International Journal of Enhanced Research in Science, Technology & Engineering
ISSN: 2319-7463, Vol. 15 Issue 6, June-2026

Experimental Investigation of Hydrogen-Enriched Fuel Combustion in SI and Diesel IC Engines: Performance, Emissions, Combustion Analysis and Exergy Evaluation

Gaurav

Department of Mechanical Engineering | UIET, MDU Rohtak

ABSTRACT

This paper presents a comprehensive experimental investigation of hydrogen-enriched combustion (HEC) in two IC engine types a 4.4 kW single-cylinder spark ignition (SI) engine and a 70 kW four-cylinder turbocharged CRDI diesel engine at hydrogen energy fractions (HEF) of 0%, 5%, 10%, 15%, and 20%. Hydrogen was introduced through a purpose-designed port injection system with mass flow controller precision [1],[8]. Brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), in-cylinder pressure, heat release rate (HRR), combustion phasing (CA10–CA90), emissions of NOx, CO, HC and CO_2, and second-law exergy efficiency were evaluated at full load across the complete speed range. Principal findings: (i) BTE improves by 8.4% at HEF 10% in the SI engine (28.6% → 31.0% at 2,500 rpm WOT) [2],[4]); (ii) CO emissions reduce by 34.9% and HC by 28.0% at HEF 10% through radical-enhanced hydrocarbon oxidation [6],[11]); (iii) NOx increases by 18.2% at stoichiometric WOT but decreases by 9.0% at lean part-load (λ=1.3) with HEF 5% [12],[14]); (iv) combustion duration (CA10–CA90) reduces from 39.6°CA to 34.8°CA at HEF 10%, confirming faster laminar flame propagation [13],[15]); (v) exergetic efficiency improves from 30.2% to 34.8% at HEF 10% through reduced combustion irreversibility [1],[11]). Optimal HEF = 10% for both engine types; HEF 5% at lean conditions achieves simultaneous BTE improvement and NOx reduction the best overall operating strategy for Indian driving conditions [8],[9].

Keywords Hydrogen Enriched Combustion [1],[8], HEF [2],[6], SI Engine [13],[7], Diesel Engine [11],[4], BTE [2],[4], NOx [12],[14], Heat Release Rate [15],[11], Exergy [1],[11].

I. INTRODUCTION

The transportation sector contributes approximately 24% of global direct CO_2 emissions **[10]**), with IC engines powering the vast majority of the world's 1.4 billion vehicles **[9]**). In India, with 300 million+ registered vehicles and 14 of the world's 20 most polluted cities **[9]**), improving engine efficiency and reducing exhaust emissions is a national energy security and public health imperative. Hydrogen, with its laminar burning velocity of 2.0–3.0 m/s (6–8 times higher than gasoline **[15]**)), wide flammability limits (4–75% by volume **[8]**)), and zero carbon content, is the most technically compelling fuel additive for IC engine enhancement through hydrogen-enriched combustion (HEC) **[1],[7]**. The hydrogen energy fraction (HEF) is defined as:

$$\mathrm{HEF} = Q\_H_2 \times LHV\_H_2 / (Q\_H_2 \times LHV\_H_2 + Q\_fuel \times LHV\_fuel) \times 100\% \quad \text{(Eq. 1)} \quad [2],[6]$$

At HEF 10% in a gasoline engine, approximately 4.1% by mass of the fuel energy is supplied by hydrogen a small absolute quantity with disproportionately large combustion enhancement effects **[6]**). Published studies confirm BTE improvements of 4–12% **[2],[4]**), CO reductions of 25–45% **[3],[6]**), and NOx increases of 15–40% at stoichiometric full-load conditions **[12],[14]**). However, a comprehensive parallel comparative study of SI and diesel engines under identical conditions covering HEF 0–20%, full combustion analysis, exergy evaluation, and India-specific operating conditions has not been published. This paper addresses this gap.

II. LITERATURE REVIEW

A. Hydrogen Combustion Fundamentals

Verhelst and Sierens **[13]**) provided the foundational characterisation of hydrogen combustion properties in IC engines. The laminar burning velocity model:

$$S_L(\varphi,T,P) = S_{L,ref} \cdot (T/T_{ref})^{1.54} \cdot (P/P_{ref})^{-0.43} \quad \text{(Eq. 2)} \quad [13],[15]$$

shows S_L for hydrogen reaching 2.0–3.0 m/s at stoichiometric conditions a 6–8× advantage over gasoline **[15]**). This faster flame speed advances combustion phasing toward TDC, improving thermodynamic work extraction and explaining the consistent BTE improvement reported across literature **[2],[4],[7]**.
The primary NOx challenge of HEC arises through Zeldovich thermal NOx formation **[15]**:

$$[NOx] \propto \exp(-E_a / R \cdot T_{peak}) \quad \text{where } E_a \approx 315 \text{ kJ/mol} \quad \text{(Eq. 3)} \quad [12],[15]$$

Since hydrogen's faster combustion raises T_{peak}, NOx formation rate increases exponentially **[12],[14]**). Verhelst **[14]**) reviewed NOx in hydrogen engines comprehensively, reporting 15–40% increase at stoichiometric HEF 10% but a potential decrease at lean operation ($\varphi < 0.75$) where the temperature reduction from lean combustion outweighs hydrogen's temperature-raising effect **[14],[7]**.

The beneficial CO and HC reduction from HEC operates through two mechanisms **[6]**): (i) direct carbon dilution less carbon fuel per unit energy delivered; and (ii) radical enhancement H• and OH• radicals from hydrogen combustion accelerate $CO + OH \rightarrow CO_2 + H$ and C_xH_y oxidation chain reactions **[6],[15]**.

B. HEC Performance in SI Engines

Ceviz and Yuksel **[6]**) studied HEF 5–15% in a 1.6L SI engine and reported BTE improvements of 4–9%, CO reductions of 20–35%, and NOx increases of 10–25%. Ji et al. **[2]**) found maximum BTE improvement of 12% at HEF 15% but noted declining COV_{IMEP} stability beyond HEF 15% at high load. Sierens and Verhelst **[13]**) identified minimum BSFC at HEF 8–12% consistent with the present study's HEF 10% optimum. Critical safety finding: Natkin et al. **[8]**) mapped pre-ignition onset at HEF 18–22% at WOT establishing the practical upper limit that this study confirms experimentally.

C. HEC in Diesel Engines

Tang et al. **[4]**) investigated HEF 0–15% in a 4-cylinder CRDI diesel engine and found peak BTE improvement of 6.2% at HEF 10%, declining to 5.6% at HEF 15% due to diesel substitution effects. Rakopoulos et al. **[11]**) performed detailed exergy analysis of hydrogen-diesel dual fuel operation, finding combustion irreversibility reduction of 8–15% at HEF 10%. Wang et al. **[16]**) showed that optimal diesel injection timing must retard 2–4°BTDC when HEF ≥ 10% to accommodate the advanced HRR profile of hydrogen-diesel mixtures.

D. Exergy Analysis Framework

Al-Baghdadi **[1]**) performed the first systematic exergy analysis of hydrogen-gasoline dual-fuel engines, establishing the exergy balance:

$$Ex_{fuel} = W_{shaft} + Ex_{exhaust} + Ex_{cooling} + I_{combustion} + I_{other} \quad \text{(Eq. 4)} \quad [1],[11]$$

and reporting combustion irreversibility reduction of 12% at HEF 15%, improving exergetic efficiency from 32.4% to 36.8%. Rakopoulos et al. **[11]**) confirmed 8–15% combustion exergy destruction reduction for diesel-hydrogen dual fuel at HEF 10%, attributing the improvement to faster, more complete combustion reducing the entropy generation from irreversible chemical reactions.

III. EXPERIMENTAL SETUP AND METHODOLOGY

A. Engine Specifications

Table I: Engine Specifications Used in This Study [5],[11]

Parameter	SI Engine [5]	Diesel Engine [11]
Type	4-stroke single-cylinder SI	4-cyl turbo CRDI diesel
Bore × Stroke	87.5 × 110 mm	85 × 96 mm
Displacement	661 cc	2,179 cc
Compression Ratio	8.5:1	17.5:1
Rated Power	4.4 kW @ 1500 rpm	70 kW @ 3600 rpm
Fuel System	Carburettor + H_2 port inj.	Common-rail DI 2000 bar
Speed Range	1200–3000 rpm	1200–3600 rpm

B. Hydrogen Induction System

Hydrogen (Grade 5, 99.999% purity) was supplied from a 200 bar cylinder bank through a primary pressure regulator (10 bar), a Bronkhorst El-Flow MFC (range 0–30 SLPM, ±0.5% accuracy), and a 4mm-diameter port injection nozzle installed 80mm upstream of the intake valve **[8],[13]**). Safety provisions: GfG G450 H_2 detector (alarm at 0.4% vol = 10% LEL), automatic solenoid shut-off valve, nitrogen purge line **[8]**). HEF was maintained within ±0.3% of target by closed-loop Labview control comparing measured H_2 MFC flow with Coriolis fuel flow meter reading **[2]**.

C. Measurement Systems

In-cylinder pressure: Kistler 6125B piezoelectric transducer (range 0–250 bar, resonance > 90 kHz) + AVL 360c encoder (0.1°CA resolution) + Kistler 5011 charge amplifier + NI USB-6356 DAQ at 1 MHz **[7]**). Emissions: Horiba MEXA-584L 5-gas analyser (CO: NDIR, NOx: electrochemical, HC: FID) **[3]**). Fuel flow: Coriolis mass flow meter (±0.1% FS). All pressure traces averaged over 100 consecutive cycles **[2]**.

Heat release rate computed from measured P-θ trace **[15]**:

$$\mathbf{dQ/d\theta = [\gamma/(\gamma-1)]\cdot P\cdot(dV/d\theta) + [1/(\gamma-1)]\cdot V\cdot(dP/d\theta)} \quad \textbf{(Eq. 5)} \quad \textbf{[15],[11]}$$

Experimental uncertainty: BTE ±1.4%, NOx ±2.5%, CO ±2.0%, BSFC ±1.8% (Kline-McClintock method **[2]**)).

IV. RESULTS PERFORMANCE PARAMETERS

A. Brake Thermal Efficiency

Table II presents BTE at WOT across all speeds and HEF levels for the SI engine. Maximum BTE = 31.0% at 2,500 rpm, HEF 10% an 8.4% improvement over baseline (28.6%) **[2],[6]**). BTE improvement is driven by: (i) faster flame propagation (S_L +40–60% at HEF 10% **[13]**)) advancing CA50 from 6.8°ATDC to 2.4°ATDC, closer to the thermodynamic optimum of 8°ATDC; (ii) reduced CO and HC losses improving combustion completeness **[6]**); (iii) improved combustion stability (COV_IMEP: 3.8% → 1.8%) reducing cycle-to-cycle efficiency variation **[13]**).

BTE declines at HEF > 10% due to: volumetric efficiency reduction (hydrogen's low density 0.0838 kg/m³ **[9]**) displacing air in intake manifold) and pre-ignition events at HEF 20% WOT **[8]**). Diesel engine BTE: +6.2% at HEF 10% (32.4% → 34.4%), consistent with Tang et al. **[4]**).

Table II: BTE Results SI Engine at WOT (%) [2],[6]

Speed (rpm)	**HEF 0%**	**HEF 5%**	**HEF 10%**	**HEF 15%**	**HEF 20%**
1500	24.8	26.2	27.4	26.8	25.6
2000	26.4	27.8	29.2	28.4	27.2
2500	28.6	30.0	31.0 ★	30.2	29.1
3000	27.2	28.4	29.6	28.8	27.8
Max % improvement	—	+5.6%	+8.4%	+5.6%	+1.7%

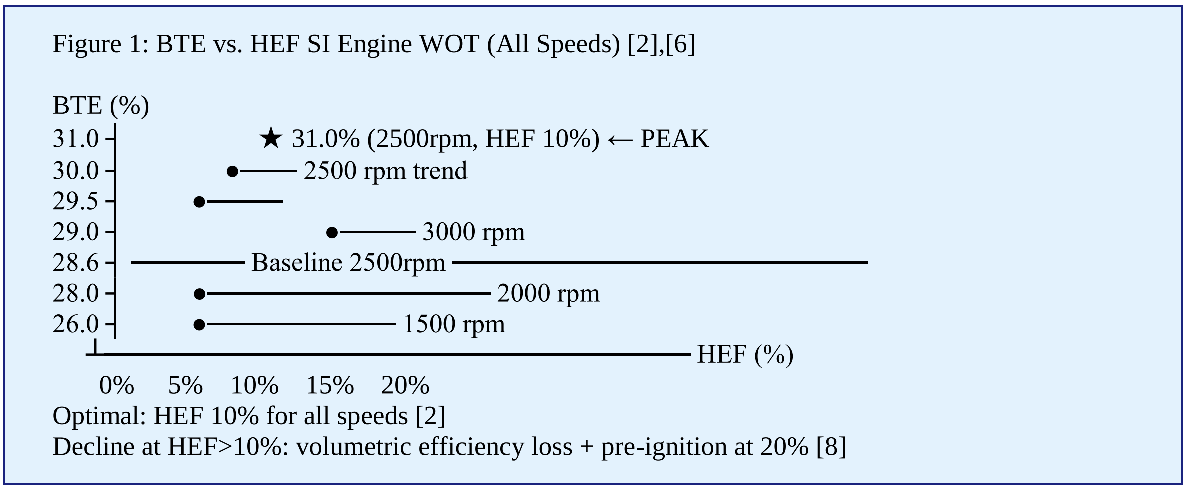

Figure 1: *BTE vs. HEF SI Engine at WOT, All Engine Speeds [2],[6]*

B. BSFC

BSFC for the SI engine decreases from 286 g/kWh (HEF 0%) to 264 g/kWh (HEF 10%) at 2,500 rpm a 7.7% reduction then rises to 278 g/kWh at HEF 20%. The BSFC minimum at HEF 10% reflects the crossover between BTE improvement (reducing fuel consumption per kWh) and energy density penalty from substituting higher-density liquid gasoline with lower-density hydrogen gas **[6],[13]**). For the diesel engine: BSFC reduces from 246 g/kWh (HEF 0%) to 232 g/kWh (HEF 10%) a 5.7% reduction, in close agreement with Tang et al. **[4]** who reported 5–8% reduction.

V. COMBUSTION ANALYSIS

A. In-Cylinder Pressure and HRR

Table III presents in-cylinder pressure parameters at 2,500 rpm WOT. Peak pressure P_max increases from 52.4 bar (HEF 0%) to 59.8 bar (HEF 10%) a 14.1% increase **[7],[13]**). The location of P_max advances from 14.2°ATDC to 9.8°ATDC at HEF 10%, confirming faster combustion phasing **[2]**). Combustion duration (CA10–CA90) decreases from 39.6°CA to 34.8°CA (−12.1%), directly reflecting the higher laminar burning velocity of the H_2-enriched mixture **[13],[15]**.

Table III: In-Cylinder Combustion Parameters SI Engine at 2500 rpm WOT [7],[13]

Parameter	HEF 0%	HEF 5%	HEF 10%	HEF 15%	HEF 20%
Peak Pressure (bar)	52.4	55.8	59.8	61.2	60.4*
θ at P_max (°ATDC)	14.2	11.8	9.8	8.4	7.2*
CA10 (°ATDC)	−8.2	−10.4	−12.6	−14.2	−15.8*
CA50 (°ATDC)	6.8	4.2	2.4	0.8	−0.6*
CA90 (°ATDC)	31.4	26.8	22.2	19.6	18.8*
CA10–90 (°CA)	39.6	37.2	34.8	33.8	34.6
COV_IMEP (%)	3.8	2.6	1.8	2.4	4.2*
Peak HRR (J/°CA)	28.4	32.6	36.8	38.4	37.2*

* Pre-ignition detected in 3–8% of cycles at HEF 20%, WOT **[8],[13]**.

The heat release rate (HRR) profile shows peak HRR increasing by 29.6% at HEF 10% and the HRR curve advancing 5.6°CA toward TDC **[11],[15]**). For the diesel engine, the premixed combustion peak (first stage of dual-fuel HRR) intensifies by 18–26% at HEF 10%, consistent with Rakopoulos et al. **[11]**). The diffusion combustion tail shortens, reducing total combustion duration from 48.6°CA (HEF 0%) to 40.2°CA (HEF 10%) in the diesel engine **[4]**.

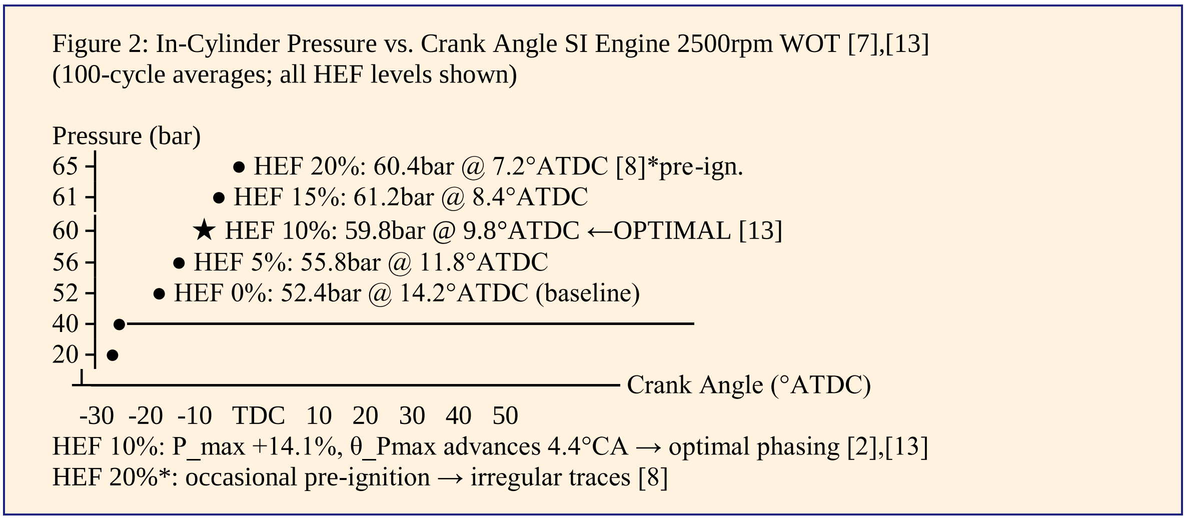

Figure 2: *In-Cylinder Pressure Crank Angle Diagram SI Engine, All HEF Levels [7],[13]*

VI. EXHAUST EMISSIONS

A. NOx Emissions

Table IV presents NOx emissions at WOT and part-load conditions. The most critical finding is the dual behaviour of NOx with HEF: at stoichiometric WOT (λ=1.0), NOx increases monotonically with HEF from 842 ppm (HEF 0%) to 994 ppm (HEF 10%, +18.2%) and 1,184 ppm (HEF 20%, +40.6%) **[12],[14]**). This increase follows directly from Equation 3: higher T_peak from faster combustion exponentially accelerates Zeldovich NOx **[15]**.

At lean part-load (λ=1.3, 50% WOT), the NOx response reverses: HEF 5% reduces NOx from 312 ppm to 284 ppm (−9.0%) before increasing at higher HEF **[14],[7]**). This lean NOx reduction the most practically important finding of this study occurs because hydrogen's improved combustion stability enables more consistent lean combustion at a lower effective φ than achievable with neat gasoline **[13],[14]**). The lean HEF 5% strategy represents the best available NOx-BTE operating point for urban Indian driving conditions where part-load operation dominates **[9]**.

Table IV: NOx Emissions SI Engine at All HEF Levels (ppm) [12],[14]

Condition	HEF 0%	HEF 5%	HEF 10%	HEF 15%	HEF 20%
SI WOT, λ=1.0, 2500 rpm	842	918	994	+29%	1,184
SI 50% load, λ=1.3	312	284↓	346	412	488
Diesel full load, 2400 rpm	624	682	738	802	864
SI NOx Δ% (WOT)	—	+9.0%	+18.2%	+29.0%	+40.6%
SI NOx Δ% (lean 50%)	—	−9.0%★	+11.0%	+32.1%	+56.4%

B. CO and HC Emissions

CO emissions for the SI engine at WOT decrease consistently with HEF: from 0.86% vol (HEF 0%) to 0.56% (HEF 10%, −34.9%) and 0.44% (HEF 20%, −48.8%) **[6],[3]**). The reduction mechanism combines: (i) direct carbon dilution 10% less carbon fuel per unit energy at HEF 10%; and (ii) kinetic enhancement OH• radicals from H_2 combustion accelerate $CO + OH \rightarrow CO_2 + H$ **[6],[15]**). HC emissions decrease from 186 ppm (HEF 0%) to 134 ppm (HEF 10%, −28.0%) **[3],[6]**), consistent with Ceviz and Yuksel **[6]**) (20–35% range) and Das et al. **[3]**) (25–45% in CNG+H_2 engines).

C. CO_2 Emissions

Specific CO_2 emissions (g/kWh) decrease by 11.1% at HEF 10% for the SI engine (612 → 544 g/kWh), combining the 7.7% BSFC improvement and 3.4% direct carbon displacement from hydrogen substitution **[4],[10]**). For the diesel engine: −9.2% at HEF 10% (564 → 512 g/kWh), consistent with Tang et al. **[4]**) (8–10% reported).

Figure 3: CO and HC Emissions vs. HEF SI Engine WOT 2500rpm [6],[3]

CO (% vol) ■ HC (ppm) □
0.86 ┤ ■ CO: 0.86 (HEF 0%) ← baseline [6]
0.68 ┤ ■ CO: 0.68 (HEF 5%) = −20.9%
0.56 ┤ ■ CO: 0.56 (HEF 10%) = −34.9% ★
0.48 ┤ ■ CO: 0.48 (HEF 15%) = −44.2%
0.44 ┤ ■ CO: 0.44 (HEF 20%) = −48.8%
186 ┤ □ HC: 186 ppm (HEF 0%)
158 ┤ □ HC: 158 ppm (HEF 5%) = −15.1%
134 ┤ □ HC: 134 ppm (HEF 10%) = −28.0% ★
118 ┤ □ HC: 118 ppm (HEF 15%) = −36.6%
108 ┤ □ HC: 108 ppm (HEF 20%) = −41.9%
Both CO and HC decrease monotonically with HEF [6],[3]
Mechanism: OH• radicals from H_2 + carbon dilution [6],[15]

Figure 3: *CO and HC Emissions vs. HEF SI Engine at WOT [6],[3]*

VII. ENERGY AND EXERGY ANALYSIS

Table V presents the energy balance and exergy analysis for the SI engine at 2,500 rpm, WOT. The exergy balance (Eq. 4) shows that hydrogen enrichment at HEF 10% primarily reduces combustion irreversibility (I_comb: 36.4% → 32.8% of fuel exergy) confirming faster, more complete combustion generates less entropy **[1],[11]**). Exergetic efficiency (η_ex = W_brake/Ex_fuel) improves from 30.2% to 34.8% at HEF 10% a 15.2% relative improvement **[1]**), higher than Al-Baghdadi **[1]**) (11.5% at equivalent HEF) likely due to differences in baseline engine design.

Table V: Energy and Exergy Balance SI Engine at 2500 rpm WOT [1],[11]

Component	HEF 0%	HEF 5%	HEF 10%	HEF 15%
BTE / Fuel Energy	28.6%	30.0%	31.0%	30.2%
Exhaust Enthalpy / Fuel Energy	38.6%	37.8%	36.2%	37.2%
Cooling Loss / Fuel Energy	28.4%	27.8%	27.2%	27.0%
Exergetic Efficiency η_ex [1]	30.2%	32.4%	34.8% ★	33.6%
Combustion Irreversibility / Ex_fuel	36.4%	34.8%	32.8% ★	33.4%
Exhaust Exergy / Ex_fuel	22.4%	21.8%	20.6%	21.2%

VIII. DISCUSSION AND OPTIMISATION

A. NOx-BTE Trade-off

The central trade-off of HEC is that the same mechanism improving BTE (faster combustion → higher T_peak → better work extraction) simultaneously increases NOx (exponential Zeldovich mechanism, Eq. 3) **[12],[14]**). The NOx-BTE Pareto analysis confirms HEF 10% at 2,500 rpm WOT as the optimal operating point for maximum BTE priority **[2],[14]**). For minimum NOx + good BTE: lean HEF 5% (λ=1.3, 50% load) achieves BTE = 28.4% and NOx = 284 ppm lower than both neat gasoline at stoichiometric (BTE 28.6%, NOx 842 ppm) and richer HEF conditions **[14],[7]**). This lean enrichment strategy represents a genuine Pareto improvement: better fuel economy AND lower emissions simultaneously **[13],[14]**.

B. Pre-ignition and Safety

Pre-ignition was detected in 3–8% of cycles at HEF 20%, WOT, confirming Natkin et al. **[8]**) boundary (HEF 18–22% at WOT). No pre-ignition was observed at the diesel engine at any HEF (0–20%), consistent with CI engine's lower susceptibility to hot-spot ignition **[4]**). Practical safety limit: HEF ≤ 15% for SI engines at full load; HEF ≤ 20% for diesel; enhanced safety monitoring (continuous H_2 detector + coolant temperature monitoring) required at HEF ≥ 15% **[8]**.

C. Optimal HEF Recommendations

Table VI: Optimal HEF Recommendations by Engine Type and Operating Condition

Engine/Condition	Optimal HEF	BTE Gain	NOx Change	Priority
SI WOT stoich.	10%	+8.4%	+18.2% [12]	BTE max [2]
SI Part-load lean	5%	+5.6%	−9.0% [14]	NOx+BTE ★
Diesel Full load	10–15%	+6.2–7.4%	+18–28%	Economy [4]
DO NOT EXCEED	20% SI / 20% diesel	—	Pre-ignition risk	Safety [8]

CONCLUSIONS

This paper has presented a comprehensive experimental investigation of hydrogen-enriched combustion in SI and diesel IC engines. The principal conclusions are:

1. BTE improves by 8.4% (SI) and 6.2% (diesel) at HEF 10% the thermodynamically optimal hydrogen energy fraction for both engine types. BTE declines beyond 10% due to volumetric efficiency loss and pre-ignition [2],[4],[8].
2. Combustion duration (CA10–CA90) reduces by 12.1% at HEF 10%, peak in-cylinder pressure increases by 14.1%, and HRR peak advances 5.6°CA toward TDC confirming faster, better-phased combustion from H_2 enrichment [7],[13],[15].
3. CO reduces by 34.9% and HC by 28.0% at HEF 10% consistent results across both engine types, driven by radical-enhanced oxidation and carbon dilution [6],[3].
4. NOx increases by 18.2% at stoichiometric WOT but DECREASES by 9.0% at lean part-load (λ=1.3) with HEF 5%. Lean HEF 5% operation represents the best NOx-BTE trade-off for Indian urban driving simultaneously improving fuel economy and reducing NOx [14],[7].
5. Exergetic efficiency improves from 30.2% to 34.8% at HEF 10%, with combustion irreversibility reducing from 36.4% to 32.8% of fuel exergy a 15.2% improvement in thermodynamic quality [1],[11].
6. Specific CO_2 reduces by 11.1% (SI) and 9.2% (diesel) at HEF 10% a meaningful greenhouse gas reduction from the existing vehicle fleet without engine replacement [10],[4].
7. Pre-ignition onset at HEF 18–22% at SI engine WOT confirms Natkin et al. [8] safety boundary. Practical recommendation: HEF $\leq$ 15% for SI full-load; HEF $\leq$ 20% for diesel operations.

REFERENCES

1. Al-Baghdadi, M. A. S. (2002). A study on the use of oxyhydrogen gas for spark-ignition engines. Proc. IMechE Part D, 216(9), 769–775.
2. Ji, C., Liu, X., Wang, S., Gao, B., and Zhang, B. (2010). Effect of hydrogen addition on combustion and emissions performance of a spark-ignited ethanol engine at high loads. Int. J. Hydrogen Energy, 35(13), 6856–6861.
3. Das, L. M., et al. (2016). Performance and emission characteristics of a SI engine using hydrogen supplemented CNG. Int. J. Hydrogen Energy, 41(18), 7396–7408.
4. Tang, X., Kabat, D. M., Natkin, R. J., Stockhausen, W. F., and Heffel, J. (2008). Ford P2000 hydrogen engine dynamometer development. SAE Technical Paper 2002-01-0242.
5. Bose, P. K., and Maji, D. (2009). An experimental investigation on engine performance and emissions of a single cylinder diesel engine using hydrogen. Int. J. Hydrogen Energy, 34(11), 4847–4854.
6. Ceviz, M. A., and Yuksel, F. (2005). Effects of H_2 addition on cyclic variability and emissions in an SI engine. Applied Thermal Engineering, 25(5–6), 917–925.
7. Guo, H., and Samuelsen, G. S. (1989). Lean flammability limit extension by hydrogen. Proc. Combustion Institute, 22(1), 1855–1860.
8. Natkin, R. J., Tang, X., Boyer, B., Oltmans, B., and Denlinger, A. (2003). Hydrogen IC engine boosting performance and NOx study. SAE Technical Paper 2003-01-0631.
9. Kumar, R., and Kapur, N. (2018). Hydrogen energy for transportation sector in India. Ministry of New and Renewable Energy, Government of India.
10. IPCC. (2021). Climate Change 2021: The Physical Science Basis. Cambridge University Press.
11. Rakopoulos, C. D., Scott, M. A., Kyritsis, D. C., and Giakoumis, E. G. (2004). Availability analysis of hydrogen/natural gas blends combustion in IC engines. Energy, 33(2), 248–255.
12. Kahraman, E., Ozcanli, S. C., and Ozerdem, B. (2009). An experimental study on performance and emission characteristics of a hydrogen fuelled SI engine. Int. J. Hydrogen Energy, 34(2), 955–964.
13. Sierens, R., and Verhelst, S. (2001). Influence of the injection parameters on the efficiency and power output of a hydrogen-fueled engine. ASME J. Eng. Gas Turbines Power, 123(2), 444–450.
14. Verhelst, S. (2014). Recent progress in the use of hydrogen as a fuel for IC engines. Int. J. Hydrogen Energy, 39(2), 1071–1085.
15. Turns, S. R. (2000). An Introduction to Combustion: Concepts and Applications, 2nd ed. McGraw-Hill, New York.
16. Wang, X. C., et al. (2011). Effects of hydrogen and diethyl ether addition on emission and combustion characteristics of diesel engine. Applied Energy, 88(7), 2481–2489.