Effect of Varying Inlet Valve Throat Diameter at Different IVO, IVC, and OVERLAP Angles on SI Engine Performance

Osama H. Ghazal Yousef S. Najjar Kutaeba J. AL-Khishali

***Abstract* —the intension in this work is to contribute towards pursuing the development of on variable valve timing (VVT), for improving the SI engine performance. Further to a previous work, this investigation covers the effect of varying the inlet throat diameter and the IVO, IVC and the Overlap angle between IVO and EVC on engine performance at the design engine speed. Power, Torque, BMEP, BSFC and Volumetric Efficiency were calculated and presented to show the effect of varying valve timing on them for all the inlet throat diameters considered.**

***Index Terms*— internal combustion engines, overlap angles, performance, variable valve timing**

NOMENCLUTURE

A	Coefficient in *Wiebe* equation**,** *Annand* open or closed cycle A coefficient	*IVC*	Inlet valve close, degree
aBDC	After Bottom Dead Center	*IVO*	Inlet valve open, degree
BMEP	Brake mean effective pressure, bar	*k*	Thermal conductivity of gas in the cylinder, W/m K
BSFC	Brake specific fuel consumption, g/kWh	*M*	Coefficient in *Wiebe* equation
bTDC	Before Top Dead Center	m_{frac}	Mole fraction
C	Carbon, Annand closed cycle coefficient	NO_x	Nitrogen oxide
CO	Carbon monoxide	O_2	Oxygen
CO_2	Carbon dioxide	*Overlap*	Overlap, degree
c_p	Specific heat at constant pressure, kJ/kg K	*P*	Brake power, kW
c_v	Specific heat at constant volume, kJ/kg K	*Re*	Reynolds number based upon mean piston speed and the engine bore
D_{cyl}	Cylinder bore B, m	*T*	Brake torque, N.m
dQ /A	Heat transfer per unit area, W/m^2	T_{gas}	Gas Temperature, Kelvin
EVC	Exhaust valve close, degree	T_{wall}	wall emperature, Kelvin
EVO	Exhaust valve open, degree	*VVT*	Variable valve timing
H	Hydrogen	*VL*	Valve lift, mm
H	Heat transfer coefficient, W/m^2 K	η_{vol}	Volumetric Efficiency, %

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I. INTRODUCTION

Both governments and car manufacturers suffer a very numerous constraints due to the current emission legislations and the large concern about the environment. Furthermore the control of green-house gas emissions has begun to add to the numerous constraints that vehicle manufacturers have to satisfy, also to consider the reduction of engine fuel consumption. Gas motion within the cylinder is a major factor in the combustion process in SI engines, fuel-air mixing and heat transfer. [Haywood page 326][1]. Since the valve is the minimum area of the flow, it has the highest velocity during the intake, where the flow produces a shear layer. Besides that unless the inlet valve flow area is of sufficient size the induction could be chocked which reduces the volumetric efficiency [1]. When dealing with engine topics exclusively, improving fuel economy to reduce CO_2 emissions means improving the engine thermal efficiency [2].This target can be met following different routes, each of them could be an effective way with different cost-to-benefit ratio. Often, it could be observed, it is helpful to adopt numerous solutions contemporaneously. As an example, fast combustion, lean burn, variable valve timing and actuation, gasoline direct injection and so long may be reminded. During most of its average life, a road engine is run under low load and low speed conditions. It is known that load reduction in spark-ignition engines is traditionally realized by introducing additional losses during the intake stroke by means of a throttle valve. In these operating points, the engine efficiency decreases from the peak values (already not very high) to values dramatically lower. The optimization of intake and exhaust valve timing can provide significant reductions in pumping losses at part load operation [3–5]. A number of papers have been sighted by [6].

Thermodynamic conditions during the closed cycle (compression, combustion and expansion) can be directly controlled by adjusting the intake valve opening IVO and closing angle (IVC), which defines the total intake mass flow rate and the effective compression ratio of the engine [7].

The objective of this paper is to contribute towards the development pursuing, among others on variable valve timing (VVT), for improving the engine performance. Furthermore, the investigation of the effect of inlet throat diameter at IVO, IVC and the Overlap angle between IVO and EVC on engine performance able to optimize engine torque and volumetric efficiency at the engine design speed. Power, BMEP and BSFC were calculated and presented to show the effect of varying inlet valve throat diameter on them.

II. THEORETICAL ANALYSIS

For the purpose of analyzing the engine characteristics the dimensions were considered with a specially designed program used to predict the gas flows, combustion and overall performance of internal combustion engines. Engine Speed was varied between 500 to 3500 rpm. Ignition was taken 10^{o} bTDC. Program Structure can be conceptualized as comprising three discrete modules. The data entry and model generation are shown in Table (1).

Table .1 Base Engine Data

<table>
<tr><td colspan="2">No. of cylinders</td><td colspan="2">Bore mm</td><td>Stroke mm</td><td colspan="2">Connecting rod length mm</td><td>compression ratio</td></tr>
<tr><td colspan="2">1</td><td colspan="2">95</td><td>85</td><td colspan="2">129.8</td><td>8</td></tr>
<tr><td colspan="8">Fuel type is Iso-Octane (C_8H_{18}) of the following properties</td></tr>
<tr><td colspan="3">Heating value</td><td colspan="2">Density</td><td colspan="2">H/C molar</td><td>molecular mass</td></tr>
<tr><td colspan="3">43000 kJ/kg</td><td colspan="2">0.75 kg/liter</td><td colspan="2">1.8</td><td>114.23 kg/k.mol</td></tr>
<tr><td colspan="8">Valves data are as follows considering single valves for inlet and exhaust valve. The original valve timing is shown in Figure (1 and 2), inlet throat diameter and valves lift</td></tr>
<tr><td>IVO angle</td><td>IVC</td><td>EVO</td><td>EVC</td><td>inlet throat dia.</td><td colspan="2">exhaust throat dia.</td><td>maximum valve lifts</td></tr>
<tr><td>54^{o} bTDC</td><td>22^{o} aBDC</td><td>22 bBDC</td><td>54 aTDC</td><td>31 mm</td><td colspan="2">26 mm</td><td>9.5 mm</td></tr>
</table>

Input data such as inlet pressure, temperature, equivalence ratio are also been introduced for all runs considered. Also the required exit data such as the back pressure are given.

The solution of the equations represents the physical processes to predict the flows between the elements of the model. It is designed to solve the energy, momentum and continuity equations as appropriate within each element to obtain the thermodynamic state variables and flow velocity at each crank angle throughout the engine cycle. The solution procedure is 'time marching' and a number of engine cycles are simulated in order to obtain a converged (cyclically repeatable) solution.

To simulate the engine the processes are broken down in such a way that a number of discrete sub-models, Such as, the thermodynamic properties model where, the program tracks the flow of gas as a mixture. For combustion the type of the fuel was as specified in the above table. The effect of gas temperature on gas properties such as c_p, c_v, and viscosity are calculated for the individual gas species and then 'averaged' using the Gibbs-Dalton relationships. Thus gas properties change appropriately with both gas composition and temperature.

The combustion process employed a single zone combustion model. The combustion rate defined via a one part *Wiebe* function. Dissociation effects (*CO* generation) were modeled through curve fits to the *Eltinge* diagram, which relates combustion products of *CO* and O_2 to user specified parameters of air-fuel ratio and mal-distribution.

The *Wiebe* function define the mass fraction burned as

$$m_{frac} = 1.0 - \exp^{-A\left[\frac{\theta}{\theta_b}\right]^{M+1}}$$

Where:

A = coefficient in *Wiebe* equation = 10 for gasoline
M = coefficient in *Wiebe* equation = 2.0 for gasoline
Θ = actual burn angle (after start of combustion) calculated by the program
Θ_b = total burn angle (0-100% burn duration)

Heat transfer was modeled in all elements. Within cylinders the empirically derived heat transfer correlation proposed by *Annand* was employed. It was chosen to be used in this analysis, the constant for such a case are available.

The connective heat transfer model proposed by *Annand* is defined as;

$$\frac{hD_{cyl}}{k} = A\ \mathrm{Re}^{B}$$

Where:
h = heat transfer coefficient [W/m^2 K]
A = *Annand* open or closed cycle, A coefficient = 0.2
B = *Annand* open or closed cycle, B coefficient = 0.8
k = thermal conductivity of gas in the cylinder [W/m K]
D_{cyl} = cylinder bore B= 9.5 [mm]
Re = Reynolds number based upon mean piston speed and the engine bore. The density that calculated for the cylinder contents at each crank angle

Thus the heat transfer per unit area of cylinder wall is defined as;

$$\frac{dQ}{A} = h\left(T_{gas} - T_{wall}\right) + C\left(T_{gas}^{4} - T_{wall}^{4}\right)$$

Where:
dQ/A = heat transfer per unit area [W/m^2]
C = *Annand* closed cycle coefficient = 0 for the case considered.

The first part of the heat transfer equation is the connective heat transfer and the second part is the radiative heat transfer.

The outputs of the analysis and of the calculated results were given in an output file. Details of the element conditions and flows at each crank angle are stored for subsequent post processing. These results include in-cylinder pressures, temperatures, volumes and fuel mass fractions burned as well as all the input data of the test.

The IVO angle was varied while all other parameters were kept constant at different engine speeds collecting the results for further processing. Also the IVC angles were varied while other parameters were kept constant and it was also the case for overlap angle variation.

The exhaust gas emission produced due to engine run, at the design speed of 2500 rpm, was investigated using the engine data and fuel utilized. The combustion pressures attained and the fuel air ratio used was fed to the Engineering Equation Solver software EES and the values of mole fraction of NO, CO, CO_2, H_2O, O_2 and N_2 were recorded and later plotted for all cases considered.

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III. RESULTS

A. Inlet Valve Opening Angle (IVO) Effect

For the engine geometry and running conditions shown in Table (1). All parameters were kept constant except the IVO angle and the inlet throat diameter.

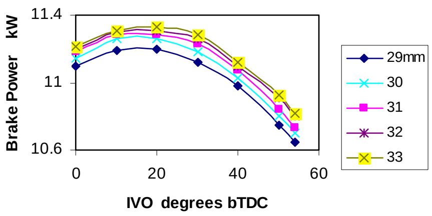

FIGURE (1) Power versus IVO angle, at different inlet throat diameters (N_d = 2500 rpm)

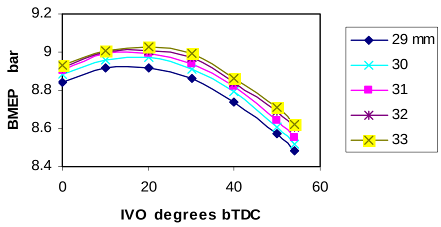

FIGURE (2) BMEP versus IVO angle, at different inlet throat diameters (N_d = 2500 rpm)

The IVO was varied from the original value 54° IVO angle bTDC opening down to 0° at TDC in steps for different inlet throat diameter values. As shown in Figure (1) the brake power is drawn versus the IVO angle opening bTDC for inlet throat diameter (29,30,31,32 and 33 mm) at the engine design speed (N_d = 2500 rpm). It showed an increase in power with the IVO angle reduction up to values of IVO angle bTDC less than 20° for all inlet throat diameters considered. But the power was decreased for lower values of IVO down to 0°. The increase in power may be due to the reduction of residual gases and backflow of exhaust into the inlet manifold. The reduction of inlet throat diameter show a considerable reduction in power may be due to the restriction on charge gas inflow to the cylinder may be due to the viscous effect of the attached jet formed which is Reynolds number dependent [9], while the increase of inlet throat diameter shows a slight increase in power over the original 31 mm throat diameter. Early IVO angle bTDC cause the high pressure exhaust gas reducing the amount of inlet mixture incoming through the inlet manifold this is quite noticed at high inlet throat diameters on early IVO opening.

Figure (2) shows the variation of BMEP versus IVO angle bTDC at different inlet throat diameters. The late opening of inlet valve showed an increase in BMEP especially at higher inlet throat diameter. A less effect on BMEP is noticed at IVO angles less than 20°. It shows a similar behavior of the considerable decrease in BMEP due to the decrease in inlet throat diameter. This was also noticed by [1] as the BMEP Increases by decreasing the IVO angle bTDC.

Figure (3) shows the variation of BSFC versus IVO angle bTDC. This showed that BSFC is hardly affected by IVO angle bTDC higher than 20° for different inlet throat diameters. But it shows an increase in BSFC for the entire inlet throat diameter considered at IVO angles less than 20°.

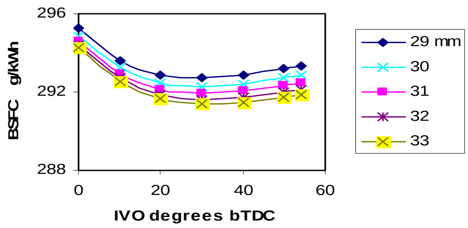

FIGURE (3) BSFC versus IVO angle, at different inlet throat diameters (N_d = 2500 rpm)

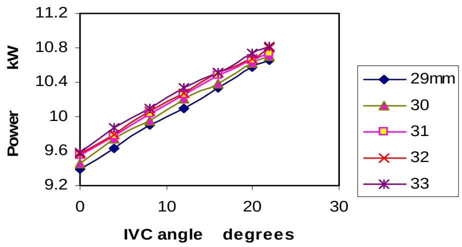

FIGURE (4) Power versus IVC angle, at different inlet throat diameters (N_d = 2500 rpm)

B. Inlet Valve Closing Angle (IVC) Effect

For the engine geometry and running conditions shown above, all parameters were kept constant except the IVC angle and the inlet throat diameter. The former was varied from the original value 22° IVC angle aBDC opening down to 0° at BDC in steps for different inlet throat diameter values. Figure (4) show the brake power drawn versus the IVC angle closing aBDC for inlet throat diameters (29, 30, 31, 32 and 33 mm) at the engine design speed (N_d = 2500 rpm). It showed a decrease in power with the IVC angle reduction for all inlet throat diameters considered. But it is less sever at larger inlet throat diameters. That was noticed by [4], it shows that, a late IVC reduce the volumetric efficiency. In contrast early IVC leads to greater reduction in volumetric efficiency, and this limits the output power. This shows a noticeable decrease in volumetric efficiency η_{vol} with reducing IVC angle aBDC, especially at smaller inlet throat diameters. This was also noticed by [8], which will lead to limit the maximum power output.

Figure (5) shows the variation of BMEP versus IVC angle aBDC at different inlet throat diameters. The late closing of the inlet valve close aBDC showed an increase in BMEP for all inlet throat diameters and of nearly similar trends but with a less effect at high inlet throat diameters.

Figure (6) shows the variation of BSFC versus IVC angle aBDC. This showed that BSFC is slightly affected by IVC angle aBDC, as it is increased slightly by reducing IVC angle aBDC.

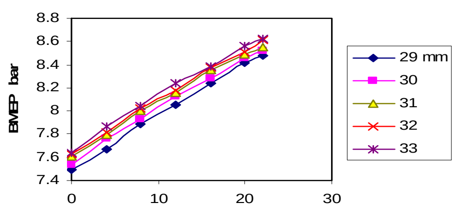

FIGURE (5) BMEP versus IVC angle, at different inlet throat diameters (N_d = 2500 rpm)

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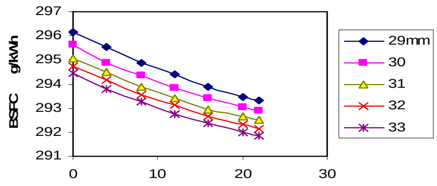

FIGURE (6) BSFC versus IVC angle, at different inlet throat diameters (N_d = 2500 rpm)

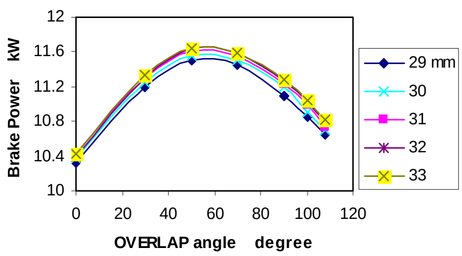

FIGURE (7) Power versus overlap angle, at different inlet throat diameters (N_d = 2500 rpm)

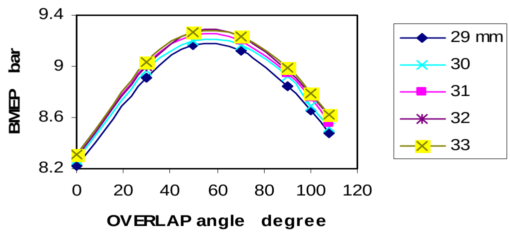

FIGURE (8) BMEP versus overlap angle, at different inlet throat (N_d = 2500 rpm)

C. *Overlap Angle (IVO - EVC) Effect*

For the engine geometry and running conditions shown above, all parameters were kept constant except the overlap angle between IVO and EVC at TDC and inlet throat diameters. The former was varied from the original value 108^o overlap angle at TDC down to 0^o at TDC in steps while for the later Five values were considered (29,30 ,31,32 and 33 mm). Figure (7) shows the brake power drawn versus the overlap angle at TDC. For all inlet throat diameters the power showed an increase with reducing overlap angle till about 50^o angle then it starts a considerable decrease to lower values at 0^o overlap. It has a larger reduction for less inlet throat diameters considered.

Figure (8) shows the variation of BMEP versus overlap angle at TDC, this showed that BMEP have a similar trend as the torque with overlap angle at TDC values for all inlet throat diameters and they nearly have a similar trend also. But the effect at high overlap angles and at large inlet throat diameters is less sever, also at small inlet throat diameters has a more sever effect.

Figure (9) shows the variation of BSFC versus overlap angle at TDC, this showed that BSFC decreases to a minimum at lower overlap angles at TDC then an increase toward the 0^o overlap angle. This is quite noticeable at small inlet throat diameters while it approaches a minimum at around overlap angle around 50^o.

D. *Engine Emissions*

Figure (10) shows the effect of IVO angle bTDC on NO emission at the design speed (2500 rpm) for different inlet throat diameters. The reduction of IVO angle causes a further reduction of NO mole fraction down to angle 10^o bTDC. The reduction in NO is quit noticeable with increasing the inlet throat diameter to 33mm.

Figure (11) shows the effect of IVC angle bTDC on NO emission at the design speed (2500 rpm) for different inlet throat diameters. An ever increase in NO with the reduction of IVC angle down to 0^o.

Figure (12) shows the effect of overlap angle bTDC on NO emission at the design speed (2500 rpm) for different inlet throat diameters. A reduction in NO could be recognized for all inlet throat diameters down to 60^o then a sharp increase in NO as the overlap angle is reduced.

Figure (13) shows the effect of IVO angle bTDC on CO emission at the design speed (2500 rpm) for different inlet throat diameters. The reduction of IVO angle causes a further reduction of CO mole fraction down to angle 10^o bTDC.

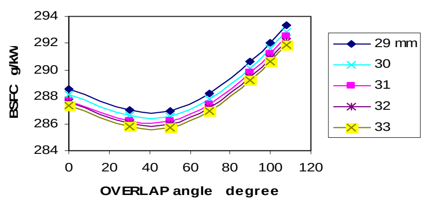

FIGURE (9) BSFC versus overlap angle, at different inlet throat diameters (N_d = 2500 rpm)

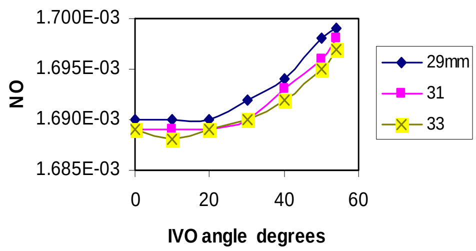

FIGURE (10) NO emission versus IVO at (N_d=2500 rpm)

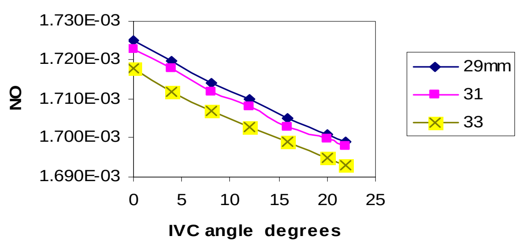

FIGURE (11) NO emission versus IVC at (N_d=2500 rpm)

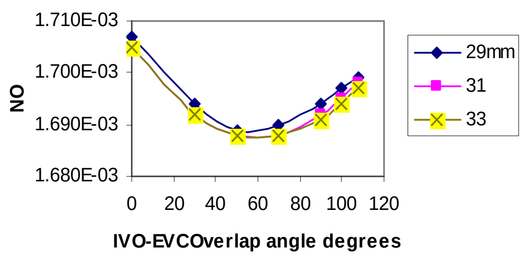

FIGURE (12) NO emission versus overlap angle at (N_d =2500 rpm)

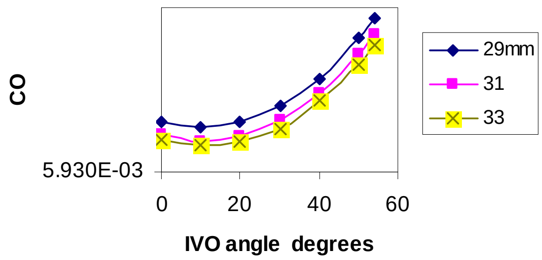

FIGURE (13) CO emission versus IVO at (N_d =2500 rpm)

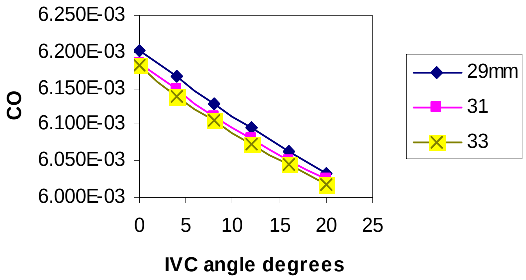

FIGURE (14) CO emission versus IVC at (N_d =2500 rpm)

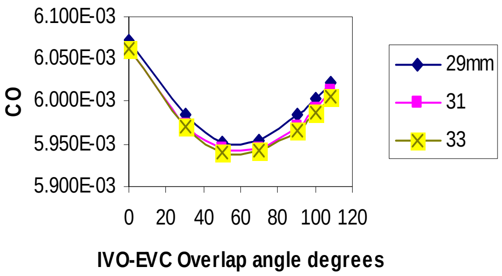

FIGURE (15) CO emission versus overlap angle at (N_d=2500 rpm)

Figure (14) shows the effect of IVC angle bTDC on CO emission at the design speed (2500 rpm) for different inlet throat diameters. An ever increase in CO with the reduction of IVC angle down to 0^o similar to NO behavior.

Figure (15) shows the effect of overlap angle bTDC on CO emission at the design speed (2500 rpm) for different inlet throat diameters. A reduction in CO could be recognized for all inlet throat diameters down to 60^o then a sharp increase in CO as the overlap angle is reduced. Values of other parameters were hardly altered like CO_2 and H_2O.

IV. CONCLUSIONS

1. The effect of IVO reduction is beneficial to power, torque, BSFC and volumetric efficiency at the design speed and also will reduce engine NO and CO emission up to 20^o degrees aTDC. Where there is a little effect on engine performance when IVO angles reduced to less than 20^o bTDC at all inlet throat diameters considered particularly at the large ones.
2. There is a reduction in engine performance by decreasing the IVC angle aBDC at all inlet throat diameters and an increase of NO and CO emissions were noticed.
3. The reduction in overlap angle between IVO and EVC shows an improvement in performance till about 50^o angle then it start to decrease It is also the case for NO and CO emissions.
4. The reduction of IVO bTDC produced a better performance, while on the contrary the IVC reduction causes a drop in the engine performance and an increase in NO and CO emissions.
5. A better increase in engine performance is gained when reducing the overlap angle between IVO and EVC also causing a reduction in NO and CO emissions.

REFERENCES

[1] J. B. Haywood, “Internal Combustion Engine Fundamentals”. New York: McGraw-Hill, 1988.
[2] G. Fontana, E. Galloni, "Variable valve timing for fuel economy improvement in a small spark-ignition engine"; Applied Energy 2009; 86:96–105
[3] T. Ahmad, M. Thobald, "A survey of variable valve actuation technology"; SAE, New York, SAE paper 89, 674, 1989.
[4] C. Gray, "A review of variable engine valve timing"; SAE, New York, SAE paper, 1988;880 386.
[5] S. Shiga, S. Yagi, M. Morita, and T. Matsumoto, T. Karasawa, and T. Nakamura, "Effects on valve close timing for internal combustion engines"; Trans Jpn Soc Mech. Eng Ser B 1996;62(596):1659–65.
[6] N. Kosuke, K. Hiroyuki, and K. Kazuya, " Valve timing and valve lift control mechanism for engines" ; Mechatronics 2006; 16:121–129
[7] J. Benajes, S. Molina, J. Martin, and R. Novella, "Effect of advancing the closing angle of the intake valves on diffusion-controlled combustion in a HD diesel engine"; Applied Thermal Engineering 2009;29:1947–1954
[8] R. Stone, "Introduction to Internal Combustion Engines"; McMillan Press,3rd edition, London, 1999.
[9] R. C. Ferguson, A. T. Kirkpatrick, "Internal Combustion Engines Applied Thermodynamics", 2nd Edition, John Wiley and sons inc., New York, 2001.
[10] H. Baur, "Gasoline-Engine Management - Basic Components"; Bosch Company, 1st edition, Stuttgart, 2001.

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