International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026. ISSN (Online): 2456-7361

Effects of Engine Load and Fuel-Injection Timing on the Combustion, Performance and Emissions of 2-Methylfuran and 2-Methyltetrahydrofuran in a DISI Engine

Rafiu K. Olalere[1,2]*, Hongming Xu[2], Sheriff Lamidi[1], Kabiru A. Olaiya[1]
[1]Department of Mechanical Engineering, Lagos State University of Science and Technology, Lagos, Nigeria
[2]Department of Mechanical Engineering, University of Birmingham, Birmingham, United Kingdom
* Corresponding author: lereraf@yahoo.com

Highlights

Neat 2-methylfuran produced the highest peak cylinder pressure and the shortest combustion duration over the tested load range.
Gasoline retained the lowest mass-specific fuel consumption because of its higher lower heating value.
Injection timing between 270 and 280°CA bTDC provided the most favourable compromise between mixture preparation, peak pressure and fuel consumption.
Fuel-bound oxygen improved combustion completeness and reduced hydrocarbon emissions, but the associated temperature response created a NOx trade-off.
The study provides a direct comparison of neat MF, neat MTHF and gasoline under identical stoichiometric DISI operating conditions.

Abstract— *Furan-derived oxygenates are attractive renewable candidates for spark-ignition engines because they possess volatility, energy density and anti-knock characteristics that are more compatible with gasoline infrastructure than many conventional alcohol fuels.. This study experimentally compares neat MF and neat MTHF with a commercial unleaded gasoline (ULG) in a single-cylinder spray-guided DISI engine operated at 1500 rpm, stoichiometric air–fuel ratio (λ = 1) and fuel-specific maximum-brake-torque spark timing. Engine-load sweeps were conducted from 3.5 to 8.5 bar indicated mean effective pressure (IMEP), while single-pulse injection-timing sweeps were performed from 180 to 300° crank angle before top dead centre (°CA bTDC) at 5.5 bar IMEP. At 8.5 bar IMEP, the peak cylinder pressure of MF was approximately 21% higher than that of ULG and 15% higher than that of MTHF, consistent with its shorter combustion duration; ULG exhibited approximately 30% lower indicated specific fuel consumption than MF and about 23–27% lower consumption than MTHF because of its higher gravimetric heating value. The most favourable injection region for the tested fuels was approximately 270–280°CA bTDC, although the optimum depended on the selected performance metric. Oxygenation reduced incomplete-combustion products, particularly hydrocarbons, but increased the sensitivity of NOx formation to combustion temperature. The results demonstrate that MF offers rapid and stable combustion, whereas MTHF provides an intermediate fuel-consumption response but requires careful control of knock and NOx at high load.*

Keywords— *2-methylfuran; 2-methyltetrahydrofuran; direct-injection spark-ignition engine; start of injection; engine load; combustion duration; indicated efficiency; regulated emissions.*

NOMENCLATURE AND ABBREVIATIONS

Symbol	Definition	Symbol	Definition
°CA	Crank-angle degree	IMEP	Indicated mean effective pressure
bTDC	Before top dead centre	ISFC	Indicated specific fuel consumption
CD	Combustion duration	MBT	Maximum brake torque
COVIMEP	Coefficient of variation of IMEP	MF	2-Methylfuran
DISI	Direct-injection spark-ignition	MTHF	2-Methyltetrahydrofuran
EGR	Exhaust-gas recirculation	NOx	Oxides of nitrogen
HC/THC	Hydrocarbon/total hydrocarbon	Pmax	Peak cylinder pressure
LHV	Lower heating value	ULG	Unleaded gasoline

I. INTRODUCTION

Battery electrification is advancing rapidly, but high-efficiency internal-combustion engines are expected to remain relevant in hybrid powertrains, specialist vehicles and regions where charging infrastructure, vehicle cost or duty-cycle constraints limit rapid fleet electrification. In this transition, renewable liquid fuels can lower life-cycle greenhouse-gas emissions while retaining the high energy density, rapid refuelling and existing distribution advantages of conventional hydrocarbons. Recent assessments therefore emphasise the need to co-optimise fuel molecular structure and advanced spark-ignition combustion systems rather than evaluate alternative fuels only as drop-in substitutes [1][2]. Gasoline direct injection has become central to modern spark-ignition engine development because it supports charge cooling, high compression ratio, turbocharging, knock control and flexible mixture preparation. These advantages are accompanied by important calibration challenges. The start of injection determines the ambient

International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026. ISSN (Online): 2456-7361

pressure and temperature into which the spray is introduced, the time available for atomisation and evaporation, the probability of wall or piston impingement, the level of mixture stratification near the spark plug and, ultimately, combustion stability and pollutant formation [3][4]. Early injection can provide a longer mixing period and a more homogeneous charge, but may increase wall wetting under some conditions. Late injection can reduce the time available for evaporation and mixing, causing locally rich regions, incomplete combustion and higher particulate or hydrocarbon emissions. Consequently, injection timing cannot be considered independently of fuel volatility, latent heat, density, laminar burning velocity and chemical reactivity [5].

Among renewable oxygenates, furan-derived fuels have attracted sustained attention because they can be synthesised from the carbohydrate fraction of lignocellulosic biomass. Catalytic conversion routes from sugars and platform molecules such as furfural or 5-hydroxymethylfurfural provide a pathway that can avoid direct competition with food production. Recent reviews of tailor-made fuels from biomass identify 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF) as promising candidates whose molecular structures and thermophysical properties can be matched to advanced engine concepts [6][7]. Compared with ethanol, MF has lower water affinity, a substantially lower enthalpy of vaporisation and a higher volumetric energy density, which can improve blend stability and reduce cold-start penalties. MF also has a high research octane number and a relatively fast burning rate, making it attractive for knock-limited spark-ignition operation. MTHF, the hydrogenated analogue of MF, has a higher hydrogen-to-carbon ratio and greater saturation, but its lower octane quality and lower vapour pressure can alter knock tendency, evaporation and mixture formation. Fundamental and engine studies have established that the two fuels do not behave as interchangeable oxygenates. MF generally exhibits high knock resistance, fast flame development, short combustion duration and good combustion stability. In a spray-guided DISI engine , MF retained a wider usable spark-timing range than gasoline at elevated load and was less sensitive to injection-timing variation. Earlier comparative testing also reported that MF could achieve higher indicated efficiency and lower hydrocarbon emissions than gasoline under selected conditions, although its high flame temperature tended to increase Nox [9][110]. MTHF offers a higher gravimetric heating value than MF and can therefore reduce the fuel-mass penalty associated with oxygenation; however, its lower octane rating may constrain high-load operation. Laminar-flame and ignition studies further show that hydrogenation of the furan ring changes reaction pathways, burning velocity and autoignition characteristics, so calibration strategies derived for MF cannot automatically be transferred to MTHF [11-14]. The present study addresses these gaps by experimentally comparing neat MF, neat MTHF and commercial unleaded gasoline in the same single-cylinder spray-guided DISI engine. Two complementary test matrices are employed. The first quantifies the effects of increasing load from 3.5 to 8.5 bar IMEP at constant speed, stoichiometric operation and fuel-specific optimum spark timing. The second examines a broad single-pulse start-of-injection sweep at an intermediate load of 5.5 bar IMEP. The measured responses include peak cylinder pressure, cycle-to-cycle variation, combustion duration, indicated specific fuel consumption, indicated and combustion efficiencies, and engine-out CO, hydrocarbon and NOx emissions. The novelty of the work lies in the controlled three-fuel comparison and in linking the observed responses to the contrasting properties of an unsaturated furan (MF), a saturated cyclic ether (MTHF) and a multicomponent petroleum gasoline. The study therefore provides a direct assessment of the effect of load on combustion phasing, stability, efficiency and emissions and identifies the injection-timing region that provides the most favourable mixture preparation for each fuel.

II. Experimental Setup

The experimental investigations were conducted using a fully instrumented gasoline direct-injection spark-ignition (DISI) engine test facility designed for steady-state combustion and emission analysis under controlled laboratory conditions. The facility was configured to permit independent variation of engine operating parameters while maintaining precise control of intake conditions, fuel delivery, ignition timing and engine cooling. The overall experimental system comprised the test engine, an eddy-current dynamometer, an engine control unit (ECU), fuel supply units, combustion-pressure measurement equipment, exhaust gas analysers, intake-air measurement devices and a high-speed data acquisition system.

2.1 Engine and instrumentation

The experiments were performed using a single-cylinder spray-guided DISI engine. The engine specifications are as listed in Table 1. The engine was naturally aspirated with a throttle controlling the intake air flow rate. A direct current dynamometer was coupled to the engine to maintain a constant speed (±1 rpm). The engine was controlled by an in-house software written in LabView. All temperatures were measured with K-type thermocouple and all tests were conducted with coolant and lubricant oil temperatures maintained at 85±2 °C and at intake air temperature of 30±2 °C. Fig.1(a) illustrates the schematic of the engine testing system. There is an exhaust plenum chamber 8 times the swept volume of the cylinder located 15 cm downstream the exhaust valves which is very common in many experimental single-cylinder engine setups. The pressures fluctuations on the exhaust gases are reduced by passing it through the exhaust plenum before passing through the Lambda meter to measure the air fuel ratio. The engine test cell of the University of Birmingham used for the experiment were indicated in fig1(b)The load test was performed over the range of 3.5bar to 8.5bar IMEP at constant fuel injection timing of 280CADbTDC. And secondly the effect of fuel injection timing was investigated across the SOI sweep of 300CAD bTDC to 180CAD bTDC at the intermittent loads of 5.5 bar IMEP. Both tests were conducted at constant engine speed of 1500 rpm (±1 rpm) and at the stoichiometric air–fuel ratio (AFR stoichiometric) or (λ = 1). In other to keep the engine speed constant irrespective of its torque output, it has a DC dynamometer coupled to it. The intake airflow was stabilised through a 100L intake buffer. The fuel

IJSES
International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026.
ISSN (Online): 2456-7361

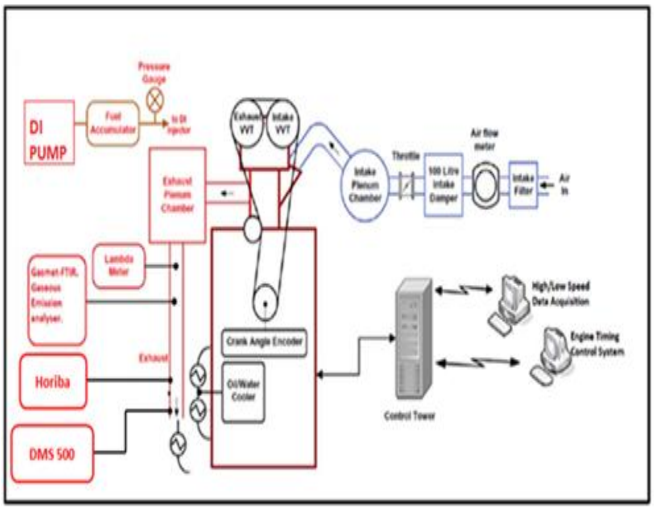


Fig 2.1 Experimental setup

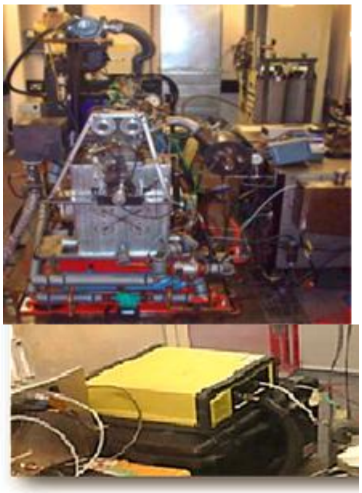


Fig 2,2: Test Cell. University of Birmingham, UK

TABLE 1: Engine Specification

Engine Type	4 Stroke ,4Valve
Displacement	497cc
Bore x Stroke	83mm x 92mm
Compression Ratio	11.5:1
Fuel Type, DI Pressure	ULG and MF , MTHF 1
Intake valve Opening	16.5^0 bTDC
Exhaust valve Closing	36.7^0 aTDC

injection pressure was maintained constant at 150 bar SGDI for the two tests. Table 2 displays the properties of the three tested fuels. The Gasoline used for the experiment was supplied by Shell Global Solutions, UK. ULG95 was chosen for this study because of its high-octane rating and its competitive characteristics market rating. The MF and MTHF used for the experiment were provided by Fisher Scientific, UK, with 99% purity level. Each of these experiments were conducted using the fuel-specific optimum spark timings, known as the Maximum Brake Torque (MBT) timings.

Test fuels Properties

TABLE 2: Base fuel Properties [5] [6-8] [15]

PROPERTIES	Gasoline	MF	MTHF
Molecular formula	C_2-C_{14}	C_5H_6O	$C_5H_{10}O$
Lower heating value MJ/Kg	42.9	31.2	34.12
Lower heating value MJ/L	31.9	27.63	28.97
Research octane number (RON)	96.8	103	88.2
Motor octane number (MON)	85.7	86	71.2
Stoichiometric air-fuel ratio	14.46	10.05	11.16
Heat of vaporisation (KJ/Kg)	373	358	375.3
Reid vapour Pressure (Kpa)	32.8	18.5	13.6
Density 20^0c (Kg/m3	744.6	913.2	854
H/C ratio	1.795	1.8	1.96
Gravimetric O_2 content.(%)	0	19.49	18.58
Initial boiling point (^{0}c)	32.8	64	80.3
Solubility in water (vol. %)	Negligible	0.3	12.1

Injection sweeps were performed for each fuel at interval ranging from 180bTDC to 280bTDC at 5.5bar IMEP. The crank shaft encoder signal determines the location of the piston relative to the top dead center (TDC) and is used by the ETCS software to control the injection, ignition, and variable valve timing. During this experiment all the control signals are based on the 1440TTL pulses per cycle clock signals (2pulse/cad). The gaseous emissions were quantified using a Horiba MEXA-7100DEGR gas tower. Exhaust samples were taken 0.3m downstream of the exhaust valve and pumped via a heated line (maintained at 191°C) to the analyser

III. Results and Discussion

3.1 Effect of Engine load.

The engine load was varied from 3.5bar IMEP to 8.5bar IMEP. The result of varying loads on the Peak cylinder Pressure Pmax, Coefficient of the combustion variation (Cov), combustion duration, fuel consumption, the indicated engine efficiency and the combustion efficiency are detailed below. The result of the gaseous emissions with the load variation was also detailed for the three fuels investigated. Horiba Mexa was used to measure the gaseous emission.

3.1.1 Cylinder Peak Pressure Pmax and Coefficient of Variation of IMEP (COV)

Figure 3.1 presents the variation of peak cylinder pressure (Pmax) and the coefficient of variation of IMEP (COVIMEP) with engine load for ULG, MF and MTHF. Peak cylinder pressure increased progressively with increasing IMEP for all three fuels, indicating enhanced combustion intensity at higher engine loads. The increase in Pmax is attributed to the greater fuel mass injected, higher in-cylinder pressure and temperature, and accelerated heat-release rate, which collectively improve combustion efficiency [16] [8]. Among the fuels,

MF consistently produced the highest peak cylinder pressure, followed by MTHF and ULG. At the highest load (8.5 bar IMEP), MF reached approximately 43 bar, compared with about 38 bar for MTHF and 35 bar for ULG. The superior performance of MF is attributed to its oxygenated molecular structure, favourable volatility and relatively high laminar flame speed, which promote rapid flame propagation and more complete combustion [8][9]. MTHF also generated higher peak pressures than ULG throughout the load range but remained slightly lower than MF. Although MTHF contains chemically bound oxygen, hydrogenation of the furan ring reduces its

International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026.
ISSN (Online): 2456-7361

combustion reactivity relative to MF, resulting in a slightly lower heat-release rate. Nevertheless, its combustion performance remained superior to conventional gasoline, demonstrating the potential of biomass-derived furan fuels to enhance combustion in DISI engines Figure 3.1(b) shows that combustion stability, represented by COVIMEP, remained below the commonly accepted stability limit of 5% for all fuels, confirming stable engine operation throughout the investigated load range [7]. ULG exhibited the highest COVIMEP values, whereas MF consistently recorded the lowest values, indicating superior cycle-to-cycle combustion stability. The improved stability of MF is attributed to its enhanced mixture preparation and faster combustion, which reduce cycle-to-cycle variations in flame development. MTHF displayed intermediate behaviour, reflecting its combustion characteristics between those of MF and ULG. The lower combustion variability observed for MF agrees with its higher peak cylinder pressure and more rapid combustion, suggesting a strong relationship between combustion stability and combustion intensity. Similar improvements in combustion stability for oxygenated furan fuels have been reported in previous DISI engine studies, where chemically bound oxygen enhanced flame propagation and reduced cyclic variability [13][11]

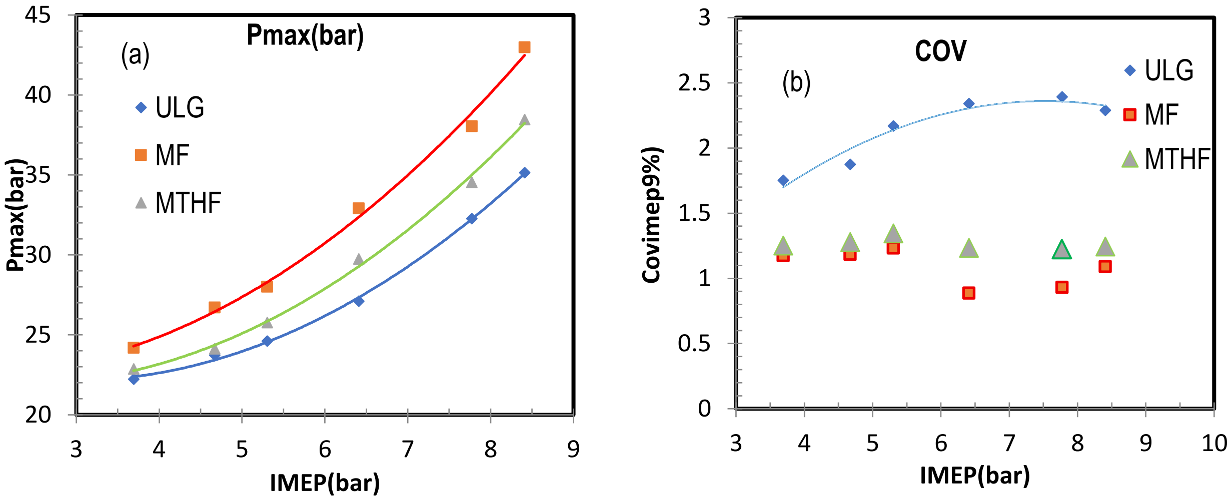


Figure 3.1. Effect of engine load (IMEP) on (a)Peak in-cylinder pressure (Pmax) and (b) Coefficient of variation of indicated mean effective pressure (COV of IMEP) for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF

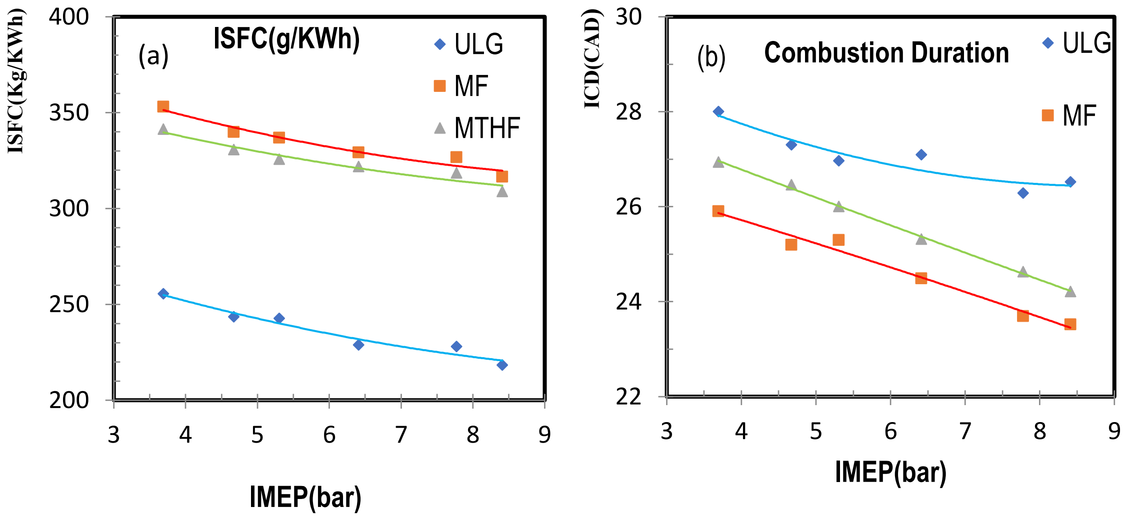


Figure 3.2. Effect of engine load (IMEP) on (a) indicated specific fuel consumption (ISFC) and (b) combustion duration (CD) for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF)

3.1.2 Effect of Engine Load on Indicated Specific Fuel Consumption and Combustion Duration.

Figure 3.2(a) presents the variation of indicated specific fuel consumption (ISFC) and combustion duration with engine load for ULG, MF and MTHF. For all fuels, ISFC decreased with increasing IMEP, indicating improved fuel conversion efficiency at higher engine loads. This behaviour is typical of spark-ignition engines because frictional and pumping losses become proportionally smaller as the engine produces more

IJSES
International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026.
ISSN (Online): 2456-7361

useful work, allowing a larger fraction of the fuel energy to be converted into indicated power [17].

Among the investigated fuels, ULG consistently exhibited the lowest ISFC, followed by MTHF and MF. The higher ISFC recorded for MF is primarily attributed to its lower lower heating value, which requires a greater mass of fuel to produce the same engine output despite its favourable combustion characteristics [17] [18]. MTHF displayed intermediate fuel consumption owing to its higher energy density relative to MF, confirming that hydrogenation of the furan ring improves fuel economy while maintaining the advantages of an oxygenated fuel. Figure 3.2(b) shows that combustion duration decreased progressively with increasing engine load for all three fuels. Higher engine loads increase in-cylinder pressure, temperature and turbulence intensity, accelerating flame propagation and shortening the combustion process [7][8]. MF consistently exhibited the shortest combustion duration, followed by MTHF and ULG. The rapid combustion of MF is attributed to its favourable volatility, chemically bound oxygen and relatively high laminar flame speed, which promote faster flame development and more complete oxidation. MTHF also burned faster than gasoline, but because its hydrogenated molecular structure reduces its combustion reactivity, it had slightly longer combustion durations than MF. The reduction in combustion duration corresponds closely with the increase in peak cylinder pressure observed in Figure 3.1, indicating that faster combustion concentrates heat release nearer top dead centre and improves pressure development during the expansion stroke. Consequently, the shorter combustion duration of MF contributes directly to its superior combustion efficiency and thermal performance. Similar improvements in combustion behaviour for oxygenated furan fuels have been reported by Wang et al. [6], who demonstrated that MF promotes faster combustion than gasoline in DISI engines.

3.1.3 Effect of Engine Load on Indicated Thermal Efficiency and Combustion Efficiency

Figure 3.3 illustrates the influence of engine load on indicated thermal efficiency (ITE) and combustion efficiency for ULG, MF and MTHF. Both parameters increased progressively with increasing IMEP, indicating more efficient conversion of fuel energy into useful work at higher loads.

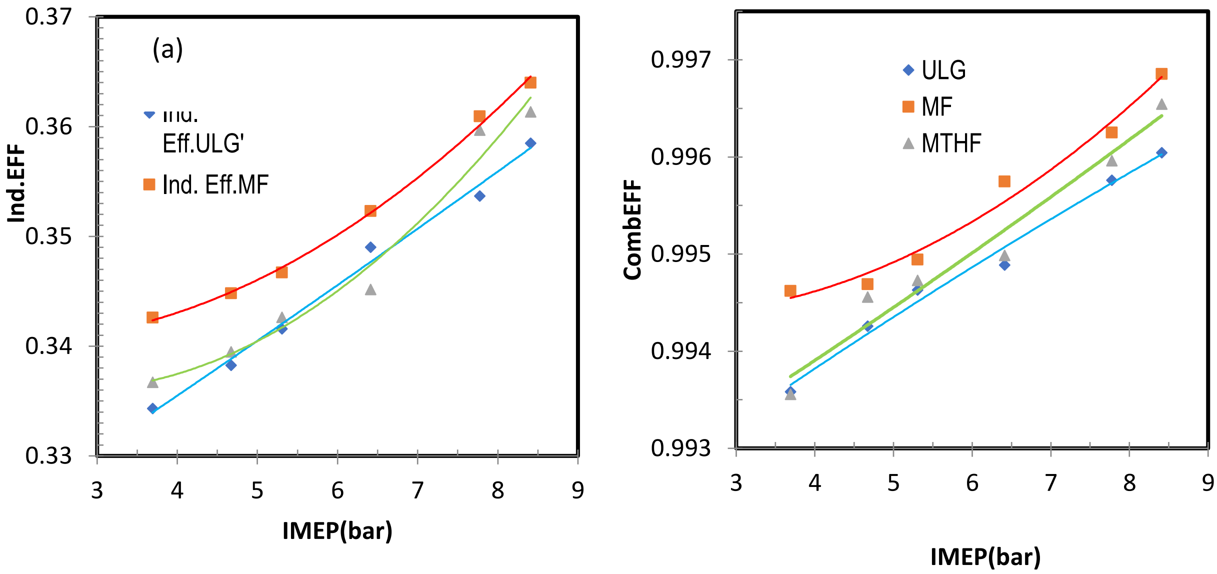


Figure 3.3. Effect of engine load (IMEP) on (a) indicated thermal efficiency (ITE) and (b) combustion efficiency for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF)

This improvement is attributed to reduced relative heat transfer and pumping losses, higher in-cylinder temperatures and pressures, and more effective combustion under increased load conditions [19]. At higher loads, a larger proportion of the fuel energy is released near top dead centre, resulting in improved thermal utilisation and engine efficiency. As indicated in figure 3.3(a) Among the fuels investigated, MF consistently exhibited the highest indicated thermal efficiency, followed by MTHF and ULG. At the maximum load, MF achieved an ITE of approximately 36%, compared with about 35% for MTHF and 34% for ULG. The superior efficiency of MF is attributed to its chemically bound oxygen, favourable volatility and high laminar flame speed, which enhance mixture preparation, accelerate flame propagation and improve heat release. Although MF possesses a lower heating value than gasoline, its improved combustion characteristics enable more effective utilisation of the available chemical energy, leading to higher thermal efficiency [9][20]. MTHF also outperformed gasoline but exhibited slightly lower efficiency than MF because its hydrogenated molecular structure reduces combustion reactivity. A similar trend was observed for combustion efficiency as indicated in figure 3.3(b), which increased steadily with engine load for all fuels. Higher combustion temperatures and improved turbulence promoted more complete oxidation of the air–fuel mixture, thereby reducing incomplete combustion losses. MF consistently recorded the highest combustion efficiency, approaching 99% at the highest load, while MTHF achieved intermediate values and ULG remained the least efficient. The superior combustion efficiency of MF reflects its enhanced oxidation capability arising from its oxygenated molecular structure, which facilitates more complete fuel oxidation and reduces the formation of incomplete combustion products such as CO and HC. These observations agree with previous studies demonstrating that oxygenated furan fuels improve combustion completeness and overall engine efficiency in DISI engines [10] [21].

3.2 Effect of Fuel Injection Timing on Combustion, Performance and Emission Characteristics

Fuel injection timing is one of the most influential calibration parameters governing the combustion process in direct-injection spark-ignition (DISI) engines because it directly controls fuel atomisation, spray penetration, droplet evaporation, air-fuel mixing and the subsequent development of combustion. Unlike port fuel injection systems, where fuel has sufficient residence time to evaporate and mix with the intake air before entering the combustion chamber, fuel preparation in DISI engines occurs entirely within the cylinder under rapidly changing pressure and temperature conditions. Consequently, even relatively small changes in injection timing can significantly modify mixture formation, combustion phasing, heat-release characteristics and pollutant formation. Optimising injection timing is therefore essential for achieving high thermal efficiency while simultaneously minimising fuel consumption and exhaust emissions [7][22].

3.2.1 Effect of Fuel Injection Timing on Peak Cylinder Pressure

Figure 3.4 illustrates the influence of fuel injection timing on peak cylinder pressure (Pmax) for ULG, MF and MTHF under different engine loads. Fuel injection timing significantly affected combustion development by influencing spray atomisation, fuel evaporation and air–fuel mixture formation before ignition. Advancing the start of injection generally increased peak cylinder pressure for all fuels, indicating improved combustion resulting from enhanced mixture preparation. Earlier injection provides additional time for fuel vaporisation and homogeneous charge formation, leading to faster flame propagation and higher in-cylinder pressure during combustion [23][8]. Among the investigated fuels, MF consistently produced the highest peak cylinder pressure, followed by MTHF and ULG over most injection timings. The superior combustion performance of MF is attributed to its chemically bound oxygen, favourable volatility and relatively high laminar flame speed, which accelerate heat release and improve combustion efficiency [24][5]. MTHF also generated higher peak pressures than ULG but remained slightly lower than MF because its hydrogenated molecular structure results in lower combustion reactivity. These observations demonstrate that fuel molecular structure strongly influences combustion characteristics even under identical injection strategies. Although advancing injection timing generally improved pressure development, excessive advancement did not always result in additional gains. Injecting fuel too early may increase spray-wall interaction and fuel film formation, reducing mixture quality and combustion efficiency. Consequently, each fuel exhibited an optimum injection timing at which peak cylinder pressure was maximised, indicating that injection timing should be optimised according to the physicochemical properties of the fuel rather than using a common calibration for all fuels. The higher peak cylinder pressures obtained with MF correspond closely with its shorter combustion duration and higher thermal efficiency discussed in the previous section, confirming that improved mixture preparation and faster combustion enhance overall engine performance. Similar behaviour has been reported in previous studies on oxygenated furan fuels, where optimised injection timing improved combustion efficiency and pressure development in DISI engines [10]251]

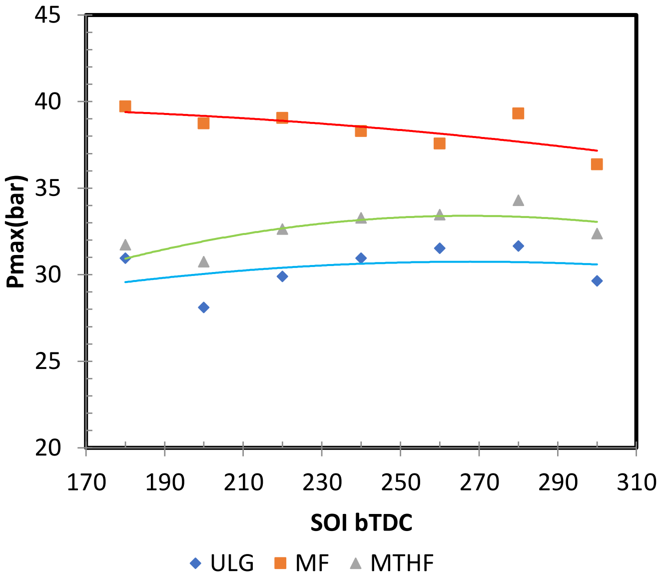


Figure 3.4. Effect of start of injection (SOI) timing on peak cylinder pressure (Pmax) for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

3.2.2 Effect of Fuel Injection Timing on Combustion Duration

Figure 3.5 shows the variation of combustion duration with fuel injection timing for ULG, MF and MTHF at different engine loads. Fuel injection timing had a significant influence on combustion duration because it directly affected fuel atomisation, evaporation and mixture homogeneity before ignition. In general, advancing the injection timing reduced combustion duration for all fuels, indicating faster combustion resulting from improved air–fuel mixing. Earlier injection provides sufficient time for fuel evaporation and homogeneous mixture formation, thereby accelerating flame propagation after spark ignition [7][8]. Among the fuels investigated, MF consistently exhibited the shortest combustion duration, followed by MTHF and ULG across most operating conditions. The superior combustion behaviour of MF is attributed to its chemically bound oxygen, favourable volatility and relatively high laminar flame speed, which promote rapid flame development and more complete combustion [9][24]. MTHF also burned faster than gasoline but showed slightly longer combustion durations than MF because its hydrogenated molecular structure lowers its combustion reactivity. Although

Volume 10, Issue 7, pp. 86-96, July 2026.

ISSN (Online): 2456-7361

earlier injection generally shortened combustion duration, excessive advancement did not always produce further improvement. Very early injection may increase spray-wall interaction and fuel film formation, leading to less favourable mixture distribution and diminishing the benefits of extended evaporation time. This indicates that each fuel possesses an optimum injection timing that balances mixture preparation with efficient combustion.

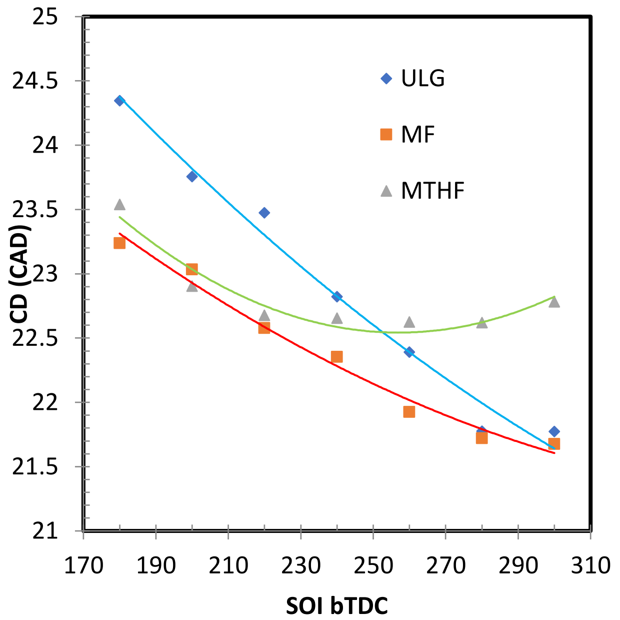


Figure 3.5. Effect of start of injection timing on combustion duration for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

3.2.3 Effect of Fuel Injection Timing on Combustion Efficiency

Figure 3.6 presents the effect of fuel injection timing on combustion efficiency for ULG, MF and MTHF under different engine loads. Combustion efficiency was strongly influenced by injection timing because it determines the quality of fuel atomisation, evaporation and air–fuel mixing before ignition. In general, advancing the injection timing improved combustion efficiency for all fuels by providing sufficient time for mixture preparation, thereby promoting more complete oxidation of the fuel [24]. Improved mixture homogeneity enhances flame propagation and reduces the formation of incomplete combustion products, resulting in more effective conversion of the fuel's chemical energy into useful heat. Among the fuels investigated, MF consistently achieved the highest combustion efficiency, followed by MTHF and ULG over most operating conditions. The superior performance of MF is attributed to its oxygenated molecular structure, favourable volatility and high laminar flame speed, which promote rapid and complete combustion [9]26]. MTHF also exhibited higher combustion efficiency than gasoline, reflecting the beneficial effect of chemically bound oxygen, although its combustion efficiency remained slightly lower than MF because of its lower combustion reactivity. While earlier injection generally enhanced combustion efficiency, excessive advancement did not produce further improvement. Very early injection may increase spray-wall interaction and fuel film formation, leading to local mixture inhomogeneity and reducing the benefits of extended evaporation time.

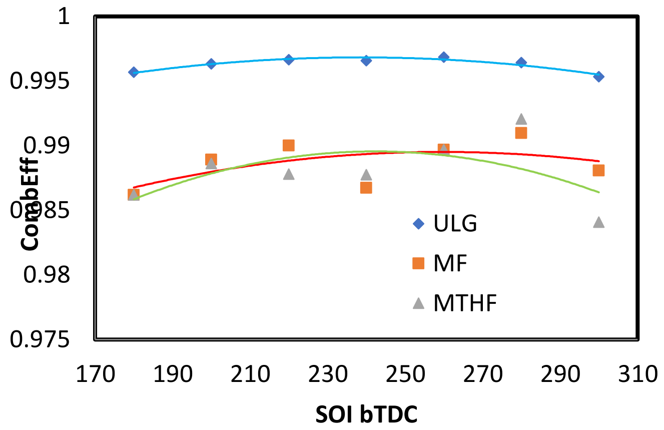


Figure 3.6. Effect of start of injection timing on combustion efficiency for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

3.2.4 Effect of Fuel Injection Timing on Indicated Specific Fuel Consumption

Figure 3.7 illustrates the influence of fuel injection timing on indicated specific fuel consumption (ISFC) for ULG, MF and MTHF under different engine loads. Fuel injection timing significantly affected fuel consumption by altering spray atomisation, evaporation and air–fuel mixture formation before ignition. In general, advancing the injection timing reduced ISFC for all fuels, indicating improved fuel utilisation due to enhanced mixture preparation and more complete combustion. Earlier injection provides additional time for fuel evaporation and charge homogenisation, resulting in faster combustion and more efficient conversion of fuel energy into useful work [27][8]. Among the investigated fuels, ULG consistently exhibited the lowest ISFC, followed by MTHF and MF across most operating conditions. The higher fuel consumption of MF is primarily attributed to its lower lower heating value, which requires a greater mass of fuel to produce the same engine output despite its superior combustion characteristics [9][5]. MTHF exhibited intermediate ISFC because its higher energy density relative to MF reduced the fuel mass required while still benefiting from the improved combustion characteristics associated with oxygenated fuels. Although advancing the injection timing generally improved fuel economy, excessive advancement did not continue to reduce ISFC. Very early injection may increase spray-wall impingement and fuel film formation, reduce combustion efficiency and offsetting the advantages of longer evaporation time. Consequently, each fuel exhibited an optimum injection timing that produced the minimum ISFC, highlighting the need for fuel-specific calibration in DISI engines. Similar trends have been reported for oxygenated furan fuels, where appropriate injection strategies improved combustion efficiency and reduced fuel consumption in DISI engines [10][21]

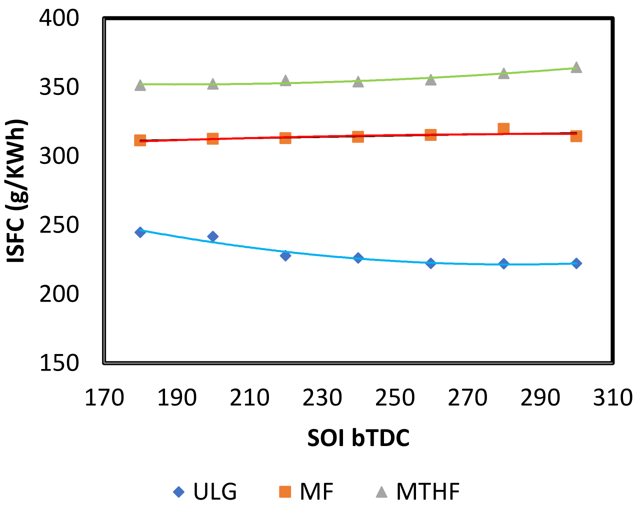


Figure 3.7. Effect of start of injection timing on indicated specific fuel consumption for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

3.2.5 Effect of Fuel Injection Timing on Indicated Thermal Efficiency

Figure 3.8 presents the effect of fuel injection timing on indicated thermal efficiency (ITE) for ULG, MF and MTHF under different engine loads. Fuel injection timing had a significant influence on thermal efficiency because it governs spray atomisation, fuel evaporation and mixture preparation before ignition.

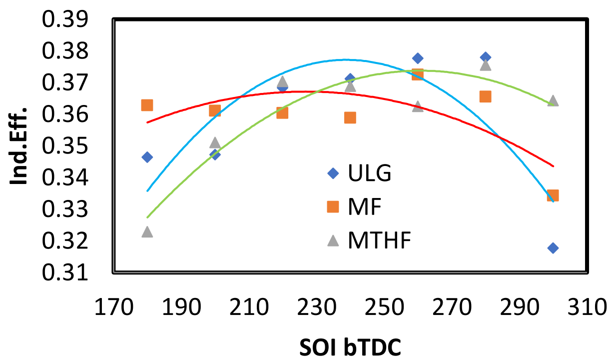


Figure 3.8. Effect of start of injection (SOI) timing on indicated thermal efficiency (ITE) for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

In general, advancing the injection timing increased ITE for all fuels, reflecting improved combustion efficiency and more effective conversion of fuel energy into useful work. Earlier injection promotes better air–fuel mixing and faster heat release, allowing a greater proportion of the combustion process to occur near top dead centre where energy conversion is most efficient [7][15]. Among the fuels investigated, MF consistently achieved the highest indicated thermal efficiency, followed by MTHF and ULG across most operating conditions. The superior performance of MF is attributed to its chemically bound oxygen, favourable volatility and relatively high laminar flame speed, which enhance mixture formation and accelerate combustion [9][7]. MTHF also exhibited higher thermal efficiency than gasoline, although its performance remained slightly below that of MF because the hydrogenation of the furan ring reduces combustion reactivity. These findings demonstrate that fuel molecular structure plays an important role in determining thermal efficiency under varying injection strategies

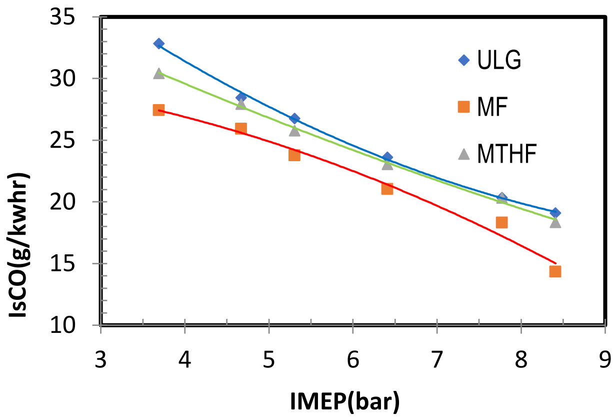
3.3 Effect of Engine Load on Gaseous Emissions of the Combusted Fuels

Although advancing injection timing generally improved thermal efficiency, excessive advancement produced only marginal gains and, in some cases, slightly reduced efficiency. This behaviour is attributed to increased spray-wall interaction and fuel film formation associated with very early injection, which can deteriorate mixture quality and reduce combustion effectiveness. Consequently, each fuel exhibited an optimum injection timing that maximised thermal efficiency, highlighting the need for fuel-specific engine calibration. The higher thermal efficiencies obtained with MF correspond closely with its higher peak cylinder pressure, shorter combustion duration and greater combustion efficiency discussed in the preceding sections, confirming that improved mixture preparation and faster combustion enhance overall engine performance. Similar improvements in thermal efficiency for oxygenated furan fuels have been reported in previous investigations of DISI engines [10][11]. The optimum SOI differs among the three fuels because of differences in volatility, latent heat of vaporisation, viscosity, density and combustion chemistry. The results therefore reinforce the necessity for fuel-specific injection calibration when renewable oxygenated fuels are introduced into modern DISI engines. Adopting a common gasoline calibration strategy for MF and MTHF would not fully exploit their combustion advantages and may result in unnecessary reductions in thermal efficiency The gaseous emissions produced during combustion provide valuable insight into the efficiency of the combustion process and the environmental performance of alternative fuels in direct-injection spark-ignition (DISI) engines. Unlike combustion and performance parameters, which primarily reflect the conversion of chemical energy into useful work, exhaust emissions are strongly influenced by the complex interactions among fuel molecular structure, combustion temperature, air-fuel mixing, oxygen availability and in-cylinder chemical kinetics. Changes in engine load significantly modify these conditions by altering cylinder pressure, combustion temperature, fuel injection quantity and residual gas concentration, thereby affecting the formation and oxidation of major gaseous pollutants such as carbon monoxide (CO), unburned hydrocarbons (HC) and nitrogen oxides (NO_x).

3.3.1 Effect of Engine Load on Carbon Monoxide (CO) Emissions

Figure 3.**9** shows the effect of engine load on indicated specific carbon monoxide (ISCO) emissions for ULG, MF and MTHF. For all fuels, CO emissions decreased with increasing engine load, indicating more complete combustion under higher load conditions

IJSES
International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026.
ISSN (Online): 2456-7361



monoxide (ISCO) emissions for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

Increased load raises the in-cylinder temperature and pressure, accelerating oxidation reactions and promoting the conversion of carbon monoxide to carbon dioxide during combustion [7][13]. Consequently, combustion becomes more complete, reducing the formation of CO. Among the investigated fuels, MF consistently produced the lowest CO emissions, followed by MTHF and ULG across the entire load range. Because of its chemically linked oxygen, which boosts local oxygen availability during combustion and encourages more complete oxidation of carbon-containing intermediates, MF has fewer CO emissions [9][5]. MTHF also emitted less CO than gasoline because of its oxygenated molecular structure, although its emissions remained slightly higher than MF due to its comparatively lower combustion reactivity. In contrast, ULG produced the highest CO emissions, reflecting its lower oxygen availability and less efficient oxidation process. The reduction in CO emissions corresponds closely with the higher combustion efficiency and thermal efficiency observed for MF and MTHF in the preceding sections. These results confirm that improved combustion quality directly reduces incomplete combustion products. Similar reductions in CO emissions from oxygenated furan fuels have been reported in previous DISI engine studies, where enhanced combustion efficiency promoted more complete fuel

3.3.2 Effect of Engine Load on Hydrocarbon (HC) Emissions

Figure 3.10 illustrates the influence of engine load on indicated specific hydrocarbon (ISHC) emissions for ULG, MF and MTHF. HC emissions decreased progressively with increasing engine load for all fuels, indicating more complete combustion under higher load conditions. As engine load increases, higher in-cylinder temperatures, pressures and turbulence enhance flame propagation and oxidation of unburned fuel, thereby reducing the amount of hydrocarbons escaping combustion [7][13]. Among the investigated fuels, MF consistently produced the lowest HC emissions, followed by MTHF and ULG throughout the load range. The superior performance of MF is attributed to its chemically bound oxygen, favourable volatility and high laminar flame speed, which improve fuel vaporisation, mixture homogeneity and flame propagation, leading to more complete oxidation of the fuel 239][5].

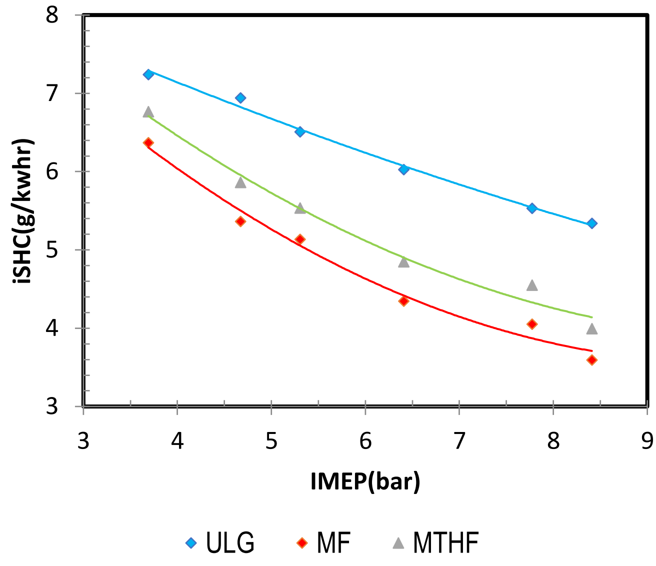


Figure 3.10. Effect of engine load (IMEP) on indicated specific hydrocarbon (ISHC) emissions for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

MTHF also emitted less HC than gasoline because its oxygenated molecular structure promotes more efficient combustion, although its emissions remained slightly higher than MF due to its comparatively lower combustion reactivity. In contrast, ULG recorded the highest HC emissions, reflecting less complete combustion associated with conventional gasoline. The lower HC emissions obtained with MF and MTHF are consistent with their higher combustion efficiency and shorter combustion duration discussed in the previous sections, demonstrating the close relationship between combustion quality and hydrocarbon formation. Similar reductions in HC emissions from oxygenated furan fuels have been reported in previous studies, where improved oxidation characteristics significantly reduced incomplete combustion products [6][11].

3.3.3 Effect of Engine Load on Nitrogen Oxides (NO_x) Emissions

Figure 3.11 presents the variation of indicated specific nitrogen oxides ($ISNO_x$) emissions with engine load for ULG, MF and MTHF. Unlike CO and HC emissions, NO_x emissions increased with increasing engine load for all fuels. This trend is attributed to the higher in-cylinder temperatures and pressures associated with increased load, which promote thermal NO_x formation through the extended Zeldovich mechanism [7][13]. Flame temperature is further increased by higher load combustion efficiency, accelerating nitrogen oxidation. Among the fuels investigated, MF consistently produced the highest NO_x emissions, followed by MTHF and ULG across the load range. The higher NO_x emissions of MF are associated with its oxygenated molecular structure and rapid combustion characteristics, which promote higher peak combustion temperatures and more complete oxidation [12][13].

International Journal of Scientific Engineering and Science
Volume 10, Issue 7, pp. 86-96, July 2026. ISSN (Online): 2456-7361

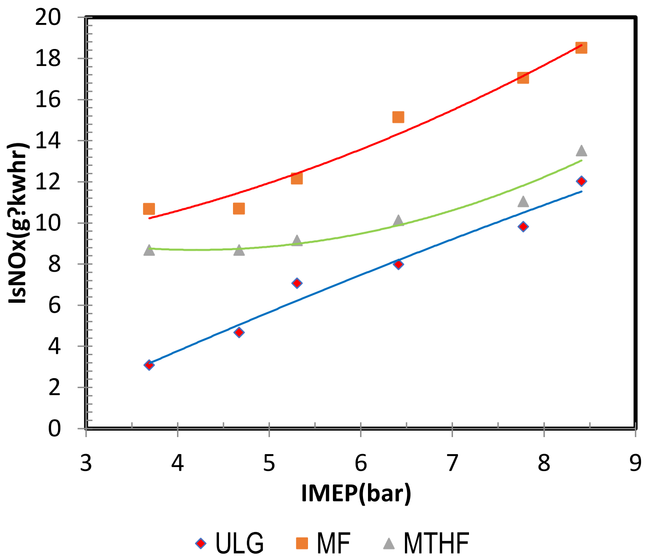


Figure 3.11. Effect of engine load (IMEP) on indicated specific nitrogen oxides ($ISNO_x$) emissions for unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF).

MTHF exhibited intermediate NO_x emissions because its combustion characteristics lie between those of MF and gasoline, while ULG consistently recorded the lowest NO_x emissions due to its comparatively lower combustion temperatures.

IV. Conclusion

This study experimentally investigated the combustion, performance and gaseous emission characteristics of unleaded gasoline (ULG), 2-methylfuran (MF) and 2-methyltetrahydrofuran (MTHF) in a direct-injection spark-ignition (DISI) engine under varying engine loads and fuel injection timings. The objective was to evaluate the suitability of biomass-derived oxygenated fuels as sustainable alternatives to conventional gasoline while identifying the combustion mechanisms responsible for their observed performance. The results demonstrate that both MF and MTHF possess considerable potential as renewable fuels for advanced gasoline engines, although their performance is strongly influenced by operating conditions and engine calibration.

Engine load exerted a significant influence on combustion development, engine performance and emission formation. Increasing IMEP consistently increased peak cylinder pressure, indicated thermal efficiency and combustion efficiency while reducing combustion duration, indicated specific fuel consumption, combustion variability, carbon monoxide (CO) and unburned hydrocarbon (HC) emissions for all investigated fuels. These improvements were attributed to enhanced in-cylinder temperature, pressure and turbulence intensity, which promoted faster flame propagation, more complete oxidation and improved conversion of chemical energy into useful work. Conversely, increasing engine load resulted in higher nitrogen oxide (NO_x) emissions because the elevated combustion temperatures favoured thermal NO_x formation through the extended Zeldovich mechanism.

Among the investigated fuels, 2-methylfuran consistently demonstrated the most favourable combustion characteristics. MF produced the highest peak cylinder pressure, the shortest combustion duration, the lowest cycle-to-cycle combustion variability and the highest indicated thermal and combustion efficiencies across most operating conditions. These improvements are attributed to its chemically bound oxygen, favourable volatility and relatively high laminar flame speed, which collectively enhanced mixture preparation, accelerated flame propagation and promoted more complete combustion. The superior combustion quality of MF also resulted in the lowest CO and HC emissions throughout the investigated load range, confirming that oxygenated renewable fuels effectively reduce incomplete-combustion products.

Although MF exhibited the highest indicated specific fuel consumption compared with gasoline, this behaviour primarily reflects its lower heating value rather than inferior combustion efficiency. In contrast, MTHF demonstrated intermediate performance, combining lower fuel consumption than MF with combustion characteristics superior to those of conventional gasoline. The results therefore indicate that hydrogenation of the furan ring modifies combustion behaviour by increasing fuel energy density while slightly reducing combustion reactivity. Consequently, MTHF provides a practical compromise between combustion efficiency, fuel economy and exhaust emissions.

. MF maintained comparatively high combustion efficiency over a relatively broad injection timing window, whereas MTHF exhibited greater sensitivity to injection timing, indicating that fuel-specific calibration is essential for maximising its combustion performance.

The emission characteristics further demonstrated the influence of fuel chemistry on combustion quality. MF consistently produced the lowest CO and HC emissions owing to its enhanced oxidation capability and rapid combustion characteristics. However, these advantages were accompanied by higher NO_x emissions than those observed for ULG and MTHF because the improved combustion efficiency increased peak flame temperatures and promoted thermal NO_x formation.

Overall, the present investigation demonstrates that biomass-derived furan fuels represent promising renewable alternatives to conventional gasoline for direct-injection spark-ignition engines. Among the fuels investigated, MF exhibited the greatest potential for improving combustion quality, thermal efficiency and reducing incomplete-combustion emissions, while MTHF provided a balanced compromise between combustion performance, fuel economy and emission characteristics. The results further confirm that optimum engine calibration, particularly fuel injection timing, is fuel-dependent and should be specifically tailored to the physicochemical properties of renewable oxygenated fuels rather than adopting conventional gasoline calibration strategies. These findings contribute to the growing body of knowledge supporting the application of sustainable oxygenated biofuels in next-generation.

Volume 10, Issue 7, pp. 86-96, July 2026. ISSN (Online): 2456-7361

V. Recommendations for Future Work

Future research should investigate the combined optimisation of fuel injection timing, spark timing and exhaust gas recirculation (EGR) to minimise the NO_x penalty associated with oxygenated fuels while preserving their combustion advantages. Additional studies employing optical diagnostics, detailed heat-release analysis and computational fluid dynamics (CFD) are also recommended to provide deeper understanding of spray evolution, mixture formation and combustion mechanisms of MF and MTHF under transient engine operating conditions. Furthermore, long-term durability studies, catalyst compatibility assessments and investigations involving higher engine speeds and loads would provide additional information necessary for the commercial implementation of biomass-derived furan fuels in future gasoline direct-injection engine.

References

[1] J. Abraham, F. V. Bracco, and R. D. Reitz, "Comparison of computed and measured premixed charge engine combustion," *Combustion and Flame*, vol. 60, pp. 309–322, 1985, doi: 10.1016/0010-2180(85)90106-5.
[2] P. Anselmi, L. Jan, M. Matrat, G. Maio, and B. Xu, "Renewable fuel and blend properties for the reduction of GHG emissions under different spark-ignited engine architectures," *Fuel*, vol. 353, Art. no. 129194, 2023, doi: 10.1016/j.fuel.2023.129194.
[3] J. Li, L. Ye, S. Gong, X. Deng, S. Wang, R. Liu, and W. Yang, "Review on the combustion progress and engine application of tailor-made fuels from biomass," *Renewable and Sustainable Energy Reviews*, vol. 192, Art. no. 114242, 2024, doi: 10.1016/j.rser.2023.114242.
[4] A. T. Hoang et al., "2-Methylfuran (MF) as a potential biofuel: A thorough review on the production pathway from biomass, combustion progress, and application in engines," *Renewable and Sustainable Energy Reviews*, vol. 148, Art. no. 111265, 2021, doi: 10.1016/j.rser.2021.111265.
[5] T. Le, H. Nguyen, A. T. Hoang, and V. V. Pham, "Methyltetrahydrofuran and 2,5-dimethyltetrahydrofuran: A review of heterogeneous catalytic approaches," RSC Advances, vol. 15, pp. 26537–26551, 2025.
[6] C. Wang et al., "Combustion characteristics and emissions of 2-methylfuran compared with 2,5-dimethylfuran, gasoline and ethanol in a DISI engine," *Fuel*, vol. 103, pp. 200–211, 2013, doi: 10.1016/j.fuel.2012.05.043.
[7] H. Wei et al., "Experimental investigation on the combustion and emissions characteristics of 2-methylfuran gasoline blend fuel in a spark-ignition engine," Applied Energy, vol. 132, pp. 317–324, 2014, doi: 10.1016/j.apenergy.2014.07.009.
[8] J. B. Heywood, Internal Combustion Engine Fundamentals, 2nd ed. New York, NY, USA: McGraw-Hill Education, 2018.
[9] H. Zhao, Ed., Advanced Direct Injection Combustion Engine Technologies and Development. Cambridge, U.K.: Woodhead Publishing, 2010.
[10] B. Sun, H. Tian, and F. Liu, "The distinctive characteristics of combustion duration in hydrogen internal combustion engine," *Int. J. Hydrogen Energy*, vol. 39, no. 26, pp. 14472–14478, 2014.
[11] M. Mohamed et al., "Impact of RON on a heavily downsized boosted SI engine using 2nd generation biofuel—A comprehensive experimental analysis," *Energy Conversion and Management*: X, vol. 22, Art. no. 100557, 2024.
[12] S. Coşman, "Experimental research of the effects of benzoylthiourea derivative fuel and gasoline mixtures on engine performance and emissions," *Int. J. Automotive Sci. Technol*., vol. 7, no. 4, pp. 403–409, 2023.
[13] B. Flach et al., EU-28 Biofuels Annual 2017, GAIN Report NL7015. Washington, DC, USA: USDA Foreign Agricultural Service, 2019.
[14] R. K. Olalere, Combustion Performance and Emissions Characteristics of 2-Methyltetrahydrofuran and Cyclopentanone in a DISI Engine. Ph.D. dissertation, University of Birmingham, Birmingham, U.K., 2025.
[15] H. Liu, R. K. Olalere, C. Wang, X. Ma, and H. Xu, "Combustion characteristics and engine performance of 2-methylfuran compared with gasoline and ethanol in a direct-injection spark-ignition engine," *Fuel*, vol. 299, Art. no. 120825, 2021, doi: 10.1016/j.fuel.2021.120825.
[16] M. Thewes et al., "Analysis of the impact of 2-methylfuran on mixture formation and combustion in a direct-injection spark-ignition engine," *Energy & Fuels*, vol. 25, no. 12, pp. 5549–5561, 2011, doi: 10.1021/ef201021a.
[17] R. K. Olalere et al., "Experimental study of combustion and emissions characteristics of low blend ratio of 2-methylfuran/2-methyltetrahydrofuran with gasoline in a DISI engine," *Fuel,* vol. 382, Art. no. 133799, 2025, doi: 10.1016/j.fuel.2024.133799.
[18] M. Pan et al., "Performance comparison of 2-methylfuran and gasoline on a spark-ignition engine with cooled exhaust gas recirculation," *Fuel*, vol. 132, pp. 36–63, 2014, doi: 10.1016/j.fuel.2014.04.054.
[19] S. R. Turns, An Introduction to Combustion: Concepts and Applications, 3rd ed. New York, NY, USA: McGraw-Hill Education, 2013.
[20] L. Liang et al., "Spray and dynamic heat transfer induced by 2-methylfuran transient spray impingement on piston surface," *Int. J. Thermal Sciences*, vol. 188, Art. no. 108154, 2023, doi: 10.1016/j.ijthermalsci.2023.108154.
[21] Y. Liang et al., "Experimental investigation of spray impingement characteristics of 2-methylfuran under gasoline direct injection conditions," Int. J. Thermal Sciences, vol. 187, 2023.
[22] Y. Román-Leshkov et al., "Production of dimethylfuran for liquid fuels from biomass-derived carbohydrates," Nature, vol. 447, no. 7147, pp. 982–985, 2007, doi: 10.1038/nature05923.
[23] H. Xu et al., "Fuel injection strategies for gasoline direct-injection engines using renewable oxygenated fuels," SAE Technical Paper, 2013.
[24] H. Zhao et al., "Metal chlorides in ionic liquid solvents convert sugars to 5-hydroxymethylfurfural," *Science*, vol. 316, no. 5831, pp. 1597–1600, 2007, doi: 10.1126/science.1141199.
[25] H. Liu et al., "An experimental study on particle evolution in the exhaust gas of a direct injection SI engine," Applied Energy, vol. 260, Art. no. 114220, 2020.
[26] E. Austin, K. T. Hoang, and collaborators, Evaluation of Gasmet™ DX-4015 Series Fourier Transform Infrared Gas Analyzer. DTIC Technical Report, 2009.
[27] R. K. Olalere, G. Zhang, and H. Xu, "Experimental investigation of gaseous emissions and hydrocarbon speciation for MF and MTHF gasoline blends in DISI engine," *International Journal of Automotive Manufacturing and Materials*, vol. 3, no. 1, p. 6, 2024.