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บทความวิจัย

Influence of Engine Oil Viscosity on the Combustion and Performance Characteristics of Gasoline Direct Injection Engines

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Abstract: The present study examines the influence of lubricating oil viscosity on the combustion dynamics and overall performance of a gasoline direct injection (GDI) engine. Experimental evaluations were conducted using three multigrade oils (SAE 10W-30, 10W-40, and 20W-50) at a constant engine speed of 2000 rpm across low (10 Nm) and medium (50 Nm) load conditions. The rheological analysis confirmed an exponential decrease in viscosity with rising temperatures, with SAE 20W-50 exhibiting the highest flow resistance. Engine performance data revealed that the lowest viscosity variant (SAE 10W-30) minimized frictional losses, thereby maximizing peak cylinder pressure, indicated mean effective pressure, and fuel economy. Furthermore, SAE 10W-30 recorded the optimal brake thermal efficiency ($\approx$25%). Although SAE 20W-50 offered enhanced film strength, it resulted in marginally reduced efficiency. Under medium-load conditions, the thermal reduction of viscosity diminished the performance variations among the tested oils. The study concludes that strategic selection of lower-viscosity lubricants can substantially improve the mechanical efficiency and fuel economy of GDI engines under partial loads, while maintaining adequate component durability.

Keywords: GDI engine; Lubricant viscosity; Combustion performance; Friction losses; Brake thermal efficiency

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1. Introduction

The rapid development of gasoline direct injection (GDI) technology has enabled higher thermal efficiency and reduced fuel consumption compared to conventional port fuel injection (PFI) engines. By injecting fuel directly into the combustion chamber, GDI engines achieve better charge cooling, precise mixture control, and improved part-load efficiency [1]. However, these advantages come at the expense of increased combustion pressure, temperature, and particulate emissions, which impose greater mechanical and thermal stresses on the lubrication system [2].

Lubricating oil plays a critical role in maintaining hydrodynamic film strength, minimizing wear, and reducing frictional losses in engine components such as piston rings, cylinder liners, and bearings. The viscosity of the oil governs its flow resistance and film thickness, which directly influence both mechanical efficiency and heat transfer characteristics of the engine [3]. Low-viscosity oils can reduce frictional and pumping losses, enhancing fuel economy, but may also lead to insufficient film formation and increased wear under high temperature and load conditions [4]. Conversely, high-viscosity oils provide stronger lubrication protection but can increase parasitic losses, particularly during cold start or low-load operation [5].

Previous studies have examined the effects of lubricant viscosity on friction and fuel consumption in spark-ignition engines. For instance, Fernando. F Rovai et al [6]. reported that using lower-viscosity lubricants significantly reduced friction mean effective pressure (FMEP) and improved brake-specific fuel consumption (BSFC). Similarly, found that viscosity influences the combustion duration and in-cylinder pressure rise rate, particularly in engines operating at low speeds or light loads. However, most of these studies were conducted using PFI configurations or simplified friction rigs, with limited data available for modern GDI engines under realistic operating conditions [7].

Recent research emphasizes that lubricant viscosity plays a pivotal role in optimizing engine dynamics and mechanical efficiency by directly influencing internal resistance. Lowering lubricant viscosity significantly reduces frictional losses within the piston assembly and bearings, which enhances the stability of in-cylinder pressure profiles and leads to an increase in Brake Mean Effective Pressure (BMEP) [8,9]. Furthermore, the rheological properties of the oil affect the engine's thermal management; high-viscosity fluids can induce localized heating that reduces intake air density, whereas optimized low-viscosity lubricants maintain a more favorable thermal gradient, thereby preserving the airflow rate and volumetric efficiency [9].

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From a thermodynamic perspective, the transition toward ultra-low viscosity oils is a primary strategy for improving thermal efficiency and reducing fuel consumption. Experimental evidence indicates that minimizing parasitic pumping work and hydrodynamic drag allows for a measurable reduction in brake-specific fuel consumption. Modern formulations, often utilizing advanced friction modifiers, enable these thinner oil films to operate effectively without compromising engine durability, ensuring that a greater portion of combustion energy is converted into effective work [8].

In GDI engines, lubricant–fuel interaction is particularly complex due to direct wall impingement, higher cylinder pressure, and increased blow-by gas recirculation. These factors can alter the viscosity and oxidation state of the oil during operation, subsequently affecting combustion stability, heat release behavior, and overall engine efficiency [10]. Therefore, understanding how base oil viscosity impacts the combustion process and energy conversion efficiency under different loads is essential for optimizing both performance and durability.

The present study aims to investigate the effect of engine oil viscosity on the combustion and performance characteristics of a GDI engine. Three multigrade lubricants SAE 10W-30, 10W-40, and 20W-50 were tested under low (10 Nm) and medium (50 Nm) load conditions at 2000 rpm. The analysis covers viscosity–temperature behavior, in-cylinder pressure development, mean effective pressures, airflow and fuel consumption, and brake thermal efficiency. The objective is to establish the correlation between oil viscosity and mechanical efficiency in GDI operation, providing insights into selecting lubricants that optimize both fuel economy and engine protection.

2. Experimental apparatus

2.1 Viscosity Measurement Setup

The viscosity characteristics of three multigrade lubricating oils—SAE 10W-30, 10W-40, and 20W-50 were experimentally determined using a falling-ball viscometer method. The schematic and photographs of the experimental setup are shown in Fig. 1. Each oil sample was poured into a cylindrical glass tube to a level slightly above two reference marks, corresponding to a vertical travel distance of 25 cm between the start and end points. The measurement procedure was performed at controlled oil temperatures ranging from 25 °C to 90 °C, in increments of approximately 10 °C.

To achieve the desired test temperatures, the glass tube containing the oil was immersed in a thermostatic water bath equipped with a thermocouple for continuous temperature monitoring and an electrical heating element to maintain uniform heat distribution throughout the sample. Once the target temperature was

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stabilized (±0.5 °C), a steel sphere with a diameter of 50.8 mm was carefully released from the upper reference mark and allowed to fall freely through the oil column under the influence of gravity.

The time required for the sphere to traverse the 25-cm path between the START and END reference points was measured using a digital stopwatch. Each experiment was repeated three times to ensure repeatability, and the average value was used for subsequent analysis. The terminal velocity of the ball was then computed and used to determine the kinematic viscosity of each lubricant according to Stokes' law, which relates viscous drag, buoyant force, and gravitational acceleration [11].

2.2 GDI Engine setup

The schematic representation of the experimental setup is shown in Fig. 2, which was designed to investigate the combustion and performance characteristics of a gasoline direct injection (GDI) engine lubricated with oils of varying viscosity grades. The test bench utilized a four-cylinder, four-stroke Mazda Skyactiv-G P3 GDI engine, which was coupled to a Lammed DW160H eddy-current dynamometer. Engine load and speed were precisely regulated using an FDJ-001 control interface, enabling steady-state operation across the designated torque and speed conditions. The key technical specifications of the GDI engine are summarized in Table 1.

Table 1 Specifications of the test engine.

Parameter	Specification
Brand/Model	Mazda / Skyactiv-G P3
Engine type	4-stroke, liquid-cooled, inline 4-cylinder
Bore × Stroke (mm)	71.0 × 82.0
Displacement (cm^3)	1298
Compression ratio	12.0:1
Rated power	69.35 kW @ 5800 rpm
Maximum torque	123 Nm @ 4000 rpm
Fuel injection type	Direct injection

In-cylinder pressure was recorded using a Kistler 6052C-3-1 (Switzerland) piezoelectric pressure transducer, synchronized with crank angle measurements obtained from a Kistler 2614CK encoder. Both signals were routed to a DEWESoft combustion analysis system (Slovenia) for high-resolution data acquisition and processing. The engine's ignition and injection parameters were managed by a Link G4+ Force GDI ECU, which provided precise control of fuel timing and pressure. The GDI injector employed was a six-hole multi-hole injector, operating at an injection pressure of 100 bar with an injection timing of 330° before top dead center (bTDC).

Commercial-grade Gasohol E20 (20% ethanol blended gasoline) served as the test fuel throughout the experiment. Its key properties are summarized in Table 2. The fuel was supplied through a dual-stage pumping system comprising a low-pressure (LP) and high-pressure (HP) fuel

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pump, integrated with a calibrated weight-based fuel measurement system to ensure accurate fuel flow monitoring.

To minimize confounding factors, each test point was recorded only after thermal stabilization. The engine was warmed up until coolant temperature and lubricant temperature at the measurement location reached steady values (within ±1–2 °C for at least 3–5 min). The dynamometer-controlled speed and load at the target setpoints (2000 rpm; 10 Nm and 50 Nm), and the same fuel (Gasohol E20) and stock engine calibration were retained throughout all tests. Combustion data were then acquired over 200 consecutive cycles and ensemble-averaged to reduce cycle-to-cycle variability. A sample size of 200 cycles has been established as statistically sufficient for reliable assessment of IMEP cycle-to-cycle variation in SI and GDI engines.

The engine speed of 2000 rpm was selected as it represents a typical part-load urban cruising condition for this class of 1.3 L GDI engine. At this speed, the engine operates well below its rated output (5800 rpm), and viscous frictional losses constitute a proportionally larger fraction of total mechanical losses. This operating point therefore maximises the sensitivity of the performance measurements to differences in lubricant viscosity, making it the most appropriate condition for isolating and quantifying oil viscosity effects.

Table 2 Properties of Gasohol E20 [11]

Property	Value
Lower heating value (MJ/kg)	39.3
Density (kg/L)	0.757
Research Octane Number (RON)	95.2
Motor Octane Number (MON)	83.4
Carbon (% mass)	78
Hydrogen (% mass)	13
Oxygen (% mass)	8
H/C ratio (molar)	2
O/C ratio (molar)	0.077

The engine was operated under closed-loop stoichiometric control (λ = 1.0) throughout all experiments. The same fuel (Gasohol E20), injection timing (330° bTDC), injection pressure (100 bar), and ECU calibration were retained across all oil grades and test conditions, ensuring that combustion chemistry remained consistent and that observed performance differences are attributable solely to lubricant viscosity.

Table 3 summarises the mean stabilised thermal conditions recorded for each oil grade and load condition. For all tests, the lubricant temperature stabilised at 90 °C ± 1 °C, and the coolant inlet temperature was maintained at 80 °C while the coolant outlet remained at 87–90 °C across all three oil grades and both load conditions, confirming that the thermal boundary conditions were controlled equally and consistently across all tests. Performance differences observed are therefore attributable to the intrinsic viscosity characteristics of each lubricant grade.

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Table 3 Mean stabilised thermal conditions for each test condition (coolant temperatures controlled equally across all three oil grades)

Oil Grade (SAE)	Load Condition	Oil Temp. (°C)	Coolant Inlet (°C)	Coolant Outlet (°C)
10W-30	Low (10 Nm)	90	80	87
10W-30	Medium (50 Nm)	91	80	90
10W-40	Low (10 Nm)	89	80	87
10W-40	Medium (50 Nm)	91	80	90
20W-50	Low (10 Nm)	90	80	87
20W-50	Medium (50 Nm)	91	80	90

2.3 Calculation and Data Analysis

2.3.1 Kinematic viscosity from the falling-ball method

A falling-ball viscometer was used to obtain the lubricant kinematic viscosity as a function of temperature under a consistent, comparative protocol. The steel sphere traveled a known distance L = 0.25 m, and the travel time (t) was recorded; the corresponding terminal velocity (V) was calculated as:

$$V=\frac{L}{t} \tag{1}$$

Assuming steady, laminar falling-sphere motion (Stokes regime), the dynamic viscosity (μ) can be expressed as:

$$\mu=\frac{2r^2g(\rho_s-\rho_f)}{9v} \tag{2}$$

where r is the sphere radius (m), g is gravitational acceleration (m s^{-2}), ρ_s is the sphere density (kg m^{-3}), and ρ_f is the lubricant density (kg m^{-3}). The kinematic viscosity (V) was then obtained from:

$$v=\frac{\mu}{\rho_f} \tag{3}$$

Each condition was repeated three times, and the average value was reported. Because the primary objective is to compare lubricants under identical conditions, the falling-ball results are used mainly to establish relative viscosity–temperature trends between SAE grades. Any potential departures from ideal Stokes assumptions (e.g., wall effects or transitional Reynolds number) are treated consistently across all oils because the same geometry and procedure were applied.

2.3.2 Indicated work and IMEP

Cycle-resolved in-cylinder pressure p (θ) was recorded and synchronized with crank angle θ. The indicated work Wi over one engine cycle was obtained from the p–V integral:

$$W_i=\oint PdV \tag{4}$$

The indicated mean effective pressure (IMEP) was calculated as:

$$IMEP=\frac{W_i}{V_d} \tag{5}$$

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where V_d is the total displaced volume of the engine (or the displaced volume of the measured cylinder, depending on the reporting basis). In this work, the combustion analysis software (DEWESoft) was used to compute indicated metrics consistently from the measured pressure traces, while the definitions above are provided for completeness and transparency.

2.3.3 Brake power, BSFC, BMEP and BTE

Brake torque(T) and speed(N) were measured by the dynamometer. Brake power (P_b) was calculated as:

$$P_b=2\pi NT \tag{6}$$

with N in revolutions per second (or converted consistently from rpm). The brake specific fuel consumption (BSFC) was calculated as:

$$BSFC=\frac{\dot{m}_f}{P_b} \tag{7}$$

Where $\dot{m}_f$ is the fuel mass flow rate.

For a four-stroke engine, brake mean effective pressure (BMEP) was calculated as:

$$BMEP=\frac{4\pi T}{V_d} \tag{8}$$

Brake thermal efficiency (BTE) was calculated from the fuel lower heating value (LHV) as:

$$BTE=\frac{P_b}{\dot{m}_f \cdot LHV} \tag{9}$$

2.3.4 Friction means effective pressure

The friction means effective pressure (FMEP) was estimated from:

$$FMEP=IMEP-BMEP \tag{10}$$

3. Results and Discussion

The kinematic viscosity of engine oils with different SAE grades (10W-30, 10W-40, and 20W-50) was experimentally evaluated using a falling-ball viscometer setup. In this test, a steel ball with a diameter of 50.8 mm was allowed to move through each oil sample over a fixed vertical distance of 25 cm under steady-state conditions, while the time required for the ball to travel this distance was recorded at various oil temperatures ranging from 30 °C to 90 °C. The velocity of the ball was subsequently calculated, and the kinematic viscosity was determined based on Stokes' law and the relationship between viscous drag, buoyant force, and gravitational acceleration [11]. As illustrated in Fig. 3a, the time required for the ball to traverse the oil column decreases exponentially with increasing temperature for all oil grades. This trend reflects the reduction in internal friction as the oil molecules gain kinetic energy at elevated temperatures. Among the samples, SAE 20W-50 exhibited the longest fall time across all temperatures, indicating the highest viscosity and resistance to flow.

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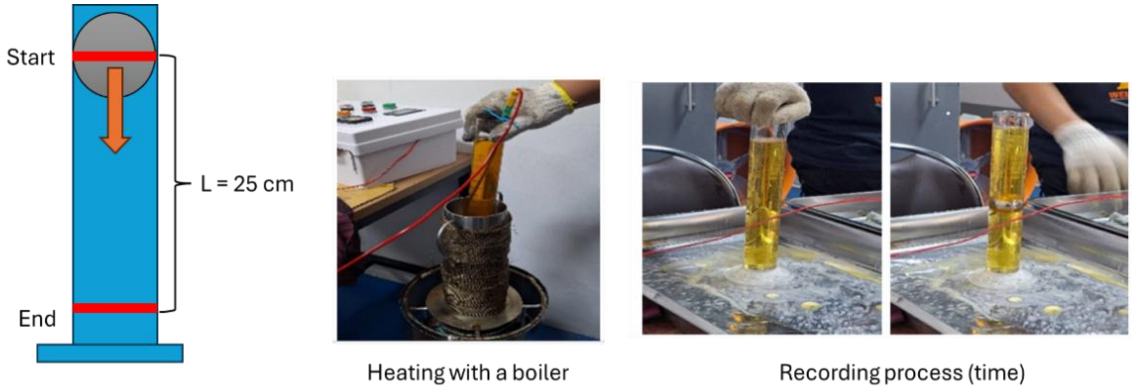

Fig. 1 Experimental procedure for the falling-ball viscosity measurement

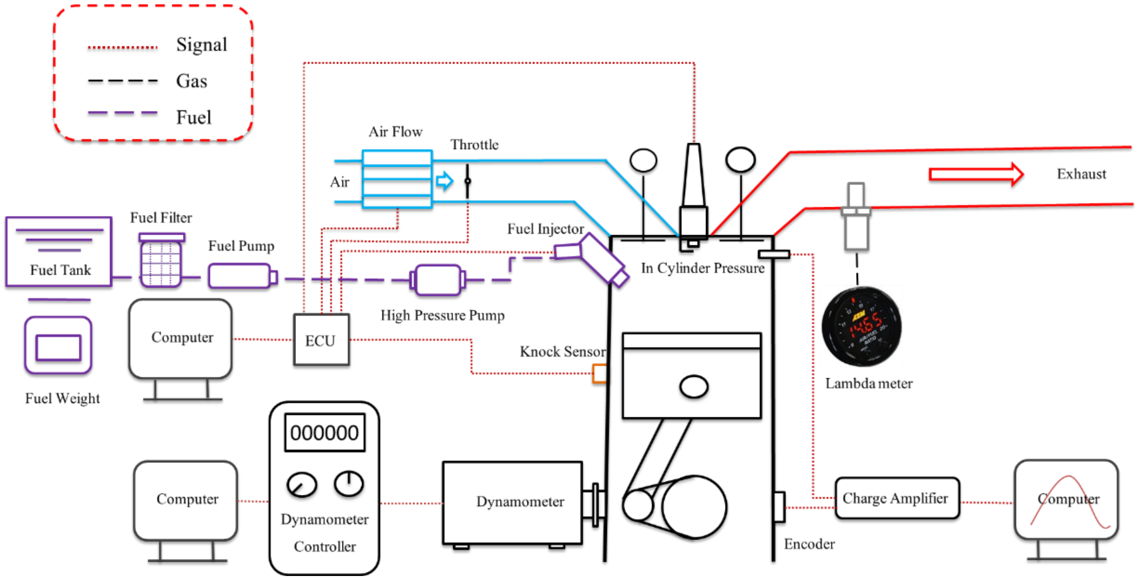

Fig. 2 The schematic of Schematic of GDI engine setup.

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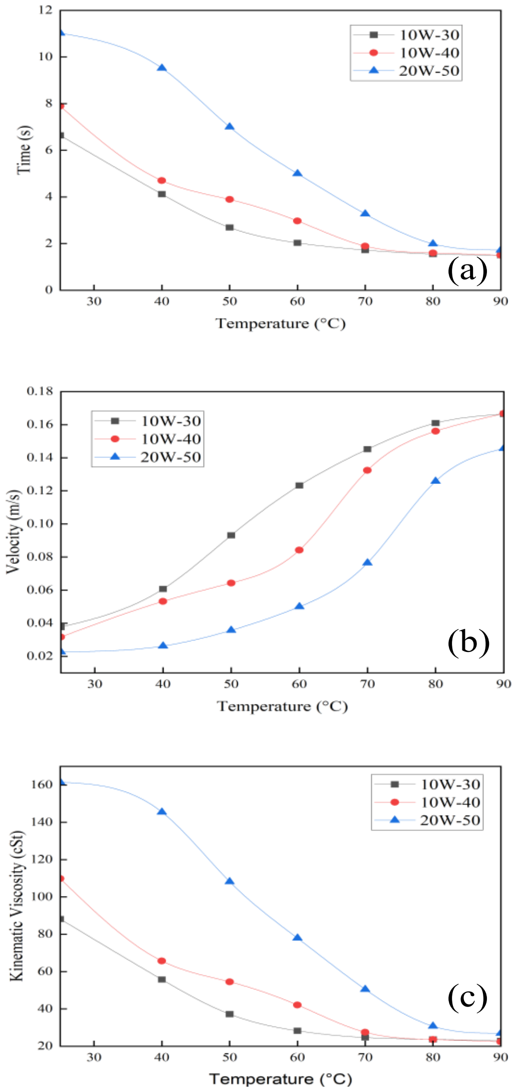

Fig. 3 Variation of (a) falling time, (b) velocity, and (c) kinematic viscosity of different engine oils (SAE 10W-30, 10W-40, and 20W-50) as a function of temperature (30–90 °C) determined using a falling-ball viscometer with a 50.8 mm steel ball over a 25 cm path length

In contrast, SAE 10W-30 showed the shortest fall time, consistent with its lower viscosity index. At 30 °C, the measured time for 20W-50 was approximately 10.5 s, compared to 8.0 s for 10W-40 and 6.5 s for 10W-30, demonstrating the sensitivity of the falling-ball method to viscosity differences among lubricants of varying grades [12].

The relationship between velocity and temperature, shown in Fig. 3b, follows an inverse trend to that of fall time. The velocity of the ball increases with temperature due to the decrease in viscous resistance. The 10W-30 oil consistently produced the highest velocity at each temperature, while 20W-50 yielded the lowest. The rate of increase in velocity was most pronounced between 30 °C and 70 °C, beyond which the change became relatively gradual, indicating a transition to near-Newtonian flow behavior at elevated temperatures. This observation aligns with previous findings that engine oils exhibit reduced sensitivity to temperature once approaching their optimal operating viscosity range [13].

The variation of kinematic viscosity with temperature is depicted in Fig. 3c. As expected, viscosity decreased exponentially with rising temperature, consistent with the Arrhenius-type dependence commonly observed in lubricating oils. At 30 °C, the 20W-50 oil exhibited a kinematic viscosity of approximately 155 cSt, which was nearly double that of 10W-30 at 78 cSt.

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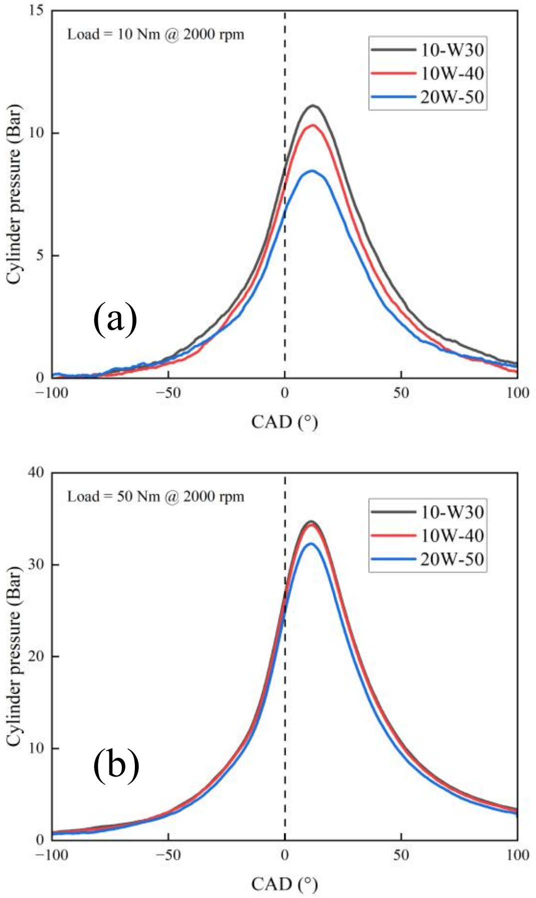

Fig. 4 In-cylinder pressure profiles of a GDI engine operating with different lubricating oils (SAE 10W-30, 10W-40, and 20W-50) at 2000 rpm under (a) low load = 10 Nm and (b) medium load = 50 Nm conditions

However, at 90 °C, which represents the typical engine operating temperature, the viscosities of all three oils converged to a similar range (20–25 cSt). This suggests that at this operating state, the influence of the oil grade is significantly diminished compared to its more pronounced effect at lower temperatures. Consequently, the difference in hydrodynamic film thickness among these grades becomes less significant under stabilized high-temperature conditions [14].

After determining the viscosity characteristics of each lubricant, combustion measurements were conducted on the 1.3-L, four-cylinder GDI engine at 2000 rpm under low-load (10 Nm) and medium-load (50 Nm) conditions. In-cylinder pressure was measured in an instrumented cylinder (one cylinder) while the engine operated in normal four-cylinder configuration. Fig. 4 presents the cylinder pressure traces at a constant engine speed of 2000 rpm, recorded at low load (10 Nm) and medium load (50 Nm), respectively.

As shown in Fig. 4a, the maximum cylinder pressure varied noticeably with the viscosity grade of the lubricant. At low load, SAE 10W-30 produced the highest peak pressure of approximately 11.8 bar, followed by 10W-40 (≈ 10.6 bar) and 20W-50 (≈ 9.2 bar). The higher peak pressure with lower-viscosity oil can be attributed to reduced hydrodynamic friction between the piston rings and cylinder wall during the compression and combustion phases. At the stabilised oil temperature of 90 °C, the viscosities of all three oils converged to a similar range (20–25 cSt), resulting in comparable ring-pack sealing

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effectiveness across all grades. The slight advantage in peak cylinder pressure observed with 10W-30 is therefore primarily attributed to reduced hydrodynamic drag rather than superior sealing. The lower viscosity minimizes pumping and viscous losses, enabling more effective compression work transfer to combustion pressure. It should be noted that the elevated peak cylinder pressure associated with SAE 10W-30 does not reflect a proportional increase in indicated IMEP. Based on the thermodynamic relationship IMEP = BMEP + FMEP and considering that BMEP was maintained constant across all test conditions through dynamometer control, any reduction in FMEP would necessarily yield a marginal decrease in IMEP. Consequently, the higher peak pressure observed with the lower-viscosity lubricant is more accurately interpreted as a result of diminished hydrodynamic resistance during the compression stroke, which facilitates more efficient in-cylinder work transfer within that specific phase of the cycle, rather than an enhancement of the total cycle-integrated indicated work. This interpretation is corroborated by the work of Abu-Nada et al. [15], who reported that a reduction in piston-assembly friction modifies the in-cylinder pressure trajectory without producing a commensurate change in the net indicated work output per cycle.

In contrast, the 20W-50 oil exhibited delayed pressure buildup and a slightly broader combustion duration, suggesting that higher viscosity hinders oil film flow at cold and light-load conditions, leading to increased mechanical friction and potentially slower heat release. Similar observations have been reported by E. Abu-Nada et al. [15], where thicker oils were associated with higher FMEP (Friction Mean Effective Pressure) and lower indicated efficiency at partial load.

Under medium engine load (Fig. 4b), the influence of oil viscosity became less pronounced. All lubricants produced similar pressure profiles, with peak pressures around 33–35 bar. The reduced sensitivity at medium load is due to elevated oil temperatures that decrease viscosity differences among the samples, allowing all lubricants to operate within the mixed- or hydrodynamic-lubrication regime. Consequently, combustion behavior was governed primarily by injection and ignition timing rather than by mechanical frictional differences.

Nevertheless, 10W-30 still exhibited a marginally higher peak pressure, implying slightly improved energy conversion efficiency. This can be linked to its lower internal resistance, which promotes more consistent ring motion and enhanced volumetric efficiency. However, excessively low viscosity may risk thinner oil films and reduced wear protection under high thermal

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stress, particularly in GDI engines known for elevated piston crown and liner temperatures [16].

The observed differences in peak pressure at low load can be attributed to the combined effects of mechanical loss and sealing/film behavior, which influence the effective indicated work available for combustion development. Importantly, the comparison was made at identical speed–load setpoints after thermal stabilization; thus, the differences are interpreted as lubricant-driven trends rather than transient thermal effects.

Following the in-cylinder pressure analysis, the effects of lubricant viscosity on engine performance were examined in terms of indicated IMEP, FMEP, and the coefficient of variation of IMEP (IMEP_COV) under both low and medium load conditions at 2000 rpm, as summarized in Fig. 5.

As illustrated in Fig. 5a, the IMEP increased significantly with engine load for all oil grades, consistent with the higher fuel energy input under medium-load operation. At the typical engine operating temperature of 90 °C, the viscosities of all three lubricants converged to a similar range, as previously shown in Fig. 4c. Consequently, the sealing effectiveness and gas leakage (blow-by) across the piston rings were nearly identical for all oil grades.

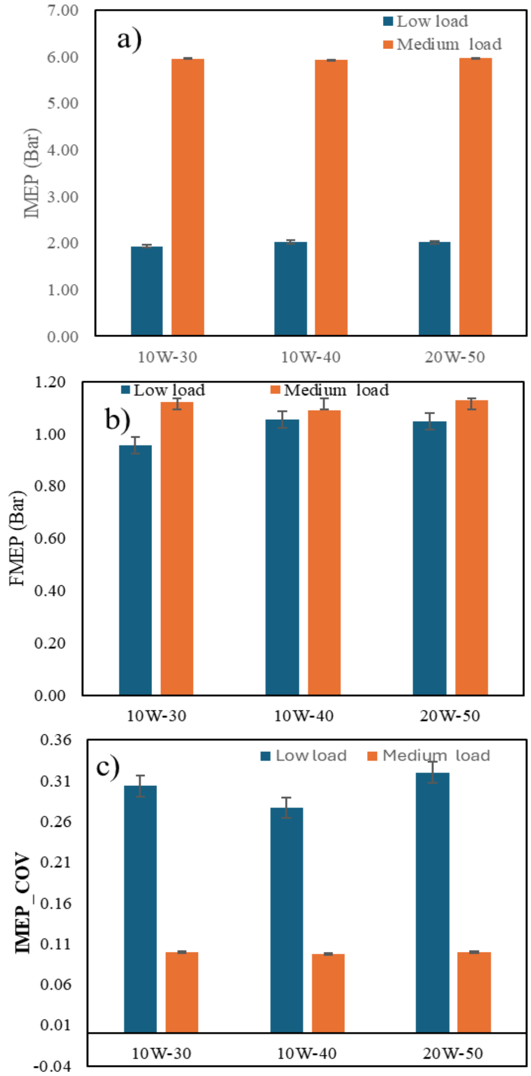

Fig. 5 Comparative analysis of (a) IMEP, (b) FMEP and (c) coefficient of variation of IMEP (IMEP_COV) for three lubricant grades (SAE 10W-30, 10W-40, and 20W-50) under low load (10 Nm) and medium load (50 Nm) conditions at 2000 rpm.

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This explains why the IMEP values yielded by 10W-30, 10W-40, and 20W-50 were very similar, with only a marginal difference observed at low load, where the slightly thinner film of 10W-30 might result in a minor reduction in pressure retention compared to the higher-viscosity grades.

At medium load (50 Nm), the IMEP values converged across all oil grades, indicating that the influence of viscosity diminishes once the oil reaches its fully warmed-up condition and enters the hydrodynamic lubrication regime. These findings are consistent with the trends observed in similar GDI engine studies, where low-viscosity lubricants improved part-load efficiency but offered minimal advantage under high-temperature conditions.

The FMEP values for all test conditions are presented in Fig. 5b. It was observed that FMEP slightly increased with load and with oil viscosity. The 20W-50 oil consistently produced the highest FMEP values, followed by 10W-40, while 10W-30 resulted in the lowest frictional losses. The differences are primarily governed by the hydrodynamic shear stress in the oil film and the pumping losses associated with thicker oils.

Although high-viscosity oils enhance the load-carrying capacity of the lubricant film, they also increase resistance to motion, particularly in the piston-ring–liner and crankshaft journal bearings. The higher FMEP, with 20W-50 under both conditions confirms that excessive viscosity leads to increased mechanical losses and reduced indicated efficiency. At medium load, however, the rise in oil temperature reduces viscosity differences among the oils, which explains the relatively small deviation in FMEP at 50 Nm.

The coefficient of variation of IMEP (IMEP_COV) was used to assess cycle-to-cycle combustion stability, as shown in Fig. 5(c). At low load, all oils exhibited higher IMEP_COV values (0.25–0.32), indicating greater combustion variability under partial-load operation. At the same speed–load point, the higher-viscosity oil (20W-50) tended to show slightly higher sensor-reported airflow compared with lower-viscosity oils. This trend is consistent with the possibility that increased mechanical losses may require a marginally higher air and fuel throughput to maintain the same brake load; however, the airflow metric is used here mainly to support relative comparisons between lubricants under identical test conditions. Among them, 10W-30 and 20W-50 showed slightly higher variations compared to 10W-40, which may be linked to differences in oil–fuel interaction and combustion chamber wall wetting at low temperatures.

As engine load increased, IMEP_COV markedly decreased (<0.1 for all cases), signifying improved combustion stability. This improvement results from higher in-cylinder temperatures and enhanced fuel

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vaporization, which mitigate the effects of oil viscosity on ignition and mixture formation. Such a trend is consistent with prior studies reporting that GDI engines exhibit more stable combustion with rising load and temperature [1].

To further evaluate the influence of lubricant viscosity on engine performance, the volumetric airflow (CFM) and fuel consumption rate were measured under both low load (10 Nm) and medium load (50 Nm) conditions at a constant engine speed of 2000 rpm, as shown in Fig. 6.

As depicted in Fig. 6a, the volumetric airflow increased markedly with engine load for all lubricant types, reflecting the greater air–fuel charge required to sustain higher torque output. At both load conditions, the 20W-50 oil produced a slightly higher airflow rate compared to 10W-30 and 10W-40, which can be attributed to the marginally higher throttle opening needed to overcome the greater frictional resistance of the higher-viscosity oil.

At low load, all oil types exhibited similar airflow behavior (≈1.8–2.0 CFM), while at medium load, the airflow increased to approximately 6.5–7.0 CFM, with 20W-50 showing the maximum. The minor increase in airflow for thicker oils suggests a compensatory rise in engine breathing effort due to higher mechanical drag. This behavior is consistent with prior studies that reported increased intake flow and pumping work when lubricating viscosity exceeds the optimal hydrodynamic range [3].

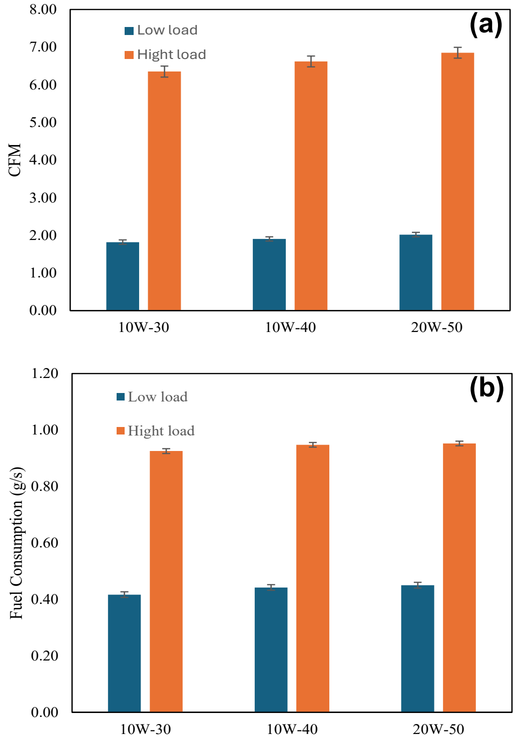

Fig. 6 (a) Variation of airflow rate (CFM) and (b) fuel consumption rate (g/s) of a GDI engine operating with different lubricants (SAE 10W-30, 10W-40, and 20W-50) at 2000 rpm under low (10 Nm) and medium (50 Nm) load conditions

The corresponding fuel consumption rates are presented in Fig. 6b. Similar to airflow trends, fuel consumption increased significantly with engine load and viscosity grade. Under medium load, the fuel flow rate rose from approximately 0.9 g/s (10W-30) to 1.0 g/s (20W-50). This increase can be attributed to additional pumping and frictional

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losses that must be compensated for higher fuel input to maintain the same brake output.

At low load, the differences among oils were less pronounced but still followed the same trend 10W-30 consumed the least fuel, followed by 10W-40 and 20W-50. The reduction in fuel flow for lower-viscosity oils is directly linked to lower frictional torque and improved thermal efficiency, particularly under light-load operation where viscous losses represent a substantial portion of total mechanical losses.

These findings emphasize that oil viscosity has a measurable impact on part-load fuel economy, even though its influence diminishes under high-temperature, medium-load conditions when viscosity differences narrow.

The brake thermal efficiency (BTE) of the GDI engine operated with three lubricant grades (SAE 10W-30, 10W-40, and 20W-50) under low (10 Nm) and medium (50 Nm) load conditions at 2000 rpm is presented in Fig. 7. BTE represents the ratio of brake power output to the chemical energy input from the fuel, thereby reflecting the combined effects of combustion efficiency, frictional losses, and pumping work.

As shown in Fig. 7, BTE increased markedly with engine load for all lubricant types, rising from approximately 10–11% at low load to 24–25% at medium load. This improvement is primarily due to higher combustion temperatures and reduced relative heat losses at elevated loads, which enhance the thermodynamic efficiency of the cycle. Furthermore, at medium load, the engine operates closer to stoichiometric conditions with improved fuel vaporization, minimizing incomplete combustion and increasing brake output.

Across all test conditions, the 10W-30 oil yielded the highest BTE, followed closely by 10W-40, while 20W-50 consistently produced the lowest values. The difference was more evident at low load, where viscous friction and pumping losses have a larger proportional effect on total mechanical losses. The lower viscosity of 10W-30 reduces hydrodynamic drag in critical interfaces such as piston-ring–liner and crank journal bearings allowing more of the combustion energy to be converted into brake work.

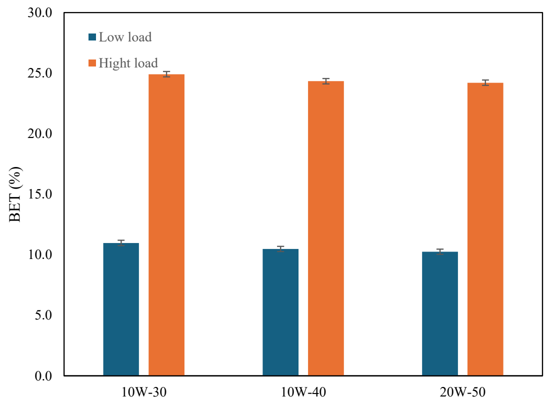

Fig. 7 BTE of a GDI engine operated with three lubricant grades (SAE 10W-30, 10W-40, and 20W-50)

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At medium load, the gap between the oils narrowed, consistent with the findings in IMEP and FMEP analyses. As the oil temperature rises, viscosity differences diminish, leading all lubricants to behave similarly within the hydrodynamic regime. These results corroborate the general understanding that oil viscosity plays a significant role in part-load efficiency but becomes less influential once thermal equilibrium is reached.

4. Conclusion

This experimental investigation elucidated the influence of engine oil viscosity on the combustion and performance characteristics of a GDI engine under low- and medium-load conditions at 2000 rpm. Three multigrade lubricants SAE 10W-30, 10W-40, and 20W-50 were evaluated based on their flow behavior, in-cylinder pressure profiles, mean effective pressures, and thermal efficiency.

The principal findings of this study are as follows: (1) At low load, SAE 10W-30 produced the highest peak cylinder pressure (~11.8 bar), greatest IMEP, and lowest FMEP, yielding the best brake thermal efficiency (~10–11%). These advantages are attributable to its lower hydrodynamic drag in the piston-ring–liner and bearing interfaces. (2) At medium load, viscosity differences diminished as oil temperature increased, resulting in comparable combustion and friction characteristics across all three grades and a peak BTE of approximately 24–25% for all lubricants. (3) SAE 10W-40 provided the most stable cycle-to-cycle combustion (lowest IMEP_COV) across both load conditions, indicating a favourable balance between oil film protection and mechanical efficiency. (4) SAE 20W-50 consistently incurred the highest frictional losses and fuel consumption, particularly under partial-load operation.

In summary, lubricant viscosity significantly affects GDI engine performance at partial loads, where SAE 10W-30 offers the most favourable fuel economy and thermal efficiency. Its impact diminishes at medium loads due to temperature-induced viscosity convergence. SAE 10W-40 represents a practical compromise for applications where both efficiency and lubrication margin are important considerations, while SAE 20W-50 should be avoided where fuel economy is a priority. These findings provide actionable guidance for lubricant selection in modern GDI engine applications operating under typical urban driving conditions.

Future research should investigate: (i) a broader range of engine speeds (e.g., 1000–4000 rpm) to establish the speed-dependency of viscosity effects; (ii) transient operating conditions and cold-start behaviour, where viscosity differences are most pronounced; (iii) long-term tribological assessment to directly quantify wear rates and durability across oil grades; and (iv) the influence of lubricant additives and synthetic oil formulations on GDI engine combustion stability and efficiency.

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Overall, the results indicate that lubricant viscosity significantly affects GDI engine behavior at low loads, influencing frictional losses, heat release, and energy conversion efficiency. However, its impact becomes minimal at medium loads, where temperature-induced viscosity reduction dominates. The 10W-30 oil provides the most favorable compromise between efficiency and protection during typical operating conditions, while 10W-40 may offer a more conservative lubrication margin than 10W-30 under conditions where film thickness is critical, with only a marginal reduction in brake thermal efficiency. Direct assessment of durability would require dedicated tribological testing beyond the scope of the present work.

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