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The Effect of Compression Ratio and Alternative Fuels on Performance and Exhaust Emission in a Diesel Engine by Modelling Engine

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ABSTRACT: This study investigates the effect of compression ratio and different fuels on engine performance and exhaust emissions in a 6.8L turbocharged industrial diesel engine. For carried out this work, a 6 cylinder four stroke engine with gamma technologies power software is modelled and the effect of compression ratio (15:1 - 19:1) and alternative fuels (Diesel, Ethanol, Methanol, Decane, Soybean biodiesel, Diesel- Ethanol) at wide open throttle and various speeds from 800-2400 rpm are presented. The results indicate that the brake specific fuel consumptions of decane fuel at a compression ratio of 17:1 is lower than those of other fuels and also the maximum brake torque obtained with decane fuel at 1400 rpm. At this engine observed that decane fuel has higher brake power as compared to other fuels used due to higher heating value content. The emission results show that diesel fuel emitted more Carbon monoxide and Carbon dioxide emissions but soybean biodiesel (B100) has less Carbon monoxide, whereas highest oxides of nitrogen is founded with soybean biodiesel. Carbon monoxide and Carbon dioxide emissions are very close to each other when used decane and diesel fuel. In general decane fuel has higher performance and soybean biodiesel had fewer emissions at a compression ratio of 17:1.

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1- Introduction

To reduce concerns on exhaust emissions of vehicles, the use of non-fossil fuels and friendly with the environment is inevitable [1, 2]. In addition to the fact that some fuels reduce pollutant emissions, the Compression Ratio (CR) also plays a significant role on the performance and reduction of pollutants in cars. [3, 4]. One of the most important operating parameters to reduce the emissions to an acceptable level in the internal combustion engines is compression ratio and the use of alternative fuels with different cetane number [5-9]. Cenk Sayin & Metin Gumus [10] showed that with the increase of compression ratio, NO_x emissions in a diesel engine increase while carbon monoxide (CO) and hydrocarbons emissions decrease. In the India University of Dindigul, the researchers indicated that in a diesel engine, the most brake power and low mean effective pressure obtained at the Compression ratio 21 and 40% biodiesel fuel in the blend [11]. A performance and exhaust emission Comparison was done by Jindal et al. [12], they prove that variable compression ratio along with the change of injection pressure increase fuel consumption, brake thermal efficiency and decrease emissions. Also, another researcher showed that to cut environmental problems in the diesel engines, variable compression ratio and use of Alternate fuels are the main goals [13]. In a diesel engine, the least brake specific fuel consumption and brake specific energy consumption was obtained at compression ratio 21 [14]. Use of huge oil methyl ester in a diesel engine cause emissions such as Smoke, Carbon monoxide, Hydrocarbon except for the oxides of nitrogen (NOx) reduce [15].
adding 20% petroleum-based diesel fuel to methanol, ethanol, and waste frying oil in a Direct Injection (DI) diesel engine resulted that the Brake Specific Fuel Consumptions (BSFC) of ester fuels were higher than those of petrodiesel [16, 17]. Also, ester fuels emitted less carbon monoxide and total hydrocarbon emissions, but they caused to produce more NO_x and Exhaust carbon dioxide (CO_2) was very close to each other [18, 19]. Ravi et al. [20] converted a single cylinder Compression Ignition (CI) engine to Spark Ignition (SI) engine to operate as a liquid pressure gas fuel, the effect of compression ratio and the blend of hydrogen fuel with 15% hydrogen substitution on engine performance, exhaust emissions, and combustion behavior were studied and compared. There was a perceptible improvement in brake thermal efficiency, brake power and considerable reductions in hydrocarbon, CO and CO2 levels [21, 22]. Another study [23-25] carried out on dual fuel diesel engine with variable compression ratio for investigation on the performance and emission characteristics using biomass-derived producer gas, the results by experimental in diesel engine fueled on 2.5% and 5% ethanol supplement demonstrate a decrease in amount of CO and smoke emissions fueled by higher percentage of ethanol. More heat release rate and cylinder pressure fueled obtained with 5% ethanol in LemonGrass Oil25 (LGO25). Hydrocarbons emission decreased about 63.62% due to the increment of the compression ratio from 12 to 18 at 3.2 kW brake power [26, 27].
By numerical and experimental results in a compression ignition engine, it was concluded that with the increase of the compression ratio, the maximum in-cylinder pressure and brake thermal efficiency increased, whereas exhaust gas temperature reduced [28, 29]. By increasing the compression ratio and reaching it to 19.5: 1, brake thermal efficiency increased about 21.18%. The use of a diesetrol fuel mixture caused the engine power and torque did not remain steady and hydrocarbon contaminants decreased by about 10% and carbon monoxide increased by about 7% [30, 31]. Earlier

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works also indicate that improvement in performance and emissions of gas engines can be obtained by enhancing the compression ratio [32, 33].

In the present study, the effects of compression ratio were studied on alternative fuels CI engine to obtain performance, emission and combustion characteristics at full throttle condition. Operation parameters like brake power, brake specific fuel consumption, exhaust emissions such as CO, CO_2 and NO_x were studied and compared. The results showed that both factors of compression ratio and fuel type are very important for optimizing engine operating conditions.

2- Simulation Setup and Engine Specification

Gamma Technologies (GT) Power software is based on one-dimensional modeling from GT-SUITE, representing the flow and heat transfer in pipes, turbocharged diesel engine and other components of an engine system. In addition, It has been used in many other specialized models can be applied for many kinds of system analysis [34, 35]. It is known that power is the maximum power that an engine can put out and described as work per unit of time. The power output of a diesel engine depends on the size and design of engine volume, but also on the speed at which it is running and the load or torque. The torque of an engine depends on the power generated by an engine. Fig. 1 presents the computational model of a six-cylinder with a turbocharger, direct injection, and CI engine.

In the present study, the effects of compression ratio were studied on a four-stroke, direct-injection diesel engine. For the purpose of selecting the appropriate design, input data corresponding to a real engine and the specifications outlined in Table 1. The GT-SUITE environment provides a useful set of high-productivity features and the data were analyzed graphically using GT-Power. The fuel properties used in this study are given in Table 2. In this table can see the difference of oxygen content, heating value and boiling points at various fuels compared to diesel.

Table 1. Specification of the 6068HF275- Industrial Diesel Engine

Engine parameters	Value
Model	6068HF275- Industrial Diesel Engine
Engine type	6.8*L*
Bore & Stroke(mm)	106x127
Aspiration	Turbocharged and air-to-air after cooled
Displacement(*L*)	6.8*L*
Number of Cylinder	6
Compression ratio	17.0:1
Ignition system	Direct injection engine

2- 1- Fluid dynamics governing equations

The flow model involves the solution of the Navier-Stokes equations, namely the conservation of continuity, momentum and energy equations. Which are calculated according to Eqs. (1) to (4). In the GT-Power software, these equations are solved in a one-dimensional fashion. This means that all the equations are in the direction of the averaging. In this study, explicit solving was used to solve the equations, in which base variables are explicitly solved in mass flow, density, and

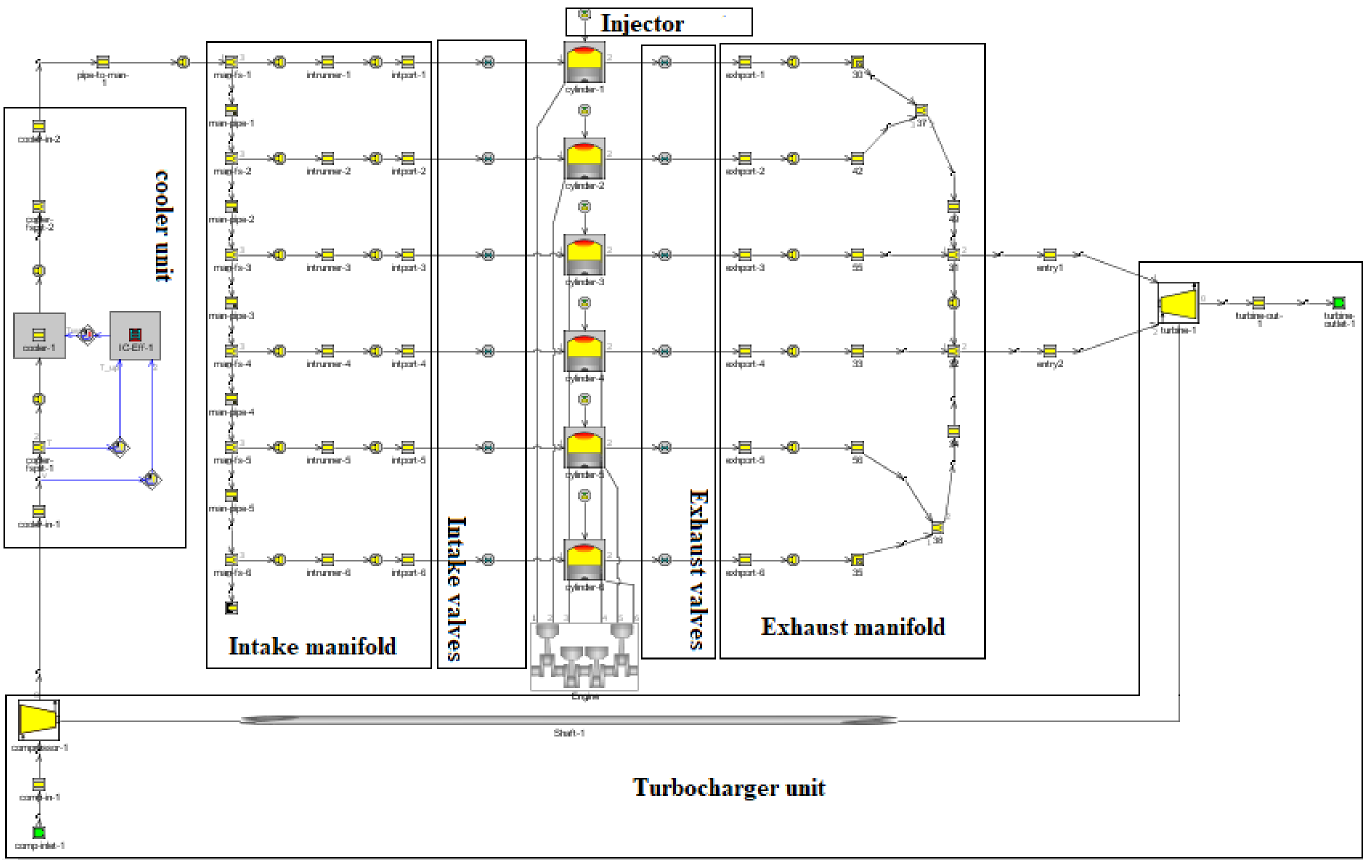

Fig. 1. Computational model of a six-cylinder, direct-injection, compression ignition engine

Table 2. The fuel properties used in this study [36]

fuel properties	unit	Diesel	Ethanol	Methanol	Biodiesel	Decane	Diesel-Ethanol
Density	kg m^{-3}	830	785	792	890	727	825
Heat vaporization at 298K	MJ kg^{-1}	0.25	0.92	1.17	0.35	0.36	0.317
Oxygen Atoms per Molecule	…	0	1	1	34.39	0	0
Lower Heating Value	MJ kg^{-1}	43.25	27.73	21.11	37.11	44.62	41.7
Critical Temperature	K	569.4	516	513	785.87	617.8	564.06
Critical Pressure	bar	24.6	6.38	79.5	12.07	21.1	22.77

internal energy. In explicit solving, the system is divided into small volumes, in which all the splitters are subdivided into a sub volume and all tubes of one volume or more. The scalar variables (pressure, temperature, density, internal energy, enthalpy, etc.) are assumed to be uniform on the boundary of each of the underlying volumes. Vector variables (mass flux, velocity, mass fraction flux, etc.) are calculated for each boundary.

$$\frac{dm}{dt} = \sum_{boundaries} \dot{m} \tag{1}$$

$$\frac{dme}{dt} = -P\frac{dV}{dt} + \sum_{boundaries} \dot{m}H - hA_s(T_{fluid} - T_{wall}) \tag{2}$$

$$\frac{dpHV}{dt} = +V\frac{dp}{dt} + \sum_{boundaries} \dot{m}H - hA_s(T_{fluid} - T_{wall}) \tag{3}$$

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{boundaries} \dot{m}u - 4C_f \frac{\rho u|u|}{2}\frac{dxA}{D} - C_p(\frac{1}{2}\rho u|u|)A}{dx} \tag{4}$$

In which $\dot{m}$ is boundary mass flux into volume, m is mass of the volume, V is volume, P is pressure, ρ is density, A is cross-sectional flow area, A_s is heat transfer surface area, e is total specific internal energy, H is total specific enthalpy, h is heat transfer coefficient, T_{fluid} is fluid temperature, T_{wall} is wall temperature, u is velocity at the boundary, C_f is Fanning friction factor and C_p is specific heat.

2- 2- Time step calculation

In this software, the choice of the time step is related to the type of solver used. In the explicit method, pressure, temperature, etc. are calculated directly and without repetition, try and error. The relationship between the time step and the length of the discretization is created through the Courant number. The time step must be selected in such a way that the Eq. (5) is established.

$$\frac{\Delta t}{\Delta x}(|u| + C) \leq 0.8 \times m \tag{5}$$

In which Δt is time step (s), Δx is minimum discretized element length, m is time step multiplier specified by the user in run setup and C is speed of sound.

2- 3- Heat transfer

2- 3- 1- Heat loss and surface roughness effect

The heat transfer from fluids inside of pipes and flow split to their walls is calculated using a heat transfer coefficient. The heat transfer coefficient is calculated at every time step from the fluid velocity, the thermo-physical properties, and the wall surface roughness. The heat transfer coefficient of smooth pipes is calculated using the Colburn analogy. Which is presented in Eq. (6).

$$h_g = \frac{1}{2}C_f \rho U_{eff} C_p \Pr^{-\frac{3}{2}} \tag{6}$$

The surface roughness attribute in pipe components can have a very strong influence on the heat transfer coefficient, especially for very rough surfaces such as cast iron or cast aluminum. In this case, first, the value of h is obtained from Eq. (6), then corrected with the help of Eq. (7).

$$h_{g,rough} = h_g(\frac{C_{f,rough}}{C_f})^n \tag{7}$$

In which $h_{g,\,rough}$ is heat transfer coefficient of rough pipe and $C_{f,\,rough}$ is Fanning friction factor of rough pipe. In which U_{eff} is effective velocity outside boundary layer and Pr is Prandtl number.

2- 3- 2- In-cylinder heat transfer model

The best and most used heat transfer model in the engine is the Woschni relationship. This model is used when the Swirl and Tumble coefficients in the model are considered zero. In this regard, the heat transfer rate is calculated as Eq. (8) [37].

$$\dot{Q} = h_c A_{cw}(T_{cw} - T_c) \tag{8}$$

The convective heat transfer coefficient for the Woschni models is defined as follows:

$$h_c = \frac{3.26p^{0.8}w^{0.8}}{B^{0.2}T^{0.53}} \tag{9}$$

where the average in cylinder gas velocity w is correlated as follows:

$$V = C_1\overline{S}_p + C_2\frac{V_d T_r}{P_r V_r}(P - P_m) \tag{10}$$

In which w is Average cylinder gas velocity, B is Cylinder bore, C_1 & C_2 are constants, $\overline{S}_p$ is Mean Piston Speed, V_d is Displaced volume, P_r is Working fluid pressure prior to combustion, T_r is Working fluid temperature prior to combustion, P_m is Motoring fluid pressure prior to combustion and V_r is Working fluid volume prior to combustion.

2- 4- Combustion modeling

The approximation of the analytic functions of combustion rate in internal combustion engines is useful and cost-effective tools for simulating the engine cycle. In this research, we used the Wiebe function to predict burns in internal combustion engines that work with different combustion and fuel systems. The purpose of using a function is to model the premixed combustion and distribute different parts of the combustion process. When the amount of pressure in the cylinder is not available or measurable to calculate the burning rate, this model provides an easy way to obtain a reasonable burning rate [38]. Also, the number of combustion temperature zones was considered to be two-zone because, according to the advice of the GT-Power software, this mode is used for most common calculations and calculates the temperature and pressure independently for burned and unburned gases. In addition, selecting this mode makes the results of NOx calculations more meaningful and accurate.

$$\begin{aligned} Comb(\theta) &= (CE)(F_p)[1-e^{-(WC_p)(\theta-SOI-ID)^{(E_p+1)}}] \\ &+(CE)(F_M)[1-e^{-(WC_M)(\theta-SOI-ID)^{(E_M+1)}}]+ \\ &(CE)(F_T)[1-e^{-(WC_T)(\theta-SOI-ID)^{(E_T+1)}}] \end{aligned} \quad (11)$$

The values of F_M , WC_p , WC_M and WC_T are obtained from following relationships.

$$F_M = 1 - F_T - F_P \quad (12)$$

$$WC_P = [\frac{D_p}{2.302^{1/(E_P+1)} - 0.105^{1/(E_P+1)}}]^{-(E_P+1)} \quad (13)$$

$$WC_M = [\frac{D_M}{2.302^{1/(E_M+1)} - 0.105^{1/(E_M+1)}}]^{-(E_M+1)} \quad (14)$$

$$WC_T = [\frac{D_T}{2.302^{1/(E_T+1)} - 0.105^{1/(E_T+1)}}]^{-(E_T+1)} \quad (15)$$

In which *SOI* is Start of Injection, *ID* is Ignition Delay, *CE* is Fraction of fuel burned, F_p is Premix Fraction, D_p is Premix Duration, D_M is Main Duration, D_T is Tail Duration, F_T is Tail Fraction, F_M is Main Fraction, E_P is Premix Exponent, E_T is Tail Exponent, E_M is Main Exponent, θ is Instantaneous Crank Angle, WC_p is Wiebe Premix Constant, WC_M is Wiebe Main Constant and WC_T is Wiebe Tail Constant.

2- 5- Friction modeling

In this research, the Chenn-Flynn model was used to model mechanical friction, which is presented in the Eq. (16).

$$\begin{aligned} FMEP &= C + (PF \times P_{\max}) + (MPSF \times Speed_{mp}) \\ &+(MPSSF \times Speed_{mp}{}^2) \end{aligned} \quad (16)$$

In which *PF* is Peak Cylinder Pressure Factor, P_{max} is Maximum Cylinder pressure, *MPSF* is Mean Piston Speed Factor, $Speed_{mp}$ is Mean Piston Speed, *MPSSF* is Mean Piston Speed Squared Factor, *FMEP* is Friction Mean Effective Pressure and *C* is Constant part of *FMEP* .

3- Model Validation

In the last years, numerical simulation has seen a great development thanks to costs reduction and speed increases of the computational systems. With these advancements, mathematical algorithms are able for performing correctly with realistic problems. To verify the quality and the problem of numerical simulation, validation models are necessary to quantify the similarity between reality and simulations. To confirm that the results of a numerical solution are fully consistent with reality, it must ensure that the numerical method solves exactly the mathematical modeling equations.

3- 1- Validation of the model using experimental and simulation results

The most common validation method is the use of experimental results due to the fact that the measurements indicate that the model is consistent with reality. To confirm the results of a simulation, it is essential to achieve the highest similarity between simulation and measurement settings. The most important issue for the computing simulation environment is to try to model the whole set or at least the most important features related to the engine. Otherwise, the simulation results do not indicate the risk of unrealistic results, and causing a validation error.

For this study, the experimental results obtained by engine test bed, have been used to validate the model. Therefore, the engine operating conditions imposed in this investigation have been used; the experimental result used for validation purposes is engine torque and power.

Fig. 2 shows the evolution of torque and power as a function of speed for validation between experimental results and calculated data from simulation in this study. Calculating percentage error allows comparing an estimate to an exact value. The percentage error gives the difference between the approximate and exact values as a percentage of the exact value and can help to see how close experimental data or estimate was to a real value. The formula for calculating percentage error is as below:

$$\text{percentage error} = \frac{|\text{simulation value} - \text{experimental value}|}{\text{experimental value}} \times 100 \quad (17)$$

So, with using of above formula and experimental and simulation data, the percentage error is low. The results show that the model fits the experimental torque and power data with an error lower than 6.12%. Thus, it is acceptable for validation of software [40-42]. Hogg studies demonstrated that 20% error in calculation has a good agreement in evaluation and validation between experiment and numerical models [43].

4- Present Work

For conducted this investigation, the effects of compression ratio and various fuels were studied in compressed ignition engine to obtain performance, exhaust emissions and combustion characteristics at full load condition.

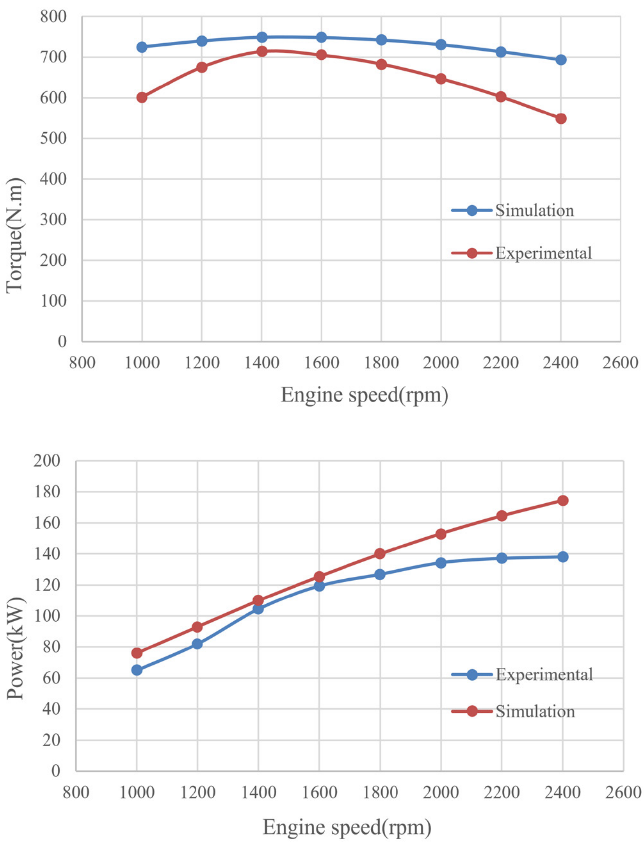

Fig. 2. Curve of engine torque and power versus engine speed (comparison of experimental and simulation data) [39]

Subsequently, a turbocharged engine with different compression ratios and alternative fuels was performed to check its combustion characteristics to obtain the best fuel and operational conditions. A six-cylinder CI engine with a peak torque of 740-930 N.m (546-686 lb-ft) at 1400 rpm was modeled to operate at different conditions. The response of enhancing the compression ratio on engine performance and emissions were studied at unit equivalence ratio. Four compression ratios namely 15:1, 16:1, 17:1, 18:1 and 19:1 were tried. Also, at full load operation used various fuels like Diesel, Ethanol, Methanol, Decane, Soybean biodiesel, Diesel-Ethanol that was found optimal fuel in this engine. Injection duration selected 20 degrees of crank angle and injected mass was adjusted 96 mg for 1400 rpm. Performance parameters like brake power, emissions of HC, CO, NO_x, and heat release rate were studied and compared.

5- Results and Discussion

5- 1- Brake specific fuel consumption

The variation of brake specific fuel consumption for different compression ratios and for various fuels are given in Figs. 3 and 4. It has been observed that brake specific fuel consumption of decane fuel is lower than that of all other fuels at compression ratio 18 and 19 and is shown in Fig. 2. This is due to fuel density, viscosity, and fuel heating value. The energy value of decane fuel is higher than other fuels and is close to diesel and diesel- ethanol, as shown in Table 2. The Brake specific fuel consumption decreases with an increase in compression ratio [43]. The brake specific fuel consumption of the decane fuel is equal 212.135 g/kWh at the compression ratio of 19 whereas for the same fuel at the compression ratio of 19 is equal 215.725 g/kWh. At other fuels, the brake specific fuel consumption increases. Low values of brake fuel consumption are obviously in Fig. 2 and the lowest amount related to methanol fuel. Can also be operated the lowest brake specific fuel consumption at 1400 rpm.

The variation of brake specific fuel consumption is presented in Fig. 4 when various fuels are compared to pure diesel fuel. As illustrated in the Fig. 3, the brake specific fuel consumption generally increased with speed due to the high engine power. As shown in Fig. 3, the reduction in BSFC is 3.03% for decane fuel and the increments are 4%, 17.4%, 63.8% and 125.52% for Diesel-Ethanol, Soybean biodiesel, Ethanol, and Methanol, respectively, compared to the results of diesel fuel at 1400 rpm. Thus, this subject shows that engine needs to more fuel to be injected into the cylinder to get the same power output.

5- 2- Engine torque

From Fig. 5 it is observed that the engine brake torque increases as the compression ratio increases. This is due to the increase in brake power at high compression ratios. The turning effect on the cylinder cranks become greater with Increase in compression ratio. This means that the engine applies a higher pressure on the piston in expansion stroke and increases engine torque and power.

The variation of brake torque is observed in Fig. 6 when various fuels are compared to pure diesel fuel. As illustrated with the Fig. 6, the brake torque generally increased with the increase of engine speed. As shown in Fig. 6, the maximum brake torque obtained with decane fuel and the increment is 2.66% compared to the results of diesel fuel at 1400 rpm engine speed.

5- 3- Engine power

The brake power values for various fuels at different compression ratios are shown in Figs. 7 and 8. The figures show that the ethanol, methanol, soybean biodiesel and diesel- ethanol except for decane fuel with standard diesel fuel have a reduction in brake power.

Brake power increases at higher compression ratio due to the conversion from the chemical energy to mechanical energy. Due to the higher heating value of the fuels and stable combustion the brake power increases. The maximum brake power for decane and diesel fuel at compression ratio 19 was 180.80 kW and 175.5 kW respectively. The other fuels are also indicated at a reduction in brake power, with higher compression ratios due to a lower heating value of the fuel. The Fig. 8 shows Variation of brake power with engine speed for compression ratio 17:1. The results indicate that with increasing engine speed, the brake power increases and the highest power obtained for decane fuel due to the higher heating value.

5- 4- Exhaust emissions

5- 4- 1- CO emission

In this part, the changes in the exhaust emissions with different engine speed and various fuels were discussed. These exhaust emissions were carbon monoxide, carbon dioxide and oxides of nitrogen.

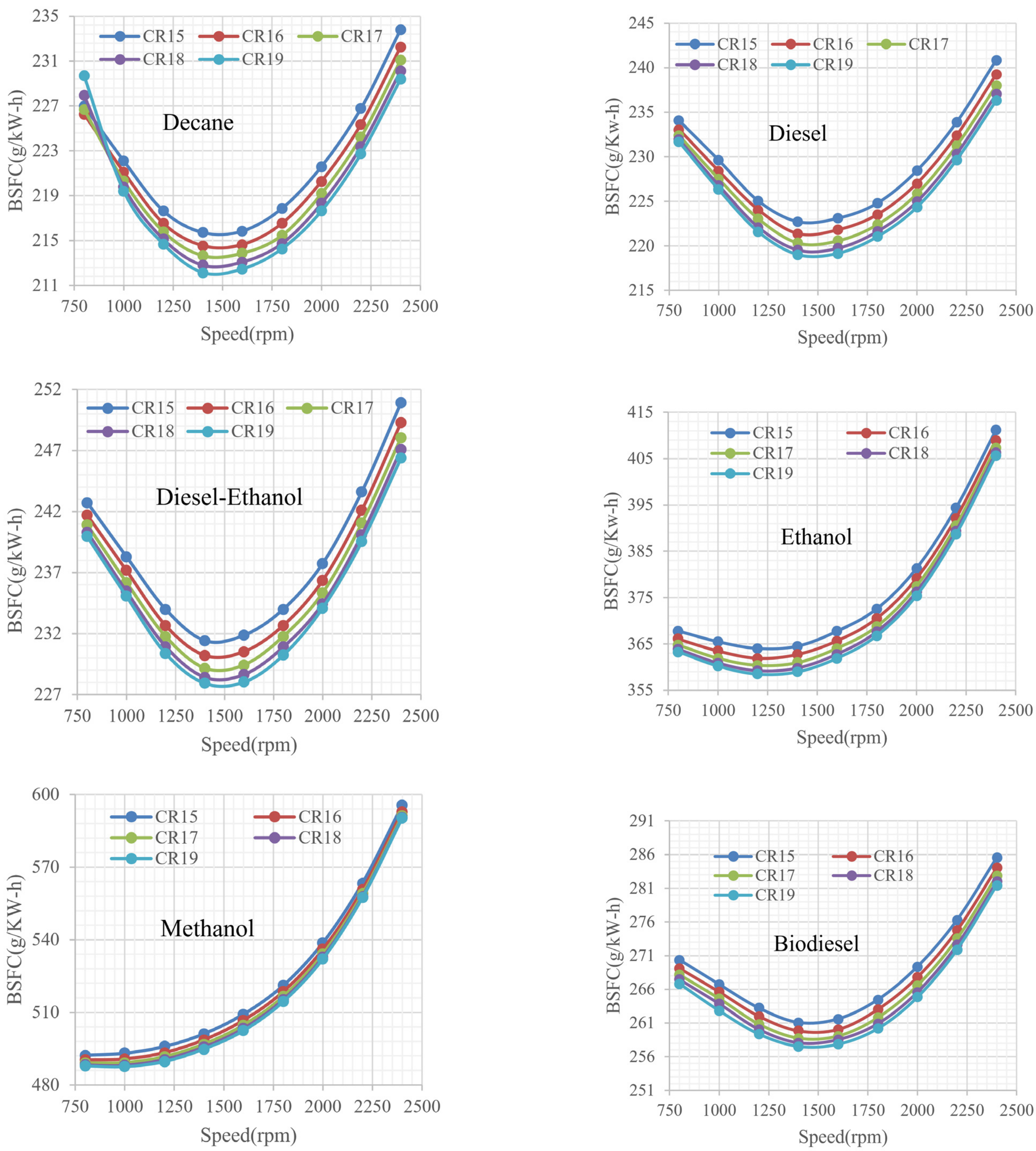

Fig. 3. Variation of brake specific energy consumption vs engine speed for different compression ratios and various fuels

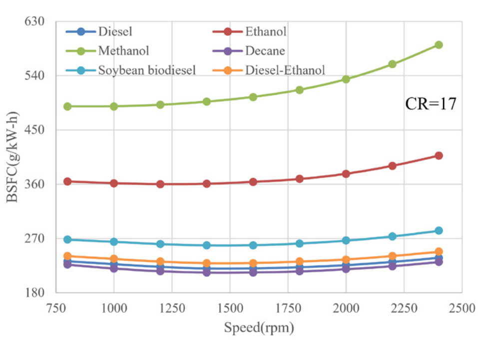

Fig. 4. Variation of brake specific fuel consumption with engine speed for compression ratio 17:1

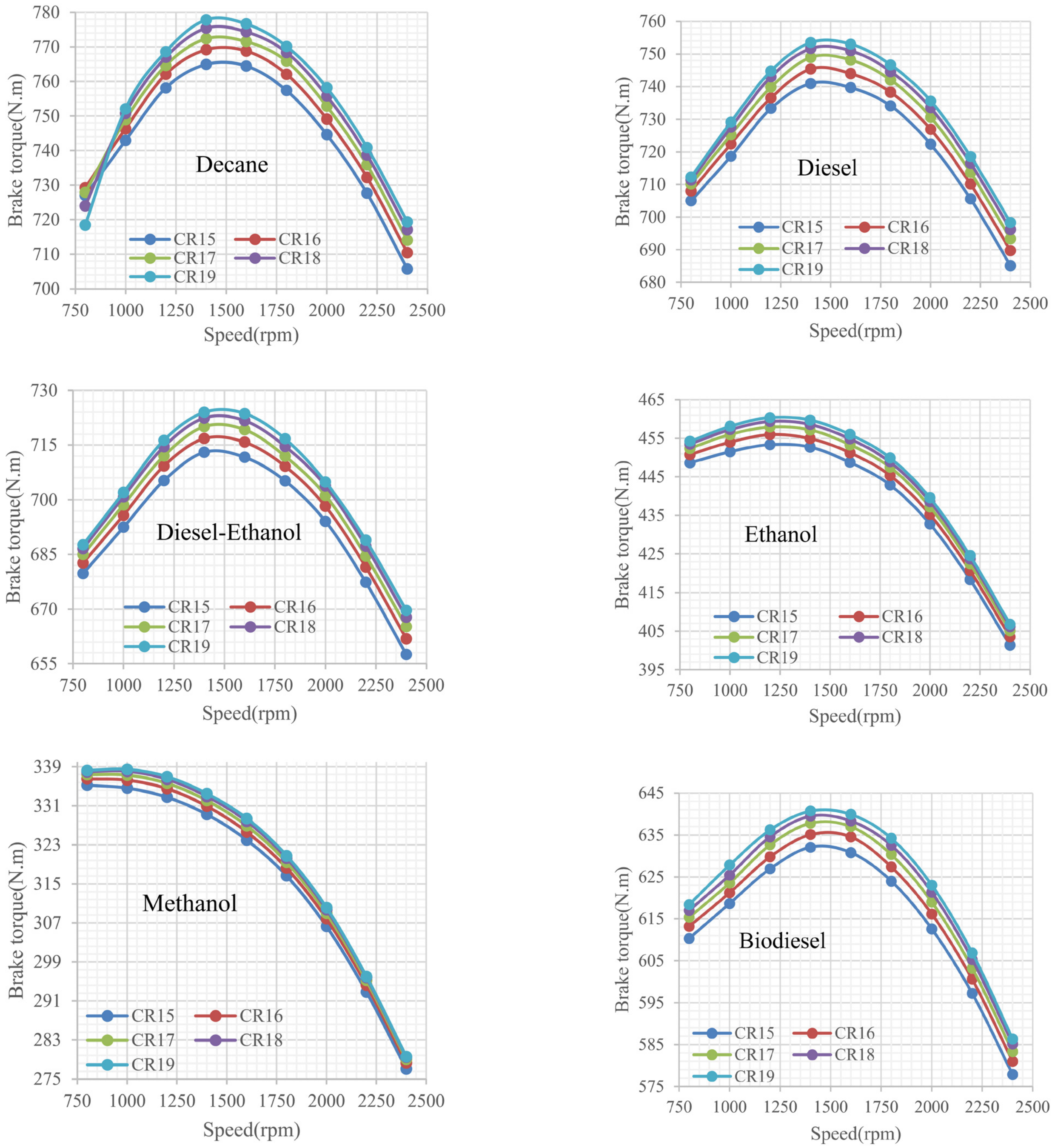

Fig. 5. Variation of brake torque vs engine speed for different compression ratios and various fuels

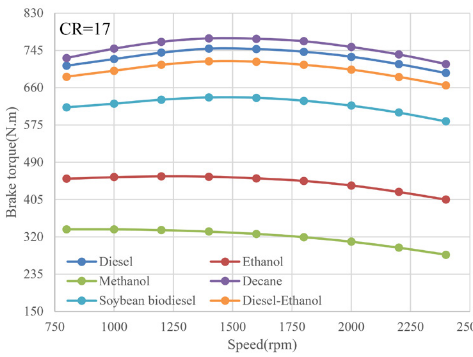

Fig. 6. Variation of brake torque with engine speed for compression ratio 17:1.

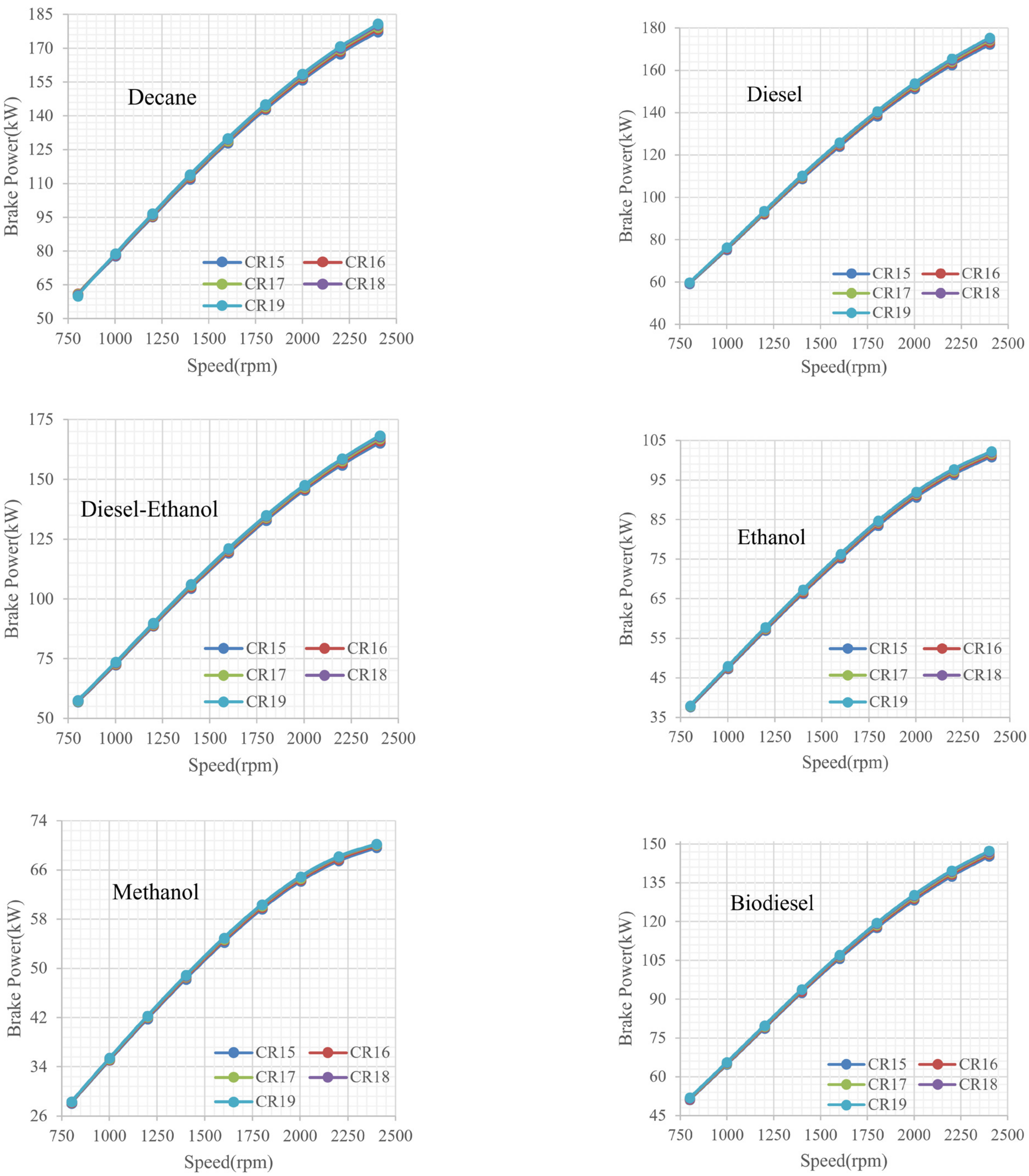

Fig. 7. Variation of brake power vs engine speed for different compression ratios and various fuels

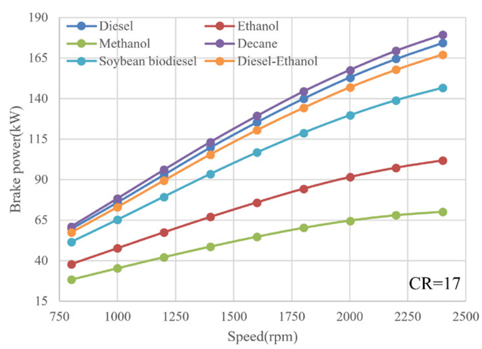

Fig. 8. Variation of brake power with engine speed for compression ratio 17:1

One of the most important parameter affecting on CO emission is Air/fuel ratio, which unburn and poor combustion is an indicator the lack of air in the fuel mixture for combustion. Furthermore, incomplete mixing of Air/fuel ratio in the combustion chamber, increase locally rich zones, so that for changing the CO to CO_2 needs to extra oxygen. As demonstrated in Fig. 9 CO emissions reduced with going up engine speed for all fuels. This reduction is more obvious for decane, diesel and blends of diesel-ethanol fuel and between 800 rpm and 1200 rpm. The high CO emissions at low engine speed related to decane and diesel fuel as result of an incomplete mixing of air and fuel due to rich fuel zones, low volumetric efficiency, fuel chemical combination, and high amount of inert exhaust gas.

Therefore, the oxygen concentration decreases in the combustion chamber and consequently combustion is affected negatively. Also, as seen in Fig. 9 Less CO emissions related to soybean biodiesel (B20%) and this is because of lean mixture, fuel chemical properties and more complete combustion. The amount of CO for methanol and ethanol fuels is very close to soybean biodiesel.

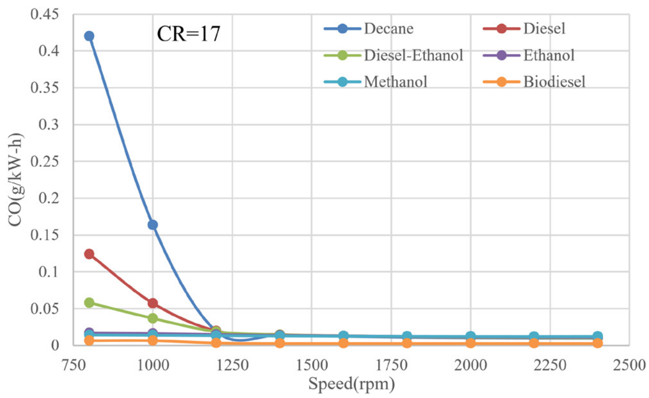

Fig. 9. Variation of CO emission with engine speed and different fuels

5- 4- 2- CO_2 exhaust emission

Today, greenhouse gases owing the wide use of fossil fuels has increased, especially carbon dioxide, which its effect on greenhouse is in high level. Fig. 10 compares the CO_2 emissions of various fuels used in the diesel engine at different speeds. The CO_2 emission increases with going up speed due to turbocharger existence. The methanol emits the highest amount of CO_2 in comparison with other fuels especially, at high speed so that while CO going up, the amount of CO_2 decreases. Diesel fuel emits a low level of CO_2 emissions. It is also founded that level of CO_2 emission for soybean biodiesel, methanol and ethanol fuel is very near to each other. More amount of CO_2 in exhaust emission is an indication of the complete combustion of fuel. This supports the higher value of exhaust gas temperature.

5- 4- 3- NOx emissions

NOx emissions from different fuels are shown in Fig. 11. NOx emission with a compression ratio of 17: 1 was observed. As shown in the figure below, high NOx emissions refer to soybean biodiesel. NOx formation is dependent on the reaction time, the reaction temperature and also prominently upon the availability of oxygen at a higher temperature [44, 45].

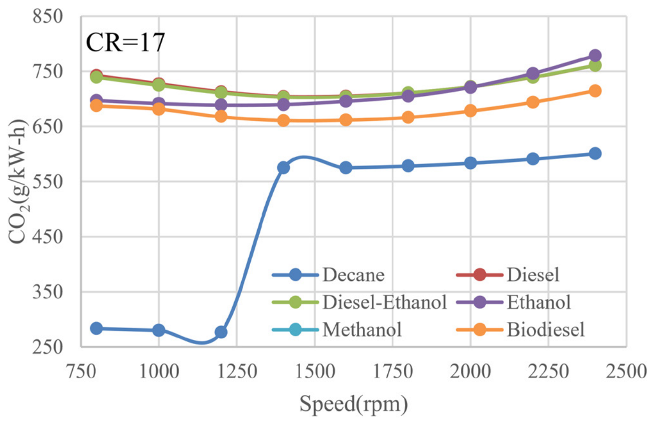

Fig. 10. Variation of CO_2 emission with engine speed and different fuels

The oxygen content of soybean biodiesel fuel increases the maximum cylinder's temperature and this tends to increase the NOx concentration. NOx concentration for soybean biodiesel compared to other fuel used in this investigation is high, especially at high speeds. In this engine, when soybean biodiesel is inducted, higher NOx formation is attributed to higher in-cylinder temperature due to faster energy release and peak cylinder pressure.

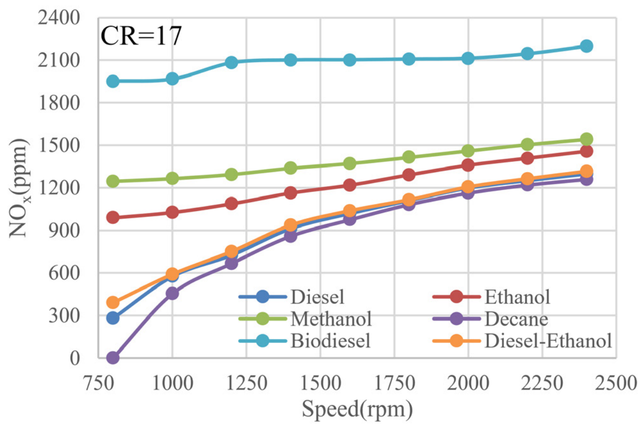

Fig. 11. Variation of NOx emission with engine speed and different fuels

6- Conclusions

In this paper, the effects of operating conditions such as compression ratio and various fuels on the engine performance and exhaust emissions of a six-cylinder diesel engine by modeling software investigated. Based on the obtained results, the conclusions can be drawn as follows:

1. The brake specific fuel consumption of the decane fuel is slightly lower than that of standard diesel at higher compression ratios and on compared to other fuels such as methanol and ethanol is strongly lower. This may be due to better combustion and increase in the heating value content of the decane fuel.
2. It was found that the increase of compression ratio has a positive effect and caused that brake power and torque increase and also results showed that with decane fuel can obtain the most power and torque in this engine compared to other fuels used.
3. Soybean biodiesel emitted less CO and more CO_2, NOx

emissions than petroleum-based diesel fuel. Exhaust emissions of methanol fuel are low and near to soybean biodiesel, but if used Exhaust Gas Recirculation (EGR) or a method which decrease NO_x, soybean biodiesel (B20%) is the best fuel in this research.

4. The results of the study showed that the decane fuel with a compression ratio of 19:1 in total had the best functional and pollutant characteristics among the six fuel used in this study. Therefore, this fuel can be the best alternative for diesel fuel.

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