NONLINEAR CONTROL OF HYDRAULIC CAMSHAFT ACTUATORS IN VARIABLE CAM TIMING ENGINES

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Abstract: This study demonstrates, from both mechanical considerations and test data, the presence of a significant nonlinearity in the variable cam timing (VCT) actuators. This motivates the inclusion of a simple nonlinear compensation term in the VCT controller allowing faster valve timing control loops than linear controllers with better robustness in practice. Nonlinear models of hydraulic VCT actuators are identified on a dual-independent VCT engine under closed-loop operation. It is shown by simulation and also verified experimentally that simple nonlinear control schemes achieve superior performance compared to linear PIDs even though the identified nonlinearity is likely to be only a basic approximation of the true behaviour.

Keywords: hydraulic actuators, closed-loop identification, nonlinear control, inversion, PID control

1. INTRODUCTION

VCT engines offer the potential to give superior fuel economy, performance and emission levels of gasoline engines (Leone *et al.*, 1996). Most of the emission benefits are due to the internal EGR mechanism realisable by VCT. Effective use of internal EGR can reduce both NO_x and HC emissions. At the same time, it reduces the pumping loses and improves the fuel economy. Furthermore, VCT permits the optimisation of the cam timing over a wide range of engine operating conditions, providing both good idle quality (minimum overlap) and improved wide-open throttle high speed performance (maximum inducted air charge).

VCT has a substantial effect on the breathing process of the engine. While it offers many advantages, it also causes a significant disturbance to cylinder air flow and air-fuel ratio (AFR) which may result in driveability problems and increased tailpipe emissions. In order to minimise the adverse effects of VCT, mean value models are developed and feedforward and feedback torque and AFR control strategies are proposed in the literature (Gorinevsky *et al.*, 1999; Jankovic *et al.*, 1998; Stefanopoulou *et al.*, 1998).

The VCT mechanism uses electronically controlled hydraulic actuators to move the inlet and exhaust valve timings relative to the crankshaft position. Published research has so far treated the VCT actuators as linear systems and used simple linear controllers for position control (Gorinevsky *et al.*,

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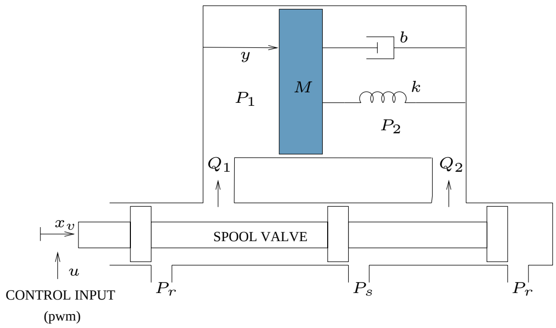

Fig. 1. Hydraulic VCT Actuator Model

1999; Jankovic *et al.*, 1998; Stefanopoulou *et al.*, 1998). On the other hand, the dynamics of hydraulic systems are highly nonlinear (deSilva, 1989).

This paper shows that VCT actuators have significant nonlinearity. The dynamics of the system obtained using the first principles are nonlinear. The test data further suggest that there is an integrator in the system dynamics. The presence of an integrator prevents open-loop operation of the actuators and makes identification a challenging task. It is proposed that the nonlinear dynamics can be captured by a static nonlinearity and therefore the model is simplified. Since many parameters in the model can not be measured directly, they are estimated using identification methods. Once the model is identified and verified, a linear and a nonlinear controller are designed for valve timing tracking. The performances of the controllers are compared during the real engine operation and it is shown by experimental data that the nonlinear controller can achieve better tracking performance.

2. MODELLING OF THE VCT ACTUATORS

The physical model of a VCT actuator is shown in Fig. 1. A spool valve, which is actuated by a pulse-width modulation (pwm) signal, controls the pressure levels P_1 and P_2 through the displacement x_v. The net force acting on the piston, $P_1A_1 - P_2A_2$, determines the displacement y, which alters the valve timing. The piston is connected to one of the walls with a spring and a viscous damping is assumed to be acting on the piston. The objective is to control the displacement y to track a desired valve timing trajectory. The physical equations governing the system can be written as (deSilva, 1989):

$$\ddot{y} = \frac{1}{M}\left(-b\dot{y} - ky + P_1A_1 - P_2A_2\right) \qquad (1)$$

$$\dot{P}_1 = \frac{\beta_e}{V_1}(Q_1 - A_1\dot{y}), \qquad (2)$$

$$\dot{P}_2 = \frac{\beta_e}{V_2}(Q_2 + A_2\dot{y}), \qquad (3)$$

where M is the mass of the piston, b is the damping constant, k is the spring coefficient, β_e is the bulk modulus of the engine oil, A_1 and A_2 are the left and right surface areas of the piston respectively and the volumes V_1 and V_2 are given by

$$\begin{aligned} V_1 &= A_1 y, \\ V_2 &= A_2(y_t - y), \end{aligned} \qquad (4)$$

where y_t is the total distance that the piston can move. The flow rates Q_1 and Q_2 are both functions of the spool position and the cylinder pressures:

$$Q_1 = \begin{cases} x_v C_{f_1}(x_v)\sqrt{\mid P_s - P_1 \mid}, & x_v \geq 0 \\ x_v C_{f_1}(x_v)\sqrt{\mid P_1 - P_r \mid}, & x_v < 0 \end{cases} \qquad (5)$$

$$Q_2 = \begin{cases} -x_v C_{f_2}(x_v)\sqrt{\mid P_2 - P_r \mid}, & x_v \geq 0 \\ -x_v C_{f_2}(x_v)\sqrt{\mid P_s - P_2 \mid}, & x_v < 0 \end{cases} \qquad (6)$$

where $C_{f_{1,2}}(x_v)$ are flow coefficients, P_s and P_r are the source and return pressures respectively. These equations are nonlinear and the parameters are uncertain. Variation of the engine oil temperature and pressure contributes to the uncertainty in the parameters. In addition, there is further complexity and uncertainty in the system due the unmodelled dynamics of the spool valve between the control input and the spool valve position x_v. All these unmodelled dynamics and uncertainty make this system a challenging identification problem. Available signals for identification are the control input (pwm) and the displacement y. The displacement y can only take values between y_{min} and y_{max}. A sample input-output data under detuned closed-loop with a sampling time of $0.01s$ is shown in Fig. 2. The engine speed is kept constant at 1500 rpm and the intake manifold pressure is around $45kPa$ during this measurement. The data suggest that there is an integrator in the system as the control input always converges to the same value at steady-state. This is likely to be due to the negligible effect of the spring on the system. The main downside of having an integrator in the plant is that it makes open-loop identification difficult. Therefore, the identification tests are performed under closed-loop operation. There are difficulties involved with closed-loop identification. First a controller is necessary in the loop . A simple detuned PI controller is used for the identification tests. It is important that the input signal to the system contains most of its

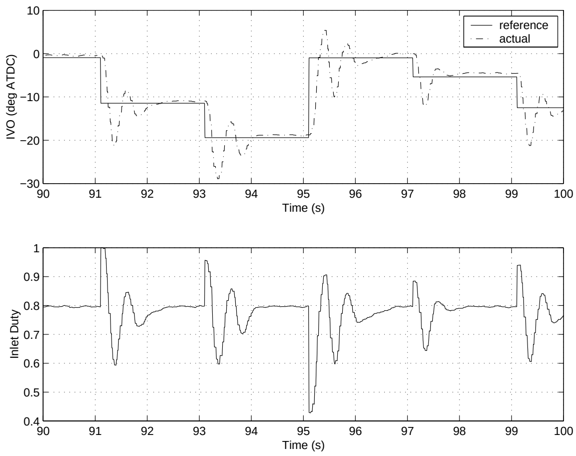

Fig. 2. A sample input-output data

energy in the frequency range of interest. Therefore PI parameters are chosen to excite the system up to the frequency of $20rad/s$. Second, conventional open-loop identification methods may not be successful for closed-loop data. Special techniques for closed-loop identification of linear systems should be employed (Forssell and Ljung, 1999). Finally, the system at hand has only one equilibrium point in terms of its input value. Hence, a linear model identified around this equilibrium point would be a poor representation of the real system, if the nonlinearity present in the system was significant. Alternatively, one can use the physical equations of the system (1)-(6) to identify a model. However, this will result in a high-order nonlinear system with significant uncertainty which would be too complicated for our purposes. One way of simplifying the model is to represent the nonlinear dynamics of the valve flows and pressure states in (2)-(6) with a static nonlinearity of the following form:

$$f(u) = \begin{cases} \overline{u}, & u \geq \overline{u}, \\ a_1(u-u_0) + a_2(u-u_0)^2, & u_0 \leq u < \overline{u}, \\ b_1(u-u_0) + b_2(u-u_0)^2, & \underline{u} \leq u < u_0, \\ \underline{u}, & u < \underline{u}. \end{cases} \tag{7}$$

Saturation of the control input is also considered in the static nonlinearity by adding $\overline{u}$ and $\underline{u}$. To add an integrator to the model, the linear part of the system is assumed to be a second order transfer function with an integrator. Moreover, a closer inspection of the data given in Fig. 2 suggests that there is a transport delay from control input to the plant output. Thus, the final form of the simplified nonlinear model can be written as:

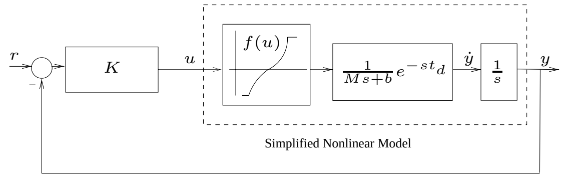

Fig. 3. Identification framework

$$\ddot{y}(t) = \frac{1}{M}\left(-b\dot{y}(t) + f(u(t-t_d))\right), \tag{8}$$

where t_d is the transport delay in seconds. There are in total 10 parameters to be estimated in this simplified nonlinear model. Block diagram representation of the simplified model is given in Fig. 3.

3. IDENTIFICATION OF THE MODEL PARAMETERS

Identifying a transfer function from u to y would seem like the natural choice. However, this choice would try to optimise the steady-state accuracy of the model as well as the transient accuracy. Instead, a transfer function from u to $\dot{y}$ would be identified to optimise the transient behaviour of the model only. Defining $\dot{y} = \tilde{y}$ in (8) and discretising the equation gives:

$$\tilde{y}(k) = \frac{M}{M+bT}\tilde{y}(k-1) + \frac{T}{M+bT}f(u(k-t_d)), \tag{9}$$

where T is the sampling period and t_d can take only integer values. The cost function for identification is chosen as the square of the error between measured output $\tilde{y}_m$ and the model output $\tilde{y}$. The objective of the identification can be written as

$$\min_{Q} \sum_{k=1}^{N} (\tilde{y}(k) - \tilde{y}_m(k))^2, \tag{10}$$

where Q is the vector of unknown parameters in the simplified model. Random step inputs of duration $2s$ are applied as a reference to the setup in Fig. 3 and the input-output data is measured for $100s$ for each test with a sampling time of $0.01s$. Since this is a nonlinear optimisation problem, numerical solvers are used to minimise the cost (10). Good initial conditions are crucial for the success of the numerical optimisation. Therefore, whenever possible, initial values of the model parameters are estimated using other identification methods. For example, the initial value of the transport delay is obtained by calculating impulse response estimate from reference to output which suggests a delay of 5 sampling times. The first $80s$ of the data is used for identification and the last $20s$ of the data is used for validation. A portion of the identification and the validation data for inlet VCT model is given in Fig. 4 (IVO stands for inlet valve opening). The

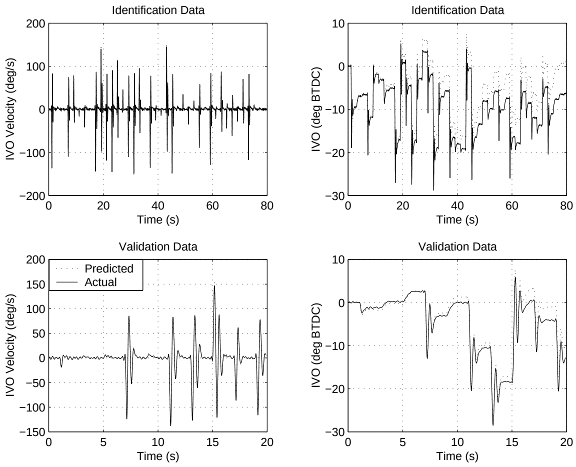

Fig. 4. Inlet VCT identification

fitting of the $\tilde{y}$, IVO velocity in the figure, to the actual data is very good both for identification and validation data. This suggests that the identified model captures the dynamic behaviour with a good accuracy. As expected, however, when the actual IVO angle is considered as the output there is a drift in the model. This represents no problem for control design purposes as model accuracy is crucial at frequencies around the closed-loop bandwidth not at steady-state, and a controller can easily correct such a slow drift. The same identification procedure is applied to exhaust VCT input-output data as well. The identified static nonlinearities are shown

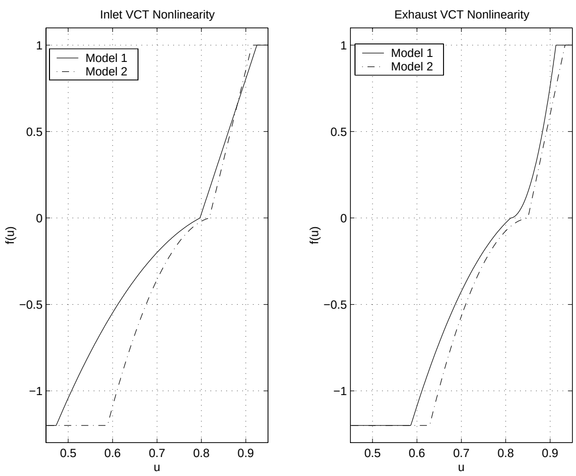

Fig. 5. Identified static nonlinearities at different operating points

in Fig. 5 (model 1). Another set of nonlinearity (model 2 in Fig. 5) is identified using a different data set taken when the engine was very warmed-up for both inlet and exhaust VCT systems. There are a few implications of these results. First, the inlet and exhaust nonlinearities are different. This might be because of their different locations on the engine or some other physical difference in their structure. Second, as the engine warms up, the steady-state value of the control input increases. This might be due to possible increase in the engine oil temperature and pressure, P_r. The results show that model parameters can change as the engine runs. Such change can be considered as the uncertainty present in the model parameters. These two different sets of models can be used to assess the robustness characteristics of the designed controllers by simulations.

4. CONTROLLER DESIGN AND EXPERIMENTAL RESULTS

In this section two different types of controllers are designed. First, a PID controller with the following structure is tuned to give the desired performance:

$$K = k_p + \frac{k_i}{s} + \frac{k_d s}{\frac{s}{\tau} + 1} \tag{11}$$

The inlet VCT model 1 is linearised around $u = 0.82$ and the transport delay is replaced with its first order Padé approximation. The aim of the controller is to track the reference as fast as possible with sufficient gain (GM) and phase margins (PM). Choosing $k_p = -0.0161$, $k_i = -0.0385$, $k_d = -4.47 \times 10^{-4}$, $\tau = 93.49$ gives a GM of 3.35 dB and a PM of 34.9° with a gain cross-over frequency of $18.067 rad/s$. In addition a classical anti-windup scheme is implemented with a gain of 4.5 to prevent integrator windup (Kothare *et al.*, 1994).

A second controller is designed by exploiting the knowledge of the static nonlinearity. The input static nonlinearity is inverted and then a PID controller is designed for the resulting linear plant to give the desired tracking properties. The parameters of the PID with nonlinear inversion (NLI) is chosen as $k_p = -0.06$, $k_i = -0.015$, $k_d = -0.002$, $\tau = 30$ to have a gain cross-over frequency of $8.25 rad/s$. This controller has a GM of 8 dB and a PM of 49° on the linearised plant. Such high robustness margins are required due to the uncertainty involved in the identified nonlinearity. The anti-windup gain is chosen as 0.2.

The controllers designed above are tested on the identified nonlinear model of the VCT mechanism. Since simulations can not offer the final validation of the controllers, only experimental results are presented. Control and data acquisition is performed using the dSPACE suite of rapid prototyping tools. The engine speed is kept constant at around 1500 rpm during the experiments. The PID controllers are discretised with a sampling time of $0.01s$. Measurements are measured in the crank angle domain every 180 degrees. In the following only the performance of the inlet VCT controller is considered for briefness.

In real operation actuators can follow step references with magnitudes up to 40 degrees (such as during tip-ins and gear changes). To have a broad comparison, one relatively small step size and one relatively large step size are applied at the references of the both controllers. The plots that follow are the averaged responses across a series of 4 step reference applications in both directions. When the step size is small and there is little saturation of the actuator as shown in Fig. 6, the linear controller suffers from high levels of overshoot and slow settling time. Whereas, the nonlinear controller has a smaller overshoot and a much faster settling time. Poor performance of the linear PID is due to the low gain of the plant around the steady-state input value u_0. When the control input is away from its steady-state value, i.e. when the plant has high gain , the linear controller can achieve high loop gains and therefore good tracking performance. As the control input gets closer to its steady-state value, i.e. as the plant gain gets smaller, the linear PID can not achieve high loop gains and its tracking performance deteriorates. On the other hand, nonlinear PID can maintain high loop gain all the time due to the inversion of the nonlinearity at the plant input, which is the cause of the change in the plant gain in the model.

The overshoot of the linear PID is significantly reduced for larger demand signals as illustrated in Fig. 7 owing to the anti-windup scheme since the intervention of the anti-windup gain prevents high overshoots. However, the linear controller still suffers from slow settling times as expected. Tracking performance of the nonlinear controller is not affected much by the change in the magnitude of the reference step. It still provides good tracking and fast settling time. Its settling time is approximately 50% faster than the settling time of the linear one. Such a fast response can be crucial for transient control problems present in VCT engines. It was cited in the literature that for adequate torque control during tip-in and tip-out, the time constant for VCT response should match the time response of the manifold filling and emptying dynamics which is in the order of $0.15s$ (Stein *et al.*, 1995). The nonlinear controller can match this requirement with a settling time of $0.4s$ whereas the linear one fails with a settling time of $0.9s$.

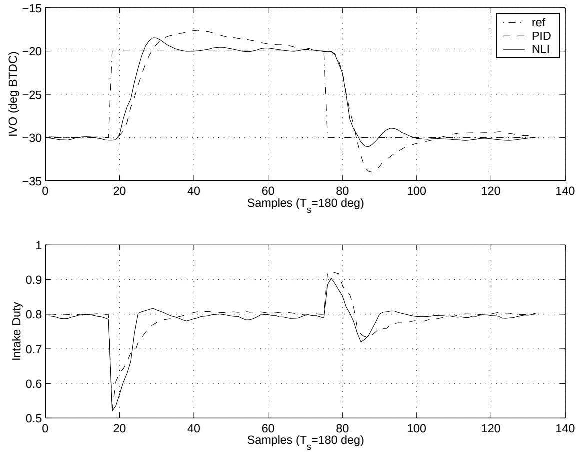

Fig. 6. Tracking performance for a 10° step in reference

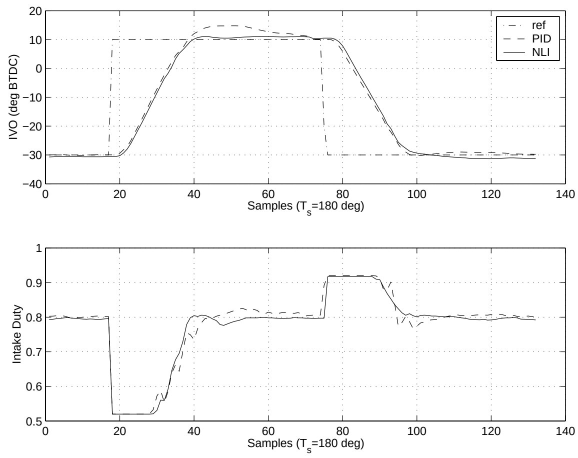

Fig. 7. Tracking performance for a 40° step in reference

5. CONCLUSION

It is shown that a simple nonlinear model can offer a better representation of the VCT actuators. Such a nonlinear model allows design of simple nonlinear controllers by inversion of the nonlinearity which achieve superior performance over their linear counterparts. Experiments indicates that the overshoot in the response can be reduced significantly and the speed of the response can be increased by more than 50% with the nonlinear controller.

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