative to the B7-only baseline. CO decreased by approximately 2.6 ppm for each percentage-point increase in the H₂/diesel energy ratio. For $f = 5\%$, the estimated NO_x increased by approximately 0.9 ppm per percentage point. Likewise, when $f = 1.2\%$ was used, it increased by 3.5 pp. Therefore, the relative changes in CO and NO_x depend on the assumed NO₂/NO_x fraction. These pollutants also have different emission limits, so their changes should not be compared directly in terms of environmental impact.

Steam output increased as CO₂ intensity decreased. These opposite trends resulted mainly from the almost constant diesel flow and the increase in steam production. System efficiency followed the non-monotonic pattern as discussed in Section 3.6. Flue-gas temperature changed only slightly across the tested H₂ ratios.

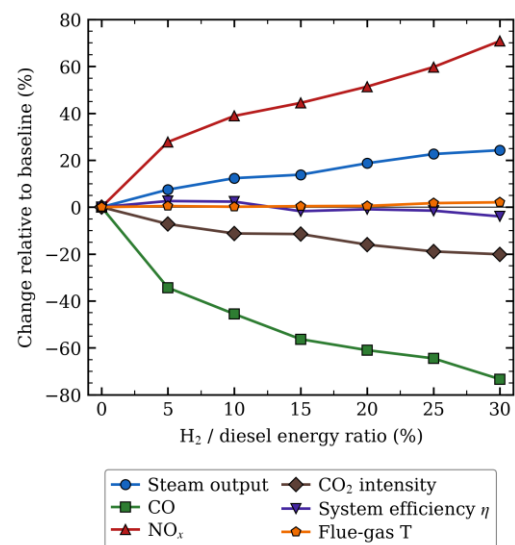


Fig. 5. Percentage changes in steam output, CO, estimated NO_x, CO₂ intensity, system efficiency, and flue-gas temperature relative to the B7-only baseline.

Fig. 6 shows the quadratic regression fits for CO, measured NO₂, CO₂ intensity, and steam output, which agree

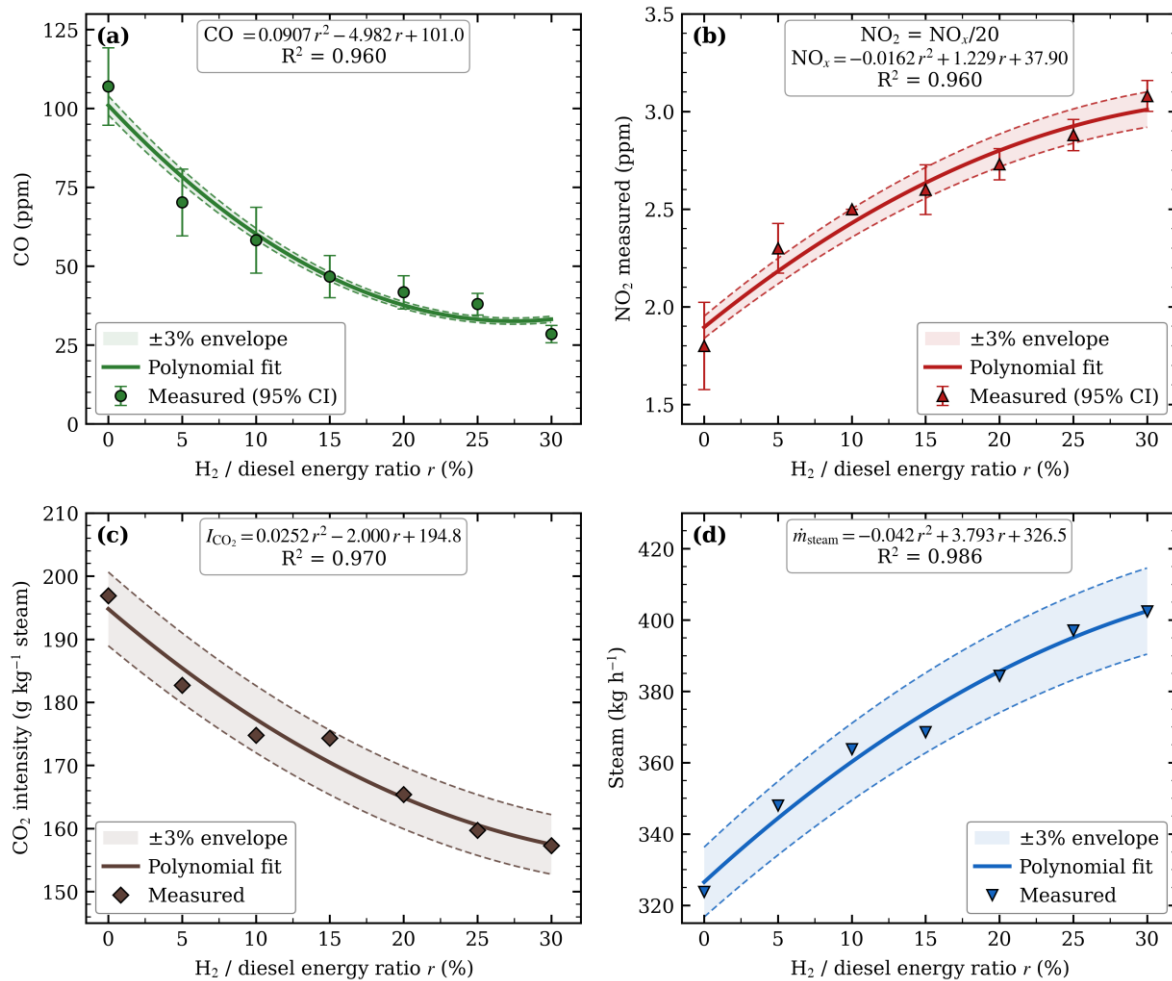


Fig. 6. Quadratic regression fits for (a) CO, (b) measured NO_x, (c) CO₂ intensity, and (d) steam output. Shaded areas show $\pm 3\%$ around each fit and are not prediction intervals. Cross-validation results are given in Table A3.

with the experimental data ($R^2 \geq 0.960$). Equations 4–7 can be used to estimate steam output, CO, NO_x, and CO₂ intensity within the tested range of $0 \leq r \leq 30$. The correlation for CO₂ intensity is:

$$CO_2 \text{ intensity} = 0.0252r^2 - 2.000r + 194.8 \quad (7)$$

where CO₂ intensity is expressed in g CO₂/kg steam and r is expressed as a percentage.

The regressions were fitted and evaluated using the same seven experimental conditions. Therefore, their R^2 values suggest that they are in good agreement with the current data rather than predictive accuracy. Leave-one-out cross-validation results are presented in Table A3. The errors were larger near the ends of the tested range, particularly for CO, which had an RMSECV of 13.3 ppm. Therefore, these correlations should be utilized only as screening tools within the tested range and should not be extended beyond that scope.

The quadratic regressions were also compared with linear models using the small-sample corrected Akaike information criterion (AICc). As shown in Table A3, the linear models were preferred for all four variables. However, this result was strongly affected by the small number of data points because AICc penalizes the additional quadratic term. For NO_x, the linear model also gave better cross-validation performance than the quadratic model ($R^2_{CV} = 0.808$ and 0.720 , respectively).

The quadratic equations were retained to maintain the same form for all four correlations. Within the tested range, the linear and quadratic NO_x estimates differed by no more than 2.0 ppm. Therefore, the simpler linear equation, $NO_x = 0.743r + 39.9$, may be used when a quadratic form is not required.

3.8 Environmental Performance Assessment

As shown in Fig. 7(a), CO₂ intensity decreased from 196.9 to 157.3 g/kg of steam, corresponding to a reduction of 20.1%. Because diesel consumption remained almost constant, this reduction mainly resulted from the increase in steam output. When normalized by total fuel heat input (Fig. 7(b)), the CO emission index decreased from 35.19 to 7.75 g/GJ, reflecting a decrease of 78.0%. In contrast, the estimated NO_x index increased from 19.45 to 27.47 g/GJ, representing a 41.3% increase. The dry flue-gas flow used in these calculations was obtained from the carbon and oxygen balance rather than the measured CO₂ concentration, as explained in Section 3.5. The increase in the NO_x index was smaller than the 70.8% increase in concentration because total fuel heat input increased by 29%.

3.9 Preliminary fuel-cost-only sensitivity analysis (excluding H₂ compression, storage, supplemental O₂, safety, and capital costs)

This analysis considers only the purchase costs of B7 diesel and hydrogen. It excludes hydrogen compression,

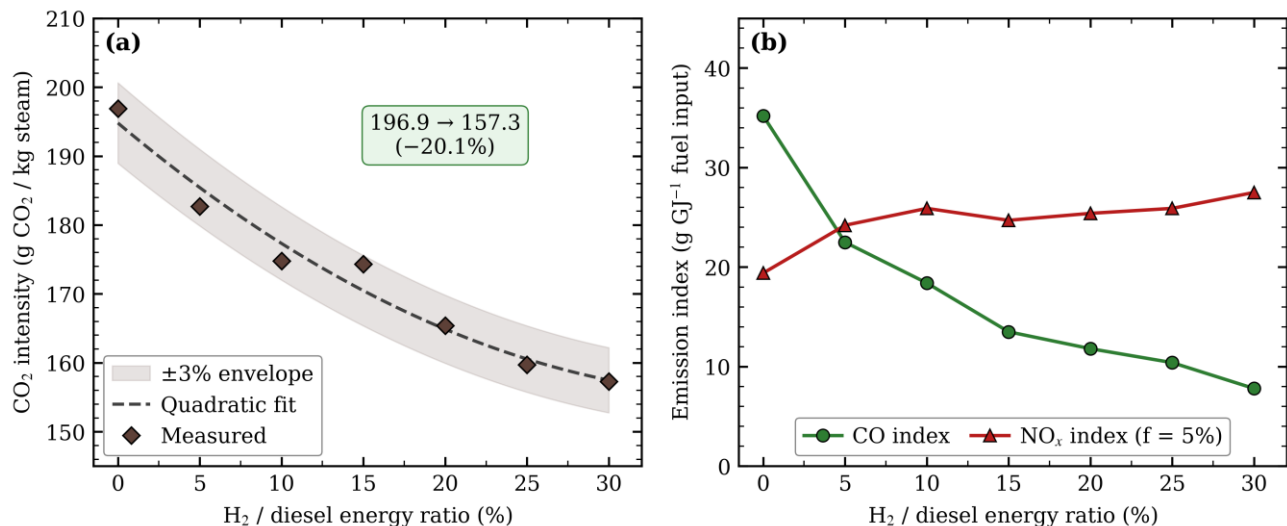


Fig. 7. Effect of the H₂/diesel energy ratio on (a) CO₂ intensity per kilogram of steam and (b) CO and estimated NO_x emission indices per unit of fuel heat input. The NO_x index was calculated using $f = 5\%$; using $f = 1.2\%$ gives values 4.17 times higher.

storage, supplemental oxygen, safety upgrades, maintenance, or capital costs. Hydrogen was supplied in packs of sixteen 50-L cylinders charged to 166 bars. Each pack cost 8,525.76 THB, including VAT. Based on the nominal gas content, the hydrogen price was approximately 753 THB/kg (23 USD/kg). When gas compressibility at 166 bars was considered, the estimated price increased to approximately 847 THB/kg. The lower value of 753 THB/kg was used in this analysis. The average price of industrial B7 diesel was 43.7 THB/L, corresponding to 52.0 THB/kg (1.59 USD/kg). At this price, the baseline fuel cost was 3.27 THB per kilogram of steam. All calculations were performed in Thai baht. USD values were converted to 32.7 THB/USD and are provided only for comparison.

Figure 8(a) shows the break-even H₂ price at different H₂/diesel energy ratios and B7 prices. At the B7 price used in this study (52.0 THB/kg), the break-even H₂ price was approximately 228 THB/kg at 5% H₂ and 185 THB/kg at 10% H₂. At higher H₂ ratios, it ranged from 120 to 140 THB/kg (3.7–4.3 USD/kg). The hydrogen price paid in this study was approximately 753 THB/kg, which was three to six times higher than the break-even prices. Therefore, hydrogen co-firing was

not cost-competitive at any tested ratio. Compared with diesel-only operation, the fuel cost per kilogram of steam increased by 16.7% at 5% H₂ and 103.3% at 30% H₂. Figure 8(b) shows the fuel-cost difference over a range of H₂ prices at the B7 price used in this study. The black line represents the break-even condition, and the dotted lines indicate the hydrogen price used in the experiment and the three future price scenarios. IRENA (2020) predicted that green hydrogen could reach approximately 1.5–2.5 USD/kg by 2030, depending mainly on electricity cost and location. At those projected prices, hydrogen co-firing at moderate ratios could approach fuel-cost parity with diesel-only operation. Carbon pricing could further improve its economic position, although carbon costs were not included in this analysis.

The price scenarios in Table A4 show that no tested H₂ ratio was cost-competitive at the cylinder price of 753 THB/kg. At 120 THB/kg (3.7 USD/kg), all tested ratios above 0% became cost-competitive, with larger savings at lower future prices. Assuming 4,000 operating hours per year, a ten-year period, and an interest rate of 8%, the fuel-cost savings at 10% H₂ could support approximately 1.25 million THB of capital

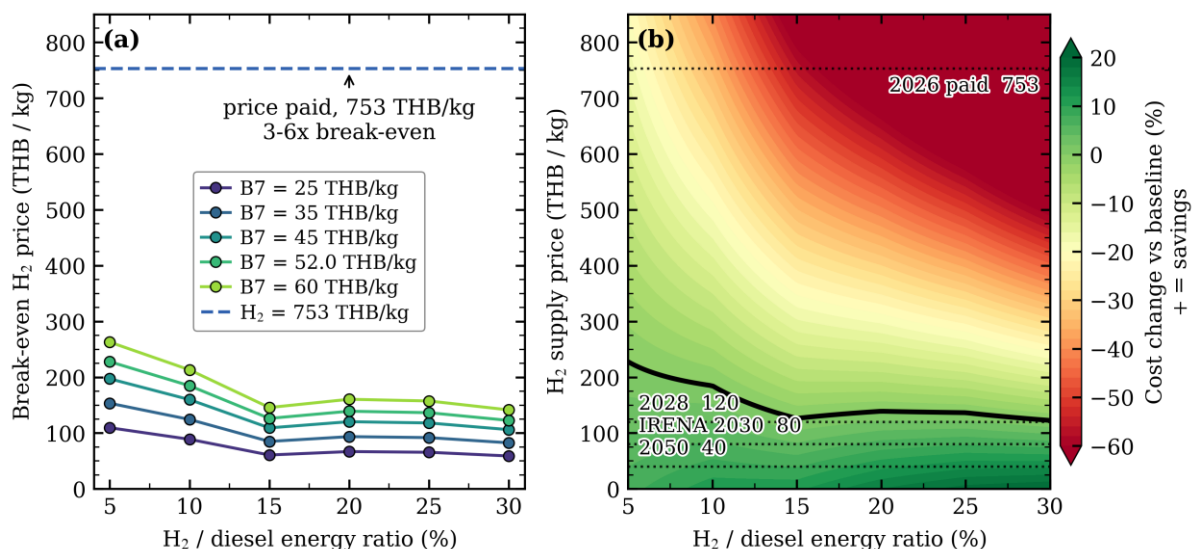


Fig. 8. Fuel-cost sensitivity analysis: (a) break-even H₂ price at different B7 prices and (b) fuel-cost savings at a B7 price of 52.0 THB/kg. The black line indicates the break-even condition

investment at an H₂ price of 120 THB/kg. At 80 THB/kg, the supported investment increased to approximately 2.0 million THB at 10% H₂ and 2.5 million THB at 30% H₂. These values are gross estimates and do not include additional maintenance costs.

The economic viability of co-firing therefore depends primarily on the price of hydrogen. The estimated break-even range was around 120–230 THB/kg (3.7–7.0 USD/kg), which was significantly lower than the cylinder price used in this study. No burner replacement was required within the tested range, but bulk or on-site hydrogen supply would likely be needed to achieve competitive fuel costs. Until then, the main reasons for using hydrogen are reduced CO and CO₂ intensity rather than fuel-cost savings. However, increased NO_x remains an important drawback. Fig. 9 compares the overall boiler performance at H₂/diesel energy ratios of 0%, 10%, 20%, and 30%, showing the gains in steam productivity, CO reduction, and CO₂-intensity reduction against the rise in NO_x on a normalized scale.

3.10 Operational Safety Observations

Hydrogen safety controls were carefully maintained throughout the experiment. These included certified flashback arresters, an injection pressure below 0.5 bar(g), two hydrogen leak detectors with audible alarms, and continuous forced ventilation. Hydrogen handling strictly followed NFPA 2 and Thai Department of Industrial Works practices.

No flashbacks, external ignition, alarm activation, or unplanned shutdown occurred during the seven test conditions. The flame remained stable at H₂/diesel energy ratios up to 30% with supplemental oxygen. The leak detectors provided alarms but did not record hydrogen concentration or leak rate. Therefore, the results confirm only that no safety incident occurred under the tested conditions. Industrial application should include continuous hydrogen monitoring with data logging, automatic fuel shutoff, and a formal hazard assessment.

3.11 Comparative Analysis with Published Work

The hydrogen co-firing studies listed in Table 1 differ in fuel type, burner design, combustion conditions, unit size, and the definition of hydrogen fraction. Premasathit *et al.* (2025) examined diesel–hydrogen combustion at a hydrogen energy fraction of 23%, which is close to the 23.1% used at $r = 0.30$ in the present study. Their model predicted higher flame temperature and lower CO₂ after hydrogen addition. These trends agree qualitatively with the calculated adiabatic flame temperature and fuel-based CO₂ intensity reported here. However, their study used CFD and experimentally validated only the pure-diesel condition. They also varied the equivalence ratio from 0.65 to 1.10, but the current study maintained the flue-gas O₂ concentration within a narrow range. Therefore, the two studies can be compared only qualitatively. Their simulated flame temperature was higher than the measured value because of heat loss to the chamber walls, indicating that adiabatic calculations should not be utilized to predict absolute emission levels.

Jung *et al.* (2024) simulated hydrogen blending in a natural-gas-fired melting furnace and reported a temperature increase of 32 K at 30 vol.% H₂. A larger increase occurred only at higher H₂ fractions. The present model predicted a smaller increase of 14 K across the tested range. Although the fuels and definitions of H₂ ratio differed, both studies showed only a moderate temperature change at low-to-moderate H₂ ratios. This may partially account for the almost linear increase in NO_x that was observed in this study.

Studies of gas-fired boilers also reported lower CO and higher NO_x after hydrogen addition (Lan *et al.*, 2024; Nemitallah *et al.*, 2024; Sun *et al.*, 2024; Wang *et al.*, 2022). Nevertheless, their numerical results cannot be compared directly with the present study. In general, natural-gas boilers have much lower baseline CO concentrations than the 107.0 ppm measured here because gas and liquid fuels undergo different mixing and combustion processes. The definitions of hydrogen fraction also

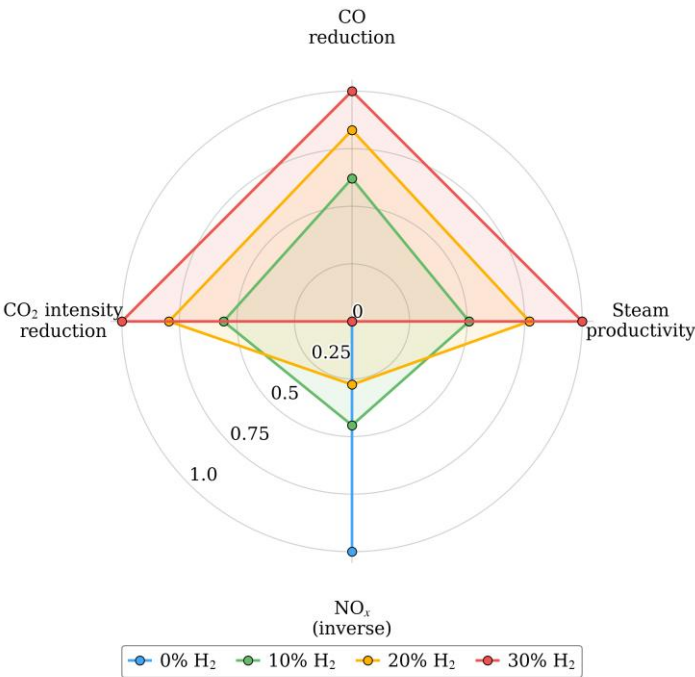


Fig. 9. Normalized comparison of steam productivity, CO reduction, CO₂-intensity reduction, and inverse NO_x at H₂/diesel energy ratios of 0%, 10%, 20%, and 30%. Each metric was scaled from 0 to 1, with 1 representing the most favorable result. System efficiency was excluded because of the measurement limitations discussed in Section 3.6

differ. A blend containing 30 vol.% H₂ in natural gas provides only about 10% of the total fuel energy, compared with 23.1% in the present study.

Dual-fuel engine studies also reported lower CO and higher NO_x after hydrogen addition (Hosseini *et al.*, 2023; Jayabal & Sivanraju, 2025; Periyathambi *et al.*, 2025). Hosseini *et al.* (2023) found that the suitable hydrogen ratio depends on the engine, fuel, and operating conditions. This study indicates 30% H₂ as the maximum ratio examined, rather than suggesting it as an optimal operating condition. However, compression-ignition engines operate at much higher temperatures and pressures than boilers. Kannan *et al.* (2021) also showed that various modeling techniques can produce large differences in predicted ignition delay. Therefore, comparisons between engine and boiler studies should be considered qualitative.

Recent studies have reported different results. Heeley *et al.* (2026) tested natural gas–hydrogen mixtures from 0% to 100% in a 300-kW pilot furnace operated at 75 kW. Radiant heat transfer was maintained. Jankowski *et al.* (2025) reported different results for two domestic condensing boilers rated at 21.9 and 25.7 kW. Adding up to 50 vol.% H₂ increased efficiency by 0.4–1.6 percentage points and reduced both NO_x and CO. The authors attributed these results to the low operating temperature of condensing boilers, with flame temperatures below 1,300 °C. This differs from the higher-temperature conditions of the present boiler and shows that the NO_x response depends on the combustion temperature, not only the H₂ ratio. Direct comparison is also limited because their emissions were corrected to 0% O₂ and NO was measured without NO₂.

3.12 Study Limitations

All seven conditions were tested at a single low-fire setting of approximately 870 MJ/h. The results may differ at medium or full load. The boiler also operated at 0 bar(g), whereas industrial boilers commonly operate at 5–15 bar(g). Therefore, the results should be verified at higher loads and pressures.

The findings are specific to the Kawasaki 15750 boiler and may not apply directly to other boiler or burner designs. Supplemental oxygen also increased with the H₂ ratio, from 1.0% to 11.9% of the total oxygen supply. Therefore, the effects of hydrogen and oxygen could not be fully separated, and the results may differ under air-only operation. Finally, the efficiency calculation did not include the energy required for hydrogen compression or supplemental oxygen production. Hydrogen compression alone may require approximately 4% of its lower heating value. NO_x was not measured directly. Therefore, NO_x was estimated using assumed NO₂/NO_x fractions of 1.2–5%, as described in Section 2.3. The 1.2% value was obtained from previous measurements of oil-fired boilers, but the kinetic calculation could not identify a single conversion factor. In addition, the measured NO₂ concentrations of 1.80–3.08 ppm was low compared with the analyzer accuracy of ±5 ppm. Therefore, the absolute NO_x values should be treated as semi-quantitative estimates. However, the measurements showed a consistent increase in NO₂, and the relative increase of 70.8% does not depend on the selected conversion factor.

Hydrogen flow was measured using a rotameter with an accuracy of ±2% of full scale. Error propagation gave an uncertainty of approximately ±0.6 percentage points at (r = 30%). A wider uncertainty of ±3 percentage points was adopted to include possible error from manual valve adjustment. System efficiency was calculated from one feedwater-totalizer measurement at each condition. The blowdown valve was closed, but no steam-flow meter was installed, and the drum water level was not recorded at the beginning and end of each

test. Therefore, changes in drum water inventory may have affected the calculated steam output and efficiency. The observed decrease of 3.60 percentage points should not be considered an exact efficiency loss, and the detailed shape of the efficiency curve was not interpreted. Future tests should include direct steam-flow measurement and drum-level monitoring. The measured dry-basis CO₂ values were also inconsistent with the carbon and oxygen balance (Section 3.5). Therefore, these values were excluded from the quantitative analysis. The CO₂ and O₂ channels should be checked using certified calibration gas.

All seven conditions were tested on the same day. Therefore, the reported variation represents only short-term repeatability and does not include day-to-day changes in boiler operation or ambient conditions. Ambient temperature ranged from 31.7 to 34.0 °C, but relative humidity and atmospheric pressure were not recorded. The time between test conditions was also limited. After the 5% H₂ condition, only 1.2–6.0 minutes passed before the next measurement window began. The boiler may therefore have continued adjusting to the new H₂ ratio during data collection. The semi-empirical model was calibrated and evaluated using the same seven conditions, so its agreement with the measurements does not represent independent validation. In addition, the H₂ ratio and oxygen enrichment were strongly correlated (correlation coefficient = 0.956). Their individual effects could not be fully separated. Future tests should vary oxygen enrichment while keeping the H₂ ratio constant.

The AICc results favored linear models over quadratic models for all four correlations (Table A3). The quadratic forms were retained to provide a consistent form across the reported correlations. However, the curvature should not be treated as a confirmed physical trend because only seven conditions were tested. The results apply only to the Kawasaki 15750 boiler at the low-fire setting. Equations 4–7 are empirical correlations for the tested range of 0–30% H₂/diesel energy ratio. They should not be extrapolated beyond this range or applied to other loads and boiler designs without further testing. The economic analysis also covers fuel purchase costs only and excludes additional capital, safety, storage, and maintenance costs.

The 20.1% reduction reported at 30% H₂ represents CO₂ intensity at the point of combustion. Upstream emissions from hydrogen production can change this result. Each kilogram of upstream CO₂ emitted per kilogram of H₂ adds approximately 5.4 g CO₂/kg steam. Gray hydrogen at 10 kg CO₂/kg H₂ would increase total CO₂ intensity to 211.4 g/kg steam, above the B7-only baseline of 196.9 g/kg. Blue hydrogen at 1 kg CO₂/kg H₂ would give 162.7 g/kg steam, and green hydrogen with near-zero upstream emissions would remain close to 157.3 g/kg steam. Therefore, low-carbon hydrogen is required to maintain the CO₂ benefit. Large-scale deployment may also be limited by hydrogen supply and regulation. Current pipeline blending limits range from 0.1 to 10 vol.% across different jurisdictions (Erdener *et al.*, 2023). These limits did not affect the present cylinder-supplied experiment but may affect future industrial scale-up.

4. Conclusions

Hydrogen co-firing was tested in a commercial B7 diesel-fired water-tube boiler at H₂/diesel energy ratios from 0% to 30%. At 30% H₂, steam output increased by 24.3%, from 323.8 to 402.4 kg/h, but the diesel flow remained relatively steady. However, total energy input increased by 29%.

CO decreased by 73.4%, from 107.0 to 28.5 ppm. Measured NO₂ increased by 70.8%, from 1.80 to 3.08 ppm. Depending on the assumed NO₂/NO_x fraction, estimated NO_x ranged from 36–

150 ppm at the baseline and 62–256 ppm at 30% H₂. The corrected values remained below the national limit of 200 ppm. However, the estimate based on the more conservative conversion fraction exceeded the Bangkok limit of 160 ppm at the higher H₂ ratios. Because NO was not measured and the NO₂ concentrations were low relative to the analyzer accuracy, the absolute NO_x values should be considered as semi-quantitative estimations. CO₂ intensity decreased by 20.1%, from 196.9 to 157.3 g CO₂/kg steam, at the point of combustion. However, the lifecycle benefit depends on how hydrogen is produced. Low-carbon hydrogen is required to maintain this reduction.

Calculated system efficiency decreased from 91.05% to 87.45%, corresponding to an apparent decrease of 3.60 percentage points. Because drum water inventory was not recorded, the exact efficiency change could not be calculated. The semi-empirical model reproduced the NO_x trend with a mean absolute percentage error of 4.0%, but it was calibrated using the same experimental data. Hence, it should only be used as a screening tool within the tested range. At the cylinder price paid in this study, approximately 753 THB/kg (23 USD/kg), hydrogen co-firing was not cost-competitive at any studied ratio. The estimated break-even hydrogen price was around 120–230 THB/kg (3.7–7.0 USD/kg). Thus, lower-cost bulk or on-site hydrogen supply would be required for economic operation.

The boiler operated without flashback, external ignition, or unplanned shutdown at all tested ratios. However, the results apply only to this boiler at the low-fire setting. Future work should include direct NO and steam-flow measurements, drum-level and steam-quality monitoring, accurate hydrogen mass-flow measurement, and tests at higher loads and pressures. Longer-term safety, durability, lifecycle, and economic assessments are also required before industrial retrofit can be recommended.

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Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data Availability

The experimental data and model calibration files supporting the findings of this study are available from the corresponding author upon reasonable request.

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Appendix A. Supporting data

Table A1. Uncertainty analysis for system efficiency.

(a) Relative sensitivity coefficients of the efficiency, $\partial \ln \eta / \partial \ln \theta$					
Variable θ	$\partial \ln \eta / \partial \ln \theta$	Comment			
Steam mass flow $\dot{m}_{st}$	+1.000	direct proportionality			
Total fuel energy $\dot{m}_{fuel} \cdot LHV$	-1.000	direct proportionality			
Steam quality x	+0.880	$x \cdot h_{fg} / (h_f + x \cdot h_{fg} - h_{fw})$			
Feedwater enthalpy h_{fw}	-0.052	small; enters only through the enthalpy rise			
(b) Instrument-related uncertainty budget for η (relative, %)					
r (%)	Steam flow	Fuel energy	Feedwater T	Combined	Combined (pp)
0%	0.154	0.300	0.343	0.48	± 0.44
5%	0.144	0.301	0.343	0.48	± 0.45
10%	0.137	0.328	0.343	0.49	± 0.46
15%	0.136	0.369	0.343	0.52	± 0.47
20%	0.130	0.417	0.343	0.56	± 0.50
25%	0.126	0.466	0.343	0.59	± 0.53
30%	0.124	0.516	0.343	0.63	± 0.55
(c) Effect of the assumed steam quality on absolute and relative efficiency					
Steam quality x	η at r = 0 (%)	η at r = 30% (%)	Relative change (%)		
0.92	88.52	85.02	-3.954		
0.95 (adopted)	91.05	87.45	-3.954		
0.98	93.58	89.88	-3.954		
1.00	95.27	91.50	-3.954		

Table A2. Parameters and calibration of the semi-empirical model of Section 2.5.

(a) Complete parameter set		
Parameter	Value	Origin
k_{scale} (Eq. 3)	fitted	matched to the $r = 0$ NO_x measurement
τ (residence time)	0.50 s	estimated from chamber volume and flow rate
β (OH enhancement)	1.74	MAPE scan over all seven conditions, $k_{\text{scale}}/\tau/T_{\text{ad}}$ held fixed
CO_{scale}	0.414	fitted to the $r = 0$ CO measurement
Dryer–Glassman pre-exponent	$2.5 \times 10^{10} \text{ s}^{-1}$	lumped value, this work
Dryer–Glassman activation energy	167.4 kJ/mol	(Dryer & Glassman, 1973)

(b) Refitted NO_x correlation (Eq. 3) against the measured condition means			
r (%)	T_{ad} (K)	NO_x measured (ppm, est.)	NO_x from Eq. 3 (ppm)
0	2202	36.0	36.0
5	2210	46.0	41.2
10	2212	50.0	45.6
15	2218	52.0	51.0
20	2215	54.5	54.5
25	2218	57.5	59.5
30	2216	61.5	63.4

Table A3. Fitted, cross-validated and information-criterion statistics for Eqs. 4–7 ($n = 7$ conditions). ΔAICc is the AICc of the quadratic form minus that of the linear form; positive values favor the linear form. K counts the estimated parameters including the residual variance, $K = 4$ for the quadratic and 3 for the linear form at $n = 7$, following Burnham and Anderson (2002). R^2_{CV} (int.) is scored on the five interior conditions only.

Eq.	Quantity	Fit R^2	R^2_{CV} (all)	RMSE_{cv}	R^2_{CV} (int.)	ΔAICc	Preferred
4	Steam rate (kg/hr)	0.986	0.945	6.0 kg/hr	0.913	+7.9	linear
5	CO (ppm)	0.960	0.704	13.3 ppm	0.689	+5.1	linear
6	NO_x (ppm, est.)	0.960	0.720	4.1 ppm	0.708	+9.8	linear
7	CO_2 intensity (g/kg steam)	0.970	0.879	4.4 g/kg	0.803	+9.2	linear

Table A4. Fuel-cost and capital sensitivity of the retrofit. Price scenarios, all for H₂ delivered, with B7 at 52.0 THB/kg throughout: 2025 is the cylinder price actually paid, 753 THB/kg, 2028 a projection at 120 THB/kg, 2030 the IRENA target at 80 THB/kg and 2050 a long-term price of 40 THB/kg (≈23, 3.7, 2.4, and 1.2 USD/kg, respectively). Panel (b) annuitizes the fuel-cost margin over ten years at 8%.

(a) Fuel-cost penalty (%) against the B7-only baseline, four H ₂ price scenarios					
r (%)	2026	2028	2030	2050	
5	+16.7	−3.4	−4.7	−6.0	
10	+34.5	−3.9	−6.4	−8.8	
15	+56.9	−0.6	−4.2	−7.8	
20	+70.5	−2.2	−6.8	−11.4	
25	+85.4	−2.3	−7.8	−13.4	
30	+103.3	−0.4	−7.0	−13.5	

(b) Capital expenditure serviceable by the fuel-cost margin (million THB), annuitized over ten years at 8%					
Hours per year	H ₂ ratio	2026	2028	2030	2050
2,000	10%	none	0.63	1.01	1.40
2,000	20%	none	0.37	1.14	1.92
2,000	30%	none	0.08	1.23	2.39
4,000	10%	none	1.25	2.03	2.80
4,000	20%	none	0.74	2.29	3.84
4,000	30%	none	0.15	2.47	4.78
6,000	10%	none	1.88	3.04	4.21
6,000	20%	none	1.11	3.43	5.76
6,000	30%	none	0.23	3.70	7.17