Friction Power Modeling As A Function Of Diesel Engine Speed And Geometry With Predicting Their Effects On Some Performance Parameters

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Abstract :

The first part of the present research is to estimating the relationship between the overall friction power and rotating speed of a 4-stroke single cylinder diesel engine practically by using an electrical motor which was directly connected to the engine with a suitable coupling, and the input power was controlled by using an inverter,(current and volts) were measured by using a digital clamp-meter for different rotating speeds of (600,1200,1800, 2400, 3000 and 3600) rpm. From the data achieved an equation (polynomial third order) was estimated which represents the friction losses as a function of engine speed. The estimated equation is used in simulation program (QBASIC), for predicting the effects of friction on some parameters (power ,efficiency and fuel consumption) . The second goal is to developing the estimated equation and using it many cylinders and many displacement volume diesel engines by multiplying it by displacement ratio and number of cylinders. Finally the results showed good agreement.

Key words :// fraction power ,diesel engine , engine performance ,simulation ,modeling .

نمذجة القدرة الاحتكاكية كدالة للسرعة الدورانية والابعاد الهندسية لمحركات الديزل واستنتاج تاثيراتها على بعض معاملات الاداء.

رفيق احمد خليفة **شيرين الياس كسكه**
مدرس **مدرس مساعد**
الكلية التقنية/ كركوك–الوقود والطاقة

الخلاصة:

الجزء الاول من هذا البحث يتعلق بايجاد العلاقة الرياضية بين السرعة الدورانية والقدرة اللازمة للتغلب على الاحتكاك لمحرك ديزل رباعي الاشواط ذو اسطوانة واحدة عمليا وذلك عن طريق ربط محرك الديزل بمحرك كهربائي مناسب (ثلاثي الطور) باستخدام قارنة ، والتحكم بالسرعة الدورانية للمحرك عن طريق منظم (inverter) وقياس القدرة الداخلة (التيار والفولتية) باستخدام (مقياس الفولتية والتيار) للسرعات (٢٤٠٠،١٨٠٠،٦٠٠،١٢٠٠، ٣٠٠٠ و ٣٦٠٠) دورة/دقيقة ، ومن خلال البيانات العملية تم استنتاج المعادلة الرياضية (متعددة الحدود) واستخدمت في برنامج محاكاة على الحاسبة الالكترونية بلغة (QBASIC) لحساب (القدرة ،الكفاءة،معدل الضغط الفعال، واستهلاك الوقود النوعي الفرملي والكفاءة الميكانيكية). والهدف الثاني هو استخدام المعادلة الرياضية بعد تطويرها لتحاكي جميع انواع محركات الديزل بعد ادخال نسبة الحجم المكتسح وعدد الاسطوانات. واخيرا اظهرت النتائج توافقا ومقبولية جيدة.

الكلمات الدالة:/ القدرة الاحتكاكية ، محركات الديزل، اداء المحركات،المحاكاة، نمذجة.

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)
dM_f	fuel mass burning during the step	(kg)
$dQ_{Conv.}$	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)
$\Delta\alpha$	Step value	(degrees of crank angle)

Introduction:

The simple deffinition of the friction power is the difference between the indicated power and brake power and has a major effects on the engine performance and it is important to know how to measure and reduce it for improving the engine performance.

The mechanical friction of the movement parts of an engine (piston ring assembly and bearings) is the major cause of power losses which about (65 – 70)% of the total losses [1]. It is well known that the engine efficiency increases by reducing the mechanical friction therefore a great efforts and many researches carried out by engineers to study the effects of friction losses by using a mathematical models or correlations which are a simple methods and effective tools for simulation method of engine performance and comparing the calculations with the results obtained by the practical method. It is a fact that a large percentage of the mechanical friction loss in engines occur on the lubricated surfaces between the skirt and the cylinder liner as well as between the cylinder rings and cylinder liner [2] .By consider energy consumption within the engine, it is found that friction loss contributes the major portion of the energy consumption developed in an engine. About two-thirds of it is caused by piston skirt friction, piston rings, and bearings, and the other third is due to the valve train, crankshaft, and gears [3]. A mathematical model was estimated and used in the simulation program and the method is showed reasonable results [4].The empirical corelations are used [5,6] to predict the friction power losses and friction losses are assumed to be of three components (mean effective pressure lost to overcome friction due to gas pressure behind the rings, mean effective pressure absorbed in friction due to wall tension of rings, and mean effective pressure absorbed in friction due to piston and rings). Another empirical formulas for friction mean effective pressure for all constituents for SI engines having a displacement volume between (845 and 2000)Cm^3 as a function of engine speed was developed and used by[7,8].

Due to the importance of this consept this research was performed and aimed to: **1-** estimating a relationship between the engine rotating speed and power consumed to overcome the friction as a mathematical model and using it in a computer simulation method. **2-** developing the estimated mathematical relationship to satisfy for many cylinders and many displacement volumes by using a displacement ratio and number of cylinders .

Experimental Steps and Instrumentation :

The first step of the practical part in this research was preparing the small 4-stroke single cylinder direct injection naturally aspirated diesel engine by removing the nozzle and controlling the valves by setting them to open during experiment time to prevent the cylinder gas content resistance . The technical specifications of the engine used are given in table (1).

Table (1) Technical specification of the C.I. engine [9]

Cylinder number	**1**
Cylinder diameter (D)	**86mm**
Stroke (S)	**72mm**
Compression ratio	**14**
Connecting rod length (L)	**125mm**
Intake valve closing	**30^o aBDC**
Exhaust valve opening	**50^o bBDC**

The second step was preparing a suitable three phases electrical motor and connecting it by a mechanical coupling to the diesel engine ,the third step was preparing an (inverter) which is a device that controls the motor rotating speed by adjusting the electrical power input (current and volts were measured by using a digital clamp meter) which is equal to the power that required to overcome the friction with taking to accounts the power factor of the motor and inverter .After preparing the system as in figure (1).

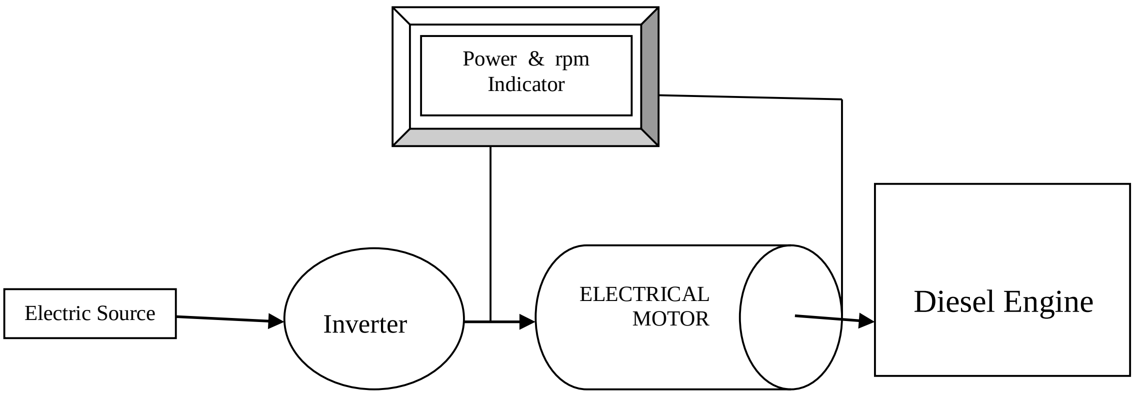

Fig.(1) Schematic Diagram of the Testing System

The testing procedures were performed for speeds of (600,1200,1800, 2400, 3000 and 3600) rpm, and the input power for each speed was measured and recorded, a relationship between the input power and rotating engine speed was achieved by curve fitting with aid of computer and a mathematical model was obtained (third order polynomial equation) as:

$FP=300(N/1000)^3-1000(N/1000)^2 + 1768(N/1000) - 400$ (1) Where

FP= the overall friction power ,and N=engine speed .

Equation (1) is the first goal of this work constructed from the practical tests and it represents the friction power as a function of engine speed for the engine which was used in this work .This equation can be used with different diesel engines after developing it by taking into account the engine geometry and cylinder numbers as:

$$FP=\frac{V_{dn}}{V_{do}}[300(N/1000)^3-1000(N/1000)^2+ 1768(N/1000)-400]*Z \quad (2)$$

Where $\frac{V_{dn}}{V_{do}}$ is the displacement volume ratio, V_{dn} is the displacement volume of any diesel engine which is wanted to test it ,V_{do}=0.000418 m^3 which is the displacement volume of the present diesel engine, and Z is the number of cylinders . Equation (2) represents the second goal of this work which is suitable for using it in simulation program .

Theoretical Analysts :

The estimated mathematical model of the friction power was used in a simulation program (QBASIC) language using the technical specification of engine and fuelwith applying the fully thermodynamic cycle starting from the moment of intake valve closing to the moment of exhaust valve opening with dividing the cycle to intravels each intravel (Δα=2 degree of crank angle) in which all parameters such as instantaneous cylinder volume , area and properties of the cylinder content were calculated as:

1-The overall combustion equation considered for the fuel with C-H-O-N is [10]:

$$CxHyOz +\lambda*ycc*(O_2+3.773N_2)\rightarrow xCO_2 +(y/2)\, H_2O +(\lambda-1)\, ycc\, O_2+\lambda*ycc*(3.773N_2) \quad (3)$$

$A/F_{\,Stoch.}$= ycc = x + (y/2) – (z/2) and λ = Excess air factor

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

$$tmr = 1+\lambda * ycc *4.773 \quad (4)$$

$$tmp = x + (y / 2)+ 3.773* \lambda * ycc +(\lambda -1)* ycc \quad (5)$$

where tmr = total mole number of reactants

tmp = total mole number of products

2-Volume at any crank angle is calculated from the following equation [10,11,12]:

$$V(\theta)=Vc+(Vs/2)*\{1-\cos(\theta)+(2L/S)-[(2L/S)^2-\sin^2\theta]^{0.5}\} \quad (6)$$

3- Initial temperature and pressure at the end of each step (step= 2deg.)were calculated as [10,11]:

$$T_2 = T_1 *[\frac{V_1}{V_2}]^{k-1} = T_1 *[\frac{V_1}{V_2}]^{\frac{R_{mol}}{C_V(T_1)}} \quad (7)$$

$$P_2 = [\frac{V_1}{V_2}]*[\frac{T_2}{T_1}]*P_1 \quad (8)$$

4- 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 [10,11]:

$$e_{i(T)} = R_{mol}((\sum_{j=1}^{5} U_{i,j} *T^j)-T) \quad (9$$

$$E_{(T)} = R_{mol}((\sum_{i=1}^{n} W_i((\sum_{j=1}^{5} U_{i,j} *T^j)-T) \quad (10)$$

Where $E_{(T)}$ = total internal energy , $U_{i,j}$ = polynomial coefficients for mixture species in the cylinder [4,10]. 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

$$W_t = \sum_{i=1}^{n} W_i \qquad (11)$$

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

5-Specific heat at constant volume and constant pressure for each species is calculated using the expression given below [11]:

$$C_{V(T)} = \frac{1}{W_t}\frac{\partial(E_{(T)})}{\partial T} \qquad (12)$$

The derivative of equation (11) is

$$W_t * C_{V(T)} = R_{mol}\sum_{j=1}^{n} W_i((\sum_{j=1}^{5} j*U_{i,j}*T^{j-i})-1) \qquad (13)$$

Where equations (9&12) represent the general expression of the internal energy & specific heat.

6- The work done during the step is calculated as [11,12]:

$dW = P\,dV = 0.5\,(P_1+P_2)\,(V_2-V_1)$ (14)

By applying the first law of thermodynamics during the compression stroke before the combustion starting.[11,12]

$dQ_{conv} - dW= dE= E(T_2) - E(T_1)$ or $f(E) = E(T_2) - E(T_1)+ dW- dQ_{conv}$ (15)

where f(E)=energe function and dQ_{con}=heat loss through the cylinder walls calculated using (Eichelberg"s equation) $dQ_{conv.} = h*A_s*(T - T_{wall})$ (16)

Where h= Heat transfer coefficient and calculated as:

$h=2.466*10^{-4}(U_p)^{1/3}(P)^{1/2}(T)^{1/2}$ (17)

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

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

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 (15) was solved numerically using "Newton Raphson's" method [13,14,15,16], and a solution was obtained when f(E) = 0 and during the combustion the energy equation can be written as[10,11]:

$E(T_2) =E(T_1) - dW - dQ_{Conv.} + dM_f\,Q_{HV}$ (18)

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

$ER =E(T_1) -E(T_2) - dW - dQ_{Conv.} + dM_f\,Q_{HV}$ (19)

If the numerical value of (ER=error) 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) [11,12] .The derivative of equation (19) is:

$$ER' = C_V(T_2)*tmp \qquad (20)$$

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:

$$(T_2)_n= (T_2)_{n-1} - (ER / ER') \qquad (21)$$

<u>Fuel injection rate</u>:

mass of the fuel injected for each crank angle interval (Δα)which was (2degrees) is calculated using the following expression[10,11]:

$$TFIR = \frac{Ma*\Phi}{\left(\frac{A}{F}\right)stoich.} \qquad (22)$$

Where TFIR is the 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 [12]. $$\Phi = \frac{\frac{A}{F}stoich.}{\frac{A}{F}actual} \qquad (23)$$

$$FIR = (TFIR/\theta_{dur.}) \qquad (24)$$

FIR is the mass of fuel injected per degree of crank angle. and $\theta_{dur.}$ is the duration of fuel injection

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) [13, 14,15]. The empirical formula used in the present model is shown in equation which was used by [10,11]. 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.

$$ID = (40/CN)^{0.69}*EXP(4644/T_2)*0.000343*N*(P_2)^{-0.388} \qquad (25)$$

The following equations are used for calculating the indicated power, indicated thermal efficiency and indicated specific fuel consumption[10,12,13]:

Indicated Power = $IP = (Imep*A*L*N*Z)/(60*n)$ (26) Where N=Engine Speed (rpm) , A=Cross Section Area of the cylinder (m^2), Z= Number of Cylinders and (n=1 for 2-stroke engines and n=2 for 4-stroke engines).[12]

Indicated Thermal Efficiency = $\eta_{ith} = IP/(\dot{m}_f*Q_{HV})$ (27)

Indicated Specific Fuel Consumption = $ISFC= \dot{m}_f / IP$ (28)

Brake Power = $BP=IP-FP$ (29)

Brake Thermal Efficiency = $\eta_{bth} = BP/(\dot{m}_f*Q_{HV})$ (30)

Mechanical Efficiency = $\eta_m =BP/IP$ (31)

Results and discussion :

The obtained results in this research are represented by the following figures :

Figure (2). Represents the three types of the power (Indicated ,Brake and Friction) versus the engine rotating speed .It is observed that the increasing of engine speed increases the friction losses and leades to decreasing the brake power, and the results which were obtained by uing equation (1) in the simulation program show good agreement with literatures .which means that the method is very easy more suitable

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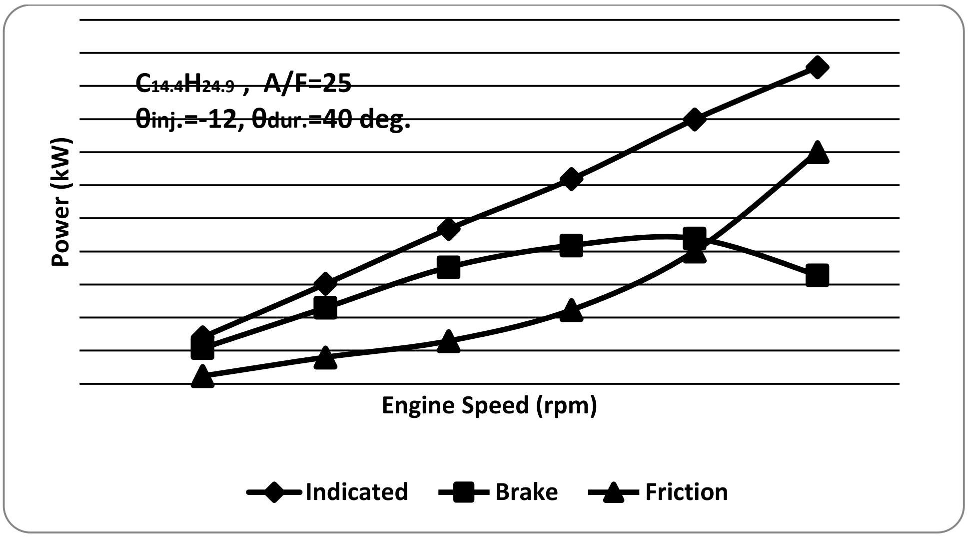

Fig. (2) The Effect of Engine Speed on the Three Types of Power.

Figure (3). Represents the relationship between two types of thermal efficiency (indicated and brake) versus the engine speed .It is observed that the brake thermal efficiency is less than indicated thermal efficiency due to friction losses .The method that was used showed good agreement again .

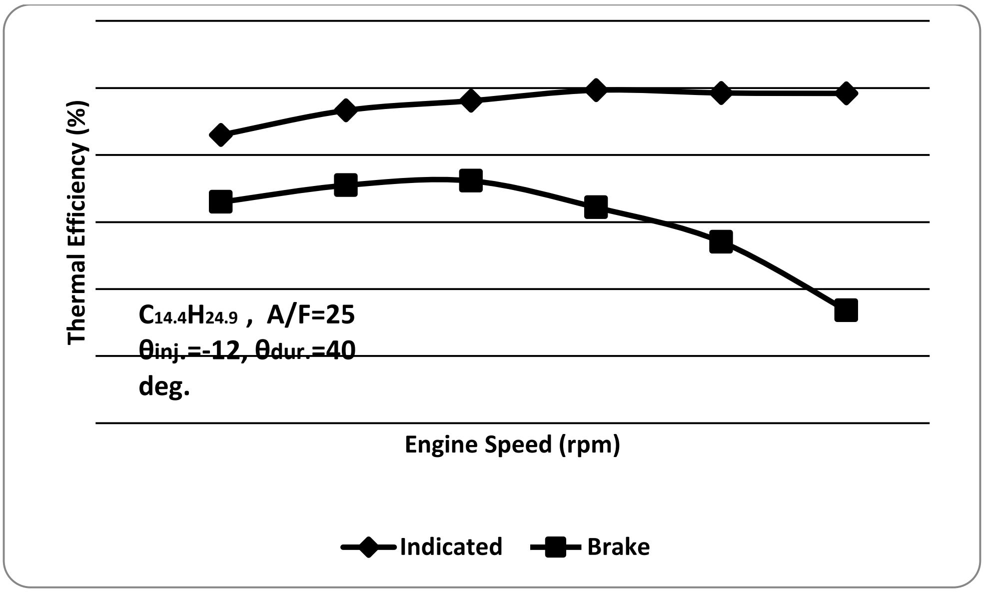

Fig. (3) The Effect of Engine Speed on the Thermal Efficiency.

Figure (4). Represents the relationship between the engine speed and specific fuel consumption (brake and indicated) .It is observed that the brake specific fuel consumption increases with increasing the engine speed, due to increasing the friction losses that leads to reducing the brake power.

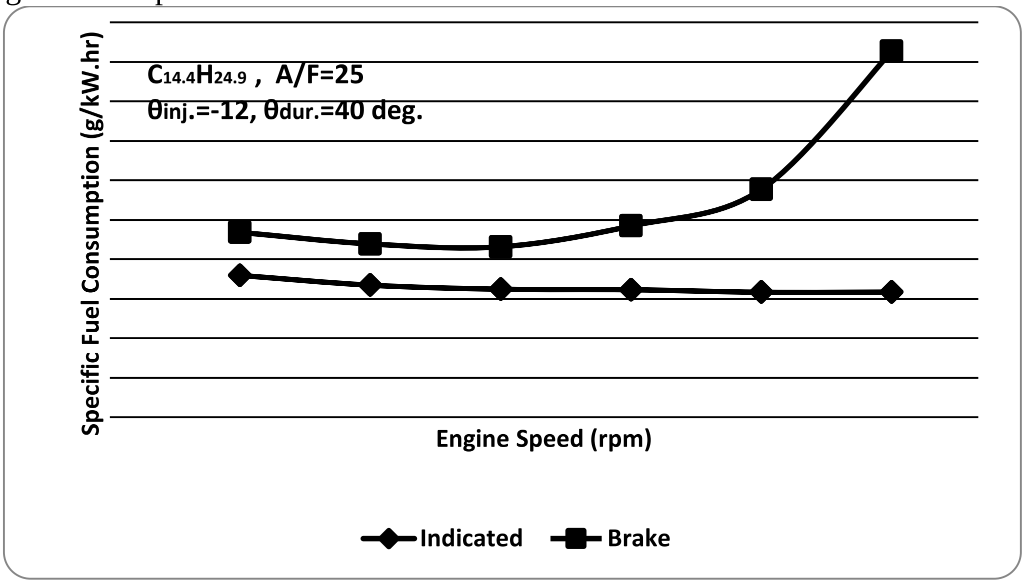

Fig. (4) The Effect of Engine Speed on the Specific Fuel Consumption .

Figure (5).represents the relationship between the engine speed and mechanical efficiency .It is observed that the mechanical efficiency decreases with increasing the engine speed ,this decreasing is due to the increase of brake power which leads to decreasing the mechanical efficiency .

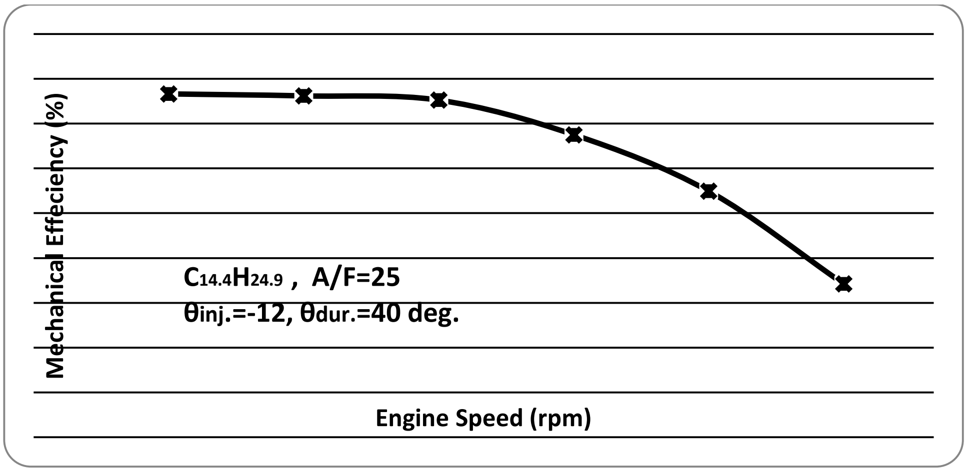

Fig. (5) The Effect of Engine Speed on the Mechanical Efficiency.

For the second goal of this research which was developing the mathematical model of overall friction power calculating, another diesel engine table (2) was tested using equation (2) , and the obtained results show good agreement.

Table (2). Specification of the engine . [17]

Bore	**120mm**
Stoke length	**160mm**
Connecting rod length	**250mm**
Compression ratio	**17:1**
Displacement volume	**1.81 liter**
Inlet valve closing (IVC)	**50°aBDC**
Exhaust valve Open (EVO)	**46° bBDC**
Fuel injection timing	**28° bTDC**

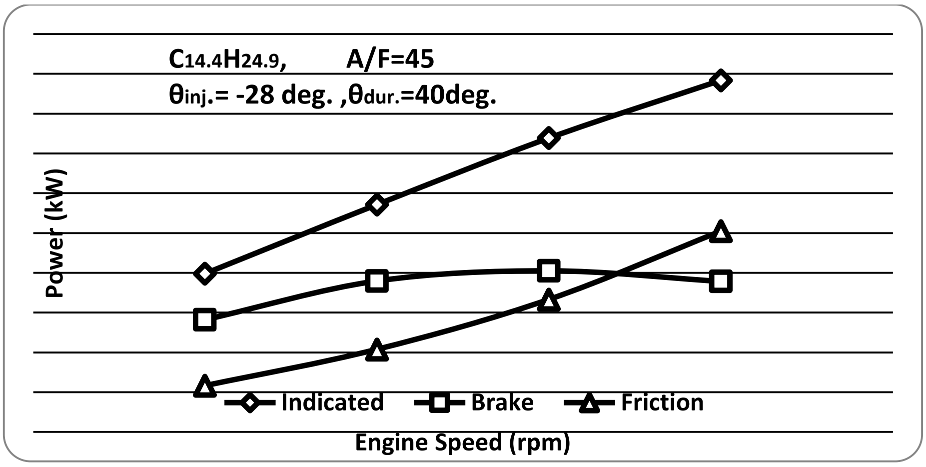

Fig. (6) The Effect of Engine Speed on the Three Types of Power.

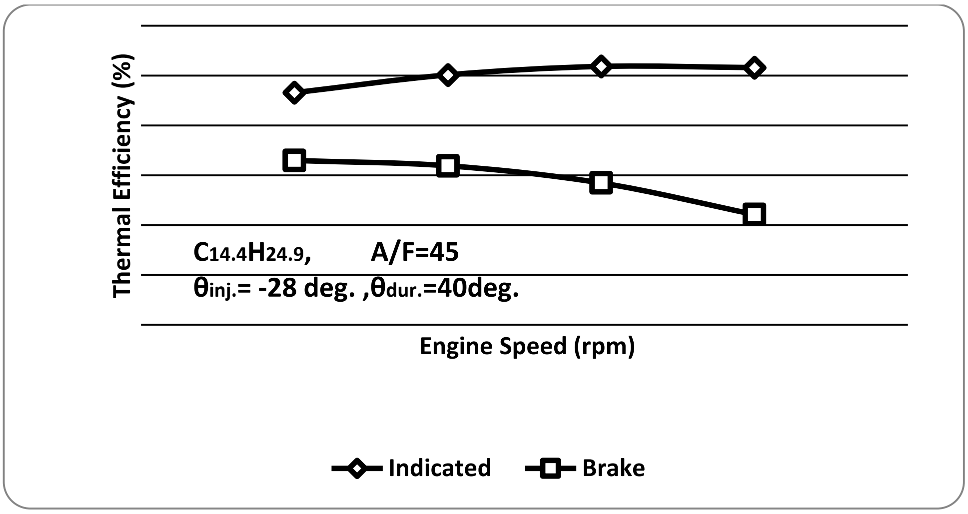

Fig. (7) The Effect of Engine Speed on the Thermal Efficiency .

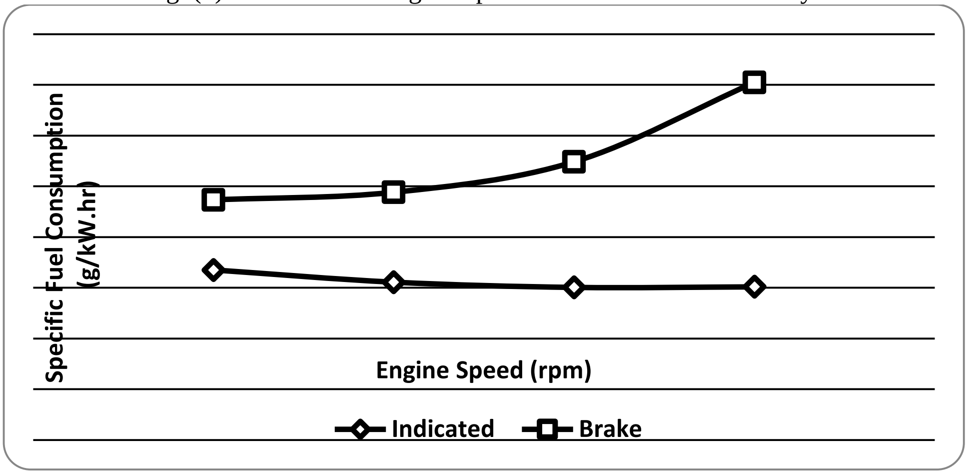

Fig. (8) The Effect of Engine Speed on the Specific Fuel Consumption .

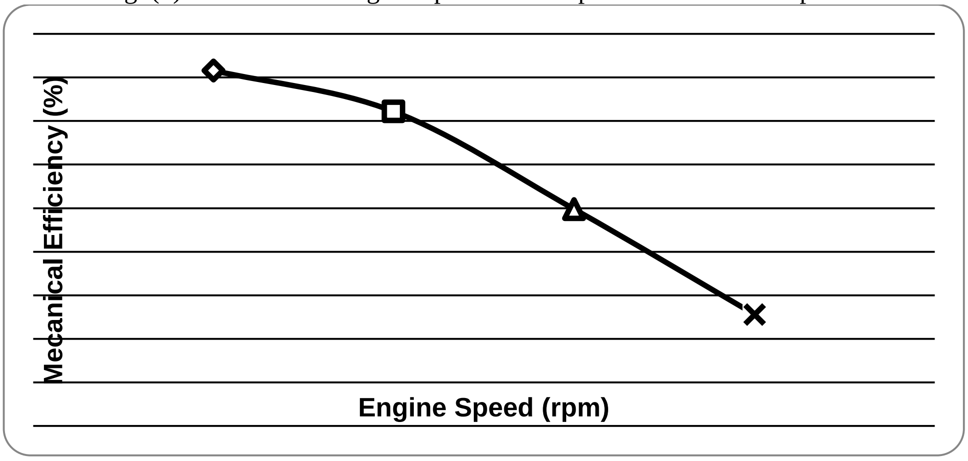

Fig. (9) The Effect of Engine Speed on the Mechanical Efficiency .

Conclusions:

1.The estimated mathematical equation can be used for calculating the friction power for many cylinders and many displacement volumes by simulation method which makes the testing process fast, easy and less pollutants with good approximation .

2.The engine design parameters can be predicted and studied for best results by running the simulation program in a short time with economy in costs .

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