A NEW STRATEGY FOR REDUCTION OF EMISSIONS AND ENHANCEMENT OF PERFORMANCE CHARACTERISTICS OF DUAL FUEL ENGINES AT PART LOADS

R. Khoshbakhti Saray*

Department of Mechanical Engineering, Sahand University of Technology
P.O. Box 51335-1996, Tabriz, Iran
khoshbakhti@sut.ac.ir

A. Mohammadi Kousha and V. Pirouzpanah

Department of Mechanical Engineering, University of Tabriz
P.O. Box 51666-14766, Tabriz, Iran
kousha@tabrizu.ac.ir - pirouz@tabrizu.ac.ir

*Corresponding Author

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Abstract Increasingly restrictive emission regulations and renewed focus on energy efficiency drive the current researches to find alternative fuels and their related better combustion strategies. In this regard, dual fuel engines, in which natural gas fuel is used as a main fuel and diesel fuel is employed as a pilot fuel, have received considerable attention. However, poor fuel utilization efficiencies and high emissions of HC and CO may be encountered at light loads. This study focuses on improving the aforementioned drawbacks. Exhaust gas recirculation (EGR) and its inherent thermal energy can be used as an effective way to improve the performance and emission parameters of these engines at part load conditions. Therefore, in the laboratory of authors, an experimental work was conducted on an IDI Lister (8-1) dual fuel engine to investigate the effects of different levels of EGR temperature on combustion process, performance and emissions of these engines. The amount of EGR conducted into the engine was altered but its temperature level was considered constant at 10 and 50 percents of full load of engine. Results of this work show that the ignition delay and combustion durations shorten sufficiently by increasing EGR percentage and its temperature to a specified level. Also, CO and UHC emissions reduce whereas NO_x emission increases but not too much for low percentage of EGR. Moreover, by employing low percentage of EGR, performance and emission parameters show better behavior in comparison with high percentage of EGR at a constant temperature.

Keywords Dual Fuel Engine, Combustion, Natural Gas, Pilot Fuel, Hot EGR

چکیده قوانین آلایندگی سخت‌گیرانه و تمرکز بر بازده انرژی محققان را برای یافتن سوخت‌های جایگزین و راهبردهای احتراقی بهتر مرتبط سوق می‌دهند. در این ارتباط، موتورهای دوگانه‌سوز که در آنها سوخت گاز طبیعی به‌عنوان سوخت اصلی و سوخت دیزل به‌عنوان سوخت آتش‌زا مورد استفاده قرار می‌گیرند، بیشتر مورد توجه قرار گرفته است. اما، موتورهای دوگانه‌سوز، در شرایط بارهای جزئی دارای عملکرد ضعیف و مقادیر بالای آلاینده‌های خروجی CO و UHC می‌باشند. این مطالعه بر بهبود معایب مذکور متمرکز می‌شود. در این راستا، بازخورانی گازهای خروجی (EGR) و انرژی حرارتی ذاتی آن می تواند به عنوان یک روش موثر در بهبود عملکرد و آلایندگی این موتورها در شرایط بارهای جزئی مورد استفاده قرارگیرد. به همین دلیل، در آزمایشگاه مولفین، به‌منظور بررسی تاثیر مقادیر مختلف دمای EGR بر روی فرآیند احتراق، عملکرد و آلایندگی این موتورها در شرایط بارهای جزئی، آزمایش های تجربی بر روی موتور LISTER(8-1) دوگانه‌سوز از نوع پاشش غیرمستقیم انجام شد. گازهای خروجی با مقادیر متفاوت در شرایط ۱۰ و ۵۰ درصد بار کامل با مقادیر دمایی ثابت بازخورانی شدند. نتایج نشان می‌دهند که با افزایش EGR، زمان تأخیر در اشتعال و دورهٔ احتراق کوتاه‌تر می‌شود. همچنین، آلاینده‌های CO و UHC کاهش یافته ولی آلایندهٔ NO_x افزایش می یابد که این افزایش به‌ازای مقادیر پایین EGR ناچیز می‌باشد. به علاوه پارامترهای عملکردی و آلایندگی به ازای استفاده از EGR با مقادیر پایین در مقایسه با مقادیر بالای آن رفتار بهتری از خود نشان می‌دهند.

1. INTRODUCTION

Nowadays, achieving simultaneous reduction of NO_x and particulate emissions without compromising engine performance is the focus of diesel engines research works. Since this is a difficult task, it is better to consider other alternatives to diesel engines. These alternatives can be spark ignited natural combustion engines (dedicated), homogeneous charge compression ignition (HCCI) engines and dual fuel combustion engines in which very small amount of pilot fuel is used as an ignition source of the gaseous fuel air mixture. Combustion process in the first category is imitated by SI engines combustion. Therefore, they lose diesel engine benefits. In the second category, diesel-like or higher efficiencies and much lower emissions are proposed. But, control of combustion in HCCI mode as well as achieving higher brake mean effective operations continue to pose serious challenges [1]. Ultimately, the dual fuel engines are shown to match diesel efficiencies and produce significantly lower particulate emissions and a little higher amount of NO_x at higher loads [2-4]. However, engine performance and emissions suffer at low loads when operating in dual fuel mode [2-6]. The main reason for this poor light load performance is due to the very lean mixtures [2-6]. The lean mixtures of natural gas and air are hard to ignite and slow to burn.

Pirouzpanah, et al [3] conducted an experimental study to determine performance and emission characteristics of an automotive direct injection dual-fuelled diesel engine. Cooled EGR was used to resolve the poor light load performance of the engine. Results showed that at part loads, the application of EGR could considerably reduce CO and UHC emissions.

Karim, et al [7] discussed briefly the nature of the combustion processes and their influence on the production of the major exhaust pollutants associated with dual fuel operation, mainly at relatively light load. Some measures such as increasing pilot fuel quantity, increasing gaseous fuel equivalence ratio and total equivalence ratio were used to control performance and emission characteristics of dual fuel engines at part loads.

Karim, et al [8-9] examined the effects of residual gases on the compression ignition and combustion processes through analyzing the cyclic variations of autoignition in a motored engine fuelled with homogeneous gaseous fuel air mixtures. It was shown that controlled EGR could enhance the autoignition processes in the gas fuelled compression ignition engines by suitably seeding the intake charge of the current cycle with the chemical species found in the exhaust gases of the previous cycle.

Abd Alla, et al [10-12] conducted some experiments on a Ricardo E6 dual fuel engine to investigate the effect of pilot fuel quantity and admission of high percentages of EGR gases on the performance and exhaust emissions of dual fuel engines at part loads. The results showed that both of the methods could improve performance and emission characteristics except for NO_x emission when using higher amounts of pilot fuel quantity.

Daisho, et al [2,6,13] studied experimentally the effect of EGR on combustion and exhaust emission characteristics of dual fuel engines with the objective of improving their drawbacks at part loads. It was shown that hot EGR with high percentages could improve performance and emission characteristics of these engines at part loads.

Krishnan, et al [14] examined the influence of engine operating variables on the performance, emissions and heat release rate in a dual fuel engine. Experimental results established the importance of increasing intake manifold pressure and temperature in improving dual fuel performance and emissions at part loads.

Papagiannakis, et al [15] investigated experimentally the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine. Results revealed the effect of pilot fuel replacement by natural gas on engine performance and emissions.

Pirouzpanah, et al [16-17] investigated theoretically the combustion phenomenon of dual fuel engines at part loads and using hot exhaust gas recirculation (EGR) to improve the aforementioned drawbacks. By employing this technique, it was found that, lower percentages of EGR considering its thermal and radical effects had the positive effect on performance and emission parameters of dual fuel engines at part loads.

The purpose of the present contribution is to investigate experimentally the emission and performance characteristics of a dual fuel engine being operated on natural gas with pilot diesel injection. As mentioned before, the dual fuel engines at part loads inevitably suffer from lower thermal efficiency and higher emission of carbon monoxide and unburned fuel. Therefore, this work is an attempt to investigate the nature of poor and complex combustion phenomenon at part loads and hot EGR has been used to solve the aforementioned problems.

2. EXPERIMENTAL ARRANGEMENT

2.1. Experimental Apparatus A schematic illustration of the experimental arrangement used to investigate dual fuel engine operation is shown in Figure 1. A single cylinder Lister indirect injection compression ignition engine was coupled to a D.C. dynamometer through a torque meter which provided both torque and speed measurements. The engine details are shown in Table 1.

Air flow rate was derived from the measured pressure drop across an orifice installed on a surge tank. Mass flow rates of diesel and natural gas fuels were measured by volumetric flow meters. Temperature of cooling water, lubricating oil, inlet air and exhaust gas were also measured to ensure proper engine operating conditions. A data acquisition system was used to collect the important data and store them in a personal computer for exact analysis. The cylinder pressure data and crank angle degrees signal were fed into the computer. The pressure transducer was Kistler-6123 piezoelectric type which was remote-mounted with the main combustion chamber. Optical Shaft encoder was used which was coupled to the engine crankshaft. A TNM-DS20080 digital oscilloscope which had two output channels was used to collect the cylinder pressure data and crank angle position.

Also, EGR line and an electric heater were designed and manufactured to enter different amounts of hot EGR to the intake charge. EGR temperature can be varied in the range of 298 to 1000 K by the electric heater. Electric heater has been constructed from thin quartz glasses in which electric elements has been placed within each of the glasses as shown in Figure 2.

Engine emissions were measured using an AVL Dicom4000-class1 exhaust gas analyzer. Unburned hydrocarbons, CO and CO_2 were measured using a non dispersive infrared detector while an electrochemical detector was used for O_2 and NO_x measurements.

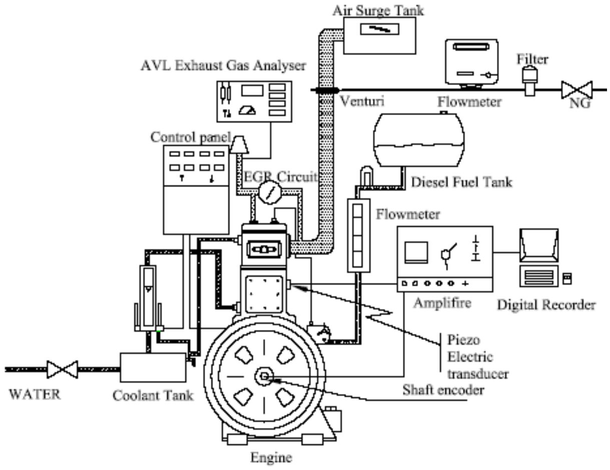


Figure 1. Experimental layout.

TABLE 1. Engine Specifications.

Engine Make	Lister (8-1)
Engine Type	Four stroke, Single Cylinder, Compression Ignition, Swirl Type IDI
Bore (mm)	114.1
Stroke (mm)	139.7
Con-Rod Length to Crank Radius	4
Displacement (Litres)	1.43
Compression Ratio	17.5:1
Max. Power	8 hp at 850 rpm
Injection Pressure (Kg/cm^{2})	91.7
Injection Timing	20°CA BTDC
Valve Timing	IVO = 5°CA BTDC
	IVC = 15°CA ABDC
	EVO = 55°CA BBDC
	EVC = 20°CA ATDC

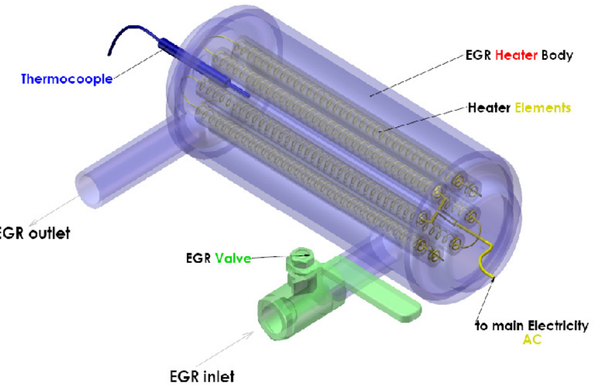


Figure 2. EGR electric heater layout.

2.2. Test Cases Examined Experiments were conducted on an IDI dual fuel engine connected with a D.C. electrical dynamometer. After starting the engine on diesel mode, different performance and emission tests conducted on the engine at diesel mode to gain the base diesel engine performance and emission characteristics at different loads. Then, the amount of diesel fuel was reduced and the rest of input energy was supplied by a venturi type mixer installed on the intake manifold which can introduce sufficient amount of natural gas fuel to operate engine in dual fuel mode at various load conditions. During engine tests at part loads, the amount of hot EGR introduced to the engine were increased progressively until the performance and emission parameters were deteriorated. It is necessary to mention that the amounts of pilot and natural gas fuels were kept constant at each set of experiments to evaluate the effect of hot EGR on the combustion, performance and emission parameters.

EGR percentage was calculated using the measured amounts of CO_2 concentration in the intake mixture and exhaust gases as following:

$$\text{EGR }\% = \frac{CO_{2,\text{intake}}}{CO_{2,\text{exhaust}}} \times 100 \tag{1}$$

Total equivalence ratio, ϕ, is defined as the ratio of the mass of the stoichiometric amount of air required for the combustion of both of the gaseous and the pilot diesel fuels to the mass of the actual amount of air drawn in:

$$\phi = (14.532\,\dot{m}_p + 16.684\,\dot{m}_{NG}) / \dot{m}_a \tag{2}$$

The gas equivalence ratio, ϕ_{gas}, is defined as the ratio of the mass of the stoichiometric amount of air required for the combustion of the gaseous fuel to the mass of the actual amount of air drawn in:

$$\phi_{Gas} = (16.684\,\dot{m}_{NG}) / \dot{m}_a \tag{3}$$

In which, $\dot{m}_a$, $\dot{m}_p$ and $\dot{m}_{NG}$ are the mass flow rate of air, pilot and natural gas fuels, respectively.

The details of examined test cases are shown in Table 2.

3. COMBUSTION ANALYSIS

The scaled absolute cylinder pressure data and the TDC pickup signal were used to estimate the ignition delay, duration of combustion and the rate of heat release. In this paper, the first and second derivatives of cylinder pressure were used to predict the start of combustion and so the ignition delay period [18]. Also, the heat release rate diagrams provided valuable information about combustion and its related parameters. The net heat release rate was determined by applying the first law of thermodynamics for each measured cylinder pressure data using the following expression [19]:

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \tag{4}$$

Where γ is the specific heat ratio. For diesel engines with lean mixtures, Egnell [20] proposed the following formula for calculating specific heat ratio:

$$\gamma = 1.38 - 0.2\exp(-900 / T(K)) \tag{5}$$

Also, for calculating specific heat ratio for dual fuel engine cases, Brunt's formula [21] was used as following:

$$\gamma = 1.338 - 6.0 \times 10^{-5}\,T + 1.0 \times 10^{-8}\,T^2 \tag{6}$$

Where T is temperature in K. The cylinder charge

TABLE 2. Quantities of Different Variables for Test Cases Examined.

Case	EGR Percentage (%)	Intake Temperature (K)	Power Output (kW)	BSEC (kJ/kW.hr)	ϕ	ϕ_{gas}	Volumetric Efficiency (%)
1	0	298	0.549	81661	0.64	0.58	80
	4.8	413	1.717	26572	0.9	0.79	57
	11.9	413	1.145	37608	0.82	0.72	55
2	0	298	2.545	19367	0.72	0.64	78
	4.1	347	3.015	17024	0.95	0.85	61
	6.6	347	2.623	18792	0.91	0.82	60

temperature was calculated by a single zone combustion model which developed for considering both diesel and dual fuel engine combustion [22]. Heat transfer was calculated by Annand's correlation [23].

The measured in-cylinder pressure data was recorded for 10 cycles in a file with a sampling rate corresponding to 0.5˚CA. These data were initially corrected for any drift of the signals obtained from the charge amplifier. A signal from an optical shaft encoder was providing the position of TDC in each cycle. Then, the raw cylinder pressure data were converted to absolute pressures by attributing the intake manifold pressure as a reference pressure at BDC. Ultimately, the mean of the cycle indicator diagrams was obtained while a light smoothing for the pressure signals was applied based on twice smoothing for each of ten data points. This seemed to offer a reasonable compromise between the valuable signal (without the loss of any information) and relatively the smooth values for the first derivative of pressure. Also, five points 4^{th} order difference scheme was used for calculations of 1^{st} and 2^{nd} derivatives of in-cylinder pressure.

4. RESULTS AND DISCUSSION

As already mentioned, dual fuel engines use very lean mixtures at part load conditions. Therefore, at these circumstances, products of combustion process in the exhaust gases will be unburned fuel and active radicals such as H, O, OH and etc. When these active radicals along with the inert gases in the exhaust gases are introduced into the intake mixture by EGR gases, they will have chemical and dilution effects respectively, on the ignition and combustion processes. Meanwhile, hot EGR will have thermal effect besides chemical and dilution effects. Therefore, this kind of EGR has profound effect (chemical and thermal) on the ignition and combustion processes [24].

To study the effects of hot EGR on the ignition and combustion processes and consequently on the performance and emission parameters of dual fuel engines at part loads, the experimental measurements covered the spectrum of loads from 10 % to 50 % of full load at a constant engine speed equal to 730 rpm. Injection timing of pilot fuel was 20˚CA BTDC in all of the experiments. Fixed amounts of pilot fuel 0.095 and 0.126 (kg/hr) have been used for 10 and 50 percents of full load, respectively. Full load is defined as the load of dual fuel engine in which the maximum amount of brake power can be produced by the engine. EGR percentage was varied by the EGR valve installed on the EGR line and its temperature was altered by an electric heater installed on the EGR line. Therefore, different levels of EGR temperature and consequently different levels of intake mixture temperature in the ranges between 298 and 500 K were achieved by adjusting the temperature setting of the electric heater.

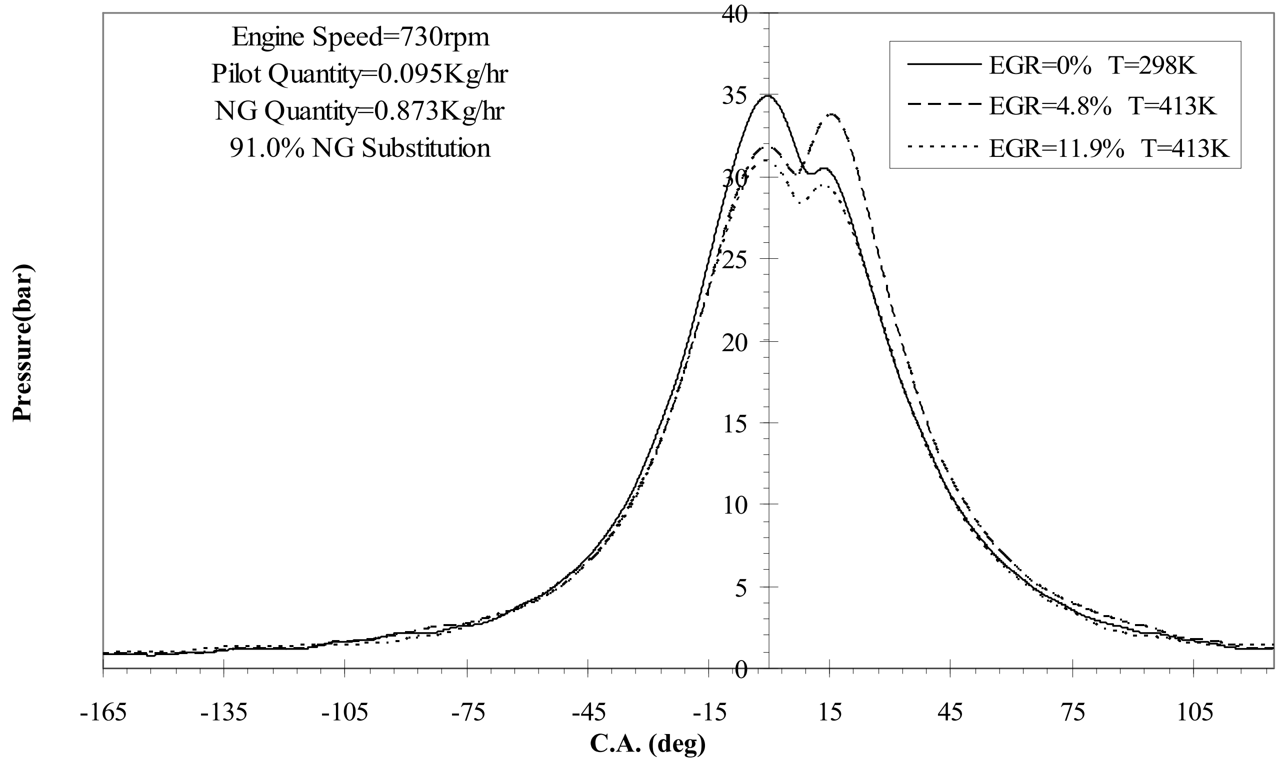
Figure 3 shows variations of the cylinder pressure with the crank position for constant intake mixture temperatures and various percentages of EGR for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. It can be observed that by increasing the EGR percentages up to 4.8 and 4.1 and increasing intake mixture temperature up to 413 K and 347 K for 10 and 50 percents of full load, respectively, peak cylinder pressure increases and shifts to the suitable crank position (about 15˚CA ATDC) to improve performance parameters. Also, it is necessary to say that increasing the EGR percentages at a fixed temperature (413 K and 347 K for 10 and 50 percents of full load, respectively) to levels more than the specified values (4.8 and 4.1 for 10 and 50 percents of full load respectively) will lead to decrease the in-cylinder pressure.

Figure 4 describes variations of the net heat release rate with crank position for constant intake mixture temperatures and various percentages of EGR for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. As indicated in this figure, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, heat release rate increases. As stated before, EGR can promote the combustion process due to increasing total equivalence ratio, the intake temperature of the charge and preparing better fuel air mixing ready for combustion. Also, apart from its thermal effect, EGR tends to improve the preignition reaction rates of the cylinder charge by suitably seeding the intake charge with partial oxidation products that are sources of fruitful active radicals. But, with increasing the EGR percentages at the specified fixed temperatures to the levels more than the aforementioned levels, heat release rates show a tendency of reduction. This trend may be originated from this description that the mentioned positive effects of EGR are moderated by two important effects which show themselves at higher EGR percentages and intake mixture temperatures. These are diluting effects of some products in EGR gases and the negative effect of higher intake temperature on the engine charging efficiency.

Figures 5 and 6 show the variations of the 1st and 2nd derivatives of cylinder pressure with crank position for constant intake mixture temperatures and various percentages of EGR for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. The minimum position of the 1st derivative of cylinder pressure after injection timing can be considered the start of combustion, whereas at that position the 2nd derivative is equal to zero and the 3rd derivative is greater than zero. It can be observed that, for both loads by increasing the EGR percentage and intake mixture temperature up to the mentioned critical levels, combustion process starts earlier than the baseline dual fuel engine in which the fuel air mixture is lean. But the combustion process is postponed when the EGR percentage and intake mixture temperature are higher than the critical levels. Therefore, ignition delay of the cylinder charge shortens for the former case and prolongs for the latter case. These trends are consistent with those observed in the heat release rate curves.

Figure 7 describes the variations of the brake power with percentage of EGR for a constant intake mixture temperature for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. It can be observed that, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, brake power increases significantly. But, with increasing the EGR percentages at the specified constant intake mixture temperatures to the levels more than the aforementioned levels, brake power shows a tendency of decrease. These trends are consistent with those observed in the heat release rate curves.

Figure 8 indicates the variations of the brake specific energy consumption (BSEC) with percentage of EGR for a constant intake mixture temperature for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a



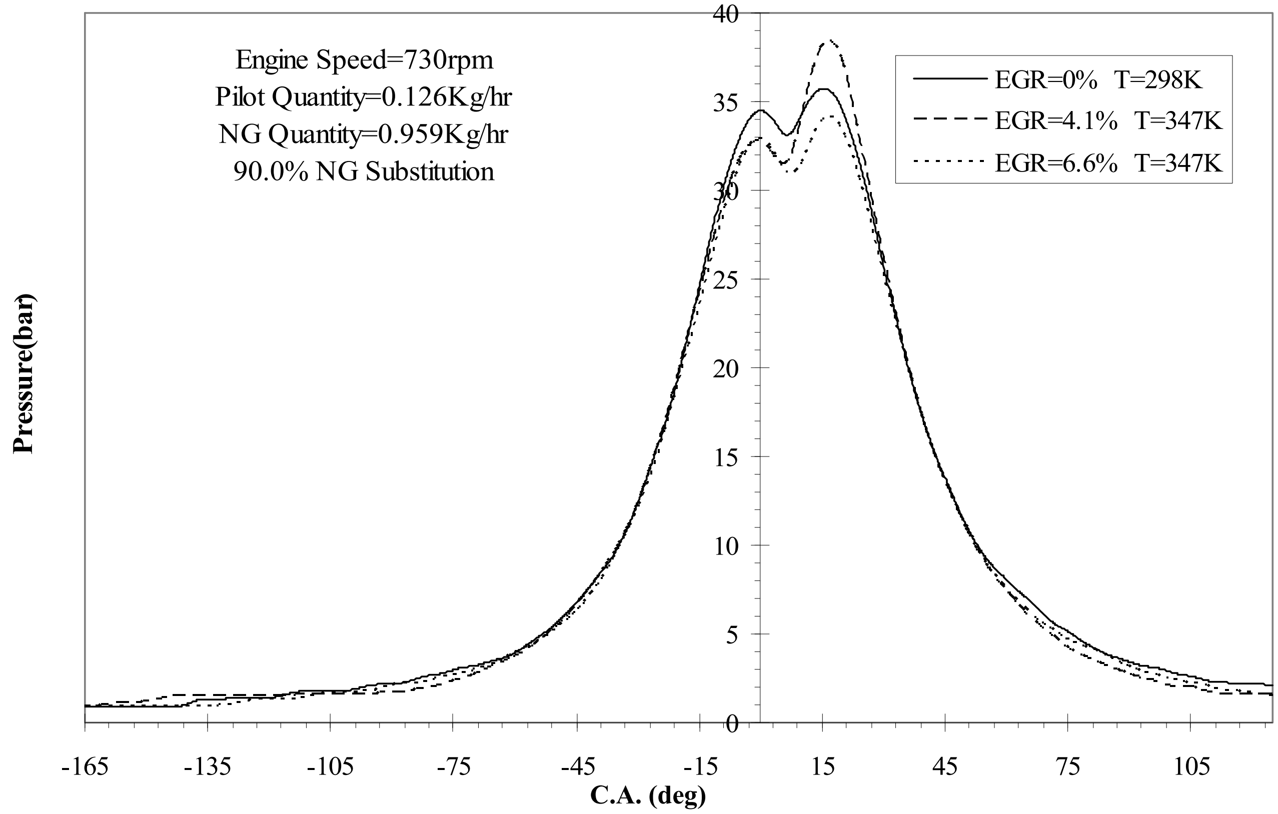
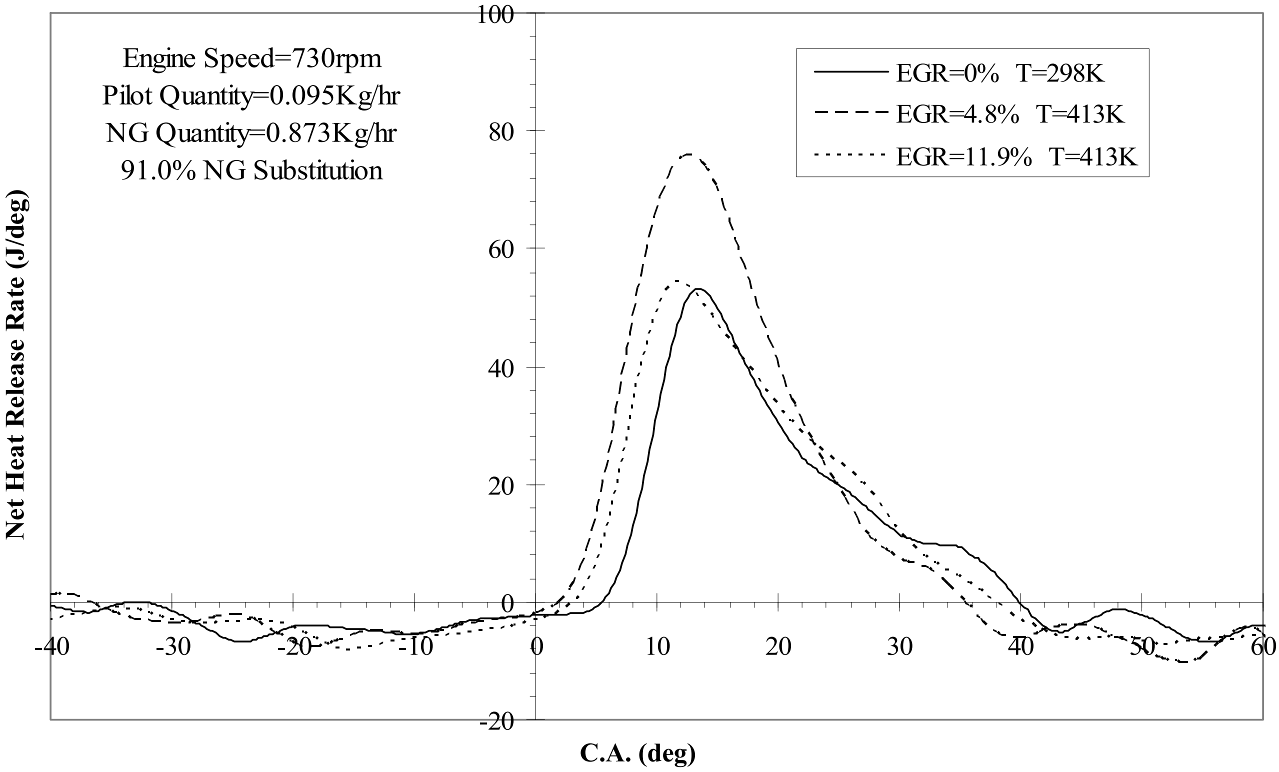


Figure 3. Variations of the cylinder pressure with crank position for constant intake mixture temperatures and different percentages of EGR at 10 and 50 percents of full load.



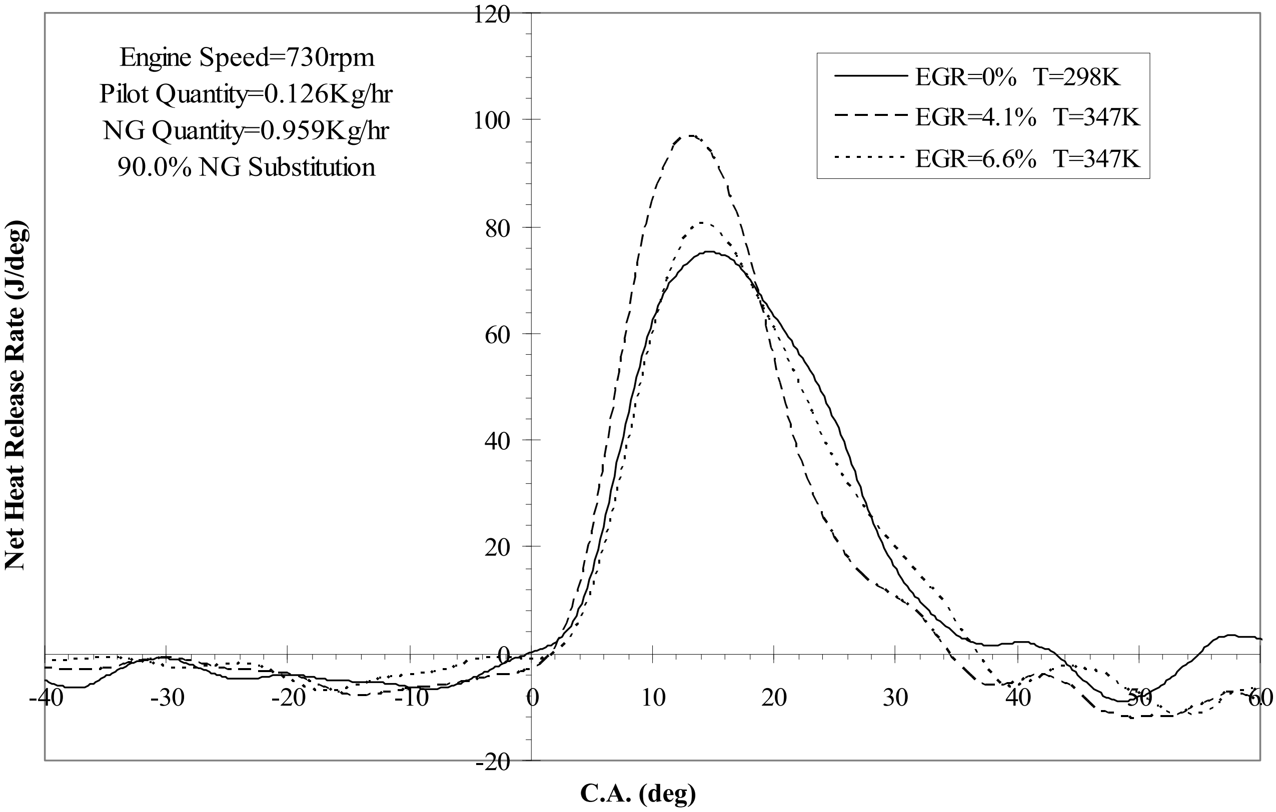


Figure 4. Variations of the net heat release rate with crank position for constant intake mixture temperatures and different percentages of EGR at 10 and 50 percents of full load.

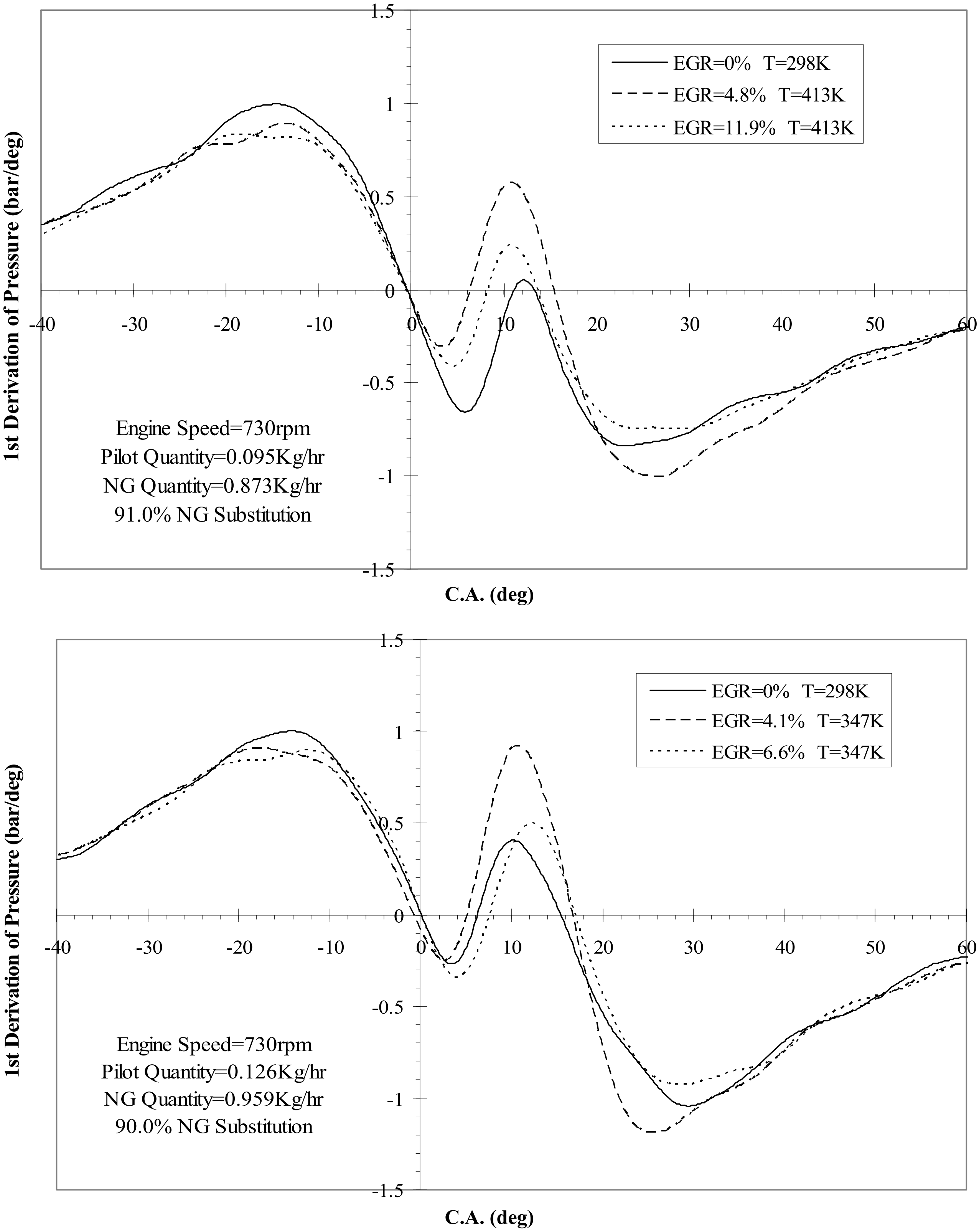


Figure 5. Variations of the 1st derivative of cylinder pressure with crank position for constant intake mixture temperatures and different percentages of EGR at 10 and 50 percents of full load.

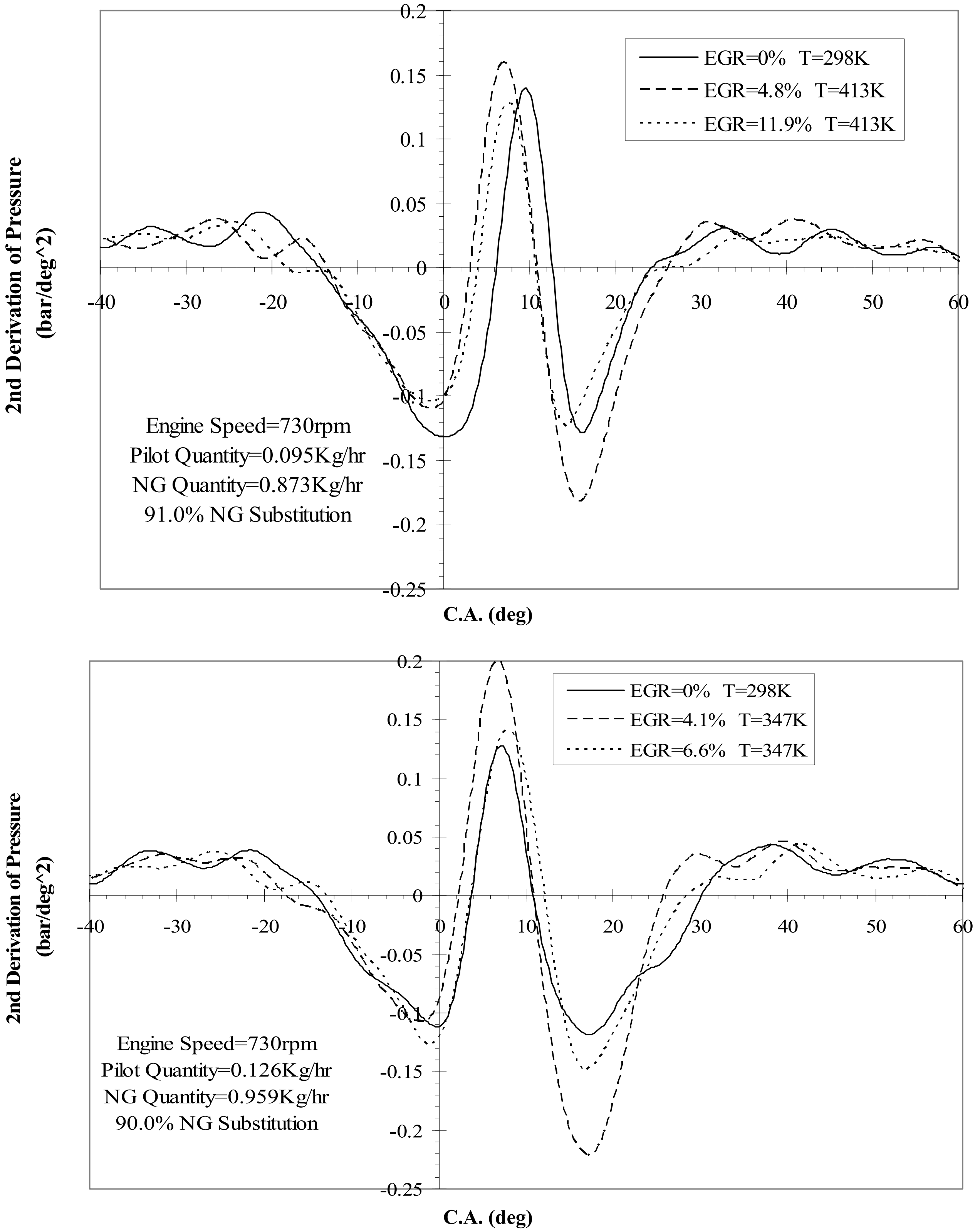


Figure 6. Variations of the 2^{nd} derivative of cylinder pressure with crank position for constant intake mixture temperatures and different percentages of EGR at 10 and 50 percents of full load.

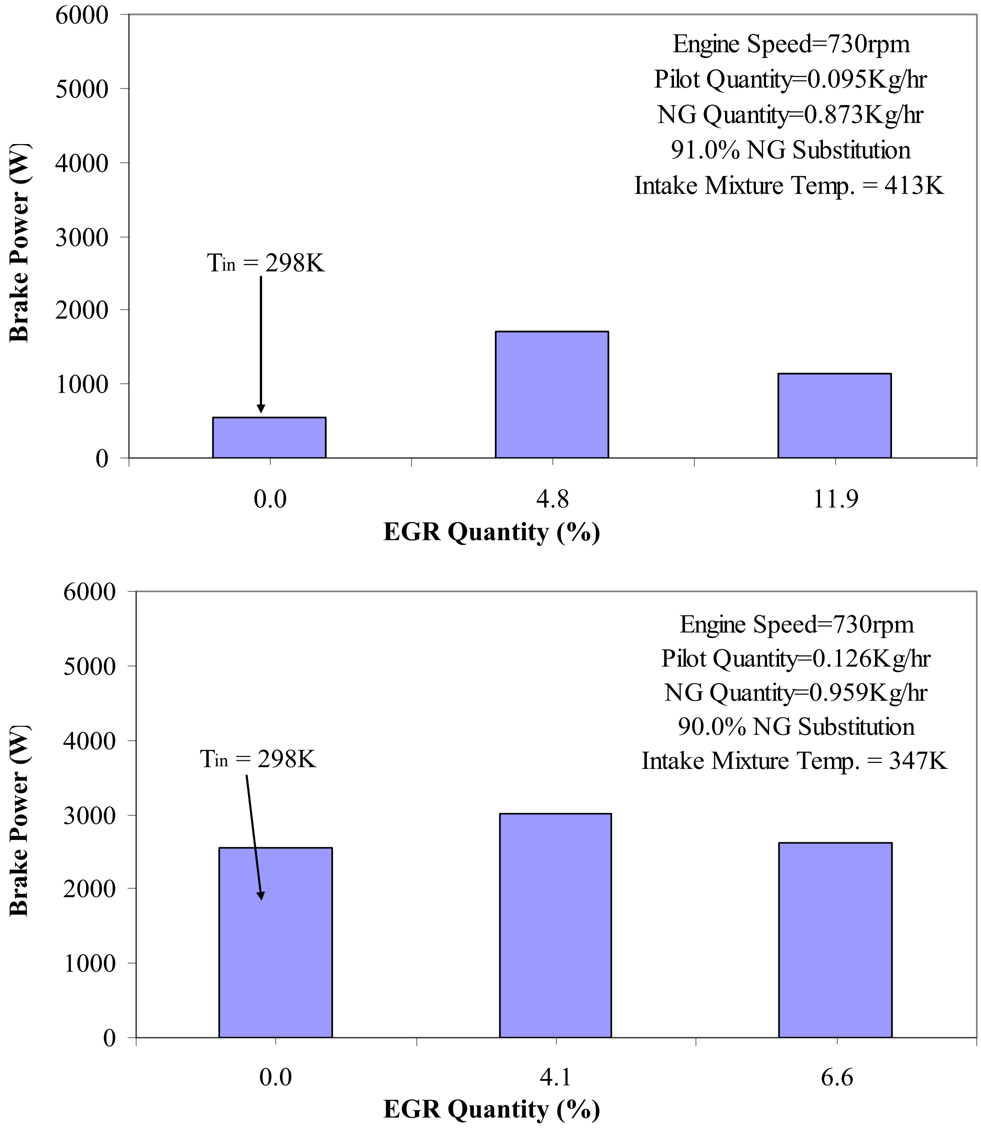


Figure 7. Variations of the brake power with EGR percentage for constant intake mixture temperatures at 10 and 50 percents of full load.

dual fuel engine. As indicated in this figure, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, BSEC decreases remarkably. But, with increasing the EGR percentages at the specified constant intake mixture temperatures to the levels more than the aforementioned levels, BSEC shows a tendency of increase. These trends are consistent with those observed in the heat release rate curves.

Figure 9 describes the variations of the unburned hydrocarbons emission with percentage of EGR for a constant intake mixture temperature for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. It can be observed that, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, UHC decreases remarkably. The main reason for this trend is perhaps due to overcoming the positive effects of EGR on its negative effects. Therefore, the combustion process promotes and the amount of UHC decreases significantly. But, with increasing the EGR percentages at the

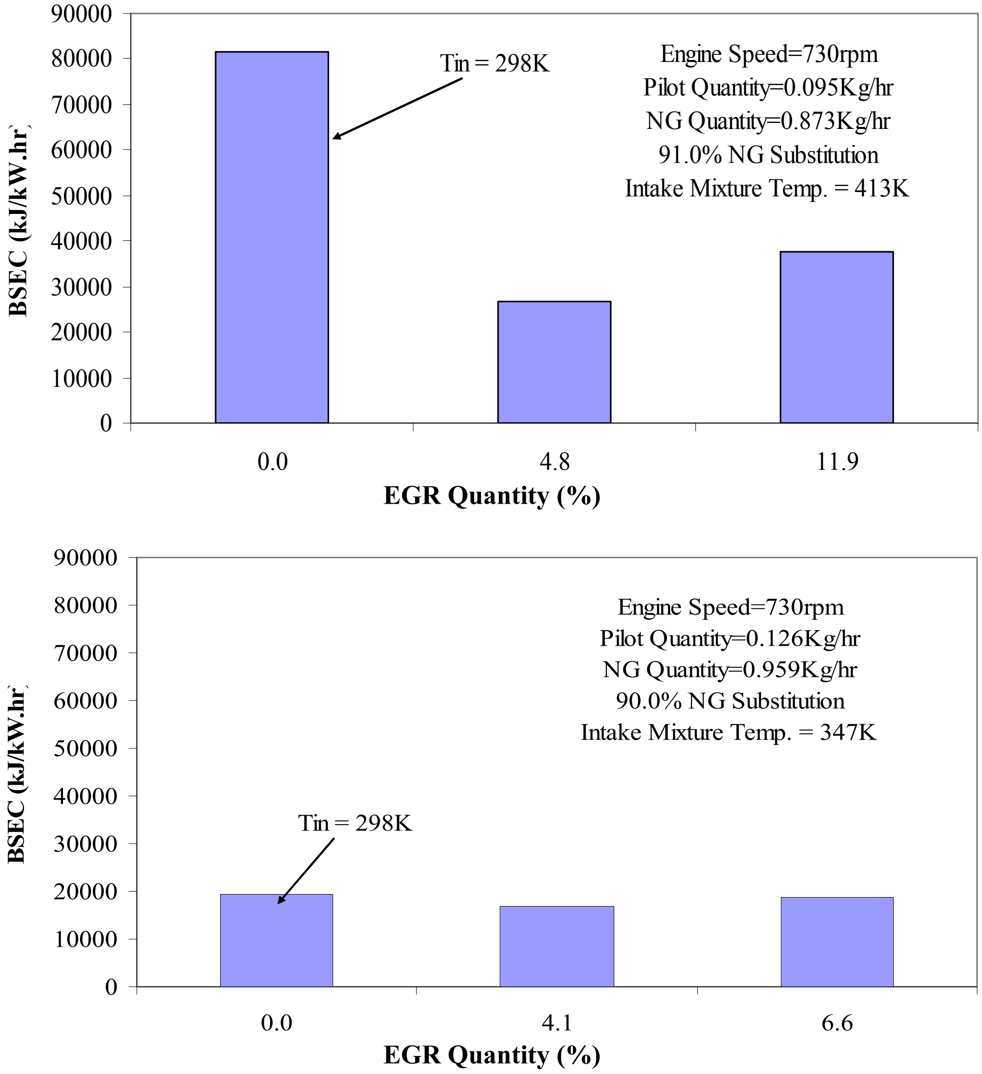
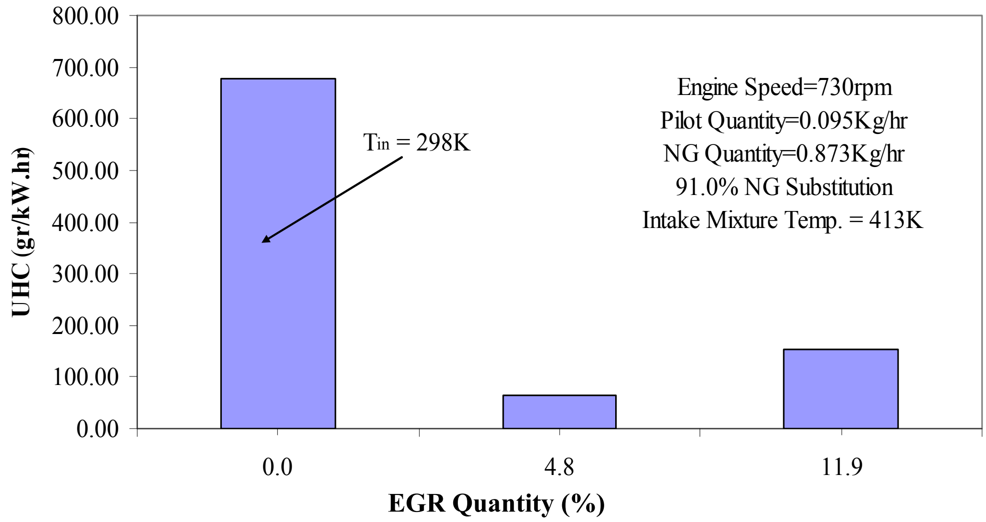


Figure 8. Variations of the brake specific energy consumption with EGR percentage for constant intake mixture temperatures at 10 and 50 percents of full load.

specified constant intake mixture temperatures to the levels more than the aforementioned levels, UHC shows a tendency of increase. This trend may be due to increasing the negative effects of EGR that can be the result of employing high percentages of EGR.

Figure 10 shows the variations of the carbon monoxide emission with percentage of EGR for a constant intake mixture temperature for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. It can be observed that, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, CO decreases remarkably. But, increasing the EGR percentages at the specified constant intake mixture temperatures to the levels more than the aforementioned levels, CO shows a tendency of increase. These trends are consistent with those observed in the UHC curves.

Figure 11 describes the variations of the nitrogen oxides emission with percentage of EGR for a constant intake mixture temperature for a fixed amount of pilot and gaseous fuels at 10 and



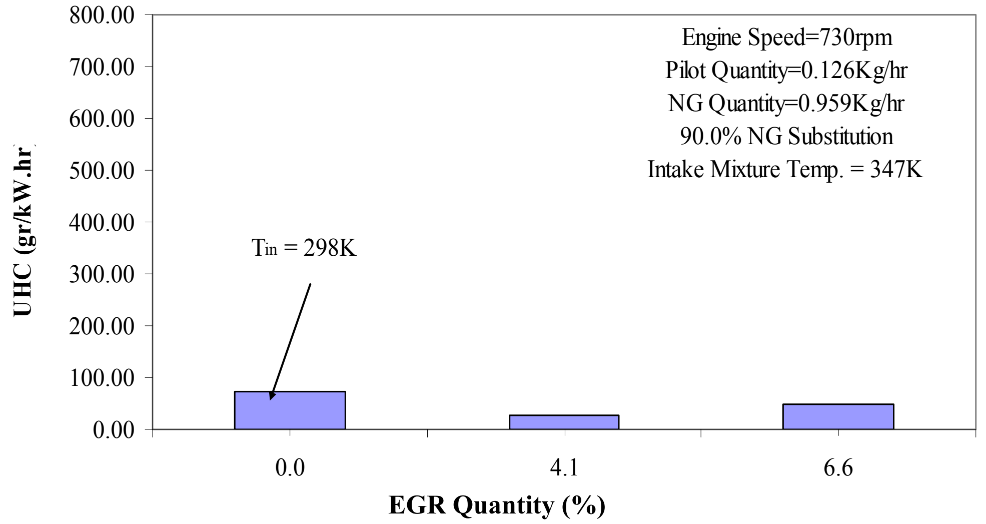
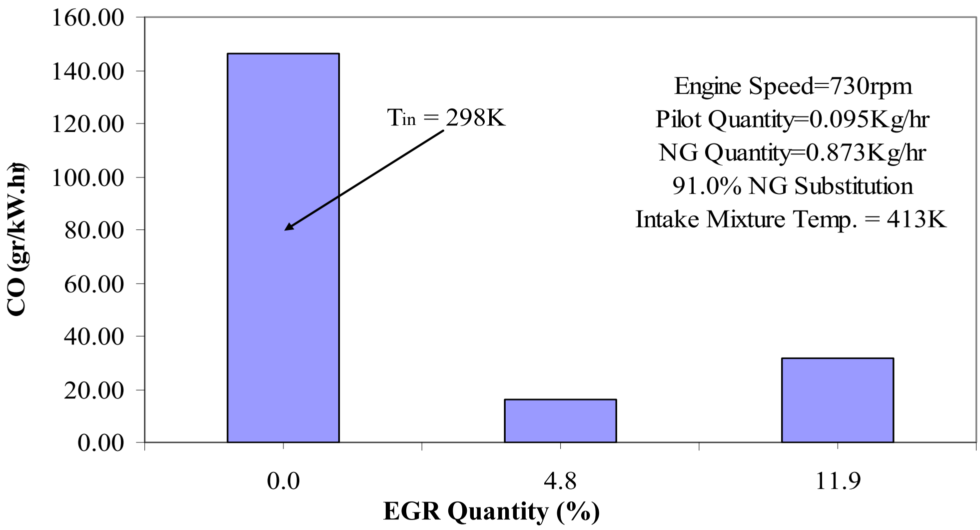


Figure 9. Variations of the unburned hydrocarbons emission with EGR percentage for constant intake mixture temperatures at 10 and 50 percents of full load.

50 percents of full load of a dual fuel engine. As indicated in this figure, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, NO_x increases but this increase is not remarkable. As mentioned before, the enhancement of the combustion process due to hot EGR increases NO_x emission. But, with increasing the EGR percentages at the specified constant intake mixture temperatures to the levels more than the aforementioned levels, NO_x shows a tendency of decrease. This trend may be due to increasing the diluting effects of EGR that can be the result of employing high percentages of EGR at a constant intake mixture temperature.

Figure 12 shows the variations of the oxygen concentration in the intake and exhaust manifolds with percentage of EGR for a constant intake mixture temperature for a fixed amount of pilot and gaseous fuels at 10 and 50 percents of full load of a dual fuel engine. It can be observed that, by increasing the EGR percentage and the intake mixture temperature to the aforementioned levels for 10 and 50 percents of full load, O_2 concentration



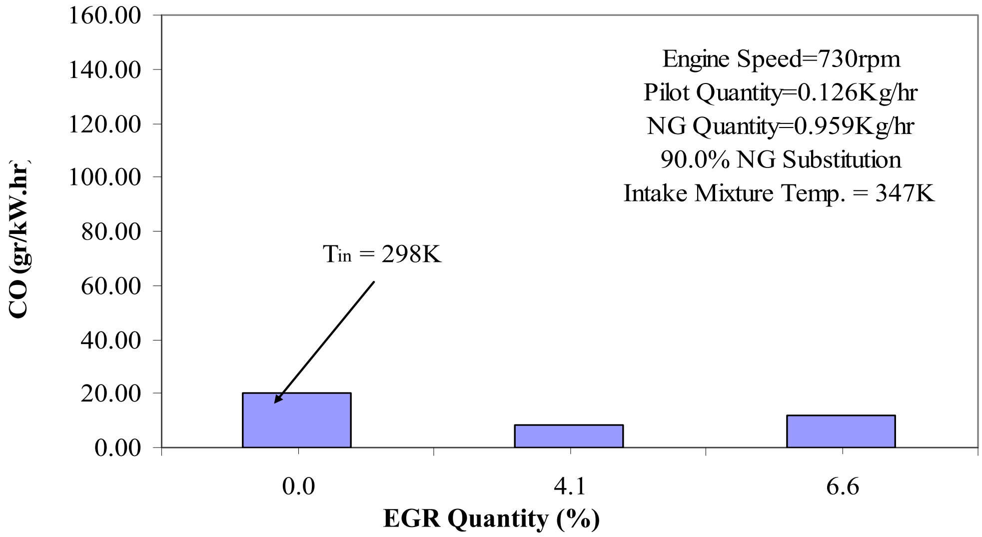
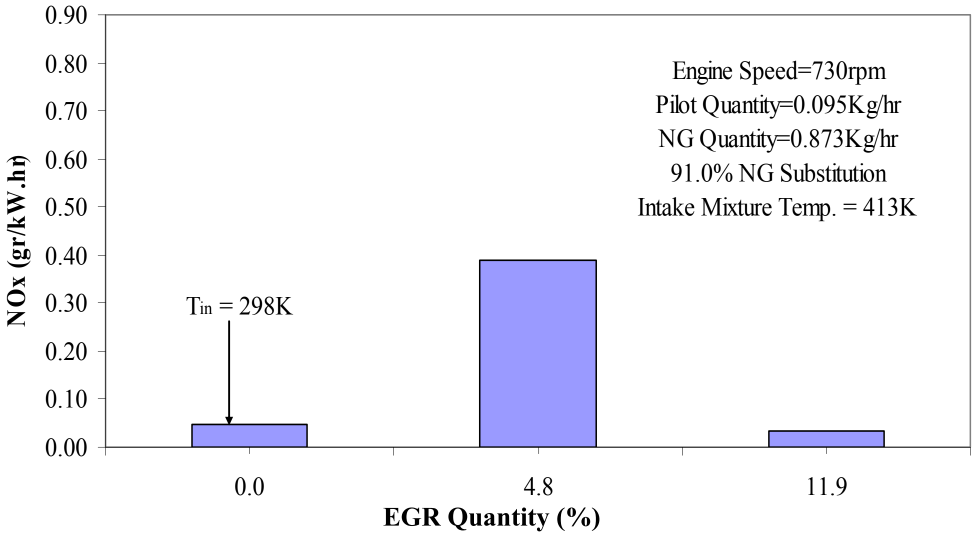


Figure 10. Variations of the carbon monoxide emission with EGR percentage for constant intake mixture temperatures at 10 and 50 percents of full load.

in the intake mixture does not decrease remarkably but it decreases in the exhaust mixture significantly. These trends are perhaps due to lower percentages of hot EGR which can have better effect on the combustion process and consequently on the performance and emission parameters. But, with increasing the EGR percentages at the specified constant intake mixture temperatures to the amounts more than the aforementioned levels, O_2 concentration in the exhaust does not decrease significantly as seen in the UHC and CO curves. This trend may be due to increasing the diluting effects of EGR that can be the result of employing high percentages of EGR at a constant intake mixture temperature.

5. CONCLUSIONS

A single cylinder indirect injection compression ignition engine was used to investigate hot EGR strategy for achieving low emissions and good performance from pilot ignited natural gas dual



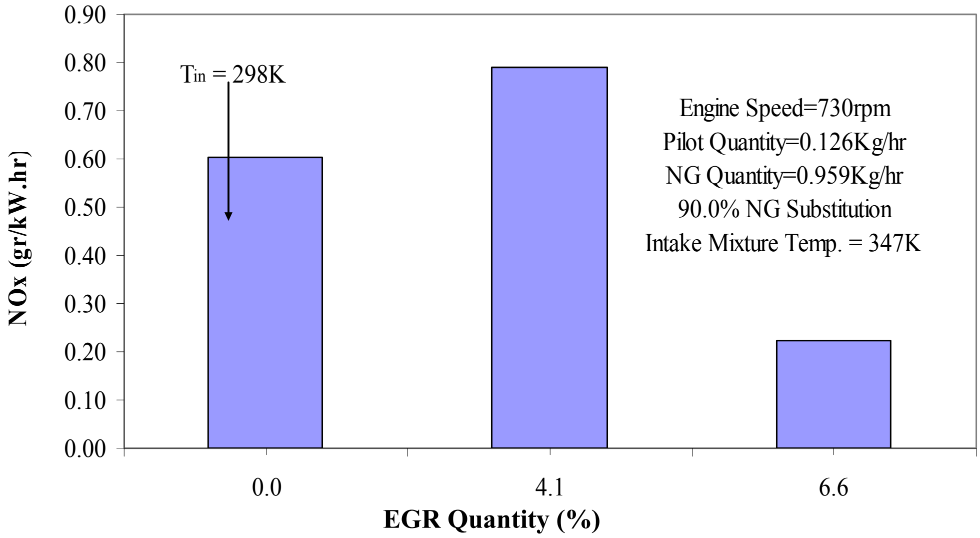


Figure 11. Variations of the nitrogen oxides emission with EGR percentage for constant intake mixture temperatures at 10 and 50 percents of full load.

fuel engines at part load conditions. The obtained results lead to the following important conclusions:

1. By employing hot EGR, the amount of EGR required to overcome the problems of dual fuel engines at part load conditions is very low. It can be observed that, by employing high percentage of EGR at a constant temperature in comparison with low percentage of EGR, combustion process, performance and emission parameters deteriorate.
2. By increasing engine load at part load conditions, the amount of EGR and its temperature to improve the combustion process and consequently the performance and emission parameters are lowered.
3. By employing low percentages of hot EGR in comparison with its high percentages at a constant temperature, combustion starts earlier and its duration is shortens.
4. Low percentage of hot EGR at a critical temperature level reduces unburned hydrocarbons as well as carbon monoxide

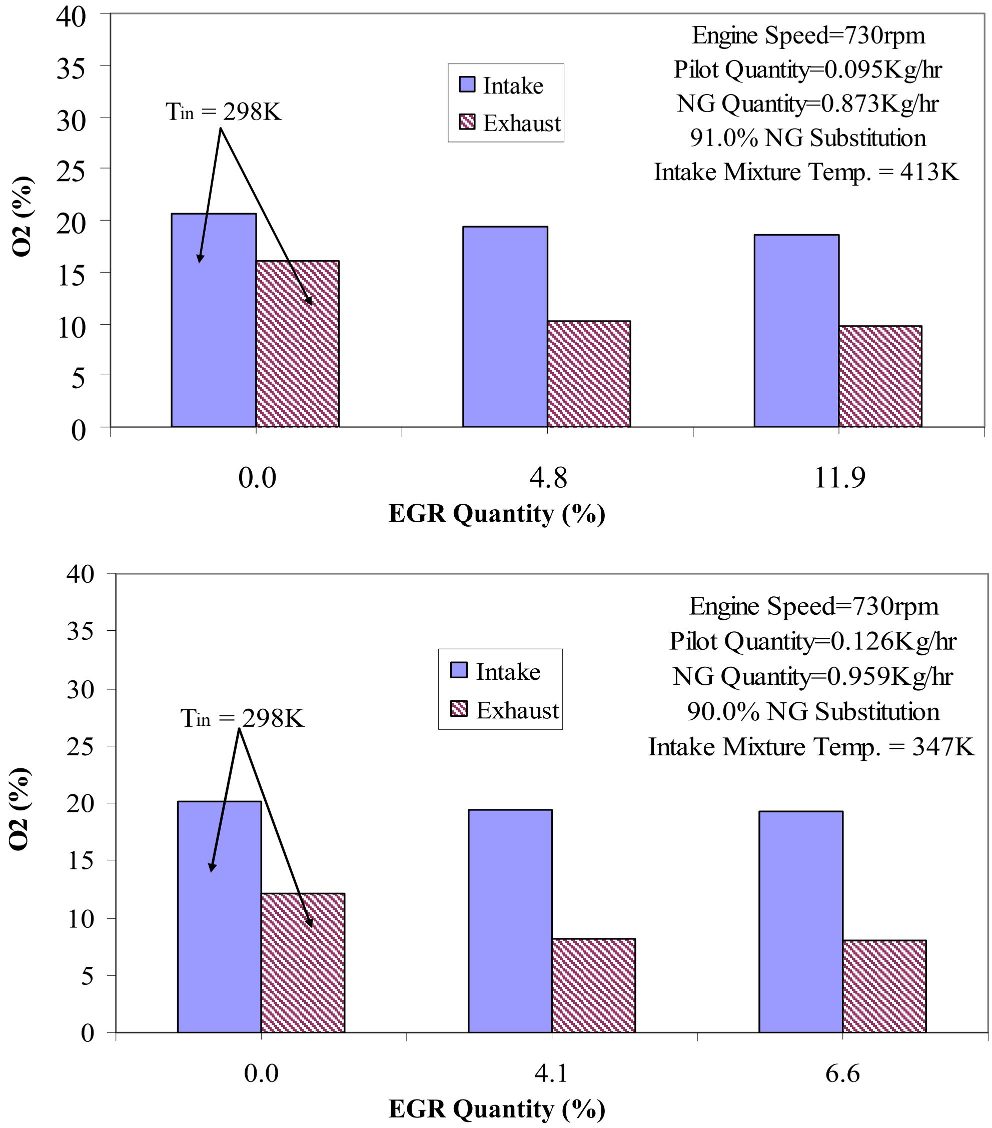


Figure 12. Variations of the intake and exhaust oxygen concentrations with EGR percentage for constant intake mixture temperatures at 10 and 50 percents of full load.

emissions sufficiently but increase in nitrogen oxides due to employing hot EGR is not remarkable.

5. High percentage of hot EGR at a critical temperature level increases unburned hydrocarbons as well as carbon monoxide emissions and reduces nitrogen oxides due to presence of higher levels of diluted gases.
6. By employing low percentage of EGR in comparison with high percentage of EGR at a constant temperature, thermal and radical effects of EGR may overcome its dilution effects and vice versa.

6. ACKNOWLEDGEMENTS

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7. ABBREVIATIONS

UHC	Unburned hydrocarbon
CO	Carbon monoxide
NO_x	Nitrogen oxides
HCCI	Homogeneous charge compression ignition
IDI	Indirect injection
EGR	Exhaust gas recirculation
HRR	Heat release rate
TDC	Top dead center
BDC	Bottom dead center
BSEC	Brake specific energy consumption
D.C.	Direct current
CO_2	Carbon dioxide
$CO_{2,int\ ake}$	CO_2 concentration in the intake mixture
$CO_{2,exhaust}$	CO_2 concentration in the exhaust mixture
ϕ	Total equivalence ratio
ϕ_{gas}	The gas equivalence ratio
$\dot{m}_a$	Mass flow rate of air
$\dot{m}_p$	Mass flow rate of pilot fuel
$\dot{m}_{NG}$	Mass flow rate of natural gas fuel
$\frac{dQ_n}{d\theta}$	The net heat release rate
γ	The specific heat ratio
T	Temperature

8. REFERENCES

1. Flowers, D.L., Aceves, S.M., Smith, R., Torres, J., Girard, J. and Dibble, R., "HCCI in a CFR Engine: Experiments and Detailed Kinetic Modeling", ***SAE Paper***, 2000-01-0328, (2000).
2. Daisho, Y., Takahashi, K., Iwashiro, Y., Nakayama, K.R. and Saito, T., "Controlling Combustion and Exhaust Emissions in a Direct-Injection Diesel Engine Dual Fueled with Natural Gas", ***SAE Paper***, 952436, (1995).
3. Pirouzpanah, V. and Khoshbakhti Saray, R., "Reduction of Emissions in an Automotive Direct Injection Diesel Engine Dual-Fuelled with Natural Gas by using Variable Exhaust Gas Recirculation", ***Proc. Instn, Mech. Engrs, Part D, J. Automobile Engineering***, Vol. 217, (2003), 719-725.
4. Karim, G.A., "Combustion in Gas Fueled Compression: Ignition Engines of the Dual Fuel Type", ***J. Engineering for Gas Turbines and Power***, Vol. 125, (2003), 827-836.
5. Liu, Z. and Karim, G.A., "A Predictive Model for the Combustion Process in Dual Fuel Engines", ***SAE Paper***, 952435, (1995).
6. Daisho, Y., Yaeo, T., Koseki, T., Saito, T. and Kihara R., "Combustion and Exhaust Emission in a Direct Injection Diesel Engine Dual Fueled with Natural Gas", ***SAE Paper***, 950465, (1995).
7. Karim, G.A., Liu, Z. and Jones, W., "Exhaust Emissions From Dual Fuel Engines at Light Load", ***SAE Paper***, 932822, (1993).
8. Liu, Z. and Karim, G.A., "An Examination of the Role of Residual Gases in the Combustion Processes of Motored Engines Fuelled with Gaseous Fuels", ***SAE Paper***, 961081, (1996).
9. Wong, Y.K. and Karim, G.A., "An Analytical Examination of the Effects of Exhaust Gas Recirculation on the Compression Ignition Process of Engines Fuelled with Gaseous Fuels", ***SAE Paper***, 961936, (1996).
10. Abd Alla, G.H., Soliman, H.A., Badr, O.A. and Abd Rabbo, M.F., "Effect of Pilot Fuel Quantity on the Performance of a Dual Fuel Engine", ***Int. J. Energy Conversation and Management***, Vol. 41, (2000), 559-572.
11. Abd Alla, G.H., Soliman, H.A., Badr, O.A. and Abd Rabbo, M.F., "A Computational Investigation of the Effect of Exhaust Gas Recirculation on the Performance of a Dual Fuel Engine", ***SAE Paper***, 2000-01-2040, (2000).
12. Abd Alla, G.H., Soliman, H.A., Badr, O.A. and Abd Rabbo, M.F., "Effect of Diluent Admissions and Intake Air Temperature in Exhaust Gas Recirculation on the Emissions of an Indirect Injection Dual Fuel Engine", ***Int. J. Energy Conversion and Management***, Vol. 42, (2001), 1033-1045.
13. Kusaka, J., Okamoto, T., Daisho, Y., Kihara, R. and Saito, T., "Combustion and Exhaust Gas Emission Characteristics of a Diesel Engine Dual- Fueled with Natural Gas", ***JSAE Review***, Vol. 21, (2000), 489-496.
14. Krishnan, S.R., Biruduganti, M., Mo, Y., Bell, S.R. and Midkiff, K.C., "Performance and Heat Release Analysis of a Pilot-Ignited Natural Gas Engine", ***Int. J. Engine Research***, Vol. 3, No.3, (2001), 171-184.
15. Papagiannakis, R.G. and Hountalas, D.T., "Experimental Investigation Concerning the Effect of Natural Gas Percentage on Performance and Emissions of a DI Dual Fuel Diesel Engine", ***Applied Thermal Engineering***, Vol. 23, (2003), 353-365.
16. Pirouzpanah, V. and Khoshbakhti Saray, R., "Enhancement of Combustion Process in Dual Fuel Engines at Part Loads using Exhaust Gas Recirculation", ***Proc. Instn, Mech. Engrs, Part D, J. Automobile Engineering***, Vol. 221, (2007), 877-888.
17. Pirouzpanah, V., Khoshbakhti Saray, R., Sohrabi, A. and Niaei, A., "Comparison of Thermal and Radical Effects of EGR Gases on Combustion Process in Dual Fuel Engines at Part Loads", ***Int. J. Energy Conversion and Management***, Vol. 48, (2007), 1909-1918.
18. Stone, R., "Introduction to Internal Combustion Engines", 3rd Ed., MacMillan, London, U.K., (1999).
19. Heywood, J.B., "Internal Combustion Engine

Fundamentals", McGraw-Hill, New York, U.S.A., (1988).
20. Egnell, R., "Combustion Diagnostics by Means of Multi Zone Heat Release Analysis and NO Calculation", ***SAE Paper***, 981424, (1998).
21. Brunt, M.F. J., Rai, H. and Emtage, A.L., "The Calculation of Heat Release Energy from Engine Cylinder Pressure Data", ***SAE Paper***, 981052, (1998).
22. Egnell, R., "The Influence of EGR on Heat Release Rate and NO Formation in a DI Diesel Engine", ***SAE Paper***, 2000-01-1807, (2000).
23. Annand, W.J.D., "Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engine", ***Proc. Instn, Mech. Engrs***, Vol. 177, (1963), 973-980.
24. Khoshbakhti Saray, R., "Enhancement of Combustion Process in Dual Fuel Engines at Part Loads by using Suitable Techniques", ***Int. J. Engineering***, Vol. 22, No. 1, (2009), 77-90.