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VARIABLE CAM TIMING: CONSEQUENCES TO AUTOMOTIVE ENGINE CONTROL DESIGN

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Abstract: One objective of this paper is to illuminate fuel economy and emission improvement mechanisms of variable cam timing systems and analyze their effects on engine control system design. By retarding or advancing the cam phase one can vary the engine volumetric efficiency, as well as the amount of exhaust gas that dilutes the air charge. Combining these effects with intake manifold and engine speed dynamics leads to a complex behavior of engine air-charge and torque that requires special handling by the engine control system. This paper reviews control algorithms for VCT engines that have been reported in the literature.

Keywords: Engine modeling and control, variable valve timing

1. INTRODUCTION

Variable valve timing is used in spark ignition automotive engines to improve fuel economy, reduce emissions, and increase peak torque and power (Duckworth and Baker, 1996; Leone et al., 1996; Stein et al., 1995). We shall consider only the variable cam phasing systems as opposed to other VVT systems such as cam profile switching (Matsuki et al., 1996), variable intake/exhaust duration (Chattopadhay, 1993), variable valve lift (Flierl and Kluting, 2000; Pierik et al., 2000), and camless (electromagnetic valve drive) engine systems (Ashab et al., 1998).

In conventional (non-VCT) engines, relative phase between the camshafts and the crankshaft is fixed at a value which represents a compromise between conflicting requirements at different operating conditions. A VCT mechanism (see (Steinberg et al., 1998) for a mechanical design of the actuator) varies the phase of the valve opening and closing relative to the crankshaft as a function of engine operating conditions. Depending on the camshaft (exhaust, intake, or both) being actuated, there are four types of variable cam timing systems: intake-only, exhaust-only, where only intake or exhaust valve timing is varied, dual-equal, where intake and exhaust timing is varied equally, and dual-independent, where the intake and exhaust timings are varied independently (Leone et al., 1996).

The main topic of this paper is to describe the effects of variable cam timing on engine operation and their consequences on engine control system design. Primary VCT effects, analyzed in detail in Section 2, can be summarized as follows: (i) retarding the exhaust cam timing increases exhaust gas residual reducing NO_x emissions and pumping losses at part load; (ii) retarding intake cam timing reduces volumetric efficiency, particularly at low engine speeds, thus reducing pumping losses; (iii) advancing intake cam timing up to a point, increases volumetric efficiency and peak engine torque at low and medium speeds; (iv) higher intake advances can also be used to recirculate exhaust gas to reduce NO_x emissions and pumping losses.

In this paper we propose a generic model of a VCT engine and review control algorithms reported in the literature. These include air-fuel ratio regulation, air-charge estimation for VCT systems with EGR backflow, and transient torque regulation.

The paper is organized as follows. Section 2 reviews the VCT effects on engine operation. Section 3 presents a model of a generic VCT engine. Section 4 reviews some of the results on VCT engine control.

2. EFFECTS OF VCT ON ENGINE OPERATION

The timing (or phase) of opening and closing of the intake and exhaust valves is determined by the valve lift profiles shown in Figure 1. In a conventional engine

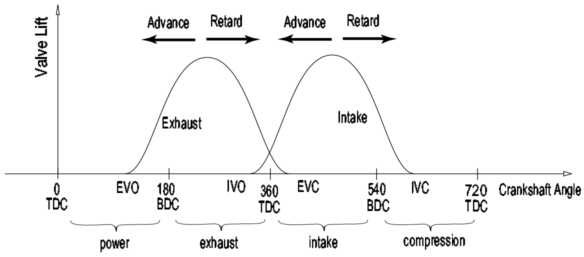

Fig. 1. Profiles of intake and exhaust valve lifts versus crank angle.

this timing is fixed at a value that represent the best compromise between conflicting requirements for idle speed quality, fuel-economy, low-speed torque, and power.

Typically, the exhaust valve opening (EVO) occurs before the end of the power stroke. This allows earlier release of the hot exhaust gas resulting in reduced pumping losses during the subsequent exhaust stroke. The exhaust valve closes just after the cylinder reaches the top dead center (TDC) at the end of the exhaust stroke. The small angle (5 - 15 degrees) the EVC trails the TDC allows the inertia of the escaping exhaust gas to empty the cylinder beyond what would otherwise be achieved (c.f. (Heisler, 1995)). Closing it either earlier or later would increase the amount of exhaust gas retained in the cylinder, in particular at lower engine speeds.

In conventional engines the intake valve opens just before the beginning of the intake stroke (see the IVO event in Figure 1). The closing of the intake valve occurs well into the compression stroke to exploit the inertia of the gas that, at high engine speed, still fills the cylinder even after the piston has started the compression stroke. The IVC value of about 50 to 60 degrees after bottom dead center (BDC) provides good engine pumping (volumetric efficiency) at high engine speeds.

By varying the cam timing with operating conditions, some of the design tradeoffs can be avoided. Depending on the cam-shaft actuated, we can retard and/or advance the intake, exhaust, or both cams. Advancing the cam timing results in earlier (in crankshaft degrees) opening and closings of the valves while retarding results in later opening and closing (see Figure 1). Note that cam phasing does not change the duration of the interval the valves are open.

2.1 *Exhaust VCT*

By retarding the EVO more exhaust gas is retained in the cylinder for combustion. The additional dilution reduces NO_x emissions and pumping losses by displacing fresh air and increasing intake manifold pressure required for a given air-charge and engine torque. The plot in Figure 2 shows the experimentally obtained relationship between the mass air flow and intake manifold pressure when the exhaust cam timing is retarded. This relationship is well approximated by straight lines described by slope and offset coefficients that depend on engine speed. It appears that only the offset coefficient depends on exhaust cam timing.

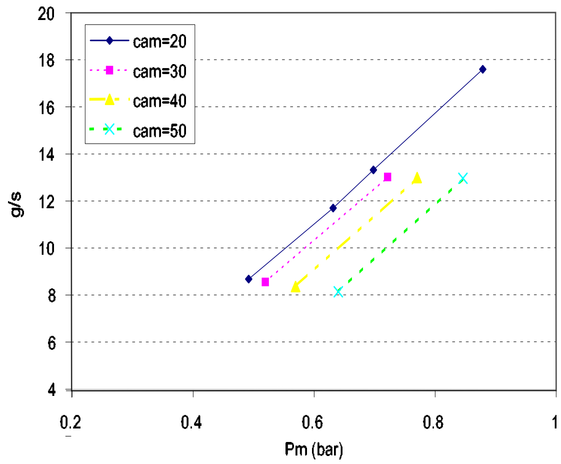

Fig. 2. Effect of exhaust cam retard on engine pumping at 1500 RPM.

The exhaust gas displaces fresh air and can be tolerated only at part load operating conditions. At a low load, the exhaust gas negatively affects combustion stability, while at high loads it reduces the amount of fresh air and thus torque production. Therefore, a typical schedule of the exhaust cam timing is the following: base timing (EVC about 10 degrees after TDC) at low load, retarded at part load, and base (or slightly retarded) at high load.

2.2 *Intake VCT*

Changes in intake cam timing have even more profound effect on engine air intake. In a typical intake VCT system, the IVO is advanced into the exhaust stroke, which means that IVC moves closer to the BDC of the intake stroke. For small intake cam advances the main effect is the increase of air charge (at a given manifold pressure) and, subsequently, increase in maximal torque at low to medium engine speeds. For an explanation of this effect we refer to Figure 3 that shows the effective cylinder volume (trapped volume) as a function of IVC. At low engine speeds, the mass of trapped air is proportional to the gas volume at the crankshaft angle the intake valve closes. As we mentioned before, the nominal intake valve closing, denoted by IVC_0, is about 50 to 60 degrees after BDC (during the compression stroke). Advancing the cam timing from IVC_0 to IVC_{adv} increases the effective cylinder volume from V_0 to V_{adv} increasing the engine air-intake capabilities at low to medium speeds. At high engine speeds, good engine air-intake is achieved by advancing the timing to about IVC_0 to exploit the ramming effect due to air inertia.

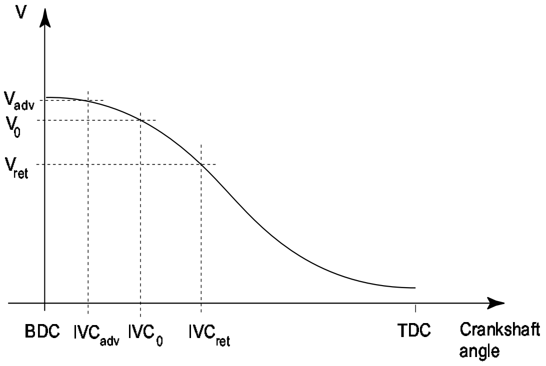

Fig. 3. Volume of trapped gas as a function of IVC.

At the same time, advancing the intake cam timing advances IVO into the exhaust stroke. As the IVO advances, more of the exhaust gas is allowed to enter the intake manifold as a backflow through the intake valve as shown in Figure 4. This effect can be used to provide an exhaust gas recirculation (EGR) mechanism reducing the NO_x emissions. At a low manifold

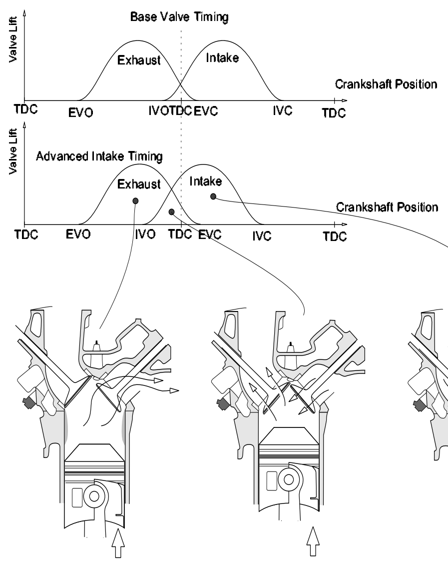

Fig. 4. EGR is provided by advancing the intake valve timing, resulting in a backflow of the exhaust gas into the intake manifold.

pressure, increase in EGR backflow dominates and the cylinder mass airflow decreases with cam advance. At high pressure, early IVC has the dominant effect and the air charge increases with cam advance. Both effects are clearly visible in Figure 5 which shows the engine cylinder mass air flow versus manifold pressure for an intake-only VCT engine. At low to part load, intake advance provides EGR. The partial pressure of EGR in the cylinder can be estimated from the offset coefficient (equal to the manifold total pressure at which the air-flow is 0). On the other side, by extending the constant-cam lines one can conclude that the cam advance of about 30 degrees provides the best air-flow at wide open throttle conditions (manifold pressure of 1 bar). Because the combustion stability does not allow EGR at low loads, a typical intake timing schedule would be

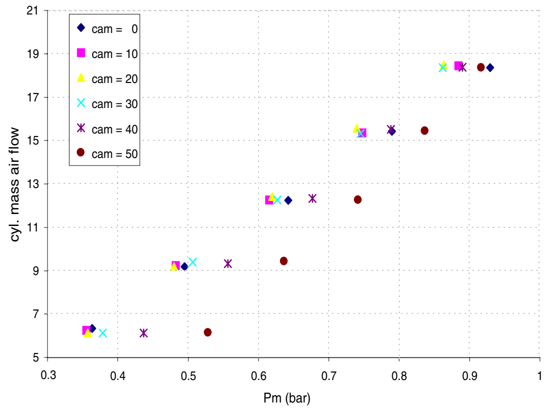

Fig. 5. Cylinder mass air flow versus intake manifold pressure and intake cam advance at 1000 RPM.

base timing at low load, fully advanced timing at part load and partially advanced timing at high load.

Another possibility to improve fuel economy is to retard the intake cam timing to reduce the effective volume (see Figure 3) and the engine air-intake at a given manifold pressure. That is, for a given cylinder air flow, retarding the intake cam timing results in an increase in manifold pressure and, subsequently, a reduction in pumping losses. The data that show the effect of intake cam retard come from a dual-independent VCT system in which both intake and exhaust cam are retarded equally. Thus, the relationship between cylinder mass air flow and manifold pressure is characterized by reduced slopes due to late IVC's and increased offsets due to late EVC's as the timing is retarded. Figure 6 shows this experimentally measured relationship at $N = 1500$ RPM and different values of cam phase retard.

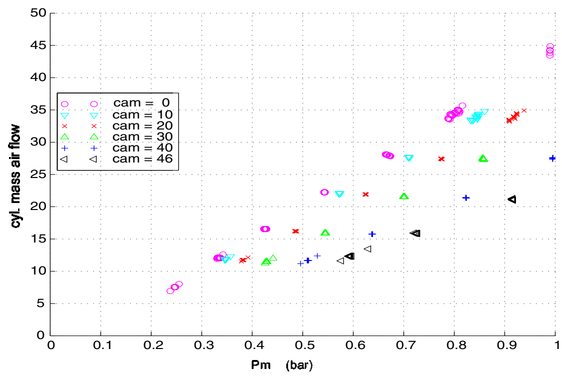

Fig. 6. Cylinder mass air flow remains an affine function of the manifold pressure at different values of dual-equal cam timing.

3. MODEL OF A VCT ENGINE

For a dual-equal VCT engine a simple model has been derived in Stefanopoulou et al. (1998) taking a mean-value model of the intake manifold from Powell and Cook (1987) as the starting point. We further refine this model by introducing separate effects of the intake and exhaust cam timing and taking into account the backflow of the exhaust gas in the case of intake cam advance. The dynamics governing the intake manifold pressure change is obtained by differentiating the ideal gas law $PV = mRT$:

$$\begin{aligned}\dot{P}_m &= \frac{RT_m}{V_m}\dot{m} + \frac{mR}{V_m}\dot{T}_m \\ &= \frac{RT_m}{V_m}(W_\theta + W_{egr}^{in} - W_{cyl} - W_{egr}^{out}) + \frac{\dot{T}_m}{T_m}P_m\end{aligned} \tag{3.1}$$

where P_m, T_m, and V_m are the intake manifold (total) pressure, temperature, and volume, R is the specific gas constant, W_θ and W_{cyl} are the throttle and cylinder mass flow rates, and $W_{egr}^{in/out}$ are the EGR backflows in and out of the intake manifold (due to intake advance). From the discussion in the previous section we obtain

$$\begin{aligned}W_{cyl} &= \alpha_1(N, \zeta_{int})P_m + \alpha_2(N, \zeta_{int}, \zeta_{exh}) \\ W_{egr}^{in/out} &= F^{in/out}(N, P_m, \zeta_{int}, \zeta_{exh})\end{aligned} \tag{3.2}$$

The intake and exhaust cam timing variables are denoted by ζ_{int} and ζ_{exh}. Their dynamic behavior is well approximated by a rate limited first order lags:

$$\begin{aligned}\dot{\zeta}_{int} &= sat_{adv}^{ret}(-a\zeta_{int} + a\zeta_{int}^{com}) \\ \dot{\zeta}_{exh} &= sat_{adv}^{ret}(-a\zeta_{exh} + a\zeta_{exh}^{com})\end{aligned} \tag{3.3}$$

where $\zeta_{int/exh}^{com}$ are the commanded values for the intake and exhaust cam timings and adv and ret are the rate limits on cam advance and retard (retard being denoted as positive direction):

$$sat_{adv}^{ret}(x) = \begin{cases} x & \text{if } adv \le x \le ret \\ adv & \text{if } x < adv \\ ret & \text{if } ret < x \end{cases}$$

The advance and retard rate limits assume typical values of 100 to 200 degrees per second that depend on hardware configuration, oil temperature, and engine speed (Moriya et al., 1996; Steinberg et al., 1998). The rate limits need not be the same for intake and exhaust cams.

The flow through the throttle valve can be obtained by using the orifice flow equation:

$$W_\theta = C_\theta(\theta)\frac{P_a}{\sqrt{RT_a}}\,\Psi\left(\frac{P_m}{P_a}\right) \tag{3.4}$$

where C_θ is the effective flow area of the throttle, and θ is the throttle angle. The pressure ratio correction factor Ψ is given by

$$\Psi(r_p) = \begin{cases} \gamma^{\frac{1}{2}}\left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} & \text{if } r_p \le \left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \\ \sqrt{\dfrac{2\gamma\left(r_p^{\frac{2}{\gamma}} - r_p^{\frac{\gamma+1}{\gamma}}\right)}{\gamma-1}} & \text{if } r_p > \left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \end{cases}$$

The constant $\gamma = 1.4$ is the ratio of specific heats for air, c_p/c_v, and r_p is the ratio of the pressures downstream and upstream of the orifice.

4. CONTROL DESIGN ISSUES

It is clear that having a variable cam timing introduces a significant change in engine operation that requires redesign of some engine control system components and/or introduction of new ones. In general burn rate depends on the dilution (percent EGR in the mixture) and gas turbulence, both of which are affected by cam timing. Thus, spark schedules must take into account the cam timing in order to provide best fuel economy and prevent possible misfire. VCT also affects the air-charge and torque response of the engine and, subsequently, vehicle drivability. The effects of VCT on air-charge estimation and torque/drivability are discussed next.

4.1 *Air-charge estimation, fuel control*

Most vehicles come equipped with three way catalysts that achieve high conversion efficiencies of hydrocarbons, carbon monoxide, and oxides of nitrogen only when engine operates at the stoichiometric air-fuel ratio (approximately equal to 14.6). Therefore, one of the key control objectives of the engine control system is to maintain the air-fuel ratio at stoichiometry by appropriately regulating the fuel injection pulse widths. Because the variable cam timing changes significantly the engine breathing dynamics, for air-charge estimation, engine volumetric efficiency coefficients (α_1 and α_2) have to dependent on cam timing, in addition to the conventional dependence on engine speed. Other modifications proposed to improve the air-fuel regulation have been reported in the literature (Jankovic et al., 2001; Stefanopoulou et al., 2000).

For dual-equal VCT engines, several different cam-timing/fuel control configurations were analyzed in (Stefanopoulou et al., 2000) including the decentralized architecture (two independent PI loops) and full MIMO architecture. The comparison of the configurations has shown that the MIMO controller achieves an improvement over the decentralized one. The cross coupling term from the cam error to the fuel injector pulse-width command is mainly responsible for the improvement and the authors have argued that it should be retained to achieve the best possible performance.

The effects of EGR backflow on air-charge estimation have been considered in (Jankovic et al., 2001). The presence of EGR backflow raises the intake manifold pressure (see equation (3.1) and complicates the air-flow to manifold pressure relationship. This backflow EGR is difficult to measure because it does not mix completely with intake manifold air. If a manifold air pressure (MAP) sensor is available, the standard "speed-density" air-charge estimation method applies assuming that the slope and offset coefficients are adjusted for cam timing. On the other hand, with the mass air flow (MAF) sensor, the EGR transients caused by cam timing negatively affect the estimation accuracy. The authors have proposed a method to determine new slope and offset coefficients that connect the cylinder mass air flow and partial pressure of air (instead of the total pressure) in the intake manifold. It has been shown that the coefficients can be identified only from transient data and two identification methods have been proposed.

4.2 *VCT transient effects*

Even if the spark timing and air-fuel ratio are regulated accurately, the effect of cam timing on engine air-intake may cause undesirable transient torque variation and drivability problems. The situation is depicted in Figure 7 in the context of dual-equal VCT. In steady state, the engine operates at the intersection of the throttle characteristic (dotted curves) determined by the throttle opening and the engine pumping characteristic determined by engine speed and cam timing (straight lines in Figure 7). If we retard the cam timing from

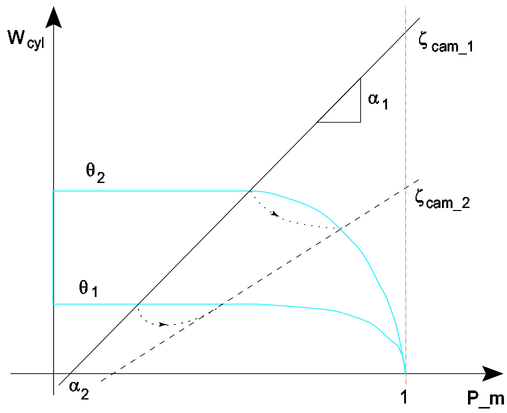

Fig. 7. Changing the cam timing at fixed throttle causes a transient change in cylinder air-charge.

ζ_{cam_1} to ζ_{cam_2}, but keep the throttle position fixed at θ_1, the beginning and final mass air flow rates are the same, but there is drop in air flow during the transition. If the cam moves instantaneously, before the manifold pressure can change, the mass air flow would fall from the amount determined by the ζ_{cam_1} line to ζ_{cam_2} line instantaneously, and then transition along the ζ_{cam_2} line to the new steady state point as the manifold pressure increases. In real engines, the cam moves with finite speed and the transient air-charge drop is less pronounced. At higher throttle openings, the final steady state also changes with cam timing (intersection of θ_2 curve with ζ_{cam_2} line. The transient effects of cam timing are clearly visible in (green-dash) traces shown in Figure 8 obtained experimentally in a dynamometer test cell. The throttle angle is held constant while the engine speed is almost constant, so the torque drop and flare are solely due to cam timing changes.

To reduce or remove the effects of cam timing on torque response and drivability, a feedforward compensation method has been proposed in (Jankovic and Frischmuth, 1997; Jankovic et al., 1998). The idea is to treat the cam timing as a known disturbance, and use the electronic throttle (or air-bypass valve) to cancel the effect of the disturbance on air-charge. Because the disturbance (cam timing) is closer to the performance output (air-charge) than the control input (electronic throttle), the control law is characterized by its use of the rate of change of the measured cam timing signal. An advantage of this approach is that it does not require additional sensors for implementation. On the other hand, because it is completely feedforward, it requires relatively accurate engine volumetric efficiency slope and offset coefficients as a function of cam timing. The compensation is implemented as an additive term θ^* to the throttle position due to the driver's request θ_0:

$$\theta_{com} = \theta_0 + \theta^*. \tag{4.1}$$

The design objective is to find a control law for θ^* such that the rate of change of W_{cyl} coincides with that of the conventional engine denoted by W_{cyl}^0 that can be generated by a reference model. The control law that accomplishes this has been derived in (Jankovic and Frischmuth, 1997):

$$\theta^* = -\theta_0 + C_\theta^{-1}\left(\frac{\Psi\left(\frac{P_m^0}{P_a}\right)C_\theta(\theta_0)}{\Psi\left(\frac{P_m}{P_a}\right)} - \frac{\left(P_m\frac{\partial\alpha_1}{\partial\zeta_{cam}} + \frac{\partial\alpha_2}{\partial\zeta_{cam}}\right)\dot{\zeta}_{cam}}{K_m\Psi\left(\frac{P_m}{P_a}\right)\alpha_1}\right)$$

where $P_m^0 = \frac{1}{\alpha_1(N,0)}(W_{cyl}^0 - \alpha_2(N,0))$ and, instead of the measured P_m signal, a feedforward estimate $\hat{P}_m = \frac{1}{\alpha_1(N,\zeta_{cam})}(W_{cyl}^0 - \alpha_2(N,\zeta_{cam}))$ has been employed. The performance of the compensator has been tested experimentally. Traces of the engine response with and without the compensator are shown in Figure 8. Note that the perfect rejection is achieved if the torque does not respond to the cam disturbance. The second plot from the top shows the actuation due to θ^* which is equal to the difference between the solid and dash curves. More details about the experimental set-up and the results can be found in (Jankovic et al., 2000).

Another approach to improve the transient torque response has been pursued in (Hsieh, et al.). A MIMO feedback controller has been designed to regulate engine torque, cam timing, and air-fuel ratio. The controller requires and in-line (crankshaft) torque sensor for implementation. This control design has also been tested experimentally. The details of the experimental configuration and the performance achieved can be found in (Hsieh and Koncsol, 2000).

5. CONCLUSION

Variable cam timing systems, used in modern automotive engines to improve fuel economy, emissions,

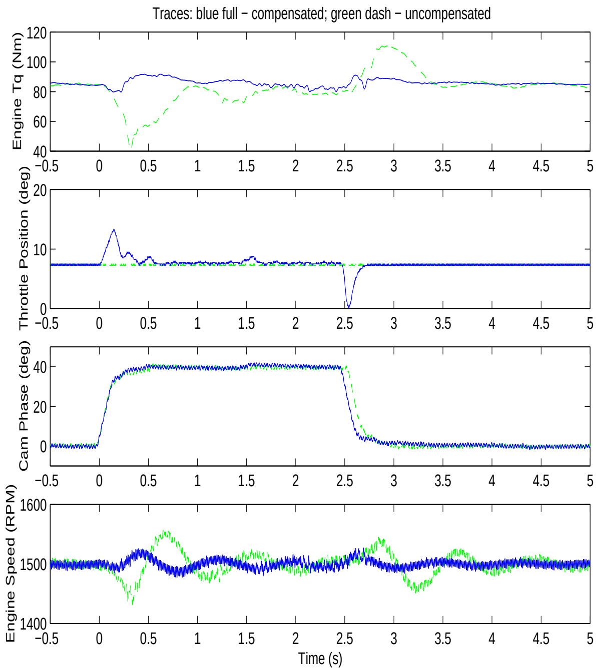

Fig. 8. The response of the VCT engine to cam phase changes with (solid line) and without (dash line) the compensator.

torque, and power, present a challenging problem to engine control designers. In this paper we have analyzed different VCT systems and their effects on engine air intake, charge dilution with the exhaust gas, and the torque production. We have proposed a model of a VCT engine and presented a review of existing results on control design.

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