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The effect of the turbocharger system and different fuels on the performance and exhaust emissions of a diesel engine by a numerical study

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A B S T R A C T

In internal combustion engines, the turbocharger and alternative fuels are two important factors affecting engine performance and exhaust emission. In this investigation, a one-dimensional computational fluid dynamics with GT-Power software was used to simulate a six-cylinder turbocharged diesel engine and the naturally aspirated diesel engine to study the performance and exhaust emissions with alternative fuels. The base fuel (diesel), methanol, ethanol, the blend of diesel and ethanol, biodiesel and decane was used. The results showed that decane fuel in the turbocharged engine has more brake power and torque (about 3.86%) compared to the base fuel. Also, the results showed that the turbocharger reduces carbon monoxide and hydrocarbon emissions, and biodiesel fuel has the least amount of carbon monoxide and hydrocarbon among other fuels. At the same time, the lowest NOX emission was obtained by decane fuel. As a final result can be demonstrated that the decane fuel in the turbocharged engine and the biodiesel fuel in the naturally aspirated engine could be a good alternative ratio to diesel fuel in diesel engines.

1. Introduction

Todays, due to environmental pollution problems and need to non-fossil fuels maximize the efficiency of fuels and increase the power output of diesel engines is considered inevitable. In Iran, especially the big cities, the pollution of buses and diesel vehicles is becoming a major dilemma. The lack of proper fuel, soot filters, and absence of a requirement for a technical inspection of diesel vehicles has led to an increase in mortality and the growth of lung cancer due to pollution. All of this shows that low-quality fossil fuels got up living costs and reduce the quality of life. Therefore, in order to reduce the incidence of lung cancer and other diseases, it is necessary to increase the fuel quality and its type according to the performance of the engine. For the solution of this crisis suggests the use of alternative fuels And the deployment of a turbocharger in the vehicle

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to increase the efficiency, engine performance and reduce emissions.

Fuels that are derived from plant and animal resources are called biofuels [1]. Alternative fuels such as biogas, bioethanol, and biodiesel are also used as biofuels [2]. Biodiesel is defined by ASTM as "fuel containing mono-alkyl esters of long-chain fatty acids derived from vegetable oils or animals fats, called B100" [3, 4]. Due to the higher ignition point of biodiesel ratio to diesel fuel, it is easier to transport and maintain, so that with combining it with diesel fuel, fuel quality increased and harmful emissions in diesel vehicles reduced. [5, 6]. Although there are several methods for producing biogas fuels, today there is a method known as the environmentally friendly method, which is the transesterification reaction of oils obtained from Planta Oilseed [7, 8]. Noorollahi et al. [9] studied the effects of diesel fuel mixes (diesel, biodiesel, and ethanol) on exhaust emissions in a diesel engine. The experimental results showed that hydrocarbon emissions reduced by about 10% and carbon dioxide was about 7% higher. And the best efficiency, and lowest emission obtained with the D91B6E3 fuel mixture. In other studies, the effect of methanol on the performance and combustion in a diesel engine was investigated. Experimental and numerical results indicated that the engine performance improved with methanol fuel and CO and HC emissions decreased, whereas, with increasing methanol, the amount of CO2 and NOX emissions were increased. [10-12]. An experimental study by Lahane, S. and K. Subramanian[13] with different percentage of diesel and biodiesel fuel on performance and emission characteristics of a diesel engine showed that engine torque with biodiesel and diesel (B100) compounds decreased by 2.7%. Also, decreased Carbon monoxide (CO) and hydrocarbon (HC), but nitrogen oxide (NOx) increased [13]. Other researchers investigated the effect of turkey fats (TRFB) for biodiesel fuel production on combustion in a diesel engine. It was observed that TRFB10, TRFB20, and TRFB50 combinations reduce smoke emissions and increase the amount of nitrogen oxide emissions [14].

The turbocharger is a very delicate air pump that delivers air to the engine by controlling the energy lost in the exhaust of the engine [15]. The compressor is often positioned between the air filter and engine manifold. In a study, researchers simulated turbocharging effects on the engine emissions. The results showed that braking power, NOX, CO, and CO2 emissions in the turbocharged engine are higher and specific emissions are lower than the naturally aspirated engine [16]. The specific fuel consumption of turbocharged tractors ITM485 and ITM800 are lower, power and tensile strength these tractors were significantly higher than that of none turbocharged tractor with ITM475 [17]. The experimental and theoretical results effect of a turbocharger on the performance and exhaust emissions of a diesel-fueled diesel engine showed that the brake thermal efficiency of the biodiesel fuel was slightly higher than that of diesel fuel [18, 19]. While the biodiesel reduced brake power, torque, and increased fuel consumption [20]. It was also observed that CO emissions in biodiesel fuel were lower than diesel fuel, while NOX emissions in biodiesel fuel were higher. The final results showed that the use of biodiesel improves performance and reduces CO emissions in a turbocharged engine than diesel [21]. Researchers investigated the use of turbocharger and

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Miller cycle in diesel engines [22, 23]. They showed that the best-known technique for increases engine performance and reduces exhaust emissions is the turbocharger. While the use of Miller cycle and steam injection method reduces brake power and engine brake thermal efficiency by 6.5% and 10%, TC boosts brake power and accelerates engine thermal efficiency by up to 18% and 12% [24].

For this study, first, a 6 cylinder diesel engine modelled by GT-POWER software and then evaluated the effect of turbocharger system and different fuels on the performance and exhaust emissions of a direct injection diesel engine. Finally, the best fuel for a turbocharged engine, with the lowest emissions and higher performance, has been identified.

2. Methodology

The research engine in this study is a six-cylinder direct injection diesel 6068HF275 made by John Deere USA. The characteristics of this engine are shown in Table 1.

2.1 Simulation Setup

In this study, the six-cylinder engine shown in Figure.1 was simulated by GT-Power software to study the effect of turbocharger and fuel type on performance and engine exhaust emissions. The GT-Power software is part of the GT-Suite software and gamma-technology company that emulates the engine and its accessories. Numerical calculations in this software are based on solving one-dimensional fluid dynamics equations including phenomena related to flow motion, heat transfer in the pipe and other components of the engine. To modeling engine, all parts of the engine were first introduced as a real six-cylinder engine, and then the required data were entered according to the actual engine conditions at the atmospheric pressure of one atm. In addition, the injected mass at per cycle engine of 96 mg and the length of the injection rate of 20 degrees has been selected, the engine speed changed from 800 to 2400 rpm, and considered injection advance at different speeds.

Table 1- Technical specifications of diesel engine John Deere 6068HF275

Engine characteristics	Value
Number of cylinders	6
Cylinder bore	106 mm
Stroke	127 mm
Connecting rod length	270 mm
Standard injection timing	4 BTDC
Compression ratio	17
Maximum torque	740-930 N.m @ 1400 RPM
Maximum power	129-187 kW @ 2000-2400 RPM

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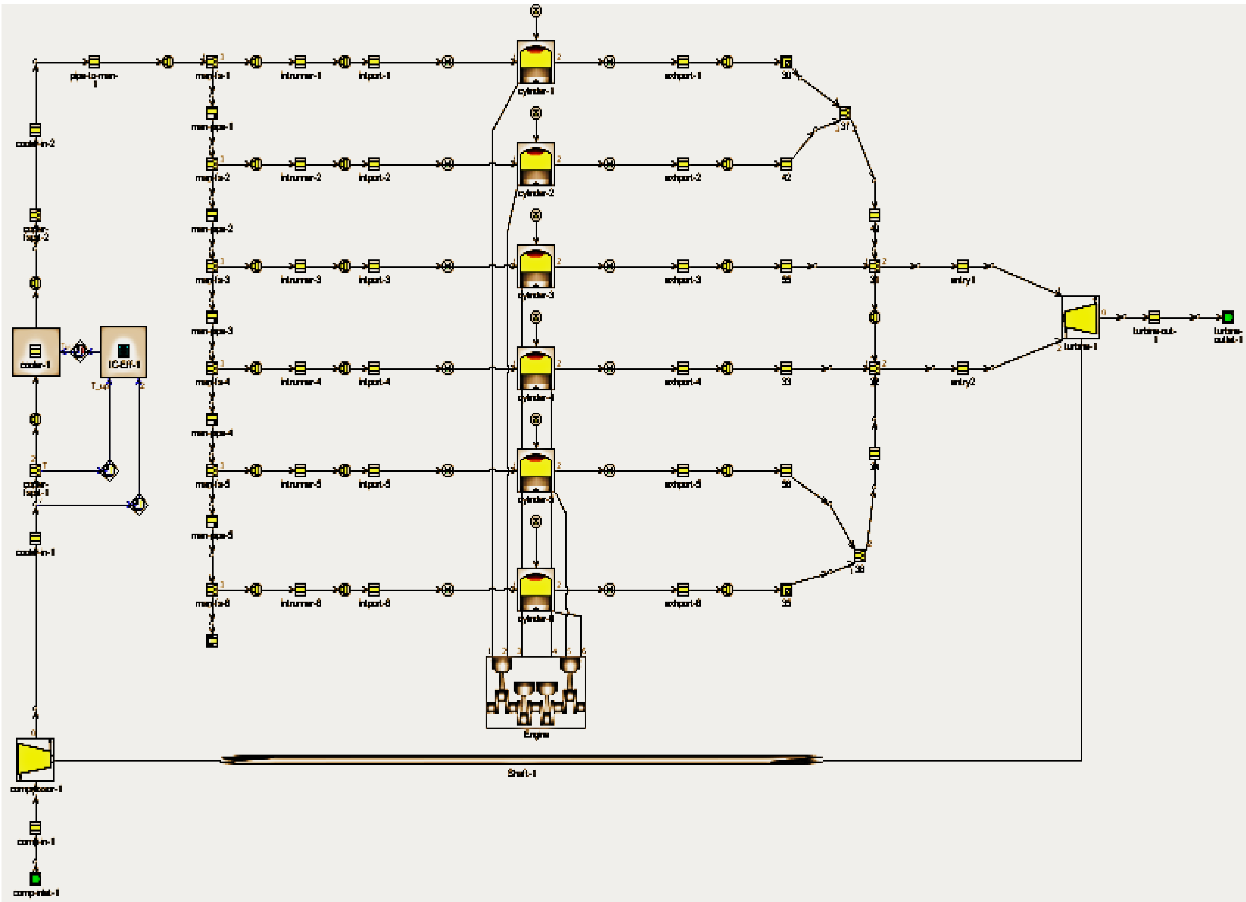

Fig.1. Computational model of a six-cylinder, direct-injection, compression-ignition engin

2.1.1 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 equations 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 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.

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

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

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

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

2.1.2 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 relationship 5 is established.

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

2.1.3 Heat transfer

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 Equation 6.

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

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 Equation 6, then corrected with the help of Equation 7.

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

2.1.4 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 Wiebe function is used to calculate the energy release rate per crank angle. This function, also known as the Ricardo function, uses the burning length and independent input of the function-shaped parameters to calculate the burning mass rate. Compacted combustible mass fraction is depicted in function of the crank angle in equation 8.

$$W = 1 - Exp\left[-AWI\left(\frac{\Delta\theta}{BDUR}\right)^{WEXP}\right] \quad (8)$$

2.1.5 Friction modeling

In this research, the Chenn-Flynn model was used to model mechanical friction, which is presented in the equation 9.

$$FMEP = C + (PF \times P_{max}) + (MPSF \times Speed_{mp}) + (MPSSF \times Speed_{mp}^2) \quad (9)$$

2.1.6 Fluid Properties

In the process of engine simulation, six type of fuel was examined. The GT-Suite software library contains many types of

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fluids and their specifications and features, and fluids that are not available in this library can be manually included in this software. In addition to introducing uncompressible fuel specifications, the software also needs to include fuel vapor characteristics, so that in case of evaporation, fluid specifications are predictable. The fuels used in this study include diesel, ethanol, 10% ethanol and diesel, biodiesel from soybean oil, petrolatum, and methanol, with some of the important properties of the fuels used in Table 2.

Table2- Important properties of used fuels

fuel properties	unit	Diesel	Ethanol	Methanol	Biodiesel	Decane	Diesel-Ethanol
Density	m s^{-1}	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
Hydrogen Atoms per Molecule	…	23.6	6	4	2	22	21.84
Carbon Atoms per Molecule	…	13.5	2	1	18.82	10	12.35
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

2.2 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.

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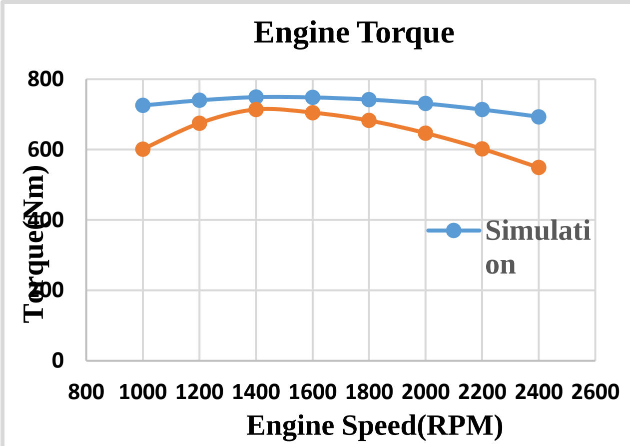

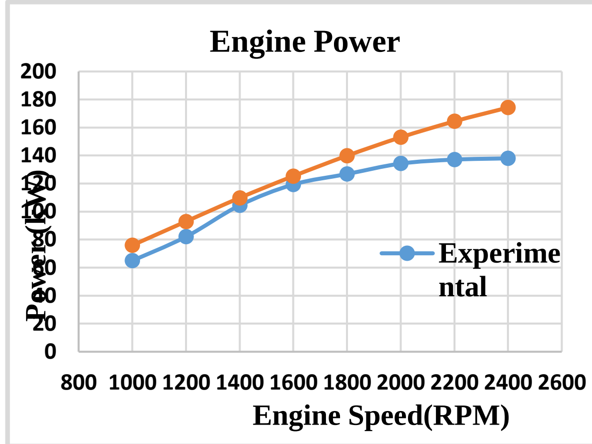

Fig.2. Curve of engine torque and power versus engine speed (comparison of experimental and simulation data)

Figure 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 equation for calculating percentage error is as below:

$$percentage\ error = \frac{|simulation\ value - experimental\ value|}{experimental\ value} \times 100$$

So, with using of above equation 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 (25-27). Hogg studies demonstrated that 20% error in calculation has a good agreement in evaluation and validation between experiment and numerical models (28).

3. Results and discussion

3.1 Brake power and brake torque

Brake torque and brake power changes in turbocharged engine and the naturally aspirated engine, in six different fuels and as a function of speed are shown in Figures 2 and 3, respectively. Figures 3 and 4 show that, with increasing speed, the brake torque first increases and then decreases and the brake power increases. Among the fuels used in the naturally aspirated engine, the maximum brake power and brake torque with biodiesel fuel was 15.42% higher than the base and lowest brake power and brake torque with 42.43% lower than base were methanol fuel, and in a turbocharged engine among the used fuels, the maximum brake power and brake torque with decane fuel was 58.76% higher than the base and the lowest brake power and brake torque with 31.33% lower than the base were methanol fuel. In decane fuel, according to Table 2, although it has a lower oxygen content than biodiesel fuel, if used in a turbocharged engine, due to sufficient oxygen supply, it provides power and torque higher than biodiesel fuel and biodiesel fuel due to Excessive oxidation is flawed and produces

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fewer power and torque than the decane fuel, and it can also be concluded that the turbocharged engine has a better performance than the naturally aspirated engine.

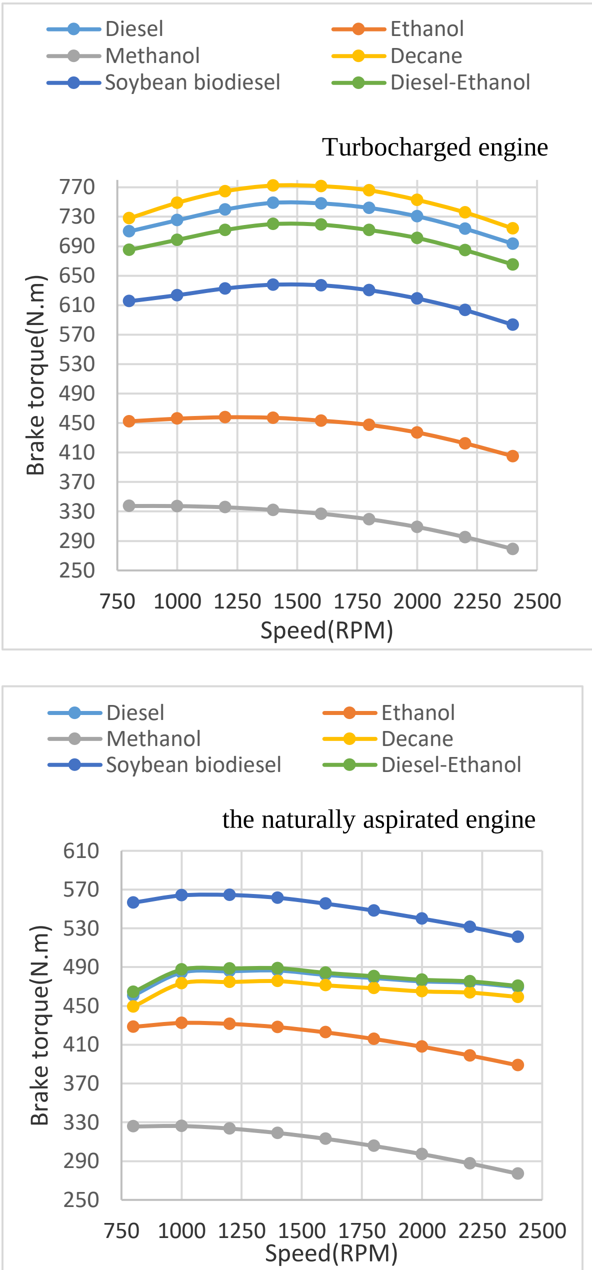

Fig.3. Brake torque in turbocharged and the naturally aspirated engine with different fuels.

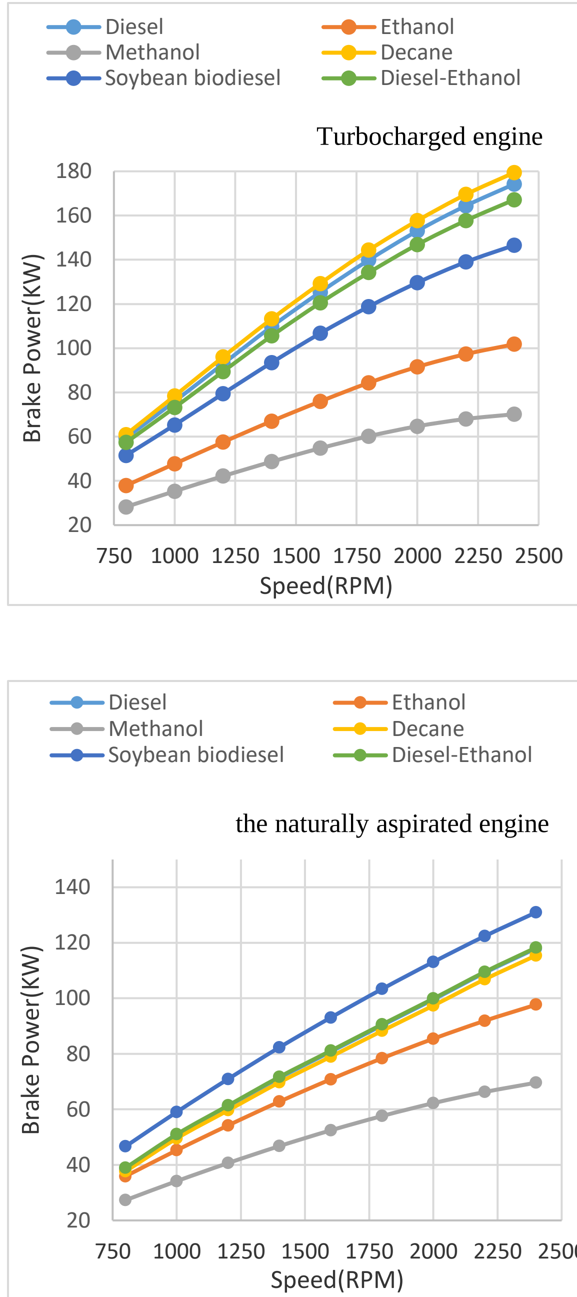

Fig.4. Brake power in turbocharged and the naturally aspirated engine with different fuels.

3.2 Brake specific fuel consumption

BSFC is defined as the ratio of fuel consumption to brake power. As shown in Fig. 5, with increasing speed, the specific fuel consumption of the brake is first

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reduced and then increased. The results showed that in the naturally aspirated engine and turbocharged engine, biodiesel fuel with 13.36% and decane fuel with 37.01% lower than the base have the least fuel consumption, which is due to higher power generation The fuel used in the engine[29, 30].

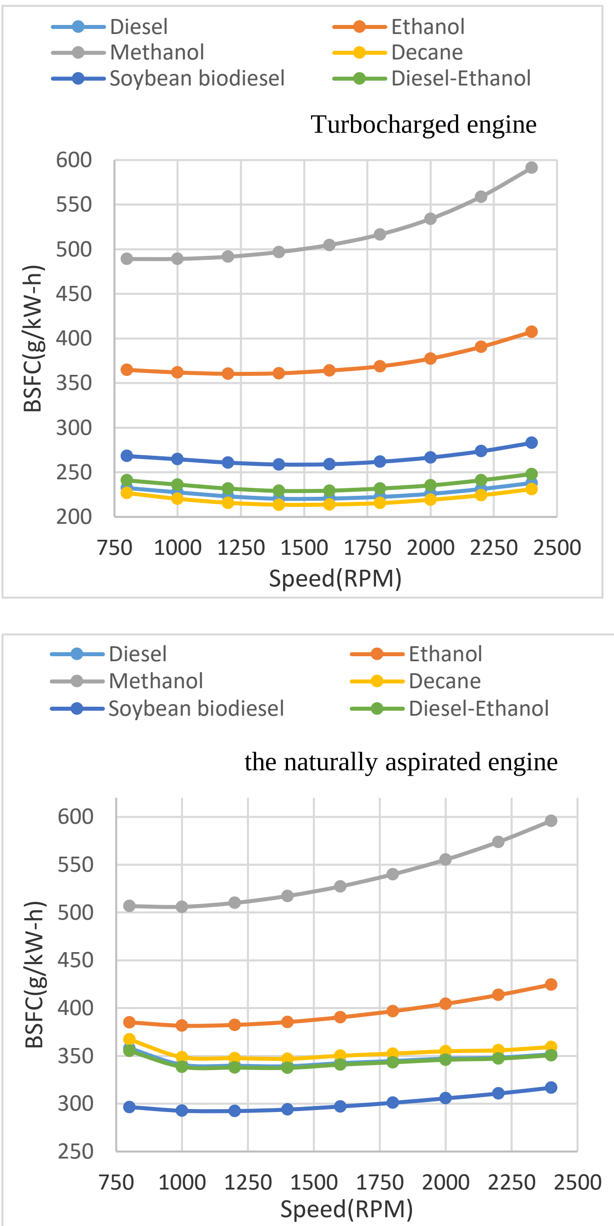

Fig.5. Brake specific fuel consumption in turbocharged and the naturally aspirated engine with different fuels.

3.3 CO emissions

The diffusion of CO from the engine largely depends on the fuel's properties, the availability of oxygen, and the fuel mix with air, temperature and turbulence inside the combustion chamber. With reference to Fig. 6, with increasing speed, the amount of co emission decreases. The results showed that the CO content of a turbocharged engine is higher than the naturally aspirated engine. We know that better and more complete combustion of fuels reduces CO emissions [31-34]. The use of a turbocharger increases air intake to the diesel engine, and fuel and air can be easily mixed in the combustion chamber, resulting in better combustion and lower emissions of CO. The results showed that the lowest CO emission is related to biodiesel fuel. Biodiesel has a higher cetane number compared to other fuels, which reduces the combustion delay and increases the amount of oxygen in the biodiesel with lower combustion delay time, better combustion and lower emissions of CO.

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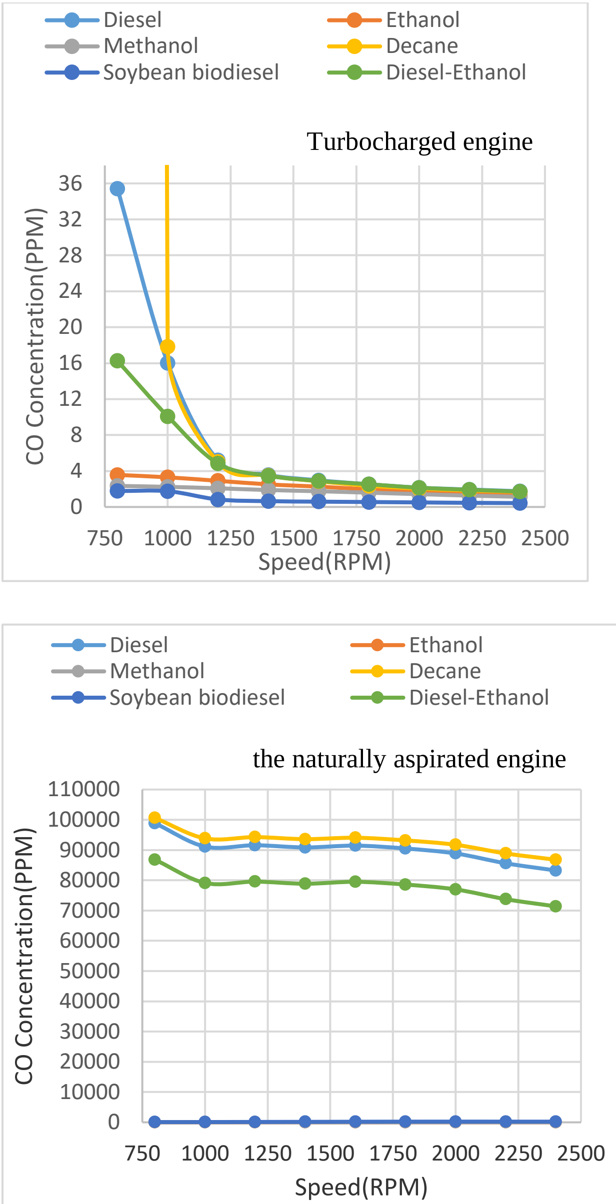

Fig.6. CO emission in turbocharged and the naturally aspirated engine with different fuels.

3.4 CO_2 emissions

With reference to Fig. 7, with increasing turbocharged engine speeds in all fuels, CO2 emissions are reduced and increased in some fuels in the naturally aspirated engine. The results showed that co2 emission in the naturally aspirated engines was higher in comparison to turbocharged engines, and decane fuel in a turbocharged engine was 26.54% more co2 than base and in the naturally aspirated engine, 15.61% produced less co2 than the base. The greater amount of CO2 represents the complete combustion of fuel in the combustion chamber. CO2 emissions are not too harmful for humans, but they increase the potential for ozone depletion and global warming [35].

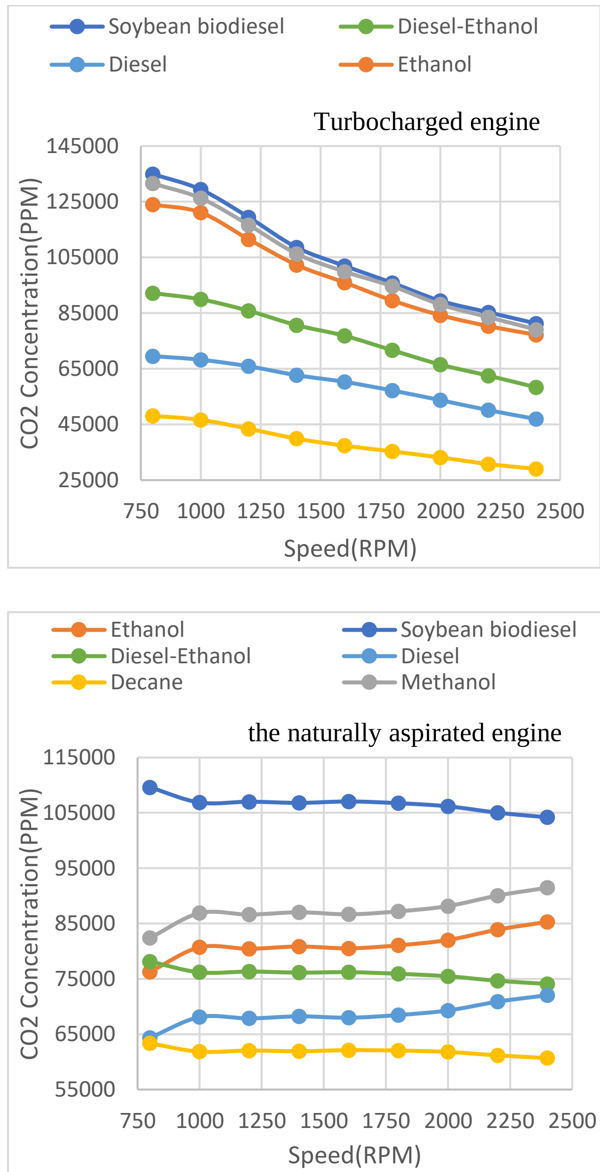

Fig.7. CO_2 emission in turbocharged and the naturally aspirated engine with different fuels

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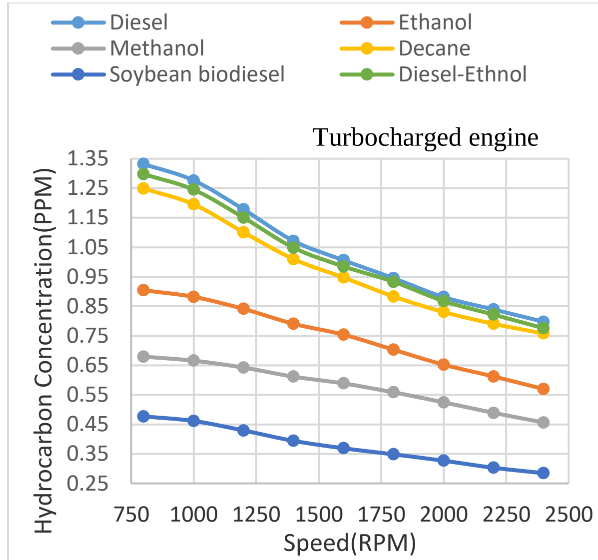
3.5 HC emissions

With regard to Fig. 8, with increasing speed, HC emission has increased and all fuels have lower HC emissions than diesel fuel. The results showed that the HC emission in the naturally aspirated engine is higher than that of a turbocharged engine, and that the HC emission of biodiesel in the naturally aspirated engine and turbocharged engines has been reduced by 63.45% and 42.47%, respectively. Because all fuels have lower HC emissions than diesel, it can be due to long-term combustion delays, as more fuel is accumulated in the combustion chamber, it can lead to higher emissions of hydrocarbons [36-39].

3.6 NO_X emissions

The fuel thermal mechanism, combustion temperature, oxygen content and the long-term availability of fuel gas are the most important factors in the formation of NO_X. With regard to Fig. 9, NO_X has increased with increasing speed. The viscosity and density of fuels have an impact on NO_X emissions, and it results in NO_X emissions due to larger droplets of fuel [40-41]. According to Figure 8, the highest NO_X emissions from biodiesel fuel are due to the high oxygen content of this fuel and the lowest NO_X emissions from decane fuel, due to the low fuel density compared to other fuels.



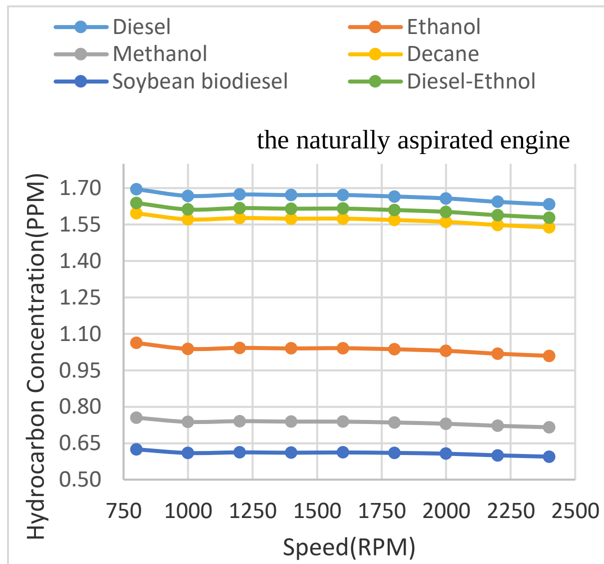
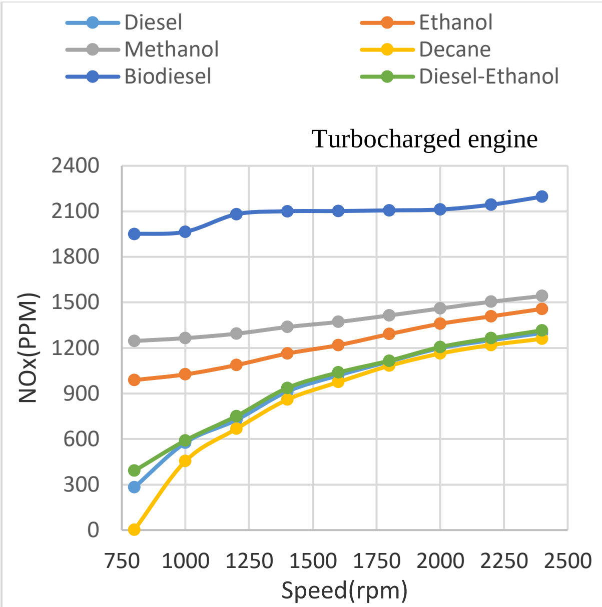


Fig.8. HC emission in turbocharged and the naturally aspirated engine with different fuels



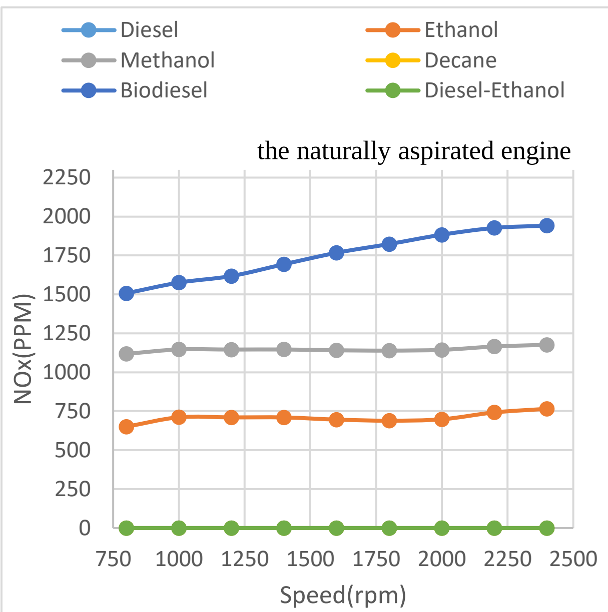


Fig.9. NO_X emission in turbocharged and the naturally aspirated engine with different fuels

4. Conclusion

In this research, the performance and emissions of a turbocharged six-cylinder engine and the naturally aspirated engine using different fuels were investigated, which included the following results:

1. The power and torque values of the turbocharged engine increase with the increase in the fuel type and its value by using diesel fuel compared to the base fuel is about 76.68%, which is the highest value in the naturally aspirated engine in biodiesel fuel, which improves its proportion The basis for fuel was about 15.42%.

2. The results showed that biodiesel fuel in the naturally aspirated engine with 13.36% and decane fuel in a turbocharged engine with 37.1% lower fuel consumption compared to baseline has the lowest fuel consumption compared to other fuels.

3. The results showed that biodiesel fuel in turbocharged engines and the naturally aspirated engine has the lowest amount of CO and HC emissions and the highest amount of NOx contamination, and with increasing speed, emissions of CO and HC and emission of NOx emissions increase.

4. The results showed that fuels with high viscosity and density have a higher NOx content. Therefore, the lowest and highest NOx values were related to decane fuel and biodiesel fuel, respectively.

5. The results showed that the turbocharger improves engine performance and reduces emissions of hydrocarbons and carbon monoxide.

6. The results of the study showed that decane fuel in turbocharged engines and biodiesel fuel in the naturally aspirated engines have the best functional and pollutant characteristics among the six fuel used in this study. So these two fuels can be the best alternative for diesel fuel.

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5. List of Symbols and Abbreviations

Abbreviation	
$\dot{m}$	boundary mass flux into volume
m	mass of the volume
V	volume
P	pressure
ρ	density
A	cross-sectional flow area
A_s	heat transfer surface area
e	total specific internal energy (internal energy plus kinetic energy per unit mass)
H	total specific enthalpy
h	heat transfer coefficient
T_{fluid}	fluid temperature
T_{wall}	wall temperature
u	velocity at the boundary
C_f	Fanning friction factor
K_p	pressure loss coefficient (commonly due to bend, taper or restriction)
D	equivalent diameter
d_x	length of mass element in the flow direction (discretization length)
d_p	pressure differential acting across dx
Δt	time step (s)
m	time step multiplier specified by the user in Run Setup (less than or equal to 1.0)
Δx	minimum discretized element length (m)
c	speed of sound (m/s)
U_{eff}	effective velocity outside boundary layer
C_p	specific heat
Pr	Prandtl number
$h_{g\ rough}$	heat transfer coefficient of rough pipe
$C_{f.rough}$	Fanning friction factor of rough pipe
w	Mass fraction of burned mixture
$\Delta\theta$	Time elapsed from ignition
AWI	Wibe function coefficient
$WEXP$	power to function Wibe
$BDUR$	Time elapsed according to the crank angle, to burn mass of the mixture 10 to 90%
$FMEP$	Friction Mean Effective Pressure
P_{max}	Maximum Cylinder pressure
$Speed_{mp}$	Mean Piston Speed
C	Constant part of FMEP
PF	Peak Cylinder Pressure Factor
$MPSF$	Mean Piston Speed Factor
$MPSSF$	Mean Piston Speed Squared Factor

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