**Journal of Agricultural Machinery**

Homepage: https://jame.um.ac.ir



**Research Article**
**Vol. 13, No. 1, 2023, p. 67-83**

# Experimental Investigation of Performance and Emissions of a Compression Ignition Engine Using a Combination of Compressed Natural Gas and Diesel Fuel

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https://doi.org/10.22067/jam.2023.79199.1127

| Received: 16 October 2022<br>Revised: 25 December 2022<br>Accepted: 07 January 2023<br>Available Online: 07 January 2023 | **How to cite this article:**<br>Niknam, Y., Mohamadzamani, D., & Gholami Pareshkoohi, M. (2023). Experimental Investigation of Performance and Emissions of a Compression Ignition Engine Using a Combination of Compressed Natural Gas and Diesel Fuel. *Journal of Agricultural Machinery*, *13*(1): 67-83. https://doi.org/10.22067/jam.2023.79199.1127 |
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## Abstract

This study presents the effects of compressed natural gas fuel on a four-cylinder compression ignition engine. Compressed natural gas as the main fuel and diesel fuel as the igniter were used to investigate performance and emissions from the dual fuel engine. According to the engine speed and load, the amount of diesel fuel as igniter was adjusted using mechanical changes in the governor, while no ignition system was used. The engine experimental tests were performed at engine speeds of 1200, 1400, 1600, 1800 and 2000 rpm, using diesel fuel and dual fuel. These data were collected in the Engine Research Center of Tabriz Motorsazan Company and experimental runs were repeated three times. The maximum torque of the engine in diesel mode was 360 N m at 1400 rpm. Compared to the diesel mode, the dual fuel mode showed the maximum torque by 334 N m at 1600 rpm, which is about 26 N m less than that gained from the diesel mode. Considering emissions analysis at 2000 rpm, it is seen that the amount of NOX, HC, $CO_2$ and CO emissions in the dual fuel mode was 20, 53, 16 and 86% more than diesel mode, respectively. However, $O_2$ and soot showed the highest reduction at 2000 rpm for dual fuel mode by 51% and 69% respectively. This study indicated that there was a considerable enhancement in exhausted emissions when the injection of the diesel fuel as igniter was done mechanically. In this regard, control the amount and time of the igniter injection could likely be helped for better control of emissions. Therefore, further research on the modification of the diesel injection system as igniter or CNG injection system is needed towards reducing emissions.

**Keywords:** Compression ignition engine, Compressed natural gas, Diesel, Pollution, Power

## Introduction

The massive energy subsidies in the transportation sector have been the serious challenges of the Iranian economy in recent decades. The dependence of the country's transportation cycle on imported petroleum products is a very vulnerable from both political and peripheral point of view. The release of gasoline and diesel prices, quotas, gas burning and all -out public transport development have been described as solutions to exit this crisis. Among these options, and in terms of the fleet's approach and a glimpse of alternatives, as well as the frequency of Iran's gas reserves as the second source of the gazette, the option of using gas -fired vehicles from other options Distinct (Mirfattah, 2010). Countries around the world have set a variety of goals in the use of Compressed Natural Gas (CNG). They use gas cars partly to reduce fuel costs and partly to reduce environmental pollution. International law and the recommendation of the international community to protect the environment are an impact on the approach to the natural way. In 1995, for example, the global Energy Authority in Tokyo officially declared compressed natural gas consumed in the transportation system (Mirfattah, 2010). In the meantime, compression ignition engines are widely used in the transportation sector, where demand for fossil fuels is high. About 60 percent of European passenger cars use these types of engines (Saravanan, Kumar, Ettappan, Dhanagopal, & Vishnupriyan, 2020). Given the effect that contamination has on human health and the environment, it can be concluded that the issue of contamination of internal combustion engines is of high importance (Assasi, Mirzaei, & Khoshbakhti Saray, 2017).

The use of compressed natural gas fuel in combustion engines based on liquid fuels is associated with problems such as reducing efficiency and increasing corrosion of the engine components. The use of compressed natural gas fuel in combustion engines based on the auto cycle (spark ignition) has passed its research and development stage for years and is now widely used in some countries, including Iran. However, the use of compressed natural gas fuel in diesel cycle engines without the use of sparking system goes through its research steps and is not widely used commercially. One of the problems with the use of compressed natural gas fuel in common compressions is the high cost of converting these engines into a spark ignition engine. This conversion requires the use of spark system components such as coil, wire and candles, and in electronic spraying systems, processors and sensors. It is also a relatively difficult task to change the cylinder head and install candles. On the other hand, the compression ignition engine that turns into a spark ignition engine can only work with a type of fuel (CNG) and cannot be used in diesel fuel (Sajedian, Mohammadzamani, & Ranjbar, 2014). In comparison with other fuels that can be used in compression ignition engines, diesel fuel has a lower Stan number, lower energy consumption from well-to-wheel, and the amount of CO and HC emissions is relatively low, but the amount of $NO_X$ and suspended particles from well-to-wheel is high (Azizi, Mahdaloui, & Hassani, 2013). Relatively few researches have been conducted on the possibility of using compressed natural gas in diesel engines without the use of an ignition system, and a significant part of the existing studies has been on systems equipped with spark ignition. In a study, the Perkins 1104D-E44TA engine heat release was studied with compressed gas and diesel fuel. The heat release property is a criterion of the combustion process, so this variable affects the engine performance indicators. In this study, heat release for a dual engine was compared with the heat release features for a diesel engine under the same operating conditions. The analysis of heat release properties was performed in the range of their effect on the concentration of $NO_2$ in the engine exhaust. The results of this study showed that the greater share of compressed natural gas in the total amount of energy released in the engine cylinders causes a greater difference in the combustion process and also reduces the concentration of $NO_2$ in the engine exhaust (Kurczynski, Lagowski, & Pukalskas, 2019). In a research to investigate the effect of ignition fuel amounts on the combustion process, performance and emissions, dual fuel diesel engines with indirect injection were evaluated in all load conditions. In this research, the ignition delay time interval and the combustion process and the formation of pollutants in the dual engine were investigated in two ways. In the first case, the amount of ignition fuel is kept constant, and by adding gas fuel, the engine load is increased, and in the second case, for partial load of 25% and full load, the increase in gas fuel and the decrease in ignition fuel shortens the length of the combustion process. and it reduces CO and HC pollutants, but in partial loads with increasing gas fuel and decreasing ignition fuel, the duration of the combustion process becomes longer and CO and HC pollutants increase strongly (Mohammadi Koosha, Piroozpanah, Khoshbakht Sarai, & Salsbili, 2008). In a study, the functional variables and emissions of diesel engine in different ratios of air-to-fuel balance ratio were discussed using diesel-channel fuel mixtures. The results showed that the increase in ethanol in the diesel and ethanol mixture increased the air -to -fuel balance ratio, resulting in a decrease in power and torque and increased engine fuel consumption (Shadidi, Haji Agha Alizade, Najafi, Moosavian, & Khazaee, 2020). A number of researchers conducted an empirical examination of a compression combustion engine to study functional and pollution parameters, using compressed natural gas fuel and diesel. The engine was modified to work with a mixture of diesel fuels and natural gas in dual -fuel state so that diesel fuel was injected into the cylinder, while natural gas into the input air pipe with two controlled injectors were injected. The energy content of the gas fuel injected in zero values (diesel fuel), 15, 40 and 75% of the total fuel energy content vary. In addition to experiments, the engine was modeled with a single-dimensional commercial software. The results showed that both $NO_X$ and soot emissions decreased by 15 and 40% and energy content rates in the gas-burn mix, respectively, compared to diesel fuel. However, an increase in carbon monoxide emissions was observed by adding 15% natural gas fuel compared to diesel fuel (Karagoz, Sandalc, Koylu, Dalkilicx, & Wongwises, 2016). In a resaerch, experimental investigation of compressed natural gas using in an indirect injection diesel engine at different conditions was conducted. The novelty of this experimental work is replacing different mass fractions of diesel fuel with natural gas in an indirect injection diesel engine and evaluating its effect on the emissions of soot and nitrogen oxides, and brake specific fuel consumption in the presence of cold exhaust gas recirculation. Experiments were done at different equivalence ratios with 1200, 2000, and 3000 rpm and at 25, 50, and 75% of the full load in each speed. Replacing 40% mass fraction of input diesel fuel by adding natural gas resulted in a maximum 74% reduction of soot; the reason is a decrease in carbon to hydrogen ratio in the mixture (Bayat & Ghazikhani, (2020)). An empirical and numerical study on the effect of diesel injection scheduling from 10 to 50 degrees before the top death center (BTDC) on combustion and greenhouse gas emissions of a dual-fuel-burner-powered engine in the engine was conducted. The results of this study showed that the highest thermal efficiency indicated was obtained in the scheduling of diesel injection 2 and 2 degrees BTDC. Finally, with increased diesel injection time from 10 degrees BTDC to 50 degrees BTDC, $NO_X$, methane and CO decreases by 65.8, 83 and 60%, respectively, while the thermal efficiency increases by 7.5% (Yousefi, Birouk, & Guo, 2017). A research was conducted on CNG-diesel dual fuel engine by optimizing the timing of CNG injection from 70° to 150° ATDC and the duration of CNG injection from 70° to 150° CA with 20° intervals at low load. The results explained that by retarded the timing of CNG injection (130° ATDC) yielding a value of cylinder pressure and heat release rate (HRR) was the highest with emissions of hydrocarbon (HC), carbon monoxide (CO) and particulate matter (PM) were lower and also the amount of CNG volume inserted into the cylinder was more large with optimal CNG injection duration of 110°CA on CNG-diesel dual fuel engine under low load (Yuvenda, Sudarmanta, Wahjudi, & Muraza, 2020).

Considering that the studies have focused on the possibility of using CNG using a spark ignition system, this study aims to determine the feasibility of using compressed natural gas in a compression ignition engine without using an ignition system and its effects on engine performance and pollutant emissions have been investigated. So the aim of the research was to investigate the effect of using two types of fuel including diesel fuel and dual fuel (diesel + CNG) in a compression ignition engine on performance indices and the amount of pollutant emissions.

## Materials and Methods

The combustion engine used in this study is a four -cylinder complication engine manufactured by Tabriz Motorsazan Company whose maximum power is at the speed of 2000 rpm, 82 hp. Other technical specifications of this engine are presented in Table 1. In the tests, the engine was started with diesel fuel and CNG fuel was not used in idle mode. In the tests, by increasing the rotational speed of the engine, CNG fuel was used as the main fuel and diesel (with Euro 4 standard) was used as the ignition fuel.

**Table 1-** Technical specifications of the engine used in the research

| Engine specifications | Value |
|---|---|
| Cylinder number | 4 |
| Piston diameter (mm) | 100 |
| Piston courses (mm) | 127 |
| Compression ratio | 17.5 |
| Maximum power (hp) at 2000 (rpm) | 82 |
| air inlet system | Turbocharger |
| Maximum torque (N m) at 1400 (rpm) | 360 |
| Volume Capacity (L) | 3.99 |

Equation (1) was used to achieve fuel consumption in the dual diesel and natural gas refueling system in each combustion chamber (Sontag, Borgnakke, & Van Wylen, 2015).

$$\dot{V}_{Fuel} = \dot{V}_{Diesel\ oil} + \dot{V}_{CNG} \quad (1)$$

In which $\dot{V}_{Fuel}$ is volume flow rate of combined $\dot{V}_{CNG}$ is volumetric flow rate of CNG fuel consumption ($l.hr^{-1}$).

The ratio of gas entry to dual ga and diesel (α) is calculated as follows:

$$\rightarrow \dot{V}_C = \alpha \dot{V}_f \longrightarrow \dot{V}_g = (1-\alpha)\dot{V}_f$$

$$\alpha = \frac{\dot{V}_c}{\dot{V}_F} \quad (2)$$

In which $\dot{V}_c$ is volumetric flow rate of gas and $\dot{V}_F$ is volumetric flow rate gas and diesel.

Due to the power of 82 hp for the compression ignition engine used in this study, the dual fuel system (diesel and CNG) must be able to supply the energy needed to produce this power. Therefore, according to the first law of thermodynamics (Equation 3 (Sontag *et al*., 2015)):

$$\dot{Q} = (\dot{U}_2 - \dot{U}_1) + \dot{W} \quad (3)$$

In which $\dot{Q}$ is the heat rate created in the process (hp), $\dot{U}_1$ is the producted power in the process (hp), $\dot{U}_2$ is the internal energy rate in the process (hp) and $\dot{W}$ is equal to 82 hp.

**Table 2-** Some physical and thermal properties of air, diesel fuel and CNG (Chala, Aziz, & Hagos, 2018)

| | ρ ($kg\ m^{-3}$) | $C_v$ ($kJ\ kg^{-1}.\dot{k}^{-1}$) | $\dot{V}_f$ |
|---|---|---|---|
| Air | 1.169 | 103 | 17.16 (1-α) $\dot{V}_f$ |
| Diesel | 750 | 214 | α $\dot{V}_f$ |
| CNG | 0.648 | 1.736 | (1-α) $\dot{V}_f$ |

$$\dot{U}_2 - \dot{U}_1 = \left[\dot{m}_{air} C_{V\,air} + \dot{m}_C C_{v\,C} + \dot{m}_g C_{v\,g}\right]\Delta T = \left[\rho_{air} \dot{V}_{air} C_{V\,air} + \rho_C \dot{V}_C C_{v\,C} + \rho_g \dot{V}_g C_{v\,g}\right]\Delta T \quad (4)$$

Next, considering the values of Table 2 and the internal energy equation for dual fuel and air:

$$\dot{U}_2 - \dot{U}_1 = \left[577.98\times10^{-3} + 9072.185\times10^{3}\alpha \dot{V}_f\right] \quad (5)$$

$$\dot{Q} = \dot{m}_f LHV \qquad (6)$$

In which: $\dot{m}_f$ is mass flow rate of dual fuel and LHV is the low heat value equal to 2 $kJ.kg^{-1}$. Mass flow rate was calculated from Relationship 7 (Sontag *et al.*, 2015):

$$\dot{m}_f = \rho \dot{V}_f \ (\frac{kg}{s}) \qquad (7)$$

$$\begin{aligned}\dot{Q} &= \dot{m}_f LHV \\ &= \left[\rho_C \dot{V}_C + \rho_g \dot{V}_g\right] LHV \\ &= \left[0.648\alpha \dot{V}_f + 750(1-\alpha)\dot{V}_f\right] LHV \\ &= \left[0.648\alpha \dot{V}_f + 750(1-\alpha)\dot{V}_f\right]\left[32016.15 + \frac{9461.68}{0.648\alpha + 750(1-\alpha)}\right]\end{aligned}$$

After simplifying the following results:

$$\rightarrow \ \dot{Q} = \dot{V}_f \left(24021572.9 - 23991366\alpha\right) \qquad (8)$$

Now, according to the relationships above, the equations (5) and (8) and their replacement in the thermodynamic equation (3) to achieve the result are followed as follows:

$$\dot{Q} = (\dot{U}_u - \dot{U}_u) + \dot{w}$$
$$\dot{V}_f (24021572.9 - 23991366\ \alpha) = \left[577.98 \times 10^{-3} + 9072.185 \times 10^{3}\ \alpha\ \dot{V}_f\right] + \left(\frac{107424}{4}\right) \qquad (9)$$

The importance of this study was that during the test when the engine only operated with diesel fuel, 2.2 cc of diesel fuel was consumed in idle engine speed for 30 seconds. Since one of the goals of the study was to reduce fuel consumption, so in the present study, the dual fuel consumption for 30 seconds considered 2 cc, thus from the previous relationships, the amount of CNG and diesel fuel consumption was calculated. Finally, the ratio of gas to dual fuel (α) was calculated equal to 0.72. This means that CNG makes up 72% and diesel fuel makes up 28% of the dual fuel at idle speed (450 rpm). According to the way of changing the mechanical structure of the governor in this research, with the increase of the rotational speed of the engine, α increased linearly. According to the structure of the engine used in the tests, by changing the mechanical structure of the governor in the injector pump, it was possible to inject a limited amount of diesel fuel as an ignitor in dual fuel mode. In this case, the ignition fuel (diesel) was injected into the compressed CNG in the combustion chamber at the end of the compression stage and combustion took place. In this research, no electronic fuel injection control system was used and fuel injection was completely done mechanically. The innovation and simplicity of the proposed method has been in not changing the structure of the fuel supply system and reducing the cost. Performance tests and pollutants were conducted at the Tabriz Motorsazan Research and Development Center. Figure 1 shows schematic of the components of the performance and the pollution tests of the engine.

The AVL 415S polluting machine was used to measure the contaminants from the combustion of the tested engine. This device is a product of AVL Ditest, manufactured by Germany and is capable of measuring the combustion products of spark ignition and compression ignition vehicles. This device can measure $NO_X$, HC, CO, $O_2$ and $CO_2$ digitally. The exhaust gases from the engine exhaust were transferred to the device by a pipe and the amount of each pollutant was measured separately by the device.

All performance and emissions tests were performed after 10 minutes of working in an idle engine speed mode and tests at five rotational speeds of 1200, 1400, 1600, 1800 and 2000 rpm and in two modes of diesel fuel and dual mode. The performance test was done in such a way that first the engine was started with diesel fuel and then the tests were performed in three repetitions to check the reproducibility of the tests. After completing the engine test in diesel mode, the tests were performed for the dual fuel mode in three repetitions. In order to analyze the data obtained from the performance and emission tests, the data were drawn as graphs by Excel software. In this research, data analysis was done by direct comparison method.

## Results and Discussion

### Effect on torque

The torque analysis for both diesel and dual fuel (diesel as igniter and compact natural gas as main fuel) modes at various engine speeds was shown in Figure 2.

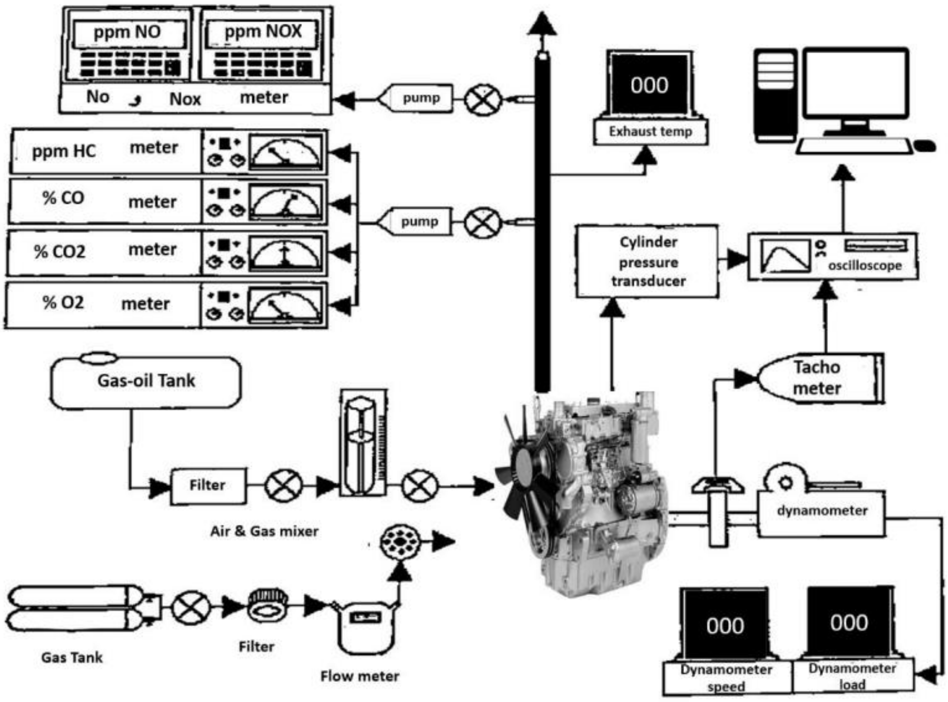
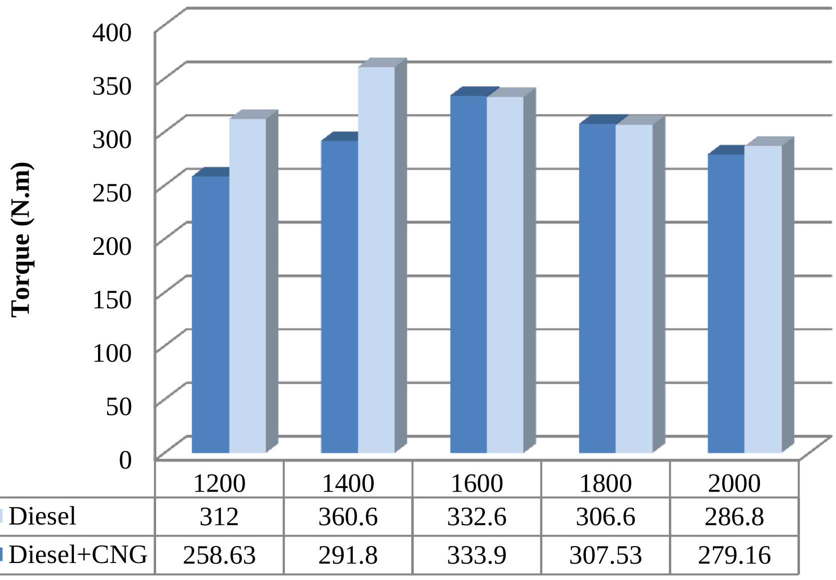



**Fig. 1.** Scheme of components of performance tests and engine pollution




| | 1200 | 1400 | 1600 | 1800 | 2000 |
|---|---|---|---|---|---|
| Diesel | 312 | 360.6 | 332.6 | 306.6 | 286.8 |
| Diesel+CNG | 258.63 | 291.8 | 333.9 | 307.53 | 279.16 |

**Fig. 2.** Variation of torque for two modes at different engine speeds

According to the Figure 2, at diesel mode the maximum torque was 360.6 N m at 1400 rpm. While at dual fuel mode, corresponding maximum torque was 334 N m at 1600 rpm. It can be observed that maximum torque take place at higher speed than diesel mode and this shows the effect of diesel fuel injection system (as igniter) in low speeds (the amount of injected fuel and diesel injection timing on combustion timing can be noticeable).

The average value of torque achieved was 312 and 258.63 N m by diesel fuel and the dual fuel at 1200 rpm, respectively. Also, the average value of torque achieved by diesel fuel and dual fuel was 360.6 and 291.8 N m at 1400 rpm, respectively. From the analysis of torque, there is an average reduction of 54 N m and 68.2 N m in torque for the dual fuel at 1200 and 1400 rpm, respectively. On the other hand, for engine speeds above 1400 rpm (from 1600 to 2000 rpm), there is no remarkable difference in the torque of dual fuel mode and diesel.

The maximum torque in both modes was achieved at different engine speeds. Therefore, brake power was calculated from torque data at the indicated speed of engine (2000 rpm). The BP of engine is 80.47 and 78.33 hp for diesel mode and dual fuel mode, respectively. According to the obtained results, the BP with diesel fuel was 6.15 hp (7.6%) more than the dual fuel at 2000 rpm.

As mentioned the maximum torque in both modes was achieved at different engine speeds. Therefore, brake power was calculated from torque data at the indicated speed of engine (2000 rpm). The BP of engine is 80.47 and 78.33 hp for diesel mode and dual fuel mode, respectively. According to the obtained results, the BP with diesel fuel was 6.15 hp (7.6%) more than the dual fuel at 2000 rpm. it is clear that BP of dual fuel mod is close to that of the diesel mode. It seems in dual fuel mode the more amount diesel fuel and advance injection time aid the combustion process, which prevents BP reduction. By decreasing the engine speed, BP reduces, which is mostly due to the impossibility of advance fuel injection and this is due to the mechanical nature of fuel injection, which is unable to provide a linear process throughout its working range.

This result is consistent with the research results of Yousefi *et al*, (2017) in which the thermal efficiency of a compression ignition engine with the use of dual fuel and in different ignition fuel injection timings was investigated.

**Diesel fuel consumption in dual fuel mode and diesel mode**

Figure 3 shows the amount of diesel fuel consumption in the dual fuel and diesel mods, at various engine speeds.

According to Figure 3, the amount of diesel fuel consumption in the dual fuel mode at engine speeds lower than 1600 rpm was relatively low. In comparison with diesel fuel consumption in the diesel mode, about 75.7% (1200 rpm), 77.5% (1400 rpm) and 72.08 % (1600 rpm) reduction was observed.

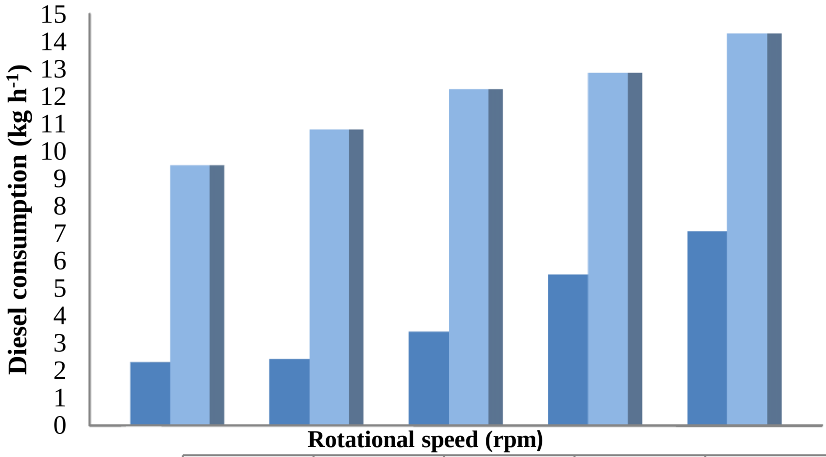
For all engine rotational speed conditions, an average of about 65% reduction in diesel fuel consumption was achieved. The reason for this reduction in diesel fuel consumption was the use of natural gas fuel as the main fuel and the use of diesel fuel as ignition fuel.

The result shows enhancement of diesel fuel consumption of both modes by increasing engine speed, which are almost the same for both modes. From the engine speed 1200 rpm to 2000 rpm, the diesel fuel consumption achieved by dual fuel mode was 4.78 kg $h^{-1}$ and the diesel mode was 4.8 kg $h^{-1}$.

Based on the results of diesel fuel consumption analysis, the enhancement in the diesel fuel consumption by increasing engine speed was observed in the dual fuel mode that confirm the significant effect of amount and timing diesel fuel injection as igniter on the torque of engine. It should be noted that power improvement in the dual fuel mode is related to sufficient availability of diesel fuel as igniter in combustion by increasing engine speed (according to the engine speed and load, the amount of diesel fuel as igniter was adjusted using mechanical changes in the governor). Therefore, the mechanism used regulate diesel fuel injection aids the combustion process, which is in line with the research results of Yusefi *et al*, (2017).

### CNG fuel consumption

The amount of CNG fuel consumption in the mode of using dual fuel was also measured in five different rotational speeds as in other tests, and the result is shown in Figure 4.




| | 1200 | 1400 | 1600 | 1800 | 2000 |
|---|---|---|---|---|---|
| Dual fuel mode | 2.3 | 2.42 | 3.42 | 5.51 | 7.08 |
| Diesel | 9.48 | 10.78 | 12.25 | 12.85 | 14.28 |

**Fig. 3.** Diesel consumption in dual fuel and diesel modes vs. engine speed

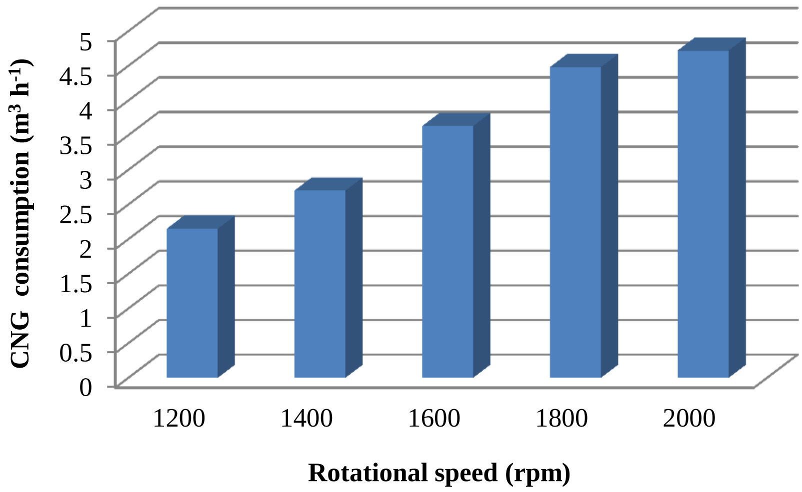



**Fig. 4.** CNG consumption for dual fuel mode vs. engine speed

CNG fuel was the main fuel in the dual fuel mode. As shown in Figure 4, for the engine speed ranging from 1400 to 1800 rpm, CNG fuel consumption increases. Increasing of the engine speed causes more CNG induction into the engine cylinder through inlet manifold. According to Fig. 2, the maximum torque for dual fuel mode is in 1600 rpm and from 1600 to 2000 rpm, torque reduce slightly. Higher injected CNG fuel quantity would result in enhancement of torque. Consequently, at higher engine speeds torque starts to decrease when CNG fuel induction into the engine cylinder through inlet manifold decreases.

**Specific consumption of diesel fuel in diesel mode**

In figure 5, the specific fuel consumption of diesel is displayed, which is presented for five rotational speeds of the engine.

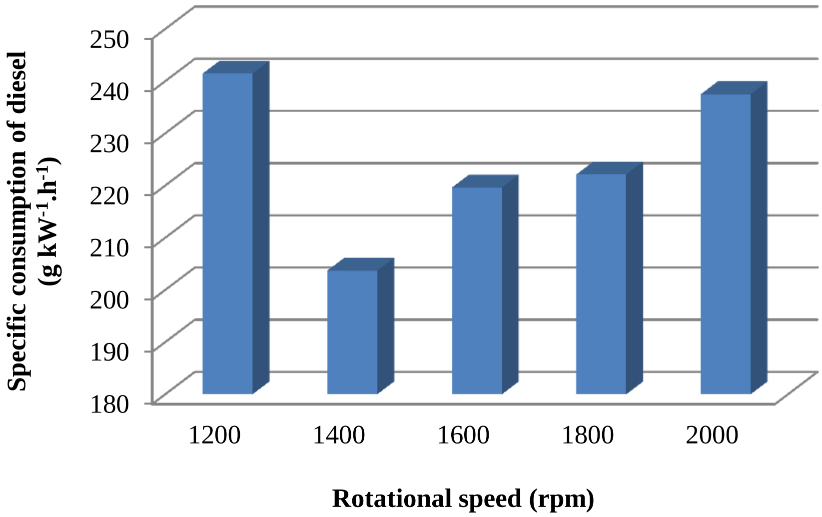



**Fig. 5.** Specific consumption of diesel fuel in diesel mode vs. engine speed

In Figure 5, the specific fuel consumption of diesel is displayed at different engine speeds. The specific fuel consumption was obtained by dividing the fuel consumption rate by the production power. According to Figure 5, the specific consumption of diesel fuel at 1400 rpm, where the maximum torque produced (Figure 2), was 203.71 g $kW^{-1}.h^{-1}$ (the lowest value the diesel specific consumption and highest torque of engine was observed at 1400 rpm).

The reason that the specific fuel consumption is low at the point of the maximum production torque is that the specific fuel consumption has an inverse relationship with the torque or in other words power.

That is, the higher the production capacity, the lower the specific fuel consumption. Also As shown in Figure 2, with the increase in rotational speed from 1400 to 2000 rpm, the engine torque has decreased in diesel fuel mode. Since specific fuel consumption has an inverse relationship with power (or, in other words, torque), therefore, with an increase in the rotational speed of the engine, the specific fuel consumption has increased. Although some observations (including fuel consumption at a rotational speed of 1200 rpm) cannot be justified.

**The specific consumption rate of diesel fuel and CNG in dual fuel mode**

Figure 6 shows the specific consumption of CNG fuel and diesel fuel in the dual fuel mode, separately. The minimum the specific consumption of CNG fuel and diesel fuel was obtained at 1400 rpm (under dual fuel mode the maximum increment in torque at 1400 rpm condition was observed).

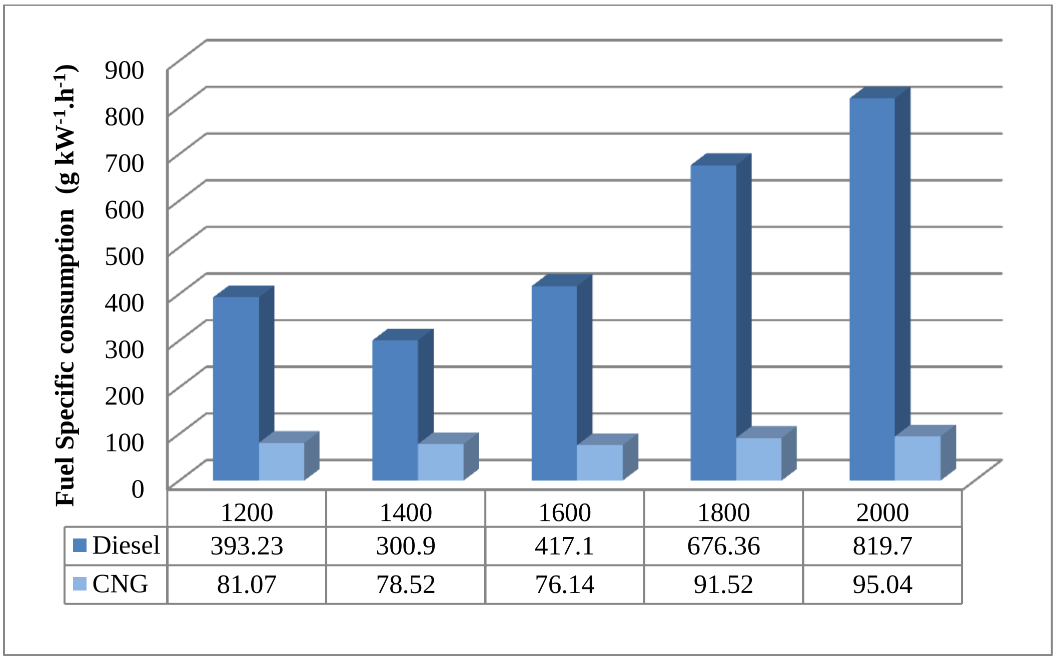



**Fig. 6.** Specific consumption of dual fuel (diesel + CNG) vs. engine speed

As previously mentioned, in the dual fuel mode, CNG is the main fuel and diesel fuel as an igniter. In Figure 6, with increasing engine speed from 1600 to 2000 rpm the specific consumption of CNG and diesel fuel increased. The lack of proper combustion natural gas-air mixture at high speeds is likely the primary factor influencing proper fuel thermal energy conversion, leading to a reduction in BP. Therefore, it is necessary to modify the mechanical structure of the engine both in the air supply system and in the fuel supply system in order to control this process.

## Effect of dual fuel mode on emissions

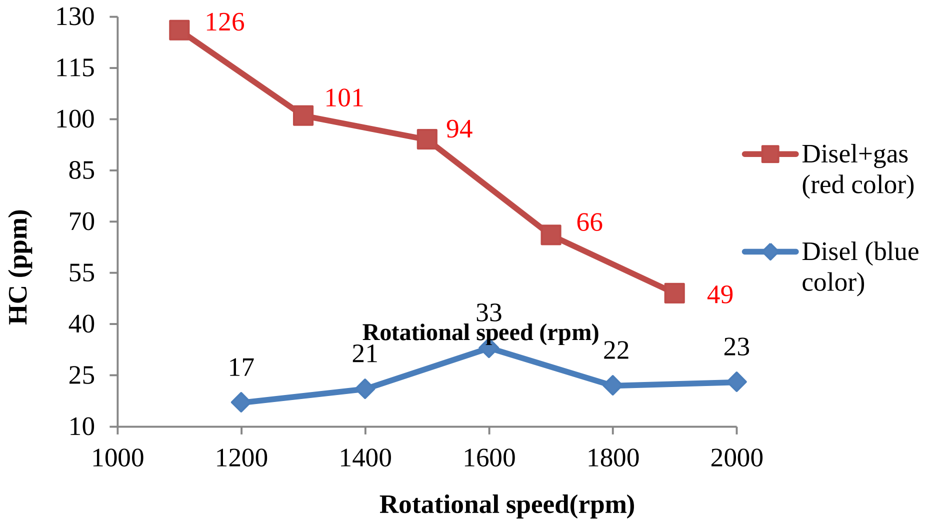



**Fig. 7.** HC emissions vs. engine speed for diesel and dual-fuel modes

Figure 7 shows the amount of HC emissions of the engine for both diesel fuel and dual fuel modes. According to Figure 7, the amount of HC emissions in dual fuel mode is higher than that of diesel fuel mode. However, the amount of this pollution decreased with the increase of engine speed, which is consistent with the research results of Piroozpanah & Abbas Alizadeh (1998). Rich fuel mixture and incomplete combustion contribute to the formation of unburned hydrocarbon emissions. The amount of unburned hydrocarbon in the case of using dual fuel is due to the richness of the fuel and the incomplete combustion of methane.

Engines that work with CNG fuel (the main part of CNG is methane) produce more HC emissions in partial loads, and the amount of this emissions decreases with the increase of engine load and speed (Yuvenda *et al.*, 2020). It is worth mentioning that the load applied by the dynamometer was proportional to the rotational speed of the engine. By increasing the rotational speed of the engine, the ratio of CNG fuel to diesel fuel (α) increased in the dual system, and as a result, more oxygen entered the combustion chamber, which resulted in proper combustion and a reduction in HC pollutant. In a constant rotational speed, HC emissions increased in dual fuel mode than diese mode. The reduced amount of pure air entering a cylinder caused by the existence of CNG fuel in the cylinder, causing a decrease in volumetric efficiency on dual fuel engine, which has a negative effect on combustion performances and emissions especially carbon monoxide and hydrocarbon which is consistent with the results of Yuvenda *et al*, (2020)'s research.

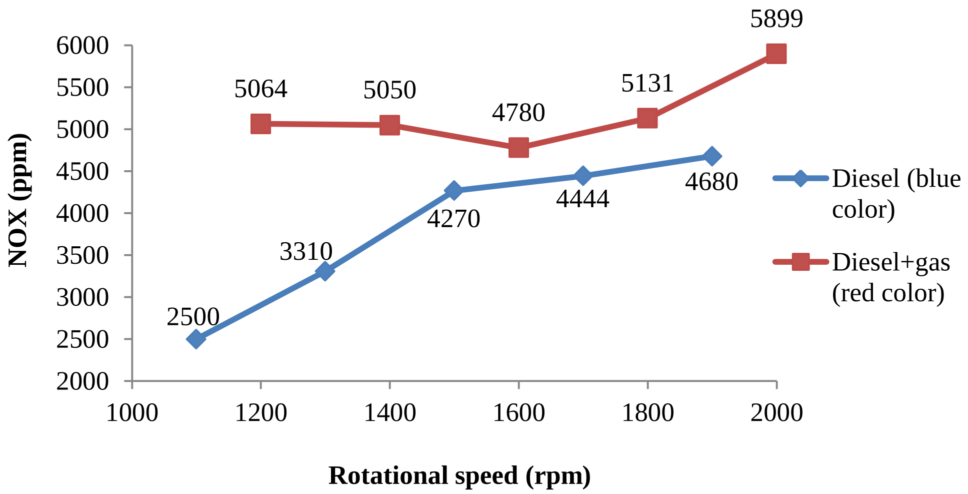



**Fig. 8.** $NO_X$ emissions vs. engine speed for diesel and dual-fuel modes

Figure 8 shows the amount of $NO_X$ emissions with both diesel fuel and dual fuel modes. According to the diagram in Figure 8, the amount of $NO_X$ produced by the engine was lower in the diesel mode than in the dual mode. Factors such as lean fuel mixture, rich mixture, increased ignition delay time and high combustion temperature contribute to the formation of particulate emissions (Yuvenda *et al.*, 2020). In the dual fuel mode, due to the richness of the fuel and the increase in ignition delay compared to the diesel mode, $NO_X$ production was higher, which is consistent with the research results of Piroozpanah & Abbas Alizadeh (1998) and Yuvenda *et al*, (2020).

The ratio of specific heat capacity of CNG is higher than that of air. Adding CNG increases the overall heat capacity of the mixture inside the cylinder, accordingly, the average temperature at the end of the compression stroke and during the overall combustion process decreases. Low combustion temperature reduces $NO_X$ formation. CNG injection reduces the amount of air and oxygen concentration in the cylinder charge, and as a result, the possibility of access to oxygen for the formation of $NO_X$ is reduced (Yuvenda *et al.*, 2020). But the release of more heat in the power cycle increases the maximum combustion temperature and this causes an increase in $NO_X$ emissions. Also, since in the dual system, the value of α increased with the increase in the rotational speed of the engine, or in other words, the amount of CNG consumption increased, so the amount of $NO_X$ also increased, which is in line with the results of Karagoz *et al*, (2016) matches.

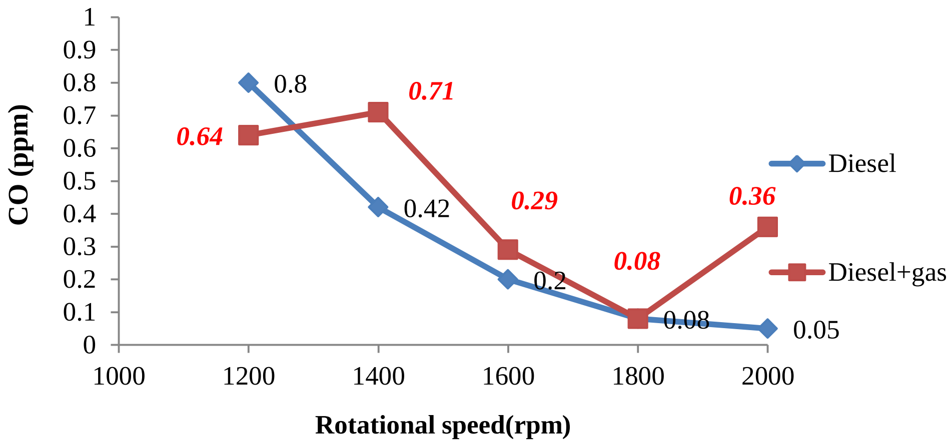



**Fig. 9.** CO emissions vs. engine speed for diesel and dual-fuel modes

Figure 9 shows the amount of CO emissions for both diesel fuel and dual fuel modes at different speeds. It is clear that increasing the engine speed decreased CO emissions for both modes. It can also be seen that at 1200 rpm, CO emissions showed the lower amount compared to diesel mode. However, compared the diesel mode, the CO emissions increased when the engine speed increased. CO emissions are caused by incomplete combustion of fuel. The most important reason for CO emissions is the use of rich fuel due to lack of oxygen. The reduced amount of pure air entering a cylinder caused by the existence of CNG fuel in the cylinder, causing a decrease in volumetric efficiency on dual fuel engine, which has a negative effect on carbon monoxide. This result has also been confirmed by Yuvenda *et al*, (2020). However, if the combustion temperature is lower than 1450 K, CO can also be produced in a lean fuel mixture (Shadidi *et al.*, 2020). This result is in agreement with the research result of Karagoz *et al.*, 2016; Cheenkachorn, Poompipatpong, & Ho, 2013 is fully compliant.

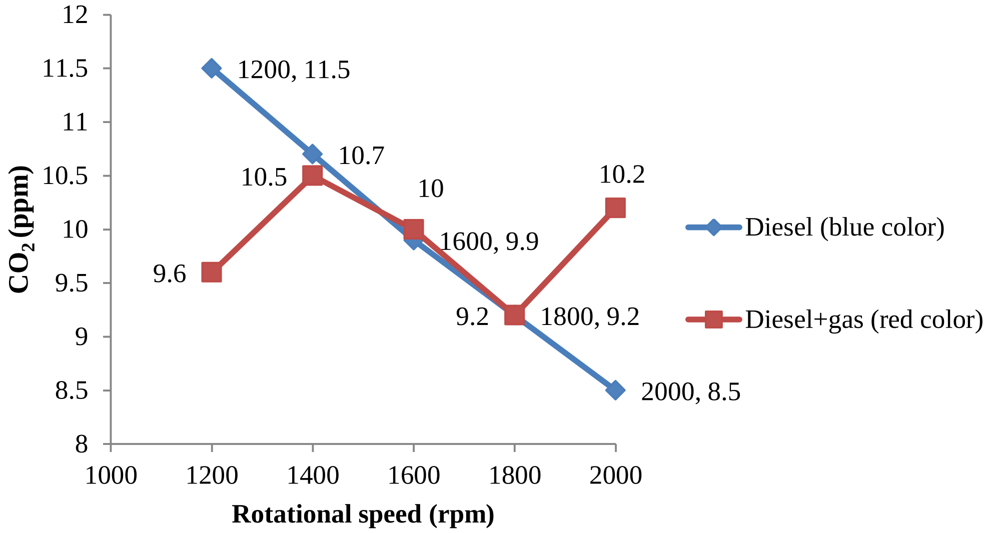



**Fig. 10.** $CO_2$ emissions vs. engine speed for diesel and dual-fuel modes

Figure 10 shows the amount of carbon dioxide ($CO_2$) produced by the engine for the diesel mode and the dual fuel mode. According to the obtained results, the amount of carbon dioxide produced by the engine is almost the same in both modes except in low and high engine speeds, and in lower rotational speeds, the carbon dioxide produced by the dual fuel is less than diesel fuel. Assuming the validity of the test data, the authors did not find convincing reasons for the higher and lower amount of carbon dioxide produced in diesel mode and dual fuel mode at 1200 and 2000 rpm, respectively.

Figure 11 shows the amount of $O_2$ produced by the engine at different rotational speeds for diesel fuel and dual fuel.

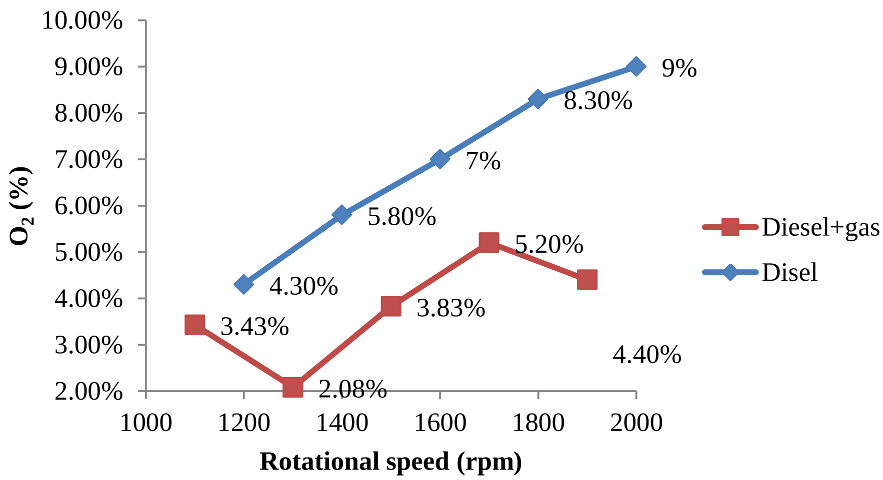



**Fig. 11.** $O_2$ emissions vs. engine speed for diesel and dual-fuel modes

According to Figure 11, the amount of $O_2$ produced by the engine in the mode of using dual fuel is on average 50% lower than the mode of using diesel fuel.

Figure 12 shows the amount of soot produced (output from the exhaust) of the engine for the diesel and dual fuel system. As it can be seen, with the increase in the rotational speed of the engine, the amount of soot emission has decreased in the dual mode which is consistent with the results of Bayat & Ghazikhani (2020)'s research. It seems at higher speeds, the increment in the amount of CNG is one of the main reasons contributing to the decrease in soot emissions. This is in good agreement with the results of Karagoz *et al*, (2016). Also as the CNG substitution ratio is increased, soot emissions get drastically reduced. The reason is the equivalence ratio distribution of air-fuel becomes more homogenous and the local fuel-rich region shrinks with increasing of CNG substitution ratios (Zhouab, Li, & Lee, 2019).

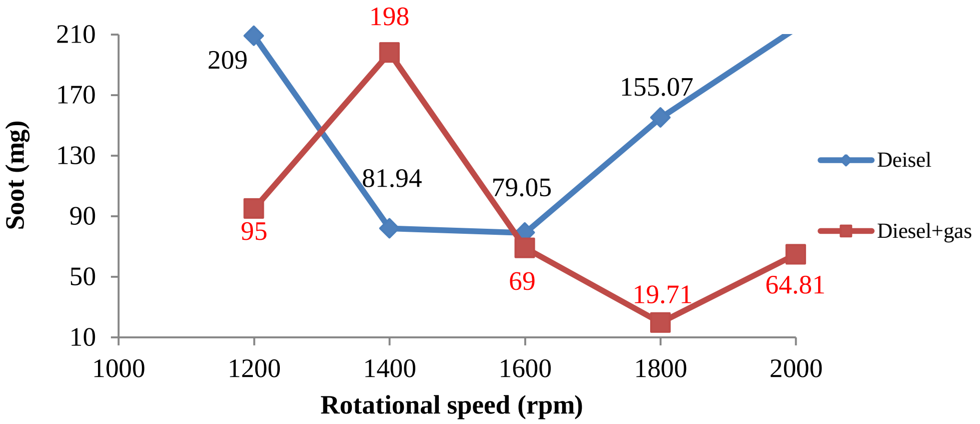



**Fig. 12.** Soot emissions vs. engine speed for diesel and dual-fuel modes

## Conclusion

The effect of a wide range of diesel engine speeds, varying from 1200 to 2000 rpm, on the performance and emissions of a dual-fuel compression ignition engine has been studied, experimentally. Compressed natural gas as the main fuel and diesel fuel as the igniter were used. In this study, a limited amount of diesel fuel as an igniter by a changing the mechanical structure of the governor in the injector pump, to inject in dual fuel mode.

Based on the performance analysis (considering torque and power), no significant difference was observed between the engine performance in dual fuel mode and diesel fuel mode. The maximum torque was 360.6 N m at 1400 rpm. While at dual fuel mode, corresponding maximum torque was 334 N m at 1600 rpm. Results showed only a 26 N m torque reduction in the dual fuel mode. At low speeds (1200 and 1400 rpm), the difference in torque produced by the engine in diesel and dual fuel modes were 53.4 and 68.8 N m, respectively. This difference is due to the fact that at low engine speeds, the amount of diesel fuel injected into the cylinders in diesel mode is close to the stoichiometric state compared to the amount of gas that enters in the form of natural suction in dual fuel mode. On the other hand, it is concluded that diesel fuel injection system (as igniter) in low speeds could be responsible for the reduction of torque. Considering high speeds, there is no remarkable difference in the torque of dual fuel mode and diesel mode.

Regarding the effect of different modes on exhaust emissions, results indicated an increment in the combustion emission including $NO_X$, HC and CO with a reduction of $O_2$ and soot emissions for dual fuel mode in comparison with diesel mode. However, effect of both modes on $CO_2$ emissions had the same trend.

Regarding mechanical injection system to inject igniter in dual fuel mode, there is a critical need for investigation of timing and quantity of fuel injection for better control of emissions. Also, CNG injection system could be considered for further research.

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جلد ۱۳، شماره ۱، بهار ۱۴۰۲، ص ۸۳-۶۷

# بررسی تجربی عملکرد موتور و انتشار آلاینده‌ها در یک موتور احتراق تراکمی با به‌کارگیری ترکیب سوخت گاز طبیعی فشرده و دیزل

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تاریخ دریافت: ۱۴۰۱/۰۷/۲۴
تاریخ پذیرش: ۱۴۰۱/۱۰/۱۷

## چکیده

این مطالعه اثرات سوخت گاز طبیعی فشرده را بر روی یک موتور احتراق تراکمی چهار سیلندر ارائه می‌کند. گاز طبیعی فشرده به‌عنوان سوخت اصلی و سوخت دیزل به‌عنوان آتش‌زنه برای بررسی عملکرد موتور و انتشار گازهای گلخانه‌ای در یک موتور دوگانه‌سوز به‌کار گرفته شد. با توجه به سرعت دورانی و بار موتور، مقدار سوخت دیزل به‌عنوان آتش‌زنه با استفاده از تغییرات مکانیکی در ناظم تنظیم شد، در حالی‌که از هیچ سامانه جرقه‌زنی الکترونیکی استفاده نشد. آزمون‌های تجربی موتور در سرعت‌های دورانی ۱۲۰۰، ۱۴۰۰، ۱۶۰۰، ۱۸۰۰ و ۲۰۰۰ دور در دقیقه با استفاده از سوخت دیزل و دوگانه‌سوز انجام شد. این داده‌ها در مرکز تحقیقات موتور شرکت موتورسازان تبریز جمع‌آوری و آزمون‌ها در سه بار تکرار انجام شد. بیشینه گشتاور موتور در حالت دیزل ۳۶۰ نیوتن‌متر در سرعت دورانی ۱۴۰۰ دور در دقیقه بود. در مقایسه با حالت دیزل، حالت دوگانه‌سوز بیشینه گشتاور را به میزان ۳۳۴ نیوتن‌متر در ۱۶۰۰ دور در دقیقه نشان داد که حدود ۲۶ نیوتن‌متر کمتر از گشتاور به‌دست آمده از حالت دیزل بود. با در نظر گرفتن آنالیز آلاینده‌ها در سرعت مشخصه ۲۰۰۰ دور در دقیقه، مشاهده شد که میزان انتشار $NO_X$، HC، $CO_2$ و CO در حالت دوگانه‌سوز به‌ترتیب ۲۰، ۵۳، ۱۶ و ۸۶ درصد بیش از سوخت دیزل بوده است. با این حال، $O_2$ و دوده بیشترین کاهش را در ۲۰۰۰ دور در دقیقه برای حالت دوگانه‌سوز به‌ترتیب ۵۱ و ۶۹ درصد نشان دادند. این مطالعه نشان داد که هنگامی که تزریق سوخت دیزل به‌عنوان آتش‌زنه به‌صورت مکانیکی انجام می‌شود، افزایش قابل‌توجهی در انتشار گازهای خروجی وجود دارد. در این راستا، کنترل مقدار و زمان تزریق احتمالا می‌تواند به کنترل بهتر انتشار گازهای گلخانه‌ای کمک کند. بنابراین، پژوهش‌های بیشتر در مورد اصلاح سامانه تزریق دیزل به‌عنوان آتش‌زن یا سامانه تزریق CNG به‌منظور کاهش انتشار گازهای گلخانه‌ای مورد نیاز است.

**واژه‌های کلیدی:** آلاینده‌ها، توان، سوخت دیزل، گاز طبیعی فشرده، موتور احتراق تراکمی

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https://doi.org/10.22067/jam.2023.79199.1127