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The Effect of Spark Timing on the Spark Ignition Engine Performance

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Abstract

In this work the effect of spark timing on the spark ignition engines is investigated by computer simulation and experimental test for speeds of (1500,2000,2500,3000 and 3500)rpm at spark timing of (20^o,30^o,40^o,50^o and 60^o) before TDC for each speed. This is done in order to find a suitable mathematical expression for spark ignition advancing with respect to the speed of the engine to predict the correct ignition advance as in real engines .The results showed that the method of using a mathematical expression is more realistic and reasonable comparing with the results obtained by other workers.

Key words: **S.I. Engine, Engine Performance, Engine Simulation, Spark Timing**

تأثير توقيت الشرارة على أداء محركات الإشعال بالقدح

الخلاصة

يتضمن البحث دراسة تأثير توقيت الشرارة على أداء محركات الإشعال بالقدح باستخدام الحاسبة الالكترونية (المحاكاة) والاختبارات العملية للسرعات (1500,2000,2500,3000,3500) دورة/دقيقة للتوقيتات (20,30,40,50,60) درجة قبل النقطة الميتة العليا لكل سرعة من السرع أعلاه وذلك لغرض استنباط معادلة رياضية مناسبة لإيجاد وتخمين زاوية القدح التي تناسب السرعة الدورانية للمحرك كما في المحركات الحقيقية. أظهرت النتائج بان استخدام المعادلة الرياضية هي اقرب إلى واقع وظروف اشتغال المحرك(من الناحية العملية) وبالمقارنة مع أعمال بحثية لآخرون .

الكلمات الدالة: محركات الإشعال بالقدح ,أداء المحركات ,محاكاة المحرك ,توقيت الشرارة

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)
N Engine Speed (R.P.M)

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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.

Introduction

A lot of researches carried out to study the effect of engine speed on the spark ignition engine performance by computer simulation without taking into account the fact of proportionality of spark timing change with the engine speed. If the optimum ignition advance angle is elected for each speed the performance of the engine will be improved. In the present work an important parameter which is the spark timing taken into account, this parameter has significant affect on the operating and performance of spark ignition engines. In modern high speed carburetor engines with compression ratio of (8-9) the maximum power is usually reached when the maximum pressure is obtained at an angle of (12-15^o) after TDC[1].If combustion is to be completed at 15^o after TDC then the ignition should occurs at about 20^o before TDC. If ignition is too early the cylinder pressure will increase to undesirable levels before TDC and work will be wasted in the compression stroke. If ignition is late, peak pressure will not occur early enough and work will be lost at the start of the power stroke due to lower pressure. Actual ignition timing is typically anywhere from 10^o to 30^o b TDC depending on the fuel used ,engine geometry and engine speed[2]. Earlier engines (carburetor) used a mechanical timing adjustment system that consisted of a spring loaded ignition distributor that changed with engine speed due to centrifugal forces .Modern engines automatically adjust ignition timing with electronic controls. These not only use engine speed to set timing but also sense and make fine adjustment for knock and incorrect exhaust emissions.[2]

By applying the first law of thermodynamics with aid of computer the process of combustion, work done, variation in internal energies and heat losses are carried out as a cycle

calculations using a thermodynamic model step by step method by taking an increment of time (crank angle)[3].

The effect of spark time is investigated to find a suitable

mathematical expression and comparing the results with that obtained by other workers and with experimental values for speeds (1500, 2000, 2500, 3000 and 3500) rpm.

The data required are cylinder geometry (bore and stroke), connecting rod length, valve timing, compression ratio, air/fuel ratio, and initial conditions such as pressure and temperature at the beginning of the calculations. Whence the above data are available the program can predict the following parameters: Power,

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Brake Specific Fuel Consumption, Thermal efficiency and Heat losses during the cycle.

Obviously a complete power cycle of a four-stroke single cylinder engine starts with closing the inlet valve and ends with the exhaust valve opening.

Aim and methodology of the Research

The aim of this work is to study the effect of spark timing and using an expression to simulate the real engine operation and performance by:

1-Experimental study for measuring the parameters of the engine.

2-Theoritical study using mathematical model with aid of a computer program simulation.

3-Comparing the results to optimizing the perfect spark timing.

Experimental work and Instrumentation

An experiment was carried out on a 4-stroke single cylinder air cooled spark ignition engine type (T D 110) to measure the torque by using the (hydraulic dynamometer) at a variable operating conditions .

The fuel consumption was measured by using the cylindrical scales (volumetric scales) with respect to time and then converted to (mass/time) measurement using the following equation:

$$\dot{m}_f = \frac{\rho_f * \dot{V}_f * 3600}{t * 1000} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

The air consumption was measured by using the viscous flow meter, which consists of air box to reduce the vibration of the air during the engine work. From the variation of the pressure between the inlet and outlet using U type manometer, the air consumption was measured with taking into account the calibration of the device by taking the correction factor of the manufacturing manual as:

$$(\dot{m}_a)_{new} = \dot{m}_a * K1 \ldots\ldots\ldots\ldots(2)$$

$$where$$

$$K1 = 3584 * p_a \left(\frac{T + 114}{T^{2.5}}\right)$$

and

$(\dot{m}_a)_{new}$ =the air consumption (kg/hr)

$\dot{m}_a$ = the amount of the air at pressure of (101.3 kpa) and temperature of (20^{o}C)

T= the air temperature

p_a =atmospheric pressure (bar)

The speed was measured by tachometer, and the temperature measured by using thermocouples type (Ni-Cr), (Ni-Al). The calibration of the instruments which used was done before the experiment begins in order to ensure correct readings.

<u>Test and measuring process</u>

The experimental part of this work is beginning after warming up time with using the required instruments.

The operating conditions of the engine was at speeds ranged (1500-3500) r.p.m with gasoline fuel (octane number of 85) (the specification of the fuel was from AL- Beije refining company) at constant A/F ratio of 15, inlet mixture temperature is assumed (20^{o}C) which was the measured inlet air temperature and at (W.O.T).

Table (1) gives the engine specification that was used for testing purpose in this work, and during the measuring process the time of the stability of the engine (load, speed) is taken into account .

Assumptions

The following assumptions are taken into account:

1- Considering the temperature of the mixture at the beginning of the calculations is the same as the measured mixture temperature of the experimental part.

2- The calculation increment ($\Delta\alpha$) is one degree of the crank during the combustion period.

3- The combustion chamber is (burnt zone), the pressure and temperature of the cylinder content is homogenous during the step.
4- The cylinder content is treated as ideal gases.
5- The temperature of the cylinder content varies at each step and so variable specific heats considered.

Theoretical analysis

The simulation of the active strokes of spark ignition engine was programmed using (Quick Basic) language where the active strokes are divided as follows:

1- Compression stroke
2- Combustion stages
3- Expansion stroke

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

1- Calculating the volume by using the following equation [4]:

$$V(\theta) = Vc + (Vs/2) * K_2 \text{.................}(3)$$

$$K_2 = [1 - \cos(\theta) + (2L/S) - K_3$$

$$K_3 = \sqrt{(2L/S)^2 - \sin^2\theta}]$$

Where
$V(\theta)$ = Instantaneous Cylinder Volume, V_S = Swept Volume and V_C = Clearance Volume.

2- Estimating the temperature at the end of the step using [5]:

$$T_2 = T_1 * [\frac{V_1}{V_2}]^{k-1} = T_1 * [\frac{V_1}{V_2}]^{\frac{R_{mol}}{C_V(T_1)}} \text{.......(4)}$$

3-Estimating the pressure at the end of the step using [5]:

$$P_2 = [\frac{V_1}{V_2}] * [\frac{T_2}{T_1}] * P_1 \text{..................(5)}$$

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

$$e_i(T) = R_{mol}((\sum_{j=1}^{j=5} U_{i,j} * T^j) - T) \text{............(6)}$$

where e_i=the specific internal energy , and the total internal energy E(T) is:

$$E(T) = R_{mol}((\sum_{i=1}^{n} W_i ((\sum_{j=1}^{5} U_{i,j} * T^j) - T) \text{................(7)}$$

Where
E(T) = the total internal energy, $U_{i,j}$=polynomial coefficients[11] for mixture species i=1 to n, and W_i = number of moles of species .The following subscripts are given for the species:

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

The total number of moles (W_t) in the mixture is given [11]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 $C_V(T) = \frac{1}{W_t}\frac{\partial(E(T))}{\partial T}$ ....(8)

Differentiating eq.(7)with respect to (T) gives:

$$W_t * C_V(T) = R_{mol} \sum_{j=1}^{n} W_i ((\sum_{j=1}^{5} j * U_{i,j} * T^{j-i}) - 1) \text{............ (9)}$$

Equations (7) &(9) are general expressions for internal energy and specific heat for the mixture[11].

5-The work done is calculated[6] by:

$dW = P\,dV = ((P_1+P_2)/2)\,(V_2-V_1)$….(10)

6-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[3] as:

$dQ_{conv} - dW = dE = E(T_2) - E(T_1)$ or
$f(E) = E(T_2) - E(T_1) + dW - dQ_{conv}$…..... (11)

where $dQ_{conv.}$ is the amount of heat transferred through the cylinder walls which is calculated by using the (Eichelberg's) equations[7]as :

$dQ_{conv.} = h*A_s*(T - T_{wall})$………..... (12)

where

$h = 2.466*10^{-4}(U_p)^{1/3}(p)^{1/2}(T)^{1/2}$…......(13)

U_P = piston speed = (2*S*N)/60 , $T = (T_1+T_2)/2$ & $P = (P_1+ P_2)/2$
$A_s = (2A_p + \pi * D * X)$ where
A_P=the top surface area of the piston, D = Cylinder Diameter and

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

7- 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 (11) was solved numerically using "Newton Raphson's" method [11], 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:

$$(T_2)_n=(T_2)_{n-1}-f(E)_{n-1}/f'(E)_{n-1} \quad \text{.........} (14)$$

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

8- The delay periods calculated by using the following equations [8] :

$$D.P = \frac{360*(N)(R_f)}{U_T} \quad \text{...............}(15)$$

Where R_f is the flame radius calculated [12] by:

$$R_f = \{\frac{0.001*V_2}{\frac{2}{3}\pi}\}^{\frac{1}{3}} \quad \text{...............}(16)$$

and U_T is the turbulent flame front speed calculated by:

$$U_T = U_L * ff \quad \text{...............}(17)$$

where ff is the turbulent flame factor calculated by:

$$ff = 1+(0.0018*N) \quad \text{...............}(18)$$

and U_L is the laminar flame front speed is determined by using the experimental equation of (Kuehl) [9].

$$U_L = [\frac{1.087*10^6}{\{(\frac{10^4}{T_b})+(\frac{900}{T_{un}})\}^{4.939}}]*P^{-0.09876} \quad \text{...........}(19)$$

for $(\phi \le 1.0)$

$$T_b = T_{un} + 2500*(\phi)(X_r) \quad \text{...............}(20)$$

for $(\phi) > 1.0$

$$T_b = T_{un} + K_4 - K_5 \quad \text{...............}(21)$$

$$K_4 = 2500*(\phi)(X_r)$$

$$K_5 = 700*(X_r)(\phi - 1)$$

Where (ϕ) is the equivalent ratio which represented [12].by :

$$(\phi) = AF_{actual} / AF_{stoichiometric}$$

and $X_r = \rho_b / \rho_{un}$

Spark angle

The spark angle is proportional to engine speed and calculated in this research work by the following new estimated formula:

$$\theta_{SPARK} = -43.958 * Ln(RPM) + 292.99 \quad \text{........}(22)$$

Combustion Duration:

The combustion duration is proportional to engine speed which is calculated [10] by:

$$\Delta\theta_c = 40 + 5[\frac{N}{600} - 1] + [166\frac{Ycc}{Y} - 1.1]^2 \quad \text{..........}(23)$$

Y_{cc} is the correct fraction of the oxygen required .

Y is the number of moles of O_2 in the mixture .

and the rate of heat release is determined by using[3] :

$$dQ_{release} = (LHV)*(m_m)*(\frac{F}{A})/K_6 \quad \text{...................}(24)$$

$$K_6 = [2*(\theta_c)(1+\frac{F}{A})]\sin[\pi\frac{\theta - \theta_{spa}}{\theta_c}]$$

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

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

or

$$f(E) = (E_2 - E_1) + dW + K_7 \quad \text{...............}(25)$$

$$K_7 = dQ_{conv} - dQ_{release}$$

and the correction values of the all variables estimated during each step ,the cycle is beginning with the start of intake valve closing and before the start of a new step by increasing the angle by ($\Delta\alpha$) ,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) is greater than the exhaust valve open angle.

Finally the program inquire to save data in a file and calculates the following [3] :

1- Indicated power:

$$IP = \oint PdV * N \quad \ldots\ldots\ldots\ldots (26)$$

2-Brake power:

$$BP = \eta_m * IP \quad \ldots\ldots\ldots\ldots (27)$$

3-Indicated specific fuel consumption:

$$i.s.f.c = \frac{\dot{m}_f}{\oint pdv * N} \quad \ldots\ldots\ldots\ldots (28)$$

4-Brake specific fuel consumption:

$$b.s.f.c = \frac{i.s.f.c}{\eta_m} \quad \ldots\ldots\ldots\ldots (29)$$

5-Thermal efficiency:

$$\eta_{th} = \frac{\oint pdv * N}{\dot{m}_f * HHV * \eta_c} \quad \ldots\ldots\ldots\ldots (30)$$

Results and Discussion

In this study the predicted results of computer simulation before and after developing the program are plotted and compared with the results obtained by experimental tests.

Figures (1,2,3,4,5,6,7,8 ,10 and 11) represent the variations of cylinder gas pressure and temperature with crank angle for different engine speeds (1500,2000,2500,3000, and 3500) rpm at spark timing of (20,30,40,50 and 60) degrees before TDC for each speed .It observed that the increasing of the engine speed without advancing the spark timing reduces the maximum pressure and maximum temperature in the cylinder with reducing the indicated mean effective pressure and increasing the exhaust temperature .

Advancing the spark timing increase the maximum pressure and temperature of the cylinder gases for all speeds and alters the pressure and temperature at the initial moment of combustion and changes the position of maximum pressure and temperature. With a too great advancing angle, combustion develops mainly before TDC, and additional work is performed at the end of combustion.

Figures (9 and 12) represents the variations of cylinder gas pressure and temperature respectively with crank angle for different speeds by using an equation to simulate the advancing or retarding process of the spark timing as in real engines. This method is reasonable to predict the engine speed affect on the cylinder gas pressure and temperature, because with an increase in speed the growth of the angle of crankshaft travel corresponding to the first stage (ignition delay) and second stage (flame propagation) of combustion phases is compensated for by a greater ignition advance angle so that the effectiveness of the combustion process in these stages does not become worse.

Figures(13,14,15and 16) represents the variations of (brake power, thermal efficiency, indicated specific fuel consumption and heat loss) respectively with engine speed of (1500,2000, 2500, 3000 and 3500) at constant spark timing of (20,30,40,50 and 60)degrees b.TDC for each speed and with using the new mathematical model to evaluate the spark timing automatically during the computer simulation for any speed with results obtained by experimental work. It observed that the increasing of the engine speed:

1-Increases the brake power and the results which are obtained by using of the new mathematical model (Variable Spark Timing) is reasonable results.

2-Increases the brake thermal efficiency until the speed of 2500 rpm and than decreases due to the heat loss by dissociation. The predicted results by the new mathematical model (Var.S.T)for spark timing evaluation is reasonable comparing with other predicted results. It is observed that the efficiency grows if the optimum ignition advance is selected for each speed

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power and then increases due to the decreasing of the indicated mean effective pressure.
3-Decreases the ISFC until speed of 2500 due to increasing the indicated.
4-Decreases the heat loss due to less time for heat transfer by the cylinder walls .It is observed that the results which are obtained by the new method (Var.S.T) more realistic.

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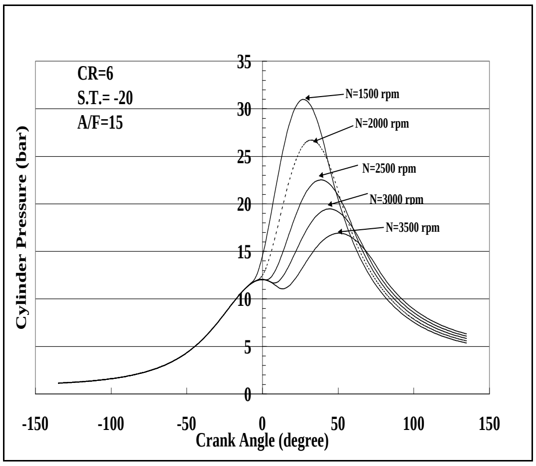

Figure(1) Effect of Engine Speed on Cylinder Gas Pressure at Spark Timing of (20 degree) b.TDC.

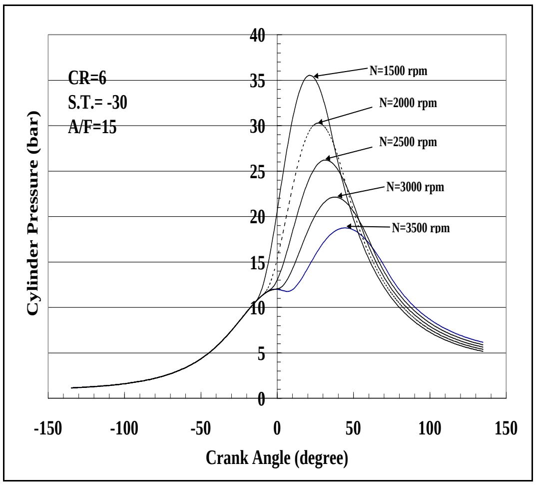

Figure (2) Effect of Engine Speed on Cylinder Gas Pressure at Spark Timing of (30 degree) b.TDC.

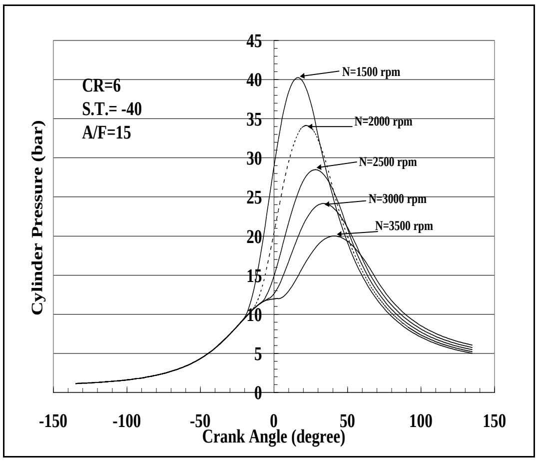

Figure (3) Effect of Engine Speed on Cylinder Gas Pressure at Spark Timing of (40 degree) b.TDC.

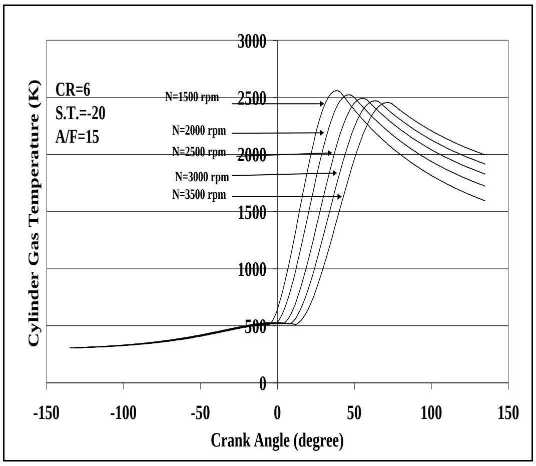

Figure (4) Effect of Engine Speed on Cylinder Gas Temperature at Spark Timing of (20 degree) b.TDC.

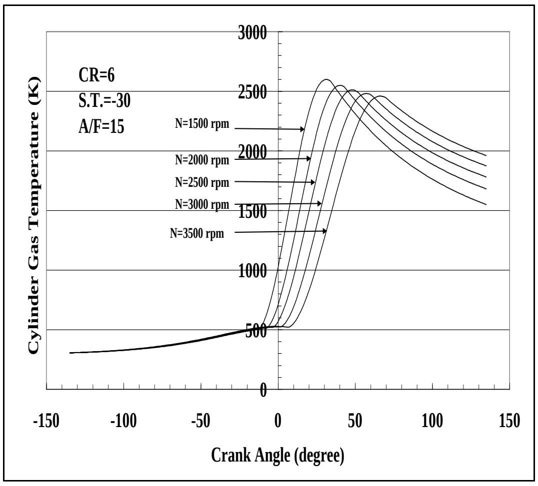

Figure (5) Effect of Engine Speed on Cylinder Gas Temperature at Spark Timing of (30 degree) b.TDC.

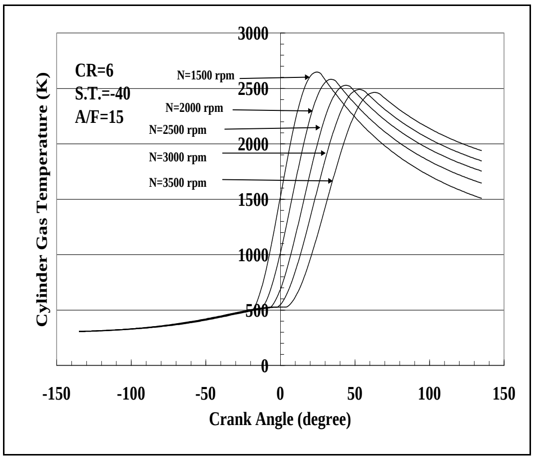

Figure (6) Effect of Engine Speed on Cylinder Gas Temperature at Spark Timing of (40 degree) b.TDC.

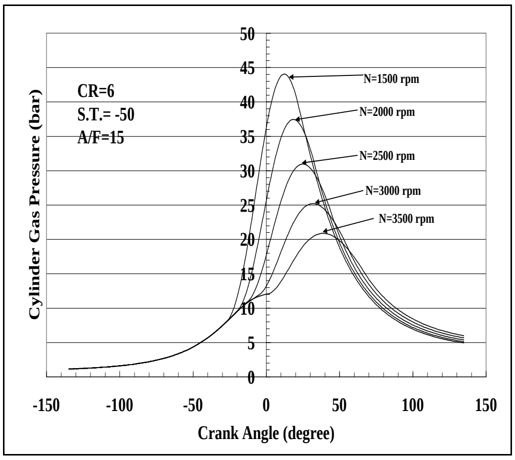

Figure (٧) Effect of Engine Speed on Cylinder Gas Pressure at Spark Timing of (40 degree) b.TDC.

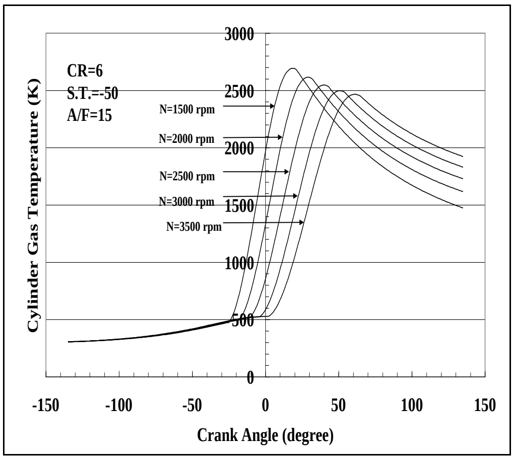

Figure (١٠) Effect of Engine Speed on Cylinder Gas Temperature at Spark Timing of (40 degree) b.TDC.

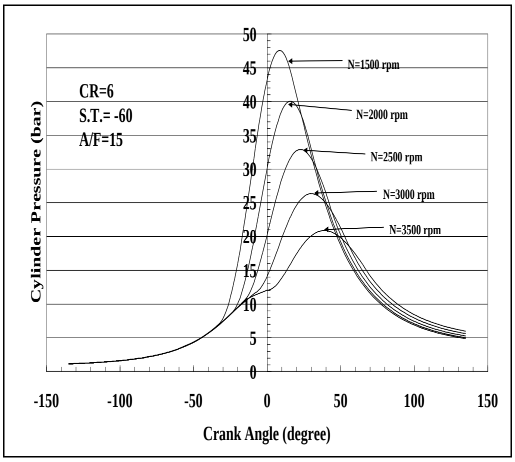

Figure (٨) Effect of Engine Speed on Cylinder Gas Pressure at Spark Timing of (40 degree) b.TDC.

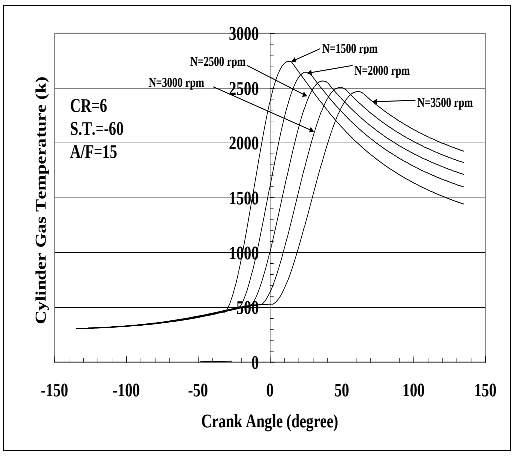

Figure (١١) Effect of Engine Speed on Cylinder Gas Temperature at Spark Timing of (40 degree) b.TDC.

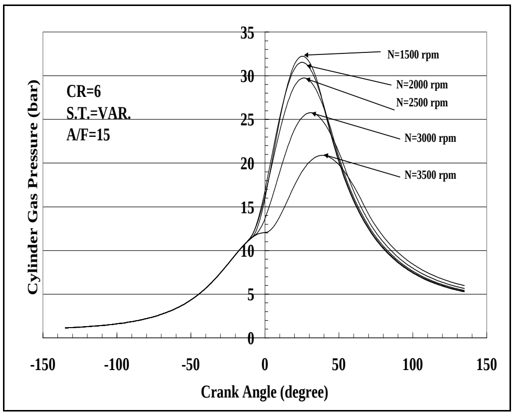

Figure (٩) Effect of Engine Speed on Cylinder Gas Pressure When Spark Timing is Variable by using the New Expression:

$$\theta_{SPARK} = -43.958 * Ln(RPM) + 292.99$$

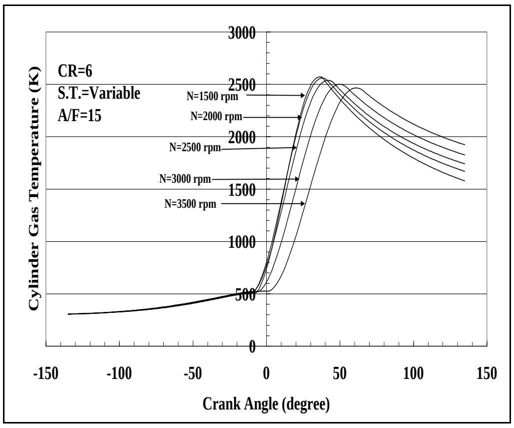

Figure (١٢) Effect of Engine Speed on Cylinder Gas Temperature When Spark Timing is Variable by using the New Expression:

$$\theta_{SPARK} = -43.958 * Ln(RPM) + 292.99$$

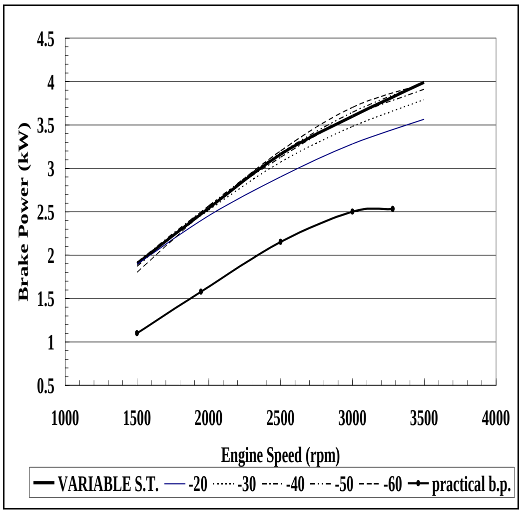

Figure (13)Effect of Engine Speed on The Brake Power at Spark Timing of(-20, -30, -40, -50,-60), Variable Spark Timing and Practical Operation of The Engine.

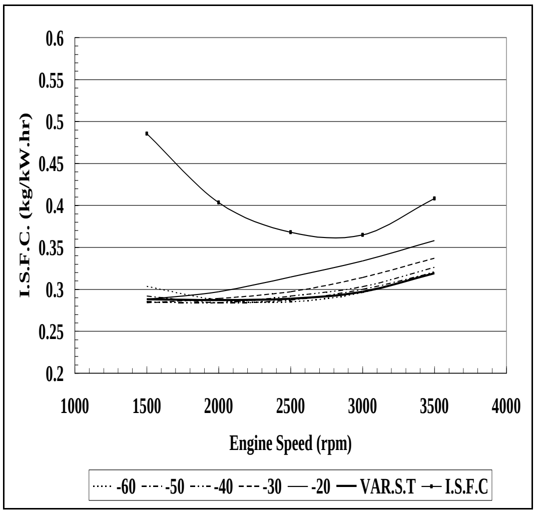

Figure (15)Effect of Engine Speed on The I.S.F.C. at Spark Timing of (-20, -30,- 40,-50 , -60), Variable Spark Timing and Practical Operation of The Engine.

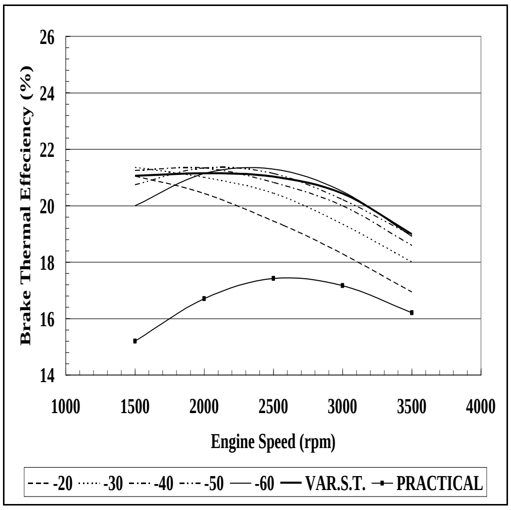

Figure (1٤)Effect of Engine Speed on The Brake Thermal Efficiency at Spark Timing of(-20, -30, -40, -50,-60), Variable Spark Timing and Practical Operation of The Engine.

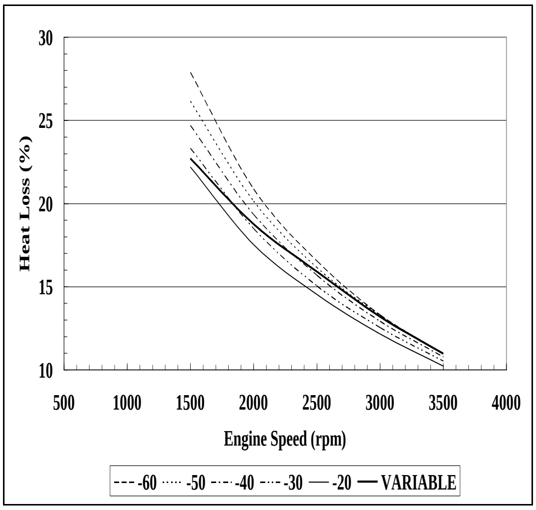

Figure (16)Effect of Engine Speed on The Heat Loss at Spark Timing of (-20,-30, -40,-50 , -60) and Variable Spark Timing.