CHAPTER 5

RESULTS AND DISCUSSION

5.1 CHOOSING BEST FLOW RATE OF OEH GAS

5.1.1 Comparison of performance, emission, and combustion characteristics of various flow rates of OEH gas

Initially the engine was tested with petroleum diesel (standard diesel) at standard engine specifications i.e., injection timing of diesel fuel as 23º BTDC, injection pressure as 200 bar, compression ratio as 17.5:1, and speed as 1800 rpm. This served as a base line operation to compare the results of other experiments.

In the first of phase of experiment, the engine was tested for the best flow rate of OEH gas by considering the facts of higher thermal efficiency and reduced engine-out emissions. For this, the OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm produced by electrochemical reaction of water was aspirated into the cylinder along with the intake air at standard engine specification.

The significant results of the present experimental investigation are presented in the following sections.

5.1.1.1 Brake thermal efficiency (BTE)

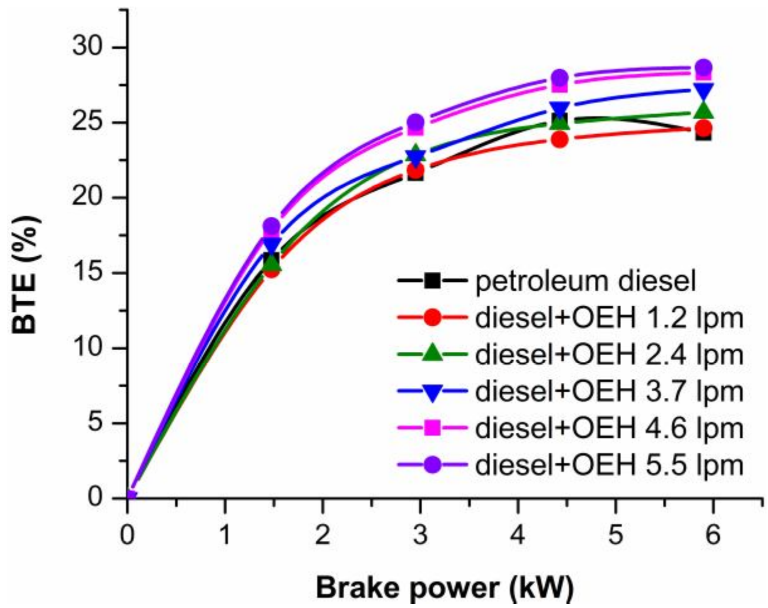

Figure 5.1 Comparison of BTE with BP for petroleum diesel and diesel with OEH gas of various flow rates

Figure 5.1 shows the comparison of brake thermal efficiency for petroleum diesel (base line operation) and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specifications.

The brake thermal of an engine is defined as the ratio of the output brake power to that of the chemical energy input in the form of fuel supply. The equation 5.1 given below was used to calculate the brake thermal efficiency.

$$Brake\ thermal\ efficiency = \frac{BP}{m_f \times LCV} \qquad (5.1)$$

On analyzing the graph, it could be concluded that the brake thermal efficiency increased when OEH gas was used as a combustion stimulant, compared to base line operation. At 100% rated load, the brake thermal efficiency for base line operation was 24.32%, whereas it increased by 1.32%, 5.59%, 11.92%, 16.45%, and 17.88% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm was added in the diesel combustion process. This increase in brake thermal efficiency was due to higher-energy content of the hydrogen present in the gas mixture, its high flame velocity and also due to the presence of atomic hydrogen and oxygen (Santilli 2006) as they were in the higher-energy state than their dual molecule counterparts. Because of this, when the ignition was initiated by petroleum diesel, they immediately started to fracture the heavier hydrocarbon molecule of diesel and initiated the chain reactions, which ultimately resulted in more efficient combustion and higher brake thermal efficiency than petroleum diesel combustion.

5.1.1.2 Brake specific energy consumption (BSEC)

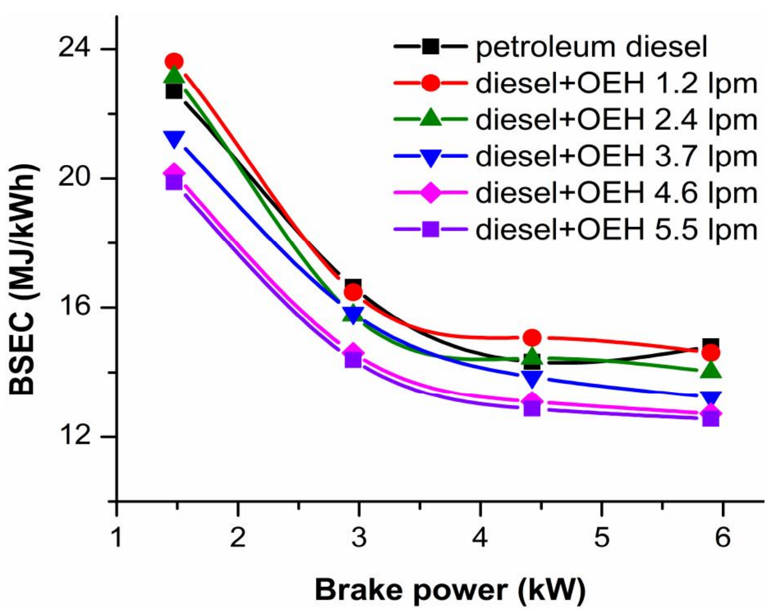

Figure 5.2 Comparison of BSEC with BP for petroleum diesel and diesel with OEH gas of various flow rates

Brake specific energy consumption is defined as the amount of energy consumed from fuel per hour for producing unit brake power. It is a strong indication of the efficiency with which the engine develops power from fuel. The following equation 5.2 was used to calculate BSEC.

$$BSEC = \frac{m_f \times LCV}{BP} \tag{5.2}$$

Figure 5.2 shows the comparison of BSEC for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. On studying the graph, it is concluded that the BSEC decreased when OEH gas was used as a combustion catalyst in the diesel combustion process, compared to base line operation. At 100% rated load of the test engine, the BSEC for base line operation was 14.8 MJ/kWh, whereas it got decreased by 1.29%, 5.29%, 10.65%, 14.12%, and 15.17% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm was added in the diesel combustion process. This decrease in BSEC was due to high catalytic nature of OEH gas (Dulger & Ozcelik 2000). This resulted in uniformity in fuel-air mixture formation and better combustion.

5.1.1.3 Carbon monoxide emission (CO)

Figure 5.3 shows the comparison of CO emission for petroleum diesel and diesel with OEH of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. The experimental results showed that for all the flow rates of OEH gas addition, CO emission was lower at part load, but increased with increase in load. The CO emission for base line operation at maximum load was 0.13% vol., whereas it got decreased by 3.84%, 6.92%, 15.38%, 18.46%, and 11.53% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm was added in the diesel combustion process.

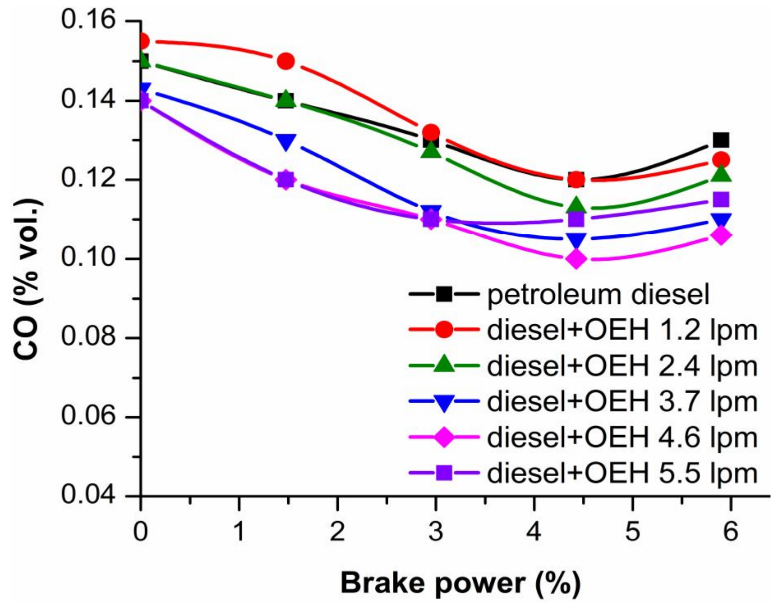

Figure 5.3 Comparison of CO with BP for petroleum diesel and diesel with OEH gas of various flow rates

This might be due to the fact that OEH gas contained oxygen in its mixture of composition. This favored comparatively better combustion for OEH gas added diesel combustion.

5.1.1.4 Carbon dioxide emission (CO_2)

Figure 5.4 shows the comparison of CO_2 emission for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. The experimental results showed that for all the flow rates of OEH gas, CO_2 emission was lower at low load conditions, but increased with increase in load. If the degree of combustion of fuel and air mixture was high, the CO_2 emission would be more. Probably, the same thing happened during the combustion influenced by the OEH gas.

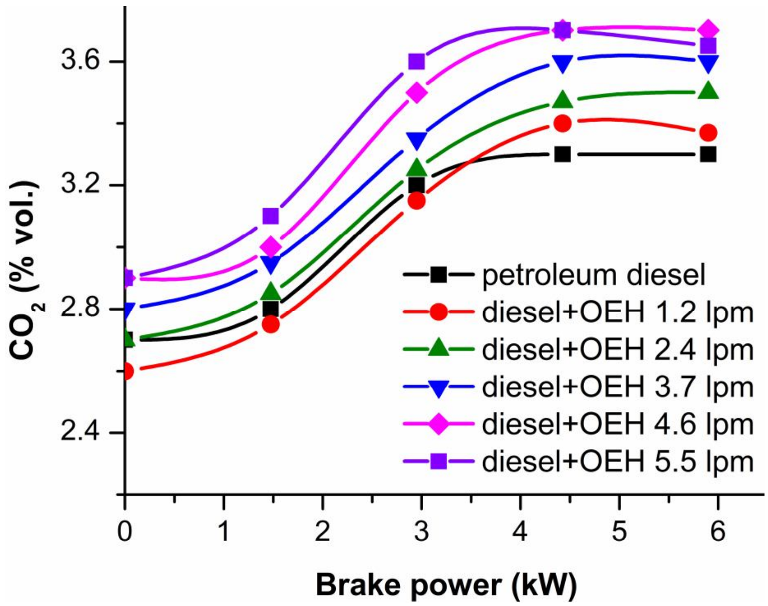

Figure 5.4 Comparison of CO_2 with BP for petroleum diesel and diesel with OEH gas of various flow rates

The CO_2 emission for diesel at maximum load was 3.3% vol., whereas it got increased by 2.12%, 6.06%, 9.09%, 12.12%, and 10.61% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm was supplemented in the diesel combustion process.

5.1.1.5 Unburned hydrocarbon emission (UBHC)

Figure 5.5 presents the comparison of UBHC emission for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. The experimental results showed that for all the flow rates of OEH gas, UBHC emission was lower at part load, but got increased with increase in load.

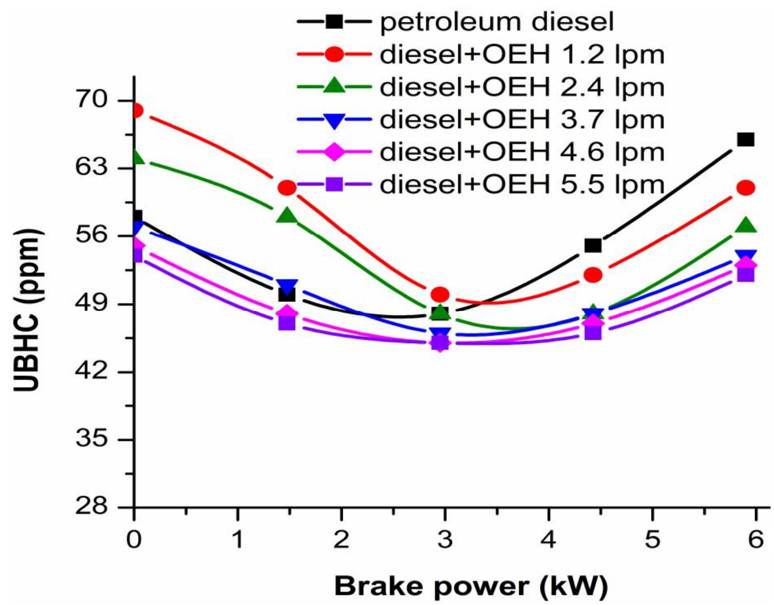

Figure 5.5 Comparison of UBHC with BP for petroleum diesel and diesel with OEH gas of various flow rates

The UBHC emission for diesel at maximum load was 66 ppm, whereas it got decreased by 7.57%, 13.63%, 18.18%, 19.69%, and 21.21% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm was inducted in the diesel combustion process. This might be due small quenching distance of hydrogen presented in the gas mixture and high oxygen index of the gas mixture. These facts favored a comparatively better combustion for OEH gas assisted diesel combustion rather than diesel combustion.

5.1.1.6 Oxides of nitrogen emission (NO_X)

Figure 5.6 depicts the comparison of NO_X emission for petroleum diesel and diesel with OEH of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification.

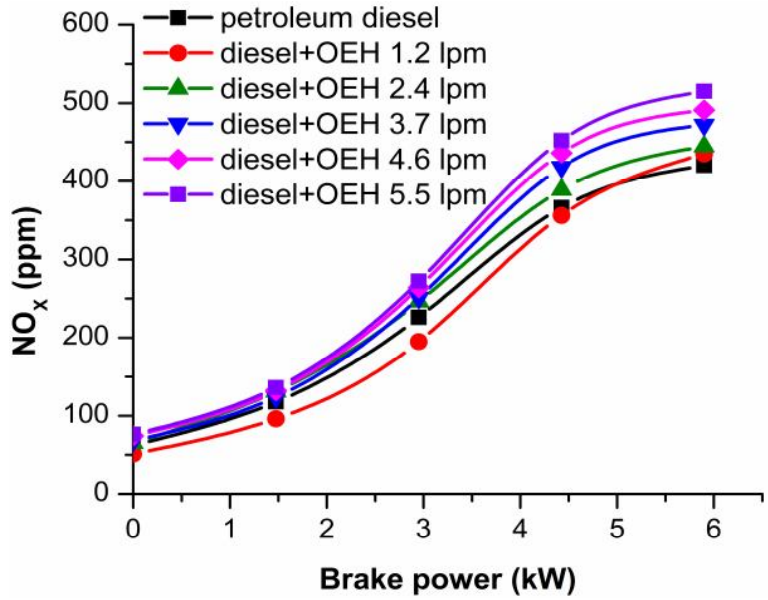

Figure 5.6 Comparison of NO_X with BP for petroleum diesel and diesel with OEH gas of various flow rates

The experimental results showed that for all the flow rates of OEH gas, NO_X emission was lower at part load, but increased with an increase in load. The presence of oxygen in the gas mixture along with spontaneous combustion of hydrogen due to its high flame velocity had led to complete combustion of fuel-air mixture better than petroleum diesel combustion. As a result, the adiabatic flame temperature inside the cylinder was more in the case of OEH gas assisted combustion than petroleum diesel combustion. This catalyzed the reactions for oxidation of nitrogen and hence NO_X emission was more for OEH gas assisted diesel combustion than petroleum diesel combustion. The NO_X emission for diesel at maximum load was 420 ppm, it got increased by 3.57%, 5.83%, 12.38%, 16.9%, and 22.62% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was added in the diesel combustion process.

5.1.1.7 Smoke emission

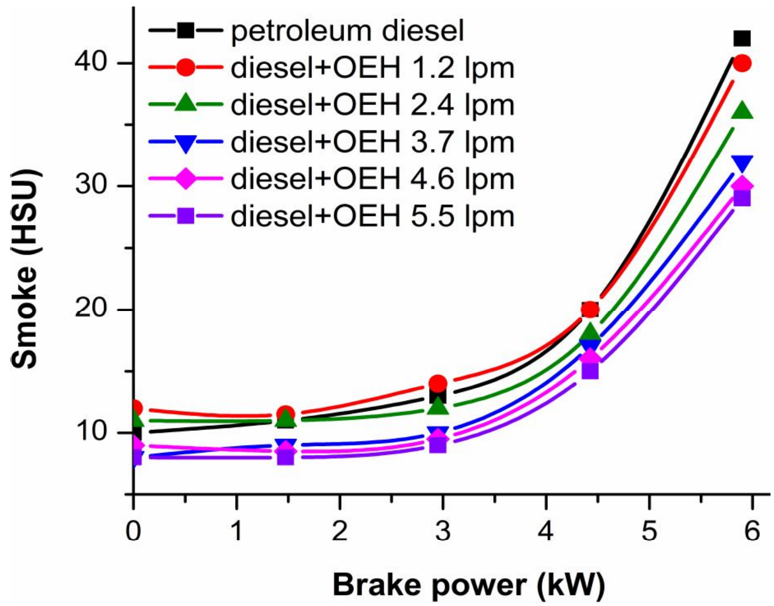

Figure 5.7 Comparison of smoke emission with BP for petroleum diesel and diesel with OEH gas of various flow rates

Figure 5.7 shows the comparison of smoke emission for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. From graph it is distinguished that the smoke emission of OEH gas aided diesel combustion was higher compared to petroleum diesel combustion. The smoke emission for diesel at rated load of the engine was 42 ppm, whereas it got decreased by 4.76%, 14.28%, 23.81%, 28.57%, and 30.95% for OEH gas addition of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm respectively at rated load of the test engine. The final concentration of the particulate matters in the exhaust was the outcome of the competition between the rate of formation and the rate of oxidation of particulates which were particularly sooty in nature (Birtas et al 2011). Both these rates were related exponentially with temperature. Thus the impact of the addition of OEH gas could be related to the influence on the temperature during the time of reactions. This resulted in conversion of long chain carbon molecules towards shorter chain volatile organics (McWilliam 2008) and resulted in the reduction of smoke emission in the exhaust of the engine.

Among all the flow rates, 5.5 lpm of OEH gas resulted in lowest smoke emission at rated load of the engine. This might be due to availability of high concentration of OH radicals generated by chemical reactions between hydrogen and oxygen (Das 1996a), which led to increase in oxidation reactions rate when compared to other flow rates of OEH gas.

5.1.1.8 Excess oxygen emission

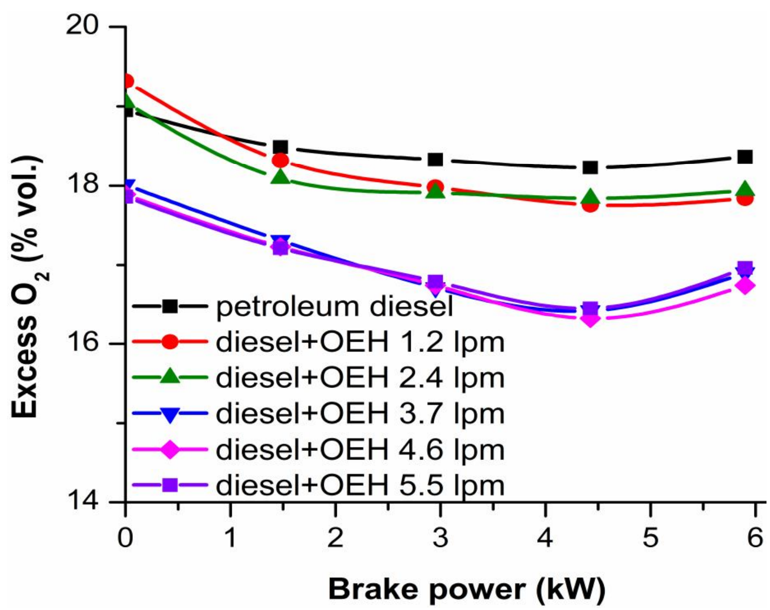

Figure 5.8 Comparison of excess oxygen emission with BP for petroleum diesel and diesel with OEH gas of various flow rates

Figure 5.8 shows the comparison of excess oxygen emission for petroleum diesel and diesel with OEH of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. The experimental results showed that for all the flow rates of OEH gas addition, excess oxygen emission was higher at low load conditions, but decreased with increase in load. The excess oxygen emission for base line operation at maximum load was 18.37% vol., where as it got decreased by 2.88%, 2.34%, 8%, 8.87%, and 7.67% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm was added in the diesel combustion process. This might be due to the inherent oxygen content of OEH gas. This assisted comparatively better combustion for OEH gas aided diesel combustion. The low amount of excess oxygen was emitted when the flow rate of OEH gas was 2.4 lpm. This might be due to proper diffusion of air-fuel mixture at this flow rate. When compared to 4.6 lpm of flow rate aided diesel combustion, the 5.5 lpm flow rate aided diesel combustion emitted more excess oxygen. This was because at this flow rate, the temperature developed inside the combustion chamber was more which resulted in the dissociation of CO_2 into CO and excess oxygen. The CO emission and the CO_2 emission at this flow rate which were discussed earlier also confirmed this.

5.1.1.9 Exhaust gas temperature (EGT)

Figure 5.9 displays the comparison of EGT for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. From graph it is well-known that the EGT characteristic of OEH gas assisted diesel combustion was higher compared to petroleum diesel combustion. The EGT for diesel at rated load of the engine was 390°C, whereas it got increased by 0.51%, 2.82%, 5.38%, 6.41%, and 3.84% for OEH gas addition of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm respectively at rated load of the test engine. This might be due to the sharp increase in combustion temperature because of enhanced premixed burning phase of OEH gas assisted diesel combustion.

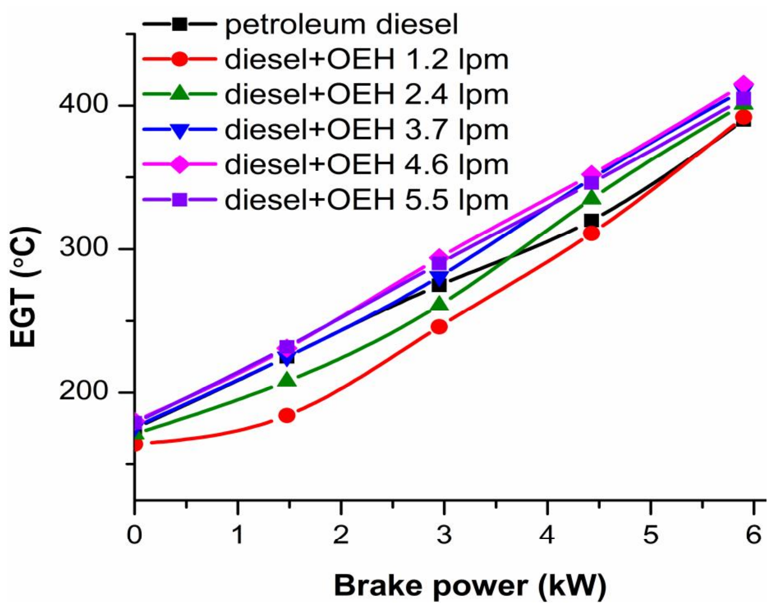

Figure 5.9 Comparison of EGT with BP for petroleum diesel and diesel with OEH gas of various flow rates

The EGT emission gradually increased with increasing flow rate of OEH gas. This might be due to the high flame velocity of hydrogen present in the gas mixture. This led to fast burning of fuel-air mixture. Among all the flow rates, 4.6 lpm of OEH gas has the highest EGT at the rated load of the engine. This might be due to the proper combustion, which led to higher combustion temperatures when compared to other flow rates of OEH gas.

5.1.1.10 Heat release rate (HRR)

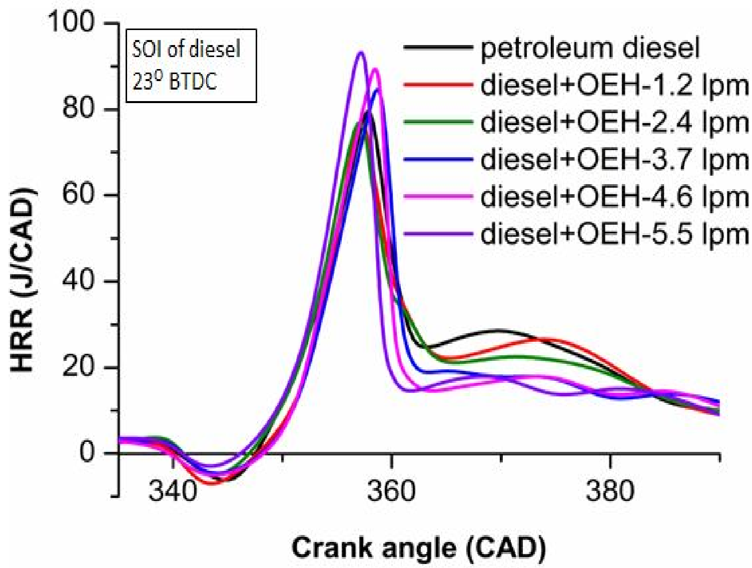

Figure 5.10 Comparison of HRR with CA for petroleum diesel and diesel with OEH gas of various flow rates at rated load

Figure 5.10 depicts the comparison of heat release rate with crank angle for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. The heat release rate is calculated on the basis of first law of thermodynamics. The following equation 5.3 was used for the same.

$$Q_{app} = \frac{\gamma}{\gamma-1} PdV + \frac{1}{\gamma-1} VdP + Q_W \tag{5.3}$$

The experimental results showed that for all the flow rates of OEH gas at rated load of the engine, heat release rate increased. This might be due to combination of diesel diffusion combustion and the premixed combustion of OEH consumed by multiple turbulent flames, which substantially enhanced the combustion process of OEH gas assisted diesel engine. The heat release rate for petroleum diesel combustion at maximum load was 80 J/CA, whereas it got increased by 2.5%, 5%, 10%, 13.75%, and 18.75% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was inducted in the diesel combustion process.

5.1.1.11 In-cylinder pressure

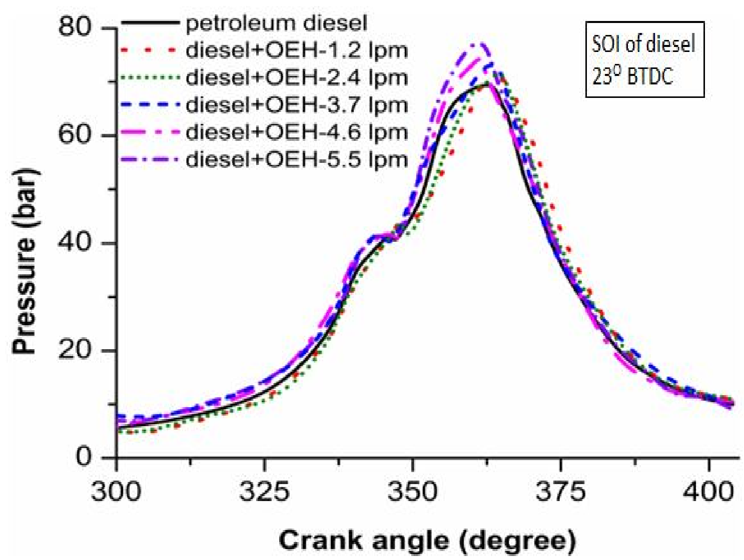

Figure 5.11 Comparison of in-cylinder pressure with CA for petroleum diesel and diesel with OEH gas of various flow rates at rated load

Figure 5.11 represents the comparison of in-cylinder pressure with crank angle for petroleum diesel and diesel with OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm , 4.6 lpm, and 5.5 lpm at standard engine specification. When OEH gas was introduced into the combustion process of petroleum diesel, the ignition delay increased by 1^o to 3^o depending upon the flow rate of OEH gas. The self-ignition temperature of OEH gas is more than pure petroleum diesel; it cannot combust on its own, it needs an assistance to start its combustion. When OEH gas was assisted by the ignition of petroleum diesel, the combustion was instantaneous and created high pressure and high temperature inside the combustion chamber. On analyzing the graph, it is evident that a small fall was followed by an immediate hike in the pressure curve; this was due to the heat observed by the fuel droplets during their vaporization from surrounding heated air present in a combustion chamber.

The peak pressure for neat diesel at the rated load of the engine was 70 bar, whereas it got increased by 1.42%, 2.14%, 4.28%, 7.14%, and 10% for OEH gas addition of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm respectively at rated load of the test engine. This might be due to the enhanced pre-mixed burning phase and proper diffusion. This resulted in higher pressure in the combustion chamber for the OEH gas assisted diesel combustion when compared to petroleum diesel combustion. Among all the flow rates of OEH gas, 5.5 lpm had higher peak pressure at the rated load of the engine. This might be due to the high concentration of gas mixture which induced more catalytic reactions in the combustion process than other flow rates assisted diesel combustion.

5.1.2 Summary of results of phase I

In the first of phase of the experiment, the engine was tested for the best flow rate of OEH gas by considering the facts of higher thermal efficiency and reduced engine-out emissions. For this, the OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm produced by electrochemical reaction of water was aspirated into the cylinder along with the intake air at standard engine specification at different load conditions of the test engine.

The test results of the engine at rated load can be summarized as follows:

- The brake thermal efficiency for base line operation was 24.32%, whereas it got increased by 1.32%, 5.59%, 11.92%, 16.45%, and 17.88% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was inducted in the diesel combustion process.

- The BSEC for base line operation was 14.8 MJ/kWh, whereas it got decreased by 1.29%, 5.29%, 10.65%, 14.12%, and 15.17% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was added in the diesel combustion process.

- The CO emission for base line operation was 0.13% vol., whereas it got decreased by 3.84%, 6.92%, 15.38%, 18.46%, and 11.53% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was inducted in the diesel combustion process.

- The CO_2 emission for diesel was 3.3% vol., whereas it got increased by 2.12%, 6.06%, 9.09%, 12.12%, and 10.61% when OEH gas of 1.2 lpm, 2.4

lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was supplemented in the diesel combustion process.

- The UBHC emission for diesel was 66 ppm, whereas it got decreased by 7.57%, 13.63%, 18.18%, 19.69%, and 21.21% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was inducted in the diesel combustion process.

- The NO_X emission for diesel was 420 ppm, whereas it got increased by 3.57%, 5.83%, 12.38%, 16.9%, and 22.62% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was added in the diesel combustion process.

- The smoke emission for diesel was 42 ppm, whereas it got decreased by 4.76%, 14.28%, 23.81%, 28.57%, and 30.95% for OEH gas addition of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm respectively in the diesel combustion process.

- The excess oxygen emission for base line operation was 18.37% vol., whereas it got decreased by 2.88%, 2.34%, 8%, 8.87%, and 7.67% when OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was added in the diesel combustion process.

- The EGT for diesel was 390°C, whereas it got increased by 0.51%, 2.82%, 5.38%, 6.41%, and 3.84% for OEH gas addition of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm respectively in the diesel combustion process.

- The heat release rate for petroleum diesel combustion load was 80 J/CA, whereas it got increased by 2.5%, 5%, 10%, 13.75%, and 18.75% when

OEH gas of 1.2 lpm, 2.4 lpm, 3.7 lpm, 4.6 lpm, and 5.5 lpm respectively was inducted in the diesel combustion process.

- The peak pressure for neat diesel was 70 bar, whereas it got increased by 1.42%, 2.14%, 4.28%, 7.14%, and 10% for OEH gas addition of 1.2 lpm, 2.4 lpm , 3.7 lpm , 4.6 lpm, and 5.5 lpm respectively in the diesel combustion process.

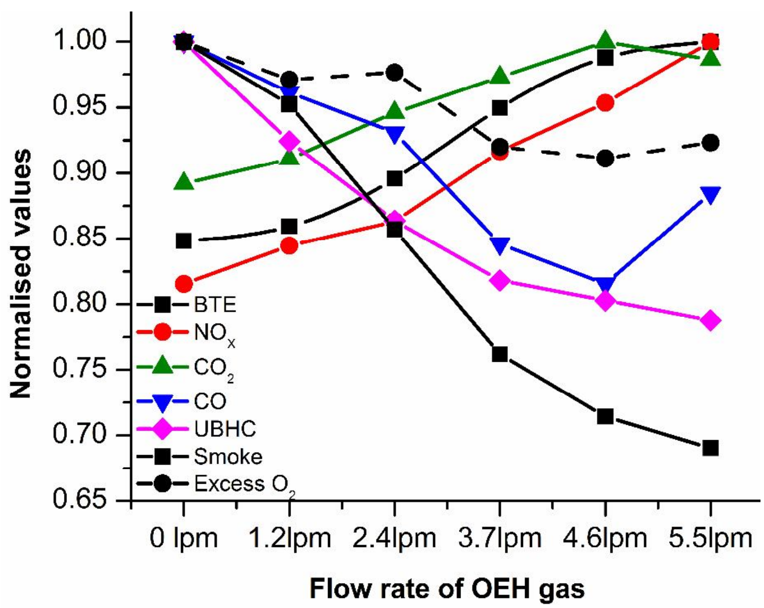

Figure 5.12 Comparison of behaviors of various flow rates of OEH gas

Figure 5.12 compares behaviors of various flow rates of OEH gas. Based on the cumulative results obtained from the Figure, the various flow rates of OEH gas can be ordered as

5.5 lpm > 4.6 lpm > 3.7 lpm > 2.4 lpm > 1.2 lpm by considering increase in brake thermal efficiency and reduction in all engine-out emissions except NO_X emission.

Table 5.1 Nutshell of the results of 4.6 lpm and 5.5 lpm OEH gas

Flow rate	BTE	CO	UBHC	Smoke	NO_X
4.6 lpm	16.45%	18.46%	19.69%	28.57%	16.90%
5.5 lpm	17.88%	11.53%	21.21%	30.95%	22.62%

From the Table 5.1, it is evident that when 4.6 lpm and 5.5 lpm of OEH gas additions in diesel combustion process were compared, the variation in performance parameter like BTE and emission parameters like CO, UBHC, and Smoke were small. But in case of NO_X emission, the 4.6 lpm emitted only 16.9% whereas 5.5 lpm emitted 22.62%. It very clearly shows the path to select for the best flow rate of OEH gas which can be used for further investigation.

5.2 EVALUATING THE PERFORMANCE OF BEST FLOW RATE OF OEH GAS UNDER VARIOUS OPERATING PARAMETERS

In the second of phase of the experiment the engine was tested for its performance, emission, and combustion characteristics when the best flow rate of OEH gas was added in the combustion process of diesel with change in operating parameters of the engine. For this phase of the experiment, six operating parameters of the engine were varied and tested. The six operating parameters varied were:

- Injection time of diesel fuel
- Injection pressure of diesel fuel
- Cooling water flow rate
- Temperature of diesel fuel
- Inlet air temperature
- Combination of injection pressure and injection time of diesel fuel

5.2.1 Varied injection timing

Injection timing plays an important role in reducing the engine-out emissions. A number of studies by several researchers indicate its significance.

In the present experimental work, the best flow rate OEH gas of 4.6 lpm was aspirated into the cylinder along with intake air at varied injection timings of diesel fuel. Three injection times were selected. One was the standard injection time of 23^{o} BTDC recommended by the engine manufacturer, second one was the retarded injection time of 19^{o} BTDC and the third one was the advanced injection time of 27^{o} BTDC. The injection times were varied by modifying the shim thickness at the link point between the pump and the engine (Khatri et al 2010).

5.2.1.1 Brake thermal efficiency (BTE)

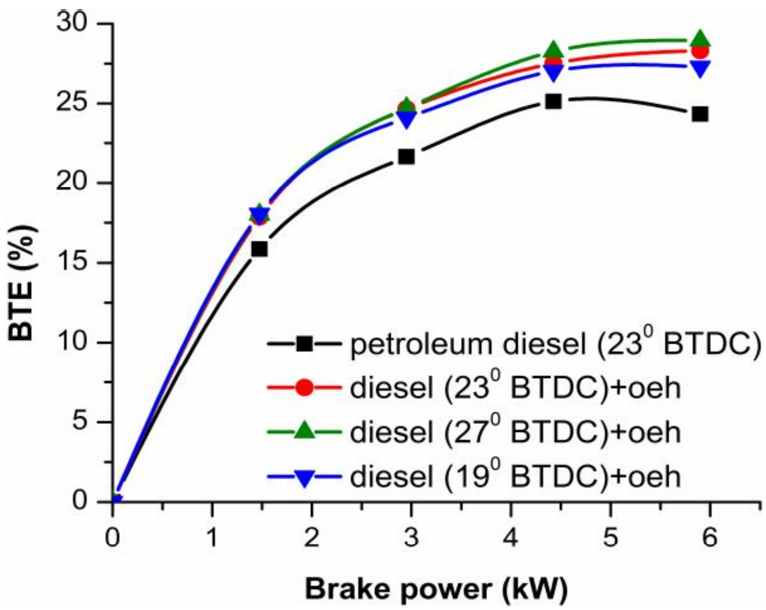

Figure 5.13 Variation of BTE with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Figure 5.13 shows the comparison of brake thermal efficiency when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings of diesel fuel. This 4.6 lpm of OEH gas accounted for an average of 9% to14% of total energy of combustion. The experimental results showed that under the influence of OEH gas at 100% rated load, the brake thermal efficiency increased by 16.45%, 12.21%, and 19.03% for standard injection timing of 23º BTDC, retarded injection timing of 19º BTDC and advanced injection timing of 27º BTDC compared to base line operation.

When the test engine was operated in retarded injection time of 19º BTDC with OEH gas, it resulted in 3.63% decrease in brake thermal efficiency compared to standard injection time operation with OEH gas and 5.73% decrease in brake thermal efficiency compared to advanced injection time operation of 27º BTDC with OEH gas. During retarded injection time operation, part of combustion took place during the expansion stroke. This was also confirmed by the in-cylinder pressure curve at this injection time. At 27º BTDC, the maximum brake thermal efficiency was obtained compared to other injection timings. The brake thermal efficiency increased by 2.22% at advanced injection time of 27º BTDC compared to standard injection time of 23º BTDC. This might be due to chemically correct fuel-air mixture which resulted in better combustion.

5.2.1.2 Brake specific energy consumption (BSEC)

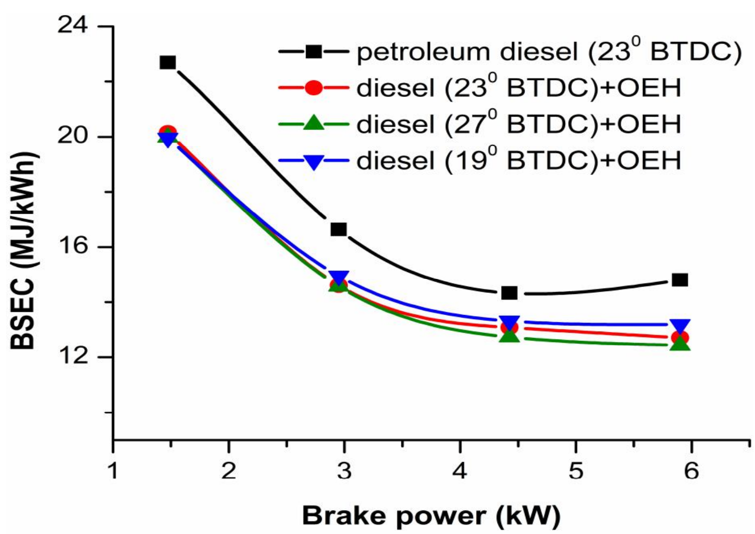

Figure 5.14 Variation of BSEC with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Figure 5.14 represents the comparison of BSEC when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings of the diesel fuel. The experimental results showed that the BSEC increased when the injection time was retarded and decreased when the injection time was advanced. Under the influence of OEH gas at 100% rated load, the BSEC got decreased by 14.12%, 10.88%, and 15.99% for standard injection timing of 23º BTDC, retarded injection timing of 19º BTDC and advanced injection timing of 27º BTDC compared to base line operation.

When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 3.77% increase in BSEC compared to standard injection time operation and 6.08% increase in BSEC compared to advanced injection time operation of 27º BTDC. This might be due to shorter ignition delay period (Mohammadi et al 2007) which resulted in low efficiency combustion. At 27º BTDC, the minimum BSEC was obtained compared to other injection timings. The BSEC got decreased by 2.22% at advanced injection time of 27º BTDC compared to standard injection time of 23º BTDC. This might be due to participation of more homogeneous mixture of fuel and air in the combustion process which resulted in improved combustion.

5.2.1.3 Carbon monoxide emission (CO)

Figure 5.15 depicts the comparison of CO emission for petroleum diesel and diesel with OEH of 4.6 lpm at different injection timings of diesel fuel. When the test engine was operated in retarded injection time of 19º BTDC at the rated load of the test engine, it resulted in 9.09% and 4.34% increase in CO emission compared to 23º BTDC and 27º BTDC. This might be due to under-mixing of fuel and air, some fuel particles in the fuel-rich zones might never react with oxygen.

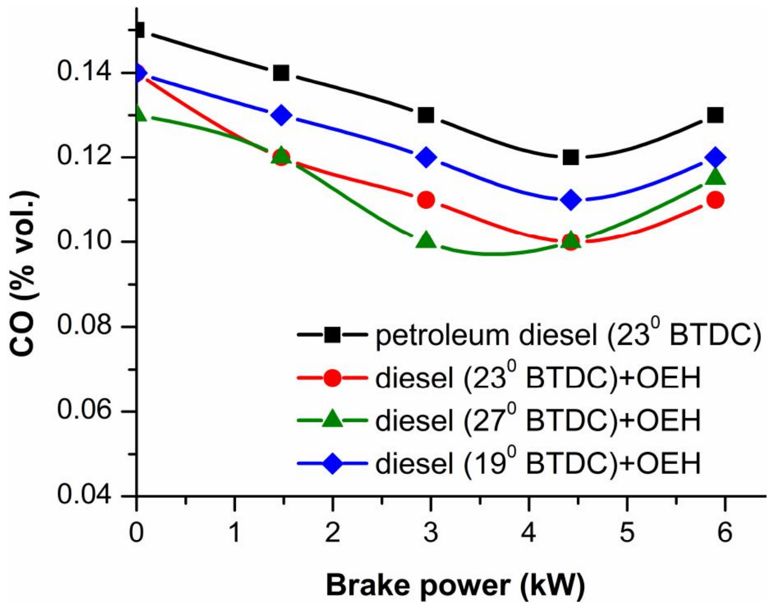

Figure 5.15 Variation of CO with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

The CO emission got decreased by 7.69%, 15.38%, and 11.53% at 19º BTDC, 23º BTDC, and 27º BTDC compared to base line operation. This might be due to high diffusing property of hydrogen and its high flame velocity resulting in intense combustion. At 27º BTDC, CO emission got increased by 4.54% compared to 23º BTDC operation. This might be due to over-mixing of air fuel mixture formation which resulted in ultra lean combustion.

5.2.1.4 Carbon dioxide emission (CO_2)

Figure 5.16 displays the comparison of CO_2 emission when OEH gas of 4.6 lpm was supplemented in the diesel combustion process at different injection timings of diesel fuel. Advancing the injection time of the diesel fuel increased the CO_2 emission whereas retarding the injection time reduced the CO_2 emission.

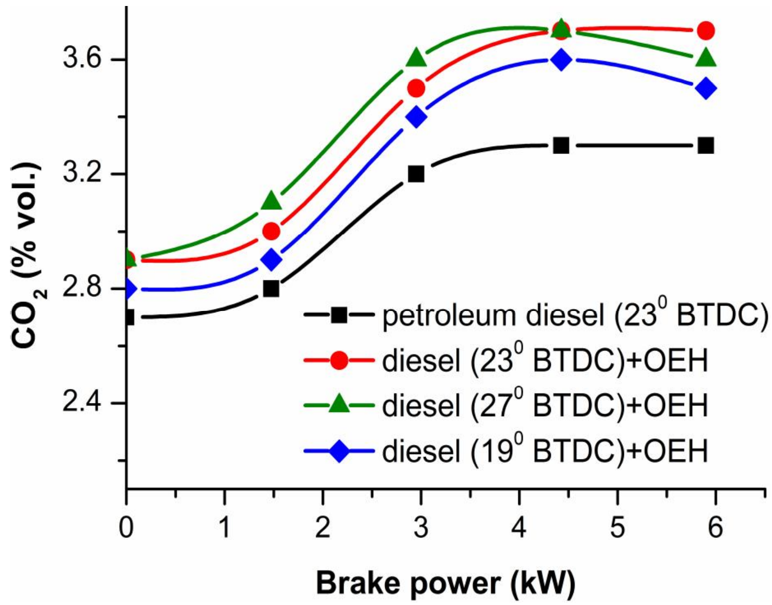

Figure 5.16 Variation of CO_2 with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Under the influence of OEH gas at full rated load of the engine, CO_2 emission got increased by 12.12% and 9.09% for standard injection timing of 23º BTDC and advanced injection timing of 27º BTDC compared to base line operation. This might be due to spontaneous combustion of OEH gas when its ignition was initiated by pilot diesel fuel. When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 5.4% and 2.77% decrease in CO_2 emission compared to 23º BTDC and 27º BTDC. This might be due to improper conversion of CO to CO_2 due to decrease in combustion temperatures and resulted in less intense combustion. At 23º BTDC, the maximum CO_2 emission was emitted from the engine compared to other injection timings. The CO_2 emission got decreased by 2.7% at 27º BTDC compared to 23º BTDC. This might be due to dissociation of CO_2 into CO and excess oxygen. The increase in CO emission at 100% load also justified this.

5.2.1.5 Unburned hydrocarbon emission (UBHC)

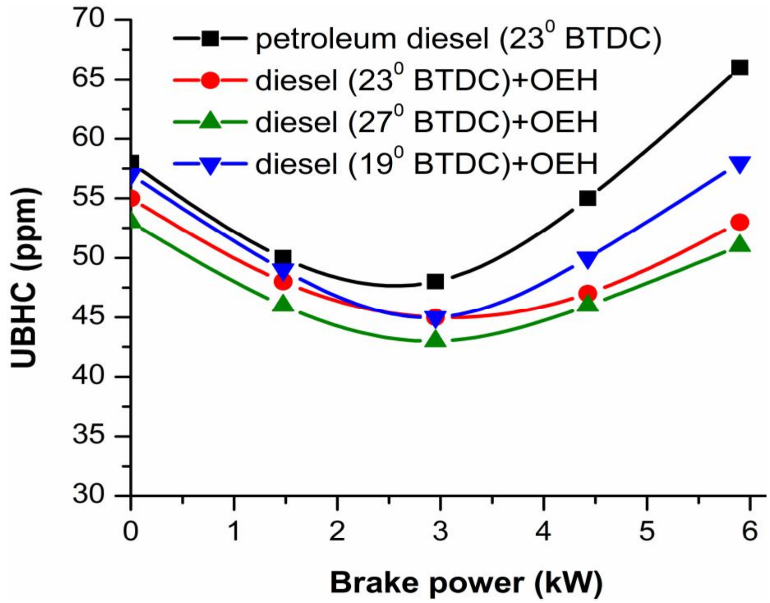

Figure 5.17 Variation of UBHC with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Figure 5.17 shows the comparison of UBHC emission when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings of diesel fuel. The advancement of injection time lessens the UBHC emission whereas retarding the injection amplifies the same. Under the influence of OEH gas at 100% rated load of the engine, UBHC emission got decreased by 19.7% and 22.72% for standard injection timing of 23º BTDC and advanced injection timing of 27º BTDC compared to base line operation. This might be due to enhanced H/C ratio in the overall fuel mixture.

At the retarded injection timing of 19º BTDC, UBHC emission got decreased by 12.12% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 9.43% and 13.72%

increase in UBHC emission compared to 23º BTDC and 27º BTDC. This might be due to low homogeneity of combustible mixture formed during the ignition delay period. At 27º BTDC, the minimum UBHC emission was exhausted from the engine compared to other injection timings. The UBHC emission got decreased by 3.77% at 27º BTDC compared to 23º BTDC. This might be due to proper mixture formation with enough oxygen to burn all the fuel particles.

5.2.1.6 Oxides of nitrogen emission (NO_X)

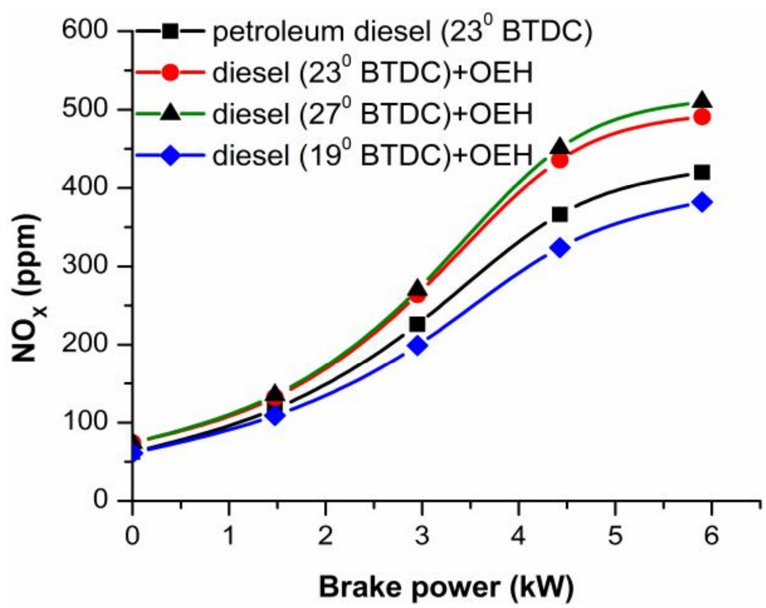

Figure 5.18 Variation of NO_X with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Figure 5.18 represents the comparison of NO_X emission when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings of diesel fuel. The advancement of injection time enhanced the NOx emission whereas retarding the injection helped to reduce the same. Under the influence of OEH gas at 100% rated load, NO_X emission increased by 16.9% and 21.42% for standard injection timing of 23º BTDC and advanced injection timing of 27º BTDC compared to base line operation. This might be due to enhanced pre-mixed burning phase as a result of instantaneous combustion of OEH gas when it was ignited by pilot diesel fuel.

At the retarded injection timing of 19º BTDC, NO_X emission got decreased by 9.04% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 22.19% and 25.09% decrease in NO_X emission compared to 23º BTDC and 27º BTDC. This might be due to low temperature atmosphere prevailing in the combustion chamber as less time was available to form homogeneous mixture during the ignition delay period which resulted in a drop in the combustion temperature (Fathi et al 2011).

At 27º BTDC, the maximum NO_X emission occurred in the engine compared to other injection timings. The NO_X emission increased by 3.86% at 27º BTDC compared to 23º BTDC. This might be due to increase in the ignition delay period. When the start of fuel injection timing was earlier, the initial air temperature and pressure would be lower. This caused ignition delay period to increase which in-turn increased the premixed burning phase, the cylinder gas temperature and the NO_X emissions (Turkcan & Canakci 2011).

5.2.1.7 Smoke emission

Figure 5.19 displays the comparison of smoke emission when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings of the diesel fuel and petroleum diesel combustion at standard injection timing. The experimental results showed that the smoke emission got increased when the injection time was retarded and got decreased when the injection time was advanced.

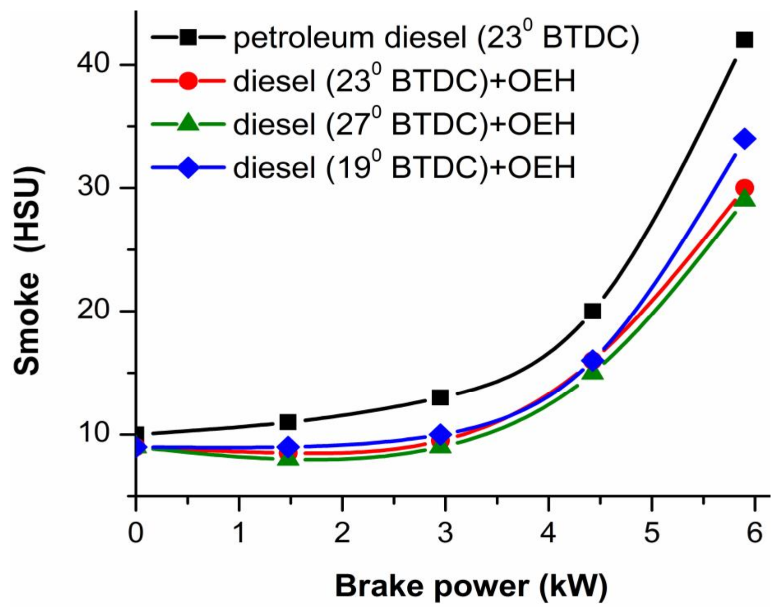

Figure 5.19 Variation of smoke emission with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Under the influence of OEH gas at 100% rated load of the engine, the smoke emission decreased by 28.57%, 19.04%, and 30.95% for standard injection timing of 23º BTDC, retarded injection timing of 19º BTDC, and advanced injection timing of 27º BTDC compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 13.33% increase in smoke emission compared to standard injection time operation and 17.24% increase in smoke emission compared to advanced injection time operation of 27º BTDC. When injection time of diesel fuel was retarded, the regions of the better air/fuel mixing got decreased. This in-turn decreased the pre-mixed combustion phase (Mohammadi et al 2007), and heat release rate. Owing to this higher smoke was emitted from the engine during retarded injection timed operation than other injection timed operations.

At 27º BTDC, the minimum smoke emission was obtained compared to other injection timings. The smoke emission got decreased by 3.33% at advanced injection time of 27º BTDC compared to standard injection time of 23º BTDC. When the diesel fuel was injected at advanced injection time, the fuel got sufficient time to mingle with air molecules. This resulted in formation of more homogeneous mixture of fuel and air. When this mixture got ignited, the combustion resulted in less smoke emission compared to other injection timed operations.

5.2.1.8 Excess oxygen emission

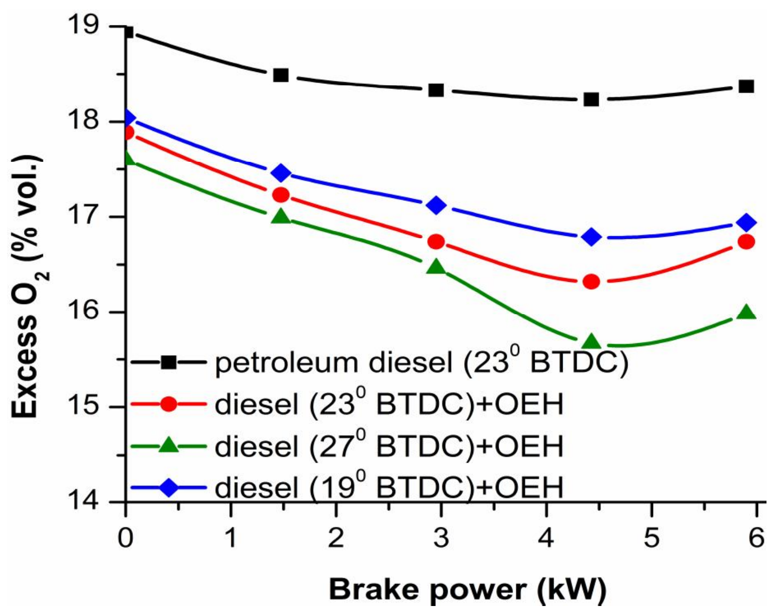

Figure 5.20 Variation of excess oxygen emission with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

Figure 5.20 depicts the comparison of excess oxygen emission for petroleum diesel and diesel with OEH of 4.6 lpm at different injection timings of diesel fuel. The experimental results showed that the excess oxygen emission increased when injection timing was retarded and decreased when injection timing was advanced. When the test engine was operated in retarded injection time of 19º BTDC at the rated load of the test engine, it resulted in 1.19% and 6% increase in excess oxygen emission compared to 23º BTDC and 27º BTDC. This might be due to existence of more fuel-rich zones at this injection timed operation.

The excess oxygen emission decreased by 7.78%, 8.87%, and 13.01% at 19º BTDC, 23º BTDC, and 27º BTDC respectively compared to base line operation. This might be due to high diffusion co-efficient of hydrogen present in the gas mixture and its low activation energy resulting in efficient combustion (Milen & Kiril 2004). At 27º BTDC, the excess oxygen emitted from the engine was lower when compared to other injection timed operations. The excess oxygen available at the exhaust of the engine at the advanced injection time of 27º BTDC got decreased by 4.54% compared to 23º BTDC operation. This might be due to the occurrence of more molecular collisions during the combustion at this injection timed operation than other injection timed operations.

5.2.1.9 Exhaust gas temperature (EGT)

Figure 5.21 illustrates the comparison of EGT of petroleum diesel combustion and when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings of the diesel fuel. Advancing the injection time facilitated to reduce EGT whereas retarding the injection time augmented the same. Under the influence of OEH gas at the maximum load of the test engine, EGT increased by 6.41% and 4.61% for standard injection timing of 23º BTDC and advanced injection timing of 27º BTDC compared to base line operation. This might be due to enhanced premixed burning phase as a result of spontaneous combustion of OEH gas which increased the average cylinder temperature.

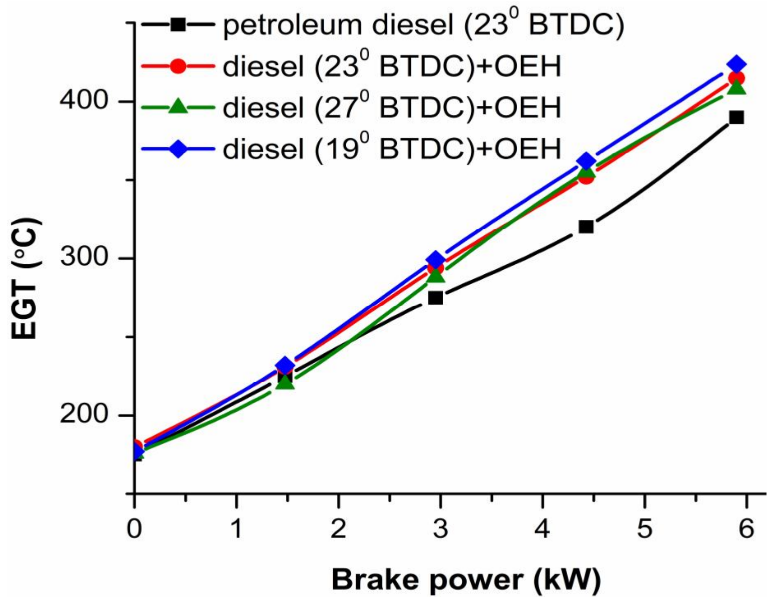

Figure 5.21 Variation of EGT with BP for different injection timings of diesel fuel with OEH gas of 4.6 lpm

At the retarded injection timing of 19º BTDC, EGT got increased by 8.71% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 2.16% and 3.92% increase in EGT compared to 23º BTDC and 27º BTDC injection timed operations. This might be due to improper expansion of combustion gases as little time was available for expansion. At 27º BTDC, the minimum EGT was exhausted from the engine compared to other injection timings. The EGT decreased by 1.68% at 27º BTDC compared to 23º BTDC. The peak pressure was achieved at around 362 degree crank angle for 27º BTDC combustion. This facilitated more complete expansion of combustion gases when compared to other injection timed operations. The heat release curve at this injection time also confirmed this statement.

5.2.1.10 Heat release rate (HRR)

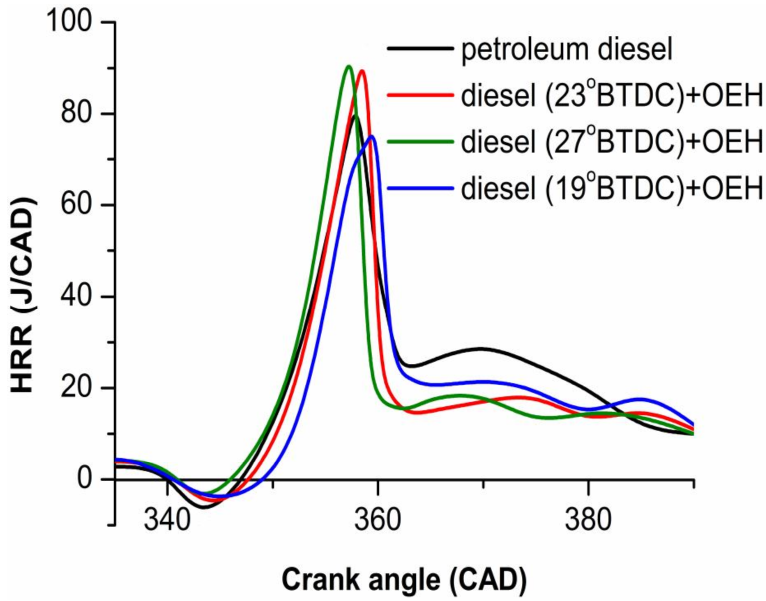

Figure 5.22 Variation of HRR with CA for different injection timings of diesel fuel with OEH gas of 4.6 lpm at rated load

Figure 5.22 compares heat release rate with crank angle when OEH gas of 4.6 lpm was inducted in the diesel combustion process at different injection timings of diesel fuel at rated load of the engine. The advancement of injection time augmented the heat release rate whereas retarding the injection time helped to decrease the same. Under the influence of OEH gas of 4.6 lpm at 100% rated load of the engine, the heat release rate increased by 13.75% and 16.25% for standard injection timing of 23º BTDC and advanced injection timing of 27º BTDC compared to base line operation. This might be due to more constant volume combustion. This led to enhanced premixed combustion phase. At the retarded injection timing of 19º BTDC, the heat release rate decreased by 7.5% compared to base line operation.

When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 18.68% and 20.43% decrease in heat release rate compared to 23º BTDC and 27º BTDC operations respectively. When the engine was operated in a retarded injection time of diesel fuel, the fuel was introduced into the cylinder at comparatively higher pressure and temperature environment. Owing to this, the ignition delay period and the pre-mixed combustion phase got reduced (Mohammadi et al 2007). At 27º BTDC, the maximum heat release rate was obtained compared to other injection timings. The heat release rate increased by 2.19% at 27º BTDC compared to 23º BTDC. This might be due to the elevated flame temperature and less heterogeneous fuel-air mixture at this injection timed operation.

5.2.1.11 In-cylinder pressure

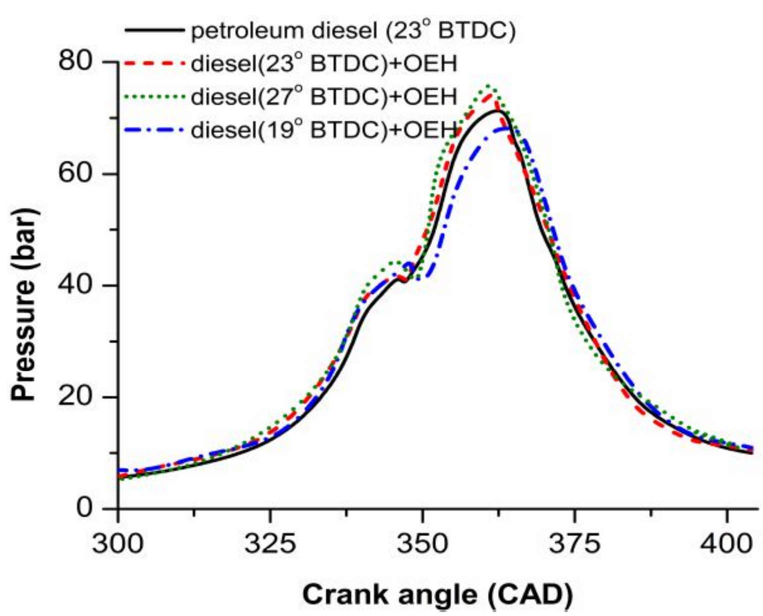

Figure 5.23 Variation of in-cylinder pressure with CA for different injection timings of diesel fuel with OEH gas of 4.6 lpm at rated load

Figure 5.23 compares in-cylinder pressure with crank angle when OEH gas of 4.6 lpm was inducted in the diesel combustion process at different injection timings of diesel fuel at rated load of the engine. Advancing the injection time of diesel fuel amplified the peak in-cylinder pressure whereas retarding the injection time helped to decrease the same. When OEH gas of 4.6 lpm was introduced to diesel combustion at 100% rated load, the peak in-cylinder pressure got increased by 5.71% and 10.72% for standard injection timing of 23º BTDC and advanced injection timing of 27º BTDC compared to base line operation. This might be due to enhanced pre-mixed burning phase.

When the pre-mixed burning was enhanced, flame propagation through the hydrogen-air mixture led to rapid heat release rates, increased the peak cylinder pressure and temperature, and improved brake thermal efficiency (Kumar et al 2003). At the retarded injection timing of 19º BTDC, the peak in-cylinder pressure got decreased by 2.85% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC, it resulted in 8.1% and 12.25% decrease in peak in-cylinder pressure compared to 23º BTDC and 27º BTDC. At 27º BTDC, the maximum peak in-cylinder pressure was obtained compared to other injection timings. The peak in-cylinder pressure increased by 4.72% at 27º BTDC compared to 23º BTDC. The peak in-cylinder pressure primarily depends on mixing rate, temperature and availability of oxidants like OH and oxygen radicals. At the injection time of 27º BTDC, all these facts were more pronounced and resulted in a more homogeneous mixture and efficient combustion.

5.2.2 Varied injection pressures

In the present experimental work, the best flow rate OEH gas of 4.6 lpm was aspirated into the cylinder along with intake air at varied injection pressures of a diesel fuel. The injection pressure was varied from 200 bar of engine manufacturer recommended specification to 180 bar, 220 bar, and 240 bar by adjusting the spring tension of the injector (Puhan et al 2009).

5.2.2.1 Brake thermal efficiency (BTE)

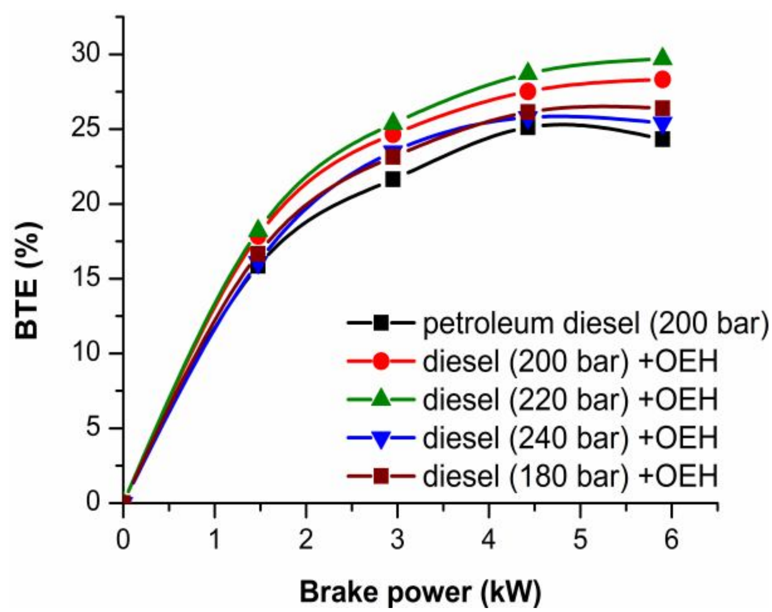

Figure 5.24 Variation of BTE with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.24 shows the graphical representation of effect of OEH gas with varied injection pressures of diesel fuel on the brake thermal efficiency at different rated loads of the test engine. From the graph, it is obvious that the brake thermal efficiency increased when OEH gas was used as an additive along with high pressure injection of diesel fuel in the combustion process of petroleum diesel. When the diesel injection pressure was increased from 180 bar to 240 bar in steps of 20 bar with OEH gas at 100% rated load, the brake thermal efficiency increased by 8.51%, 16.45%, 22.08%, and 4.48% respectively compared to base line operation. This increase in brake thermal efficiency was due to smaller droplet size of fuel produced by high injection pressure, higher calorific value of hydrogen present in the gas mixture, and its high flame velocity (Wang et al 2012b). These facts accounted for increase in brake thermal efficiency.

When the test engine was operated at the injection pressure of 180 bar, it resulted in 6.81% and 11.11% decrease in brake thermal efficiency compared to 200 bar and 220 bar injection pressure operations. This might be due to the poor atomization and mixture formation before combustion. The maximum brake thermal efficiency was witnessed at 220 bar injection pressure compared to other injection pressures. The brake thermal efficiency increased by 4.83% at the injection pressure of 220 bar compared to 200 bar injection pressure. At the injection pressure of 240 bar, the brake thermal efficiency got decreased by 10.27% and 14.41% when compared to 200 bar and 220 bar injection pressure operations. This might be due to shorter ignition delay period. So, the possibilities of homogeneous mixing of air and fuel got decreased and combustion efficiency got reduced (Murayama 1994, Ghazikhani & Darbandi 2010, Bakar et al 2008).

5.2.2.2 Brake specific energy consumption (BSEC)

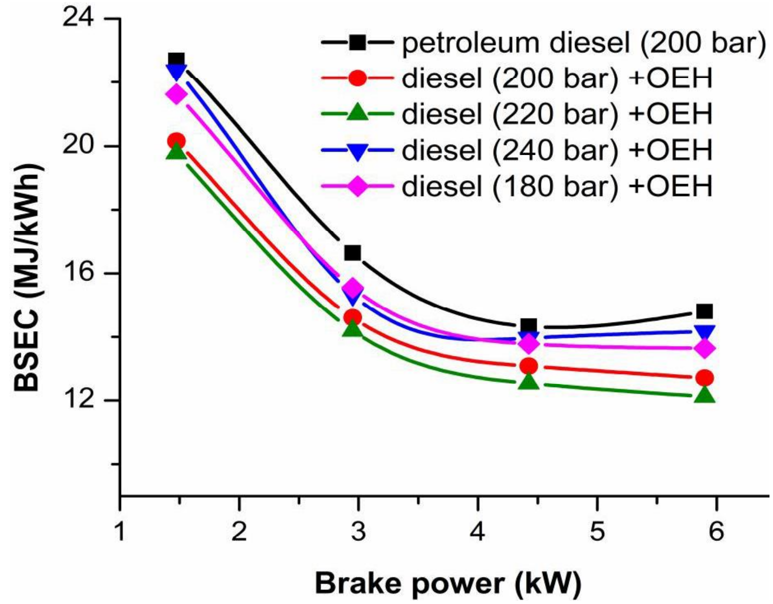

Figure 5.25 Variation of BSEC with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.25 illustrates the comparison of BSEC when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection pressures of the diesel fuel at different rated loads of the test engine. From the graph, it is seen that the BSEC decreased when OEH gas was used as an additive along with high pressure injection of diesel fuel in the combustion process of diesel. When the diesel injection pressure was 180 bar, 200 bar, 220 bar, and 240 bar, under the influence of OEH gas at maximum load of the engine, the BSEC got decreased by 7.84%, 14.12%, 18.08%, and 4.28% respectively compared to base line operation. This decrease in BSEC was a twin effect of OEH gas addition and high injection pressure of diesel fuel. When OEH gas was utilized in the diesel combustion process, due to its fast combustion rates, the energy extracted from diesel fuel was more. In addition to this, the high injection pressure of diesel fuel generated fine droplets of fuel at the nozzle exit which increased surface area of fuel droplets exposed to air. This resulted in more intimacy between the air and the fuel droplets. When they got combusted, it resulted in good combustion and more utilization of energy of fuel.

When the test engine was operated at the injection pressure of 180 bar, it resulted in 7.31% and 12.5% increase in BSEC compared to 200 bar and 220 bar injection pressure operations. This might be due to the bigger droplets available at the exit of the nozzle when the engine was operated at 180 bar injection pressure of diesel fuel. The minimum BSEC was observed at 220 bar injection pressure compared to other injection pressures. The BSEC got decreased by 4.61% at the injection pressure of 220 bar compared to 200 bar injection pressure. This might be due to increase in the relative velocity of fuel injection (Ommi et al 2008). At the injection pressure of 240 bar, the BSEC got increased by 11.45% and 16.84% when compared to 200 bar and 220 bar injection pressure operations. This is because at this pressure, the concentration of the spray was so high and it might result in evaporative cooling (Rutland & Wang 2006).

5.2.2.3 Carbon monoxide emission (CO)

Figure 5.26 depicts the comparison of CO emission for petroleum diesel and diesel with OEH of 4.6 lpm at different injection pressures of diesel fuel. The experimental results showed that the CO emission increased when injection pressure was decreased and decreased when injection pressure was increased. When the test engine was operated at injection pressure of 180 bar, CO emission got increased by 9.09% compared to 200 bar operation. This might be due to the decreased cone angle which resulted in incomplete combustion. The CO emission got decreased by 9.09% at 220 bar.

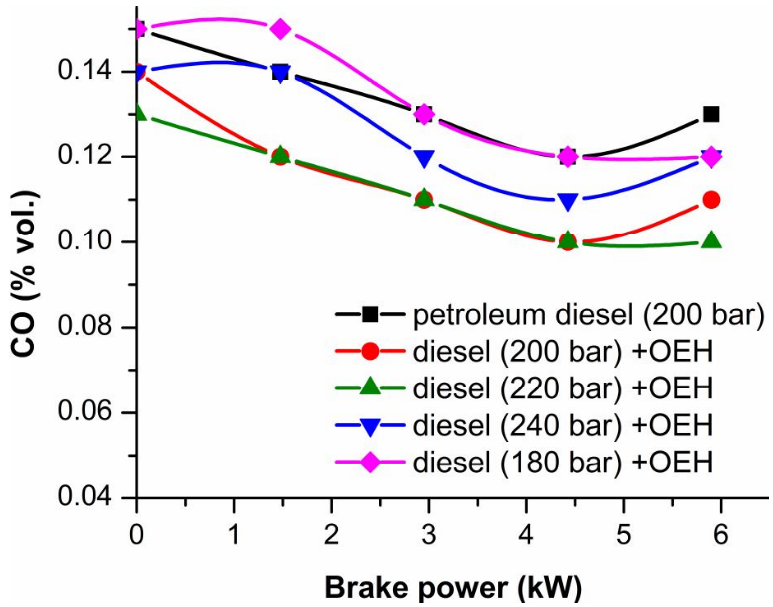

Figure 5.26 Variation of CO with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

At 220 bar, the minimum CO emission was obtained compared to other injection pressures. The main reason for the decrease was the completeness of combustion process and sufficiency of oxygen (Mohammed et al 2011). At 240 bar, CO emission got increased by 9.09% and 20% when compared to 200 bar and 220 bar operations. This might due to the fact that at high injection pressure, the fuel droplets travelled with a high velocity and that might hit the wall of the combustion chamber and this led to low oxidation reactions and less intense combustion (Ofner et al 1999, Ghazikhani & Darbandi 2010).

5.2.2.4 Carbon dioxide emission (CO_2)

Figure 5.27 shows the graphical display of the effect of OEH gas with varied injection pressures of diesel fuel on CO_2 emission at different rated loads of the test engine.

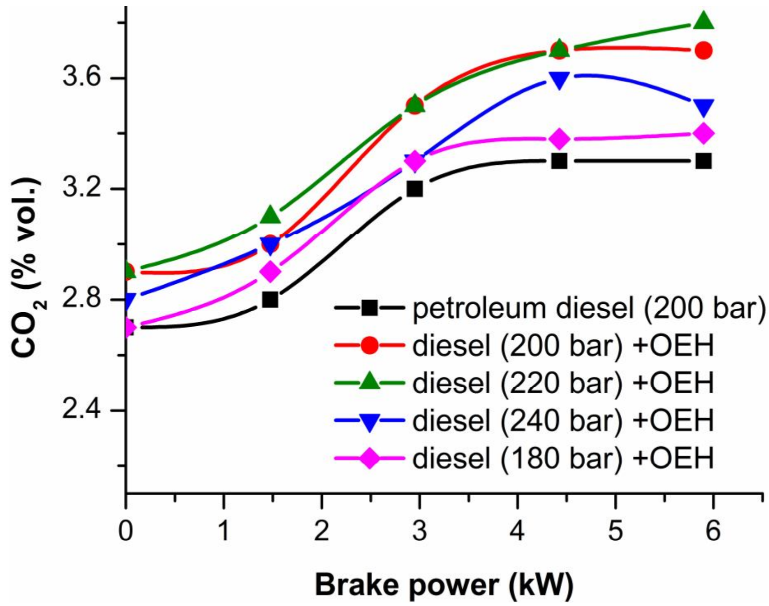

Figure 5.27 Variation of CO_2 with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

From the graph, it is obvious that when the diesel injection pressure was 180 bar, 200 bar, 220 bar, and 240 bar, under the influence of OEH gas at 100% rated load of the engine, CO_2 emission got increased by 3.03%, 12.12%, 15.15%, and 6.06% compared to base line operation. This increase in CO_2 emission was due to good atomization and the better spray characteristics of a diesel fuel along with high diffusion rate of hydrogen present in the gas mixture, which resulted in efficient combustion. At the injection pressure of 180 bar, CO_2 emission got reduced by 9.09% and 20% compared to 200 bar and 220 bar injection pressure operations. This might be due to the poor spray penetration because of higher surface tension of diesel fuel when compared to higher injection pressure fuel.

The maximum CO_2 emission was exhausted from the engine at 220 bar injection pressure operation compared to other injection pressures. The CO_2 emission got increased by 2.7% at the injection pressure of 220 bar compared to

200 bar injection pressure. This might due to proper diffusion that took place at this injection pressure, because of this more amount of fuel-air mixture got combusted. The heat release rate curve at this injection pressure also confirmed this. At the injection pressure of 240 bar, CO_2 emission got decreased by 5.4% and 7.89% when compared to 200 bar and 220 bar injection pressure operations. This is because when the injection pressure of diesel fuel was increased along with OEH gas, the fuel droplets travelled with a high velocity and formed a thick boundary layer at the wall of the combustion chamber. This might have resulted in a low degree of combustion. The lower brake thermal efficiency obtained at this injection pressure also confirmed this statement.

5.2.2.5 Unburned hydrocarbon emission (UBHC)

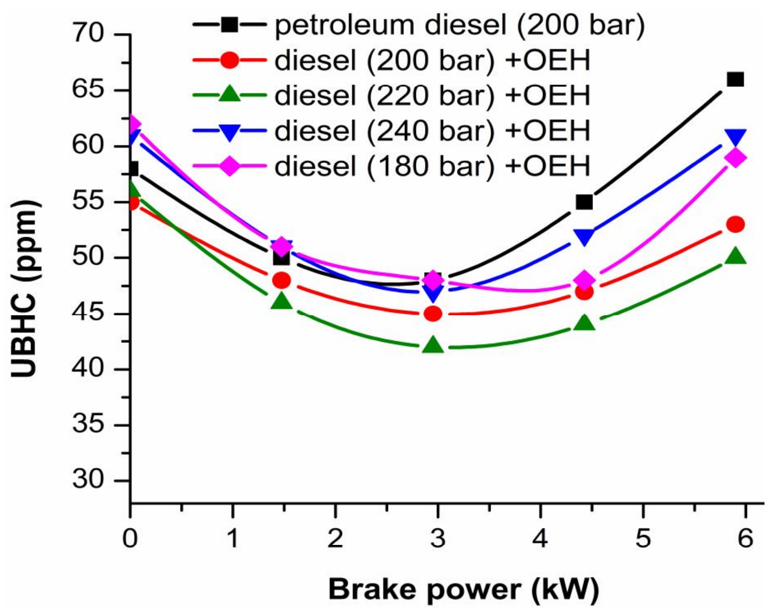

Figure 5.28 Variation of UBHC with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.28 demonstrates the graphical representation of effect of OEH gas with varied injection pressures of diesel fuel on UBHC emission at different rated loads of the test engine. From the graph, it is distinguished that when the diesel injection pressure was 180 bar, 200 bar, 220 bar, and 240 bar, under the influence of OEH gas at 100% rated load of the engine, UBHC emission got decreased by 10.6%, 19.7%, 24.24%, and 7.57% compared to base line operation. This decrease in UBHC emission was due to low quenching distance of hydrogen present in the gas mixture, its fast burning velocity, and the wider cone angle of spray of diesel fuel at high injection pressures.

At the injection pressure of 180 bar, UBHC emission got increased by 11.32% and 18% compared to 200 bar and 220 bar injection pressure operations. This might be due to decrease in turbulence intensity prevailing in the combustion chamber which resulted in incomplete combustion. The minimum UBHC emission was exhausted from the engine at 220 bar injection pressure operation compared to other injection pressures. The UBHC emission got decreased by 5.66% at the injection pressure of 220 bar compared to 200 bar injection pressure. This might be due to fewer over-rich regions present in the combustion chamber (Julien 2006) and this also resulted in higher brake thermal efficiency. The heat release rate diagram at this injection pressure also confirmed this. At the injection pressure of 240 bar, UBHC emission got increased by 15.09% and 22% when compared to 200 bar and 220 bar injection pressure operations. A very high injection pressure like 240 bar might led to fine droplets, and this could have negatively affected fuel distribution in air (Balusamy & Marappan 2007).

5.2.2.6 Oxides of nitrogen emission (NO_X)

Figure 5.29 exhibits the graphical representation of the effect of OEH gas with varied injection pressures of diesel fuel on NO_X emission at different rated loads of the test engine.

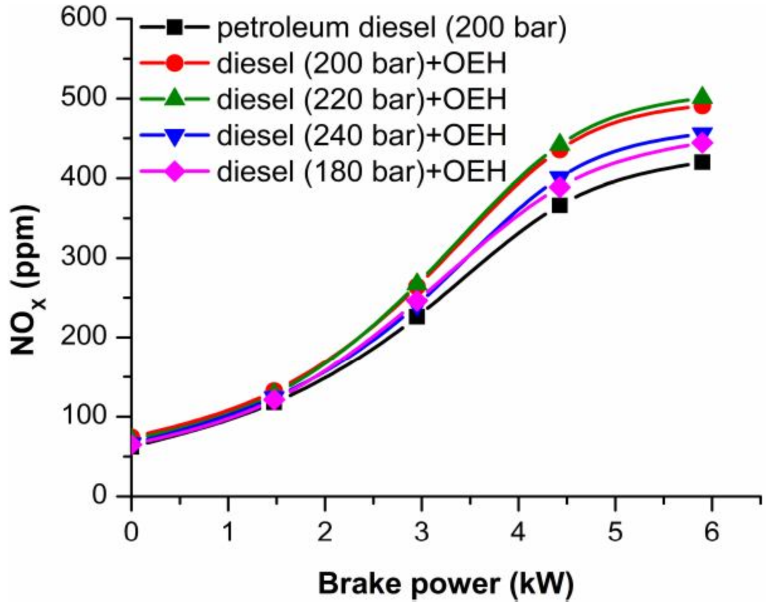

Figure 5.29 Variation of NO_X with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

From the graph, it is apparent that when the diesel injection pressure was 180 bar, 200 bar, 220 bar, and 240 bar, under the influence of OEH gas at maximum rated load, NO_X emission got increased by 5.83%, 16.9%, 19.29%, and 8.57% respectively compared to base line operation. This increase in NO_X emission was due to the high combustion rate of OEH gas and a high injection pressure of diesel fuel. The high injection pressure contributed to produce more developed sprays with shorter injection durations (Karimi 2007) and improved pre-mixed combustion phase. As a consequence, NO_X concentrations got increased.

At the injection pressure of 180 bar, NO_X emission got reduced by 9.47%, 11.27%, and 2.52% compared to 200 bar, 220 bar, and 240 bar injection pressure operations. This might be due to the deprived atomization and reduced flame temperature which resulted in poor combustion. The maximum NO_X emission from the engine occurred at 220 bar injection pressure operation compared to other injection pressures. The NO_X emission increased by 2.03% at the injection pressure of 220 bar compared to 200 bar injection pressure. This might be due to both high temperature and more available oxygen in the formed mixture that had caused NO_X emissions to rise (Heywood 1988). The heat release rate diagram at this injection pressure also confirmed this.

5.2.2.7 Smoke emission

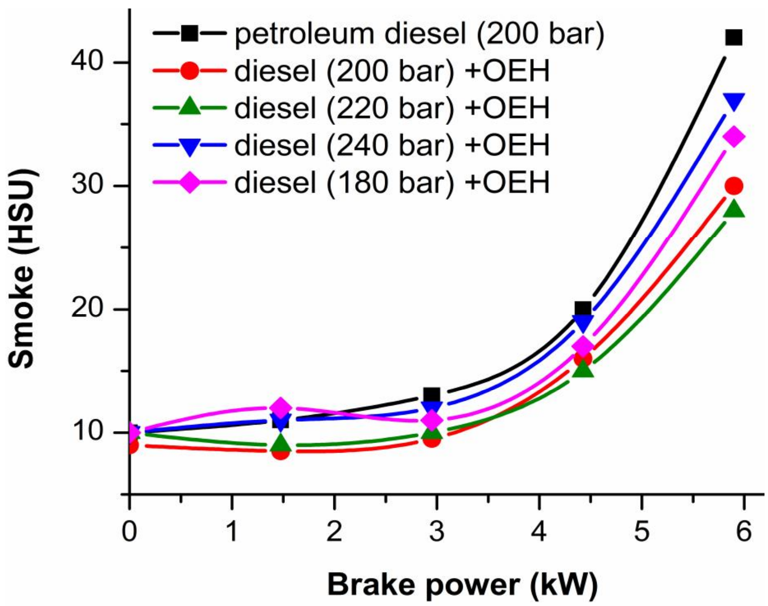

Figure 5.30 Variation of smoke emission with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.30 shows the comparison of smoke emission for petroleum diesel and diesel with OEH gas of 4.6 lpm at different injection pressures of diesel fuel. The experimental results showed that the smoke emission got augmented when injection pressure was decreased and got deprived when injection pressure was increased. When OEH gas was inducted into the combustion process, the smoke reduced substantially. The smoke was emitted from the engine due to the incomplete combustion of the fuel-air mixture. When the injection pressure of the diesel fuel was 220 bar, the smoke emission decreased from 42 HSU to 28 HSU, i.e., by 33.33% at 100% rated load of the engine compared to base line operation. At the injection pressure of 240 bar, 200 bar, and 180 bar, the smoke emission got reduced by 11.91%, 28.57%, and 19.04% respectively compared to base line operation.

This reduction in smoke emission was due to amalgamated effect of OEH gas and the high-pressure injection of diesel fuel. During the OEH gas influenced diesel combustion, the heavier hydrocarbon fuel molecule structure was fractured into lighter and smaller hydrocarbon structures in less time. The high pressure injection improved diesel fuel penetration and evaporation. This perked up the mixing of fuel and air and formed the combustible mixture in a very short time. This improved fuel-air mixture reduced engine-out smoke emission substantially when got combusted.

When the test engine was operated at injection pressure of 180 bar, the smoke emission got increased by 13.33% compared to 200 bar injection pressure operation. This might be due to the dwindled cone angle of spray which resulted in incomplete combustion. The Smoke emission got decreased by 17.64% at 220 bar when compared to 180 bar injection pressure operation. At 220 bar, the minimum smoke emission was obtained compared to other injection pressures. This might be due to optimal sauter mean diameter (SMD) of diesel fuel spray which improved the mixing process of fuel with air. At 240 bar, smoke emission got increased by 23.33% and 32.14% when compared to 200 bar and 220 bar injection pressure operations. This is because at high injection pressure the fuel droplets travel with high momentum. This adversely affects the mixing of fuel with air resulting in poor combustion.

5.2.2.8 Excess oxygen emission

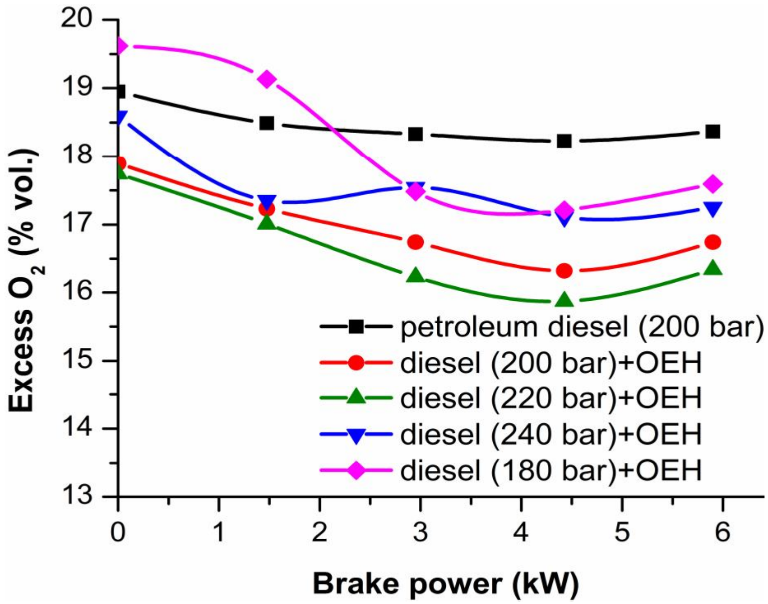

Figure 5.31 Variation of excess oxygen emission with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.31 compares excess oxygen emission for petroleum diesel and diesel with OEH of 4.6 lpm at different injection pressures of diesel fuel. The experimental results showed that the excess oxygen emission increased when the injection pressure was decreased and decreased when the injection pressure was increased. When OEH gas was inducted into the combustion process, the excess oxygen available in the exhaust of the engine got reduced significantly. The excess oxygen was emitted from the engine due to the incomplete combustion of the fuel-air mixture. When the injection pressure of the diesel fuel was 220 bar, the excess oxygen emission decreased by 11.05% at maximum load of the engine compared to petroleum diesel combustion. At the injection pressure of 240 bar, 200 bar, and 180 bar the excess oxygen emission got reduced by 6.09%, 8.87%, and 4.24% respectively compared to base line operation.

This is due to the combined effect of high combustion rates assisted by high diffusivity of hydrogen and fine atomization of fuel droplets due to high injection pressure of diesel fuel made the combustible mixture more homogeneous. This created instantaneous combustion when OEH gas was ignited by pilot petroleum diesel. The increased droplet velocity (Siebers 1999, Naber & Siebers 1996), decreased droplet size (Lee & Park 2002), and shortened ignition delay resulting in higher initial combustion (Purushothaman & Nagarajan 2009) and higher heat release rate.

When the test engine was operated at injection pressure of 180 bar, the excess oxygen emission got increased by 5.07% and 7.65% compared to 200 bar and 220 bar operations respectively. This was because, when the injection pressure was decreased, ignition delay period before start of the combustion got increased (Bakar et al 2008). This situation caused an inferior combustion which resulted in more excess oxygen emission available in the engine exhaust. At 220 bar, the minimum excess oxygen emission was obtained compared to other injection pressures. When the injection pressure of the diesel fuel was increased, the flame lift off length got increased and this enhanced the air entrainment (Siebers & Higgins 2001). At 240 bar, the excess oxygen emission got increased by 3.04% and 5.56% when compared to 200 bar and 220 bar operations. If injection pressure was too high like 240 bar, the ignition delay period became shorter. So, the potential of homogeneous mixing got reduced (Celikten 2003). This resulted in deprived combustion efficiency. Therefore, the in-cylinder pressure also got decreased.

5.2.2.9 Exhaust gas temperature (EGT)

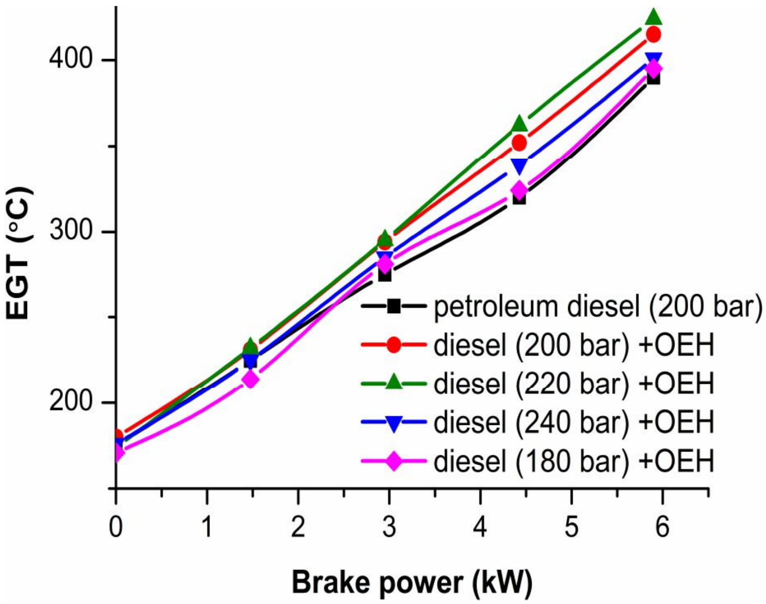

Figure 5.32 Variation of EGT with BP for different injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.32 shows the comparison of EGT for petroleum diesel and diesel with OEH gas of 4.6 lpm at different injection pressures of diesel fuel at different rated loads of the test engine. From the graph, it is well-known that EGT at the maximum load of the test engine got increased by 1.28%, 6.41%, 8.72%, and 2.82% compared to base line operation when the diesel injection pressure was 180 bar, 200 bar, 220 bar, and 240 bar respectively. This increase in EGT was due to fast combustion rates of hydrogen presented in the gas mixture and the superior spray characteristics of a diesel fuel at high injection pressures.

At the injection pressure of 180 bar, EGT got reduced by 4.81%, 6.83%, and 1.49% compared to 200 bar, 220 bar, and 240 bar injection pressure operations respectively. This might be due to the inferior quality of combustion. The maximum EGT was exhausted from the engine at 220 bar injection pressure operation compared to other injection pressures. The EGT increased by 2.16% at the injection pressure of 220 bar compared to 200 bar injection pressure. This is might be due to higher adiabatic flame temperature developed in the combustion process when compared to other injection pressure combustions. The heat release rate curve at this operating condition also confirmed this. At the injection pressure of 240 bar, EGT got decreased by 3.37% and 5.42% when compared to 200 bar and 220 bar injection pressure operations. This might be due to high momentum of the spray which disturbed the formation of homogeneous mixture of fuel and air and resulted in low flame temperature in the combustion process.

5.2.2.10 Heat release rate (HRR)

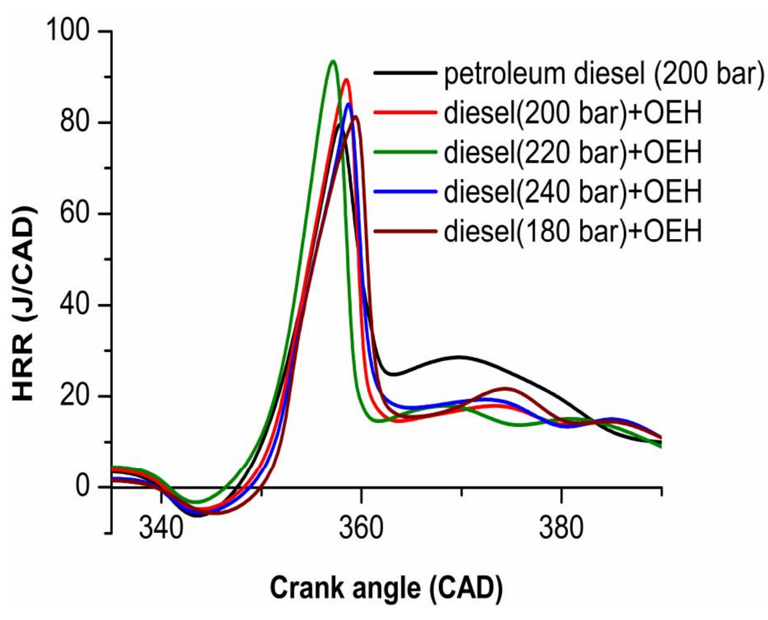

Figure 5.33 Variation of HRR with CA for different injection pressures of diesel fuel with OEH gas of 4.6 lpm at rated load

Figure 5.33 represents the effect of OEH gas with varied injection pressures of diesel fuel on the heat release rate with crank angle at rated load of the test engine. When the diesel injection pressure was 180 bar, 200 bar, 220 bar, and 240 bar, under the influence of OEH gas at 100% rated load of the engine, the heat release rate increased by 3.75%, 13.75%, 20%, and 6.25% compared to base line operation. This increase in heat release rate was due to faster combustion rates of hydrogen present in the gas mixture, enhanced pre-mixed combustion, and the better spray characteristics of a diesel fuel at high injection pressures. At the injection pressure of 180 bar, heat release rate got reduced by 8.79%, 13.54%, and 2.35% compared to 200 bar, 220 bar, and 240 bar injection pressure operations respectively. This might be due to long life of fuel droplets which reduced spray diffusion and finally resulting in deprived combustion (Rutland & Wang 2006).

The maximum heat release rate occurred at 220 bar injection pressure operation compared to other injection pressures. The heat release rate increased by 5.49% at the injection pressure of 220 bar compared to 200 bar injection pressure. This might be due to larger flame length developed at this injection pressure (Wang et al 2011). Owing to this smoke emission also got decreased. At the injection pressure of 240 bar, heat release rate got decreased by 6.59% and 11.45% compared to 200 bar and 220 bar injection pressure operations respectively. This might be due to an excessively early onset of self-ignition (Themel et al 1998).

5.2.2.11 In-cylinder pressure

Figure 5.34 depicts the effect of OEH gas with varied injection pressures of diesel fuel on the in-cylinder pressure with crank angle at rated load of the test engine. When the diesel fuel injection pressures were 180 bar, 200 bar, 220 bar, and 240 bar, under the influence of OEH gas at 100% rated load of the engine, the peak in-cylinder pressure got increased by 2.14%, 5.71%, 12.86%, and 3.57% respectively compared to base line operation.

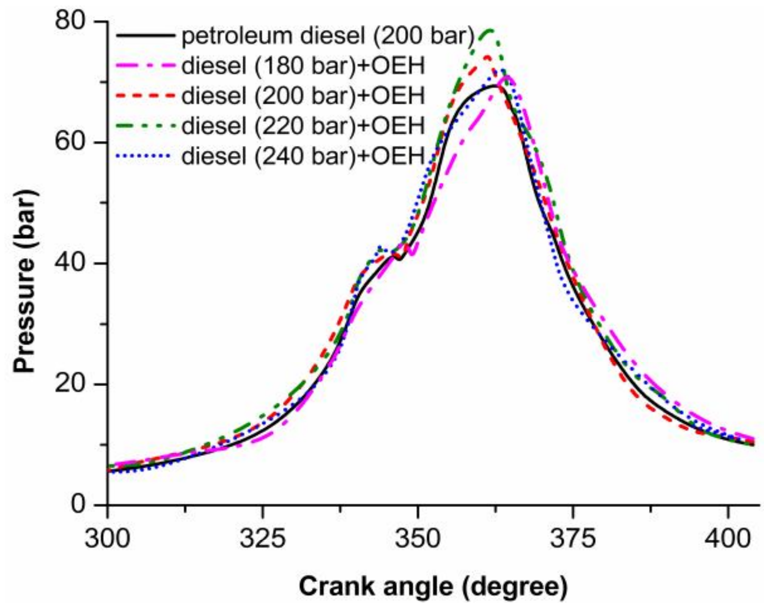

Figure 5.34 Variation of in-cylinder pressure with CA for different injection pressures of diesel fuel with OEH gas of 4.6 lpm at rated load

When going through the graph, it is observed that there was always advancement in occurrence of peak pressure in OEH gas assisted high injection pressure diesel combustion. The reason was that the mixture of fuel and air had undergone instantaneous combustion. This resulted in high pressure and high temperature in the combustion process.

At the injection pressure of 180 bar, the peak in-cylinder pressure got reduced by 3.37%, 9.49%, and 1.38% compared to 200 bar, 220 bar, and 240 bar injection pressure operations respectively. This might be due to decrease in surface area to volume ratio of the droplets (Rutland & Wang 2006). This resulted in more heterogeneous mixture of fuel and air and low efficiency combustion. The maximum peak in-cylinder pressure occurred at 220 bar injection pressure operation compared to other injection pressures. The peak in-cylinder pressure increased by 6.75% at the injection pressure of 220 bar compared to 200 bar injection pressure. According to Julien (2006), this was due to increase in vapour propagation rate which was due to decrease in the SMD of fuel particles and these facts resulted in increase in the degree of formation of homogeneous mixture of fuel and air.

At the injection pressure of 240 bar, peak in-cylinder pressure got decreased by 2.02% and 8.22% compared to 200 bar and 220 bar injection pressure operations. This might be due to high momentum of fuel droplets at a very high injection pressure like 240 bar. This resulted in the impinging of spray on the cylinder walls (Ofner et al 1999, Ghazikhani et al 2007a) which negatively affected the combustion process.

5.2.3 Varied flow rates of cooling water

In the present experimental work, the best flow rate OEH gas of 4.6 lpm was aspirated into the cylinder along with intake air at varied flow rates of cooling water. The cooling water flow rate was varied from 100% to 90%, and to 75% by controlling the outlet valve of the cooling water system.

5.2.3.1 Brake thermal efficiency (BTE)

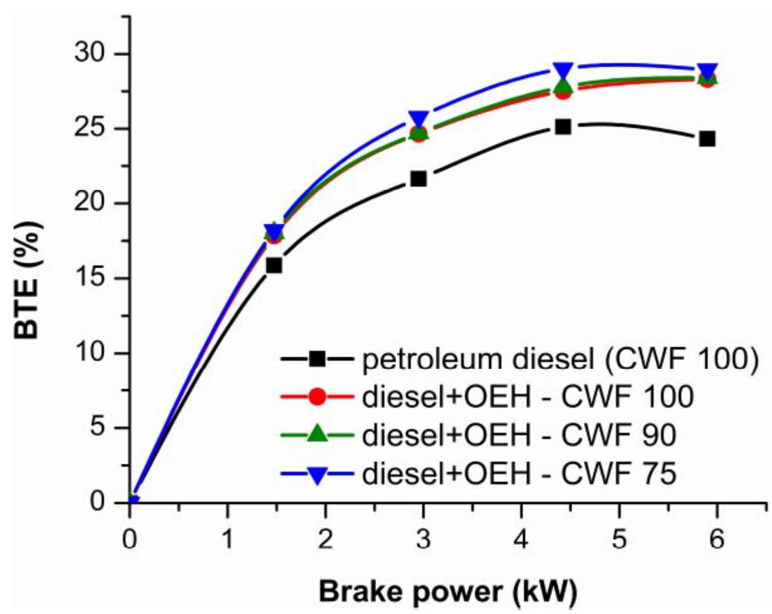

Figure 5.35 Variation of BTE with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

Figure 5.35 displays the variation of brake thermal efficiency with brake power for OEH gas of 4.6 lpm of flow rate with cooling water flow rate of 75% (CWF75), 90% (CWF90), and 100% (CWF100). On analyzing the graph, it is evident that the brake thermal efficiency increased, when OEH gas was used in combustion of pure petroleum diesel. When the cooling water flow rate was CWF75, CWF90, and CWF100 at 100% rated load, the brake thermal efficiency increased by 19%, 16.82%, and 16.45% respectively compared to base line operation. This increase in brake thermal efficiency was due to combined effect of catalytic action of OEH gas (Dulger & Ozcelik 2000) and the reduction in cooling loss of the engine caused by reducing the cooling water flow rate (Shudo et al 2001).

When hydrogen was added in a combustion process, due to its fast burning rate the heat transfer to the combustion chamber wall was more (Shudo et al 2001). When the combustion heat was confined by reducing the cooling water flow rate, the engine might act as a semi-adiabatic engine. When the cooling water flow rate was CWF100, the brake thermal efficiency decreased by 0.32% and 2.19% compared to CWF90 and CWF75 respectively. On comparing the brake thermal efficiency during CWF90 and CWF75, CWF90 resulted in a decrease in brake thermal efficiency by 1.83%. This might be due to more heat transfer from the combustion chamber to the cooling medium. The maximum brake thermal efficiency was obtained at CWF75 compared to other flow rates. This might be due to maximum retention of heat within the combustion chamber which might enhance the oxidation reactions of fuel mixture.

5.2.3.2 Brake specific energy consumption (BSEC)

Figure 5.36 displays the variation of BSEC with brake power for OEH gas of 4.6 lpm of flow rate with cooling water flow rate of CWF75, CWF90, and CWF100. From the graph, it is obvious that the BSEC increases, when OEH gas was used in combustion of petroleum diesel.

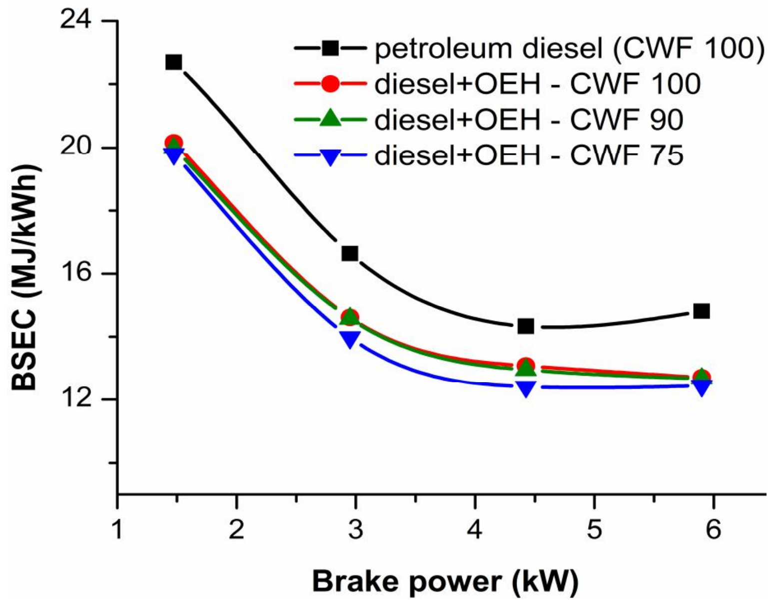

Figure 5.36 Variation of BSEC with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

When the cooling water flow rate was CWF75, CWF90, and CWF100 at maximum load of the test engine, the BSEC decreased by 15.96%, 14.4%, and 14.12% compared to base line operation. This decrease in BSEC was due to high diffusion index of OEH gas and the reduced flow rate of cooling water. When the cooling water flow rate was reduced, it increased the average temperature of the combustion process. These facts resulted in efficient combustion and reduction in engine-out emissions. When the cooling water flow rate was CWF100, BSEC got increased by 0.32% and 2.19% compared to CWF90 and CWF75.

On comparing the BSEC during CWF90 and CWF75, CWF90 resulted in increase in BSEC by 1.87%. This might be due to less combustion rates compared to CWF75. The heat release rate curve at this condition also confirmed this. The minimum BSEC was obtained at CWF75 compared to other flow rates.

This might be due to more homogeneous mixture formed during this flow rate which resulted in more extraction of energy from the fuel.

5.2.3.3 Carbon monoxide emission (CO)

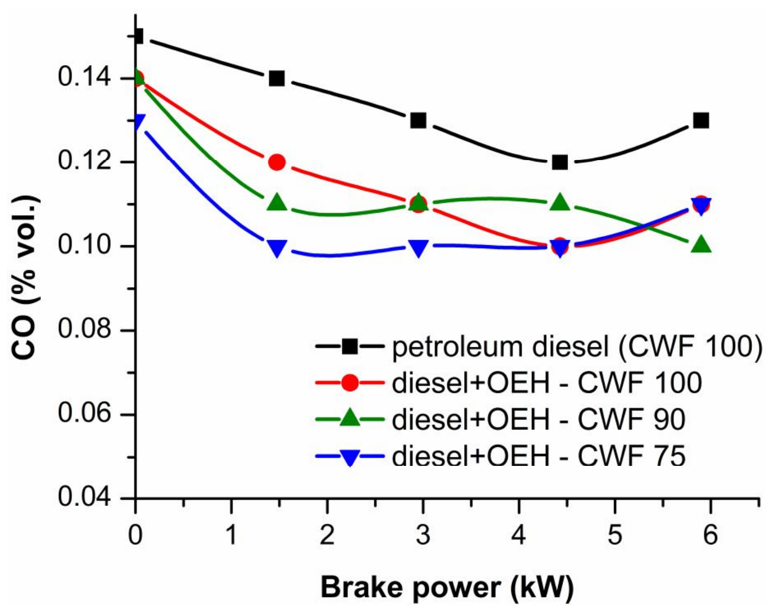

Figure 5.37 Variation of CO with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

Figure 5.37 shows the comparison of CO emission for petroleum diesel and diesel with OEH of 4.6 lpm at different flow rate of cooling water. The experimental results showed that the CO emission got increased when the cooling water flow rate was increased and got decreased when cooling water flow rate was decreased. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load, CO emission got decreased by 15.38%, 23.08%, and 15.38% compared to base line operation.

At CWF100, CO emission got increased by 10% compared to CWF90. This might be due to more over-rich regions that existed during the combustion process at this flow rate. CO emission got decreased by 9.09% at CWF90 compared to CWF75 and CWF100. At CWF90, the minimum CO emission was obtained when compared to other flow rates. This might be due to the more homogeneity of fuel-air mixture which resulted in better combustion. The reduction of CO emission at CWF100 and CWF75 was the same. This might be due to the dissociation of CO_2 into CO and oxygen at CWF75. The CO_2 emission diagram shown in Figure 5.37 also confirmed this.

5.2.3.4 Carbon dioxide emission (CO_2)

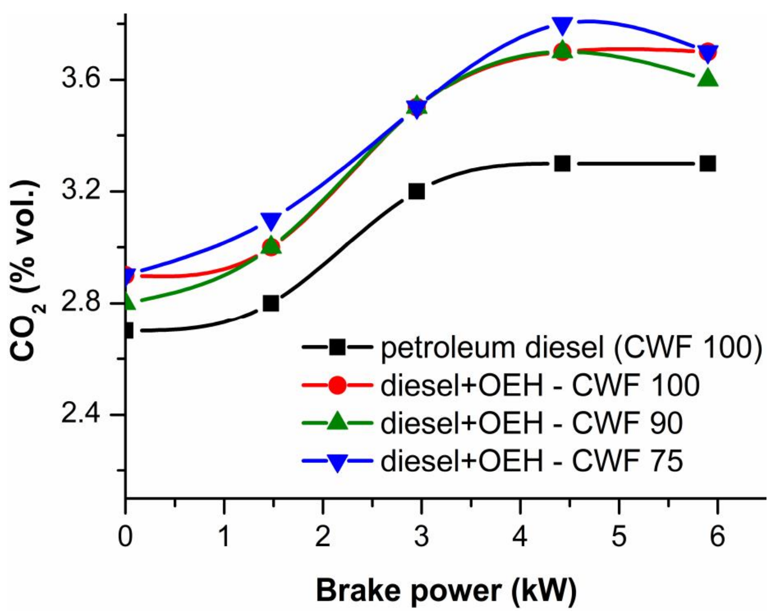

Figure 5.38 Variation of CO_2 with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

Figure 5.38 illustrates the comparison of CO_2 emission for petroleum diesel and diesel with OEH gas of 4.6 lpm at cooling water flow rate of CWF75, CWF90, and CWF100. On investigating the graph, it is learned that the CO_2 emission increased, when diesel combustion was assisted by OEH gas. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load of the engine, CO_2 emission got increased by 12.12%, 9.09%, and 12.12% compared to base line operation. This increase in CO_2 emission was a twin effect of reduction of heat loss to cooling medium and the high oxygen index of OEH gas.

At the cooling water flow rate of CWF100, CO_2 emission got decreased by 2.78% compared to CWF90. On comparing the CO_2 emission during CWF90 and CWF75, CWF90 resulted in a decrease in CO_2 emission by 2.79%. This might be due to low adiabatic flame temperature compared to CWF75 combustion. The heat release curve at this flow rate confirmed this statement. The maximum CO_2 emission was obtained at CWF75 compared to other flow rates. This might be due to an increase in CO final oxidation kinetics (Apostolescu & Chiriac 1998).

5.2.3.5 Unburned hydrocarbon emission (UBHC)

Figure 5.39 depicts the comparison of UBHC emission for petroleum diesel and diesel with OEH gas of 4.6 lpm at different flow rates of cooling water. The experimental results showed that the UBHC emission got increased when cooling water flow rate was increased and got decreased when cooling water flow rate was decreased. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load of the engine, UBHC emission got decreased by 27.27%, 21.21%, and 19.7% compared to base line operation. At CWF100, UBHC emission got increased by 1.92% and 10.42% compared to CWF90 and CWF75.

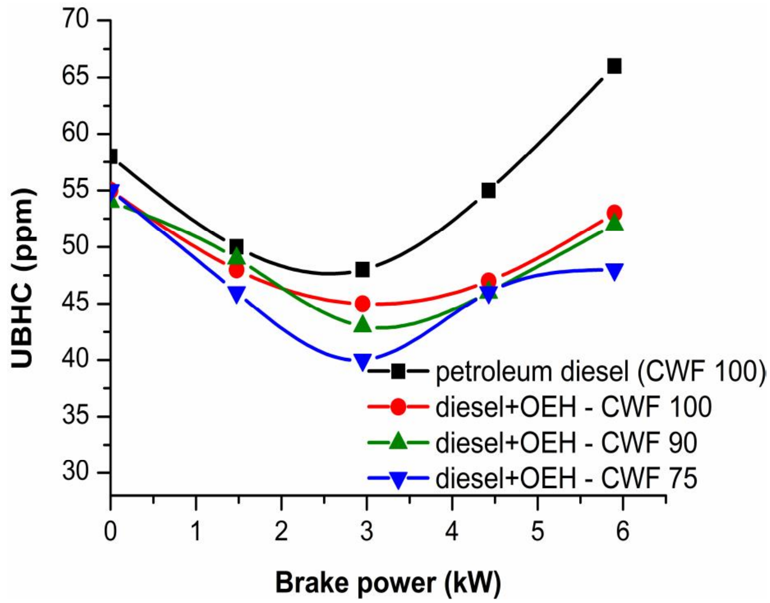

Figure 5.39 Variation of UBHC with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

At CWF75, the minimum UBHC emission was obtained when compared to other flow rates. At CWF75, UBHC emission got decreased by 8.33% and 9.43% compared to CWF90 and CWF100. This might be due to low quenching distance of hydrogen present in the gas mixture along with reduction in heat loss from the combustion chamber to the cooling medium resulting in a thin boundary layer of UBHC on the combustion chamber walls. This presumably led to more complete combustion of the injected fuel.

5.2.3.6 Oxides of nitrogen emission (NO_X)

Figure 5.40 shows the comparison of NO_X emission for petroleum diesel and diesel with OEH gas of 4.6 lpm at cooling water flow rate of CWF75, CWF90, and CWF100.

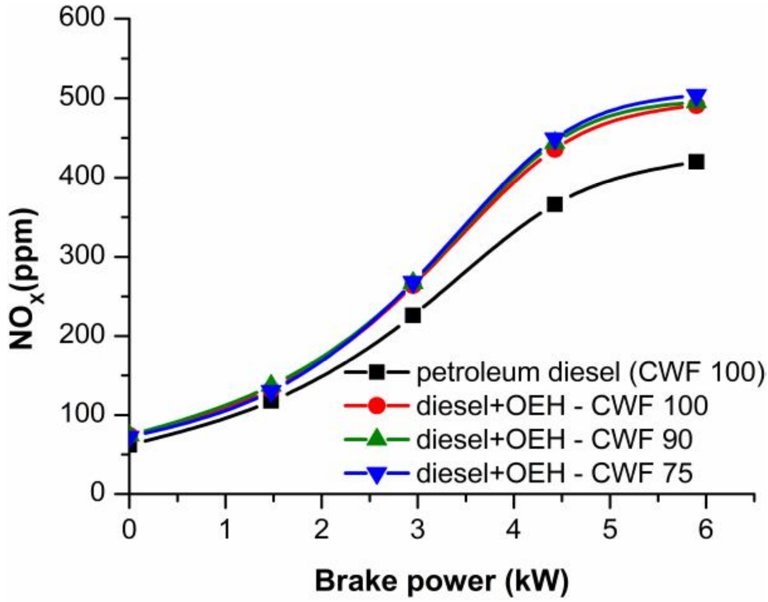

Figure 5.40 Variation of NO_X with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

On examining the graph, it is learned that the NO_X emission increased, when diesel combustion was assisted by OEH gas. When the cooling water flow rate was CWF75, CWF90, and CWF100 at 100% rated load, NO_X emission increased by 20%, 17.86%, and 16.9% respectively compared to base line operation. This increase in NO_X emission was due to instantaneous combustion of OEH gas and high temperature atmosphere prevailing in the combustion chamber. This resulted in enhanced pre-mixed burning phase during the OEH gas associated diesel combustion.

At the cooling water flow rate of CWF100, NO_X emission decreased by 0.81% and 2.58% compared to CWF90 and CWF75. On comparing the NO_X emission during CWF90 and CWF75, it could be observed that CWF90 resulted in decrease in NO_X emission by 1.79%. This might be due to less intensity of combustion compared to CWF75 combustion. The maximum NO_X emission was obtained at CWF75 compared to other flow rates. This might be due to slightly higher peak pressure and hence temperatures resulting from OEH gas addition (Birtas et al 2011) along with CWF75 would have favoured NO_X formation mechanism through extended Zeldovich reactions. This eventually results in enhanced premixed burning phase which is a prime area where the NO_X is produced.

5.2.3.7 Smoke emission

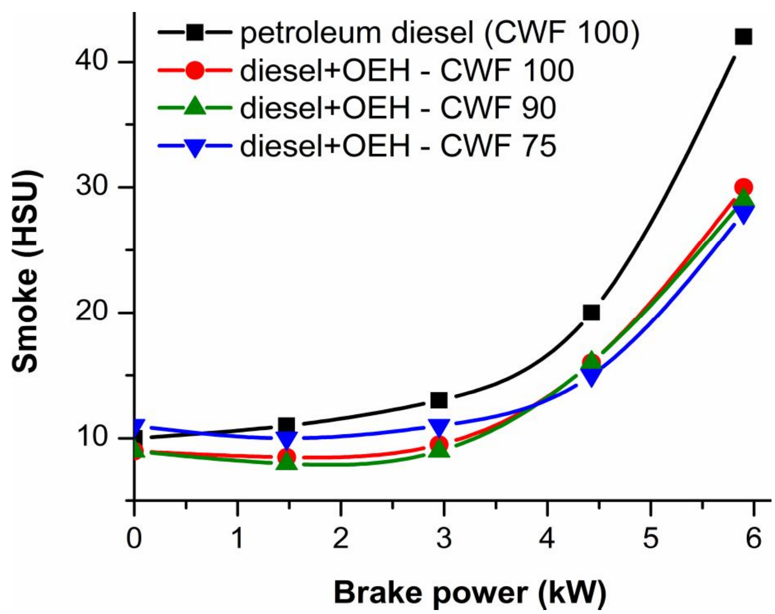

Figure 5.41 Variation of smoke emission with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

Figure 5.41 shows the comparison of smoke emission for petroleum diesel and diesel with OEH of 4.6 lpm at different flow rate of cooling water. The experimental results showed that the smoke emission increased when cooling water flow rate was increased and decreased when cooling water flow rate was decreased. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load of the engine, the smoke emission got decreased by 33.33%, 30.95%, and 28.57% respectively compared to base line operation. This might be due to two reasons: 1. The high burning velocity caused rapid flame propagation in hydrogen combustion engines resulting in an intense convection of the burning gas and a large heat transfer from the burning gas to the combustion chamber walls (Shudo et al 2001). 2. Availability of high concentration of OH radicals generated by chemical reactions between hydrogen and oxygen (Das 1996a). At CWF100, the smoke emission got increased by 3.45% compared to CWF90. Smoke emission got decreased by 3.45% and 6.67% at CWF75 compared to CWF90 and CWF100 respectively. At CWF75, the minimum smoke emission was obtained when compared to other flow rates. This might be due to increase in the transportation rate of flame front which resulted in the creation of high turbulence and high efficiency combustion.

5.2.3.8 Excess oxygen emission

Figure 5.42 shows the comparison of excess oxygen emission for petroleum diesel and diesel with OEH of 4.6 lpm at different flow rate of cooling water. The experimental results showed that the excess oxygen emission increased when cooling water flow rate was increased and decreased when cooling water flow rate was decreased. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load of the engine, the excess oxygen emission got decreased by 8.87%, 10.62%, and 10.18% respectively compared to base line operation. At CWF100, the excess oxygen emission increased by 0.49% compared to CWF90. When compared with CWF100, CWF75 operation emitted more excess oxygen. This might be due to high rate of dissociation of CO_2 into CO and oxygen. The CO emission at this flow rate also confirmed the same.

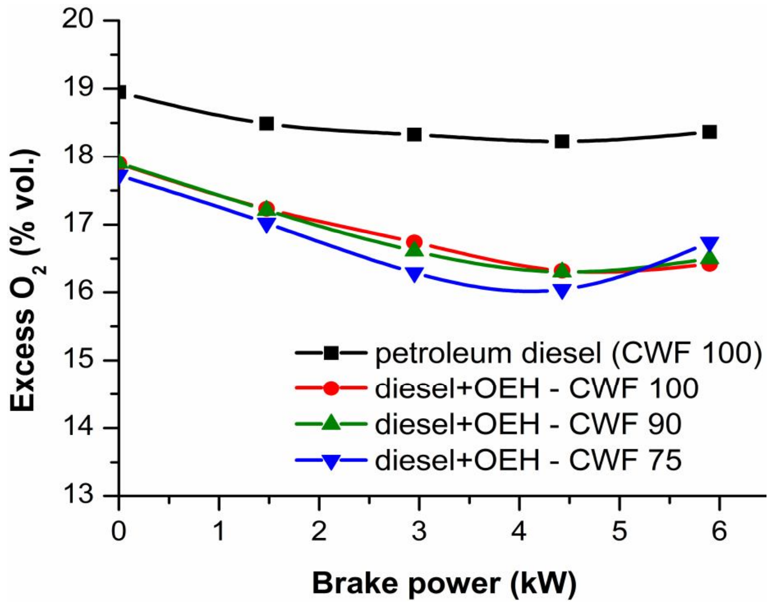

Figure 5.42 Variation of excess oxygen emission with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

On comparing the excess oxygen emission during CWF90 and CWF75 operations, the excess oxygen emission got decreased by 1.43% at CWF90 compared to CWF75. At CWF90, the minimum excess oxygen emission was obtained compared to other flow rates. This might be due to optimum quantity of heat that got transferred from combustion chamber to cooling medium at this flow rate.

5.2.3.9 Exhaust gas temperature (EGT)

Figure 5.43 shows the comparison of EGT for petroleum diesel and diesel with OEH gas of 4.6 lpm at cooling water flow rate of CWF75, CWF90, and CWF100. It is learnt from the graph that the EGT increased when diesel combustion was influenced by OEH gas. When the cooling water flow rate was

CWF75, CWF90, and CWF100 at rated load of the engine, EGT increased by 10%, 6.92%, and 6.41% compared to base line operation.

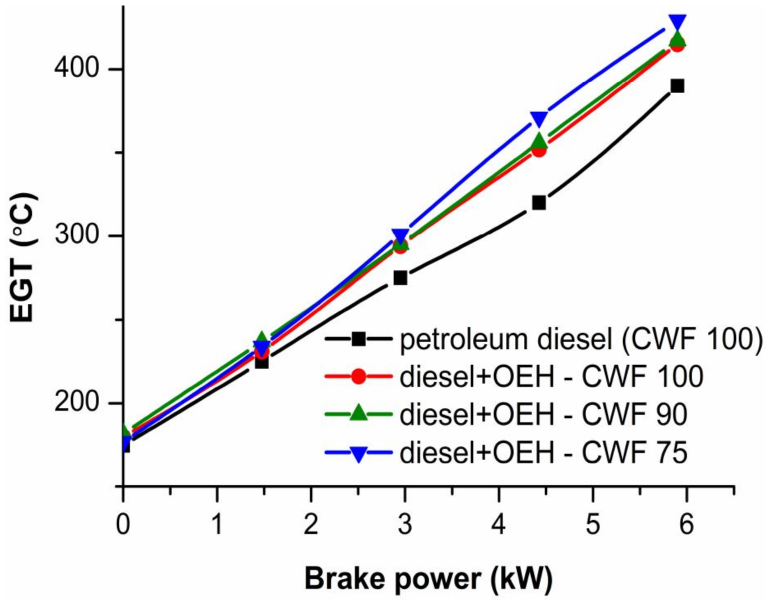

Figure 5.43 Variation of EGT with BP for different flow rates of cooling water with OEH gas of 4.6 lpm

This increase in EGT was due to combined effect of OEH gas and variation in cooling water flow rate. When heavier diesel fuel molecules were fractured into tiny hydrocarbon structures, the surface exposed to air molecules was more. This increased oxidation reactions. This along with prevention of cooling loss from the combustion chamber to the cooling medium by varying the flow rate of cooling water resulted in high temperature combustion. At the cooling water flow rate of CWF100, EGT decreased by 0.48% and 3.26% compared to CWF90 and CWF75.

On comparing the EGT during CWF90 and CWF75, CWF90 resulted in decrease in EGT by 2.8%. This might be due to less intensity of combustion compared to CWF75 combustion. The maximum EGT was obtained at CWF75 compared to other flow rates. This might be due to increase in the flame length which improved the combustion phenomena.

5.2.3.10 Heat release rate (HRR)

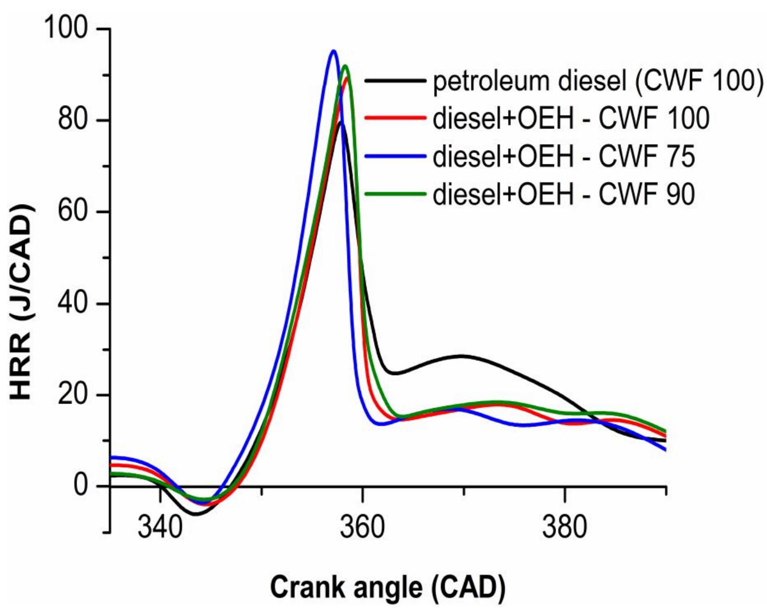

Figure 5.44 Variation of HRR with CA for different flow rates of cooling water with OEH gas of 4.6 lpm at rated load

Figure 5.44 displays the variation of heat release rate with crank angle for OEH gas of 4.6 lpm flow rate with cooling water flow rate of CWF75, CWF90, and CWF100. From the graph, it is known that the heat release rate increased, when diesel combustion was assisted by OEH gas. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load of the engine, heat release rate increased by 21.25%, 16.88%, and 13.75% compared to base line operation. This increase in heat release rate was due to high temperature produced by reduced flow rate of cooling water associated with wide flammability limits of hydrogen which caused more ignition centers during combustion (Sankaranarayanan & Pugazhvadivu 2012).

Owing to enhanced pre-mixed combustion phase, the combustion took place instantaneously. When the pre-mixed combustion phase was enhanced, the temperature and the pressure developed in the combustion process became high. At the cooling water flow rate of CWF100, heat release rate decreased by 2.67% and 6.19% compared to CWF90 and CWF75. On comparing the heat release rate during CWF90 and CWF75, CWF90 resulted in decrease in heat release rate by 3.61%. The maximum heat release rate was obtained at CWF75 compared to other flow rates. This might be due to an increase in the evaporation rate of fuel droplets at this flow rate.

5.2.3.11 In-cylinder pressure

Figure 5.45 exhibits the variation of in-cylinder pressure with crank angle for OEH gas of 4.6 lpm flow rate with cooling water flow rate of CWF75, CWF90, and CWF100. From the diagram, it is learned that the in-cylinder pressure increased, when diesel combustion was assisted by OEH gas. When the cooling water flow rate was CWF75, CWF90, and CWF100 at rated load, peak in-cylinder pressure increased by 12.86%, 7.14%, and 5.71% respectively compared to base line operation. This increase in peak in-cylinder pressure was due to high temperature atmosphere prevailing in the combustion chamber. This was due to high turbulence created by OEH gas in association with reduced flow rate of cooling water.

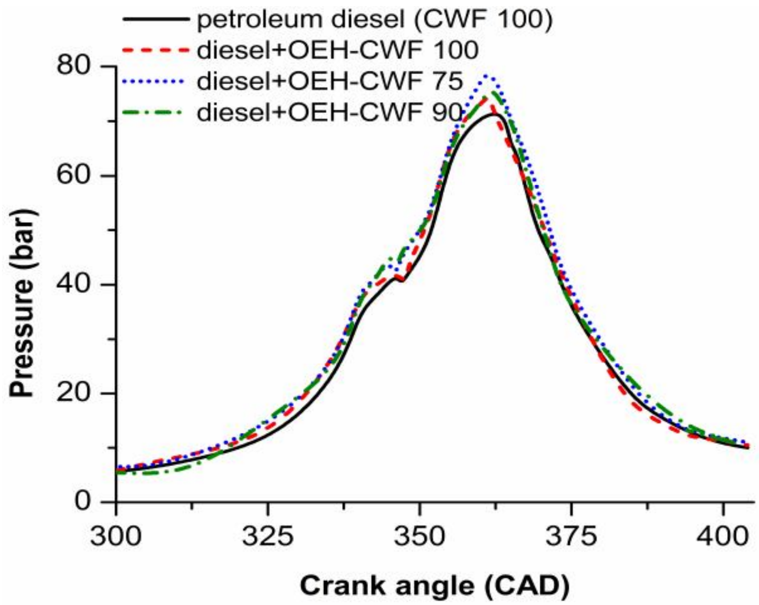

Figure 5.45 Variation of in-cylinder pressure with CA for different flow rates of cooling water with OEH gas of 4.6 lpm at rated load

At CWF100, peak in-cylinder pressure got decreased by 1.33% and 6.33% compared to CWF90 and CWF75. On comparing the peak in-cylinder pressure during CWF90 and CWF75, it could be conceded that CWF90 resulted in a decrease in peak in-cylinder pressure by 5.06%. This was, perhaps, because of the reduction in low level oxidation reactions during ignition delay period. The maximum peak in-cylinder pressure was obtained at CWF75 compared to other flow rates. This might be due to more constant volume combustion resulted due to more homogeneous mixture of fuel and air. Owing to this, the brake thermal efficiency also got increased at this flow rate.

5.2.4 Varied temperatures of diesel fuel

In the present experimental work, the best flow rate OEH gas of 4.6 lpm was aspirated into the cylinder along with intake air at varied temperatures of diesel fuel. The diesel fuel was pre heated by an in-house made heater. In this study diesel fuel was pre heated to 35ºC. It was 10ºC higher than the normal operating temperature of diesel fuel. Due to increase in temperature of diesel fuel, its viscosity got decreased from 2.94 cSt into 2.66 cSt.

5.2.4.1 Brake thermal efficiency (BTE)

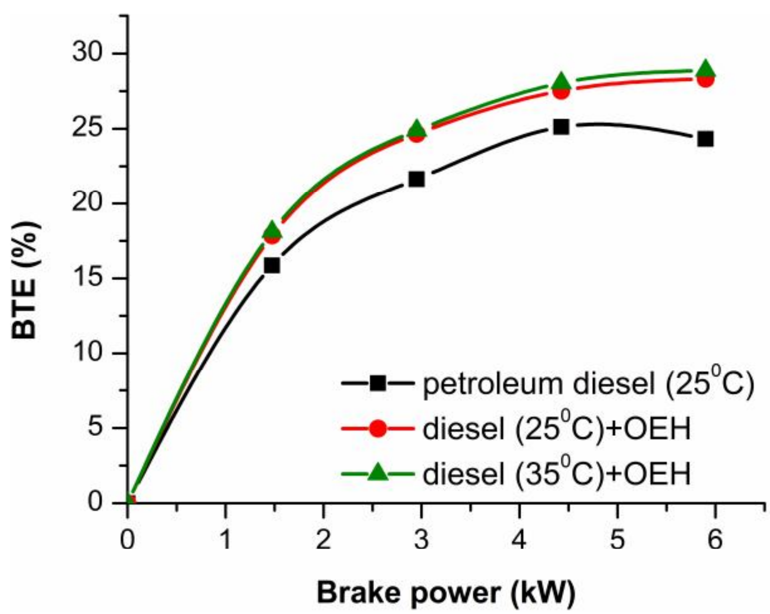

Figure 5.46 Variation of BTE with BP for different temperatures of diesel with OEH gas of 4.6 lpm

Figure 5.46 represents the effectiveness of OEH gas of 4.6 lpm at diesel fuel temperature of 25ºC (FT25) and 35ºC (FT35) at various load conditions of the test engine in regard to the brake thermal efficiency. At rated load of the test engine, OEH gas with FT25 resulted in an increase in brake thermal efficiency by 16.45% compared to base line operation. When OEH gas with FT35 was used at the same rated load condition of the engine, brake thermal efficiency got increased by 18.71%. On analyzing the graph, it is clear that the brake thermal efficiency increased when OEH gas with FT25 and FT35 was used in the engine.

This increase in brake thermal efficiency was due to the combined effect of OEH gas and the warm diesel fuel. When the diesel fuel temperature was increased, very fine sized fuel droplets were obtained at the exit of injector nozzle, which enhanced the intimacy of fuel droplets with air molecules (Shepherd 1982). Also, due to high heating value of hydrogen present in the gas mixture, operation of the hydrogen-fueled engine at the leaner equivalence ratios (Saravanan et al 2008) resulted in an increase in brake thermal efficiency. On comparing the brake thermal efficiency during FT35 and FT25, it could be said that FT25 resulted in a decrease in brake thermal efficiency by 1.94%. This might be due to poor mixing of fuel and air which resulted in lower combustion.

5.2.4.2 Brake specific energy consumption (BSEC)

Figure 5.47 represents the effect of OEH gas of 4.6 lpm with diesel fuel temperature of FT25 and FT35 at various load conditions of the test engine on the BSEC. At rated load, OEH gas with FT25 resulted in a decrease in BSEC by 14.12% compared to base line operation. When OEH gas with FT35 was used at the same rated load condition of the engine, BSEC got decreased by 15.76%. On analyzing the graph, it is clear that the BSEC decreased, when OEH gas with FT25 and FT35 were used in the engine.

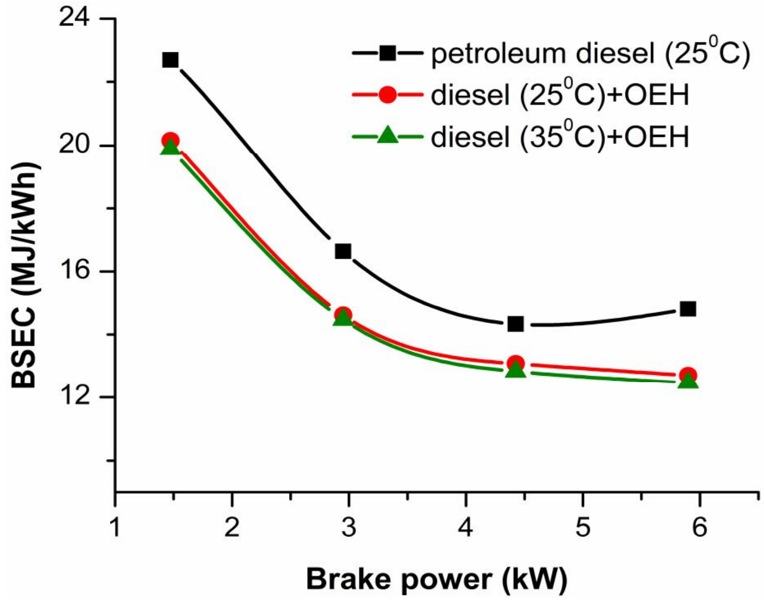

Figure 5.47 Variation of BSEC with BP for different temperatures of diesel with OEH gas of 4.6 lpm

When OEH gas was introduced in the warm diesel combustion process, due to its faster combustion rate and high catalytic nature the energy extracted from the diesel fuel was more than the base line operation which resulted in a decrease in BSEC. On comparing the BSEC during FT35 and FT25 operations, FT25 resulted in an increase in BSEC by 1.91%. This might be due to higher surface tension of diesel fuel than FT35 diesel fuel (Mangalla & Enomoto 2013). This resulted in low efficiency combustion compared to FT35 combustion.

5.2.4.3 Carbon monoxide emission (CO)

Figure 5.48 illustrates the comparison of CO emission for petroleum diesel and OEH gas of 4.6 lpm with diesel fuel temperatures FT25 and FT35.

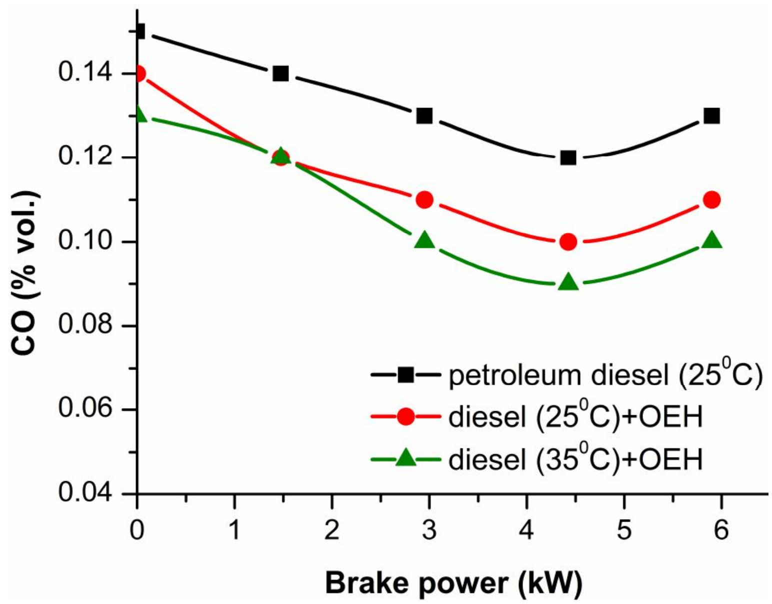

Figure 5.48 Variation of CO with BP for different temperatures of diesel with OEH gas of 4.6 lpm

The experimental results showed that the CO emission increased when the diesel fuel temperature was decreased and decreased when diesel fuel temperature was increased. At rated load of the test engine, OEH gas with FT25 resulted in a decrease of CO emission by 15.38% compared to base line operation. At FT35, the reduction was 23.08%. This decrease in CO emission was due to decrease in sauter mean diameter of fuel droplets which enhanced the evaporation rate of fuel-air mixture (Huong et al 2010). This with high burning rate of hydrogen present in the gas mixture enhanced the combustion phenomena thereby reducing the CO emission. When the test engine was operated at FT25, CO emission got increased by 9.09% compared to FT35 operation. This might be due to less intense combustion because of bigger sized droplets of fuel available at the exit of injector nozzle.

5.2.4.4 Carbon dioxide emission (CO_2)

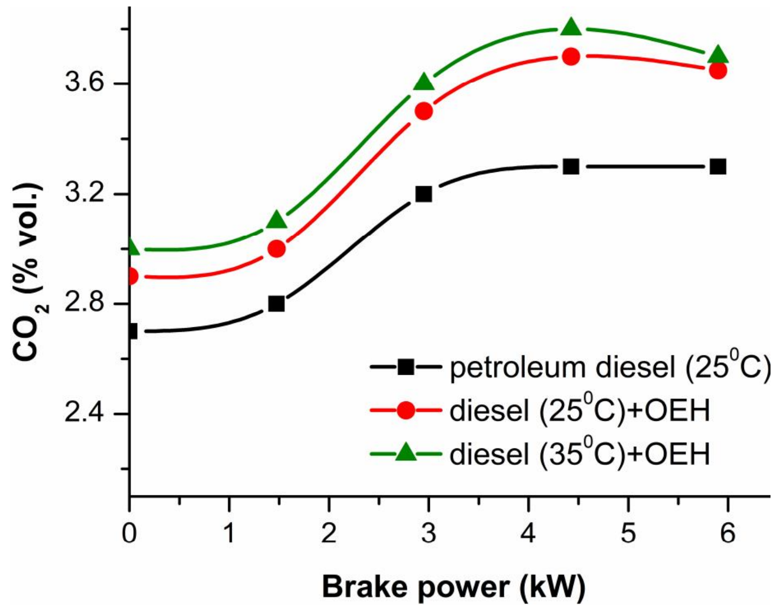

Figure 5.49 Variation of CO_2 with BP for different temperatures of diesel with OEH gas of 4.6 lpm

Figure 5.49 shows the effect of OEH gas of 4.6 lpm with diesel fuel temperature of FT25 and FT35 at various load conditions of the test engine on the CO_2 emission. At rated load of the test engine, OEH gas with FT25 and FT35 resulted in an increase of CO_2 emission by 10.61% and 12.12% compared to base line operation. On analyzing the graph, it is evident that the CO_2 emission got increased, when OEH gas with FT25 and FT35 was used in the engine. This increase in CO_2 emission was due to wider spray cone angle because of warm diesel fuel and high oxygen index of OEH gas. This resulted in high conversion rate of carbon molecules into CO_2 molecules. On comparing the CO_2 emission during FT35 and FT25, FT25 resulted in a decrease in CO_2 emission by 1.37%.

This might be due to poorer spray characteristics at this temperature of diesel fuel compared to FT35 diesel fuel.

5.2.4.5 Unburned hydrocarbon emission (UBHC)

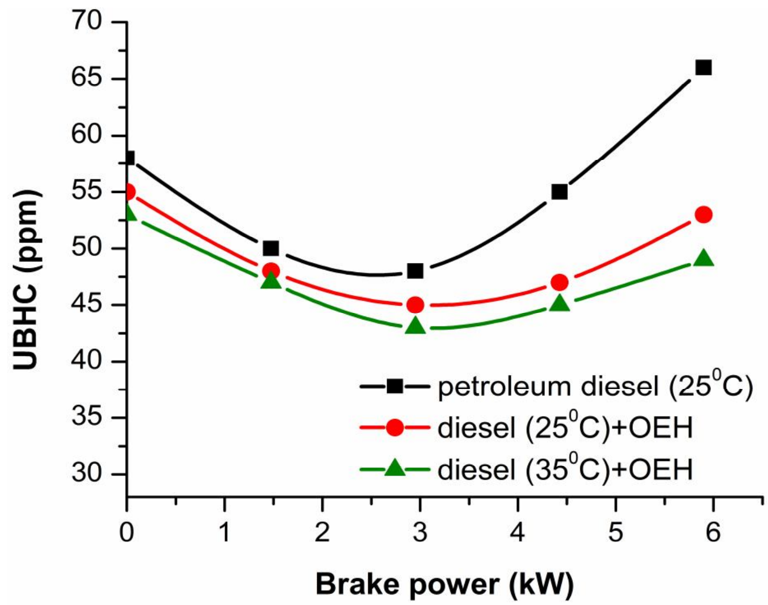

Figure 5.50 Variation of UBHC with BP for different temperatures of diesel with OEH gas of 4.6 lpm

Figure 5.50 shows the comparison of UBHC emission for petroleum diesel and OEH gas of 4.6 lpm with diesel fuel temperatures FT25 and FT35. The experimental results showed that the UBHC emission decreased when the diesel fuel temperature was increased and increased when diesel fuel temperature was decreased. At rated load of the test engine, OEH gas with FT25 resulted in a decline in UBHC emission by 19.7% compared to base line operation. At FT35, it got reduced by 25.76%. This decrease in UBHC emission was due to twofold effect of OEH gas and the warm diesel fuel. When the diesel fuel temperature was increased, it weakened the chemical bonds existing in the structure of the fuel. This along with the tendency of increasing the chain reactions of OEH gas due to its stimulant nature boosted the combustion phenomena and reduced the UBHC emission caused by the crevice effect (Ji et al 2012, Wang et al 2011). When the test engine was operated at FT25, UBHC emission got increased by 8.16% compared to FT35 operation. This might be due to less intense combustion because of high viscosity of diesel fuel when compared with FT35 (Huong et al 2010).

5.2.4.6 Oxides of nitrogen emission (NO_X)

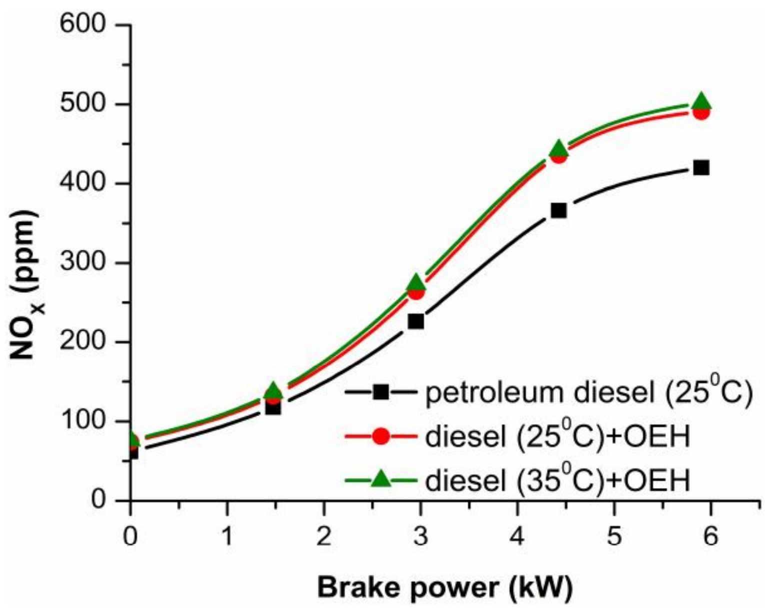

Figure 5.51 Variation of NO_X with BP for different temperatures of diesel with OEH gas of 4.6 lpm

Figure 5.51 shows the effect of OEH gas of 4.6 lpm with diesel fuel temperature of FT25 and FT35 at various load conditions of the test engine on the

NO_X emission. At rated load of the test engine, OEH gas with FT25 and FT35 resulted in an increase of NO_X emission by 16.9% and 18.71% compared to base line operation. On analyzing the graph, it is obvious that the NO_X emission increased, when OEH gas with FT25 and FT35 was used in the engine. When the diesel fuel temperature was increased, the low oxidation reactions during ignition delay period got increased. This along with high diffusion index of OEH gas created more ignition centers during combustion. This resulted in an increase in the combustion temperature and created the atmosphere which favored the formation of NO_X. On comparing the NO_X emission during FT35 and FT25, it is clear that FT35 resulted in an increase of NO_X emission by 2.23%. This might be due to enhanced pre-mixed combustion at FT35. The heat release rate curve at this condition also confirmed the same.

5.2.4.7 Smoke emission

Figure 5.52 represents the effectiveness of OEH gas of 4.6 lpm with diesel fuel temperature of FT25 and FT35 at various load conditions of the test engine on the smoke emission. At rated load of the test engine, OEH gas with FT25 resulted in a decrease in smoke emission by 28.57% compared to base line operation. When OEH gas with FT35 was used at the same rated load condition of the engine, smoke emission got decreased by 33.33%. From the graph, it is apparent that the smoke emission got decreased, when OEH gas with FT25 and FT35 was used in the engine. This decrease in smoke emission was due to an increase in the ratio of H/C. This with an increase in the concentration of smaller sized droplets of fuel resulted in an increase in flame area during combustion. On comparing the smoke emission during FT35 and FT25 combustions, FT25 resulted in an increase in smoke emission by 6.67%. This might be due to lower overall H/C ratio than FT35 operation. This eventually resulted in a low adiabatic flame temperature compared to FT35 combustion. Owing to this the BSEC also got increased at this condition.

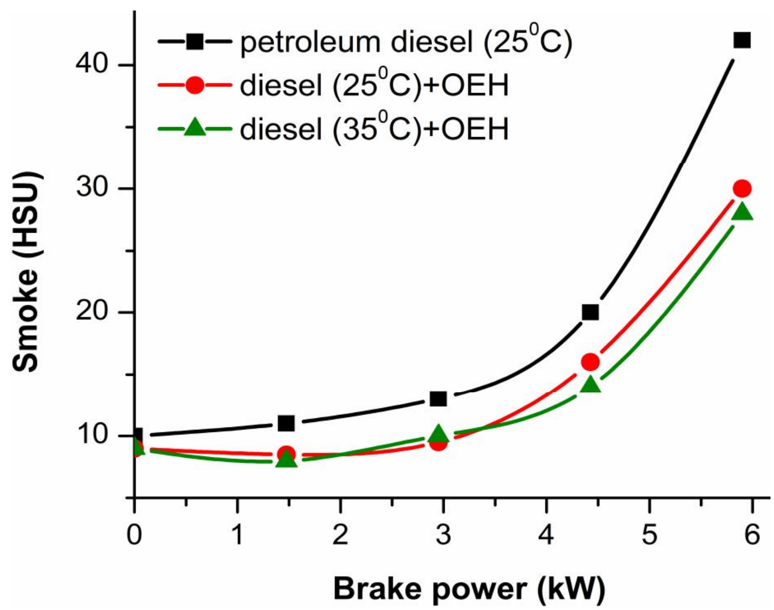

Figure 5.52 Variation of smoke emission with BP for different temperatures of diesel with OEH gas of 4.6 lpm

5.2.4.8 Excess oxygen emission

Figure 5.53 shows the comparison of excess oxygen emission for petroleum diesel and OEH gas of 4.6 lpm with diesel fuel temperatures FT25 and FT35. The experimental results showed that the excess oxygen emission increased when the diesel fuel temperature was decreased and decreased when diesel fuel temperature was increased. At rated load of the test engine, OEH gas with FT25 resulted in a decrease of excess oxygen emission by 8.87% compared to base line operation. At FT35, the reduction was 9.2%. This decrease in excess oxygen emission was due to increase in lower oxidation rates during ignition delay period and the low flame quenching distance of hydrogen present in the gas mixture. This decreased the unburned fuel in the quenching layer formed on the combustion chamber wall.

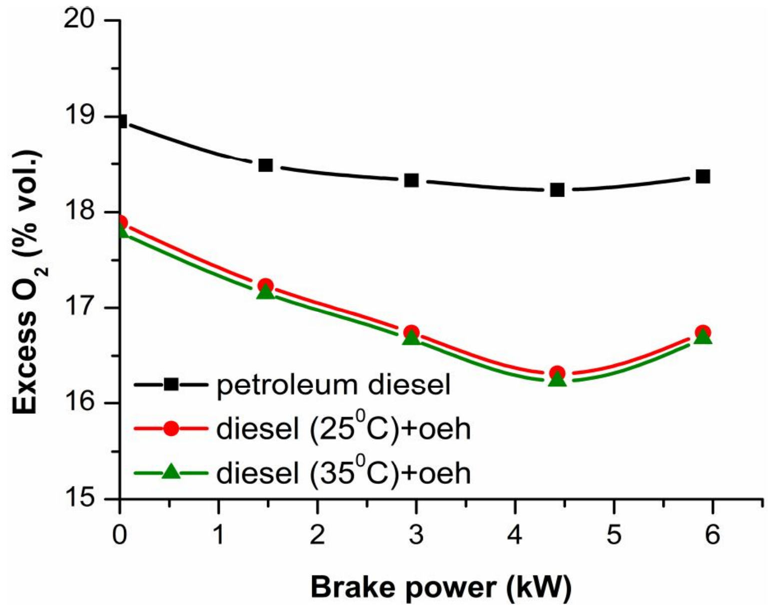

Figure 5.53 Variation of excess oxygen emission with BP for different temperatures of diesel with OEH gas of 4.6 lpm

This enabled the flame to propagate into the top land crevice and burn the fuel in the crevice (Huang et al 2006). These facts resulted in reduction of excess oxygen emission in the exhaust of the engine. When the test engine was operated at FT25, the excess oxygen emission got increased by 0.36% compared to FT35 operation.

5.2.4.9 Exhaust gas temperature (EGT)

Figure 5.54 shows the comparison of EGT for petroleum diesel and diesel with OEH gas of 4.6 lpm at varied diesel fuel temperature of FT25 and FT35 at various load conditions of the test engine. At maximum load of the test engine, OEH gas with FT25 and FT35 resulted in an increase of EGT by 6.41% and 9.23% compared to base line operation. On analyzing the graph, it is obvious that the EGT increased when OEH gas with FT25 and FT35 were used in the engine.

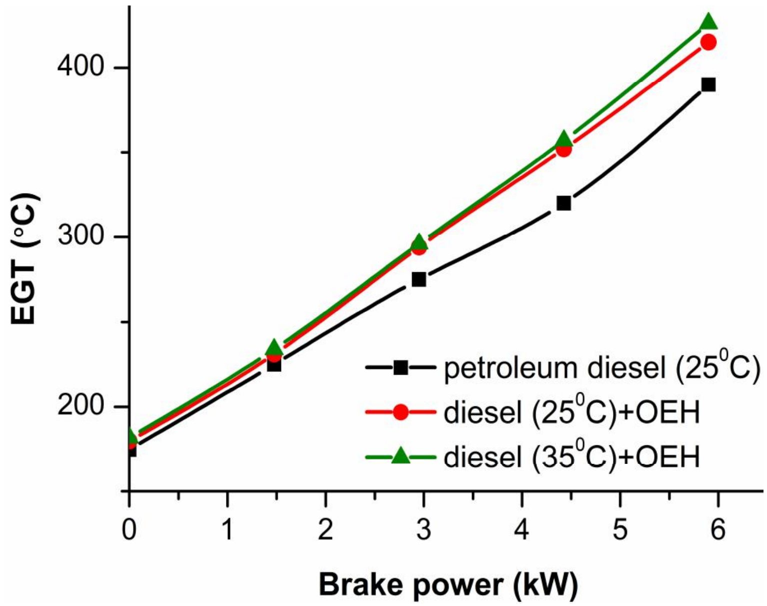

Figure 5.54 Variation of EGT with BP for different temperatures of diesel with OEH gas of 4.6 lpm

This increase in EGT was due to high temperatures developed during the combustion of OEH gas assisted warm diesel fuel combustion. When the OEH gas was ignited by pilot diesel fuel, the combustion was spontaneous, this developed high pressure and temperature in the combustion process. On comparing the EGT during FT35 and FT25, FT25 resulted in a decrease in EGT by 2.58%. This might be due to smaller spray cone angle resulted during FT25 operation than FT35.

5.2.4.10 Heat release rate (HRR)

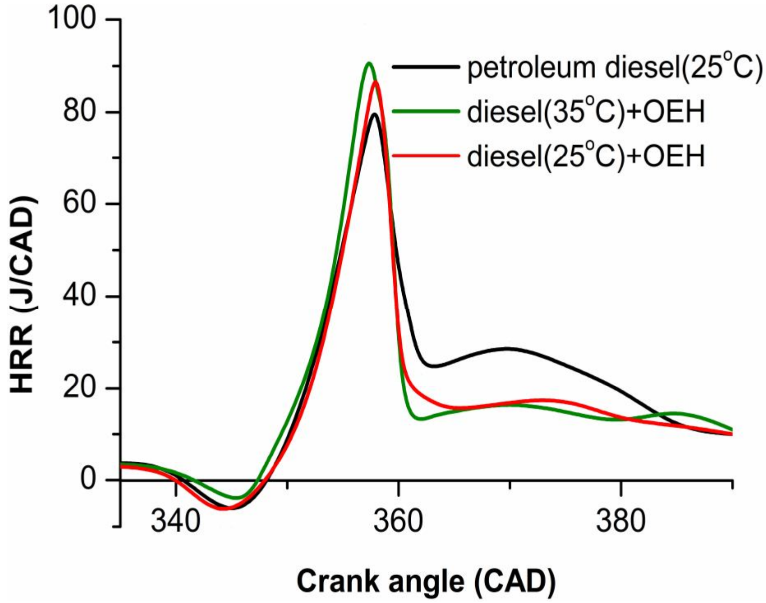

Figure 5.55 Variation of HRR with CA for different temperatures of diesel with OEH gas of 4.6 lpm at rated load

Figure 5.55 shows the effect of OEH gas of 4.6 lpm with diesel fuel temperatures of FT25 and FT35 at various load conditions of the test engine on the heat release rate with crank angle. At rated load of the test engine, OEH gas with FT25 and FT35 resulted in an increase of heat release rate by 13.75% and 16.25% compared to base line operation. On studying the graph, it is apparent that the heat release rate increased, when OEH gas with FT25 and FT35 were used in the engine operation. This increase in heat release rate was due to dual effect of OEH gas and the warm diesel fuel. When the diesel fuel temperature was increased, the rate of low level molecular collisions between the fuel and air got improved. This along with impulsive combustion of OEH gas due to its high diffusivity increased the combustion temperature and the pre-mixed combustion phase. On evaluating the heat release rate during FT35 and FT25, FT25 resulted in decrease in heat release rate by 2.15%. This might be due to low intense combustion due to less chain reactions than FT35 combustion.

5.2.4.11 In-cylinder pressure

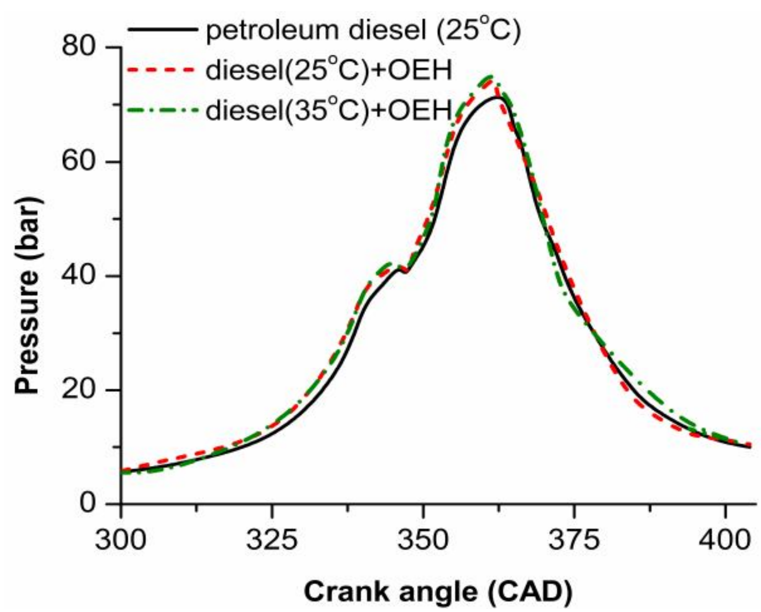

Figure 5.56 Variation of in-cylinder pressure with CA for different temperatures of diesel with OEH gas of 4.6 lpm at rated load

Figure 5.56 shows the effect of OEH gas of 4.6 lpm with diesel fuel temperatures of FT25 and FT35 at rated load of the test engine on the in-cylinder pressure. At rated load of the test engine, OEH gas with FT25 and FT35 caused an increase in peak in-cylinder pressure by 5.71% and 6.43% compared to the base line operation. On studying the graph, it was apparent that the peak in-cylinder pressure increased, when OEH gas with FT25 and FT35 was used in the engine operation.

This increase in peak in-cylinder pressure was due to higher heat release rates developed during OEH gas assisted warm diesel fuel combustion. The heat release rate is the energy conversion in the cylinder (Bunes & Einang 2000). On evaluating the peak in-cylinder pressure during FT35 and FT25, it was observed that FT25 caused a decrease in peak in-cylinder pressure by 0.67%. This might be due to lower energy conversion rate at this operating condition than FT35. The heat release curve at this operating condition also confirmed this.

5.2.5 Varied temperatures of inlet air

In the present experimental work, the best flow rate OEH gas of 4.6 lpm was aspirated into the cylinder along with change in inlet air temperatures. The temperatures were varied from operating temperature of 30ºC to 35ºC and to 25ºC. The tests were carried out by operating the engine at three different climatic conditions.

5.2.5.1 Brake thermal efficiency (BTE)

Figure 5.57 displays the variation of brake thermal efficiency with brake power for OEH gas of 4.6 lpm flow rate with inlet air temperatures of 35ºC (IAT35), 30ºC (IAT30), and 25ºC (IAT25). From the graph, it could be concluded that the brake thermal efficiency increased, when the combustion process was influenced by OEH gas when the inlet air temperatures were IAT35, IAT30, and IAT25. When the engine was operated at rated load and at IAT35, IAT30, and IAT25 with OEH gas, the brake thermal efficiency increased by 16.37%, 16.45%, and 17.02% respectively compared to base line operation.

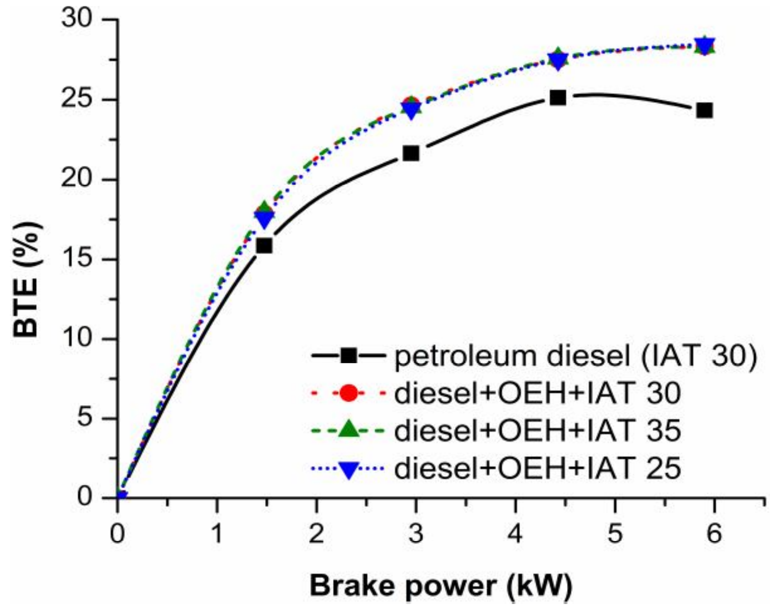

Figure 5.57 Variation of BTE with BP for different inlet air temperatures with OEH gas of 4.6 lpm

This increase in brake thermal efficiency was due to the dual effect of OEH gas and the change in inlet air temperatures. The change in inlet air temperature affected the ignition delay due to its effect on overall charge conditions during the ignition delay period. When the temperature of inlet charge air was increased, ignition delay period got decreased. This is because of higher inlet air temperature reducing the time to vaporize the fuel to make a combustible mixture (Alam et al 2005). Heat release rate diagram shown in Figure 5.65 at these conditions evidenced this. When comparing with other conditions, IAT25 resulted in a higher brake thermal efficiency. This might be due to an increase in the mass of air inducted into the combustion process at this condition. It resulted in an increase in overall oxygen concentration of fuel-air mixture and effected higher brake thermal efficiency.

5.2.5.2 Brake specific energy consumption (BSEC)

Figure 5.58 presents the variation of BSEC with brake power for OEH gas of 4.6 lpm flow rate with inlet air temperatures of IAT35, IAT30, and IAT25. From the graph, it is obvious that the BSEC decreased, when the diesel combustion process was assisted by OEH gas. When the inlet air temperatures were IAT35, IAT30, and IAT25 at rated load of the engine, the BSEC got decreased by 14.06%, 14.12%, and 14.55% compared to base line operation.

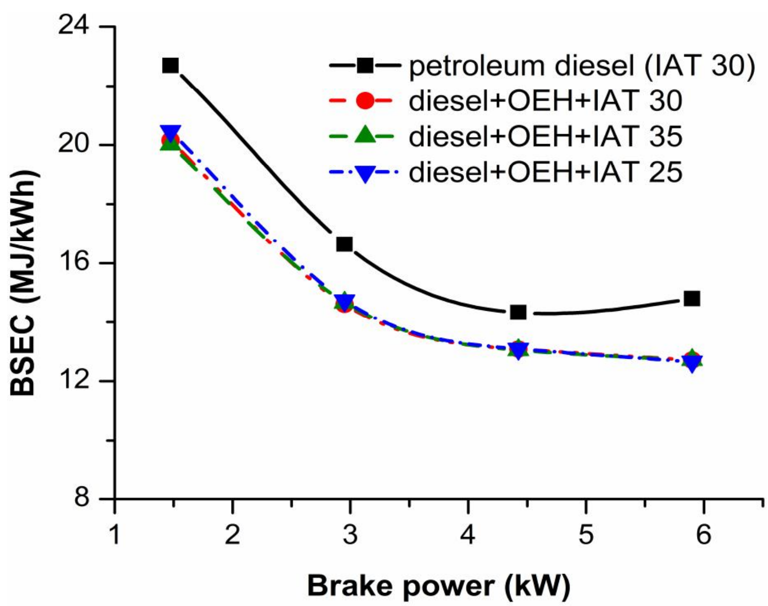

Figure 5.58 Variation of BSEC with BP for different inlet air temperatures with OEH gas of 4.6 lpm

This decrease in BSEC was due to effective utilization of energy available in the diesel fuel. This might be due to spontaneous combustion of OEH gas along with increase in penetrating length of fuel spray due to increase in inlet air temperatures (Leick et al 2007). When the test engine was operated at IAT35, it resulted in 0.57% increase in BSEC compared to IAT25 and when the test engine was operated at IAT30, it resulted in 0.49% increase in BSEC compared to IAT25 operation. This might be due to low density of inlet air inducted during these operations.

5.2.5.3 Carbon monoxide emission (CO)

Figure 5.59 shows the comparison of CO emission for petroleum diesel and OEH gas of 4.6 lpm with inlet air temperatures of IAT35, IAT30, and IAT25. The experimental results showed that the CO emission decreased when the diesel combustion was influenced by OEH gas with IAT35, IAT30, and IAT25.

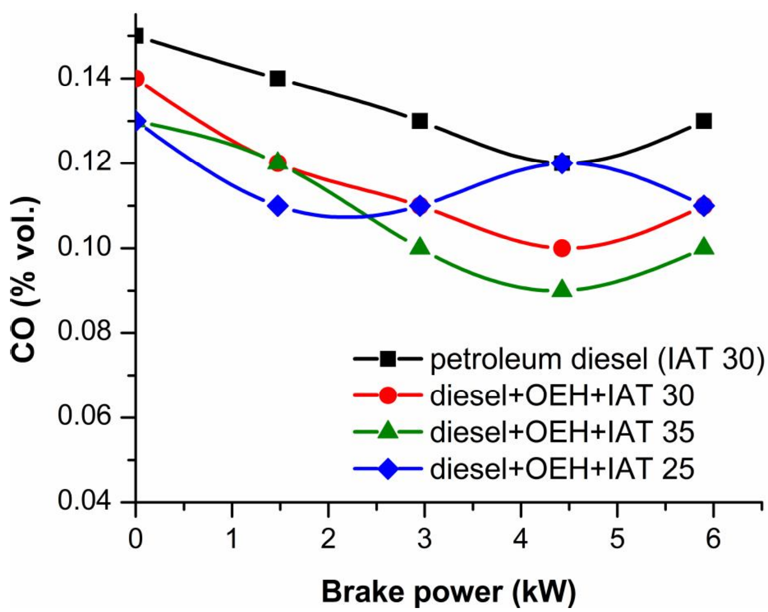

Figure 5.59 Variation of CO with BP for different inlet air temperatures with OEH gas of 4.6 lpm

At rated load of the test engine, OEH gas with IAT35, IAT30, and IAT25 resulted in a decrease in CO emission by 23.08%, 15.38%, and 15.38% respectively when compared to base line operation. This decrease in CO emission was due to the combined effect of OEH gas and the change in inlet charge temperature. When the inlet charge temperature was increased along with OEH gas, the rate of chain reactions got increased which in turn along with high diffusivity of hydrogen presented in the gas mixture reduced the percentage of heterogeneous mixture of fuel and air. When this fuel mixture got ignited, the combustion was of higher efficiency. At the inlet air temperature of IAT25, CO emission increased by 10% compared to IAT35. This might be due to increase in the aerodynamic drag on the droplets (Greco, 2008).

5.2.5.4 Carbon dioxide emission (CO_2)

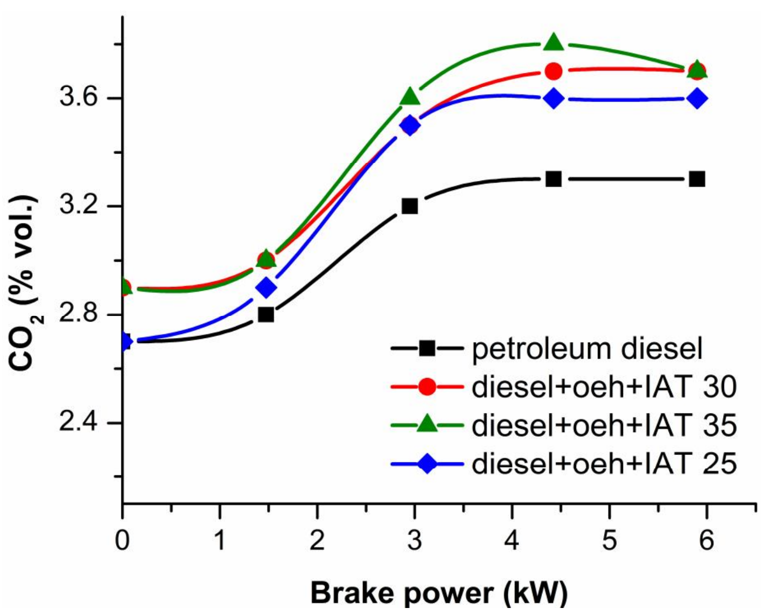

Figure 5.60 Variation of CO_2 with BP for different inlet air temperatures with OEH gas of 4.6 lpm

Figure 5.60 exhibits the variation of CO_2 emission for OEH gas of 4.6 lpm flow rate with inlet air temperatures of IAT35, IAT30, and IAT25. From the graph, it is concluded that the CO_2 emission increased, when the diesel combustion process was assisted by OEH gas. When the inlet air temperatures were IAT35, IAT30, and IAT25 at rated load of the engine, CO_2 emission got increased by 12.12%, 12.12%, and 9.09% compared to base line operation. This increase in CO_2 emission was due to high oxygen index of overall fuel-air mixture and increase in combustion rate of fuel-air mixture due to the presence of hydrogen in the gas mixture. At IAT25, CO_2 emission decreased by 2.7% compared to IAT35 and IAT30. This might be due to low degree combustion. The heat release rate curve at this load condition also confirmed this.

5.2.5.5 Unburned hydrocarbon emission (UBHC)

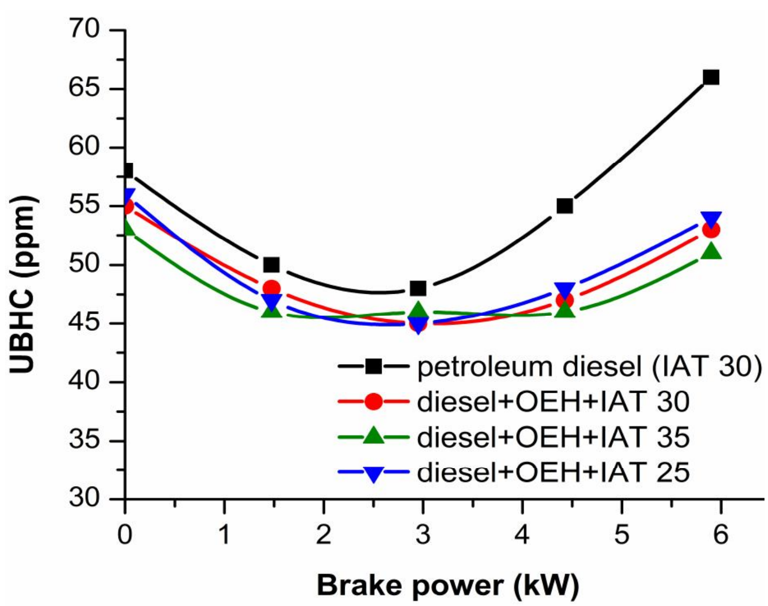

Figure 5.61 Variation of UBHC with BP for different inlet air temperatures with OEH gas of 4.6 lpm

Figure 5.61 illustrates the comparison of UBHC emission for petroleum diesel and OEH gas of 4.6 lpm with inlet air temperatures of IAT35, IAT30, and IAT25. The experimental results showed that the UBHC emission decreased when the diesel combustion was influenced by OEH gas with IAT35, IAT30, and IAT25. At the rated load of the test engine, OEH gas with IAT35, IAT30, and IAT25 resulted in a decrease in UBHC emission by 22.73%, 19.7%, and 18.18% respectively when compared to base line operation.

This might be due to the reduction in droplets mean diameter and increase of droplets surface contact with air. This made much vaporization rate of fuel droplets and formed the homogeneous mixture of fuel and air. When this fuel mixture got ignited, the combustion was of more constant volume combustion resulting in efficient combustion. At the inlet air temperature of IAT25, UBHC emission got raised by 5.88% compared to IAT35. This might be due to reduced pre-flame reactions during ignition delay period (Themel et al 1998).

5.2.5.6 Oxides of nitrogen emission (NO_X)

Figure 5.62 displays the variation of NO_X emission for OEH gas of 4.6 lpm of flow rate with the inlet air temperatures of IAT35, IAT30, and IAT25. From the graph, it is concluded that the NO_X emission increased, when the diesel combustion process was influenced by OEH gas. When the inlet air temperatures were IAT35, IAT30, and IAT25 at rated load, NO_X emission increased by 18.1%, 16.9%, and 15.48% respectively compared to base line operation.

This increase in NO_X emission was due to increase in average cylinder temperature (Bazari & French 1993). It resulted as a corollary of enhanced pre-mixed burning phase and more oxygen concentration of overall fuel mixture.

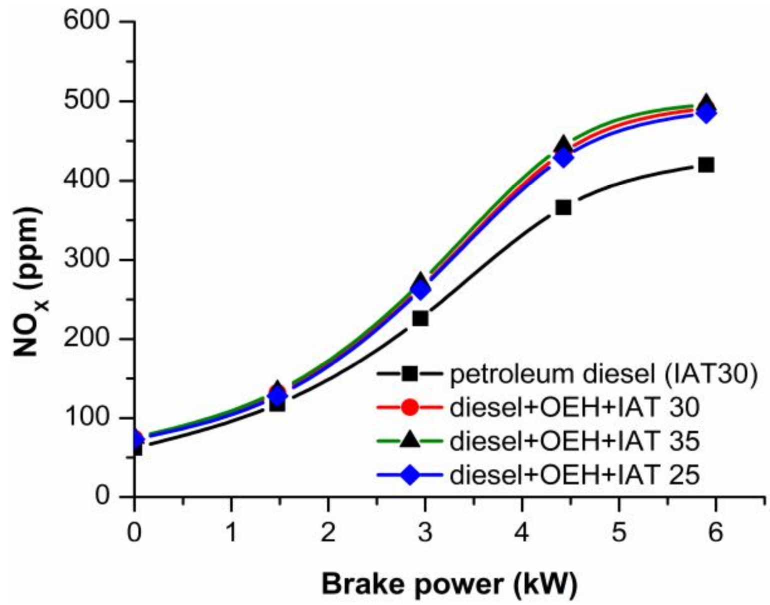

Figure 5.62 Variation of NO_X with BP for different inlet air temperatures with OEH gas of 4.6 lpm

When these factors existed in the combustion process, according to extended Zeldovich thermal NO mechanism, the NO_X formation was more. At IAT35, the NO_X emission increased by 1.02% and 2.27% compared to IAT30 and IAT25. This might be due to higher intake temperatures resulting in faster fuel combustion and shorter combustion durations (Masood et al 2007). Also, the NO_X emission is Arrhenius dependence on temperature of inlet air condition (Naber & Siebers 1998).

5.2.5.7 Smoke emission

Figure 5.63 exhibits the variation of smoke emission with brake power for petroleum diesel and diesel with OEH gas of 4.6 lpm flow rate with inlet air temperatures of IAT35, IAT30, and IAT25. It is evident from the graph that the smoke emission got decreased when the diesel combustion process was backed by

OEH gas. When the inlet air temperatures were IAT35, IAT30, and IAT25 at rated load of the engine, the smoke emission got decreased by 30.95%, 28.57%, and 23.81% respectively compared to base line combustion. This decrease in smoke emission was due to fissuring of heavier diesel fuel molecule structure into lighter and smaller hydrocarbon structure by OEH gas and increase in vaporizing rate of fuel droplets due to wider flame area.

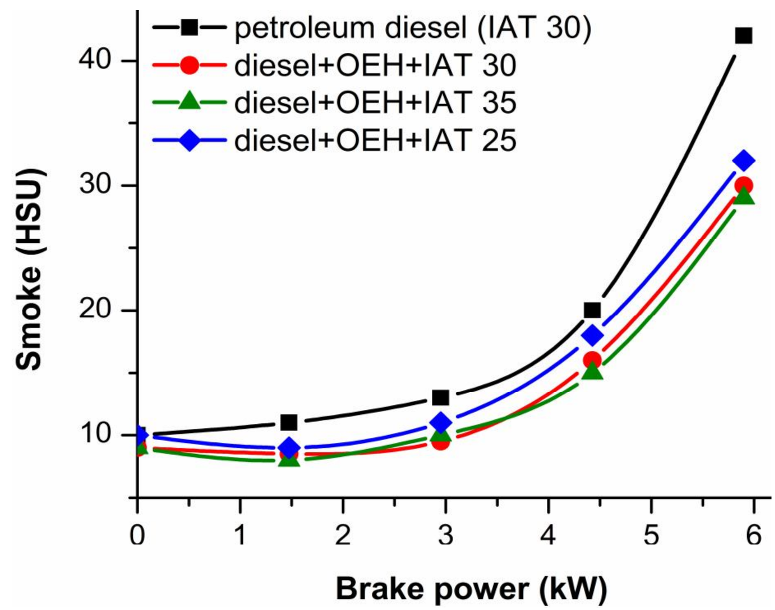

Figure 5.63 Variation of smoke emission with BP for different inlet air temperatures with OEH gas of 4.6 lpm

When the test engine was operated at IAT25, it resulted in 6.67% and 10.34% increase in smoke emission compared to IAT30 and IAT35 operations. This is because, at IAT25, the overall charge temperature got deprived which might lead to formation of more heterogeneous mixture in the combustion process and little longer diffusion mode of combustion compared to IAT30 and IAT35 combustion operations. The heat release curve of this configuration also confirmed this. When the inlet air temperature was IAT35, the smoke emission got decreased by 3.33% compared to IAT30.

5.2.5.8 Excess oxygen emission (O_2)

Figure 5.64 shows the comparison of excess oxygen emission for petroleum diesel and OEH gas of 4.6 lpm with inlet air temperatures of IAT35, IAT30, and IAT25. The experimental results showed that the excess oxygen emission decreased when the diesel combustion was influenced by OEH gas with IAT35, IAT30, and IAT25.

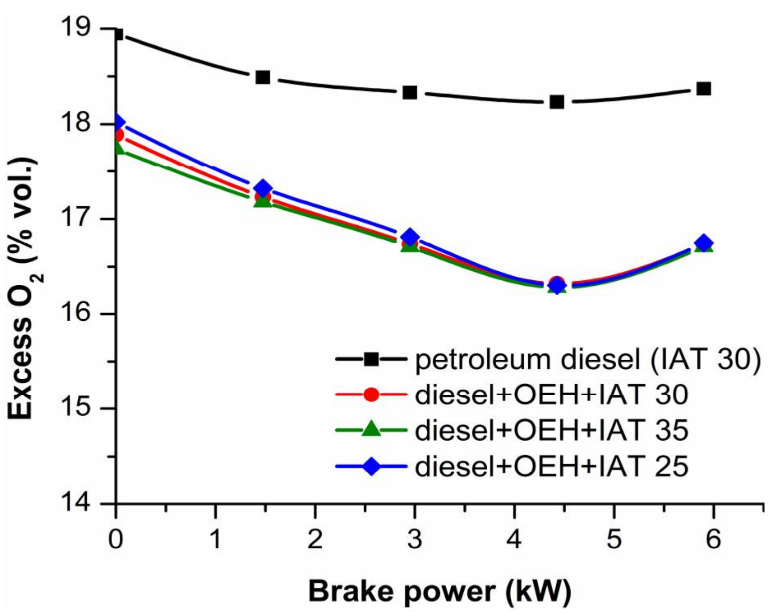

Figure 5.64 Variation of excess oxygen emission with BP for different inlet air temperatures with OEH gas of 4.6 lpm

At rated load of the test engine, OEH gas with IAT35, IAT30, and IAT25 resulted in a decrease in excess oxygen emission by 9.04%, 8.87%, and

8.82% respectively when compared to base line operation. This decrease in excess oxygen emission was due to combined effect of OEH gas and the change in inlet charge temperature. When the inlet air charge temperature was increased, it resulted in an increase in vapour propagation rate. This with high impulsive combustion nature of hydrogen present in the gas mixture enhanced the combustion of diesel fuel. At the inlet air charge temperature of IAT25, the excess oxygen emission got increased by 0.24% compared to IAT35. This might be due to reduction in the rate of molecular activity between the fuel and the air. This resulted in less powerful combustion when compared to IAT35 combustion.

5.2.5.9 Exhaust gas temperature (EGT)

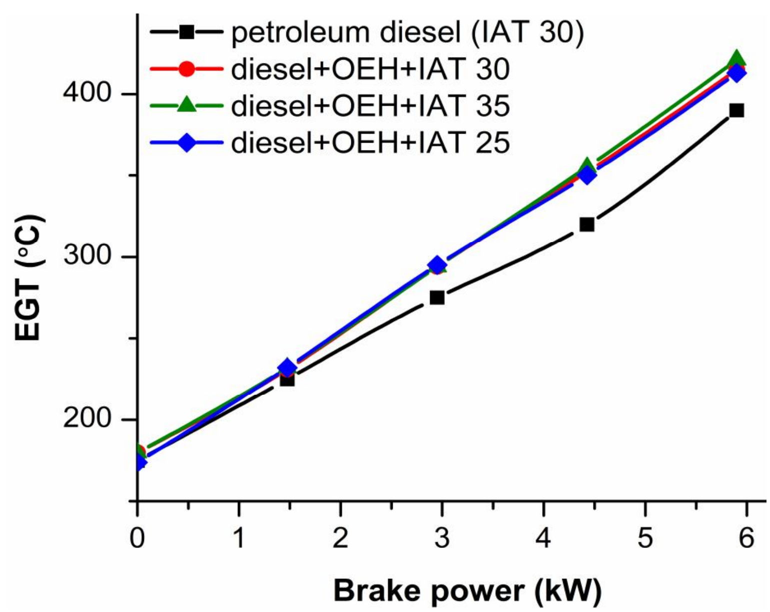

Figure 5.65 Variation of EGT with BP for different inlet air temperatures with OEH gas of 4.6 lpm

Figure 5.65 depicts the comparison of EGT for petroleum diesel and diesel with OEH gas of 4.6 lpm at varied inlet air temperatures of IAT35, IAT30, and IAT25. From the diagram, it is accomplished that the EGT increased, when the diesel combustion process was influenced by OEH gas. When the inlet air temperatures were IAT35, IAT30 and IAT25 at rated load of the engine, EGT increased by 7.95%, 6.41% and 5.9% compared to base line operation. This increase in EGT was due to overall increase in the average temperature of combustible mixture and more pre-mixed combustion due to presence of hydrogen in the gas mixture.

At IAT35, the EGT increased by 1.45% compared to IAT30. At IAT25, EGT decreased by 1.9% and 0.48% compared to IAT35 and IAT30. This might be due to low temperature atmosphere prevailing in the combustion chamber at this condition. This also resulted in increase in CO emission at this condition.

5.2.5.10 Heat release rate (HRR)

Figure 5.66 compares the variation of heat release rate with crank angle for petroleum diesel and diesel with OEH gas of 4.6 lpm of flow rate with inlet air temperatures of IAT35, IAT30, and IAT25. From the graph, it is observed that the heat release rate increased, when the diesel combustion process was influenced by OEH gas. When the inlet air temperatures were IAT35, IAT30, and IAT25 at rated load of the engine, heat release rate increased by 15.63%, 13.75%, and 12.5% respectively compared to base line operation. This might be due to faster combustion rates and heightened pre-mixed burning phase resulted as a consequence of spontaneous combustion of OEH gas with change in inlet charge temperatures. When the pre-mixed combustion phase got increased, it resulted in higher in-cylinder pressure and temperature in addition to higher NO_X emissions.

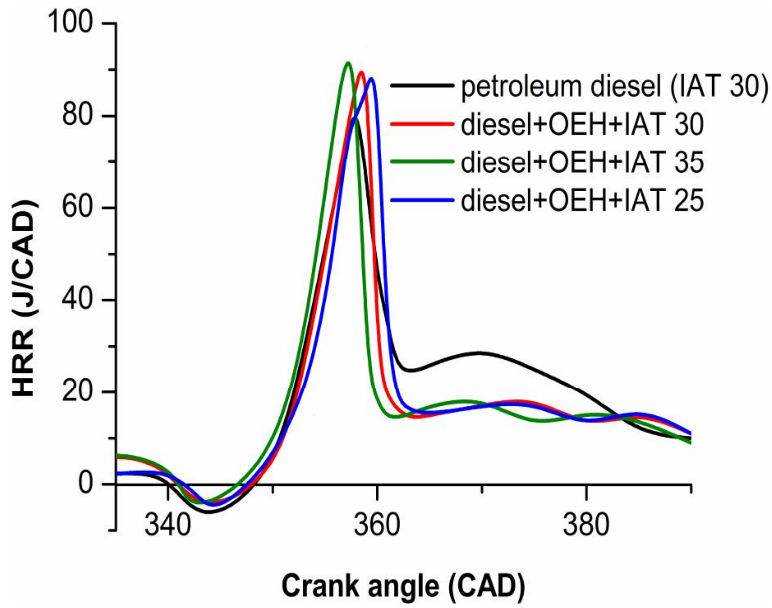

Figure 5.66 Variation of HRR with CA for different inlet air temperatures with OEH gas of 4.6lpm at rated load

At IAT25, heat release rate decreased by 2.7% and 1.1% compared to IAT35 and IAT30. This might be due to poorer fuel-air mixture formation which resulted in inferior combustion temperature. At IAT35, the heat release rate increased by 1.65% compared to IAT30. This might be due to enhancement in molecular reactivity between fuel and air compared to IAT30.

5.2.5.11 In-cylinder pressure

Figure 5.67 compares the variation of in-cylinder pressure with crank angle for petroleum diesel and diesel with OEH gas of 4.6 lpm flow rate with inlet air temperatures of IAT35, IAT30, and IAT25. From the graph, it was observed that the peak in-cylinder pressure increased, when the diesel combustion process was influenced by OEH gas. When the inlet air temperatures were IAT35, IAT30, and IAT25 at rated load of the engine, peak in-cylinder pressure increased by 7.14%, 5.71%, and 4.29% compared to the base line operation.

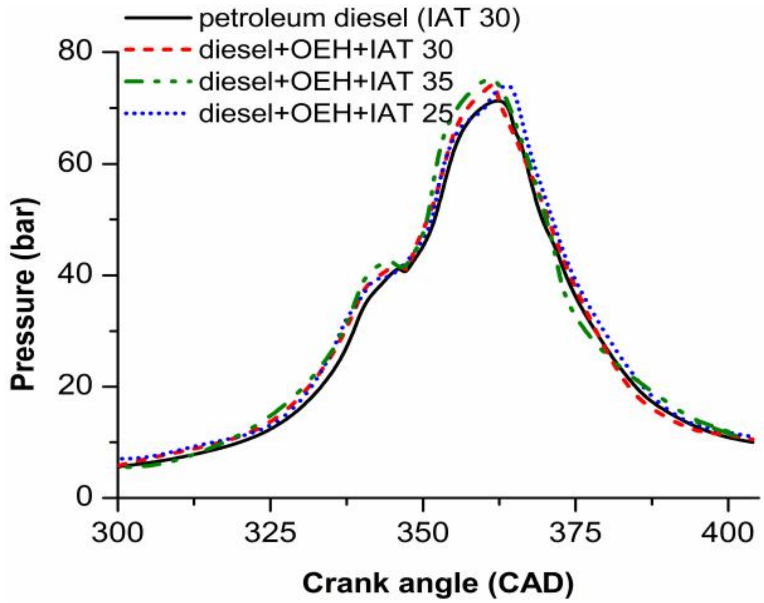

Figure 5.67 Variation of in-cylinder pressure with CA for different inlet air temperatures with OEH gas of 4.6lpm at rated load

Owing to high combustion efficiency of OEH gas and the increase in the penetration length of fuel spray which enhanced the droplet evaporation around the periphery of the penetrating spray (Julien 2006) resulted in an increase in the rate of heat release and also the combustion pressure. At IAT25, peak in-cylinder pressure decreased by 2.67% and 1.35% compared to IAT35 and IAT30. This might be due to lowered rate of pre-flame reactions during ignition delay period. At IAT35, the peak in-cylinder pressure increased by 1.35% compared to IAT30. This might be due to boosted combustion phenomena as a consequence of increase in the rate of chemical reactions. The heat release rate curve at this configuration also confirmed this.

5.2.6 Varied injection pressures and injection timings

In the present experimental work, the best flow rate OEH gas of 4.6 lpm was aspirated into the cylinder along with intake air at varied injection pressures and injection timings of a diesel fuel. The combinations of injection pressure and the injection timing selected for this investigation were:

- 200 bar injection pressure with injection time of 23^o BTDC
- 220 bar injection pressure with injection time of 23^o BTDC
- 220 bar injection pressure with injection time of 19^o BTDC

The combinations were selected based on the results discussed in the sections 5.2.1 and 5.2.2.

5.2.6.1 Brake thermal efficiency (BTE)

Figure 5.68 shows the comparison of brake thermal efficiency when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings and at different injection pressures of diesel fuel. The experimental results proved that the brake thermal efficiency decreased when the injection time was retarded and increased when the injection time was advanced. Also, the brake thermal efficiency decreased when the injection pressure was reduced and increased when the injection pressure was increased. Under the influence of OEH gas at the maximum load condition of the test engine, the brake thermal efficiency increased by 16.45%, and 22.08% for standard injection timing of 23^o BTDC, when the injection pressures of diesel fuel were 200 bar and 220 bar respectively compared to base line operation.

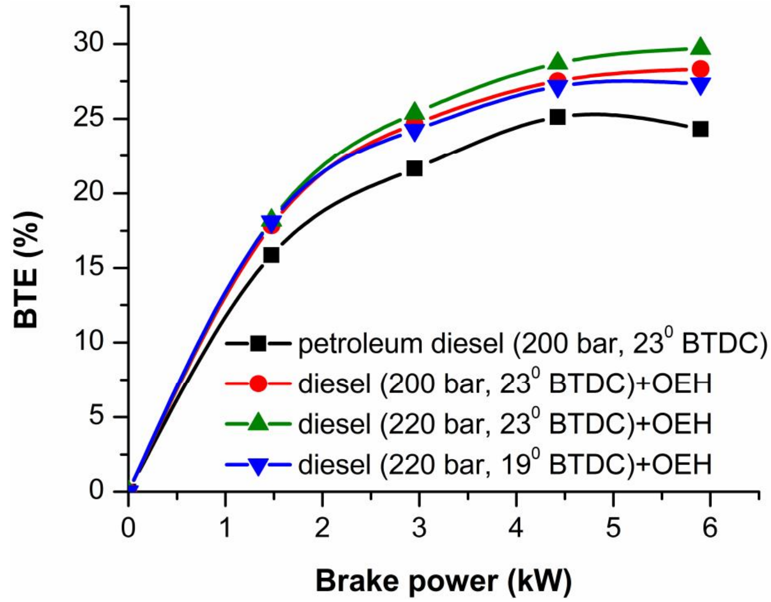

Figure 5.68 Variation of BTE with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

At the same time when injection timing of the diesel fuel was retarded to 19º BTDC and with the injection pressure of diesel fuel as 220 bar, the brake thermal efficiency increased by 12.24% compared to base line operation. This might be due to twin effect of OEH gas and the high injection pressure of diesel fuel. When the injection pressure of the diesel fuel was increased, the exposed surface of the diesel fuel droplet to air molecules got increased because of the fine droplet size of the fuel molecule. This increased the intimacy between the air and the fuel molecules. Along with this the high burning rate of hydrogen presented in the gas mixture, increased the efficiency of combustion.

When the test engine was operated in retarded injection time of 19º BTDC and with the diesel fuel injection pressure of 220 bar, it resulted in 3.53% and 7.98% decrease in brake thermal efficiency compared to standard injection timed operation of 23º BTDC with diesel fuel injection pressures of 200 bar and 220 bar respectively. This might be due to the fact that when the diesel fuel was injected at the retarded injection time, it resulted in the shorter ignition delay period (Mohammadi et al 2007). This reduced the adiabatic flame temperature of the combustion process. At 23º BTDC with the diesel fuel injection pressure of 220 bar resulting in maximum brake thermal efficiency compared to other injection timed operation. The brake thermal efficiency increased by 4.84% at the injection time of 23º BTDC and with diesel fuel injection pressure of 220 bar compared to standard injection time of 23º BTDC and with diesel fuel injection pressure of 200 bar. This might be due to more homogeneity of fuel and air mixture which resulted in an increase in brake thermal efficiency. The heat release rate curve at this combination of injection time and the injection pressure also confirmed this.

5.2.6.2 Brake specific energy consumption (BSEC)

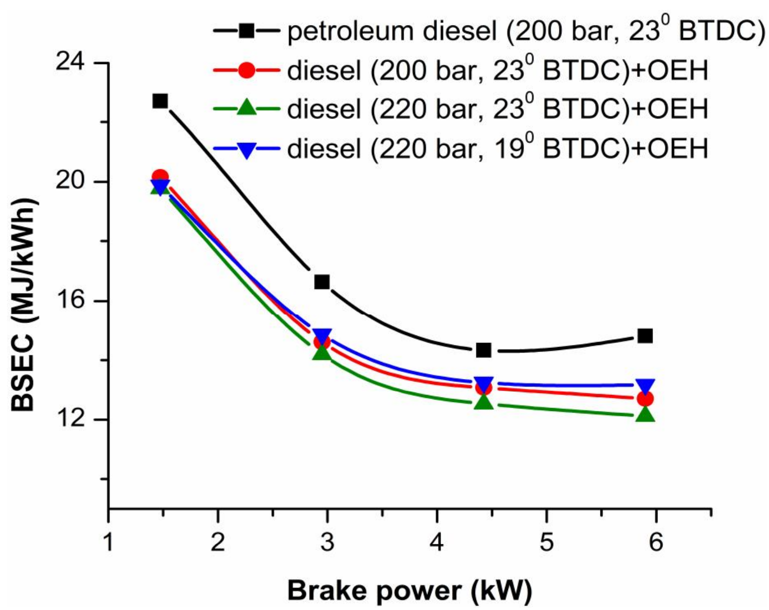

Figure 5.69 Variation of BSEC with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.69 depicts the comparison of BSEC when OEH gas of 4.6 lpm was inducted in the diesel combustion process at different injection timings & different injection pressures of the diesel fuel. The experimental results showed that the BSEC decreased when the injection time was retarded and increased when the injection time was advanced. Also, the BSEC decreased when the injection pressure was reduced and increased when the injection pressure was increased. Under the influence of OEH gas at 100% rated load of the engine, the BSEC decreased by 14.12% and 18.09% for standard injection timing of 23º BTDC and with the diesel fuel injection pressures of 200 bar and 220 bar respectively compared to base line operation. At retarded injection timing of 19º BTDC with the diesel fuel injection pressure of 220 bar, the BSEC decreased by 10.98% compared to base line operation.

This decrease in BSEC was due to the combined effect of catalytic action of OEH gas (Dulger & Ozcelik 2000) addition and high injection pressure of diesel fuel. When OEH gas was utilized in the diesel combustion process, due to its high diffusivity and low flame quenching distance major part of air-fuel mixture got combusted. In addition to this, the high injection pressure of diesel fuel generated fine droplets of fuel at the nozzle exit which formed the homogeneous mixture of air and fuel at quick time. When this homogeneous mixture got combusted, it resulted in good combustion and extracted more energy from the fuel during the combustion process. When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure of 220 bar, it resulted in 8.67% and 3.66% increase in BSEC compared to standard injection time with diesel fuel injection pressures of 220 bar and 200 bar operations.

At 220 bar injection pressure of diesel fuel and with injection timing of 23º BTDC, the minimum BSEC was obtained compared to other injection timings. The BSEC decreased by 4.61% at the diesel fuel injection pressure of 220 bar and with injection timing of 23º BTDC compared to diesel fuel injection pressure of 200 bar and with standard injection time of 23º BTDC. This might be due to increase in the spray cone angle which resulted in enhanced combustion.

5.2.6.3 Carbon monoxide emission (CO)

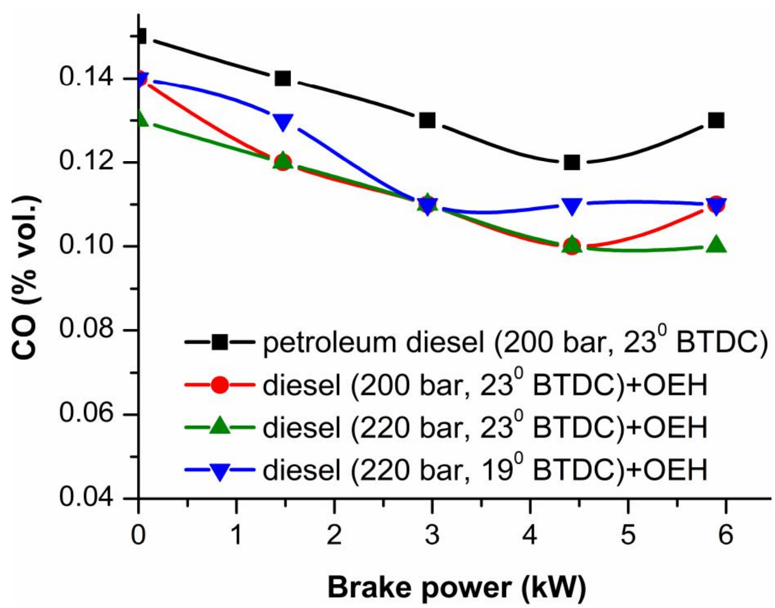

Figure 5.70 Variation of CO with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.70 illustrates the comparison of CO emission for petroleum diesel and diesel with OEH of 4.6 lpm at different injection timings and with different injection pressures of diesel fuel. The experimental results showed that the CO emission increased when injection timing was retarded and decreased when injection timing was advanced. Also, CO emission increased when injection pressure was decreased and decreased when injection pressure was increased. When the test engine was operated in retarded injection time of 19º BTDC with diesel fuel injection pressure of 220 bar at the rated load of the test engine, it resulted in 10% increase in CO emission compared to diesel fuel injection pressure of 220 bar and injection time of 23º BTDC. This might be due to under-mixing and some fuel particles in the fuel-rich zones might never react with oxygen. The CO emission decreased by 23.08% and 15.38% at the injection timings of 19º BTDC and 23º BTDC with the injection pressure of 220 bar compared to base line operation. This might be due to improvement in the completeness of combustion process and sufficiency of oxygen (Mohammed et al 2011). This resulted in intense combustion and reduced CO emission. When the injection pressure of the diesel fuel was 200 bar and with the injection time of 23º BTDC, CO emission got increased by 10% compared to diesel engine operation of 23º BTDC and with the diesel injection pressure of 220 bar. This might be due to decrease in the relative velocity of fuel injection (Ommi et al 2008).

5.2.6.4 Carbon dioxide emission (CO_2)

Figure 5.71 demonstrates the comparison of CO_2 emission when OEH gas of 4.6 lpm was supplemented in the diesel combustion process at different injection timings and the different injection pressures of the diesel fuel. Advancing the injection time of the diesel fuel increased the CO_2 emission whereas retarding the injection time reduced the CO_2 emission. Also, CO_2 emission increased when injection pressure was increased and decreased when injection pressure was decreased. Under the influence of OEH gas at full rated load of the engine, CO_2 emission increased by 12.12% and 15.15% for standard injection timing of 23º BTDC and injection pressures of 200 bar and 220 bar respectively compared to base line operation. This might be due to spontaneous combustion induced by atomic hydrogen and oxygen present in the OEH gas. They fissured heavy diesel molecules into tiny structured molecules in a very short time and resulted in high efficiency combustion.

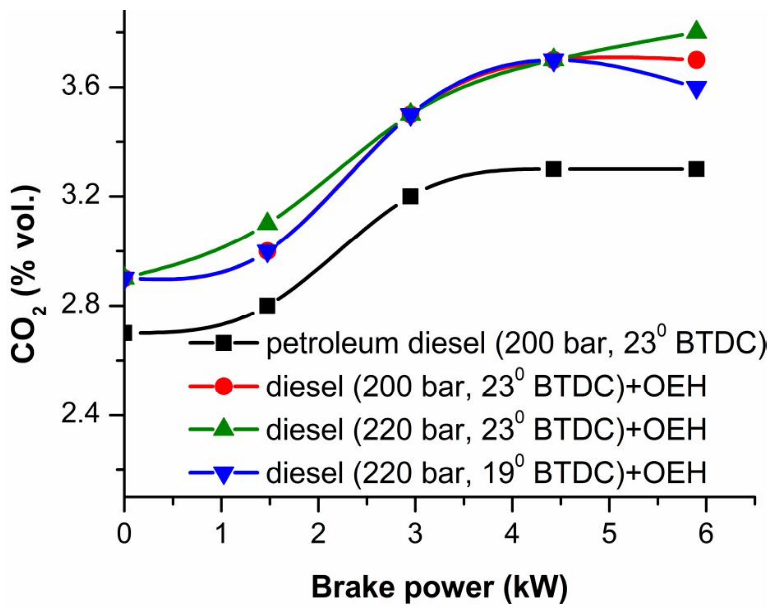

Figure 5.71 Variation of CO_2 with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure of 220 bar, it resulted in 5.26% and 2.7% decrease in CO_2 emission compared to the operation of 23º BTDC injection time with the diesel fuel injection pressures of 220 bar and 200 bar respectively. This might be due to improper conversion of CO to CO_2 due to decrease in combustion temperatures and resulted in less intense combustion. At 220 bar injection pressure of diesel fuel with 23º BTDC injection time, the maximum CO_2 emission was emitted from the engine compared to other injection timings and the injection pressure combinations. The CO_2 emission increased by 2.7% at 23º BTDC injection time with 220 bar injection pressure compared to 200 bar injection pressure with 23º BTDC injection time. This might be due to increase in turbulence intensity prevailing in the combustion chamber which resulted in more complete combustion (Ofner et al 1999).

5.2.6.5 Unburned hydrocarbon emission (UBHC)

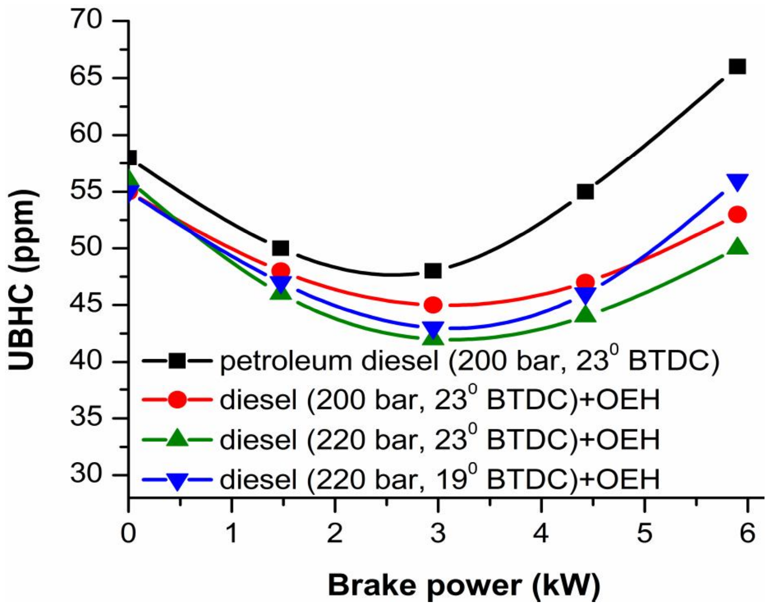

Figure 5.72 Variation of UBHC with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.72 shows the comparison of UBHC emission when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings and at different injection pressures of diesel fuel. The advancement of injection time lessened the UBHC emission whereas retarding the injection amplified the same. Also, UBHC emission increased when injection pressure was decreased and decreased when injection pressure was increased. Under the influence of OEH gas at 100% rated load of the engine, UBHC emission decreased by 19.7% and 24.24% for standard injection timing of 23º BTDC with the diesel fuel pressures of 200 bar and 220 bar respectively compared to base line operation.

This decrease in UBHC emission was due to low quenching distance of hydrogen present in the gas mixture, its fast burning velocity, and the wider cone angle of spray of diesel fuel at high injection pressures (Mangalla & Enomoto 2013). At the retarded injection timing of 19º BTDC with the diesel fuel pressure of 220 bar, UBHC emission decreased by 15.15% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel pressure of 220 bar, it resulted in 12% and 5.66% increase in UBHC emission compared to 23º BTDC injection time with the diesel fuel pressures of 220 bar and 200 bar respectively. This might be due to low homogeneity of combustible mixture formed during the ignition delay period. This was due to insufficient time available for proper mixing.

At 23º BTDC with the diesel fuel pressure of 220 bar, the minimum UBHC emission was exhausted from the engine compared to other injection timings and the injection pressures combination. The UBHC emission decreased by 5.66% at 23º BTDC with the diesel fuel pressure of 220 bar compared to 200 bar injection pressure of diesel fuel at the same injection time. This might be due to proper diffusing of air-fuel mixture at this combination. This resulted in more complete combustion of fuel mixture. The heat release rate curve at this combination also confirmed this.

5.2.6.6 Oxides of nitrogen emission (NO_X)

Figure 5.73 explicates the comparison of NO_X emission when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings and at different injection pressures of diesel fuel. The advancement of injection time enhanced the NO_X emission whereas retarding the injection helped to reduce the same. Also, NO_X emission increased when injection pressure was increased and decreased when injection pressure was decreased.

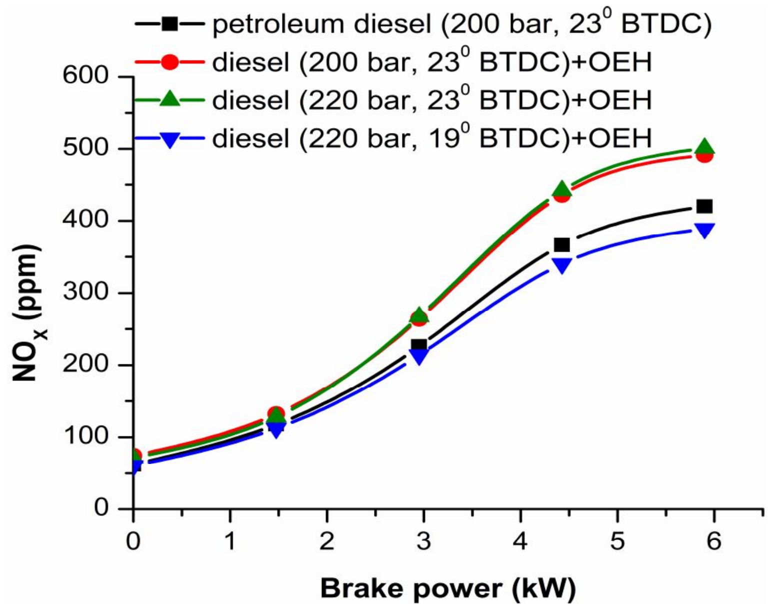

Figure 5.73 Variation of NO_X with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

Under the influence of OEH gas at 100% rated load of the engine, NO_X emission increased by 16.9% and 19.29% for standard injection timing of 23º BTDC with the diesel fuel injection pressures of 200 bar and 220 bar respectively compared to base line operation. The higher NO_X emissions are generally attributed to higher gas temperatures. The presence of hydrogen in the gas mixture could also contribute significantly to NO formation, according to the extended Zeldovich kinetic mechanism (Chiriac et al 2006). At the retarded injection timing of 19º BTDC with the diesel fuel injection pressure of 220 bar, NO_X emission got reduced by 7.38% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure 220 bar, it resulted in 22.36% and 20.779% decrease in NO_X emission compared to 23º BTDC with the diesel fuel injection pressures of 220 bar and 200 bar respectively. This might be due to extension of combustion process into expansion stroke which resulted in low temperature atmosphere during retarded injection time.

At 23º BTDC with the diesel fuel injection pressure of 220 bar, the maximum NO_X emission was exhausted from the engine compared to other injection timings. The NO_X emission increased by 2.04% at 23º BTDC with the diesel fuel injection pressure of 220 bar compared to 23º BTDC with the diesel fuel injection pressure of 200 bar. This might be due to fewer over-rich regions present in the combustion chamber (Julien 2006) and this also resulted in higher brake thermal efficiency than other combinations of operation.

5.2.6.7 Smoke emission

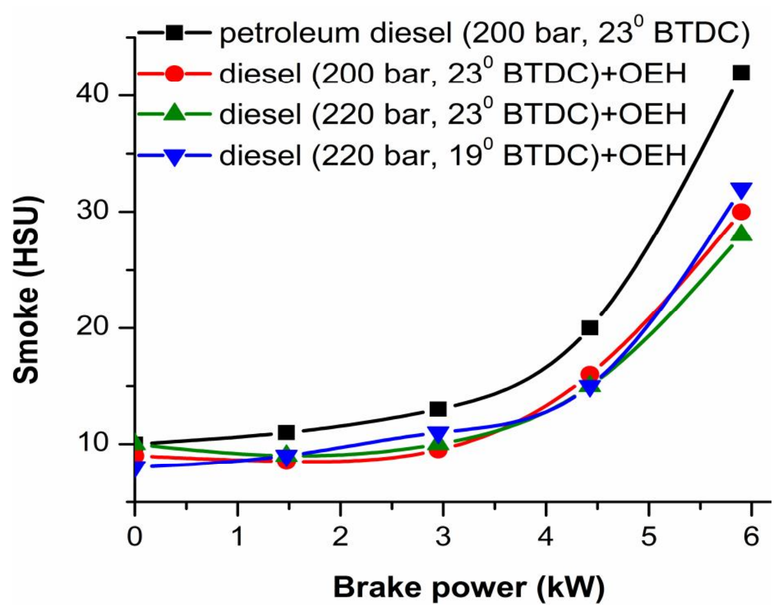

Figure 5.74 Variation of smoke emission with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.74 shows the comparison of smoke emission when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings and with different injection pressures of the diesel fuel and petroleum diesel combustion at standard engine specification. The experimental results showed that the smoke emission increased when the injection time was retarded and decreased when the injection time was advanced. Also, the smoke emission increased when the injection pressure was decreased and decreased when the injection pressure was increased. Under the influence of OEH gas at the maximum load of the engine, the smoke emission decreased by 28.57% and 33.33% for standard injection timing of 23º BTDC with the diesel fuel injection pressures of 220 bar and 200 bar respectively compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure of 220 bar, it resulted in 23.81% decrease in smoke emission compared to base line operation. Also, it increased by 14.29% and 6.69% when compared to standard injection time of 23º BTDC with diesel fuel injection pressures of 220 bar and 200 bar operations respectively.

When the injection time of diesel fuel was retarded, the fuel was introduced into the cylinder at a relatively higher pressure and temperature atmosphere. This decreased the pre-mixed combustion phase (Mohammadi et al 2007) and resulted in higher smoke emission compared to other configurations of operation. At 23º BTDC with diesel fuel injection pressure of 220 bar, the minimum smoke emission was obtained compared to other combinations of injection timings and injection pressures. The smoke emission got decreased by 6.67% at the injection time of 23º BTDC with the injection pressure of diesel fuel being 220 bar compared to standard injection time of 23º BTDC with the injection pressure of diesel fuel of 200 bar. The quantity of smoke emitted was linear with increasing injection pressure (Siebers & Pickett 2002). When the diesel fuel was injected at increased injection pressure, the fuel got mingled with air molecules to a great extent. This resulted in the formation of more homogeneous mixture of fuel and air. When this mixture got ignited, the combustion resulted in less smoke emission compared to other combinations of injection time and injection pressure.

5.2.6.8 Excess oxygen emission

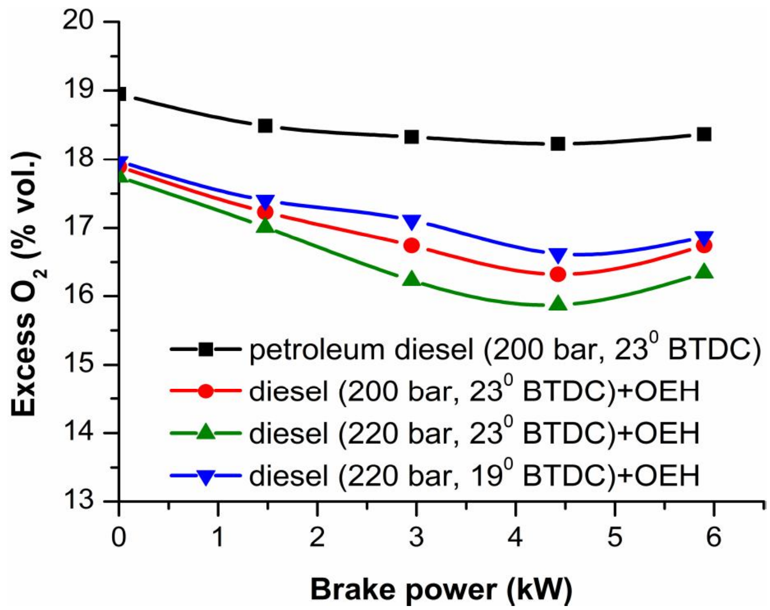

Figure 5.75 Variation of excess oxygen emission with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

Figure 5.75 illustrates the comparison of excess oxygen emission for petroleum diesel and diesel with OEH of 4.6 lpm at different injection timings and at different injection pressures of diesel fuel. The experimental results showed that the excess oxygen emission increased when the injection timing was retarded and decreased when injection timing was advanced. Also, the excess oxygen emission increased when injection pressure was decreased and decreased when injection pressure was increased. When the test engine was operated in retarded injection time of 19º BTDC with the injection pressure of diesel fuel of 220 bar at the rated load of the test engine, it resulted in 3.24% and 0.78% increase in excess oxygen emission compared to 23º BTDC injection time with the injection pressure of diesel fuel of 220 bar and 200 bar respectively.

This might be due to reduced H/C ratio in the overall fuel mixture. The excess oxygen emission decreased by 11.05% and 8.87% at 23º BTDC with the injection pressure of diesel fuel of 220 bar and 200 bar respectively compared to base line operation. When the diesel fuel was introduced at the retarded injection time of 19º BTDC with the injection pressure of diesel fuel of 220 bar, the excess oxygen emission decreased by 8.17% compared to base line operation. At 23º BTDC with the diesel fuel injection pressure of 220 bar, the excess oxygen emitted from the engine was lower when compared to other combinations of injection time and injection pressure operations.

The excess oxygen available at the exhaust of the engine at the injection time of 23º BTDC with the diesel fuel injection pressure of 220 bar got decreased by 2.39% compared to operation of 23º BTDC with the diesel fuel injection pressure of 220 bar. This might be due to availability of optimum sized fuel droplets at the exit of the injector nozzle at this configuration and optimum mixing of air and fuel due to high diffusion co-efficient of hydrogen present in the gas mixture.

5.2.6.9 Exhaust gas temperature (EGT)

Figure 5.76 displays the comparison of EGT of petroleum diesel combustion and when OEH gas of 4.6 lpm was added in the diesel combustion process at different injection timings and at different injection pressures of diesel fuel. Advancing the injection time facilitated to reduce EGT whereas retarding the injection time augmented the same. Also, increasing the injection pressure amplified EGT whereas reducing the injection pressure dropped the same. Under the influence of OEH gas at the maximum load of the test engine, EGT increased by 6.41% and 8.72% for standard injection timing of 23º BTDC with the diesel fuel injection pressure of 200 bar and 220 bar respectively compared to base line operation.

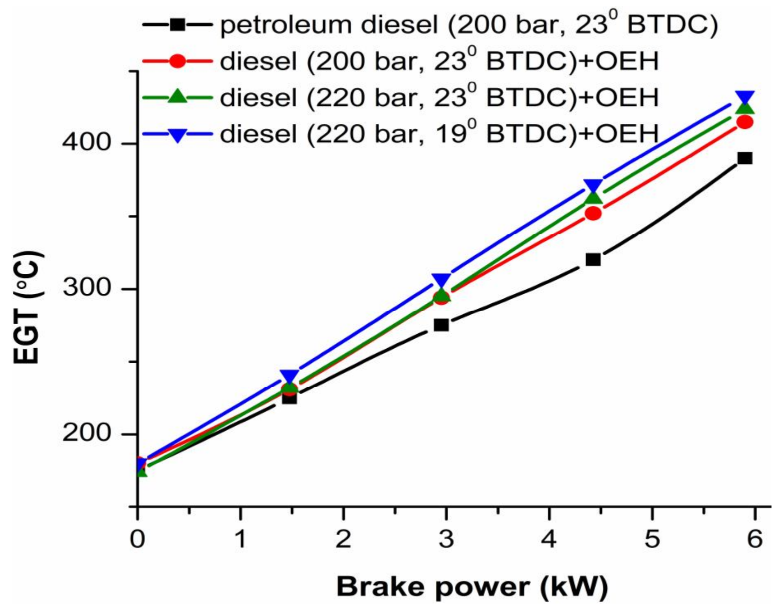

Figure 5.76 Variation of EGT with BP for different injection timings & injection pressures of diesel fuel with OEH gas of 4.6 lpm

This increase in EGT was due to fast burning velocity of hydrogen present in the gas mixture and the superior spray characteristics of a diesel fuel at increase in injection pressures. These factors resulted in enhanced premixed burning phase as a result of spontaneous combustion of OEH gas when its ignition was assisted by pilot diesel fuel. At the retarded injection timing of 19º BTDC with the diesel fuel injection pressure of 220 bar, EGT increased by 8.72% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure of 220 bar, it resulted in 2.12% and 4.34% increase in EGT compared to 23º BTDC with the diesel fuel injection pressures of 220 bar and 200 bar operations respectively. This might be due to less pressure prevailing in the combustion process during the expansion stroke (Fathi et al 2011).

At 23º BTDC with the diesel fuel injection pressure of 200 bar, the minimum EGT was exhausted from the engine compared to other combinations of injection timings and injection pressures. The EGT decreased by 2.12% at 23º BTDC with the diesel fuel injection pressure of 200 bar compared to 220 bar injection pressure with same injection time. This might be due to the fact that the peak pressure and the temperature developed during 220 bar operation were higher compared to 200 bar operation. The heat release rate curve and the in-cylinder pressure curve at 220 bar operation confirmed this.

5.2.6.10 Heat release rate (HRR)

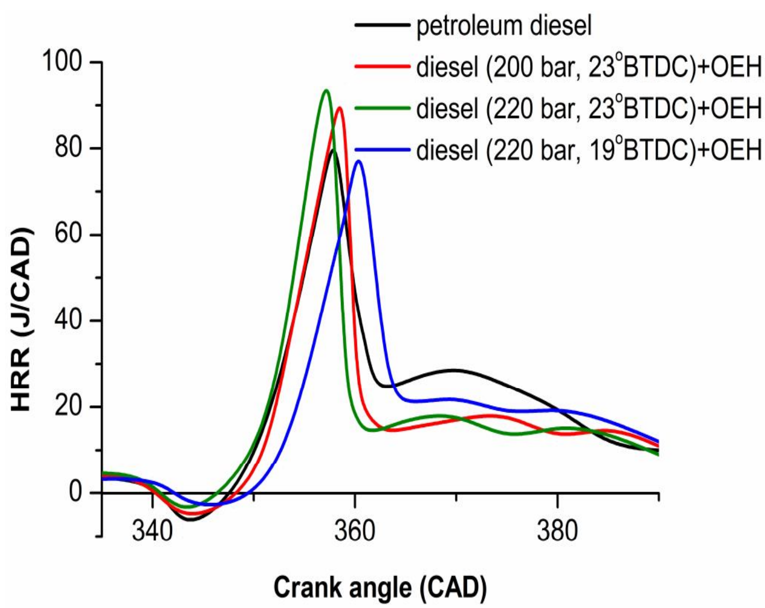

Figure 5.77 Variation of HRR with CA for different injection timings & injection pressures of diesel fuel at rated load

Figure 5.77 compares heat release rates of petroleum diesel combustion and when OEH gas of 4.6 lpm was inducted in the diesel combustion process at different injection timings and at different injection pressures of diesel fuel. The advancement of injection time augmented the heat release rate whereas retarding the injection time helped to decrease the same. Also, increasing the injection pressure intensified heat release rate whereas reducing the injection pressure plummeted the same. Under the influence of OEH gas of 4.6 lpm at the maximum load of the engine, the heat release rate increased by 20% and 13.75% for standard injection timing of 23º BTDC with the diesel fuel injection pressure of 200 bar and 220 bar respectively compared to base line operation. This might be due to significantly enlarged flammable region and extended flammability limit of fuel-air mixture. This resulted in improved pre-mixed burning phase and increase in NO_X emission.

At the retarded injection timing of 19º BTDC with the diesel fuel injection pressure of 220 bar, the heat release rate decreased by 5% compared to base line operation. When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure of 220 bar, it resulted in 20.83% and 16.48% decrease in heat release rate compared to 23º BTDC with the diesel fuel injection pressures of 200 bar and 220 bar respectively. At standard injection time of 23º BTDC with the diesel fuel injection pressure of 220 bar, the maximum heat release rate was obtained compared to other injection timing and injection pressure combinations. The heat release rate increased by 5.49% at 23º BTDC with the diesel fuel injection pressure of 220 bar compared to 23º BTDC with the diesel fuel injection pressure of 200 bar. This might be due to the elevated flame temperature and a smaller amount of heterogeneous fuel-air mixture present at this combination of injection pressure and injection time combustion compared to other combinations of injection pressure and injection time combustions.

5.2.6.11 In-cylinder pressure

Figure 5.78 compares in-cylinder pressures with crank angle for petroleum diesel and diesel with OEH gas of 4.6 lpm at different injection timings and at different injection pressures of diesel fuel.

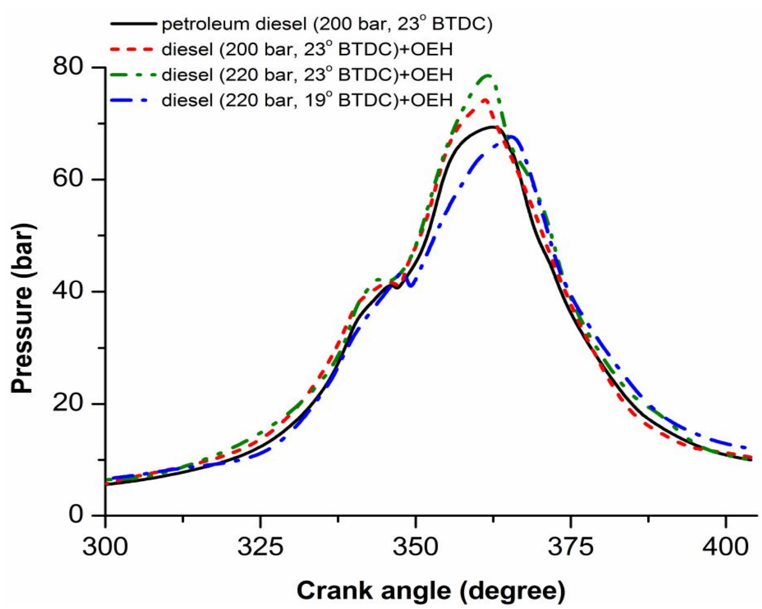

Figure 5.78 Variation of in-cylinder pressure with CA for different injection timings & injection pressures of diesel fuel at rated load

The experimental results showed that the in-cylinder pressure decreased when the injection time was retarded and increased when the injection time was advanced. Also, increasing the injection pressure increased peak in-cylinder pressure whereas reducing the injection pressure plunged the same. Under the influence of OEH gas at the maximum load of the engine, the peak in-cylinder pressure increased by 5.71% and 11.43% for standard injection timing of

23º BTDC with the diesel fuel injection pressures of 200 bar and 220 bar respectively compared to base line operation.

When the test engine was operated in retarded injection time of 19º BTDC with the diesel fuel injection pressure 220 bar, it resulted in 1.43% decrease in in-cylinder pressure compared to base line operation. Also, it got decreased by 11.54% and 6.76% compared to standard injection time of 23º BTDC with diesel fuel injection pressures of 220 bar and 200 bar respectively. When the fuel was injected at retarded injection time, the local temperature and the pressure available in the combustion chamber at the time of injection were higher. This in-turn reduced the homogeneous mixing of fuel and air.

At 23º BTDC with the diesel fuel injection pressure of 220 bar, the maximum in-cylinder pressure was obtained compared to other combinations of injection timings and injection pressures. The peak in-cylinder pressure increased by 5.41% at the injection time of 23º BTDC with the diesel fuel injection pressure of 220 bar compared to standard injection time of 23º BTDC with the diesel fuel injection pressure of 200 bar. This might be due to increase in the rate of chain reactions between air and fuel mixture and reduced SMD of droplets of fuel created due to high injection pressure of diesel fuel. These facts accounted for enhanced pre-mixed burning phase resulting in increase in peak in-cylinder pressure of diesel combustion when it was assisted by OEH gas.

5.2.7 Summary of results of phase II

In the second of phase of the experiment the engine was tested for its performance, emission, and combustion characteristics when the best flow rate of OEH gas was added in the combustion process of diesel with change in operating parameters of the engine. For this phase of experiments, six operating parameters of the engine were varied and tested. The six operating parameters varied were:

- Injection time of diesel fuel
- Injection pressure of diesel fuel
- Cooling water flow rate
- Temperature of diesel fuel
- Inlet air temperature
- Combination of injection pressure and injection time of diesel fuel

The tests results of the engine at varied operating parameters can be summarized as follows.

5.2.7.1 Varied injection timings (19º BTDC, 23º BTDC, and 27º BTDC)

- The brake thermal efficiency at the injection timing of 27º BTDC was higher compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The BSEC at the injection timing of 27º BTDC was lower compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The CO emission was lower at the injection timing of 23º BTDC compared to injection timings of 27º BTDC, 19º BTDC, and also to the base line operation.

- The CO_2 emission was higher at the injection timing of 23º BTDC compared to injection timings of 27º BTDC, 19º BTDC, and also to the base line operation.

- The UBHC emission was lower at the injection timing of 27º BTDC compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The NO_X emission was higher at the injection timing of 27º BTDC compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The smoke emission was lower at the injection timing of 27º BTDC compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The excess oxygen emission was lower at the injection timing of 27º BTDC compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The EGT was higher at the injection timing of 19º BTDC compared to injection timings of 27º BTDC, 23º BTDC, and also to the base line operation.

- The HRR was higher at the injection timing of 27º BTDC compared to injection timings of 23º BTDC, 19º BTDC, and also to the base line operation.

- The peak in-cylinder pressure was lower at the injection timing of 19º BTDC compared to injection timings of 27º BTDC, 23º BTDC, and also to the base line operation.

5.2.7.2 Varied injection pressures (180 bar, 200 bar, 220 bar, and 240 bar)

- The brake thermal efficiency at the injection pressure of 220 bar was higher compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The BSEC at the injection pressure of 220 bar was lower compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The CO emission was lower at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The CO_2 emission was higher at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The UBHC emission was lower at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The NO_X emission was higher at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The smoke emission was lower at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The excess oxygen emission was lower at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The EGT was higher at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The HRR was higher at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

- The peak in-cylinder pressure was higher at the injection pressure of 220 bar compared to injection pressures of 180 bar, 200 bar, 240 bar, and also to the base line operation.

5.2.7.3 Varied flow rates of cooling water (CWF75, CWF90, and CWF100)

- The brake thermal efficiency at CWF75 was higher compared to CWF90, CWF100, and also to the base line operation.

- The BSEC at CWF75 was lower compared to CWF90, CWF100, and also to the base line operation.

- The CO emission at CWF90 was lower compared to CWF75, CWF100, and also to the base line operation.

- The CO_2 emissions at CWF75 and CWF100 were higher compared to CWF90 and also to the base line operation.

- The UBHC emission at CWF75 was lower compared to CWF90, CWF100, and also to the base line operation.

- The NO_X emission at CWF75 was higher compared to CWF90, CWF100, and also to the base line operation.

- The smoke emission was lower at CWF75 compared to CWF90, CWF100, and also to the base line operation.

- The excess oxygen emission was lower at CWF90 compared to CWF75, CWF100, and also to the base line operation.

- The EGT was higher at CWF75 compared to CWF90, CWF100, and also to the base line operation.

- The HRR was higher at CWF75 compared to CWF90, CWF100, and also to the base line operation.

- The peak in-cylinder pressure at CWF75 was higher compared to CWF90, CWF100, and also to the base line operation.

5.2.7.4 Varied temperatures of diesel fuel (FT25 and FT35)

- The brake thermal efficiency at FT25 was higher compared to FT35 and also to the base line operation.

- The BSEC at FT25 was lower compared to FT35 and also to the base line operation.

- The CO emission at FT35 was lower compared to FT25 and also to the base line operation.

- The CO_2 emission at FT35 was higher compared to FT25 and also to the base line operation.

- The UBHC emission at FT35 was lower compared to FT25 and also to the base line operation.

- The NO_X emission at FT35 was higher compared to FT25 and also to the base line operation.

- The smoke emission was lower at FT35 compared to FT25 and also to the base line operation.

- The excess oxygen emission was lower at FT35 compared to FT25 and also to the base line operation.

- The EGT was higher at FT35 compared to FT25 and also to the base line operation.

- The HRR was higher at FT35 compared to FT25 and also to the base line operation.

- The peak in-cylinder pressure at FT35 was higher compared to FT25 and also to the base line operation.

5.2.7.5 Varied temperatures of inlet air (IAT35, IAT30, and IAT25)

- The brake thermal efficiency at IAT25 was higher compared to IAT35, IAT30, and also to the base line operation.

- The BSEC at IAT25 was lower compared to IAT35, IAT30, and also to the base line operation.

- The CO emission at IAT35 was lower compared to IAT25, IAT30, and also to the base line operation.

- The CO_2 emissions at IAT30 and IAT35 were equal and higher compared to IAT25 and also to the base line operation.

- The UBHC emission at IAT35 was lower compared to IAT25, IAT30, and also to the base line operation.

- The NO_X emission at IAT35 was higher compared to IAT25, IAT30, and also to the base line operation.

- The smoke emission was lower at IAT35 compared to IAT25, IAT30, and also to the base line operation.

- The excess oxygen emission was lower at IAT35 compared to IAT25, IAT30, and also to the base line operation.

- The EGT was higher at IAT35 compared to IAT25, IAT30, and also to the base line operation.

- The HRR was higher at IAT35 compared to IAT25, IAT30, and also to the base line operation.

- The peak in-cylinder pressure at IAT35 was higher compared to IAT25, IAT30, and also to the base line operation.

5.2.7.6 Varied injection pressures and injection timings of diesel fuel (200 bar & 23^o BTDC, 220 bar & 23^o BTDC, 220 bar & 19^o BTDC)

- The brake thermal efficiency at 220 bar & 23^o BTDC was higher compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The BSEC at 220 bar & 23^o BTDC was lower compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The CO emission at 220 bar & 23^o BTDC was lower compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The CO_2 emission at 220 bar & 23^o BTDC was higher compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The UBHC emission at 220 bar & 23^o BTDC was lower compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The NO_X emission at 220 bar & 19^o BTDC was lower compared to 200 bar & 23^o BTDC, 220 bar & 23^o BTDC, and also to the base line operation.

- The smoke emission was lower at 220 bar & 23^o BTDC compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The excess oxygen emission was lower at 220 bar & 23^o BTDC compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The EGT was higher at 220 bar & 19^o BTDC compared to 200 bar & 23^o BTDC, 220 bar & 23^o BTDC, and also to the base line operation.

- The HRR was higher at 220 bar & 23^o BTDC compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.

- The peak in-cylinder pressure at 220 bar & 23^o BTDC was higher compared to 200 bar & 23^o BTDC, 220 bar & 19^o BTDC, and also to the base line operation.