INTEGRATED SYSTEM IDENTIFICATION AND CONTROL DESIGN FOR AN IC ENGINE VARIABLE VALVE TIMING SYSTEM

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ABSTRACT
This paper applies integrated system modeling and control design process to a continuously variable valve timing (VVT) actuator system that has different control input and cam position feedback sample rates. Due to high cam shaft torque disturbance and high actuator open-loop gain, it is fairly difficult to maintain the cam phase at the desired constant level with an open-loop controller. As a result, multirate closed-loop system identification is a necessity. For this study, multirate closed-loop system identification, PRBS $q$-Markov Cover, was used for obtaining linearized system models at different engine operational conditions; and the output covariance constraint (OCC) controller; an $H_2$ controller, was designed based upon the identified model and evaluated on the VVT test bench. Performances of the designed OCC controller was compared with those of the baseline PI controller on the test bench. Results show that the OCC controller uses less control effort and has less overshoot than those of PI ones.

INTRODUCTION
Continuously variable valve timing (VVT) system was developed in early nineties [1]. The benefits of using VVT for internal combustion engines include improved fuel economy with reduced emissions at low engine speed, as well as increased power and torque at high engine speed. Vane-type VVT system [2] is a hydraulic mechanic actuator controlled by a solenoid. Electric motor driven cam phase actuators become available recently due to its fast responses [3]. This paper studies the modeling and control of hydraulic VVT systems.

There are two approaches to obtain a control oriented VVT model for model-based control development and validation: physics based system modeling [4] and system identification. In this paper, the closed-loop system identification approached is employed to obtain the VVT system model. System identification using closed-loop experimental data was developed in seventies [5] and it has been widely used in engineering practice [6-8]. Closed-loop system identification can be used to obtain the open-loop system models when the open-loop plant cannot be excited at the conditions ideal for system identification. For instance, the open-loop plant could be unstable. In this paper, the closed-loop identification method was selected due to many factors. The main reason is that VVT actuator is disturbed by engine speed, load, oil pressure and temperature, which made it impossible to maintain the cam phase at a desired value. Therefore, open-loop system identification at a desired cam phase is not practical. In order to maintain at a desired operational condition for identifying the VVT actuator system, closed-loop identification was selected.

The first step of this paper is to obtain linearized system models for VVT actuator system at different operational conditions using the indirect closed-loop system identification approach discussed in [7]. The $q$-Markov COVariance Equivalent Realization ($q$-Markov Cover) system identification method [9-11] using PRBS (Pseudo-Random Binary Signals) was used to obtain the closed-loop system models. The $q$-Markov cover theory was originally developed for model reduction. It guarantees that the reduced order system model preserves the first $q$ Markov parameters of the original system. The realization of all $q$-Markov Covers from input and output data of a discrete time system is useful for system identification. Q-Markov Cover for system identification uses pulse, white noise, or PRBS as input excitations. It can be used to obtain the linearized model representing the same input/output sequence for nonlinear systems [11]. It was also been extended to identify multirate discrete-time systems when input and output sampling rates are different [12].

For the proposed study, the multirate system identification is required due to event based cam phase sampling (function of
engine speed) and time based control sampling. For our test bench setup, the cam position was sampled four times per engine cycle. For instance, when the engine is operated at 1500 RPM, the cam position sample period is 20ms, and the control input is updated at a fixed sample period of 5ms. For this study, multirate PRBS q-Markov Cover was used for closed-loop system identification on the VVT test bench. System models at different engine operational conditions were identified using closed-loop multirate identification. The OCC (Output Covariance Constraint) control problem [13-15] minimizes control effort subject to multiple performance constraints on output covariance matrices. An iterative algorithm with guaranteed convergence can be used for finding a controller satisfying the optimality conditions. The OCC control scheme was used in aerospace control problems for minimal control effort [13-15]. In this paper, a nominal model was selected for control design using OCC design method. Multiple OCC controllers were designed based on closed-loop identified models, and their performance was compared against that of PI controller. In order to eliminate steady state error, system control input was increased to add integration to the OCC controllers. The results show that OCC controllers were able to achieve the similar system performance to PI ones with much less control effort.

The paper is organized as the following. First, framework and formulation of closed-loop identification for the VVT actuator system are provided. Then OCC design framework is presented. The identification and controller design results obtained from the test bench are showed, along with the discussions of the experiment results. Conclusions are provided in the last section.

SYSTEM IDENTIFICATION FRAMEWORK

Consider a general form of linear time-invariant closed-loop system in Fig. 1, where \( r \) is the reference signal, \( n \) is the measurement noise, \( u \) and \( y \) are input and output.

As discussed in the Introduction section, there are many approaches for the closed-loop identification, which are categorized as direct, indirect, and joint input-output approaches. In this paper, we utilize the knowledge of the controller to calculate open-loop plant from identified closed-loop plant, which is called indirect approach. To ensure the quality of identified plant, controller in this paper is set to be proportional ([16] and [17]).

\[
y = H \cdot r = GK(I + GK)^{-1}r
\]  

(1)

Let \( \hat{H} \) be identified closed-loop transfer functions from \( r \) to \( y \). The open-loop system model \( G_{ID} \) can be calculated using identified \( \hat{H} \), assuming that \( (I - \hat{H})^{-1} \) is invertible. The closed-loop controller transfer function is used to solve for the open-loop system models. We have

\[
G_{ID} = \hat{H}(I - \hat{H})^{-1}K^{-1}
\]  

(2)

PRBS signal is used in the identification of the system. The most commonly used PRBS are based on maximum length sequences (called \( m \)-sequences) [18] for which the length of the PRBS signals is \( m = 2^n - 1 \), where \( n \) is an integer (order of PRBS). In this paper, inverse PRBS [11] is used in the \( q \)-Markov Cover identification algorithm. For convenience, in the rest of this paper, the term “PRBS” is used to represent the inverse PRBS. In this paper, system models are identified in discrete time domain using PRBS-GUI [12] developed for multirate PRBS \( q \)-Markov Cover. The advantage of using the GUI is that the number of Markov parameter and the order of the identified model can be adjusted based upon identification results.

OUTPUT COVARIANCE CONTROL (OCC)

Consider the following linear time-invariant system (11)

\[
x_p(k + 1) = A_p x_p(k) + B_p u(k) + D_p w_p(k)
\]

\[
y_p(k) = C_p x_p(k)
\]

\[
z(k) = M_p x_p(k) + v(k)
\]

(11)

where \( x_p \) is the state, \( u \) is the control, \( w_p \) represents process noise, and \( v \) is the measurement noise. The vector \( y_p \) contains all variables whose dynamic responses are of interest. The vector \( z \) is a vector of noisy measurements. Suppose that we apply to the plant (11) a strictly proper output feedback stabilizing control law below:

\[
x_c(k) = A_c x_c(k) + F z(k)
\]

\[
u(k) = G x_c(k)
\]

(12)

Then the resulting closed-loop system is

\[
x(k + 1) = Ax(k) + Dw(k)
\]

\[
y(k) = \begin{bmatrix} y_p(k) \\ u(k) \end{bmatrix} = \begin{bmatrix} C_y \\ C_u \end{bmatrix} x(k) = C x(k)
\]

(13)

where \( x = [x_p^T \ x_c^T]^T \) and \( w = [w_p^T \ v^T]^T \). Formulas for \( A \), \( C \), and \( D \) are easy to obtain from (11) and (12).
Consider the closed-loop system (13). Let \( p_W \) and \( V \) denote positive definite symmetric matrices with dimensions equal to the process noise \( w_p \) and measurement vector \( z \), respectively. Define \( W = \text{block diag}[p_W, V] \) and let \( X \) denote the closed-loop controllability Gramian from the input \( W^{-1/2}w \). Since \( A \) is stable, \( X \) is given by:

\[
X = AXA^T + DWD^T
\]  
(14)

In this paper we are interested in finding controllers of the form (12) that minimize the (weighted) control energy \( \text{trace}(RC_uXC_u^T) \) with \( R > 0 \), and satisfy the constraints:

\[
Y = CXC^T \leq \bar{Y}
\]  
(15)

where \( \bar{Y} \geq 0 \) are given and \( X \) solves (14). This problem, which we call the output covariance constraint (OCC) problem, is defined as: Find a full-order dynamic output feedback controller for system (11) to minimize the OCC cost

\[
J_{\text{occ}} = \text{trace}(RC_uXC_u^T), \quad R > 0
\]  
(16)

subject to (14) and (15).

The OCC problem may be given several interesting interpretations. For instance, assume first that \( w_p \) and \( v \) are uncorrelated zero-mean white noises with intensity matrices \( W_p > 0 \) and \( V > 0 \). Let \( E \) be an expectation operator, and

\[
E[w_p(k)] = 0; \quad E[w_p(k)w_p^T(k-n)] = W_p\delta(n)
\]

\[
E[v(k)] = 0; \quad E[v(k)v^T(k-n)] = V\delta(n)
\]  
(17)

Defining \( E_{\epsilon}[\cdot] := \lim_{k \to \infty} E[\cdot] \) and \( W = \text{block diag}[W_p, V] \), it is easy to see that the OCC is the problem of minimizing \( E_{\epsilon}u^TRu \) subject to the OCC constraint \( Y := E_{\epsilon}y(k)y^T(k) \leq \bar{Y} \). As is well known, the constraint may be interpreted as constraint on the variance of the performance variables or lower bounds on the residence time (in a given ball around the origin of the output space) of the performance variables [19].

The OCC problem may also be interpreted from a deterministic point of view. To see this, define the \( L_{\epsilon} \) and \( L_2 \) norms:

\[
\|y\|_{\infty}^2 := \sup_{k \geq 0} y^T(k)y(k)
\]

\[
\|w\|_2^2 := \sum_{k=0}^{\infty} w^T(k)w(k)
\]  
(18)

and define the (weighted) \( L_2 \) disturbance set

\[
\mathcal{W} := \left\{ w : R \rightarrow R^{n_w} \text{ and } \|W^{-1/2}w\|_2^2 \leq 1 \right\}
\]  
(19)

where \( W > 0 \) is a real symmetric matrix. Then, for any \( w \in \mathcal{W} \), we have [20, 21]:

\[
\|y\|_{\infty}^2 \leq \sigma(Y), \quad \text{and} \quad \|u_i\|_{\infty}^2 \leq \left[C_uX_uC_u^T\right]_{ii}, \quad i = 1, 2, \ldots, n_u
\]  
(20)

where \( n_u \) is the dimension of \( u \). (Here, \( \sigma[\cdot] \) denotes the maximum singular value and \( [\cdot]_{ii} \) is the \( i \)-th diagonal entry.) Moreover, references [20, 21] show that the bounds in (20) are the least upper bounds that hold for any signal \( w \in \mathcal{W} \).

Thus, if we define \( \bar{Y} := I\epsilon^2 \) in (15) and

\[
R = \text{dial}[\tau_1, \tau_2, \ldots, \tau_{n_u}] \in (16), \text{ the OCC problem is the problem of minimizing the (weighted) sum of worst-case peak values on the control signals given by}
\]

\[
J_{\text{occ}} = \sum_{i=1}^{n_u} \tau_i \left\{ \sup_{w \in \mathcal{W}} \|u_i\|_{\infty}^2 \right\}
\]  
(21)

subject to constraints on the worst-case peak values of the performance variables of the form

\[
\sup_{w \in \mathcal{W}} \|y\|_{\infty}^2 \leq \epsilon^2
\]  
(22)

This interpretation is important in applications where hard constraints on responses or actuator signals cannot be ignored, such as space telescope pointing and machine tool control.

Detailed proof can be found in [15]. The controller system matrices \( A_c, F, \) and \( G \) can be calculated using an iteration algorithm introduced in [13] and [15].

**VVT SYSTEM BENCH TESTS SETUP**

**System configuration**

![VVT SYSTEM BENCH TESTS SETUP](image)

**FIG. 2: VVT PHASE ACTUATOR TEST BENCH**

Closed-loop system identification and control design test were conducted on VVT test bench (Fig. 2). A Ford 5.4L V8 engine head was modified and mounted on the test bench. The cylinder head has a single cam shaft with a VVT actuator for
two intake valves and one exhaust valve. These valves introduce cyclic torque disturbances to the cam shaft. The cam shaft is driven by an electrical motor through a timing belt.

An encoder is installed on the motor shaft (simulating the crank shaft), which generates crank angle signal with one degree resolution, along with a so-called gate signal (360 degrees per pulse). A plate with magnets was mounted at other side of the extended cam shaft. These magnets pass two cam position sensors when the engine is running. One sensor is used to determine engine combustion TDC position, along with the encoder signals. The other is used to determine the cam phase, and the cam position signal updates 4 times per cycle.

The cam phase actuator system consists of a solenoid driver circuit converting DC voltage command to PWM signal, a solenoid actuator, and hydraulic cam actuator. An electrical oil pump was used to supply pressurized oil for both lubrication and as hydraulic actuating fluid of the cam phase actuator. The cam actuator command voltage signal is generated by the Opal-RT prototype controller and sent to the solenoid driver. The PWM duty cycle is proportional to input voltage with maximum duty cycle (99%) corresponding to 5V. The solenoid actuator controls the hydraulic fluids (engine oil) flow and changes the cam phase. The cam position sensor signal is sampled by the Open-RT prototype controller and the corresponding cam phase is calculated.

The cam phase actuator has an output range of ±30 degrees. Fig. 4 shows an open-loop step response of the VVT phaser. Input to the system is a step of 0V (0% duty cycle) and 5V (99% duty cycle). It can be found that the cam phase system has a settling time about 1.5 seconds for advancing (rising) and 1.0 second for retard (falling), demonstrating its nonlinear characteristics. This is mainly due to the fact that the VVT actuator has different dynamics for advancing and retarding. For advancing, the actuating torque generated by the oil pressure overcomes the cam load torque and moves cam phase forward; and for retarding, the oil trapped in the actuator bleeds back to the oil reserve when the cam phase is pushed back by the cam shaft load. This difference leads to the response characteristics difference for advance and retard operations, which makes the system nonlinear. This phenomenon will be discussed in bench test section.

![FIG. 4: CAM PHASE ACTUATOR OL STEP RESPONSE](image)

**FIG. 4: CAM PHASE ACTUATOR OL STEP RESPONSE**

Fig. 5 shows the VVT system steady-state responses via open loop constant inputs of 0.1V interval. It can be observed that for open-loop control, the cam phase actuator behaves almost like a binary state and it is very difficult for the VVT actuator to maintain at a desired non-saturated cam timing position due to the actuator hysteresis characteristics, cam load and engine oil pressure variation. This indicates that open loop system identification, which requires to hold the actuator operate at a desired location during the system identification process, is almost impossible. Therefore, closed-loop system identification is adopted in this research. In order to ensure the closed loop system identification, a proportional controller is selected for the closed loop system identification to have good system identification accuracy ([16] and [17]).

![FIG. 5: CAM PHASE ACTUATOR OPEN-LOOP STEADY-STATE RESPONSES](image)

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**VVT Open-loop properties**

The cam phase actuator has an output range of ±30 degrees. Fig. 4 shows an open-loop step response of the VVT phaser. Input to the system is a step of 0V (0% duty cycle) and 5V (99% duty cycle). It can be found that the cam phase system...
BENCH TEST RESULTS
Closed-loop Identification

The operating point and controller gain need to be selected carefully due to the system property. The solenoid drive circuit has an operational range of 0 and 5 volts that corresponding to 1 and 99 percent of the solenoid PWM duty cycle. Therefore, in order to avoid saturation, we have to select the phase actuator operation condition carefully; otherwise, the control input might be saturated, leading to high system identification error. Therefore, the PRBS signal magnitude is selected to be 12°, nominal operational condition is centered at -14° cam phase, and the controller proportional gain is 0.1 (volt/degree). To obtain a family of system transfer functions, the system identification bench tests were conducted at different engine speeds and oil pressures. In this case we selected two engine speeds (1000 and 1500 rpm) and a constant oil pressure of 60psi. Recorded reference signals and system response data are processed using MATLAB PRBS-GUI [12].

TABLE 1 SYSTEM IDENTIFICATION PARAMETERS

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>1000</th>
<th>1500</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Sample Rate (ms)</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Output Sample Rate (ms)</td>
<td>30</td>
<td>20</td>
</tr>
<tr>
<td>Output/Input Sample Ratio</td>
<td>0.167</td>
<td>0.25</td>
</tr>
<tr>
<td>PRBS order</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>Signal length (s)</td>
<td>81.88</td>
<td>81.88</td>
</tr>
<tr>
<td>Markov parameter, #</td>
<td>90</td>
<td>60</td>
</tr>
<tr>
<td>1D open-loop model order</td>
<td>4</td>
<td>2 and 4</td>
</tr>
</tbody>
</table>

Number of Markov parameters to be match by system identification was used to optimize the identification accuracy, and identified system model order is determined by the dominant dynamics of PRBS response data (see Fig. 6). Fig. 6 shows the order selection diagram produced during PRBS system ID at 1500rpm. The chart shows a dominant 1st order dynamic because the order index chart has the largest gap between the 1st and 2nd dots. Gap between 4th and 5th dots are larger than the gap between 2nd and 3rd order gaps. Therefore, the order of the identified model is selected to be 4 in order to keep the model order low without losing major system dynamics. The rest of system identification parameters are shown in Tab. 1.

![FIG. 6: IDENTIFIED MODEL ORDER SELECTION](image)

Using the identified closed-loop model and equation (2), a 4th order open-loop plant model (see Appendix A) at 1500rpm is obtained. Continues time models are used to represent the physical system's continuous property. The corresponding open loop Bode diagram (Fig. 7) shows that there exists a dynamic mode at around 12.5 Hz, which is equal to the engine cycle frequency of 12.5 Hz (80ms/cycle) at 1500rpm. We believe this dynamic mode is associated with the cyclic cam load introduced by both the valves and cam timing belt. For the purpose of phase controller design, the dynamics introduced by the cam cyclic load can be eliminated. Therefore, a second order model is obtained by eliminating the 12.5Hz mode. Plant model calculated from the identified 2nd order model has almost identical behavior to the 4th order model without the 12.5 Hz mode (see Fig. 7 and Appendix A).

![FIG. 7: BODE DIAGRAM OF OL PLANT AT 1500RPM](image)

4th order closed-loop model is identified at 1000rpm. Similar to the case at 1500rpm, the identified model has a dynamic mode at about 8 Hz, which is corresponding to engine cycle period (8.3 Hz, 120ms/cycle). However, in this case, a 2nd order open loop model was not obtained directly from system identification. Note that in this case, after calculating open-loop plant from the identified closed-loop model there exists a pair of non-minimal phase zeros shown in the root locus at the frequency close to engine cycle frequency. To eliminate the dynamics at this frequency, a 2nd order model is obtained by removing the pole-zero pairs from the 4th order plant model (see Appendix A). The 2nd order plant model has very similar frequency response as the 4th order plant model except without the dynamics introduced by the cyclic engine cam load (Fig. 8).

![FIG. 8: BODE DIAGRAM OF OL PLANT AT 1000RPM](image)
Validation of Identified Model

To evaluate the accuracy of these identified models, their step responses are compared with these obtained from the bench tests. Since the open-loop step response cannot be obtained for the VVT actuator, their closed-loop responses are compared in this study. The same proportional control gain of 0.1 Volt/degree was used for the step responses. A step input of 12 crank degrees is used. For the identified models, simulations were conducted in Simulink under the same conditions. The normalized step responses are compared in Fig. 9 at 1000rpm and Fig. 10 at 1500rpm. Notice the oscillations in the recorded response and it demonstrates the difficulty for a proportional controller to maintain the cam phase at a desired level.

Family of system models were obtained from bench tests at different engine speeds and oil pressures. A $2^{\text{nd}}$ order model (23) was selected as the nominal model for control design below:

$$G = \frac{0.0003s^2 - 0.06s + 647.2}{s^2 + 7.615s + 20.67}$$

OCC controller with single input

In this section, controllers were designed and validated on the test bench. Step input was used as reference signal and varies between -20 and 0 degree. A PI controller was tuned for the VVT system on the test bench for comparison purpose. The PI tuning process was done at different engine speeds and oil pressures. The tuned PI (24) achieves good balance between fast response time and little oscillations at different conditions:

$$K_{\text{base}}(s) = 0.2 + \frac{0.1}{s}$$

For OCC design, system matrices of nominal model (23) are:

$$A_p = A = \begin{bmatrix} -7.62 & -20.68 \\ 1 & 0 \end{bmatrix}, \quad B_p = D_p = B = \begin{bmatrix} 1 \\ 0 \end{bmatrix}$$

$$C_p = M_p = C = \begin{bmatrix} -0.063 \\ 647.39 \end{bmatrix}, \quad D = 0$$

Controller design parameters are selected as:

$$W_p = 1, \quad V = 0.01, \quad R = [1]$$

Using the iterative design algorithm in [15], an OCC controller was obtained:

$$K_{\text{occ}}(s) = \frac{194.8s + 2701}{s^2 + 131s + 8582}$$
\[ K_{occ-i} = \frac{-239.9s^2 - 2751s - 1.1\times10^4}{s(s^3 + 51.3s^2 + 1305s + 1.97\times10^4)} \]  

(28)

The OCC control with integrator has a large overshoot with oscillations (Fig. 11). In order to eliminate steady-state error and reduce response time, a multi-input control design with proportional and integral inputs are proposed.

\[ K_{occ-2ir} = \frac{84.5s^3 + 935.2s^2 + 1164.5s + 220}{s^4 + 122.2s^3 + 7464.7s^2 + 3022s} \]  

(29)

FIG. 12: OCC DESIGN FRAMEWORK WITH INTEGRATOR

**OCC Controller Design with Multi-input**

For the dual-input control design, the controller has an additional integrator input to the plant (see Fig. 13). Noise intensity matrices \( W_p \) and \( V \) were the same as (26). Input weighting matrix was selected as \( R = \text{diagonal}(1,20) \). Note that in this case the input effort cost ratio between direct control and integral control is 1 to \( \sqrt{20} \). The dual-input controller was designed and shown in (29) and its performance at 900rpm with 45 PSI oil pressure is compared with base PI controller in (17). Fig. 14 shows that both controllers have very similar response times and steady state errors and OCC controller has less overshoot.

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(29)

FIG. 13: MULTI-INPUT OCC DESIGN FRAMEWORK

**Controller Performance Comparison**

Table 2 shows response comparison between PI and multi-input OCC controller. Both controllers have zero steady-state error, with oscillation magnitude of 1 degree (lowest possible and limited by measurement resolution). Both controllers have very similar 95% settling time and 10–90% rising time. Compared with PI controller, the OCC controller has much lower overshoot. In some cases, OCC controller reduces PI controller’s overshoot by 50%. In advance step (from -20 to 0 degree), multi-input OCC controller uses less control effort than PI. In retard step, control effort difference is smaller (Fig. 15). The reason is that in advance step, all the effort is made by the actuator; while in the retard step, engine oil pressure is working with actuator. At steady-state, multi-input OCC controller shows higher effort spikes than PI controller. This is due to the fact that the designed OCC controller has a higher gain than PI and therefore is more sensitive to the change in error signal, which has the resolution of one crank degree in the experiment. Small step response (Fig. 16) test shows the controller is able to overcome system hysteresis.

**TABLE 2: CONTROLLER PERFORMANCE COMPARISON**

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>Oil Pre. (psi)</th>
<th>Overshoot (deg)</th>
<th>Advanced/Retarded Performance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Settling Time (s)</td>
<td>Rising Time (s)</td>
</tr>
<tr>
<td>900</td>
<td>7/4</td>
<td>2.16/2.38</td>
<td>0.39/0.19, 0.34/0.17</td>
</tr>
<tr>
<td>1200</td>
<td>5/5</td>
<td>2.39/1.86</td>
<td>0.36/0.17, 0.32/0.15</td>
</tr>
<tr>
<td>1500</td>
<td>5/5</td>
<td>2.01/1.91</td>
<td>0.32/0.20, 0.28/0.21</td>
</tr>
<tr>
<td>1800</td>
<td>6/4</td>
<td>2.10/1.82</td>
<td>0.33/0.18, 0.26/0.21</td>
</tr>
<tr>
<td>900</td>
<td>6/6</td>
<td>2.26/2.57</td>
<td>0.33/0.17, 0.24/0.14</td>
</tr>
<tr>
<td>1200</td>
<td>6/5</td>
<td>2.49/1.82</td>
<td>0.30/0.20, 0.20/0.20</td>
</tr>
<tr>
<td>1500</td>
<td>6/5</td>
<td>1.71/1.84</td>
<td>0.28/0.16, 0.18/0.21</td>
</tr>
<tr>
<td>1800</td>
<td>5/5</td>
<td>1.61/1.84</td>
<td>0.25/0.16, 0.18/0.20</td>
</tr>
</tbody>
</table>

**FIG. 14: STEP RESPONSE COMPARISON**

**FIG. 15: CONTROL EFFORT COMPARISON @45PSI 900RPM**

**FIG. 16: SMALL STEP RESPONSE COMPARISON**

**CONCLUSION**

This paper applies integrated system modeling and control design process to a continuously variable valve timing (VVT) actuator system. The experiment conducted serves as the first step in the iterative design process. Constrained by the sample rate of the crank-based cam position sensor (a function of engine speed) and time based control scheme, the actuator control sample rate is different from cam position sensor one.
Due to cam shaft torque load disturbance and high actuator open-loop gain, it is also difficult to maintain the cam phase at the desired level with an open-loop controller. The closed-loop multi-rate system identification is required. Closed-loop system identification using PRBS q-Markov Cover was applied to obtain open-loop system models of a VVT cam actuator system. The proposed closed-loop system identification approach provides models whose time responses are fairly close to bench responses. An output covariance constraint controller was designed based on the identified model and tested on the test bench. The controller has an extra integrator for better response performance. Comparing with PI controller, the multi-input OCC controller uses less energy and has similar closed-loop response time. OCC controller also reduces overshoot up to 50%.

ACKNOWLEDGMENTS
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REFERENCES

APPENDIX A
IDENTIFIED PLANT MODELS

<table>
<thead>
<tr>
<th>Engine Speed(rpm)</th>
<th>Identified Open-loop Plant (4th Order)</th>
<th>Identified Open-loop Plant (2nd Order)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>( G(s) = \frac{3.56 \times 10^{-4} s^4 - 5.27 s^3 + 585.6 s^2 - 1.8 \times 10^4 s + 2.12 \times 10^6}{s^4 + 11.6 s^3 + 2780 s^2 + 3.21 \times 10^5 s + 4.6 \times 10^4} )</td>
<td>( G(s) = \frac{3.56 \times 10^{-4} s^2 - 5.27 s + 592.2}{s^2 + 11.6 s + 16.7} )</td>
</tr>
<tr>
<td>1500</td>
<td>( G(s) = \frac{0.012 s^4 - 3.09 s^3 + 1354 s^2 - 2.88 \times 10^3 s + 9.023 \times 10^6}{s^4 + 14.54 s^3 + 5971 s^2 + 8.54 \times 10^5 s + 2.38 \times 10^5} )</td>
<td>( G(s) = \frac{0.0124 s^2 - 2.04 s + 1582}{s^2 + 16.78 s + 34.82} )</td>
</tr>
</tbody>
</table>