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Study a Model Close to the Actual Cycle Of Internal Combustion Engines

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Abstract— Using simulating models for internal combustion engine cycles is appreciable method for predicting the engines performance for saving the time and the effort. Fuel air ratio and gas variable specific heats are taken into account in the present work. Irreversibilities resulted from nonisentropic compression and expansion processes and heat loss through the cylinder wall are also taken into account in the present model. Finite difference method is applied for estimating the states through the heat addition process and compression and expansion strokes. Computer program is designed for the model includes all the above conditions and the cycle parameters. Experimental test was carried out on a single cylinder constant speed diesel engine to verify the obtained results using the present model. The obtained results show a good agreement with the corresponding data recorded from the experimental tests. Other Comparisons are done with the corresponding results of an actual engine model results which published for gasoline and diesel engines. The obtained results from the model show a good agreement with the corresponding data in researches. The effect of the cycle parameters (inlet air temperature, inlet air pressure, air fuel ratio, compression ratio, and compression and expansion efficiencies) on the power output and thermal efficiency are studied. It is shown that the power and thermal efficiency increase with the increase of compression and expansion efficiencies, inlet air pressure and compression ratio. For gasoline engine cycle the optimum value of compression ratio is around10 to be prevented from detonation, and for diesel the optimum value is around 20. With increasing air fuel ratio the power output increase then decrease and the thermal efficiency increases, so the optimum value of air fuel ratio for gasoline engine cycle is around 13 and for diesel around 15. With increasing the inlet air temperature the power output and thermal efficiency are decreased. The Specific Fuel Consumption decreases with increasing power for the two cycles. The benefit from the research is that optimum parameters for operating are predicted by the model. The obtained results would be more realistic and implemented on the performance evaluation of the internal combustion engine.

Keywords: Otto, Diesel, Dual, Irreversible, Combustion, Performance, heat transfer

1 Introduction

Most studies of the internal combustion engines use air-standard power cycle model to perform their thermodynamics analyses. Such models are used in order to show the effect of varying engine parameters and conditions on the performance, this type of analyses provide exceedingly generous predictions of the performance for the heat engines. In order to provide a more reasonable estimation of the performance potential of a real cycle, one should take the main irreversibilities and the heat losses into account to be more close to practice. Heat transfer losses through the cylinder wall were considered by[1]. Also the effect of heat transfer losses and compression and expansion efficiencies on the performance were studied [2 -4]. Zhao et al. [5] studied the influence of the multi-irreversibilities mainly resulting from the adiabatic processes, finite time processes and heat loss through the cylinder wall on the performance of the cycle. In the studies of air-standard power cycles [1-4]the working fluid was only assumed to be air. In [1-3] the air was assumed as the working fluid as an ideal gas with constant specific heats. Many researchers studied the performance of air standard power cycles by assuming variable specific heats as [4, 6, 7]. The influence of the compression and expansion efficiencies, variable heat capacities, heat loss and other parameters on the performance of the cycle are studied in[8]. An irreversible Dual cycle model which is more closed to practice is established in [9]. In the irreversible Otto cycle model, the non-linear relation between the specific heat of the working fluid and its temperature, the friction loss computed according to the mean velocity of the piston, the internal irreversibility described by using the compression and expansion efficiencies and the heat transfer loss are described in[10]. Taking the real mixture of air and fuel will be more realistic.

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NOMENCLATURE

Alphabetic symbols

a	Moles of gas at the end of process
b	Moles of gas at the beginning of process
C_v	Specific heat at constant volume(J/kmol.K)
E(T)	Internal energy(J/kmol)
Δh	Enthalpy(J/kmol)
M	Number of moles of mixture
n_l	Number of moles of specimen l
P	Pressure(Pa)
P_m	Mean effective pressure (Pa)
P'	Dimensionless power output
Q_{gained}	Actual heat released (J)
$\Delta Q_{combustion}$	Total heat released by combustion(J)
Q_{vs}	Heat of reaction(J/kmol)
R	General gas constant(J/kmol.K)
r	Crank radius(m)
T	Temperature(K)
T_o	Average cylinder wall temperature(K)
V_s	Stroke volume(m^3)
W	Work done (J/kmol)
X	Carbon atoms in fuel
XB	Mass fraction burned
Y	Hydrogen atoms in fuel
z	Number of moles of fuel

Greek symbols

β	Constant related to heat transfer (J/ K)
η_{comp}	Compression efficiency(%)
η_{exp}	Expansion efficiency(%)
η_{th}	Thermal efficiency(%)
θ	Crank angle

Abbreviations

(A/F)	Actual air fuel ratio
$(A/F)_{st}$	Stoichiometric air fuel ratio
B.D.C	Bottom dead center
DSFC	Dimensionless specific fuel consumption
RPM	Revolution per minute
T.D.C	Top dead center

In the present study the effects of irreversibilities resulting from compression and expansion efficiencies and heat transfer loss through cylinder walls are taken into account. Also mixture of air and fuel with variable specific heats are considered for the present study. Finite difference method is applied for estimating the performance parameters. Computer program is designed for studying the effect of various parameters on the performance characteristics of the internal combustion engines under different operating conditions. The results are verified with an experimental work and published results.

2 ANALYSIS

The actual internal combustion engine models are used to perform thermodynamics analysis for the internal combustion engines as shown in Fig. 1.The internal combustion engine cycle can be approximated into three processes. The first is a compression process (1-2), when the piston moves from the bottom dead center, B.D.C, to the starting of combustion process, implements a compression stroke. The second process, at which combustion is occurred, is equivalent the heat additions in heat engines after the starting point of combustion (point 2) passing through the top dead center ,B.D.C, reaches to the end point of combustion point(3). Then expansion (3-4) will be occurred when the piston moves to the B.D.C, producing the expansion stroke process. The last process is the exhaust stroke, which is equivalent to the heat rejection in the heat engine.

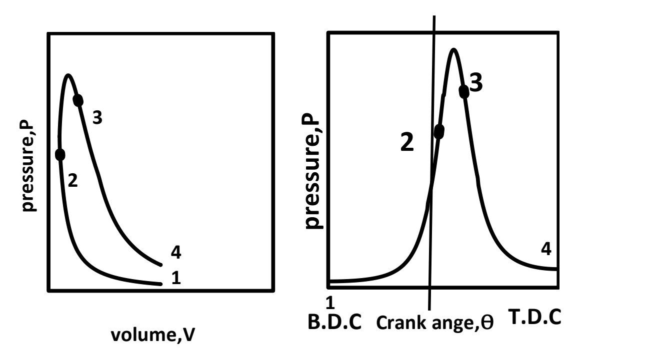

Fig: 1. P-V and P-θ schematically diagram for the internal combustion engine cycle.

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2.1 Compression Process

The piston is moved from the bottom dead center, B.D.C, to the top dead center, T.D.C, making compression stroke. To calculate the changes in the pressure P and temperature T states through the compression stroke, the stroke volume is subdivided into a number of intervals with every crank angle as in Fig. 2. A smaller increment in the interval volume gives a more accurate calculation. The subscripts i and i+1 are used to define the states at the beginning and the end of the volume element.

The cylinder volume can calculated at every crank angle from the relation [11]

$$V=V_C + AREA*(r\cos\Theta + l^2 - r^2 \sin^2 \Theta)^{1/2}$$

Where V, VC, l, r, AREA and Θ are the cylinder volume at any crank angle, clearance volume, connecting rod length, crank radius, cylinder area and crank angle.

At the beginning and end of each interval, the first law of thermodynamics is applied.

$$Q_{combustion} - {}_iW_{i+1} = E(T_{i+1}) - E(T_i) + {}_iQ_{i+1} \qquad (1)$$

Where ${}_iQ_{i+1}$, ${}_iW_{i+1}$,E(T) and $Q_{combustion}$ are heat transfer loss, work done, internal energy, and heat of reaction, respectively.

Considering a pure air is compressed from B.D.C to T.D.C with no fuel burn. Equation (1) becomes:

$$-{}_iW_{i+1} = E(T_{i+1}) - E(T_i) + {}_iQ_{i+1} \qquad (2)$$

Due to small difference in the P through each element, the work term can be approximately estimated from:

$${}_iW_{i+1\ actual} = (P_i + P_{i+1})\frac{(V_{i+1}-V_i)}{2\eta_c} \qquad (3)$$

Where η_c is the compression efficiency.

At the beginning of the compression stroke the cylinder wall has a high temperature compared with the inlet air temperature then the air temperature is gradually increased to a temperature higher than the wall temperature before the end of the compression stroke. So very small amount of heat transfer loss is expected through the compression stroke and thusiQi+1 can be neglected.

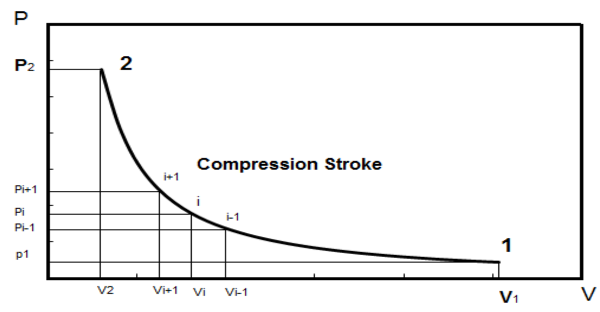

Fig. 2. P-V schematically diagram of the compression stroke.

Then equation (2) becomes:

$$E(T_{i+1}) - E(T_i) + (P_i + P_{i+1})\frac{V_{i+1}-V_i}{2\eta_c} = 0 \qquad (4)$$

In this expression the initial conditions at subscript i, is known but P i+1 and Ti+1areunknown. Then If T i+1 is known, the E(T i+1) can be calculated. Firstly, applying the equation of state on states i and i+1:

$$P_{i+1} = \frac{V_i}{V_{i+1}} * \frac{T_{i+1}}{T_i} * P_i \qquad (5)$$

Equations (4), (5) can’t be solved analytically, so numerical solution must be applied, and the numerical method may be the Newton-Raphson method as shown:

$$(T_{i+1})_n = (T_{i+1})_{n-1} - \frac{f(E)_{n-1}}{f'(E)_{n-1}}$$

Where n-1, n are the previous trail and the current trail of temperature.

Where f(E)from equation (4) :

$$f(E) = E(T\ i+1) - E(Ti) + (Pi + P\ i+1)\frac{V_{i+1}-V_i}{2\eta_{comp}} = 0$$

$$f'(E) = \frac{df(E)}{dT}$$

$$f'(E) = \frac{dE(T)_{i+1}}{dT}$$

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Since$\frac{dE(T)_i}{dT}$ = 0, the work term is not very sensitive toT_{i+1}

As shown below in equations (7), (8) that:

$f'(E) = \frac{dE(T)_{i+1}}{dT} = M * Cv\ (T)_{i+1}$

Where M is the number of moles of mixtures

So

$(T_{i+1})_n = (T_{i+1})_{n-1} - \frac{f(E)_{n-1}}{M * Cv\ (T_{i+1})_{n-1}}$

Ti+1 is estimated for the first trail by assuming an isentropic change from the state conditions at Ti as

$$T_{i+1} = T_i \left(\frac{V_i}{V_{i+1}}\right)^{k-1} = T_i \left(\frac{V_i}{V_{i+1}}\right)^{R/Cvi} \quad (6)$$

The internal energies $E(T_i)$, $E(T_{i+1})$ and the specific heats Cv (Ti), $Cv(T_{i+1})$ are calculated from the gas composition and temperatures as in [12].

$$E(T) = R \sum_{l=1}^{l=s} n_l[(\textstyle\sum_{j=1}^{j=7} U_{l,j} T^j) - T] \quad (7)$$

$$C_v(T) = \frac{dE\ \ (T)}{dT} = \frac{R\sum_{l=1}^{l=s} Jn_l[(\sum_{j=1}^{j=7} U_{l,j}T^{j-1}) - 1]}{\sum_{l=1}^{j=N} n_l} \quad (8)$$

The coefficients $U_{l,J}$ are given by Benson and Whitehose[12],Where nl and N are the number of moles of gas l and number of mixture specimens , respectively.

Then T_{i+1} and $P_{i+1}$are calculated from equations (4) and (5) respectively. Also E(T)and $C_v(T)$are calculated from equations (7) and(8) respectively. These calculations are repeated until the change in bothT_{i+1} and P_{i+1} values are too small.

And the previous steps for all the intervals are applied on the compression process.

2.2 Combustion Process

When the ignition begins the combustion process is subdivided into number of intervals of crank angle according to the combustion duration, the beginning of the interval is, i, and the end is, i+1.

Since the volume is constant, no work is done, dW= 0, the first law for the combustion period becomes:

Since the volume is constant, no work is done, dW= 0, the first law for the combustion period becomes:

$$-P\ (V_{i+1} - V_i) = E\ (T_{i+1}) - E(T_i) - \Delta Q_{combustion} + \Delta Q_{loss} \quad (9)$$

Where $\Delta Q_{combustion}$ and ΔQ_{loss} are heat of combustion in interval and heat transfer loss.

The last equation is solved by the Newton-Raphson method until it satisfied, as shown:

$(T_{i+1})_n = (T_{i+1})_{n-1} - \frac{f(E)_{n-1}}{f'(E)_{n-1}}$

Where n-1, n are the previous trail and the current trail of temperature.

Where f(E)from equation (9) :

$f(E) = E(T_{i+1}) - E(Ti) - \Delta Q_{ombustion} + \Delta Q_{loss} = 0$

$f'(E) = \frac{df(E)}{dT}$

$f'(E) = \frac{dE(T)_{i+1}}{dT}$

Since$\frac{dE(T)_i}{dT}$ = 0,$\frac{\Delta Q_{loss}}{dT}$has very small value so it can be neglected

As shown in equations (7),(8) that:

$f'(E) = \frac{dE(T)_{i+1}}{dT} = M * Cv\ (T)_{i+1}$

Were M is the number of moles of mixtures

So

$(T_{i+1})_n = (T_{i+1})_{n-1} - \frac{f(E)}{M * Cv\ (T_{i+1})_{n-1}}$

Ti+1 is estimated for the first trail as:

$T\ i+1 = T_i + \frac{1 * z * Q_{vs}}{M * CvT_i}$

The heat of combustion for each interval period can be estimated according to the combustion equation:

$\Delta b_5\ C_xH_y + \Delta\ b_3O_2 + \Delta\ b_4\ N_2 \rightarrow \Delta\ a_1\ CO_2 + \Delta a_2\ H_2O +$

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$\Delta\ a_4\ N_2$ (10)

Where Δ, bi and ai are the difference between numbers of moles of this interval and the previous one, number of moles of specimen at the beginning of the interval and number of moles of specimen at the end of the interval.

(13)

At the beginning of the combustion interval the numbers of moles are b_1, b_2 ,b_3 , b_4 and b5for substances CO_2, H_2O, O_2, N_2 and CxHy respectively:

Thus, b_1= XB_{i-1}*z *x, b_2= XB_{i-1}* z *(y/2), b_3= z*(x+y/4) *((A/F)/(A/F)st - XB_{i-1}), b_4= 3.76 b3, b_5= (XB_i- XB_{i-1}) * z

At the end of combustion period, at the temperature T2, the numbers of moles are a1, a2, a3, a4 and a5for substances CO2, H2O, O2, N2 and CxHy respectively:

Thus, a_1= XB_i *z*x, a_2= XB_i* z*(y/2), a_3= z*(x+y/4) *((A/F)/(A/F)st - XB_i), a_4=b4, a_5=0

Where XB is mass fraction burned at every crank angle θ and given by a Weibe function and is used to represent the mass fraction burned versus crank angle:

$$XB=1-exp[-d(\frac{\theta-\theta_o}{\Delta\theta})^{g+1}]$$

Where θ is the crank angle, θo is the start of combustion, ΔΘ is the total combustion duration, XB changes from zero to 1,d and g are adjustable parameter. Varying d and g changes the shape of the curve significantly. Actual mass fraction burned curves have been fitted with d=5 and g =2[11, 13, 14, 15].

The numbers of moles of gases Mi and Mi+1 before and after combustion interval are given by:

$$M_i=\sum_{l=1}^{l=5} b_l \quad , M_{i+1}=\sum_{l=1}^{l=5} a_l$$

To get the heat released by combustion at constant volume:

$$\Delta Q_{combustion}= \Delta\ a_1\ CO_2 + \Delta\ a_2\ \ H_2O- \ \Delta b_5\ C_xH_y - \Delta\ b3O \qquad (11)$$

So the actual heat gained at the process (Qgained) is:

$$\Delta Q_{gained} = \Delta Q_{combustion} - \Delta Q_{loss} \qquad (12)$$

$$\Delta Q_{loss=}\beta\ (T_i+T_{i+1}- 2To)$$

Where β, T_i ,$T_{i+1}$and To are constant related to heat transfer, temperature at the beginning of combustion interval, temperature at the end of combustion interval and average cylinder wall temperature respectively.

And for the general cases β/CV>0 [5, 8], and always β/CV= 0.1[5].

And the final pressure can be given from this relation [11, 16]:

$$\frac{dQ}{dt}=P\,\frac{dV}{dt}\frac{k}{k-1} + V\,\frac{dP}{dt}\frac{1}{k-1} \qquad (13)$$

And k=1.4 – 7.18 ×10-5 × T[11]

Then T_{i+1} and $P_{i+1}$are calculated from equations (9) and (13) respectively. Also E(T)and Cv(T)are calculated from equations 7 and 8 respectively. These calculations are repeated until the change in bothT_{i+1} and P_{i+1} values are too small.

And the previous steps for all the intervals of combustion duration.

2.3 Expansion Process

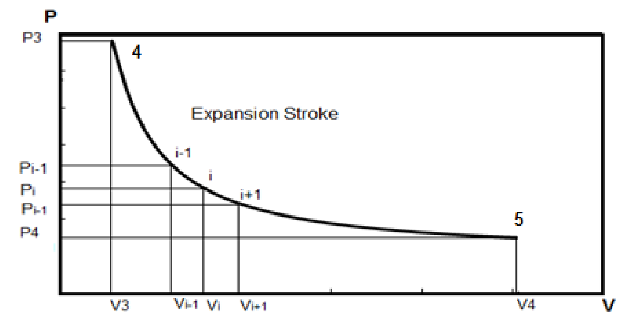

Fig. 3. P-V schematically diagram of the expansion stroke.

During the expansion stroke the composition of the cylinder contents is constant and the calculations are the same as compression stroke but from end of combustion bottom dead center, except for the work.

$$dW=PdV=\frac{P_i+P_{i+1}}{2}\,(V_{i+1} - V_i) * \eta_e \qquad (14)$$

So equation (2) becomes:

$$E(T\ i+1) - E(Ti)+dW = 0 \qquad (15)$$

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In this expression the initial conditions at subscript i, is known but P_{i+1} and T_{i+1} are unknown. Then if T_{i+1} is known, the $E(T_{i+1})$ can be calculated.

The pressure can be given from the relation [11, 16]

$$\frac{dQ}{dt}=P\frac{dV}{dt}\frac{k}{k-1}+V\frac{dP}{dt}\frac{1}{k-1} \quad (16)$$

And k=1.4 – 7.18 ×10-5 × T[11]

And dQ =0 at the expansion stroke.

Equations (14), (15) can't be solved analytically, so numerical solution must be applied, and the numerical method may be the Newton-Raphson method as shown.

$$(T_{i+1})_n=(T_{i+1})_{n-1}-\frac{f(E)_{n-1}}{f'(E)_{n-1}}$$

Where n-1, n are the previous trail and the current trail of temperature.

Where f(E)from equation (15) :

$$f(E)= E(T_{i+1}) - E(T_i)+dW= 0$$

$$f'(E)=\frac{df(E)}{dT}$$

$$f'(E)=\frac{dE(T)_{i+1}}{dT}$$

Since $\frac{dE(T)_i}{dT}=0$, the work term is not very sensitive to T_{i+1}

As shown in equations (7), (8) that:

$$f'(E)=\frac{dE(T)_{i+1}}{dT}=M\times Cv\,(T)_{i+1}$$

Where M is the number of moles of mixtures

$$(T_{i+1})_n=(T_{i+1})_{n-1}-\frac{f(E)_{n-1}}{M\times Cv\,(T_{i+1})_{n-1}}$$

Ti+1 is estimated for the first trail by assuming an isentropic change from the state conditions at Ti as

$$T_{i+1}= T_i\left(\frac{V_i}{V_{i+1}}\right)^{k-1}=T_i\left(\frac{V_i}{V_{i+1}}\right)^{R/Cvi}$$

Then T_{i+1} and P_{i+1} are calculated from equations (15) and (16) respectively. Also E(T) and C_v(T)are calculated from equations 7 and 8 respectively. These calculations are repeated until the change in both T_{i+1} and P_{i+1} values are too small.And the previous steps for all the intervals of the expansion process.

Work and thermal efficiency of the cycle:

The work done in the cycle is obtained by summing the work terms for each step in compression and expansion processes and the mean effective pressure is:

$Pm=\frac{W}{Vs}$ where Vs is the stroke volume

Power = cycle work* (RPM /60)/2000 Kw

To make the power dimensionless:

$P' = Power/ (m_{air}\times Cp\times \Delta T)$

The thermal efficiency is given by:

$$\eta th=\frac{W}{Qvs * z}$$

Dimensionless specific fuel consumption:

$$DSFC=\frac{1}{P'\times(A/F)}$$

Where P' is the dimensionless power.

3 IMPLEMENTATION

For verification of the model results, comparison is done with the corresponding results of an actual engine. So an experimental test was carried out on a single cylinder diesel engine (Crossly) excited in heat lab of mechanical engineering department, Assuit University. The main specifications of the tested engine are given in Table 1. Table 2 shows a comparison between the experimental results and the corresponding results obtained from computer program for the present model.

TABLE 1

MAIN SPECIFICATIONS OF THE SINGLE CYLINDER DIESEL ENGINE (CROSSLY)

Cylinder diameter (m) 0.146	Engine fuel C12H26
stroke (m) 0.279	Stoichiometric Air fuel ratio 15.121
Compression ratio 14.19	
Normal rating speed (rpm) 475	

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TABLE 2:

COMPARISON BETWEEN EXPERIMENTAL RESULTS AND PRESENT MODEL

Air inlet temp. ˚C	(A/F)actual.	Present model			Experimental results		
		IMEP (kPa)	Indicated Power (IP/CpT1)	Thermal eff. (%)	IMEP (kPa)	Indicated Power (IP/CpT1)	thermal eff. (%)
57	16.647	405.04	1.6423	14.03	403.5	1.85	21.7

The comparison shows that the present model is closed to the experimental data and the deviation is small in the indicated mean effective pressure and power. Therefore the proposed model would be accepted. But for thermal efficiency we find that the present model has higher deviation than the indicated mean effective pressure and power. The deviation between these results can be due to many factors such as efficiency of combustion, value of compression and expansion efficiencies, equation used for heat transfer calculations, value of heat transfer coefficient and errors in measuring instruments.
Comparisons also are done with the corresponding results of an actual engine model results which published in international researches for gasoline and diesel engines [4, 17]. The main specifications of the tested engines are given in Table 3. Figures 4, 5 show a comparison between P-V diagrams and Figures 6, 7show a comparison between P-θ of the published researches results and the corresponding results obtained from computer program for the present model for gasoline and diesel engines.

Figures 4, 5, 6, 7 give comparisons between the corresponding results from the present model and the published researches results for gasoline and diesel engines. The comparison shows that the obtained results from the present model are closed to the published data and the deviation is small. Therefore the proposed model would be accepted. But the present model has deviation on some points in the P-V and P-θ diagrams, and the deviation is the smallest in the combustion region.

The small deviation between these results can be due to many factors such as efficiency of combustion, value of compression and expansion efficiencies, equation used for heat transfer calculations, errors in ignition timing, value of heat transfer coefficient and difference in models equations.

TABLE 3:

MAIN SPECIFICATIONS OF THE OTTO AND DIESEL ENGINES

	Diesel engine[17]	Gasoline engine[4]
Fuel	$C_{10.8}H_{18.7}$	C_8H_{18}
Compression ratio	16.4	8.3
Cylinder bore(m)	0.112	0.0864
Stroke(m)	0.115	0.0674
Connecting rod length(m)		0.13
Crank radius(m)	0.0575	0.0337
Engine speed(RPM)	2100	5000
Inlet pressure (bar)	1	1
Inlet temperature (K)	300	300
Equivalence ratio	0.47	1
Ignition timing	22° BTDC	−25° BTDC
Duration of combustion		70°

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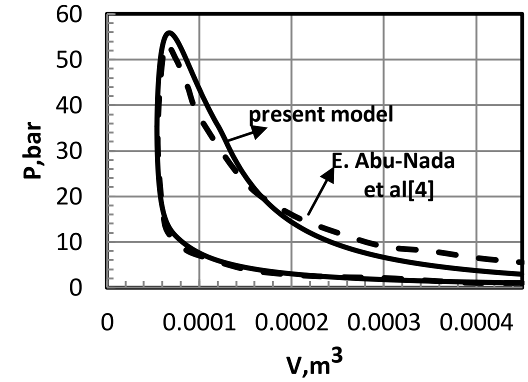

Fig. 4. P-V diagram Comparison for gasoline engine

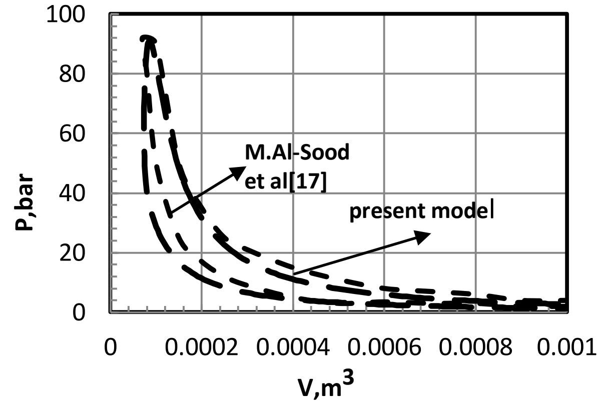

Fig. 5. P-V diagram Comparison for Diesel engine

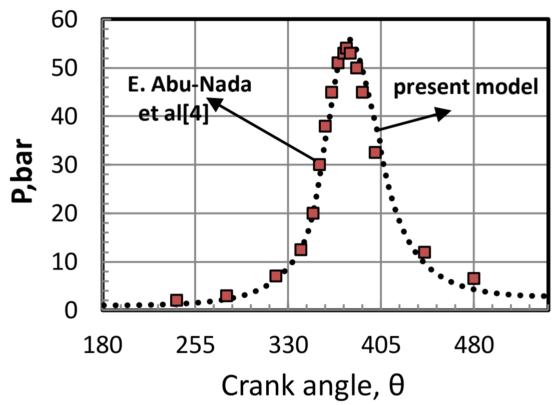

Fig. 6. P-θ diagram Comparison for gasoline engine

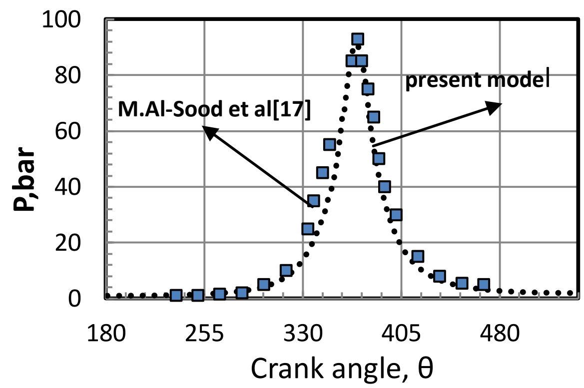

Fig. 7. P-θ diagram Comparison for Diesel engine

4 RESULTS AND DISCUSSION

The proposed model is applied to study the effects of certain parameters on the performance of the internal combustion engines. Using the proposed model, working internal combustion engine cycle types are used under different operating conditions. A study of results of the two engine types is carried out.

The engine performance is presented in a form of relations of the dimensionless power and the thermal efficiency with different parameters. These parameters are inlet air temperature and pressure, heat transfer coefficient, compression and expansion efficiencies, compression ratio and air fuel ratio and specific fuel consumption. Conditions under which these studies done for diesel and gasoline engines are in table 2 except the focused parameter.

In figure 8 (P-V) diagrams for gasoline and diesel engines for one state from the model with various compression ratios and the same bore and stroke engine and the same RPM and $C_{10.8}H_{18.7}$ for diesel and C_8H_{18} for Otto. In this state air fuel ratios for Otto and diesel are 15, 18 respectively.

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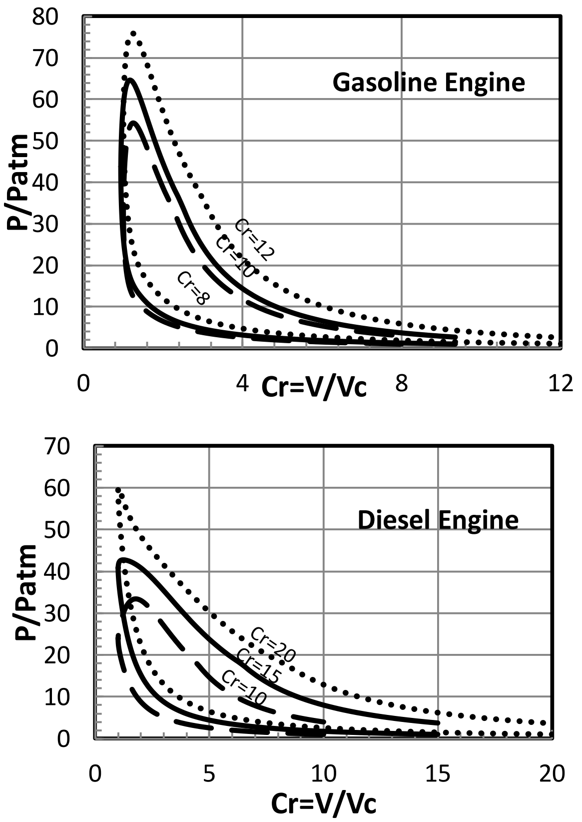

Fig. 8. P-V diagram of Gasoline and Diesel engines respectively for different compression ratios.

TABLE: 4

CONDITIONS UNDER WHICH THIS STUDY DONE FOR DIESEL AND GASOLINE EXCEPT THE FOCUSED PARAMETER

Parameter	gasoline	Diesel
fuel	C_8H_{18}	$C_{10.8}H_{18.7}$
Inlet temperature (K)	300	300
Inlet pressure(bar)	1	1
Heat transfer coefficient	0.1×Cv	0.1×Cv
compression expansion efficiency%	0.97	0.97
Compression ratio	8.3	16.4
Air fuel ratio	15	18
Speed(RPM)	2100	2100

2.1 Effect of Inlet Air Temperature on the Engine Performance

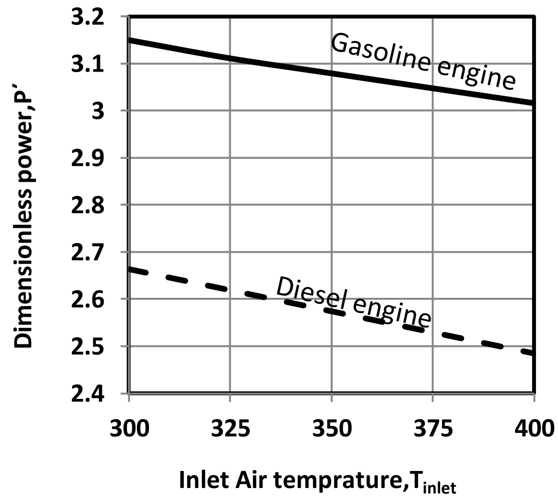

Fig. 9. Effect of inlet air temperature, T_{inlet}, ˚C, on power, P'.

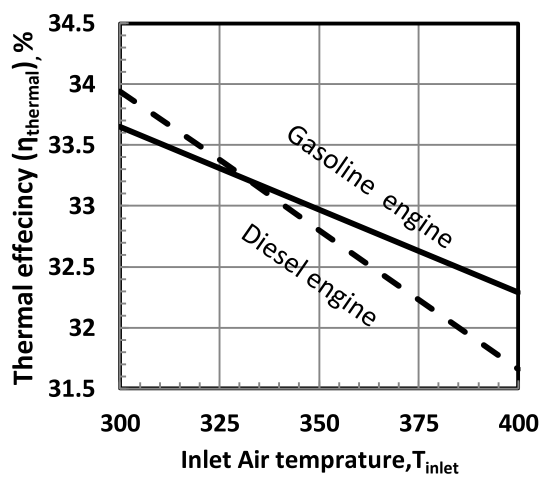

Fig. 10. Effect of inlet air temperature, T_{inlet}, ˚C, on thermal efficiency, η_{th}.

Figures 9, 10 show that when the inlet air temperature increases the power output and thermal efficiency decrease as in Hou, S.-shyurng[1] as expected.

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2.2 Effect of Inlet Air pressure on the Engine Performance

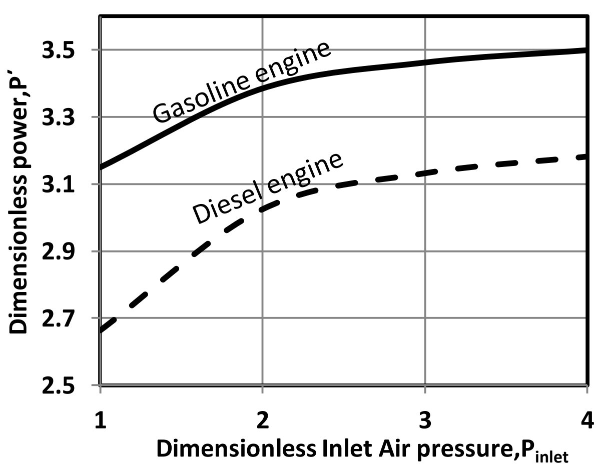

Fig. 11. Effect of inlet air relative pressure, Pinlet, on power, P'.

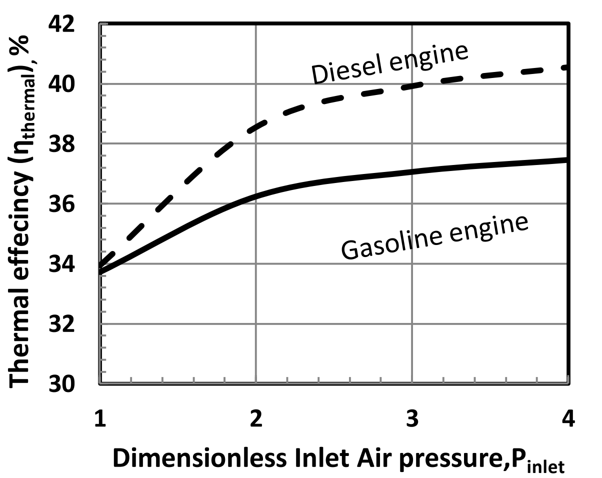

Fig. 12. Effect of inlet air relative pressure, P_{inlet}, on thermal efficiency, η_{th}.

Figures 11, 12 show that the power output and thermal efficiency increase with increasing the inlet pressure for the two types as expected [18]. The increasing in thermal efficiency is optimum up to 2 bar. So turbocharging and supercharging is preferred in some cases. So turbo-charging and supercharging are preferred in some cases.

2.3 Effect of Heat Transfer Coefficient on the Engine Performance

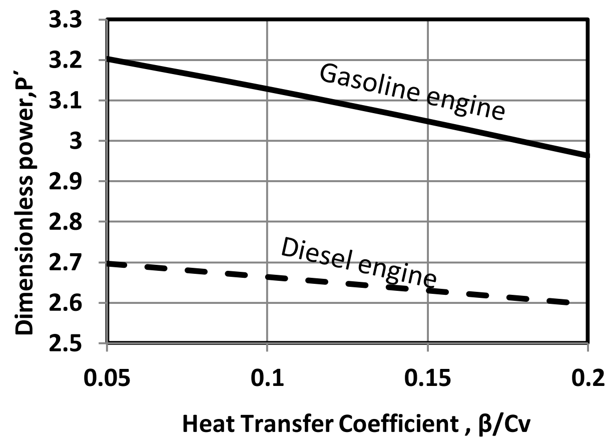

Fig. 13. Effect of heat transfer coefficient, β, on power, P'.

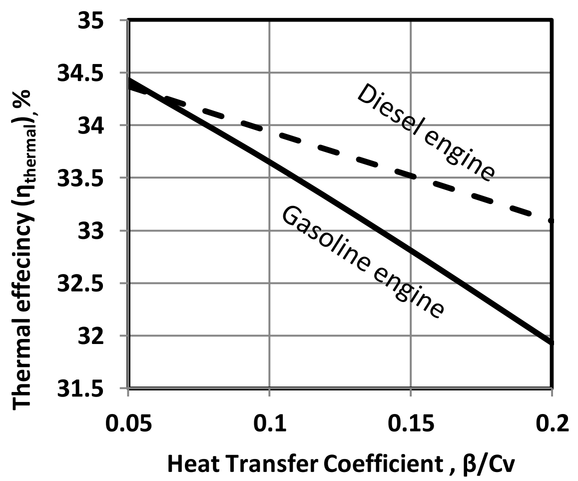

Fig. 14. Effect of heat transfer coefficient, **β,** on thermal efficiency, η_{th}.

Figures 13, 14 show that the heat transfer coefficient affects the performance of the heat engine. The power and thermal efficiency are decreasing with the increase of heat transfer coefficient. The rate of decreasing of power is larger for Otto cycle than Diesel cycle. This is due to that the Otto cycle has larger maximum cycle temperature, which is consequently increasing the amount of heat loss.

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2.4 Effect of Compression and Expansion Efficiencies on the Engine Performance

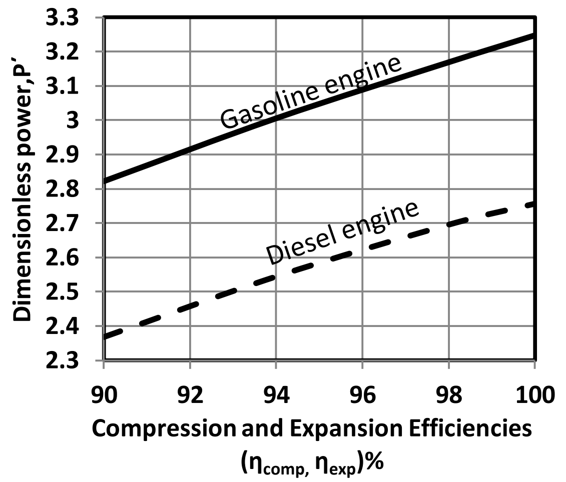

Fig. 15. Effect of Compression and expansion efficiencies, $\eta_{comp,exp}$, on power, P'.

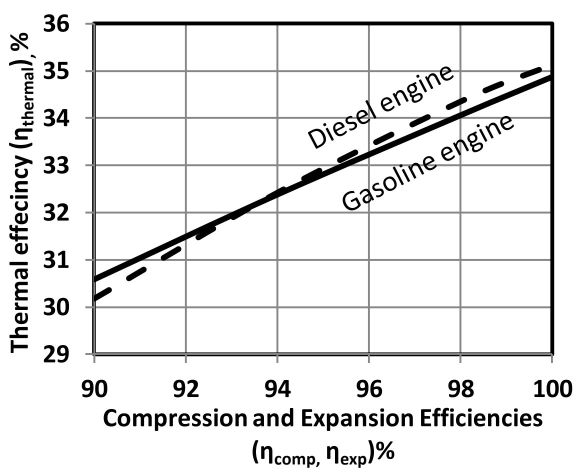

Fig. 16. Effect of Compression and expansion efficiencies, $\eta_{comp,exp}$, on thermal efficiency ,η_{th}.

Figures 15, 16 show that the power and thermal efficiency increase with increasing compression and expansion efficiencies for the two types of cycle as expected

2.5 Effect of Compression Ratio on the Engine Performance

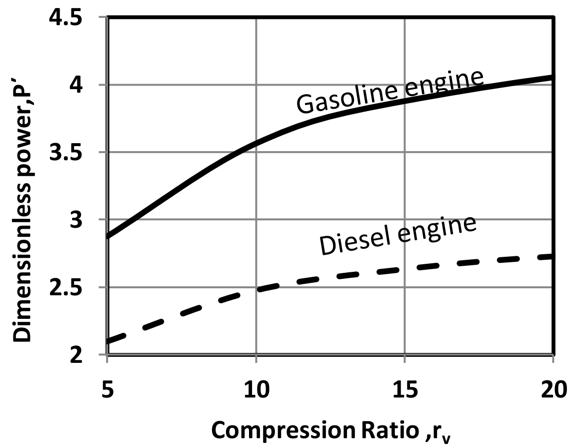

Fig. 17. Effect of Compression ratio, rv, on power, P'.

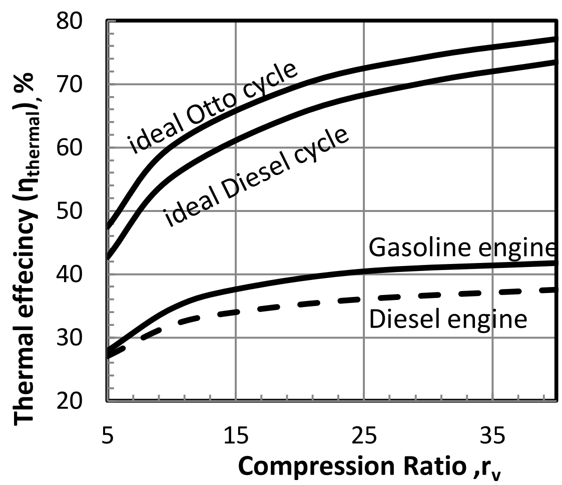

Fig. 18. Effect of Compression ratio, r_v, on thermal efficiency, η_{th}.

Figures 17, 18 show that with increasing the compression ratio the power output and thermal efficiency increase, but when these values of compression ratio reach high values it increase slowly as shown. For Otto cycle after r_v=10 detonation will be done [19]. For diesel cycle, also the power and thermal efficiency increase up to value of compression ratio, r_v=20 as in [19], and the dual cycle is in between. Also, Figure 18 shows a

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comparison between ideal thermal efficiency for Otto and diesel cycles, according to the ideal relation of thermal efficiency for Otto and diesel, $\eta_{otto}=1-rv^{1-\gamma}$, $\eta_{Diesel}=1-r_v^{1-\gamma}*\frac{r_c^{\gamma}-1}{\gamma(rc-1)}$, where γ,rc are specific heat ratio and cut-off ratio, and results from the model. It is shown that these values in the ideal cycle are larger than results from model due to the irreversibilities. And at high values of compression ratios the power output and thermal efficiency decrease as [2, 20].

4.6 Effect of Air fuel ratio on the Engine Performance

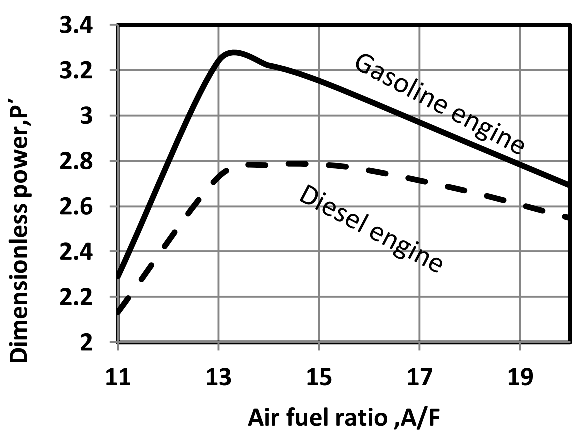

Fig. 19. Effect of Air fuel ratio, A/F, on power P'.

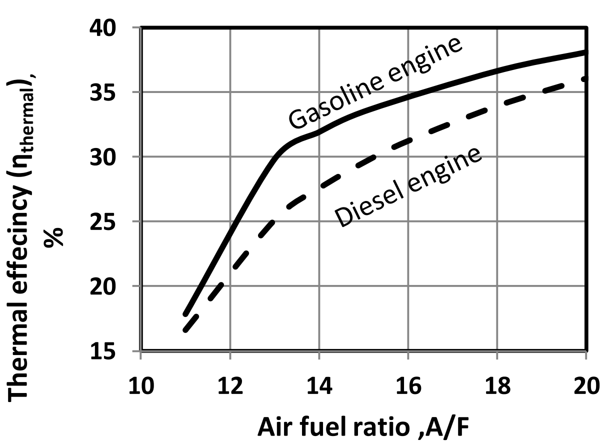

Fig. 20. Effect of Air fuel ratio, A/F, on thermal efficiency, ηth.

Figure 19, 20 show that the three cycles with increasing air fuel ratio the power output increase to a certain value after which they decrease and the thermal efficiency increase continually as the amount of fuel decrease with increasing the air fuel ratio after this certain value, and for diesel this value is around 15. The optimum value of air fuel ratio for Otto is around 13.

4.7 Effect of Specific Fuel Consumption

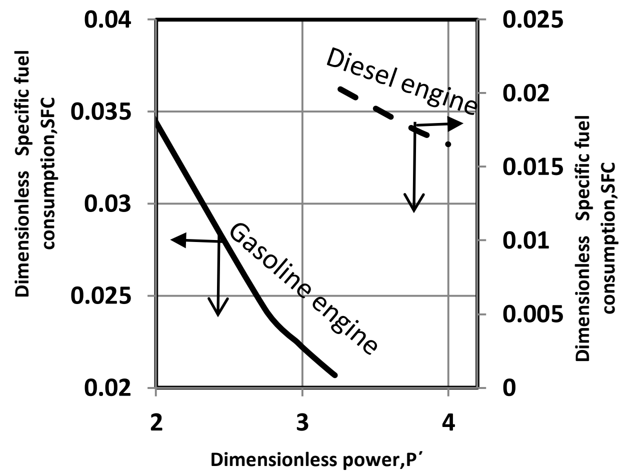

Fig. 21. Effect of Specific Fuel Consumption, SFC, (kg/ hr), on power, P'.

Figure 21 shows that the Specific Fuel Consumption decreases with increasing power as expected [21, 22].

5 CONCLOSION

An irreversible gas cycle model was done and verification of this model had been done with comparison to experimental results from a single cylinder compression ignition engine. Signed Experimental test was carried out on cylinder constant speed diesel engine to verify the obtained results using the present model. The obtained results show a good agreement with the corresponding data recorded from the experimental tests. Other Comparisons are done with the corresponding results of an actual engine model results which published in international researches for gasoline and diesel engines. Some irreversibilities of internal combustion engine cycles like compression and expansion efficiencies and heat transfer loss had taken into account. Computer program is

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designed for the proposed model. Effects of some parameters as inlet air temperature, inlet air pressure, heat transfer coefficient, compression ratio, air fuel ratio and compression and expansion efficiencies had studied with numerical examples for the two types of cycles (gasoline and Diesel engines). It is found that the power and thermal efficiency increase with the increase of compression and expansion efficiencies and inlet air pressure. With increasing air fuel ratio the power output decrease and the thermal efficiency increase to a certain value then it decreases, so the optimum value of air fuel ratio for gasoline engine cycle is around 13 and for diesel around 15. With increasing the inlet air temperature and the heat transfer coefficient the power output and thermal efficiency are decreased. With increasing the compression ratio the power output and thermal efficiency increase but for gasoline the optimum value is 10 to be prevent from detonation, and for diesel the optimum value is around 20. The Specific Fuel Consumption decreases with increasing power.

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