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Experimental investigation of combustion performance in the dual-fuel gasoline-NG SI engine in the free of residual burned gas condition with skip-fire technique

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ABSTRACT

Spark ignition engines are among the common apparatuses for generating power for various applications. By employing dual-fuel spark ignition engines using gasoline and natural gas and analyzing the variations in burned residual gases within the combustion cycles, the operational efficiency of the engine can be improved. Experimental data were obtained from a one-cylinder spark ignition engine operating at different spark advances, a compression ratio of 9, and a fuel mixture of 60% gasoline by mass and 40% natural gas by mass under stoichiometric conditions. Measurements were taken under both skip-fire and no-skip scenarios. The raw recorded data were processed to derive the in-cylinder pressure versus crank angle diagram and to calculate the Indicated Mean Effective Pressure (IMEP). Analysis of the ensemble average cycle derived from experimental runs at optimal spark advances revealed that the rate of pressure changes within the cylinder before reaching peak pressure is higher under skip fire conditions. Additionally, a difference of 5 degrees in crankshaft angle was observed between the optimum spark advance with and without skip fire mode. The standard deviation and coefficient of variation for the IMEP were found to be lower in the skip fire condition, indicating a reduction in cyclic variation with skip fire. Furthermore, the rate of change of the mass fraction burned was higher under skip fire conditions compared to no skip fire mode, suggesting faster combustion in the skip fire condition.

Keywords: Gasoline, Skip Fire, SI Engine, Burned Residual Gas, Natural Gas, Dual Fuel.

1. Introduction

Spark ignition engines are common for producing power for diverse applications. The high cost of producing and consuming gasoline, compared to natural gas, particularly in Iran, has significantly amplified the necessity for strategies to reduce fuel consumption. Additionally, gasoline-fueled engines exhibit lower knock resistance and generate more power at the same compression ratio [1]. Considering these factors, the performance of engines can be enhanced by utilizing dual-fuel spark ignition engines with both gasoline and natural gas and by examining the variations in burned residual gases within the combustion cycles. Thus, the analysis of combustion parameters and the functional effects of spark ignition engines, particularly through examining changes in the amount of burned residual gases under combustion conditions, represents a novel and effective area of research.

Burned residual gases in naturally aspirated spark ignition engines is unavoidable, even under full load conditions. The quantities of these gases generally decrease with increasing compression ratio and increase with decreasing engine load [2,

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3]. The presence of burned residual gases in the fuel-air mixture leads to changes in burning velocity, the duration of the combustion period, the Indicated Mean Effective Pressure (IMEP), the fraction of exhaust gas species, and the optimum spark advance. Burned residual gases act as a diluent for the air-fuel blend, effectively reducing the amount of incoming fuel per cycle and ultimately decreasing the energy content per mass unit of the mixture. This issue can adversely affect the quality of flame propagation, the rate of energy release, and prolong the combustion duration [2]. Conversely, the high temperature of burned residual gases elevates the temperature of the air-fuel blend, potentially enhancing combustion speed and optimizing spark advance. Hence, investigating the impacts of burned residual gases on the operational and combustion aspects of the engine is essential [4, 5]. Therefore, in verifying and validating thermodynamic simulation codes for spark ignition engines [6, 7], it is advisable to conduct the code validation process initially using experimental cycles that are free from burned residual gases. This approach ensures a higher degree of accuracy and reliability for the thermodynamic simulation codes.

The extraction of experimental results in spark ignition engines under conditions where some of the burned residual gases remain in the combustion cycle is a common practice [8, 9]. However, obtaining experimental data free of burned residual gases necessitates more complex structures and equipment. One effective method for extracting cycles free of burned residual gases is the skip fire technique. In the skip fire technique with conventional engine fueling, ignition occurs only in the initial cycle for every m cycles, while the remaining m-1 cycles involve engine run-up. In the first run-up cycle following the combustion cycle, burned residual gases are present. However, when intake and exhaust processes occur in the run-up cycle, the concentration of burned gases in the residual gases for the subsequent cycle is significantly reduced. The goal of using the skip fire technique is to achieve combustion cycles free from residual gases, and it is advisable to initially focus on the behavior of combustion cycles free from residual gases when researching experimental combustion cycles. Additionally, in the context of validating simulation models for engine cycles, validating against combustion cycles free from residual gases is of special importance.

Sellnau et al. [10] conducted studies using practical methodologies to estimate the presence of burned residual gases in internal combustion engines. CO2 sensing sensors were employed inside the cylinder along with the skip fire technique to estimate the fraction of residual gases within the cylinder. Mohammadi [11] utilized the skip fire technique to extract experimental results of pressure-crank angle cycles free from burned residual gases. Focusing on IMEP at various skip fire ratios, it was found that at a skip fire ratio of 4 to 1—where four cycles include one combustion cycle and three run-up cycles—there is sufficient assurance of the absence of burned residual gases. Robinet and Higelin [12] used the skip fire technique in their research and estimated that in this technique, there is one ignition cycle followed by three skip cycles for every four cycles. In the worst-case scenario, approximately 0.1% of burned residual gas may be present in the current ignition cycle. Abdi-Aghdam and Ataei [13] explored the impact of skip fire cycles on a natural gas-fueled spark ignition engine, revealing that skip fire and non-skip fire cycles exhibit different optimal spark advances. In recent years, researchers have utilized the skip fire technique to diagnose incomplete combustion [14], reduce emissions [3], decrease fuel consumption at low engine loads and speeds [15], and achieve ignition cycles free of residual gases. Rakopoulos et al. [16] modeled cyclic variations on a spark ignition engine fueled by methane on a cycle-by-cycle basis and validated the model with experimental data based on maximum cylinder pressure and IMEP. The results showed that variations in the equivalence ratio impact cyclic variations.

Given that the initial phase of combustion, where the flame growth is more rapid than the laminar burning velocity (LBV) of the air-fuel blend, becomes altered with the combination of two different fuels, the significance of the LBV becomes evident and crucial. Ballou et al. investigated LBV and flame inconstancy using pure hydrocarbons such as iso-octane and methane. It was demonstrated that adding methane to iso-octane increased the LBV [17, 18] in the lean combustion range. Abdi-Aghadam et al. [19], using common fuels gasoline and natural gas, investigated the effect of adding natural gas to gasoline in a combustion vessel at an initial pressure of 5 bar. Based on experimental data, the LBV of the mixture was calculated using the pressure-based method, and it was demonstrated that increasing the percentage of natural gas in the blend enhances the rate of LBV.

Ramasamy et al. [20], in their study on a natural gas-gasoline spark ignition engine operating in dual-fuel mode with natural gas as the predominant fuel, compared the engine's performance parameters in dual-fuel mode with single-fuel conditions using natural gas. It was observed that increasing the proportion of gasoline in the mixture relative to NG resulted in increased torque. In a spark ignition engine with port injection using natural gas and gasoline in dual-fuel mode, Mowahed et al. [21] conducted a study on a fully-loaded turbocharged engine. It was demonstrated that growing the fraction of natural gas in the dual-fuel mode led to decreased levels of CO and HC pollutants. Yekani et al. [22, 23] investigated the experimental study of a dual-fuel (gasoline-natural gas) engine. It was found that as natural gas was added to the air-fuel mixture, the knock intensity decreased. Sarabi and Abdi Aghdam [24, 25] examined the performance of a natural gas-gasoline dual-fuel spark ignition engine at compression ratios of 9 and 10. It was demonstrated that adding natural gas to the mixture reduced IMEP and overall pollutants. Additionally, Yekani et al. [26,27] found that with the addition of natural gas to gasoline in the dual fuel and lean-burn conditions, pollutant levels decreased compared to using gasoline single fuel.

In port injection spark ignition engines, fuel and air are blended before entering the chamber, where conditions can be assumed to be homogeneous, facilitating the analysis of combustion parameters. Flame growth studies were conducted by Moxey et al. [28] in a spark ignition engine using flame imaging, with a focus on gasoline with added ethanol and butanol. It was shown that the coefficient of variation of IMEP is lowest for the gasoline-butanol blend, and the duration of rapid combustion is shortest for gasoline using the Rassweiler-Withrow method [29]. Nadaleti et al. [30] added hydrogen gas to methane base biogas in a spark ignition engine and demonstrated that adding hydrogen to the mixture reduces the period of combustion by calculating the burned mass fraction using the Rassweiler-Withrow method. Hotta et al. [31], in a one-cylinder spark ignition engine fueled by biogas with gasoline, assumed homogeneity in the fuel-air mixture and estimated the burned mass fraction. A combustion simulation model for spark ignition engines was developed by Benjamin et al. [32, 33] to control the fraction of residual gas by optimizing the combustion phase. The model calculates thermal efficiency and the fraction of residual gas using the mean in-cylinder pressure.

In research on in-cylinder combustion, it is important to consider conditions free from residual gases to investigate combustion behavior and performance when the entire cylinder chamber is filled with fresh air-fuel mixture. This issue is crucial for validating simulation models for combustion engines, as the simulation model must first be validated under residual gas-free conditions. The best method for preliminary validation of combustion simulations in spark ignition engines is undoubtedly to compare simulation performance with experimental results in the absence of residual gases from the previous cycle, allowing for accurate experimental condition definition in the simulation. One necessary technique for enabling experimental data collection in residual gas-free conditions is the skip fire technique, which requires both special software for managing engine conditions during skip fire and appropriate hardware equipment capable of executing and withstanding the associated conditions during research. Recent studies [20, 25] have increasingly focused on creating conditions for combined combustion and utilizing the benefits of dual fuels simultaneously. Research on laboratory systems that allow for the experimental investigation of combustion cycles free from residual gases is limited in the technical literature, likely due to constraints in data collection and laboratory equipment. Considering these points, the present study involves the necessary equipment for experimental data collection under dual-fuel combustion conditions and in the absence of residual gases from the previous cycle, representing a novelty of this research.

In this article, experimental results of pressure variations inside the cylinder were extracted under two conditions: cycles with residual gases from the previous cycle and cycles without such gases, under dual-fuel conditions of 60%

gasoline and 40% natural gas, with constant engine speed, compression ratio, and equivalence ratio. The optimal advance was determined using the skip fire strategy, and the cyclic variations of IMEP and peak pressure at different advances were examined.

Numerical

ATDC	After bottom dead center
BTDC	Before top dead center
CAATDC	Crank angle after top dead center
CABTDC	Crank angle before top dead center
CAD	Crank angle degree
CAFIT	Crank angle from ignition timing
COV	Coefficient of variation
DOV	Deviation of value
IMEP	Indicated mean effective pressure
LBV	Laminar burning velocity
NG	Natural gas
OHV	Over head valve
P	Cylinder pressure
TDC	Top dead center

Greek Letter

σ	Standard deviation
θ	Crank angle

2. Experimental setup

In this study, the Gunt CT300 model, manufactured by a German company, was utilized as the testing platform. The platform comprises a spark ignition single-cylinder research engine coupled with an asynchronous dynamometer capable of adjusting speed. Fueling systems have evolved significantly in research, transitioning from carburetor and mechanical modes to electronic injectors that are capable of adjusting injection start times and fuel injection durations. Spark advance control has been enhanced by researchers [34-36], enabling operation with either gasoline or natural gas fuel, as well as in dual-fuel modes (gasoline-natural gas). Both the natural gas and gasoline fueling systems are configured as port fuel injection systems into the intake manifold, with the amount of fuel injected being controlled by the engine management system, which can be adjusted by the user. Additionally, the engine is equipped to operate under skip fire conditions with various modes, allowing for the determination of consecutive combustion cycles and consecutive run-up cycles within each cycle category. In this study, a set of 7 cycles (3 consecutive firings and 4 consecutive motoring in each cycle category) with 200 alternations were utilized to extract skip fire cycles. The initial firing cycles from each category were selected as cycles free from burned residual gases, referred to as cycles with skip fire. Data acquisition was conducted under skip fire and non-skip fire conditions at various spark advances, followed by the selection of the optimum spark advance for each mode. The specifications of the research engine mentioned above are summarized in Table 1, and a perspective of this single-cylinder research engine is illustrated in Fig.1.

The representation of the test platform is represented in Fig. 2, and the components of Fig. 2 are listed in Table 2.

Table 1. Specifications of CT300 research engine

Specification	Descriptions
Piston diameter	90 mm
course	74 mm
Displacement volume	470 cm^2
Compression ratio	9
Spark system	Electronic with adjustable ignition timing
Fuel injection system	Adjustable injection of NG and gasoline
Lubrication system	Spray and compressor systems
Cooling system type	Singe-phase water cooling system
The number and position of valves	2 OHV
Intake valve opening and closing angle	Open angle 0^o TDC, and close 50^o after BDC
Exhaust valve opening and closing angle	Open angle 40^o before BDC, and close 8^o after BDC
Breathing system	Natural
Combustion chamber shape	Disc – shaped

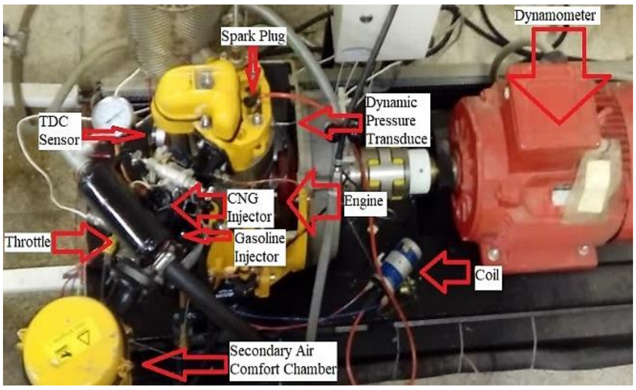


Fig.1. A view of single-cylinder research engine

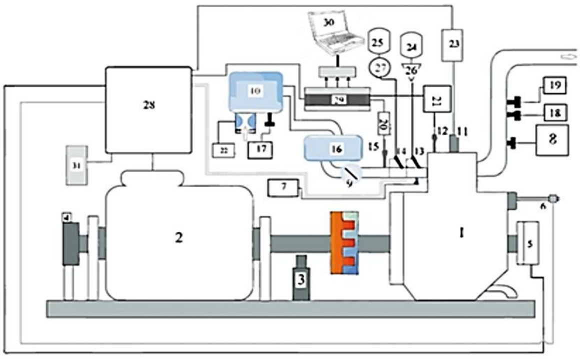


Fig.2. Test platform, measurement systems and control assembly [24]

Table 2: Description of specified components

Number	Specification
1	Engine
2	dynamometer
3	engine speed sensor
4	torque sensor
5	shaft encoder
6	Suction TDC sensor
7	inlet mixture temperature sensor
8	exhaust gas temperature sensor
9	throttle
10	primary air comfort chamber
11	spark plug
12	dynamic pressure transducer
13	NG injector
14	Gasoline injector
15	absolute pressure transducer
16	secondary air comfort chamber
17	temperature sensor of comfort tank
18	gas analyzer A
19	gas analyzer B
20	absolute pressure transduce amplifier
21	dynamic pressure transducer amplifier
22	inlet air flow sensor
23	ignition system
24	NG tank
25	gasoline tank
26	NG pressure regulator
27	gasoline pump
28	management system
29	Data – Logger
30	Computer
31	input electricity power

In spark ignition engines, cyclic variations in cylinder pressure diagrams concerning crank angle, as well as pressure in consecutive cycles under identical test conditions, are not unexpected. Typically, the use of an ensemble average cycle derived from experimental cycles under similar conditions is employed in engine literature. The cylinder pressure at any crank angle θ of the ensemble average cycle can be calculated using Eq. (1) from the pressure of the experimental cycles under identical conditions at the same angle [13]:

$$P_{eq}(\theta) = \frac{1}{N}\sum_{i=1}^{N} P_i(\theta) \tag{1}$$

where N represents the number of experimental cycles, and P_{eq} and P_i denote the pressures of the equivalent cycle and the i-th experimental cycle at the crank angle θ, respectively.

Equation (2) expresses the dependency of the work of the ensemble average cycle on the work of the experimental cycles under identical conditions.

$$W_{c,eq} = \oint P_{eq}dV = \frac{1}{N}\sum_{i=1}^{N}\oint P_i dV \tag{2}$$

$$W_{c,eq} = \frac{1}{N}\sum_{i=1}^{N} W_{c,i} \tag{3}$$

where $W_{c.eq}$ and $W_{c.i}$ denote the work of the ensemble average cycle and the i-th experimental cycle, respectively. The IMEP can be expressed using Eq. (4).

$$IMEP = \frac{W_c}{V_d} \tag{4}$$

where V_d is the displaced cylinder volume.

In this engine, natural gas and gasoline are injected into the inlet port. The injection rate of each injector (injection duration), injection start angle, and spark advance are adjusted using the engine management system. Given the potential impact of injection location pressure, the injectors for gasoline and natural gas engines were calibrated under various conditions [35]. In this study, the representative formula for gasoline, $C_{7.76}H_{13.1}$, with 746 kg/m³ density was considered [36]. The composition of natural gas was utilized as described in Table 3, based on the volume percentage of species [19], with the average chemical formula of the hydrocarbon part being $C_{1.04}H_{3.97}$. Approximately 91.94% of the natural gas composition consisted of hydrocarbons, with the remainder being impurities such as CO_2 and N_2.

Assuming that the volume percentage in natural gas can be used as the mole percentage, the stoichiometric relationship between air and the gasoline-natural gas dual-fuel can be expressed using Eq. (5) below:

$$\tilde{x}C_{7.76}H_{13.1} + (1-\tilde{x})(0.9481C_{1.04}H_{3.97} + 0.0069CO_2 + 0.045N_2) \tag{5}$$

where $\tilde{x}$ is the mole fraction of gasoline. Using the average molecular mass of gasoline (M_G) and natural gas (M_{NG}), the relationship between mass (x) and mole ($\tilde{x}$) parts of gasoline in the dual-fuel condition is given by Eq.(6):

$$\tilde{x} = \frac{xM_{NG}}{(1-x)M_G + M_{NG}x} \tag{6}$$

Table 3. Major Constituent components of NG [19]

Percentage	Constituent
88.33 %	CH_4
4.67 %	C_2H_6
1.14 %	C_3H_8
0.49 %	C_4H_{10}
0.18 %	C_5H_{12}
0.69 %	CO_2
4.5 %	N_2

3.Method of experimentation

Given that most modern spark ignition engines are designed to operate on gasoline, a notable approach to reducing emissions is to utilize a blended fuel mixture of gasoline and natural gas, with gasoline remaining the predominant fuel. Previous research [22-27] has explored various gasoline-natural gas blend ratios, consistently finding that a blend comprising 60% by mass gasoline and 40% by mass natural gas yields the lowest emissions and minimal knocking intensity. Consequently, this study adopted a fuel blend of 60% gasoline and 40% natural gas by mass. The engine in this study was operated with engine nominal speed of 1800 rpm a compression ratio of 9, using the specified dual-fuel mixture of 60% gasoline and 40% natural gas by mass under full load conditions and at a stoichiometric equivalence ratio. Experimental data were gathered under steady-state conditions, both with and without skip fire, and at various spark advances. Data acquisition involved 200 consecutive cycles at different spark advances.

In the skip fire mode, a protocol consisting of 7 cycles was used, which included 3 consecutive firing cycles followed by 4 consecutive motoring cycles, with alternating fuel conditions. This process was repeated 200 times, resulting in a total of 1400 consecutive cycles recorded for each spark advance under skip fire conditions.

Data were acquired at various spark advances in the skip-fire condition. The recorded data were processed using FORTRAN language code to generate the P-θ diagram (in-cylinder pressure versus crank angle) and to calculate the IMEP for each cycle, as well as to determine the mean values across the cycles.

4.Results and discussion

As detailed in the preceding section, experimental data were collected under two conditions: with skip fire and without skip fire, using a dual-fuel mixture of 60% gasoline by mass and 40% natural gas by mass. The collected data were processed using Fortran code to analyze the results.

4.1. Ensemble Average Cycle

In the collection of experimental data from spark ignition engines under identical test conditions, cyclic variations in consecutive cycles are not unusual. To address this issue, a common approach in engine literature [13] is the use of an ensemble average cycle. Figure 3 illustrates both the real consecutive cycles and their corresponding ensemble average cycle. Additionally, Fig.3 depicts the pressure changes relative to the crank angle for real cycles with and without skip fire, alongside their corresponding ensemble average cycle. The spark advance for the skip fire condition is set to 24 degrees BTDC, while it is 29 degrees BTDC in the condition without skip fire. The graph indicates that the intensity of pressure changes before reaching the peak pressure is higher in the skip fire condition compared to the no skip fire condition. This difference is attributed to the absence of burned residual gases in the fuel-air mixture within the cylinder before the ignition of the spark plug.

Figure 4 displays the variations in pressure relative to the crank angle for both skip fire and no skip fire conditions. It is observed that the pressure peak and the area under the curve are greater in the skip fire mode. This increase can be attributed to the absence of residual gases in the skip fire mode, which contributes to a higher pressure and more effective combustion within the cylinder.

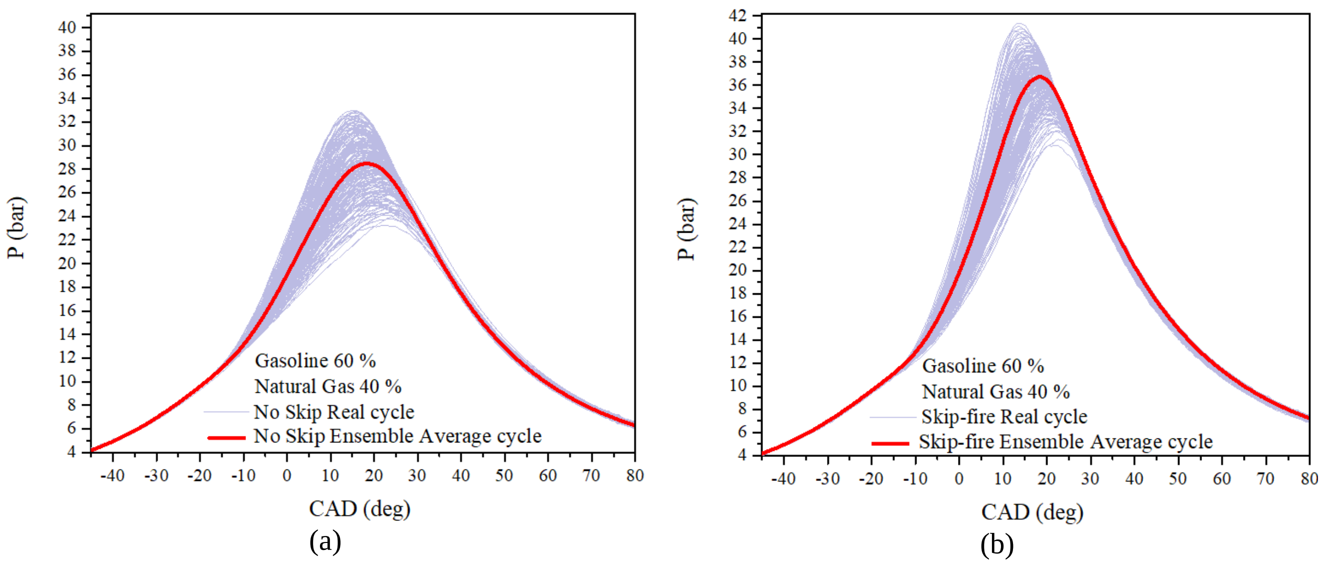


Fig. 3. Pressure changes in relation to crank angle for real cycles, alongside their corresponding ensemble average cycle under conditions: a) without skip fire and b) with skip fire.

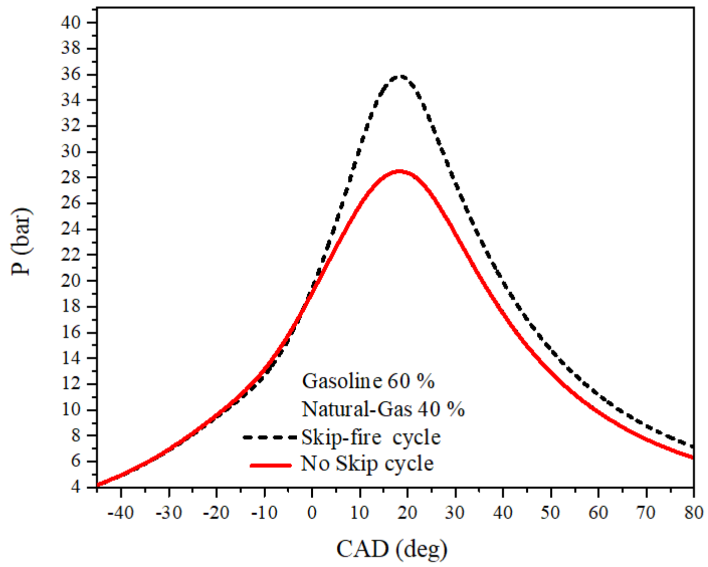


Fig.4. Pressure changes depending on the crank angle of the cycle without skip fire and skip fire mode

4.2. IMEP Changes

One of the primary parameters for evaluating the operational cycles of spark ignition engines is the IMEP. Using Eq. (4), the IMEP values were computed for each spark advance setting under both skip-fire and non-skip technique modes, as depicted in Fig. 5. Figure 5 illustrates the IMEP values as a function of spark advance for a fuel mixture comprising 60% gasoline by mass and 40% natural gas by mass, under both conditions. The optimum spark advance with skip fire is 24 degrees before top dead center (BTDC), whereas in the non-skip fire condition, it is 29 degrees BTDC. The observed lower optimum spark advance in the skip fire condition, with a difference of 5 degrees, is attributed to the complete filling of the cylinder with a fresh fuel-air mixture. Which leads to a faster burning rate and consequently a shorter combustion period.

Considering that a primary objective and innovation of this study is the collection of experimental data under conditions free from residual gases, and noting that the absence of residual gases allows for a greater amount of fresh air-fuel mixture into the combustion chamber compared to conditions without skip fire mode, it can be inferred that the input energy to the cycle will be higher.

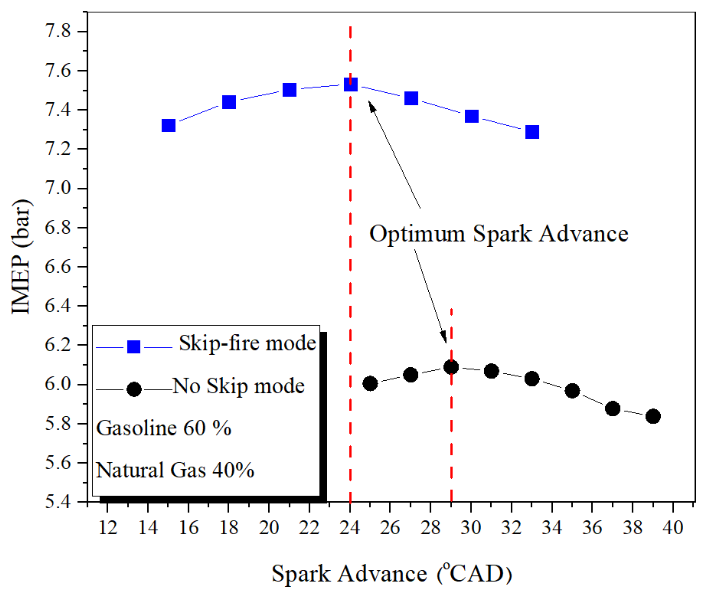


Fig.5. IMEP changes with spark advance in the state with and without skip fire

Figure 6 displays the IMEP values at the optimum spark advance for both skip-fire and non-skip modes. The IMEP in the skip fire mode is approximately 23.6% higher than in the non-skip fire mode.

4.3. Cyclic Variation & P_{max} Changes

Based on the previous subsection, where the ensemble average cycle was derived from real cycles and the IMEP values for each cycle were calculated using the relationships outlined in the methodology section, an analysis of cyclic variations under skip-fire and non-skip mode is presented. Figure 7 illustrates the IMEP values and their percentage changes from the average value at the optimum spark advance over 200 consecutive cycles, plotted against the cycle number for both skip fire and non-skip fire conditions. The percentage of data variations from the average value is indicated by the Coefficient of Variation (DOV), which is expressed as a percentage according to Eq. (7).

$$DOV(\%) = \frac{|\bar{y} - y_i|}{\bar{y}} \times 100 \quad (7)$$

Here, $\bar{y}$ is denoted as the value of IMEP and y_i represents the IMEP value of each cycle it can be observed that the percentage of IMEP variations relative to the average value is lower in the skip fire condition compared to the no skip fire condition. Additionally, the standard deviation and coefficient of variation are reduced under skip fire conditions. This reduction in cyclic variation is due to the absence of residual gases in the air-fuel mixture during skip-fire operation, which allows the cylinder to be filled with fresh fuel and air.

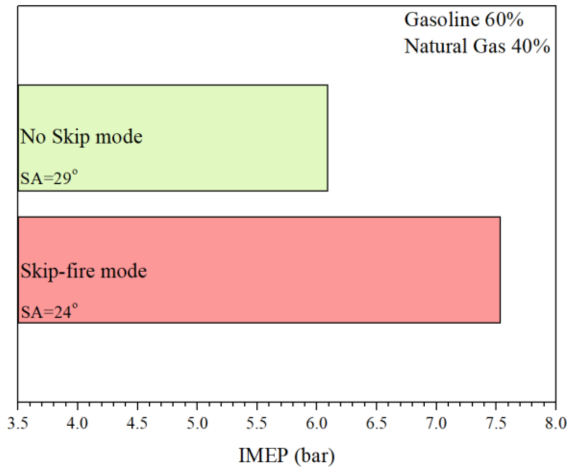


Fig.6. The IMEP values in the optimum spark advance in the skip fire and No skip mode

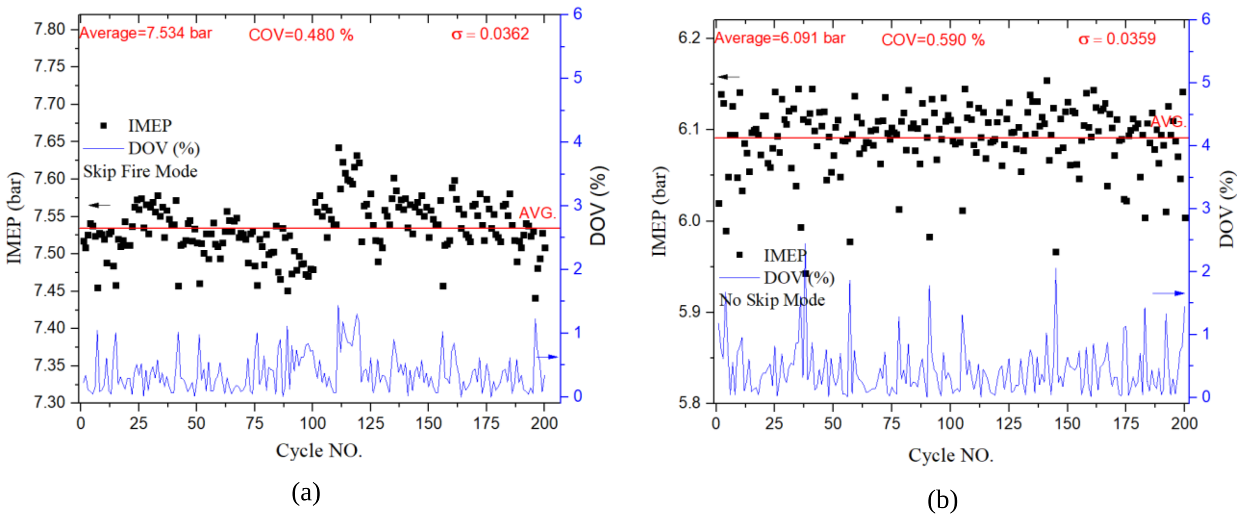
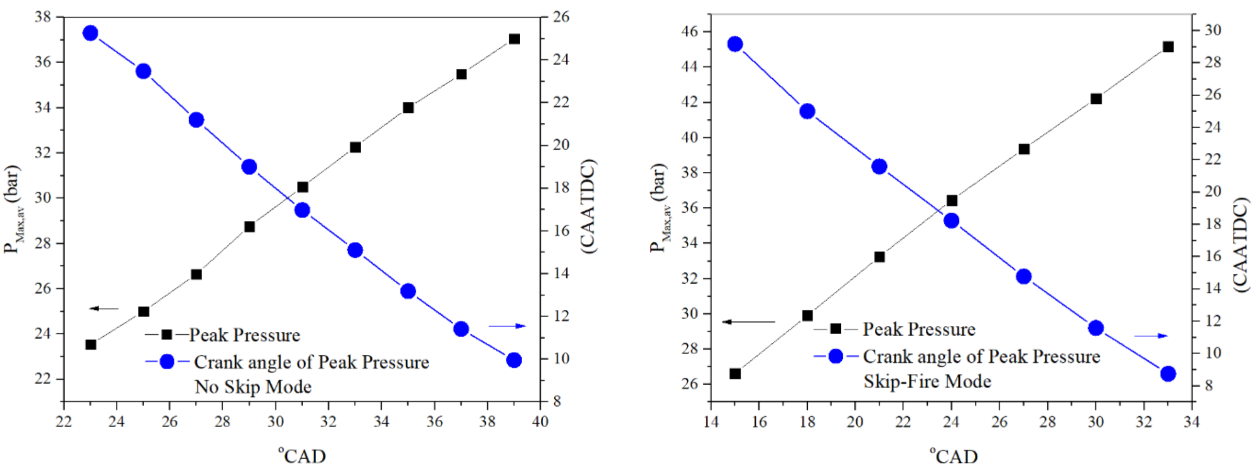


Fig.7. IMEP and DOV changes according to the cycle sequence number at the optimum spark advance in the states: a) with skip fire and b) No skip fire.

The increase in the amount of fuel and air in the cylinder results in higher input energy, leading to a higher IMEP value in the skip fire condition. Furthermore, the removal of residual gases eliminates the diluent effect, resulting in improved fuel-air mixing and more complete combustion. This enhancement in combustion quality contributes to the observed decrease in cyclic variations under skip fire conditions.

Figure 8 illustrates the changes in the pressure peak of the ensemble average cycle and the position of its occurrence relative to spark advance in both skip fire and non-skip fire conditions for a dual-fuel mode consisting of 60% gasoline by mass and 40% natural gas by mass. It was observed that in the non-skip fire condition, the pressure peak occurs at the optimum spark advance, approximately 19 degrees after the top dead center (ATDC). Conversely, in the skip fire condition, the pressure peak is observed at approximately 18 degrees ATDC. During the expansion phase, heat transfer and piston movement contribute to a negative pressure rate, while ongoing combustion generates a positive pressure rate. The absence of burned residual gases in the skip fire condition leads to a higher energy mixture compared to the non-skip fire condition. Consequently, in skip fire mode, the greater positive pressure rate from combustion results in a higher pressure peak, and the peak pressure occurs closer to top dead center of the piston compared to the no skip fire mode.

From the analysis of changes in cylinder pressure in terms of crank angle and engine geometrical characteristics, the changes in the mass fraction burn during combustion were estimated using the Rassweiler-Withrow method [29] under skip fire and no skip fire conditions. The mass fraction burn about the crank angle relative to the crank angle from ignition timing (CAFIT) under skip fire and no skip fire conditions, in the dual-fuel mode of 60% gasoline and 40% natural gas, at the optimum spark advance, is depicted in Figure 9. Given that the spark timing marks the beginning of the initial flame formation and the start of the combustion phase, and since the spark advance varies across different conditions, a parameter is needed to standardize the ignition reference point across all cases. This parameter, expressed as CAFIT (Crank Angle from Ignition Timing), is used to denote the angle from the moment of ignition. The combustion duration as a function of CAFIT is shown in Fig.9. It can be seen that the mass fraction burn graph in the skip fire condition is higher than in the no skip fire condition. The flame development angle, defined as the interval from start of spark ignition to 10% of the mass fraction burn [20] in terms of crankshaft angular steps, as well as the duration to reach 90% of the mass fraction burn (total combustion period) [37], is higher in the skip fire mode.



(a) (b)

Fig.8. Pressure peak changes and its occurrence in the states a) Skip fire and b) No skip fire

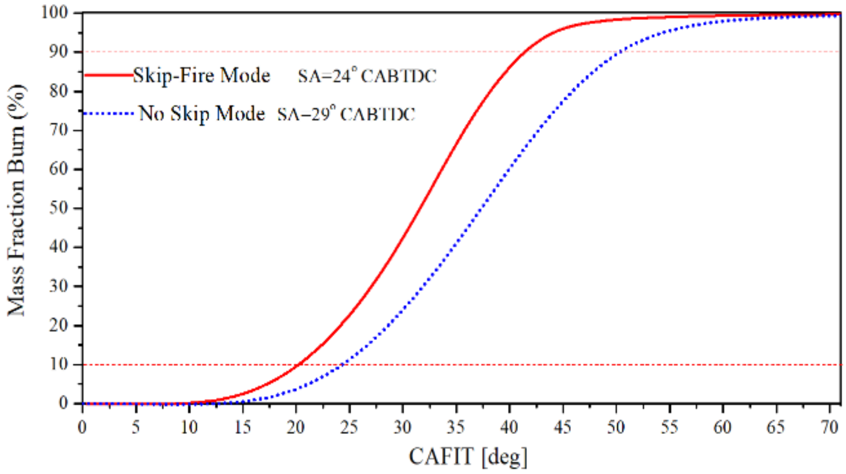


Fig. 9. Mass fraction burn changes with regard to the crank-angle in the states with skip fire and no skip fire

5.Conclusion

The research engine used in the present study was a dual-fuel single-cylinder spark ignition engine operating on a gasoline-natural gas blend and equipped with an electronic control system. The required experimental data were extracted from this engine under skip fire and no skip fire conditions, using a dual-fuel mode of 60% gasoline and 40% natural gas at various spark advances. Data from cycles free of burned residual gas were collected using the 3 to 4 skip fire technique (3 firing and 4 motoring). The following results were obtained from the analysis of the data for the two conditions: skip fire and no skip fire.

- By examining the graphs of in-cylinder pressure in terms of the crank angle of the ensemble average cycle resulting from the experimental cycles at optimum spark advance under conditions with and without skip fire, it was observed that the rate of pressure changes inside the cylinder before the pressure peak was higher in the condition with skip fire.
- A difference of 5 crank degrees was observed in the optimum spark advance between the conditions with and without skip fire. It was noted that the optimum spark advance in the condition with skip fire was nearer to the top dead center of the piston.
- The standard deviation and coefficient of variation for IMEP were found to be lower in the condition with skip fire compared to the condition without skip fire. Another way to say, less cyclic variation was observed in the condition with skip fire mode.
- It was observed that the rate of change of the mass fraction burn was higher in the condition with skip fire compared to no skip fire, indicating that faster combustion occurred in the condition with skip fire.

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