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EXPERIMENTAL STUDY ON THE INJECTION PRESSURE EFFECT ON *BSFC* AND EMISSIONS IN A TURBOCHARGED DIESEL ENGINE

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ABSTRACT

Fuel injection pressures in diesel engines play an important role in engine performance and emissions obtaining treatment of combustion. In the present diesel engines such as common rail diesel engine, the injection pressures can be increased about 1500 – 2000 bars. In this study, the effects of injection pressure on *bsfc* and exhaust emissions have been investigated. Experiments have been performed on a direct-injection turbocharged diesel engine. Emissions and engine *bsfc* values have been measured in 13 speed and load conditions based on ECE-R49 test by changing injection pressure from 200 to 300 bars. Results show that higher injection pressure reduces HC and *bsfc*, due to better mixing, causes faster combustion. The results also reveal that NO_X and CO_2 are not substantially changed by injection pressure. Smoke behaves differently due to injection pressure. This is because of injection pressure effect on ignition delay and exhaust temperature. These parameters affect re-burning soot at the late combustion phase. In term of emission control in turbocharged diesel engine, this research work proposes variable injection pressure instead of constant injection pressure.

KEYWORDS: Diesel engine; injection pressure; exhaust emissions; brake specific fuel consumption (*bsfc*).

INTRODUCTION

In a diesel engine it is important to distribute the fuel jet quickly and to form a uniform gas mixture after fuel injection in order to economize fuel consumption and reduce harmful emissions. That is why in recent years sprays have been actively investigated [1, 2, 3].The diesel engine produces lower amounts of HC, CO, and NO_X than the comparable gasoline engine. HC and CO are lower because of the more complete combustion of the fuel and air. NO_X is lower because the peak temperature is not maintained very long [4]. Smoke, or particulate emission, occurs when there is insufficient air to completely burn the fuel. There are several factors that the engine designer varies to provide low emission levels with high performance and good fuel economy. Some of these factors are the shape of the combustion chamber, the location and angle of the nozzle, the injection rate and nozzle

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spray pattern, injection timing, and camshaft timing [4]. Another method for improving the diesel engine performance and exhaust emissions is to use the turbocharger which is driven by exhaust gas from the engine cylinder [5-7]. Also the inlet manifold pressure of a turbocharged engine is a very effective parameter on improving the soot emission [8].

In the present diesel engines, fuel injection systems are designed to obtain higher injection pressure. When fuel injection pressure is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. When injection pressure is increased fuel particle diameters will become small [2]. Since formation of mixing of fuel to air becomes better during ignition period, smoke level and CO emission will be less. However, if injection pressure is too high, ignition delay period becomes shorter. So, possibilities of homogeneous mixing decrease and combustion efficiency falls down. Therefore, smoke is formed at the exhaust of the engine [9].

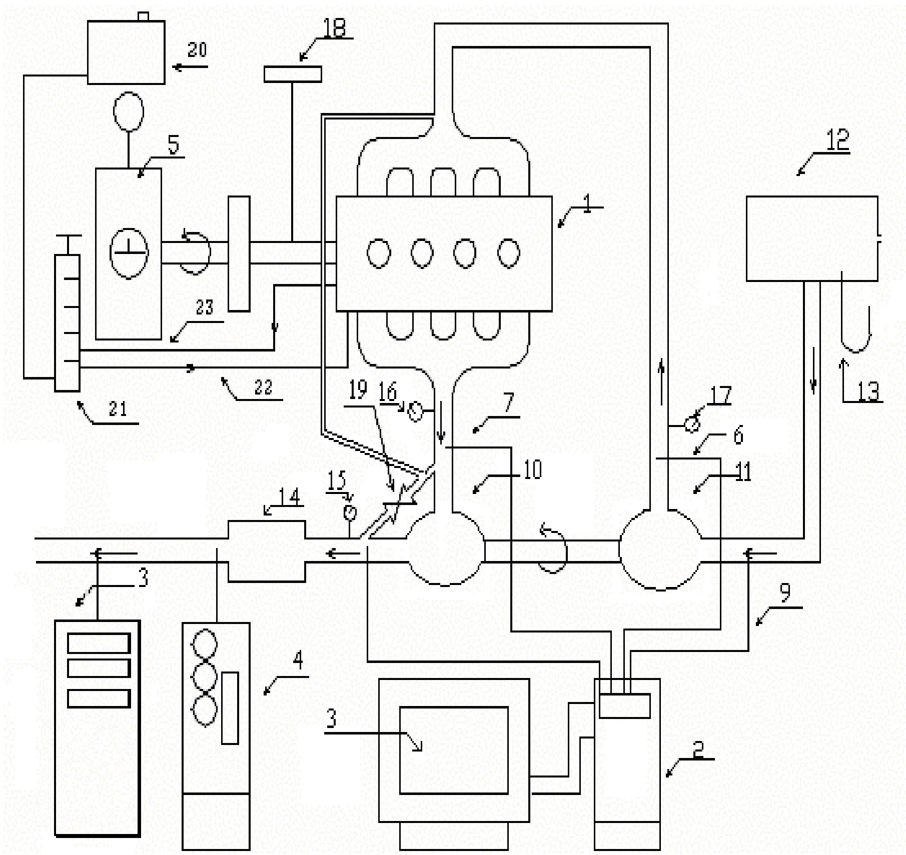
In high pressure common rail diesel injection, the diesel spray structure has been studied using optical diagnostics. An increase of injection pressure is found to enhance the atomization at the nozzle outlet, resulting in a more distributed vapour phase, hence resulting in better mixing [10].

In this study, the effects of injection pressure on *bsfc* and exhaust emissions have been investigated on a direct-injection turbocharged diesel engine. In each test injection pressure increased from initial injection pressure (200 bars) to injection pressures 220, 250, and 300 bars. In the tests the amount of fuel injections was constant in different injection pressures. Emission values and *bsfc* were measured based on ECE-R49 standard test at different injection pressures.

1. EXPERIMENT

1.1. ENGINE TEST RIG

Figure 1 shows a schematic diagram of the test rig and measuring system. OM314 turbocharged diesel engine (1) was used in the present work. Engine characteristics are given in Table 1. The engine is turbocharged by a turbocharger (10, 11) equipped with a waste gate (19), supported by Borg Warner Company to convert the aspirated engine to a turbocharged diesel engine. The test rig is equipped with a 112 kW DXF Heenan & Froude hydraulic dynamometer (5). A Plint-RE205 gas analyzer (3) measures unburned hydrocarbons as C_6H_{14}, CO, CO_2 and O_2. The experimental apparatus is composed of AVL-415 smoke meter (4), thermocouples type K (6, 7, 8, 9) by using an interface connected to a PC (2), volumetric fuel meter (21), air meter (surge tank and orifice manometer) (12, 13) pressure gages (15, 16, 17) and electrical engine speed meter (18).



1 – ***Engine***
2 – ***Computer***
3 – ***Plint-RE205 gas analyzer***
4 – ***AVL-415 smoke meter***
5 – ***Hydraulic dynamometer***
6 – ***Thermocouple type [K]***
7 - ***Thermocouple type [K]***
8 - ***Thermocouple type [K]***
9 - ***Thermocouple type [K]***
10 – ***Turbocharger turbine***
11 – ***Turbocharger compressor***
12 – ***Surge tank***
13 – ***Manometer***
14 – ***Exhaust silencer***
15 – ***Pressure gage***
16 - ***Pressure gage***
17 - ***Pressure gage***
18 – ***Electrical speed meter***
19 – ***waste gate***
20 – ***Fuel reservoir***
21 – ***Fuel gage***
22 – ***Fuel inlet***
23 – ***Reverse fuel***

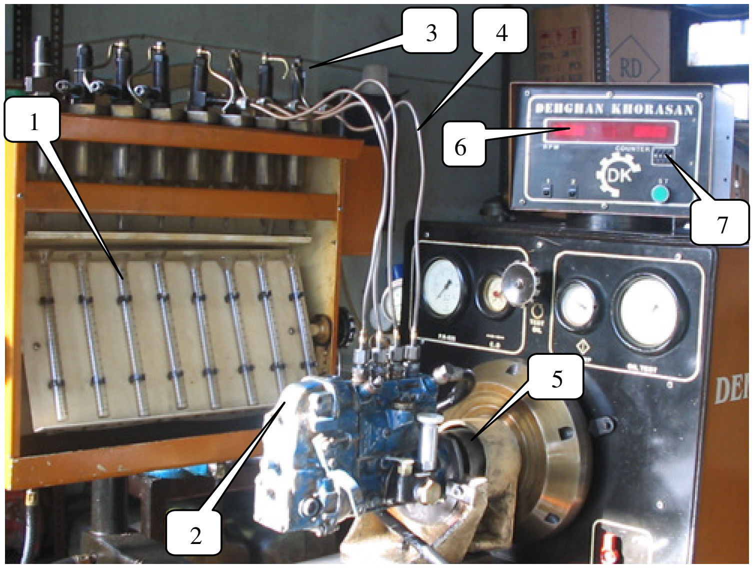
Figure 1. Block diagram of the test set up

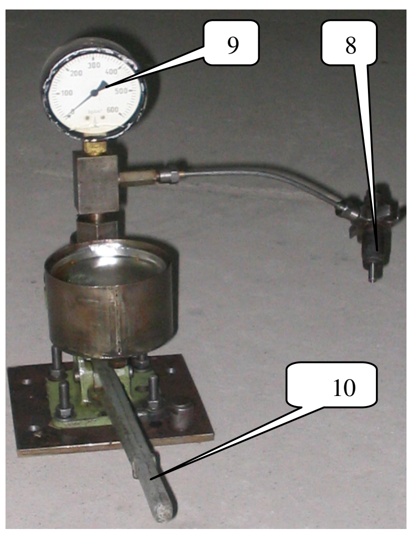
Engine and turbocharger	Specification
Engine type	4-stroke diesel engine
Number of cylinders	4
Combustion chamber	Direct injection
Bore x stroke (mm)	97 x 128
Piston displacement (cc)	3784
Compression ratio	17 : 1
Maximum power (HP)	85
Maximum torque (Nm)	235
Maximum speed (rpm)	2800
Mean effective pressure (bar)	6.8@2800 rpm
Turbocharger turbine	Radial type
Turbocharger compressor	Centrifugal type

Table 1. Specifications of experimental engine and turbocharger

1.2. REGULATION OF DISTRIBUTOR PUMP AND NOZZLE

Figure 2 shows distributor pump and nozzle regulator units. In each test, injection pressure of nozzles was regulated with the nozzle regulator unit. By adding specific washer at the bottom of the spring nozzle the injection pressure increased. Also in each test, the distributor pump and nozzles were regulated by the distributor regulator unit. In each test, the distributor regulator unit regulates the amount of fuel injected at the speed of 600 rpm and 100 stroke of the plunger inject 5 cc of fuel.





1 – Glass
2 – Distributor pump
3 – Nozzles
4 – Pipe
5 – Shaft
6 – Electrical speed meter
7 – Counter
8 – Nozzle
9 – Pressure gage
10 – Actuator

Figure 2. Regulator of distributor pump and nozzle units

1.3. EXPERIMENTAL SET UP AND MEASUREMENTS

Experiments were conducted on a diesel engine connected with a Froude hydraulic dynamometer. Before starting the engine, the nozzles were taken off and adjusted to 200, 220, 250 and 300 bars. In each adjustment, one or more washers were used to change the nozzle pressures. After that, the adjusted nozzles were tested with the distributor pump in the regulator unit of the distributor pump and in all injection pressure mass flow rate of fuel and adjusted to 5cc at 600 rpm and 100 stroke of the plunger. Then the nozzles and the distributed pump were fitted to the engine, and the engine was run.

Engine tests were done based on ECE-R49 test in 13 speed and load conditions, as shown in Tables 2 and 3. In the experiments, the fuel flow rate, the air flow rate, CO, CO_2, O_2, HC and smoke level were measured by exhaust probes connected to a tailpipe. These measurements were repeated in injection pressures of 200, 220, 250 and 300 bars.

Mode No.	Speed	Load (%)	Weighting factors
1	Idle	0	0.25/3
2	Maximum torque speed	10	0.08
3		25	0.08
4		50	0.08
5		75	0.08
6		100	0.25
7	Idle	0	0.25/3
8	Rated power speed	100	0.10
9		75	0.02
10		50	0.02
11		25	0.02
12		10	0.02
13	Idle	0	0.25/3

Table 2. 13-mode ECE-R49 Test

No	Speed (rpm)	Torque (N.m)
1	920	3
2	1760	20.4
3	1760	51
4	1760	102
5	1760	153
6	1760	204
7	920	3
8	2400	187
9	2400	140
10	2400	93
11	2400	46
12	2400	19
13	920	3

Table 3. 13-mode ECE-R49 Test based on engine performance

1.4. EXHAUST EMISSIONS CALCULATION

In the tests, CO, CO_2, UHC and O_2 were measured by Plint-RE205 gas analyzer. Soot was also measured by AVL-415 smoke meter in each test. Other species in exhaust gas were calculated with the following chemical equation:

$$mC_xH_y + z(O_2 + 3.76N_2) \longrightarrow v_1C + v_2CO + v_3CO_2 + v_4N_2 + v_5NO + v_6O_2 + v_7C_6H_{14} + v_8H_2O \qquad (1)$$

The molecular weight of fuel and its molecular formula are considered as a heavy diesel fuel (200 kg/kmol, $C_nH_{1.7n}$), therefore the formula of fuel is $C_{14.59}H_{24.86}$ [11].

The unknown coefficients are v_1, v_4, v_5, v_8, z, m.. Exhaust density is calculated with equation (2). Fuel and air flow are also measured respectively with equations (3) and (4) by orifice, manometer and fuel gage measurement and characteristics. Then soot flow rate is calculated with equations (5) and (6). The unknown coefficients are calculated by using of equations (7), (8), (9), (10), (11) and (12).

Equation (13) is an extra checking equation.

$$\rho_e = \frac{P_e}{(287 \times [T_e + 273])} \tag{2}$$

In equation (2), T_e and T_e are the turbine outlet temperature and pressure.

$$\dot{m}_{air} = c_d \times A_o \times \sqrt{2 \times 9.81 \times \rho_l \times \Delta h_{orifice} \rho_{air}} \tag{3}$$

$$\dot{m}_f = \frac{50 \times 10^{-6}}{t_f} \times \rho_f \tag{4}$$

$$\dot{V}_e = \frac{\dot{m}_a + \dot{m}_f}{\rho_e} \tag{5}$$

$$\dot{m}_{soot} = \rho_{soot} \times \dot{V}_e \tag{6}$$

$$\frac{12v_1}{z * 4.76 * 29} = \frac{\dot{m}_{soot}}{\dot{m}_{air}} \tag{7}$$

$$\frac{200m}{z * 4.76 * 29} = \frac{\dot{m}_f}{\dot{m}_{air}} \tag{8}$$

$$mx = v_1 + v_2 + v_3 + 6v_7 \tag{9}$$

$$my = 14v_7 + 2v_8 \tag{10}$$

$$2z = v_2 + 2v_3 + v_5 + 2v_6 \tag{11}$$

$$2 * 3.76z = 2v_4 + v_5 \tag{12}$$

Plint-Re205 measures emissions in a mixture of CO, CO_2, N_2, NO, O_2, C_6H_{14} and H_2O, so

$$v_2 + v_3 + v_4 + v_5 + v_6 + v_7 + v_8 = 100 \tag{13}$$

Specific soot emission is calculated in ECE-R49 standard test with equation (14) [12].

$$S_{soot} = \frac{\sum \dot{m}_{soot} \times w_f}{\sum p_b \times w_f} \tag{14}$$

The mass flow species are calculated by using of equations (15), (16) and (17)

$$M_m = \sum y_i \, M_i \tag{15}$$

In equation (15), $M_m, y_i \; and \; M_i$ are molecular weight of mixture, mole fraction and molecular weight of species in exhaust flow respectively.

$$\dot{m}_m = \dot{m}_f + \dot{m}_a \tag{16}$$

In equation (16), $\dot{m}_m$ is the mass flow of exhaust.

$$\dot{m}_i = \frac{ppm \times \dot{m}_m \times M_i}{M_m} \tag{17}$$

In equation (17), *ppm* is the particle per million of species.

Specific NO_x emission, specific HC emission, specific CO emission, specific O_2 emission, and specific CO_2 emissions are calculated in ECE-R49 standard test by equations (18-22).

$$S_{\mathrm{NO_X}} = \frac{\sum \dot{m}_{\mathrm{NO_X}} \times w_f}{\sum p_b \times w_f} \tag{18}$$

$$S_{\mathrm{HC}} = \frac{\sum \dot{m}_{\mathrm{HC}} \times w_f}{\sum p_b \times w_f} \tag{19}$$

$$S_{\mathrm{CO}} = \frac{\sum \dot{m}_{\mathrm{CO}} \times w_f}{\sum p_b \times w_f} \tag{20}$$

$$S_{\mathrm{O_2}} = \frac{\sum \dot{m}_{\mathrm{O_2}} \times w_f}{\sum p_b \times w_f} \tag{21}$$

$$S_{\mathrm{CO_2}} = \frac{\sum \dot{m}_{\mathrm{CO_2}} \times w_f}{\sum p_b \times w_f} \tag{22}$$

Brake specific fuel consumption is calculated with equation (23).

$$bsfc = \frac{\sum \dot{m}_f \times w_f}{\sum p_b \times w_f} \tag{23}$$

In equations (14), (18), (19), (20), (21), (22) and (23), w_f is the weighting factor related to ECE-R49 test. p_b is brake power in each mode.

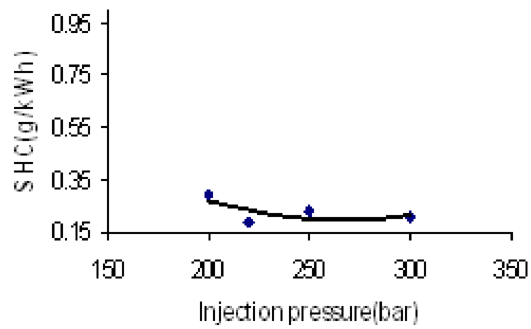


Figure 3.The effect of injection pressure on specific HC

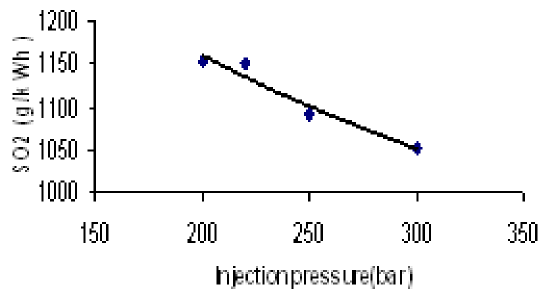


Figure 4.The effect of injection pressure on specific O_2

2. TEST RESULTS AND DISCUSSIONS

Exhaust emission measurements were made from injection pressure of 200 bars (initial injection pressure) to injection pressures OF 220, 250 and 300 bars. The variation of $S_{\mathrm{HC}}, S_{\mathrm{O_2}}, S_{\mathrm{CO}}, S_{\mathrm{CO_2}}, S_{\mathrm{NO_X}}$and S_{soot} against injection pressure are shown in Figures 3, 4, 5, 6, 7, and 8 respectively. Figure 3 shows that specific HC emission is reduced when injection pressure is increased due to a better mixing of air and fuel, but if injection pressure is too high because of impinging on the body of the cylinder and cooling, its value increases. As it can be seen from Figure 4, oxygen emission is reduced when injection pressure is increased due to lower ignition delay and higher combustion rate at better fuel air mixing. As shown in Figure 5, CO values increase as injection pressure is increased. This is because of higher exhaust temperature (Figure 9) (or higher combustion temperature) which causes dissociated CO_2 to CO and O_2 with higher injection pressures. As Figure 6 shows, the amount of specific CO_2 emission is different from the increase in injection pressure. As long as fuel air mixture is improved, CO_2 emission increases because of better combustion efficiency. But with further injection pressure, exhaust temperature increases (Figure 9). This causes dissociation of CO_2 and lower CO_2. So the result shows that specific CO_2 emission behaves differently with injection pressures (Figure 6).

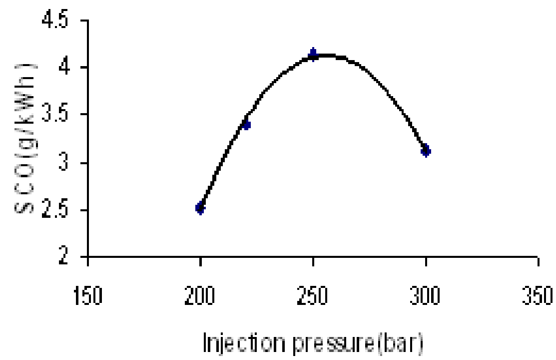


Figure 5.The effect of injection pressure on specific CO

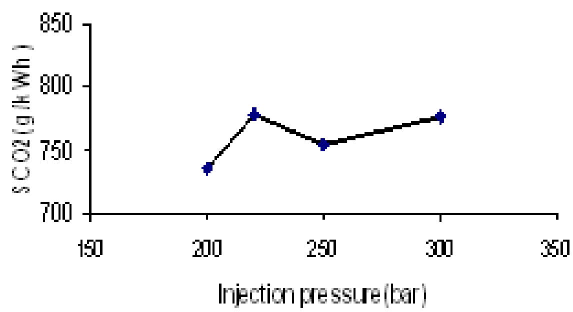


Figure 6.The effect of injection pressure on specific CO_2

Specific NO_x emission as shown in Figure 7 is almost constant in spite of variation of injection pressure because the peak temperature is not changed too much. As it can be seen in Figure 8 the highest soot value (0.91g/kWh) and the lowest value (0.51g/kWh) are obtained at 300 and about 235 bars respectively.

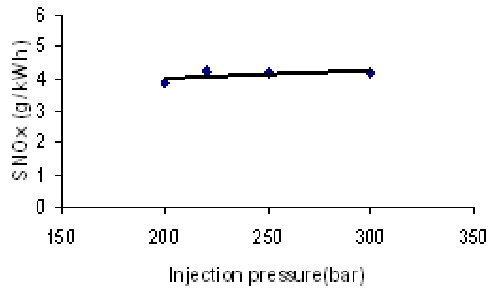


Figure 7.The effect of injection pressure on specific NO_X

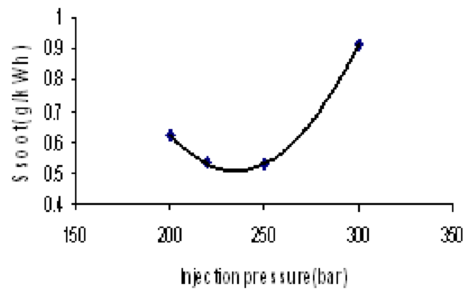


Figure 8.The effect of injection pressure on specific soot

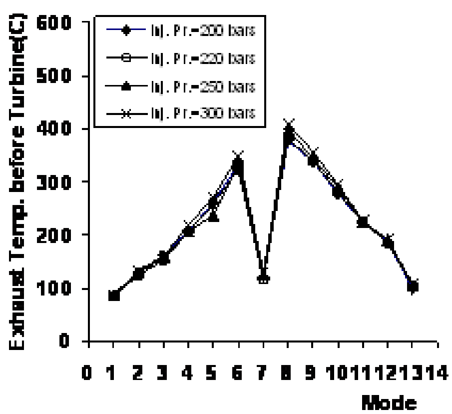


Figure 9.The effect of injection pressure on exhaust temperature

Smoke level is reduced when injection pressure is increased to 235 bars; after that smoke level increases when injection pressure increases to 300 bars. Since formation of the mixing of fuel to air becomes better during the ignition period, smoke level emission will be less. This is because lower ignition delay and better combustion rate prepare more time for the soot to be re-burned at late combustion phase. But, when injection pressure increases to more than 235 bars, ignition delay period becomes shorter because cylinder temperature increases with higher injection pressure. Accordingly, possibilities of homogeneous mixing decrease [9]. Therefore, smoke level is increased. As shown in Figure 10, engine specific fuel consumption (*bsfc*) is improved with higher injection pressure due to lower ignition delay and higher combustion rate at better fuel air mixing.

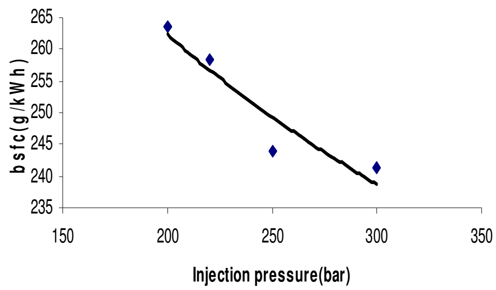


Figure 10.The effect of injection pressure on brake specific fuel consumption

CONCLUSIONS

Engine tests have been done based on ECE-R49 test in 13-speed-and-load conditions. It is shown that additional injection pressure is necessary for reducing *bsfc* and HC, which causes faster combustion. Variations of CO are generally different according to the injection pressure due to effect on combustion temperature which causes dissociated CO_2 to CO and O_2. NO_x emission is not substantially changed by injection pressure because the peak temperature is not changed too much. The amount of CO_2 emission depends on two parameters (i.e. combustion rate and exhaust temperature); so CO_2 emissions are not at exact levels in different injection pressures. Smoke level has differently changed according to injection pressure. It is shown in this study that minimum smoke level is achieved in 235 bars by variation of 200 to 300 bars injection pressure. This is because of injection pressure roles in ignition delay and exhaust temperature effects on re-burning soot at late combustion phase. Fuel economy is very important for engines. But environmental conditions are more important than the economy. As a result, the control of injection pressures is very important in diesel engines. So this research work proposes variable injection pressure instead of constant injection pressure.

NOMENCLATURE

w_f	weighting factor
p_b	brake power (kW)
$\dot{m}$	mass flow rate (kg/s)
$\dot{V}$	volume flow rate(m^3/s)
y	mole fraction of species
P	pressure (kPa)
C_d	discharge coefficient of orifice
$\Delta h_{orifice}$	monometer pressure drop
ρ	density (kg/m^3)
S	specific emission (g/kWh)
T	temperature(K)
b	brake
e	exhaust
f	factor
i	species

ABBREVIATIONS

bsfc	brake specific fuel consumption (g/kWh)
DI	direct injection

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