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Experimental and Numerical Investigation of the Effect of Exhaust Gas Recirculation on Performance and Emissions of a Dual Fuel (Diesel–Gas) Engine

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Abstract: In recent years, the major concern about diesel engines is that they pollute the air, which is a special dilemma in those areas having difficulty meeting health–based outdoor air quality standards. In one hand, natural gas resources are abundant and its combusttion results in low levels of pollutant formation and resource abundance on the other hand, conventional compression ignition engine can be easily converted to dual fuel engines in order to use natural gas as main fuel and diesel as pilot injection. In this study, experimental and numerical investigation on performance and emissions of a dual fuel engine was perfomed. The experimental work in this paper was conducted on a new heavy duty locally-designed 12 cylinder diesel engine, called D87, in dual fuel mode of operation. In order to further investigation of combustion and emissions formation processes in dual fuel mode, a CFD simulation for flow field, spray, combustion process and emission formation have been performed. The main goal of the study was to investigate the possibility of decreasing exhaust emissions and fuel consumption. Results indicate that cold EGR has better influence on performance and exhaust emission characteristics of dual fuel engine. Furthermore, investigation on different EGR ratios indicates that applying 5% EGR is the best EGR ratio in terms of performance and emissions characteristics which enhances engine performance and results in a reduction in HC emission.

Key words: Dual fuel engine, Combustion, Emission, CFD, Numerical simulation, EGR.

INTRODUCTION

Nowadays, there are lots of diesel engines in the world which are used in power plants; train sets, marine application and so on, Which have a major drawback in terms of pollution.

To overcome this defficiency, many researches are being done on alternative fuels and combustion enhancement.

Natural gas is one of the most important fuels that is available in large quantity producing lower emission than diesel. To use natural gas in diesel mode, there is a need to have a high compression ratio (about 40). According to engine design limitations, it is difficult to reach this ratio. Using a diesel fuel as pilot is one of the best ways spark plug, which have been done in previous paper (Karim, G.A. and Z. Liu, 1992).

There are many investigations and analyses to develop a dual fuel engine. So, the compression ignition engine of the dual fuel type has been employed in a wide range of applications to utilize various gaseous fuel resources and minimize exhaust gas emissions without excessive increase in cost from that of conventional diesel engines (Karim, G.A. and Z. Liu, 1992). Theoretical study the effects of engine parameters on performance and emissions of a pilot ignited natural gas diesel engine has been investigated by Papagiannakis *et al.* (2009). An existing two–zone phenomenological model has been used to examine the effect of the increase in pilot fuel quantity accompanied with intake charge temperature, on performance and pollutant emissions of a direct-injection dual-fuel diesel–natural gas engine The comparison between normal diesel and normal dual fuel operation reveals that the simultaneous use of diesel and natural gas leads to higher values of the brake specific fuel consumption. Also, Papagiannakis *et al.* (2009) has investigated emission characteristics of high speed, dual fuel operating in a wide range of natural gas/diesel fuel ratios to reduce pollutant emissions from diesel engines. It was shown that the reduction in total relative air–fuel ratio, caused by the increase of diesel fuel supplementary ratio, results in a lower brake thermal efficiency compared to the one under normal diesel operation.

Natural gas sources are spread throughout the world, which dampens the risk of energy crisis. Therefore, nowadays, there is an ever–increasing market demand for both new dual–fuel engine concepts and for the conversion of existing single fuel direct injection diesel engines to satisfy environmental and commercial constraints.

Numerical simulations of a pilot ignited natural gas direct injection in diesel engines has been done by Li *et al.* (1999). Statistical analysis was done to evaluate the expected value and variance of "closeness" between

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diesel sprays and natural gas jets for different injector tip configurations. The results indicated that this affects the heat release rate. Uyehara (1995) discussed how to control dual fuel engine emissions by studying the combustion of both the diesel and dual fuel engines. Carlucci *et al.* (2008) converted a single–cylinder diesel engine into a dual–fuel. The engine was tested on a wide range of operating conditions spanning different values of engine load and speed. During tests, the engine was operated at two different conditions and for each of them, methane and diesel fuel injection pressure, together with pilot fuel amount was varied. Regarding nitrogen oxides (NOx), it was found that the penetration of the jet is as important as the quantity of pilot fuel injected. The more the jet penetrated into the combustion chamber, the more its combustion spread into the same chamber, and then the local temperatures had been closer in value to the bulk temperature. Similar conclusions could be drawn for CO and HC emission levels. Since this kind of engine is newer than Otto and Diesel one it needs much tudy and survey to solve its problem and develop it.

In order to reduce emission levels in internal combustion engine, new technologies have been identified and can be classified into two main groups: in-cylinder emission reduction and exhaust after-treatment. Within the first group, exhaust gas recirculation (EGR) has shown to be extremely efficient in DI diesel engines. Many researchers have investigated the effect of EGR on diesel engine by experiments and simulation, (Pierpont, D.A., 1995; Ladommatos, N., 1998; Sasaki, S., 2000; Baik, D.S.Y.C. Han and S.K. Oh, 2001; Akihama, K., 2001; Khatamnezhad, H., 2011; Mahla, S.K., 2010).

The effects of equivalence ratio of natural gas and EGR rate on ignition and burning rate of natural gas as well as the knock limit were investigated experimentally by Ishida *et al.* (Liu, A.B. and R.D. Reitz, 1993). But, the differences in combustion between this concept and conventional dual fuel combustion must be investigated to determine their effects on combustion characteristics as well as emissions. This is possible by CFD simulation of the flow field in combustion chamber.

The main target of this contribution is to improve a new heavy duty locally-designed 12 cylinder diesel engine, called D87, in dual fuel mode of operation (Diesel and gas) by the use of EGR concept via CFD simulation. The experimental work done of this study was conducted on diesel mode and dual fuel mode of operation (20% diesel-80% gas) at constant speed (1500 rpm) and full load condition. The experimental results have also reported and compared with the simulated data. were used one and three-Dimensional simulation in parallel way in order to analyze the performance and combustion process of a dual fuel engine. The exhaust gas recirculation has been entered combustion chamber and the results that contain performance and emission are expressed. In order to investigate the effect of EGR on combustion process, at different ratios of EGR from 2% to 8% at hot and cold condition were used.

Experimental Setup:

The engine used in the present research was a 12 cylinder, four-stroke, direct-injection diesel engine with the specifications given in Table 1. The base engine is a DI diesel engine. In order to operate at dual fuel mode, compression ratio was decreased from 15:1 to 11.5:1 and gaseous fuel port was added to the base engine.

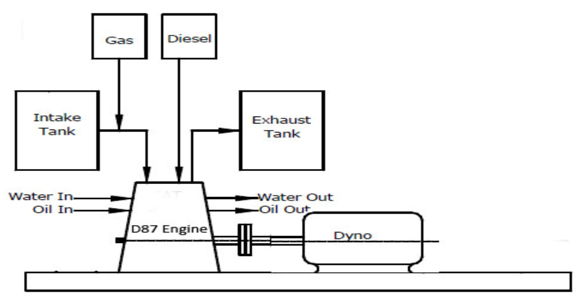


Fig. 1: Schematic of the engine setup in test cell.

Table 1: D87 Engine.

Engine type	Diesel Mode	Dual Fuel Mode
Engine speed	1500 rpm	1500 rpm
Bore × stroke	150×180 mm	150×180 mm
Power	1000 kW	850 kW
Compression ratio	15:1	11.5:1
Injector type	Common-rail	Common-rail
Number of nozzle holes	8	8

Fig. 1 shows a schematic of the engine setup in test cell. The engine was coupled to a water brake dynamometer and the experiments were carried out at 1500 rpm while maintaining a power of full load condition. Crank angle resolved in-cylinder pressure was measured using a piezoelectric pressure transducer.

The primary engine fuel, natural gas, was a mixture of 90.3 percent methane, 4.3 percent ethane, and less than 6 percent nitrogen and carbon dioxide. natural gas was injected into the air manifold. Fig. 2 shows D87 dual fuel engine in test cell.

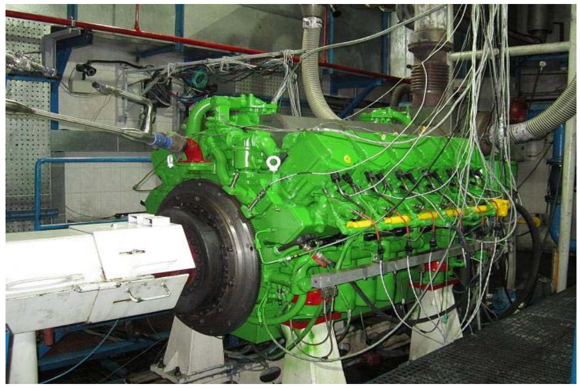


Fig. 2: D87 dual fuel engine in test cell.

Numerical Model:

Diesel engine combustion modeling is a complicated task due to the fact that many competing processes are occurring simultaneously. These include fuel injection, spray break-up, evaporation, auto-ignition, combustion, turbulence generation and dissipation, mixing and heat transfer.

CFD simulation solves the compressible, turbulent, three dimensional transient conservation equations for reacting multi-component gas mixtures with the flow dynamics of an evaporating liquid spray. The turbulent flows within the combustion chamber are simulated using the k-ε turbulence model, modified for variable-density engine flows.

Spray Model:

The spray module is based on a statistical method referred to as the discrete droplet method. This operates by solving ordinary differential equations of the trajectory, momentum, heat and mass transfer of single droplets, each being a member of a group of identical non–interacting droplets termed a parcel. Thus the behavior of one member of the group represents the behavior of the complete parcel.

The wave model was selected to represent spray breakup (Uyehara., O.A., 1995). In this model the growth of an initial perturbation on a liquid surface is linked to its wavelength and to other physical and dynamic parameters of the injected fuel and the domain fluid.

$$\frac{dr}{dt} = \frac{(r - r_{stable})}{\tau_a} \tag{1}$$

Where τ_a is the break-up time of the model, which can be calculated as:

$$\tau_a = \frac{3.726.C_2.r}{\Lambda.\Omega} \tag{2}$$

The constant C_2 corrects the characteristic breakup time and varies from one injector to another. r_{stable} is the radius of the product droplet, which is proportional to the wavelength Λ of the fastest growing wave on the liquid surface:

$$r_{stable} = C_1.\Lambda \tag{3}$$

The wavelength Λ and wave growth rate Ω depend on the local flow properties.

The Dukowicz model (Dukowicz, J.K., 1979) was applied for treating the heat-up and evaporation of the droplets. This model assumes a uniform droplet temperature. The rate of droplet temperature change is determined by the heat balance, which states that the heat convected from the gas to the droplet either heats up the droplet or supplies heat for vaporization.

Combustion Model:

The extended coherent flame model (CFM) has been used in order to describe combustion in diesel and gasoline engines. The CFM is applicable to both premixed and non-premixed conditions on the basis of a laminar flamelet concept, whose velocity S_L and thickness δ_L are mean values, integrated along the flame front, only dependent on the pressure, the temperature and the richness in fresh gases. Such a model is attractive since a decoupled treatment of chemistry and turbulence is considered. All flamelet models assume that reaction takes place within relatively thin layers that separate the fresh unburned gas from the fully burnt gas. Using this assumption, the mean turbulent reaction rate is computed as the product of the flame surface density Σ and the laminar burning velocity S_L via:

$$\rho.\dot{r}_{fu} = -\omega_l.\Sigma \tag{4}$$

ω_l is the mean laminar fuel consumption rate per unit surface along the flame front. For lean combustion:

$$\omega_l = \rho_{fu,fr}.S_l \tag{5}$$

$$\rho_{fu,fr} = \rho_{fr}.y_{fu,fr} \tag{6}$$

In this equation $\rho_{fu,fr}$ is the partial fuel density of the fresh gas, ρ_{fr} the density of the fresh gas and $y_{fu,fr}$ is the fuel mass fraction in the fresh gas. The diesel-ignited gas engine model combines homogeneous premixed gas combustion with Diesel ignition. The auto-ignition model is chosen automatically. Auto-ignition is calculated in regions which are richer than the homogeneous mixture.

Emission Models:

The extended Zeldovich mechanism (Zeldovich, Y.B., 1947) has been implemented to describe nitric oxide (NOx) formation. The classical extended Zeldovich scheme as follows by:

$$N_2 + O \underset{K_2}{\overset{K_1}{\leftrightarrow}} NO + N \tag{7}$$

$$N + O_2 \underset{K_4}{\overset{K_3}{\leftrightarrow}} NO + O \tag{8}$$

$$N + OH \underset{K_6}{\overset{K_5}{\leftrightarrow}} NO + H \tag{9}$$

NOx formation has been found to be very sensitive to small changes in the computed in-cylinder gas temperature field. The relation between NO production and in-cylinder *temperature is* as follow:

$$\frac{d[NO]}{dt} = \frac{6\times 10^{16}}{T^{1/2}}\exp\left(\frac{-69090}{T}\right)[O_2]^{\frac{1}{2}}[N_2] \tag{10}$$

Model Description:

The commercial CFD softwares AVL-FIRE and GT power were used to perform the numerical simulation of engine performance, combustion and emission formation in D87 engine.

Combustion characteristics and its influence on flow field and emission formation was investigated by three dimensional CFD model in Fire code. A three dimensional combustion chamber model has been shown in Fig. 4. As there are 8 holes in the used nozzle, the constructed model contains one hole and 45 degree. This takes advantage of the symmetry of the chamber geometric setup, which significantly reduces computational runtime. At TDC, the whole mesh number is 19385 cells. This mesh size will be able to provide good spatial resolution for the distribution of most variables in the combustion chamber. Calculations are carried out on the closed system from IVC at -150°CA ATDC to EVO at 120°CA ATDC.

The generated model in Fire code is a closed cycle in compression and expansion stroke of engine. Therefore, EGR and turbocharging effects cannot be simulated on the initial condition of engine such as boost pressure and initial temperature. Also, engine performance parameters such as IMEP, BSFC and etc. cannot be achieved in closed cycle simulation.

GT power software has a one dimensional CFD model that can simulate the four stroke of engine operation. Therefore, GT Power software has been used in this paper to determine the initial condition in different cases as well as engine performance. In order to simulate the in-cylinder combustion process in one dimensional CFD model, the measured heat release rate in three dimensional simulation has been imported as a burn rate profile. So, the heat release rate was imported to GT and the constructed model according to Fig. 3 was utilized to solve it. Results of one dimensional including inlet pressure, inlet temperature and equivalence ratio were imported to the three dimensional model and re-run with new condition. This procedure had continued to reach a small residual between two simulations. Therefore, combustion process, emission formation and engine performance was investigated in different conditions in parallel simulation of one and three dimensional CFD simulation.

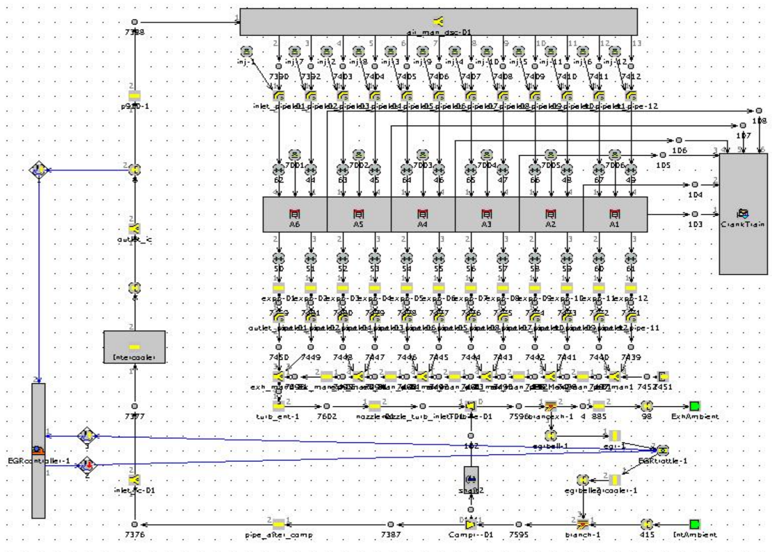


Fig. 3: Computational model in GT Power software.

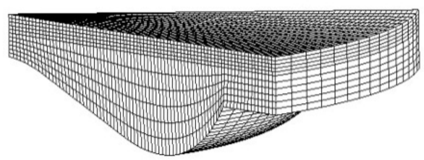


Fig. 4: Computational mesh at 20 degree CA BTDC.

RESULTS AND DISCUSSION

In this research, parametric study has been conducted to investigate the effect of exhaust gas recirculation on combustion and emission in D87 dual fuel in parallel simulation with Fire and GT power code.

Before using the CFD model to examine the effect of EGR on combustion process and emissions, it is necessary to validate its predictive ability. In order to do this, experimental data of the above-mentioned target engine were used with FIRE and GT Power.

Fig. 5 indicates the comparison of simulated and experimental in-cylinder pressures against the crank angle for the D87 diesel engine with SOI at 12 degree CA BTDC At full load. The good agreement between predicted in-cylinder pressure and the experimental data can be observed. It is due to time step and computational grid independency of the results. Also, the simulated in-cylinder pressure compared to experimental value is illustrated in Fig. 6 for dual fuel operation mode (20% diesel-80% gas) at full load. There is a good agreement between the results. Therefore, the CFD and combustion simulation model performed in this study are able to represent the real combustion process inside engine and have the capability to be implemented for further calculation.

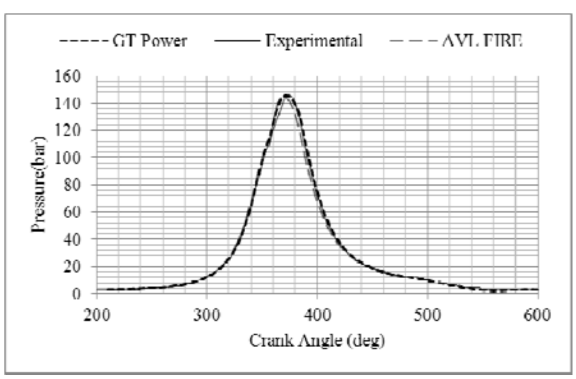


Fig. 5: Comparison between experimental with 1D and 3D simulation data in diesel engine.

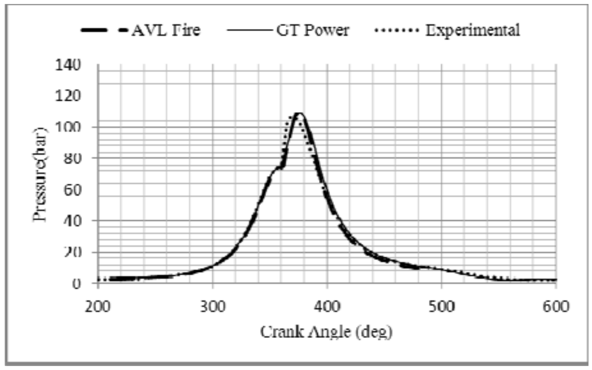


Fig. 6: Comparison between experimental with 1D and 3D simulation data in dual fuel engine.

In order to investigate the effect of EGR cooling as well as different EGR rates, cold and hot EGR were simulated. For cold EGR simulation, exhaust gas is recirculated after the turbocharger and driven into the intake port before compressor by a pipe. However, in hot EGR, a pipe imports some of the exhaust gases from the exhaust manifold into the intake manifold where it is mixed with the fresh air. The pipe contains a valve to regulate the amount of EGR. To quantify the amount of EGR, the EGR percentage is defined by:

$$EGR\% = \frac{m_{EGR}}{m_{air} + m_{fuel} + m_{EGR}} \tag{11}$$

According to that, 2%, 5% and 8% of EGR has been simulated in hot and cold condition. closed cycle Results of 3D CFD simulation is shown in this section. It can be seen from Fig. 7-10 that the in-cylinder pressure and temperature increase with increase of EGR rate in both conditions. peak in-cylinder pressure and temperature in hot EGR condition is more than those cold EGR. Also, results show that unlike the cold EGR, hot EGR increases the exhaust temperature. In order to investigate this occurrence, it needs to analyse combustion process in cylinder using heat release rate .

Rate of heat release traces are compared for different cases in Fig. 11 and Fig. 12. The first peak is for diesel injection and after a short time; premixed combustion for gas fuel is started. The second peak is for natural gas and as it can be seen, it is much more than diesel heat release rate.

Results show that EGR can promote the combustion process due to increasing the total equivalence ratio. Because, EGR substitutes part of fresh air and causes a reduction in air fuel ratio. Also, more free radicals in EGR tends to decrease ignition delay and increase combustion rate as well as increase the flame speed on the burning of gas fuel combustion.

Since hot EGR has high free radicals, according to heat release rate profiles the ignition delay period was decreased by EGR rate increase. But, ignition delay is approximately equal in cold EGR due to constant temperature. In hot EGR condition, due to higher temperature than cold EGR in one hand, and more free radicals at the other hand, heat release rate has been increased. Therefore, in-cylinder pressure and temperature will be higher in hot EGR in comparison with cold EGR at the same ratio.

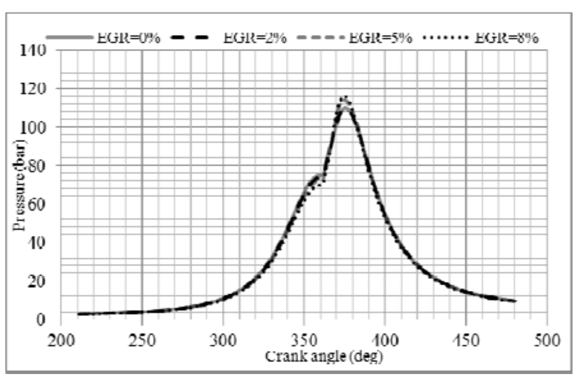


Fig. 7: Pressure alteration with different cold EGR.

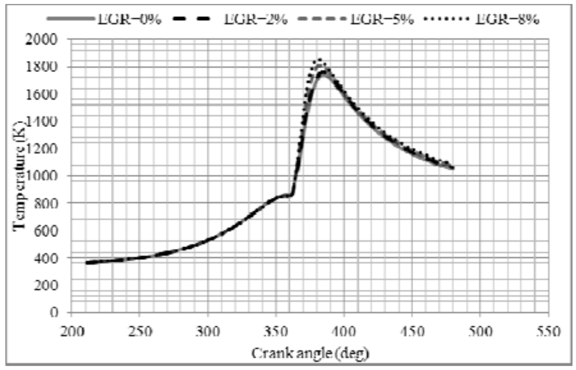


Fig. 8: Temperature alteration with different cold EGR.

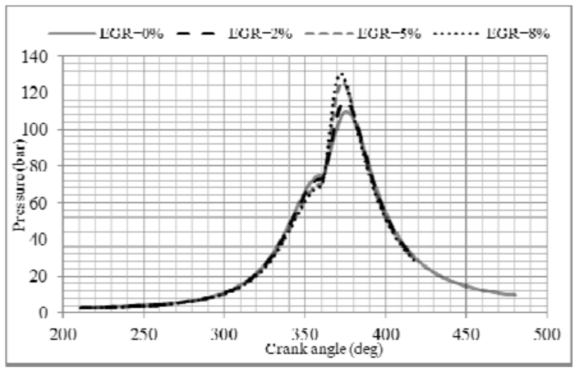


Fig. 9: Pressure alteration with different hot EGR.

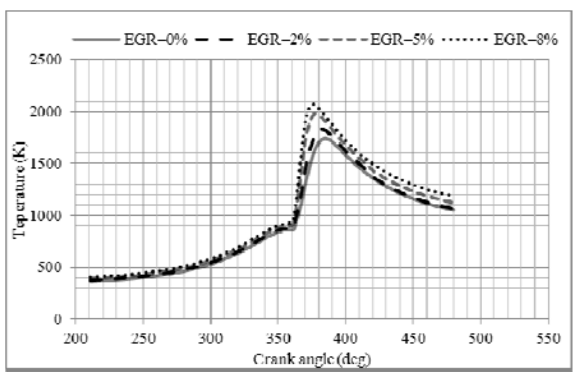


Fig. 10: Temperature alteration with different hot EGR.

Fig. 13 to Fig.14 show the effects of EGR on HC and NO exhaust emission at EVO in dual fuel engine. As can be seen, HC emission is decreased in higher EGR rates. It is due to improvement of combustion efficiency and in-cylinder temperature increase. It can obviously be seen that the HC in the hot EGR is lower than cold EGR due to higher heat release rate which results in higher in-cylinder temperature.

Fig. 14 indicates the variations of the mass fraction of NO with different cases of EGR for a fixed pilot fuel quantity of the dual fuel engine. NO is formed inside the combustion chamber in post–flame combustion process in the high temperature region. It can be seen that with increasing percentages of EGR, the NO emission is increased. Hot EGR can raise the charge temperature more than cold EGR and thereby, it can influence the combustion process and, hence, high level of NO emission. With EGR increasing, proportional with temperature, NO mass fraction increases with 2% EGR. For 5% of EGR, rate of increase will be quicker due to a higher temperature and when the EGR mass fraction increases to 8%, NO will increase with a lower rate. The main reason for increasing NO emission with lower percentage of EGR is originated from the dilution and thermal effects of EGR. In higher percentage of EGR, the effect of air can be seen apparent and with a lower level of available N and O_2, we will have a reduction in NO increment.

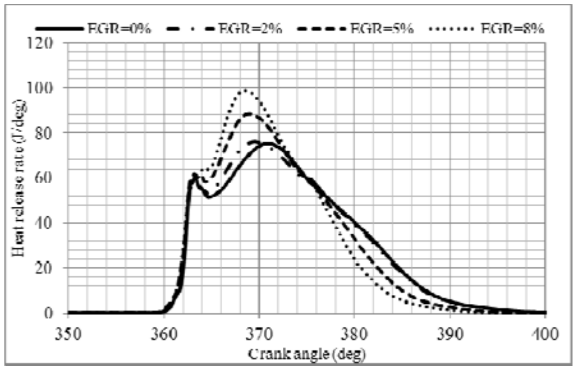


Fig. 11: Heat release rate in different cold EGR.

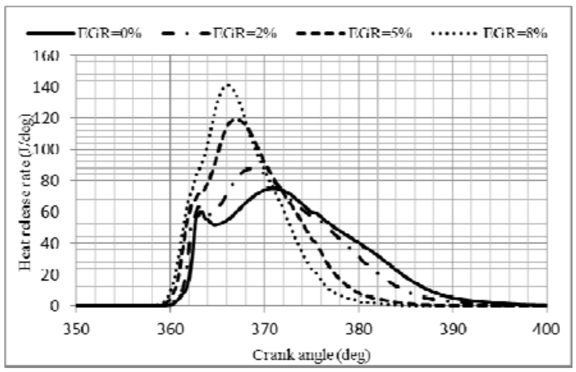


Fig. 12: Heat release rate in different hot EGR.

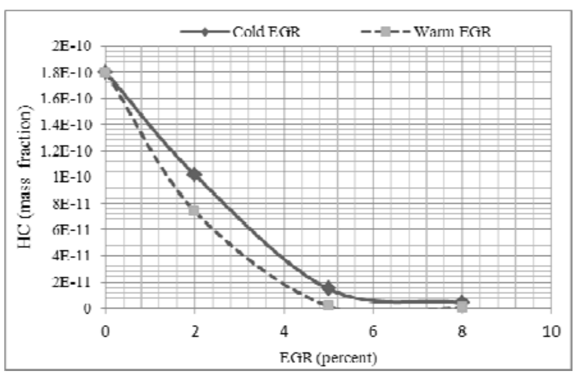


Fig. 13: HC mass fraction in different EGR percentage.

2D contours for equivalence ratio and in-cylinder temperature are shown in different EGR with different crank angle in Fig. 15 to Fig. 18. There is a homogenous air fuel ratio in the base mode. When EGR percentage is increased, especially for high level of EGR, there is a non-homogeneity in the combustion chamber. Therefore the maximum temperature will be increased in higher EGR levels. Results indicate that in-cylinder temperature for 8% EGR are obviously more than other cases in 370 and 380 crank angle. Also, hot EGR has higher temperature region in comparison to cold EGR.

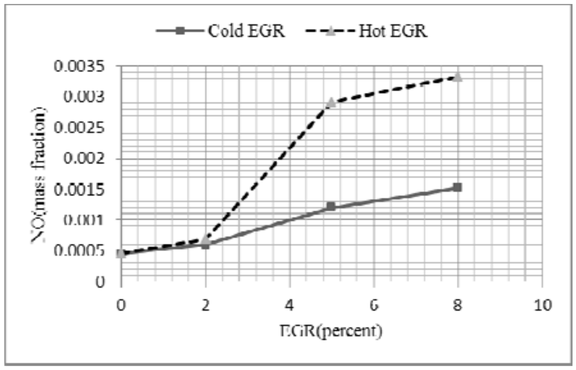
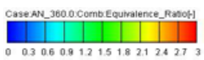


Fig. 14: NO mass fraction in different EGR percentage.



Crank angle	0%	2%	5%	8%
360				
370				
380				

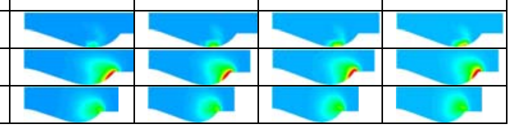
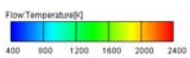


Fig. 15: 2D contour for equivalence ratio in different cold EGR.



Crank angle	0%	2%	5%	8%
360				
370				
380				

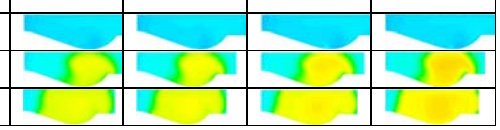


Fig. 16: 2D contour for equivalence ratio in different cold EGR.

Crank angle	0%	2%	5%	8%
360				
370				
380				

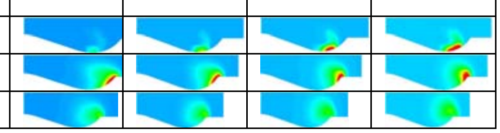


Fig. 17: 2D contour for equivalence ratio in different cold EGR.

Crank angle	0%	2%	5%	8%
360				
370				
380				

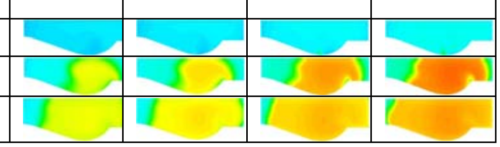


Fig. 18: 2D contour for equivalence ratio in different cold EGR.

Table 2 and Table 3 show engine performance characteristic in different EGR percentages that has obtained by one dimensional simulation.

Where $\dot{W}$ is engine power, BSFC represents brake specific fuel consumption which is according to equation (12):

$$BSFC = \frac{\dot{m}_{fuel}}{\dot{W}} \tag{12}$$

Peak pressure for 0, 2, 5 and 8 percentage of EGR occurs in 17.8, 17.47, 16.8 and 16.5 crank angle after top dead center. However the peak pressure in hot EGR occurs in 18.33, 14, 13.2 and 12 respectively. Although higher pressure make higher output power and EGR makes the maximum pressure higher than base engine but with exhaust gas recirculation to the cylinder and especially for hot EGR, the peak point will approach to TDC. This makes higher pressure occur in a less volume thus providine less output power than was expected. Therefore, brake specific fuel consumption decreased to 5% EGR rate.

Table 2: Engine performance characteristic with different cold EGR percentage.

EGR percentage	0	2	5	8
Power(kw)	850	848	854	853
BSFC	191.12	191	190.24	191.5

Table 3: Engine performance characteristic with different hot EGR percentage.

EGR percentage	0	2	5	8
Power(kw)	850	852	852	840
BSFC	191.12	190.4	190	196

Conclusion:

Environmental affairs, economy and using alternative fuels are today's worldwide important issues, both in academia and industries. Dual fuel (gas and diesel) engine has a good position for power plant and transportation systems. In this research, different EGR mass fraction has been investigated in a dual fuel engine to improve the engine performance and decrease of exhaust emission.

The results show that dual fuel engine has lower power in comparison with diesel engine. To increase the performance of duel fuel engine, 2%, 5% and 8% EGR was utilized. In this study we use these values by the concept of cold and hot EGR. In addition, the result demonstrates that in both cold and hot EGR, by increasing the EGR from 0 to 5%, the peak cylinder pressure and temperature will increase. This happened because of increasing heat release rate in combustion chamber. So, the performance characteristics for 5% has the best output.

Moreover, HC emission is decreased by adding EGR to 5%, but because of increasing the in-cylinder temperature, the rate of thermal NO formation will increase and therefore the value of NO emission at EVO will rise.

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