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SIMULATION OF SI ENGINE PERFORMANCE AND NO_x EMISSIONS

Abdorouf M. Naas, Fatima M. Ellafi, and Salem A. Farhat

* Department of Mechanical and Industrial Engineering, Faculty of Engineering,
University of Tripoli, Libya
E-mail: A.naas@uot.edu.ly

الملخص

تشكل أكاسيد النيتروجين (NO_x) المنبعثة من محركات الإشعال بالشرارة (SI) تهديدًا مباشراً على صحة الإنسان، وخاصةً التي يتم فيها استخدام الوقود السائل كالبنزين، وهي المسؤول الأساسي في الغالب عن تلوث الهواء. عليه، يجب معالجة هذه الظاهرة المدمرة للبيئة وتقليلها من خلال تحسين أداء محركات الإشعال بالشرارة. تم في هذه الورقة التركيز على دراسة انبعاثات أكاسيد النيتروجين (NOx) من محرك اشعال بالشرارة. لأجل تم تطوير برنامج كمبيوتر باستخدام لغة FORTRAN، هذه المحاكاة تستند إلى العلاقات الديناميكية الحرارية لمحركات الاحتراق الداخلي الترددية. يهدف هذا البحث إلى دراسة معدلات الأداء والانبعاثات من محرك الاشعال بالشرارة (تم اختيار محرك عالي السرعة أحادي الأسطوانة من الأبحاث المنشورة مسبقًا من أجل إجراء بعض المقارنة والتحقق من نتائج المحاكاة) بتغيير بعض ظروف تشغيل المحرك، وقد أظهرت نتائج المقارنة أنها متطابقة ومتفقة تماما أثناء شوطي الانضغاط والتمدد. تم دراسة تكوين أكاسيد النيتروجين (NOx) في ظل ظروف مختلفة؛ بما في ذلك تأثير درجة حرارة الهواء الداخل (T_i) لاسطوانة المحرك بشكل طبيعي دون استخدام شحن جبري، وتأثير توقيت الاشعال (θ_{ig}) لشمعة الإشعال، وكذلك تأثير نسبة التكافؤ بين الهواء والوقود (ϕ) على تكوين أكاسيد النيتروجين. أظهرت النتائج أن درجة حرارة الهواء الداخل للاسطوانة لها تأثير كبير جداً في تقليل معدلات تكوين أكاسيد النتروجين، حيث انخفض بنسبة حوالي 20% عند انخفاض درجة حرارة الهواء الداخل للاسطوانة من 350K إلى 283K. كما أن لزمن الاشعال له تأثير مباشر على تكوين أكسيد النيتروجين، وذلك بسبب ارتفاع درجة حرارة غازات الاحتراق داخل الاسطوانة، وفي هذه الحالة، زمن الاشعال المبكر تسبب في حدوث أقصى ضغط لغازات الاحتراق في الأسطوانة عند النقطة الميتة العليا (TDC)، مما نتج عنه انخفاض في الكفاءة الحرارية بشكل كبير وارتفاع درجة حرارة غازات الاحتراق يعطي فرصة أكبر لتكوين أكاسيد النيتروجين. أيضاً، يتسبب تأخير زاوية الإشعال في حدوث انخفاض في الحد الأقصى للضغط داخل الأسطوانة بزاوية متأخرة جدًا بعد النقطة الميتة العليا (ATDC) مما يتسبب عنه انخفاض في قدرة المحرك، أظهرت النتائج أن تغير توقيت زاوية الإشعال من 20^o إلى 40^o (BTDC) نتج عنه انخفاض في تكوين (NO_x) بنسبة 74%. كذلك بينت النتائج أن تأتير نسبة الهواء للوقود أدت إلى إنخفاض واضح في تكوين (NOx). عندما تم تشغيل المحرك بخليط فقير جداً فإن درجة حرارة غازات الاحتراق انخفضت بشكل كبير مما نتج عنه انخفاض في تكوين أكسيد النيتروجين، حيث انخفض بنسبة 96% عند انخفاض نسبة التكافؤ من $\phi = 0.85$ إلى $\phi = 0.6$.

ABSTRACT

Nitrogen oxides (NO_x) poses a direct threat to human health during peak pollution and is mainly emitted from vehicles powered by spark ignition (SI) engines, especially in which liquid fuel is used such as gasoline, and is mostly responsible for air pollution, therefore, this destructive phenomenon of the environment must be addressed and reduced by improving the performance of spark-ignition engines.

In this paper, the focus was on the emissions of nitrogen oxides (NO_x) from the spark-ignition engine, a computer program was developed using a FORTRAN language, the simulation based on the thermodynamics relations of piston internal combustion engines, so the aim of the present paper intends to study the performance and emission rates of a spark-ignition engine (Single-cylinder high-speed engine was selected from previously published research in order to make some comparison and verification of the results of the simulation) by changing some of operating conditions of the engine. The formation of nitrogen oxides (NO_x) have been investigated under different conditions; including the effect of the intake air temperature (T_i) naturally aspirated into the engine cylinder, the effect of the ignition timing (θ_{ig}) of the spark plug, and also the effect of the equivalence ratio of air to fuel (ϕ) on the formation of nitrogen oxides. The results showed that the intake air temperature (T_i) has a strong effect on the reduction of nitrogen oxides (NO_x) formation when the intake air temperature decreased from 350K to the temperature of 283K, the emission of nitrogen oxide decreased by about 20%. Also, the spark ignition angle has a strong effect on the formation of NO_x, that is because of the high temperature of gases produced from the combustion inside the cylinder, in this case, an early spark ignition angle may not provide a good performance of the engine because of occurring the maximum pressure in the cylinder at the top dead center (TDC), which causes a decrease in thermal efficiency, high temperature, and a decrease in power. Also, the ignition angle delay causes a drop in the maximum pressure inside the cylinder at a very late angle After Top Dead Center (ATDC). The results show that by changing the ignition from 40^o to 20^o Before Top Dead Center (BTDC), the formation of NO_x decreased by 74%. The effect of air-fuel ratio, on the formation of $NO_{x,}$ significantly decreased on the formation of NO_x when the engine was operated with a very lean mixture because the temperature of the gas at very lean is very low. Nitrogen oxide formation decreased from $\phi = 0.85$ to $\phi = 0.6$ by 96%.

KEYWORDS: Air Pollution; NOx; Spark Ignition Angle; Ignition Timing; Air to fuel Ratio.

INTRODUCTION

Pollutants are all biological, physical, or chemical variables that are released into the environment through engine emissions, especially fossil-fuel heat engines, which generates many toxic gases, including carbon dioxide, carbon monoxide, nitrogen oxide, Sulphur oxide, and unburned fuel, and also acid rain. Acid rain is one of the polluting causes of air due to the high emission of heat engines of Sulphur oxide SO_x and NO_x in the atmosphere, through the chemical reactions of these environmentally stable oxides, H_2SO_4 and HNO_3 are produced. Acid rain can cause significant damage to the earth's environment such as water, fish, and even humans.

In this paper, the focus will be on the heat engine of the type of spark-ignition engine, through mathematical simulation based on thermodynamics relationships to improve engine performance and reduce its NO_x emissions into the environment.

Many previous researches were done by studying and analyzing engine performance, and its emissions, including experimental and theoretical studies. Most studies have focused on designing a geometry of combustion chamber to improve engine performance and reduce environmental pollution, and some research concerned about controlling some different engine variables, such as valve timing and ignition timing, fuel ratio, and compression ratio, etc. reference [1] studied an experimental test for a spark-type engine using two commercial software for modeling to control engine performance, several laboratory experiments were conducted in their investigation, in terms of pressure, and temperature inside the cylinder as function of the engine crank angle at different engine speeds had been recorded. This reference is used to compare the results of mathematical analysis in this paper of the pressure and temperature of gases in the cylinder as a function of crank angle with the same engine specifications.

For spark ignition (SI) engines, a common approach to reduce engine-out emissions and fuel consumption by application of diluted combustion. This concept is based on the addition of air or combustion residuals to the stoichiometric air to fuel mixture. Most of the researches changed the ignition time and adjusting valves timing, the results in their study showed a decrease in fuel consumption but also caused an increase in NO_x emissions, so it requires some treatment to the system to reduce NO_x emissions as much as possible [2], also the Miller Cycle has been used to reduce the emissions of NO_x from the spark ignition engines. The Miller Cycle shows good results in the reduction of NO_x with slight loss of engine power when operating the engine at Maximum load compared with an Otto-cycle engine [3].

The use of natural gas fuel in spark-ignition engines has also contributed significantly to improved engine performance as well as a significant reduction in emitted pollutants. Many research have focused on this type of study, by improving performance and reducing NO_x emissions of the engine. When natural gas is used as an alternative fuel, it allows for an increased compression ratio without the occurrence of the knocking phenomenon. The engine is also operated at an equivalence air fuel ratio from 0.95 to a very low level of lean mixture compared to the use of gasoline liquid fuel. The NO_x levels increase with increasing compression ratio, and engine speed on the lean side. Retarding spark timing reduced the NO_x levels highly at the lean side, and the maximum formation of NO_x at A/F ratio at the rich side. NO_x levels increased with compression ratio increase at fixed spark timing operation, but when an optimum spark timing was used the compression ratio impact was reversed to the spark timing effect [4].

Some researchers describe the results of the thermodynamic evaluation of methanol as dedicated alternative fuel for gasoline based spark-ignition engines. The investigations have been done for the octane demand of the engine for methanol under variable load and rated speed conditions. The thermodynamic results showing the lower octane demand of the engine for methanol implies that gasoline operated spark ignition engine needs changes in its design in order to operate gasoline spark ignition engine by using methanol fuel, the change in its design is needed for better performance. They concluded that there is a good possibility of using alternative sources of fuel such as Methanol as a spark-ignition engine fuel, the higher compression ratio will help in reducing the gap between the performances of this engine under gasoline and methanol modes. Their results were also generated for comparative CO, HC, and NO_x emissions characteristics of the engine [5].

Gong. et, al [6] use the classical extended Zeldovich mechanism, and N_2O pathway for the NO_x formation mechanism is employed as the chemical mechanism in the model. Their results show in the model that the air-fuel mixing and inhomogeneity of the charge. Since the temperature has a dominant effect on NO_x emission, a flame temperature correlation was developed to model the flame temperature during the combustion for NO_x. The effects of fuel injection timing, injection pressure, spark timing, overall engine Air Fuel Ratio(AFR), and intake air temperature on NO_x emission was examined, and well captured by the model. A comparison of all the NO_x emission data with the model indicates that the NO_x model is a good predictive tool for NO_x emissions in spark ignition direct injection (SIDI) engines.

In this paper, simulation has been developed based on the book titled "Internal Combustion Engine Applied Thermodynamics" by C.R. FERGUSON [7] for studying the effect of the three variables of spark ignition engine on the engine performance and emissions using Zeldovich mechanism. These variables are: first; the temperature of the air surrounded the engine which will be entering into the cylinder, second; ignition timing of the spark, and third; the air to fuel ratio.

MATHEMATICAL MODEL OF THE ENGINE PERFORMANCE AND NO_x EMISSION

Design parameters that substantially affect the value of NO_x are calculated based on the book "Internal Combustion Engine Applied thermodynamics" By C.R.FERGUSON, WILEY & SONS (Ferguson. 1986) [7]. The governing equations of this simulation program for combustion process are applied on the combustion chamber, and the combustion model is done based on the energy conservation equation given by:

$$m\frac{du}{d\theta}+u\frac{dm}{d\theta}=\frac{dQ}{d\theta}-P\frac{dV}{d\theta}-\frac{\dot{m}_i h_i}{\omega} \tag{1}$$

The specific internal energy of the system is given by:

$$u=\frac{U}{m}=xu_b+(1-x)u_u \tag{2}$$

The specific volume of the system is given by:

$$v=\frac{V}{m}=xv_b+(1-x)v_u \tag{3}$$

The relation between v, T, and P is:
For burned gas:

$$v_b=v_b(T_b,P) \tag{4}$$

For unburned gas:

$$v_u=v_u(T_u,P) \tag{5}$$

Differentiating Eqn. (4) & (5) with respect to crank angle:

$$\frac{dv_b}{d\theta}=\frac{\partial v_b}{\partial T_b}\frac{dT_b}{d\theta}+\frac{\partial v_b}{\partial P}\frac{dP}{d\theta} \tag{6}$$

$$\frac{dv_u}{d\theta}=\frac{\partial v_u}{\partial T_u}\frac{dT_u}{d\theta}+\frac{\partial v_u}{\partial P}\frac{dP}{d\theta} \tag{7}$$

From the thermodynamics relations of gausses of piston and cylinder arrangement and after a series of integrations and derivations, we summarize the equations as follows:

$$\alpha=\frac{1}{m}\left(\frac{dV}{d\theta}+\frac{VC}{\omega}\right) \tag{8}$$

μ= $H\frac{\left(\frac{\pi.B^2}{2}+\frac{4V}{B}\right)}{\omega m}\left[\frac{v_b}{cp_b}\frac{\partial \ln v_b}{\partial \ln T_b}\sqrt{x}\frac{T_b-T_w}{T_b}+\frac{v_u}{cp_u}\frac{\partial \ln v_u}{\partial \ln T_u}(1-\sqrt{x})\frac{T_u-T_w}{T_w}\right]$ (9)

γ = $-(v_b-v_u)\frac{dx}{d\theta}-v_b\frac{\partial \ln v_b}{\partial \ln T_b}\frac{h_u-h_b}{cp_bT_b}\left[\frac{dx}{d\theta}-\frac{(x-x^2)C}{\omega}\right]$ (10)

λ = $x\left[\frac{v_b^2}{cp_bT_b}\left(\frac{\partial \ln v_b}{\partial \ln T_b}\right)^2+\frac{v_b}{P}\frac{\partial \ln v_b}{\partial \ln P}\right]$ (11)

ψ = (1-x)$\left[\frac{v_u^2}{cp_uT_u}\left(\frac{\partial \ln v_u}{\partial \ln T_u}\right)^2+\frac{v_u}{P}\frac{\partial \ln v_u}{\partial \ln P}\right]$ (12)

$$\frac{dP}{d\theta}=\frac{\propto+\mu+\gamma}{\lambda+\psi} \quad (13)$$

$$\frac{dT_b}{d\theta}=\frac{-H\left(\frac{\pi B^2}{2}+\frac{4V}{B}\right)\sqrt{x}(T_b+T_w)}{\omega m\, cp_b\, x}+\frac{v_b}{cp_b}\frac{\partial \ln v_b}{\partial \ln T_b}\left(\frac{\propto+\mu+\gamma}{\lambda+\psi}\right)+\frac{h_u+h_b}{X\, cp_b}\left\{\frac{dx}{d\theta}-(x-x^2)\frac{C}{\omega}\right\} \quad (14)$$

$$\frac{dT_u}{d\theta}=\frac{-H\left(\frac{\pi B^2}{2}+\frac{4V}{B}\right)(1-\sqrt{x})(T_u+T_w)}{\omega m\, cp_u\,(1-x)}+\frac{v_u}{cp_u}\frac{\partial \ln v_u}{\partial \ln T_u}\left(\frac{\propto+\mu+\gamma}{\lambda+\psi}\right) \quad (15)$$

$$\frac{dQ}{d\theta}=\frac{H}{\omega}\left(\frac{\pi}{2}B^2+\frac{4V}{B}\right)\left[\sqrt{x}.(T_b-T_u)+(1-\sqrt{x})(T_u-T_w)\right] \quad (16)$$

$$\frac{dh_i}{d\theta}=\frac{Cm}{\omega}\left[(1-x^2)h_u+x^2h_b\right] \quad (17)$$

$$\frac{dW}{d\theta}=P\frac{dV}{d\theta} \quad (18)$$

The heat into the system, the first term on right - hand side of eq. (1):

$$\frac{dQ}{d\theta}=\frac{-\dot{Q}_b-\dot{Q}_u}{\omega} \quad (19)$$

$$\dot{Q}_b=HA_b(T_b-T_w) \quad (20)$$

$$\dot{Q}_u=HA_u(T_u-T_w) \quad (21)$$

$$A_b=\left(\frac{\pi.B^2}{2}+\frac{4V}{B}\right)\sqrt{x} \quad (22)$$

$$A_u=\left(\frac{\pi.B^2}{2}+\frac{4V}{B}\right)(1-\sqrt{x}) \quad (23)$$

The formation of nitric oxide (NO_x):

The formation of NO_x in a piston engine is more complex because it is dependent on a series of reactions such as the Zeldovich mechanism:

$$N + NO \underset{k_1^-}{\overset{k_1^+}{\Leftrightarrow}} N_2 + O \tag{24}$$

$$N + O_2 \underset{k_2^-}{\overset{k_2^+}{\Leftrightarrow}} NO + O \tag{25}$$

$$N + OH \underset{k_3^-}{\overset{k_3^+}{\Leftrightarrow}} NO + H \tag{26}$$

k_i^+, k_i^- are the forward and reverse rate constants.

The balance equation for NO is:

$$\frac{d(NO)}{dt} = 2(1-\beta^2)\frac{S_1}{1+\beta\dfrac{S_1}{S_2+S_3}} \tag{27}$$

Where:

$\beta = \dfrac{[NO]}{[NO]_e}$ ===> NO concentration divided by the equilibrium NO concentration.

S_i is the rate of forward reaction.

$S_1 = k_1^+ [NO]_e [N]_e$

$S_2 = k_2^+ [O_2]_e [N]_e$

$S_3 = k_3^+ [OH]_e [N]_e$

The square brackets *[ ]* means the Concentration, and *[ ]$_e$* is the Equilibrium concentration.

The strong temperature dependence of the NO formation rate can be demonstrated by considering the initial values of d[NO]/dt when $[NO]/[NO]_e << 1$ then from eq. (27)

$$\frac{d[NO]}{dt} = 2S_1 = 2k_1^+[N]_e[NO]_e = 2k_1^-[O]_e[N_2]_e \tag{28}$$

The equilibrium oxygen concentration is given by:

$$[O]_e = \frac{K_{p(O)}[O_2]_e^{\frac{1}{2}}}{(RT)^{\frac{1}{2}}} \tag{29}$$

$K_{p(O)}$is the equilibrium constant for the reaction ==>$\frac{1}{2}O_2 = O$

And given by:

$$K_{p(O)} = 3.6\times10^3 \exp\left(\frac{-31{,}090}{T}\right)P^{\frac{1}{2}} \tag{30}$$

R is the gas constant $\left[\frac{kJ}{kgK}\right]$

P is the pressure in [atm].

T is the temperature in [K]

The initial NO formation rate is:

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{T^{\frac{1}{2}}} exp\left(\frac{-69{,}090}{T}\right)[O_2]_e^{\frac{1}{2}}[N_2]_e \tag{31}$$

The strong dependence of d[NO]/dt on temperature, high temperature, and high oxygen concentrations result in high NO formation rates, then the formation rate of NO is a function of air/fuel ratio. Also, the formation dependent on the time of reactions that means the high flame speed will reduce the NO_x formation, maximum flame speed occurred at the stoichiometric ratio of the mixture $(\phi = 1)$.

RESULTS AND DISCUSSIONS

The internal combustion engines "spark ignition type" are among the main engines used in different fields, such as transportation, and also is one of the most important sources of environmental pollution of air from carbon dioxide (CO_2) and nitrogen oxides(NO_x). Therefore, it is necessary to improve its performance and reduce its polluting emissions, especially toxic ones such as nitrogen oxides(NO_x). In this paper, high-speed four-stroke engine is selected from previous research [1] to study environmental pollution causes from the emission of (NO_x) which emitted from an engine with the following specifications as shown in Table (1):

Table 1: Engine input data parameters

Parameter	**Data**
Compression ratio R	10
Bore B	10 [cm]
Stroke S	8 [cm]
Connecting rode length L	16 [cm]
Engine speed N	2000 [rpm]
Intake Temperature T_i	350 [K]
Wall temperature T_w	420 [K]
Intake pressure P_i	1 [bar]
Combustion duration	60 [Deg]
Equivalence ratio $\emptyset$	0.8
Air Fuel ratio	14.7 kg air/kg fuel
Ignition timing	35 [Deg] BTDC
Fuel	C_8H_{18}

The Expected output of the calculation is the following relations:

1-	Burned gas temperature as a function of crank angle	$(T_b - \theta)$
2-	Unburned gas temperature as a function of crank angle	$(T_u - \theta)$
3-	Average Temperature as a function of crank angle	$(T_{av.} - \theta)$
4-	Pressure as a function of crank angle	$(P - \theta)$
5-	NO_x as a function of crank angle	$(NO_x - \theta)$

Engine performance parameters:

6-	Thermal efficiency	η_{th}
7-	Indicated mean effective pressure	$imep[MPa]$
8-	Total amount of Nitric Oxide per cycle	$NO_X[PPM]$

By using the equations described in the mathematical model section, the results were divided into three parts, the first part: study the effect of the air intake temperature on engine performance and the formation of (NO_x), the intake air temperature that means the air temperature entering into the cylinder, where the air in the cylinder is charged using naturally aspirated. The second part: study the effect of the spark ignition angle on engine performance and (NO_x) formation. The third part: study the effect of air-to-fuel ratio (A/F) on the formation of(NO_x).

Before studying the effect of the intake air temperature, timing ignition, and air to fuel ratio on the NO_x, the engine performance should be identified. The engine data chosen from reference [1] was selected to be entered into the mathematical model program for comparing and verification of the results produced at intake air temperature (Ti=350 K).

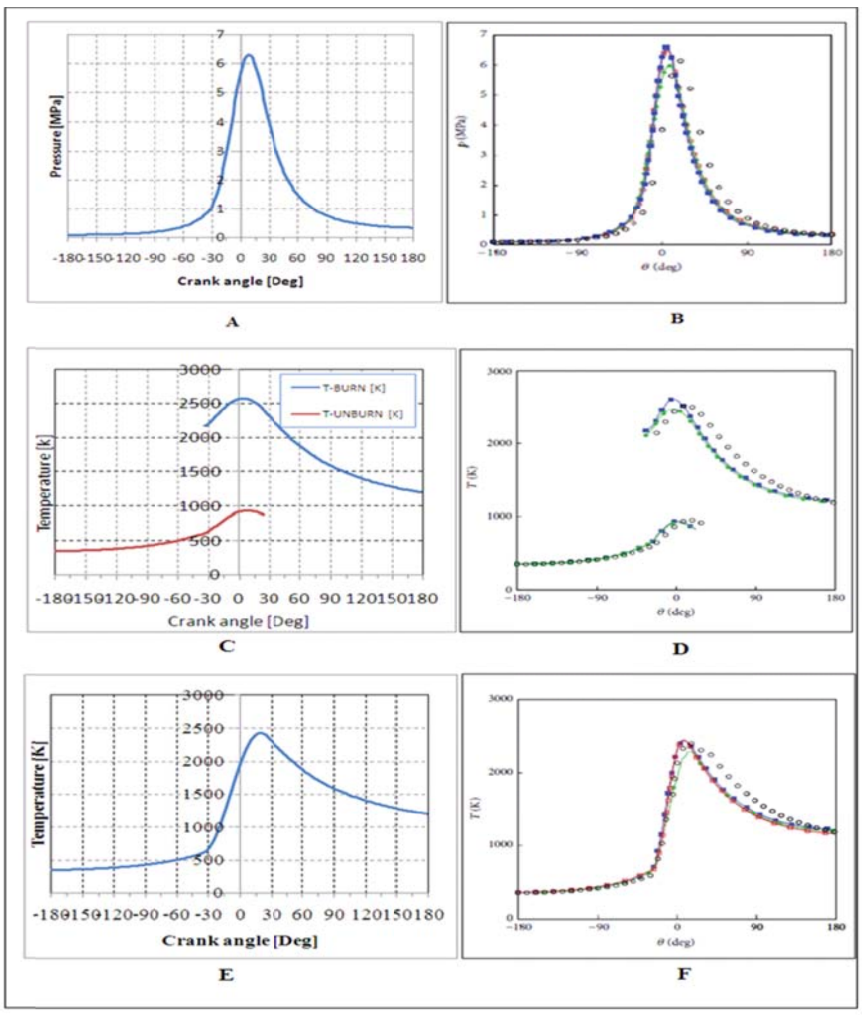

Figure 1: Comparison of Pressure of gases, burned and unburned temperature of gases, and the average temperature of gases in the cylinder as a function of crank angle(θ), left-hand side sub-figures (A, C & E) for this paper and-right hand side sub-figures (B, D & F) for reference [1].

Figure (1) shows a comparison between the results of the engine chosen from reference [1] and the results of the mathematical analysis used in this paper. The sub-figures (A, C, and E) on the left hand side of Figure (1) illustrate the results of this paper, while sub-figures (B, D), and F on the right hand side of Figure (1) illustrates the results in reference [1]. These results compared the relationship of the pressure of gases in the cylinder with the angle of the crankshaft shown in sub-figures (A and B), and the temperature relationship of both burned and unburned gases in the cylinder with the angle of the crankshaft shown in sub-figures (C and D), and the relationship of the average temperature of the gases inside the cylinder with the angle of the crankshaft shown in sub-figures (E and F). Also Figure (1), shows that the results of simulations of reference [1] and this paper simulation are almost the same. Due to the importance of the direct effect of the temperature inside the cylinder on the formation of NO_x emissions, these relationships must be first ascertained it is compatible with previous studies.

A - Effects of the intake air temperatures.

In order to evaluate the performance of any piston internal combustion engine, the indicated mean effective pressure (imep) must be known, through which the thermal efficiency and indicated power of the engine is calculated. As it is known that the indicated power and thermal efficiency are directly proportional to (imep), there are several factors affecting the indicated mean effective pressure, including variable parameters such as the intake air temperature, the air fuel ratio and the timing of the ignition with fixed parameters such as the dimensions of the engine geometry, and compression ratio.

Figure (2) presents the relationship of the effect of the intake temperature of the air on the indicated mean effective pressure (imep). Air temperatures have been changed from 10°C, which is cold to 77°C, which is hot (inlet air temperature of the selected engine from ref [1] Ti=77°C). It is clear from the results, that the air temperature of the environmental has a strong effect on the imep of the engine, where the effect of intake air temperature on the imep is an inversely proportional relationship, the indicated mean effective pressure (imep) increased by 27.5% when the intake air temperature decreased from 77°C to 10°C.

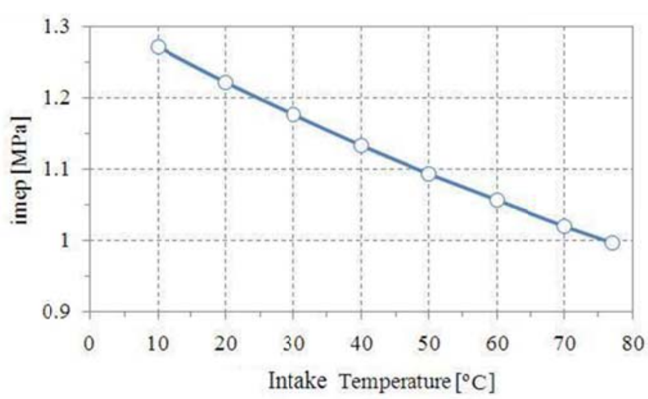

Figure 2: Indicated mean effective pressure (imep) as a function of the intake temperature.

Figure (3) shows the effect of the intake air temperature on thermal efficiency. From results the thermal efficiency at low air entry temperature of 10°C is increased by about 3% compared with air entry temperature of 77°C.

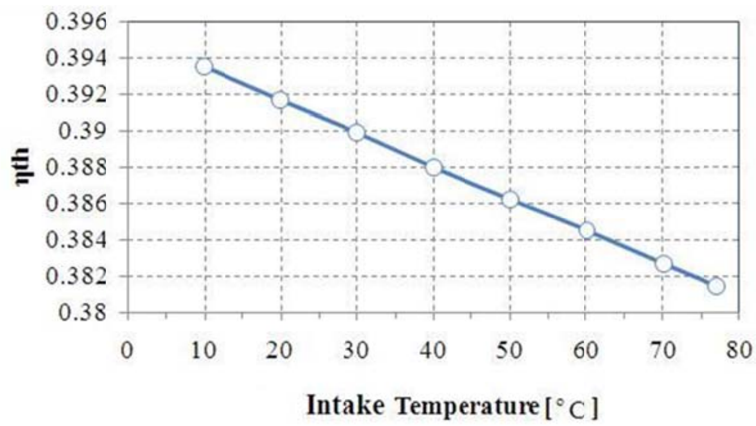

Figure 3: Thermal efficiency (η_{th}) as a function of the intake temperature.

Formation of the NO_x depends on the temperature of the combustion process and is not formed at low temperatures. Therefore, the combustion temperature is the design key to control the amount of Nitrogen Oxides (NO_x) formation, and the temperature of combustion inside the cylinder is very sensitive by changing the intake air temperature into the cylinder. The higher air entry temperature led to a higher combustion temperature in the cylinder, causing NO_x to form in greater quantity.

Figure (4) shows the effect of the intake air temperature on NO_x formation during compression and power strokes, it can be seen that the NO_x formation process starts from the beginning of the ignition time to the maximum temperature of the gas inside the cylinder, which occurs after the top dead center with a very small crank angle, which is about 18 [deg] ATDC in this case.

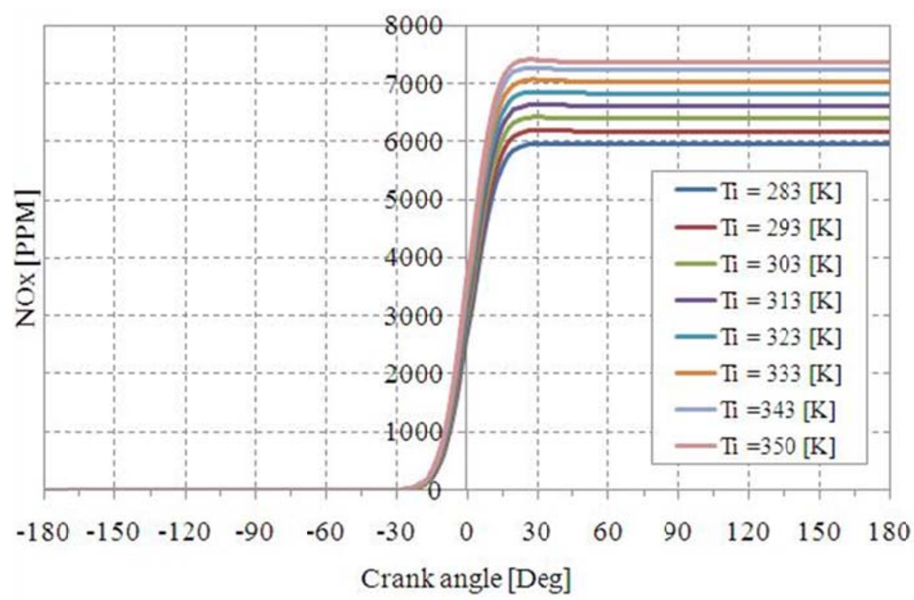

Figure 4: NO_x emission as a function of the crank angle at different intake temperature.

Also, from Figure (4), it can be seen that at the highest air entry temperature (350 K) was formed the largest amount of NO_x, and has been noted that NO_x is not formed before the ignition angle (-35) because the temperatures are very low, only the formation begins after the ignition angle, in the presence of combustion process. Figure (5) shows that the formation of NO_x decreased by 20% when the intake air temperature dropped from 77°C to 10°C.

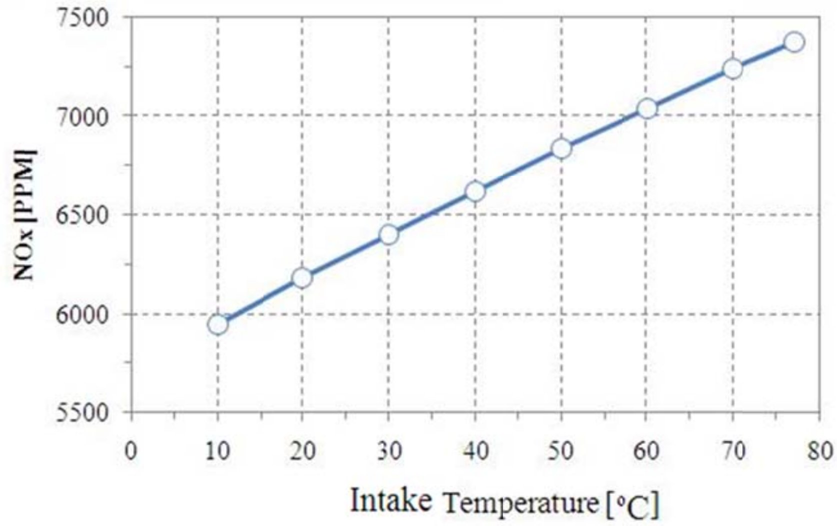

Figure 5: NO_x emission as a function of the intake temperature.

B - Ignition timing effects

Ignition timing is one of the most important parameters in controlling the performance of the engine and its emissions “Specially polluting gases”. In this paper, the effect of the ignition timing on pressure, temperature, and formation of NO_x gases inside the cylinder was studied. The ignition timing angle of the spark ignition engines is very sensitive because it has a direct impact on the performance of the engine, its emissions, and also the maximum pressure (Pmax) of gases inside the cylinder. The timing of the ignition on this type of engines for different conditions of operation such as speed, load and others, are usually between 5 to 20 degrees of crank angle (CA) after top dead center (ATDC). The maximum pressure (Pmax) of the gases inside the cylinder is presented due to excessive delay of ignition timing angle, which causes a drop in the mean effective pressure, and resulting in a decrease in thermal efficiency and indicated power. At very early of the ignition timing that will cause a maximum pressure very close to the top dead centre and sometimes at the TDC, which causes a major problem such as the phenomenon of knocking on the piston and cylinder wall, the excessive temperature rises, with a significant reduction in power and thermal efficiency, and increases in NO_x formation in the emissions, which cause a damage of the engine or reduction of its life. So it’s important to choose the suitable ignition timing for the best engine performance and also less NO_x in the emissions.

Figure (6) shows the pressure inside the cylinder as a function of crank angle at different ignition timing. The combustion time in the cycle is 60 degrees starting from a different ignition angle, in this paper the ignition angle was chosen from 15 to 60 degrees BTDC including the real angle of the engine chosen from the reference [1], which is 35 degrees BTDC. A negative sign that precedes the ignition angle means BTDC in the compression stroke. It is clear from Figure (6) that the pressure increases as the ignition timing is early with a noticeable change in the angle of the maximum pressure occurrence, the results show that, at the ignition angle 60 degrees BTDC the pressure was very high with the occurrence of maximum pressure at the TDC, this is not allowed by the reciprocating internal combustion engines. In the case of excessive ignition angle delay at the crank of 15^o BTDC, it causes low (Pmax) inside the cylinder ATDC, and that will cause decrease in the indicted mean effective pressure (imep) and decrease in thermal efficiency.

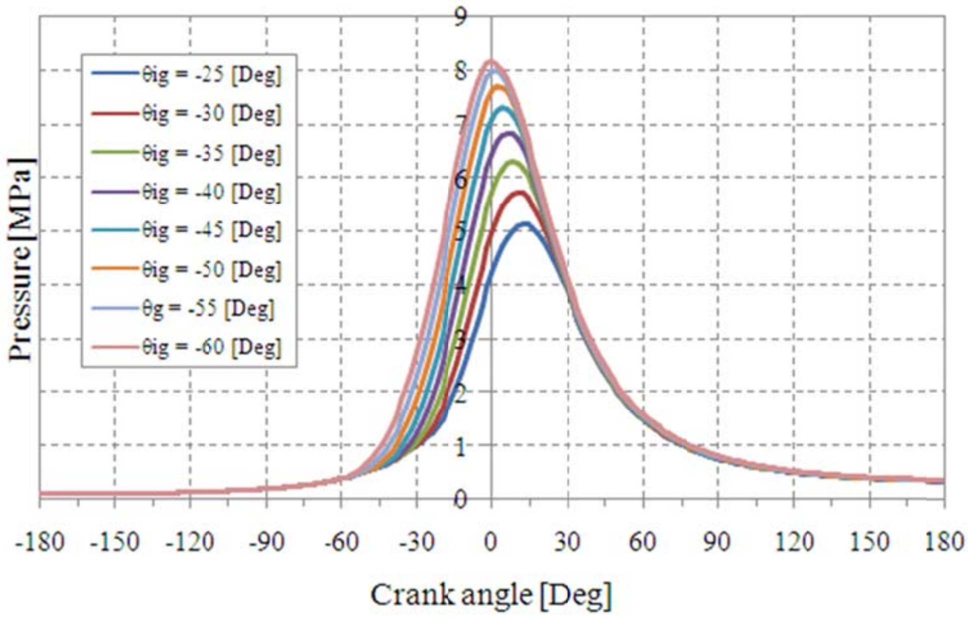

Figure 6: Pressure of the gas in the cylinder as a function of the crank angle at different ignition timing$(\boldsymbol{\theta_{ig}})$.

Figure (7) shows the indicated mean effective pressure (imep) versus the ignition timing, as seen in the figure the maximum imep occurred at the ignition timing of crank angle at 30^o BTDC, at very early ignition angle (60^o) the imep was reduced by 22%, while when the ignition time delay at 15^o BTDC was decreased by 4%. Figure (8) shows the thermal efficiency at different ignition timing angles. Efficiency follows the same behavior as the imep shown in Figure (7). The optimum efficiency was at an ignition timing of 30 degrees BTDC. Figure (9) shows the effect of ignition timing on maximum pressure of the gases inside the cylinder, it's clearly from the figure that the maximum pressure of gases inside the cylinder occurred at the ignition timing at 60^o BTDC of crank angle.

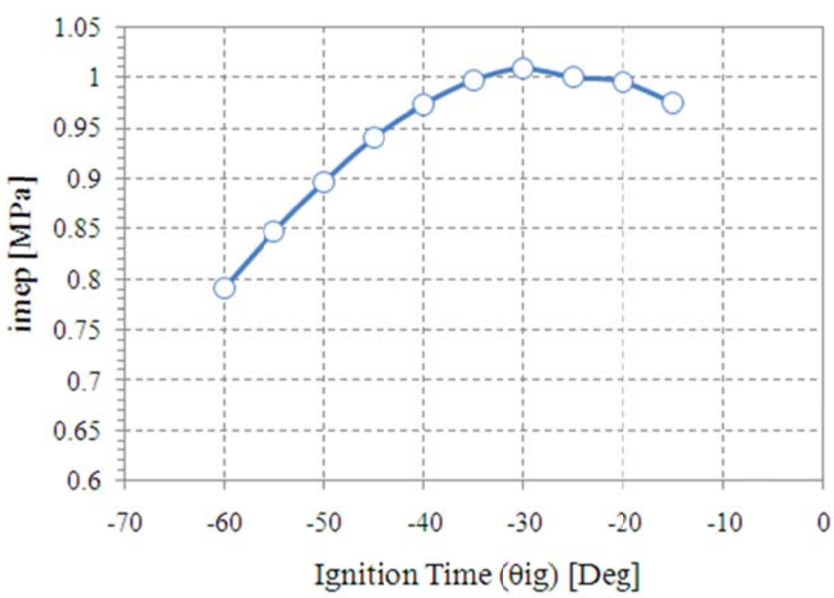

Figure 7: Indicated mean effective pressure (imep) as a function of the ignition timing $(\boldsymbol{\theta_{ig}})$.

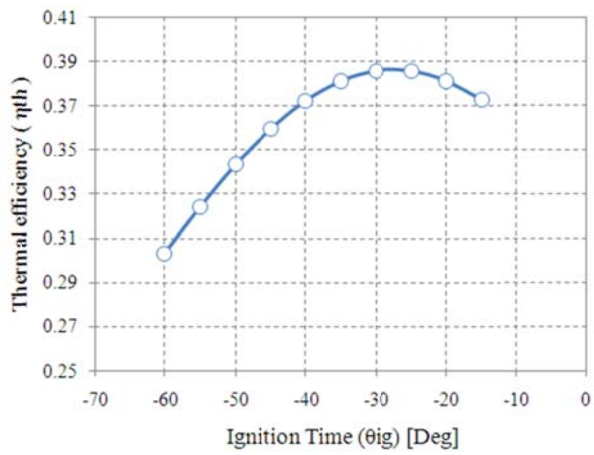

Figure 8: Thermal efficiency (η_{th}) as a function of the ignition timing (θ_{ig}) .

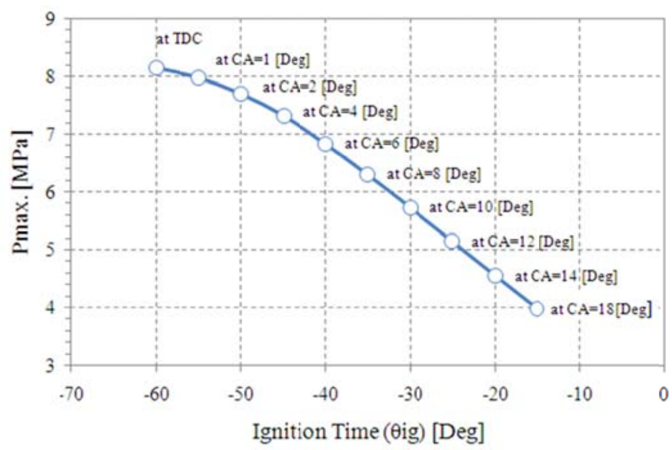

Figure 9: Maximum pressure of the gases inside the cylinder as a function of the ignition timing(θ_{ig}), labeled each value of maximum pressure occurrence at the crank angle (CA).

Figure (10) presents the relationship between the average temperature of gases inside the cylinder versus the crank angle at different ignition Timing. The results show a rise in temperature at the early ignition timing, at an early ignition angle (60°).

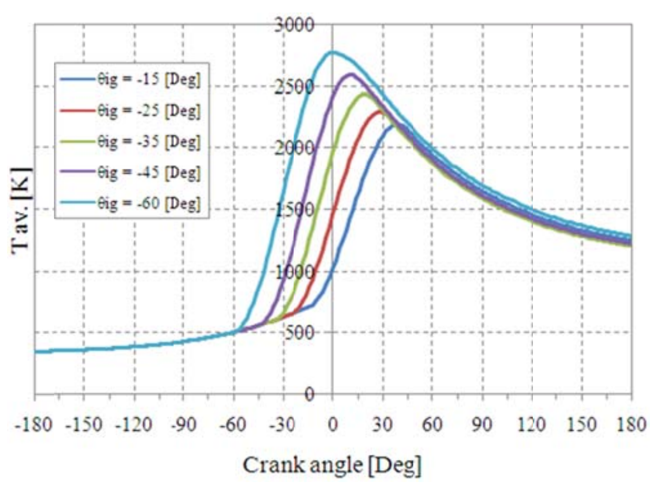

Figure 10: Average temperature of the gases inside the cylinder as a function of crank angle (θ)at different ignition timing(θ_{ig}).

Figure (11) shows that the maximum temperature in the cycle occurrence at TDC, and causing a maximum temperature range from 2213 [K] at the crank angle of 38 degrees ATDC to 2770 [K] at TDC. This range in temperature gives an opportunity to form NO_x in larger quantities. Figure (11) shows the maximum temperature in the cycle at different angles of the ignition timing, also showing the location of occurrence on the crank angle.

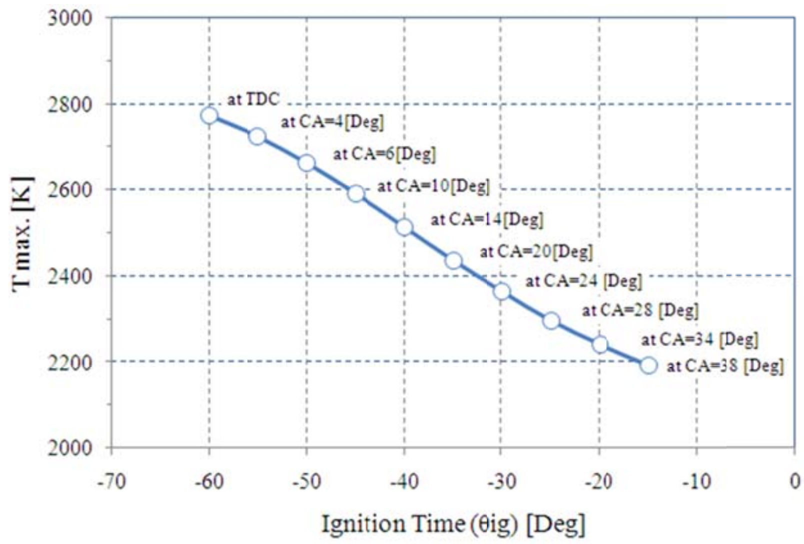

Figure 11: Maximum average temperature of the gas in the cylinder as a function of the ignition timing(θ_{ig}), labeled each value of maximum temperature occurrence at the crank angle (CA).

Figure (12) shows the amount of NO_x as a function of crank angle at different ignition timing. Due to the high average temperatures of gases inside the cylinder as an early ignition angle, it caused an increase in the formation of larger amounts of nitrogen oxides.

Figure (13) shows the NO_x at a different ignition time. It is clear from the results that the relationship between NO_x formation and ignition time is not linear, NO_x increased by 86% at ignition timing from 15 degrees to 60 degrees BTDC.

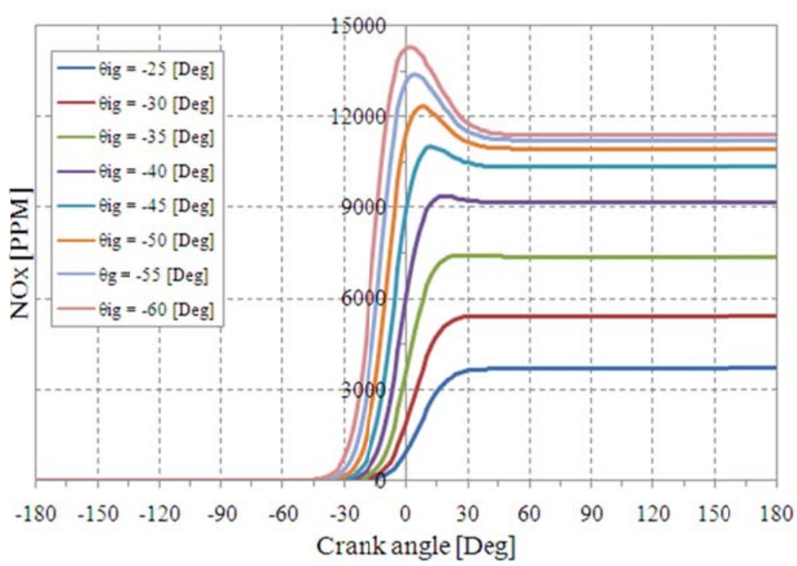

Figure 12: NO_x emission as a function of the crank angle at different ignition timing(θ_{ig}).

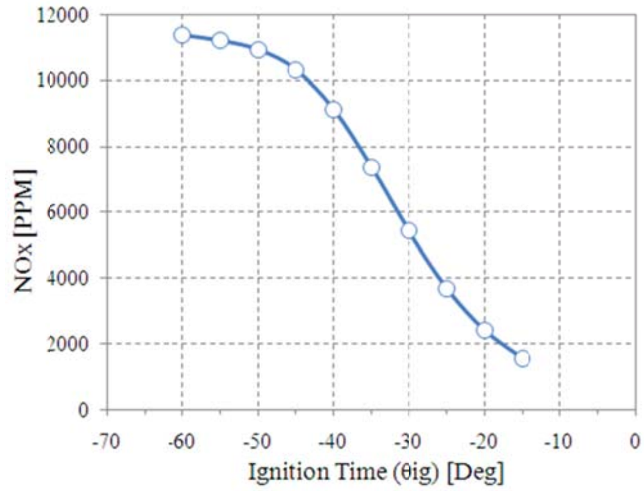

Figure 13: NO_x emission as a function of the ignition timing(θ_{ig}).

C - Air to Fuel ratio effects

The air-to-fuel ratio is a key to the design of any hydrocarbon-fueled heat engine, as it has a strong impact on engine performance and environmental pollution. To increase thermal efficiency, the case required to reduce specific fuel consumption, and environmental pollution, the engine must be operated at a very lean mixture as possible as you can, including the reciprocating spark ignition engine. In this part of the results, the effect of air-fuel ratio on engine performance and also on NO_x formation is studied, where the equivalence ratio $(\emptyset)$ of air and fuel was changed from the stoichiometric equivalence ratio $(\emptyset = 1)$ to a very lean mixture at equivalence ratio of 0.6, the specifications of the engine chosen from the reference [1] are as follows: $(\emptyset = 0.8)$, Air-to-Fuel ratio is 14.7, and octane fuel type (C_8H_{18}).

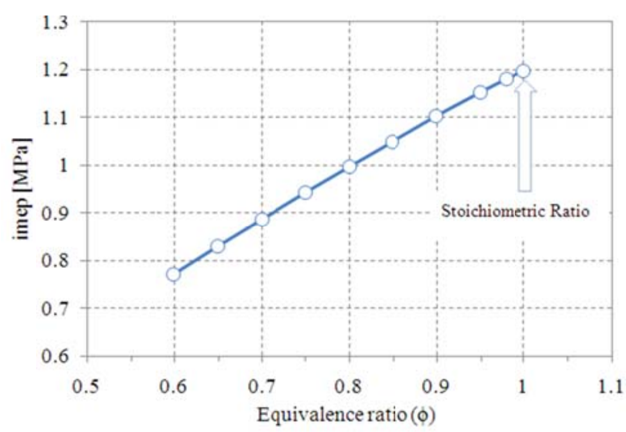

Figure 14: Indicated mean effective pressure (imep) as a function of the equivalence ratio(ϕ).

Figure (14) shows the effect of air-fuel ratio on the indicated mean effective pressure (imep) inside the cylinder, is directly proportionally relationship between the equivalence ratio $(\emptyset)$ and imep of the engine, maximum imep at stoichiometric equivalence ratio $(\emptyset = 1)$, considering that the maximum temperature of combustion is at stoichiometric equivalence ratio, resulting in higher pressure and greater power. The indicated mean effective pressure increased by 55% when the equivalence ratio $(\emptyset)$ increased from 0.6 to 1.0, this increase because of the more amount of fuel in the mixture, but the thermal efficiency of the engine decrease to the minimum values when the fuel in the mixture increases to the stoichiometric ratio, Figure (15) shows the reduction in thermal efficiency when the engine is operated at stoichiometric ratio, where the thermal efficiency decreased by 6% compared to that at $(\emptyset = 0.6)$.

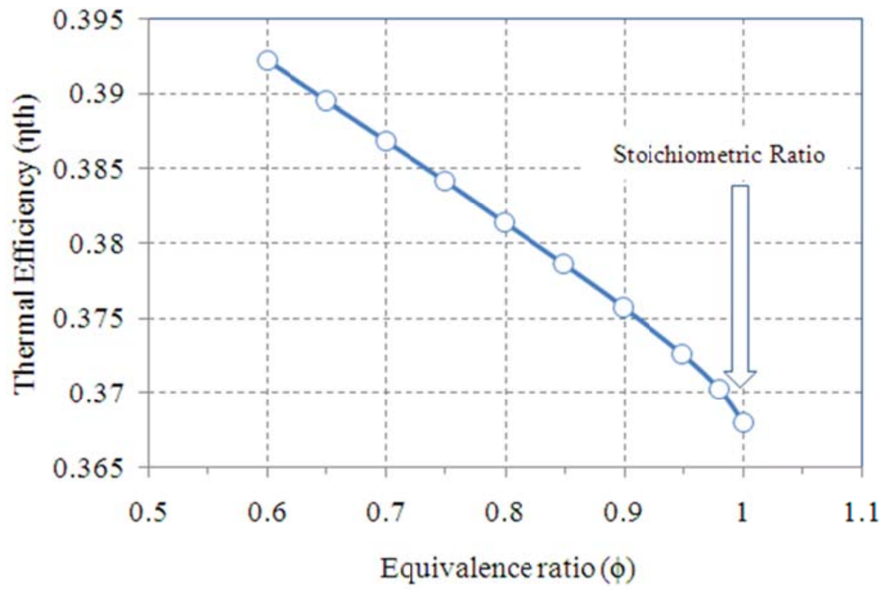

Figure 15: Thermal efficiency (η_{th}) as a function of the equivalence ratio (ϕ).

Figure (16) illustrates the relationship of the average temperature of gases inside the cylinder with the crank angle for different equivalence ratios$(\emptyset)$. The results show that the maximum temperature of gases inside the cylinder recorded when engine operation at the stoichiometric equivalence ratio$(\emptyset = 1)$, and decreases to its lower levels when the engine operates at very lean mixture. The maximum temperature dropped to 2113 [K] at $(\emptyset = 0.6)$ compared to 2651 [K] at stoichiometric equivalence ratio$(\emptyset = 1)$, as shown in Figure (17) .

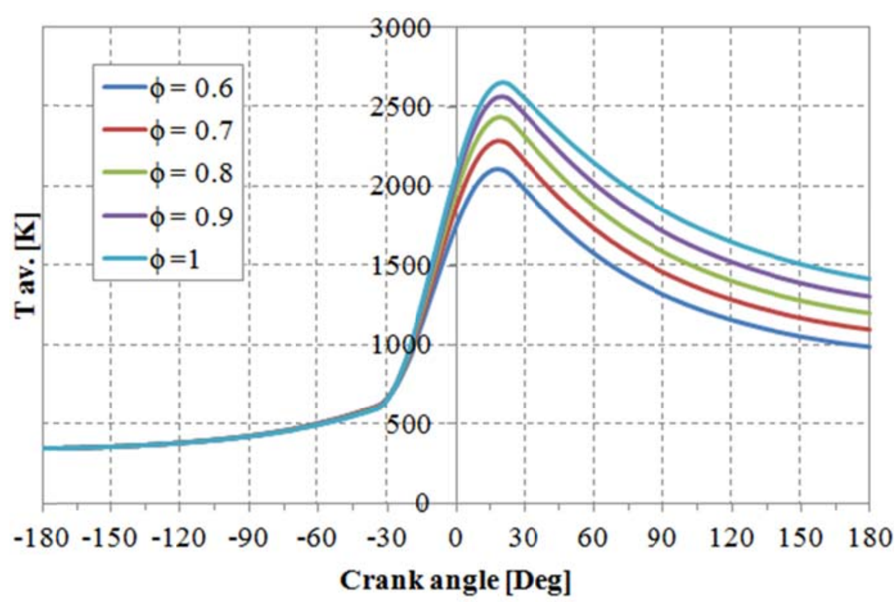

Figure 16: Average temperature of the gas in the cylinder as a function of crank angle at different equivalence ratio(ϕ).

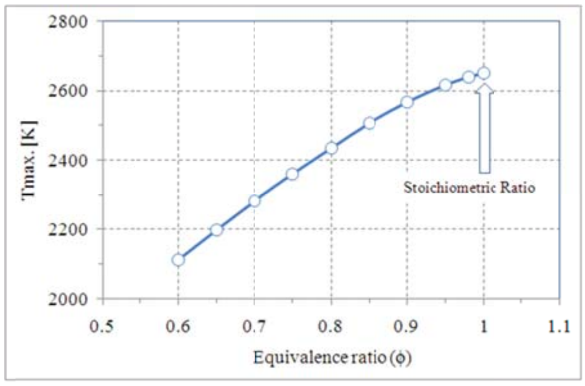

Figure 17: Maximum average temperature of the gases inside the cylinder as a function of the equivalence ratio(ϕ). All maximum values of temperature occurrence at 18 [Deg} ATDC.

The formation of nitrogen oxides at different equivalence ratios ($\emptyset$)is shown in Figures (18) and Figure (19). The formation of NO_x was based on NO_x model consists of a chemical mechanism of the classical extended Zeldovich mechanism. The results show the strong effect of air-to-fuel ratio on NO_x formation, as a result of the reduction of the gases temperature inside the cylinder when the engine is operated by a equivalence ratios ($\emptyset$) for a very lean mixture, resulting in a very strong reduction in NO_x quantities formed. Also the amount of NO_x decreases slightly as the mixture approaches the stoichiometric equivalence ratio, that was observed in Figure (19), the amount of NO_x reduced when operating at a slightly lower than the stoichiometric equivalence ratio, although the maximum temperature is recorded in this ratio, and the reason for this: because the speed of flame at this ratio is very high, i.e. the combustion time will be very small, it is known that the formation of NO_x needs enough time to complete the chemical reactions, longer time gives greater chance in formation of NO_x, also as a result of the very high temperature closed to the stoichiometric equivalence ratio causes the dissociation of nitrogen oxide and its transformation into another composite of gases, as seen in Figure (18), when operating the engine at very close to the stoichiometric equivalence ratio a peak can be seen and then decreases slightly, meaning that, the amount of NO_x has dissociated causing it to decrease. The maximum NO_x amount was at $\emptyset = 0.85$, where NO_x decreased by 96% at $\emptyset = 0.6$

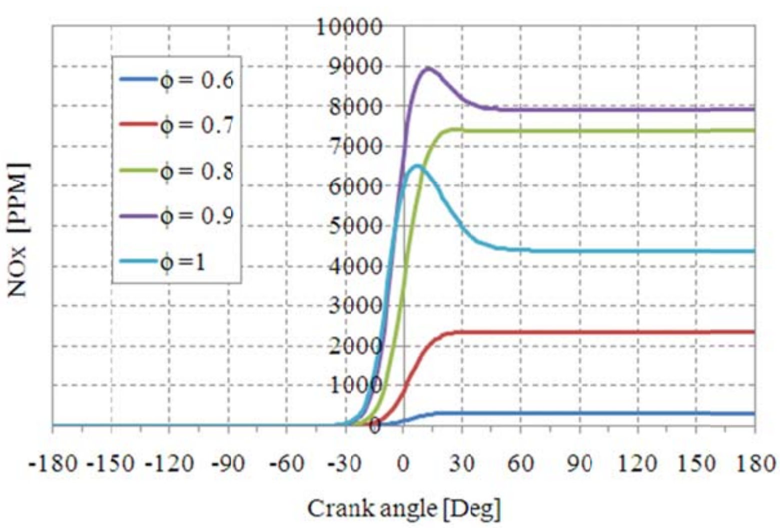

Figure 18: NO_x emission as a function of the crank angle at different equivalence ratio(ϕ).

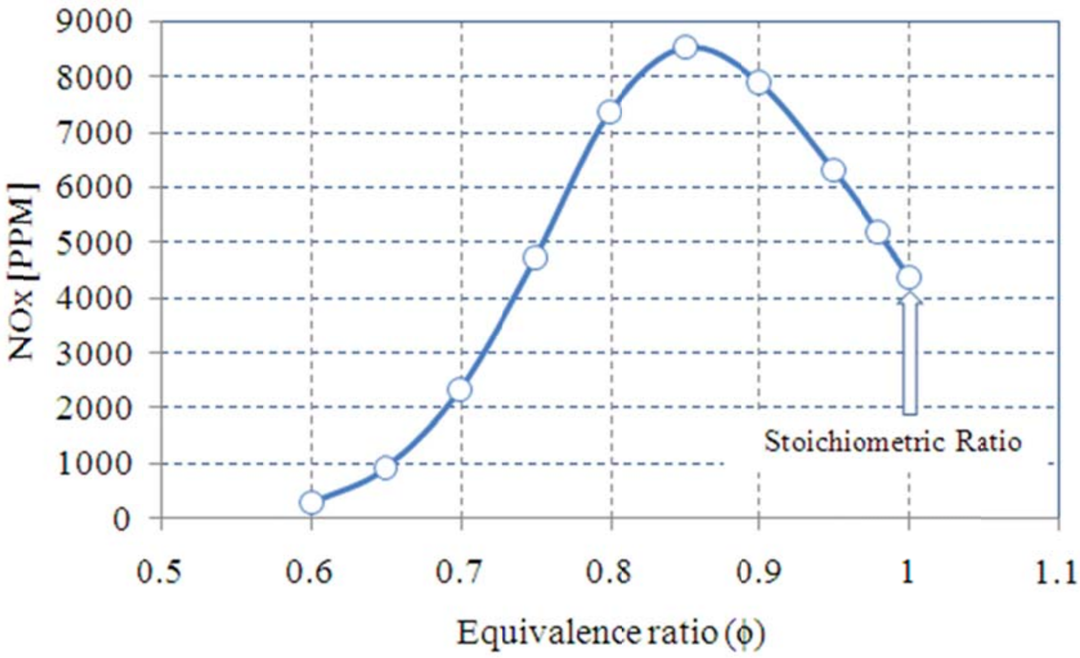

Figure 19: NO_x emission as a function of the equivalence ratio(ϕ).

CONCLUSIONS

- The results showed that the environment temperature of the air has a strong effect on the reduction of nitrogen oxide (NO_x) Formation, when the air temperature of the environment decreases as its entering into the engine cylinder from 350 [K] to a temperature of 283 [K], nitrogen oxide decreased by 20%.
- The effect of the spark ignition angle has a significant effect on the formation of NO_x due to high combustion temperature in the cylinder. Here in this case an early angle may not be provided resulting in maximum pressure at the top dead center (TDC) which causes a decrease in thermal efficiency, a high temperature, and a decrease in power. Also the ignition angle delay causes a drop in the maximum pressure inside the cylinder at a very late angle after the top dead center (ATDC), which also causes a decrease in the thermal efficiency of the engine, and also a decrease in power, the maximum pressure occurred between the angle of the crank shaft from 5 to 15 degrees after the top dead center (ATDC). In these results under these conditions for the permitted ignition angle from -20 to -40 before the top dead center(BTDC), the formation of NO_x decreased by 74%.
- Effect of air-fuel ratio: The formation of NO_x decreased significantly when the engine operated at very lean mixture, because of low combustion temperature during the combustion process. It is known that the temperature of combustion is as high as possible at the stoichiometric equivalence ratio ($\phi = 1.0$), but the results show that the maximum amount of NO_x was at the equivalence ratio of 0.85, and did not occur at equivalence ratio of 1.0 (stoichiometric ratio), because the formation of NO_x takes time through a series of reactions. At stoichiometric ratio($\phi = 1.0$), the flame speed is very high, which does not give a chance to produce NO_x in greater quantity, and also due to dissociation at very high temperature of nitrogen oxide and its transformation into another composite of

gases. Nitrogen oxides formation is reduced by 96% for equivalence ratio from $0.85\ to\ 0.6$.

NOMENCLATURE

mi	mass in the cylinder [kg]	v_b	specific volume of burned gas $\left[\frac{m^3}{kg}\right]$
u	specific internal energy $\left[\frac{kJ}{kg}\right]$	v_u	specific volume of unburned gas $\left[\frac{m^3}{kg}\right]$
θ	crank angle [Deg]	T_b	burned gas temperature [K]
P	Pressure [Pa].	T_u	unburned gas temperature [K]
V	cylinder volume $[m^3]$	W	Work [J]
Q	Heat loss [kJ].	C_{pb}	specific heat at constant pressure of burned gas $\left[\frac{kJ}{kgK}\right]$
ω	engine frequency $\left[\frac{rad}{sec}\right]$	C_{pu}	specific heat at constant pressure of unburned gas $\left[\frac{kJ}{kgK}\right]$
x	mass fraction.	C	Blow-by coefficient
u_b	specific internal energy of burned gas $\left[\frac{kJ}{kg}\right]$	H	heat transfer coefficient by convection
u_u	specific internal energy of Unburned gas $\left[\frac{kJ}{kg}\right]$	A_b	surface area of burned gas in contact with the cylinder wall $[m^2]$
β	concentration ratio	A_u	surface area of unburned gas in contact with the cylinder. wall $[m^2]$
Kp	equilibrium constant for the reaction	T_w	wall temperature [K]
S_i	Rate of forward reaction.		

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