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The effect of using hydrogen at partial load in a diesel-natural gas dual fuel engine

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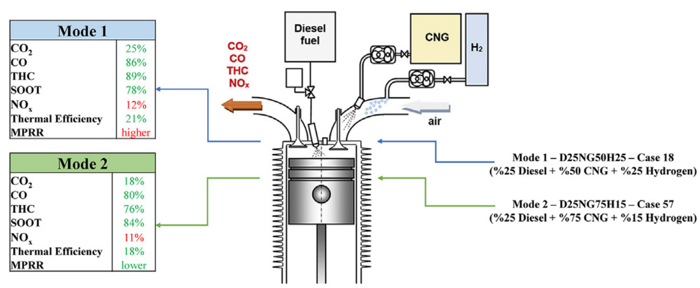
HIGHLIGHTS

- Optimum performance with low injection advance for dual fuel mode at partial load.
- Wide flammability limit with hydrogen-enriched natural gas.
- Tendency to knock at 50% or more hydrogen energy fraction in gas mixture.
- Lower UHC, CO and CO_2 emissions with hydrogen enrichment.
- NO_x emission control with natural gas-hydrogen mixture.

GRAPHICAL ABSTRACT



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ABSTRACT

Alternative fuels are extremely important in terms of both performance and emission values. Natural gas and hydrogen are the foremost effective alternative fuels due to the high energy density and the environmentalist. In the present study, the effects of adding the hydrogen on the performance and emission values in a single-cylinder natural gas-diesel dual fuel engine at partial load was numerically investigated by ANSYS Forte CFD program. In addition, taking into account the effects of different diesel fuel injection advances, the study was expanded. Analyzes were based on two different modes. The first mode (Mode 1) was based on the sharing of energy between gas fuels (natural gas-hydrogen). The second mode (Mode 2) included the effects of hydrogen enrichment on natural gas. Natural gas and hydrogen mixtures in appropriate proportions provided improvements in performance and emission. Increasing the diesel fuel injection advance (after 30° CA BTDC) caused results to deteriorate, especially at high hydrogen ratios. The fact that the hydrogen ratio in the gas mixture was above 50% caused the engine to operate with knock tendency. In the study, highly effective improvements were achieved without knock tendency in the proper diesel injection advance (10°-18° CA BTDC) and gas mixtures (below 50% hydrogen). There was a 21% and 30% improvement in power and total BSFC values for Mode 1 (14° CA BTDC of SOI and D25NG50H25 mixture), respectively, and 36% and 23% for Mode 2 (10° CA BTDC of SOI and D25NG75H15 mixture). On the other hand, NO_x emission was found to be increased by 12% and 11%. The main reason for the increase in

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NO_x emissions was that hydrogen increased local temperatures. However, CO, UHC and PM emissions were sharply reduced as a result of the increase the hydrogen content in the gas fuel. For Mode 1 and Mode 2, CO, UHC and PM emissions decreased by 86%, 89%, 78% and 80%, 76%, 84%, respectively. While increasing the hydrogen ratio was not very effective on ignition delay at low injection advances (10°- 14° CA BTDC), it was extremely effective for thermal efficiency. The thermal efficiency showed an improvement of 21% and 18% for the optimum cases of Mode 1 and Mode 2, respectively.

Introduction

The main reason for air pollution is the release of harmful gases to the air due to energy production. Internal combustion engines (ICEs) are the most used vehicles in energy generation. The energy requirement of ICEs is provided mainly by liquid hydrocarbon fuels. Gasoline and diesel fuels are typical examples. It is predicted that the existing reserves of liquid hydrocarbon fuels will be exhausted after about 50 years. The increase in air pollution is gradually restricted by tightening exhaust emission standards. As such, diesel and gasoline engines have difficulty in responding to emission standards. The use of alternative fuels has become extremely necessary in terms of reserve problems and emission limits. It is preferred that the alternative fuel to be used should not require a major structural change in the engine, be environmentalist and have high energy density. It is very practical to use natural gas and hydrogen as alternative fuel in diesel and gasoline engines.

Scientists have conducted various experimental and numerical studies about the effects of using gas fuels in diesel engines on performance and emission values. Liu and Dumitrescu studied the effect of different H/C ratios in natural gas content on combustion and emissions [1]. Kakee et al. [2] examined the effects of intake air temperature and engine speeds at various methane levels in a diesel engine on performance and emissions. Zhang et al. [3] investigated natural gas mixing ratios at different engine loads and speeds. Li et al. [4] studied the effect of fuel equivalence ratio on combustion properties, thermal efficiency and exhaust gas emissions in a heavy-duty diesel engine. An improvement in thermal efficiency of 27.9% was achieved. Natural gas is effective on combustion temperatures.

When comparing with pure diesel combustion, it has been stated that local temperatures in the cylinder decreased as a result of the slow burning of natural gas by Mittal et al. [5] for dual fuel engine. Therefore, NO_x emissions also tend to decrease. The diesel fuel on dual fuel engines on performance and emission values is vitally important. Mikulski and Wierzbicki [6] studied the effect of burning methane with a high-atom hydrocarbon fuel. Decreasing the diesel fuel injection duration increased the fuel injection pressure and combustion noise. By delayed the injection, higher heat was generated in the cylinder. The raise in injection pressure and advance provided fuel economy [7,8]. The pilot diesel fuel quantity is one the most important element for the start of the dual fuel combustion process to be adjusted optimally [9,10]. Mikulski et al. [11] suggested that the amount of natural gas fuel should be increased while the pilot diesel fuel was injected in two stages in order to reduce unburned HC emissions and increase thermal efficiency. Performing pilot fuel injection in two stages and not delaying the injection causes increase in thermal efficiency and decrease in NO_x emissions [10,12–15]. In the partial load area, the amount of pilot diesel fuel should be boosted in order to ascend engine performance and reduce emissions. As the combustion of natural gas improves, it has been observed that the unburned products, CO and HC emissions have decreased [16]. Pilot diesel fuel injection has extremely important effects on in-cylinder pressure and combustion stability. The cetane number (CN) and aromatic content of pilot diesel fuel are also key factors in terms of ignition delay and thermal efficiency on combustion performance [17]. It has been observed that in natural gas-diesel dual fuel engines, it is possible to control emissions as well as performance. Experimental tests were performed at different air-fuel ratios, various loads and engine speeds in a single-cylinder dual fuel engine using different gas proportions, emission values were investigated. It has been shown that NO_x and PM emissions can be controlled in the dual fuel combustion mode [18]. Studies have shown that in general, natural gas is an environmentalist fuel and causes reasonable results in performance values. However, UHC, CO and CO2 emissions from natural gas need to be further reduced.

In some studies, It is also seen that hydrogen gas alone is used as dual fuel in diesel engines. The effect of adding hydrogen to the intake air of a single cylinder four-stroke diesel engine on performance and emission values was investigated experimentally [19,20]. It was observed that CO and PM emissions decrease with increasing hydrogen ratio. For 22% and 53% hydrogen ratios, CO emissions improved by 67.3% and 69.2%, respectively, while improvements were observed in PM emissions by 43.6% and 58.6%, respectively. In the hydrogen-diesel dual fuel combustion mode, at different at a constant engine speed, hydrogen is sent to the combustion chamber at different volumetric ratios, performance and emission values were obtained [21]. Increasing the amount of hydrogen at all loads resulted in an increase in thermal efficiency. CO, UHC and SOOT emissions are drastically reduced by adding hydrogen at all loads.

According to the results of a numerical study investigating the effects of direct injection of hydrogen into the combustion chamber under partial load at a diesel-hydrogen dual fuel mode, it was stated that early injection of hydrogen improved the performance parameters but increased NO_x emissions [22]. Jamrozik et al. [23] experimentally investigated engine performance and emission values for different ratios of hydrogen in hydrogen-diesel dual fuel mode in a diesel engine. Increasing the hydrogen amount to 30% at full load resulted in 13% increase in maximum combustion pressure, 46% increase in heat release rate and 35% increase in pressure increase rate (bar/CA). Yang et al. [24] simulated the effects of hydrogen in a diesel engine. At rates varying between 6% and 20%, hydrogen was sent to the combustion chamber. It was seen that the rate of pressure and heat release first increased and then decreased after a certain hydrogen rate. PM emissions decreased while NO_x emissions increased. Khairallah and Koylu [25] analyzed numerically the effects of hydrogen-diesel dual fuel operation on the performance and emission values of a single cylinder four-stroke diesel engine. Since hydrogen structurally does not contain carbon atoms (C), it was observed that the addition of hydrogen fuel reduced CO and CO_2 gas emissions compared to diesel fuel combustion conditions. Suzuki and Tsujimura [26] conducted experimental and numerical study using hydrogen as a dual fuel in a diesel engine. The effects of different EGR ratios and pilot diesel fuel injection advances in dual fuel operating conditions with hydrogen were also examined. It was emphasized that increasing EGR rates reduced NO_x emissions, but there was a slight loss in thermal efficiency. Hydrogen dual fuel combustion has resulted in significant improvements in CO, CO2 and UHC emissions in general, while increasing NO_x emissions.

In dual fuel engines, it is known that using hydrogen-natural gas fuel mixture lead to more positive results on engine performance and emission values. The effect of adding hydrogen to natural gas (HCNG), which is a premix fuel, on the performance and emission values in a diesel-natural gas dual fuel engine was investigated experimentally [27–29]. The increase in the volumetric ratio of hydrogen in the mixture caused the maximum pressure and heat release rates to increase inside the cylinder, while the ignition delay decreased. The unburned products CO and HC emissions were reduced by increasing oxidation as a result of increased thermal efficiency. Arslan and Kahraman [30] experimentally investigated the effects of natural gas and hydrogen-enriched natural gas on the performance and emission values of a diesel engine. It has been stated that the crank angle, where the maximum incylinder pressure is seen, with the increase in engine speed, moves further to the right from the top dead center. The in-cylinder pressure value, which increased by 4.5% under diesel-natural gas conditions, increased by 6.5% with the addition of hydrogen. The effects of methane (CH_4), carbon dioxide (CO_2) and hydrogen (H_2) mixtures on the performance and emission values of a common rail four-cylinder diesel engine in dual fuel combustion mode were carried out with an experimental study [31]. As a result of enrichment of synthetic biogas with hydrogen, HC emissions decreased by 4.97%–30.92% and CO_2 emissions by 5.16%–10%. Tutak et al. [32] experimentally investigated the effects of hydrogen enriched natural gas on performance and emissions in a natural gas-diesel dual fuel engine. According to the test results, as a result of the enrichment of natural gas with hydrogen, the combustion time was 30% shorter than the case without enrichment, and NO_x emissions increased. The critical value of hydrogen in the mixture was determined as 19%. More than 19% hydrogen has been reported to cause engine knock and pressure variation from cycle to cycle. A three-cylinder turbocharged diesel engine is numerically modeled in a CFD program (AVL Fire) to examine the effects of hydrogen and CNG fuels on performance and emission values in dual fuel combustion mode [33]. The results showed that the addition of hydrogen had a positive effect on engine power and thermal efficiency. It also reduced fuel consumption, HC emissions and PM emissions. On the other hand, NO_x emissions increased as a result of the increase in local temperatures. A study was conducted by Ouchikh et al. [34], a singlecylinder air-cooled diesel engine was tested as a conventional diesel engine and then it was operated in the dieselnatural gas dual fuel mode and in the hydrogen addition mode to the natural gas. It was emphasized that adding hydrogen to natural gas fuel caused the improvement of the gas mixture combustion phase, thus causing increases in cylinder peak pressures and heat release rates, especially at high loads. The hydrogen ratio shortened the combustion phase of gas fuel and increased the combustion stability. It was stated that the ignition delay did not change much at all gas mixing ratios.

When the studies are examined, it is quite complex to provide performance and emission control. However, if the thermodynamic and chemical properties of fuels can be combined in appropriate proportions, reasonable results in terms of performance and emission values are obtained. Natural gas is a hydrocarbon fuel with high octane number, low C/H ratio and high energy potential per unit mass. A high octane number indicates that it is resistant to knock tendency and can be used in diesel engines. It is known that natural gas causes a decrease in in-cylinder local temperatures. Since the flammability limits of hydrogen extend to very wide fuel mixture ratios, it has a positive effect on engine efficiency. The combustion rate of hydrogen is very high and the in-cylinder regional temperatures are directly affected by the combustion rate. By enriching the natural gas with hydrogen, the performance is increased as well as the emission values are reduced. In the current study, unlike the diesel-natural gas, diesel-hydrogen and diesel-natural gas-hydrogen studies conducted in the literature, the effects of the energy fraction distribution of gas fuels (natural gas-hydrogen) on the combustion process were investigated by keeping the diesel fuel constant (25% energy fraction) as an ignition element. Since diesel fuel has a high carbon ratio, it causes high emission values. The effects of diesel fuel injection advance on the fixed energy fraction were also investigated. The ignition of natural gas and hydrogen mixtures in a CI engine is provided with a low amount of diesel fuel, and the power was supplied with gaseous fuels. In present study, the results of an experimental study were taken as reference [35–37]. The numerical model of the diesel engine was established by a CFD program. The model was validated by comparing the numerical results with the experimental results. Dual fuel combustion at different energy ratios of hydrogen-methane gas mixture was numerically investigated at partial load.

Numerical model

In present study, the effect of alternative fuels in partial load regions were numerically investigated on performance and emissions. Therefore, a numerical model were created by considering the experimental study for the low load region. The test cases conducted by Guo et al. [35–37] were used for setup the numerical model. Test cases were carried out by operating a dual fuel engine at a fixed engine speed (910 rpm) under partial load (25% load-4.05 bar BMEP). 75% of the energy was supplied from natural gas and the remaining 25% was from diesel fuel. Performance and emission values were obtained in different diesel fuel injection advances (10°, 14°, 18°, 22°, 26°, 30°, 34°, 38°, 42°, 46° and 50° CA BTDC).

Numerical model was generated with the ANSYS Forte CFD program. ANSYS Forte is a CFD program that models closed process in-cylinder combustion [39,40]. The process includes the time between closing the intake valve and opening the exhaust valve. The program assumes that the fluids are newtonian fluids.

All chemical substances included in the combustion process as liquid and gas are calculated with the help of the conservation of species equation [40]. The calculation of fluids is governed by the form of the basic conservation equations for compressible fluids. Basic conservation equations are mass conservation, momentum conservation and energy conservation equations. Chemkin Pro solvent for the combustion model is included in the ANSYS Forte program.

The reactions of the species according to the activation energies are gradually resolved by the program with the reduced mechanisms obtained from detailed chemical mechanisms. N-heptane (n-C_7H_{16}) represented diesel fuel with similar properties, and methane (CH_4) represented natural gas fuel. The reduced mechanism obtained by reduction of skeletal kinetic mechanisms was used for n-heptane and methane. The mechanism consists of 137 species and 1022 reactions [41]. In order to be closer to diesel fuel properties, n-heptane fuel were made to behave like n-tetradecane (n-$C_{14}H_{30}$) in physical properties [40]. The spray properties were obtained using the radius of influence model for the droplet collision model, and the radius of influence was set at 0.2 cm [40]. Sector mesh was applied since the injector cylinder center and the nozzle holes were symmetrical with respect to each other. Since the number of injector nozzle holes is 6, the 60° part of the combustion chamber is taken into account geometrically. In CFD programs, it is recommended that cells be as square as possible and the cell dimensions should be minimum 0.5 mm and maximum 2 mm [40,42]. The number of cells should be denser around the injector where the injection is made. Sector combustion chamber with approximately 75,000 cells was included in simulations. Numerical results remained stable after 75,000 meshes. The combustion chamber is shown in Fig. 1. The start and finish timings of the analysis are the closing of the intake valve and the opening of the exhaust valve respectively. Considering the fact that it is a work carried out under partial load conditions, it was estimated that the swirl ratio value was not very high. The initial swirl ratio value was assumed to be 0.5 [38,43,44]. The spray angle was taken as 130° and the spray cone angle as 15°. The engine specifications are shown in Table 2. Table 3 and Table 4 shows initial conditions and boundary conditions respectively.

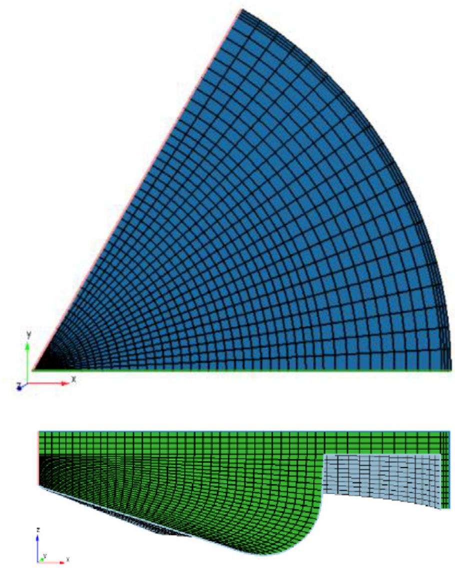

Fig. 1 – Combustion chamber for numerical model.

Model validation

The validation of the model was successfully demonstrated by comparing the obtained numerical data with the

Table 1 – Test Conditions [35–37].

Cases	Pintake	Tintake	Engine Speed	SOI	EOI	Air Flow	Diesel Flow	Naturalgas Flow
–	(bar)	(K)	(rpm)	(BTDC)	(BTDC)	(kg/h)	(kg/h)	(kg/h)
1				10°	3.57	66.93	0.5238	1.470
2				14°	7.71	67.21	0.4948	1.389
3				18°	11.81	67.57	0.4710	1.306
4				22°	15.89	67.45	0.4503	1.259
5				26°	19.95	67.08	0.4370	1.237
6	1.05	313	910	30°	23.86	66.70	0.4552	1.234
7				34°	27.99	67.24	0.4337	1.215
8				38°	32.01	67.62	0.4314	1.199
9				42°	36.04	67.73	0.4167	1.190
10				46°	40.12	67.65	0.4122	1.179
11				50°	44.15	67.85	0.4134	1.164

Table 2 – Engine specifications.

Number of Cylinders	1
Bore x Stroke	137.2 mm × 165.1 mm
Compression Ratio	16.25:1
Stroke Volume	2.44 L
Number of Valves	2 intake & 2 exhaust
Diesel Injection	Direct Injection
Natural Gas Injection	Port Injection
Maximum Power	74.6 kW (@2100 rpm)

Table 3 – Initial conditions.

Intake Pressure (bar)	1.02
Intake Temperature (K)	360
Turbulent Kinetic Energy (m^2/s^2)	10
Turbulent Length Scale (m)	0.003
Turbulent Model	Rans RNG k-epsilon
Swirl Ratio	0.5
Spray Angle	130°
Spray Cone Angle	15°
Nozzle Hole Number	6
Nozzle Hole Diameter (mm)	0.23
IVO (CA BTDC)	358.3°
IVC (CA BTDC)	169.7°
EVO (CA ATDC)	145.3°
EVC (CA ATDC)	348.3°

Table 4 – Boundary conditions.

Cylinder Head	Wall, 425 K
Piston	Mesh Movement, 450 K
Liner	Wall, 400 K
Segment Cut	Periodic inlet outlet

experimental data [35–37]. The numerical model was created by considering the fuel (diesel and natural gas) and air flow rates used in the test conditions. Fuel and air flow rates are shown in Table 1. While validating the established numerical model, both performance and emission values were investigated. In the validation study, the in cylinder pressure and HRR values were compared. In addition, the compatibility of NO_x, BMEP and ignition delay was also monitored. The compatibility of experimental and numerical results with each other was shown in Fig. 2.

Ignition delay and thermal efficiency values, which are extremely important at the start of the combustion process and for the continuation of the combustion, have also been compared. Table 5 shows the error analyzes between the test and numerical results for performance values. Reasonable results were obtained in all injection advances, and the parametric study phase was started.

Parametric study

Numerical studies were carried out in dual fuel mode in which natural gas and hydrogen were used at different diesel fuel injection advances. The case matrix for parametric study was shown in Table 6. In numerical studies, the results were evaluated by considering two modes. In the analyzes performed on the first mode (Mode 1), provided that the injected diesel fuel was kept constant (25% energy fraction), the remaining 75% of energy was shared between hydrogen and natural gas fuels. Gradually, the amount of natural gas fuel was reduced and transferred to hydrogen fuel. Energy balance was achieved according to the following eq. (1) by considering the methane as natural gas and n-heptane as diesel fuel.

$$\%CH_4 = \frac{m_{CH_4} xLHV_{CH_4}}{m_{diesel} xLHV_{diesel} + m_{CH_4} xLHV_{CH_4} + m_{H_2} xLHV_{H_2}} x100 \tag{1}$$

The abbreviations in the table indicate the energy distribution (%) of diesel, natural gas and hydrogen fuels. For example; The operating point D25NG65H10 shows that 25% of the energy was obtained from diesel fuel, 65% from natural gas fuel and the remaining 10% from hydrogen fuel. D25NG65H10 belongs to the Mode 1 set of operating points. In this way, by keeping the total energy constant at 100%, only an energy sharing was made between natural gas and hydrogen.

In the second mode, extra energy transfer was provided by adding hydrogen to the system. According to Mode 2, 25% energy fraction diesel fuel amount and 75% energy fraction natural gas fuel were kept constant and extra hydrogen fuel in different energy fractions was sent to the combustion chamber. Energy balance for Mode 2 was provided by the following Eq (2).

$$\%H_2 = \frac{m_{H_2} xLHV_{H_2}}{m_{diesel} xLHV_{diesel} + m_{CH_4} xLHV_{CH_4}} x100 \tag{2}$$

D25NG75H20 case belongs to Mode 2 operating conditions. For D25NG75H20 case, it means providing an extra 20% hydrogen fuel to the system (into the cylinder) by keeping as fixed the natural gas energy fraction of 75% with the 25% diesel energy fraction under experimental conditions. As can be seen, the total energy ratio is not 100%, it contains 20% extra energy input. For Mode 1 and Mode 2 gas mixtures, the effects of diesel fuel injection advances were investigated with 4° CA increments in the range of 10°-38° CA BTDC. The representative scheme of the Mode 1 and Mode 2 analyzes performed in the numerical study is shown in Fig. 3.

Results and discussion

Natural gas can be used alone as fuel in the air-gas premix charge [43,45]. Similarly, hydrogen fuel can be used alone [21–23,46]. Natural gas and hydrogen have different thermodynamic effects on engine performance and exhaust gas emissions. The fact that natural gas is a slow burning gas causes a decrease in the local temperatures in the cylinder. It is known that nitrogen oxide (NO_x) emissions decline and the proportion of harmful gases formed as a result of dissociation decreases with decreasing local temperatures.

Hydrogen is a fuel that has high mass energy density and can be easily burned and does not cause harmful emission release as combustion product. As a result of hydrogen combustion, harmful gases such as CO, CO_2, SO_x and unburned hydrocarbons (UHC) do not occur. It only emits water vapor

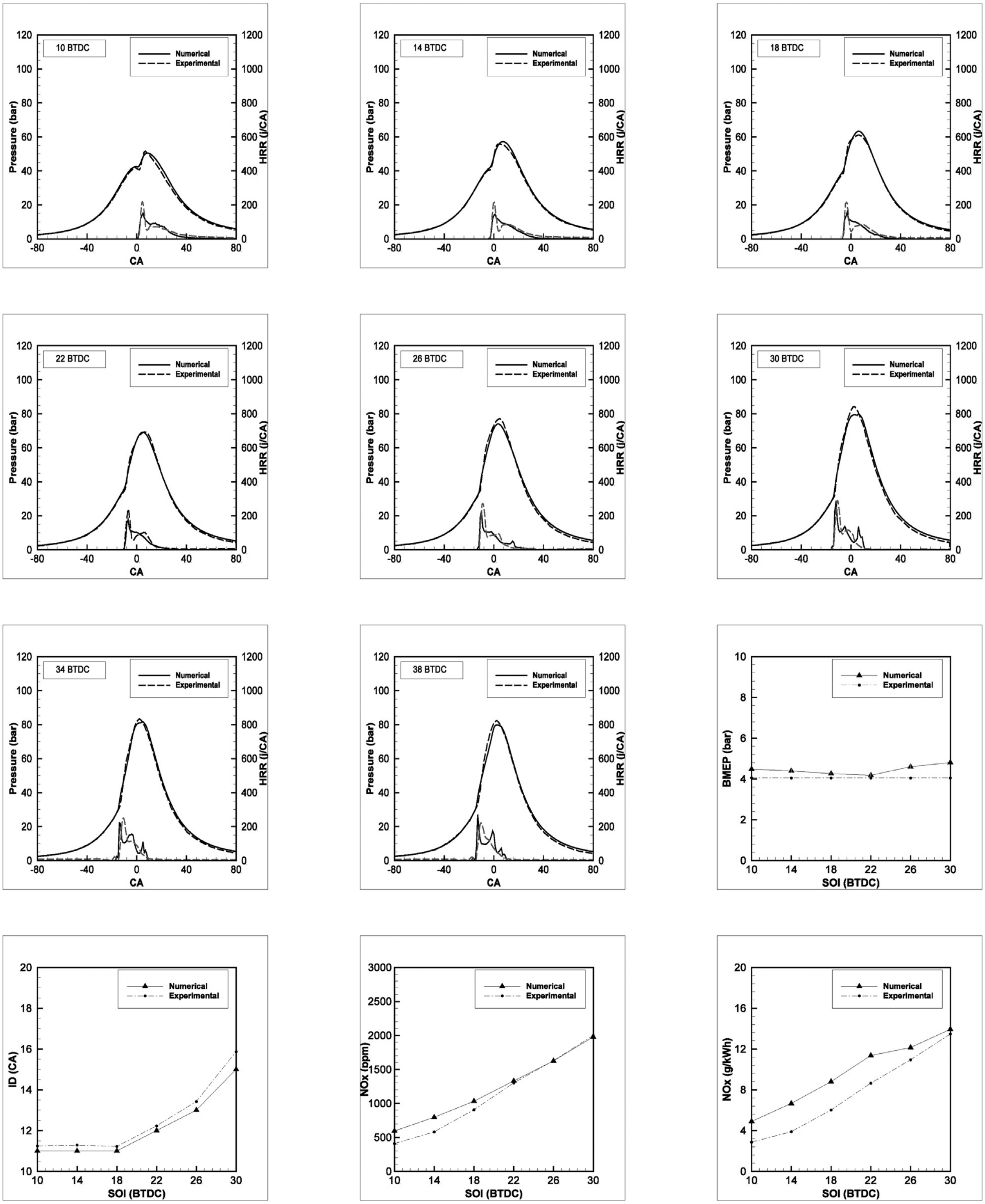

Fig. 2 – Comparison of numerical and test results [35–37].

(H_2O) that does not harm the environment. Hydrogen ignition range extends to very wide fuel mixture ratios.

Wide flammability limit of hydrogen among gaseous fuels have a positive effect on engine efficiency. Hydrogen can be flamed between 0.14 and 4.35 values of excess air ratio (λ). The local temperatures inside the cylinder are directly affected by the burning rate. The level of nitrogen oxide (NO_x) components formed as a result of hydrogen combustion is higher than other fuels due to the increase in local temperature levels during the combustion process [23,32,33].

Both gas fuel types (natural gas and hydrogen) have advantages and disadvantages. It has been observed that using

Table 5 – Relative error between test cases and numerical model.

		SOI (CA BTDC)							
		10°	14°	18°	22°	26°	30°	34°	38°
BMEP (bar)	Experimental	4.05	4.05	4.05	4.05	4.05	4.05	4.05	4.05
	Numerical	4.47	4.39	4.26	4.18	4.59	4.8	4.53	4.35
	% Rel. Error	10.37	8.39	5.18	3.21	13.33	18.51	11.85	7.41
ID (CA)	Experimental	11.25	11.29	11.22	12.23	13.43	15.87	19.52	23.73
	Numerical	11.02	11.02	11.02	12	13.02	15.01	17.01	20.02
	% Rel. Error	2.014	2.39	1.78	1.88	3.05	5.42	12.85	15.63
Thermal Efficiency (%)	Experimental	30.82	33.02	34.62	35.45	35.88	35.02	36.1	36.62
	Numerical	30.83	32.02	32.93	33.56	37.6	39	37.78	36.67
	% Rel. Error	0.03	3.03	4.88	5.33	4.79	11.36	4.65	0.13

Table 6 – Parametric case matrix.

Mode 1								
SOI (CA BTDC) = >	10°	14°	18°	22°	26°	30°	34°	38°
D25NG75H00	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
D25NG65H10	Case 9	Case 10	Case 11	Case 12	Case 13	Case 14	Case 15	Case 16
D25NG50H25	Case 17	Case 18	Case 19	Case 20	Case 21	Case 22	Case 23	Case 24
D25NG25H50	Case 25	Case 26	Case 27	Case 28	Case 29	Case 30	Case 31	Case 32
D25NG00H75	Case 33	Case 34	Case 35	Case 36	Case 37	Case 38	Case 39	Case 40
Mode 2								
SOI (CA BTDC)= >	10°	14°	18°	22°	26°	30°	34°	38°
D25NG75H00	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
D25NG75H05	Case 41	Case 42	Case 43	Case 44	Case 45	Case 46	Case 47	Case 48
D25NG75H10	Case 49	Case 50	Case 51	Case 52	Case 53	Case 54	Case 55	Case 56
D25NG75H15	Case 57	Case 58	Case 59	Case 60	Case 61	Case 62	Case 63	Case 64
D25NG75H20	Case 65	Case 66	Case 67	Case 68	Case 69	Case 70	Case 71	Case 72
D25NG75H25	Case 73	Case 74	Case 75	Case 76	Case 77	Case 78	Case 79	Case 80

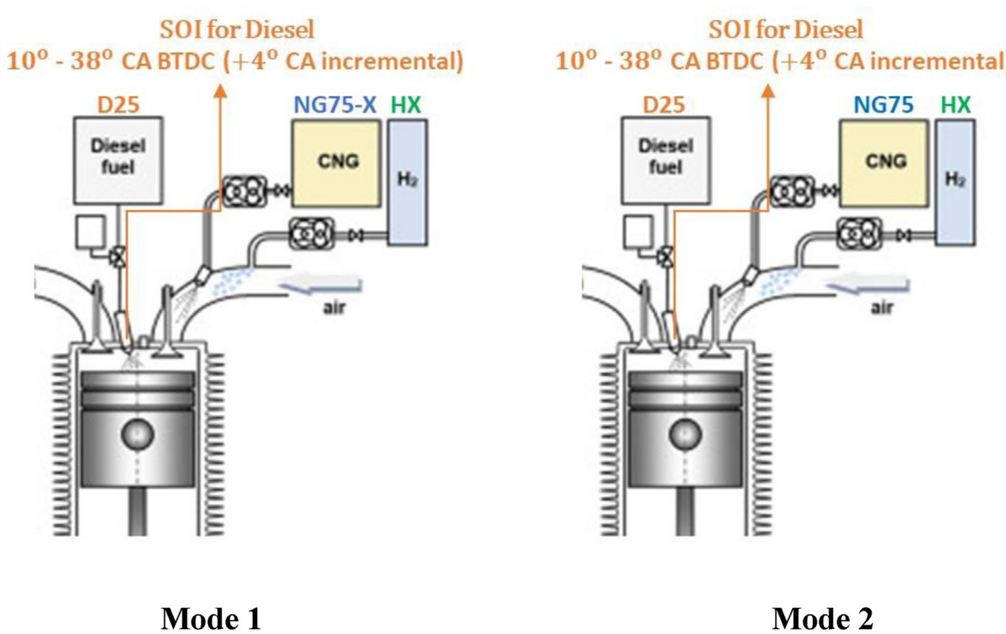

Fig. 3 – Representative numerical model scheme for Mode 1 and Mode 2.

them together as a mixture gives more positive results on performance and emission values [21,32,33]. There are many reasons why hydrogen and natural gas are preferred together. It is known that hydrogen is a fast burning fuel, unlike natural gas that burns slowly. The burning rate can be controlled by using two gas fuels in the form of a mixture. When considered

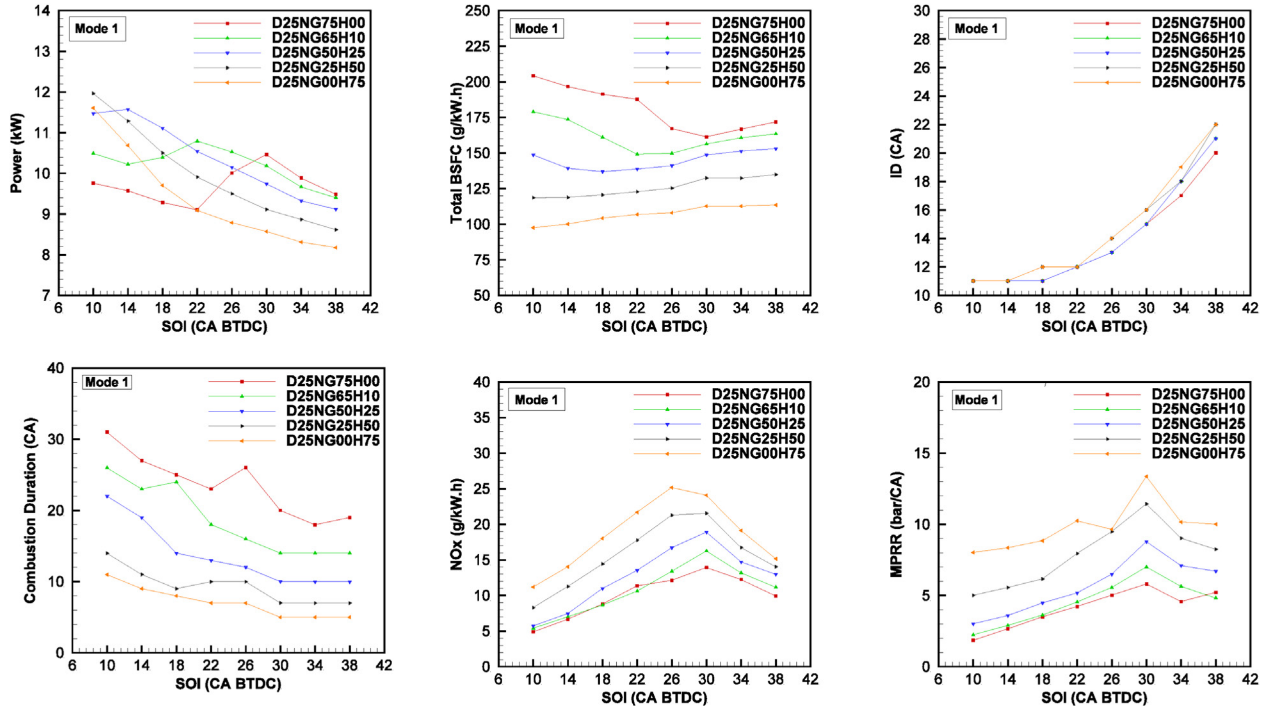

Fig. 4 – Mode 1 Results - The Effect of SOI for Fixed Gas Mix (Hydrogen and Natural gas).

in terms of emissions, it was seen that positive improvements were achieved in many studies [23,33,46]. It were observed that in-cylinder regional temperatures were high with the combustion of hydrogen [19–21,25], but when used with natural gas, the local temperatures decreased [34].

The increase in NO_x emissions resulting from hydrogen combustion can be reduced by using hydrogen together with natural gas [28,34]. Hydrogen is not a hydrocarbon as a fuel. Therefore, it does not form unburned hydrocarbon (HC) and carbon monoxide gas (CO), which are an incomplete combustion product, in combustion products. Emission types resulting from natural gas combustion can be reduced by applying hydrogen and natural gas as a mixture. Considering situations, using hydrogen and natural gas fuels together has an extremely important effect on performance and emission values.

In present study, analyzes were based on two different modes. The first mode (Mode 1) was based on the sharing of energy between gaseous fuels (Natural gas-Hydrogen). The second mode (Mode 2) included the effects of hydrogen enrichment on the system. 40 analysis results were obtained separately for each mode and interpreted for 80 different cases in total. It is aimed to obtain optimum performance and emission values by examining the thermodynamic and chemical properties of hydrogen and natural gas under different modes. Therefore, the addition of hydrogen in the form of energy sharing and as an extra energy input to the cylinder has been investigated separately. In addition, the effects on combustion and performance are revealed by comparing both modes at the same injection advance. In all cases of Mode 1 and Mode 2, the amount of diesel fuel injected was kept constant, and the performance and emission values were observed with the changes of natural gas-hydrogen energy fractions and injection advance.

Mode 1

By keeping the 25% energy obtained from diesel fuel constant, the remaining 75% of energy was shared between natural gas and hydrogen at different energy rates. Improvements were achieved on performance and emission values with different pilot diesel fuel injection advances.

As a result of increasing the hydrogen ratio in the gas mixture, it was observed that the performance values increased first in low injection advances (10° and 18° CA BTDC), but decreased after a certain hydrogen ratio (50% of hydrogen ratio and above). The variation of the performance according to the hydrogen ratio and injection advance is shown by Fig. 4 and Fig. 5.

Hydrogen is a fuel that burns faster compared to natural gas. Therefore, the increasing combustion rate caused the maximum in-cylinder pressure to regress towards the top dead center [23,30,32]. As a result of the increase in the hydrogen energy ratio, in-cylinder temperature fluctuations and consequently in-cylinder pressure increases were observed [47]. The variation of the in-cylinder pressure value according to the crank angle is shown by Fig. 9 and Fig. 10. Crank angle at which maximum pressure is seen should be around 7°-15° CA ATDC [48–53]. The maximum pressure that occurs before or after interval means power loss. If the hydrogen ratio is desired to be increased, the injection advance should be reduced. At lower diesel fuel injection advances, BSFC was significantly decreased by increasing the hydrogen energy fraction ratio (Fig. 5). The hydrogen ratio for

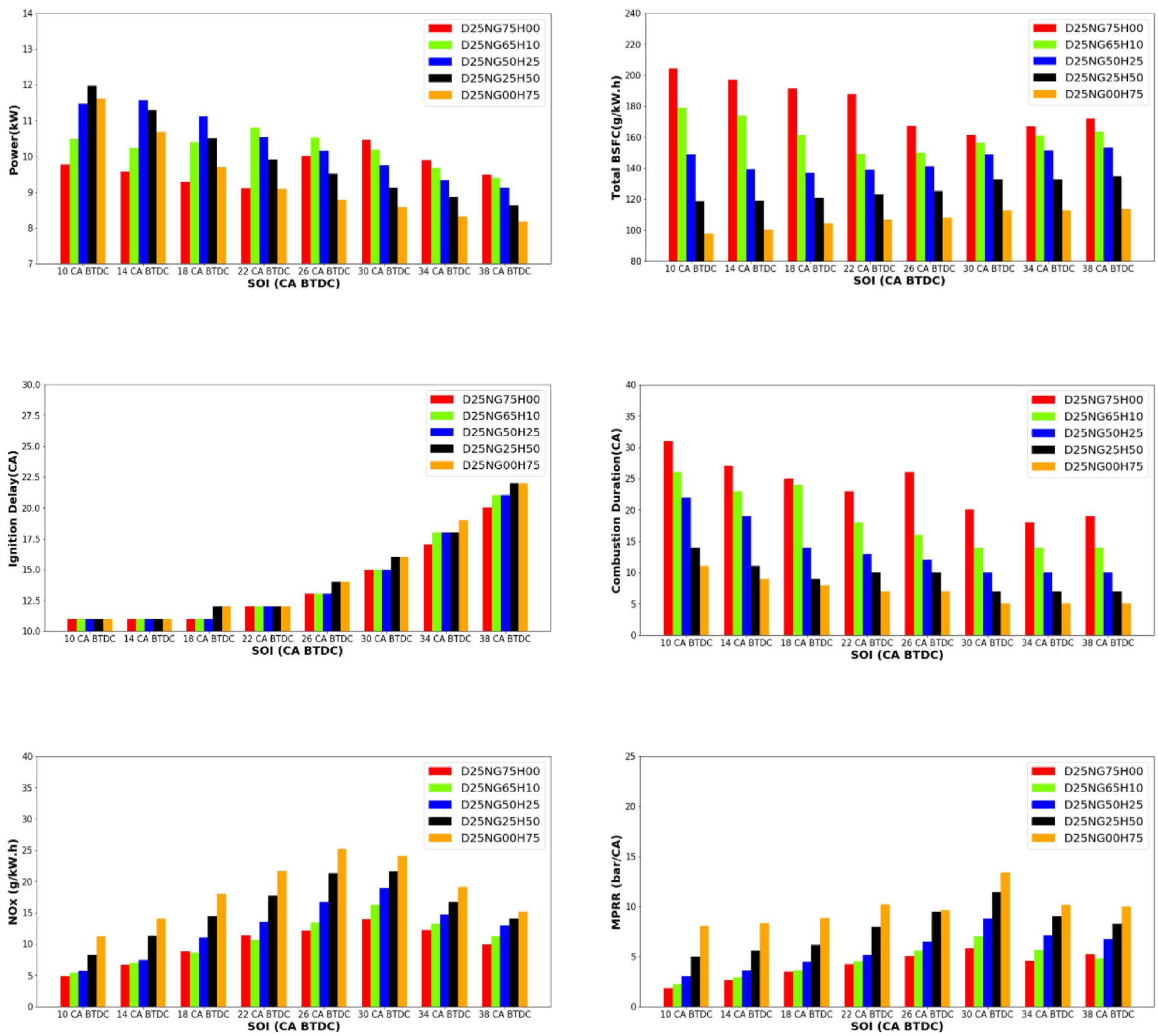

Fig. 5 – Mode 1 Results - The Effect of Gas Mix (Hydrogen and Natural gas) for Fixed SOI.

Table 7 – Comparison of optimum case for Mode 1 and test case.

	Case 2	Case 18
Power (kW)	9.575	11.58
Torque (N.m)	100.477	121.47
BSFC (g/kW.h)	196	136.27
Ignition Delay (CA)	11°	11°
Thermal Efficiency (%)	32.02	38.71
Combustion Duration (CA)	27°	19°
MPRR (bar/CA)	2.66	3.6
Max. Pressure (bar)	57.25	67.45
CA for Max. Pres. (ATDC)	7° CA	8° CA
Max. Mean Temperature (K)	1504	1779
NO_x (g/kW.h)	6.66	7.46
SOOT (g/kW.h)	0.000766	0.000172
CO (g/kW.h)	9.94	1.42
HC (g/kW.h)	35.21	4.04

lower injection advances had almost no effect on ignition delay. It played a more active role in high injection advances (before 30° CA BTDC). At low diesel fuel injection advances (10°-18° CA BTDC), it did not affect the ignition delay but increased the combustion rate. Excessive increase in combustion rate after 50% hydrogen ratio in the gas mixture led to a decrease in performance values (Fig. 5). The negative effects of excess hydrogen ratio have also been seen in studies in the literature [32,54].

At 22°-30° CA BTDC injection advances, more better performance was achieved with the presence of only natural gas in gas fuel. With the addition of hydrogen to the gas mixture, the performance values decreased (Fig. 4). The main reason is that although the mass energy density of hydrogen is high, its volumetric energy density is low. Since the amount of hydrogen taken in response to the decreased natural gas energy occupied more space in the cylinder, the volumetric efficiency decreased and the ignition delay increased. At high injection advances (before 30° CA BTDC), performance values decreased in all cases (Fig. 4). The main reason was that since diesel fuel started to be injected at a lower temperature, the ignition event in the cylinder started later. Ignition delay increased and a higher rate of fuel accumulated in the cylinder until ignition occurred. The increased ignition delay suddenly increased the in-cylinder peak pressure and temperatures and caused the burning time to decrease. Therefore, it is not

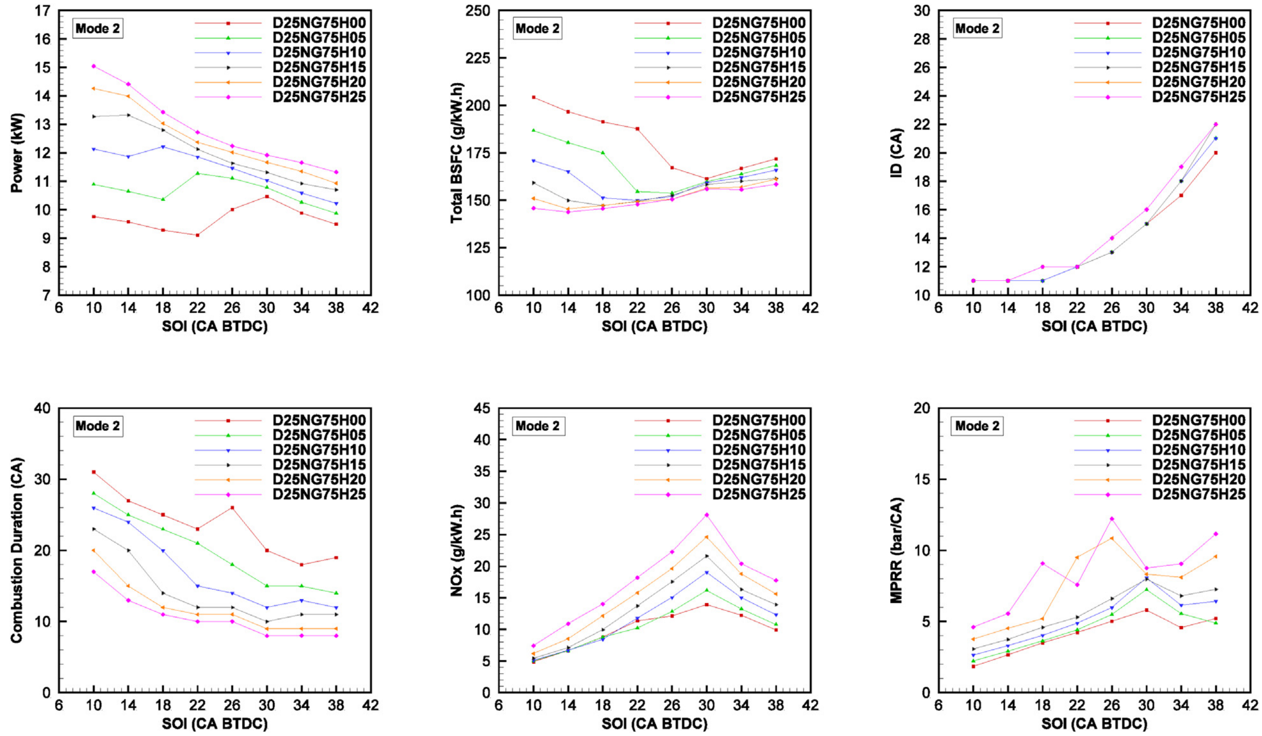

Fig. 6 – Mode 2 Results - The Effect of SOI for Fixed Gas Mix (Hydrogen and Natural gas).

appropriate to increase the injection advance too much in order to not loss the performance values. Considering the performance values, it was observed that low injection advances and gas mixture cases at maximum 50% hydrogen ratio achieved better results for Mode 1.

In terms of emission values, reducing the injection advance improved all emissions except NO_x emission at all gas mixture ratios [55]. The increase in the hydrogen ratio caused an increase in NO_x emissions as it increased the local temperature values in the cylinder (Fig. 4). In order to reduce NO_x emission, the natural gas ratio in the gas mixture should be increased. Thus, desired results are obtained in low load regions in terms of performance and emission values. Since natural gas burns more slowly than hydrogen, it prevents the rise of local temperatures inside the cylinder. NO_x emissions as a function of temperature are reduced. Since the in-cylinder peak pressures increased with the increase of the hydrogen ratio in the gas mixture, the hydrogen ratio in the mixture should be kept at a certain value in order to avoid knock. Tendencies and engine noise problems [23,30,32].

The knock tendency is shown in Fig. 4 as Rate of Maximum Pressure Rise (MPRR). Increasing the hydrogen ratio led to the pressure curves into a more pointed structure (Fig. 9). This means that the burning time was shortened and sudden pressure increases occurred. The rapid development of the combustion led to a loss in performance values, considering that the area under the pressure curve is the power value. Thus, the energy balance must be provided with natural gas and hydrogen fuel with a suitable mixing ratio. The high thermal energy and rapid burning feature of hydrogen reduced the rate of the regions in the cylinder where combustion does not occur. A lower unburned zone means lower emissions of CO and UHC, which are incomplete combustion products [23,31]. In addition, the absence of carbon atoms in hydrogen led to improvements in SOOT emissions (Table 7). For Mode 1, the optimum value of diesel fuel injection advance in terms of performance and emission values was determined as 14° CA BTDC. Considering the Mode 1 analyzes, it was seen that the optimum case in terms of both performance and emission values was 50% natural gas and 25% hydrogen gas mixture at 14° CA BTDC injection advance (Case 18). It provided an improvement of 21% in engine power and 30% in specific fuel consumption. It was observed that there was an increase of approximately 12% in the NO_x emission value. Remarkable results were obtained for other types of emission. A reduction of 89%, 78% and 86% was achieved in HC, SOOT and CO emissions, respectively (Table 7).

Mode 2

Extra energy input to the system was provided by hydrogen fuel by keeping fixed the energy of 25% and 75%, respectively, from diesel and natural gas. With different diesel injection advances, improvements were observed in performance and emission values for Mode 2 as well. Hydrogen addition for all injection advances improved performance values, especially at the low injection advances (10° and 14° CA BTDC), while the hydrogen enrichment process had little effect on the ignition delay (Fig. 6). It was found that the SOI (Start of Injection) had a greater effect on the ignition delay (Figs. 6 and 7). While the increase in hydrogen ratio caused an increase in thermal efficiency at low injection advances, it lost its effect on thermal efficiency at high injection advances. It was observed that the thermal efficiency decreased in cases after the injection

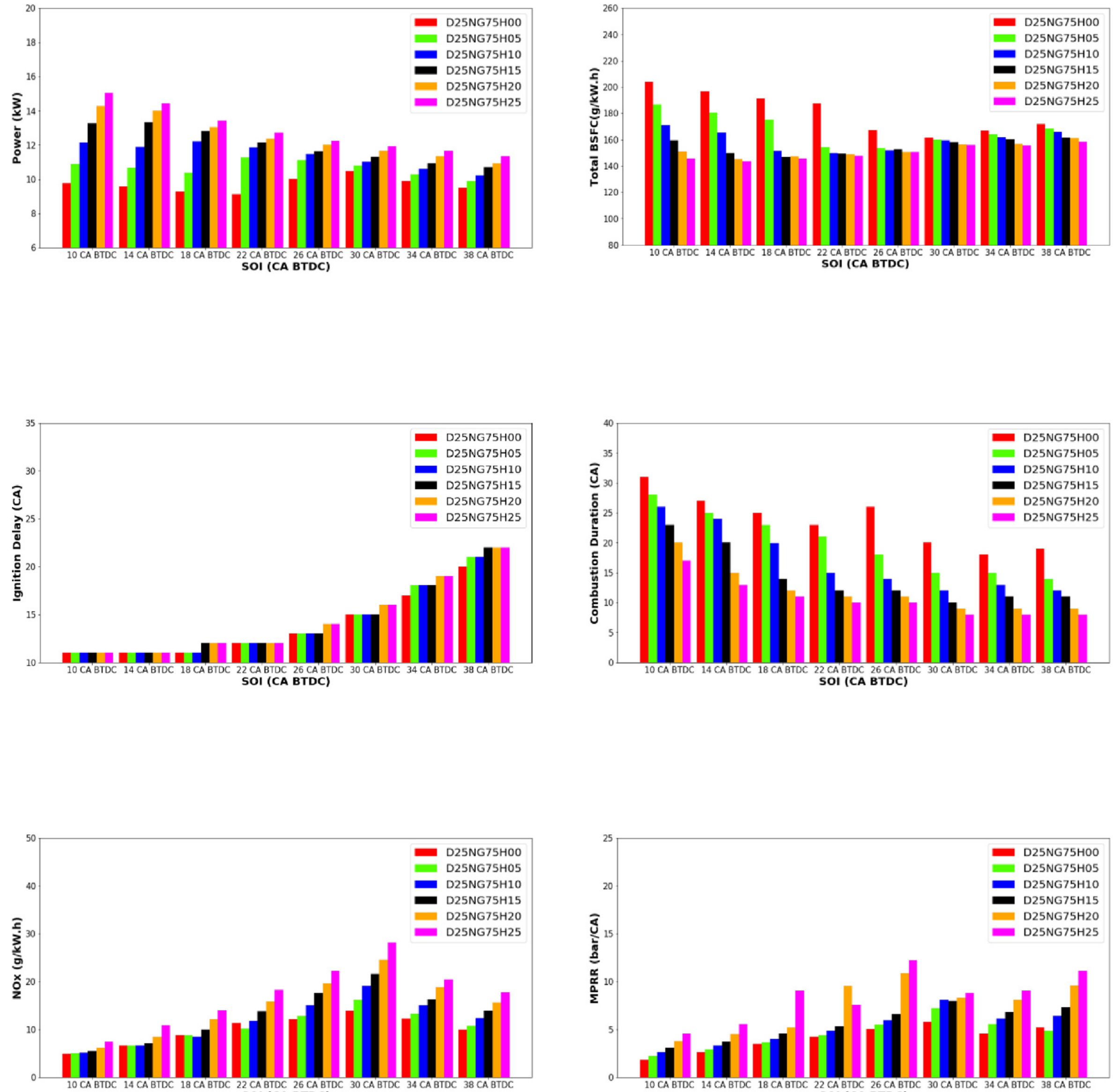

Fig. 7 – Mode 2 Results - The Effect of Gas Mix (Hydrogen and Natural gas) for Fixed SOI.

Table 8 – Comparison of optimum case for Mode 2 and test case.

	Case 1	Case 57
Power (kW)	9.76	13.28
Torque (N.m)	102.42	139.34
BSFC (g/kW.h)	203.85	156
Ignition Delay (CA)	11°	11°
Thermal Efficiency (%)	30.84	36.48
Combustion Duration (CA)	31°	23°
MPRR (bar/CA)	1.85	3
Max. Pressure (bar)	50.55	60.35
CA for Max. Pres. (ATDC)	9° CA	12° CA
Max. Mean Temperature (K)	1477	1809
NO_x (g/kW.h)	4.9	5.44
SOOT (g/kW.h)	0.0012	0.000196
CO (g/kW.h)	10.25	2.019
HC (g/kW.h)	36.73	8.76

Table 9 – Comparison of optimum cases for Mode 1 and Mode 2.

	Mode 1	Mode 2
	Case 18	Case 57
Power (kW)	11.58	13.28
Torque (N.m)	121.47	139.34
BSFC (g/kW.h)	136.27	156
Ignition Delay (CA)	11°	11°
Thermal Efficiency (%)	38.71	36.48
Combustion Duration (CA)	19°	23°
MPRR (bar/CA)	3.6	3
Max. Pressure (bar)	67.45	60.35
CA for Max. Pres. (ATDC)	8° CA	12° CA
Max. Mean Temperature (K)	1779	1809
NO_x (g/kW.h)	7.46	5.44
SOOT (g/kW.h)	0.000172	0.000196
CO (g/kW.h)	1.42	2.019
HC (g/kW.h)	4.04	8.76

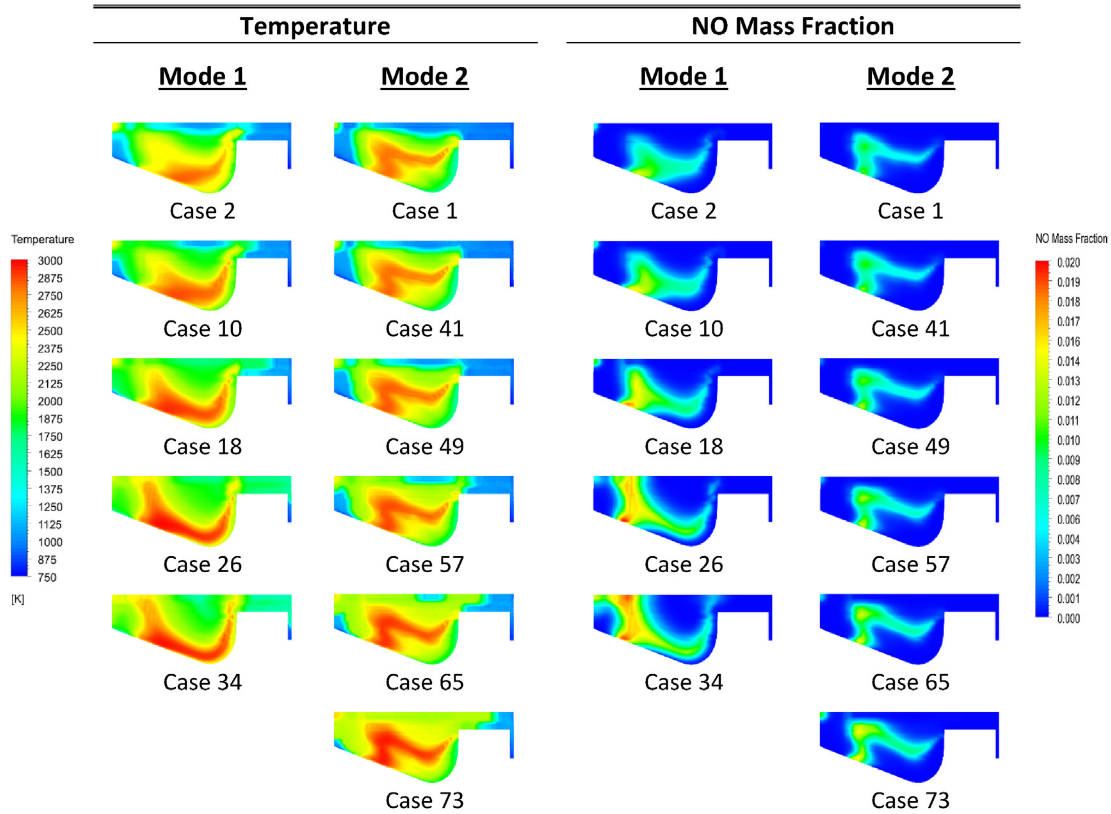

Fig. 8 – The Effect of Hydrogen for fixed SOI - Mode 1 (for 14° CA BTDC of SOI) and Mode 2 (for 10° CA BTDC of SOI) Temperature and NO_x Contours for 10° CA ATDC of piston position.

advance 30° CA BTDC. Similarly, while the total specific fuel consumption decreased with the increase of the hydrogen ratio in low injection advances, the dependence on the hydrogen ratio disappeared with the increase in the injection advance (Figs. 6 and 7). When examined in terms of emissions, just like the results of Mode 1, all other emissions except NO_x emissions decreased with the increase of the injection advance. The increase of the hydrogen rate at fixed diesel fuel injection advances also caused the same effect as Mode 1. At low injection rates, NO_x emission increased with the increase of hydrogen in the gas fuel, but increase was very small compared to high injection rates. If there is no injection strategy or EGR application, it should be known that increasing the hydrogen ratio leads to negative results after a certain value [56–59]. Excessive hydrogen rate causes knock [23,30,32], noise and vibration problems [32], internal cylinder thermal and mechanical deformations, which have extremely harmful effects, and NO_x emissions increase (Figs. 6 and 7).

In order to reduce NO_x emissions, water vapor injection processes are also carried out by increasing the humidity of the combustion chamber [60–62]. Thus, it is aimed to improve NO_x emissions by lowering the in-cylinder local temperatures. For Mode 2, the optimum value of diesel fuel injection advance in terms of performance and emission values was determined as 10° CA BTDC.

Considering the Mode 2 analyzes, the optimal situation in terms of both performance and emission values is obtained in the case of 15% hydrogen gas enrichment and 10° CA BTDC injection advance (10° CA BTDC D25NG75H15 = Case 57). It provided an improvement of 36% in engine power and 23% in specific fuel consumption. It was observed that there was an increase of approximately 11% in the NO_x emission value. Remarkable results were obtained for other types of emission. A reduction of 76%, 84% and 80% was achieved in HC, SOOT and CO emissions, respectively (Table 8).

Comparison of Mode 1 and Mode 2

The increase in performance values as a result of adding hydrogen as extra energy to the system in the same injection advances (Mode 2) was greater compared to the cases of sharing the energy between gas fuels (Mode 1). The ratio of natural gas in the gas mixture at Mode 2 operating points was higher compared to Mode 1. For example, the case with 14° CA BTDC injection advance and 10% hydrogen ratio in Mode 1 condition (Case 10) has more mass hydrogen content in Mode 2 compared to 14° CA BTDC injection advance and extra 10% hydrogen reinforced case (Case 50). In fact, the hydrogen ratio of Case 50 operating point is 9%.

Similarly, when compared in terms of natural gas ratios in Case 10 and Case 50 gas mixture, it is 65% and 68% respectively. In terms of diesel fuel content, it is 25% and 23%, respectively. Therefore, the proportionally low ratio of hydrogen and diesel makes Mode 2 advantageous in terms of NO_x emissions (Table 9). In addition, its relatively high natural gas ratio increased the combustion duration. Increasing the

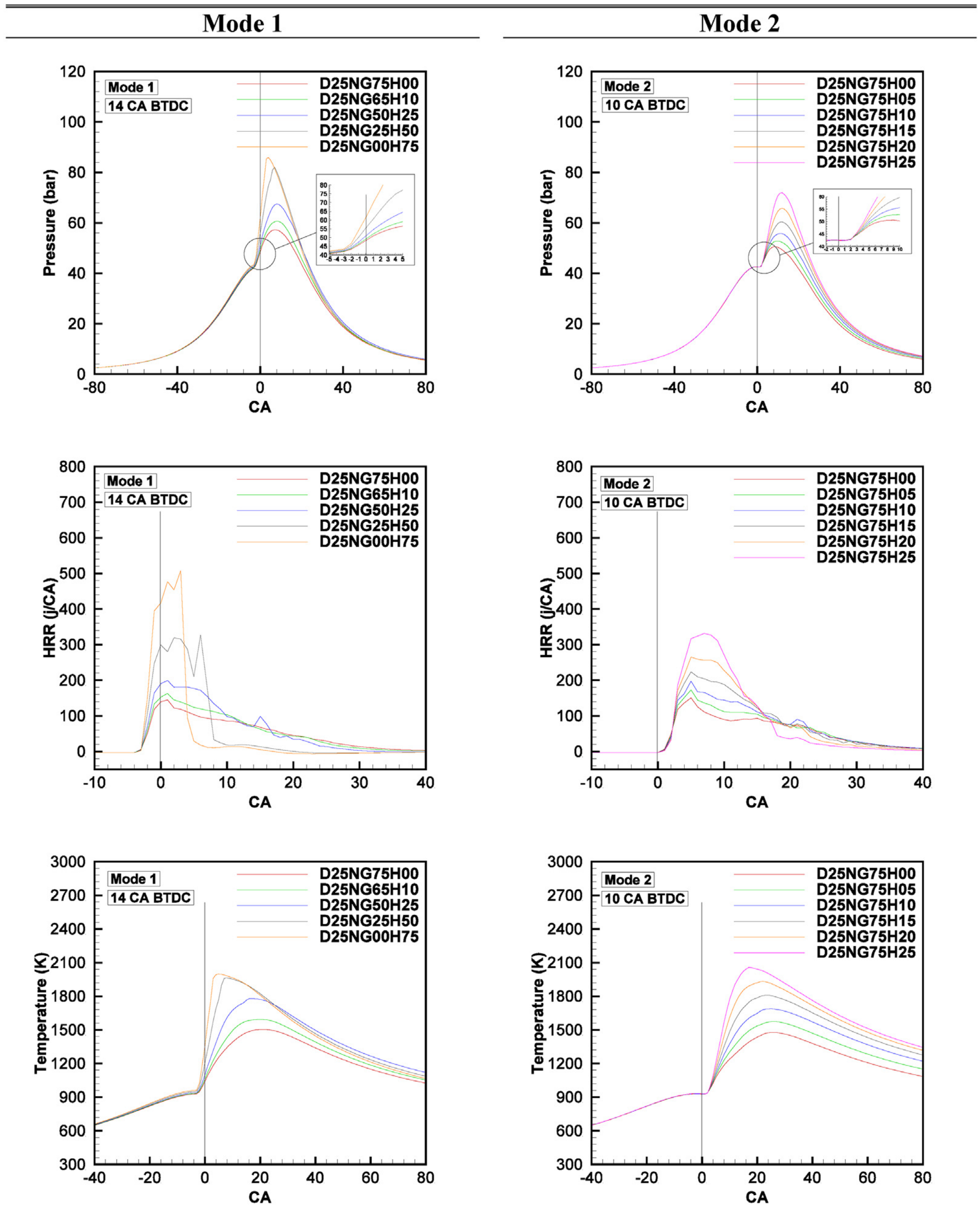

Fig. 9 – The effect on pressure, HRR and temperature of hydrogen for optimum SOI of Mode 1 (14° CA BTDC) and Mode 2 (10° CA BTDC).

ratio of hydrogen in the gas mixture shortens the combustion time [32,63]. Therefore, it provided more effective results in terms of power and torque values (Table 9). However, as extra fuel input was provided, it resulted in higher specific fuel consumption values than Mode 1 conditions [33] (Table 9).

The increase in the amount of hydrogen in the gaseous fuel was extremely effective on the in-cylinder pressure values. Maximum pressure values were seen for 10° CA ATDC piston position in all cases at optimum injection advance value. In particular, the MPRR value increased with the increase in the amount of hydrogen. As the pressure increased, temperature values and therefore NO_x emission values increased. Temperature and NO_x distributions are shown at the crank angle (10° CA ATDC) where the maximum pressure occurs (Fig. 8).

The pressure curve appeared to be sharper under Mode 1 conditions, but more flattened under Mode 2 conditions (Fig. 9). The ignition delay was lower in the Mode 1 state due to the higher hydrogen content in the gas mixture (Table 9). It caused the desired maximum pressure value to occur earlier [30] (Fig. 9). Although Mode 2 appeared to be more advantageous compared to Mode 1 in some cases, Mode 1 offered promising results compared to cases where hydrogen is not included (experimental measurements). For example, the

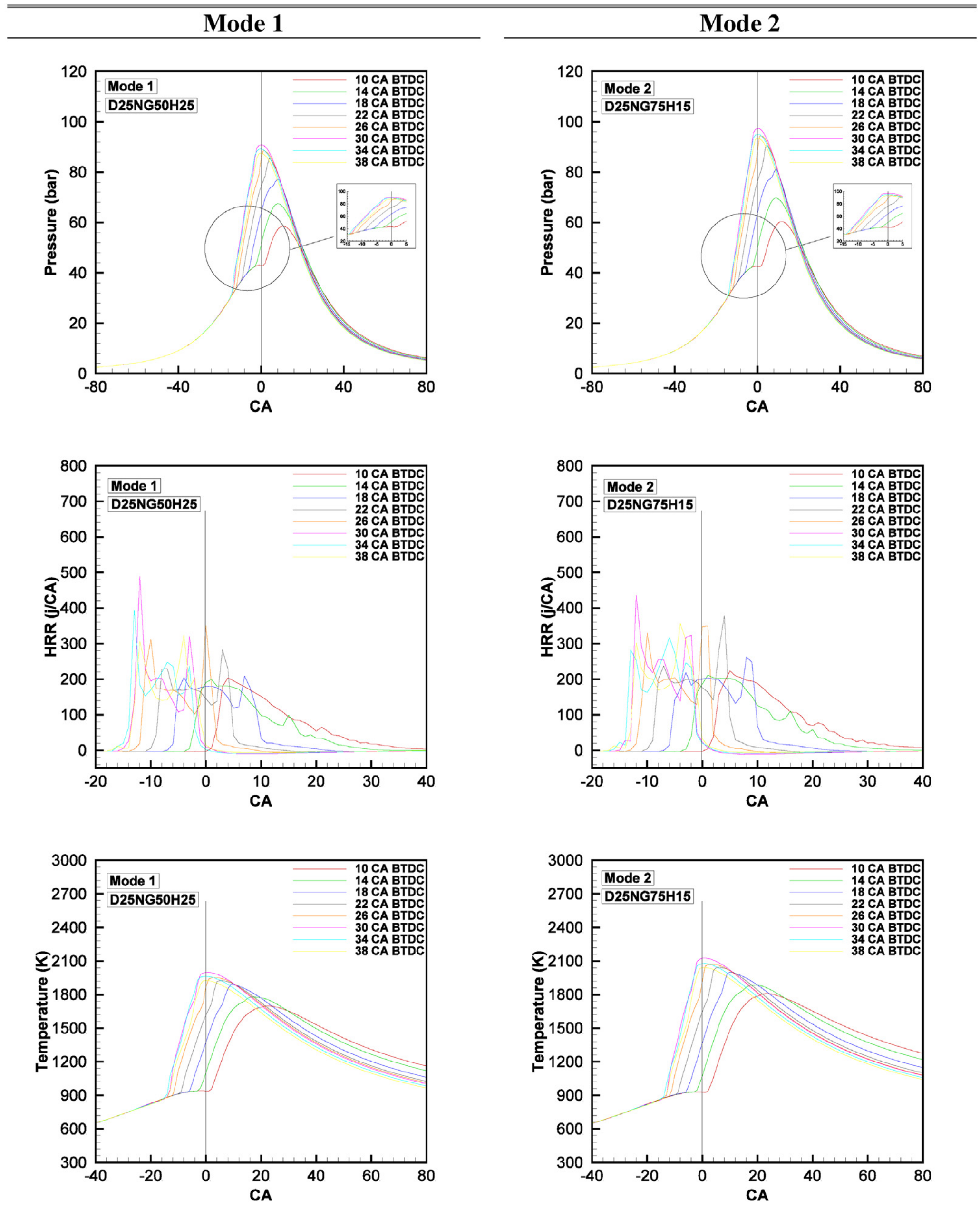

Fig. 10 – The effect on pressure, HRR and temperature of SOI for optimum hydrogen content of Mode 1 (D25NG50H25) and Mode 2 (D25NG75H15).

case with 25% hydrogen ratio with 14° CA BTDC injection advance (Case 18) resulted in higher performance values compared to test results (Case 2) [35–37].

It can be seen in Table 7 that Mode 1 was more advantageous than the test results. Mode 1 also made improvements in terms of emissions. Incomplete combustion products at partial loads are a big problem especially in diesel engines. In addition, natural gas is a hydro carbon fuel in natural gas-diesel dual fuel engines, although incomplete combustion products have been reduced compared to diesel operating conditions. Therefore, PM, CO and HC emissions are seen among the exhaust gas emissions. Hydrogen produces only water vapor (H_2O) as a combustion product. By including hydrogen in the gas mixture, a reduction in incomplete combustion products was achieved. Similar results were obtained by comparing the Mode 2 condition with the experimental measurements. For example, the case (Case 57) with 15% extra hydrogen ratio (about 12% of all energy content) with an injection advance of 10° CA BTDC resulted in higher performance values compared to the test results (Case 1) [35–37]. It can be seen in Table 8 that Mode 2 was more advantageous than the test results. It also made improvements in terms of emissions (Table 8). Comparing the optimum cases of Mode 1 and Mode 2 conditions, more optimum results were obtained when a slightly lower hydrogen ratio was provided in the gas mixture versus a higher natural gas ratio. Thanks to

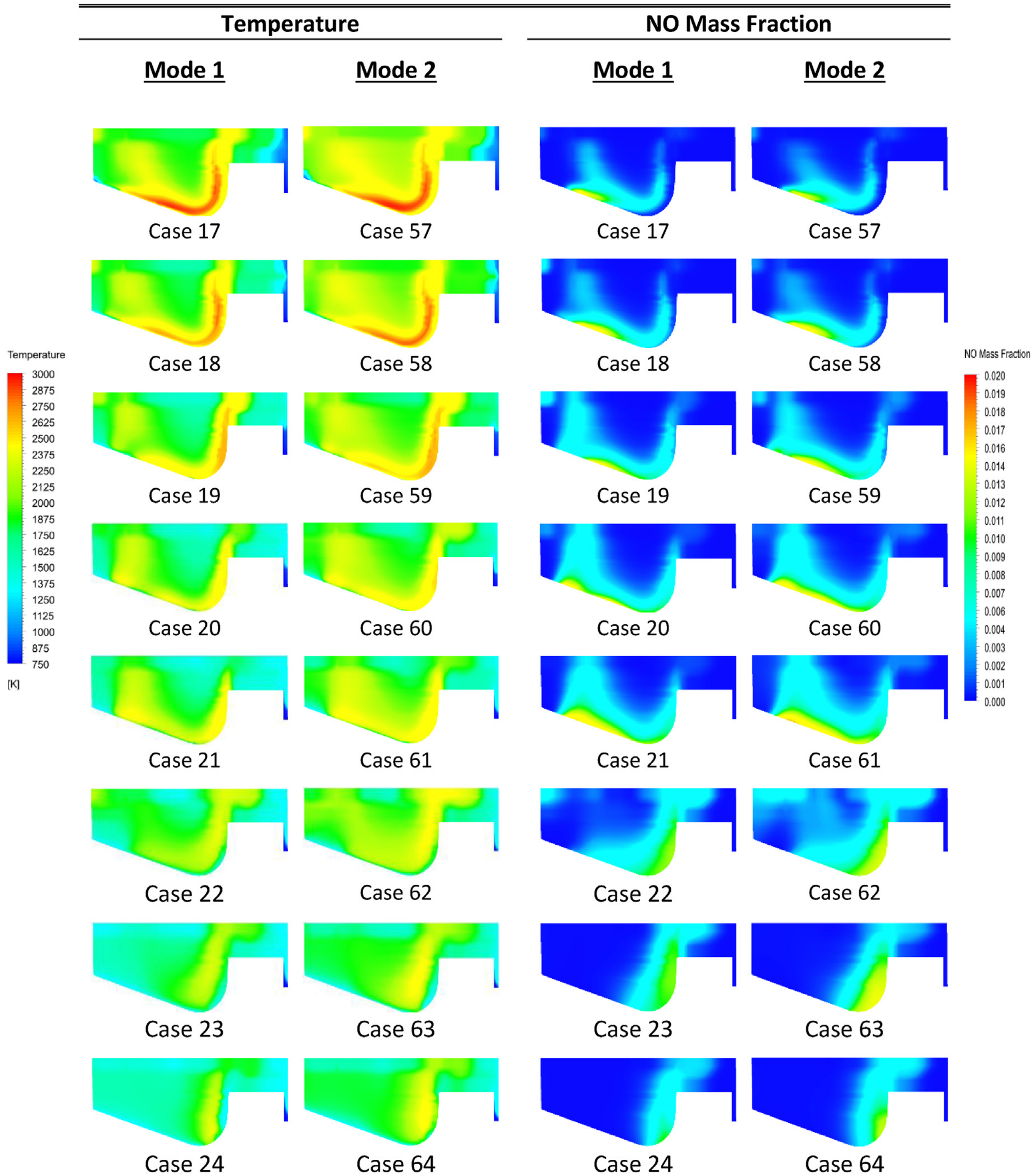

Fig. 11 – The Effect of SOI for fixed Hydrogen Content - Mode 1 (for D25NG50H25) and Mode 2 (for D25NG75H15) Temperature and NO_x Contours for 20° CA ATDC of piston position.

the slow burning mechanisms, the higher the natural gas rate in the mixture, the lower the temperature. Accordingly, it was seen that NO_x emissions also decreased. For Mode 1, the situation with 50% natural gas and 25% hydrogen was the optimal gas mixture, while for Mode 2, 75% natural gas and 15% hydrogen mixture was the optimal. Increasing the injection advance of diesel fuel too much resulted in knocking tendencies and shortening of combustion time. Therefore, gas fuel energy balance should be arranged between hydrogen and natural gas for optimal injection advances [23,32,33]. The in-cylinder temperature and NO_x distributions of increasing the hydrogen ratio in the gas mixture at optimum injection advances for Mode 1 and Mode 2 were shown in (Fig. 8). The increase in the amount of hydrogen increased the in-cylinder temperatures, thus causing higher NO_x emission values.

For other types of emissions it is advantageous in all respects to increase the hydrogen content (Table 9). What should be considered here is the optimum energy balance of hydrogen and natural gas fuel for NO_x control. As mentioned earlier, injection advance of 14° CA BTDC for Mode 1 and 10° CA BTDC for Mode 2 had more positive results. The effect of increasing the diesel fuel injection advance on the in-

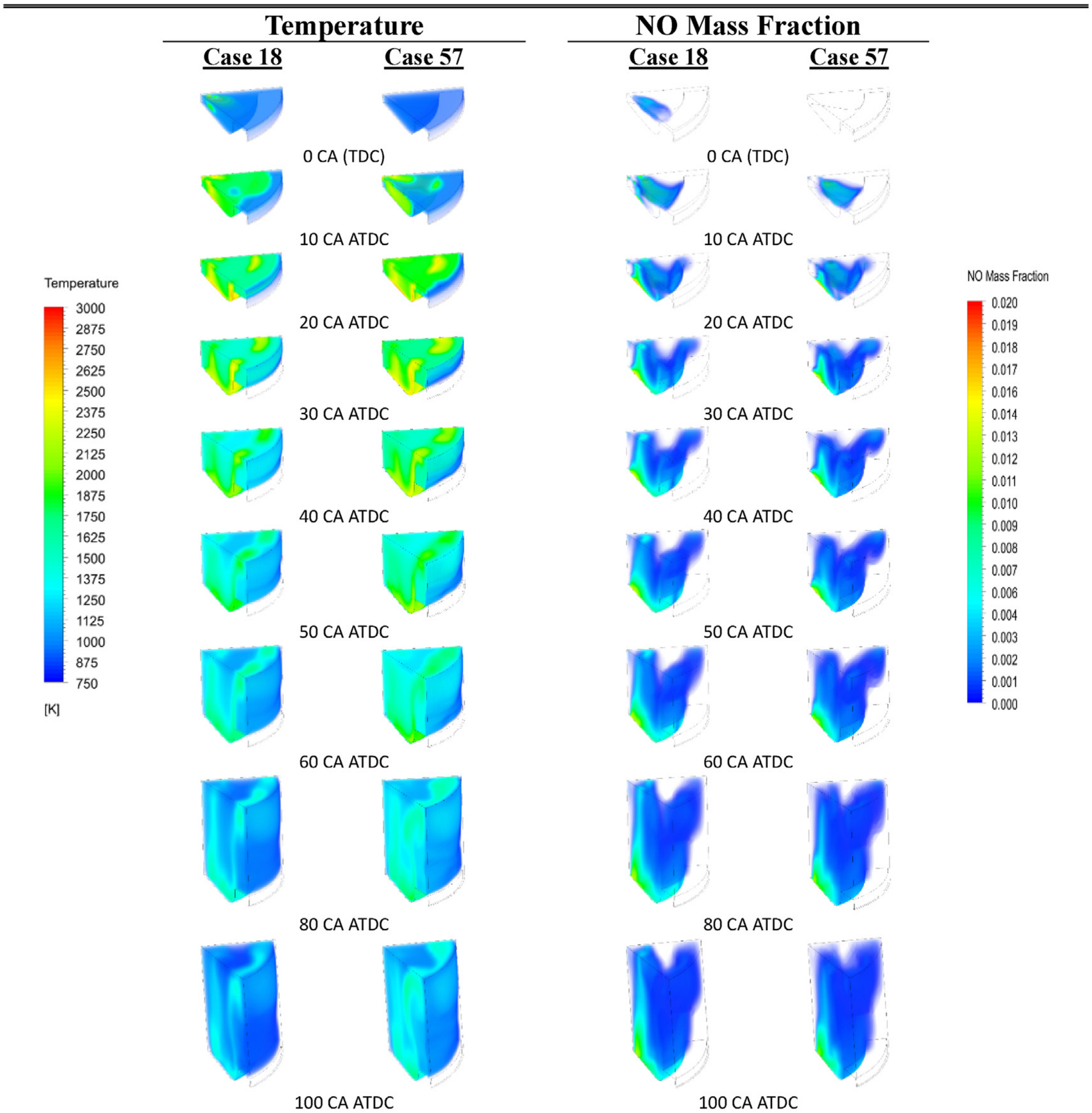

Fig. 12 – Comparison of Optimum Cases of Mode 1 and Mode 2 for Different Combustion Chamber Positions in terms of both Temperature and NO_x.

cylinder temperature and NO_x emissions distribution for both modes at the optimum gas mixing ratios is shown in Fig. 11. The crank angle at which the maximum mean temperature occurs was 20° CA ATDC of the piston position for all cases. Therefore, the effect of the variation of the injection advance on the temperature distribution was demonstrated at the crank angle at which the maximum mean temperature occurred. As can be seen, the increase in the injection advance reduced the local temperature distribution and NO_x emissions. The decrease in temperatures adversely affected the thermal efficiency and thus caused the in-cylinder combustion to deteriorate, thus reducing the performance values. In addition, the comparison of the temperature and NO_x distributions of the optimum operating points of Mode 1 and Mode 2 according to the piston position is shown in Fig. 12. In Mode 1 and Mode 2 cases, in-cylinder local temperatures must be lowered in order to control NO_x emissions. Another method that can be recommended for lowering local temperatures is the inclusion of water vapor injection into the combustion chamber. It has been observed that remarkable results have been obtained in studies where the method has been applied before [64]. Especially at higher hydrogen content at dual fuel diesel engine, it is expected that water vapor injection would give extremely positive results in terms of NO_x emissions.

Conclusions

In the present study, the numerical model of the engine was established with the help of a CFD program (ANSYS Forte), taking the experimental results from literature as a reference [35–37]. The numerical model was verified by comparing the obtained numerical results with experimental results. Within the scope of the study, analyzes were based on two different modes. The first mode (Mode 1) was based on the sharing of energy between gas fuels (natural gas and hydrogen). The second mode (Mode 2) included the effects of the hydrogen enrichment process as extra energy input into the cylinder. The results obtained can be summarized as follows:

- Injection advance had an extremely important effect on ignition delay. Higher injection advance, higher ignition delay especially before 22° CA BTDC.
- Injection advance should not be increased too much in partial load conditions. Optimum injection advance is 14° CA BTDC and 10° CA BTDC for Mode 1 and Mode 2, respectively.
- There was a rapid drop in performance values before 22° CA BTDC injection advance for Mode 1. It was observed that the performance values were close to each other at 10°, 14°, 18° and 22° CA BTDC values of the injection advance.
- For Mode 2, performance values decreased in all injection advances before 10° CA BTDC.
- Increasing the hydrogen ratio in the gas mixture, especially in the high injection advances (before 22° CA BTDC), had negative effect on the engine performance. The local temperatures in-cylinder were increased.
- In order to balance the cylinder temperature, the natural gas fuel fraction in the gas mixture should not be too low (minimum 50%).
- The increase in the natural gas fraction in the gas mixture had also led to an increase in unburned HC, SOOT and CO emissions. HC, SOOT and CO emissions should be improved by enriching natural gas with hydrogen.
- As the hydrogen fraction in the gas mixture was increased, there were improvements in performance values and all emissions except NO_x emissions. However, the situation was reversed when the hydrogen in the gas mixture exceeded 50%.
- The optimum fraction of natural gas and hydrogen for Mode 1 is 50% and 25% (D25NG50H25), respectively, while for Mode 2 it is 75% and 15% (D25NG75H15), respectively.
- After 50% hydrogen energy fraction, as MPRR reached 10 bar/CA, knock tendency started to appear. The hydrogen fraction in the gas mixture should be limited in order to avoid knock tendency.
- With the increase in the hydrogen fraction, it was ensured that the thermal efficiency increased about 21% and 18% for the optimum cases of Mode 1 (Case 18) and Mode 2 (Case 57), respectively.
- There was a 21% and 30% improvement in power and total BSFC values for Mode 1 (Case 18), respectively, and 36% and 23% for Mode 2 (Case 57).
- NO_x increased by 12% for Mode 1 (Case 18), 11% for Mode 2 (Case 57) compared to test cases.
- By including hydrogen in the gas mixture, the emissions were reduced to even lower values. In the study, 89%, 86% and 78% improvement was achieved in HC, CO and SOOT emissions in Mode 1 condition, respectively, and 76%, 80% and 84% for Mode 2 condition, respectively.

As a result, comparing two different modes (Mode 1 and Mode 2), Mode 1 has lower fuel consumption, CO, CO_2 and UHC emissions, but higher NOx emissions and knock tendency. Mode 2 showed more effective results in terms of performance (higher power, lower MPRR and maximum cylindir pressure) and NO_x emissions. However, both modes showed improved results compared to diesel-natural gas dual fuel combustion. Higher mass ratio of hydrogen led to higher temperatures and NO_x emissions. Different pilot diesel fuel injection strategies (pilot injection, main injection and post injection) or EGR (Exhaust Gas Recirculation) were recommended to prevent the increase in NO_x emissions. The effects of water vapor injection into the combustion chamber can also be considered. Thus, both performance values are kept at high values and improvements can be achieved in all emission types, including NO_x emissions.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Not: Bu makalenmin başlıca eserim olması nedeniyle Doktora öğrencisi Ferha Ekin ' in danışmanı olduğumu gösteren belge aşağıda yer almaktadır.

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Adı geçenin isteği üzerine bilgi edinilmesini rica ederim.

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