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AN INVESTIGATION ON USING EGR IN NATURAL GAS SI ENGINES

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ABSTRACT
In this study, a computer model of the four-stroke, spark-ignition natural gas engine thermodynamic cycle was developed. This model was constructed based on the mass and energy conservation principles and the combustion process was analyzed using a two-zone combustion model. The combustion angle was calculated by using relationships derived from a turbulent model. In addition, a kinetic model based on the extended Zeldovich mechanism was developed in order to show the ability of Exhaust Gas Recirculation (EGR) on reducing NO emissions. The aim of this study is to investigate the effect of adding EGR to a stoichiometric mixture on engine performance and NO emissions. It was found that using EGR reduced NO emission by about 80% and improved fuel consumption compared to a stoichiometric non EGR mixture condition.

Keywords: Combustion, Engine, Natural Gas, EGR.

1. INTRODUCTION
Natural gas is one of the cleanest economically available fuels for engines. Studies around the world have shown that engines running on natural gas emit significantly lower emissions compared to engines running on conventional fuels such as gasoline or diesel [1, 2]. In addition, natural gas is available in many parts of the world that have poor oil reserves. Using natural gas as an alternative clean fuel will decrease the dependence on imported oil in such countries. In addition, the world sources of natural gas are far larger than the petroleum oil, thus the research in utilizing natural gas in engines represents an investment for the future.

One of the natural gas engine combustion technologies, which begun in the early 1980's, is the "lean burn" combustion technique. This technology became dominant in gas engine industry as it led to high engine efficiency accompanied with longer durability and lower cost. Today after almost a quarter century of continuous lean burn engine development and investment, most of the conventional gas engines operate with lean burn mode. According to the Engine Manufacturers Association, USA 2004, over 80% of all heavy duty stationary natural gas engines sold in the USA employs lean burn combustion technology [3]. Most of the research conducted in the lean-burn strategy basically focused on extending the maximum burning lean limit in order to reduce NOx emissions to satisfy the increased emission restrictions. That usually was achieved by designing fast-burning combustion chambers and/or employing the stratified charge concept, usually by using either a combustion pre-chamber or direct fuel injection.

Table 1: Emission standards, g/kWh [4]

Year	Standard	CO	HC	NOx	PM
1996	Euro2	4	1.1	7	0.15
2000	Euro3	2.1	0.66	5	0.1
2005	Euro4	1.5	0.46	3.5	0.02
2008	Euro5	1.5	0.46	2	0.02

Recently, laser ignition systems have been developed in order to ignite extremely lean fuel air mixtures, which require high ignition energy.

Currently, increasingly stringent ambient air quality standards demand engine emissions to be extremely low; see Table 1 [4]. In order for the engine under the lean burn mode to produce lower NOx emissions, it has to operate with a leaner mixture. In other words, the engine has to operate near the misfire limit to produce relatively lower NOx emissions. As the engine operates near the misfire limit, the engine stability deteriorates, the hydrocarbon (HC) and CO emissions increase, and the engine efficiency decreases. Another way to control NOx emissions is to retard the spark timing, which also leads to a decrease in engine efficiency and an increase in HC emissions. Therefore, it seems that any efforts towards a future decrease in NOx emissions would lead to an increase in HC emission and a decrease in engine thermal efficiency. At the end, a compromise must be made between the increase in NOx emissions and the decrease in engine efficiency. It has become obvious that it would be difficult for the conventional gas engine operating on lean burn mode to meet the stringent future emission standards, such as 2010 emission standards, especially for NOx emissions without using exhaust gas after-treatment.

The current emission reduction technologies used for the NOx emission after-treatment in lean burn engines such as the selective catalytic reduction (SCR) devices are expensive and add some complexity to the engine use. For example, the SCR technique consists of ammonia storage, feed, injection system and a catalyst. In this system, the ammonia is injected in the exhaust gases upstream of the catalyst. In order for this system to operate properly, a certain exhaust gas temperature range must be maintained [5]. In addition, an oxidation catalyst would also be necessary to reduce both the HC and CO emissions.

It could be concluded that in order for the engines to meet the future emission standards, some alternative techniques must be investigated and developed. One of these alternative techniques is the use of a three way catalyst (TWC) to reduce NOx, HC, and CO emissions. The three way catalyst technology was developed in the 1970s for the automobile industry to reduce the gasoline engine emissions. The TWC is capable of reducing the three emissions at the same time and it is much less expensive than the SCR devices used in lean burn engines. However, in order for the TWC to operate efficiently, the engine must operate at near stoichiometric fuel-air ratio (i.e. without excess air). When the engine operates near the stoichiometric mixture, the in-cylinder temperature increases, and consequently, the thermal stresses and the knocking tendency increase. This would lead to some restrictions on the use of turbocharging, high compression ratio, and optimum spark advance timing. As a result, the engine would operate less efficiently than a similar lean burn engine.

In order to reduce the in-cylinder temperature, an inlet charge dilution must be employed. One of the methods used to dilute the inlet charge is to recycle some of the exhaust gases back into the cylinder intake with the inlet mixture. This method is called Exhaust Gas Recirculation (EGR). Using EGR with the stoichiometric inlet mixture will lead to a decrease in the in-cylinder temperature and a decrease in knocking tendency and could permit the engine to use turbocharging, relatively higher compression ratio, and optimum spark advance timing to achieve a relatively higher thermal efficiency compared to non diluted stoichiometric mixture operation. In addition, adding EGR to the inlet mixture will reduce the oxygen partial pressure in the inlet mixture, and consequently the in-cylinder NOx production will decrease. Furthermore, as EGR will be added to a stoichiometric mixture, the use of a TWC for necessary emission reductions is also possible.

Although the concept of using EGR in engines, especially for petrol and diesel engines, is not new, it is believed that natural gas SI engine operation employing stoichiometric mixture with EGR has not been fully optimized yet [3]. Further research is still needed in the current time and in the future to investigate and to assess this concept before being used as a certified option and an alternative to the lean burn combustion strategy to achieve extremely low emissions accompanied with high engine efficiency and good combustion stability.

For this purpose, a computer model of the four-stroke spark-ignition natural gas engine thermodynamic cycle was developed in order to study the effect of using EGR on natural gas SI engine performance parameters such as power and fuel consumption in addition to NO emissions. This will help in providing guidance for future experimental work. The model has been validated by experimental results and a good agreement between the results was found.

2. MODEL CONSTRUCTION

The following assumptions and approximations have been considered for simplification:

1. The contents of the cylinder are fully mixed and spatially homogeneous in terms of composition and properties during intake, compression, expansion, and exhaust processes. Thus, the thermodynamic properties vary only with time (or crank angle).
2. For the combustion process, two zones (each is spatially homogeneous) are used. The two zones are the unburned and the burned zones. The two zones are separated from each other by the flame front (see Figure 1).
3. The intake and exhaust manifolds are assumed to be infinite plenums containing gases at constant temperature and pressure.
4. The flow rates in both the intake and exhaust processes are determined from quasi-steady one-dimensional flow rate equations.
5. All gases are considered to be ideal gases during the engine thermodynamic cycle.
6. All crevice effects are ignored, and the blow-by is assumed to be zero.
7. The cylinder wall temperature is assumed to be constant (400 K) and the heat transfer is determined using Woschni correlation [6].
8. The engine is in steady state such that the thermodynamic state at the beginning of each thermodynamic cycle (two crankshaft revolutions) is the same as the end state of the cycle.

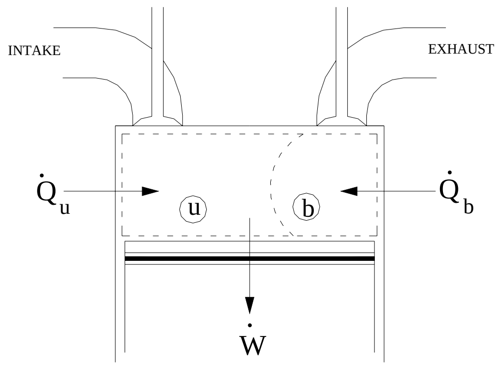

Fig 1: Schematic of the two-zone combustion modeling

In the present model, the thermodynamic cycle simulation starts with assumed guesses of the values of pressure and temperature of the contents within the cylinder at the instant the intake valve opens. After two crankshaft revolutions (720 crank angle degrees), the calculated values of pressure and temperature are compared to the initial guesses. If the calculated values are not within an acceptable tolerance to the initial guesses, the simulation is repeated using the final calculated values as initial guesses.

The Combustion Process:

The following assumptions have been assumed during combustion:

- The flame front thickness is assumed to be negligible.
- The cylinder pressure is assumed to be the same in the burned and unburned zones.
- Only the convective heat transfer mode, between the cylinder contents and the cylinder wall, is considered.
- The heat transfer between the two zones is neglected.
- For the burned zone, ten species (CO_2, H_2O, CO, N_2, O_2, OH, NO, H, O, and H_2) are considered in chemical equilibrium during combustion and expansion.
- The combustion chamber wall area in contact with the burned gases is assumed to be proportional to the square root of the burned mass fraction to account for the greater volume filled by burned gases against the unburned volume as suggested by Ferguson [7].

The Thermodynamic Formulations

Figure 1 is a schematic of the engine cylinder during combustion, which shows the cylinder heat transfer from both the unburned (u) and burned (b) zones, and the piston work. The basic relations used in the development of the present cycle simulation are the first law of thermodynamics, the conservation of mass law, and the ideal gas law.

These three principles have been applied to both the unburned and burned control volumes in order to derive expressions for the time (or crank angle) derivative of the unburned and burned gas temperatures and volumes in addition to the cylinder pressure during combustion. These expressions are expressed in terms of engine design parameters and operating conditions. The Euler numerical solution technique as described by Caton [8] was used to solve the differential equations to determine the in-cylinder pressure and temperature.

The Burning Rate

The S-shaped mass fraction burned profile, the Wiebe function, was used to determine the burning rate:

$$X_b = 1 - exp\left[-a\left(\frac{\theta-\theta_o}{\Delta\theta}\right)^{m+1}\right] \tag{1}$$

Where θ is the crank angle, θ_o is the crank angle at the start of combustion, $\Delta\theta$ is the total combustion duration (from $X_b = 0$ to $X_b = 1$), and a and m are adjustable parameters which fix the shape of the curve. Actual mass fraction burned curves have been fitted with $a \approx 5$, and $m \approx 2$ as suggested by Heywood [6].

The Combustion angle

A turbulent flame propagation model developed by Tabaczynski and coworkers [9] was used by Hires and coworkers [10] to obtain explicit relations for the flame development angle, $\Delta\theta_d$, and the rapid burning angle, $\Delta\theta_b$, as function of engine design and operating variables:

$$\Delta\theta_d = C\left(\bar{S}_p v\right)^{1/3}\left(\frac{L}{S_l}\right)^{2/3} \tag{2}$$

$$\Delta\theta_b = C'\left(\frac{B}{L^*}\right)\left(\frac{\rho_i}{\rho_u^*}\right)^{10/9}\left(\bar{S}_p v^*\right)^{1/3}\left(\frac{L_i}{S_l^*}\right)^{2/3} \tag{3}$$

Where v is the kinematic viscosity, L is the distance between cylinder head and piston, S_l is the laminar burning velocity, $\bar{S}_p$ is the mean piston speed, ρ is the density, and B is the cylinder bore. The subscript i denotes the value at ignition and t he subscript u refers to the unburned mixture, whereas the superscript (*) denotes the value at cylinder conditions where $X_b = 0.5$. C and C' are constants, which depend on engine geometry.

Both equations (2) and (3) have been used in the present model in order to calculate the combustion duration ($\Delta\theta = \Delta\theta_d + \Delta\theta_b$) at different operating conditions. The combustion duration is then used to determine the burned mass fraction using the Wiebe function.

NO Formation Kinetic Model

The extended Zeldovich mechanism [6] was used to determine the rate of change of NO mole fraction during combustion and expansion processes.

3. RESULTS AND DISCUSSION

The prescribed thermodynamic model was used to predict the performance of a 507 cc single cylinder Ricardo E6 engine. Table 2 shows the Ricardo engine specifications.

Table 2: Engine specifications

Number of cylinders	1
Bore, mm	76.2
Stroke, mm	111.125
Capacity, cc	507
Compression ratio	8

The predicted torque speed engine characteristic for the wide-open throttle (WOT), stoichiometric fuel-air mixture, and maximum brake torque (MBT) spark-timing engine operating conditions was compared to experimental results conducted by Mustafi and coworkers [11] for Ricardo E6 natural gas engine. Figure 2 shows that there is a very good agreement between the predicted and experimental results, which indicates that the model has been well constructed.

All of the following theoretical performance investigations on adding recycled exhaust gases to the inlet fresh stoichiometric fuel-air mixture were predicted by the model for the wide open throttle (WOT), and maximum brake torque (MBT) spark timing operating conditions. The percentage of exhaust gas recirculation (EGR) was calculated as a percentage from the total mass inlet mixture.

Figure 3 shows the effect of adding the exhaust gases to the inlet fresh mixture on engine brake power at 1000 rpm and three different inlet conditions. Firstly, when the exhaust gas with a temperature comparable to the average exhaust gas temperature (1000 K) was added to the fresh fuel-air mixture, the brake power dropped from 3.3 kW to about 1.6 kW at 20% EGR. This expected decrease in engine power was due to the decrease in volumetric efficiency with the admission of EGR in the inlet mixture as the exhaust gases replaced a part of the fuel-air mixture at a constant inlet pressure. Secondly, when the total inlet mixture was cooled to an inlet temperature of 333 K, the decrease of the power was less severe due to the improvement of the volumetric efficiency. Finally, the simulated supercharged cooled inlet mixture condition was studied. Thus, the inlet pressure was increased with the increase of EGR in order to keep the engine power constant corresponding to the no EGR admission condition. To achieve this, the inlet pressure was increased from 98 kPa at no EGR condition to 122 kPa at 20% EGR admission condition.

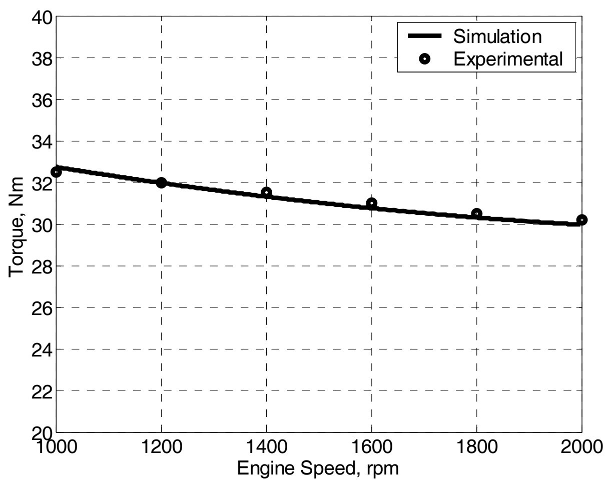

Fig 2: A comparison between modeling and experimental [11] results for Ricardo natural gas engine at WOT, stoichiometric fuel-air mixture, and MBT spark timing operation.

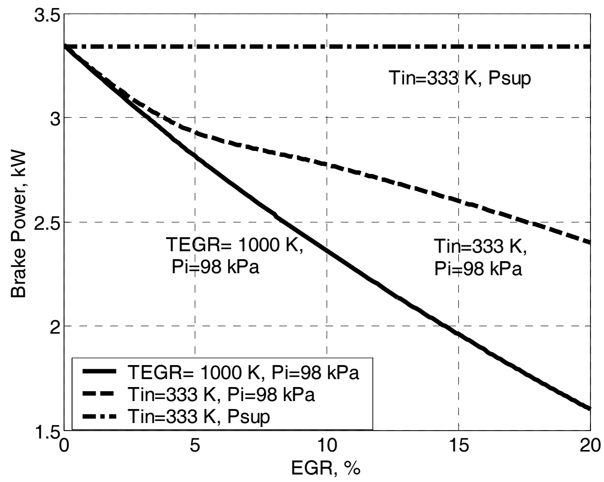

Fig 3: The variation of brake power with EGR admission at 1000 rpm and different inlet mixture conditions.

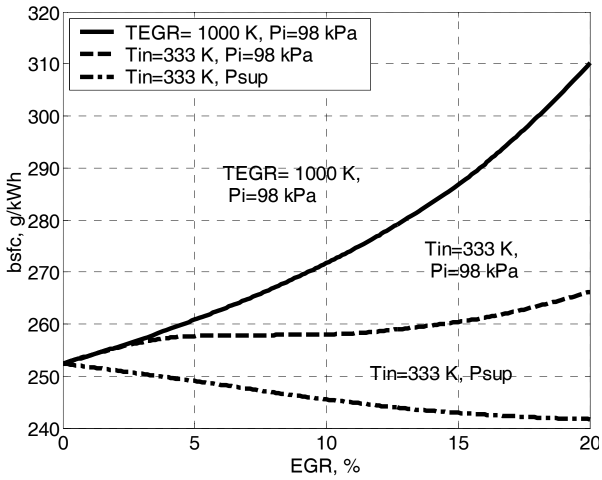

Fig 4: The variation of brake specific fuel consumption with EGR admission at 1000 rpm and different inlet mixture conditions

Figure 4 shows the variation of the brake specific fuel consumption, bsfc, with EGR for the prescribed three inlet conditions at 1000 rpm. The supercharged cooled inlet mixture condition led to a decrease in the fuel consumption with EGR admission. The fuel consumption was reduced by about 4% at 20% EGR condition. For the supercharged cooled inlet mixture condition, the cylinder heat transfer was reduced with the admission of EGR compared to the no EGR inlet condition. That was because of the decrease of the in-cylinder temperature due to EGR admission. The addition of EGR to the inlet mixture led to a decrease in the burned fuel mass and an increase in the total in-cylinder mass, which includes inert gases with high specific heat such as H_2O and CO_2. Furthermore, the pumping friction loss decreased with the admission of EGR due to the increase in the inlet pressure.

In addition to the decrease in the in-cylinder temperature, the admission of EGR in the inlet mixture resulted in a decrease in the oxygen partial pressure. Both of these conditions led to a significant decrease in the in-cylinder NO emission for the three prescribed inlet mixture conditions as shown in Figure 5.

These results agree well with several experimental results such as the results obtained by Lumsden and coworkers [12], who studied the effect of adding EGR to a stoichiometric inlet mixture at constant load and speed on SI engine performance at part load conditions. They found that EGR strategy led to a small but significant improvement in fuel economy in addition to an excellent reduction in NOx emissions.

It can be concluded that the use of EGR with supercharged cooled inlet mixture condition resulted in both an improvement in fuel consumption and a significant reduction in the in-cylinder NO emission. Therefore, this specified inlet condition was selected for further investigations.

In order to show the effect of engine speed when EGR is added to the inlet mixture for a supercharged cooled inlet condition, similar studies were conducted at 2000 and 3000 rpm. For each speed, the inlet pressure was increased with the increase of EGR in order to keep the power the same as the corresponding no EGR power value condition.

The significant reduction of NO emission was achieved at all engine speeds as shown in Figure 6. However, the percentage of fuel consumption improvement was dependent on both engine speed and %EGR as shown in Figure7. The addition of EGR to the inlet mixture increased the combustion angle, and consequently the burning rate decreased. Moreover, the combustion angle increased further with the increase of engine speed. The high value of combustion angle predicted at high engine speeds and high inlet mixture dilution with EGR resulted in an increase in the cylinder heat transfer as higher portions of the fuel mass burned away from the piston top dead centre. That led to a loss in the fuel consumption improvement at 3000 rpm and high inlet mixture dilution as shown in Figure 7.

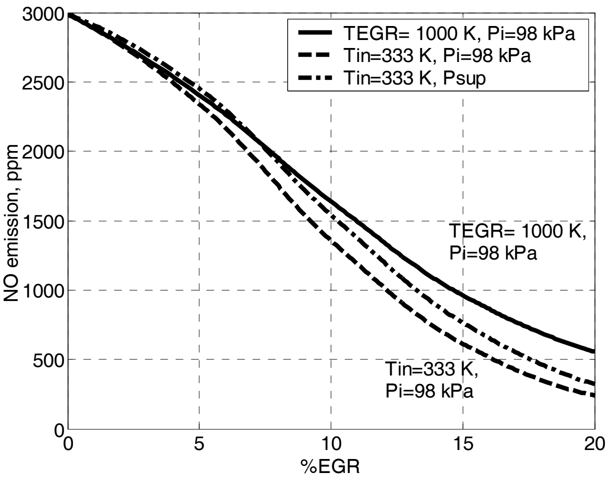

Fig 5: The variation of NO emission with EGR admission at inlet mixture temperature of 333 K, supercharged inlet condition, and different engine speeds.

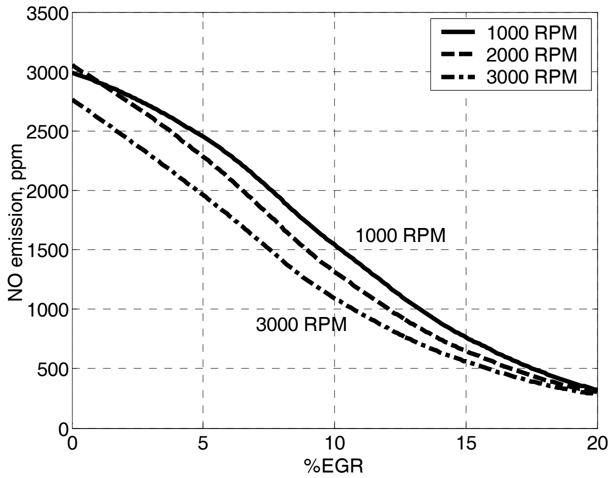

Fig 6: Variations of NO emissions with EGR at different speeds.

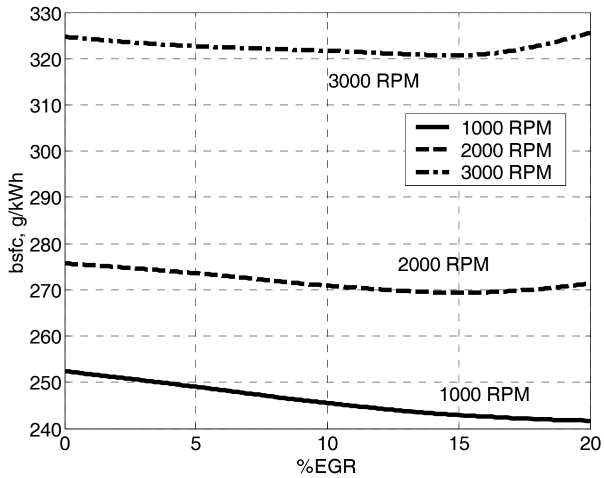

Fig 7: Variation of brake specific fuel consumption with EGR admission at inlet mixture temperature of 333 K, supercharged inlet condition, and different engine speeds

5. CONCLUSIONS

A two-zone combustion model was developed in order to study the effect of adding EGR to a stoichiometric fuel-air mixture on engine performance and NO emissions under several inlet conditions. It was found that The use of Exhaust Gas Re-circulation (EGR) can reduce the in-cylinder NO emission by about 80% and leads to an improvement in fuel consumption compared to a stoichiometric non EGR mixture condition.

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