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Effect of Inlet Valve Variable Timing in the Spark Ignition Engine on Achieving Greener Transport

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***Abstract*—**The current emission legislations and the large concern about the environment produced very numerous constraints on both governments and car manufacturers. Also the cost of energy increase means a reduction in fuel consumption must be met, without largely affecting the current engine production and performance. It is the intension to contribute towards the development and pursuing, among others on variable valve timing (*VVT*), for improving the engine performance. The investigation of the effect of (*IVO*) and (*IVC*) to optimize engine torque and volumetric efficiency for different engine speeds was considered. Power, BMEP and BSFC were calculated and presented to show the effect of varying inlet valve timing on them for all cases. A special program used to carry out the calculations. The analysis of the results shows that the reduction of 10% of (*IVO*) angle gave an improvement of around 1.3% in torque, BSFC, and volumetric efficiency, while a 10% decrease in (*IVC*) caused a 0.1% reduction in power, torque, and volumetric efficiency.

***Keywords*—**Green transportation, inlet valve variable timing, performance, spark ignition engines.

NOMENCLATURE

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

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I. INTRODUCTION

THE control of green-house gas emissions has begun to add to the numerous constraints that vehicle manufacturers have to satisfy. The reduction of engine fuel consumption becomes a primary requirement as well as meeting current and future emission legislations. Naturally, talking about reduction of engine fuel consumption means to keep unvaried, sometimes improved, the performance level of current engine production [1]-[3]. Dealing with engine topics exclusively to improve fuel economy hence reduce CO_2 emissions means improving the engine thermal efficiency [1] .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. 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 [5]–[7].

TABLE I
BASE ENGINE DATA, FUEL, AND VALVES DATA

No. of cylinders	1
Bore [mm]	95
Stroke [mm]	85
Connect. rod length [mm]	129.5
compression ratio	8
Fuel type	(C_8H_{18})
IVO angle, degree	54^o bTDC
IVC angle, degree	22^o aBDC
EVO angle, degree	22 bBDC
EVC angle, degree	54 aTDC
inlet throat dia. [mm]	31
exhaust throat dia. [mm]	26
maximum valve lifts [mm]	9.5

Variable valve timing (VVT) is used in spark ignition automotive engines to improve fuel economy, reduce NOx gases, and increase peak torque and power. A number of papers have been cited by [8]. They have been reported for the effects when variable timing cam is used.

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 [9].

The objective of this paper is to contribute towards the development and pursuing, of variable valve timing (VVT), for improving the engine performance. Furthermore, the investigation of the effect of IVO, IVC on engine performance to optimize engine torque and volumetric efficiency, and the effect of engine speed has also been considered.

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].

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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 I.

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 model. The combustion rate defined via a one part *Wiebe* function [10]. 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 [11].

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 2500rpm, 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.

III. Results

A. Inlet Valve Opening Angle (IVO) Effect

For the engine geometry and running conditions shown in Table I. All parameters were kept constant except the IVO angle. It was varied from the original value54^{o} IVO angle bTDC opening down to 0^{o} in steps. As shown in Fig. 1 the brake power IVO angle for different engine speeds between (1000 - 5000rpm). It shows slight change in power with the IVO. This effect is more recognized with higher engine speeds (3000-5000rpm) [4]. He said that engine performance is fairly insensitive to inlet valve opening. The increase in power may be due to the reduction of residual gases and backflow of exhaust into the inlet manifold [13]-[16]. Fig. 2 shows the variation of torque versus IVO angle bTDC. The late opening of inlet valve shows an increase in brake torque especially at lower engine speeds (1000-2000rpm), whereas reduced effect on torque at higher engine speeds (2500-3500rpm) are noticed. Also this figure shows the insensitivity of IVO angle less than 25^{o}. Fig. 3 shows the variation of BSFC versus IVO angle. This shows that BSFC is hardly affected by IVO angle higher than 25^{o} for high engine speeds (3000-5000rpm) but it was insensitive to IVO angle for lower engine speeds (1000-2000rpm).

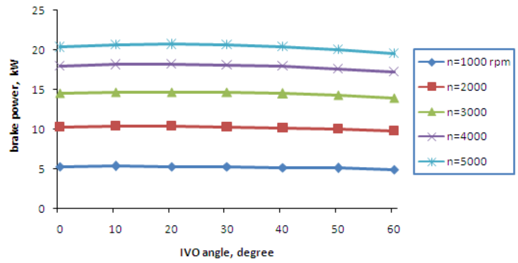

Fig. 1 Power versus IVO angle for different speeds

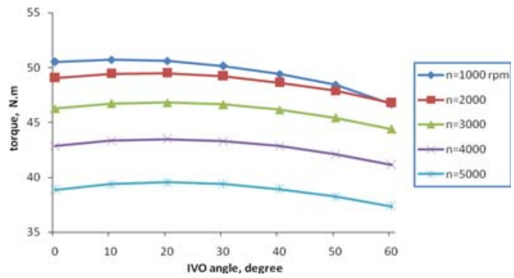

Fig. 2 Torque versus IVO angle for different speeds

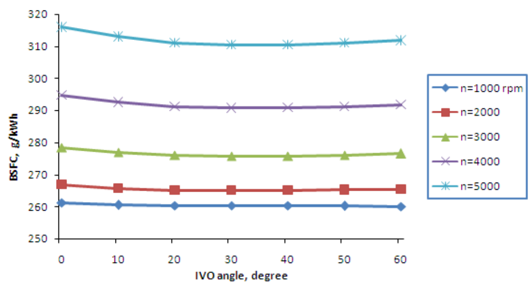

Fig. 3 BSFC versus IVO angle for different speeds

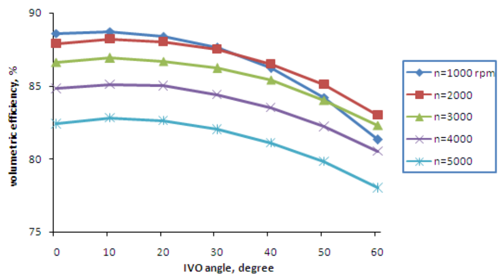

Fig. 4 Volumetric efficiency versus IVO angle for different speeds

Fig. 4 shows the effect of IVO angle on volumetric efficiency η_{vol}. This shows a noticeable increase in volumetric efficiency η_{vol} with reducing IVO angle especially at lower engine speeds. It hardly shows a variation with IVO angle less than 20^{o}. This might be due to lower induction flow speeds at low engine speeds. Early IVO angle causes the high pressure exhaust gas reducing the amount of inlet mixture incoming through the inlet manifold.

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. It was varied from the original value 22° aBDC down to 0° at BDC in steps. Fig. 5 shows the brake power drawn versus the IVC angle for different engine speeds between (1000 – 5000rpm). It shows a decrease in power with the IVC reduction for all engine running speeds. But it is less severe at lower engine speeds (1000-2000rpm).That was noticed by [4], who showed that, at low speeds, a late IVC reduces the volumetric efficiency. In contrast at high engine speeds early IVC leads to greater reduction in volumetric efficiency, and this limits the output power.

Fig. 6 shows the variation of torque versus IVC angle at different engine running speeds between (1000 – 5000rpm). The late closing of the inlet valve shows an increase in brake torque for all engine speeds and of nearly similar trends. Fig. 7 shows the variation of BSFC versus IVC; this shows that BSFC is slightly affected by IVC, as it is increases slightly by reducing IVC angle.

Fig. 8 shows the effect of IVC angle on volumetric efficiency η_{vol}. This shows a noticeable decrease in volumetric efficiency η_{vol} with reducing IVC angle aBDC, especially at higher engine speeds (3000 – 5000rpm) and at a lower slop for lower engine speeds (1000 – 2000rpm). This was also noticed by [12], which will lead to limit the maximum power output.

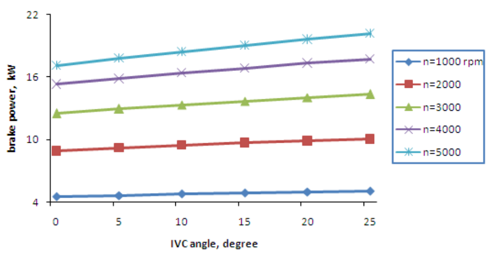

Fig. 5 Power versus IVC angle for different speeds

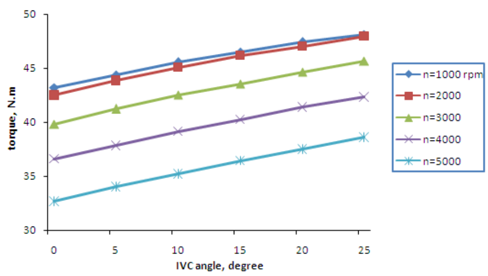

Fig. 6 Torque versus IVC angle for different speeds

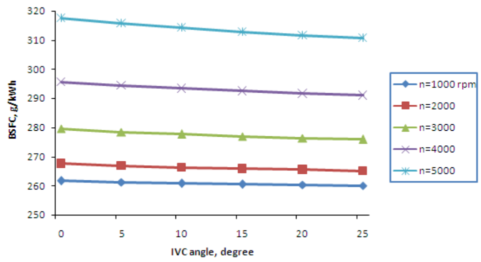

Fig. 7 BSFC versus IVC angle for different speeds

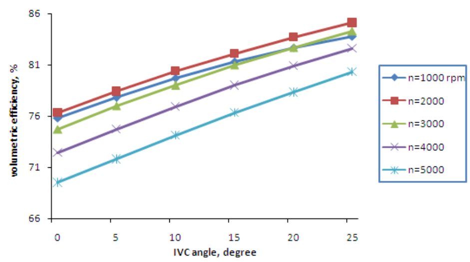

Fig. 8 Volumetric Efficiency versus IVC angle for different speeds

C. Engine Emissions

Fig. 9 shows the effect of IVO angle bTDC on NO emission for different engine speed The reduction of IVO angle causes a further reduction of NO mole fraction down to angle 10° bTDC. The reduction in NO is quit noticeable with increasing the inlet throat diameter to 33mm.

Fig. 10 shows the effect of IVO angle bTDC on CO emission for different engine speed. The reduction of IVO angle causes a further reduction of CO mole fraction down to angle 10° bTDC.

Fig. 11 shows the effect of IVC angle bTDC on NO emission for different engine speed. An ever increase in NO with the reduction of IVC angle down to 0°.

Fig. 12 shows the effect of IVC angle bTDC on CO emission for different engine speed. A reduction in CO could be recognized for all inlet throat diameters down to 60° then a sharp increase in CO as the overlap angle is reduced.

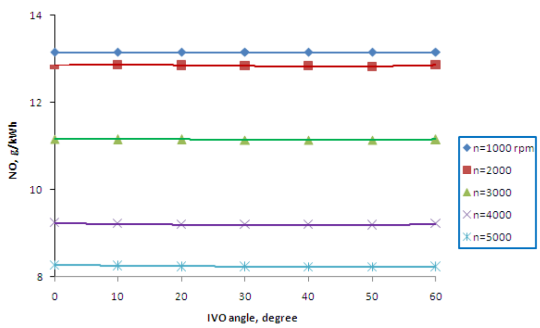

Fig. 9 NO emission versus IVO angles for different speeds

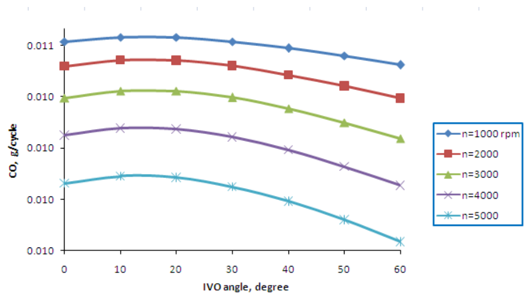

Fig. 10 CO emission versus IVO angles for different speeds

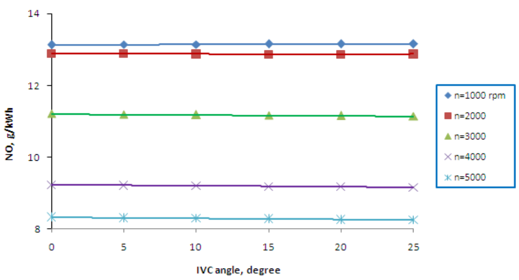

Fig. 11 NO emission versus IVC angles for different speeds

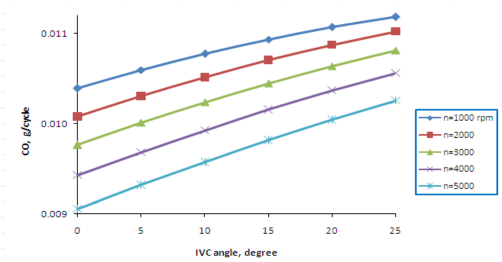

Fig. 12 CO emissions versus IVC angles for different speeds

IV. Conclusions

The effect of IVO reduction is beneficial to power, torque, BSFC and volumetric efficiency at design speed and also reduces engine NO and CO emissions. But, there is a little effect on engine performance when IVO angles reduced to less than 25° bTDC.

There is a reduction in power, torque, BSFC and volumetric efficiency by decreasing the IVC angle aBDC particularly at higher engine speeds with increase of NO and CO emissions. On the contrary the IVC reduction causes a drop in the power, torque, BSFC and volumetric efficiency and an increase in NO and CO emissions.

References

[1] Y. Najjar, and H. Abu Kayyas, "Performance of Automotive Reciprocating Engines with Different Fuels", Journal of Fuel Science and Technology International, Vol. 6, No. 3, pp. 315-327, 1988.
[2] Y. Najjar, and A. Al-Turki, "Modeling and Simulation of Spark-Ignition Engines", Journal of Fuel Science and Technology International., Vol.14, No. 8, pp993-1018, 1996.
[3] Y. Najjar, and A. Al-Turki, "Modeling and Simulation of Compression-Ignition Engines", Journal of Fuel Science and Technology International, Vol. 14, No.8, pp1019-1035, 1996.
[4] G. Fontana, E. Galloni, "Variable valve timing for fuel economy improvement in a small spark-ignition engine"; Applied Energy 86 (2009) 96–105.
[5] T. Ahmad, and M. Thobald, "A survey of variable valve actuation technology"; SAE, New York, SAE paper 89, 674, 1989.
[6] C. Gray, "A review of variable engine valve timing"; SAE, New York, SAE paper, 880 - 386, 1988.
[7] S. Shiga, S. Yagi, M. Morita, 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.
[8] N. Kosuke, K. Hiroyuki, and K. Kazuya, "Valve timing and valve lift control mechanism for engines" ;Mechatronics 16 (2006) 121–129
[9] 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 29 (2009) 1947–1954
[10] Habempirische Formel fur die Verbrennungsgeschrwindigkeit Verlag der Akademie der Wissenschaften der VdSSR I.Wiebe Moscow (1956)R.
[11] L. Eltinge, "Fuel-Air Ratio and Distribution from Exhaust Gas Composition", SAE 680114. 1968.
[12] W. Annand, "Heat Transfer in the Cylinder of Reciprocating Internal Combustion Engines", Proc.I.Mech.E, 177-973, 1963.
[13] R. Stone, "Introduction to Internal Combustion Engines"; McMillan Press, 3rd edition, London, 1999.
[14] R. Ferguson, and A. Kirkpatrick,"Internal combustion engines applied thermodynamics", 2nd Edition, John Wiley and sons inc., New York, 2001.
[15] H. Baur, "Gasoline-engine Management - Basic components"; Bosch Company, 1st edition, Stuttgart, 2001.
[16] J.B.Heywood, "Internal Combustion Engines Fundamentals", McGraw Hill, 1988.