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Theoretical and Experimental Analysis of Electromagnetic Variable Valve Timing Control Systems for Improvement of Spark Ignition Engine Performance

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Abstract

The objective of this paper is to meet the requirements of higher torque values at all engine speeds. This can be achieved by varying the valve timing automatically using a new variable valve timing system (VVT), which gives continuously variable valve actuation at all engine speeds. A model engine is designed using dimensional analysis methods and then implemented to verify the proposed control system. Moreover, microcontroller and computer-aided control systems are constructed and used to modify the variable valve timing control in the laboratory. In this paper, a mathematical model of variable valve timing is developed to obtain the best volumetric efficiency with optimum valve timing at different engine speeds. From this model, the look-up table is created at all ranges of the engine speed. A single cylinder engine is used to estimate engine performance characteristics for conventional camshaft. In addition, a model engine is designed and constructed to apply the Variable Valve Timing control system. The investigations show that the system is flexible throughout the entire range of operation speeds and is able to alter valve timing concerning both valve opening and closing. The ability of valve opening and closing can be realized with rates higher than these of the conventional timing mechanisms.

Keywords: Variable valve timing; Spark ignition engine; Control System; Experimental test; Theoretical approach; Performance

Introduction

The concept of control system to improve the performance and to satisfy constant engine torque for internal combustion engines is well established. However, the variable valve timing is one of the ways used to improve the performance and satisfy constant engine torque requirements. Numerous methods have been studied theoretically and experimentally for the control of valve timing.

The optimization of valve train actuation for internal combustion engines through the fixed geometry camshaft is a compromise of the required torque, fuel consumption, idle characteristics and exhaust emissions. Varying the engine valve-open duration, lift and phasing are known ways to improve engine performance, increase fuel economy and reduce emissions.

With the essential need to enhance overall vehicle efficiency there has been considerable efforts to improve the performance of spark ignition engine (SIE) over the past years. A main disadvantage of the conventional SIE is decreasing of volumetric efficiency at most engine speeds that cause reducing in engine torque [1,2]. To improve the volumetric efficiency two-intake valves are used in cylinder to widen the inlet charge port; however, widening the inlet charge port is not the only parameter for improving the volumetric efficiency. Engine volumetric efficiency depends on the actual mass charge entering the cylinder in the admission stroke [3,4]. This mass is a function of engine speed, throttle valve position (engine load) and valve timing, therefore, for a conventional engine the maximum volumetric efficiency will occur at a specific engine speed.

In order to increase the range of speeds in which the maximum value of the torque will remain constant, some techniques should be used to vary the valve timing. Proper valve timing is a function of the engine speed at which maximum torque is required.

Production VVT systems typically vary the relative camshaft position (cam phasing angle) [5-9] or using more than cam profile and select one among them [10,11] depending on engine operating conditions. These systems seek to reduce the impact of the traditional fixed valve timing compromise, which penalizes low volumetric efficiency versus high speed and low speed combustion stability. However, they cannot control the valve opening duration and lift, and considered as a stepped control systems. The most flexible actuators usually entail the separate actuation of each valve. The actuator, then, uses hydraulic system [12] or electromagnetic motor [13,14]. Also electromagnetic actuators are used [15-19] however, they are still not applicable since they require a lot of power to open and close. Similar work has been conducted by Vu and Pyung [20] to develop a new hybrid magnet engine valve actuator using shorted turn for fast initial response since the electromagnetic actuator using solenoids is the most advance system to provide the most flexibility to valve timing, but it has critical drawback of high power consumption [20].

Some of the above systems are capable of fast valve motion; however, they are very complicated and cannot be worked at high engine speeds such as hydraulic system. Using solenoid as an electromagnetic actuator has been proven as a suitable construction for variable valve actuation,

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where the valve spring fixes the closing position of the valve.

Theoretical analysis

In this section, there are three theoretical studies accomplished to calculate the actual engine volumetric efficiency, to evaluate the optimum volumetric efficiency at all engine conditions, and to estimate the proposed model engine size.

Calculation of actual engine volumetric efficiency

The function of the throttle body is to restrict the airflow into the engine in order to control the torque output. A simple model with an empirically determined discharge coefficient is presented to give reasonable flow prediction results. The throttle body model is based on a one-dimensional steady, isentropic, compressible flow equation for flow across an orifice. This equation is defined [21] as:

$$m_a^* = \begin{cases} C_d A(h) \frac{P_o}{\sqrt{R_a T_o}} \left(\frac{P}{P_o} \right)^{\frac{1}{k}} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{P}{P_o} \right)^{\frac{k-1}{k}} \right] \right\}^{\frac{1}{2}}, & \frac{P}{P_o} > \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \\ C_d A(h) \frac{P_o}{\sqrt{R_a T_o}} k^{\frac{1}{2}} \left(\frac{2}{k+1} \right)^{\frac{k+1}{2(k-1)}}, & \frac{P}{P_o} \leq \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \end{cases} \quad (1)$$

where C_d is orifice discharge coefficient, $A(h)$ is cross-sectional area of throttle body (m^2), P_o is stagnation pressure (kPa), R_a is ideal gas constant (kJ/kg·K), T_o is stagnation temperature (K), P is intake manifold pressure (kPa) and k is ratio of specific heats.

In the throttle body there is negligible pressure recovery downstream of the minimal cross-sectional area, so the throat pressure can be approximated by the intake manifold plenum pressure. The throttle body is not truly a one-dimensional orifice, and information reflecting this fact is contained in the discharge coefficient. This coefficient has been shown to be a function of the throttle lift (i.e., throat geometry) and the pressure ratio across the throttle body. Furthermore, steady-state flow tests have shown that these two influences are independent within the operating range of the engine.

The cross-sectional area of the throttle body $A(h)$ as shown in Figure 1, is a function of the throat diameter, lift and geometry of the plunger. At large throttle lift the geometry of the plunger has a significant effect on the throttle cross-sectional area. The final cross-sectional area of the

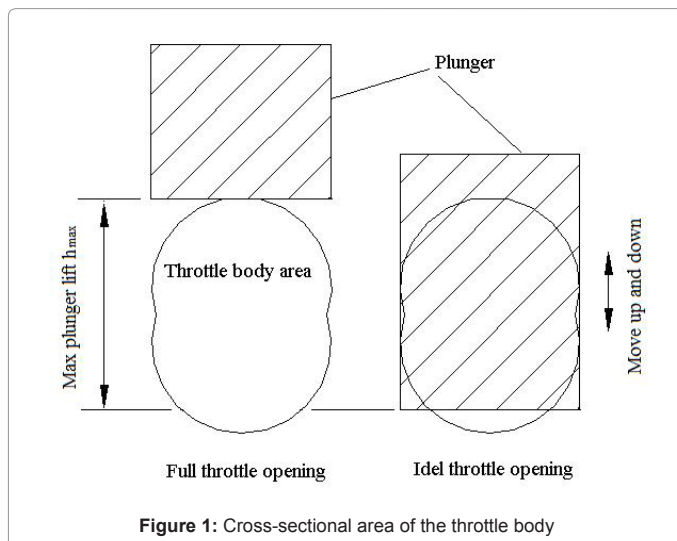


Figure 1: Cross-sectional area of the throttle body

throttle is deduced experimentally as follows:

$$A(h) = -0.022h^3 + 0.86h^2 + 9.36h - 3.41 \quad (2)$$

The volumetric efficiency then can then be calculated using the following mathematical expression:

$$\eta_v = \frac{120m_a^*}{NV_s \rho_a} \quad (3)$$

where η_v is the engine volumetric efficiency, N is the engine speed, V_s is the engine volume and ρ_a is the air density.

In the throttle body, there is negligible pressure recovery downstream of the minimal cross-sectional area, so the throat pressure can be approximated by the intake manifold plenum pressure. The throttle body is not truly a one-dimensional orifice, and information reflecting this fact is contained in the discharge coefficient. This coefficient has been shown to be a function of the throttle lift (i.e., throat geometry) and the pressure ratio across the throttle body. Furthermore, steady-state flow tests have shown that these two influences are independent within the operating range of the engine.

Evaluation of the theoretical volumetric efficiency

Theoretical analysis is used to evaluate the engine volumetric efficiency by considering the mass flow rate of charge through the engine valves. Following is the mathematical model and its associated equations. The mass flow rate is mainly influenced by the pressure drop through the valve and valve characterizing parameters (shape and dimensions). The following model assumptions are considered in this study:

1. The model is a quasi-steady thermodynamic, flame propagation model.
2. The combustion chamber is generally divided into burned and unburned regions. Each zone is uniform in composition, local specific heats and temperature through the calculation step.
3. The volume of flame reaction zone is negligible.
4. There is no heat transfer between the burned and unburned zones.
5. The cylinder charge is frequently assumed to be composed of frozen mixture of air, fuel vapor and residual gases, in the unburned gas zone.
6. The burned gases in the burned zone are assumed to be a mixture of reacting gases assumed to be in chemical equilibrium.
7. The first law of thermodynamics, equation of state, and conservation of mass and volume are applied to the burned and unburned zones.
8. The pressure is assumed to be uniform throughout the cylinder charge.

The first-order ordinary differential equations for pressure, mass, volume, composition, heat transfer and temperature of burned and unburned zones are given below.

The rate of changes of mass through the inlet or exhaust valves, and the cylinder temperature, volume and pressure can be calculated using the following:

$$\frac{dm}{d\phi} = A \cdot p_{up} \cdot \sqrt{\frac{2}{R_{up} \cdot T_{up}}} \cdot \sqrt{\frac{k}{k-1} \left[\left(\frac{p_d}{p_{up}} \right)^{\frac{2}{k}} - \left(\frac{p_d}{p_{up}} \right)^{\frac{k+1}{k}} \right]} \quad (4)$$

$$\frac{dm_c}{d\phi} = \frac{dm_i}{d\phi} - \frac{dm_e}{d\phi} \quad (5)$$

$$\frac{dT_c}{d\phi} = \left(\frac{dQ_c}{d\phi} - p_c \cdot \frac{dV_c}{d\phi} \right) / (C_{V_c} \cdot m_e) \quad (6)$$

$$\frac{dV_c}{d\phi} = \left(\frac{A_p \cdot r \cdot \pi}{180} \right) \left[\sin\phi + \sin 2\phi / \left(2\lambda \sqrt{1 - \left(\frac{\sin\phi}{\lambda} \right)^2} \right) \right] \quad (7)$$

$$\frac{dp_c}{d\phi} = p_c \left(\frac{dm_c}{d\phi} / m_c + \frac{dT_c}{d\phi} / T_c - \frac{dV_c}{d\phi} \right) \quad (8)$$

$$\frac{dQ_{j_u}}{d\phi} = h_{j_u} A_{j_u} (T_{j_u} - T_{g_j}) / (6N) \quad (9)$$

$$\frac{dR_2}{d\phi} = U_t / (6N), \dots, \frac{dV_b}{d\phi} = S_f \frac{dR_2}{d\phi} + \frac{dV_c}{d\phi} \frac{A_{b_r}}{A_p} \quad (10)$$

$$\frac{dm_b}{d\phi} = m_b \left(\frac{1}{p_c} \frac{dp_c}{d\phi} + \frac{1}{V_b} \frac{dV_b}{d\phi} - \frac{1}{T_b} \frac{dT_b}{d\phi} \right), \dots, \frac{dm_u}{d\phi} = - \frac{dm_b}{d\phi} \quad (11)$$

$$\frac{dT_b}{d\phi} = \frac{1}{m_b C_{V_b}} \left[\frac{dQ_b}{d\phi} - p_c \frac{dV_b}{d\phi} + \frac{dm_b}{d\phi} (h_b - u_b) \right] \quad (12)$$

$$\frac{dT_u}{d\phi} = \frac{1}{m_u C_{V_u}} \left[\frac{dQ_u}{d\phi} - p_c \frac{dV_u}{d\phi} + \frac{dm_u}{d\phi} (h_u - u_u) \right] \quad (13)$$

$$\frac{dp_c}{d\phi} = \frac{p_c}{V_c} \left[\left(\frac{V_b}{m_b} - \frac{V_u}{m_u} \right) \frac{dm_b}{d\phi} + \frac{m_u \cdot R_u}{p_c} \frac{dT_u}{d\phi} + \frac{m_b \cdot R_b}{p_c} \frac{dT_b}{d\phi} - \frac{dV_c}{d\phi} \right] \quad (14)$$

where m is the charge mass (kg), ϕ is crank angle (degree), A is valve area (m^2), p is pressure (bar), R and T are the gas constant (kJ/kg.K) and temperature (K) respectively, k is ratio of specific heats, Q is gas heat (kJ), V is volume (m^3), C_v specific heat at constant vol. (kJ/kg.K), r is crank radius (m), λ is ratio of con-rod length to crank radius, h is coefficient of heat transfer (kJ/kg.K), N is engine speed (rpm), R_2 flame front radius (m), U_t turbulent flame speed (m/s) and S_f is flame front area (m^2). The Suffixes meaning are as follow, up means upstream conditions, d downstream conditions, c cylinder, i inlet, e exhaust, p piston, j cylinder, burned or unburned, u unburned zone, g gas, j cylinder, burned or unburned and burned zone.

The calculation is based on an energy balance of charge flowing through engine cylinder. Knowing the pressure drop through the valve and discharge coefficients of engine valve being obtained as a function of the valve lift, the flow through the valve is calculated. The analysis starts at a point where the pressure and volume of the gas inside the engine cylinder are known. Then it would employ step by step method and the accumulated mass will be calculated inside the cylinder and hence the engine volumetric efficiency will be determined.

Size determination of the model engine

Various physical phenomena like fluid flow, heat transfer, involve

complex problems, which cannot be solved by analytical methods alone and one has to rely on experimental data. Before any large engineering project is undertaken, the performance of the component parts is studied by conducting series of experiments on a small-scale model of the system prototype. py theorem, geometric similarity, kinetic similarity, kinematic similarity, and dynamic similarity methods [22] are used to find the size of the model engine.

Experimental Setup

In this section, two test rigs are constructed; first test is related to a single cylinder SIE and the second is related to a proposed model engine. To estimate the SIE engine performance characteristics, the engine test rig measurements included electronic devices and data acquisition system are used as shown in engine Figure 2.

The test rig for microcontroller control method and computer-aided control method are shown in Figures 3a and 3b, respectively. It consists of an electric motor, model engine, inverter, power supply, and driving control circuit. The model engine is constructed and sized according to the dimensional analysis study. Model engine valve, its bush and cylinder head construction drawing are shown in Figures 4-6, respectively. The model engine is driven by an electric motor which its rotating speed can be changed using the inverter. Driving card is designed and fabricated to control the valve timing and duration of a model engine. The inverter is used to control the electric motor speed

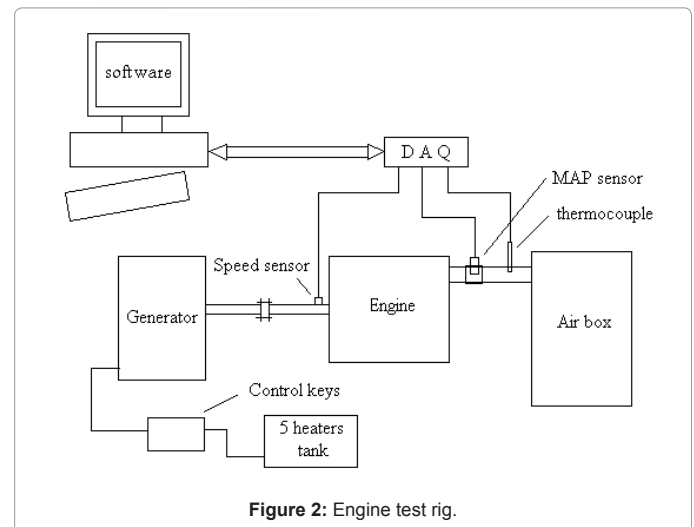


Figure 2: Engine test rig.

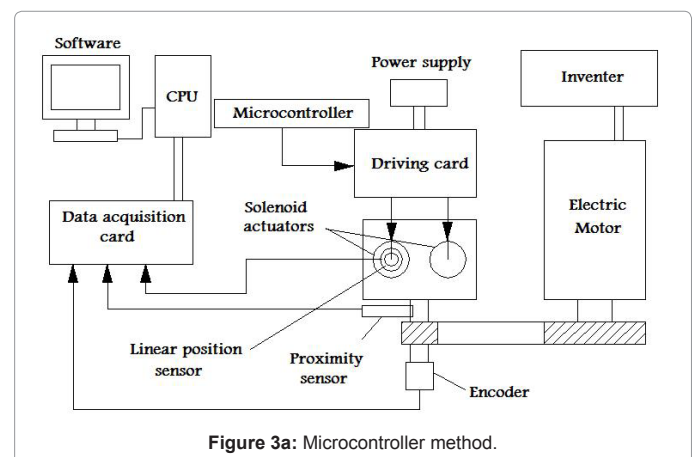
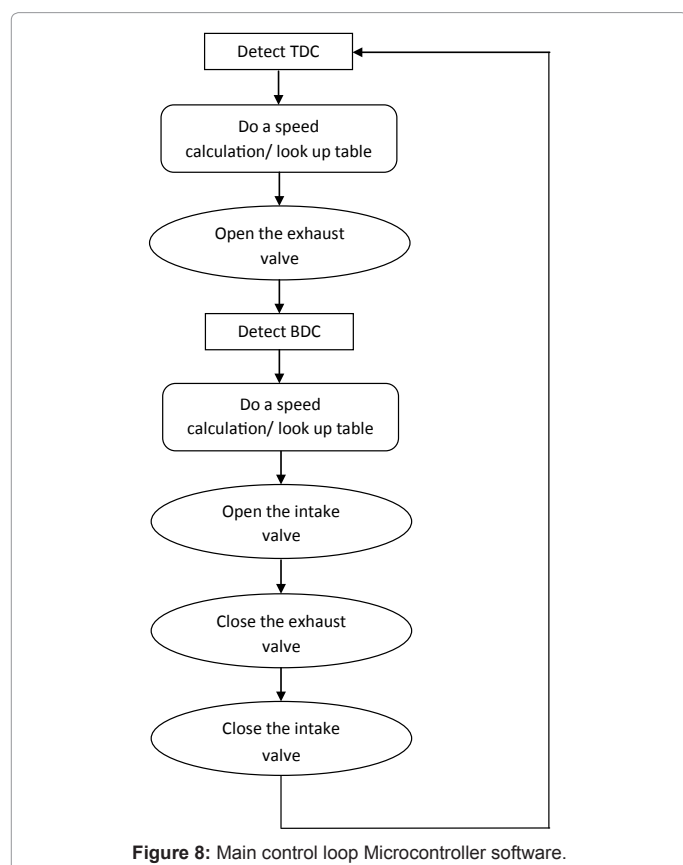
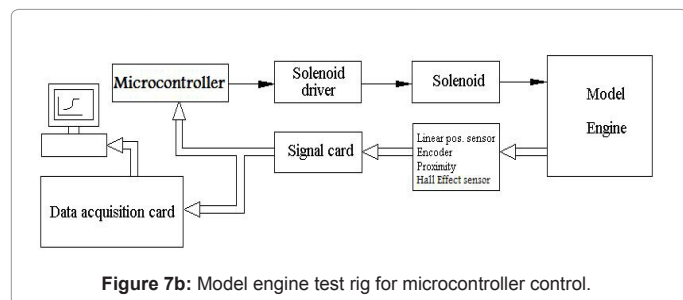
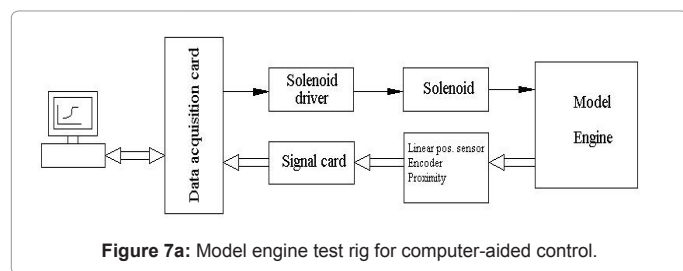


Figure 3a: Microcontroller method.



microcontroller control, Hall Effect sensors measure the engine speed and crank angle position and feed the input signals to the microcontroller. The microcontroller then generates digital output signals that operate the solenoid actuators to control the valve timing mechanism. For the computer control, Encoder sensor measures the engine speed and crank angle position, proximity sensor measures the Top Dead Center (TDC), and feed these inputs to the computer software. The computer generates digital outputs to the solenoid actuators.

The control sequence for the microcontroller is triggered by some incoming signals and input instructions. In this work, digital input/output system is chosen to speed up the control response. The engine speed and the crank angle position are measured by Hall Effect sensor, transfers the speed and the position measurements to the microcontroller memory through the multi I/O driving card.

When the control power is turned on the software initializes the microcontroller PIC's (Peripheral Interface Controller) I/O lines. The PIC controller is programmed to measure the engine speed by measuring the time each half revolution of the crankshaft takes. This is carried out using the PIC's timer 0 (TMR0). It measures the time from one rising edge of the SENSOR pulse to the next. Timer is programmed using the pre-scaled so that it rolls over every 2.048 milliseconds. By counting the TMR0 roll over, kept in the variable roll count (rollcnt), the engine speed, N_e , can be determined from.

$$N_e = 2.048 \times \text{rollcnt} + \text{TMR0 value, when the SENSOR pulse rises.} \quad (15)$$

The software doesn't actually calculate speed. Instead it uses the rollcnt as an index into look-up tables. These tables provide the time from TDC, from when the SENSOR pulse rises, to the next value. This provides good timing as long as the engine speed is constant.

The main control loop shown in Figure 8 consists of a series of sequentially executed routines. The normal sequence is to open the exhaust valve, detect Bottom Dead Centre (BDC), do a speed calculation/table look-up, close the exhaust valve, open the intake valve, detect TDC, do a speed calculation/look-up, and close the intake valve. Each of these events occurs at a specific time. The software again used TMR0 to measure the time since TDC and to determine when this time catches the appropriate speed dependent times from the look-up tables.

The exhaust valve will be opened when rollcnt reaches the exhaust valve open count (EVOCnt) and the TMR0 reaches the EVOTMR0 value. These two values come from the table look-ups in the speed calculation routine.

The exhaust valve open sequence checks for EVOCnt or TMR0 roll over and increments the variable rollcnt if one has occurred. It then saves the current value of TMR0 so that it can do the roll over check next time this first loop is executed. The software then checks to see if the rollcnt matches the exhaust valve open count (EVOCnt). If it doesn't the software continues looping and checking for TMR0 roll over until the match occurs. The software then enters a second loop where it waits until TMR0 reaches or exceeds the exhaust valve open TMR0 value (EVOTMR0). The time for the end of the 2.5 millisecond valve pulse is calculated and the software loops in the next sequence waiting for this time to be reached.

In the personal computer method electrical signals are transferred to the computer in the form of binary numbers. Contents of the input port of the digital card are thus brought into the computer memory. The software used to show the valve timing and valve displacement with crank angle degree manipulates input data.

The computer software consists of two measurement programs. First program is to measure the response of the model engine system (start of valve opening, duration, closing, and valve lift with the crank angle degree). When that program started the software initialize to measure the engine speed by measuring the time each revolution of the crankshaft takes.

The valve open, close and lift are determined by using the analog signals (volt signal) from the linear position sensor and convert it to

a position in (mm). The software is adjusted to operate for a specific period of time namely one second. At the end of each second it sends the results to the Microsoft excel file. After that the relation between the crank angle degrees and valve position can be monitored.

The Lab View computer software, as shown in Figure 9, is used for controlling the model engine valve timing. The software starts if it detects the engine TDC; hence it starts the main control loop. The main control loop for computer control consists of series of sequentially executed routines. The normal sequence is to detect TDC, open and close each valve at crank positions determined by the speed calculation routine and look-up table values. Each of these events occurs at a specific time.

The intake or exhaust valve will open or close when the encoder sensor approach the proposed crank angle as given in the look-up table corresponding to the speed calculated by the speed calculation routine.

It is worth noting that the engine is put under control of the microcontroller or computer software so that the engine is always maintained very close to a certain desired valve timing corresponding to the best volumetric efficiency regardless of speed changes. Continuous sampling of engine speed and continuous correction of the valve timing do this.

The control loop of the microcontroller or computer software is mainly composed of input instructions; digital output instructions

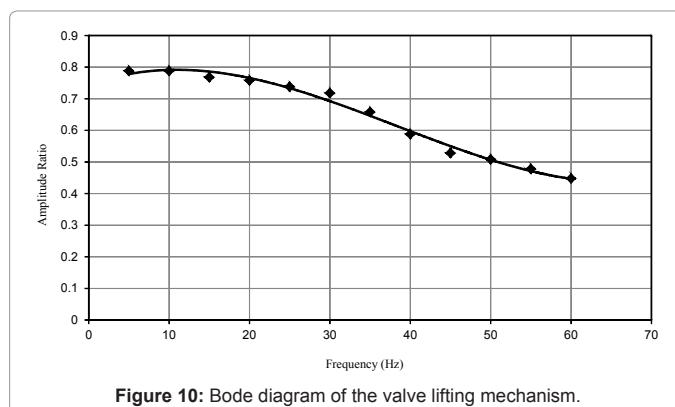


Figure 10: Bode diagram of the valve lifting mechanism.

Engine speed rpm	I.V.O B.T.D.C	I.V.C A.B.T.C	Duration
800 to 1000	10°	30°	220°
1500 to 1700	50°	70°	300°
2000 to 2150	80°	90°	350°

B.T.D.C: before top dead center

A.B.T.C: after bottom dead center

Table 2: sample of valve timing and duration.

I.V.O, I.V.C: intake valve opening and closing respectively

look-up table and control instructions. The input instructions are sent by speed measurement port to transfer the measured speed value to the microcontroller or computer memory. The microcontroller or computer generates digital output instructions. For the required pulse width for the solenoids, the output signals are transferred to both solenoid actuators.

The new concept for self-regulation of optimum valve timing corresponding to the best volumetric efficiency was carried out. The new concept is done by the way of creating a look-up table of the optimum valve timing against speed. A look-up table is a list of related values stored in the memory unit of a microcontroller or computer; this table relates the output solenoid signal settings given by the microcontroller or computer to the input signals received from a given sensors.

Results and Discussions

From the bode diagram shown in Figure10 bandwidth of the solenoid can be determined as follow: Bandwidth = frequency at 0.7 of the maximum amplitude ratio. So the Bandwidth is equal 30 HZ, where bandwidth is the solenoid operating range of frequency. This result means that the solenoid actuator can be operated carefully from 30 Hz frequencies, i.e. around 4000 rpm. Then it will be operated from 4000 to 6000 rpm with less efficiency (less valve lift).

The theoretical study showed that the described theoretical method of volumetric efficiency and torque is a method to forecast the values of volumetric efficiency η_v and torque T at different valve timings. Therefore, a better valve timing can be predicated at each engine speed. Table 2 shows the sample of recorded valve timings and duration that fed to the microcontroller and computer-based controls. It indicates that while the engine speed increased the duration of valve opening must be increased (that is not offered by conventional cam engines).

Figures 11-13 show the variation of timing and duration of valve at different speeds with microcontroller control, Figures 14-16 show that variation when using the computer-aided control. These figures show an accurate achievement of the look-up table values of the optimum

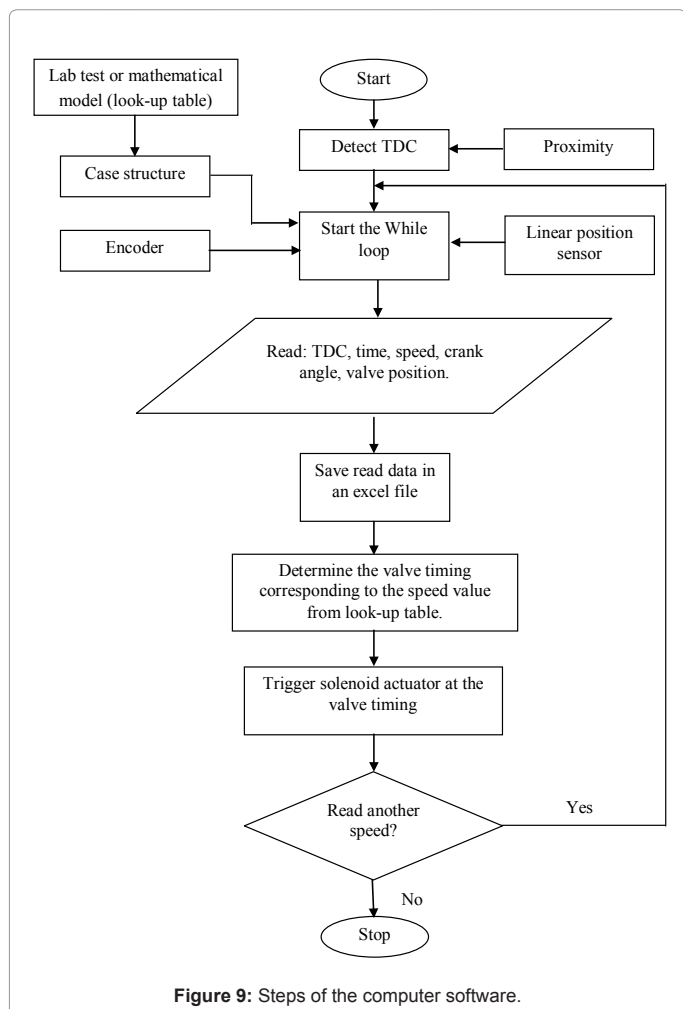


Figure 9: Steps of the computer software.

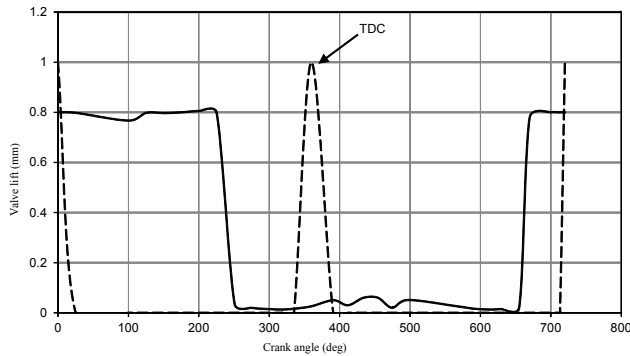


Figure 11: Engine valve timing response with crank angle at 825 rpm.

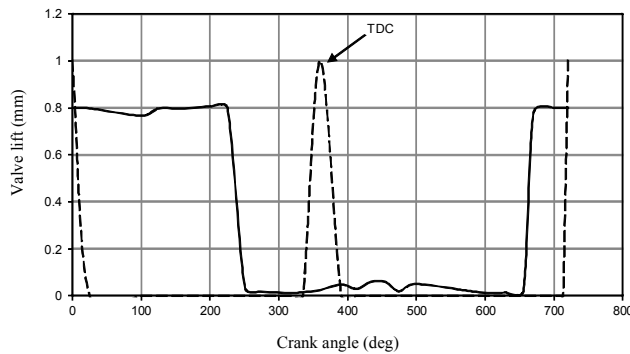


Figure 12: Engine valve timing response with crank angle at 1636 rpm.

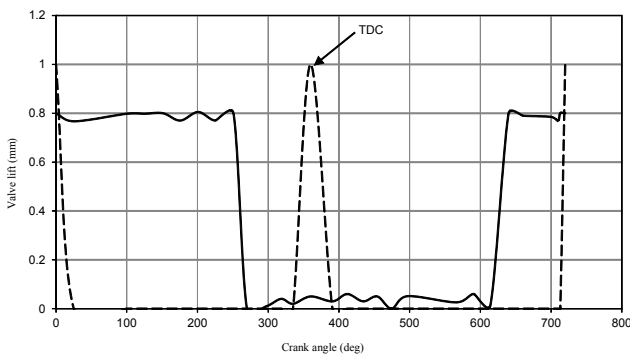


Figure 13: Engine valve timing response with crank angle at 2090 rpm.

valve timing covering completely the operation speed ranges. The slope of valve lift to the crank angle decreases as the engine speed and, consequently, the valve opening duration increases. A steep decay of the valve closing velocity is noticed and the rate increases at higher speeds. Figures 17-19 show a proposed and measuring valve response at various engine speeds.

It is possible to notice that the electromagnetic control system is extremely sensitive to its microcontroller and computer-aided control program which alters the required optimum valve timings, and this will help to obtain an accurate valve timing for any type of engine by calculating the optimum valve timing to achieve the corresponding improved engine torque values at different engine speeds [23-26].

The effect of the electromagnetic control system on the valve opening, duration and closing is as follows:

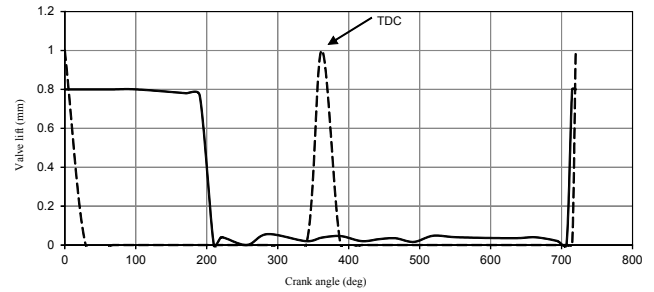


Figure 14: Engine valve timing response with crank angle at 1000 rpm.

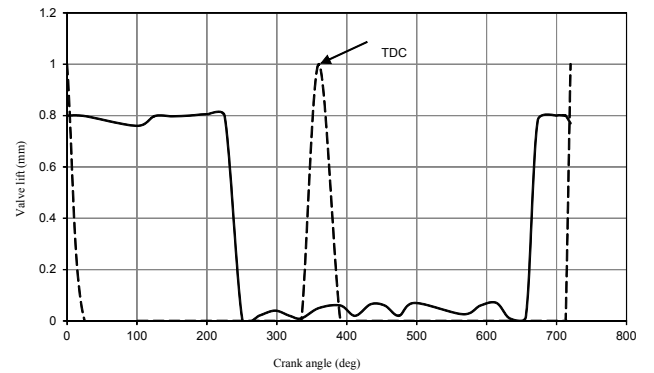


Figure 15: Engine valve timing response with crank angle at 1680 rpm.

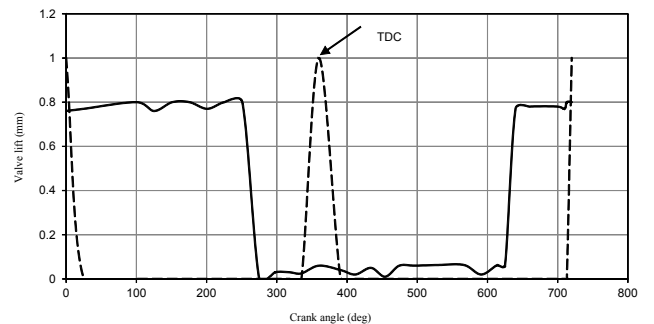


Figure 16: Engine valve timing response with crank angle at 2140 rpm.

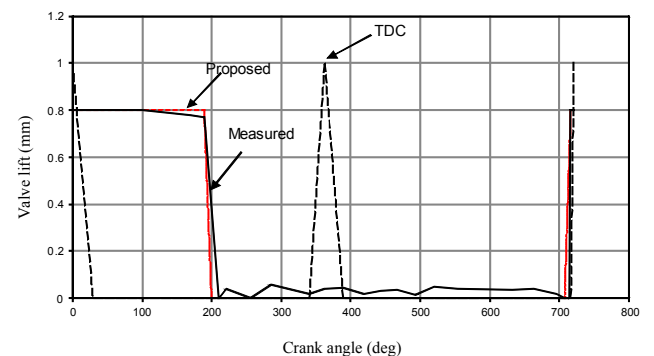


Figure 17: Proposed and measuring valve response at 1350 rpm.

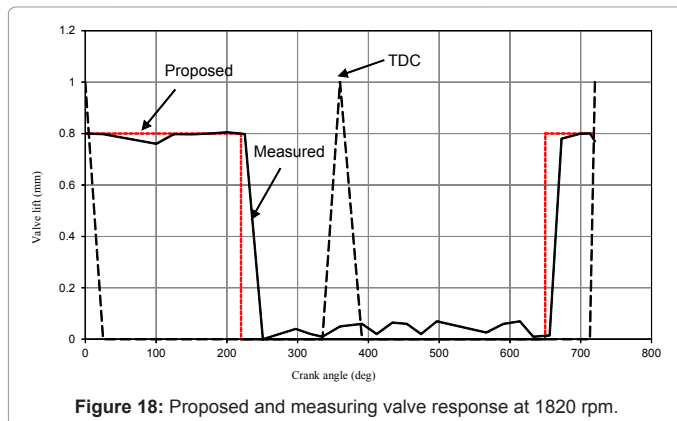


Figure 18: Proposed and measuring valve response at 1820 rpm.

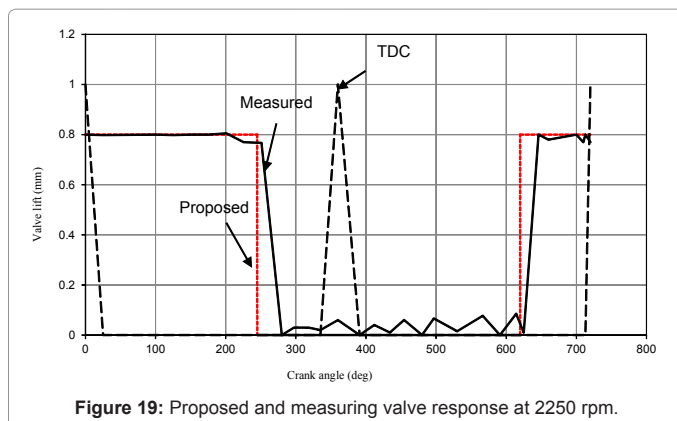


Figure 19: Proposed and measuring valve response at 2250 rpm.

At valve lift

When the induced coil is driven with a constant current, the steep decay of the velocity shows the hard impact with about 0.8 m/s, which causes substantial acoustical noise and leads to a deterioration of the lifetime of the actuator. Consequently, a control system should fulfill the following requirements:

- The landing speed of the armature should be in the range of 0.1 m/s.
- The system must be robust against variations of cylinder pressure and variation of the supply voltage

At valve opening

In this case, the system is very stable at all duration of valve opening. The full valve lifting at most of duration that cause an improvement in volumetric efficiency related to increase the average valve lift and to reduce the average gas velocities. Therefore, the flow resistance in the intake and exhaust ports is reduced.

At valve falling (or closing)

In this case, when the volt drops and the induced coil is released, the stored energy in the coil spring will be relieved and a steep decay of the velocity shows the hard impact with about 0.6 m/s and resonance of spring will occur at high speeds when the valve closes. Consequently, a damper system must be used to avoid this problem.

Conclusions

A continuously electromagnetic variable valve timing system

is developed and implemented to meet the requirements of a higher volumetric efficiency and torque under all ranges of engine running conditions. The major conclusions are as follows:

1. An electromagnetic camless valve train is developed for a camless model engine. This development confirmed its functional ability to control the valve timing and its duration.
2. The proposed system demonstrated a full flexibility and quick response in the variable timing and duration.
3. By applying the new valve timing of intake valve at specific speed an improvement of the volumetric efficiency is achieved. This improvement is obtained over the whole range of engine speed by changing the suitable valve timing at any speed.
4. New concept for self-regulation of optimum valve timing is applied. This concept is done by the way of creating a look-up table of the optimum valve timing against speed.
5. The proposed electromagnetic variable valve timing system has the following advantages:
 - Reduced friction losses by minimized a number of additional components such as the camshaft, the rocker arms, and the push rod.
 - Maximum valve lifts most of the period of valve opening
6. The results show a good agreement between that of microcontroller system technique and that of computer-aided control technique.

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