

Article

Performance, Emission, Energy, and Exergy Assessment of Gasoline–1-Decanol, Gasoline–DIPE, and JP-8 Fuels Toward Sustainable Two-Stroke UAV Propulsion

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Abstract

This study experimentally investigates the performance, emission, energy, and exergy characteristics of six fuels in a two-cylinder, air-cooled, two-stroke unmanned aerial vehicle (UAV) engine, using neat gasoline (G100) as the reference fuel. The test matrix comprised G100, neat JP-8, gasoline–1-decanol blends containing 10 and 20 vol.% decanol (G90D10 and G80D20), and gasoline–diisopropyl ether blends containing 10 and 20 vol.% DIPE (G90E10 and G80E20). Experiments were conducted at shaft speeds of 3250, 3750, 4500, 5250, and 6250 rpm under matched speed–load conditions. Fuel consumption, exhaust-gas temperature, excess-air ratio, gaseous emissions, and first- and second-law thermodynamic performance were evaluated. The inclusion of G100 enabled the effects of the oxygenated additives to be quantified directly. At 5250 rpm, G90E10 reduced BSFC by 0.73% relative to G100 while increasing energy and exergy efficiencies by 1.69% and 1.61%, respectively. G80E20 produced larger reductions in brake-specific CO and HC emissions, 6.63% and 14.58%, respectively, and increased energy and exergy efficiencies by 2.43% and 2.27%, respectively, although both DIPE blends increased NO_x. In contrast, increasing the decanol fraction increased BSFC and reduced thermodynamic efficiency relative to G100 despite moderate reductions in CO. JP-8 showed the strongest operating-point dependence; at 5250 rpm its BSFC was 9.52% lower than G100 and its energy and exergy efficiencies were 12.33% and 12.84% higher, respectively, whereas its advantage diminished markedly at the highest speed. Overall, the results demonstrate that oxygenate type and concentration produce distinct trade-offs between fuel utilization and gaseous emissions, while the suitability of JP-8 for gasoline-oriented two-stroke UAV propulsion is strongly dependent on operating conditions. From an operational-sustainability perspective, the results show that improvements in fuel utilization and incomplete-combustion emissions are strongly fuel and operating-point dependent. However, fuel production pathways, life-cycle greenhouse-gas emissions, durability, economic impacts, and mission-level environmental effects were outside the scope of this study and should be addressed before broader sustainability conclusions are drawn.



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conditions of the [Creative Commons](https://creativecommons.org/licenses/by/4.0/)[Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.**Keywords:** two-stroke UAV engine; JP-8; diisopropyl ether; 1-decanol; energy analysis; exergy analysis; sustainable UAV engine

1. Introduction

Unmanned aerial vehicles (UAVs) have become increasingly important in military surveillance, border security, environmental monitoring, precision agriculture, mapping, disaster assessment, cargo delivery, and emergency-response operations. The propulsion requirements of these platforms vary considerably depending on payload, mission duration, cruise altitude, flight speed, and take-off conditions. Electric propulsion systems are widely used in small UAVs because of their low noise, mechanical simplicity, and rapid control response. Nevertheless, battery-powered systems remain constrained by their relatively limited specific energy, especially in applications requiring long endurance, high payload capacity, or continuous operation. Internal combustion engines therefore continue to be an important propulsion option for small and medium UAVs because liquid fuels provide high gravimetric energy density, rapid refueling, and comparatively long operating endurance [1–3].

Among reciprocating propulsion systems, two-stroke engines offer several advantages for UAV applications, including high power-to-weight ratio, compact construction, low component count, and the ability to generate a power stroke during every crankshaft revolution. These characteristics enable a relatively high shaft output to be obtained from a lightweight engine package [4,5]. However, the conventional two-stroke cycle also presents significant drawbacks. The simultaneous opening of the intake and exhaust ports during scavenging may allow a portion of the fresh air–fuel charge to escape directly into the exhaust. Charge short circuiting, residual-gas retention, wall quenching, lubricant combustion, and incomplete oxidation can consequently increase unburned hydrocarbon (HC) and carbon monoxide (CO) emissions while reducing fuel-conversion efficiency [6–8]. The severity of these effects depends on engine speed, scavenging efficiency, mixture preparation, fuel volatility, and combustion duration.

Previous investigations of small two-stroke engines have demonstrated that fuel composition strongly affects exhaust temperature, delivery ratio, scavenging behavior, combustion stability, and pollutant formation. Ghazikhani et al. [6] reported that gasoline–ethanol blends changed the exhaust-gas temperature, delivery ratio, performance, and emission characteristics of a two-stroke spark-ignition engine. Their subsequent second-law assessment indicated that alcohol addition also influenced combustion irreversibility and exergy utilization [7]. Magnusson et al. [8] showed that the use of ethanol and ethanol–gasoline blends in a two-stroke engine affected not only regulated emissions but also oxygenated organic compounds such as aldehydes and ketones. These studies confirm that fuel-bound oxygen can promote oxidation; however, the resulting emission trends remain dependent on mixture strength, evaporation, scavenging losses, and combustion temperature.

The increasing use of UAVs has also intensified interest in safer and logistically compatible fuels. Gasoline provides favorable evaporation and ignition characteristics for spark-ignition engines, but its high volatility and low flash point create disadvantages during transportation, storage, and operation under military or remote-field conditions. Kerosene-based aviation fuels have higher flash points, lower volatility, and wider logistical availability within military fuel infrastructures. Consequently, substantial research has been directed toward converting gasoline UAV engines to aviation kerosene or other heavy fuels [9–14]. The main technical difficulty is that fuels such as JP-8 evaporate more slowly than gasoline and may produce wall wetting, heterogeneous mixtures, delayed flame development, poor cold-start performance, and unstable combustion when used in an engine originally designed for a volatile gasoline fuel.

Studies of aviation heavy-fuel engines have shown that the fuel delivery system and combustion chamber configuration must be adapted to compensate for the low volatility of kerosene-based fuels. Qiao et al. [9] compared different injection arrangements for a

two-stroke heavy-fuel light aero-engine and demonstrated the importance of fuel injection strategy. Wang et al. [10] optimized the injection control system of a two-stroke UAV aero-engine, while Wang et al. [11] numerically converted a port-fuel-injected gasoline engine into a direct-injection aviation-kerosene engine. Their results showed that injection timing, spray targeting, ignition location, scavenging flow, and combustion chamber geometry play decisive roles in mixture formation and thermal efficiency. Further optimization research confirmed that compression ratio and injection parameters must be selected together to increase power and efficiency without causing excessive knock [12]. Air-assisted injection has also been proposed to improve the atomization and combustion of low-volatility aviation kerosene in two-stroke spark-ignition engines [13,14].

Experimental studies have further demonstrated that heavy-fuel operation in UAV engines is feasible, although its effectiveness varies with operating conditions. Liu et al. [15] compared gasoline and RP-3 aviation kerosene in a two-stroke spark-ignition UAV engine and observed that the combustion and cyclic behavior of kerosene were particularly sensitive to engine load. A related investigation showed that the knock tendency of RP-3 depended on the mixture formation and combustion conditions [16]. Hooper [17] examined the cold-starting behavior of a spark-ignition UAV engine supplied with low-volatility fuel, whereas subsequent studies addressed the effects of trapped compression ratio and fuel preparation on heavy-fuel operation [18,19]. These findings indicate that heavy fuel cannot be assessed only by its lower heating value; volatility, ignition quality, lubricant compatibility, compression ratio, and thermal state must also be considered.

JP-8 is particularly relevant to military UAV applications because it is a widely used kerosene-based aviation fuel. Duddy et al. [20] converted a spark-ignited aircraft engine to JP-8 and showed that acceptable engine operation could be obtained after appropriate modifications. For a two-stroke engine, fuel lubricity is an additional concern because the lubricant is normally supplied with the fuel. Lee et al. [21] investigated methods of increasing JP-8 lubricity for midsized two-stroke UAV engines and demonstrated the importance of selecting a compatible lubricant or lubricity-enhancing component. JP-8 therefore offers advantages in handling safety and single-fuel logistics, but its implementation in a gasoline-designed two-stroke UAV engine may involve compromises in evaporation, cold starting, fuel consumption, exhaust temperature, and combustion completeness.

A second strategy for improving two-stroke engine operation involves the use of oxygenated components in gasoline. Recent UAV engine studies have shown that alcohol selection and molecular structure can significantly affect engine response. Özer et al. [22] investigated gasoline–octanol and gasoline–hexanol blends in a two-stroke UAV engine and reported distinct effects on specific fuel consumption, emissions, energy losses, exergy destruction, and exergy efficiency. A study using JP-5 and hydroxy gas similarly demonstrated that performance improvement should be evaluated together with energy, exergy, environmental, and economic indicators [23]. More recently, Tunçer et al. [24] performed experimental and thermodynamic analyses of a two-stroke UAV engine operated with gasoline–methanol blends. Together, these investigations demonstrate that oxygenated fuels can improve certain combustion and emission characteristics, but the magnitude and direction of the response depend on oxygen content, chain length, volatility, viscosity, and heating value.

Diisopropyl ether (DIPE) is an oxygenated ether that has attracted attention as a gasoline component because of its favorable volatility, oxygen content, and mixture formation characteristics. In a multi-point fuel-injection spark-ignition engine, Dhamodaran et al. [25] found that DIPE–gasoline blends increased brake thermal efficiency, in-cylinder pressure, and heat-release rate while reducing HC and CO emissions. However, the enhanced oxidation process and higher combustion temperature could also increase CO₂

and NO_x emissions. This response highlights the trade-off between improved combustion completeness and temperature-dependent pollutant formation.

The effects of DIPE are also strongly linked to engine operating conditions and blend fraction. Sivakumar et al. [26] tested DIPE concentrations between 10 and 25 vol.% in a variable-compression-ratio spark-ignition engine and showed that engine load, compression ratio, and DIPE concentration interact in determining fuel consumption and emissions. Sathyanarayanan et al. [27] similarly used response-surface methodology to predict and optimize the performance of a gasoline engine supplied with DIPE–gasoline blends. The resulting optimum condition represented a compromise among brake thermal efficiency, specific fuel consumption, and regulated emissions. In homogeneous-charge compression-ignition operation, DIPE altered ignition timing, operating range, combustion phasing, and incomplete-combustion emissions [28]. Comparative emission studies with different higher-alcohol and ether oxygenates have also indicated that the chemical class of an oxygenated additive influences CO, HC, CO₂, and NO_x formation differently [29]. Accordingly, results obtained from four-stroke or homogeneous-charge engines cannot be transferred directly to a carbureted two-stroke UAV engine.

Higher alcohols provide a different combination of advantages and limitations. As the carbon-chain length increases, alcohol fuels generally exhibit improved miscibility with hydrocarbon fuels and higher energy density compared with methanol and ethanol. At the same time, their viscosity and boiling temperature increase and their volatility decreases [30–33]. These changes can reduce evaporative losses and improve storage safety, but they may also delay mixture preparation in an engine with limited time for fuel vaporization. Research with n-pentanol–gasoline blends has demonstrated that fuel-bound oxygen can increase brake thermal efficiency and modify combustion pressure and heat release, while the lower heating value and evaporation behavior affect fuel consumption and exhaust emissions [30]. Butanol–gasoline studies have similarly shown that blend ratios near 10–20 vol.% can be used with limited engine modification, whereas high alcohol fractions can produce more pronounced changes in combustion duration, power, and specific fuel consumption [31].

1-Decanol is a long-chain alcohol with substantially lower volatility and higher viscosity than the short-chain alcohols commonly used in gasoline engines. Although direct studies of gasoline–decanol blends in two-stroke UAV engines are extremely limited, research in compression-ignition engines provides useful evidence regarding its mixture and combustion behavior. Reviews of higher-alcohol fuels emphasize that chain length, fuel-bound oxygen, cetane or octane behavior, latent heat of vaporization, and viscosity jointly govern ignition, efficiency, and emissions [32,33]. Rajasekaran et al. [34] investigated the combined effects of 1-decanol, injection pressure, and exhaust-gas recirculation and concluded that its performance depended strongly on mixture preparation and combustion control. Ashok et al. [35] comparatively evaluated hexanol and decanol as oxygenated additives and showed that the longer-chain alcohol could affect efficiency and fuel consumption differently from a shorter-chain alcohol. Nanthagopal et al. [36] also reported that decanol addition changed fuel properties, combustion, and emission behavior in a ternary blend. Long-chain alcohol experiments have confirmed that 1-decanol can provide favorable combustion efficiency under suitable injection and thermal conditions, but its viscosity and low volatility remain important limitations [37].

The available literature indicates that DIPE, 1-decanol, and kerosene-type aviation fuels influence two-stroke engine operation through fundamentally different physicochemical mechanisms. DIPE is characterized by high volatility and fuel-bound oxygen, which can promote mixture preparation and oxidation, whereas 1-decanol has a substantially higher boiling temperature and viscosity that may limit vaporization during the short prepara-

tion period of a two-stroke engine. JP-8 presents a different challenge because its broad distillation range and low volatility can restrict mixture formation in a gasoline-oriented spark-ignition engine. Consequently, comparisons among these fuels should account not only for conventional performance and emissions but also for differences in fuel-energy utilization and useful work potential.

Despite the available studies on oxygenated gasoline blends and aviation heavy fuels, the targeted literature review did not identify an experimental study that directly compares neat gasoline, gasoline–1-decanol blends, gasoline–DIPE blends, and neat JP-8 within the same two-stroke UAV engine while simultaneously evaluating performance, gaseous emissions, energy, and exergy characteristics over the same operating speed range. Previous investigations have generally focused on a single oxygenate family, a single aviation fuel, or engines specifically modified for heavy-fuel operation. In addition, the absence of a neat-gasoline reference in many comparative fuel studies limits the ability to quantify whether observed changes arise from the oxygenated component itself or from differences among alternative fuels. The inclusion of G100 in the present study therefore provides a common baseline against which the effects of decanol, DIPE, and JP-8 can be directly quantified.

Accordingly, the present study experimentally evaluates six fuels: neat gasoline (G100), gasoline–1-decanol blends containing 10 and 20 vol.% decanol (G90D10 and G80D20), gasoline–DIPE blends containing 10 and 20 vol.% DIPE (G90E10 and G80E20), and neat JP-8. All fuels are evaluated in the same two-cylinder, air-cooled, two-stroke UAV engine at shaft speeds of 3250, 3750, 4500, 5250, and 6250 rpm. Fuel consumption, brake output, exhaust-gas temperature, excess-air ratio, and gaseous emissions are examined together with first- and second-law thermodynamic analyses. The principal objective is to quantify the changes produced by the oxygenated blends and JP-8 relative to G100 and to determine how fuel volatility, viscosity, oxygen content, and heating value influence the trade-offs among fuel consumption, emissions, energy efficiency, and exergy efficiency. This integrated comparison is intended to clarify both the potential advantages and the operating limitations of the investigated fuels for two-stroke UAV propulsion. Energy and exergy analyses have also been applied to evaluate the thermodynamic performance and emissions of gasoline engines operated with different fuels [38].

2. Materials and Methods

The experimental platform used in the present investigation was based on the two-stroke UAV engine test system previously developed by the authors [22]. The engine, loading mechanism, fuel consumption measurement arrangement, temperature sensors, and exhaust-gas analysis system were retained, whereas the fuel matrix was expanded to include neat gasoline (G100), gasoline–1-decanol blends, gasoline–diisopropyl ether (DIPE) blends, and neat JP-8. The methodology below therefore describes the six fuels evaluated under the common experimental procedure adopted in the present study.

2.1. Experimental Engine and Test Facility

Experiments were conducted on a purpose-built test bench equipped with a two-cylinder, air-cooled, spark-ignition, two-stroke engine intended for UAV propulsion (Baykuş, Erin Motor, İstanbul, Türkiye) (Figure 1). The engine was connected to an adjustable loading system. The reaction force produced by the loading unit was measured using an S-type load cell, and shaft speed was monitored continuously using a laser tachometer/rotary encoder. The experimental arrangement also incorporated a volumetric fuel consumption unit, a K-type exhaust thermocouple, a thermal imaging camera, an exhaust sampling line, and a BOSCH BEA 060 (Robert Bosch GmbH, Gerlingen, Germany) exhaust-gas analyzer. Fuel consumption was measured using a volumetric time-based

measurement system. The system consisted of a calibrated graduated fuel vessel connected to the engine fuel-supply arrangement and a digital chronometer. During each stabilized operating condition, the time required for the engine to consume a fixed fuel volume of 20 mL was recorded. The volumetric fuel flow rate was subsequently calculated from the consumed volume and elapsed time, and the corresponding mass flow rate was obtained using the density of the relevant test fuel. This arrangement allowed the same measurement principle to be applied consistently to all six fuels.

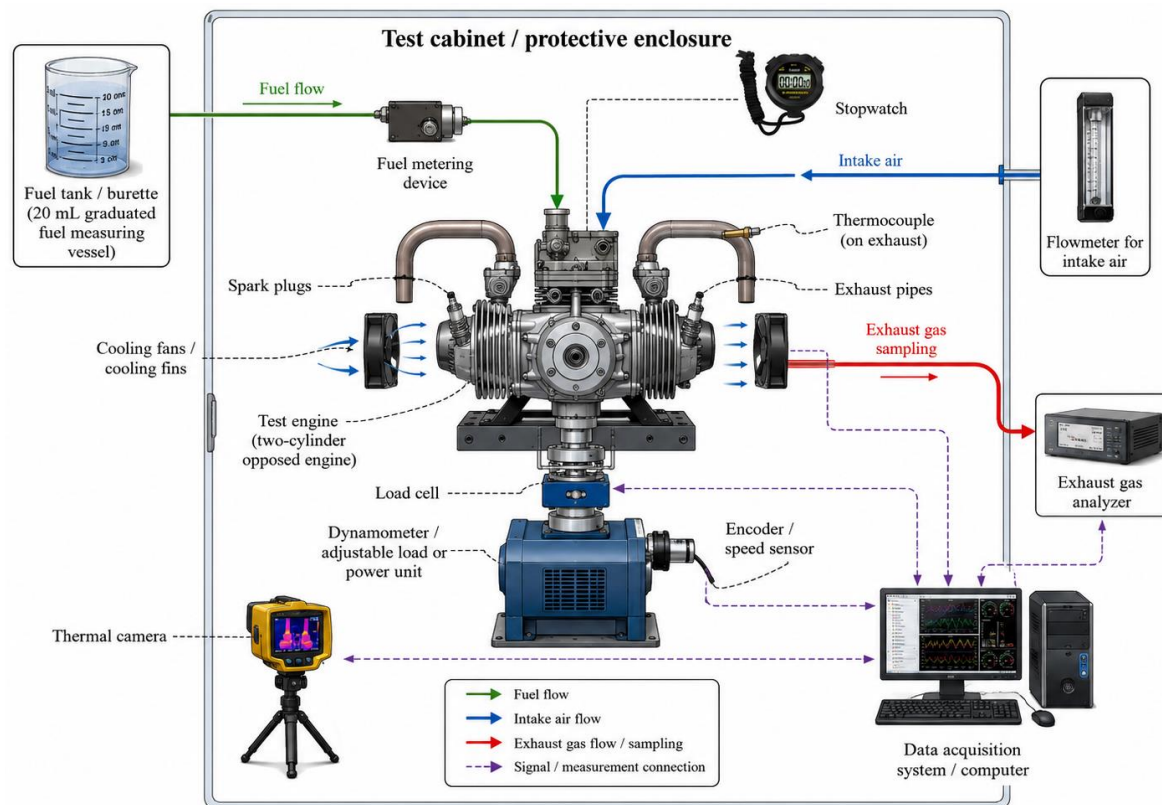


Figure 1. Schematic representation of the experimental test setup.

The main specifications of the engine are presented in Table 1. The engine had a total displacement of 132 cm³, a bore of 48 mm, a stroke of 36.5 mm, and a compression ratio of 10.4:1. Its rated maximum power was 12 hp at 8000 rpm, while the nominal idle speed was approximately 1800 rpm [22].

Table 1. Technical specifications of the two-stroke UAV engine.

Parameter	Unit	Specification
Manufacturer and model	–	Erin Motor–Baykuş
Engine configuration	–	Two-cylinder, two-stroke, spark-ignition
Cooling system	–	Air-cooled
Bore	mm	48
Stroke	mm	36.5
Total displacement	cm ³	132
Compression ratio	–	10.4:1
Maximum power	hp	12 at 8000 rpm
Nominal idle speed	rpm	1800

The engine was tested at shaft speeds of 3250, 3750, 4500, 5250, and 6250 rpm. These points were selected to cover low-, intermediate-, and relatively high-speed regions within the stable operating envelope of the engine without approaching either idle instability or the maximum power limit.

2.2. Test Fuels and Blend Preparation

Six fuels were investigated. Neat gasoline was tested as the reference fuel and denoted G100. The same gasoline was separately blended with 1-decanol and DIPE at oxygenate volume fractions of 10% and 20%, producing G90D10, G80D20, G90E10, and G80E20, respectively. JP-8 was tested in neat form without gasoline addition. All six fuels were evaluated using the same engine configuration and experimental procedure. The fuel compositions and abbreviations used are listed in Table 2.

Table 2. Volumetric compositions and abbreviations of the test fuels.

Fuel No.	Fuel Composition	Abbreviation
1	90 vol.% gasoline + 10 vol.% 1-decanol	G90D10
2	80 vol.% gasoline + 20 vol.% 1-decanol	G80D20
3	90 vol.% gasoline + 10 vol.% DIPE	G90E10
4	80 vol.% gasoline + 20 vol.% DIPE	G80E20
5	100 vol.% JP-8	JP8
6	100 vol.% Gasoline	G100

The oxygenated blends were prepared immediately before each experimental series. Gasoline and the relevant oxygenate were measured separately using calibrated volumetric glassware and transferred to a clean, dry, sealed vessel. For a nominal 1 L batch, G90D10 and G90E10 were prepared using 900 mL gasoline and 100 mL oxygenate, whereas G80D20 and G80E20 were prepared using 800 mL gasoline and 200 mL oxygenate. The containers were closed immediately and mixed until no visible stratification was present. This procedure was particularly important for DIPE because of its low boiling temperature and high volatility [39]. JP-8 was tested in neat form (100 vol.% JP-8) and was supplied directly to the engine without gasoline blending. Its inclusion in the experimental matrix was intended to assess the feasibility and operating characteristics of a kerosene-based aviation fuel in the investigated spark-ignition UAV engine; it was not considered or represented as a gasoline specification fuel.

The volumetric fraction of component *i* was defined as Equation (1):

$$X_{v,i} = \frac{V_i}{V_{\text{total}}} \quad (1)$$

where V_i is the volume of component V_i and V_{total} is the final blend volume.

The prepared blends were kept in sealed containers away from direct heat and ignition sources. Before each test, the blend was visually inspected for phase separation, suspended matter, or contamination. The fuel tank, fuel line, and carburetor chamber were drained when changing fuels. A small quantity of the next fuel was passed through the supply system before the engine was restarted, thereby minimizing carryover from the preceding test.

2.3. Fuel Properties and Calculation of Blend Properties

The properties adopted for the constituent fuels are summarized in Table 3. Gasoline data were taken from the authors' published characterization of the fuel used with the

same test platform [22]. The gasoline used in the present study was a RON 95 commercial gasoline with the properties reported in Table 3. Therefore, the quantitative results obtained for G100 and the gasoline-based blends are specific to the investigated gasoline batch and should not be interpreted as universally applicable to all commercial gasoline formulations. Nevertheless, the use of the same gasoline batch as the common base fuel for G100, G90D10, G80D20, G90E10, and G80E20 enables the effects of 1-decanol and DIPE addition to be evaluated consistently within the present experimental matrix. The molecular formulae, phase-change data, and standard combustion enthalpies of DIPE and 1-decanol were obtained from the current NIST Chemistry WebBook [39,40]. Density, boiling point, and flash-point data were checked against current Merck/Millipore product and safety documentation [41,42]. JP-8 limits were taken from the official MIL-DTL-83133L, Amendment 1 specification [43]. Recent engine research also confirms that the comparatively high viscosity and low volatility of 1-decanol must be considered when interpreting mixture formation and combustion behavior [34].

Table 3. Thermophysical and chemical properties of the constituent fuels used to prepare the test fuels.

Property	Gasoline	1-Decanol	DIPE	JP-8
Chemical nature/formula	Multicomponent gasoline [22]	$C_{10}H_{22}O$ [40]	$C_6H_{14}O$ [39]	Kerosene-type mixture [43]
Density	747.6 kg/m ³ at 15 °C [22]	830.6 kg/m ³ at 20 °C [42]	725 kg/m ³ at 25 °C [41]	775–840 kg/m ³ at 15 °C [43]
Kinematic Viscosity	0.61 mm ² /s at 40 °C [22]	14.5 mm ² /s at 25 °C; [40,42]	0.43 mm ² /s at 25 °C; [39,41]	≤8.0 mm ² /s at −20 °C [43]
Lower heating value	43.40 MJ/kg	41.8 MJ/kg	39.3 MJ/kg	≥42.7 MJ/kg [43]
Oxygen content	0 wt.% [22]	10.11 wt.%	15.66 wt.%	≈0 wt.%
Stoichiometric air–fuel ratio	≈14.70	13.07	12.14	≈14.6
Boiling characteristic	Batch-specific ASTM D86 range	229–232 °C [40,42]	68–69 °C [39,41]	T ₁₀ ≤ 205 °C; FBP ≤ 300 °C [43]
Flash point	Batch-specific	≈95 °C [42]	≈−28 °C [41]	≥38 °C [43]
Ignition-quality descriptor	RON 95 [22]	Blend RON not inferred	Blend RON not inferred	Batch-specific calculated cetane index [43]

The density values reported in Table 3 correspond to the reference temperatures specified in the respective sources. Since fuel density is temperature dependent, values reported at different temperatures should not be interpreted as directly comparable properties. The viscosity values listed in Table 3 were obtained at different reference temperatures and were therefore used only for qualitative interpretation. Flash point, distillation characteristics, and octane or cetane indices are non-linear mixture properties; consequently, these parameters were not estimated for the prepared blends by simple volumetric interpolation. Since fuel consumption was determined volumetrically, fuel density was used to convert the measured volumetric flow rate to mass flow rate. Accordingly, the density value adopted in the calculation directly affects BSFC and the subsequent energy and exergy calculations.

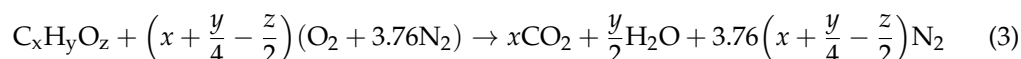
The lower heating values of pure 1-decanol and DIPE were obtained from the standard liquid-phase combustion enthalpies reported by NIST. The higher-to-lower heating value

conversion was performed by subtracting the enthalpy required to retain the combustion-generated water in the vapor phase as Equation (2):

$$LHV_i = \frac{|\Delta_c H_i^\circ| - n_{H_2O} \Delta h_{vap,H_2O}}{M_i} \quad (2)$$

where $\Delta_c H_i^\circ$ is the standard molar combustion enthalpy, n_{H_2O} is the number of moles of water formed per mole of fuel, $\Delta h_{vap,H_2O}$ is the enthalpy of vaporization of water at the reference condition, and M_i is the molar mass of the fuel.

For an oxygenated compound represented by $C_xH_yO_z$, the stoichiometric combustion relation was expressed as Equation (3):



The constituent mass fraction in a volumetrically prepared blend was calculated from Equation (4):

$$X_{m,i} = \frac{X_{v,i}\rho_i}{\sum_j X_{v,j}\rho_j} \quad (4)$$

Assuming additive liquid volumes, the estimated blend density was expressed as Equation (5):

$$\rho_{blend} = \sum_i X_{v,i}\rho_i \quad (5)$$

The blend lower heating value, oxygen mass fraction, and stoichiometric air requirement were calculated on a mass fraction basis:

$$LHV_{blend} = \sum_i X_{m,i}LHV_i \quad (6)$$

$$O_{blend} = \sum_i X_{m,i}O_i \quad (7)$$

$$AFR_{st,blend} = \sum_i X_{m,i}AFR_{st,i} \quad (8)$$

2.4. Experimental Procedure

Before each fuel series, the engine was inspected for leakage, the fuel measurement line was purged, and the load-cell and speed measurement signals were checked. The engine was then operated with the fuel assigned to that series until stable thermal conditions were obtained. Measurements were initiated only after shaft speed, measured reaction force, exhaust-gas temperature, and exhaust concentrations showed no systematic drift.

For each fuel, data were collected at 3250, 3750, 4500, 5250, and 6250 rpm. Engine speed was adjusted using the loading system and maintained at the prescribed value during the measurement interval. The fuel metering setting recorded for a given fuel was retained throughout that fuel's test sequence. Because the fuels differed considerably in volatility and stoichiometric air demand, mixture strength was evaluated from the measured exhaust composition and lambda value rather than being assumed identical.

The compression ratio, ignition system, spark plug, intake geometry, exhaust system, cooling arrangement, and moment-arm geometry were kept unchanged throughout the comparative study. The engine was allowed to reach the same stability criterion before the next data set was recorded.

At each operating point, shaft speed, reaction force, fuel consumption time, exhaust-gas temperature, engine surface temperature, and exhaust concentrations were recorded after stable operation had been achieved. The arithmetic mean of three consecutive readings

was used in the calculations. The reaction force measured by the load cell was converted to torque using the effective moment-arm length of the test system:

$$T = F L \quad (9)$$

where T is the torque in N·m, F is the measured reaction force in N, and L is the effective moment-arm length in m [22].

Brake power was calculated as Equation (10):

$$P_b = \frac{2\pi NT}{60000} \quad (10)$$

where P_b is in kW and N is the shaft speed in rpm.

Fuel consumption was determined using the volumetric measurement unit shown in Figure 1. After stable engine operation had been established at each test point, the elapsed time required to consume a fixed volume of 20 mL was measured using a digital chronometer. The same nominal measurement volume and procedure were used for G100, G90D10, G80D20, G90E10, G80E20, and JP-8. The volumetric fuel flow rate was calculated from the measured volume and consumption time, and the mass flow rate was subsequently obtained using the density of the corresponding fuel. The volumetric and mass flow rates were correspondingly calculated as Equations (11) and (12):

$$\dot{V}_f = \frac{V_f}{\Delta t} \quad (11)$$

$$\dot{m}_f = \rho_f \dot{V}_f \quad (12)$$

where V_f is the consumed fuel volume, Δt is the measured fuel consumption time, and ρ_f is the density of the relevant test fuel. For G90D10, G80D20, G90E10, and G80E20, the blend densities were estimated from the constituent densities listed in Table 3 using the volumetric fractions given in Table 2. For JP-8, the batch-certificate or experimentally measured density was used rather than the midpoint of the specification range.

Brake-specific fuel consumption was determined from Equation (13):

$$BSFC = \frac{3600\dot{m}_f}{P_b} \quad (13)$$

where BSFC is expressed in kg/kWh; multiplication by 1000 gives g/kWh. Since the test fuels had different densities, fuel consumption comparisons were conducted on a mass basis rather than solely from volumetric consumption.

Exhaust-gas temperature was measured using a K-type thermocouple installed in the exhaust line at a fixed distance and insertion depth relative to the exhaust outlet. The same installation geometry was maintained for all fuels. Engine surface temperature was monitored using a thermal imaging camera with a nominal measurement range from -10 °C to 1000 °C and an accuracy of ± 2 °C [22]. Camera position, viewing angle, and emissivity setting were kept constant during the tests.

A BOSCH BEA 060 gas analyzer was connected to the exhaust sampling line to measure CO, CO₂, HC, O₂, and NO_x concentrations. The analyzer was zeroed and calibrated before each test series according to the manufacturer's procedure. The sampling probe was positioned to minimize ambient air entrainment, which is especially important in small two-stroke exhaust systems. Concentration readings were recorded after the analyzer signal became stable.

The excess-air ratio was defined as:

$$\lambda = (A/F)_{\text{actual}} / (A/F)_{\text{st}}$$

A value below unity represents fuel-rich operation, whereas a value above unity represents lean operation. The fuel-specific stoichiometric air–fuel ratios calculated from the constituent compositions were considered when interpreting lambda. The excess-air ratio was calculated using the actual air–fuel ratio and the fuel-specific stoichiometric air–fuel ratio determined from the constituent compositions.

Because the fuels differed in viscosity, volatility, density, and stoichiometric air requirement, identical mixture strength was neither assumed nor imposed. The resulting lambda value was recorded at each stabilized operating point. As shown in Table 4, lambda ranged from 0.799 to 0.933, indicating globally fuel-rich operation throughout the experimental matrix.

Table 4. Measured excess-air-ratio (λ) values for all fuels and operating points.

Fuel	3250	3750	4500	5250	6250 rpm
G100	0.823	0.833	0.812	0.801	0.903
JP-8	0.803	0.812	0.801	0.799	0.888
G90D10	0.832	0.855	0.822	0.839	0.888
G80D20	0.841	0.859	0.834	0.846	0.898
G90E10	0.832	0.841	0.832	0.826	0.933
G80E20	0.813	0.823	0.812	0.802	0.892

The theoretical adiabatic flame temperature was calculated for each fuel and operating point using a constant-pressure equilibrium approach. The measured excess-air ratio at each operating condition was converted to the equivalence ratio using $\phi = \frac{1}{\lambda}$, thereby accounting for the differences in mixture strength among the test fuels. For neat gasoline (G100), a representative RON95 ternary surrogate consisting of 13.7 mol.% n-heptane, 42.8 mol.% iso-octane, and 43.5 mol.% toluene was adopted from Pera and Knop [44]. For JP-8, the representative average empirical formula $C_{11}H_{21}$ was adopted following the kerosene/JP-8 surrogate literature summarized by Dagaut and Cathonnet [45]. The molecular formulas $C_{10}H_{22}O$ for 1-decanol and $C_6H_{14}O$ for DIPE were used directly. The blend surrogate compositions were determined from the volumetric blend ratios and the constituent densities. The reactants were initialized at the representative experimental ambient conditions of 289.95 K and 99.9 kPa. The equilibrium product composition was obtained by Gibbs-free-energy minimization considering CO_2 , CO , H_2O , H_2 , O_2 , and N_2 . The adiabatic flame temperature was obtained from the adiabatic enthalpy balance, and the resulting values were used only as theoretical thermochemical indicators rather than direct measurements of peak in-cylinder temperature.

Representative surrogate formulations used in the calculation:

$$\text{G100 surrogate} = 0.137 \text{ n} - C_7H_{16} + 0.428 \text{ i} - C_8H_{18} + 0.435 C_7H_8 \quad (14)$$

$$\text{JP} - 8 \approx C_{11}H_{21}$$

Adiabatic enthalpy condition:

$$\sum n_r h_r(T_{\text{in}}) = \sum n_p h_p(T_{\text{ad}}) \quad (15)$$

where T_{in} is the initial reactant temperature and T_{ad} is the calculated adiabatic flame temperature.

CO and CO₂ were initially measured in vol.%, whereas HC and NO_x were measured in ppm vol. The measured concentrations were subsequently converted into brake-specific emissions and reported in g/kWh. The combined uncertainty of a calculated parameter was determined using the root-sum-square propagation method. For a dependent variable $R = f(x_1, x_2, \dots, x_n)$, the absolute uncertainty was calculated as Equation (16):

$$w_R = \left[\sum_{i=1}^n \left(\frac{\partial R}{\partial x_i} w_{x_i} \right)^2 \right]^{1/2} \quad (16)$$

where w_{x_i} represents the uncertainty of each independent measurement. Relative uncertainty was expressed as Equation (17):

$$U_R = \frac{w_R}{R} \times 100 \quad (17)$$

The uncertainties of torque, brake power, fuel mass flow rate, BSFC, energy efficiency, and exergy efficiency were determined using the root-sum-square propagation method. The measurement ranges, resolutions, and instrument accuracies used in the experimental assessment are summarized in Table 5.

Table 5. Measurement instruments, resolutions, and stated uncertainties.

Parameter	Instrument/Method	Resolution or Sensitivity	Accuracy/Tolerance
Shaft speed	Laser tachometer	1 rpm	±0.5% [22]
Reaction force	S-type load cell	±0.01 kg	±0.1% [22]
Fuel volume	Graduated fuel vessel	±0.1 mL	±0.1 mL [22]
Fuel consumption time	Digital chronometer	±0.1 s	±0.1 s [22]
Engine surface temperature	Thermal camera	Device resolution	±2 °C [22]
Exhaust-gas temperature	K-type thermocouple	0.1 °C	According to calibration certificate
CO	BOSCH BEA 060	0.001 vol.%	±0.005 vol.% [22]
CO ₂	BOSCH BEA 060	0.01 vol.%	±0.2 vol.% [22]
HC	BOSCH BEA 060	1 ppm vol.	±12 ppm [22]
O ₂	BOSCH BEA 060	0.01 vol.%	±0.4 vol.% [22]
NO _x	BOSCH BEA 060	According to instrument range	According to calibration certificate

Using the stated instrument accuracies and the root-sum-square propagation method, the relative uncertainty in brake power was approximately ±0.51%. The propagated uncertainty in fuel mass flow rate varied between approximately ±0.54% and ±0.75%, depending mainly on the measured fuel consumption time. Consequently, the overall propagated uncertainty in BSFC ranged from approximately ±0.74% to ±0.91%. Under the adopted calculation framework, the corresponding propagated uncertainties in energy efficiency and exergy efficiency were also approximately ±0.74–0.91%. These values include the uncertainties of the directly measured quantities entering the calculations; uncertainties

associated with literature-derived density, LHV, and chemical-exergy factors were not included because consistent uncertainty bounds were not available for all fuels. The propagated uncertainties of the principal calculated parameters are summarized in Table 6.

Table 6. Propagated uncertainties of the principal calculated parameters.

Calculated Parameter	Relative Uncertainty
Brake power	$\pm 0.51\%$
Fuel mass flow rate	$\pm 0.54\text{--}0.75\%$
BSFC	$\pm 0.74\text{--}0.91\%$
Energy efficiency	$\pm 0.74\text{--}0.91\%$
Exergy efficiency	$\pm 0.74\text{--}0.91\%$

3. Thermodynamic Analysis

3.1. Energy Analysis

The engine was treated as a steady-flow control volume. Changes in kinetic and potential energy were neglected. The principal energy input was the chemical energy of the fuel, whereas the outputs consisted of brake power, exhaust-gas energy, and unaccounted energy loss. Intake air energy was neglected because the inlet air was close to the reference ambient state.

The steady-state energy balance was written as Equation (18):

$$\dot{E}_{\text{fuel}} = P_b + \dot{E}_{\text{exh}} + \dot{E}_{\text{loss}} \quad (18)$$

Fuel-energy input was determined by Equation (19):

$$\dot{E}_{\text{fuel}} = \dot{m}_f LHV_f \quad (19)$$

where LHV_f is the lower heating value of the relevant test fuel. The sensible energy carried by the exhaust gas was calculated as Equation (20):

$$\dot{E}_{\text{exh}} = \dot{m}_{\text{exh}}(h_{\text{exh}} - h_0) \quad (20)$$

where $\dot{m}_{\text{exh}}$ is the exhaust mass flow rate, h_{exh} is the specific enthalpy at the measured exhaust temperature, and h_0 is the specific enthalpy at the reference condition.

The residual energy term was obtained from Equation (21):

$$\dot{E}_{\text{loss}} = \dot{E}_{\text{fuel}} - P_b - \dot{E}_{\text{exh}} \quad (21)$$

This residual includes heat rejected from the engine surface and cooling air, friction, incomplete-combustion effects, and other energy flows that were not quantified separately. It was therefore defined as the unaccounted energy loss.

Energy efficiency, equivalent to brake thermal efficiency under the adopted control volume, was calculated as Equation (22):

$$\eta_{\text{en}} = \frac{P_b}{\dot{m}_f LHV_f} \quad (22)$$

3.2. Exergy Analysis

Exergy analysis was used to quantify the conversion of the fuel's maximum useful-work potential and the irreversibilities associated with combustion, heat transfer, mixing, friction, and exhaust discharge. The reference environment was defined as $T_0 = 298.15 \text{ K}$ and $P_0 = 101.325 \text{ kPa}$ for all energy and exergy calculations [46,47].

The steady-state exergy balance was expressed as Equation (23):

$$\dot{E}x_{\text{fuel}} = P_b + \dot{E}x_{\text{exh}} + \dot{E}x_Q + \dot{E}x_D \quad (23)$$

The specific chemical exergy of each fuel was determined from Equation (24), where β_f is the dimensionless chemical-exergy factor. The fuel-exergy input was calculated using Equation (24), where β_f represents the dimensionless chemical-exergy factor. For fuels with known elemental compositions, this factor was estimated using the Kotas correlation given in Equation (25) [46]:

$$\dot{E}x_{\text{fuel}} = \dot{m}_f e_{\text{ch},f} = \dot{m}_f \beta_f \text{LHV}_f \quad (24)$$

where $e_{\text{ch},f}$ is the specific chemical exergy and β_f is the chemical-exergy factor. For fuels with known elemental composition, the chemical-exergy factor was estimated using the Kotas correlation (Equation (25)) [46]:

$$\beta_f = 1.0401 + 0.1728 \left(\frac{H}{C} \right) + 0.0432 \left(\frac{O}{C} \right) + 0.2169 \left(\frac{S}{C} \right) \left[1 - 2.0628 \left(\frac{H}{C} \right) \right] \quad (25)$$

Because gasoline and JP-8 are multicomponent fuels, representative elemental compositions were used in determining their chemical-exergy factors. No single pure-component molecular formula was assigned to these fuels.

The physical component of exhaust exergy was calculated as Equation (26):

$$\dot{E}x_{\text{exh,ph}} = \dot{m}_{\text{exh}} [(h_{\text{exh}} - h_0) - T_0 (s_{\text{exh}} - s_0)] \quad (26)$$

When the measured gas composition was incorporated, exhaust chemical exergy was estimated from Equation (27):

$$\dot{E}x_{\text{exh,ch}} = \dot{n}_{\text{exh}} RT_0 \sum_i y_i \ln \left(\frac{y_i}{y_{i,0}} \right) \quad (27)$$

where y_i is the molar fraction of component i in the exhaust and $y_{i,0}$ is its molar fraction in the reference environment. Total exhaust exergy was then as follows:

$$\dot{E}x_{\text{exh}} = \dot{E}x_{\text{exh,ph}} + \dot{E}x_{\text{exh,ch}} \quad (28)$$

The exergy associated with heat transfer from the engine surface was estimated as Equation (29):

$$\dot{E}x_Q = \dot{Q}_{\text{loss}} \left(1 - \frac{T_0}{T_s} \right) \quad (29)$$

where T_s is the representative absolute surface temperature obtained from the thermal-camera measurements.

Exergy destruction was calculated from the exergy balance with Equation (30):

$$\dot{E}x_D = \dot{E}x_{\text{fuel}} - P_b - \dot{E}x_{\text{exh}} - \dot{E}x_Q \quad (30)$$

Exergy efficiency was determined as Equation (31):

$$\eta_{\text{ex}} = \frac{P_b}{\dot{E}x_{\text{fuel}}} \quad (31)$$

Exergy destruction was obtained from the difference between fuel-exergy input and the sum of brake power, exhaust exergy, and heat transfer exergy.

4. Results and Discussion

The calculated adiabatic flame temperatures ranged from approximately 2155 to 2341 K across the investigated fuels and operating points (Table 7). The decanol-containing blends generally exhibited comparatively high theoretical flame temperatures, whereas JP-8 produced the lowest calculated values over the investigated range. At 6250 rpm, the calculated flame temperature increased for all fuels because the measured mixtures approached stoichiometric conditions more closely. G90E10, for example, reached approximately 2341 K at a measured excess-air ratio of 0.933. Nevertheless, the ranking of calculated adiabatic flame temperature did not directly reproduce the measured exhaust-gas-temperature ranking. This difference is expected because exhaust-gas temperature is governed not only by the thermodynamic flame temperature potential but also by combustion phasing, exhaust-port timing, gas mass flow, wall heat transfer, mixture preparation, and the extent of late oxidation. Accordingly, the calculated adiabatic flame temperatures were used as supporting thermochemical indicators rather than as direct representations of measured exhaust-gas temperature or peak in-cylinder temperature. The lower heating values, chemical-exergy factors, and specific chemical exergy values adopted for the investigated fuels are summarized in Table 8.

Table 7. Calculated adiabatic flame temperatures (K) for the six test fuels based on measured excess-air-ratio values and representative surrogate formulations.

Shaft Speed (rpm)	JP-8	G90D10	G80D20	G90E10	G80E20	G100
3250	2160 K	2272 K	2292 K	2270 K	2258 K	2250 K
3750	2172 K	2296 K	2310 K	2280 K	2270 K	2262 K
4500	2157 K	2260 K	2284 K	2270 K	2257 K	2237 K
5250	2155 K	2279 K	2297 K	2263 K	2245 K	2223 K
6250	2260 K	2324 K	2338 K	2341 K	2333 K	2324 K

Table 8. Lower heating value, chemical-exergy factor, and specific chemical exergy used in the exergy calculations.

Fuel	LHV (MJ/kg)	β_f	$e_{ch,f}$ (MJ/kg)
G100	43.400	1.07363	46.595
JP-8	42.700	1.06883	45.639
G90D10	43.221	1.07423	46.430
G80D20	43.047	1.07481	46.267
G90E10	42.994	1.07443	46.193
G80E20	42.586	1.07537	45.796

Figure 2 shows the variation in brake-specific fuel consumption (BSFC) and exhaust-gas temperature (EGT) with shaft speed for G100, neat JP-8, gasoline–1-decanol blends, and gasoline–DIPE blends. G100 was used as the common reference fuel to quantify the effects of oxygenate addition and to provide a direct baseline for comparison with neat JP-8. For all fuels, BSFC decreased markedly between 3250 and 5250 rpm and increased again at 6250 rpm. The minimum BSFC values were therefore obtained at 5250 rpm, indicating that this speed provided the most favorable balance between fuel consumption and brake power output for the investigated engine. At 3250 rpm, the high brake-specific values were mainly associated with the low useful power output, weak charge motion, relatively poor fuel evaporation, and unfavorable scavenging conditions. Increasing

shaft speed enhanced in-cylinder turbulence and mixture preparation, while the increase in brake power was greater than the corresponding increase in fuel consumption. Similar speed-dependent behavior has been associated with variations in delivery ratio, trapping efficiency, residual-gas fraction, and fresh-charge short circuiting in two-stroke engines [6,48]. At 6250 rpm, BSFC increased for every fuel, although the increase was considerably greater for JP-8 than for the oxygenated gasoline blends. At this speed, the reduced time available for evaporation, flame propagation, and post-flame oxidation, together with increasing mechanical friction and possible deterioration in cylinder filling and trapping efficiency, limited the benefit of the higher shaft speed. Thus, 5250 rpm represented the optimum operating point within the tested range, whereas operation at 6250 rpm imposed additional fuel consumption penalties. Comparable trends have been reported in two-stroke UAV engines fueled with higher-alcohol blends, where thermodynamic efficiency improved up to an intermediate-high speed before decreasing as thermal, frictional, and gas exchange losses became more influential [22]. The inclusion of G100 provides a direct reference for evaluating the oxygenated blends. G100 exhibited BSFC values of approximately 1.797, 0.794, 0.610, 0.474, and 0.496 kg/kWh at 3250, 3750, 4500, 5250, and 6250 rpm, respectively. Similar to the other fuels, its minimum BSFC occurred at 5250 rpm. At this operating point, G90E10 and G80E20 showed slightly lower BSFC than G100, whereas G90D10 and G80D20 showed higher values. This comparison indicates that DIPE addition provided a modest fuel consumption benefit relative to neat gasoline, while increasing the 1-decanol fraction imposed a BSFC penalty under the investigated operating conditions. Among the gasoline-based fuels, G90E10 generally produced the lowest BSFC, with values of 1.796, 0.793, 0.613, and 0.493 kg/kWh at 3250, 3750, 4500, and 6250 rpm, respectively. Increasing the DIPE fraction from 10 to 20 vol.% caused only minor changes in BSFC and did not provide a clear additional fuel economy advantage. DIPE has a low boiling temperature and high volatility, which favor rapid evaporation and homogeneous mixture formation during the short preparation period available in a two-stroke engine. Its molecular oxygen may also promote the oxidation of locally rich regions. Previous gasoline–DIPE studies have similarly reported improved thermal efficiency and combustion development resulting from enhanced mixture preparation [25,49]. However, the higher DIPE ratio also reduced the blend heating value and changed the stoichiometric air requirement. Under unchanged carburetor and ignition settings, these effects may have offset the expected benefit of increasing the DIPE content to 20 vol.%. The gasoline–decanol blends exhibited higher BSFC than the corresponding DIPE blends, and increasing the decanol fraction from 10 to 20 vol.% further increased fuel consumption. At 5250 rpm, BSFC increased from 0.482 kg/kWh for G90D10 to 0.491 kg/kWh for G80D20. This behavior was primarily related to the high boiling temperature and viscosity of 1-decanol. Compared with DIPE, decanol evaporates more slowly and requires greater thermal energy for vaporization. In a small two-stroke engine, these properties can promote droplet persistence, wall-film formation, and mixture heterogeneity, thereby limiting the effective conversion of fuel energy into brake work. Studies involving 1-decanol and other long-chain alcohols have similarly emphasized the strong influence of volatility, viscosity, and operating temperature on fuel utilization [34,37]. JP-8 produced the highest BSFC at 3250, 3750, and 4500 rpm because its broad distillation range and low volatility restricted evaporation and mixture homogenization under relatively low thermal conditions. Nevertheless, JP-8 achieved the lowest overall BSFC of 0.429 kg/kWh at 5250 rpm. This operating-point-specific result suggests that the higher engine temperature and stronger charge motion at 5250 rpm were sufficient to improve JP-8 evaporation and combustion, allowing its relatively high mass-based energy content to be utilized more effectively. Heavy-fuel engine studies

also show that kerosene combustion is highly dependent on fuel preparation, engine temperature, injection or carburetion conditions, ignition strategy, and scavenging behavior [11,12,48,50]. However, the JP-8 advantage disappeared at 6250 rpm, where BSFC increased sharply to 0.542 kg/kWh, confirming its greater sensitivity to shaft speed. EGT increased continuously with shaft speed for all fuels. G100 exhibited EGT values of 71, 86, 92, 103, and 108 °C as shaft speed increased from 3250 to 6250 rpm. Compared with G100, G90E10 generally produced lower exhaust temperatures, whereas the decanol blends, particularly G80D20, produced higher EGT values. At 5250 rpm, for example, the EGT was 103 °C for G100, compared with 99 °C for G90E10, 102 °C for G80E20, 105 °C for G90D10, 109 °C for G80D20, and 108 °C for JP-8. The increase resulted from the higher fuel-energy release rate, greater exhaust mass flow, and reduced time available for heat transfer during each engine cycle. G90E10 produced the lowest EGT range, from 78 to 103 °C, whereas G80D20 generated the highest values, from 83 to 116 °C. The comparatively low EGT of G90E10 was consistent with its favorable BSFC and suggests that the high volatility of DIPE promoted earlier and more effective combustion, leaving less sensible energy in the exhaust. In contrast, the higher EGT of the decanol blends was consistent with slower evaporation and a longer effective combustion process, which may have shifted part of the heat release toward the late expansion period. Overall, G90E10 provided the most consistent compromise between BSFC and EGT over the complete speed range. Increasing DIPE to 20 vol.% produced little additional BSFC benefit, whereas increasing decanol from 10 to 20 vol.% increased both BSFC and EGT. JP-8 exhibited the strongest speed dependence, with unfavorable performance at most speeds but a distinct optimum at 5250 rpm. These results demonstrate that the engine response was governed by the combined effects of shaft speed, fuel volatility, viscosity, heating value, oxygen content, mixture formation, and two-stroke scavenging characteristics rather than by a single fuel property. The calculated adiabatic flame temperature results further demonstrate that EGT cannot be interpreted solely from the intrinsic fuel properties. For G100, the calculated adiabatic flame temperatures were approximately 2250, 2262, 2237, 2223, and 2324 K at 3250, 3750, 4500, 5250, and 6250 rpm, respectively. At 5250 rpm, the calculated values were approximately 2223 K for G100, 2279 K for G90D10, 2297 K for G80D20, 2263 K for G90E10, 2245 K for G80E20, and 2155 K for JP-8. These differences reflect not only fuel composition but also the experimentally observed variation in equivalence ratio. Therefore, the measured EGT trends should be interpreted as the combined result of fuel chemistry, mixture strength, combustion development, and heat transfer rather than as a direct measure of adiabatic flame temperature. It should be emphasized that the present experimental system did not include crank-angle-resolved in-cylinder pressure measurements. Consequently, combustion phasing, apparent heat release rate, IMEP, CA10/CA50, and flame propagation characteristics were not determined. Therefore, combustion-related explanations given below are used only as possible interpretations of the measured performance, temperature, lambda, and emission trends and should not be considered direct measurements of combustion timing or flame development.

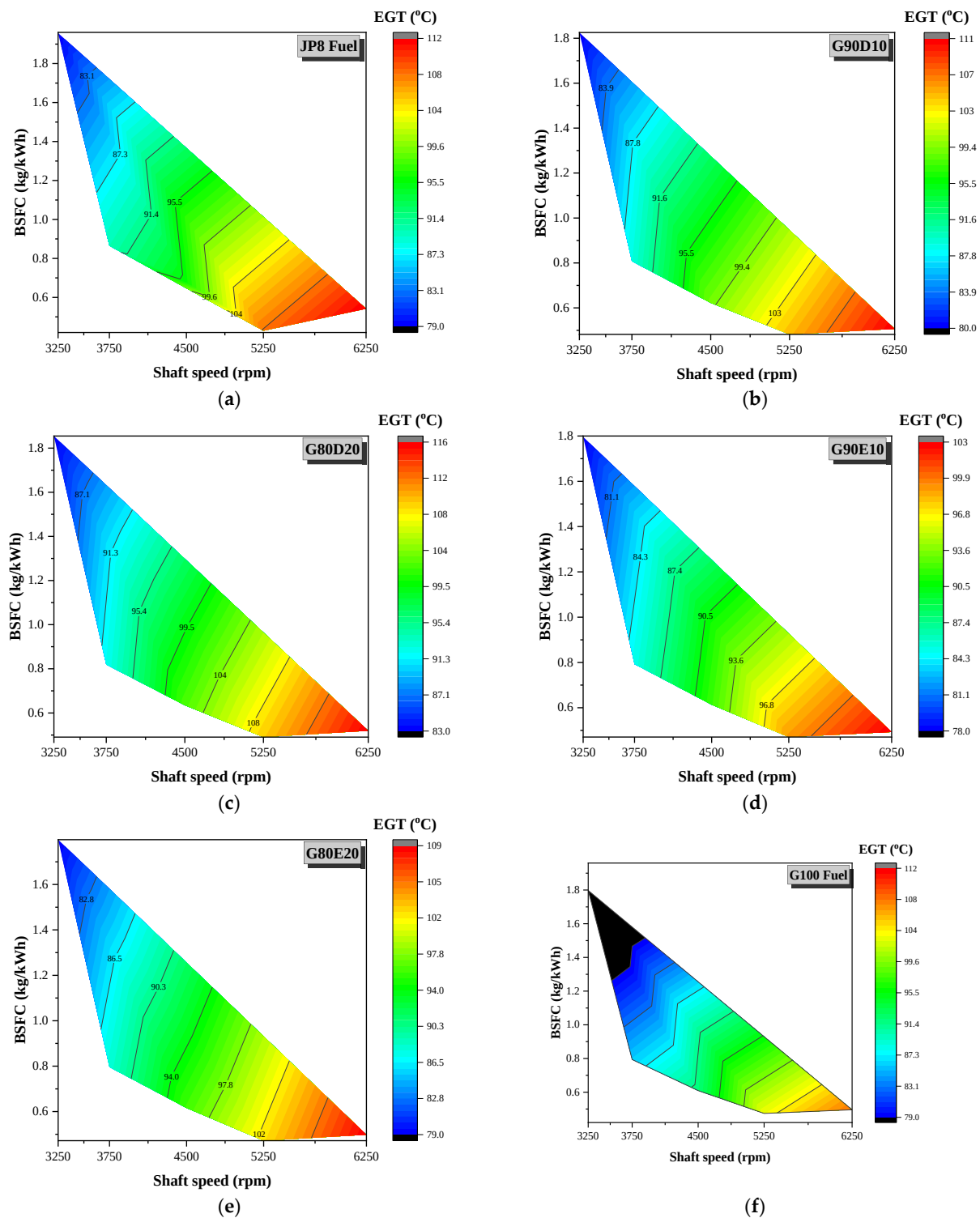


Figure 2. Combined variation of brake-specific fuel consumption and exhaust-gas temperature with shaft speed for (a) JP-8, (b) G90D10, (c) G80D20, (d) G90E10, (e) G80E20, and (f) G100 fuels.

Figure 3 illustrates the variation in CO and CO₂ emissions with shaft speed for G100, neat JP-8, and the gasoline–oxygenate blends. The incorporation of G100 enables the effect of fuel-bound oxygen on carbon oxidation to be evaluated directly relative to neat gasoline. A common non-monotonic pattern was observed for all fuels. CO decreased between 3250 and 3750 rpm, rose at 4500 rpm, and then declined toward 6250 rpm. CO₂ followed the opposite tendency, reaching its lowest values at 4500 rpm before increasing at the two highest speeds. The simultaneous CO peak and CO₂ minimum at 4500 rpm indicate less

complete oxidation of carbon at this operating point. The relatively high CO values at 3250 rpm were partly related to the low brake output, since the emissions were normalized by power. Low thermal conditions and weaker in-cylinder motion may also have limited evaporation and oxidation, while residual-gas dilution and short circuiting of the fresh charge could have increased the proportion of incompletely burned fuel leaving the cylinder. As the speed increased to 3750 rpm, CO fell by 7.95% for JP-8 and by 12.07–14.02% for the gasoline-based blends. The reduction coincided with higher brake output and more intense charge motion, which likely supported faster mixing and carbon oxidation. Comparable speed-dependent changes have been reported for two-stroke engines in which scavenging and trapping efficiency directly influence incomplete-combustion emissions [6,22,49]. At 4500 rpm, CO increased by 14.52% for JP-8 and by approximately 18–19% for the oxygenated blends relative to 3750 rpm. JP-8 produced the highest value, 288.36 g/kWh, while CO₂ fell to 293.87 g/kWh. The same response appeared for every fuel, suggesting that this local maximum was linked mainly to a change in engine operation rather than to one particular fuel. A less suitable air–fuel distribution, increased residual-gas retention, or a temporary reduction in trapping quality may have created locally rich regions in which CO oxidation was incomplete. Since cylinder pressure and heat release data were not available, the exact cause of this anomaly cannot be established. From 4500 to 6250 rpm, CO declined by approximately 32–34%, whereas CO₂ increased by 33–42%. The higher speed likely intensified turbulence and raised the thermal level of the combustion and exhaust processes, promoting the conversion of CO into CO₂. G100 followed the same general speed-dependent pattern as the oxygenated blends, with CO decreasing from the low-speed region, increasing locally at 4500 rpm, and subsequently decreasing toward 6250 rpm, while CO₂ exhibited the opposite tendency. This similarity indicates that part of the non-monotonic CO–CO₂ behavior was associated with the engine operating conditions rather than solely with the oxygenated components. The increase in brake power also contributed to lower specific CO values. Although the residence time for evaporation and oxidation became shorter, the measured results show that the improvement in oxidation was dominant over this part of the operating range. G80E20 produced the lowest CO emission at every speed, with a mean value of 242.69 g/kWh. This was 2.26% lower than JP-8 and 0.99% lower than G90E10. DIPE evaporates readily and contains oxygen within its molecular structure; therefore, the 20 vol.% blend likely formed a more uniform combustible mixture and supplied additional oxygen to fuel-rich zones. Earlier studies on DIPE–gasoline mixtures have also reported lower CO and HC emissions together with improved combustion development [25–27,48]. However, the lower CO of G80E20 was accompanied by the highest mean CO₂ value, 414.21 g/kWh. This pairing indicates that more of the fuel carbon reached the fully oxidized state. Accordingly, the result reflects better combustion completeness, but also greater direct CO₂ production per unit brake work. The decanol blends showed a smaller response to increasing oxygenate concentration. Mean CO decreased only from 248.02 g/kWh for G90D10 to 247.45 g/kWh for G80D20. Although decanol contributes chemically bound oxygen, its high boiling point and viscosity can slow vaporization and encourage droplet persistence or wall-film formation. These physical limitations appear to have weakened the benefit expected from the additional oxygen [34–37]. Mean CO₂ increased from 404.43 to 409.51 g/kWh as the decanol fraction rose from 10 to 20 vol.%, showing a modest improvement in oxidation, but the change remained less pronounced than with DIPE. JP-8 yielded the lowest mean CO₂ emissions, 383.36 g/kWh, but this should not be regarded automatically as the best environmental outcome. At 4500 rpm, its lowest CO₂ value occurred together with the highest CO value, indicating that a portion of the carbon remained only partially oxidized. The broad distillation range and low volatility of JP-8 can hinder vaporization in a gasoline-oriented

engine, particularly under less suitable thermal conditions. At 6250 rpm, JP-8 CO fell to 191.11 g/kWh while CO₂ rose to 417.00 g/kWh, showing that the higher-speed condition improved carbon conversion. In summary, G80E20 provided the lowest CO emissions, whereas JP-8 produced the lowest CO₂ values. The former result was associated with stronger oxidation, while the latter was partly linked to incomplete conversion of carbon at certain speeds. The trends therefore reflect differences in how each fuel responded to the engine's changing thermal and gas exchange conditions rather than a simple ranking based only on oxygen content or heating value [51].

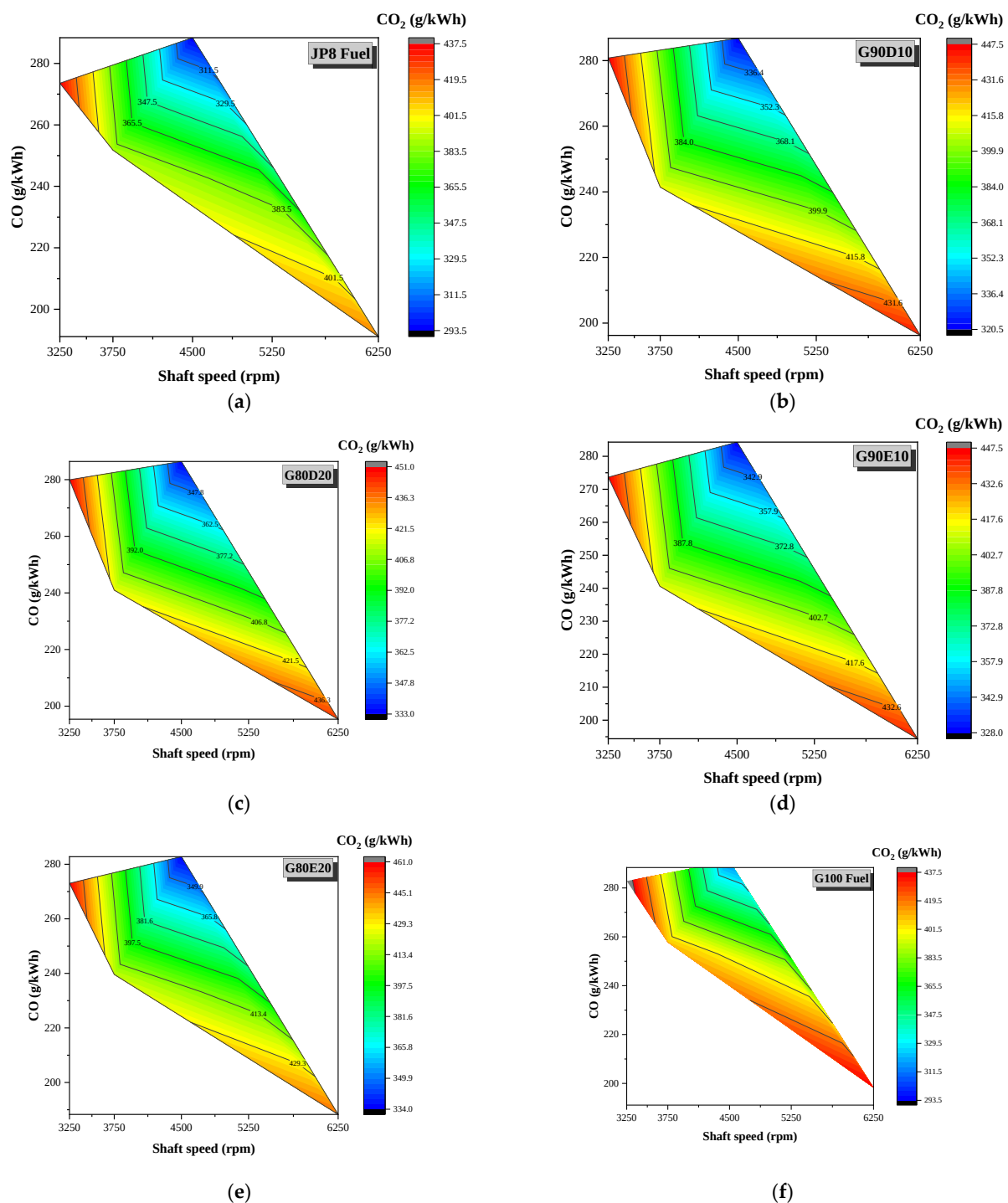


Figure 3. Combined variation of brake-specific CO and CO₂ emissions with shaft speed for (a) JP-8, (b) G90D10, (c) G80D20, (d) G90E10, (e) G80E20 and (f) G100 fuels.

Figure 4 shows the variation in HC and NO_x emissions with shaft speed for G100, neat JP-8, and the four oxygenated gasoline blends. The G100 data provide a direct baseline for distinguishing the effect of oxygenate addition from the general influence of engine speed. HC followed the same general trend for all fuels, decreasing from 3250 to 5250 rpm and rising again at 6250 rpm. At the lowest speed, HC ranged from 13.796 g/kWh for JP-8 to 14.480 g/kWh for G80D20. These relatively high values were partly associated with the low brake output used in the specific emission calculation. Low cylinder-wall temperature, weak charge motion, flame quenching, and the escape of part of the fresh mixture during scavenging may also have increased the quantity of unburned fuel in the exhaust. Such losses are characteristic of port-controlled two-stroke engines and depend strongly on trapping efficiency and residual-gas behavior [49]. The HC behavior of G100 followed the same general trend observed for the other fuels, decreasing markedly up to 5250 rpm and increasing again at 6250 rpm. At 5250 rpm, G100 exhibited an HC value of 6.54 g/kWh in the current brake-specific representation. The lower HC values obtained with the DIPE blends, particularly G80E20, indicate that DIPE addition improved the oxidation of unburned hydrocarbons relative to neat gasoline. In contrast, the response of the decanol blends was considerably smaller, indicating that the beneficial effect of fuel-bound oxygen was partly offset by the lower volatility and higher viscosity of 1-decanol. Between 3250 and 5250 rpm, HC decreased by 55.54–60.23%, depending on the fuel, and all mixtures reached their minimum values at 5250 rpm. The increase in engine temperature and turbulence over this range likely improved vaporization and flame development, while the rise in brake power further reduced the emissions expressed per unit output. The agreement between the HC and BSFC trends suggests that 5250 rpm offered the most effective fuel utilization within the tested range. At 6250 rpm, however, HC increased by approximately 34–39%. The shorter time available for vaporization and oxidation, together with possible deterioration in charge trapping, appears to have outweighed the advantage of stronger flow motion at the highest speed. G80E20 yielded the lowest mean HC emission, 9.292 g/kWh, and produced the minimum values at four of the five operating points. Raising the DIPE fraction from 10 to 20 vol.% reduced mean HC by 2.55%. DIPE evaporates readily and contributes oxygen directly within the fuel molecule, both of which can reduce locally over-rich zones and support the oxidation of unburned species. Similar reductions in HC and CO have been reported for gasoline–DIPE mixtures in spark-ignition engines [25,27]. The improvement was most evident at 5250 rpm, where G80E20 emitted 5.584 g/kWh compared with 5.980 g/kWh for G90E10. Nevertheless, its BSFC remained slightly higher than that of G90E10 at several speeds, showing that lower HC did not translate directly into lower fuel consumption. The decanol blends produced mean HC emissions of 9.891 g/kWh for G90D10 and 10.053 g/kWh for G80D20. Increasing the decanol content therefore raised HC by 1.64%. The additional oxygen supplied by 1-decanol was insufficient to compensate for its high boiling temperature and viscosity. Slow evaporation can leave larger droplets or surface films within the intake and cylinder, increasing mixture non-uniformity and the probability of incomplete burning or direct discharge during scavenging. Long-chain alcohol studies likewise show that their emission response is strongly influenced by vaporization and fuel-delivery conditions [34,37]. JP-8 gave a mean HC value of 9.500 g/kWh, lower than both decanol blends and close to the DIPE results. Its minimum of 6.056 g/kWh occurred at 5250 rpm, where JP-8 also achieved its lowest BSFC. The thermal state of the engine at this speed may have been sufficient to improve the vaporization of the kerosene-range components. At 6250 rpm, HC rose to 8.254 g/kWh, indicating that the shorter preparation and reaction periods again limited complete fuel utilization. NO_x responded differently from HC. For the four gasoline-based blends, emissions changed only moderately between 3250 and 5250 rpm

but increased sharply at 6250 rpm. At 5250 rpm, values ranged from 0.1128 g/kWh for G90D10 to 0.1526 g/kWh for G80E20. When the speed increased to 6250 rpm, they rose to 0.3318–0.3783 g/kWh. This increase coincided with higher EGT and may reflect a greater occurrence of high-temperature oxidation zones. The oxygenated components can also increase local oxygen availability once evaporation is complete. However, EGT alone cannot represent peak cylinder temperature, and the absence of pressure and heat-release data prevents a direct assessment of thermal NO formation. Among the oxygenated blends, G90D10 recorded the lowest mean NO_x emission, 0.1858 g/kWh, followed by G90E10 at 0.1951 g/kWh. Increasing DIPE from 10 to 20 vol.% raised mean NO_x by 13.61%, while the corresponding increase for decanol was 16.43%. Thus, the higher oxygenate fractions generally favored NO_x formation. G80E20 combined the lowest HC with higher NO_x than G90E10, revealing a trade-off between the oxidation of unburned hydrocarbons and the formation of nitrogen oxides. G80D20, in contrast, produced more HC and more NO_x than G90D10, implying that mixture non-uniformity may have created both incompletely burned regions and hotter oxygen-rich zones within the same cycle. JP-8 produced NO_x values of 0.1924, 0.2323, 0.1934, 0.7897, and 0.3915 g/kWh as shaft speed increased. The peak at 5250 rpm was much higher than the values of the other fuels and coincided with the minimum JP-8 BSFC. G100 also provides an important reference for the NO_x response. At 5250 rpm, G100 exhibited a NO_x value of approximately 0.086 g/kWh, which was lower than the corresponding values of the oxygenated blends. Thus, although the oxygenated fuels generally reduced CO and, particularly for DIPE, HC emissions, these improvements were accompanied by increased NO_x formation relative to neat gasoline. This result demonstrates a clear oxidation–NO_x trade-off. Because no in-cylinder pressure or heat-release measurements were available, the higher NO_x levels cannot be attributed directly to changes in peak combustion temperature or combustion phasing; they are therefore interpreted only as being consistent with changes in local oxygen availability and thermal conditions. This may indicate an operating condition in which improved JP-8 vaporization and energy release generated locally higher temperatures. Aviation-kerosene studies have shown that NO_x formation can change markedly with mixture preparation, equivalence ratio, ignition strategy, and engine thermal state [52,53]. Even so, the magnitude of the measured peak should be treated cautiously because it was more than four times the value at 4500 rpm and approximately twice that at 6250 rpm. The original analyzer reading, emission-conversion procedure, brake power input, and repeatability of this operating point should therefore be checked before drawing a firm conclusion. Taken together, G80E20 was the most effective fuel for limiting HC, whereas G90D10 gave the lowest mean NO_x among the blends. Raising either oxygenate concentration increased NO_x, while only DIPE produced a clear reduction in HC. JP-8 showed competitive HC performance but an unusual NO_x response at 5250 rpm that requires experimental confirmation.

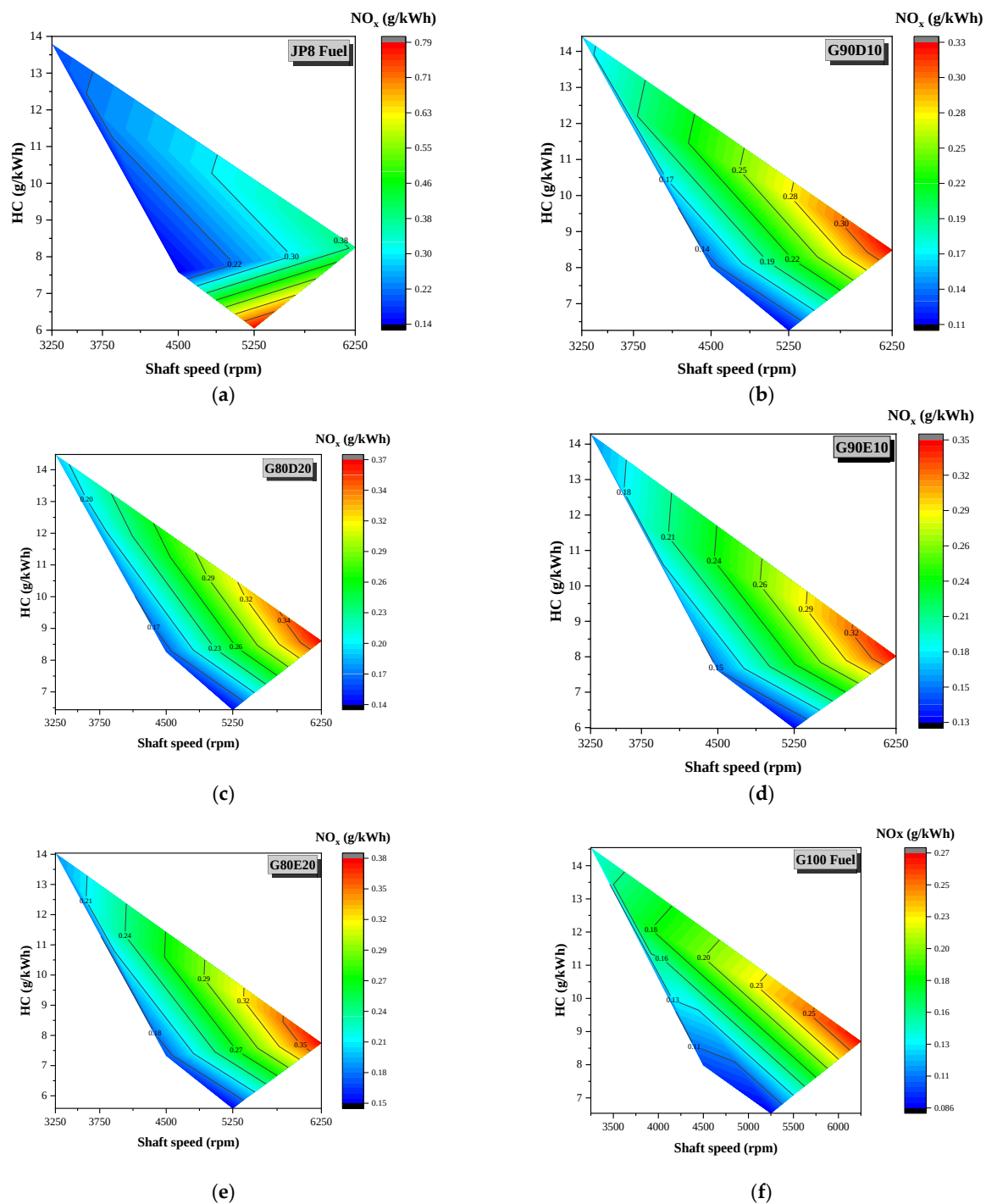


Figure 4. Combined variation of brake-specific HC and NO_x emissions with shaft speed for (a) JP-8, (b) G90D10, (c) G80D20, (d) G90E10, (e) G80E20, (f) G100 fuels.

Figure 5 presents the variations in fuel-energy input, brake power, exhaust-energy rate, residual/unaccounted energy, and energy efficiency for all six fuels. G100 was included as the common thermodynamic reference, allowing changes in fuel-energy demand caused by 1-decanol and DIPE addition to be quantified directly. Brake power was identical for all fuels at a given speed, increasing from 0.613 kW at 3250 rpm to 5.890 kW at 6250 rpm. The comparison was therefore conducted under matched speed-load conditions, and the differences in efficiency mainly reflected the amount of fuel energy required to produce the same output. Between 3250 and 4500 rpm, G80E20 required the lowest fuel-energy input, whereas JP-8 required the highest. At 4500 rpm, the corresponding values were 18.365

and 20.385 kW, giving G80E20 an approximately 9.9% lower energy demand. The lower input of the DIPE blends agreed with their BSFC results. DIPE evaporates rapidly and can improve charge uniformity during the short mixture-preparation period of a two-stroke engine. Its chemically bound oxygen may also assist oxidation in fuel-rich regions, enabling the required brake output to be produced with less supplied energy. Similar improvements in thermal efficiency and combustion development have been reported for gasoline–DIPE mixtures [25]. Increasing the DIPE fraction from 10 to 20 vol.% produced only a small additional reduction in fuel-energy input. For instance, the values at 5250 rpm were 22.048 kW for G90E10 and 21.880 kW for G80E20, while at 6250 rpm they were nearly identical. The higher DIPE ratio therefore provided little further benefit. Its lower heating value and altered stoichiometric air requirement likely counterbalanced part of the gain associated with easier evaporation, particularly because the carburetor and ignition settings were unchanged. The decanol blends required more fuel energy than the corresponding DIPE mixtures. At 5250 rpm, G90D10 and G80D20 consumed 22.726 and 23.045 kW, respectively. Increasing the decanol fraction consistently raised the energy demand. The fuel-energy input of G100 was approximately 22.51 kW. At the same operating point, G90E10 and G80E20 required approximately 22.05 and 21.88 kW, respectively, whereas G90D10 and G80D20 required approximately 22.73 and 23.05 kW, respectively. Thus, DIPE addition slightly reduced the energy required to produce the prescribed brake output relative to G100, whereas 1-decanol addition increased it. JP-8 showed a distinctly lower fuel-energy input of approximately 20.04 kW at this operating point, although this advantage was not maintained at 6250 rpm. The high boiling temperature and viscosity of 1-decanol can delay vaporization and promote larger droplets or wall films, reducing mixture uniformity and limiting the proportion of the supplied energy converted into useful work. Engine studies involving 1-decanol have likewise shown that its efficiency response is strongly dependent on fuel preparation and operating temperature [34]. JP-8 produced the highest fuel-energy input at 3250, 3750, 4500, and 6250 rpm. Its low volatility and wide distillation range make vaporization more difficult in an engine configured for gasoline-based fuels. A different response occurred at 5250 rpm, where the fuel-energy input fell to 20.036 kW despite the increase in brake power to 3.958 kW. This value was 9.2–15.0% lower than those of the oxygenated blends and corresponded to the minimum JP-8 BSFC. The engine thermal condition at this speed appears to have improved the utilization of the kerosene-range components. However, the JP-8 input rose sharply to 37.682 kW at 6250 rpm, showing that this advantage was restricted to one operating point. Previous thermodynamic studies also indicate that aviation fuel performance can vary markedly with engine speed and combustion system conditions [23,51]. Exhaust energy increased with shaft speed for every fuel. G90E10 produced the lowest values throughout the range, rising from 0.527 to 1.424 kW, whereas G80D20 produced the highest, from 0.571 to 1.671 kW. This order matched the EGT results. The lower exhaust-energy discharge of G90E10 suggests that a greater share of the released energy was converted before exhaust-port opening. By contrast, slower evaporation of the decanol-rich blend may have shifted part of the heat release toward the late expansion period, leaving more sensible energy in the exhaust gases. Although the absolute exhaust-energy rate increased, its fraction of the fuel input did not always follow the same trend. For JP-8, exhaust energy represented approximately 6.16% of the input at 5250 rpm but only 4.18% at 6250 rpm. This reduction in percentage did not indicate a smaller exhaust loss; it resulted from the much larger increase in total fuel-energy supply at the highest speed. The unaccounted energy loss was obtained by subtracting brake power and exhaust energy from the fuel input. It therefore includes surface heat rejection, friction, incomplete combustion, short-circuited fuel, and other unmeasured contributions and should not be interpreted as heat transfer alone. At 3250 rpm, this term ranged

from 11.753 kW for G80E20 to 12.995 kW for JP-8. Over the full speed range, the DIPE blends generally produced the lowest residual losses, whereas JP-8 produced the highest values except at 5250 rpm. A marked rise occurred at 6250 rpm. Relative to 5250 rpm, unaccounted energy loss increased by about 103.6% for JP-8 and by 59.4–61.3% for the gasoline-based blends. Brake power increased by 48.8%, but fuel-energy input rose much faster, particularly for JP-8. Greater friction, shorter vaporization and reaction periods, and less effective charge trapping may have prevented the additional energy from being converted proportionally into shaft work. Similar two-stroke UAV studies have shown that the efficiency benefit of increasing speed can be reversed when thermal and gas exchange losses become dominant [22,23]. The G100 energy efficiency increased from approximately 4.64% at 3250 rpm to a maximum of 17.59% at 5250 rpm and then decreased to 16.82% at 6250 rpm. Relative to G100, the DIPE blends provided a modest improvement in energy efficiency around the optimum operating region, whereas the decanol blends generally showed lower efficiency. This comparison confirms that the apparent thermodynamic benefit of the oxygenated fuels depends strongly on oxygenate type rather than simply on the presence of fuel-bound oxygen. Energy efficiency increased from 3250 to 5250 rpm for all fuels and then declined at 6250 rpm. G80E20 gave the highest efficiencies between 3250 and 4500 rpm, reaching 13.856% at 4500 rpm. At 5250 rpm, JP-8 achieved the maximum value of the entire data set, 19.756%, compared with 17.177–18.091% for the blends. This result arose from the unusually low JP-8 fuel-energy input at that speed rather than from a consistent advantage across the test range. At 6250 rpm, JP-8 efficiency fell by 20.87% to 15.632%, whereas the reductions for the blends remained between 4.51 and 5.54%. G90E10 and G80E20 retained efficiencies of 17.144% and 17.140%, respectively. On an average basis, G80E20 gave the highest energy efficiency, at 12.912%, followed closely by G90E10 at 12.848%. Increasing decanol content reduced the mean efficiency, while the higher DIPE fraction produced only a modest improvement. Thus, the DIPE blends offered the most consistent energy utilization, whereas JP-8 showed a pronounced but isolated optimum at 5250 rpm.

Figure 6 presents the fuel-exergy input, exhaust exergy, heat-transfer-related exergy term, exergy destruction, and exergy efficiency for all six fuels. G100 was used as the common reference for evaluating the effect of oxygenate addition on the useful work potential of the supplied fuel. Because brake power was kept identical at each shaft speed, the differences mainly reflect how much fuel exergy was required to obtain the same output. Fuel exergy exceeded fuel-energy input because chemical exergy represents the maximum theoretical work obtainable as the fuel reaches equilibrium with the reference environment [46,47]. Fuel exergy generally increased with shaft speed. At 3250 rpm, JP-8 required the highest input, 15.116 kW, whereas G80E20 required the lowest, 13.873 kW. The same ordering continued at 3750 and 4500 rpm; at 4500 rpm, G80E20 consumed approximately 9.35% less fuel exergy than JP-8. The lower values of the DIPE blends were consistent with their BSFC and energy analysis results. Rapid DIPE evaporation may have improved charge preparation, allowing the prescribed brake output to be produced with a smaller fuel flow. Increasing DIPE from 10 to 20 vol.% provided only a minor additional reduction, and the two ether blends reached nearly identical values at 6250 rpm. The lower heating value of the higher-DIPE mixture and its different stoichiometric requirement therefore appear to have limited the gain obtained from its greater volatility [22,25]. The decanol blends required more fuel exergy than the corresponding DIPE blends. At 5250 rpm, G100 required approximately 24.16 kW of fuel exergy. The corresponding inputs were approximately 23.69 kW for G90E10 and 23.53 kW for G80E20, whereas G90D10 and G80D20 required approximately 24.41 and 24.77 kW, respectively. The comparison therefore shows that DIPE addition slightly reduced the supplied fuel exergy relative to neat gasoline,

while increasing the decanol fraction increased the exergy requirement. Raising the decanol content increased the exergy demand at every tested speed. Although 1-decanol contains chemically bound oxygen, its high boiling temperature and viscosity can slow vaporization and leave a less uniform charge, thereby increasing the fuel quantity needed for the same brake output [34,37]. JP-8 showed a distinct response at 5250 rpm. Its fuel-exergy input decreased from 21.787 kW at 4500 rpm to 21.415 kW, while brake power rose from 2.545 to 3.958 kW. At this point, JP-8 required 8.99–13.54% less fuel exergy than the blends. This agrees with its minimum BSFC and maximum energy efficiency and suggests that the engine temperature at 5250 rpm was particularly suitable for the vaporization and use of the kerosene-range fuel. The advantage disappeared at 6250 rpm, where JP-8 fuel exergy increased to 40.274 kW, well above the gasoline-based fuels. Its behavior was therefore highly dependent on the operating point rather than consistently superior across the speed range. Exhaust exergy increased steadily with engine speed. Average values were lowest for G90E10 and G80E20, at approximately 1.502 kW, and highest for JP-8, at 1.579 kW. At 6250 rpm, JP-8 discharged 2.544 kW of exergy through the exhaust, whereas the DIPE blends remained near 2.36 kW. The lower exhaust exergy of G90E10 corresponded with its lower EGT, while the relatively high value of G80D20 was consistent with its hotter exhaust. Slower evaporation of the decanol-rich blend may have shifted part of the heat release closer to exhaust-port opening, increasing the useful-work potential carried out of the cylinder. Unaccounted energy loss remained much smaller than the other terms, ranging from approximately 0.006 to 0.020 kW and accounting for less than 0.08% of fuel exergy. It increased with shaft speed as engine surface temperature and heat rejection rose. G80E20 produced the largest value at 6250 rpm, 0.02019 kW. The small magnitude does not indicate negligible heat loss, because low-temperature heat carries limited useful work potential. Heat rejected at a relatively low temperature contains limited work potential even when its energy content is appreciable. These values also depend on the assumed heat transfer rate, representative surface temperature, and reference environment. Exergy destruction decreased slightly from 3250 to 3750 rpm and then increased at 4500 rpm. At that speed, JP-8 produced the highest value, 17.664 kW, whereas G80E20 produced the lowest, 15.749 kW. The initial reduction indicates that the increase in brake output was greater than the rise in irreversibilities. The lower destruction values of the DIPE blends were mainly associated with their smaller fuel-exergy inputs rather than with a large difference in exhaust or unaccounted energy loss. At 5250 rpm, JP-8 exergy destruction fell to 15.826 kW, while the blend values increased to 17.878–19.048 kW. This unusual result arose from the sharp improvement in JP-8 fuel utilization at that speed. At 6250 rpm, destruction increased by approximately 101% for JP-8 and by 59–61% for the blends. The increase in supplied exergy was therefore not matched by a proportional rise in brake output. Greater mechanical losses, shorter preparation and reaction periods, and less effective charge trapping may have contributed to this deterioration. Similar speed-dependent reversals have been reported in thermodynamic assessments of two-stroke UAV engines [22,23]. G100 exergy efficiency increased from approximately 4.32% at 3250 rpm to 16.38% at 5250 rpm before decreasing to 15.66% at 6250 rpm. At 5250 rpm, both DIPE blends exhibited slightly higher exergy efficiencies than G100, whereas the decanol blends were lower. JP-8 achieved the highest value at this operating point, but its pronounced decrease at 6250 rpm demonstrates that this was an operating-point-specific advantage rather than a consistent superiority over G100. Exergy efficiency increased with shaft speed up to 5250 rpm and decreased at 6250 rpm for all investigated fuels. This reduction can be explained primarily by the disproportionate increase in fuel-exergy input relative to brake power output at the highest engine speed. Although brake power continued to increase from 5250 to 6250 rpm, the corresponding increase in fuel consumption caused the supplied fuel exergy to rise more

rapidly, thereby reducing the ratio of useful brake work to supplied exergy. This behavior was particularly pronounced for JP-8. Its fuel-exergy input increased from approximately 21.4 kW at 5250 rpm to 40.3 kW at 6250 rpm, while its exergy efficiency decreased from approximately 18.5% to 14.6%. The deterioration at 6250 rpm is also consistent with the increase in BSFC observed at the same operating point. At the higher shaft speed, the shorter time available for fuel evaporation, mixture preparation, flame development, and post-flame oxidation, together with potentially greater frictional and gas exchange losses and less effective charge trapping, may contribute to the higher fuel requirement per unit brake output. These effects are expected to be more critical for the lower-volatility JP-8 fuel. However, because in-cylinder pressure and heat-release measurements were not available in the present study, these mechanisms are interpreted as plausible contributors rather than directly measured combustion phenomena.

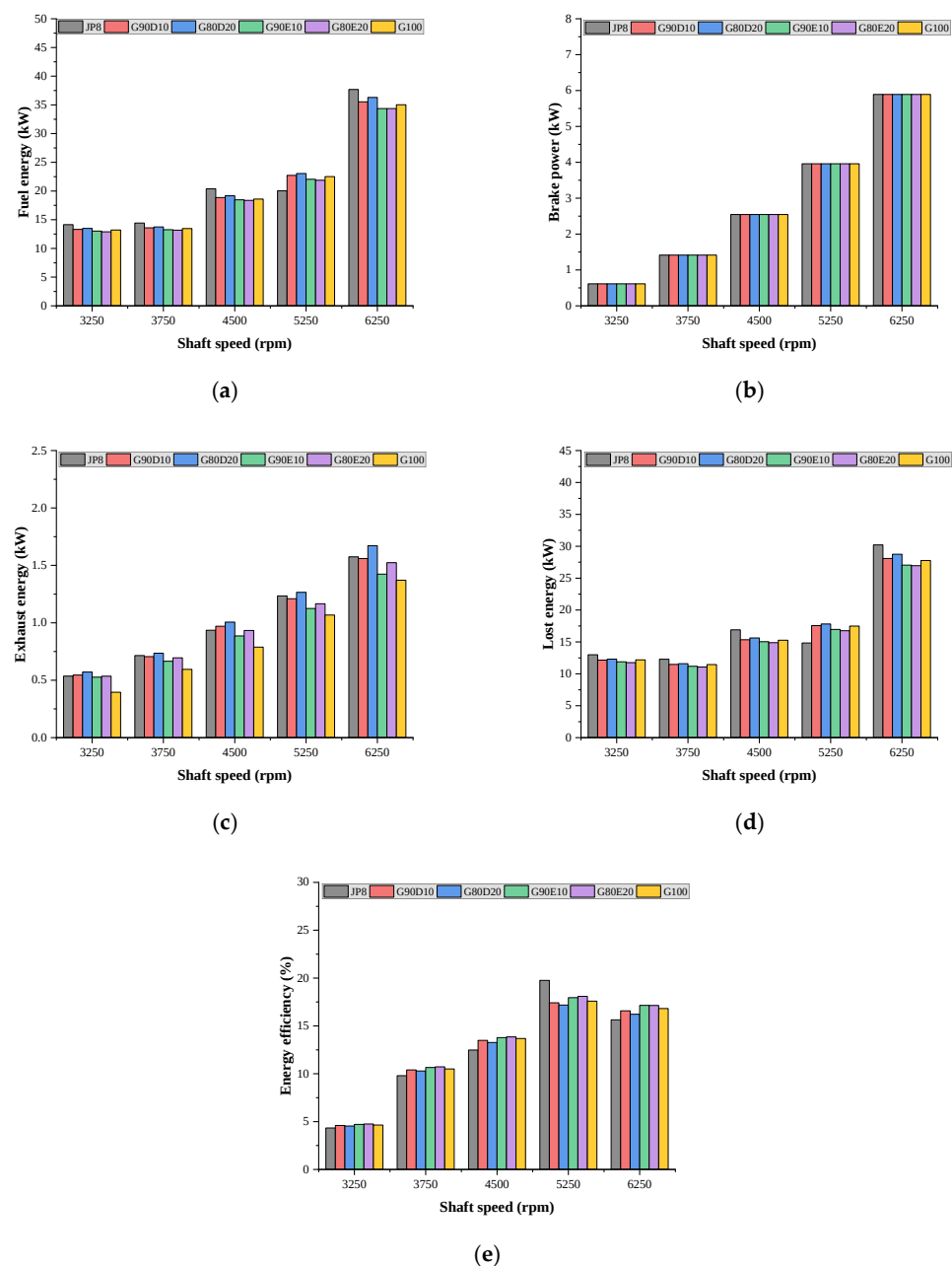


Figure 5. Variations in (a) fuel-energy input, (b) brake power, (c) exhaust-energy rate, (d) lost-energy rate, and (e) energy efficiency with shaft speed for neat JP-8, G90D10, G80D20, G90E10, G80E20 and G100 fuels.

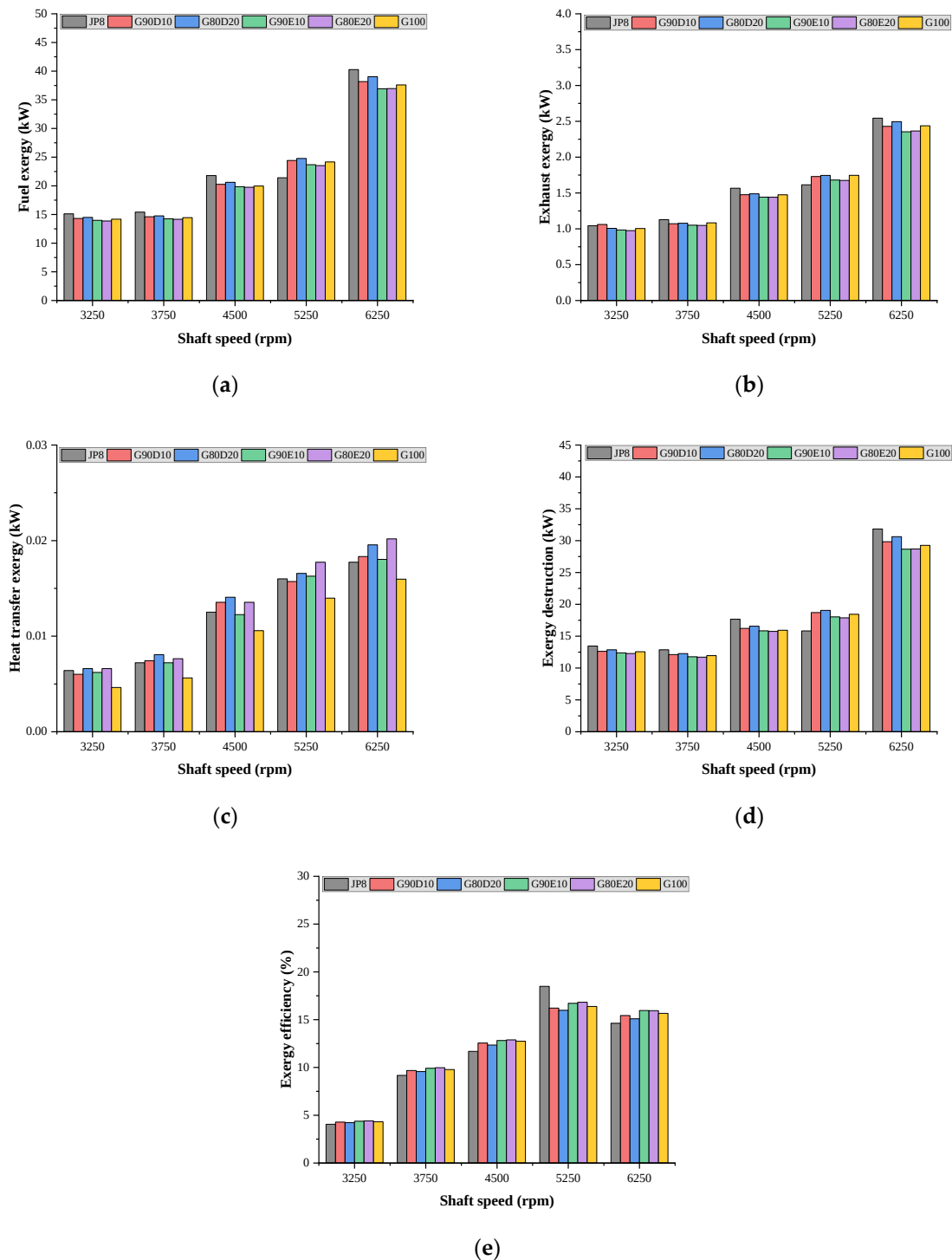


Figure 6. Variations in (a) fuel-exergy input, (b) exhaust exergy, (c) heat transfer exergy, (d) exergy destruction, and (e) exergy efficiency with shaft speed for neat JP-8, G90D10, G80D20, G90E10, G80E20 and G100 fuels.

5. Conclusions

This study experimentally compared neat gasoline (G100), gasoline–1-decanol and gasoline–DIPE blends, and neat JP-8 in a two-stroke UAV engine over 3250–6250 rpm. The incorporation of G100 as a common reference enabled the effects of the oxygenated components and JP-8 to be quantified directly under matched operating conditions. The main conclusions are the following:

- All fuels reached their minimum BSFC at 5250 rpm. At this operating point, G90E10 and G80E20 reduced BSFC by 0.73% and 0.51%, respectively, relative to G100. G80E20 also increased energy and exergy efficiencies by 2.43% and 2.27%, respectively. In contrast, increasing the 1-decanol fraction increased BSFC and reduced thermodynamic efficiency, indicating that its lower volatility and higher viscosity adversely affected mixture preparation under the investigated conditions.
- The oxygenated fuels exhibited clear emission trade-offs. At 5250 rpm, G80E20 reduced brake-specific CO and HC emissions by 6.63% and 14.58%, respectively, relative to G100, but increased NO_x. G90D10 and G80D20 also reduced CO but provided no corresponding thermodynamic advantage over G100. These results indicate that improved oxidation of CO and HC does not necessarily coincide with reduced NO_x formation.
- JP-8 showed the strongest dependence on operating conditions. At 5250 rpm, its BSFC was 9.52% lower than that of G100, while its energy and exergy efficiencies were 12.33% and 12.84% higher, respectively. However, this advantage was not sustained over the complete speed range, demonstrating that JP-8 operation in a gasoline-oriented two-stroke engine is highly sensitive to mixture preparation and engine operating conditions.
- Overall, the results demonstrate that DIPE addition can provide modest improvements in fuel utilization together with lower CO and HC, whereas increasing the decanol fraction imposes a fuel consumption and efficiency penalty under the present engine calibration. The comparison with G100 further shows that the apparent benefit of an alternative fuel should be evaluated across the complete operating range rather than from a single optimum condition.

The principal limitations of the study are the absence of crank-angle-resolved in-cylinder pressure and heat-release measurements, particulate-emission measurements, and fuel-specific optimization of ignition timing and air–fuel ratio. The quantitative findings are also specific to the RON 95 gasoline batch used as the baseline fuel. Therefore, the proposed combustion mechanisms should be regarded as interpretations of the measured trends rather than direct observations of combustion phasing or flame propagation. Future studies should include in-cylinder pressure acquisition, apparent heat-release and CA10/CA50 analyses, IMEP and cycle-to-cycle combustion assessment, particulate characterization, fuel-specific calibration, validation with gasoline fuels of different compositions, and flight-based evaluation. From a sustainability perspective, the present results address operational fuel utilization and gaseous emissions only; life-cycle fuel production, greenhouse-gas emissions, durability, economic impacts, and mission-level environmental effects remain outside the present scope.

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suggestions concerning calculations, scientific interpretation, manuscript structure, or references were independently checked, verified, and revised by the authors. The authors take full responsibility for the scientific content and final version of the manuscript.

Conflicts of Interest: Author Erdal Tunçer is employed by Erin Engine Corporation Inc. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as potential conflicts of interest.

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