SIMULATION MODEL FOR SPARK IGNITION ENGINE FUELED WITH NATURAL GAS

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الملخص

أدى استخدام الغاز الطبيعي كوقود لمحركات الاحتراق الداخلي رباعية الاشواط التي تعمل بالشرارة إلى تقليل انبعاثات الملوثات مثل أكاسيد النيتروجين والهيدروكربون الغير محترق واكاسيد الكربون والمواد العضوية الغير محترقة إذا ما قورنت بالوقود الهيدروكربوني التقليدي، كما أن ارتفاع رقم الاوكتان العالي للغاز الطبيعي يسمح بزيادة نسبة الانضغاط في محركات الإشعال بالشرارة التي تعمل بالغاز الطبيعي وبالتالي تحسن كفاءتها.

تعد هذه الورقة امتداد للنموذج الذي قدمه Ferguson [1] والذي ناقش تأثير نسب الانضغاط على أداء وكفاءة المحرك وايضا تأثير نسب الهواء الى الوقود وتأثير سرعة المحرك على الانبعاثات وعلى كفاءة محرك الاحتراق الداخلي رباعي الاشواط يعمل بالشرارة باستخدام وقود الغاز الطبيعي. ولتأكيد صحة نتائج الدراسة قدر الامكان وبعيدة عن الفرضيات التي قد تكون بعيدة على الواقع تم أخذ نتائج معملية لمحرك رباعي الاشواط يعمل بالشرارة وإجراء محاكاة له بواسطة برنامج فورتران وكانت النتائج جيدة جدا ومتقاربة حيث تم اخذ نسب انضغاط 8:1، 10:1 و12:1 وسرعة محرك 1200 و1800 دورة في الدقيقة وقد تمت الدراسة بتغيير نسب الوقود الى الهواء بنسب مختلفة.

تشير نتائج الدراسة الى انه كلما زادت نسب الانضغاط زادت الكفاءة وإن نسبة الانضغاط 12:1 أفضل نسبة انضغاط لمحرك رباعي الاشواط يعمل بالغاز الطبيعي

ABSTRACT

Using natural gas as a fuel for internal combustion engine (spark ignition) results in reduction of pollutant emission such as NO_x, *HC*, *CO* and *SOOT* if compared with conventional hydrocarbon fuel in addition the high-octane number of natural gas allows an increase of compression ratio in spark ignition engine and consequently improves their efficiency.

In this paper an extension of the model presented by Ferguson [1], is applied using *FORTRAN* language. A comprehensive performance data is presented for equivalence ratios, compression ratios, and engine speed. The contributions were to develop and validate a computer code simulation for 4-stroke cycle SI engine by using different fuel and compare the results with previous experimental studies. The results of the computer code are in very good agreement with the experimental studies under various conditions.

Experiments showed that a compression ratio of 12:1 is a reasonable value for a compressed-natural-gas direct-injection engine to obtain a better thermal efficiency without a large penalty of emissions.

KEYWORDS: Air-Fuel Model; Thermodynamic Models; Spark Ignition Engines; Natural Gas.

INTRODUCTION

The depletion of the world oil reserves together with the environmental pollution, and global warming problems caused by large-scale use of fossil-fuels are deriving interest in alternative fuels for automotive engines. Alcohols and gaseous fuels are two categories of alternative fuels that have received much consideration, Alcohols such as ethanol and methanol can be produced from renewable bio-resources and give less polluting exhausts. Gaseous fuels, such as natural gas and liquefied petroleum gas, offer cleaner combustion due to improved fuel-air mixture preparation and higher hydrogen to carbon ratios than in conventional liquid fuels [2]. Although the suitability of these types of alternative fuels has been demonstrated by many research groups [3, 5], the drive to convert to an alternative fuel has been stronger in some countries than others depending on the availability of the alternative fuel at a competitive cost.

Natural gas is thought to be one of the most promising alternatives to traditional vehicle fuels for engines since it has cleaner combustion characteristics and plentiful reserves. Natural gas (*NG*) is widely used in taxis and city buses all over the world and the natural-gas-fuelled engine has been realized in both the spark ignition engine and the compression ignition engine. Furthermore, its high octane value and good anti-knock property permits a high compression ratio (*R)* leading, of course, to higher thermal efficiency in the high-load condition. Previous studies showed low emissions by using natural gas. Engines fuelled with natural gas emit less carbon monoxide (*CO*) and non-unburned hydrocarbons (*HCs*) compared with gasoline engines [4, 6].

Nowadays, there are mainly two kinds of operating mode for engines fuelled with natural gas in actual applications (or during routine operation). In the first operating mode, the homogeneous natural gas is ignited by pilot injection of the diesel fuel before the top dead centre (*TDC*). This needs two separate fuel systems and makes the system complicated. Meanwhile, *HC* emissions still remain high for light loads. In the second mode, the homogeneous mixture of natural gas and air is ignited by a spark plug as in the traditional homogeneous gasoline spark ignition engine. As natural gas occupies some fraction of intake charge, it has the disadvantage of low volumetric efficiency, and this decreases the amount of fresh air into the cylinder, leading to a decreased power output compared with that of a gasoline engine. The homogeneous charge combustion makes it difficult to burn the lean mixture.

These engines have a lower thermal efficiency because engine knock is avoided and because of the unavoidable throttling at the intake for a partial load [7]. So-called homogeneous lean combustion engines have appeared. They can realize a higher thermal efficiency owing to the lower pumping loss, the lower heat loss, and the increase in the specific heat ratio, at the expense of the moderately higher nitrogen oxide (*NOx*) emissions due to the ineffectiveness of the existing catalyst. The large cycle-by-cycle variation, however, restricts the lean operation limit of this type of homogeneous mixture engine [8, 9].

Thus, the aim of this work is to develop a simulation of operating characteristics on a spark ignition engine using gaseous fuel i.e. compressed natural gas and hydrogen and compares the results obtained with NG. Spark ignition internal combustion engine operates on either a four stroke or a two-stroke cycle. This work only focuses on four-stroke spark ignition engine cycle. Thermodynamic analysis will be done on a four-stroke spark ignition engine. With this thermodynamic knowledge as a basis, study will be extended to determine the performance of an engine. The studies of engine performance will emphasis on pressure-crank angle, effect of speed, compression ratio and equivalence ratio on engine performance.

This paper is an attempt to shed some light on the effect of combustion duration and compression ratio and engine speed (which was varied by varying the equivalence ratio) on engine performance using analytical model. The analytical model developed was tested and verified against experimental data of several engines. It was used to study the effect of various operating parameters on the combustion duration as well as the effect of combustion duration on the engine performance and emission characteristics to try to get a better understanding of the interaction between these parameters.

MATHEMATICAL FORMULATION OF THE MODEL

The principal governing equations of the model are the mass and energy conservation relations, the equations of state, and the second-law of thermodynamics. In the mathematical formulation of governing equations the crank angle is taken as the independent variable. Thus, the differential form of the energy conservation equation (the first-law of thermodynamics) applied to an open system encasing the cylinder contents are [1, 10]:

$$m\frac{du}{d\theta}+u\frac{dm}{d\theta}=\frac{dQ}{d\theta}-P\frac{dV}{d\theta}-\frac{\dot{m}_l h_l}{\omega} \quad (1)$$

Where (***θ*** is the crank angle, ***m*** is the mass, ***u*** is the internal energy, ***Q*** is the net heat added to the system, ***P*** is the pressure, ***V*** is the volume and ***ω*** is the rotational speed). in the last term of the equation $\boldsymbol{m_l}$ stands for the instantaneous leakage or blow-by rate. Early in the combustion process, unburned gas leaks past the rings while late in the combustion process, burnt gas leaks past the rings.

The model assumes the leakage to be always out of the cylinder and taking with it gas characterized by the enthalpy $\boldsymbol{h_l}$ of the cylinder contents (which in turn depends on the temperature, pressure, and composition of the fluid). A two-zone combustion model is adopted whereby the combustion chamber is divided into two zones containing unburned gases and burned gases [10, 11]. Accordingly, the specific internal energy of the system ***u*** for spark-ignition engine assumed to be given by:

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

where (***x*** is the mass fraction of the cylinder contents that have been burned, $\boldsymbol{u_b}$ is the specific internal energy of the burnt gas that is at a temperature $\boldsymbol{T_b}$ and $\boldsymbol{u_u}$ is the specific internal energy of the unburnt gas at a temperature $\boldsymbol{T_u}$. (the classical two-zone model), The mass fraction ***x*** is determined from an empirical burning law, such as:

$$x=\begin{cases}0 & \theta<\theta_s\\ \frac{1}{2}\left\{1-\cos\left[\frac{\pi(\theta-\theta_s)}{\theta_b}\right]\right\} & \theta_s<\theta<\theta_s\\ 1 & \theta>\theta_s+\theta_b\end{cases} \quad (3)$$

Likewise, the specific volume of the system ***v*** is expressed as:

$$v=\frac{V}{m}=xv_b+(1-x)v_u \quad (4)$$

The enthalpy of the mass loss due to blow-by: Early in the combustion process, unburned gas leaks past the rings. Late in the combustion process, burnt gas leaks past the rings it is assumed that:

$$h_l = (1 - x^2)h_u + x^2 h_b \qquad (5)$$

Unlike standard gas cycles, the model takes into consideration the relative timing of the heat addition by using an empirical relationship that expresses the fraction of the heat added at any time to the crank angle. Empirical relationships are also used to express the terms involving $\frac{\boldsymbol{dm}}{\boldsymbol{d\theta}}$ and $\frac{\boldsymbol{dQ}}{\boldsymbol{d\theta}}$ in (1), the energy equation (1) is thus seen to be a relationship among three parameters and their derivatives, i.e. the equation can be put in the form:

$$f\left(\theta, \frac{dP}{d\theta}, \frac{dT_b}{d\theta}, \frac{dT_u}{d\theta}, P, T_b, T_u\right) = 0 \qquad (6)$$

Therefore, two more equations are needed to complete the mathematical formulation of model. One of the requisite equation is derived by differentiating (2) for the specific volume of the system. The second requisite equation comes from introduction of the un-burnt gas entropy into the analysis. The three equations are then rearranged in the standard form used to numerically integrate a set of ordinary differential equations (*ODEs*):

$$\frac{d\xi_i}{d_\theta} = f_i(\theta, P, T_b, T_u) \qquad where\ i = 1,2,3 \qquad (7)$$

Where ($\boldsymbol{\xi_1}$, $\boldsymbol{\xi_2}$, $\boldsymbol{\xi_3}$ refer to $\boldsymbol{P, T_b, T_u}$, respectively). The three equations are supplemented by three other equations for the work done, the heat loss, and the enthalpy loss. The model then consists of a set of six ordinary differential equations describing the rates of change of six parameters with respect to crank angle. By simultaneously integrating these equations from the start of compression until the end of expansion, the indicated efficiency and the indicated mean effective pressure can be determined.

The model needs thermodynamic properties of the combustion reactants and products at different stages of the engine's cycle. To evaluate these properties, the following formulae are used [9, 11]:

$$\frac{c_p}{R} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 \qquad (8a)$$

$$\frac{h}{RT} = a_1 + \frac{a_2}{2}T + \frac{a_3}{3}T^2 + \frac{a_4}{4}T^3 \frac{a_5}{5}T^4 + a_6\frac{1}{T} \qquad (8b)$$

$$\frac{s}{R} = a_1 \ln T + a_2T + \frac{a_3}{2}T^2 + \frac{a_4}{3}T^3 + \frac{a_5}{4}T^4 + a_7 \qquad (8c)$$

Where ($\boldsymbol{c_p}$ is the specific heat at constant pressure, $\boldsymbol{R}$ is the gas constant, $\boldsymbol{T}$ is the temperature in Kelvin, $\boldsymbol{h}$ is the specific enthalpy and $\boldsymbol{s}$ is the specific entropy). The coefficients $\boldsymbol{a_6}$ in the enthalpy equation and $\boldsymbol{a_7}$ in entropy equation are constants resulting from the relevant integration of (8a) [10, 11]. To evaluate the thermodynamic properties of fuels (in vapor phase) simplified versions of Eqs. (8) is used in which both $\boldsymbol{a_4}$ and $\boldsymbol{a_5}$ are set to zero.

THE COMPUTER PROGRAM

The engine operating conditions chosen for the FORTRAN code simulation with the purpose of comparing it against the experimental data are dedicated i.e. 1200 and 1800 rpm, respectively. The engine operating conditions consider certain variations in the intake temperature, injection timing, and injection duration and spark ignition timing as listed in Table (1)

Table 1: Experimental engine specifications [15]

Item	Specification
Engine type	4 stroke (Ricardo)
Number of cylinders	1
Bore (mm)	100
Stroke (mm)	115
Length connecting rod (mm)	190
Compression ratio	8,10,12
Engine speed (rpm)	1200,1800
Displacement (l)	0.903
Combustion chamber	Bowl-in-shape
Ignition source	Spark plug
Initial pressure (kPa)	99
Residual mass fraction	0.05

Ferguson [1] provided FORTRAN computer program for air-fuel models of both fuel-inducted and fuel-injected engines. For the solution of the system of *ODEs*, he adopted the subroutine DVERK from the (IMSL) package which uses fifth and sixth order Runge-Kutta-Verner method. In order to model the *SI* engine cycle, the thermodynamic zero-dimensional, two zones model was developed. Such a model belongs to the second generation of the engine models and was fully developed by Heywood [10], Ferguson [1] and others researchers [12].

This model was extended with the NOx production Zeldovich model [14]. A program called ZINOX was developed and proved in this work. As shown on Figure (1), the models allow single or multi runs to be performed. If a single-cycle simulation is chosen, the model will proceed to integrate the *ODEs* starting from a crank angle of -180^o until 180^o to obtain the variation of the cycle parameters with crank-angle over the complete cycle. At the end of the cycle the simulation also gives the values of four overall parameters which are the indicated thermal efficiency (η), the indicated mean effective pressure (*IMEP*), the work (*W*), *NOx* emission (*NOx*), the error in the conservation of mass (Error 1), and error in the conservation of energy (Error 2).

The results are stored in a normal text file. The multi-cycle option gives the variation of the eight overall parameters (η, *IMEP*, W, *NOx* emission ... ,Error 1, and Error 2), If this simulation is selected, the model does the cycle integration for each value of the selected parameter all stores the values of the parameters for the cycle in a second text file. The input data to the *ZINOX* -1 engine model are given below:

Following the specification of the fuel and engine properties, the model may be triggered to run the required simulation mode by pressing the "enter" button. The results can then be plotted with Grapher Golden software. The code input data for a natural gas as fuel shown in Figure (1). The other Figures show the variation of the pressure, work, temperature, and heat leakage with the crank angle.

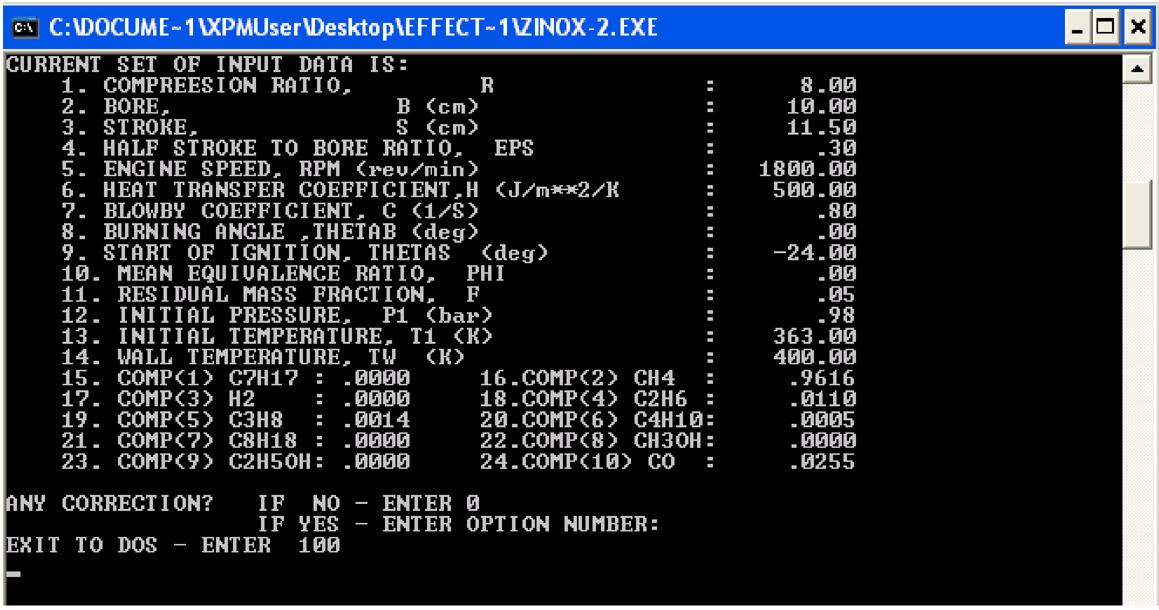

Figure 1: Engine specifications and operating conditions

Results and Discussions

The results obtained from the thermodynamic simulation model are presented. The simulate Pressure, Temperature, Work, Heat Transfer and Total combustion duration for a single cylinder; four stroke SI engine running on natural gas.

Figures (2a – 2b) show the maximum cylinder pressure versus the compression ratio *R*. The maximum cylinder pressure increases with increasing compression ratio R. Two reasons are responsible for this behaviour. One is the rises in the unburned gas temperature and pressure with increasing compression ratio *R*. The other is the compact heat release process on increasing the compression ratio *R*. Both factors favour an increase in the maximum cylinder pressure.

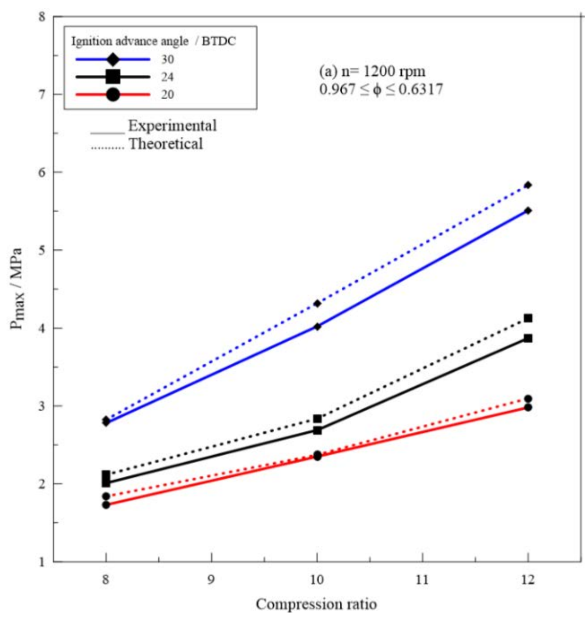

Figure 2a: Effect of compression ratio on Maximum cylinder gas pressure (1200 rpm).

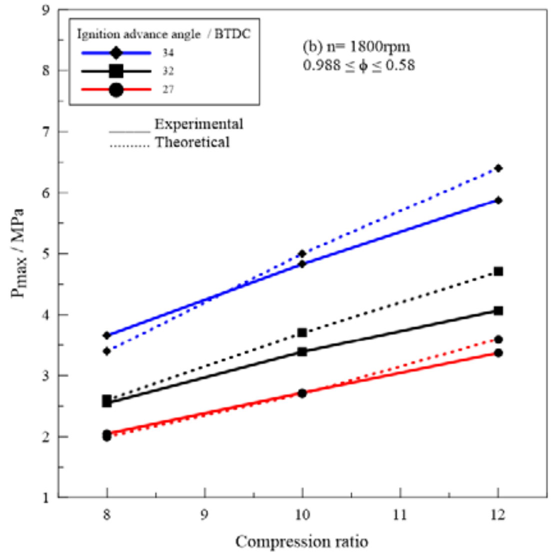

Figure 2b: Effect of compression ratio on Maximum cylinder gas pressure (1800 rpm).

Figures (3a – 3b) give the total combustion duration versus the compression ratio R. The total combustion duration decreased with increase in the compression ratio R. This is reasonable because both flame development duration and rapid combustion duration decreased on increasing the compression ratio R.

The flame development duration is decreased with increase in the compression ratio *R*. Moreover, the stratified mixture will increase the burning velocity and high turbulence is presented at high compression ratios Rs, all these contribute to a decrease in the flame development duration as the compression ratio R is increased.

The increase in the unburned gas temperature and turbulence as well as mixture stratification contributes to the decrease in the rapid combustion duration on increasing the R. Similarly, the decrease in the residual gas fraction on increasing the R also increases the flame propagation speed

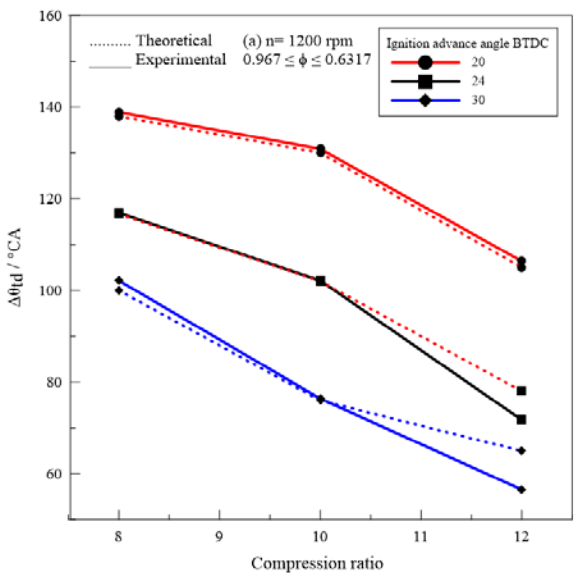

Figure 3a: Effect of compression ratio on combustion total duration (1200 rpm).

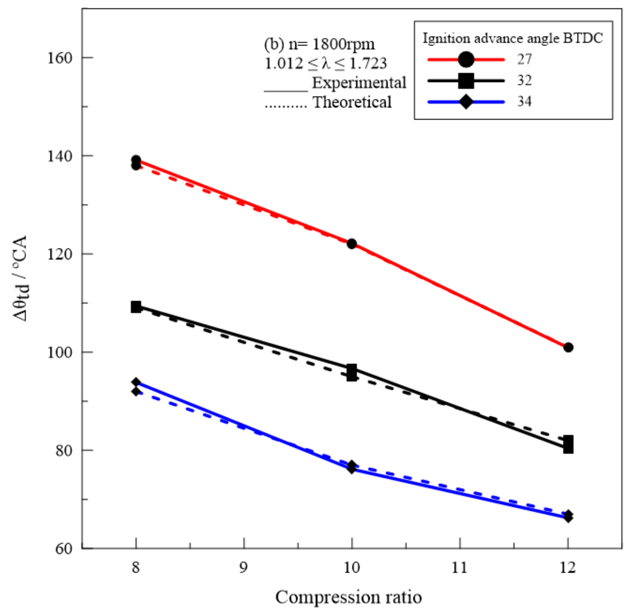

Figure 3b: Effect of compression ratio on combustion total duration (1800 rpm).

Figures (4a – 4b) show the distributions of temperatures of burned and unburned mixture during compression and combustion strokes. Ignition starts at -20 to -34 degrees before TDC, when the combustion model starts to work in the burned zone. Calculations in the unburned zone continue without addition of heat due to combustion. Temperature profiles corresponding to the burned and unburned mixtures are overlapped during the combustion interval.

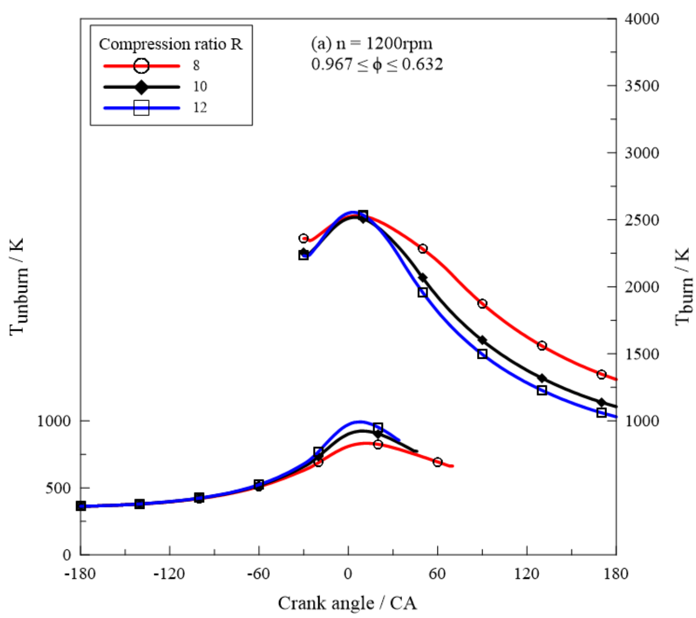

Figure 4a: Effect of compression ratio on T burn and T unburn (1200 rpm).

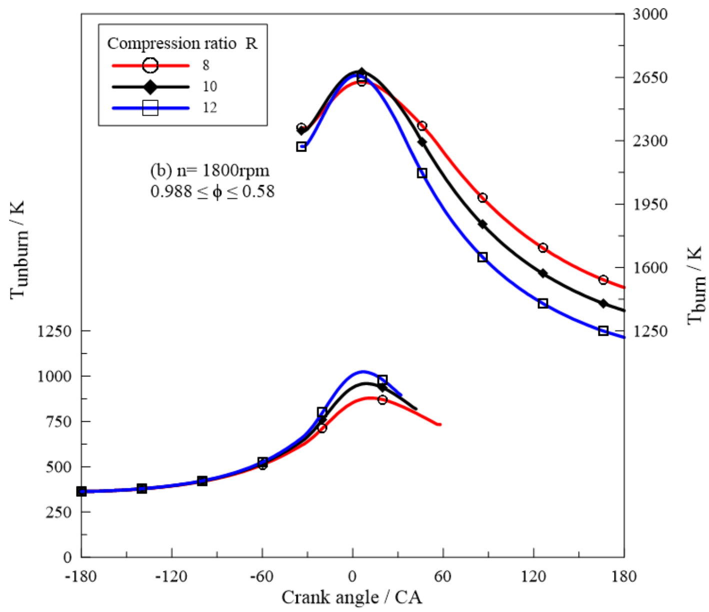

Figure 4b: Effect of compression ratio on T burn and T unburn (1800 rpm).

Similar to the behaviour of the maximum cylinder pressure, the maximum mean gas temperature also presents an increasing trend on increasing the compression ratio R. This is reasonable since the temperature at the moment of ignition increases with increasing R. The increase in the burning velocity with increasing R will also contribute to the increase in the maximum cylinder gas temperature. The results support the fact that the heat transfer increases with increasing R since the engine will undergo higher heat transfer at higher gas temperatures.

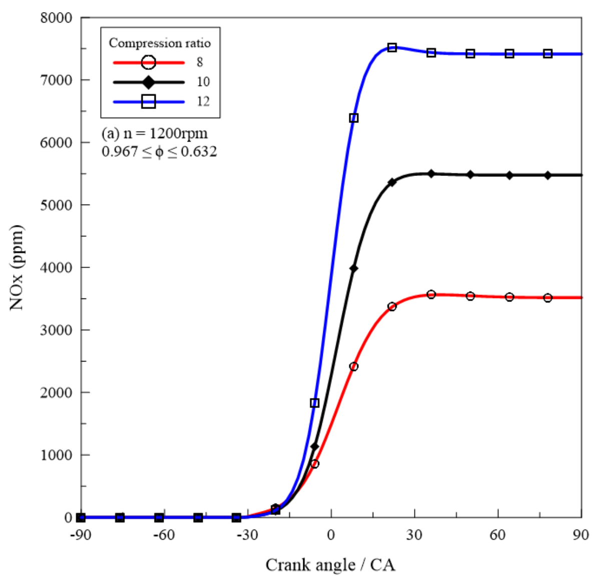

Figure 5a: Effect of compression ratio on *NOx* (1200rpm)

Figures (5a - 5b) give the exhaust *NOx* concentration versus the crank angle. At low and middle engine loads, the *NOx* concentration has a low value and it does not vary on increasing the *R*. The lean mixture combustion results in this behavior. However, at high engine loads, the *NOx* concentration shows an increasing trend with increasing *R*. The relatively high burning rate of the overall rich mixture combustion and a high-temperature environment contribute to the increase in the *NOx* emissions with increasing R at high engine loads. The *NOx* concentration increases with increase in the R at fixed ignition timings and *NOx* emissions show an increasing trend and then a decreasing trend for MBT timing.

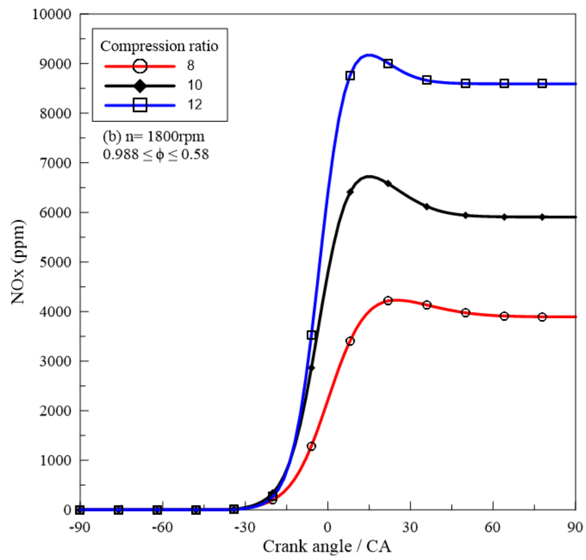

Figure 5b: Effect of compression ratio on *NOx* (1800rpm)

Figure (6) for Work, it is the same effect as torque just only interest in field of speed. It is clearly that the power is increased to the maximum power around 1800 rpm and then slightly decreases at every compression ratios. For the effect of engine speed, it has not directly affected on the performance of engine but it increases the circulation rate of fuel higher when increasing speed and changes in ignition timing at different speed which has the effect to the performance of the engine.

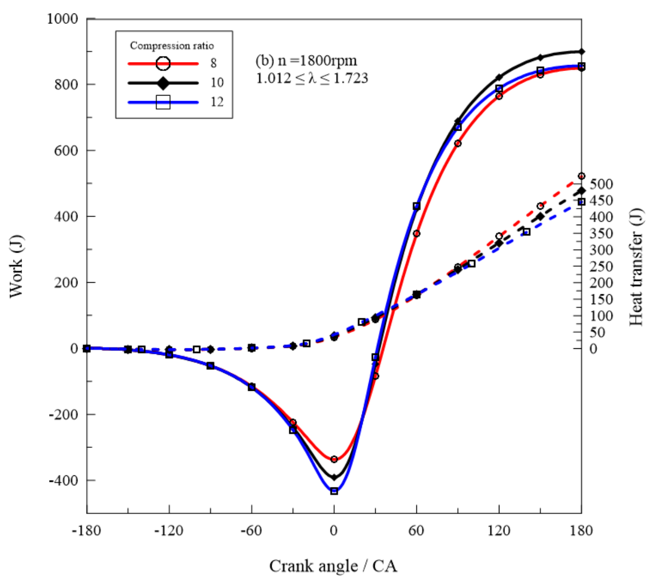

Figure 6: Effect of compression ratio on work and Heat transfer (1800rpm)

We also note the cumulative work is initially negative due to the piston compression and becomes positive on the expansion stroke. The heat transfer loss is very small during compression, indicating a nearly isentropic compression process, and is somewhat linear during the expansion process.

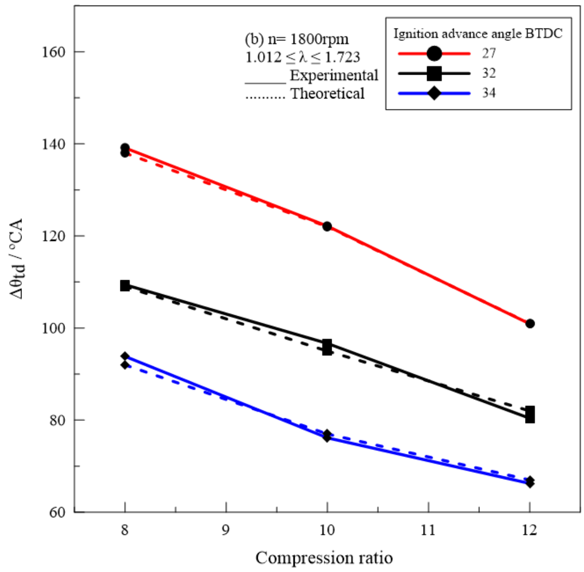

Figure 7: Effect of compression ratio on thermal efficiency (1200 and 1800rpm)

The thermal efficiency η versus the compression ratio *R* for various engine speeds are shown in Figure (7). The thermal efficiency increases with increase in the R at high engine speed. However, the thermal efficiency is increased with increasing R. The following reasons are considered to influence the brake thermal efficiency from the variation in the compression ratio (*R).*

1. Increasing the *R* will increase the cylinder gas pressure, temperature, and mixture concentration at the end of the compression stroke. This speeds up the chemical reactions, resulting in increases in the burning rate and the thermal efficiency.
2. The expansion ratio is increased with increase in the *R*, and this decreases the exhaust gas temperature and reduces the energy carried away by the exhaust gases, leading to an increase in the thermal efficiency
3. Increasing the *R* increases the cylinder gas temperature at the end of the compression stroke and decreases the exhaust gas temperature during the late expansion and exhaust stroke. This will extend the temperature difference between high temperature and low-temperature cycles, resulting in an increase in thermal efficiency [10].
4. With increasing *R*, the clearance volume is decreased. The reduction in the clearance volume also improves the combustion, resulting in an increase in the flame temperature, which will increase the thermal efficiency.

CONCLUSIONS

From the comparison between Zheng et al [15] experimental results and the present computer code Zinox-1 calculations for the same engine parameters with compression ratio 8:1, 10:1 and 12:1, excess air ratio in the range of 1.012 to 1.723, engine speed 1200 to 1800 and variations in ignition timing from (20 to 34) BTDC, Zinox-1 model results are slightly different and this is probably caused by the following:

- In the computer model limited different fuel compositions used than those of Huang et al [16] fuel options which are shown in Figure (1) this might be one of the possible explanations of the difference between this calculation and his experimental results.
- The results achieved here are a little bit differ from those of Zheng et al [15] because he used special measurement instruments, which are very accurate and specific for his experimental research engine (direct injection engine made by Hitachi). However, the Constants considered in this study the computer model (thermodynamic data for elements and combustion products) shown in Figure (1) may have some effect on the results.
- Regarding NOx emissions, the model gives the most predictions of NOx emissions, which shows the correct trends.
- The results show that the compression ratio has a large influence on the engine performance, combustion, and emissions. Increased of maximum pressure the penetration distance of the natural gas jet is decreased and relatively strong mixture stratification is formed as the compression ratio is increased, giving a fast burning rate and a high thermal efficiency, especially at low and medium engine loads. Experiments showed that a compression ratio of 12:1 is a reasonable value for a compressed natural gas direct injection engine to obtain a better thermal efficiency without a large penalty of emissions.

This paper confirms that the computer code Zinox-1 is a reliable program for the calculations of maximum pressure, efficiency and NOx emission of any engine.

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NOMENCLATURE

A	Area
C_p	Specific heat at constant pressure
C_v	Specific heat at constant volume
E	Total energy
n	Engine speed
NG	Natural gas
p	Cylinder fluid pressure
P_{max}	Maximum cylinder gas pressure
$\dot{q}$	Heat flux rate
T	Temperature
T_b	Temperature of burned particles
T_{max}	Max maximum mean gas temperature

T_u	Temperature of un-burned particles
V	Volume of Combustion Chamber
v	Specific volume
U	Internal energy
u	Specific internal energy
u_u	Specific internal energy of un-burned particles
u_b	Specific internal energy of burned particles
W	Work done
x_b	Mass fraction
λ	Relative air: fuel ratio
ATDC	After top dead centre
BTDC	Before top dead centre
CO	Carbon Monoxide
R	Compression ratio
ºCA	Degree crank angle
IC	Internal combustion
NO_x	Nitric oxide, nitrous oxide and nitrogen dioxide
SI	Spark Ignition
rpm	Revolutions per minute.
HC	unburned hydrocarbon