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Relative Change in SI Engine Power and Economy with Variable Valve Timing: Simulation and ANOVA Analysis

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Abstract

In this research, a comparison between the performance of Ricardo E6/T variable compression ratio engine was conducted at constant and variable valve timing. This was done at three different speeds. The inlet and exhaust valves timing was varied and the performance was tested using specialized software. Results showed strong dependency of engine power and economy on valve timing and lift for all speeds. Further, higher lifts and durations are favorable for good power and the opposite are favorable for good economy. Valve overlap is affecting the engine power and economy in opposite direction as other factors.

Keyword: VVT, Engine performance, engine emissions, CVT, DOE, ANOVA

1. Introduction

A major goal of engine manufacturers is to minimize specific fuel consumption and emissions from engines. One solution is by the independent actuation of the inlet and exhaust valves at any position of the piston, with no more need for a camshaft (Hong et al., 2004).

A major disadvantage of conventional spark ignition (SI) engines results from the energy losses during inhaling of the sub-atmospheric gases during the suction and the expelling of exhaust gases to the atmosphere during the exhaust stroke. These pumping losses depend on the opening and closing position of the throttle valve. The losses are high when the throttle valve tends to close intake and are low at wide-open throttle. Thus, the pumping losses are inversely proportional with the engine load.

Without a throttle valve, control of the air–fuel mixture can be realized by variation of the intake valve-opening period; therefore, variable valve timing (VVT) has great potential for reducing pumping losses.

Variable valve operating methods can be traced as far back as the steam age. As for conventional automotive internal combustion engines, variable valve timing (VVT) was first patented by Fiat in the late 1960's. However, Alfa Romeo was the first manufacturer to introduce VVT into a production vehicle in their 1980 Spider 2000. Since then, many manufacturers have incorporated the principles of VVT into their designs. The array of VVT variations from manufacturers has generated many different VVT system names/acronyms, such as VTEC and VVT-i.

Engines without VVT have non-adjustable camshafts, therefore, the valve lift, duration and timing are fixed. Once the camshafts and crankshaft are set, the valve timing cannot vary. Variable valve timing is used to aid performance, fuel economy, and lower emissions by enabling the optimization of engine performance under different loads and operations. With VVT, larger valve overlap, valve lift, duration, and timing adjustments can be achieved depending on engine speed, load, and temperature. For instance, at low engine speed the valve timing can be advanced to help throttle response and engine torque, whilst under load. The valve timing can be retarded to help reduce exhaust emissions and increase power at a higher RPM, where valve opening times are greater.

In customary internal combustion engines, the intake and exhaust valve-timing are fixed. The timing is selected such that an optimal performance is achieved at a single well defined design point (Atashkari et al., 2007 and Dresner et al., 1989). In order to increase the performance of internal combustion engines, several investigations have been conducted. One of the most important of these investigations is the one that tries to optimize the amount of timing of intake and exhaust valves for all intervals of engine load and speed in SI engines (Maekawa et al., 1989; Asmus,1991; Nakayasu et al., 2001). Valve control is one of the most important parameters for optimizing efficiency and emissions, permitting internal combustion engines to conform to the advent of the more recent federal gas mileage and emission requirements with their emphasis on lower engine speed and low pollution emissions (sher et al., 2002).

Variable valve-timing relates to both the opening time and opening duration. Control of the intake valve provides optimal filling of the cylinder at all engine speeds. This natural supercharging and the improved engine torque and power that accompany it, make it possible to downsize engine capacity and, thus, reduce fuel consumption at all operating conditions (Kohany et al., 1999). An early intake closing time (before bottom dead center (bBDC)) will cause the fresh mixture to expand until bBDC, and therefore, its temperature at the commencement of the compression stroke will be lower.

As a result, lower amounts of NOx, but higher amounts of HC are expected to be emitted. A way of controlling the load while improving fuel economy was suggested by (Ma, 1988).

Controlling valves' events can improve the torque curve, the brake power curve, or the indicator power curve of a given engine design. Variable valve timing can also be used to reduce fuel consumption and to a small extent the engine emissions (Nagumo et al, 1995). This is achieved by controlling the in-cylinder maximum temperature, and the amount of residuals remaining at the commencement of the compression stroke (EGR control).

2. The Study

The Ricardo E6/T variable compression engine, supplied by Ricardo, was used to verify the simulation model for this study. It is a single-cylinder, poppet valve, four-stroke type, and has a bore of 76.2mm and a stroke of 111.1mm respectively. The normal speed range is 1000-3500 rev/min. The compression ratio could be varied from 4.5:1 to 20:1.

The engine was coupled to a Laurence Scott 'NS' type Swinging Field AC Dynamometer. The dynamometer could be used for motoring, measuring power output and running constant engine speed.

It is supplied with a 3-phase, 440 volts, and mains power supply. The speed of the dynamometer was achieved by a separated oil-cooled regulator controlled by a hand wheel. Speed measurement was undertaken using the clock output of the crankshaft encoder; this was checked against a wall-mounted manual speed meter. Table (1) show some of the engine design data.

Table (1). Engine design and operating parameters

Parameter	Specification
Diameter	76.2mm
Stroke Length	111.1mm
Compression ratio	8.5
Engine speed	1000, 2000 and 3500 RPM
Inlet valve Opening/Closing	8 bTDC/36 aBDC
Exhaust valve Opening/Closing	43 bBDC/6 aTDC
Load	Full load

At each engine speed, with fixed throttle opening, stoichiometric air-fuel ratio and ignition timing, the model was run while changing the valve opening, closing and lift values. The angles were changed from 0 to 60 degrees while the lift was varied from 7-12mm. This process was repeated for each engine speed value.

3. Model Description

The combustion chamber was generally divided into burned and unburned zones separated by a flame front (Figure 1). The first law of thermodynamic, equation of state and conservation of mass and volume were applied to the burned and unburned zones. The pressure was assumed to be uniform throughout the cylinder charge. A system of first order ordinary differential equations were obtained for the pressure, mass, volume, temperature of the burned and unburned zones, heat transfer from burned and unburned zone, and mass flow into and out of crevices.

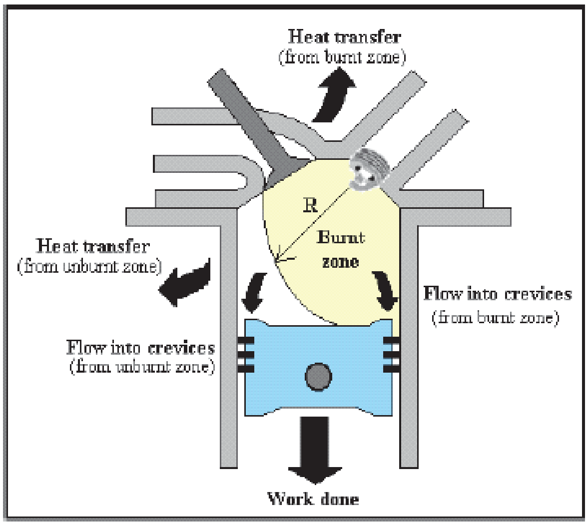


Figure 1. Two zone thermodynamic model for combustion.

The instantaneous cylinder volume measured from BDC position, which is a function of the crank-angle and the geometry of tile slider-crank, can be expressed as.

$$V(\theta) = Vs.\left[\left(\frac{CR}{CR-1}\right) - \left(\frac{1-\cos(\theta)}{2}\right) + \left(\frac{CRL}{S}\right) - \frac{1}{2}.\sqrt{\left(\frac{2*CRL}{S}\right)^2 - \sin^2(\theta)}\right] \quad (1)$$

By differentiating the equation (7), the rate of change of cylinder volume with crank angle is given by,

$$\frac{dV}{d\theta} = \frac{V_s}{2}\sin\theta\left(\frac{\cos\theta}{\sqrt{\left(\frac{2*CRL}{S}\right)^2 - \sin^2(\theta)}} - 1\right) \quad (2)$$

3.1 Compression

The following assumptions have been made during the calculations of compression stroke: (1) The mixing between fresh charge and residual gases is perfect, (2) No chemical reaction occurs during compression.

The calculation procedure starts with the trapped mass of fuel, air and residuals. The pressures and temperatures in this stroke are then calculated using the first law of thermodynamics equations and the equation of state (Yamin et al., 2000 and Al-Baghdadi, 2008):

$$\frac{dp}{d\theta} = \left\{\frac{R}{C_v}\left(\frac{dQ}{d\theta}\right) - p\frac{dV}{d\theta}\left(\frac{R}{C_v} + 1\right)\right\}\left(\frac{1}{V}\right) \quad (3)$$

and

$$\frac{dT}{d\theta} = T\left(\frac{1}{V}.\frac{dV}{d\theta} + \frac{1}{p}.\frac{dp}{d\theta}\right) \quad (4)$$

The work done by the reciprocating piston is given by:

$$\frac{dW}{d\theta} = p\left(\frac{dV}{d\theta}\right) \quad (5)$$

This continues till the nominal spark time, when combustion period is said to commence.

The heat transfer rate from the gas to wall is calculated using Annand's equation (Al-Baghdadi, 2008) for convective heat transfer:

$$\frac{Q}{A_p} = \frac{a\,k_q}{D}(R_e)^b (T_u - T_w) + \sigma\left(T_u^4 - T_w^4\right) \tag{6}$$

The instantaneous heat interaction between the cylinder content (burned and unburned zones) and its walls was calculated by using the semi-empirical expression for a four stroke engine:

$$-\frac{dQ}{dt} = A\left[0.26\frac{K_q}{D}\left(\frac{Du_p}{\mu}\right)^{0.7}(T - T_w) + 0.69\sigma(T^4 - T_w^4)\right] \tag{7}$$

The crevices are the volume between the piston, piston rings and cylinder wall (figure 1). Gases flow into and out of these volumes during the engine operating cycle as the cylinder pressure changes. The instantaneous energy flows to the crevices was calculated by using the semi-empirical expression of (Gatowski et al., 1984) for a spark ignition engine;

$$\frac{dQ_{cr}}{d\theta} = (u + R\,.\,T).\frac{dm_{cr}}{d\theta} \tag{8}$$

Where, dmcr > 0 when flow is out of the cylinder into the crevice, dmcr < 0 when flow is from the crevice to the cylinder, and (u+R.T) is evaluated at cylinder conditions when dmcr > 0 and at crevice conditions when dmcr < 0.

3.2 The Combustion Model

The rate of mass burning is calculated from the following equation given by (Heywood, 2018):

$$\frac{dM_f}{dt} = \rho_u A_f u_t \tag{9}$$

where, A_f = area of flame front calculated based on geometrical model

The turbulent entrainment flame speed is calculated using Keck equation (Yamin, 1999):

$$u_t = 0.08\ u_i \quad (\rho_u / \rho_b)^{0.5} \tag{10}$$

where, $u_i = \eta_{vol}\ (A_{iv} / A_p)$

3.3 Expansion with Two Zone

For this process, the major assumptions are:

1. The original charge is homogeneous;
2. Uniform pressure all throughout the cylinder at an time
3. Negligible flame front volume compared with cylinder volume
4. The products of combustion (except Nitrogen species) are in full equilibrium
5. Uniform local specific heat for both burned and unburned gases
6. The burned gases are frozen at the original composition

Though these assumptions show that the model is extremely simplified, however, experience show that they are well justified.

With reference to Figure (2) above, the total energy for the system is given by

$$U = m_u\ u_u + m_b\ u_b \tag{11}$$

Hence, the temperature and pressure of the burned and unburned mixtures can be obtained by applying the first law of thermodynamics, energy equation, flame speed and the geometry of the burned zone in relation to the combustion chamber.

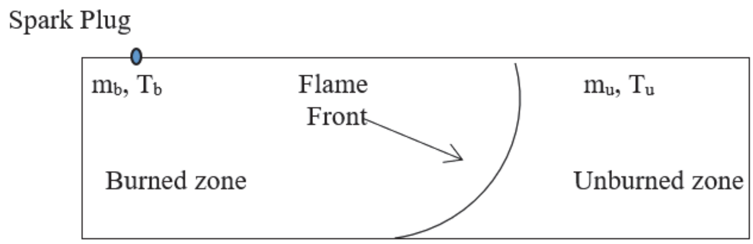


Figure 2. Combustion zone shape

The following equations are obtained:

$$\frac{dT_u}{d\theta} = \left(\frac{V_u}{m_u C_{P_u}}\right)\frac{dP}{d\theta} + \left(\frac{1}{m_u C_{P_u}}\right)\frac{dQ_u}{d\theta} \tag{12}$$

$$\frac{dT_p}{d\theta} = \frac{P}{m_p C_{P_p}}\left[\frac{dV}{d\theta} - \left(\frac{R_p T_p}{P} - \frac{R_u T_u}{P}\right)\frac{dm_p}{d\theta} - \left(\frac{R_u}{P}\frac{V_u}{C_{P_u}}\right)\frac{dP}{d\theta} - \left(\frac{R_u}{P C_{P_u}}\right)\frac{dQ_u}{d\theta} + \left(\frac{V}{P}\right)\frac{dP}{d\theta}\right] \tag{13}$$

$$\frac{dP}{d\theta} = \frac{\left\{\begin{array}{c}\left(1+\frac{C_{v_u}}{R_p}\right)P\frac{dV}{d\theta}+\left(\left(u_p-u_u\right)-C_{v_p}\left(T_p-\frac{R_u}{R_p}T_u\right)\right)\frac{dm_p}{d\theta} \\ +\left(\frac{C_{v_u}}{C_{p_u}}-\frac{C_{v_p}}{R_p}\frac{R_m}{C_{p_m}}\right)\frac{dQ_u}{d\theta}-\frac{dQ}{d\theta}\end{array}\right\}}{\left[\frac{C_{v_p} R_u}{C_{p_u} R_p}V_u-\frac{C_{v_u}}{C_{p_p}}V_u-\frac{C_{v_p}}{R_p}V\right]} \tag{14}$$

3.4 Valves Timing and Area of Opening

A review of the theory and model proposed by Benson indicates that the model is not compatible to other engines. Further, study of the effect of valve lift and flow area on engine performance is not possible using Benson's model.

Since the aim of the present work was to develop a program which is compatible to different types of 4-stroke S. I. engines, the following relations were introduced to calculate the valve lift and flow area as a function of crank rotational angle using the model presented in (Heywood, 2018). All the parameters can be understood with the help of Figure (3) below.

The instantaneous valve flow area depends on valve lift and the geometric details of the valve head, seat, and stem. There are three separate stages to the flow area development as valve lift increases as shown in Figure (4) below.

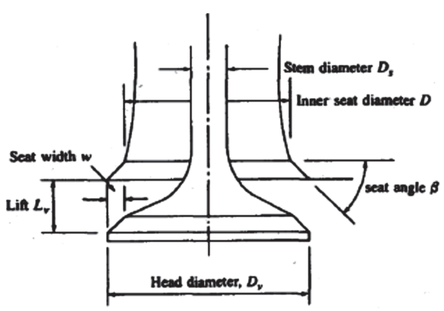


Figure 3. Valve geometrical parameters

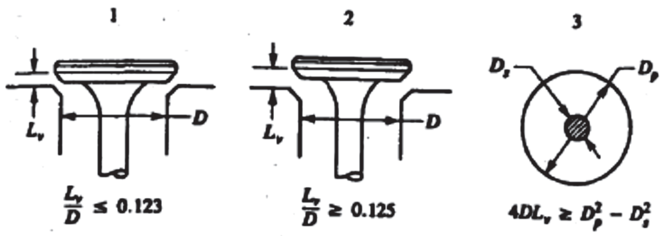


Figure 4. Stages of valve lift

For low valve lifts, the minimum flow area corresponds to a frustrum of a right circular cone where the conical face between the valve and the seat, which is perpendicular to the seat, defines the flow area. For this stage:

$$\frac{w}{\sin\beta\cos\beta} > L_{iv} > 0 \tag{15}$$

and the minimum area is:

$$A_m = \pi L_{iv} \cos\beta \left[D_v - 2w + \frac{L_{iv}}{2} \sin 2\beta \right] \tag{16}$$

where β is the valve seat angle, Liv is the valve lift, D_v is the valve head diameter (the outer diameter of the seat), and w is the seat width (difference between the inner and outer seat radii).

Second Stage: At this stage, the minimum area is still the slant surface of a frustrum of a right circular cone, but, this surface is no longer perpendicular to the valve seat. The base angle of the cone increases from (90 -β)^o towards that of a cylinder, 90^o. For this stage:

$$\sqrt{\left[\left(\frac{D_p^2 - D_s^2}{4 D_m}\right) - w^2\right]} + w \tan\beta \geq L_{iv} > \frac{w}{\sin\beta\cos\beta} \tag{17}$$

and the minimum area is :

$$A_m = \pi D_m \sqrt{\left[(L_{iv} - w \tan\beta)^2 + w^2\right]} \tag{18}$$

where Dp is the port diameter, Ds is the valve stem diameter, and Dm is the mean seat diameter (Dv - w).

Finally, when the valve lift is sufficiently large, the minimum flow area is no longer between the valve head and seat; it is the port flow area minus the sectional area of the valve stem.

Thus, for this stage

$$L_{iv} > \sqrt{\left[\left(\frac{D_p^2 - D_s^2}{4 D_m}\right) - w^2\right]} + w \tan\beta \tag{19}$$

and the minimum flow area is:

$$A_m = \frac{\pi}{4}\left(D_p^2 - D_s^2\right) \tag{20}$$

3.5 Species Formation

It is assumed that only 12 species are present in the combustion products both inside the cylinder as well as the exhaust. These are: H2O, H2, OH, H, N2, NO, CO2, CO, O2, O and A.

The governing equations for the mechanism of NO formation are based on Lavoie model (Winterbone et al., 2015):

$$\left.\begin{array}{llll}
(1)\ N + NO & \Leftrightarrow N_2 + O & K_{f1} = 3.1*10^{10} * E^{(-160/T)} & m^3/kmols \\
(2)\ N + O_2 & \Leftrightarrow NO + O & K_{f2} = 6.4*10^{6} * T * E^{(-3125/T)} & m^3/kmols \\
(3)\ N + OH & \Leftrightarrow NO + H & K_{f3} = 4.2*10^{10} & m^3/kmols \\
(4)\ H + N_2O & \Leftrightarrow N_2 + OH & K_{f4} = 3.0*10^{10} * E^{(-5350/T)} & m^3/kmols \\
(5)\ O + N_2O & \Leftrightarrow N_2 + O_2 & K_{f5} = 3.2*10^{12} * E^{(-18900/T)} & m^3/kmols \\
(6)\ O + N_2O & \Leftrightarrow NO + NO & K_{f6} = K_{f5} &
\end{array}\right\} \quad (21)$$

$$(7)\ N_2O + M \Leftrightarrow N_2 + O + M \quad K_{f7} = 10^{12} * E^{(-30500/T)} \quad m^3/kmols$$

These equations are used to give the rate of formation of (NO) as:

$$\frac{1}{V}\cdot\frac{d}{dt}[[NO].V] = 2.(1-\alpha^2)\left[\frac{R_1}{1+\alpha\frac{R_1}{R_2+R_3}} + \frac{R_6}{1+\frac{R_6}{R_4+R_5+R_7}}\right] \quad (22)$$

Where:

$$\beta = \frac{R_1+\alpha(R_2+R_3)}{(\alpha R_1+R_2+R_3)} \quad \text{and} \quad \gamma = \frac{R_4+R_5+\alpha^2 R_6+R_7}{(R_4+R_5+R_6+R_7)} \quad (23)$$

With R1 through R7 are the rates of reactions 1 to 7 respectively. The detailed method is given in (Winterbone et al., 2015).

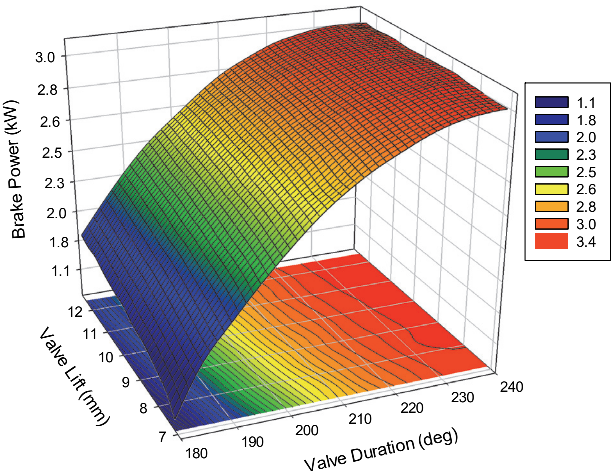
4. Results and Discussion

4.1 Engine Power

4.1.1 Case (I): Low speed (1000 RPM)

Figure (5) shows the variation of brake power with valve duration and lift. The figure shows clearly that the effect of valve lift has very little or no effect on engine brake power at low speed.

Figure (5) clearly show that, within the engine speed and valve duration studied, at low engine speed, the effect of valve lift is insignificant to brake power compared with inlet valve duration. This effect becomes more insignificant at higher valve durations. As clearly shown, the throttling effect (shown as pumping MEP) is least for higher valve durations and lift. The negative value for the PMEP curve is mere indication of the direction of the work (into cylinder). Valve lift, on the other hand has little effect on throttling loses at low valve durations. Similar findings was shown by (Sher et al., 2002 and Cairns et al., 2013).



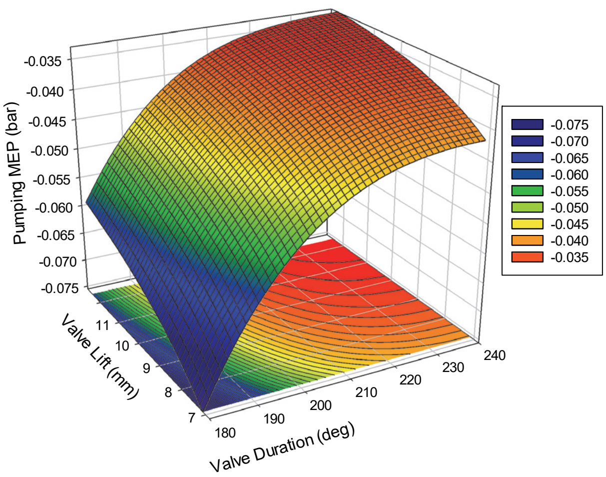


Figure 5. Variation of brake power and pumping MEP with valve duration and lift at low speed

Table 2. Statistical analysis of low speed effect

Source of Variation	DF	SS	MS	F	P
Valve Duration	12	93.591	7.799	60.819	<0.001
Valve Lift	12	1.523	0.127	0.990	0.456
Valve Duration x Valve Lift	144	0.529	0.00367	0.0286	1.000
Residual	2028	260.065	0.128		
Total	2196	355.707	0.162		

To determine, statistically, this effect, an ANOVA analysis was made on the data using Minitab 18. The data is summarized in table (2) above. The difference in the mean values among the different levels of Valve Duration is greater than would be expected by chance after allowing for effects of differences in Valve Lift. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The difference in the mean values among the different levels of Valve Lift is not great enough to exclude the possibility that the difference is just due to random sampling variability after allowing for the effects of differences in Valve Duration. There is not a statistically significant difference (P = 0.456).

The effect of different levels of Valve Duration does not depend on what level of Valve Lift is present. There is not a statistically significant interaction between Valve Duration and Valve Lift. (P = 1.000)

4.1.2 Case (II): Medium speed (2000 RPM)

Now, let us study the same effect at higher engine speeds. This is shown below in Figure (6). Though the effect of valve timing and lift is quite similar to that at lower speed, however, the throttling effect shows different behavior.

The effect of the two valve parameters are significant in throttling effect. This time the major effect is shown to be at relatively medium durations for all lifts. The maximum effect of throttling is shown to be at maximum valve lift for all durations. Hence we can conclude that the effect of inlet valve parameters is more effecting at medium speeds compared with lower ones. Perhaps the time factor during which the valve remains open plays the key role in this effect.

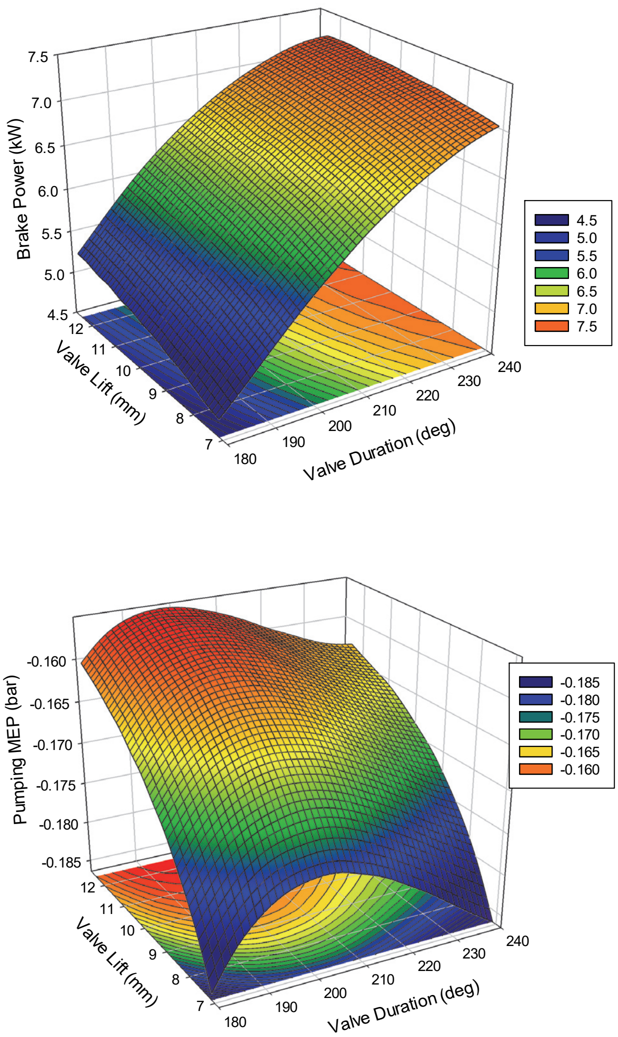


Figure 6. Variation of brake power with valve duration and lift at medium speed

Table 3. Statistical analysis of medium speed effect

Source of Variation	DF	SS	MS	F	P
Valve Duration	12	1015.514	84.626	76.428	<0.001
Valve Lift	12	43.161	3.597	3.248	<0.001
Valve Duration x Valve Lift	144	3.825	0.0266	0.0240	1.000
Residual	2028	2245.550	1.107		
Total	2196	3308.050	1.506		

Again, an ANOVA analysis was made on the data to determine this effect from statistical point of view. The data is summarized in table (3) above the difference in the mean values among the different levels of Valve Duration is greater than would be expected by chance after allowing for effects of differences in Valve Lift. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The difference in the mean values among the different levels of Valve Lift is greater than would be expected by chance after allowing for effects of differences in Valve Duration. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The effect of different levels of Valve Duration does not depend on what level of Valve Lift is present. There is not a statistically significant interaction between Valve Duration and Valve Lift. (P = 1.000)

4.1.3 Case (III): High Speed (3500 RPM)

Finally, let us study the same effect at higher engine speeds. This is shown below in Figure (7). As before, the effect of valve duration is dominant compared with valve lift. However, from throttling point of view, the effect of throttling is least at lower open durations with little effect for valve lift.

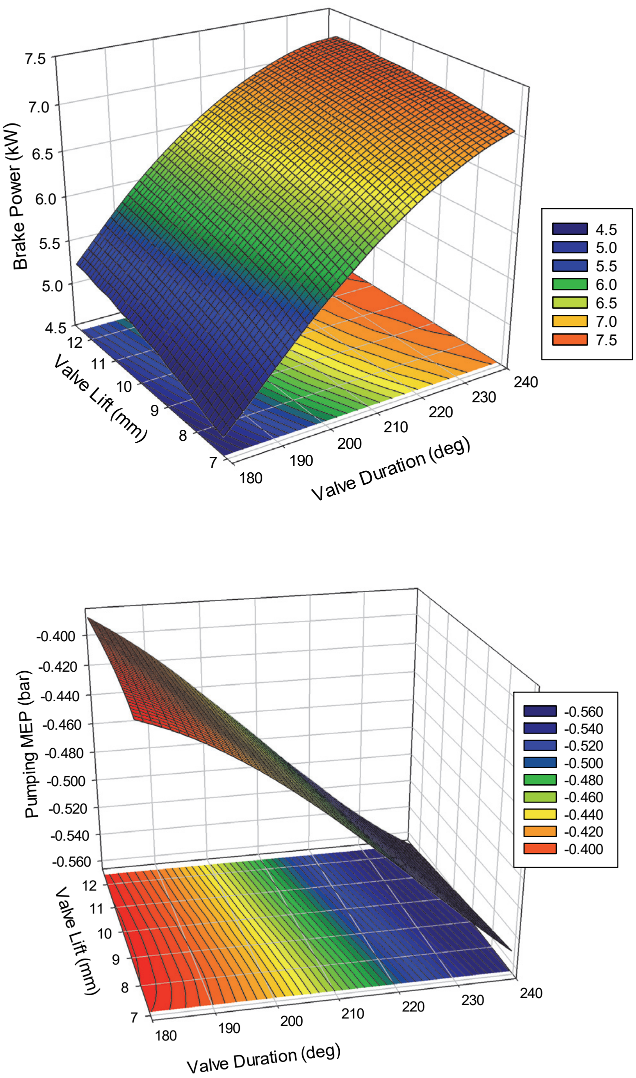


Figure 7. Variation of brake power with valve duration and lift at high speed

Table 4. Statistical analysis of high speed effect

Source of Variation	DF	SS	MS	F	P
Valve Duration	12	4812.109	401.009	96.734	<0.001
Valve Lift	12	467.313	38.943	9.394	<0.001
Valve Duration x Valve Lift	144	1.764	0.0122	0.00295	1.000
Residual	2028	8407.031	4.145		
Total	2196	13688.217	6.233		

From statistical point of view, Table (4) summarizes this effect. The difference in the mean values among the different levels of Valve Duration is greater than would be expected by chance after allowing for effects of differences in Valve Lift. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The difference in the mean values among the different levels of Valve Lift is greater than would be expected by chance after allowing for effects of differences in Valve Duration. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The effect of different levels of Valve Duration does not depend on what level of Valve Lift is present. There is not a statistically significant interaction between Valve Duration and Valve Lift. (P = 1.000).

4.2 Fuel Economy

4.2.1 Case (I): Low speed (1000 RPM)

From engine economy point of view, at lower engine speeds, the effect of valve durations is more noticeable compared with valve lift. The effect of higher durations is more dominant than the lower ones, while the lower valve lifts seem to be more favorable for fuel economy than higher ones. Similar results are shown in (Fontana t al., 2009).

These results are shown in the statistical analysis (shown in table (5)) with both being effective, however, more effect is in favor of valve durations.

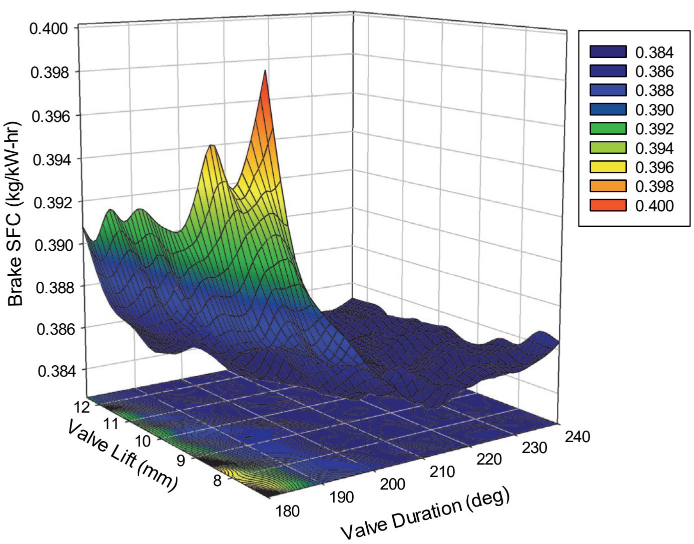


Figure 8. Variation of brake SFC with valve duration and lift at low speed

Table 5. Statistical analysis of high speed effect

Source of Variation	DF	SS	MS	F	P
Valve Duration	12	0.0135	0.00113	21.049	<0.001

Valve Lift	12	0.00126	0.000105	1.959	0.024
Valve Duration x Valve Lift	144	0.00130	0.00000901	0.168	1.000
Residual	2028	0.108	0.0000535		
Total	2196	0.125	0.0000567		

The difference in the mean values among the different levels of Valve Duration is greater than would be expected by chance after allowing for effects of differences in Valve Lift. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The difference in the mean values among the different levels of Valve Lift is greater than would be expected by chance after allowing for effects of differences in Valve Duration. There is a statistically significant difference (P = 0.024). To isolate which group(s) differ from the others use a multiple comparison procedure.

The effect of different levels of Valve Duration does not depend on what level of Valve Lift is present. There is not a statistically significant interaction between Valve Duration and Valve Lift. (P = 1.000)

4.2.2 Case (II): Medium Speed (2000 RPM)

At medium engine speeds, the effect of higher valve durations is more noticeable accompanied with lower valve lift. Valve lift effect is more significant at lower valve durations. Perhaps the lower pumping effect caused more of the fuel to be spent for power development, hence improved specific fuel consumption. This effect is also shown in Table (6). It shows that both parameters are effective with no interaction between them.

The difference in the mean values among the different levels of Valve Duration is greater than would be expected by chance after allowing for effects of differences in Valve Lift. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

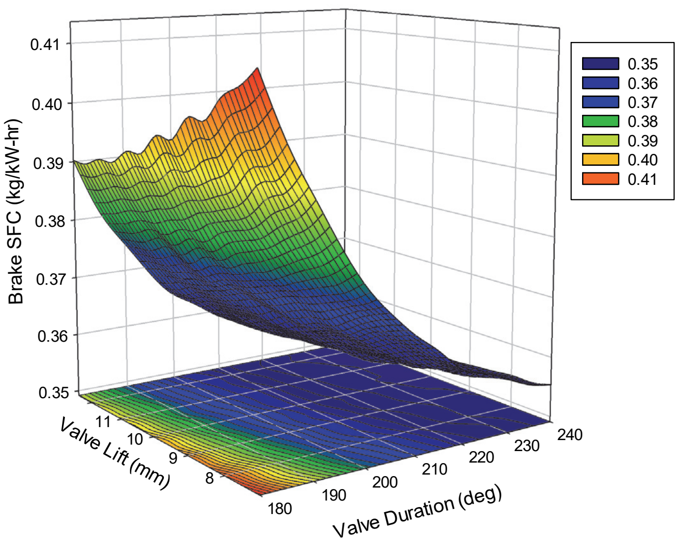


Figure 9. Variation of brake SFC with valve duration and lift at medium speed

Table 6. Statistical analysis of high speed effect

Source of Variation	DF	SS	MS	F	P
Valve Duration	12	0.413	0.0344	61.760	<0.001
Valve Lift	12	0.0222	0.00185	3.316	<0.001
Valve Duration x Valve Lift	144	0.00709	0.0000493	0.0883	1.000
Residual	2028	1.131	0.000558		
Total	2196	1.574	0.000717		

The difference in the mean values among the different levels of Valve Lift is greater than would be expected by chance after allowing for effects of differences in Valve Duration. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The effect of different levels of Valve Duration does not depend on what level of Valve Lift is present. There is not a statistically significant interaction between Valve Duration and Valve Lift. (P = 1.000)

4.2.3 Case (III): High speed (3500 RPM)

The above effect shown by the engine at medium speeds is also shown at higher speeds, however, the degree of significance of the valve lift is more at higher speeds. This is again related to the major drop of throttling effect at higher speeds with valve lift and duration. Table (7) also shows the same with no interaction between the parameters.

The difference in the mean values among the different levels of Valve Duration is greater than would be expected by chance after allowing for effects of differences in Valve Lift. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The difference in the mean values among the different levels of Valve Lift is greater than would be expected by chance after allowing for effects of differences in Valve Duration. There is a statistically significant difference (P = <0.001). To isolate which group(s) differ from the others use a multiple comparison procedure.

The effect of different levels of Valve Duration does not depend on what level of Valve Lift is present. There is not a statistically significant interaction between Valve Duration and Valve Lift. (P = 1.000)

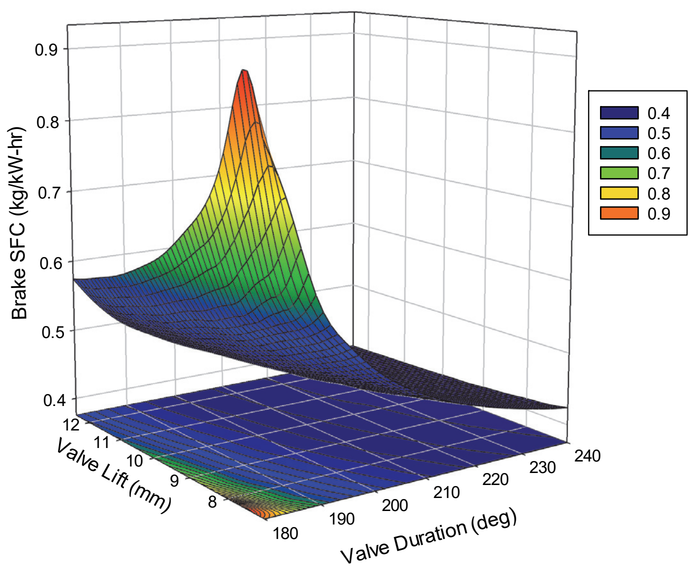


Figure 10. Variation of brake SFC with valve duration and lift at high speed

Table 7. Statistical analysis of high speed effect

Source of Variation	DF	SS	MS	F	P
Valve Duration	12	13.545	1.129	40.774	<0.001
Valve Lift	12	1.494	0.124	4.497	<0.001
Valve Duration x Valve Lift	144	1.572	0.0109	0.394	1.000
Residual	2028	56.140	0.0277		
Total	2196	72.750	0.0331		

Table 8. Correlation between various factors and engine power and SFC

	Engine speed	*Valve Duration*	*Value LIFT*	*Valve overlap*	*Brake Power*	*BSFC*
Engine speed	1.0000					
Valve duration	0.0000	1.0000				
Value lift	0.0000	0.0000	1.0000			
Valve overlap	0.0000	0.0000	0.0000	1.0000		
Brake power	0.7202	0.3112	0.0821	-0.4083	1.0000	
Brake SFC	0.3625	-0.2421	-0.0821	0.2921	-0.1865	1.0000

The above effects are shown and summarized by this correlation table (8). It is clear that there is no interaction between the inlet valve parameters. However, there is positive effect of both of them on engine power i.e. as they increase, the engine brake power increases. This is for all engine speeds studied. Except for valve overlap where the effect is negative. As for the fuel economy, this effect is negative which means there is shift towards lower durations and lift. This is not the case with valve overlap.

5. Conclusion

A good conclusion of the above work can be summarized in that the effect of inlet valve parameters depends, strongly on engine speed. Higher lifts and durations are favorable for good power and the opposite are favorable for good economy.

Valve overlap is affecting the engine power and economy in opposite direction as other factors.

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