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Control of engines with fully variable valvetrains

Mrdjan Jankovic and Stephen Magner

Abstract**—**To improve performance, fuel economy, and emissions of automotive engines, automakers have introduced a number of devices intended to vary a previously fixed engine parameter with operating condition. Such devices include variable cam timing and valve lift, variable compression ratio, intake manifold tuning, etc. Combining these devices in an engine promises improved fuel economy, but raises the issue of scheduling and coordination to achieve the benefit and prevent undesirable failure modes. This paper introduces the problem of controlling a fully variable valvetrain configuration, and shows an algorithm for synchronizing variables that have different rates of response with the goal of preventing potential piston-valve interference in mechanically unprotected engines.

I. Introduction

A number of engine parameters, including valve profiles, valve timing, compression ratio, intake manifold geometry, etc, used to be fixed at the design stage. The parameter values selected represented the best compromise between conflicting requirements at different operating conditions. For example, to achieve peak engine power it is beneficial to close the intake valve well into the compression stroke to take advantage of the ramming effect of high speed air flowing into the cylinder. On the other hand, at low engine speeds late closing of the intake valve allows fuel and air to be pushed back into the intake port thus reducing low speed torque.

In modern engines at least some of these parameters are allowed to vary with operating conditions by implementing hydraulic or electrical actuators and a control system that manages the positioning of each actuator and the interactions between them. We assume that the optimal combinations are known, obtained from engine mapping and optimization procedure. Optimizing an engine with many degrees of freedom is a challenging task [1], but is not a topic of this paper. Here we would like to address some issues related to coordination of the actuators during transients. In general, the actuators respond with different rates. If there is a region in the parameter space that results in undesirable operation (such as poor combustion) or may lead to a failure mode, it is essential to have the control system coordinate the response of the actuators to maintain their transients in the desirable operating region. Such a control system will be illustrated on an engine with variable valve timing, variable valve lift, and a high compression ratio.

In the first part of the paper we shall discuss the advantages of a fully variable valvetrain in optimizing fuel consumption of an engine [2,3,4]. By combining variable cam timing (VCT) and continuously variable valve lift (CVVL), the control system can adjust valve opening profile to get the right amount of air, fuel, and residual dilution, while minimizing pumping losses. Generally speaking, engines with a variable valve lift mechanism can run unthrottled, or almost unthrottled. For other reasons, including noise suppression, and fuel evaporation during cold starts, an electronically controlled throttle (ETC) may still be needed.

In the second part of the paper we shall discuss a specific control issue related to fully variable valve lift engine operation. Achieving the optimal fuel consumption requires that the valve lift and intake cam timing are carefully synchronized. However, the two actuators, in general, respond with different speeds. In this paper we propose a simple controller that prevents the valve-lift, cam-timing combination from entering a part of the operating region that may result in piston-valve interference. The operation of the system is illustrated by simulations.

II. Continuously variable valve lift

In fixed valvetrain engines, the timing of opening and closing of intake and exhaust valves is determined to provide a stable and reliable engine combustion under most demanding conditions (such as cold temperatures, low loads). Another set of considerations governs the selection of intake valve closing as discussed in the introduction.

If the intake or exhaust cam timing can be varied with operating conditions, typically engine speed, load (torque),

Manuscript received September 10, 2004.
S. W. Magner, Ford Research and Advanced Engineering, Dearborn MI, 48121, phone 313-323-1057, fax: 313-845-0962, smagner@ford.com
M. Jankovic, Ford Research and Advanced Engineering, Dearborn MI, 48121

ambient and/or coolant temperature, some of the restrictions of the fixed timing system can be overcome. By opening intake valves earlier (during the exhaust stroke) or closing exhaust valves later (during the intake stroke), more burned gas is retained in the cylinder for combustion resulting in improved combustion efficiency and lower NOx emissions. A fuel economy benefit is also realized if the intake valve is closed later (during compression stroke). Late closing allows a portion of the cylinder gas to escape back into the intake port, lowering the amount of air (and fuel) that is left for combustion. This means that for a given amount of air-fuel mixture in the cylinder, retarded (late) cam timing has to operate at higher manifold pressure, which results in lower pumping losses. One constraint in VCT design and operation is that the range of VCT is limited because opening an intake valve too early, or closing an exhaust valve too late, may lead to physical interference between the valve and the piston. The duration of intake and exhaust event remains fixed. Another constraint is that the valve lift profile and, hence, duration remain the same. This means that if we want to achieve late closing of the intake valve and increase manifold pressure for a given air-charge, we are also getting late opening of the intake valve potentially resulting in a pumping penalty as both valves remain closed during the initial part of the intake stroke. Some details of the operation of engines with variable cam timing can be found in [5].

Introduction of the variable valve lift actuator removes the constraint on event duration. Combined with VCT, variable valve lift allows the control system to independently control valve opening and closing times. The main objective of a CVVL mechanism is to control the air-charge with intake event duration, or equivalently, with intake valve closing time. In this way, the intake manifold pressure may remain at or close to ambient, potentially eliminating pumping losses. A P-V diagram of the typical cycle of a CVVL equipped engine is shown in Fig. 1.

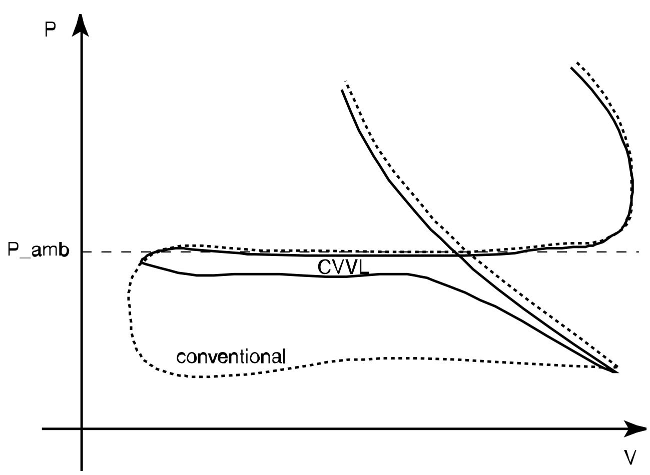

Fig 1: Pumping loop comparison of CVVL and conventional engines.

The reduction in pumping is proportional to the difference in areas on the P-V diagram enclosed by each curve. The reduction in pumping for the CVVL engine is achieved by closing the intake valve earlier (before the end of the intake stroke), while keeping the valve opening approximately the same as in the conventional engine. After the intake valve closes, the gas (air, fuel, and residual mixture) is expanded and compressed resulting in a little work done and lower pumping losses (Fig. 1 shows a small loss due to heat absorption by the gas). More details on different modes of operation and various tradeoffs in CVVL engine design can be found in [4].

In one implementation of the CVVL mechanism, changing valve lift does not change the centerline of the valve profile [2]. The intake valve opening profile versus crank angle for such a mechanism is shown in Fig. 2. However, the low lift profile would not achieve pumping improvement, unless the intake valve opening/closing time is shifted earlier (advanced). The reason is that the first part of the intake stroke would happened at closed valves, creating cylinder vacuum, even deeper than that of the conventional engine (Fig. 1), and initially large pumping losses. The desired positioning of the intake valve opening profile is shown by the red-dotted curve in Fig. 2. Thus, a CVVL mechanism of this kind, referred to as the center or neutrally biased, has to be combined with intake VCT.

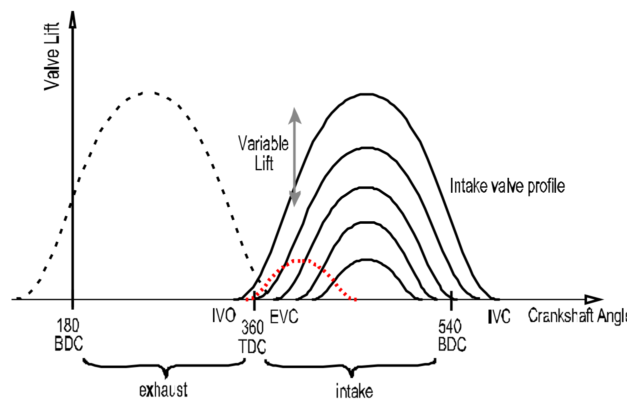

Fig. 2: Intake valve profiles versus crank angle for different values of lift.

Obviously, achieving the potential benefit of this fully variable valvetrain configuration requires accurate positioning and synchronization of the intake valve timing and valve lift. Moreover, the combined effect of the two actuators, in an engine with a high compression ratio significantly increases a chance of piston-to-valve interference. To appreciate this point, consider what may happen if the engine operates under a low load condition with the valve profile shown by the red-dotted line in Fig. 2. If the driver requests more torque, the lift has to increase, while the cam timing has to retard. If the lift, typically controlled by an electric motor, responds much faster than the hydraulically operated cam timing, the intake valve opening occurs much earlier than desired creating a potential for collision. This possibility can be removed at the engine design stage by reducing compression ratio or limiting lift or cam phase shift range. Either way, some of the benefits of the hardware will be lost. Another possibility is to rely on the control system to coordinate the two actuators. The latter approach is discussed in the remainder of this paper.

III. VALVE TIMING AND LIFT CONTROL

A controls solution to the valve timing and lift synchronization has to be found within a framework of the overall engine control architecture. The structure of the controller is dominated by the feedforward component mostly because the performance variables (torque, brake specific fuel consumption, emissions) are not directly measured. A typical structure is shown in Fig. 3. At the top of the hierarchy is the control block that interprets driver demand, handles mode transitions, FMEM, and system reconfiguration. In the middle is the scheduler-coordinator, the purpose of which is to find the best way to satisfy driver demand, while reducing fuel consumption and emissions. To do this, it issues commands to each engine actuator that controls a device. At the lowest level of the hierarchy, each device is typically controlled by a closed loop system. Finally, we would like to point out that there is an outer feedback loop that goes from the driver demand, through engine torque, vehicle acceleration and speed, and back to the driver.

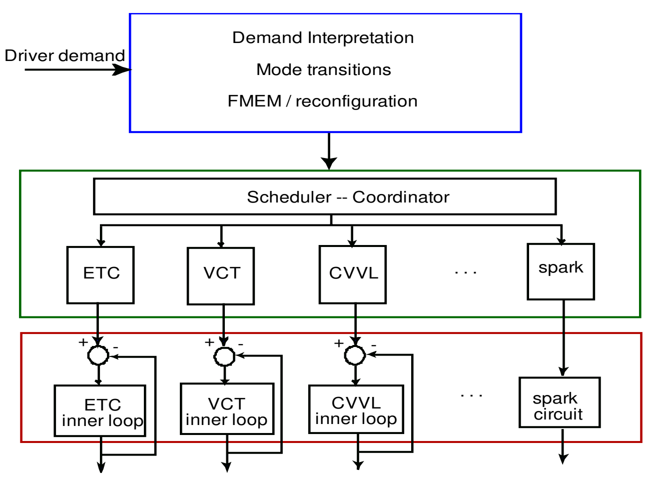

Fig 3: A hierarchical block diagram of a typical engine control system.

The cam timing is controlled by a hydraulic actuator, while the lift is controlled by an electric motor [2, 3]. We exploit the fact that there is little dynamic interaction between the two, at least in the frequency range of each actuation system, and model them separately. Then we demand that their outputs satisfy an inequality which assures that there is no interference between the piston and the valve.

A stand-alone dynamic model of the VCT actuation system has been derived in [6]. The model consists of an input nonlinearity (saturation) followed by an integrator and a first order lag with delay. Alternatively, one may consider the VCT behavior in a closed loop with a PID controller and simplify the closed loop VCT model to a rate limited first order lag [5]:

$$\dot{\zeta} = \begin{cases} -r_{inc} & \text{if} \quad -a\zeta + a\zeta_{des} < -r_{inc} \\ -a\zeta + a\zeta_{des} & \text{if} \quad -r_{inc} < -a\zeta + a\zeta_{des} < r_{dec} \\ r_{dec} & \text{if} \quad r_{dec} < -a\zeta + a\zeta_{des} \end{cases} \tag{1}$$

ζ : is the actuator position, such as VCT or CVVL

$-a\zeta + a\zeta_{des}$: is the close loop error multiplied by gain a

r_{inc} , r_{dec} : rate of increase and decrease of the actuator.

In the case of the VCT actuator the inertial forces retard the timing without actuation force, and springs added to the system assist in advancing the timing, typically resulting in non-symmetric rates of advance and retard.

The lift control system consists of a mechanism that directly controls the valve lift, an electric motor that rotates it through a set of gears, and a sensor that measures the rotational angle (and indirectly valve lift). With a relatively large gear ratio, the inertial loading on the engine is dominated by the rotor inertia. Because we are interested here in a gross motion of the lift (and not fine positioning), the motor torque is likely to saturate when required to move the valve lift through its full range. We again adopt the first order, rate limited system as the model of the valve lift described in Equation 1.

Finally, the regulated outputs of the CVVL and VCT systems must satisfy an inequality constraint $f(y_{lift}, \zeta_{cam}) > 0$. That is, the engine is required to operate away from the undesirable region in the valve timing - valve lift space. To accomplish this, the controller is allowed to temporarily modify the commands (desired values) to each subsystem from their values computed to achieve the best fuel consumption, emissions, or performance.

We have exploited the decoupled structure of the VCT and CVVL dynamics and their minimum phase, low-overshoot (or no overshoot) behavior in the closed loop to propose a simple solution to the non-interference problem in the next section. If either of these assumptions is substantially violated, one would have to consider using reference governor [7], or model predictive control [8].

IV. PREVENTING MECHANICAL INTERFERENCE

Several scenarios exist where different relative actuator speeds may result in minimum clearance violation (MCV) during transition:

MCV occurs because VCT is faster than CVVL:
Lift is commanded from high to low and intake timing must advance to maintain desired IVO. MCV results if VCT advance gets ahead of reduction in lift, allowing the valve to be too far extended when the piston is near top dead center (TDC). Fig. 4 shows the un-coordinated results of this scenario.

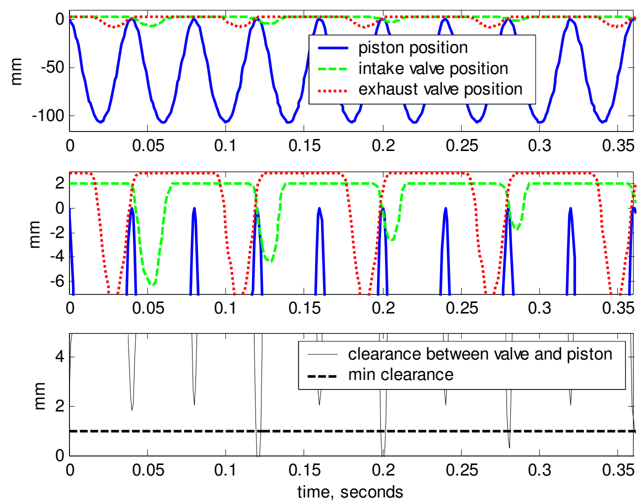

Fig. 4: VCT gets ahead of lift, causing interference.

MCV occurs because CVVL is faster than VCT:
Lift is commanded from low to high and intake timing must retard to maintain desired IVO resulting in a violation if lift increase gets ahead of VCT retard, allowing the valve to be too far extended when the piston is near TDC. Fig. 5 shows the un-coordinated results of this scenario.

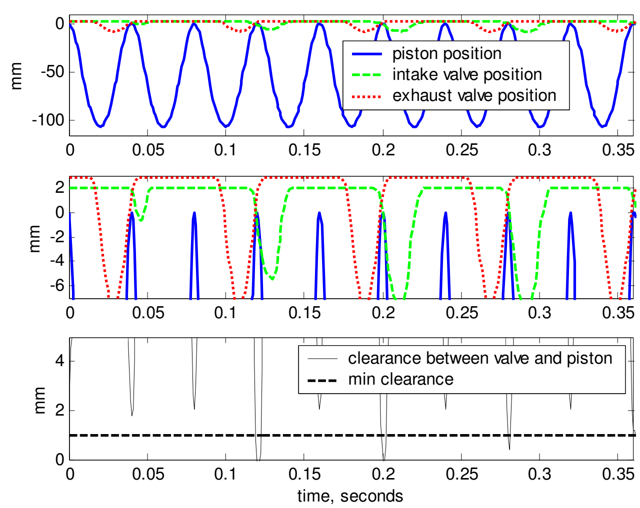

Fig. 5: Lift gets ahead of VCT, causing interference.

CVVL and VCT Reference Control to maintain minimal clearance:
A method to maintain minimal clearance (MC) is to place an intermediate stage between the reference signals of desired lift and VCT and the close loop control stage that regulates each actuator's position. Fig. 6 indicates where this new sub-system is added into the CVVL/VCT system. The VCT and Lift Coordinator (labeled *VCT_Lift_Coord*) is placed between the *Powertrain Controller*, that provides reference signals for VCT and lift, and actuator controllers (*VCT_Controller* and *Lift_Controller*) that enforce the powertrain's reference signals.

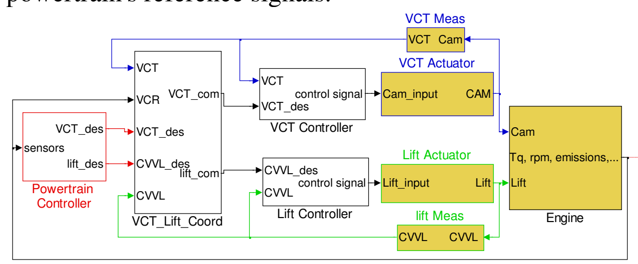

Fig. 6: Block diagram of system showing a new strategy block, *VCT_lift_Coord*, inserted into a standard control system.

We obtain MC by creating a map that relates inputs of VCT and lift to an output of clearance. Fig. 7 shows that for a given minimum clearance, there is curve that relates lift and VCT at TDC. High lift and advanced VCT result in interference, lower lift and retarded VCT result in more clearance. The exact path of the curves is dependent on engine geometry. Either a PCM implemented estimator or a mapped function can provide clearance.

$$CLR = Fn_{clr}\left(CVVL, VCT, VCR\right) \qquad (2)$$

where *CVVL* is either expressed as a fraction of maximum lift (used in these examples) or a linear measurement of CVVL position, *VCT* is the cam timing relative to TDC of the piston position of an induction stroke, *VCR* is the compression ratio the engine is operating at, and *CLR* is the clearance distance that exists between the piston at TDC and the intake valve. If a function is used, it only needs to contain values of *CLR* assuming the piston is at TDC.

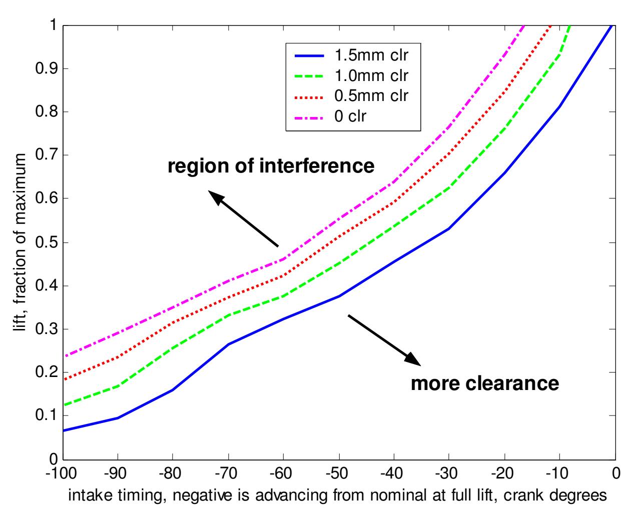

Fig. 7: TDC clearance curves in the VCT and valve lift plane, fixed VCR.

The second step compares the calculated *CLR* with a minimal clearance, which accounts for physical assembly tolerances. If the *CLR* is less than or equal to the minimal clearance, the valve position is considered in violation of the specified clearance and a flag is set that will be used to halt one or both of the actuators. For some systems a necessary modification to add an additional tolerance, or a buffer, to the minimal clearance, especially if a static map is employed to determine *CLR*. If the actuators' travel rate is sufficiently fast relative to the update rate of the algorithm, the required clearance should be increased to account for actuator travel between controller updates. One approach is to assume the travel between actuator updates is the worst case. This may be too prohibitive of actuator response. An alternative approach is to track the change of position between updates, assuming that the next change in position will be similar to the last. The direction of movement of actuator can be taken into account if so desired: movement towards interference would be considered, otherwise ignored. More sophisticated projections of actuator position may be needed depending on the dynamics of the actuators.

$$CLR_{total} = Fn_{clr}(CVVL, VCT, VCR) + CLR_{buffer} \quad (3)$$

The logic in the *VCT_Lift_Coord* assumes two continuously variable actuators: VCT and CVVL. VCR is assumed, in these examples, to be a switching device. VCR state is accounted for in this example only when calculating the replacement value for the offending actuator.

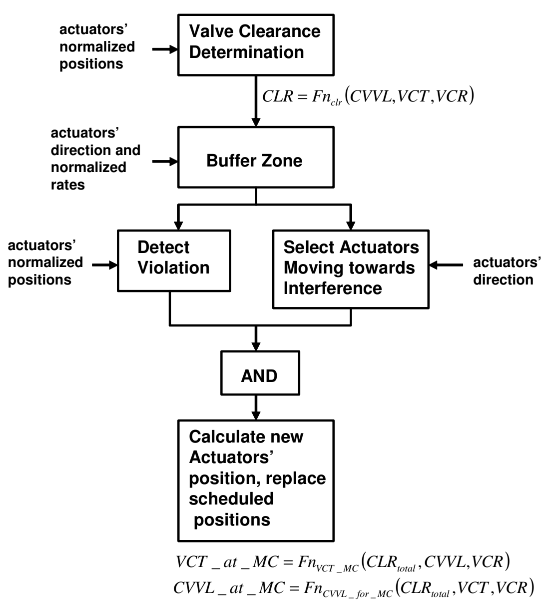

Fig. 8: Flow of control in the *VCT_Lift_coord* block.

We have separated the detection and correction of an immanent MCV to allow the strategy to more readily accommodate additional devices. The next step is to determine which actuator(s) should be restricted to avoid interference. Because the steady state scheduling will not specify an infeasible point, some actuators are moving away from the interference curve. Those actuators that are moving toward interference curve must be restricted until the interference is no longer immanent.

Each actuator desired direction is used to set a flag indicating movement towards interference. If an immanent interference exists, then this actuator specific flag is passed on to the preventive strategy.

Two new functions, using the same body of data used in *Function_lift_vct_clr*, provide MC positions for a given actuator based on the desired CLR_{total}, the current position of the other actuator, and *VCR*:

$$VCT\_at\_MC = Fn_{VCT\_MC}(CLR_{total}, CVVL, VCR) \quad (4)$$

$$CVVL\_at\_MC = Fn_{CVVL\_for\_MC}(CLR_{total}, VCT, VCR) \quad (5)$$

The final step is either pass the PCM scheduled position to the close loop controller of the actuator(s) that are not in violation and to restrict the commands to the actuators that are moving towards interference. Fig. 9 shows in the actuator position space three possible paths for a scheduled change in CVVL and VCT. The green dashed path is ideal from an interference perspective. The safe actuator, the retarding VCT, outruns the CVVL increasing lift. This may be good for a slowly changing schedule, where desired operating points actually fall on the path. A sudden change in scheduling might demand that the actuators reach new positions as fast as possible. The red dotted path is a faster CVVL lift than VCT retard, resulting in piston to valve interference. The black dash-dot "restricted" path is case where CVVL is out pacing the VCT, but once it crosses into the buffer zone, defined by the blue curve, it is forced to take on a new commanded position, just outside the blue curve. This keeps the position well away from the magenta curve, where interference occurs. The overall response is as prompt as the "safe" actuator allows.

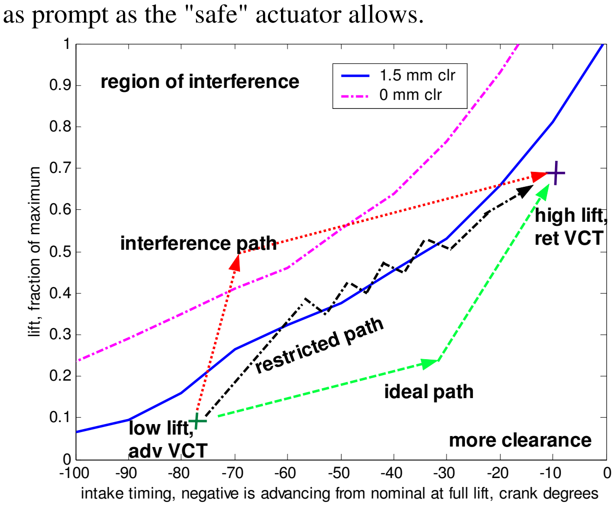

Fig. 9: Possible paths that the two actuators may take when moving from the lower left to the upper right steady state scheduled operating point.

V. SIMULATION RESULTS

We have constructed a model that uses crank angle to produce positions of the engine piston, intake, and exhaust valves relative to the center of the gasket that separates the cylinder head and the engine block. The compression of the engine is relatively high, providing a large range of potential valve to piston interference. The VCT and CVVL actuators and the feedback controls that regulate these mechanisms are represented by a first order filter structures in which the integrator in the structure, the value of which is the actuator position, can be initialized at the beginning of the simulation. The actuator speeds, set by gain before the integrator, are relatively high, so that the simulation can produce a large range of actuator movement in only a few engine cycles.

In Fig. 10, the lift actuator is set to high rate, VCT actuator set to low rate, lift schedule goes from 0.3 to 1.0 of maximum lift, VCT retards from 70 degrees advanced (indicated as -70) to 0 degrees advanced. Lift increases too fast for VCT retard, so the VCT Lift Coordinator interrupts the desired signal to the CVVL mechanism when the minimum clearance is violated (second row, green trace falls below red trace). The third row plot shows the lift commanded value (red) is jumping to the calculated MC lift position (cyan). Fourth row plot, blue trace, shows the lift position, which progresses normally when at MC, and reduces towards MC when in violation.

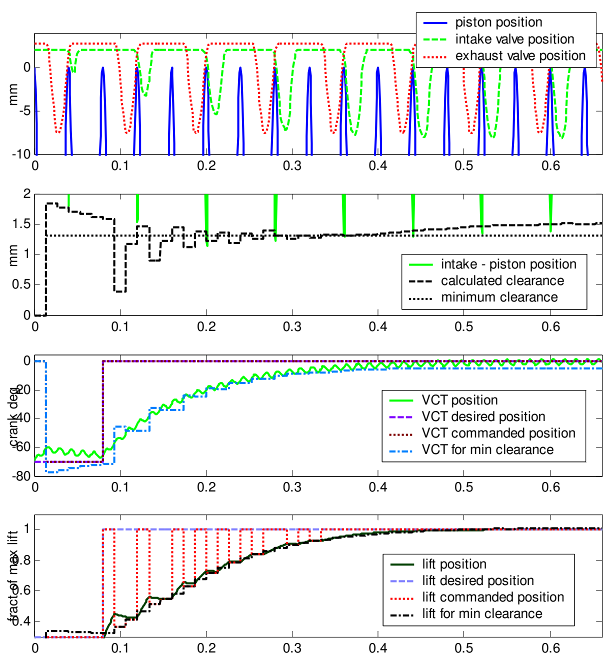

Fig. 10: CVVL lift is too fast and is restricted.

In Fig. 11, the lift actuator is set to low rate, VCT actuator set to high rate, lift schedule goes from 0.9 to 0.3 of maximum lift, VCT advances from 0 degrees advanced to -70 degrees advanced. VCT advances too fast for lift reduction, so the VCT Lift Coordinator interrupts the desired signal to the VCT mechanism when the minimum clearance is violated (second row, green trace falls below red trace). The third row plot shows the VCT commanded value (red) jumping to the calculated MC VCT (cyan). Third row plot, blue trace, shows the VCT, which progresses normally when at MC, and moves towards MC when in violation. The VCT has combustion frequency oscillation of +/-2 degrees about the average VCT position.

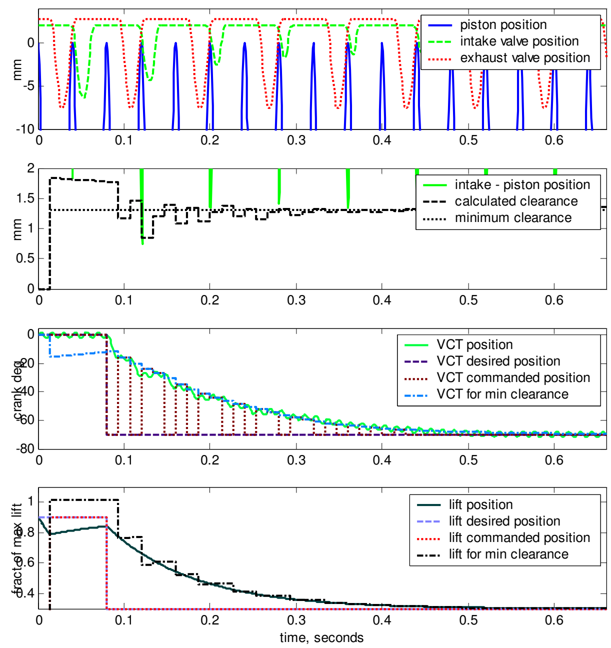

Fig. 11: VCT is too fast and is restricted.

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