EVALUATION OF THE DIESEL ENGINE PERFORMANCE UNDER EFFECT OF DIFFERENT PATTERNS OF THE FUEL INJECTION RATES [+]

Rafeq A. Khalefa [*]

Abstract :

The objective of this research is to study the effect of different fuel injection patterns on the diesel engine performance. This was achieved by developing a computer program and constricting mathematical models to simulate three patterns (Boot ,Ramp and Square) of fuel injection profiles .The performance parameters (Indicated Power, Indicated Thermal Efficiency, Indicated Fuel Consumption and Heat Loss) are all predicted at speeds of (1500,2000,2500,3000 and 3500) rpm and at Air/Fuel ratios of (20,30 and 40) . Comparing and analyzing of computational results, it can be seen that some of the fuel injection patterns (profiles) has a positive effects on the diesel engine performance parameters. In addition, both, cylinder gas temperature and pressure were found to be clearly affected too.

Key words: **fuel injection, diesel engine performance, diesel engine simulation.**

تقييم أداء محركات الديزل تحت تأثير أنماط مختلفة من معدلات حقن الوقود

رفيق احمد خليفة

المستخلص :

الهدف من هذا البحث هو دراسة تأثير النماذج المختلفة لحقن الوقود على أداء محركات الديزل من خلال إنشاء برنامج حاسبة الكترونية وبناء نماذج رياضية لمحاكاة ثلاثة انماط (تمهيدي ، متصاعد و مربع) لحقن الوقود على أداء محركات الديزل (القدرة البيانية ، الكفاءة الحرارية البيانية ، استهلاك الوقود النوعي البياني ، والخسائر الحرارية) للسرعات (3500,3000,2500,2000,1500) دورة\دقيقة ونسبة هواء\وقود (40,30,20). تحليل ومقارنة النتائج المحسوبة أظهرت بان بعض أنماط حقن الوقود ذو تأثير ايجابي على أداء محركات الديزل . إضافة إلى ذلك فان درجة الحرارة وضغط الغازات داخل الاسطوانة تأثرت أيضا بشكل واضح .

الكلمات الدالة : حقن الوقود، أداء محركات الديزل، محاكاة محركات الديزل

[+] **Received on 8/1/2014 , Accepted on 17/8/2014**
[*] **Lecturer / Technical College / Kirkuk**

Abbreviation		Units
A_s	Instantaneous Heat Transfer Area	(m^2)
A_p	Piston Surface 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	(kJ/kg)
E	Absolute Internal Energy	(kJ)
$E(T_1)$	Internal Energy at The Beginning of the Step	(kJ)
$E(T_2)$	Internal Energy at The End of the Step	(kJ)
h	Convection Heat Transfer Coefficient	$(kW/m^2.k)$
IP	Indicated Power	(kW)
Imep	Indicated Mean Effective Pressure	(kPa)
Isfc	Indicated Specific Fuel Consumption	(g/hr)
L	Connecting Rod Length	(m)
$\dot{m}_f$	The Fuel Consumption	(kg/s)
P_1	Cylinder pressure at the Beginning of the Step	(kPa)
P_2	Cylinder pressure at the End of the Step	(kPa)
Q_{HV}	Low Heating Value	(kJ/kg)
$R_{mol.}$	Gas Constant	(kJ/kg.mole.k)
S	Stroke Length	(m)
T_1	Cylinder Temperature at the Beginning of the Step	(k)
T_2	Cylinder Temperature at the End of the Step	(k)
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)

1. Introduction :

Diesel engines occupy a prominent role in both , present transportations and power generation sectors. Due to superior characteristics such as lower fuel consumption, higher engine power output and lower emissions such as (CO and HC) compared with gasoline operated engines.A great efforts are consumed by engineers to develop the performance of this type. The object is to make it more economical by lowering its fuel consumption, emissions and increasing the power output and efficiency. There have been many methods used and tried to improve the diesel engine performace.

Computer simulation has contributed enormously towards a new evaluation in the field of internal combustion engines. Mathematical tools have become very popular in recent years owing to the continuously increasing improvement in computational power. Simulation models can be used as an active toll to understand and develop the diesel engine performance. These models can reduce the number of experiments.Depending upon the various possible applications, different types of models for diesel engine combustion process have been used. [1,2,3, 4,5,6,7,8,9,10].

The performance of diesel engines is heavily influenced by their injection. In fact, the most notable advances achieved in diesel engines resulted directly from superior fuel injection system designs. Fuel injection plays an integral role in the development of the combustion in the engine cylinder. It is responsible for the distribution and mixing of fuel within the combustion chamber which in turn has an impact on the efficiency of the combustion and the resultant pollutants [11]. While the main purpose of the system is to deliver fuel to the cylinders of a diesel engine, it is how that fuel is delivered which makes the difference in engine performance, emissions, and noise characteristics. Most fuel injection systems use electronics to control the opening and closing of the nozzle. Electrical signals are converted into mechanical forces using some type of actuator. Commonly, these actuators can be either electromagnetic solenoids or active materials such a piezoelectric ceramics. However, it is still not enough to deliver an accurately metered amount of fuel at the proper time to achieve good combustion. Additional aspects are critical to ensure proper fuel injection system performance including: fuel atomization, bulk mixing and air utilization. Studies have been performed on the variation of the rate of injection (ROI) profile and its impact on fuel spray and combustion [12,13,14,15,16] . Because of the different dynamics or operations , the engine optimal ROI profiles have been identified for certain conditions corresponding to speed and load in engine operation [17]. A comprehensive study of "boot" shaped (ROI) profiles was performed and their effect on combustion, engine performance, and emissions predicted using a modified pump-line nozzle system [18,19].

2. Theoretical analysis :

The aim of the present research is to predicting the performance parameters under effect of three patterns (Boot, Ramp and Square) of fuel injection as in figure (1).This was accomplished by constricting a mathematical model for each pattern and using it in the simulation program .

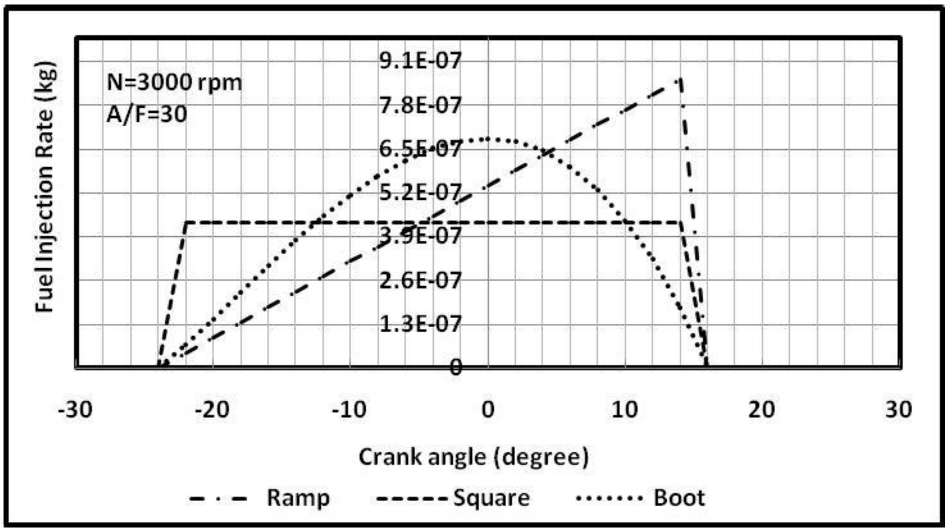

Figure (1): the three typical fuel injection patterns at speed of 3000 rpm and A/F ratio of 30

The rapid development of computer technology narrow down the time consumption for engine test through the simulation techniques. Simple models accounting for the basic features of engine operation, treating the cylinder contents as a uniform mixture, which are

usually termed single-zone models, have been developed and still continue to exist due to their simplicity and low computational cost combined with reasonable accuracy [20].
The present theoretical study was conducted using a computer simulation with "Quick Basic" language on a 4-stroke single cylinder direct injection diesel engine Table (1).

Table (1): Specification of the engine and fuel used

Bore	69mm
Stoke length	62mm
Connecting rod length	104mm
Compression ratio	22:1
Displacement volume	0.232 liter
Inlet valve closing (IVC)	38°aBDC
Exhaust valve Open (EVO)	48° bBDC
Fuel injection timing	38° bTDC
Diesel Fuel	$C_{14.4}H_{24.9}$
Heating Valve of the Fuel	39MJ
Fuel Cetane Number	49

By using first law of thermodynamics the properties of the cylinder content at each crank angle interval ($\Delta\alpha$) will be evaluated. With regards to calculation of number of moles of reactants ,the following simulation will be achieved .At the start of combustion , the moles of different species are considered includes O_2, N_2 from intake air and CO_2, H_2O, N_2 and O_2 from the residual gases. The overall combustion equation considered for the fuel with C-H-O-N is [1]:

$$CxHyOz + \lambda * ycc * (O_2 + 3.773N_2) \rightarrow xCO_2 + (y/2)\, H_2O + (\lambda - 1)\, ycc\, O_2 + \lambda * ycc * (3.773N_2) \quad \ldots \quad 1$$

$$A/F_{Stoch.} = ycc = x + (y/4) - (z/2)$$

Total number of reactants and products during the start of combustion as well every degree crank angle was calculated from the equations.

$$tmr = 1 + \lambda * ycc * 4.773 \qquad \ldots\ldots 2$$

$$tmp = x + (y/4) + 3.773 * \lambda * ycc + (\lambda - 1) * ycc \qquad \ldots\ldots 3$$

where tmr = total mole number of reactants

tmp = total mole number of products and

λ = Excess air factor

Volume at any crank angle is calculated from the following equation [1,4,9,10,20]:

$$V(\theta) = Vc + (Vs/2) * [1 - \cos(\theta) + (2L/S) - \sqrt{(2L/S)^2 - \sin^2\theta}\,] \quad \ldots\ldots\ldots\ldots 4$$

Initial pressure and temperature at the beginning of the compression process is calculated as follows [1,4,9,10,20,21]:

$$T_2 = T_1 * [\frac{V_1}{V_2}]^{k-1} = T_1 * [\frac{V_1}{V_2}]^{\frac{R_{mol}}{C_V(T_1)}} \quad \ldots\ldots\ldots\ldots 5$$

$$P_2 = [\frac{V_1}{V_2}] * [\frac{T_2}{T_1}] * P_1 \quad \ldots\ldots\ldots\ldots 6$$

The specific internal energy for each species and overall internal energy are calculated as a function of (T_1&T_2) from the expressions given below [1,4,9,10,20,21]:

$$e_{i(T)} = R_{mol} ((\sum_{j=1}^{j=5} U_{i,j} * T^j) - T) \quad \ldots\ldots\ldots\ldots 7$$

$$E_{(T)} = R_{mol} ((\sum_{i=1}^{n} W_i ((\sum_{j=1}^{5} U_{i,j} * T^j) - T) \quad \ldots\ldots\ldots\ldots 8$$

Where $E_{(T)}$ = total internal energy , $U_{i,j}$ = polynomial coefficients for mixture species in the cylinder [1,2,3]. i =1 to n & W_i= species mole number where

Species	CO_2	H_2O	O_2	N_2	C_nH_m
Subscript (i)	1	2	3	4	5

$$W_t = \sum_{i=1}^{n} W_i \quad \ldots\ldots\ldots\ldots 9$$

Where (W_t) is the total mole number of the mixture

Specific heat at constant volume and constant pressure for each species is calculated using the expression given below

$$C_{V(T)} = \frac{1}{W_t} \frac{\partial(E_{(T)})}{\partial T} \quad \ldots\ldots\ldots\ldots 10$$

The derivative of equation (5) is

$$W_t * C_V(T) = R_{mol} \sum_{j=1}^{n} W_i ((\sum_{j=1}^{5} j * U_{i,j} * T^{j-i}) - 1) \quad \ldots\ldots 11$$

Where equations (7&10) represent the general expression of the internal energy & specific heat.

The work done during the step is calculated as [1,4,9,10,20]:

$dW = P\, dV = 0.5\, (P_1+P_2)\, (V_2-V_1)$ ……………………. ……………………………12

By applying the first law of thermodynamics during the compression stroke before the combustion starting.

$dQ_{conv} - dW = dE = E(T_2) - E(T_1)$ or $f(E) = E(T_2) - E(T_1) + dW - Q_{conv}$……………13

where dQ_{con}=heat loss through the cylinder walls calculated using

(Eichelberg"s equation) $dQ_{conv.} = h*A_s*(T - T_{wall})$…… …………………..…14

Where h= Heat transfer coefficient and calculated as:

$h=2.466*10^{-4}(U_p)^{1/3}(p)^{1/2}(T)^{1/2}$…………………………………………….…..…15

$U_P = (2*S*N)/60$, $T = (T_1+T_2)/2$ & $P= (P_1+ P_2)/2$

$A_s=(2A_p+\pi *D*X)$

$$X = L + (S/2) * (1 - \cos\theta + [L^2 + S^2/4(\cos^2\theta - 1]^{0.5}$$

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 (13) was solved numerically using "Newton Raphson's" method [4,9,10,20], and a solution was obtained when f(E) = 0 and during the combustion the energy equation can be written as[1,4,20]:

$E(T_2) = E(T_1) - dW - dQ_{\text{Conv.}} + dM_f\, Q_{HV}$ ……………………………..…………16

To find the correct value of T_2, both sides of the above equation should be balanced so the above equation is rearranged as shown below

$ER = E(T_1) - (T_2)\, E - dW - dQ_{\text{Conv.}} + dM_f\, Q_{HV}$ ……………………………………17

If the numerical value of ER is less than the accuracy required, then the correct value of (T_2) has been established, otherwise a new value of T is calculated for new internal energy and C_V values by using (Newton-Raphson technique) [1,4,20] .

$ER' = C_V(T_2)*tmp$……………………………………………………………..…..18

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:

$(T_2)_n = (T_2)_{n-1} - (ER / ER')$ ………………………………………………………….…..19

3. Fuel injection duration :

The duration of fuel injection is calculated [4] by:

$\theta_{dur.} = -3.72 * Ln(A/F) + 49.46$ ……………………………………………………….20

mass of the fuel injected for each crank angle interval ($\Delta\alpha$)which was (2degree) is calculated using the following expression[4,20]:

$TFIR = \frac{Ma * \Phi}{(\frac{A}{F})stoich.}$ ……….. ……………………………………………………..…….21

Where

TFIR is the total mass of fuel injection per cycle,

M_a = mass of air = $(n_{CO2}+n_{H2O}+ n_{O2} + n_{N2})*28.96$

where n_{CO2} , n_{H2O} , n_{O2} & n_{N2} are the number of moles of CO_2 ,H_2O,O_2 and N_2 respectively in the atmospheric air.

Φ is the equivalence ratio [21]. $\Phi = \frac{A/F_{stoich.}}{A/F_{actual}}$ ……………………………….22

$FIR = \left(\frac{TFIR}{\theta_{dur.}}\right)$ …………………………………...…………………………………23

FIR is the mass of fuel injected per degree of crank angle.

Three mathematical equations are used in a simulation program for calculating the rate of fuel injection per crank angle interval ($\Delta\alpha$).These equations are correlated and concluded by using both, the computer characteristics and the developed computer program as follows:

(i)- BOOT model:

$$ROI = \left(\frac{TFIR}{13.25525}\right) * ((-5.32 * 10^{-5}) * (RAF)^3 + (6.81 * 10^{-4} * (RAF)^2 + (5.87 * 10^{-2} * (RAF) - (16.4)^{-3}) \quad \ldots (24)$$

(ii)-RAMP model:

ROI= $\left(\frac{TFIR}{\theta_{dur.}}\right) * \left(\frac{(RAF)}{9.6}\right)$ 25

(iii)-SQUARE model:

ROI= $\left(\frac{TFIR}{\theta_{dur.}}\right)*(\Delta\alpha)$ 26

where

θ_{dur} is the fuel injection duration (degree)
ROI is the rate of injection per interval ($\Delta\alpha$) which was (2 degree) in this work.
RAF= θ_{inj}- θ , θ_{nj} is the angle at which fuel injection starts (degrees)
θ is the crank angle (degrees)

Ignition delay can be defined as the time interval between the start of fuel injection and the start of combustion. Many researchers have given correlations for predicting ignition delay in which the earliest was Wolfer (1938) [6, 16,17]. The empirical formula used in the present model is shown in equation which was used by [9,20]. This equation gives the value of ignition delay in terms of values of degrees of crank angle using temperatures (T) in Kelvin and pressure (P) in kPa. The following equation has been used to predict the ignition delay.

ID = $(40/CN)^{0.69}*EXP(4644/T_2)*0.000343*N*(P_2)^{-0.388}$ 27

The following equations are used for calculating the indicated power, brake power, brake thermal efficiency and brake specific fuel consumption:

IP = (Imep*A*L*N*Z)/(60*n) 28

η_{ith} = IP/($\dot{m}_f$*Q_{HV}) 29

ISFC= $\dot{m}_f$ / IP 30

4. Results and discussions :

Figures 2,3 and 4 show the variation of indicated power with respect to engine speed at A/F of (20,30 and 40) for three fuel injection rate patterns .It observed that the maximum indicated power is obtained by using boot or ramp profile. The square profile gives lower indicated power due to variation of the fuel amount injected and burned before and after (T.D.C).This will lead to increasing or decreasing the negative or positive indicated work.

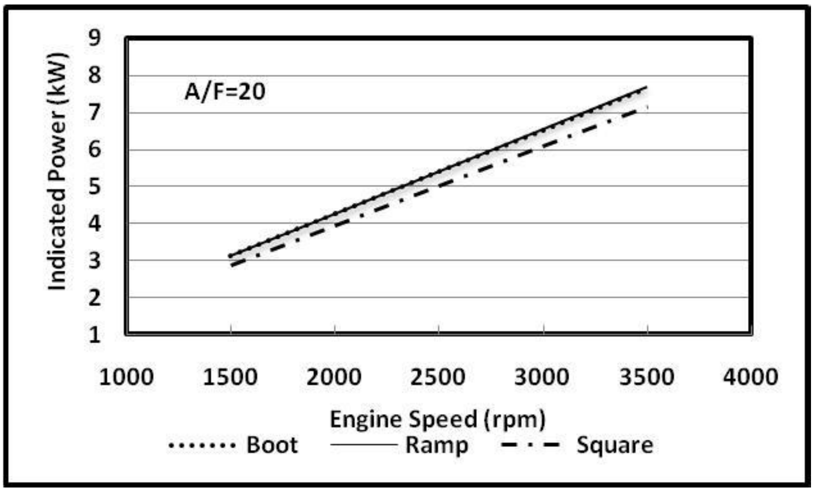

Figure (2): The effect of fuel injection patterns on the indicated power

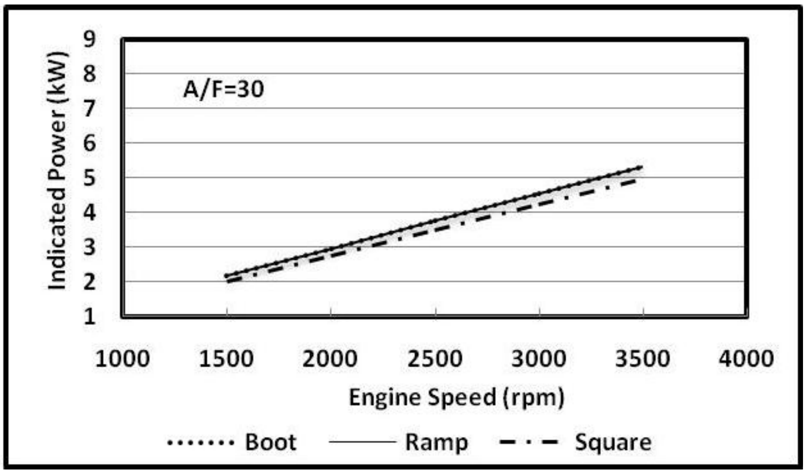

Figure (3): The effect of fuel injection patterns on the indicated power

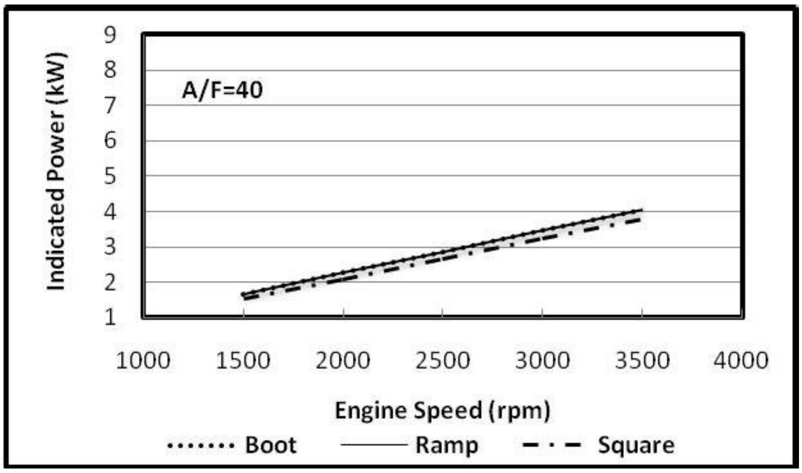

Figure (4): The effect of fuel injection patterns on the indicated power

Figures 5,6 and 7 show the variation of indicated thermal efficiency with respect to engine speed at A/F of (20,30 and 40) for three injection rate patterns**.** It can be shown that the maximum indicated efficiency is obtained by using boot or ramp profile .The square profile gives lower indicated efficiency due to variation of the fuel amount injected and burned before and after (T.D.C).This will lead to increasing or decreasing the negative or positive indicated thermal efficiency.

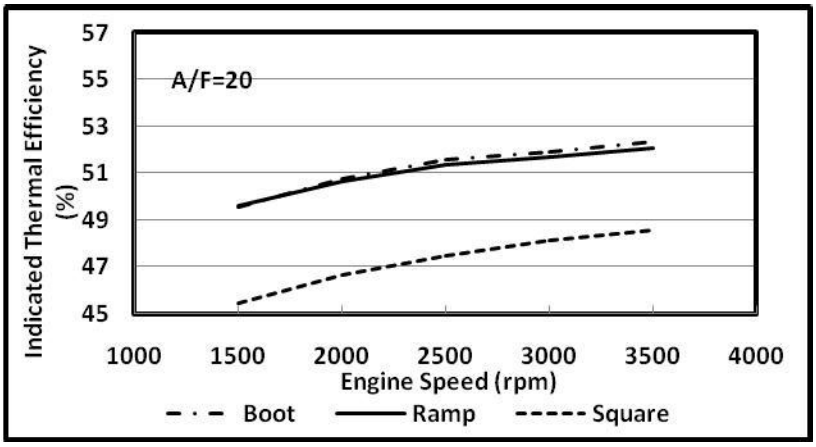

Figure (5): The effect of fuel injection patterns on the indicated thermal efficiency

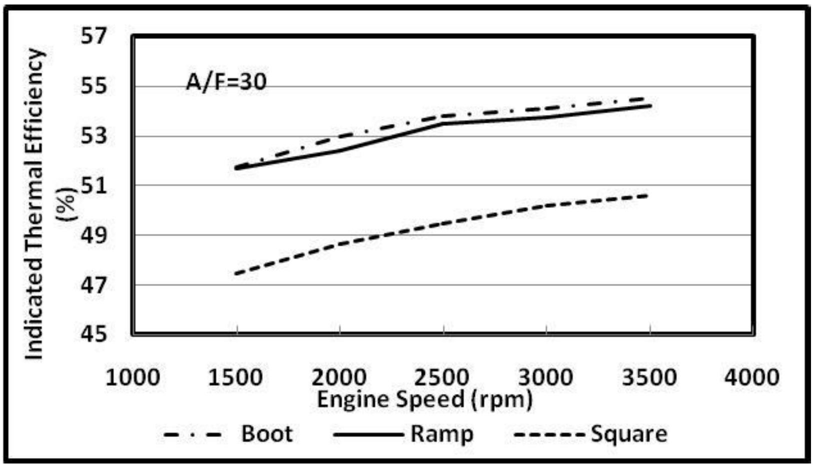

Figure (6): The effect of fuel injection patterns on the indicated thermal efficiency,

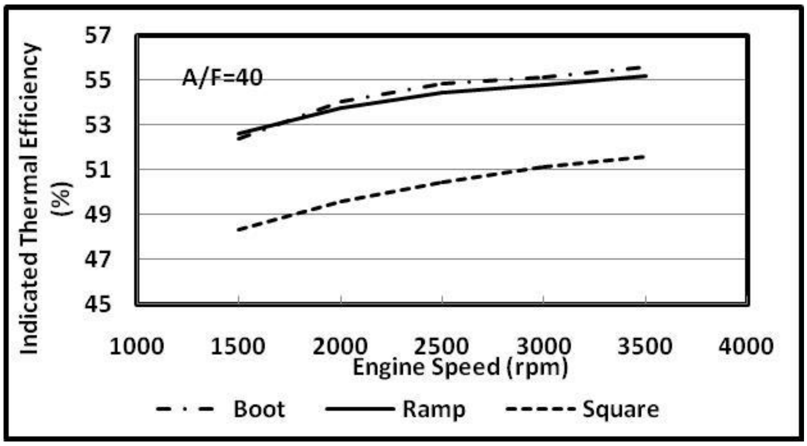

Figure (7): The effect of fuel injection patterns on the indicated thermal efficiency.

Figures (8,9 and 10) show the variations of the indicated specific fuel consumption for three patterns of fuel injection rate with respect to engine speeds at A/F of (20,30 and 40).It can be seen that the square profile leads to more indicated specific fuel consumption .The (boot and ramp) profiles have less fuel consumption which make these two profiles more economically and powerful

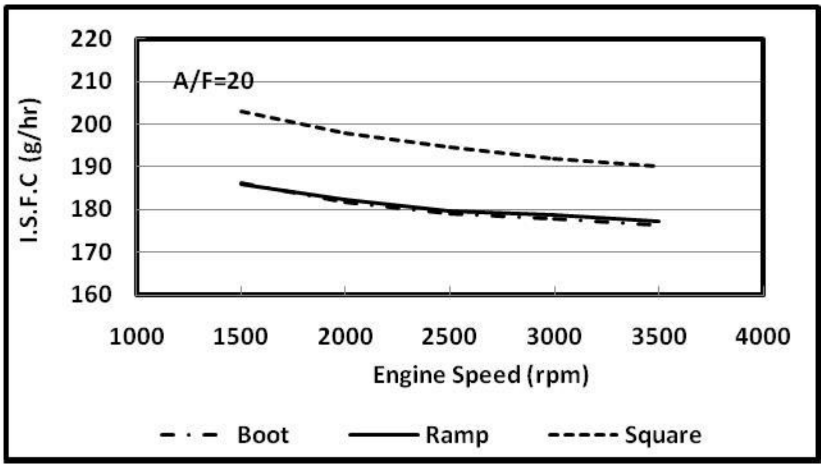

Figure (8): The effect of fuel injection patterns on the indicated specific fuel consumption.

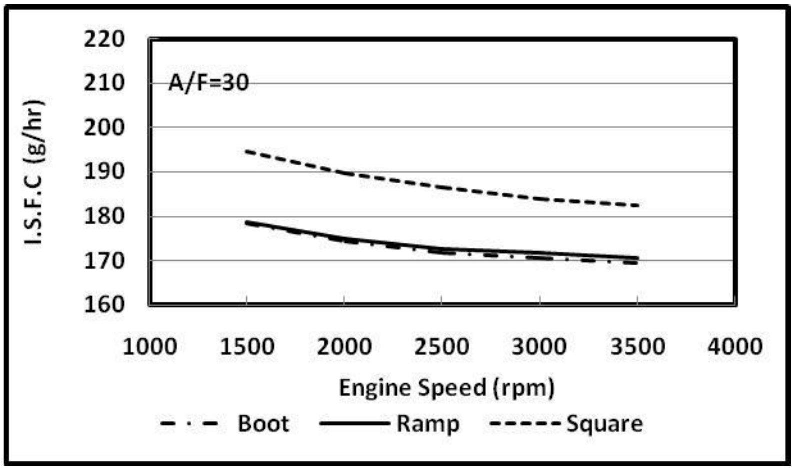

Figure (9): The effect of fuel injection patterns on the indicated specific fuel consumption.

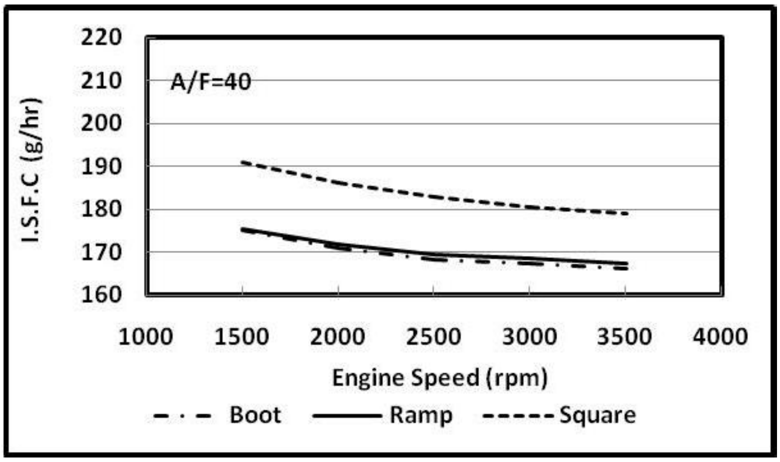

Figure (10): The effect of fuel injection patterns on the indicated specific fuel consumption.

Figures (11,12 and 13) show the variations of the heat loss for three patterns of fuel injection rate with respect to engine speeds at A/F of (20,30 and 40).It can be seen that the ramp profile leads to less heat loss .The (square and boot) profiles have more heat loss due to the more time availability for heat transferring .

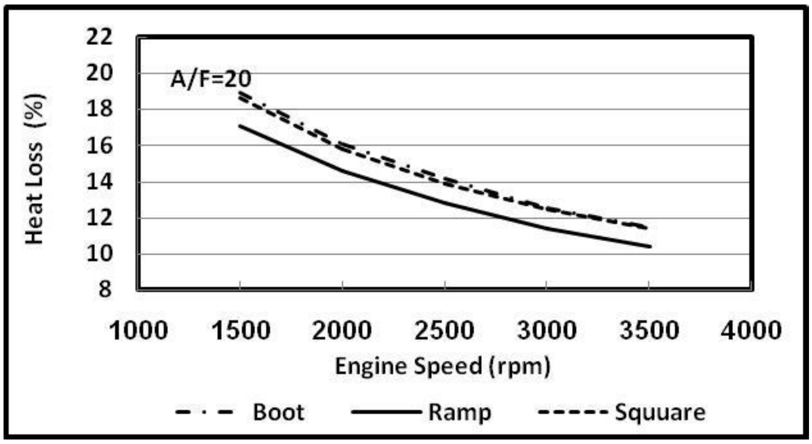

Figure (11): The effect of fuel injection patterns on the indicated specific fuel consumption.

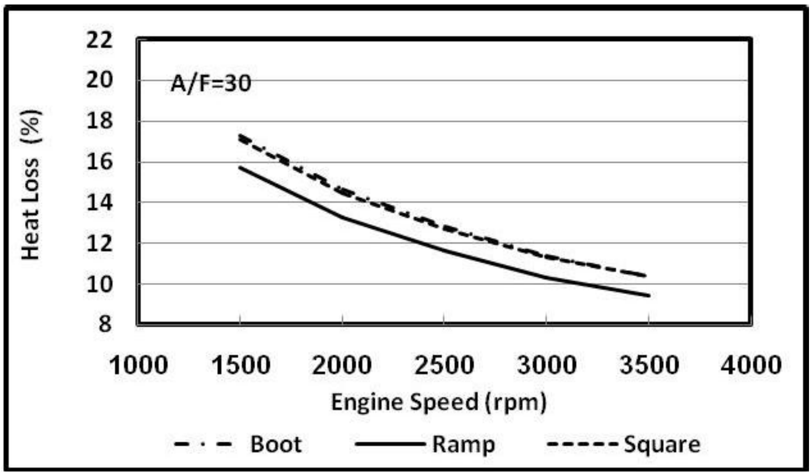

Figure (12): The effect of fuel injection patterns on the indicated specific fuel consumption.

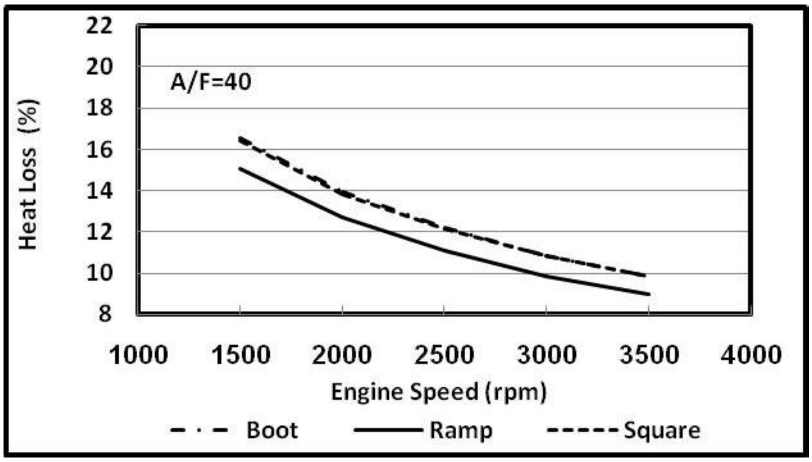

Figure (13): The effect of fuel injection patterns on the indicated specific fuel consumption.

Figures (14 and 15) show the variations of the cylinder gas temperature for three patterns of fuel injection rate at speed of (2500) rpm and A/F of (20 and 30) with respect to crank angle as a typical case. It can be observed that the boot profile leads to highest temperature while the (square and ramp) profiles have less effect.

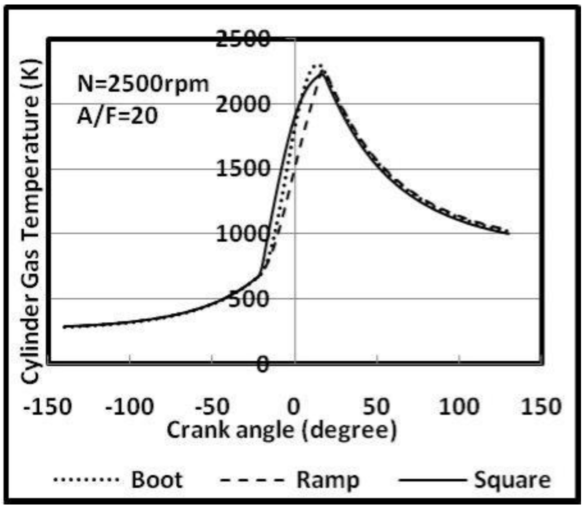

Figure (14): The effect of fuel injection patterns on the cylinder gas temperature.

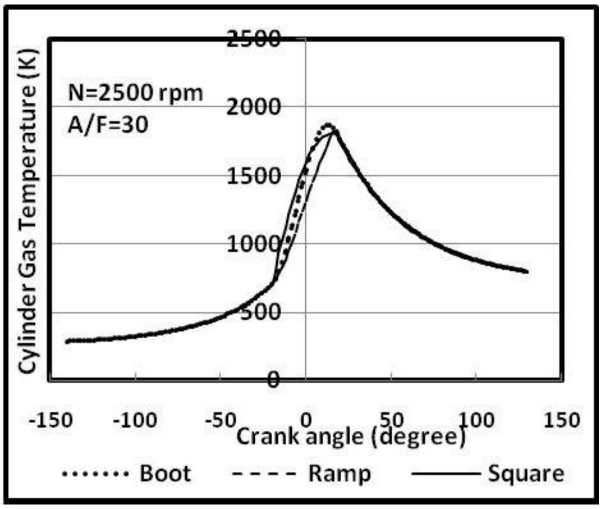

Figure (15): The effect of fuel injection pattern on the cylinder gas temperature.

Figures (16 and 17) indicate the variations of the cylinder gas pressure for three patterns of fuel injection rate at speed of (2500) rpm and A/F of (20 and 30) with respect to crank angle. It can be observed that the boot and square profiles lead to highest pressure while the (ramp) profile has less effect due to the small amount of fuel injection at the first half of the injection duration which cause smaller burning rate.

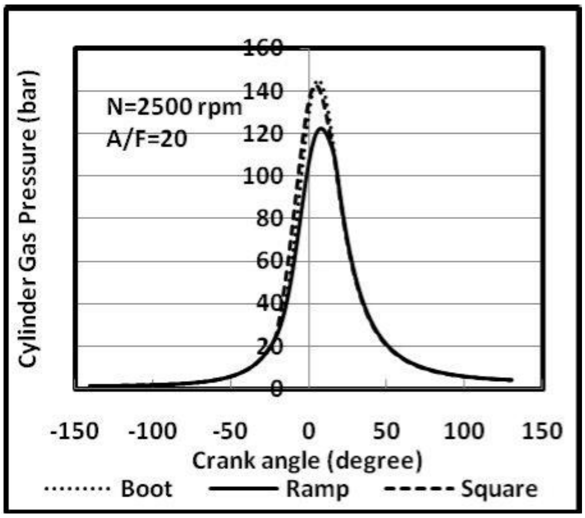

Figure (16): The effect of fuel injection patterns on the cylinder gas pressure

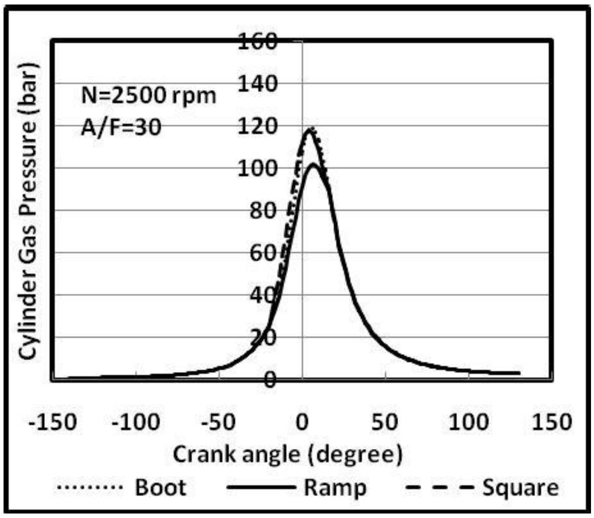

Figure (17): The effect of fuel injection patterns on the cylinder gas pressure

5. Conclusions :

From the theoretical results that have been obtained in the present study for diesel engine performance by using computer simulation method; the following conclusion can be stated and as follows:

1- The indicated power is greater relatively in cases of using (boot & ramp) profiles.
2- The thermal efficiency was improved relatively by using (boot and ramp) profiles.
3- The (boot and ramp) profiles have a positive effect on specific fuel consumption.
4- Minimum heat loss is obtained when (ramp) profile is used.
5- The maximum pressure and temperature in the cylinder is obtained when the boot model is used.
6- Finally, it can be concluded that using (boot or ramp) profile has a positive effect on diesel engine performance.

6. References :

1. B. Rajendra Prasath et al "Two-zone modeling of diesel / biodiesel blended fuel operated ceramic coated direct injection diesel engine" International Journal of Energy and Environment, Vol. 1, Issue 6, 2010 pp.1039-1056.
2. C.D. Rakopoulos "The Effect of Various Dynamic, Thermodynamic and Design Parameters on the Performance of a Turbocharged Diesel Engine Operating under Transient Load Conditions" SAE Int. 2004.
3. C. D. Rakopoulos and E. G. Giakoumis "Prediction of friction development during transient diesel engine operation using a detailed model Int. J. Vehicle Design, Vol. 44, No. 12, 2007.
4. Khalefa R. A. "Construction of A Mathematic Model for Simulating the Effect of Loads (A/F) on Fuel Injection Duration and Performance of the 4-stroke single cylinder diesel engine type (ANDORIA) "*the twelfth Conference of the Foundation of Technical Education*, Baghdad,Iraq,2010.
5. Dushyant Pathak " *Use of a Thermodynamic Cycle Simulation to Determine the Difference Between a Propane-Fuelled Engine and an ISO-Octane-Fuelled Engine*", M.sc, Thesis, Texas A&M University, 2005.
6. Donepudi Jagadish, Ravi Kumar Puli and K. Madhu Murthy"Zero Dimensional Simulation of Combustion Process of a DI Diesel Engine Fuelled With Biofuels " *World Academy of Science, Engineering and Technology*,56, 2011.
7. E. Abu-Nada, et al ,"Performance of Spark Ignition Engine Under Effect of Friction Using Gas Mixture Model" *Jour. of the energy*, institute vol. 82,No. 3,2009.
8. Rahim Ibrahim, et. Al. "Performance of an Otto Engine with Volumetric Efficiency" Journal of American science. 6,3,2010.
9. Yousif S. Philip "*Simulation of a Single Cylinder Engine Including Intake and Exhaust Systems*", M.sc, Thesis, University of Technology, Baghdad,1989.
10. Abbas S. Hussin "*Prediction of Heat Release in Diesel Engine* "A Thesis Submitted for Degree of Master , University of Mosul, College of Engineering ,Mosul, 1984.

11. Heywood, J.B."*Internal Combustion Engine Fundamentals*" McGraw-Hill, New York, 1988.
12. Nehmer D.A. Reitz R.D. “Measurement of the Effect of Injection Rate and Split Injections on Diesel Engine Soot and NOx Emissions,” *SAE Technical Paper* 940668, 1994.
13. Ghaffarpour M. Baranescu R. “ NOx Reduction Using Injection Rate Shaping and Intercooling in Diesel Engines” *SAE Technical Paper* 960845, 1996
14. Tanabe K. Kohketsu S. Mori K. Kawai K. “Innovative Injection Rate Control with Next Generation Common Rail Fuel Injection System ” *Proceedings of Seoul FISITA World Automotive Congress,* 2000.
15. Ikegami M. Koichiro N. Tanaka S. Yamane K. “Fuel Injection Rate Shaping and Its Effect on Exhaust Emissions in a Direct- Injection Diesel Engine Using a Spool Acceleration Type Injection System,” *SAE Technical Paper* 970347, 1997.
16. Benajes J. Payri R. Molina S. Soare V. “Investigation of the Influence of Injection Rate Shaping on the Spray Characteristics in a Diesel Common Rail System Equipped with a Piston Amplifier,” *Transactions of the ASME*, 2005.
17. Erlach H. Chmela F. Cartellieri W. Herzog P. “Pressure Modulated Injection and Its Effect on Combustion and Emissions of a HD Diesel Engine,” *SAE Technical Paper* 952059 1995.
18. Desantes J. M. Benajes, J. Molina S.,Gonzalez C. A."The Modification of Fuel Injection Rate in Heavy-Duty Diesel Engines Part 1:Effect on Engine Performance and Emissions "*Journal of Applied Thermal Engineering* ,24,pp2701-2714,2004.
19. Desantes J. M. Benajes, J. Molina S. Gonzalez C. A."The Modification of Fuel Injection Rate in Heavy-Duty Diesel Engines Part 2:Effect on Engine Performance and Emissions "*Journal of Applied Thermal Engineering* ,24,pp2715-2726,2004.
20. Khalefa R. A. " Performance of a Small 4-Stroke Single Cylinder Diesel Engine Under Effect of the Friction Losses" RIP, *International Journal of Applied Engineering Research,* Vol. 8, No.6 (2013) pp. 675-687.
21. Willard. Pulkrabek " *Engineering Fundamentals of the Internal Combustion Engines*" , 1997, Prentice-Hall-Inc.