SIMULATION OF THE COMPRESSION RATIO EFFECTS ON THE COMBUSTION CHARACTERISTICS AND PERFORMANCE OF A SPARK IGNITION ENGINE +

Rafeq A.Khalefa *

Abstract:

This study investigates the effects of compression ratio variations (7, 7.5, 8, 8.5 and 9) on the spark ignition engine performance (indicated power ,indicated thermal efficiency ,indicated specific fuel consumption and heat losses) and combustion characteristics (ignition delay ,turbulent flame speed and mass burning) by simulation method for the speeds of (1800, 2400 and 3300) rpm with a spark timing of (-20, -25 and -32.5) deg. for each speed respectively .

The results showed that the compression ratio has a great influence on some parameters of combustion characteristics and engine performance. Where any increase in this ratio increases most combustion characteristics except mass burning rate which is affected only by engine speed and spark timing ,while the power, efficiency and (I.S.F.C) are improved significantly , with a little increasing of heat losses through the combustion chamber walls .

Key words : SI engines ,performance ,compression ratio ,simulation, combustion characteristics .

محاكاة تأثير نسبة الانضغاط على خصائص الاحتراق وعلى أداء محرك إشعال بالقدح

رفيق احمد خليفة

المستخلص :

يتضمن هذا البحث دراسة تأثيرات تغيير نسبة الانضغاط للنسب (7, 7.5, 8, 8.5 , 9) على أداء محركات الإشعال بالقدح (القدرة البيانية ،الكفاءة الحرارية البيانية ،استهلاك الوقود النوعي البياني والخسائر الحرارية) و على خصائص الاحتراق (فترة تأخر الإشعال ، معدل احتراق الوقود وسرعة اللهب الاضطرابية) للسرعات (1800، 2400 و 3300) دورة/دقيقة مع توقيتات الشرارة (20- و 25- و 32.5-) درجة على التوالي. وأظهرت النتائج إن نسبة الانضغاط تؤثر بشكل كبير على بعض خصائص الاحتراق وعلى أداء محركات الإشعال بالقدح حيث إن أي زيادة في هذه النسبة تؤدي إلى الزيادة في خصائص الاحتراق المذكورة أعلاه ماعدا معدل احتراق الوقود التي تتأثر بالسرعة الدورانية للمحرك وتوقيت الشرارة ، بينما نجد تحسنا ملحوظا في القدرة والكفاءة واستهلاك الوقود بزيادة النسبة مع زيادة بسيطة في الخسائر الحرارية من خلال جدران غرفة الاحتراق.

الكلمات الدالة:/ محركات الإشعال بالقدح ، أداء المحركات ، محاكاة المحرك ، نسبة الانضغاط، خصائص الاحتراق.

+ Received on 27/10/2014, Accepted on 23/8/2016 .

* Lecturer/ Technical College/Kirkuk

Abbreviations:

A_s Instantaneous Heat Transfer Area (m^2)
A/F Air/Fuel Ratio
CR Compression Ratio
D Cylinder Bore (m)
$dQ_{Conv.}$ Heat Losses During the Step (kJ)
$dQ_{release.}$: Rate of Heat Release During the Step(kJ)
dw Work Done During the Step (kJ)
e Specific Internal Energy
E Absolute Internal Energy
$E(T_1)$ Internal Energy at The Beginning of the Step
$E(T_2)$ Internal Energy at The End of the Step
F/A Fuel/Air Ratio
H.H.V High Heating Value (kJ/kg)
h Convection Heat Transfer Coefficient ($kW/m^2.k$)
L Connecting Rod Length (m)
L.H.V Low Heating Value (kJ/kg)
$\dot{m}_f$ The Fuel Consumption (kg/s)
m_m Mass of Mixture (kg)
m_b Mass burned
m_{un} Un burned mass
N Engine Speed (R.P.M)
P_1 Cylinder pressure at the Beginning of the Step (kPa)
P_2 Cylinder pressure at the End of the Step (kPa)
$R_{mol.}$ Gas Constant (kJ/kg.mole.k)
R_f Flame Radius (m)
S Stroke Length (m)
T_1 Cylinder Temperature at the Beginning of the Step
T_2 Cylinder Temperature at the End of the Step
T_b Burned Zone Temperature (k)
T_{un} Unburned Zone Temperature (k)
t Time (sec)
$U_{i,j}$ Polynomial Coefficients
U_L Laminar Flame Front Speed (cm/sec)
U_T Turbulent Flame Front Speed (cm/sec)
V_1 Volume of the Cylinder at the Beginning of the Step (m^3)
V_2 Volume of the Cylinder at the End of the Step (m^3)
U_P Piston Speed (m/sec)
X_r Mass Fraction of the Burned Gasses.
θ_{Sp} Spark Ignition Angle (degree)
α_1 the angle at step1
α_2 the angle at step2

Introduction:

Simulation method is an effective tool which is used at a wide range in a development and designing process for a lot of applications, because this method is very fast , simple, more economic , and lesser pollutants. The internal combustion engines are one of these applications. The empirical methods for building ,designing and testing of engines and systems are very difficult because these methods need more time and high costs ,due to this fact and due to lots of uncontrollable parameters, such as physical geometries, ignition advanced, valve timing and combustion characteristic, influencing the performance, the development of spark ignition engines for the last 100 years were very slow [1]. This method has been used for predicting the effects of variation of valve lift [2], and (Bore/Stroke) ratio [3] on the engine performance , It was found that the results achieved by simulation method show reasonable agreement when all results when compared with experimental method .

The simulation method is based on the thermodynamic analysis of the cylinder content during the engine cycle . In this mathematical model, the 1st law of thermodynamics is applied and by the solution of mathematical model, change in performance is researched as each of the engine parameters changed as wanted, it is possible to determine the engine characteristics. The effect of friction on the spark ignition engine performance using a gas mixture model performed and the results show that it is more realistic to consider the effect of friction especially when running at high speed and rich mixture [4]. The peak pressure is higher with higher compression ratio. Increase in compression ratio leads to high heat release rate. [5]. An experimental and theoretical investigation of the influence of the compression ratio on the brake power, brake thermal efficiency, brake mean effective pressure and specific fuel consumption of the (Ricardo) variable compression ratio spark ignition engine was performed and the results showed that as the compression ratio increases, the actual fuel consumption decreases , brake thermal efficiency improves and brake power also improves and the small values of the percentage errors between the theoretical and experimental values show that there is agreement between the theoretical and experimental performance characteristics of the engine combustion characteristics [6].

The experimental study showed that the brake thermal efficiency increases with increasing the compression ratio and brake specific fuel consumption decreases with increasing compression ratio[7].

In this study the compression ratios (7, 7.5, 8, 8.5 and 9) and engine speeds of (1800, 2400 and 3300) rpm were investigated, with choosing the corresponding spark timing of (-20, -25 and -32.5) respectively as in real engines with stoichiomatric A/F ratio. The objective of this research is to develop a computer program to simulate the full thermodynamic cycle of a spark ignition engine and predict the effects of compression ratio on the combustion characteristics (ignition delay period , fuel mass burn rate, combustion duration and turbulent flame speed) and engine performance parameters (power, thermal efficiency, specific fuel consumption and heat loss), for a spark ignition engine whose technical specification as in table (1) .

Table (1) Technical specification of the S.I. engine and Fuel Used [2]

Cylinder number	**1**
Cylinder diameter (B)	**95mm**
Stroke (S)	**85mm**
Connecting rod length (L)	**129.8mm**
Compression Ratio	**8**
Intake valve closing	**22^o aBDC**
Exhaust valve opening	**22^o bBDC**
Fuel Type (ISO-Octane)	**C_8H_{18}**
Heating Value	**43000 kJ/kg**

Theoretical Analysts :

After the trapped conditions are established, the cycle calculation is carried out according to the following steps:

1- The equations of volume and area that relate to crank angle are described as following equation [8]:

$$V(\theta)=\frac{V_d}{CR-1}+\frac{V_d}{2}[\frac{L}{a}+1-\cos\theta-(\{\frac{L}{a}\}^2-\sin^2\theta)^{\frac{1}{2}}] \qquad (1)$$

$$A(\theta)=\frac{\pi}{2}b^2+\pi b\frac{S}{2}[\frac{L}{a}+1-\cos\theta+(\{\frac{L}{a}\}^2-\sin^2\theta)^{\frac{1}{2}}] \qquad (2)$$

Where

$V(\theta)$ = Instantaneous Cylinder Volume, V_d = Displacement Volume, a =Crank offset

$A(\theta)$ = Instantaneous Cylinder Area depending on piston position .

2-Estimating the temperature and pressure of the cylinder content at the end of the step (the step was 2 degrees of crank angle) using [9]:

$$T_2=T_1*[\frac{V_1}{V_2}]^{k-1}=T_1*[\frac{V_1}{V_2}]^{\frac{R_{mol}}{C_V(T_1)}} \qquad (3)$$

$$P_2=[\frac{V_1}{V_2}]*[\frac{T_2}{T_1}]*P_1 \qquad (4)$$

3-The work done is calculated [10] by:

$$dW = P\,dV = (\frac{P_1+P_2}{2})\,(V_2-V_1) \qquad (5)$$

4-The first law of thermodynamics is simply written as:

$$\delta Q-\delta W=dU \qquad (6)$$

Therefore, by using the definition of work, the first law can be expressed as [11]:

$$\delta Q_{in}-\delta Q_{loss}-(PdV)=dU \qquad (7)$$

For an ideal gas the equation of state is expressed as :

$$PV= m\,R_g\,T_g \qquad (8)$$

Where

R_g= gas constant (kJ/kg.K) & T_g= Cylinder Gas Temperature

By differentiating the equation (8) we can get:
$PdV + VdP = m R_g dT_g$ (9)
Also for an ideal gas with constant specific heats the change in internal energy is expressed as [11]: $dU = mC_v dT_g$ (10)
By substituting equation (10) into (9) then
$dU = (C_v/R_g)(PdV+VdP)$ (11)
By substituting equation (11) into (7), following equation is obtained:
$\delta Q_{in} - \delta Q_{loss} - (PdV) = (C_v/R_g)(PdV+VdP)$ (12)

5-The total amount of heat input to the cylinder by combustion of fuel in one cycle is:
$Q_{in} = m_f LHV$ (13)
The total heat added from the fuel to the system until the crank angle position reaches angle θ is given as[2]: $Q_{(\theta)} = Q_{in} X_b$ (14)
Where X_b is the Weibe function that used to determine the combustion rate of the fuel and is expressed as[1,11]:

$$X_b = 1 - \exp\{-5(\frac{\theta - \theta_S}{\Delta\theta})^3\} \quad (15)$$

6-The heat loss to the wall is given by [1]: $\delta Q_{loss} = h\delta A(T_g - T_{wall})$ (16)
By substituting equations (16) and (14) into equation (12) the pressure over crank angle now becomes:

$$\frac{dP}{d\theta} = \frac{k-1}{V}[Q_{in} X_b \frac{hA}{6N}(T_g - T_{wall})] - k\frac{P}{V}\frac{dV}{d\theta} \quad (17)$$

Where N = Engine Speed (rpm) , k = Specific Heat Ratio and h is the heat transfer coefficient, the Eichelberg Equation is selected to calculating its value in this work which is [10]:
$h = 2.466 * 10^{-4} (UP)^{1/3} (P)^{1/2} (T_g)^{1/2}$ (18)
where $Up = 2NS/60$ [12] (19)

7-Calculation of the internal energy and specific heats as a function of the temperatures (T_1 & T_2) can be calculated[11] from:

$$E(T) = R_{mol}((\sum_{i=1}^{n} W_i((\sum_{j=1}^{5} U_{i,j} * T^j) - T) \quad (20)$$

Where E(T) = the total internal energy, $U_{i,j}$=polynomial coefficients[13] for mixture species i=1 to n, table (2) , and W_i = number of moles of species .

8- The total number of moles (W_t) in the mixture is given [13]by:

$W_t = \sum_{i=1}^{n} W_i$ and the specific heat at constant volume $C_V(T)$for the mixture is:

$C_V(T) = \frac{\partial e(T)}{\partial T}$ hence the specific heat is $C_V(T) = \frac{1}{W_t}\frac{\partial(E(T))}{\partial T}$ (21)

Differentiating eq.(20)with respect to (T) gives:

$$W_t * C_V(T) = R_{mol} \sum_{j=1}^{n} W_i ((\sum_{j=1}^{5} j * U_{i,j} * T^{j-i}) - 1) \quad (22)$$

Equations (20) &(21) are general expressions for internal energy and specific heat for the mixture[13].

9-The first law of thermodynamics is applied then the correct values of (T_2 and P_2) are determined during the compression stroke and before the combustion stages[14] as:

$dQ_{conv} - dW= dE= E(T_2)-E(T_1)$ or $f(E) =E(T_2)-E(T_1)+dW- dQ_{conv.}$ (23)

where $dQ_{conv.}$ is the amount of heat transferred through the cylinder walls which is calculated by using the (Eichelberg's) equations[13]as :

$dQ_{conv.} = h*A_s*(T - T_{wall})$ (24)

Where $T= (T_1+T_2)/2$ and $P = (P_1+P_2)/2$

$A_s=(2A_p+\pi*D*X)$ where A_p=the top surface of the piston , D=Cylinder Diameter and $X=L+S/2(1-cos\theta +[L^2+S^2 /4(cos^2\theta-1]^{0.5}$ (X is the piston position with respect to θ)

10- In order to satisfy the first law of thermodynamics at each step, (T_2) should be corrected to give the correct value of internal energy and heat loss, equation (23) was solved numerically using "Newton Raphson's" method [13], and a solution was obtained when f(E) = 0 ; In this method if $(T_2)_{n-1}$ was the estimated value of(T_2) then a better approximation $(T_2)_n$ is given by:

$(T_2)_n=(T_2)_{n-1}-f(E)_{n-1}/f'(E)_{n-1}$ (25)

This procedure was done until the piston reaches a position that spark ignition starts .Then the delay period had to be calculated.

11- The delay periods calculated by using the following equations[15]:

$$D.P = \frac{6*(N)}{U_T} * R_f \quad (26)$$

Where R_f is the flame radius calculated [15] by:

$$R_f = \sqrt[3]{\frac{0.001 * V_2}{\pi}} \quad (27)$$

and U_T is the turbulent flame front speed calculated by:

$$U_T = U_L * ff \quad (28)$$

where ff is the turbulent flame factor calculated by[15]:

$$ff = 1+(0.0018*N) \quad (29)$$

and U_L is the laminar flame front speed is determined by using the experimental equation of (Kuehl) [16].

$$U_L = [\frac{1.087*10^6}{\{(\frac{10^4}{T_b})+(\frac{900}{T_{un}})\}^{4.939}}]*P^{-0.09876} \quad (30)$$

for $(\Phi \le 1.0)$

$$T_b = T_{un} + 2500*(\Phi)(X_r) \quad (31)$$

for $(\Phi) > 1.0$

$$T_b = T_{un} + 2500*(\Phi)(X_r) - 700*(X_r)(\Phi - 1) \quad (32)$$

Where (Φ) is the equivalent ratio which represented [17].by :

$(\Phi) = \frac{AF_{actual}}{AF_{stoichiometric}}$ and $X_r = \frac{m_b}{m_{un}}$

12.Combustion Duration:

The combustion duration is proportional to engine speed which is calculated [1] by:

$$\Delta\theta_c = -1.6189*(\frac{N}{1000})^2 + 19.886*(\frac{N}{1000}) + 39.951 \qquad (33)$$

and the rate of heat release is determined by using[13] :

$$dQ_{release} = (LHV)*(m_m)*(\frac{F}{A})/[2*(\theta_c)(1+\frac{F}{A})]\sin[\pi\frac{\theta-\theta_{spa}}{\theta_c}] \qquad (34)$$

The first law of thermodynamics during the combustion stages will be:

$dQ_{release}=E(T_2) - E(T_1) + dW + dQ_{conv.}$

or

$f(E)=(E_2-E_1) + dW + dQ_{conv} - dQ_{release}$ (35)

and the correction values of the all variables estimated during each step ,the cycle is beginning with the start of intake valve closing(α_1) and before the start of a new step by increasing the angle by ($\Delta\alpha$) which was (2 degree of crank angle in this work),a summation of the work done and heat loss will be done, the conditions at the end of the previous step are set up as the initial conditions for the new step .The calculation is completed when the angle (α_2= the angle at step2) is greater than the exhaust valve open angle.

13- Finally the simulation program (Q.BASIC language) which was developed in this work inquire to save data in a file and calculates the following [14] :

13- 1- Indicated power:

$$IP = \oint PdV * N \qquad (36)$$

13- 2-Indicated specific fuel consumption: $i.s.f.c = \frac{\overset{o}{m}_f}{IP}$ (37)

13- 3-Indicated Thermal efficiency: $\eta_{thi} = \frac{IP}{\dot{m}f * LHV * \eta_c}$ (38) where η_c is the combustion efficiency .

<u>Results and Discussion :</u>

This work was performed and the results were achieved and studied for all parameters at compression ratios of (7, 7.5, 8, 8.5 and 9) and speeds of (1800, 2400 and 3300)rpm, with a suitable spark timing for each speed as (20, 25, and 32.5) deg. b.TDC respectively [2].

Figure (1) represents the variations of indicated power with respect to compression ratio. It observed that the increasing of compression ratio increases the indicated power at all speeds. This improvement is about (0.5 , 1, and 1.5)kW for increasing the compression ratio from (7) to (9) ,this improving is due to increase in indicated work .

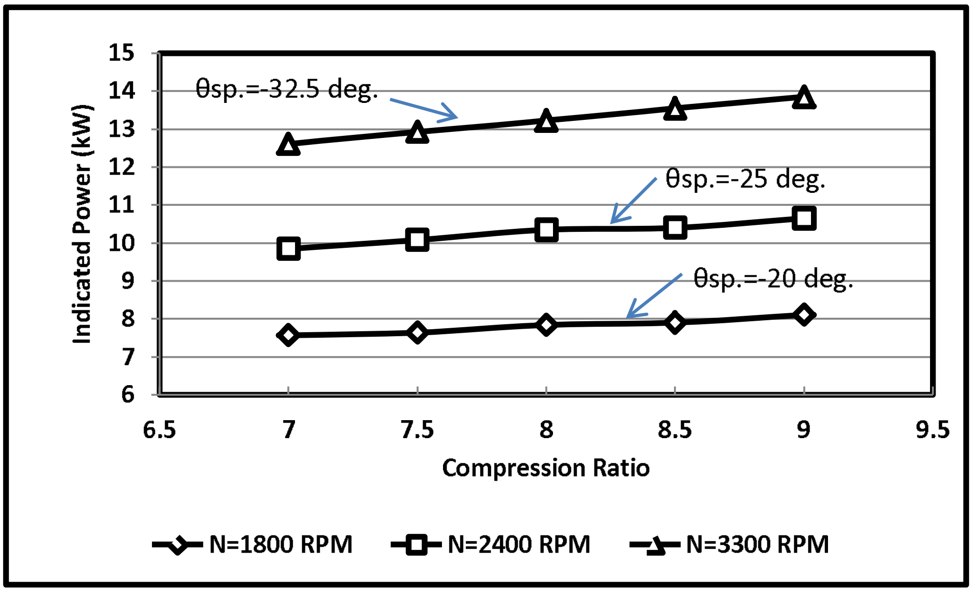


Fig.(1) The Effect of Compression Ratio on the Indicated Power.

Figure (2). shows the relationship between indicated thermal efficiency and compression ratio. It observed that the increasing of compression ratio improves the indicated thermal efficiency about (3 – 3.5) percent at this test conditions .

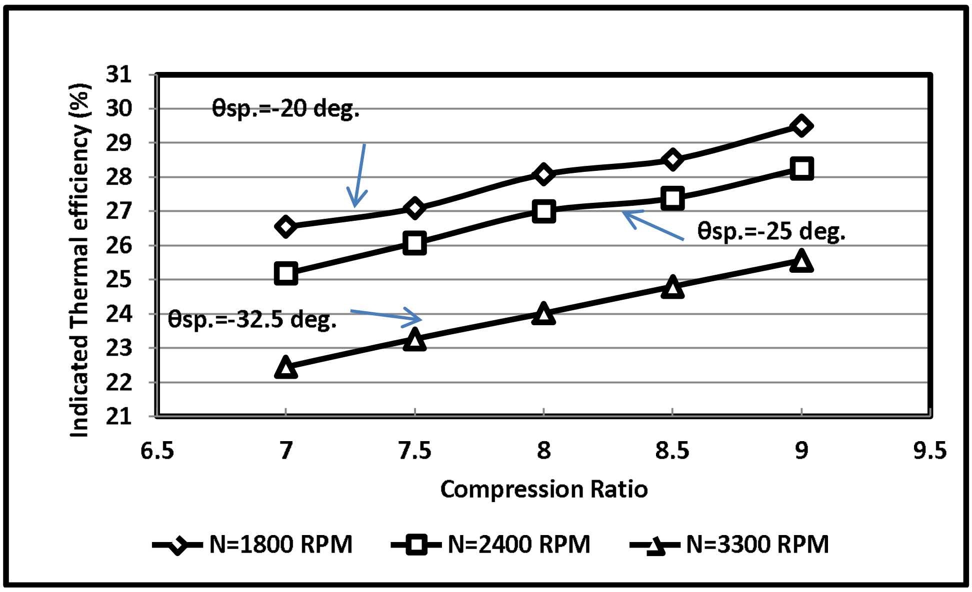


Fig.(2) The Effect of Compression Ratio on the indicated Thermal Efficiency.

Figure (3). represents the relationship between indicated specific fuel consumption and different tests of compression ratios. It observed that the increasing of compression ratio improves the indicated specific fuel consumption, which means that the relation is inversely proportional, where any increase in compression ratio decreases the indicated specific fuel consumption.

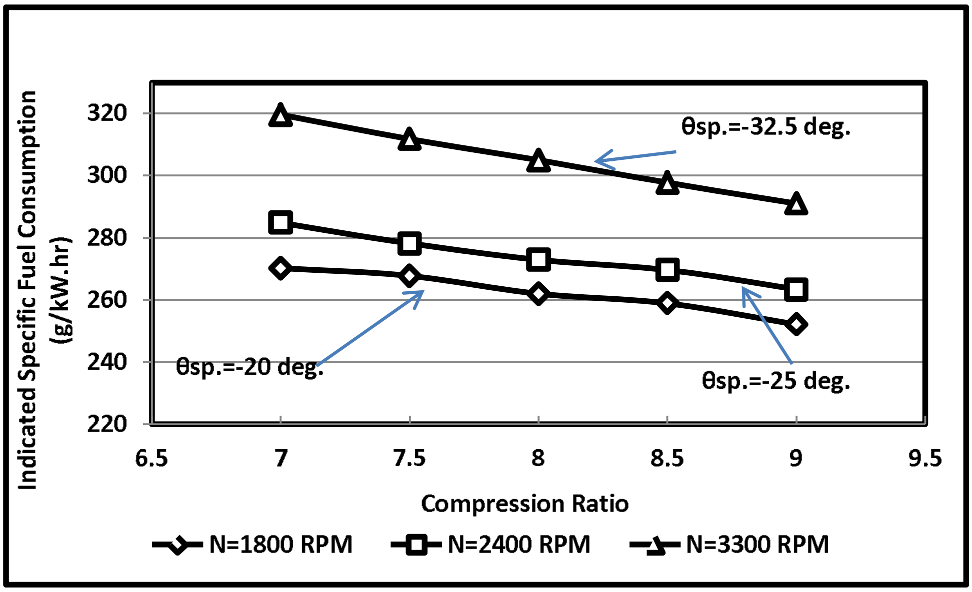


Fig.(3) The Effect of Compression Ratio on the Indicated Specific Fuel Consumption .

Figure (4). represents the relationship between heat losses and different tests of compression ratios. It observed that the increasing of compression ratio increases the heat losses , this losses are due to the increasing of the cylinder geometry (bore and stroke) which increases the heat losses and also due to increase in the cylinder pressure and temperature .

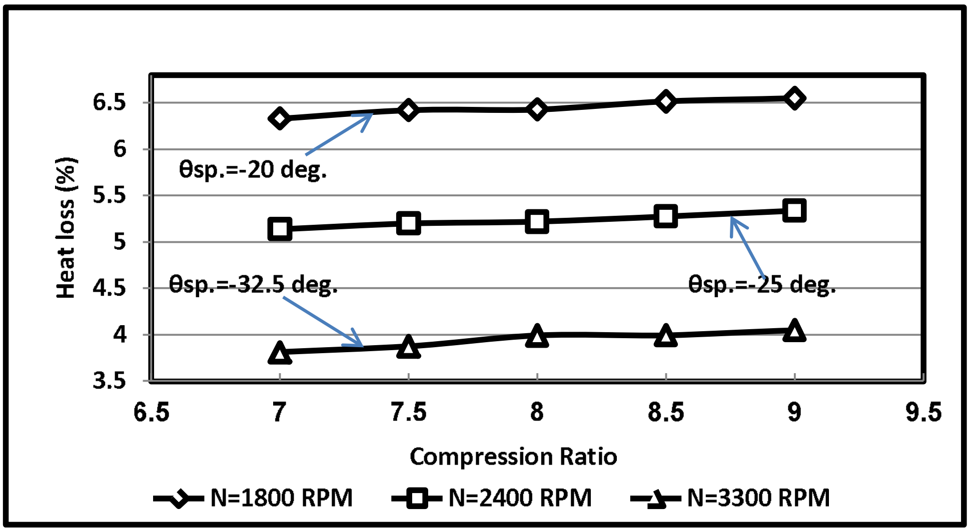


Fig.(4) The Effect of Compression Ratio on the Heat Loss.

Figure (5). represents the relationship between ignition delay and different tests of compression ratios. It observed that the increasing of compression ratio improves the ignition , which means that the relation is inversely proportional, where any increase in compression ratio decreases the ignition delay for all speeds, due to increasing the initial cylinder gas pressure and temperature .

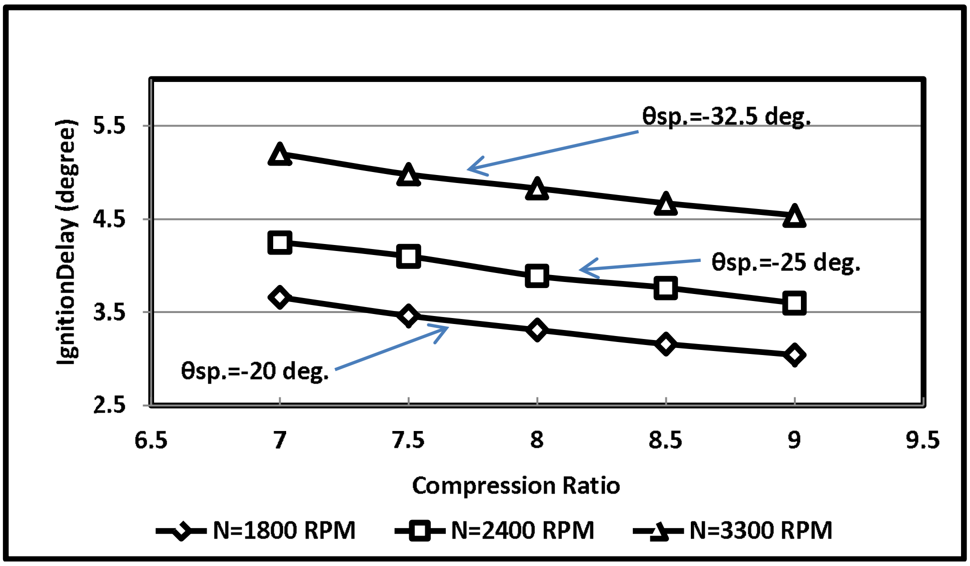


Fig.(5) The Effect of Compression Ratio on the Ignition Delay .

Figure. (6) represents the relationship between turbulent flame speed and different tests of compression ratios. It observed that the increasing of compression ratio increases the turbulent flame speed, due to increasing the cylinder gas pressure and temperature .

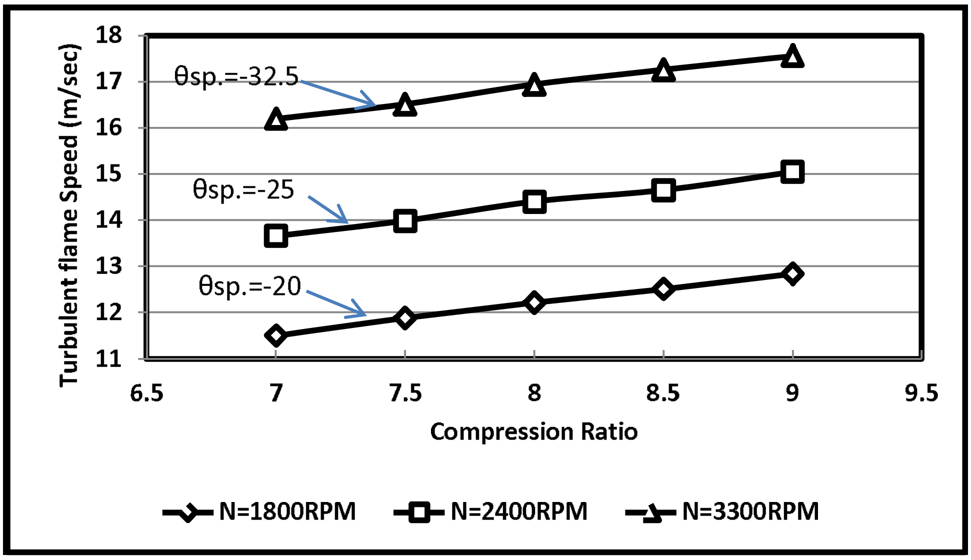


Fig.(6) The Effect of Compression Ratio on the Turbulent Flame Speed.

Figure (7). represent the variations of the fuel mass burning rate for different tests of compression ratio. It observed that the mass burning rate wasn't greatly affected by increasing or decreasing the compression ratio, but significantly affected with speed variations .

.

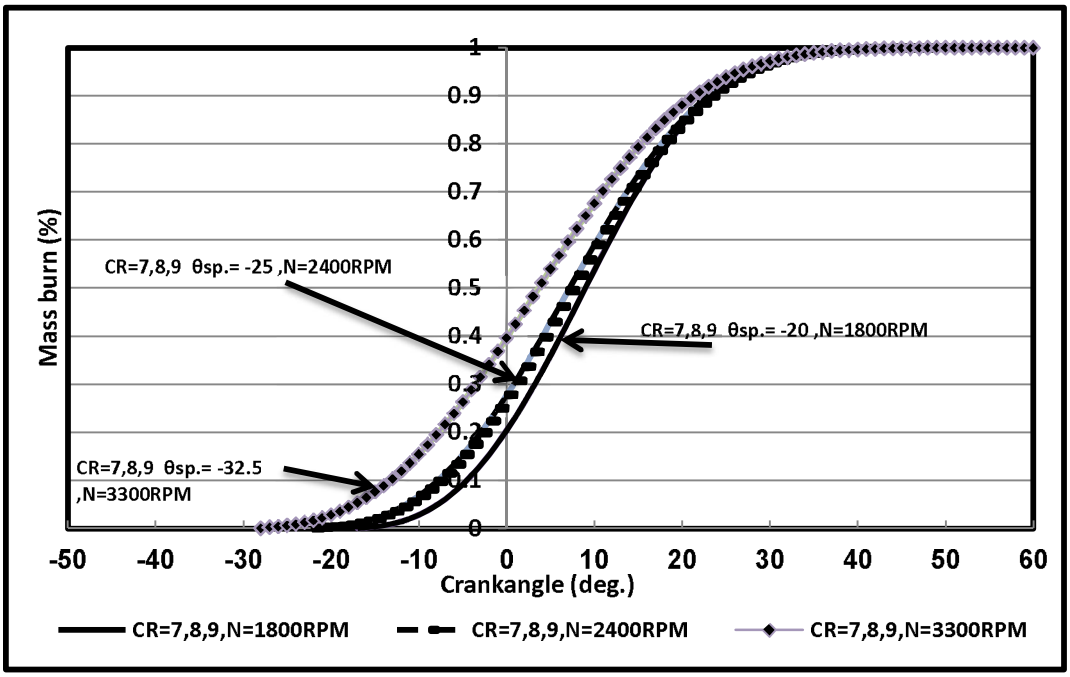
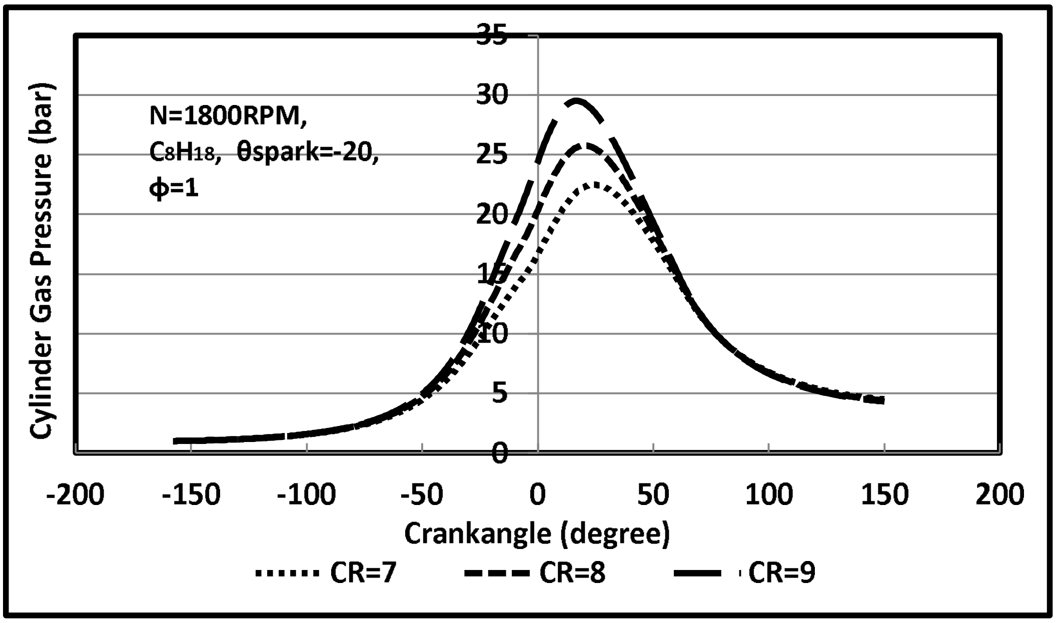
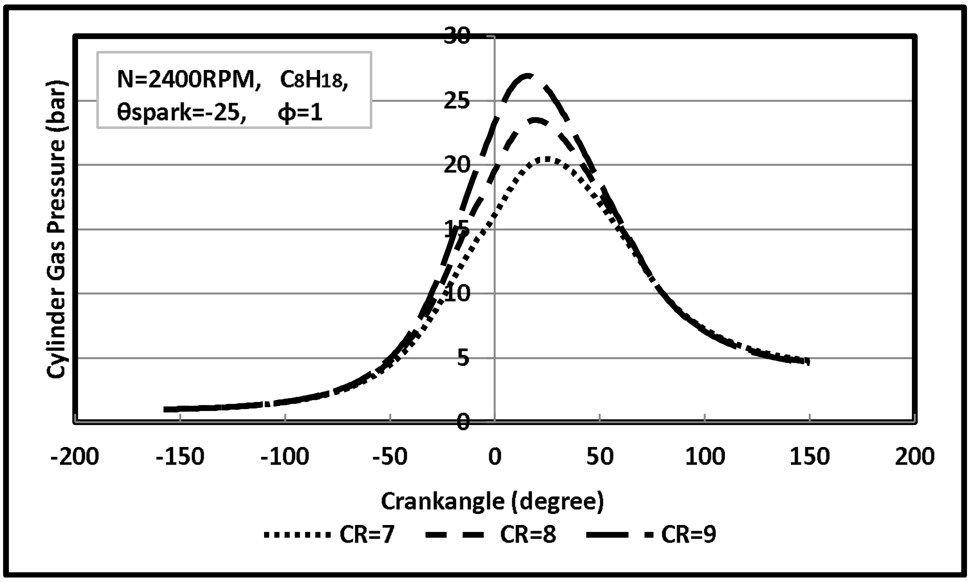


Fig.(7) The Effect of Compression Ratio on the Mass burning rate .

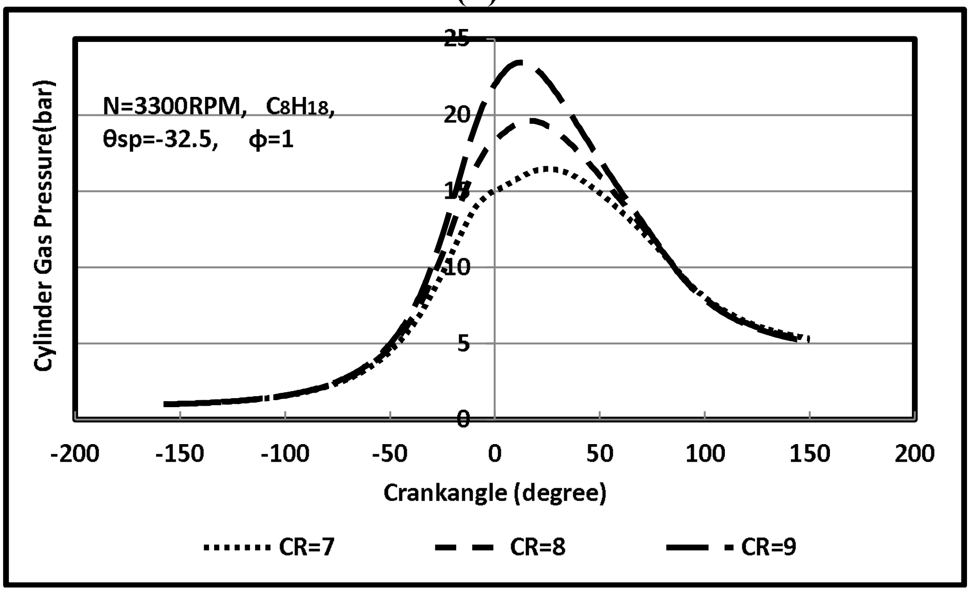
Figure (8) A, B and C .represent the effects of compression ratios (7, 8 and 9) on the cylinder gas Pressure with spark timing of (20,25, and 32.5) deg. b.TDC for speeds of (1800, 2400 and 3300) rpm. It observed that the increasing of the compression ratio increases the cylinder gas pressure strongly if the correct choose of spark timing was done , and it observed too that the cylinder gas pressure (peak) at low engine speed (1800 rpm) is greater than others (2400, 3300)rpm. At the same conditions due to availability of more time for fuel combustion near the (TDC).



(A)



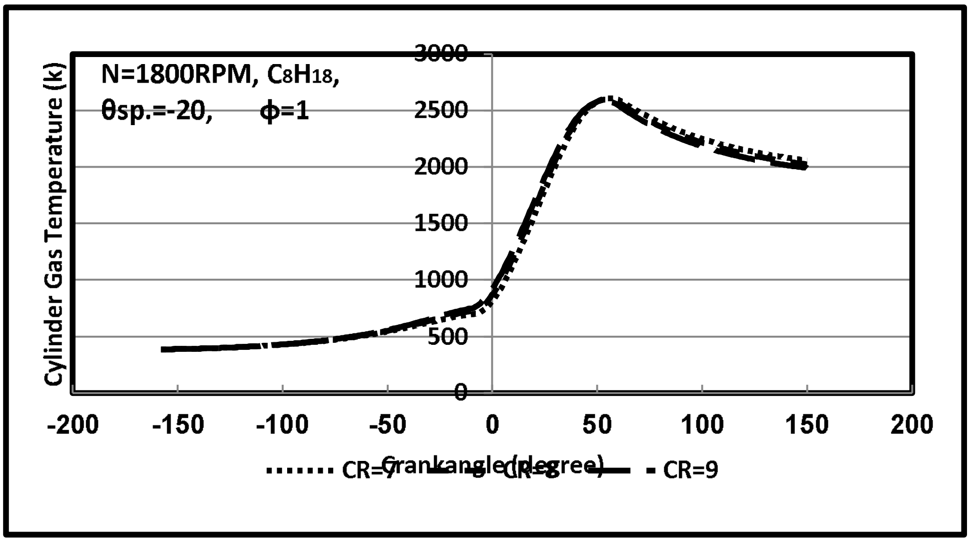
(B)



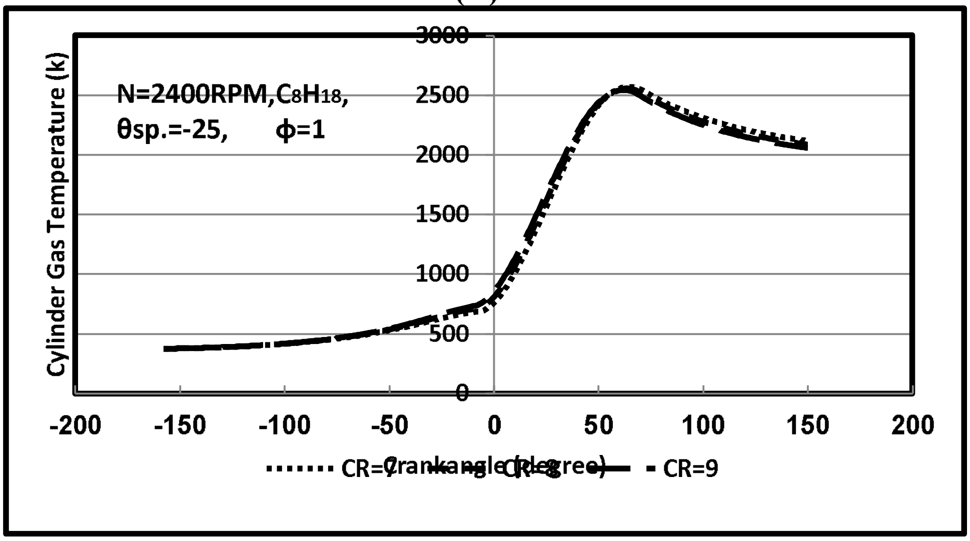
(C)

Fig.(8)A, B and C. The Effect of Compression Ratio on the Cylinder Gas Pressure at Speeds of 1800, 2400 and 3300 rpm .

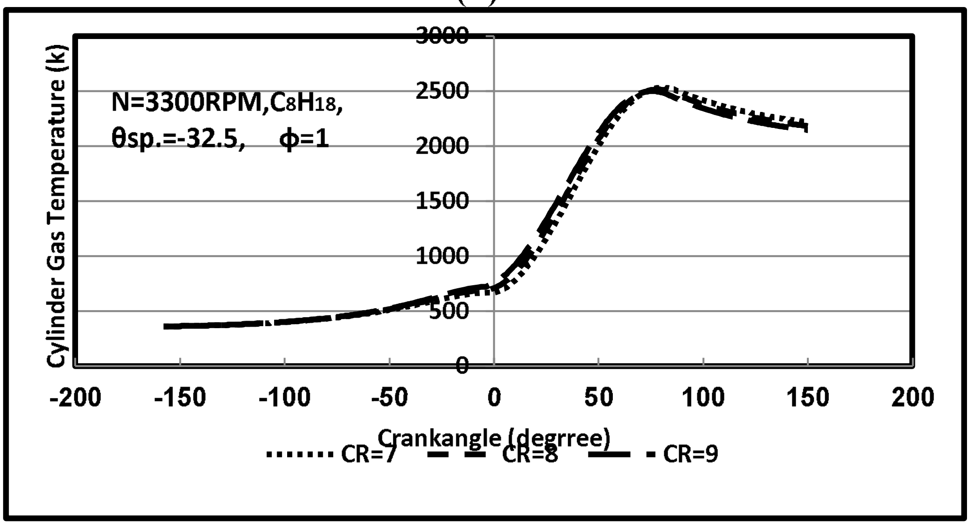
Figure (9) A, B and C. represent the effects of compression ratios (7,8 and 9) on the cylinder gas temperature with spark timing of (20,25, and 32.5) deg. b.TDC for speeds of (1800, 2400 and 3300) rpm. It observed that the cylinder gas temperature wasn't greatly affected with increasing the compression ratio in this work due to the correct choose of spark timing and it observed that the cylinder gas temperature(peak) at low engine speed (1800 rpm) is greater than others (2400, 3300)rpm. at the same conditions due to availability of more time for fuel combustion near the (TDC).



(A)



(B)



(C)

Fig.(9) A, B and C . The Effect of Compression Ratio on the Cylinder Gas Temperature at Speeds of 1800, 2400 and 3300 rpm .

Conclusions :

A simulation model was developed for analyzing the effects of compression ratio on combustion and performance characteristics of spark ignition engine. This method has been developed in such way that it can be used for predicting the effect of any parameter. The result shows that increasing the compression ratio has a positive effects on the combustion and performance characteristics. This model predicted the engine performance characteristics in closer approximation to that of experimental results.

References:

1- Sitthichok Sitthiracha et al "An Analytical Model of Spark Ignition Engine for Performance prediction" ,*The 20th Conference of Mechanical engineering Network of Thailand ,October 2006,Nakhon Ratchasima, Thailand.*

2- Kutaeba J.M. AL-Khishali et al " Effect of valve lift at different IVO ,IVC and OVERLAP angles on SI Engine performance "*The 7thJordanian International Mechanical Engineering Conference. September 2010, Amman – Jordan.*

3- Khalefa R.Ahmed " Simulation of the (Bore/Stroke) Ratio Effects on the Combustion Characteristics and Performance of a Spark Ignition Engine" *3rd Scientific conference ,Najaf, 2013.*

4- E. Abu-Nada, B.Akash,I. Al-Hinti and A. Al-Sarkhi "Performance of Spark Ignition Engine Under Effect of Friction Using Gas Mixture Model" Journal of the Energy Institute, Vol. 82 ,No. 3, 2009.

5- E. Porpatham, et al. ." Effect of Compression Ratio on the performance and combustion of a biogas fuelled spark ignition engine"Jour. Fuel ,Elsevier ,Vol.95. 2012, pp247-257.

6 – Aina T., Folayan C. O. and Pam G. Y " Influence of compression ratio on the performance characteristics of a spark ignition engine" Jour. of Advances in Applied Science Research, Vol.3, No.4,2012, pp1915-1922

7- Indira Priyadarsini, et al." Study of Impact of Spark Timing and Compression Ratio on Performance of SI Engine" International Journal of Recent Technology and Engineering (IJRTE) Vol.3,No.6, January 2015 .

8- Sitthichok Sitthiracha et al "An Analytical Model of Spark Ignition Engine for Performance prediction" *The 20th Conference of Mechanical engineering Network of Thailand* ,October 2006,Nakhon Ratchasima, Thailand.

9- Ismail Altin , Ismet Sezer ,and Attila Bilgin "Effects of the Stroke/Bore Ratio on the Performance Parameters of a Dual Spark –Ignition Engine" Jour. of Energy and Fuels, No. 23, 2009, pp 1825-1831 .

10- Bilgin,A. "Geometric Features of the Flame Propagation Process for an SI Engine Having Dual-Ignition System" Int.J.Energy Res.No.26,Vol. 11,2002 pp 987-1000.

11- Perihan Sekmen and Yakup Sekmen "Mathematical Modeling of a SI Engine Cycle With Actual Air-Fuel Cycle Analysis" Jor. of Mechanical and Computational Applications,Vol. 12, No.3, pp161-171 ,2007.

12- Willard W. Pulkrabek "Engineering Fundamentals of the Internal Combustion Engines" 1997 – Prentice- Hail-Inc.[pp 235-237].

13- موسى مصطفى ويس كهية "دراسة تأثير درجة حرارة الشحنة الداخلة على أداء وملوثات محرك احتراق داخلي يعمل بالشرارة" رسالة ماجستير ، جامعة تكريت، 2005 .

14- E. Abu-Nada, B.Akash,I. Al-Hinti and A. Al-Sarkhi "Performance of Spark Ignition Engine Under Effect of Friction Using Gas Mixture Model" Journal of the Energy Institute, Vol. 82 ,No. 3, 2009.

15-Maher A.R. AL-BAGHDADI "A Simulation Model for a Single Cylinder Four-Stroke Spark Ignition Engine Fueled with alternative Fuels" Turkish J. Eng. Env. Sci. Vol. 30, 2006, pp331-350.

16- Kuehl .D.K "Laminar Burning Velocities of Propane –Air Mixture "8th Symposium (International) on Combustion, pp.510, 1962.

17- Obert E. F. "Internal Combustion Engine and Air Pollution"3rd Ed. ,In text Educational Publishers,1973.

Table (2)Polynomial Coefficient U(i,j)Temperature Range (500-3000) K, PO=101325 pa.[7]

i	Species j	1	2	3	4	5	6	7
1	CO_2	3.0959	2.7311E-3	-7.8854E-7	8.6600E-11	0.00	6.5839	-3.9364
2	H_2O	3.7429	5.6559E-4	4.9524E-8	-1.8180E-11	0.00	0.96514	-2.3922E8
3	O_2	3.3443	6.5235E-4	-1.4952E-7	1.5389E-11	0.00	5.7123	0.000
4	N_2	3.4332	2.9426E-4	1.9530E-9	-6.574E-12	0.00	3.7586	0.000
5	H_2	3.5017	-8.181E-4	9.6699E-8	-1.4439E-11	0.00	-3.8447	0.000
6	NO	2.4990	2.9938E-4	-9.5880E-9	-4.9903E-12	0.00	5.1134	8.996E7
7	N	3.4502	2.8744E-	-2.4481E-9	6.1515E-13	0.00	4.18504	4.7165
8	OH	3.4502	3.6752E-4	-2.4435E-8	-3.0487E-12	0.00	4.78123	3.9458E7
9	CO	3.3170	3.769E-4	-3.2200E-8	-2.1945E-12	0.00	4.62384	-1.139E8
10	H	2.5000	0.0000	0.0000	0.0000	0.00	-4.5931	2.1623E8
11	O	2.7640	-2.514E-4	1.0018E-7	-1.3867E-11	0.00	3.73309	2.4707E8
12	AR	2.5000	0.0000	0.0000	0.0000	0.00	0.00	0.0000
13	$C_n H_m O_r$	-0.719	4.6426E-2	-1.6838E-5	2.6700E-9	0.00	0.00	-4.929E8