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Experimental study of NO_x emissions and injection timing of a low heat rejection diesel engine

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Abstract

In this study, an experimental study of the effects of injection timing on nitrogen oxide (NO_x) emissions of a low heat rejection (LHR) turbocharged direct injection diesel engine was conducted. The injection timing and brake specific fuel consumption (BSFC) trade-off must be considered together in performance and NO_x emissions analysis. For the original injection timing of the 20° before top dead centre, the brake specific fuel consumption value of the LHR engine was approximately 6% lower than the original engine. NO_x emissions were also higher (about 9%) than the original engine. To reduce NO_x emissions released by diesel engines, the method of injection timing retard was utilized. Thus, the LHR engine was tested for two different injection timings; 18° and 16° crank angle before top dead centre (BTDC), at the same engine speeds and load conditions. The results showed that the BSFC and NO_x emissions were reduced 2% and 11%, respectively by retarding the injection timing. Optimum injection timing for the LHR engine was obtained through decreasing by 2° BTDC.

Keywords: LHR diesel engine; Injection timing; NO_x emissions; Fuel consumption; Equivalence ratio

1. Introduction

The quest for increasing the efficiency of an internal combustion engine has been going on ever since the invention of this reliable workhorse of the automotive world. In recent times, much attention has been focused on achieving this goal by reducing energy lost to the coolant during the power stroke of the cycle. A cursory look at the internal combustion engine heat balance indicates that the input energy is divided into roughly three equal parts: energy converted to useful work, energy transferred to coolant and energy lost to exhaust. The reduction in heat losses also results in increased exhaust enthalpy. This extra exhaust energy can be utilized to drive a compounding turbine or a bottoming cycle, thus improving the overall system efficiency [1,2].

The motivating force behind the low heat rejection (LHR) engine has been the prospect to decrease of cooling load. That system is there to keep engine-operating temperatures down to levels tolerated by currently used constructional materials and lubricants. If the energy normally rejected to the coolant could be recovered instead on the crankshaft as useful work, then a substantial improvement in fuel economy would be obtained. Increased thermal efficiency and elimination of the cooling system are the major promises of the LHR engine [3]. On the other hand, the LHR engine designs promise to meet the increasingly stringent regulations in the areas of fuel economy and permissible emissions levels [3,4]. At the same time, exhaust energy rise, which accompanies this, can be effectively used in turbocharged engines. Higher temperatures in the combustion chamber can also have a positive effect on diesel engines, due to the self-ignition delay drop [5,6].

Exhaust emissions and fuel economy should be considered together. The regulated emissions include unburned hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO_x), and exhaust particulates (or in some cases, smoke). Of these, only CO is not a potential problem for the diesel, because of the overall lean stoichiometry it employs. One might expect lower HC from LHR diesel because the shortened ignition delay associated with higher temperatures decreases to opportunity for over mixing of fuel beyond the lean combustion limit and be-

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Table 1
Comparison of experimental results of low heat rejection (LHR) engine with those of original engines

Investigator(s)	Test model	Thickness of coating	Operational constraints	Performance of LHR engine compared to standard engine
Assanis et al. [1]	Single cylinder, direct injection diesel engine	500 μm and 1000 μm	Constant load and various speeds for both	10% improvement in BSFC; on average 15% increase NO_x
Kvernes et al. [9]	Single cylinder, two stroke diesel engine	500 μm	Constant BMEP and various speeds for both	5% decrease BSFC; 3% increase exhaust gas temperature
Sudhakar [10]	Turbocharged diesel engine	600 μm	Constant air/fuel ratio and different speeds for both	9.2% increase HC emission; NO_x emissions don't change; 1.7% improvement BSFC
Morel et al. [11]	Turbocharged heavy duty and light duty engines	Different levels of insulation	Constant peak pressure and air/fuel ratio for both	Thermal efficiency increased with the level of insulation at all loads for both heavy and light engines; brake thermal efficiency improves 8% which groves to 13% with Rankine bottoming cycle for truck engine
French [12]	Turbocharged and/or turbocompounded engine	Various levels of insulation	–	9% improvement in BSFC with turbocharge and 17% increase in thermal efficiency with turbocompounding; 2–4% improvement BSFC due to reduced friction in LHR engine

cause the generally higher cylinder-gas temperature during the expansion stroke should encourage oxidation of any unburned fuel escaping the primary process. However, a review of the limited published data on measured emissions from LHR diesel shows that while there are instance where HC did decrease, there are others where it increased [3].

The most important factors in determining the NO_x emissions produced by the combustion process are stoichiometry and flame temperature. If combustion took place at the overall lean stoichiometry in the engine cylinder, the NO_x problem might be relieved, but it does not. Because of the diffusive mixing of fuel and air occurring along the spray envelope, combustion is dominated by near-stoichiometric burning, where production of nitric oxide is high [3].

The flame temperature has a particularly strong effect on the production of nitric oxide, and the higher combustion temperature of the LHR engine would be expected to increase NO_x emissions. In nearly all published experiments, higher NO_x emissions have indeed been measured. Bryzik and Kamo [4] showed, however, that in lowering flame temperature by retarding injection timing, NO_x emissions could be reduced to the level of the conventionally cooled base line engine. Delaying combustion from the optimum in this manner sacrifices some fuel economy, of course, but at the selected operating point in their engine, there was still gain in fuel consumption relative to the conventionally cooled baseline. At Hino, experiments were conducted on five different LHR configurations of a heavy-duty diesel, in each case with the injection retarded in this manner to give same NO_x emissions as a cooled base line engine [3]. Some of the important experimental results are listed in Table 1.

NO_x emissions are particularly important. One of potential techniques for NO_x emissions reduction from diesel engines is injection-timing retard [7,8,13]. In the LHR diesel engines, injection-timing retard decreases NO_x emissions without increasing fuel consumption and particulate emissions [4]. Bryzik et al. [4] and Alkidas [14] reported that NO_x levels with an LHR engine were lower when on NO_x emissions versus brake specific fuel consumption (BSFC) trade off was made. Delaying combustion from the optimum in this manner sacrifices some fuel economy, of course, but at some specific operating conditions, there was still a gain in fuel relative to the original engine [4]. Suziki et al. [15] conducted experiments on five different LHR configurations of heavy-duty diesel engines in order to obtain the same NO_x emissions as that of the cooled baseline engine with retarded injection in the same manner. Delaying combustion from the optimum in the manner mentioned above sacrifices some fuel economy. However, for the selected operating conditions, they reported that there was still a gain in fuel consumption relative to the conventionally cooled baseline engines.

Kamo et al. [5] aimed at experimental determination whether a thin layer thermal barrier coated engine could improve its performance when a high pressure injector unit was used and found that the current trend toward high pressure fuel injection system was apropos, while they underlined that a thermal barrier coating (TBC) offered higher efficiency by 5–6% compared to the standard engine. They also reported significant improvement in fuel economy by insulating the diesel combustion chamber.

TBCs for diesel engines have generally been accepted to improve engine thermal efficiency and reduce emissions as well as specific fuel consumption because of their ability to provide thermal insulation to the engine components. The generally known principle that increased operation temperatures in energy conversion systems lead to an increase in efficiency, fuel savings and reduced emissions as particles, carbon monoxides (CO), hydrocarbons (HC) and limited reductions of NO_x emissions have, over many decades, promoted R&D activities in the field of TBCs development [8].

In the diesel engines, particulate and NO_x emissions are very important parameters [1,4]. For the LHR engine, the effects of particulate emissions have already been published in another paper [16] by the present author and his colleagues. Particulate and NO_x emissions mainly depend on oxygen concentration and cylinder gas temperature. For a detailed assessment, injection timing known as pump injection timing was retarded from 20° to 16° (BTDC). The brake specific fuel consumption, NO_x emissions and injection timing were considered as the key parameters of the LHR engine for the experimentations. When the LHR engine was operated at an injection timing of 20° BTDC (the optimal value for the original engine), it was found that NO_x emissions increased about 9%. When equivalence ratio was increased, brake specific fuel consumption (BSFC) values lowered for the whole engine operating conditions. At the equivalence ratio of particularly 0.6, BSFC decreased rapidly for both 20° BTDC and 18° BTDC of the LHR engine. Thus, by retarding the injection timing, an additional 1–2.8% savings in fuel consumption was obtained. 11% lower NO_x emissions were obtained than that of the original engine for 18° BTDC injection timing.

Although the NO_x emissions of a particular diesel engine have been a subject for long time for many researchers, there is still no reliable generalized method or procedure to predict these emissions. Therefore, the main aim of this study is to provide experimental data taken under relatively different working conditions with different engine, and it is hoped that the new data presented here will help in developing new predictive methods for the actual problem. The present study is investigated the effect of injection timing and valve adjustment on NO_x emissions of the LHR engine. The investigations were carried out using a six-cylinder, direct-injection diesel engine with an intercooler system. The results showed that the BSFC and NO_x emissions were reduced 2% and 11%, respectively by retarding the injection timing. Optimum injection timing for the LHR engine was obtained through decreasing by 2° BTDC.

2. Experimental setup and procedure

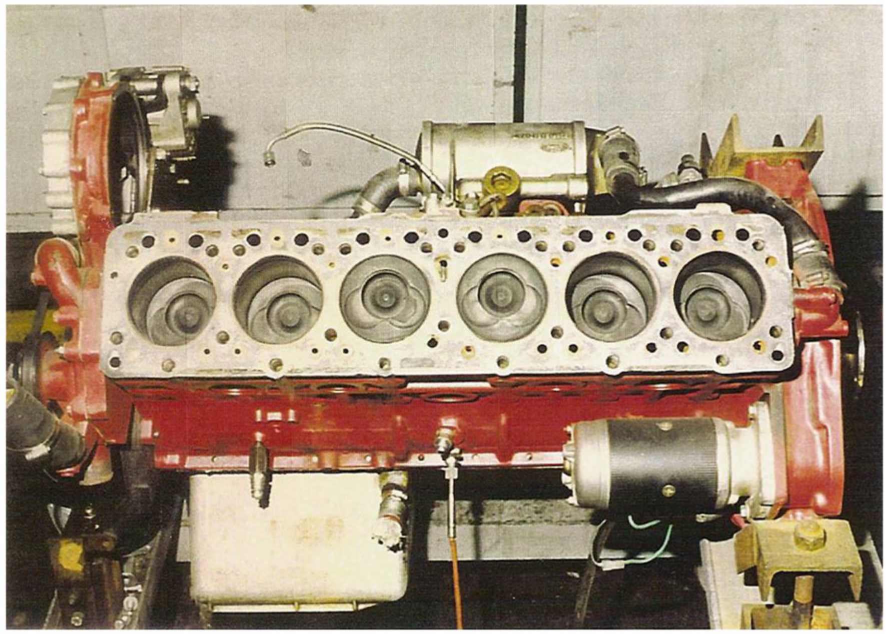
The investigations were carried out using a six-cylinder, direct injection diesel engine with an intercooler system. The important engine characteristics are summarized in Table 2. Fig. 1 shows the photo of the ceramic coated engine tested. The test setup is shown schematically in Fig. 2. It consists of a hydraulic brake dynamometer, intercooler unit, inclined manometer, NO_x emissions measuring device and a temperature sensor with a resolution of 1 °C. The measured parameters and corresponding locations which are shown in Fig. 2 are summarized in Table 3. Prior to the coating processes, the standard pistons were machined to remove material equal to the desired coating thickness. Then, the pistons, valves and cylinder head of the diesel engine were coated with ceramic materials. Atmospheric plasma spray was used as the coating method. The cylinder head and valves were coated with a 350 µm thickness $CaZrO_3$ over a 150 µm thickness NiCrAl bond coat (Metco 443, melting point 2140 °C, spray distance 170 mm). $MgZrO_3$ (Metco 210 BNS, melting point 1420 °C, spray distance 100 mm) was used as piston coating material. With the spray coating applied, the original dimensions of the coated parts of the engine were restored.

Table 2
Engine specifications

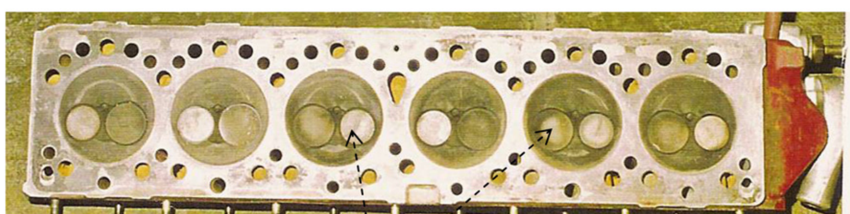
Engine type	Ford 6.0 lt. T/C, intercooling, direct injection
Cylinder number	6
Cylinder diameter (mm)	104.77
Stroke (mm)	114.9
Compression ratio	16.5:1
Maximum power (kW)	136 (2400 rpm)
Displacement (cc)	5947
Valve adjustment (mm)	0.46
Injector nozzle	Bosch
Number of holes	4
Hole diameter (mm)	0.32
Needle lift (mm)	0.35
Crack pressure (bar)	200
Injection timing (BTDC)	20°

Tests for the standard and the LHR engines were run at different load conditions and various speeds. First, the original engine was tested, as equipped with a water cooled intercooler, without coating. Then, the coated engine tests were conducted at the same working conditions. In order to measure the BSFC and NO_x emissions during operation, the engine was installed on a test stand and connected o a hydraulic dynamometer with capacity of 900 kW. Fuel amount was measured with a chronometer in a graduated cylinder that had a total volume of 250 cm^3 at different BMEP conditions and engine speeds, and then the brake specific fuel consumption was calculated. The exhaust gas temperatures were measured by a digital thermocouple (PT100) which can measure up to 1000 °C. NO_x emissions were measured by employing a gas analyzer (MRU 95/3 CD type) measurement range 0–4000 ppm with an accuracy of 300 ppm ±5% of full scale. The emission values were measured from the turbine exhaust exit. The engine speed was measured by a digital tachometer with a resolution of 1 rpm up to 10 000 rpm.

The experimental study consisted of two stages. For the first stage, the original engine was run at different speeds, changing from 1000 to 2400 rpm with an interval of 200 rpm and brake mean effective pressures ranging from 2.2 bar to 8.2 bar in order to determine the BSFC, brake mean effective pressures, equivalence ratio and NO_x emissions. At the second stage, before coating process, a 500 µm thickness was removed from the surfaces of the pistons, valves and cylinder head parts of the original engine. These parts were then coated with ceramic materials of the same thickness and mounted back to the engine. As a result, each of the ceramic coated pistons resulted in the same compression ratio (16.5:1). The same experimental measurements were repeated, as in the first stage, on the LHR



(a)



(b) supaps

Fig. 1. The photo of the coating engine tested: (a) pistons, (b) cylinder head.

version of the same engine. At the final stage, the injection delay timing of the LHR engine was retarded from 20° BTDC to 18° BTDC and, after that, to 16° BTDC. It was adjusted from fuel pump with cylinder comparator device. And then, original valve adjustment of the LHR engine was decreased from 0.46 to 0.38 mm and experiment was repeated for different injection timings. Experimental conditions of the original and the LHR engines are shown in Table 4.

3. Results and discussion

A comparative evaluation for both the original engine and the LHR engine has been made based upon the brake specific fuel consumption, the brake mean effective pressure, thermal efficiency, NO_x emissions, equivalence ratio, and injection timing.

3.1. Brake specific fuel consumption

Figs. 3, 4 and 5 show the variations in the experimental results of the brake specific fuel consumptions (BSFC) with engine speed for low, medium and full loads. Because of the higher surface temperatures of its combustion chamber, BSFC values of the insulated engine were lower than the original engine. The decrease of the BSFC was about 1–6% [1,9,10,12]. But, at full load and high-speed conditions, there was only a little increase of specific fuel consumption (Figs. 5 and 6). Because, at these conditions, the low heat rejection engine suffer significantly from loss of volumetric efficiency and the power in comparison to that of the original engine. Some of other investigation results are listed for BSFC in Table 1. The reason for this reduction was the increase of maximum exhaust gas temperature (about 65 °C) [16,17]. It was also noted that NO_x emission values were higher.

3.2. Brake mean effective pressure

Brake mean effective pressure (BMEP) is a measure of the engine performance. BMEP is directly proportional to torque which indicates the engine's ability to do work. Assanis et al. [1] investigated that the variations in BMEP with engine speed at full load for 1 mm and 0.5 mm coating thicknesses. The rise in BMEP, as engine speed decreased, results from the increased burned fraction prior to top dead center (TDC), which was realized with longer times between the start of injection and TDC. That working showed that the peak pressures of the LHR engine were decreased with increasing insulation for all

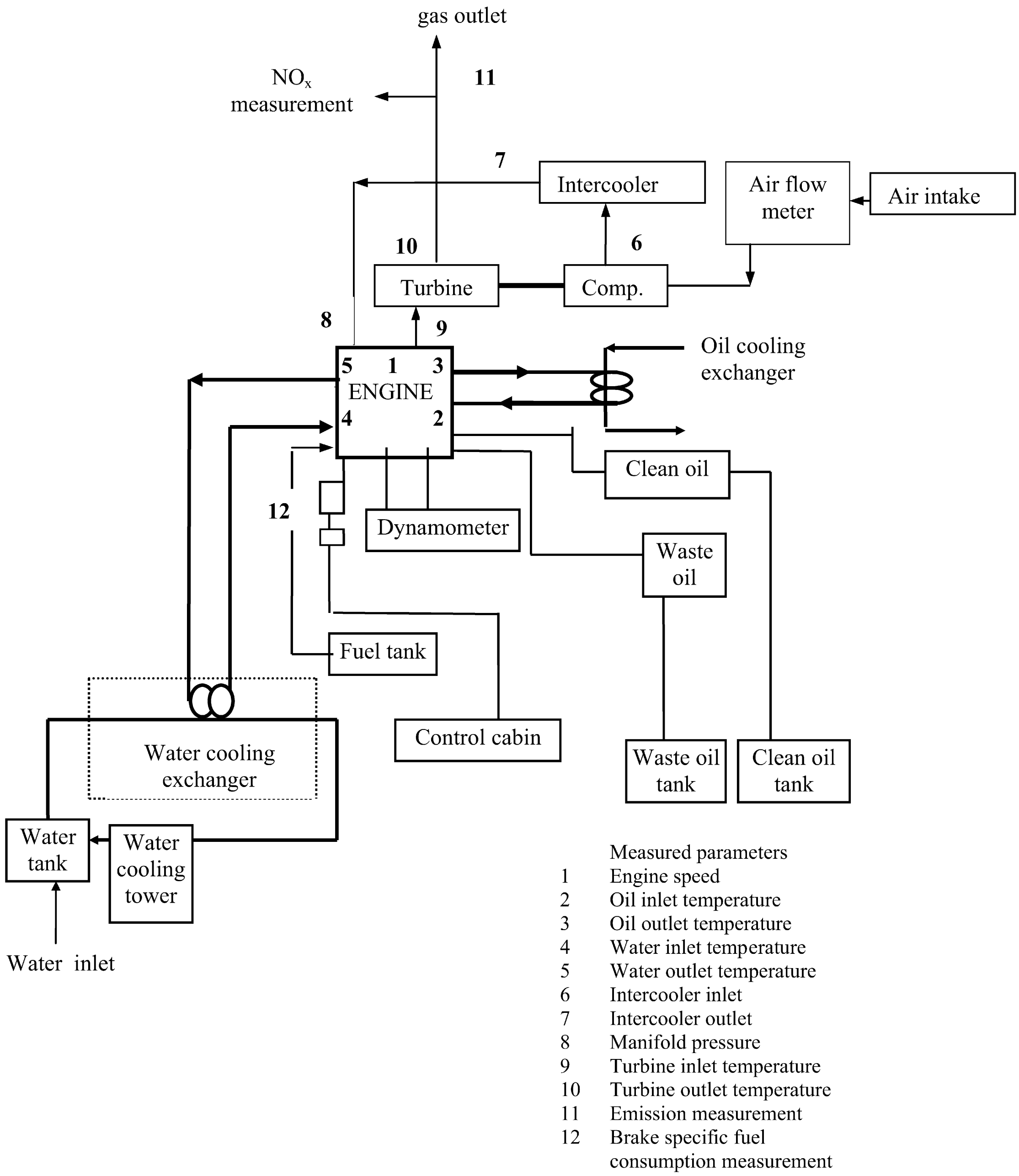


Fig. 2. The test setup.

speeds tested at full load. Part of the pressure drop is due to the lower compression ratio of 16:1 associated with the ceramic-coated pistons as opposed to 16.5:1 for the metal piston. The degraded combustion and lower peak pressures with the 1 mm coating were reflected in between 3% and 7% lower than baseline BMEP over the speed range.

In this study, before coating process, a 500 µm thickness was removed from the surfaces of the pistons, valves and cylinder head parts of the original engine. These parts were then coated with ceramic materials of the same thickness and mounted back to the engine. As a result, each of the ceramic coated pistons resulted in the same compression ratio (16.5:1).

3.3. Thermal efficiency

The variation of thermal efficiency over the speed range at full load for each of the original and the LHR engines is shown in Fig. 7. Thermal efficiency of the LHR engine is 16% higher than the original engine at low engine speeds. This efficiency is 8% better at high speeds. Alkidas et al. [1] determined that thermal efficiency of the engine with 0.5 mm ceramic coated piston was 10% better than baseline. But, thermal efficiency of the engine with 1 mm ceramic coated piston was lower 10% than baseline. This is a direct consequence of the more retarded heat release in the insulated engine [1]. Some of the important experimental results are listed in Table 1.

Table 3
Measured parameters

Label	Measured parameters	Units	Measurement range and accuracy	Device type
1	Engine speed	rpm	0–10 000 ± 1	Digital tachometer
2	Oil inlet temperature	°C	0–150 ± 1	PT-100
3	Oil outlet temperature	°C	0–150 ± 1	PT-100
4	Water inlet temperature	°C	0–150 ± 1	PT-100
5	Water outlet temperature	°C	0–150 ± 1	PT-100
6	Intercooler inlet	°C	0–150 ± 1	PT-100
7	Intercooler outlet	°C	0–150 ± 1	PT-100
8	Manifold pressure	mbar	0–7 ± 0.1	ST2100G2
9	Turbine inlet temperature	°C	0–1000 ± 1	PT-100
10	Turbine outlet temperature	°C	0–1000 ± 1	PT-100
11	Emission measurement	ppm	0–4000 ± 5	NO_x (MRU 95/3 CD)
12	Fuel measurement	cm^3/s	0–250 ± 5	Graduated cylinder

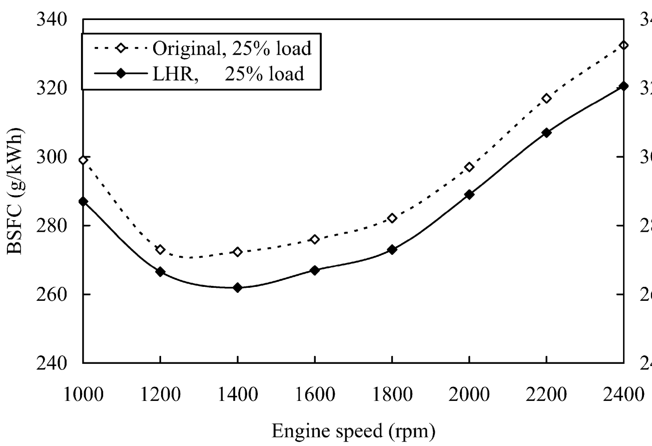


Fig. 3. Variation of brake specific fuel consumption with engine speed at low load.

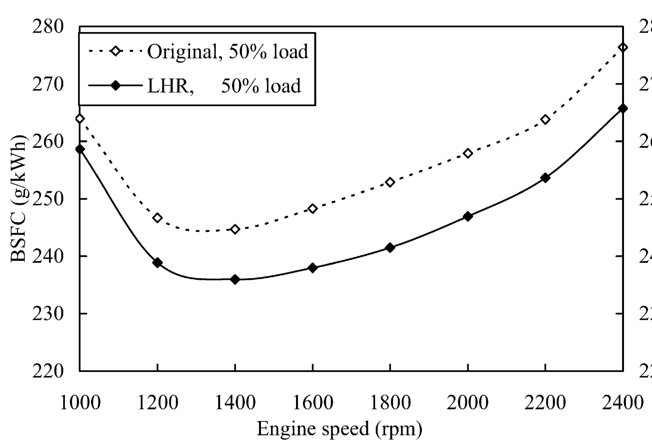


Fig. 4. Variation of brake specific fuel consumption with engine speed at medium load.

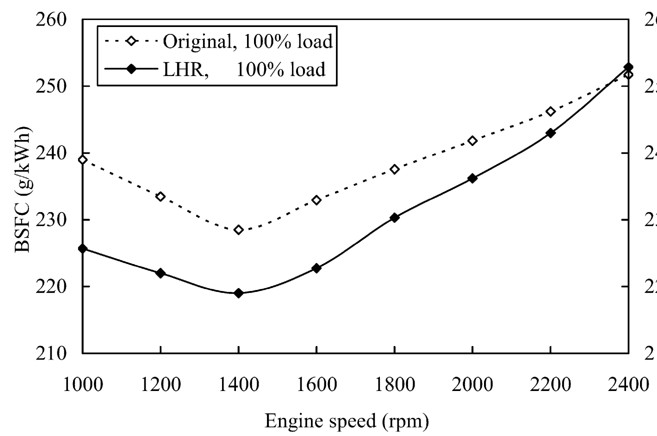


Fig. 5. Variation of brake specific fuel consumption with engine speed at full load.

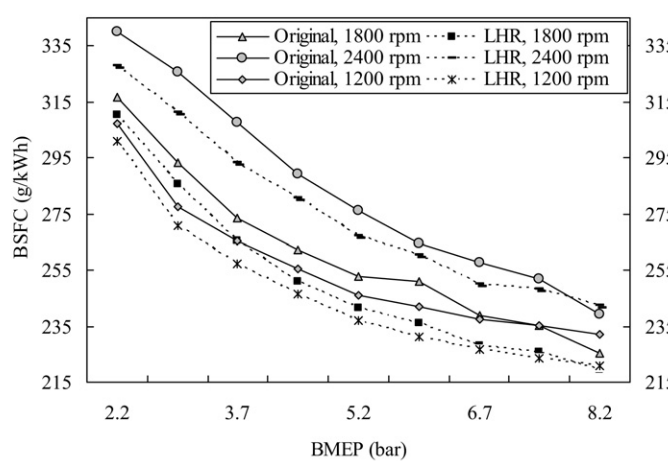


Fig. 6. Variation of brake specific fuel consumption with brake mean effective pressure for original engine and LHR engine at 1200, 1800 and 2400 rpm and 20° BTDC.

3.4. NO_x emissions

The higher NO_x emissions from the LHR engine, in comparison to the original engine, are attributed partly to the higher combustion temperature and partly to the shorter combustion duration in the LHR engine. Increasing the combustion duration increases the fraction of the fuel which burns later in the cycle and consequently decreases the emissions index of NO_x [17]. The higher NO_x emissions from the LHR engine agree with the findings of other studies [14,18,19]. For the same brake mean effective pressure and different engine speeds, the LHR engine produced more NO_x emissions at especially high speeds and low, medium and high loads, but it also had a lower BSFC at all engine operating conditions [20]. On the other hand, for

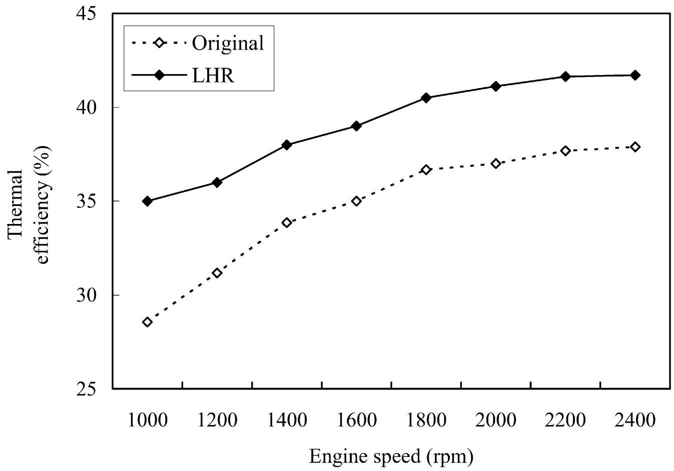


Fig. 7. Variation of thermal efficiency with engine speed at full load.

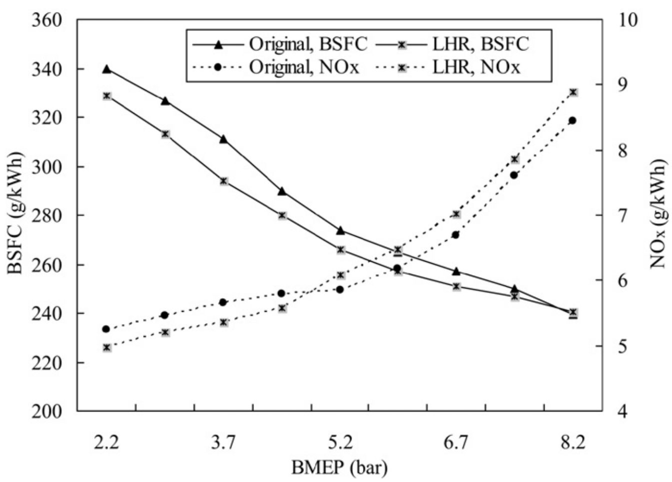


Fig. 8. BSFC-NO_x emissions trade-off for various brake mean effective pressures at 2400 rpm and injection timing of 20° BTDC.

the same engine speed and different brake mean effective pressures, the LHR engine produced lower NO_x emissions at low and medium engine loads, while higher NO_x emissions at large engine load and BSFC at whole load conditions (Fig. 8). It has been found that LHR engine provided almost no advantage at low load and speed conditions in terms of NO_x emissions. An average of 2.44% reduction in the NO_x emissions was recorded. However, when operating at low speeds–high load conditions, the NO_x emissions were measured to be 9% on average more compared to original engine. This is presumably due to the high gas temperatures caused by higher loads. On the other hand, for 8.2 bar BMEP and 2400 rpm, this value reduced to 2.5% in parallel to the deterioration in the overall volumetric efficiency (Fig. 8).

In the diesel engines, particulate emissions are of particular interest [1,4,16]. In a previous paper of the present author, due to the higher temperatures at the full load conditions, particulate levels were found to be slightly lower than the LHR engine. LHR engine provides relatively low exhaust gas temperatures for low loads at low speeds compared to the original engine, which causes lower turbocharger performances. Therefore, the volumetric efficiency of the covered engine is remarkably limited by the low turbocharger performance at low load and speed conditions. On the other hand, for full load at higher speeds, the LHR engine gives as nearly as 40% reduction in the particulate emission owing to the relatively higher gas temperatures. Bryzik et al. [1,4] reported that this reduction could be up to 86% depending upon the operating conditions of the engine.

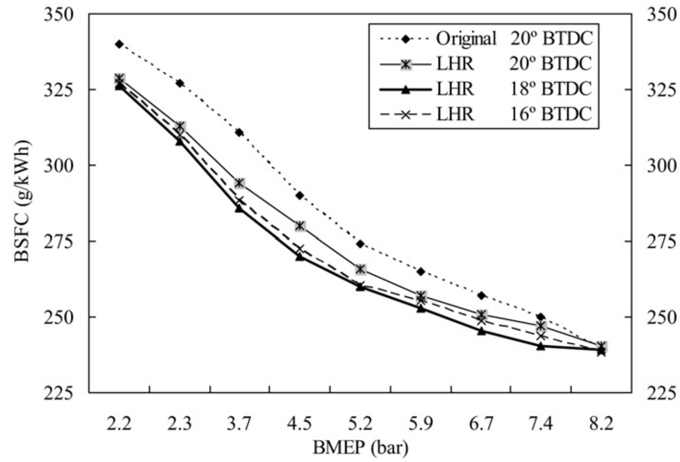


Fig. 9. Variation of brake specific fuel consumption with brake mean effective pressure for original engine with 20° BTDC and LHR engine with 20° BTDC, 18° BTDC, 16° BTDC at 2400 rpm.

3.5. *Effects of injection timing on NO_x emissions*

Potential techniques for NO_x emissions reduction from diesel engines are: exhaust gas recirculation, water injection, slower burning rate, reduced intake air temperature and injection timing retard [21,22]. In this study, injection timing retard method was used. Injection timing retard can decrease NO_x emissions without very much increasing the fuel consumption. However, it is not possible to make a prior prediction of how much injection retard is necessary to meet a certain emission target. LHR engine NO_x emissions data were obtained at various engine speeds and loads. The brake mean effective pressure and equivalence ratios were calculated. Thus, the optimum injection timing used in this study was determined.

Figs. 9 and 10 show BSFC and NO_x emissions trade-off for various brake mean effective pressures at 2400 rpm for original engine with 20° BTDC and the LHR engine with 20° BTDC, 18° BTDC, 16° BTDC at 2400 rpm. For 18° BTDC injection timing, a relative reduction of 1–2.8% in the specific fuel consumption was recorded. In addition, at 2400 rpm and 8.2 bar BMEP, the deterioration in the fuel consumption for 20° BTDC was seen to disappear for 18° BTDC as shown in Fig. 9. However, 16° BTDC of injection timing has been observed to increase the brake specific fuel consumption despite reduction in NO_x emissions at the same conditions (Fig. 10). Thus, the 18° BTDC could be regarded as the optimum setting for the LHR engine as illustrated in Fig. 9.

3.6. *Effects of equivalence ratio on NO_x emissions*

At the air excessive coefficient measurement, it was used air measurement set which composed of air tank, sharp edge nozzle

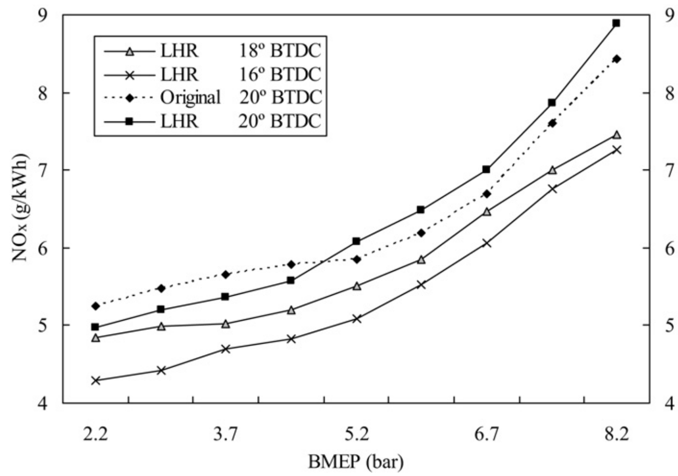


Fig. 10. Variation of NO_x emissions with brake mean effective pressure for original engine with 20° BTDC and LHR engine with 20° BTDC, 18° BTDC, 16° BTDC at 2400 rpm.

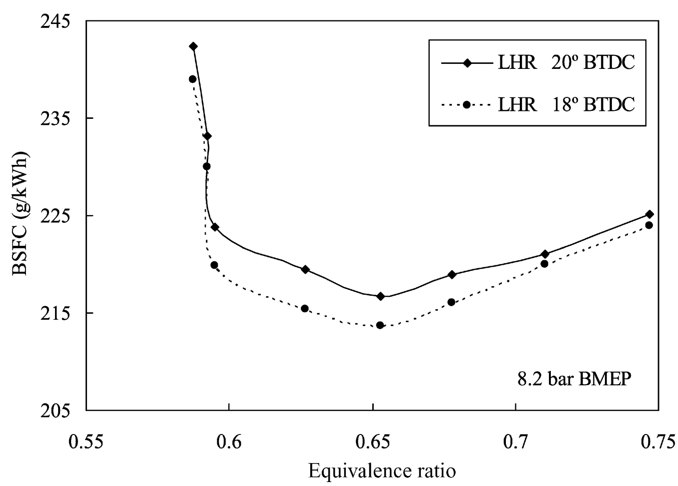


Fig. 11. Variations of brake specific fuel consumption with equivalence ratio for LHR engine at 8.2 bar BMEP and different injection timings (°BTDC).

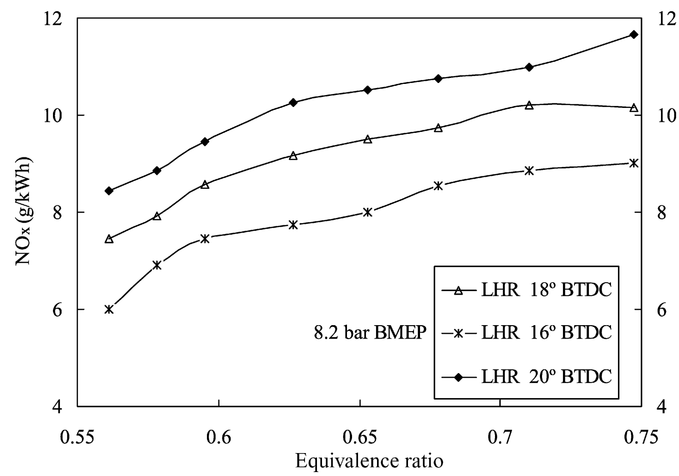


Fig. 12. Variations of NO_x emissions with equivalence ratio of the LHR engine at 8.2 bar BMEP and different injection timings (°BTDC).

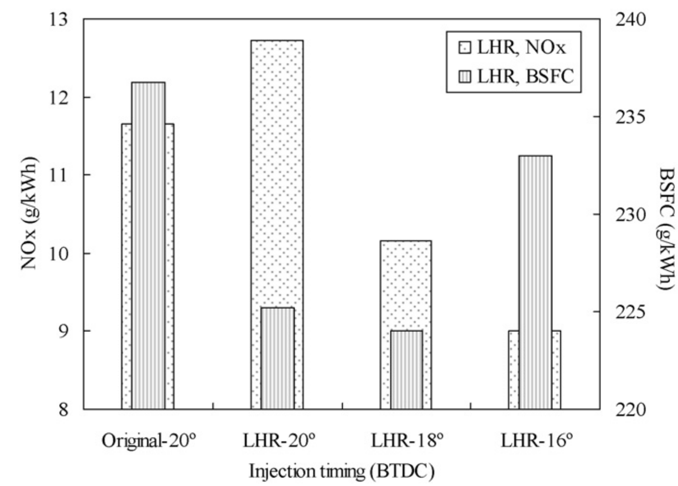


Fig. 13. Variations of brake specific fuel consumption and NO_x emissions with injection timing of the LHR engine at 1000 rpm and 8.2 bar of brake mean effective pressure.

and inclined manometer. Volume of air tank is 300 liter. Equivalence ratio was calculated from the air excessive coefficient. For fuel mass flow rate, the brake specific fuel consumption was measured by a graduated cylinder that had a total volume of 250 cm^3 at different engine load and speed conditions. Then, fuel mass flow rate was determined by means of engine power and BSFC values.

Figs. 11 and 12 show variations of the brake specific fuel consumption and NO_x emissions with equivalence ratio (i.e., the actual fuel-air ratio divided by its stoichiometric value) corresponding to 16°, 18° and 20° BTDC at 8.2 bar BMEP. The production of the oxides of nitrogen in the cylinder depends on the maximum temperature in the cycle and the oxygen concentration in the cylinder. For a fixed equivalence ratio, amount of pilot fuel and intake temperature, advancing the injection timing has a great effect on the maximum mean gas temperature in the cylinder [23]. NO_x emissions increased exponentially with the increasing oxygen concentration. Lida et al. [24] ignition delay is decreased by oxygen enriched charging and this allows injection timing to be retarded and hence NO_x emissions to reduce without increasing the specific fuel consumption, if the oxygen concentrations are less than 25%. In Fig. 11, BSFC values decreased for the whole engine operating conditions. At particularly equivalence ratio of 0.6, BSFC decreased rapidly for both 20° BTDC and 18° BTDC of the LHR engine.

For a given equivalence ratio, the maximum mean gas temperature decreases as the injection timing is retarded. The net effect is a reduction in NO_x emissions as shown in Fig. 12. For the same conditions, a decrease in BSFC was especially noted at 18° BTDC injection timing as shown in Fig. 11.

Figs. 13–16 show variations of the brake specific fuel consumption and NO_x emissions with the injection timing of the LHR engine at 1000, 1400, 1800 and 2400 rpm and 8.2 bar of brake mean effective pressure. NO_x emissions are strongly affected by injection timing. NO_x emissions are decreased 25% at low-medium speeds, and 13% at high speeds. The sensitivity of NO_x emissions to injection timing in diesel engines is no surprise. Since the injection timing is one of the key emission

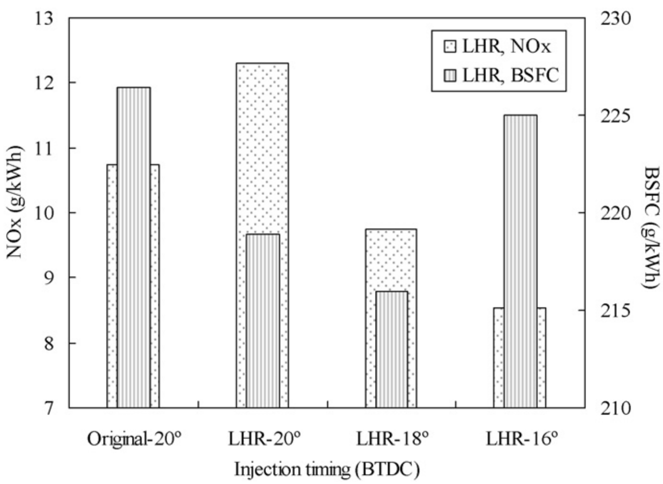


Fig. 14. Variations of brake specific fuel consumption and NO_x emissions with injection timing of the LHR engine at 1400 rpm and 8.2 bar of brake mean effective pressure.

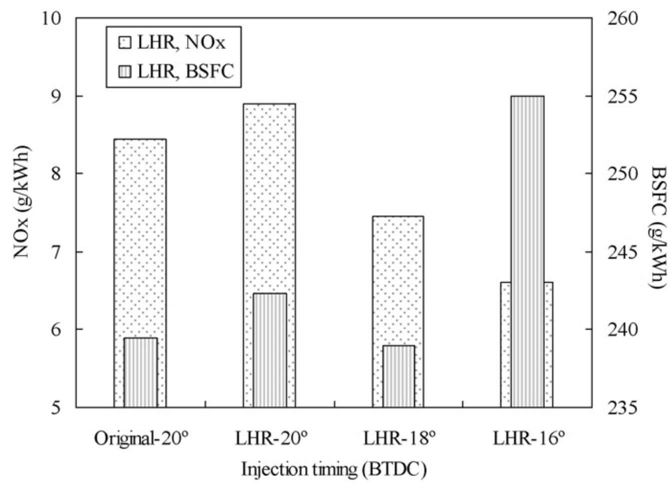


Fig. 16. Variations of brake specific fuel consumption and NO_x emissions with injection timing of the LHR engine at 2400 rpm and 8.2 bar of brake mean effective pressure.

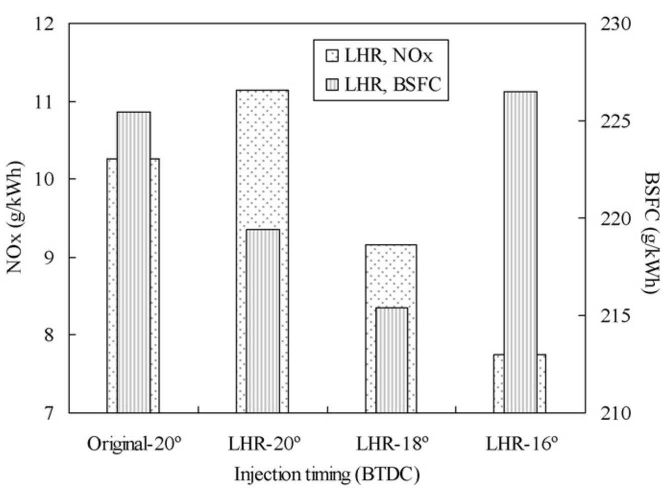


Fig. 15. Variations of brake specific fuel consumption and NO_x emissions with injection timing of the LHR engine at 1800 rpm and 8.2 bar of brake mean effective pressure.

control parameters. At a given injection timing, NO_x emissions tend to be lowered for the higher speed of 2400 rpm.

In the LHR engine with the injection timings of 18° BTDC at 1000 rpm, NO_x emissions and BSFC were 15% and 5.6% lower respectively than the original engine. For the injection timing of 16° BTDC, NO_x emissions and BSFC reduction were found as 30% and 1.6% respectively. Although BSFC of LHR engine with 20° BTDC was 0.5% lower than that of 18° BTDC, it was 3.5% higher than that of 16° BTDC (Fig. 13). In the LHR engine with the injection timings of 18° BTDC at 1400 rpm, NO_x emissions and BSFC were 10% and 5% lower respectively than the original engine. For the injection timing of 16° BTDC, NO_x emissions and BSFC reduction were found as 26% and 0.6% respectively. Although BSFC of LHR engine with 20° BTDC was 1.4% lower than that of 18° BTDC, it was higher by 2.7% than that of 16° BTDC (Fig. 14).

In the LHR engine with the injection timings of 18° BTDC at 1800 rpm, NO_x emissions and BSFC were found to be lower by 12% and 4.6%, respectively than the original engine. For the injection timing of 16° BTDC, NO_x reduction was about 32%, but BSFC increased about 0.6%. Although BSFC of LHR engine with 20° BTDC was 1.9% lower than 18° BTDC, it was 3% higher than 16° BTDC (Fig. 15). In the LHR engine with the injection timings of 18° BTDC at 2400 rpm, NO_x emissions and BSFC were 13% and 0.2% lower respectively than the original engine. For the injection timing of 16° BTDC, NO_x reduction was about 27%, but BSFC increased about 6%. Although BSFC of LHR engine with 20° BTDC was 1.3% lower than 18° BTDC, it was higher 5% higher than 16° BTDC (Fig. 16). According to these results, NO_x emissions decreased drastically for the injection timing of 16° BTDC at higher speeds and BMEP conditions, but the BSFC was further diminished. According to above mentioned results, evaluating BSFC and NO_x emissions together at 1000, 1400, 1800 and 2400 rpm speeds and full load, optimum injection timing of the LHR engine is determined 18° BTDC.

Fig. 17 shows the variation of valve adjustment on NO_x emissions with BMEP of the original engine for 20° BTDC and the LHR engine for 18° BTDC and 16° BTDC at 2400 rpm. Decreasing valve adjustment of the LHR engine from 0.46 mm to 0.38 mm for 20° BTDC, NO_x emissions of the LHR engine are lower 8.4% on average. NO_x emissions of the original engine are 4.7% on average at 0.46 mm valve adjustment. Moreover, NO_x emissions of the LHR engine are diminished at 18° BTDC and 16° BTDC for 0.38 mm valve adjustment. But, NO_x emission values of the LHR engine with 0.46 mm valve adjustment at 20° BTDC, 18° BTDC and 16° BTDC are lower than 0.38 mm. It is clearly seen that valve adjustment modifications increase to NO_x emissions of the engine. But valve adjustment indicated that such modifications had almost no effect on BSFC of the engine [16]. It can be concluded that optimization of the injection system and injection timing would play a remarkable role for LHR engines. However, it has been observed that the delay in injection timing without modification in the valve adjustment reduced both the specific fuel consumption and NO_x

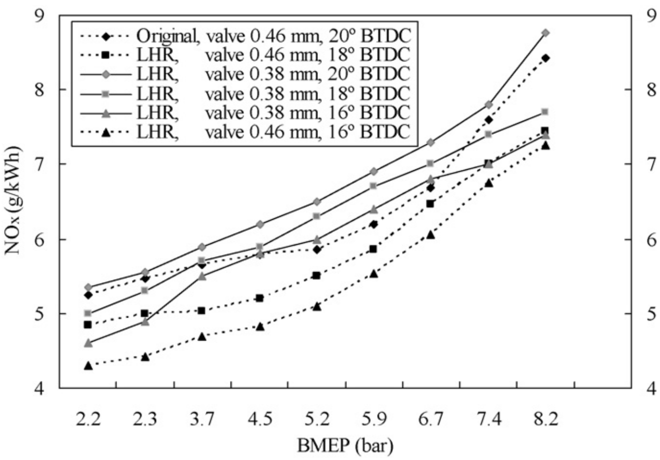


Fig. 17. Variation of NO_x emissions with BMEP for 0.46 mm valve adjustment and 20° BTDC injection delay timing of original engine and 0.4 mm valve adjustments with 18° and 16° BTDC injection delay timing and 0.38 mm valve adjustments with 20, 18° and 16° BTDC injection delay timing of the LHR engine at 2400 rpm.

emissions. Similar conclusions were also reported in the literature [21,25].

4. Conclusions

This study focused on the injection timing effect on NO_x emissions from the LHR diesel engine. Experiments were carried out using six cylinders, turbocharged direct injection diesel engine which was operated under different speeds and BMEP conditions (Table 4). The major results of this experimental study are as follows.

1. The brake specific fuel consumption values of the LHR engine were found to be approximately 6% lower than the original engine at especially medium speeds and effective pressure. For 18° BTDC of injection timing, a relative reduction of 1–2% in the specific fuel consumption was recorded. On the other hand, fuel consumption was decreased about 8%. However, 16° BTDC exhibited almost the same fuel consumption with the original engine at 2400 rpm and different brake mean effective pressures.
2. At the brake mean effective pressure of 8.2 bar, NO_x emissions were diminished by almost 31% on average for between 1000–1400 rpm (low speeds), and it was decreased by 21% on average for between 1800–2400 rpm (high speeds) when original injection timing was retarded by 2° BTDC for the LHR engine. Furthermore, NO_x emissions deduction was 5.3% on average compared to the original engine. Particulate emissions decreased clearly in the insulated engines at the full load and 2400 rpm. These reductions were up to 40%.
3. NO_x emissions were diminished in the range of 35–53% when original injection timing was retarded by 4° BTDC for the LHR engine. But, particulate emissions and BSFC are deteriorated.
4. It was observed that the NO_x emissions increased exponentially with the increasing oxygen concentration for 20° BTDC, 18° BTDC and 16° BTDC of the LHR engine. When operating at low equivalence ratios at high BMEP, the NO_x emissions were measured about 9% more compared to original engine. This is presumably due to the high gas temperatures caused by high BMEP. However, NO_x emission values decreased at 0.75 equivalence ratio for both 18° BTDC and 16° BTDC.
5. When equivalence ratio was increased, BSFC values decreased for the whole engine operating conditions. At particularly 0.6 of equivalence ratio, BSFC decreased more quickly for both 20° BTDC and 18° BTDC of the LHR engine.
6. In the light of above mentioned results, evaluating BSFC and NO_x emissions together at 1000, 1400, 1800 and 2400 rpm speeds and full load, optimum injection timing of the LHR engine is determined 18° BTDC.
7. Reducing the valve adjustment for the LHR engine from 0.46 mm to 0.38 mm gave generally lower NO_x emissions

Table 4
Experimental conditions of the original and the LHR engines

Tests	Speed (rpm)	Load (N)	Injection timing (°BTDC)	Valve adjustment (mm)	Compression ratio
Original engine tests	1000 1200 1400 1600 1800 2000 2200 2400	150 200 250 300 350 400 450 500 550	20	0.46	16.5:1
Low heat rejection engine tests 1	1000 1200 1400 1600 1800 2000 2200 2400	150 200 250 300 350 400 450 500 550	20	0.46	16.5:1
Low heat rejection engine tests 2	1000 1200 1400 1600 1800 2000 2200 2400	550	18 16	0.46	16.5:1
Low heat rejection engine tests 3	2400	150 200 250 300 350 400 450 500 550	20 18 16	0.38	16.5:1

regardless of considering the injection timing (about 2%) at full load. However, 0.38 mm of valve adjustment was observed to remarkably increase as compared to its original value (0.46 mm) at full load. NO_x emissions generally are lower than 0.38 mm valve adjustments at different injection timings and 2400 rpm. For example, they are lower 16% than 0.38 mm valve adjustment at 16° BTDC and 4.5 bar BMEP. However, they are higher 8% in 20° BTDC than in 16° BTDC at 0.38 mm valve adjustment.

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