Diesel Engine Control Using Parameter Varying Methods

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Abstract: Traditionally, engine speed regulation is achieved using classical gain-scheduled PID control to address the variable operating conditions of the engine. However, this approach provides no guarantees for closed-loop system stability or performance. In this work, a model-based linear parameter varying (LPV) approach is applied to address the fast operating condition changes of the engine and to guarantee system stability and optimized torque load rejection in the presence of variable transport delays and fuel saturation constraints. The design method is formulated in terms of linear matrix inequalities (LMIs) that can be solved efficiently using recently developed interior point convex optimization algorithms. The proposed control designs are validated using both nonlinear off-line simulations and real-time hardware-in-the-loop (HIL) implementation for a Cummins diesel engine model.

Key-Words: - Engine control, Linear Parameter Varying Control, Optimization

1 Introduction

The operation of modern internal combustion engines for automotive use must satisfy a diverse set of conflicting constraints imposed by performance, economical and environmental factors, such as, maintaining good transient and steady-state performance, providing increased fuel economy, and minimizing exhaust emissions. Based on the dramatic evolution in microprocessor capabilities, control technology can be a driving force to guarantee these objectives [2]. However, effective engine control is a challenging problem since these systems exhibit nonlinear behavior, multiple operating conditions and variable time-delays in the feedback loop.

The engine speed control problem consists of regulating the engine speed to follow a desired reference trajectory despite the presence of torque load disturbances due to road slope changes, wind and accessory loads (power steering pump, transmission shifts, charging system, air conditioner compressor engagements, etc.). The reference speed is determined by the throttle position (driver pedal) and the speed controller regulates the fuel amount injected to the engine. Engine speed control has significant impact in many important vehicle attributes, such as, performance, fuel economy, and emissions [9].

Most prior work on engine speed regulation has been concentrated on idle speed control (ISC). Methods based on linearization about the idle operating condition have been used extensively in combination with classical and modern model-based and non-model-based control design schemes. Traditional ISC strategies are based on a PI or PID control action where the integral portion ensures that the engine speed tracks the desired reference speed. Recently, Linear Quadratic approaches [16], H_∞ control, l_1 control [4], and QFT methodologies [13] have been proposed for ISC, along with nonlinear schemes based on sliding mode [7],[5] and adaptive control methods [14]. A detailed survey of different ISC approaches is presented in [12].

To address the speed regulation problem for the full operating envelope of the engine, a standard approach is to tune or schedule controllers that have been designed at different operating points. However, this scheduling approach provides no guarantees of stability or performance in case of fast operating condition changes. In addition, a challenging issue in the engine speed control problem is the presence of variable time delays in the feedback loop. These delays are combinations of fueling and transport delays and should not be neglected since they often constitute the dominant dynamics of the resulting closed loop system. The variability of the delays is nonlinear with larger delays occurring at low engine speeds.

In this work, we focus on a diesel engine speed control problem for the entire operating envelope of the engine. We seek to design controllers that provide reference speed tracking despite the presence of torque load disturbances, operating condition changes and variable time delays in the feedback loop. The proposed control design scheme is based on systematic gain-scheduling using a Linear Parameter Varying (LPV) methodology [15],[3],[1]. LPV systems are systems that depend on unknown but measurable time-varying parameters, such that the measurement of these parameters provides real-time information on the variations of the plant's characteristics. LPV controllers are parameter-dependent controllers that are scheduled or adapted in real-time based on this measurement information. The analysis and control of LPV systems has received significant attention recently to provide a systematic gain-scheduling control approach for nonlinear systems [**?**]. The LPV analysis and control synthesis problems can be formulated as linear matrix inequality (LMI) constraints that can be solved efficiently using recently developed interior-point optimization algorithms [6],[17]. It is emphasized that the LPV control design approach, in contrast to the traditional gain scheduling approach, provides guarantees of stability and performance over the full operating range of the system.

In our proposed approach, we formulate the engine speed control problem as an L_2 gain optimization problem for a simplified LPV engine model. The model contains a speed-dependent engine friction and a speed-dependent time delay. The variable delay is approximated using a first-order Pade approximation. In order to obtain zero steady state error, an integrator is appended to the engine speed error signal in the design model. The designed dynamic controller is scheduled based on the real-time measurement of the engine speed. The performance of the proposed speed controller is validated using a detailed nonlinear dynamic engine model provided by the Cummins Engine Company. The controller is implemented and tested using a hardware-in-the-loop (HIL) configuration.

2 Simplified Engine Model

We start our analysis with the development of a simplified nonlinear engine model for speed regulation. The engine speed ω is governed by the rotational dynamics of the crankshaft as follows:

$$J\frac{d\omega}{dt} = T_e - T_f - T_d \qquad (1)$$

where J is the effective moment of inertia of the engine, ω is the engine speed, T_e is the engine generated torque, T_f is the friction torque, and T_d is the disturbance torque load. The engine friction torque T_f is a nonlinear function of the engine speed, and for the engine of interest, this function can be curve-fitted from experimental data as follows:

$$T_f = \frac{\omega^2}{\alpha\omega^2 + \beta\omega\sqrt{\omega} + \gamma} \quad \text{(ft-lb)} \qquad (2)$$

where $\alpha = -0.0106$, $\beta = 0.71248$ and $\gamma = -1680.5$ and ω is the engine speed in rpm. Figure 1 shows the curve-fitted friction torque as a function of the engine speed.

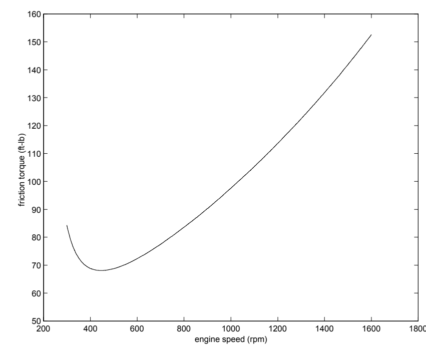

Figure 1: Engine friction torque as a function of engine speed.

The engine generated torque T_e is assumed to be proportional to the delayed fueling input, that is,

$$T_e(t) = K_1 f(t - \tau) \qquad (3)$$

where for the engine of interest $K_1 = 5.102$ and the fueling input f has a variable transport delay as follows

$$\tau(\omega) = 0.0029996 + \frac{29.166}{\omega} \quad \text{(sec)} \qquad (4)$$

Figure 2 shows the variability of this input delay as a function of the engine speed. Fuel input is limited by the actuation saturation which is assumed to have a limit of 300 $mm^3/stroke$.

Based on the above formulation, the simplified nonlinear engine speed model is given by the SIMULINK diagram in Figures 3 and 4 that contains the engine inertia dynamics, the variable fueling delay and the fuel input saturation. This model has as inputs the fueling and the torque load, and provides as output the engine speed. In this model, the engine idle speed has been set to 650 rpm and K_2 is the inverse value of the engine inertia J. The engine inertia dynamics subsystem is given by the SIMULINK

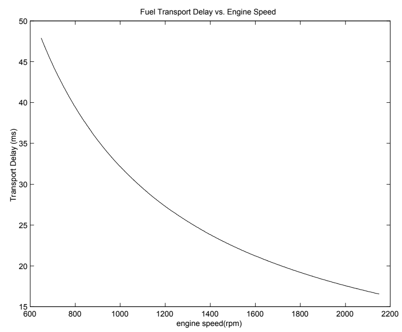

Figure 2: Fueling delay as a function of the engine speed

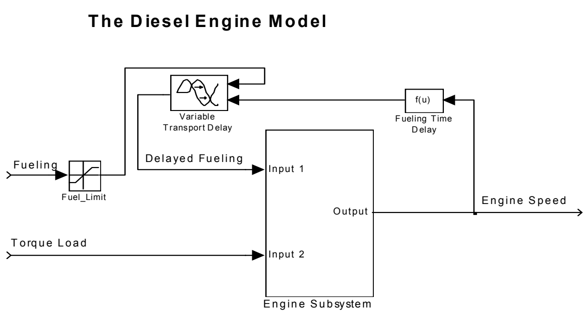

Figure 3: Diesel Engine Speed SIMULINK Model

block in Figure 4 and it contains the speed-dependent friction characteristics.

To obtain a rational expression for the delay term, we use the following first-order Pade approximation

$$e^{-s\tau(\omega)} \backsimeq \frac{1}{1+\tau(\omega)s} \tag{5}$$

that does not introduce a right-half-plane zero. The above formulation results in a second-order nonlinear dynamic model of the engine speed characteristics that will be used for parameter dependent control design.

3 Control Design Objectives and Control Strategy

The engine speed regulation system should satisfy a wide range of conflicting objectives as follows:

- Tracking performance: The engine should track the reference speed as quick as possible and the steady state error to a step input should be to minimal.

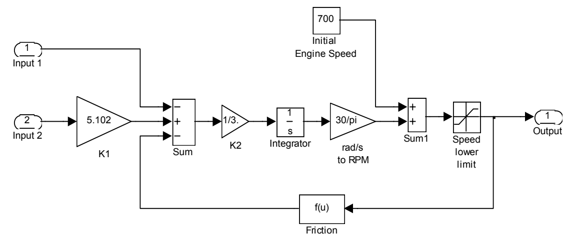

Figure 4: Diesel Engine Inertia Dynamics Subsystems

- Transient response: The engine speed percentage overshoot to a fueling step input should be small.
- Disturbance rejection: The engine speed control system should provide disturbance rejection to uncertain torque load disturbances. The maximal speed drop to a step load disturbance should be minimized and the engine speed should return back to its reference value quickly.
- Fuel economy: The fuel consumption should be minimized.

A Linear Parameter Varying (LPV) control strategy is implemented in this work to address the variability of the engine dynamics as a function of the engine speed. LPV systems are described by linear state-space equations with state matrices that depend on a time-varying parameter vector $\rho(t)$. It is assumed that the parameter vector $\rho(t)$ can be measured in real time during system operation and the control strategy can exploit this measurement to increase performance. Hence the LPV controller is self-scheduled based on $\rho(t)$ to adjust the control dynamics to the variations in the plant dynamics [15],[3],[1]. In this context, LPV control design provides a systematic methodology for gain scheduling. However, in contrast to traditional gain scheduling, the LPV control design provides guarantees of stability and performance along all possible trajectories of $\rho(t)$. For the case of LPV systems with state matrices that depend affinely on the parameter vector $\rho(t)$ the synthesis problem for L_2 gain disturbance rejection (H_∞ control problem) with a single quadratic Lyapunov function is reduced to solving a system of Linear Matrix Inequalities (LMIs) [15],[3]. This computational problem can be attacked efficiently and reliably using recently developed interior point convex optimization algorithms [6].

In this work, an H_∞ LPV controller is considered to provide disturbance rejection in the presence of unknown torque loads. The objective of the H_∞ controller is to minimize the effect of the torque disturbance on the speed error, that is, on the difference between the measured engine speed and the reference speed. The proposed LPV controller is self-scheduled based on the engine speed measurement, i.e., we are seeking a dynamic LPV controller of the form

$$\begin{aligned} \dot{x}_c &= A_c(\omega(t))x_c + B_c(\omega(t))z \\ f &= A_c(\omega(t))x_c + B_c(\omega(t))z \end{aligned}$$

where $x_c(t)$ is the controller state vector, $z(t)$ is the measurement vector, $f(t)$ is the engine fuel input and A_c, B_c, C_c and D_c are the controller state matrices that vary based on the real-time measurement of the engine speed $\omega(t)$.

To satisfy the desired engine speed regulation objectives the following control strategy is implemented:

- To eliminate the steady-state tracking error, the engine system type is increased by one. A proportional-integral (PI) term is added to the engine dynamics at the fueling command input. The PI gains are constants to be tuned by the designer.
- For fuel economy, the output vector of the engine system is augmented to include the speed error and the fuel input. Hence, fuel is penalized in the control design
- To minimize the engine speed overshoot caused by integrator windup due to fuel actuator saturation, a traditional anti-windup strategy is implemented [11]. That is, an extra feedback path is added after the control design to measure the error between the actuator output and the control output, and feeding this signal back to the integrator through a constant gain. Hence, the error signal is zero when the actuator is not saturated.
- To provide a trade-off between control effort and performance, constant weighting variables are introduced in the design.

Based on the above discussion the SIMULINK block of the augmented engine model that is used for the LPV design is show in Figure 5.

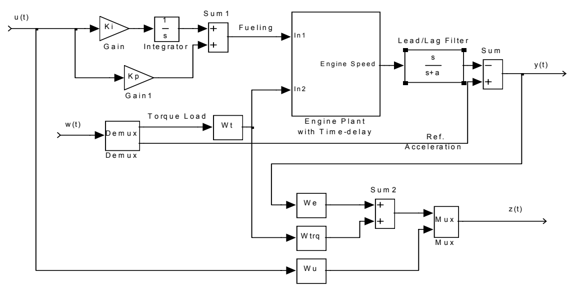

Figure 5: Augmented SIMULINK Engine Model

4 Control Design and Validation

The LPV controller is designed to optimize L_2-gain quadratic performance over all parameter trajectories based on the quadratic LPV control theory. The LMI Control Toolbox is used for control synthesis [8]. For the real time implementation, the obtained LPV controller is discretized with a sampling rate of 50 Hz. The LPV control structure augmented with the PI block and an anti-windup control scheme to address fuel saturation is shown in the SIMULINK diagram in Figure 6.

Satisfactory closed-loop performance requires a step-input overshoot percentage below 3 %, settling time below 5 seconds, steady-state error percentage below 1 %, steady-state fuel oscillation below 7 mm^3/stroke and a maximal speed drop to a 50% load (800 ft-lb) below 5 %. A nonlinear closed-loop simulation and a hardware-in-the-loop (HIL) controller implementation are used to validate that the desired closed-loop characteristics are met. It is noted that traditional gain-scheduled PID control designs currently used for control do not meet the above design specifications.

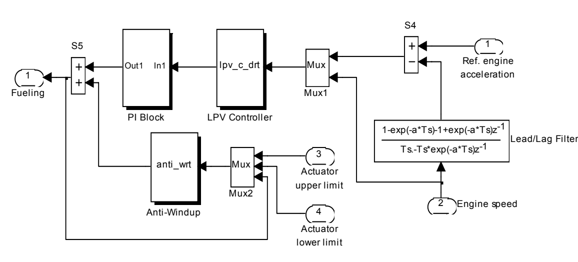

Figure 6: Augmented LPV Controller

Nonlinear Simulation: The off-line simulation of the closed-loop system is performed with a detailed nonlinear engine speed dynamics SIMULINK model provided by Cummins Engine Company. In this simulation, the fueling is selected as input and the engine speed is selected as output of the plant. The external torque load input is selected as a disturbance to the plant. The external torque load was selected as a typical load of a step change from 100 ft-lb to 800 ft-lb. A low-pass filter dynamics $\frac{1}{1.5s+1}$ is added in the torque load input side to simulate a realistic torque load profile. This torque profile is shown in Figure 7. The reference engine speed was selected as a step increase from 1200 rpm to 1800 rpm and then a drop to 1000 rpm as shown in Figure 7. To simulate a realistic operation and test the disturbance rejection of the LPV controller, a $\pm$10 ft-lb band-limited white noise torque disturbance is added to the external torque load for control simulation purposes. Two design cases with different values for the PI gains and the control design weights are selected for the simulation.

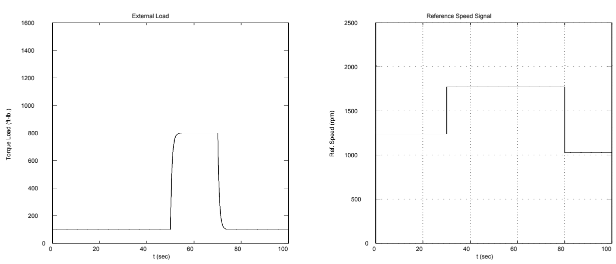

Figure 7: Torque toad and reference engine speed for control simulation

4.0.1 Design Case 1

The following PI gains and weight values are used: $k_i = 0.5,\ k_p = 0.3,\ w_u = 30,\ w_e = 10,\ w_{trq} = 7,\ w_t = 1$. The corresponding closed-loop fueling and engine speed responses are shown in Figure 8. Table 2 shows the closed-loop performance characteristics obtained with this design.

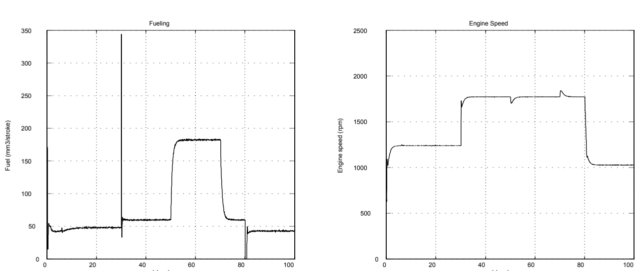

Figure 8: Fueling command and actual engine speed for Design I

Overshoot %	Settle Time	S. S. Error %
0 %	< 3 sec	about 0%

S. S. Fuel Oscillation	Max. Speed Drop
$< 2.5\ mm^3/stroke$	$70/1770\ rpm$

Table 2: Closed-loop system performance for Design I

We observe that the engine speed follows quite closely the desired reference speed. The performance characteristics are well below the desired ones.

4.0.2 Design Case 2

The following PI gains and weight values are used: $k_i = 0.5,\ k_p = 0.3,\ w_u = 30,\ w_e = 10,\ w_{trq} = 6,\ w_t = 1.2$. The corresponding fueling and engine speed response are shown in Figure 9. Table 3 shows the closed-loop performance characteristics obtained with this design.

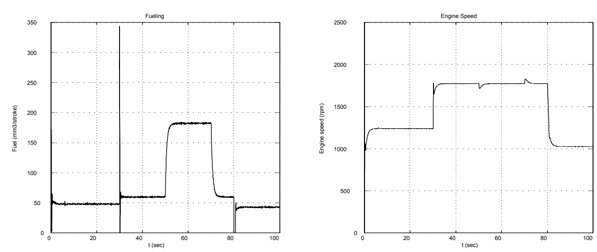

Figure 9: Fueling command and actual engine speed for Design 2

Overshoot %	Settle Time	S. S. Error %
0 %	< 3 sec	about 0%

S. S. Fuel Oscillation	Max. Speed Drop
$< 3\ mm^3/stroke$	$55/1770\ rpm$

Table 3: Closed-loop system performance for Design 2

From the above closed-loop simulations we see that the performance in terms of overshoot percentage, settling time, steady-state speed error and steady-state fuel oscillations all exceed our design requirement. As we can see, the PI gains and weights allow us to provide a trade-off between the performance characteristics, such as, maximum speed drop versus steady-state fuel oscillations.

Hardware-in-The-Loop Implementation: Real-time hardware-in-the-loop (HIL) simulation has been recently adopted by the automotive industry to test and validate control laws implemented in the engine control unit (ECU). In HIL, the engine is replaced by a detailed real-time simulation of its behavior that is coupled with the actual ECU hardware where the control law is programmed [10]. This configuration allows rapid testing and redesign of control laws. To this end, the proposed LPV control configuration has been implemented at the Applied Dynamics International (ADI) Real Time Station (RTS) at Cummins Engine Company for HIL simulation.

Compared with the off-line simulation, the overall fueling time delay in the ADI RTS implementation is higher by 40 ms due to the extra datalink communication time delay. The torque load disturbance applied to the engine model is the ideal step input as shown in Figure 10 without a low-pass filter in the torque load input side, since no such filtered implementation is allowed in this configuration. Using the same LPV controllers designed above, the system responses are shown in Figures 11 and 12. The performance results are shown in Tables 3 and 4.

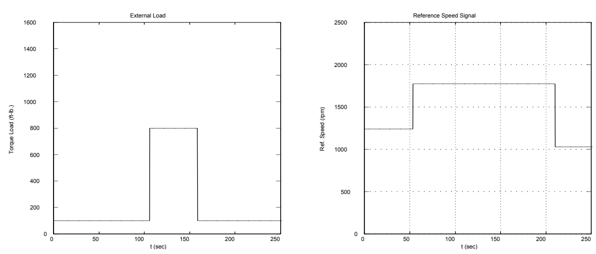

Figure 10: Torque load and reference engine speed for real-time simulation

4.0.3 Design 1

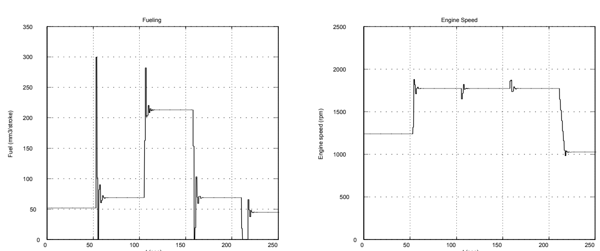

Figure 11 Fueling command and actual engine speed for Design 1

Overshoot %	**Settle Time**	**S. S. Error %**
6 %	$<$ 4 sec	about 0%

S. S. Fuel Oscillation	**Max. Speed Drop**
0 $mm^3/stroke$	123/1770 rpm

Table 4 Closed-loop system performance for Design I

4.0.4 Design 2

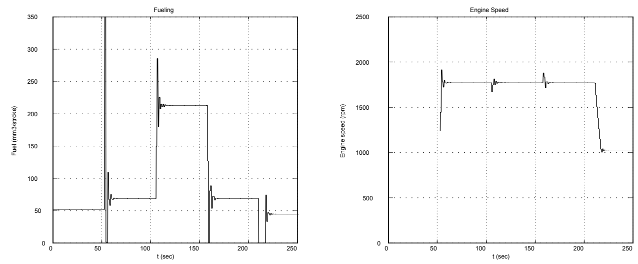

Figure 12 Fueling command and actual engine speed for Design 2

Overshoot %	**Settle Time**	**S. S. Error %**
7.1 %	$<$ 4 sec	about 0%

S. S. Fuel Oscillation	**Max. Speed Drop**
0 $mm^3/stroke$	102/1770 rpm

Table 5 Closed-loop system performance for Design I

We observe that the engine speed follows closely the desired reference speed. The engine speed percentage overshoot in this real-time simulation is above the desired bound of 3%. However, this response corresponds to the ideal torque load step input shown in Figure 10 instead of the filtered one shown in Figure 7 that appears in a realistic situation. In summary, the response of the real-time simulation is consistent with the off-line nonlinear simulation and demonstrates the validity of the proposed control designs.

5 Conclusions

A systematic LPV methodology with guaranteed L_2 gain quadratic performance is applied for engine speed regulation in a diesel engine model. The proposed controller is scheduled based on real-time engine speed measurements, and it provides guaranteed torque load disturbance rejection despite operating condition changes and variable fueling delays. A detailed nonlinear closed-loop simulation and a hardware-in-the-loop implementation are used to demonstrate the controller effectiveness. The proposed LPV methods can be extended to address multivariable engine control problems such as air handling control in diesel engines. Such methods are currently under investigation.

6 Acknowledgment

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