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Enhancing Performance and Emission Characteristics of LHR CI Engine Using TSOME-CSOME Blend by Varying Fuel Injection Pressure and Timing

D Baswaraj [1] P V Krishna Murthy [2] K Prasanna Lakshmi [3]
[1]Associatet professor, [2]Professor, [3]Professor
Dept of Mechanical Engineering,Keshav Memorial Institute of Technology, Narayanaguada, Hyderabad,India

Abstract

Abstact : This study explores the enhancement of performance and reduction of emission characteristics in a Low Heat Rejection (LHR) Compression Ignition (CI) engine using blends of Tobacco Seed Oil Methyl Ester (TSOME) and Cotton Seed Oil Methyl Ester (CSOME). The investigation focuses on the impact of varying fuel injection pressure (FIP) and fuel injection timing (FIT) on critical engine performance metrics, including brake thermal efficiency (BTE), specific fuel consumption (SFC), and exhaust gas temperature, along with emissions such as nitrogen oxides (NOx), carbon monoxide (CO), hydrocarbons (HC), and smoke opacity. The experimental setup involved a single-cylinder, four-stroke, water-cooled CI engine equipped with a Super Ni90 insert fastened on the piston crown to enhance thermal retention and efficiency. The engine was operated using a 50:50 blend of TSOME and CSOME, with tests conducted at FIPs of 190, 230, and 270 bar, and FITs of CA 27°, 29°, and 31° bTDC. Results reveal that optimizing FIP 230bar and FIT CA 29° bTDC, markedly improves engine performance while reducing emissions with enhanced BTE by 85% and decrease BSFC by 82% due to superior fuel atomization and combustion quality. Advanced injection timing led to a reduction in CO and HC emissions.

Key words: FIP, FIT, LHR, TSOME, CSOME

1. Over view

Since the advent of the diesel engine, it has been recognized as one of the most efficient internal combustion engines for transportation, industrial use, and agricultural applications due to its excellent fuel economy, consistent power output, ability to handle heavy loads, and long-term energy efficiency [1]. Over recent decades, fossil fuels have been the primary energy source, with diesel engines consuming a significant portion of these fossil fuels [2]. Despite their advantages, diesel engines produce a range of pollutants, including nitrogen oxides (NO_X), sulfur oxides (SO_X), carbon monoxide (CO), carbon dioxide (CO_2), unburned hydrocarbons (HC), smoke, and particulate matter (PM). These pollutants contribute to harmful substances in the atmosphere, which undergo chemical, physical, and biological transformations, posing serious risks to both ecological systems and human health [3–6]. Concerns about energy depletion and the environmental impact of fossil fuel consumption have driven researchers to seek alternative fuels for diesel engines. The goal is to reduce pollutant emissions while maintaining engine efficiency [7 & 13–15]. Researchers have tested various alternative fuels, often experimenting with blends of diesel and biodiesel, as well as other supplementary fuels.

The combustion, performance and emission characteristics of a diesel engine depend on various factors such as fuel injection timing, the quantity of fuel injected, the shape of the combustion

chamber, the size of the nozzle hole and fuel injection pressure. The function of the fuel injection system in a diesel engine is to atomize the injected fuel to a high degree for better penetration and evaporation in a very short time for achieving higher combustion efficiency. At higher injection pressure, the spray penetration distance becomes longer [1] which results in the maximum utilization of the air injected into the cylinder. Also, the fuel particle diameter becomes smaller which also results in better air–fuel mixing [6]. The combined effect of the aforementioned factors improves the combustion efficiency of the injected fuel. On the other hand, at lower fuel injection pressure, larger diameter fuel particles are formed and the ignition delay becomes longer [1]. A too high of injection pressure is also not desirable because in that case the ignition delay period becomes shorter and thus the possibility of the formation of a homogeneous mixture decreases [18], which adversely affects the combustion efficiency [19, 20]. The momentum of fuel droplets increases with the increase in fuel injection pressure, and the fuel spray tip penetration length also becomes higher. Thus, at a higher injection pressure, the fuel droplets collide with the cylinder wall and form a fuel oil film on it [6, 21]. This fuel oil film may not be evaporated and combusted completely and may create deposits on the combustion chamber wall [21]. One can conclude from the aforementioned information that the fuel injection pressure has a significant effect on the combustion, performance and emissioncharacteristics of a compression ignition engine.

Gumus et al. [8] found that brake thermal efficiency decreased for diesel operation when fuel injection pressure increased from 18 to 24 MPa, but improved for biodiesel under similar conditions, with 32.1% for diesel and 41.3% for biodiesel at full load. They also noted a reduction in nitrogen oxides emissions with increasing fuel injection pressure. Agarwal et al.
[9] suggested that retarding injection timing can reduce nitrogen oxides emissions without affecting engine performance. Suh et al. [10] observed that advancing injection timing lowers carbon monoxide emissions for both biodiesel and diesel fuels. Kuti et al. [11] observed a shorter ignition delay with increased fuel injection pressure and further reduction for biodiesel due to its higher cetane index. Advancing injection timing results in earlier combustion and higher power output [12].

Several studies have examined the effects of injection pressure and timing variations on biodiesel engine performance. Mahanggi et al. [20] found that increasing injection pressure improves spray characteristics, performance, and emissions. Sita and Sudarmanta [21] reported optimal performance at 250 kg/cm² with biodiesel-syngas fuel, showing increased thermal efficiency and reduced SFC. Mutyalu et al. [22] observed that 210 bar injection pressure yields the best performance and emissions for shea olein biodiesel. Mendonca et al. [23] found that retarding injection timing to 15.1° BTDC improves performance in simarouba biodiesel engines, while Hirkude et al. [24] reported optimal results at a compression ratio of 18, injection pressure of 250 bar, and timing at 27° BTDC, with further timing adjustments influencing smoke opacity. Murali Krishna et al. [25] said LHR combustion chamber demonstrated a 6% increase in peak BTE. At full 100% load operation, it showed a 2% reduction in BSFC, a 16% decrease in exhaust gas temperature, and an 11% reduction in coolant load. Additionally, air intake efficiency dropped by 6%, noise levels were reduced by 8%, PM emissions decreased by 45%, and NO_x emissions fell by 46%.

S.V. Karthic et al. [26] found that optimizing fuel injection pressure and timing significantly affects engine performance and emissions. Advancing timing by 2° and increasing pressure to240 bar improved brake thermal efficiency by up to 17.85% for diesel and 16.66% for B30, while reducing hydrocarbon and CO emissions by 46.15% and 15.9%, respectively, and smoke emissions by 28.7%. However, NOx emissions increased. Sudarmanta et al. [27] reported that increasing fuel injection pressure to 230 kg/cm² and retarding timing to 16° bTDC improved spray characteristics, resulting in a 3.9% increase in power, a 13.9% rise in thermal efficiency,and a 45.2% reduction in smoke emissions.

The aim of this research is to enhance the performance and reduce the emission characteristics of a Low Heat Rejection (LHR) Compression Ignition (CI) engine by optimizing fuel injection pressure and timing when using a blend of Tobacco Seed Oil Methyl Ester (TSOME) and Cotton Seed Oil Methyl Ester (CSOME). This involves evaluating how different injection parameters affect key performance indicators, such as brake power and thermal efficiency, while simultaneously assessing their impact on emissions including nitrogen oxides ($NO\chi$),carbon monoxide (CO), hydrocarbons (HC), and smoke opacity. The goal is to determine theoptimal injection settings that maximize engine performance and fuel efficiency while minimizing harmful emissions.

2. Methodology

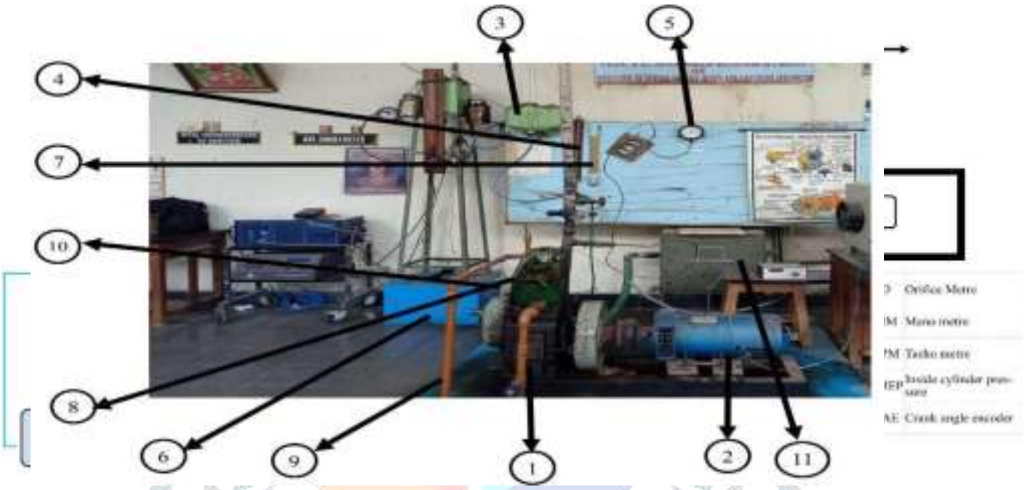
The experimental setup utilized for investigating LHR diesel engines [25] with TSOME and CSOME blend as fuels is depicted in Figure 1 and 2, while the configurations of the engines are detailed in Table 1.

Table 3 Testing Engine Technical details

Engine parameters		**Specifications**
Engine Type		Single cylinder 4 stroke CI Engine
Manufacturer		Kirloskar
Maximum power		3.8 kW at 1500RPM
Bore/Stroke		80/110mm
Specific volume		0.552 liter
Compression ratio		16.5:1
Cooling type		Water cooling
Air Gap		3mm
Insulated insert	Material	Ni90
	Thickness	5mm

To measure its brake power and eddy current dynamo metre is mounted to engine, while the fuel consumption and air consumption was determined using the burette and air-box method with naturally aspirated and engine was fitted with a water-cooling system, and the inlet watertemperature was controlled at 30°C by regulating the flow rate. Experiments were conducted at the various speeds from 3000 to 1000 and a CR of 16.5:1. The engine was manually cranking to start engine, with conventional fuel supplied initially until a steady state. The water flow rate to the engine cooling jacket was maintained at approximately 9 LPM and AVL 5-Gas analyzersetup used for measurement of emissions from engine. Once the engine attained a steady state, the test fuel was introduced into the engine from a separate fuel tank. Load application was achieved using an eddy current dynamometer and load conditions of 0 to 7kgs by increasing 20% gradually for each experiment cycle using TSOME and CSOME blend. Variousparameters such as manometer readings, load, and fuel consumption were recorded under all conditions. In this experimental work, fuel injection pressure was investigated with increasing pressures such as 190, 230 and 270 bars. The primary purpose behind increasing the fuel injection pressure is to enhance the atomization. The fuel was supplied under certain pressure with the help of injection pump so that it produces enough force against the spring to lift the nozzle valve and spray the fuel inside the cylinder with proper atomization. By varying the spring tensions, the injection pressure was changed. Injection timing was changed by adding the shims on fuel injection pump body. With 0.2 mm thickness, the Shim changed the injectiontiming by 2°CA.

Fig. 1. Experimental Schematic layout



1. Engine, 2. Electrical Dynamo meter, 3. Fuel tank, 4. Burette,
5. Piezo-electric pressure transducer
6. Air box, 7. U-tube water manometer, 8. Air inlet, 9. Outlet-jacket water flow,
10. Orifice meter, 11. Gas Analyzer

Fig. 2 Experimental setup

3. Outcomes of work

This section explores the results of an experimental study conducted on a modified LHR CI engine running on a TSOME-CSOME blend. The aim of the study was to enhance the engine'sperformance while reducing harmful emissions by varying the fuel injection pressure and timing. The performance of a modified compression ignition engine can be optimized by adjusting its injection parameters.

3.1 Performance analysis

In this experimental study, fuel injection pressure (FIP) and fuel injection timing (FIT) were the two main variables selected to improve engine performance. The effects of these parameters on critical performance indicators, such as brake power (BP), brake thermal efficiency (BTE),specific fuel consumption (SFC), and mechanical efficiency, are analysed.

3.1.1 Variation of brake power

The key performance parameter of the diesel engine being analyzed in this study is Brake Power (BP), which is heavily influenced by fuel injection pressure and timing.

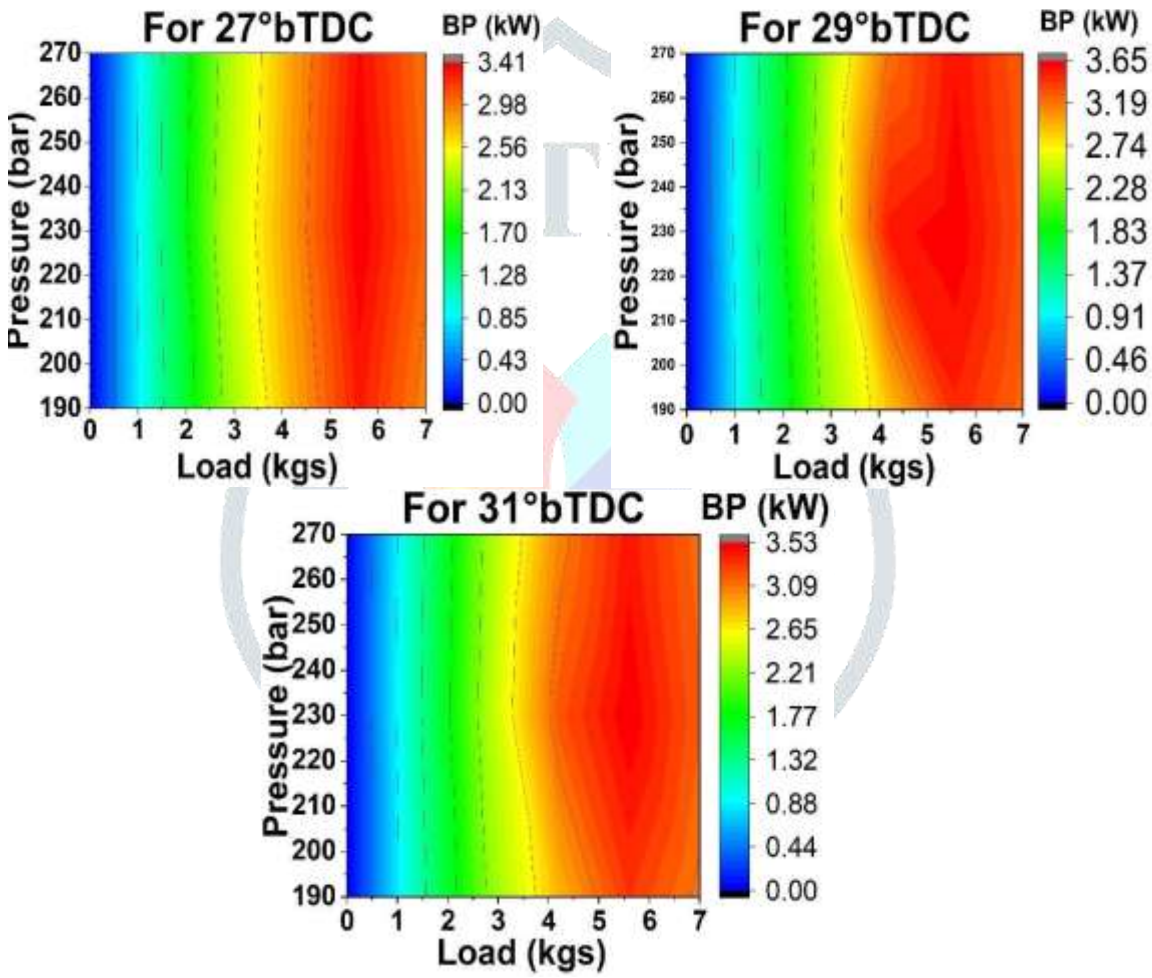


Fig. 3 Variation of brake power

Injection pressure refers to the force with which fuel is injected into the cylinder, while injection timing denotes the point at which fuel begins to enter the combustion chamber. Thisstudy investigates the effects of using a biodiesel blend of Tobacco Seed Methyl Ester(TSOME) and Cotton Seed Methyl Ester (CSOME) at different injection pressures and timings,as illustrated in Figure 3. The fuel injection timing settings of 27˚, 29˚, and 31˚ crank angle before top dead center (CA-bTDC), paired with injection pressures of 190, 230, and 270 bar, were tested. Results indicate that the highest brake power was achieved at 230 bar injection pressure and 27˚ CA-bTDC injection timing at 80% engine load, yielding 3.64 kW. In comparison, at 190 bar injection pressure and 29˚ CA-bTDC timing with the same load, the brake power was 3.48 kW, and at 270 bar and 31˚ CA-bTDC, the brake power was 3.56 kW. These results demonstrate that both higher FIP and advanced injection timing led to lower SFC.The minimum SFC of 0.41 kg/kWh was achieved at 230 bar FIP and CA 29° bTDC under 80%load, representing a 29.7% reduction compared to the baseline. This reduction in fuel consumption is attributed to the improved atomization and combustion efficiency at these optimal injection parameters, which helped to mitigate the effects of biodiesel's high viscosity.

3.1.2 Variation of SFC

An opposite trend was observed for specific fuel consumption (SFC) across all fuel injection timings (FITs) and fuel injection pressures (FIPs) shown in Figure 4.

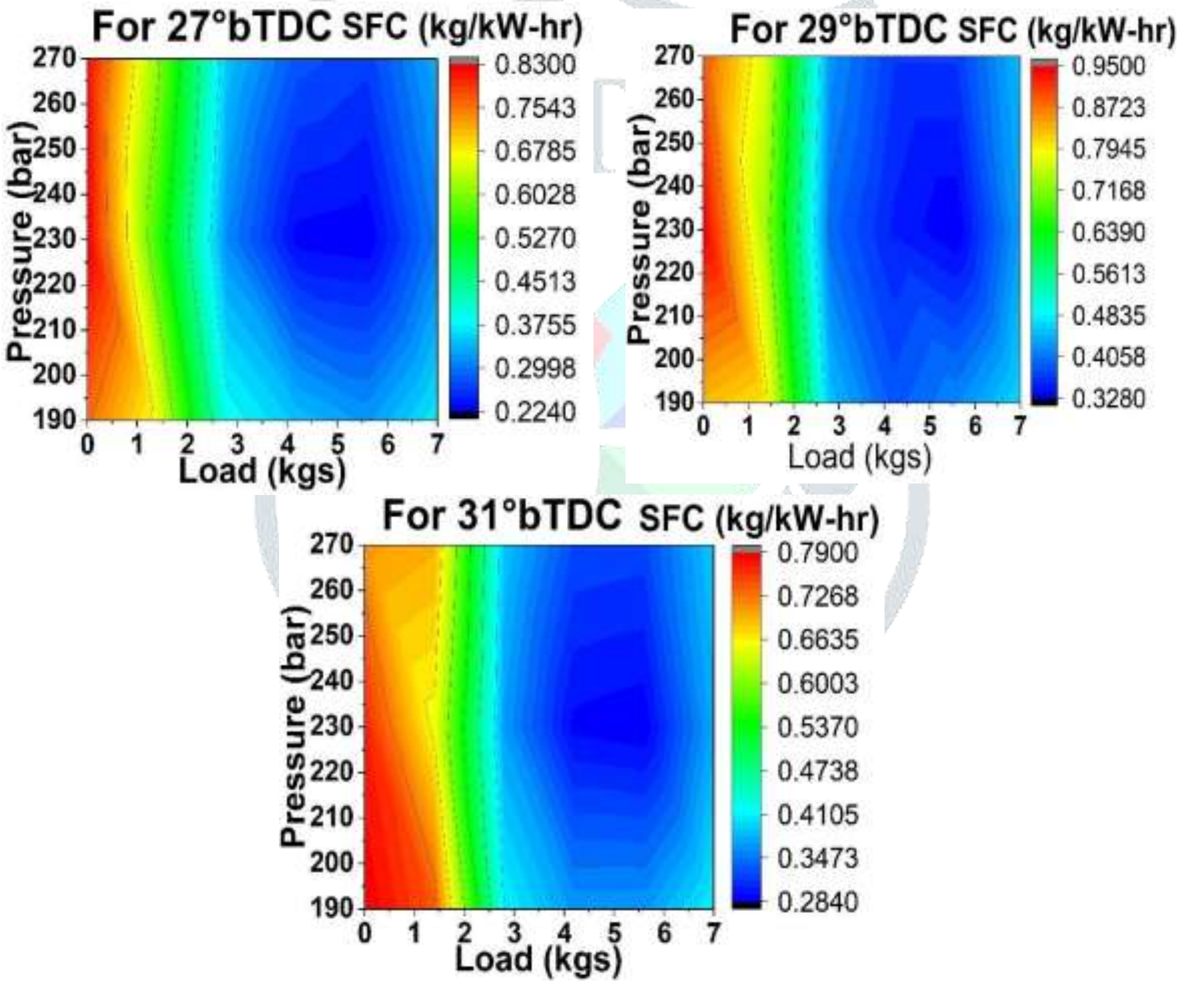


Fig. 4 Variation of SFC

As previously discussed, the trend for brake-specific fuel consumption (SFC) contrasts with that of brake thermal efficiency. Other studies have shown that BSFC is typically higher for biodiesel and its blends compared to diesel due to biodiesel's higher density and lower heatingvalue. Figure 4 illustrates the variations in BSFC as the engine load was increased from 0% to100%. At an average load and 190 bar FIP, the SFC was 0.56 kg/kWh at 27° bTDC, 0.48 kg/kWh at 29° bTDC, and 0.51 kg/kWh at 31° bTDC. When the FIP was raised to 230 bar, theSFC decreased to 0.53 kg/kWh at 27° bTDC, 0.41 kg/kWh at 29° bTDC, and 0.46 kg/kWh at 31° bTDC. Further increasing the FIP to 270 bar resulted in SFC values of 0.55 kg/kWh at 27° bTDC, 0.45 kg/kWh at 29° bTDC, and 0.48 kg/kWh at 31° bTDC. These results demonstrate that both higher FIP and advanced injection timing led to lower SFC.The minimum SFC of

0.41 kg/kWh was achieved at 230 bar FIP and 29° bTDC under 80% load, representing a 29.7% reduction compared to the baseline. This reduction in fuel consumption is attributed to the improved atomization and combustion efficiency at these optimal injection parameters, whichhelped to mitigate the effects of biodiesel's high viscosity.

3.1.3 Variation of BTE

The engine's BTE behavior from 0% to full load is presented in Figure 5. Under 80% load, thebrake thermal efficiency (BTE) was 21% at 190 bar FIP and 27° bTDC FIT, 28% at 190 bar FIP and 29° bTDC FIT, and 24% at 190 bar FIP and 31° bTDC FIT.

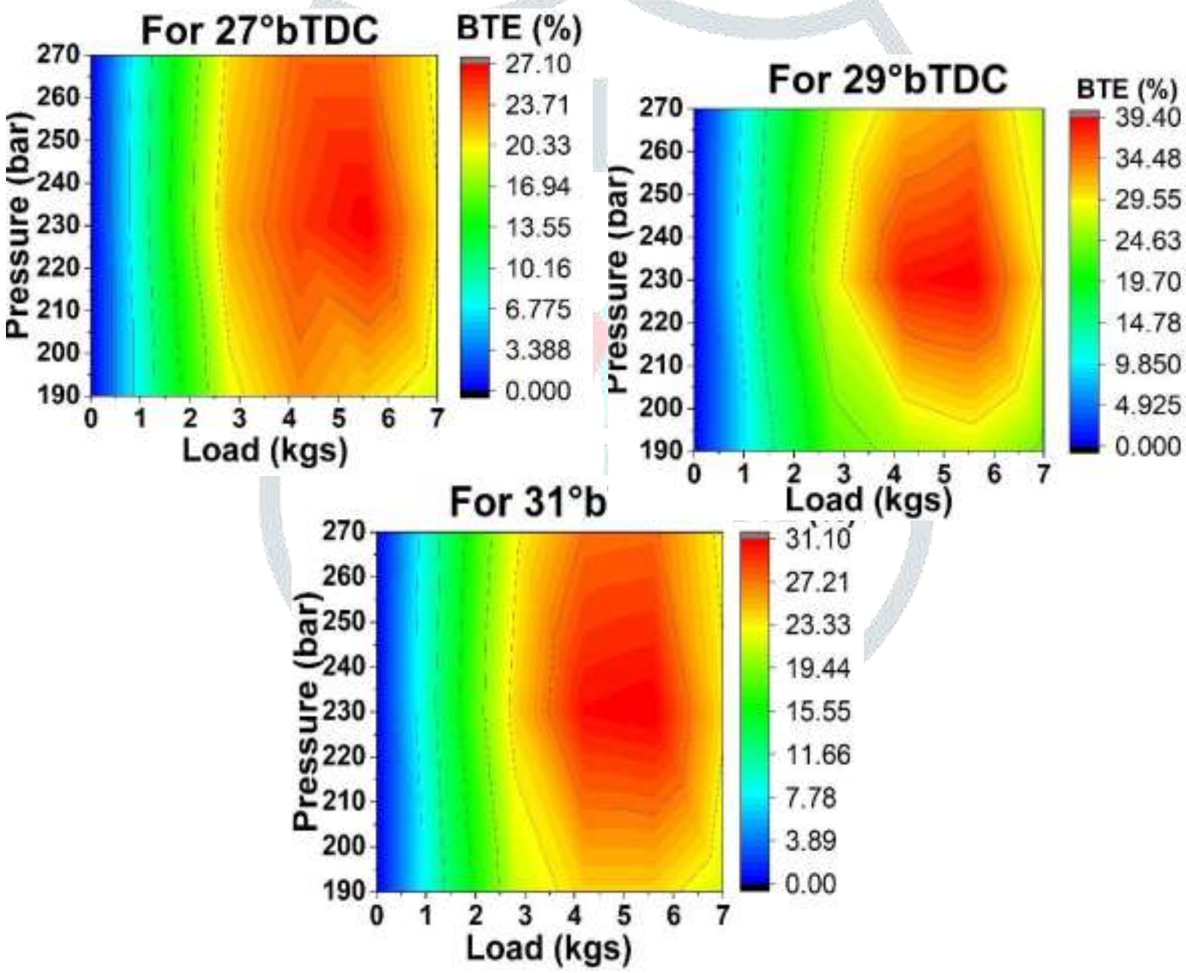


Fig. 5 Variation of BTE

When the FIP was increased to 230 bar, the BTE rose to 27% at CA 27° bTDC, 39% at CA 29° bTDC, and 31% at CA 31° bTDC FIT under the same load condition. Further increasing the FIP to 270 bar resulted in BTE values of 24% at CA 27° bTDC FIT, 33% at CA 29° bTDCFIT, and 28% at CA 31° bTDC FIT. These results show that both increasing the FIP and advancing the injection timing improved BTE. The maximum BTE of 39.33% was achieved at230 bar FIP with injection timing set at 29° bTDC and 80% engine load, representing a 42.3%improvement compared to the baseline, beyond the fuel injection pressure (FIP) of 230 bar andfuel injection timing (FIT) of 29° bTDC, a noticeable drop in brake thermal efficiency (BTE) was observed across all injection timings. This decrease is primarily due to the longer time required to build up pressure at higher injection pressures (Mohan et al., 2014). The primary reason for this decline is the increase in mechanical losses within the engine, which negativelyimpacts overall performance. The improvement in BTE is attributed to enhanced atomization from the increased FIP and advanced timing, which improved combustion efficiency by enhancing the premixed combustion phase. This led to a significant rise in BTE at 230 bar FIP and 29° bTDC FIT.

3.1.4 Variation of Mechanical Efficiency

The engine's mechanical efficiency across varying loads, from 0% to 100%, is shown in Figure 6. At average load conditions, mechanical efficiency values were 56.06%, 58.81%, and 58.13%for a fuel injection pressure (FIP) of 190 bar with injection timings of 27°, 29°, and 31° bTDC,respectively.

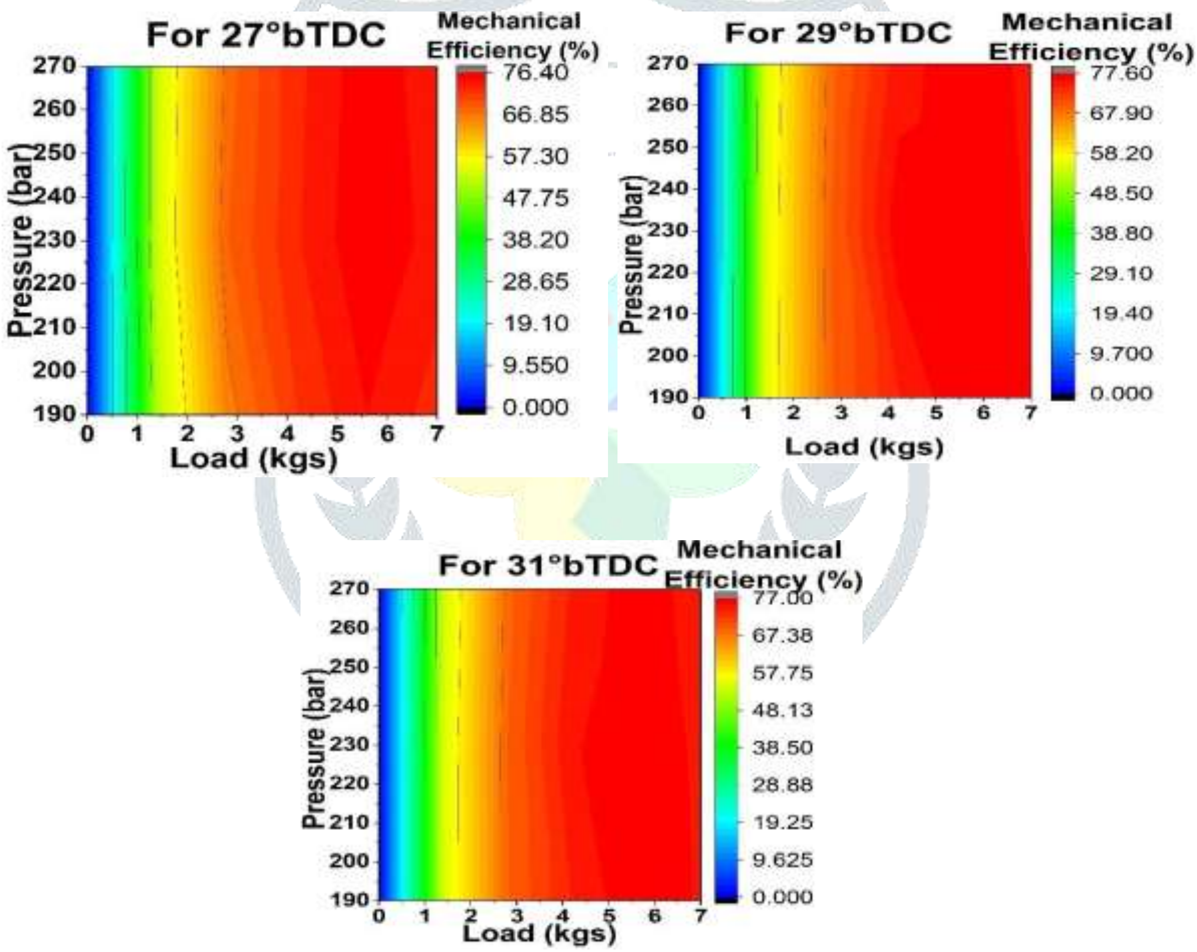


Fig. 6 Variation of Mechanical Efficiency

Increasing the FIP to 230 bar further improved mechanical efficiency to 58%, 59%, and 58.39%at the same injection timings. When the FIP was raised to 270 bar, the mechanical efficiency slightly dropped to 57%, 58.6%, and 57.9% for 27°, 29°, and 31° bTDC, respectively. These results indicate that both higher injection pressures and advanced injection timings contribute to improved mechanical efficiency. The maximum mechanical efficiency of 77.5% was achieved at 230 bar FIP with an injection timing of 29° bTDC at 80% engine load, representinga 1.2% increase over the baseline. However, when the FIP exceeded 230 bar and the injection timing advanced beyond 29° bTDC, a noticeable reduction in mechanical efficiency was observed. This decline is mainly attributed to the increased time needed to generate higher pressure, which results in greater mechanical losses and negatively impacts engine performance.

The improvement in mechanical efficiency at optimal settings is linked to reduced frictional losses and enhanced combustion efficiency, particularly in the premixed combustion phase, leading to better engine performance at 230 bar FIP and 29° bTDC.

3.2 Emission Analysis

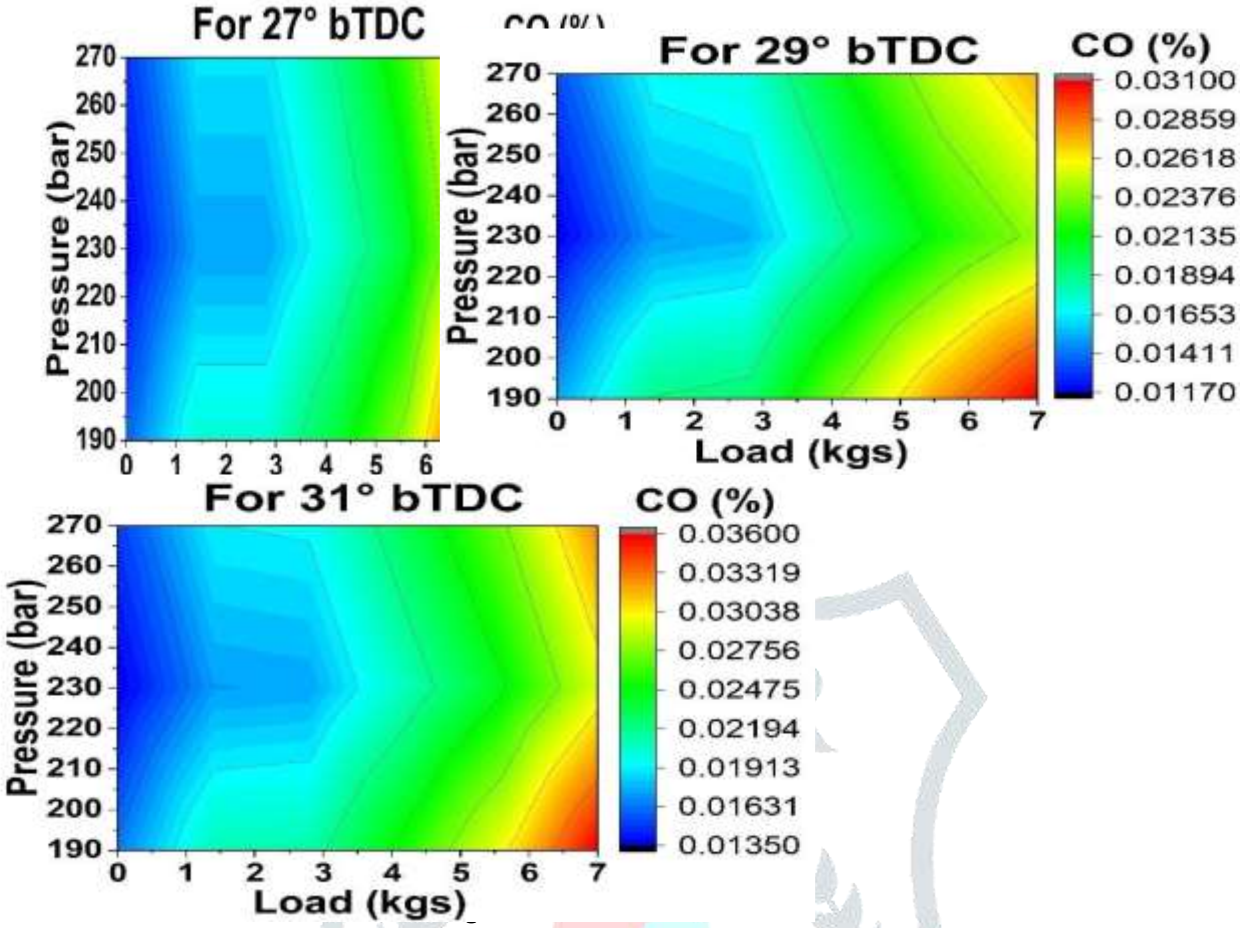
In this experimental study, fuel injection pressure (FIP) and fuel injection timing (FIT) werethe primary variables chosen to reduce engine emissions. The impact of these parameters onkey performance metrics, specifically the emission characteristics, is analyzed in detail. Thisincludes examining the levels of nitrogen oxides (NOx), carbon monoxide (CO), hydrocarbons (HC), and smoke opacity (SOP).

3.2.1 CO Emission Analysis

Carbon monoxide (CO) is emitted due to insufficient oxygen during combustion. CO poses serious health risks, as it can displace oxygen in the bloodstream, potentially leading to fatal outcomes. It is mainly produced as a result of incomplete combustion, particularly when thereis a lack of oxidants, inadequate temperature, and insufficient residence time. For complete combustion, CO should ideally be oxidized to CO_2 at the end of the process, but when this oxidation does not occur due to insufficient oxidants, CO is emitted.

As shown in Figure 7, increasing the fuel injection pressure (FIP) to 230 bar led to a slight reduction in CO emissions. However, when the FIP was further increased to 270 bar at 27° bTDC, CO emissions increased. Higher nozzle opening pressure improves atomization, leading to better combustion, but engine speed and spray penetration limits also affect the air- fuel mixture and combustion quality. When FIP is raised, deeper spray penetration can result in wall quenching, which negatively impacts combustion.

Moreover, advancing the injection timing from 27° bTDC to 29° bTDC led to a reduction in CO emissions across all pressure conditions. However, when the injection timing was further advanced to 31° bTDC, CO emissions increased again due to incomplete combustion. The optimal injection timing of 29° bTDC showed lower CO emissions compared to 27° and 31° bTDC for all FIP levels, likely due to higher cylinder temperatures and increased oxidation efficiency. Retarding the fuel injection timing resulted in higher CO emissions compared to both 27° and 31° bTDC, which may be attributed to wall impingement that caused poor air- fuel mixing, thereby promoting CO formation. Overall, CO emissions were reduced by 33.33%with an FIP of 230 bar and injection timing of 29° bTDC compared to the



baseline.

3.2.2 HC Emission Analysis

Hydrocarbon emissions from diesel engines are primarily caused by either excessive orinsufficient mixing of air and fuel, as well as the presence of large fuel droplets. As shown in Figure 8, hydrocarbon emissions decreased as fuel injection pressure increased from 190 bar to 230 bar due to improved air-fuel mixing. However, beyond 230 bar, hydrocarbon emissionsincreased due to incomplete combustion, resulting in lower peak heat release rates and peak pressures.

While higher injection pressures create a finer fuel spray, factors such as engine speed and theextent to which the spray penetrates into the compressed air play a significant role in air-fuel mixing and combustion. Increasing the injection pressure also extends the spray penetration length, which raises the likelihood of wall impingement. At full load, the air-fuel mixture became lean, limiting flame propagation through the mixture. Consequently, partial oxidationoccurred, leading to hydrocarbon emissions. Additionally, fuel impingement on the cylinder walls reduced the air-fuel mixing rate, further contributing to unburnt hydrocarbons.

Advancing the injection timing allows for more time for air-fuel mixing, which initiates combustion earlier and leads to more efficient combustion with lower hydrocarbon emissions. The earlier start of combustion increases in-cylinder temperatures, promoting complete

coxidation and reducing hydrocarbon emissions while decreasing the flame quenching layer thickness. At 27° bTDC, biodiesel blends reduced hydrocarbon emissions across all FIP levels.However, retarding the injection timing limited the time available for proper air-fuel mixing, resulting in poor vaporization and larger fuel droplets. The results show that unburnt hydrocarbon emissions increased at 31° bTDC compared to 29° bTDC.

3.2.3 Smoke opacity analysis

Diesel engines produce smoke due to poor combustion, often caused by an overly rich air-fuelmixture or partially evaporated fuel. Smoke opacity emission analysis of modified CI engine is shown in Figure 9, advancing fuel injection timing reduces smoke emissions, while increasing injection pressure tends to raise them. Specifically, advancing injection timing to 29° bTDC reduced smoke opacity by 9.09%. However, further advancing the timing to 31° bTDC caused a 6% increase in smoke opacity compared to 29° bTDC at pressure of 190bar and average load condition.

When the fuel injection pressure was increased from 190 bar to 230 bar at 29° bTDC, smoke opacity decreased by 24.24%. This reduction is attributed to the finer fuel droplets achieving nanostructure sizes, which hindered the formation of a homogeneous air-fuel mixture, leadingto incomplete combustion due to local air-fuel mixture limitations. Further increasing the pressure to 270 bar resulted in a 16% rise in smoke opacity, primarily due to lower flame propagation and decreased combustion efficiency.

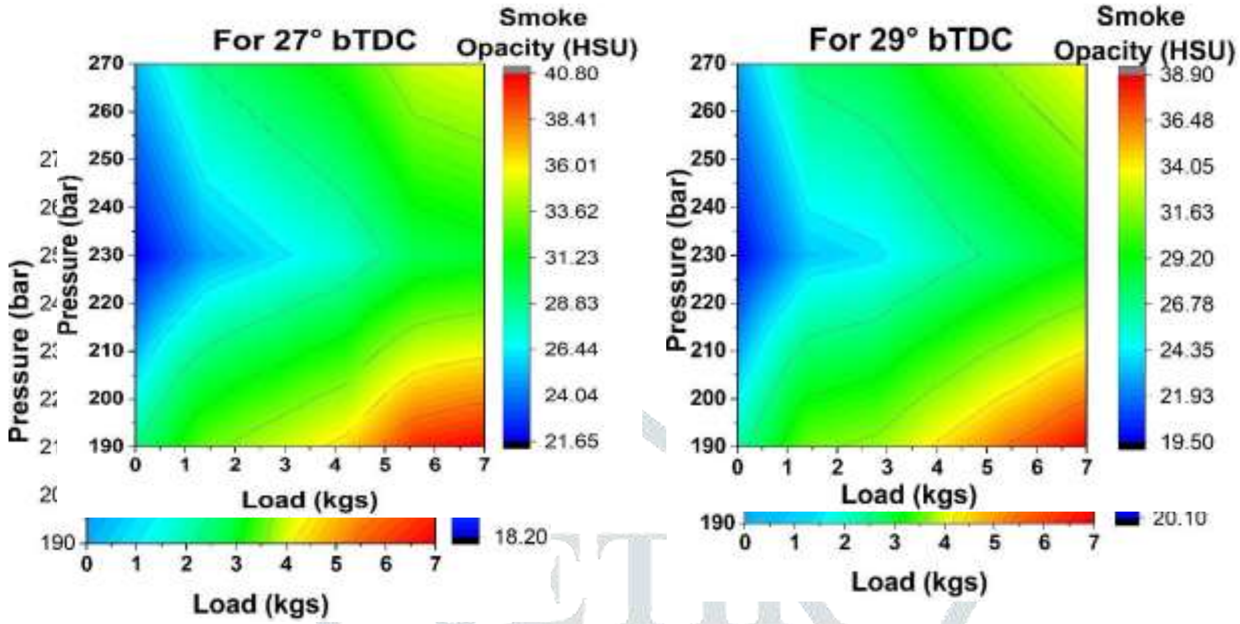
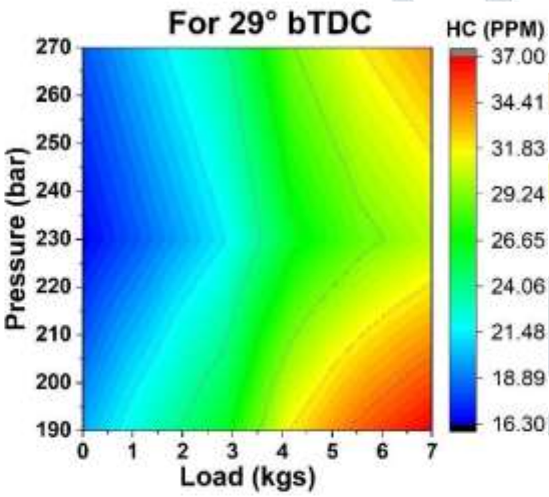


Fig. 8 Variation of Hydro Carbon emission



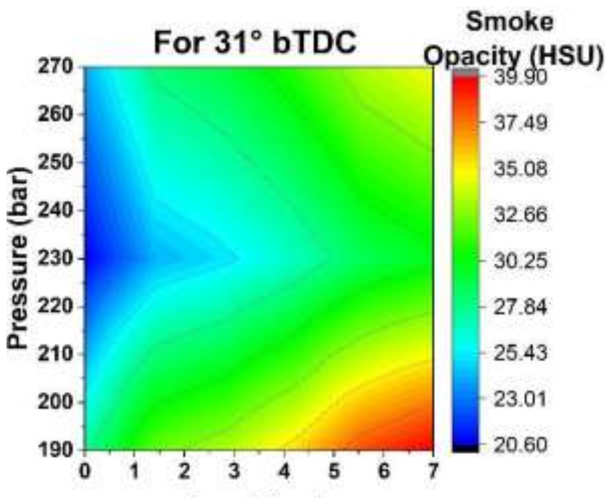


Fig.9 Variation of Smoke Capacity

Advancing the fuel injection timing resulted in lower smoke emissions because of the higher heat release during the premixed combustion phase. At CA 29° bTDC, a larger amount of fuelaccumulated during the longer ignition delay period, allowing for better air-fuel mixing. This led to a higher heat release rate during the premixed combustion phase, contributing to reducedsmoke emissions. However, the earlier occurrence of the mixing-controlled combustion phasealso led to higher smoke emissions due to less efficient combustion.

3.2.4 Analysis of Oxides of Nitrogen

Nitric oxide (NO) emissions are primarily dependent on the combustion chamber's temperature, with the latent heat of vaporization being a key factor. Fuels with higher latent heat of vaporization absorb more heat during the ignition delay period, leading to a reduction in in-cylinder temperature and, consequently, lower NO emissions. As illustrated in Figure 10,increasing the fuel injection pressure (FIP) to 230 bar resulted in a slight decrease in NO emissions. However, when the FIP was raised further to 270bar at 27° bTDC, NO emissions increased.

While higher nozzle opening pressures enhance atomization, improving combustion quality, increased FIP can lead to excessive spray penetration, causing wall quenching and negatively affecting combustion. Advancing the injection timing from 27° bTDC to 29° bTDC reduced NO emissions across all pressure conditions. Yet, further advancing the timing to 31° bTDC caused NO emissions to rise again, likely due to incomplete combustion.

The optimal injection timing of 29° bTDC produced lower NO emissions compared to both 27° and 31° bTDC across all FIP levels, likely because of improved oxidation efficiency and higher cylinder temperatures. Retarding the injection timing led to increase NO emissions whencompared to both 27° and 31° bTDC. Overall, NO emissions were reduced by 24.17% with anFIP of 230 bar and an injection timing of 29° bTDC compared to the baseline.

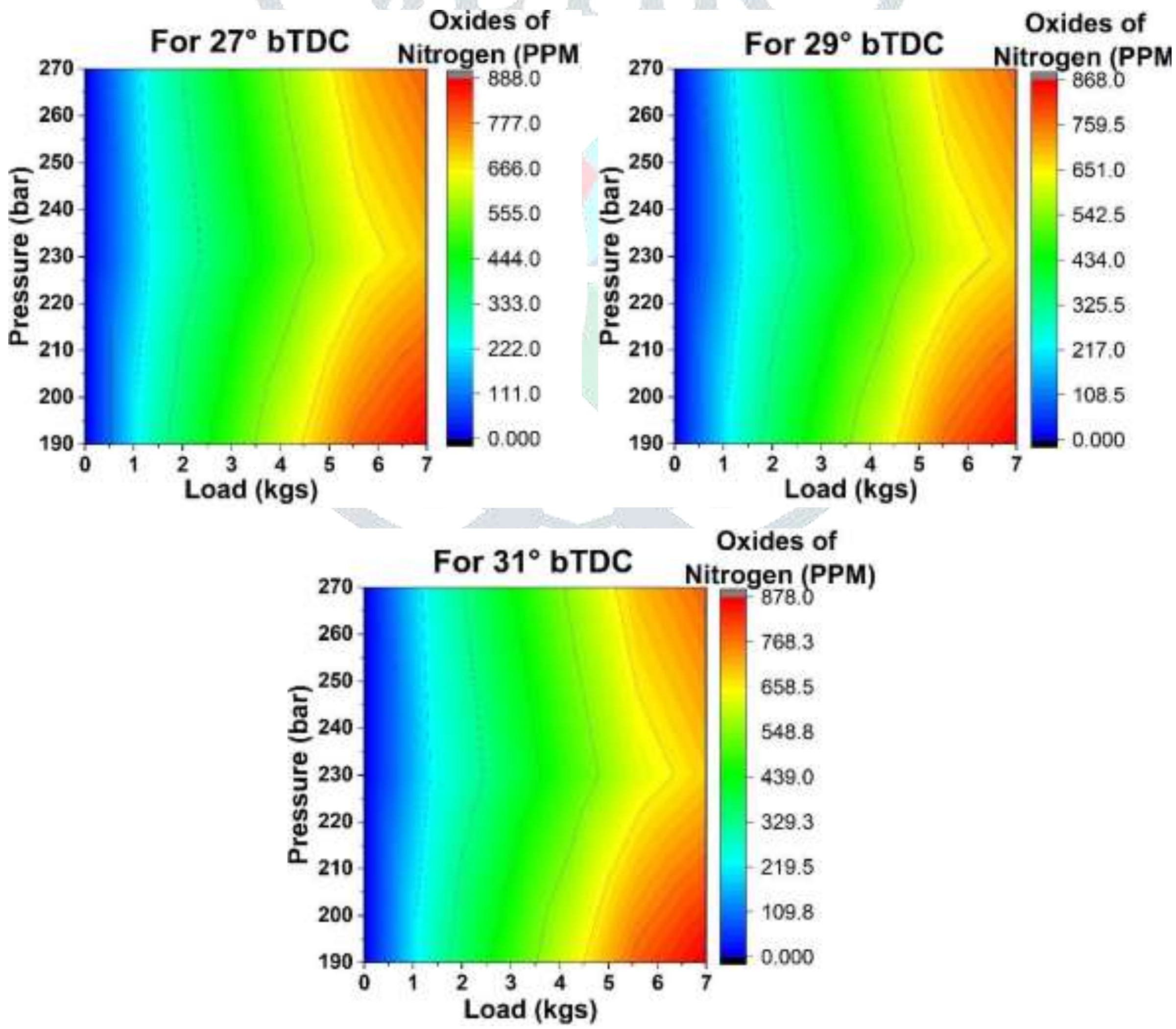


Fig. 10 Variation of Oxides of Nitrogen

4. Conclusions

In this experimental study, the performance and emission characteristics of a Low Heat Rejection (LHR) Compression Ignition (CI) engine were investigated using a blend of TobaccoSeed Methyl Ester (TSOME) and Cotton Seed Methyl Ester (CSOME). The study aimed to identify the optimal settings for maximizing engine efficiency while minimizing harmful emissions, contributing to a more sustainable and cleaner combustion process using biodiesel blends.

- The optimal fuel injection pressure (FIP) and fuel injection timing (FIT) for maximizingbrake power were found to be 230 bar and 27° crank angle before top dead center (CA-bTDC), respectively. Under these conditions, the brake power reached a peak of 3.64 kW at 80% engine load. Advancing the injection timing beyond 29° bTDC resulted in a decline in brake power due to incomplete combustion and increased mechanical losses.
- The lowest specific fuel consumption (SFC) of 0.41 kg/kWh was achieved at 230 bar FIP and 29° bTDC under 80% engine load, indicating a significant 29.7% reduction compared to the baseline. The improved atomization and combustion efficiency at these settings contributed to reduced fuel consumption, even with biodiesel's higher viscosity.
- The highest brake thermal efficiency (BTE) of 39.33% was achieved at 230 bar FIP and 29° bTDC, which represents a 42.3% improvement over the baseline. Increased FIP and advanced injection timing enhanced atomization and combustion efficiency, although efficiency declined at higher pressures and advanced timings due to increasedmechanical losses.
- A maximum mechanical efficiency of 77.5% was recorded at 230 bar FIP and 29° bTDC, demonstrating a 1.2% increase over the baseline. Higher injection pressures andadvanced injection timings generally improved mechanical efficiency, although the efficiency decreased when FIP exceeded 230 bar due to increased mechanical losses.
- CO emissions were reduced by 33.33% at 230 bar FIP and 29° bTDC due to improved combustion efficiency and oxidation. Advancing the injection timing and increasing FIP helped to reduce CO emissions, but emissions increased again at higher pressures and advanced timings beyond optimal settings.
- Hydrocarbon emissions decreased as FIP increased up to 230 bar due to better atomization and air-fuel mixing. However, HC emissions rose when FIP exceeded 230bar, likely due to wall impingement and incomplete combustion.
- The lowest smoke opacity was achieved at 230 bar FIP and 29° bTDC, with a reduction of 24.24%. Higher injection pressures reduced smoke emissions up to a point, but excessive pressures led to increased opacity due to incomplete combustion.
- NOx emissions were reduced by 24.17% at 230 bar FIP and 29° bTDC. While advancing the injection timing reduced NOx emissions, higher pressures and timings resulted in higher emissions due to increased combustion temperatures and wallquenching effects.

The most favourable combination of fuel injection pressure and timing for both performance and emission reduction was found to be 230 bar FIP and 29° bTDC. This combination resulted in the best balance between maximizing power and efficiency while minimizing emissions, particularly CO, HC, SOP, and NOx.

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