Comparative Evaluation of the Effects of Intake Air Nitrogen-Enrichment and EGR on the Operational and Environmental Behavior of a SI Heavy Duty Natural Gas Engine

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Abstract. In natural gas spark-ignited engines operating under lean conditions, low temperature combustion is identified as one of the pathways to meet the mandatory ultra low NOx emissions levels set by the regulatory agencies. Exhaust gas recirculation (EGR) has proved to be an effective methodology to reduce in-cylinder combustion temperature and hence NO_x emissions. Nitrogen enrichment of the inducted air is an effective, alternative to EGR, methodology to reduce NOx emissions, since the introduction of inert diluents, such as nitrogen, into a fuel-air mixture results in reduction of the in-cylinder combustion temperature. Since nitrogen-enrichment of intake-air and exhaust gas recirculation (EGR) are two important methods that mostly affect the combustion process occurring within the combustion chamber of natural gas spark-ignited engines, the present work studies their effect on the performance and exhaust emissions of a multi-cylinder, four-stroke, turbocharged, spark-ignited engine fuelled with natural gas. Hence, a theoretical investigation is conducted by using a comprehensive, two-zone, phenomenological model. The model has been properly modified and substantially improved to describe, in a detailed way, the combustion process of the gaseous fuel taking into account the aforementioned techniques. The results concerning engine performance characteristics, NO and CO emissions, with : (i) normal oxygen mass fraction of the inducted air (i.e. normal engine operation NEO), (ii) nitrogen-enriched inducted air (NEIA) and (iii) EGR operating modes, for various engine operating conditions, come from the application of the model. The main objectives are to record and also to comparatively evaluate the relative impact that each one of the above mentioned methods has on the engine performance characteristics and emitted pollutants. Thus, comparing the theoretical results when engine operates with (i) NEIA or (ii) EGR mode, as against with NEO mode, a considerable effect on engine performance and emission characteristics are revealed. Another objective of this assessment is to quantify NO reduction benefit achieved with each of the two strategies examined. The conclusions of the specific investigation will be extremely valuable for the application of the examined technologies on an existing spark-ignited natural gas engine.

Keywords: natural gas, spark ignition, nitrogen enrichment, exhaust gas recirculation.

INTRODUCTION

For meeting stringent imposed emissions regulations, engineers working in the automotive industry or belonging to the research community have focused their interest either on the domain of engine- or fuel-related techniques, such as alternative gaseous fuels or oxygenated fuels that can mitigate emissions, used either in diesel or spark ignition engines that are well established as dominating power-train solution in the world market [1-9]. For the majority of heavy-duty spark-ignited engines, natural gas is most usually introduced with the air during the induction stroke. The majority of this type of engines features a homogeneous natural gas–air mixture compressed rapidly below its auto-ignition condition and ignited around top dead center (TDC) position by the existence of a spark plug. Under constant engine speed, the power output of the specific type of engine is controlled by changing the total amount of the inducted mixture (i.e. air and gaseous fuel). Substantial improvements of these engines in terms of brake specific fuel consumption and reduction of pollutant emissions has been achieved, over the last years, by adopting various engine-related techniques, such as the homogeneous charge compression ignition (HCCI), the micro pilot combustion, the exhaust gas recirculation (EGR), the nitrogen enrichment of the inducted air with the use of air separation membrane (ASM) [10-13] etc. While, each concept has its advantages and disadvantages, exhaust gas recirculation (EGR) has proved to be a very effective tool to reduce the nitrogen oxide emissions. However, exhaust gas recirculation has some significant demerits, such as increased CO emissions, decreased brake thermal efficiency, combustion contamination, greater control system complexity, application variability, material durability, lubricant contamination and increased PM emissions. On the other hand nitrogen enrichment of the inducted air (NEIA) could be an effective strategy, alternative to EGR, without the undesired consequences [10-13]. This strategy could be achieved using a mature technology which involves selective permeation of gases using an air separation membrane [10-13]. Introduction of inert diluents such as nitrogen, into a fuel-air mixture, slow down the reaction rates of participating chemical species, which eventually leads to lower combustion temperatures and hence lower nitrogen oxides. This process gives an added control parameter to reduce combustion temperature in advanced engines.

It is true that experimental work concerning fuel economy and low pollutant emissions from internal combustion engines includes successive changes of each of the many parameters involved, which is very demanding in terms of money and time. Today, the development of powerful digital computers leads to the obvious alternative of simulating the engine performance by mathematical modeling, where the effect of various design and operation changes can be estimated in a fast and inexpensive way. The need for accurate predictions of emitted pollutants has forced researchers to develop two-zone combustion models, accompanied with studies of the significant process of heat transfer in engines, many aspects of which are still unexplored. Eventually, some multi-zone combustion models have appeared where the detailed analysis of fuel-air distribution permits the calculation of exhaust gas composition with reasonable accuracy, but under the rising of computing time cost when compared to lower zones combustion models. At this point it is mentioned that multi-dimensional models have proved useful in examining problems characterized by the need for detailed spatial information and complex interactions of many phenomena simultaneously, but they are limited by the relative inadequacy of sub-models for turbulence, combustion chemistry and by computer size and cost of operation, to crude approximations to the real flow and combustion processes. Thus, it is felt that a good choice for the present study is a two-zone model, which includes the effect of changes in engine design and operation on the details of the combustion process, via a phenomenological model where the geometric details are fairly well approximated by detailed modeling of the various mechanisms involved. This is to have the advantage of relative simplicity and reasonable computer time cost. Numerous experimental and theoretical investigations concerning the effect of (i) nitrogen-enrichment of intake-air [10-16] and (ii) exhaust gas recirculation (EGR) [17-23] on performance and exhaust emissions of an engine have been reported in the international literature.

The primary objective of the present work is to examine, using a theoretical model, the effect of nitrogen-enrichment of intake-air (NEIA) and exhaust gas recirculation (EGR) on the performance and exhaust emissions of an existing stationary, spark-ignited, engine fuelled with natural gas. The theoretical results are generated using a two-zone phenomenological combustion model, which predicts in-cylinder pressure and heat release rate histories, as well as NO and CO concentration profiles. Several of the model predictions on performance and emissions characteristics of the examined engine, have been presented already in the past [14-16]. For the current investigation, the simulation model has been properly modified and improved substantially to describe more accurately the complicated natural gas combustion process in a spark ignition engine environment, taking into account the details of the processes concerning the nitrogen-enrichment of intake-air (NEIA) and exhaust gas recirculation (EGR) and hence, their effects on the combustion process.

The theoretical results corresponding to engine operation without EGR and nitrogen-enriched inlet air (i.e. normal engine operation – NEO operating mode), are validated against respective experimental values obtained from a a multi-cylinder, turbocharged, water-cooled, spark-ignited engine operating under NEO mode at various engine operating points (i.e., load and engine speed) with natural gas. From the comparison of computed and experimental findings, it is revealed that the simulation model developed predicts adequately the engine performance and pollutant emissions trends with engine load under natural gas operation. Furthermore, taking into account data from the international literature, it is shown that the developed model could be used to examine the effect each one of the examined engine parameters on engine performance and pollutant emissions. In any case, it can be used safely in the present work, which performs a comparative assessment by using the simulation results concerning the relative impacts of the examined parameters on engine performance characteristics and emitted pollutants. The results reported herein concern the calculated maximum combustion pressure, ignition delay, duration of combustion, total brake specific fuel consumption, and the calculated brake specific NO and CO emissions, for intermediate and high engine loading conditions at 1500 rpm engine speed. The comparative assessment is accomplished through the comparison of results corresponding to engine operation with EGR or air inlet nitrogen-enrichment, with the corresponding ones corresponding to normal engine operation. From the theoretical findings, important information is derived revealing both the applicability each one of the examined techniques on an existing spark-ignition engine operating with natural gas and also the effect of each technique on engine performance and pollutant emissions.

Consequently, the information derived from the present work is extremely valuable regarding the implementation one of the two strategies examined in the present work for improving the environmental behavior of an existing heavy-duty spark-ignited engine fuelled with natural gas, without deteriorating seriously its performance.

BRIEF DESCRIPTION OF THE MODEL

In the present work only an outline of the model is given since its details have been presented in previous publications [14,16]. The simulation model used is a phenomenological two-zone one, examining the closed part of the engine cycle. At the start of the compression stroke, the cylinder charge is assumed to be a homogeneous mixture of air and gaseous fuel, which has been properly premixed during the induction stroke. During the compression stroke, the entire charge of the mixture is treated as a single zone up to the initiation of combustion. Here, at each instant of time, the perfect gas law describes the state of the mixture inside the cylinder, while there is uniformity in space of pressure, temperature and composition. As is well known, the combustion process of a SI engine is divided into two processes, that is, the ignition process and the stable flame propagation process [24-28]. The former one is made of the flame kernel, which is formed by a spark electric discharge and the unstable flame propagation of the kernel. This is, however, treated as a SI delay period. In the proposed model, the initiation of combustion is assumed to take place when a finite volume of the burned mixture (i.e. the volume of the flame kernel formed by the spark discharge) exceeds 0.001 times the total cylinder volume, that is, displacement plus clearance volume (see Figure 1).

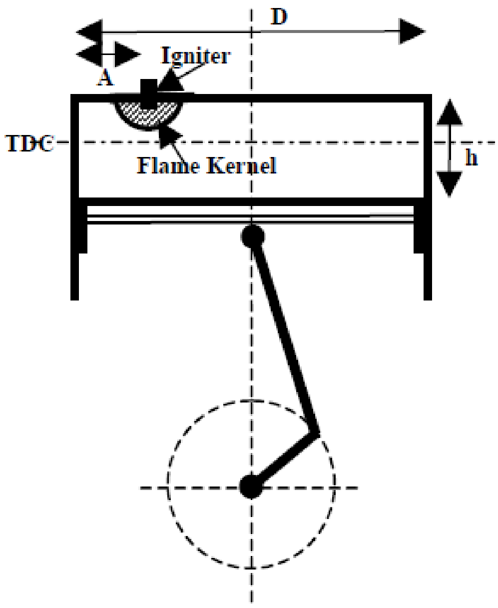

FIGURE 1. Schematic diagram of the flame kernel formation.

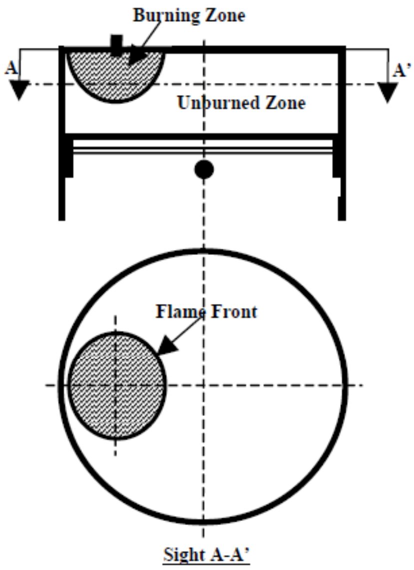

FIGURE 2. Two-zone thermodynamic model of combustion process.

When combustion is initiated, a two-zone phenomenological model is considered to exist for the rest of the closed part of the engine cycle. The first zone consists of air–gaseous fuel mixture (unburned zone), while the second one (burning zone) consists of combustion products and excess air depending on the AFR (air to fuel ratio) (see Figure 2). In each zone, there is uniformity in space of pressure, temperature and composition, at each instant of time, neglecting heat exchange between the zones. After the spark plug ignition, the two zones are separated by a thin flame front that has the shape of a sphere. During the combustion process, the instantaneous cylinder volume and its derivative with respect to crank angle, the flame front area, and the area of the combustion chamber in contact with the unburned and burning zones, are computed through the respective geometric sub-models. Assuming that the geometry of the flame front formed inside the chamber of a spark-ignited engine has a spherical form [2-3], the determination of the instantaneous flame geometry after spark discharge is achieved in the present work by applying a model developed by Annand [28]. The flame front spreads towards the unburned zone, having a flame speed that is calculated by taking into account both the turbulent flame propagation mechanism and the percentage of the nitrogen-enriched inducted-air [27]. After the initiation of combustion, the volume of the burning zone changes due to the existence of the flame front which spreads towards the unburned zone having a turbulent flame speed. The volume change of the burning zone leads to the computation of the quantity of the gaseous fuel and air entrained into the zone. Since the laminar flame thickness under engine conditions is infinitesimal [2-3], in the present model the flame is treated as negligibly thin. The present model assumes that a flame front of negligible thickness propagates into the unburned zone, having a direction perpendicular to the outer spherical surface of the burning zone formed after the initiation of combustion. The gaseous fuel entrained inside the burning zone, due to the flame front spread, is transformed into products. Thus, the combustion rate depends actually on the turbulent flame speed. Heat exchange rate for each zone is calculated by employing the Annand formula [29]. After the initiation of combustion, each zone possesses its own temperature and composition, while the pressure is uniform inside the cylinder. Dissociation of combustion products is taken into account by using the Vickland et al. [30-31] method, incorporating 11 chemical species. For the formation of NO, the extended Zeldovich chain reaction mechanism is considered [30-31].

Model Modifications - Improvements

Below, is given an analysis of the most important modifications performed to the engine simulation model regarding the definition of the calculation of the laminar flame speed and the definition of the charge mixture composition at inlet valve closure (IVC) corresponding to air inlet nitrogen-enrichment and EGR operating modes.

Laminar Flame Velocity

The laminar flame velocity is defined as the relative velocity, with which the unburned gas moves inside the flame front and is transformed to products [2-4]. It is an important intrinsic property of a combustible mixture. The laminar flame velocity developed inside a combustion chamber depends mainly on the equivalence ratio, the temperature of the unburned gas and the pressure [2-4]. Since methane is the main constituent of natural gas, the laminar flame velocity in the present model is obtained by applying a correlation proposed by Karim [32], which simulates adequately the burning velocity of Methane – Air mixtures. The mathematical formula has as follows:

$$S_l = A + \left(F_1 \cdot F_2\right) \cdot \left(F_3 + F_4\left(\varphi_u - 1.036\right) + F_5\left(\varphi_u - 1.036\right)^2\right) \tag{1}$$

where A, F_1, F_2, F_3, F_4 and F_5 are correlations given in [32] that take into account the gaseous fuel equivalence ratio of unburned zone (φ_u), the cylinder pressure and the temperature of the unburned zone. The main advantage of the proposed correlation is that it can predict adequately the laminar burning velocity for non-stoichiometric region.

Definition of the Actual Mass Flow Rate of the Inducted Air at Inlet Valve Closure (IVC)

Under EGR operating mode, the percentage of the exhaust gas re-circulated (x_{EGR}) is defined via the formula:

$$x_{EGR}\left(\%\right) = \frac{\dot{m}_{EGR}}{\dot{m}_{mix,IVC}} \cdot 100\% \tag{2}$$

where ($\dot{m}_{EGR}$) represents the mass flow rate of exhaust gas re-circulated, and ($\dot{m}_{mix,IVC}$) represents the total mass flow rate of gaseous mixture inside the cylinder at inlet valve closure. The latter is calculated by taking into account the pressure, temperature and concentration of the mixture at inlet valve closure. The pressure is assumed to be the one after the air-compressor as this is a turbocharged spark ignited heavy duty engine. It is well known that re-circulated exhaust gas displaces some of the air entering the combustion chamber. Thus, the actual mass flow rate of the inducted air at inlet valve closure is calculated as:

$$\dot{m}_{air,IVC} = \left(1 - x_{EGR}\right) \cdot \dot{m}_{mix,IVC} \tag{3}$$

Then, for specific total air excess ratio, the actual mass flow rate of natural gas is calculated by taking into account the actual mass flow rate of the inducted air as follows:

$$\dot{m}_{NG} = \frac{\dot{m}_{air,IVC}}{AFR_{st} \cdot \lambda} \tag{4}$$

where AFRst corresponds to the stoichiometric air to fuel ratio (by mass) for natural gas. It must be stated here that the mixture at inlet valve closure is assumed to be an ideal homogeneous mixture with uniform composition and thermodynamic properties. It consists of fresh air, gaseous fuel, and re-circulated exhaust gas. The re-circulated exhaust gas is assumed to consist of CO_2 and H_2O, the concentrations of which are obtained from the composition of the cylinder charge at exhaust valve opening event. Moreover, the temperature of the re-circulated exhaust gas is calculated at the exhaust valve opening condition by running the in-house made software for the baseline Normal Engine Operation (NEO). Under nitrogen-enriched air inlet (NEIA) operating mode, nitrogen enrichment alters the chemical composition of the inducted air because N_2 displaces a portion of the inducted air that would otherwise be utilized during combustion, resulting in lowering of the total air excess ratio. In the proposed model, for each load examined (i.e. air excess ratio value), the actual mass flow rate of the air at inlet valve closure ($\dot{m}_{air,IVC}$) is calculated taking into account the percentage of nitrogen enrichment in the inducted mixture combined with the air mass flow rate corresponding to NEO operating mode. The percentage of nitrogen enrichment in the inducted mixture (Δy_{N_2}) is defined as :

$$\Delta y_{N_2}(\%) = \frac{y_{\mathrm{N_2}}^{\mathrm{NEIA}} - y_{\mathrm{N_2}}^{\mathrm{NEO}}}{y_{\mathrm{N_2}}^{\mathrm{NEO}}} \cdot 100 \tag{5}$$

where ($y_{\mathrm{N_2}}^{\mathrm{NEIA}}$) represents nitrogen concentration (by volume) in the inducted mixture under NEIA operating mode while ($y_{\mathrm{N_2}}^{\mathrm{NEO}}$) represents nitrogen concentration (by volume) in the inducted mixture under NEO operating mode. Furthermore, the gaseous fuel consumption ($\dot{m}_{NG}$) is calculated by EQ(4).

TEST CASES EXAMINED

In the present work, the simulation model is used to investigate the relative impact of air inlet nitrogen enrichment and exhaust gas recirculation on the performance characteristics and exhaust emissions of a heavy duty, spark-ignited natural gas engine fuelled with natural gas. Thus, at partial (i.e. 65% of full load) and high (i.e. 100% of full load) load conditions, the percentage of nitrogen concentration in the inducted air was increased by 2, 4 and 6 per cent (Δy_{N2} = 2%, 4% and 6%) relative to the convectional operating case (i.e. 79% by volume in air). Thus, for each loading point examined, besides the oxygen mass fraction of the inducted mixture corresponding to the conventional operating case ($x_{O_2} = 23,2\%$), three different oxygen mass fractions were estimated. Specifically, oxygen mass fraction of the inducted mixture was decreased from $x_{O_2} = 23,2\%$ to $x_{O_2} = 18\%$, at both engine loading conditions examined. Moreover, for each loading point examined, three different EGR percentages were examined herein. Specifically, for each engine load condition, the appropriate EGR percentage was estimated in order that the inducted mixture oxygen mass fraction resulting from the EGR application becomes equal to the respective oxygen mass fraction derived from the application of nitrogen-enrichment operating mode. Thus, under EGR operating mode, EGR percentage was increased from $x_{EGR} = 0\%$ to $x_{EGR} = 15\%$ at partial load conditions, while at high load conditions it was increased up to $x_{EGR} = 10\%$. It must be stated here that the test case corresponding to $x_{O_2} = 23,2\%$ (by mass) nitrogen concentration in the inducted air and $x_{EGR} = 0\%$ is referred as 'normal engine operating point' (NEO point). Moreover, for each test case examined, the injection advance was kept constant. In Table 1 more details are given about the test cases examined in the present work.

Table 1. Test Cases Examine

$x_{O_2}(\%)$	65% Load & λ_α = 1,66 & 1500 rpm			100% Load & λ_α = 1,85 & 1500 rpm		
	λ_{O_2}	$\Delta y_{N_2}(\%)$	$x_{EGR}(\%)$	λ_{O_2}	$\Delta y_{N_2}(\%)$	$x_{EGR}(\%)$
23,2	1,73	0	0	1,92	0	0
21,5	1,60	2	5	1,78	2	6
19,8	1,47	4	10	1,63	4	12
18	1,34	6	18	1,49	6	20

MODEL VALIDATION

Results obtained from an extended experimental investigation conducted in the past on a 'GE Jenbacher 320', multi-cylinder, spark-ignited engine [15] are used to experimentally validate the predictions of the simulation model. The basic data related to the simulated engine are presented in Table 2 [15].

Table 2. Basic data of the Test Engine.

Engine Type	**V-70^0, 20 Cyl., 4-Stroke, SI, T/C**
Bore	135 mm
Stroke	170 mm
Connecting Rod	320 mm
Compression Ratio	11:1
Engine Displacement Volume	48,7 lt
Normal SOI (at 100% load)	23 ^{o}CA before TDC

This is a four-stroke, turbocharged, water-cooled, spark-ignited engine fuelled with natural gas. Since the specific engine is used as an electric power generator (actually in a co-generation mode), the normal speed of the engine is fixed at 1500 rpm and it is kept constant for the entire range of loads examined. A comparison between experimental and calculated pressure traces under NEO operating mode is given in Figure 3. Also, in figure 4 are compared experimental and calculated values of (i) engine power, (ii) engine efficiency and (iii) NO emissions. All experimental data were taken at four different engine loads corresponding to 40, 65, 85 and 100% of full engine load and at 1500 rpm engine speed. Examining these figures, it is observed that there is a good coincidence between calculated and experimental values at all test cases examined. This proves the ability of the specific model to predict adequately performance characteristics and the exhaust emissions of a spark ignition engine operating under natural gas fuel mode with normal chemical composition of the inducted air.

Furthermore, taking into account experimental data it is shown that the simulation model manages to predict with adequate accuracy the trend of the engine performance characteristics with the change each one of the examined techniques. Thus, the specific model can be used to look into the effect each one of the examined parameters on performance and pollutant emissions of the specific engine. It is emphasized here that the values of the present model's constants are held constant for all strategies examined in the present work (i.e. NEIA and EGR).

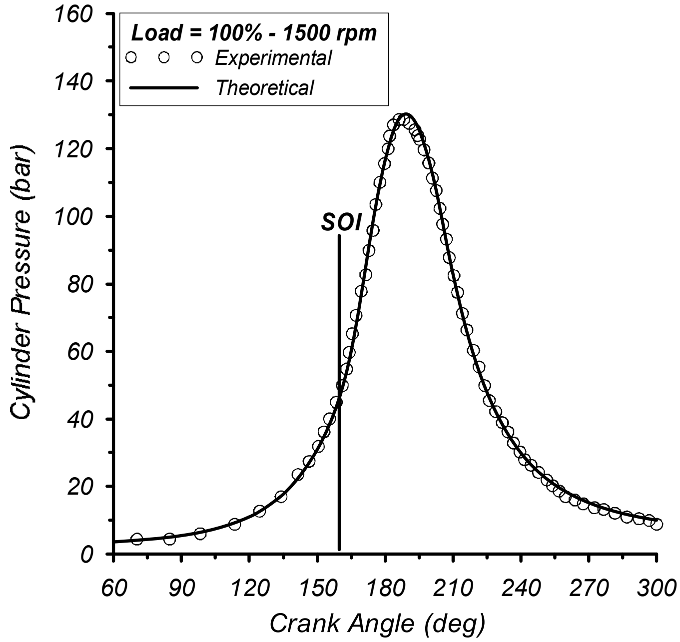

FIGURE 3. Comparison of experimental and computed pressure traces at the engine speed of 1500 rpm and 100% load.

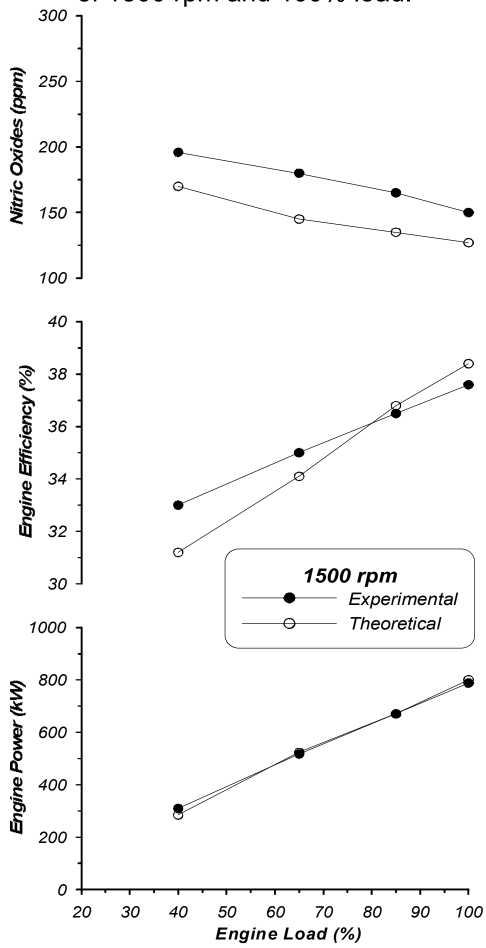

FIGURE 4. Computed and experimental values of (i) Power, (ii) Efficiency and (iii) NO emissions, versus engine load, at 1500 rpm engine speed.

RESULTS AND DISCUSSION

Comparative Evaluation of the Effects of Intake-Air Nitrogen-Enrichment and EGR on SI Natural Gas Engine Performance Characteristics

In this section, the predictive capabilities of the phenomenological model are explored. The predicted effects of the two strategies examined herein on some basic performance characteristics and pollutant emissions of a heavy-duty spark-ignited natural gas engine are examined, for two engine operating points corresponding to 65% and 100% of full engine load at 1500 rpm engine speed.

Figures 5-6 provide the predicted cylinder pressure and total heat release traces for x_{O2} = 19,75 (%) oxygen mass fraction of the cylinder charge at inlet valve closure event, at 100% of full engine load at 1500 rpm engine speed, using the two methodologies examined herein i.e. nitrogen enrichment of the inducted air (NEIA) and exhaust gas recirculation (EGR), respectively. In these figures the predicted cylinder pressure and total heat release traces are also given corresponding to normal engine operating (NEO) mode. Observing figure 5 it can be seen that the air inlet nitrogen-enrichment affects the cylinder pressure history. Thus, by raising the nitrogen mass fraction of the inducted mixture, the rate of cylinder pressure rise during the initial stage of the combustion process becomes lower, while the peak cylinder pressure occurs slightly later compared to the respective values observed under NEO condition. This is the result of both the later initiation of combustion and the lower combustion rate of the gaseous fuel, especially during the initial stages of combustion process, which occurred due to the lower cylinder charge temperatures. As far as the heat release rate curves are concerned (Figure 6), it is observed that the nitrogen enrichment affects also the burning rate. It is shown that the initiation of combustion observed with increased nitrogen mass fraction at IVC starts later compared to the respective one under NEO mode. This is attributed to the increase of ignition delay and also due to the fact that the combustion of the gaseous fuel has not yet progressed enough, since the cylinder charge conditions (i.e. cylinder charge temperature, etc.) do not favor the propagation of the flame front.

Observing figures 5-6, it is obvious that the presence of the exhaust gas re-circulated in the cylinder charge, affects both cylinder pressure and total burning rate. Thus, for high engine loading point, the increase of EGR affects the values of the cylinder pressure compared to the respective one observed under normal engine operating mode. The difference observed during the last stages of the compression stroke is the result of the higher specific heat capacity of the cylinder charge mixture (i.e. air-natural gas-EGR), compared to that of the air-natural gas in the case of NEO mode. Moreover, for all engine operating points examined, as EGR increases the rate of cylinder pressure rise during the first stage of the combustion process becomes lower, while the peak cylinder pressure occurs slightly later compared to the respective values observed without EGR. As far as the heat release rate curve is concerned (Figure 6), it is revealed that for the same oxygen mass fraction at IVC the initiation of combustion observed with EGR operating mode starts later compared to the respective one observed with air inlet nitrogen enrichment (NEIA mode). This is due to the fact that the cylinder charge with EGR (i.e. gaseous fuel – air – EGR mixture) has higher overall specific heat capacity compared to the respective one without EGR. Furthermore, as EGR increases the burning rate observed during the first stage of combustion decreases. This is due to the lower cylinder charge temperature that affects negatively the combustion process of the gaseous fuel, which has not yet progressed enough since the cylinder charge conditions during the specific phase do not favor the existence of the flame front.

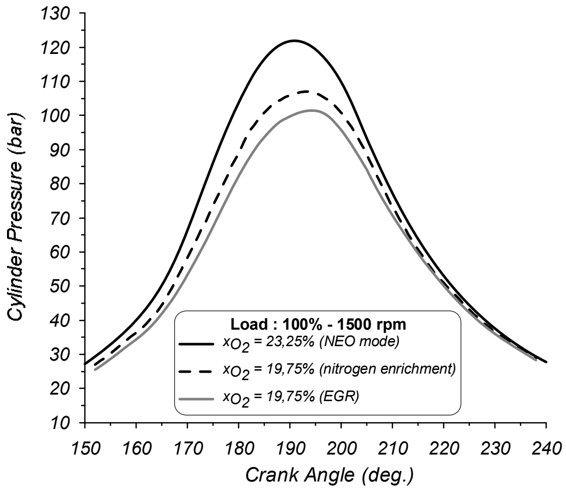

FIGURE 5. Calculated cylinder pressure traces at 1500 rpm and 100% load, for 19.75 (%) oxygen mass fraction in the inducted mixture using (i) air inlet nitrogen enrichment and (ii) EGR.

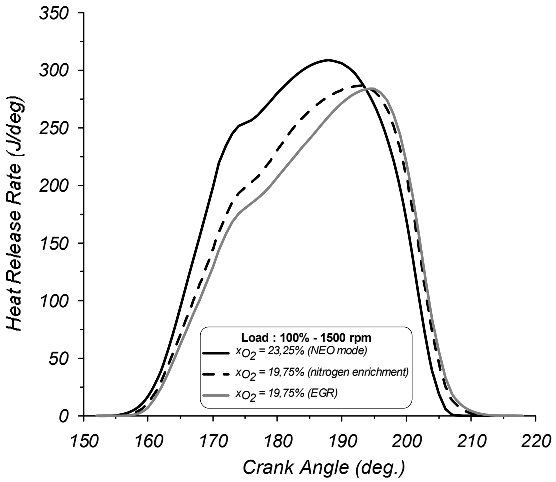

FIGURE 6. Calculated heat release rate traces at 1500 rpm and 100% load, for 19.75 (%) oxygen mass fraction in the inducted mixture using (i) air inlet nitrogen enrichment and (ii) EGR.

Figures 7A-B and 8A-B illustrate the variation of the calculated peak cylinder pressure and maximum cylinder temperature as a function of the oxygen mass fraction of the inducted mixture. The results correspond to 65% and 100% of full engine load conditions, for two strategies examined i.e. nitrogen enrichment and EGR, at 1500 rpm engine speed. In the same figures the normal engine operating point is also given, that corresponds to 23.2 (%) oxygen mass fraction (NEO mode). Since both maximum cylinder pressure and temperature are critical parameters affecting the mechanical and the thermal strength of engine structure, the study of the effect of the examined strategies on maximum cylinder pressure and temperature is of particular interest. By examining these figures, it is revealed that the maximum cylinder pressure decreases with the increase of the percentage of air nitrogen-enrichment resulting thus in lower cylinder charge temperatures. This is attributed to the decrease of the gaseous fuel combustion rate, due to the retardation of the flame front. The effect becomes more evident at full engine load conditions, where the decrease of the maximum cylinder temperature is up to 10%. For both loads examined, the decrease of the maximum cylinder pressure is not so severe (up to 10% at part load, and up to 8% at high load). Regarding the effect of EGR on both maximum cylinder pressure and temperature, it is observed that for all test cases examined, the increase of EGR leads also to the decrease of the maximum cylinder pressure and temperature. Despite the smooth decrease of the maximum temperature, the maximum cylinder pressure starts to decrease slightly with an increase of EGR until a certain limit, where a further increase of EGR leads to a more intense decrease of the maximum combustion pressure. The effect becomes more evident at high load. Eventually, it should be mentioned, that under EGR operating mode, the lower heat release rate and the higher specific heat capacity of the cylinder charge are the main reasons for the lower and delayed appearance of the maximum combustion pressure, compared to the respective values observed under NEO and air inlet nitrogen enrichment operating modes.

Figures 9A-B depict the variation of the calculated duration of combustion as a function of the oxygen mass fraction of the inducted mixture. The results correspond to 65% and 100% of full engine load conditions, for two strategies examined i.e. intake-air nitrogen-enrichment and EGR, at 1500 rpm engine speed. In the same figures the normal engine operating point is also given, that corresponds to 23.2 (%) oxygen mass fraction (NEO mode). Examining these figures, it is observed that for both engine loads examined, both strategies examined herein lead to a longer duration of combustion as compared to the respective one observed under NEO mode.

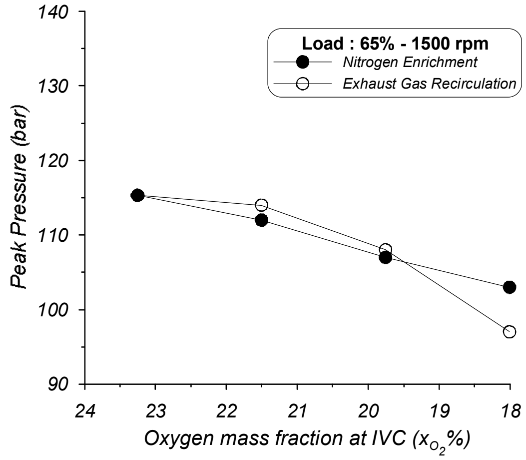

FIGURE 7A. Maximum pressure as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 65% load, for (i) NEIM and (ii) EGR operating modes.

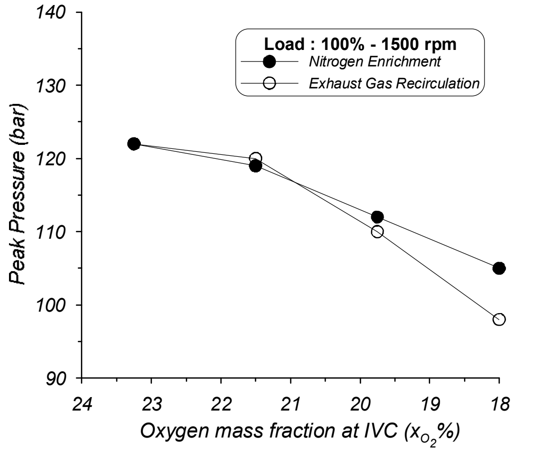

FIGURE 7B. Maximum pressure as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 100% load, for (i) NEIM and (ii) EGR operating modes.

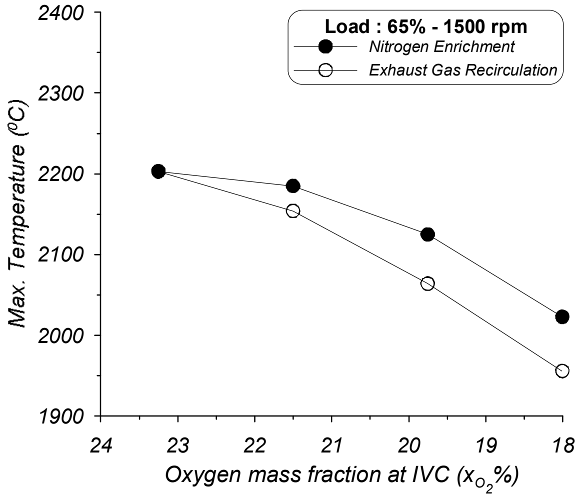

FIGURE 8A. Maximum temperature as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 65% load, for (i) NEIM and (ii) EGR operating modes.

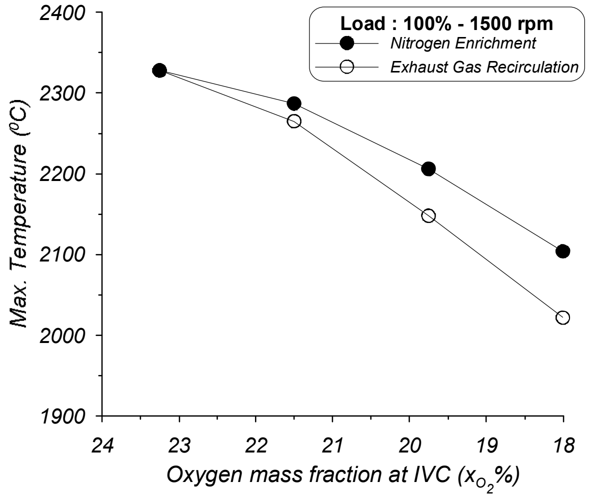

FIGURE 8B. Maximum temperature as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 100% load, for (i) NEIM and (ii) EGR operating modes.

Specifically, nitrogen enrichment prolongs the duration of combustion and the specific effect becomes more intense at high load and high nitrogen percentages in the inducted mixture. This may be attributed, primarily, to the fact that nitrogen enrichment delays slightly the initiation of combustion. This emanates from the higher specific heat capacity of the mixture accompanied by the lower cylinder charge conditions occurring at the spark timing, compared to the respective values under NEO operating mode.

According to Figures 9A-B, it is observed that, despite the fact that the lower total air excess ratios, caused by the presence of EGR, is a critical factor favoring the flame propagation mechanism contributing, thus, to an improvement of the natural gas combustion quality, as EGR percentage increases the duration of combustion also increases. This specific effect is ascribed primarily to the low cylinder charge temperature, due to the higher specific heat capacity of the cylinder charge and also to the slower combustion rate of the natural gas.

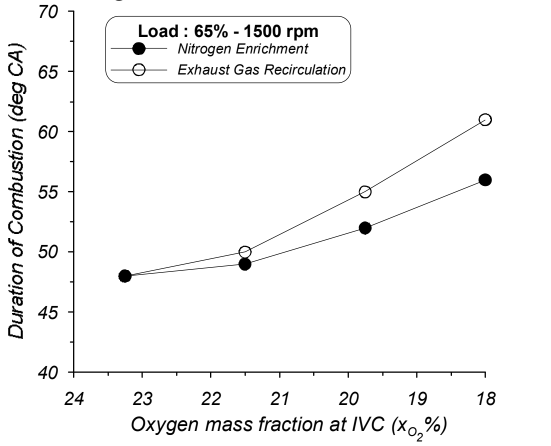

FIGURE 9A. Duration of Combustion as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 65% load, for (i) NEIM and (ii) EGR operating modes.

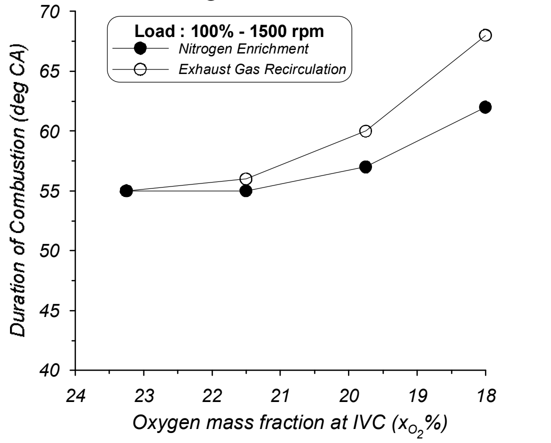

FIGURE 9B. Duration of Combustion as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 100% load, for (i) NEIM and (ii) EGR operating modes.

Figures 10A-B illustrate the variation of the calculated brake specific fuel consumption (bsfc) as a function of the oxygen mass fraction of the inducted mixture. The results correspond to 65% and 100% of full engine load conditions, for two strategies examined i.e. nitrogen enrichment and EGR, at 1500 rpm engine speed. In the same figures the normal engine operating point is also given (NEO mode). It must be noted here that the computed bsfc is estimated from the calculated brake power output and the calculated mass flow rate of the natural gas. Moreover, brake power output is estimated from the calculated indicated power output (i.e. calculated from cylinder pressure diagram) and the mechanical efficiency that is predicted through a simple simulation sub-model. Observing these figures, it is revealed that for both loads examined, the increase of the nitrogen percentage in the inducted mixture results to an increase of bsfc since the combustion of the fuel becomes more ineffective. As far as the effect of EGR percentage on bsfc is concerned, it is revealed that the presence of EGR affects also the brake engine efficiency. Specifically, for both loads examined, the increase of the EGR percentage leads initially to a slight increase of the total brake specific fuel consumption, while a further increase of the EGR beyond a critical percentage results to a more intense deterioration of engine efficiency. This specific deterioration is ascribed primarily to the longer ignition delay period, which affects negatively the heat release rate, especially during the initial stages of combustion process. At the same time, the improvement of the gaseous fuel combustion quality, which is caused by the reduction of the total air excess ratio (i.e. lower total air excess ratio leads to faster flame speed), does not contribute considerably to the improvement of engine efficiency.

Comparative Evaluation of the Effects of Intake-Air Nitrogen-Enrichment and EGR on SI Natural Gas NO and CO Emissions

Figures 11A-B illustrate the variation of the calculated specific NO emissions as a function of the oxygen mass fraction of the inducted mixture. Theoretical results are presented at65% and 100% of full engine load conditions, for the two strategies examined herein i.e. nitrogen enrichment and EGR, at 1500 rpm engine speed. In the same figures predictions for the normal engine operating point are also given. It is well known [2-4] that the formation of NO is favored in general by high gas temperatures and near stoichiometric mixture conditions towards the lean.

Observing figures 11A-B, it is revealed that, for both loads examined, specific NO concentration under nitrogen enriched inducted mixture (NEIM) operating mode is lower compared to the one observed under NEO mode. For the same load, the burning temperature observed under NEIA mode is lower compared to the respective one under NEO mode. Moreover, nitrogen enrichment depletes effectively oxygen concentration in the cylinder charge. Thus, the lower oxygen concentration in combination with the lower burning temperatures provides a possible explanation about the lower NO concentrations observed under NEIM operating modes compared to the respective ones under NEO operating mode.

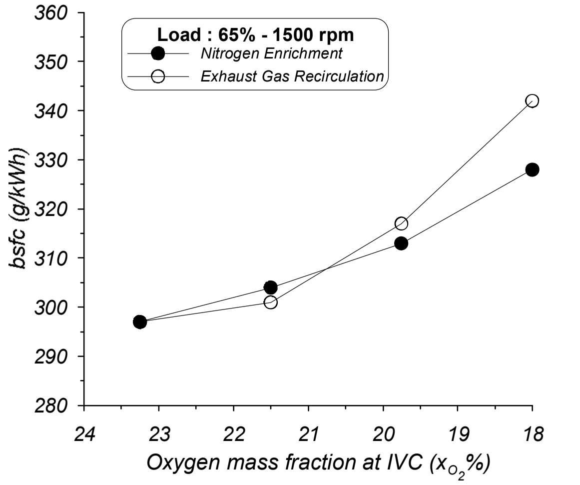

FIGURE 10A. Brake specific fuel consumption as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 65% load, for (i) NEIM and (ii) EGR operating modes.

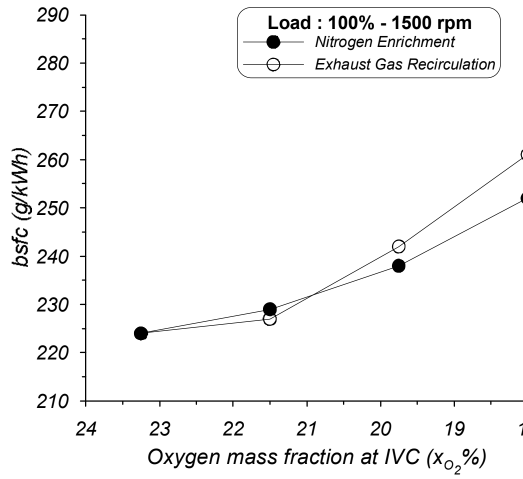

FIGURE 10B. Brake specific fuel consumption as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 100% load, for (i) NEIM and (ii) EGR operating modes.

As far as the effect of EGR on specific NO emissions is concerned, it is revealed that the increase of EGR percentage results also in a decrease of specific NO emissions. This may be attributed to the delayed, relative to TDC, initiation of combustion due to the increase of the ignition delay period. Furthermore, the increase of EGR percentage results to lower charge temperature caused by the higher specific heat capacity of the cylinder charge and, moreover, it leads to a reduction of the oxygen availability in the cylinder charge. The aforementioned parameters restrain the NO formation mechanism. By examining figures 11A-B it is revealed that for high load examined, NO emissions seems to be more sensitive to EGR rather than to nitrogen enrichment of the inducted air. Inferentially, by comparing the results obtained from both strategies examined, it is revealed that for an existent SI engine, running at high load, the curtailment of the emitted NO without serious deterioration of engine efficiency may be achieved with nitrogen enrichment of the inducted air instead of using EGR.

Figures 12A-B show the variation of the calculated specific CO emissions as a function of the oxygen mass fraction of the inducted mixture. The results correspond to 65% and 100% of full engine load conditions, for two strategies examined i.e. nitrogen enrichment and EGR, at 1500 rpm engine speed. In the same figures predictions for the normal engine operating point are also given. As known [2-4], CO formation rate depends on the relative air/fuel ratio, the unburned gaseous fuel availability and the cylinder charge temperature. The latter two parameters control the rate of fuel decomposition and oxidation [2-4].

Observing figures 12A-B, it is revealed that the decrease of oxygen mass fraction of the inducted mixture using nitrogen enrichment results in a negligible variation of CO emissions. Specifically, the increase of nitrogen concentration in the charge mixture causes a slight reduction of the total air/fuel ratio. This leads to a negligible effect on both CO formation and oxidation rates, due to the decreased charge temperature. Thus, the emitted CO concentrations observed under NEIA operating mode seem to be almost the same with the respective ones observed under normal engine operating mode. On the other hand, the increase of EGR percentage results to an increase of CO emissions. Specifically, the increase of EGR percentage causes an increase of the ignition delay period, which suppresses the progress of the gaseous fuel combustion process, a situation that affects negatively (i.e. increase) the emitted carbon monoxide. On the other hand, the increase of EGR percentage promotes slightly CO oxidation rate, due to the lower total air-fuel excess ratio, which leads to a slight acceleration of the flame front and, thus, to a slight improvement of the gaseous fuel combustion rate. Nonetheless, the aforementioned improvement contributes insignificantly to the reduction of the emitted CO, since it occurs late. Observing the results, it is revealed that for high engine operating point the effect of EGR on CO emissions is more intense compared to the respective effect caused by the nitrogen enrichment of the inducted air.

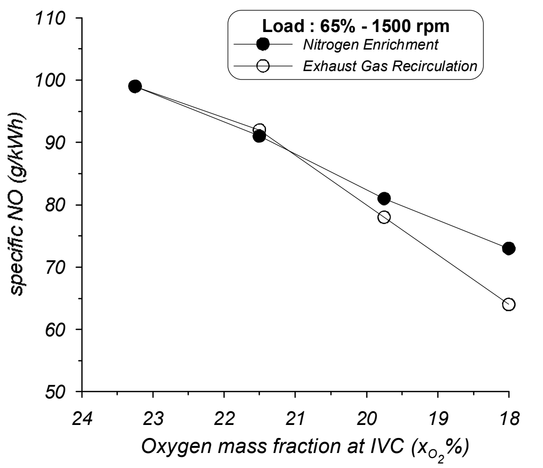

FIGURE 11A. Specific NO concentration as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 65% load, for (i) NEIM and (ii) EGR operating modes.

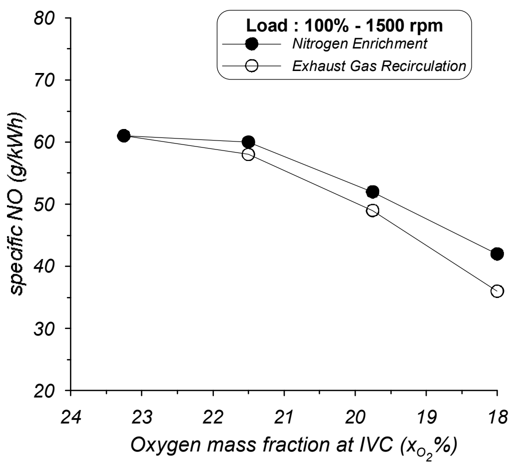

FIGURE 11B. Specific NO concentration as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 100% load, for (i) NEIM and (ii) EGR operating modes.

CONCLUSIONS

In the present work, an existing two-zone phenomenological model has been used to examine the effect of (i) nitrogen enrichment of the inducted air and (ii) EGR, on performance characteristics and pollutant emissions of a natural gas spark ignited engine. A good coincidence between calculated and measured values under normal composition of the inducted air (NIA) operation was observed for performance characteristics, NO and CO emissions. Specifically, the model predicts with reasonable accuracy the absolute values but most important it predicts the trends of the combustion and pollutants formation mechanisms with various engine operating parameters.

Acknowledging the predictive ability of the two-zone combustion model, it was used to examine the effect of the aforementioned strategies on engine performance parameters, NO and CO emissions. From the evaluation of the theoretical findings, the following conclusions can be summarized as below:

- the increase of nitrogen mass fraction in the inducted mixture, results to:
    - deterioration of engine efficiency. The effect is more evident at intermediate load and high N_2 mass fractions in the inducted mixture.

- o decrease of the maximum cylinder pressure, which at high N_2 mass fractions in the inducted mixture is up to 8%.
- o decrease of the specific NO concentration. The effect is more evident at high N_2 mass fractions in the inducted mixture for both loads examined.
- o an almost negligible variation of the specific CO concentration.
- the increase of EGR percentage, results to:
  - o deterioration of engine efficiency. The effect is more evident at high load and high EGR percentages.
  - o decrease of the maximum cylinder pressure, which at high EGR percentage is up to 12%.
  - o decrease of the specific NO concentration. The effect is more evident at high load and high EGR percentages.
  - o increase of the specific CO concentration. At high load and high EGR percentage the specific increase is up to 30%.

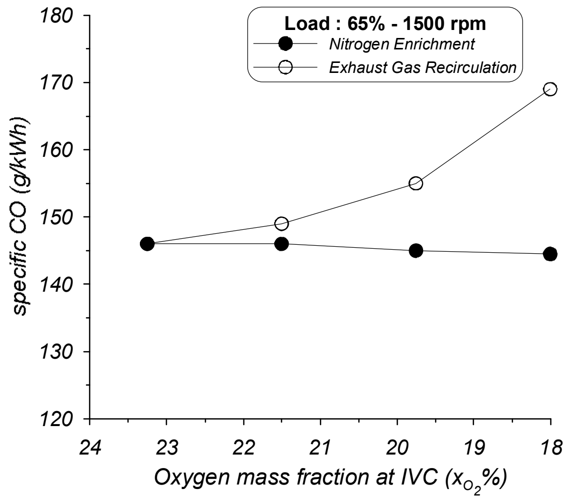

FIGURE 12A. Specific CO concentration as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 65% load, for (i) NEIM and (ii) EGR operating modes.

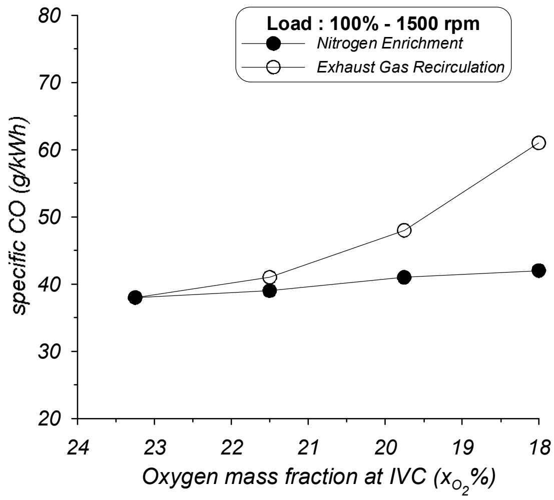

FIGURE 12B. Specific CO concentration as a function of oxygen mass fraction of the inducted mixture, at 1500 rpm and 100% load, for (i) NEIM and (ii) EGR operating modes.

In general, the increase of nitrogen percentage in the inducted mixture could be a promising solution for reducing NO emissions as compared to EGR. At low percentages, the specific strategy does not bring serious problems to engine performance characteristics. However, at high engine loads, the excessive increase of the nitrogen enrichment percentage beyond a certain limit may be proven to be harmful to engine performance characteristics (i.e. brake efficiency, engine power output). At the same time, the simultaneous increase of both parameters, at both low and high engine load conditions, does not bring any serious problem to engine operational lifetime, since the maximum cylinder pressure is lower compared to the respective one observed under normal engine operation. The results of this preliminary investigation are encouraging and urge us to prolong our theoretical investigation to examine the combined effect of other engine parameters (i.e. ignition timing, etc) on performance characteristics of an existing spark-ignited engine fuelled with natural gas. This is currently under progress and results will be given in the near future. Even though it is difficult to generalize the findings of the current preliminary investigation, we believe that they are important since the reduction of NO emissions on existing SI natural gas engines is extremely important.

REFERENCES

1. Korakianitis T, Namasivayam AM, Crookes RJ. Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions. Prog Energy Combust Sci 2011;37:89-112.
2. Heywood J.B. Internal Combustion Engine Fundamentals. New York: McGraw–Hill, 1988.
3. Ferguson, C.R. Internal Combustion Engines – Applied Thermosciences, John Wiley, New York, 1986
4. Benson, R.S. and Whitehouse, N.D. Internal Combustion Engines, Pergamon Press, Oxford, 1979
5. Karim, G.A. and Ali, I.A. Combustion, knock and emission characteristics of a natural gas fuelled spark ignition engine with particular reference to low intake temperature conditions, Proceedings of the Institute of Mechanical Engineers, Vol. 189, pp.139–147, 1975.
6. Krishnan S.R., Biruduganti M., Mo Y., Bell S.R., Midkiff K.C., Performance and Heat Release Analysis of a Pilot-Ignited Natural Gas Engine. J. of Engine Research 2002;3:171-183.
7. Srinivasan K.K., Krishnan S.R., Midkiff K.C., Improving Low Load Combustion, Stability and Emissions in Pilot-Ignited Natural Gas Engines. In Proceedings of IMechE 2006, J. of Automobile Engineering 2006;220:229-239.
8. Liss, W.E. and Thrasher, W.H., Natural Gas as a Stationary Engine and Vehicular Fuel, SAE Paper No 912364.
9. Unich, A., Bata, R.M. and Lyons, D.W., Natural Gas: A Promising Fuel for I.C. Engines, SAE Paper No 930929.
10. Poola R.B., Stork, K.C., Sekar, R.R., Callaghan, K. and Nemser, S., Variable Air Composition With Air Seperation Membrane: A New Low Emissions Tool for Combustion Engines, SAE Transactions, 1998, 106, pp. 332-346.
11. Wong, H.C., Beck, N.J. and Chen, S.K., The Evolution of Compression Ignition Natural Gas Engines for Low Emission Vehicles, 2000, ASME Paper No 2000-ICE-318.
12. Biruduganti, M., Gupta, S. and Sekar, R., Low Temperature Combustion Using Nitrogen Enrichment to Mitigate NOx from Large Bore Natural Gas Fuelled Engines, ASME Paper No ICES2008-1616.
13. Biruduganti, M., Gupta, S., McConnel, S. and Sekar, R., Nitrogen Enriched Combustion of a Natural Gas Engine to Reduce NOx Emissions, ASME Paper No ICEF2004-843.
14. Papagiannakis R.G., Rakopoulos C.D., Hountalas D.T and Giakoumis E.G., Study of the performance and exhaust emissions of a spark-ignited engine operating on syngas fuel, Int. J. Alternative Propulsion, Vol. 1, No. 2/3, 2007
15. GE Jenbacher – Technical Report Laminar Flame Speed versus Air Fuel Ratio – Experience in Woodgas Plants, June 2005, Jenbach, Austria.
16. Papagiannakis R.G., Zannis T.C., Hountalas D.T. and Kotsiopoulos P.N., Study of Performance and Exhaust Emissions of a Spark-Ignited Engine operating with Nitrogen Enrichment of Intake Air, ECOS Paper, 2011.
17. Srinivasan K. K., Krishnan S. R., Qi Y., Midkiff K. C. and Yang H.., Analysis of diesel pilot-ignited natural gas low temperature combustion with hot exhaust gas recirculation. Journal of Combustion Science and Technology, 2007, 179:9,1737 -1776
18. 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, J. of Energy Conversion and Management, 2007, 48, 1909-1918
19. Pirouzpanah V., Khoshbakhti Sarai 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, 2003, 217, 719-725.
20. Krishnan, S.R., Srinivasan, K.K., Singh, S., Bell, S.R., Midkiff, K.C., Gong, W., Fiveland, S. and Willi, M., Strategies for reduced NOx emissions in pilot-ignited natural gas engines., ASME-WA Meeting, Proc. ICEF, 2002, 39, 361-367.
21. Kusaka J., Okamoto T., Daisho Y., Kihara R., Saito T., Combustion and exhaust gas emission characteristics of a diesel engine dual- fueled with natural gas. Society of Automotive Engineers of Japan, Inc. and Elsevier Science, JSAE Review, 2000, 21, 489-496
22. Ishida M, Tagai T, Ueki H. Effect of EGR and preheating on natural gas combustion assisted with gas-oil in a diesel engine. JSME Int J, 2003;46:124-30.

23. Abd Alla GH. Using exhaust gas recirculation in internal combustion engines: a review. Energy Convers Manage, 2002, 43:1027-42.
24. Blizard, N.C. and Keck, J.C. Experimental and Theoretical Investigation of Turbulent Burning Model for Internal Combustion Engines, SAE Paper No. 740191.
25. Lucas, G.G. and James, E.H., Computer Simulation of a Spark Ignition Engine, SAE Paper No. 730053.
26. Lavoie, G. Correlations of Combustion Data for a SI Engine Calculations-Laminar Flame Speed, Quench Distance and Global Reaction Rates, SAE Paper No. 780229.
27. Bayraktar H. Experimental and theoretical investigation of using gasoline-ethanol blends in spark ignition engines. J. of Renewable Energy 30, 11, pp. 1733-1747, 2005.
28. Annand, W.J.D. Geometry of spherical flame propagation in a disc-shaped combustion chamber, Journal of Mechanical Engineering Science, Vol. 12, pp.146–149, 1970.
29. Annand, W.J.D. Heat transfer in the cylinders of reciprocating internal combustion engines, Proceedings of the Institute of Mechanical Engineers, Vol. 177, pp.973–990, 1963
30. Vickland, C.W., Strange, F.M., Bell, R.A. and Starkman, E.S. A consideration of the high temperature thermodynamics of internal combustion engines, Transactions of the SAE, Vol. 70, pp.785–793, 1962
31. Lavoie, G.A., Heywood, J.B. and Keck, J.C. Experimental and theoretical study of nitric oxide formation in internal combustion engines, Combustion Science and Technology, Vol. 1, pp.313–326, 1970
32. Al-Himyary TJ, Karim GA. A correlation for the burning velocity of methane – Air mixtures at high pressures and temperatures. ASME Trans, J Eng Gas Turbines Power 1987;109:439-42.