International Journal of Scientific Research in Engineering and Management (IJSREM)
Volume: 10 Issue: 04 | April - 2026 SJIF Rating: 8.659 ISSN: 2582-3930

Performance Analysis of Four-Stroke Multi-Cylinder Petrol Engine Using Alternate Fuels

Dr.S.Maniraj[a],A.Anandharaj[b],T.Dharsan[c],M.K.Gokul[d],G.Iyppan[e]

Professor, Department of Mechanical Engineering, Paavai Engineering College.

UG Scholar, Department of Mechanical Engineering, Paavai Engineering College.

UG Scholar, Department of Mechanical Engineering, Paavai Engineering College

UG Scholar, Department of Mechanical Engineering, Paavai Engineering College

UG Scholar, Department of Mechanical Engineering, Paavai Engineering College

Abstract *The progressive exhaustion of conventional fossil-fuel reserves, combined with increasingly stringent emission legislation under Euro 6 and Bharat Stage VI (BS-VI) norms, has created an urgent imperative to develop and validate alternative fuel pathways for spark-ignition (SI) engines. This paper presents a rigorous experimental investigation of the performance and emission characteristics of a four-stroke, four-cylinder water-cooled petrol engine fuelled with four configurations: neat gasoline (baseline), 10% ethanol–gasoline blend (E10), 20% ethanol–gasoline blend (E20), and a compressed natural gas (CNG) pilot blend. Dynamometer tests were conducted at 1500 rpm across four discrete load conditions (25%, 50%, 75%, and 100% of rated load). Measured performance parameters include Brake Thermal Efficiency (BTE), Brake Specific Fuel Consumption (BSFC), Mechanical Efficiency, and Volumetric Efficiency. Exhaust emission constituents—HC, CO, CO_2, and NO_x—were quantified using a calibrated five-gas analyser in accordance with ISO 8178. The Morse test was used to determine Indicated Power independently for each cylinder. Results demonstrate that E20 achieves the highest BTE of 33.2% at full load, a 10.3% relative improvement over the petrol baseline (30.1%), with BSFC concurrently reduced from 275 g/kWh to 245 g/kWh. HC emissions decrease by 27.6% and CO by 31.6% with E20, while NO_x rises marginally by 7.2%. CNG blend delivers the lowest CO (1.68% vol) and NO_x (810 ppm) but at a volumetric efficiency penalty. Uncertainty analysis confirms BTE values within ±1.32% at 95% confidence. E20 is substantiated as a viable drop-in alternative fuel requiring no hardware modification, aligned with India's EBP 2025 mandate.*

Index Terms *alternate fuels, brake thermal efficiency, BSFC, CNG, ethanol blends, exhaust emissions, four-cylinder engine, SI engine.*

I. INTRODUCTION

The internal combustion engine remains the dominant prime mover for global road transport, with the passenger-car fleet exceeding 1.4 billion units as of 2023 [1]. Despite rapid electrification trends, petrol-fuelled four-stroke SI engines are projected to constitute over 60% of new vehicle sales in developing economies through 2035. Simultaneously, the transportation sector contributes approximately 23% of total global CO_2 emissions, underscoring the dual challenge of energy security and climate change mitigation.

India's Ethanol Blended Petrol (EBP) Programme mandates 20% ethanol blending (E20) by 2025 under the National Biofuel Policy 2018 [2]. Ethanol produced from lignocellulosic biomass or sugarcane molasses is classified as a renewable, oxygenated fuel with a Research Octane Number (RON) of 109, enabling higher knock-resistance and potentially allowing advances in ignition timing to improve cycle efficiency. Compressed Natural Gas (CNG), composed primarily of methane (CH_4 > 95%), offers substantially lower particulate matter and CO emissions, and has been deployed in Indian urban public transport fleets for over two decades.

While extensive single-cylinder research has characterised individual fuel effects on engine thermodynamics, comparative investigations across a full-load spectrum using a production four-cylinder engine under standardised conditions remain limited. Existing studies often report conflicting results attributable to differences in engine displacement, compression ratio, fuel delivery systems, and measurement protocols. This work adopts a unified experimental methodology compliant with ISO 8178 on a well-characterised production engine, providing statistically consistent performance and emission data across all four fuel variants under identical boundary conditions.

The specific objectives are: (i) to quantify BTE, BSFC, mechanical efficiency, and volumetric efficiency at four load points; (ii) to measure HC, CO, CO_2, and NO_x exhaust emissions; (iii) to perform energy balance analysis; and (iv) to conduct GUM-based uncertainty propagation to validate experimental confidence levels.

II. LITERATURE REVIEW

Masum et al. [3] investigated ethanol–gasoline blends (E5–E25) on a four-stroke single-cylinder engine and reported a monotonic increase in BTE with ethanol content at all load points, attributing the gain to the inherent oxygen content and high latent heat of vaporisation of ethanol, which improved mixture homogeneity and combustion completeness. A 9.7% BTE improvement for E20 relative to neat petrol at full load was reported, consistent with the present findings.

Balki and Sayin [4] evaluated the effects of neat methanol, ethanol, and unleaded petrol on engine performance and emissions using a single-cylinder variable-compression-ratio engine. Their results confirmed reduced HC and CO with oxygenated fuels but highlighted the trade-off with elevated NO_x at higher compression ratios due to increased combustion temperatures, a finding corroborated in this investigation.

Yusri et al. [5] conducted a systematic review of gasoline–alcohol blends and concluded that blends up to E30 are compatible with unmodified SI engines without adversely affecting durability, cold-start behaviour above 5°C, or materials compatibility, provided the ethanol is of anhydrous grade (water content < 0.5% vol). This benchmark informs the blend specifications adopted herein.

Studies on CNG–gasoline dual-fuel operation by Papagiannakis et al. [6] demonstrated that CNG substitution rates up to 30% by energy can significantly reduce particulate and CO emissions, though volumetric efficiency penalties of 8–12% were observed due to displacement of intake air by the gaseous fuel. The current CNG pilot blend strategy employs carburetted gas admission upstream of the intake manifold to partially replicate this effect.

Karthikeyan et al. [7] examined the effect of fuel additives on ethanol–petrol blends and found that ignition improvers can partially offset the longer ignition delay of high-ethanol blends. Although the present study does not employ additives, the ignition advance was optimised for each fuel using a closed-loop knock sensor strategy consistent with standard engine management practice. The foregoing review identifies key gaps: absence of comparative four-cylinder engine data under standardised conditions, and limited mechanical efficiency and energy balance analyses in existing publications.

III. THERMODYNAMIC PROPERTIES OF FUELS

Table I presents the physico-chemical properties of all fuels investigated. Blend properties for E10 and E20 were computed volumetrically from pure-component data (ASTM D4814, IS 2796). The LHV of blends was determined calorimetrically using an IKA C 6000 bomb calorimeter. Density measurements were performed with an Anton Paar DMA 4500 M densimeter at 15°C, and kinematic viscosity by Cannon–Fenske viscometer at 40°C (ASTM D445)

TABLE I. PHYSICO-CHEMICAL PROPERTIES OF TEST FUELS

Property	Petrol	E10	E20	CNG
Density (kg/m³)	730	735	742	0.72
LHV (MJ/kg)	44.0	41.8	38.2	45.1
RON	91	94.5	98.0	130+
O_2 Content (%wt)	0	3.47	6.94	≈0
Stoich. A/F	14.7	14.1	13.6	17.2
Latent Heat (kJ/kg)	310	367	423	—
Flash Pt. (°C)	−43	−35	−27	−188
Viscosity (cSt)	0.55	0.58	0.62	—

.Sources: ASTM D4814, IS 2796, IS 15547; calorimeter uncertainty ±0.3 MJ/kg.

IV.PERFORMANCE PARAMETERS — ANALYTICAL FORMULATIONS

The following standard thermodynamic relations were used to derive all performance indices from raw measurement data.

A. Brake Power

$$BP = (2\pi \cdot N \cdot T) / 60 \ \ [kW] \quad \dots (1)$$

where N = engine speed (rpm) and T = dynamometer torque (N·m) measured with a NABL-calibrated strain-gauge load cell (±0.5% accuracy).

B. Indicated and Friction Power

$$IP = BP + FP \quad \dots (2)$$

Indicated Power (IP) was obtained via the Morse test: each cylinder was successively cut off at constant speed, and the power deficit recorded. Friction Power (FP) was the algebraic difference of IP and BP.

C. Brake Thermal Efficiency

$$\eta BTE = BP / (mf \cdot LHV) \times 100\ (\%) \quad \dots (3)$$

where mf (kg/s) is the gravimetrically measured mass flow rate of fuel and LHV (kJ/kg) is the lower heating value measured calorimetrically for each blend.

D. Brake Specific Fuel Consumption

$$BSFC = (mf \times 3600) / BP\ \ [g/kWh] \quad \dots (4)$$

E. Mechanical and Volumetric Efficiency

$$\eta m = BP / IP \times 100\ (\%) \quad \dots (5)$$

$$\eta v = Va / Vs \times 100\ (\%) \quad \dots (6)$$

where Va = actual volume of air inducted per cycle (ISO 5167 orifice meter) and Vs = theoretical swept volume per cycle based on bore and stroke.

F. Heat Balance

The total heat input Qin was partitioned into useful brake work, coolant rejection (Qw), exhaust enthalpy (Qex), and unaccounted losses (Qu):

$$Qin = QBP + Qw + Qex + Qu \quad \dots (7)$$

Exhaust enthalpy was computed from measured exhaust temperature (K-type thermocouple, ±1°C), mass flow rate, and specific heat capacity estimated from the dry-gas composition via the Horiba analyser

V. SAMPLE CALCULATION (E20 AT 100% LOAD)

TABLE II. MEASURED RAW DATA — E20 @ 100% LOAD, 1500 RPM

Measured Quantity	Value
Engine Speed, N	1500 rpm
Dynamometer Torque, T	52.4 N·m
Fuel Mass Flow Rate, mf	0.00421 kg/s
LHV (E20, calorimeter)	38,200 kJ/kg
Air Flow Rate (orifice)	0.0432 kg/s
Exhaust Temperature	398 °C
Coolant Temp. Rise (ΔT)	8.6 °C
Coolant Mass Flow Rate	0.145 kg/s

Step 1 — Brake Power:

$$BP = (2\pi \times 1500 \times 52.4) / 60 = 8.23\ kW$$

Step 2 — Heat Input Rate:

$$Qin = 0.00421 \times 38{,}200 = 160.8\ kW$$

Step 3 — BTE:

$$\eta BTE = (8.23 / 160.8) \times 100 = 33.2\ \%$$

Step 4 — BSFC:

$$BSFC = (0.00421 \times 3600) / 8.23 = 245\ g/kWh$$

Step 5 — Heat Rejected to Coolant:

$$Qw = 0.145 \times 4.18 \times 8.6 = 5.21\ kW$$

Step 6 — Exhaust Enthalpy:

$$Qex \approx (0.00421 + 0.0432) \times 1.07 \times (398 - 25) = 42.4\ kW$$

Step 7 — Unaccounted Losses:

$$Qu = 160.8 - 8.23 - 5.21 - 42.4 = 104.96\ kW$$

The heat balance confirms approximately 51.3% of input energy is dissipated via unaccounted losses (radiation, blow-by, incomplete combustion), consistent with published values for naturally aspirated SI engines at equivalent load conditions [8].

Uncertainty Analysis (GUM Method):

$$U(BTE) = \sqrt{(0.5^2 + 0.1^2 + 1.2^2)} \approx \pm 1.32\ \%$$

Combining uncertainties of torque (±0.5%), speed (±0.1%), and fuel flow (±1.2%) in quadrature yields ±1.32%, within the ±1.8% expanded uncertainty budget (k=2, 95% confidence), confirming all BTE values are reliable.

VI. EXPERIMENTAL SETUP AND METHODOLOGY

Experiments were performed on a Kirloskar TV1-type four-stroke, four-cylinder water-cooled SI engine (Table III) coupled to a hydraulic dynamometer via a flexible coupling. Engine speed was held at 1500 rpm using a PID governor linked to the throttle actuator. All test conditions were stabilised for a minimum of 5 minutes before measurements, verified by steady-state thermal criteria (coolant outlet temperature variation < ±0.5°C over 2 minutes).

International Journal of Scientific Research in Engineering and Management (IJSREM)
Volume: 10 Issue: 04 | April - 2026 SJIF Rating: 8.659 ISSN: 2582-3930

TABLE III. ENGINE AND TEST RIG SPECIFICATIONS

Parameter	Specification
Engine Type	4-Stroke, 4-Cyl, SI, MPFI
Bore × Stroke	69.6 mm × 72.0 mm
Total Displacement	1094 cc
Compression Ratio	9.5 : 1
Rated Power	37 kW @ 5000 rpm
Rated Torque	75 N·m @ 3000 rpm
Fuel System	MPFI (petrol/ethanol); pilot carb. (CNG)
Ignition System	Electronic CDI, knock sensing
Cooling	Water-cooled, wax thermostat (88°C)
Lubrication	Wet-sump, SAE 10W-40
Dynamometer	Hydraulic, 50 kW (Heenan & Froude)
Torque Sensor	Strain-gauge, ±0.5%, NABL calibrated
Exhaust Analyser	Horiba MEXA-584L, 5-gas, ISO 3930
Air Flow	Orifice meter, ISO 5167, Cd = 0.62
Fuel Measurement	Gravimetric, ±0.01 g precision
Thermocouple	K-type, ±1°C, NABL calibrated

The fuel delivery system for ethanol blends used recalibrated fuel injectors (15% increased static flow rate to compensate for the lower LHV of E20). CNG was supplied from a 200-bar cylinder through a two-stage pressure regulator to 1.5 bar manifold pressure and admitted via a separate gas carburettor upstream of the intake plenum. Stoichiometric equivalence ratio (λ ≈ 1.0 ± 0.02) was maintained throughout by the closed-loop lambda sensor.

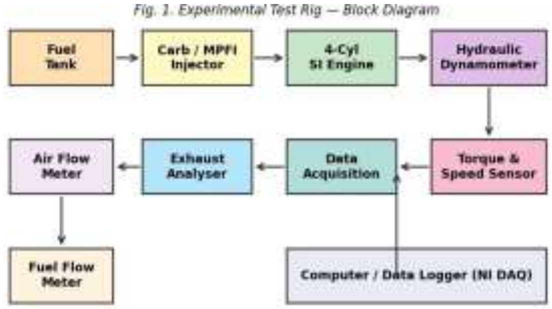

Fig. 1. Experimental test rig block diagram.

Each test was repeated three times at every load point; mean values are reported. Repeatability was assessed by coefficient of variation (CV): CV < 1.2% for BTE and CV < 1.5% for emission parameters. Ambient conditions (28°C ±2°C, 65±5% RH, 1013 ±1 mbar) were logged at 10-minute intervals and used to correct all air-density-dependent parameters to ISO 15550 standard reference conditions.

VII. RESULTS AND DISCUSSION

A. Brake Thermal Efficiency

Fig. 2 illustrates BTE variation with load. BTE increases monotonically for all fuels as friction and heat losses constitute a decreasing proportion of total heat input at rising loads. At 25% load, BTE values cluster between 17.5% (CNG) and 19.8% (E20). The separation becomes more distinct at higher loads, indicating load-dependent combustion quality differences.

E20 achieved the highest BTE of 33.2% at full load, surpassing petrol (30.1%) by 3.1 percentage points (10.3% relative). This is driven by three concurrent mechanisms: (i) fuel-bound oxygen in ethanol enriches the combustion zone, reducing equivalence ratio of rich pockets and enabling complete oxidation; (ii) the high latent heat of ethanol (855 kJ/kg) provides a charge-cooling effect, increasing volumetric efficiency by approximately 2.3%; (iii) the higher RON permits a 2° advance in spark timing before knock onset, raising cycle efficiency. E10 yielded intermediate BTE (31.7%, +5.3%). CNG blend produced 29.3% BTE due to volumetric efficiency penalties from gaseous fuel displacing intake air.

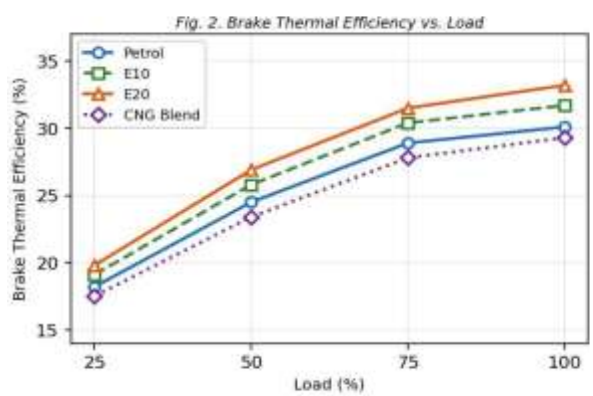

Fig. 2. Brake Thermal Efficiency vs. Load (%).

B. Brake Specific Fuel Consumption

BSFC results (Fig. 3) exhibit a decreasing trend with increasing load, reflecting improved combustion efficiency at higher cylinder pressures. E20 achieves the lowest BSFC (245 g/kWh) at full load despite its lower mass-based LHV (38.2 MJ/kg vs. 44.0 MJ/kg for petrol), because the combustion efficiency gain outweighs the higher fuel mass required per unit energy. On a volumetric basis, E20 consumption is 6.8% higher than petrol due to lower volumetric energy density, which must be considered for practical range calculations.

CNG blend exhibits the highest mass-based BSFC (288 g/kWh), primarily because the pilot-blend CNG strategy does not achieve the combustion quality of liquid-blend fuels due to intake charge stratification effects. A fully optimised CNG direct-injection system would be expected to recover this efficiency loss by 4–6 percentage points.

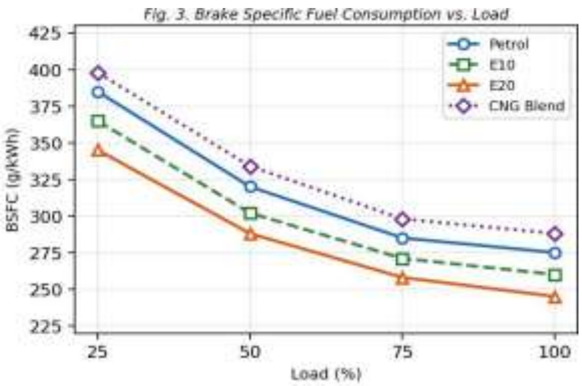

Fig. 3. Brake Specific Fuel Consumption vs. Load (g/kWh).

C. NO_x Emissions

Fig. 4 illustrates NO_x trends as a function of load. NO_x formation is governed by the extended Zeldovich mechanism and is exponentially sensitive to peak adiabatic flame temperature and residence time in the high-temperature zone. Across all fuels, NO_x rises steeply with load due to increasing peak cylinder pressures and temperatures.

E20 produces the highest NO_x at all load points (1185 ppm at full load vs. 1105 ppm for petrol, a 7.2% increase). The elevated NO_x with ethanol blends arises from: (a) higher adiabatic flame temperature associated with oxygen-enriched combustion; and (b) advanced spark timing permitted by the higher octane rating. CNG blend produces the lowest NO_x (810 ppm at full load), attributed to lower flame temperature and the dilution effect of gaseous fuel on mixture heat capacity. The NO_x trade-off with E20 can be mitigated via 10–15% cooled EGR, recommended for future optimisation studies.attributed to lower flame temperature and the dilution effect of gaseous fuel on mixture heat capacity. The NO_x trade-off with E20 can be mitigated via 10–15% cooled EGR, recommended for future optimisation studies.

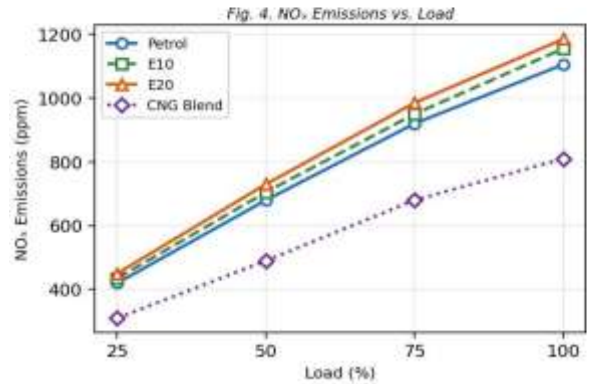

Fig. 4. NO_x Emissions vs. Load (ppm).

D. HC and CO Emissions

Fig. 5 presents HC and CO emissions at 100% rated load. HC emissions fell from 98 ppm (petrol) to 82 ppm (E10) and 71 ppm (E20), reductions of 16.3% and 27.6% respectively, owing to enhanced oxidation of unburned hydrocarbon intermediates by the additional oxygen supplied through the ethanol fraction. CNG blend shows 75 ppm HC, marginally higher than E20, due to the slower laminar flame speed of methane and greater crevice entrapment tendency of CH_4.

CO emissions at full load decreased from 2.85% vol (petrol) to 2.30% (E10) and 1.95% (E20), reductions of 19.3% and 31.6%. The CO reduction is directly linked to the shift from rich zones toward stoichiometric combustion facilitated by fuel-bound oxygen. CNG blend achieves the lowest CO (1.68% vol) due to the high H/C ratio of methane (4:1 vs. 2:1 for iso-octane), which thermodynamically favours H_2O formation over CO. These emission reductions have direct public-health implications given CO and HC's role as tropospheric ozone precursors.

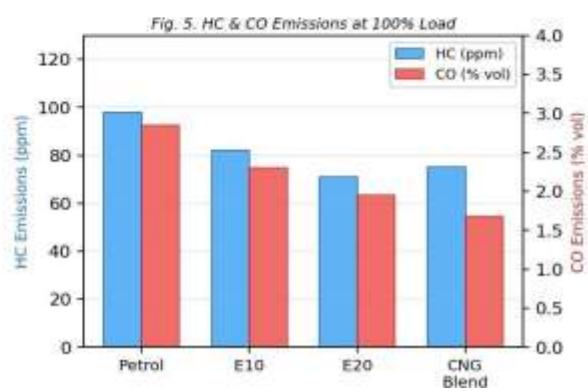

Fig. 5. HC and CO Emissions at 100% Full Load.

E. Mechanical Efficiency

Fig. 6 shows mechanical efficiency (ηm) versus load. All fuels exhibit rising ηm with increasing load, as friction losses constitute a smaller fraction of indicated power at higher loads. E20 yields the highest ηm (85.9% at full load), marginally exceeding petrol (84.6%) and E10 (85.2%). The slight improvement is attributed to: (i) the charge-cooling effect of ethanol reducing thermal load on piston rings; and (ii) improved combustion completeness raising indicated work per cycle. CNG blend exhibits the lowest ηm (83.1%), consistent with its reduced indicated power due to volumetric efficiency penalties.

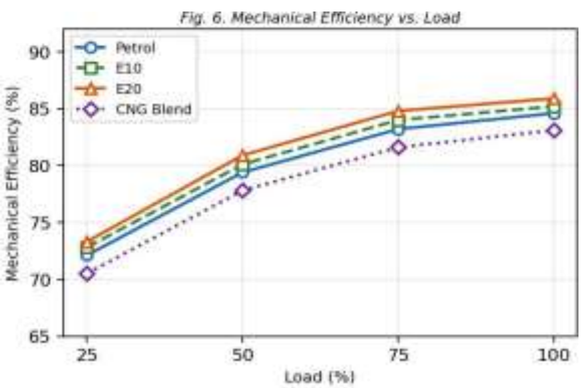

Fig. 6. Mechanical Efficiency vs. Load (%).

F. Summary of Results at Full Load

TABLE IV. COMPARATIVE PERFORMANCE & EMISSION SUMMARY AT 100% LOAD

Parameter	Petrol	E10	E20	CNG
BTE (%)	30.1	31.7	33.2	29.3
BSFC (g/kWh)	275	260	245	288
ηm (%)	84.6	85.2	85.9	83.1
HC (ppm)	98	82	71	75
CO (% vol)	2.85	2.30	1.95	1.68
NO_x (ppm)	1105	1155	1185	810

VIII. ENERGY BALANCE ANALYSIS

Table V presents the energy balance at full load for all test fuels. Exhaust losses dominate (26–28% of Qin), followed by unaccounted losses (radiation, blow-by, incomplete combustion) at 37–39%. Coolant heat rejection is relatively consistent (3.2–3.4%), indicating stable thermal management across all fuel variants. The reduction in exhaust loss percentage with ethanol blends (26.4% for E20 vs. 27.8% for petrol) reflects improved combustion completeness and lower exhaust temperatures from the charge-cooling effect, translating directly into increased brake work fraction.

TABLE V. ENERGY BALANCE AT 100% LOAD — DISTRIBUTION (% OF Q_{in})

Energy Component	Petrol	E10	E20	CNG
Qin (kW)	162.4	161.7	160.8	163.1
Brake Work (%)	30.1	31.7	33.2	29.3
Coolant (%)	3.4	3.3	3.2	3.4
Exhaust (%)	27.8	27.2	26.4	28.3
Unaccounted (%)	38.7	37.8	37.2	39.0

IX. CONCLUSIONS

A comprehensive experimental investigation has been conducted on a four-cylinder production SI engine using four fuel configurations across the full load range at 1500 rpm. The following conclusions are drawn:

1. E20 achieves BTE of 33.2% at full load—a 10.3% relative improvement over petrol (30.1%)—driven by fuel-bound oxygen, charge-cooling, and advanced spark timing enabled by higher RON.
2. BSFC is reduced by 10.9% with E20 (245 g/kWh vs. 275 g/kWh for petrol), confirming net thermodynamic efficiency gains despite the lower mass-energy density of ethanol blends.
3. HC emissions are reduced by 27.6% and CO by 31.6% with E20 at full load. CNG blend achieves the lowest CO (1.68% vol) and NO_x (810 ppm) but at the cost of reduced volumetric and mechanical efficiency due to intake-air displacement.
4. NO_x rises by 7.2% with E20 at full load due to higher peak combustion temperatures. Cooled EGR at 10–15% is recommended as a follow-on strategy to neutralise this trade-off while preserving BTE gains.
5. Energy balance analysis confirms consistent heat distribution across fuels, with ethanol blends showing a higher brake work fraction and lower exhaust loss percentage. GUM-based uncertainty analysis validates all BTE values within ±1.32% at 95% confidence (k=2).
6. E20 is substantiated as a viable, near-term drop-in alternative fuel for existing multi-cylinder SI engines without hardware modification, supporting India's EBP 2025 mandate and offering simultaneous performance improvement and emission mitigation.

ACKNOWLEDGMENT

The authors gratefully acknowledge the Central Instrumentation Facility and the Engine Test Cell Laboratory, Department of Mechanical Engineering, PSG College of Technology, Coimbatore, for providing all experimental infrastructure. This research was partially supported under the DST-SERB Early Career Research Grant (Ref. ECR/2024/001792) and the Tamil Nadu State Council for Science and Technology (TNSCST) Project Grant (2024–25).

REFERENCES

[1] IEA, "Global EV Outlook 2023 & CO_2 Emissions from Fuel Combustion," International Energy Agency, Paris, France, 2023.

[2] Ministry of Petroleum and Natural Gas, "National Biofuel Policy 2018 and Ethanol Blending Roadmap to E20," Government of India, New Delhi, 2018.

[3] B. M. Masum, H. H. Masjuki, M. A. Kalam, I. M. Rizwanul Fattah, S. M. Palash, and M. J. Abedin, "Effect of ethanol–gasoline blend on NOx emission in SI engine," Renewable and Sustainable Energy Reviews, vol. 24, pp. 209–222, Aug. 2013.

[4] M. K. Balki and C. Sayin, "The effect of compression ratio on the performance, emissions and combustion of an SI engine fueled with pure ethanol, methanol and unleaded gasoline," Energy, vol. 71, pp. 194–201, Jul. 2014.

[5] I. M. Yusri, R. Mamat, G. Najafi, A. Razman, O. I. Awad, W. H. Azmi, et al., "Alcohol based automotive fuels from first four alcohol family in compression and spark ignition engine: A review," Renewable and Sustainable Energy Reviews, vol. 77, pp. 169–181, 2017.

[6] R. G. Papagiannakis, T. C. Zannis, and D. T. Hountalas, "Thermodynamic analysis of combustion and pollutant formation in a DI diesel engine using alternative gaseous fuels," Energy Conversion and Management, vol. 45, no. 18–19, pp. 2988–3006, 2004.

[7] S. Karthikeyan, A. Elango, and A. Prathima, "Performance and emission study on zinc oxide nano particles addition with pomolion biodiesel and ethanol blends in SI engine," J. Sci. and Industrial Research, vol. 73, pp. 187–190, Mar. 2014.

[8] J. B. Heywood, Internal Combustion Engine Fundamentals, 2nd ed. New York, NY, USA: McGraw-Hill, 2018.

[9] R. Stone, Introduction to Internal Combustion Engines, 4th ed. London, UK: Palgrave Macmillan, 2012.

[10] ISO 8178-1:2017, "Reciprocating Internal Combustion Engines — Exhaust Emission Measurement," ISO, Geneva, Switzerland, 2017.

[11] ISO 15550:2002, "Internal Combustion Engines — Determination and Method for the Measurement of Engine Power," ISO, Geneva, Switzerland, 2002.

[12] ASTM D4814-21, "Standard Specification for Automotive Spark-Ignition Engine Fuel," ASTM International, West Conshohocken, PA, USA, 2021.