THE EFFECT OF VALVE TIMING ON THE LOW SPEED DIESEL ENGINE PERFORMANCE [+]

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Abstract:

In this research, it was aimed to study the effect of valve timing on the low speed direct injection 4-stroke diesel engine (Indoria) performance . For this purpose the simulation method was used by developing a computer program (QBASIC Language) to predict the effect of (IVC) and (EVO) on the performance parameters (Indicated power, Indicated thermal efficiency, Indicated specific fuel consumption and heat loss) at speeds of (800, 1000 and 1200)rpm. The results showed that the valve timing has a significant effect on the performance parameters. The best results were obtained at (60^o) a.BDC for intake valve closing and (40^o) b.BDC for exhaust valve opening.

Key word: diesel engine ,valve timing ,low speed engines, performance parameters , simulation .

تأثير توقيت الصمام على أداء محركات الديزل ذات السرعات الواطئة

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المستخلص:

ان الهدف من هذا البحث هو دراسة تأثير توقيت الصمامات على أداء محرك ديزل رباعي الأشواط ذو سرعة واطئة و حقن مباشر (اندوريا) . لهذا الغرض تم استخدام طريقة المحاكاة عن طريق تطوير برنامج حاسبة الكترونية (لغة QBASIC) لاستنتاج تأثير توقيت غلق صمام الإدخال وتأثير توقيت فتح صمام العادم على معاملات الأداء (القدرة البيانية ،الكفاءة الحرارية البيانية،استهلاك الوقود النوعي البياني والخسائر الحرارية) للسرعات(800، 1000،1200) دورة/دقيقة. وأظهرت النتائج تأثيرا ملحوظا على معاملات ادء المحرك .وتم الحصول على أفضل النتائج عند غلق صمام الإدخال بعد النقطة الميتة السفلى ب(60°) وعند (40°) لفتح صمام العادم قبل النقطة الميتة السفلى لجميـــع السرعات.

كلمات الفتاح: محركات الديزل ،توقيت الصمامات، محركات ذات السرعة الواطئة،معاملات الأداء،محاكاة.

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)

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$dQ_{Conv.}$	Heat Losses During the Step	(kJ)
$dQ_{release.}$	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)
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)
W	Engine Work	(kJ)
W_i	Species Mole Number	(kmol)
x	Carbon Atomic Number	(kmol)
y	Hydrogen Atomic Number	(kmol)
ycc	Air to Fuel ratio	----
θ	Crank Shaft Angle	(degree)
θ_{dur}	Duration of the Injection	(degree)
α	Interval of Crank Angle	(degree)
λ	Excess Air Factor	-------

Introduction:

The increase in power requirements for operating and development process of the different sectors make the internal combustion engines (spark ignition engines and compression ignition engines) an important source .Because of their simplicity ,ruggedness and high power/weight ratio, these two types have found wide application in transportation (land, sea, and air) and the power generation [1]. A lot of researches are conducted and a great of efforts are consumed for improving the diesel engine performance (increasing power and efficiency , reducing fuel combustion , heat loss and emissions). The practical method is very difficult and requires more time and it is very expensive ,due to these reasons the simulation method was used by a lot of researchers [2,3,4,5,6,7,8,9,10,11] which is simple, fast , more economic, and less pollution with reasonable results .

The valve timing of an engine has a significant effects on the engine performance because the thermodynamic conditions during the closed cycle (compression ,combustion and expansion) can be directly controlled by adjusting the intake valve opening and closing which defines the total intake mass flow rate and the effective compression ratio of the engine[12] . Camshaft of a four stroke cycle rotates at half speed of the crank shaft . The different speeds of the crankshaft and camshaft can be the cause of some confusion when describing the timing of valve opening and closing with angles, as 360°of crankshaft rotation is equivalent to 180°of camshaft rotation.

The objective of this research is to predict the effect of inlet and exhaust valve timing on the performance(indicated power, indicated thermal efficiency, indicated specific fuel consumption and heat losses) of a 4-stroke, single cylinder, direct injection which is relatively low speed (maximum speed is limited at 1200rpm type indoria [13])

Theoretical analysis:

In the present research the fully thermodynamic cycle is considered by using a proper equations for analyzing the cylinder content beginning with intake valve closing (compression, combustion and expansion) until the exhaust valve opening , during this period the thermodynamic properties are calculated for each crank angle interval ($\Delta\alpha$) which is (2 deg.). 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 [14].

The present theoretical study was conducted using a computer simulation with "Quick Basic" language on an engine has the technical specification as in table (1).

Table (1). Specification of the Engine and Fuel Used.[13]

Bore	**120mm**
Stroke Length	**160mm**
Connecting Rod Length	**250mm**
Compression Ratio	**17:1**
Displacement Volume	**1.81Liter**
Inlet Valve Closing (IVC)	**50°aBDC**
Exhaust Valve Opening (EVO)	**46° bBDC**
Fuel Injection Timing	**28° bTDC**
Diesel Fuel	**$C_{14.4}H_{24.9}$**
Heating Value of the Fuel	**39MJ**
Fuel Cetane Number	**49**

The equation that represents of fuel combustion with C-H-O-N is [14]:

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

$$\left(\frac{A}{F}\right)_{Stoch.} = ycc = x + \left(\frac{y}{4}\right) - \left(\frac{z}{2}\right)$$

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

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

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

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 [4,14,15,16,17]:

$$V(\theta) = Vc + \left(\frac{Vs}{2}\right)[1 - \cos(\theta) + \left(\frac{2L}{S}\right) - \sqrt{(\frac{2L}{S})^2 - \sin^2\theta}] \quad (4)$$

The initial pressure and temperature at the beginning of the compression process is calculated as [4,14,15,16,17]:

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

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

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 [4,9,10,14 and 15]:

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

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

Where $E_{(T)}$ is the total internal energy , $U_{i,j}$ is the polynomial coefficients for mixture species in the cylinder [2,3,14]. i =1 to n & W_i is the species mole numbers where:

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

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

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

The specific heats at constant (volume & pressure) for each species is calculated by[11]:

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

The derivative of equation (5) is

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

Where the general expression for the internal energy and specific heat are represented by equations (7&10).

The work done during the step is calculated as [4,9,10 and 15]:

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

The first law of thermodynamics can be applied during the compression stroke before the combustion starting as:

$$dQ_{conv} - dW = dE = E(T_2) - E(T_1) \text{ or } f(E) = E(T_2) - E(T_1) + dW - Q_{conv}. \quad (13)$$

Where dQ_{con} is the heat loss through the cylinder walls calculated by (Eichelberg's equation) as: $dQ_{conv.} = h*A_s*(T - T_{wall})$ (14)

Where h is the heat transfer coefficient and calculated as:

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

$U_P = \frac{2SN}{60}$, $T = 0.5(T_1 + T_2)$ & $P = 0.5(P_1 + P_2)$

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

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

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 [9,10,14,15], and a solution was obtained $f(E) = 0$. And during the combustion energy equation can be written as [4,14,15]:

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

To finding the correct value of (T_2), both sides of the above equation should be balanced and as shown below :

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

By solving the equation (17) numerically and finding the value of (ER) than comparing it with the accuracy required , if the obtained value of (ER) is less than the accuracy then the correct value of (T_2) has been established , otherwise a new value of (T) is calculated for new internal energy and (Cv) values by using (Newton-Raphson technique) [4,14,15] .

$$ER' = C_V *(T_2)* tmp \qquad (18)$$

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:

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

Fuel injection rate:

Calculating of the mass of fuel injected per each crank angle interval ($\Delta\alpha$)which was (2degree) can be done by using the following expression [14,15]:

$$TFIR = \frac{Ma*\Phi}{\left(\frac{A}{F}\right)stoich.} \qquad (20)$$

Where

TFIR = total mass of fuel injection per cycle, and

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 [16].

$$\Phi = \frac{\frac{A}{F}stoich.}{\frac{A}{F}actual} \qquad (21)$$

$$FIR = \left(\frac{TFIR}{\theta_{dur.}}\right) \qquad (22)$$

FIR = mass of fuel injected per degree of crank angle.

The time interval between the start of(fuel injection and combustion)called (Ignition delay) . There are many correlations for calculating and predicting the ignition delay in which the earliest was Wolfer (1938) [19, 20,21]. The mathematical model which was used in the present work shown in equation which was used by [15,17]:

$$ID = \left(\frac{40}{CN}\right)^{0.69} * EXP\left(\frac{4644}{T_2}\right) * 0.000343 * N * (P_2)^{-0.388} \qquad (23)$$

Where

ID = ignition delay (degree of crank angle) ,T=temperature (k) and p=pressure (kpa)

The equations below can be used for calculating the indicated (power, thermal efficiency and specific fuel consumption specific fuel consumption)[11]:

$$IP = (Imep*A*L*N*Z)/(60*n) \qquad (24)$$

$$\eta_{ith} = IP/(\dot{m}_f*Q_{HV}) \qquad (25)$$

$$ISFC = \dot{m}_f / IP \qquad (26)$$

Results and discussions:

Figures (1,2) represent the variation of indicated power and indicated thermal efficiency with respect to intake valve timing for speeds of (800, 1000, 1200)rpm at A/F of (45) **.**It observed an increase in indicated power and indicated thermal efficiency with the reduction of inlet valve timing until the maximum indicated (power and thermal efficiency) are obtained between (55-60)degree (a.BDC). The increase in power and efficiency may be due to increasing the time for charging of the cylinder, which means that the intake valve remains open during the early part of a compression stroke to allow charging of the cylinder. If the intake valve remains open for a long period (after optimum point) it will reduces the compression ratio and leads to decreasing the indicated power and indicated thermal efficiency .

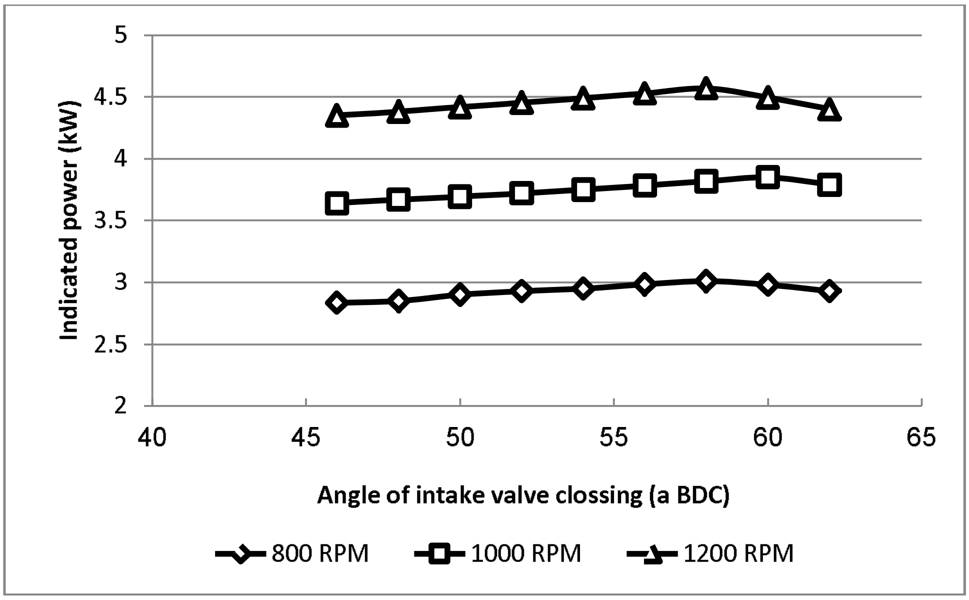

Fig.(1) the effect of intake valve timing on the indicated power.

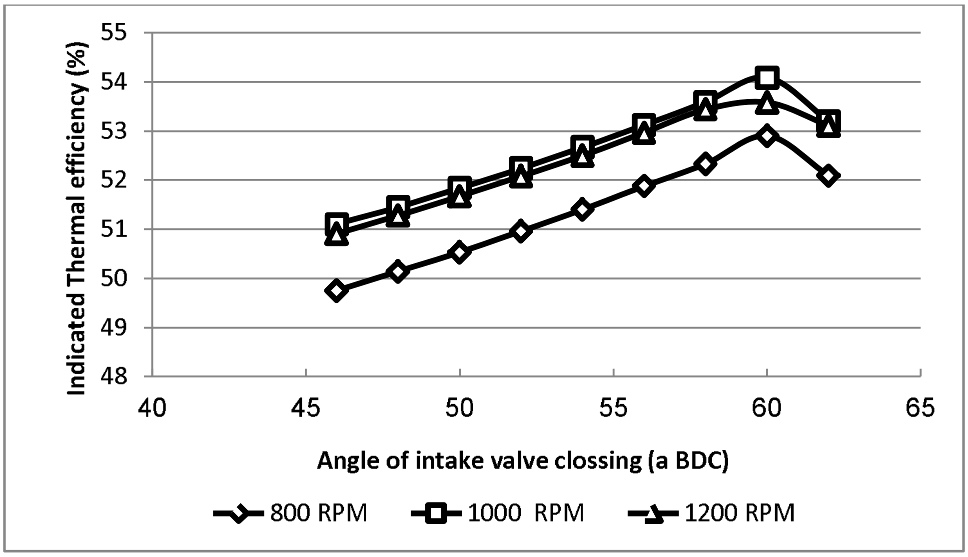

Fig.(2) the effect of intake valve timing on the indicated thermal efficiency .

Figure.(3) represents the variation of indicated specific fuel consumption and with respect to intake valve timing for speeds of (800, 1000, 1200)rpm at A/F of (45) **.**It observed that the minimum indicated specific fuel combustion (optimum point) is obtained at(60degree) (a.BDC) which means that the indicated work is greater . If the intake valve remains open for a long period (after optimum point) it will reduces the compression ratio and leads to decreasing the indicated power and increasing the fuel combustion .

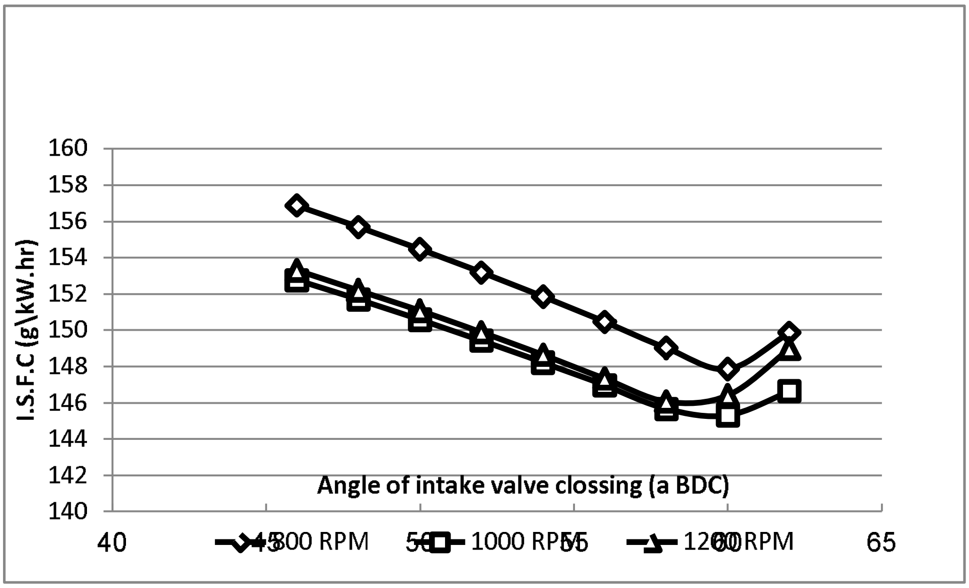

Fig.(3) the effect of intake valve timing on the indicated specific fuel combustion .

Figure.(4) represents the variation of heat losses with respect to intake valve timing for speeds of (800, 1000, 1200)rpm at A/F of (45) **.**It observed that the minimum heat losses (optimum point) is obtained at(60degree) (a.BDC) which means that the heat losses greater before the optimum point due to long time of compression stroke . If the intake valve remains open for a long period (after optimum point) it will reduces the heat losses .

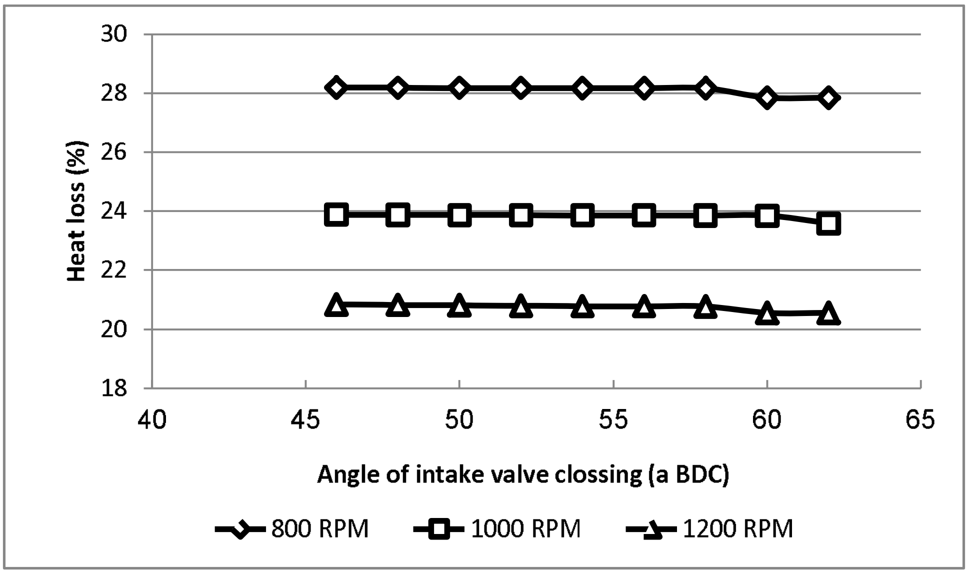

Fig.(4) the effect of intake valve timing on the heat loss .

Figures (5 and 6) represent the variation of indicated power and indicated thermal efficiency with respect to exhaust valve timing for speeds of (800, 1000, 1200)rpm at A/F of (45) **.**It observed that the maximum indicated (power and thermal efficiency) (optimum point) are obtained at (40)degree (b.BDC) which means that the earlier opening of the exhaust valve leads to maximum indicated (power and thermal efficiency) . If the exhaust valve remains closed until the BDC leads to increasing the power requirements for pumping the exhaust gases out of cylinder ,therefore the exhaust valve is opened before BDC to allow the most gases to escape in the blowdown process before the exhaust stroke is beginning .

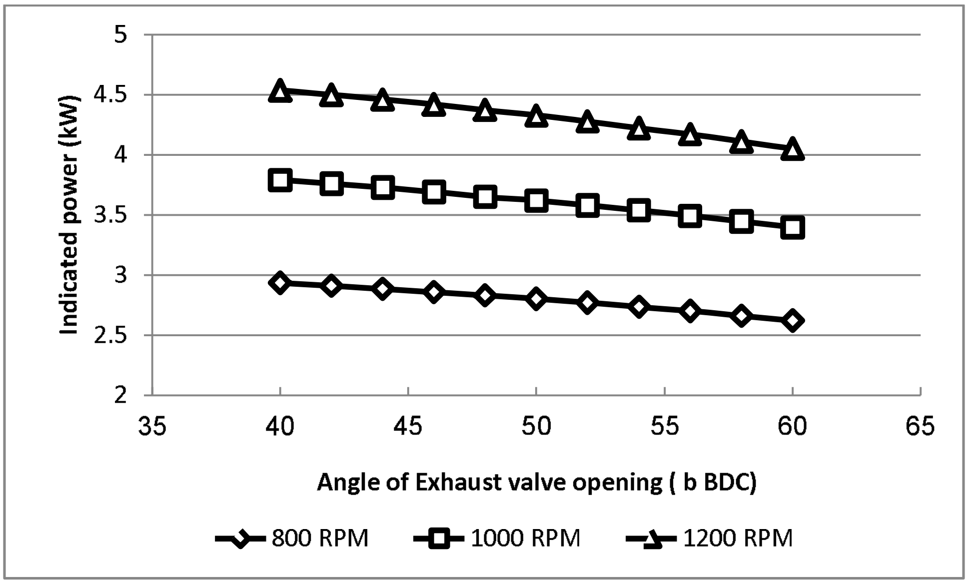

Fig.(5) the effect of exhaust valve timing on the indicated power.

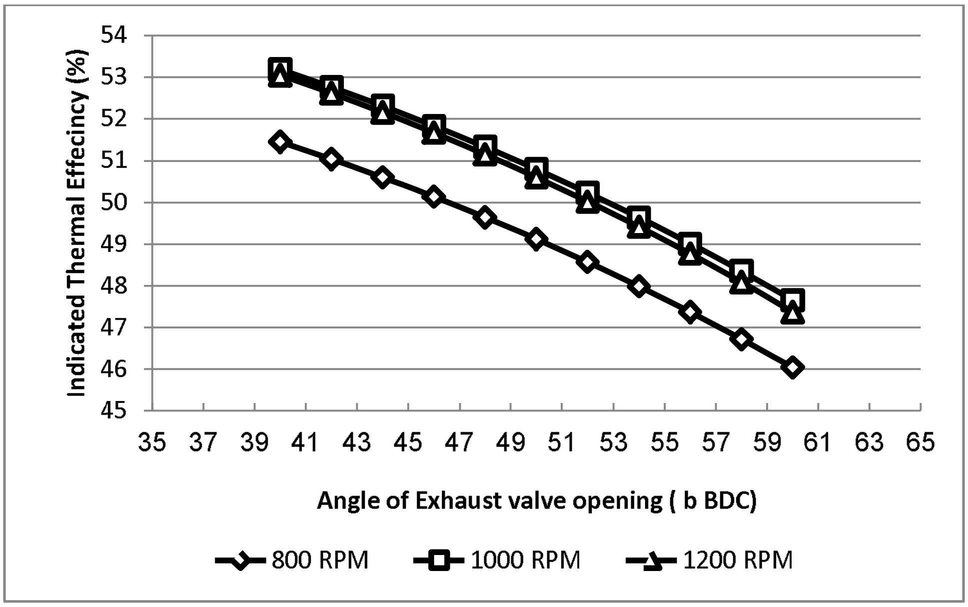

Fig.(6) the effect of exhaust valve timing on the indicated thermal efficiency .

Figure (7) represent the variation of indicated specific fuel combustion with respect to exhaust valve timing for speeds of (800, 1000, 1200)rpm at A/F of (45) **.**It observed that the minimum indicated specific fuel combustion is at (40 degree) bBDC and speeds of (800 and 1000)rpm, which means that the exhaust valve timing has a great effect on the indicated fuel combustion .

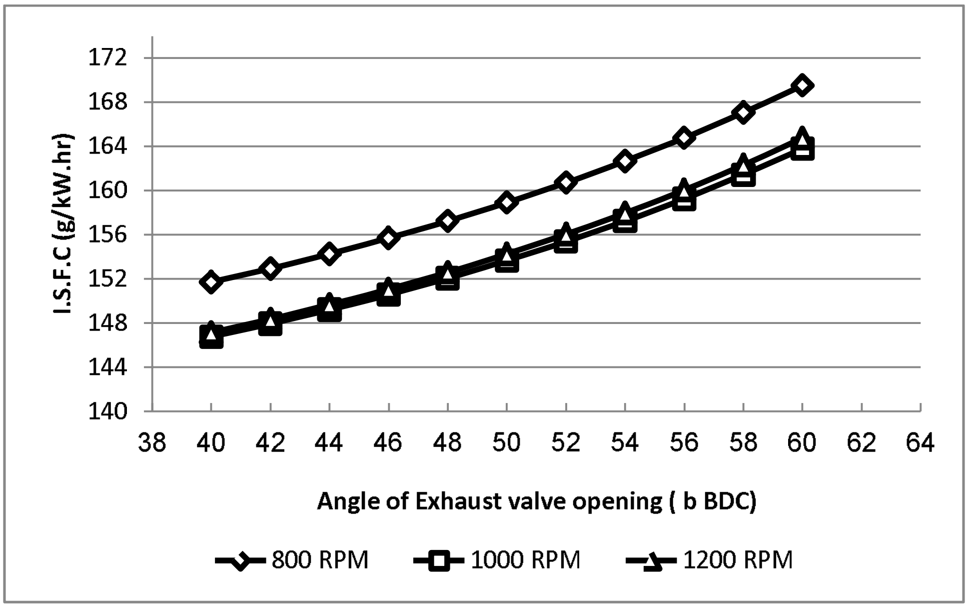

Fig.(7) the effect of exhaust valve timing on the indicated specific fuel combustion .

Figure (8) represent the variation of heat losses with respect to exhaust valve timing for speeds of (800, 1000, 1200)rpm at A/F of (45) **.**It observed that the minimum heat losses is at (60 degree) bBDC and speed of (1200)rpm, which means that the earlier opening of the exhaust valve reduces the heat losses due to reducing the heat transfer time as well as the speed of engine .

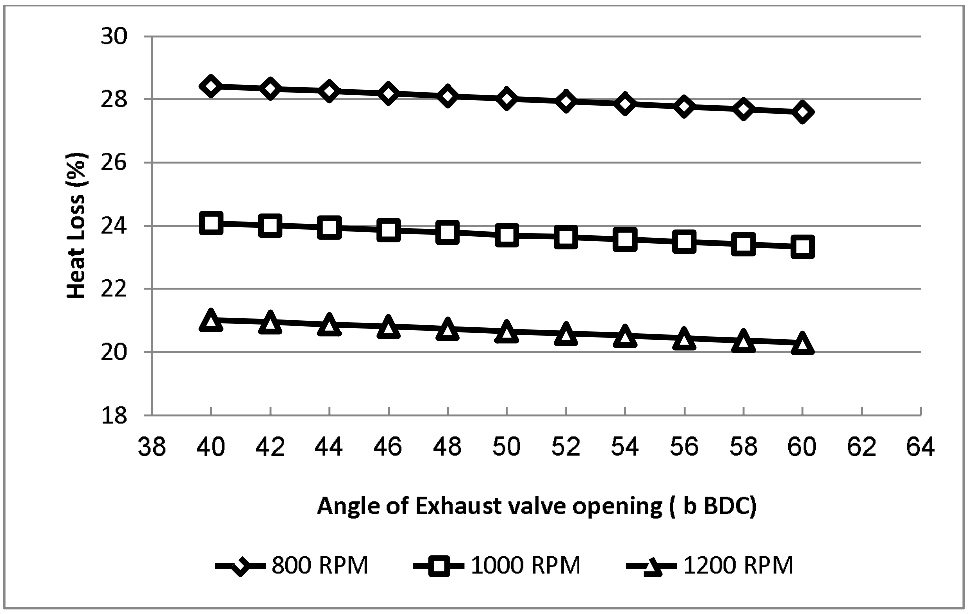

Fig.(8) the effect of exhaust valve timing on the heat loss .

Conclusions:

1. Increasing the angle of intake valve closing (a.BDC) improves the diesel engine performance, until the certain angle which was about (60^{O}) in this work , This improvement maybe due to improving the volumetric efficiency [22] and high charging of the cylinder .
2. Decreasing the angle of exhaust valve opening (b.BDC) improves the diesel engine performance until the (BDC),but the exhaust valve must be opened at a suitable angle which was (40^{o}) in this research to allow the exhaust gases to escape in order to prevent the resistance of the cylinder gases during the piston raising and reducing the power requirements for sweeping .

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