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Electronic Throttle and Wastegate Control for Turbocharged Gasoline Engines

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Abstract— **Turbocharging of gasoline engines has gained renewed popularity as a means to improve fuel economy and CO_2 emissions. Modern engines have advanced technology actuators such as electronic throttle and variable valve timing, in addition to wastegate. Proper control of system actuators is critical to achieving the full benefit of this technology. This paper presents system analysis and control development for a turbocharged gasoline engine equipped with wastegate and electronic throttle. A nonlinear mean value model for the system is presented. The nonlinear model is linearized at multiple operating points and its behavior analyzed using the systems approach. A gain scheduled, decentralized controller is designed to track the intake manifold and boost pressures to meet transient, as well as steady state requirements. A multi-variable control approach is also considered to alleviate tradeoffs inherent in decentralized design.**

I. INTRODUCTION

The ever increasing government regulation of emissions and driver demands for fuel economy and drivability emphasize the need for advanced engine technology and control. Boosting of engine intake pressure is being proposed as a possible solution to reducing CO_2 emission levels and improving the fuel economy of the engine. The torque developed by a conventional gasoline engine is proportional to the air that is supplied to the cylinders. The density of air entering a boosted engine is higher than that for a naturally aspirated engine, hence for the same maximum power, a boosted engine is smaller in size. In automotive applications, operating conditions vary within a wide range, which can lead to inadequate boost at low speed and loads, while creating an over-boost situation at high speed and loads [1]. Therefore, the amount of boost delivered by a turbocharger is typically controlled by a wastegate. Thus the advantages of boosting are accompanied by an increase in complexity of the control design and calibration.

Computer aided control system design (CACSD) methods, which rely on control oriented models, are well suited to this type of problem. Applicable models of turbocharged gasoline systems are well documented in [2] and [3], as are component based models such as the turbocharger model presented in [4]. Literature that pertains to the control of wastegate in gasoline applications, such as [5], [6] and [7], refers to systems with mechanical throttle and provides some insight into the problem.

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This paper deals with control of a fixed geometry turbocharged gasoline engine equipped with a wastegate and electronic throttle[1]. As illustrated in Figure 1, a turbine along with a wastegate is located in the exhaust path of the engine. The wastegate controls the exhaust gases flowing to the turbine by diverting part of the flow. The energy extracted by the turbine from the exhaust gases drives the compressor via the turbocharger shaft to boost ambient air. The compressed air is passed through an intercooler to reduce the temperature thereby increasing the density of the charge entering the engine and reducing the possibility of knock. The electronically controlled throttle regulates the flow of this charge into the intake manifold. For a conventional gasoline engine, fuel is injected such that stoichiometry is maintained at the engine exit. The torque delivered by the engine is therefore highly dependent on the cylinder flow.

The turbocharged engine is expected to deliver desired torque to the crankshaft, while satisfying demands for driveability and fuel economy, subject to emission constraints. Improper choice of controller structure or parameters, however, can lead to undesirable torque response. A controller with minimal structural complexity is required for ease of implementation and calibration.

A model-based analysis and control approach is adopted in this paper to develop a gain scheduled linear controller. A control oriented nonlinear model is developed in section II. The control problem is defined in section III, with open loop system characteristics presented in section IV. A decentralized controller is proposed in section V. In section VI, a multi-variable control design approach is used to design and analyze various controller structures. The paper concludes with a summary and recommendations for future research in section VII.

II. TURBOCHARGED GASOLINE ENGINE MODEL

A control oriented model of the system shown in Figure 1 is presented in this section. The components included in the model are the intercooler, engine, throttle, wastegate, turbine, compressor and various connecting volumes. Representations of exhaust gas emissions are not available.

Since the dynamics under consideration act on a timescale that is long compared to a cycle duration, a mean value model, [2], [3], is used here. In addition, the transport delays and injection to torque delays are ignored.[2] Furthermore, ideal gas properties are assumed in

[1] Electronic throttle allows the throttle opening to be controlled independent of the driver's pedal position.

[2] The model structure allows these delays to be included to provide a more accurate model.

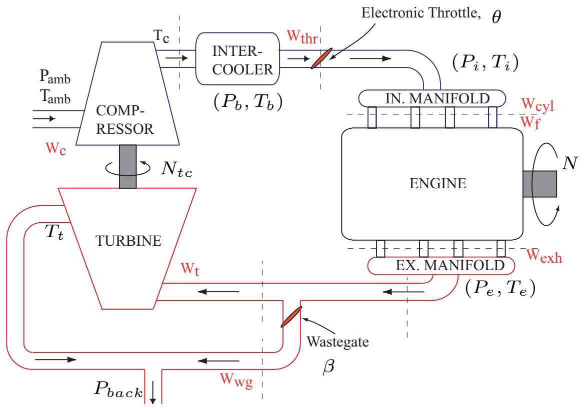


Fig. 1. Schematic of turbocharged gasoline engine

all thermodynamic equations.

A. Intercooler

The gas temperature at the exit of the intercooler, T_b, is given by

$$T_b = T_c - \epsilon\ (T_c - T^{in}_{coolant}), \tag{1}$$

where T_c is the temperature of compressed air at the exit of the compressor, ϵ is the effectiveness of the intercooler, and $T^{in}_{coolant}$ is the coolant temperature. A static model of the effectiveness, as described in [3], is used here.

$$\epsilon = f\left(\frac{W_c}{W_{coolant}}, T_c, T_{coolant}\right), \tag{2}$$

where W_c is the mass flow rate from the compressor and $W_{coolant}$ is the mass flow rate of the coolant.

This model does not account for a pressure drop in the intercooler. The effects of filling and emptying of the intercooler volume, however, are accounted for within the boost manifold model.

B. Connecting Volumes

The model contains four connecting volumes: 1) The boost manifold extends from the compressor exit to the throttle and its thermodynamics states are defined as boost pressure and temperature, P_b and T_b. 2) The intake manifold extends from the throttle to the engine inlet and its thermodynamics states are defined as intake pressure and temperature, P_i and T_i, where it is assumed that $T_i = T_b$. 3) The exhaust volume extends from the engine exit to the turbine inlet and its thermodynamics states are defined as exhaust pressure and temperature, P_e and T_e, where T_e is assumed to equal the engine exit temperature. 4) The system exit volume is the volume downstream of the turbine and the wastegate, where pressure is defined as the system exit pressure, P_{back}.

An isothermal model is used to represent the emptying and filling dynamics of the boost, intake and exhaust manifolds.

$$\dot{P}_{manifold} = \frac{RT_{manifold}}{V_{manifold}}(W_{in} - W_{out}), \tag{3}$$

where $P_{manifold}$ and $T_{manifold}$ are the thermodynamic states of the gas in the manifold; $V_{manifold}$ is the volume of the manifold; R is the ideal gas constant; γ is the specific heat ratio and W_{in} and W_{out} are flows into and out of the manifold, respectively.

P_{back} is given as a function of the engine exhaust flow, W_{exh}, and ambient pressure, P_{amb}, as

$$P_{back} = f(W_{exh}) + P_{amb}. \tag{4}$$

C. Mass Flows

The flows through the throttle and wastegate valve are represented by the standard orifice model. If P_1 and P_2 are the pressures before and after an orifice, respectively, then the mass flow through the orifice, W, is given by

$$W = C_D A \frac{P_1}{\sqrt{RT}}\phi \tag{5}$$

$$\phi = \begin{cases} \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma-1}\left(1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)} & \frac{P_2}{P_1} > 0.528 \\ \sqrt{\gamma\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} & \frac{P_2}{P_1} \leq 0.528 \end{cases},$$

where $C_D A$ is the collective value of the area of the opening of the orifice and the coefficient of discharge. $C_D A_{thr}$ varies with the throttle opening, θ, and $C_D A_{wg}$ with wastegate valve lift, β. The temperature of gas flowing through the orifice, T, is assumed to equal the temperature of the upstream manifold.

The flow of charge into the cylinder, W_{cyl}, is modelled using the speed density equation [8]. A correction term in the form of scaling factor ω_b is used to account for the effects of boost. The resultant model is described as

$$W_{cyl} = (\omega_{offset}(N) + \omega_{scale}(N)P_i)\,\omega_b\left(T_i, N, \frac{P_i}{P_e}\right), \tag{6}$$

where N is engine speed.

Fuel flow, W_f, is governed by the requirement for stoichiometry typical of conventional gasoline engines. It is given by

$$W_f = \frac{W_{cyl}}{AFR_{stoic}}, \tag{7}$$

where AFR_{stoic} is the air-fuel ratio at stoichiometry.

The flow out of the engine, W_{exh}, is given by

$$W_{exh} = W_{cyl} + W_f. \tag{8}$$

D. Engine

The torque model utilized here is described in detail in [8]. The brake torque, TQ_b, is given as

$$TQ_b = W_f\ f(P_i, N) - TQ_{fric} \tag{9}$$

$$TQ_{fric} = f(P_i, P_e, N), \tag{10}$$

where the friction torque, TQ_{fric}, is composed of pumping and mechanical friction torques. It is assumed here that the spark timing is perfectly controlled to provide maximum brake torque, therefore torque dependence on spark is not included.

The dynamics of engine shaft speed, N, are given by

$$\dot{N} = \frac{TQ_b - TQ_{load}}{J_{engine}}, \tag{11}$$

where J_{engine} is the inertia of the engine and TQ_{load} is the external load torque.

The engine exhaust temperature, T_e, is given by

$$T_e = f(N, W_f). \tag{12}$$

E. Turbocharger

The compressor and turbine are represented using the model structures presented in detail in [4]. Therefore, only the primary dependencies are provided here.

For this model, ambient conditions (P_{amb}, T_{amb}) are assumed at the inlet of the compressor, and the compressor exit pressure is assumed equal to the boost pressure, P_b. The mass flow rate through the compressor, W_c, is given by

$$W_c = f\left(\frac{P_b}{P_{amb}}, N_{tc}, T_{amb}\right), \tag{13}$$

where N_{tc} is the turbocharger shaft speed. The compressor exit temperature can be calculated as

$$\begin{aligned} T_c &= T_{amb}\left[1 + \frac{1}{\eta_c}\left(\left(\frac{P_b}{P_{amb}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)\right] && (14) \\ \eta_c &= f\left(\frac{P_b}{P_{amb}}, N_{tc}, T_{amb}\right), && (15) \end{aligned}$$

where η_c is the isentropic efficiency of the compressor.

The inlet conditions of the turbine are assumed equal to the exhaust manifold conditions (P_e, T_e). The turbine exit pressure is assumed equal to the system exit pressure, P_{back}. The mass flow through the turbine is modelled as

$$W_t = \frac{P_e}{\sqrt{T_e}} f\left(\frac{P_{back}}{P_e}, \frac{N_{tc}}{\sqrt{T_e}}\right). \tag{16}$$

The turbine exit temperature, T_t, is given by

$$\begin{aligned} T_t &= \left[1 - \left[1 - \left(\frac{P_{back}}{P_e}\right)^{\frac{\gamma-1}{\gamma}}\right]\eta_t\right] T_e && (17) \\ \eta_t &= f\left(\frac{P_{back}}{P_e}, \frac{N_{tc}}{\sqrt{T_e}}\right), && (18) \end{aligned}$$

where η_t is the isentropic efficiency of the turbine.

The power consumed by the compressor, $Power_C$, and the power generated by the turbine, $Power_T$, are calculated from the first law of thermodynamics, as

$$\begin{aligned} Power_C &= c_{p,c} W_c (T_c - T_{amb}) && (19) \\ Power_T &= c_{p,t} W_t (T_e - T_t), && (20) \end{aligned}$$

where $c_{p,c}$ and $c_{p,t}$ are the specific heats at constant pressure of the gases in the compressor and turbine, respectively.

The dynamics of the turbocharger shaft are given by

$$\dot{N}_{tc} = \frac{Power_T - Power_C}{J_{tc} N_{tc}}, \tag{21}$$

where J_{tc} is the inertia of the turbocharger.

III. CONTROL PROBLEM FORMULATION

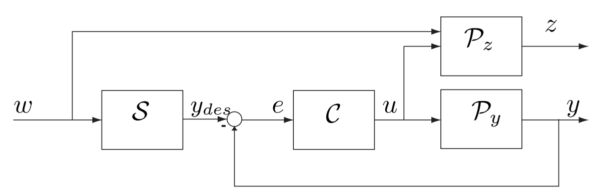
The mean value model of the turbocharged system developed in section II has five dynamical states, namely boost pressure, P_b, intake manifold pressure, P_i, exhaust manifold pressure, P_e, turbocharger speed, N_{tc}, and engine speed, N. The inputs to the system are throttle angle, θ, and wastegate valve lift, β. The primary objective of this system is to deliver desired brake torque, TQ_b, at best fuel economy, subject to emissions constraints. Due to the absence of emissions models, these constraints are not considered in this analysis. To meet the torque objective, a feedback controller is desired to enhance the response of the turbocharged system.

A formulation of this control problem is summarized in Figure 2. The engine speed is a slowly varying state which is measured. Hence, the engine speed and the desired torque, TQ_{des}, can be considered exogenous inputs and are mapped into desired intake and boost pressures using a nonlinear map, $\mathcal{S}$. These two pressures, for which measurements are available, are tracked to attain the desired steady state operating condition. The performance variable for this analysis is brake torque.

The four state model in Figure 2 is linearized to produce

$$\dot{x} = Ax + Bu, \quad y = Cx + Du. \tag{22}$$

This linear model is used for system analysis and development of various controllers for the turbocharged system. The five state nonlinear model is used for analysis and implementation to capture the effects of changing engine speed, as well as nonlinearities.



States	$x = \begin{bmatrix} P_i & P_b & P_e & N_{tc} \end{bmatrix}^T$
Control Inputs	$u = \begin{bmatrix} \theta & \beta \end{bmatrix}^T$
Output Variables	$y = \begin{bmatrix} P_i & P_b \end{bmatrix}^T$
Exogenous Inputs	$w = \begin{bmatrix} N & TQ_{des} \end{bmatrix}^T$
Performance Variables	$z = \begin{bmatrix} TQ_b \end{bmatrix}^T$

Fig. 2. Control problem formulation

Transient performance criteria are derived from the effects of engine torque on the drivability of the vehicle. The dynamics of the system, primarily due to the turbocharger inertia, cause a lag in the intake pressure response and therefore the brake torque. Authors in [9] refer to a "load acceptance test", which characterizes the ability of the engine to increase torque at a constant engine speed. The speed of response is judged via comparison of the closed loop response of the intake manifold pressure to the fastest intake pressure rise that can ideally be achieved with a wide open throttle and closed wastegate. Overshoot of the intake manifold pressure is also considered. These concepts are illustrated in Figure 3.

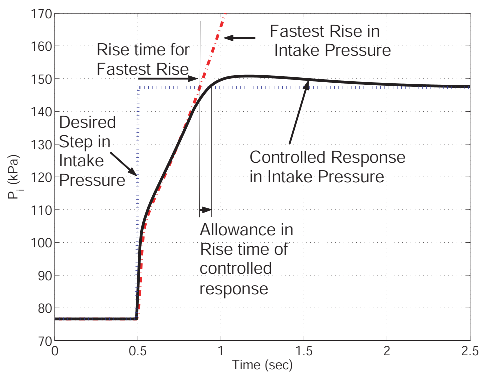


Fig. 3. Transient response specifications are quantified in terms of intake manifold pressure.

IV. Open Loop System Characteristics

Insight into system behavior is critical to developing a suitable controller simple enough for feasible implementation in an automotive system.

First consider the response to throttle, whose behavior is very similar to the behavior of typical naturally aspirated gasoline engines. Torque and pressures exhibit a positive steady state gain to throttle opening. This applies to boost pressure, P_b, as well as intake pressure, P_i, and exhaust pressure, P_e, due to the natural feedback present in the system via the turbocharger.

Now focus on the wastegate. Figure 4 shows steady state values of the boost pressure at various wastegate positions for constant throttle opening and engine speed. The DC gain from wastegate lift to boost pressure, given by the slope of the curves shown in Figure 4, is negative and its absolute value increases with engine speed. In addition, the DC gain approaches zero when the wastegate lift is large, indicating the wastegate has lost authority at these conditions.

When considering transient behavior, the torque response to a change in wastegate valve lift is non-minimum phase. Wastegate movement causes a rapid change in exhaust pressure, which affects the cylinder flow via two phenomena. The first is through the formation of an adverse pressure gradient across the engine, see equation (6). The second is due to the effect on P_i via boosting and is governed by the dynamics of the turbocharger. The two phenomena have opposing effects and since the slower phenomenon related to the turbocharger is dominant in steady state, cylinder flow, hence torque, exhibits a non-minimum phase behavior in response to wastegate. The character of the system dynamics shows little variation with operating condition. The magnitude of the response, however, does vary significantly with engine speed and load.

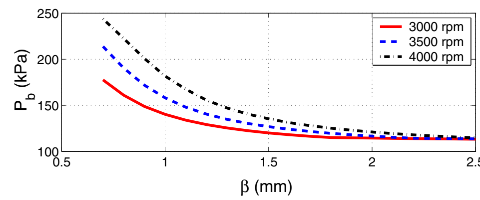


Fig. 4. The steady state response of boost pressure to wastegate position at different engine speeds, with the throttle held fixed.

V. Decentralized Controller

A two input two output (*TITO*) decentralized controller is presented as the baseline controller for this system. The throttle is used to track intake manifold pressure and the wastegate is used to track boost pressure. The requirement of steady state tracking of P_i and P_b suggests a need for integral control, therefore *proportional-integral (PI)* controllers are implemented in both tracking loops. Feed-forward of the desired steady state wastegate command, filtered at $10\ \frac{rad}{s}$, is included in the wastegate controller.

The character of the dynamics does not change with operating condition, hence, the same controller structure is used at all operating points. The controller gains vary with the system gain, and are scheduled with engine speed and load. Since the actuator movement is limited, an anti-windup scheme is applied to both integrators.

Due to the non-minimum phase zero in the relation between wastegate lift and torque, the wastegate loop is tuned to have a slow bandwidth and the throttle loop is designed to have a higher bandwidth. This approach allows the throttle loop to compensate for the non-minimum phase effect from wastegate, such that it is imperceptible in the torque response. The solid line in Figure 6 illustrates that this leads to an acceptable torque and intake pressure response. The actuator commands are, however, larger than desired and the wastegate saturates. Further detuning of the wastegate controller to address this problem results in unacceptable tracking of P_b, which can impact fuel consumption and emissions. Thus this control structure imposes a tradeoff between aggressive wastegate control and settling time of P_b.

VI. Multi-variable Control Design

Multi-variable control is investigated as a potential means to alleviate the tradeoff imposed by the decentralized control structure.

A. *Full State Feedback with Integral Control*

The full state feedback control architecture shown in Figure 5 is considered. As illustrated, the system dynamics are augmented with integrals of the errors between the desired and measured pressures to satisfy the requirement of steady-state tracking of the intake and boost pressures.

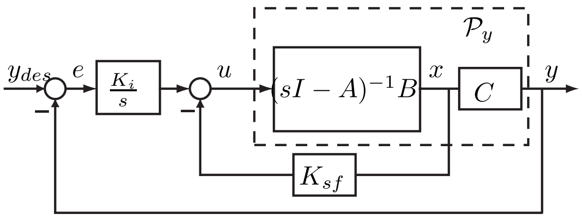


Fig. 5. Control architecture for full state feedback with integral control

The controller parameters are obtained using a linear quadratic regulator (*LQR*) approach for optimal design. The quality of torque response is evaluated for the purposes of weight iteration. High weights are used to penalize the integrals of the errors in tracking of P_i and of P_b. In regions where the wastegate loses authority (see Figure 4), the weight on error in P_b is kept low, effectively using the actuators to track P_i, as P_i has the larger effect on torque. The weights on P_i and P_b are low and tuned for damping, while those on the exhaust pressure, P_e, and the turbocharger speed, N_{tc}, are kept zero. The wastegate actuator is prone to saturation and loss of authority, therefore the cost on the wastegate control is high. This also reduces the bandwidth of the wastegate actuator, which is beneficial, as the rapid movement of wastegate shows up unfavorably in the torque response as non-minimum phase behavior.

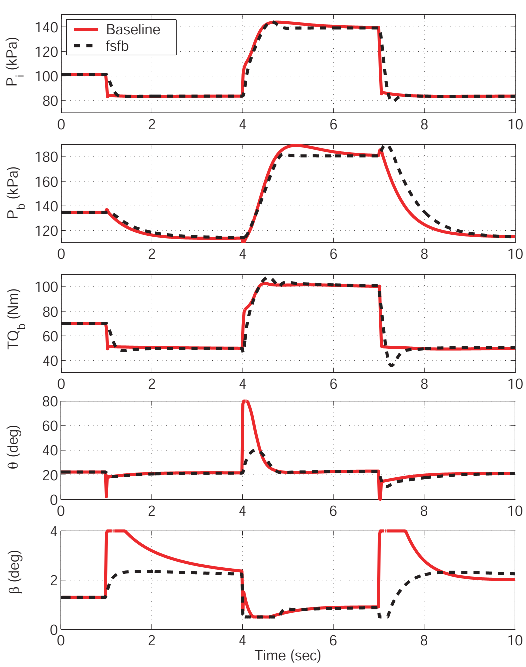


Fig. 6. The closed loop response of the nonlinear system. Solid: decentralized control; Dashed: full state feedback with integral control

The gain scheduled linear controller is augmented with filtered feed-forward on the wastegate in the same fashion as the decentralized control, and anti-windup on the integrators is included. The response with this controller when compared to the response of the baseline controller, shown in Figure 6, does not show a large improvement in response of intake pressure or torque. The actuator signals are, however, significantly smaller and saturation of the wastegate is minimized.

Further examination of the controller shows that the state feedback gains corresponding to the P_e are negligible and could be set to zero. The gains corresponding to the N_{tc} are significant, however, and this state must be measured or estimated. This proves to be a hurdle in implementing this controller architecture directly in an automotive application, as N_{tc} is not a measured quantity and an observer adds unacceptable complexity.

B. Equivalent Controller Derived from Full State Feedback

The full state feedback ("*fsfb*") controller designed in VI-A can be reformulated as an output feedback controller called an "equivalent controller", $\mathcal{C}_{eq}$. The equivalent controller, shown in Figure 7, characterizes the performance of a *fsfb* controller when implemented in the absence of any disturbance on a linear plant. $\mathcal{C}_{eq}$ depends on the gains of the state feedback controller and dynamics of the plant. The equivalent representation of a *fsfb* with integral control, on a linear plant, is given by

$$\mathcal{C}_{eq}(s) = \left(-K_{sf}\left(sI - (A - BK_{sf})\right)^{-1} B + I\right)\frac{K_i}{s}, \tag{23}$$

where K_{sf} is the gain corresponding to the plant states and K_i is the gain corresponding to the error states. Both are obtained from full state feedback design. A and B are from equation (22). The feed-forward term does not affect the calculation of the controller in feedback, hence is not included in the representation of $\mathcal{C}_{eq}$.

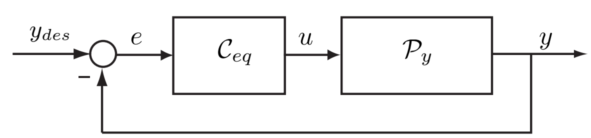


Fig. 7. Equivalent controller architecture

Although $\mathcal{C}_{eq}$ is too complex for calibration and implementation, the controller structure allows evaluation of the classical properties of a *fsfb*. A reduced order equivalent controller can be identified to approximate the *fsfb* with a simple, easy to calibrate, output feedback controller.

C. Reduced Order Approximation of Full State Feedback

The equivalent controller discussed in section VI-B has an order equal to the sum of the order of the plant and number of integrators augmented to it. A low order controller consisting of integrators, *PI* and simple lead and lag elements is desired for practical implementation. Model reduction via balanced realization is applied to $\mathcal{C}_{eq}$ with the integrators removed. Although such a reduction gives the best results, it has too many tuning parameters. The

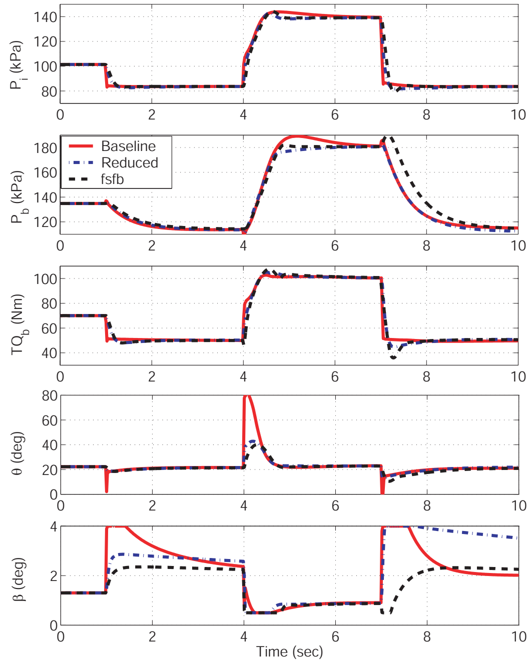


Fig. 8. The closed loop response of the nonlinear system. Solid: decentralized control; Dashed: full state feedback with integral control; Dash Dot: reduced order equivalent controller

dominant poles and zeros of the reduced order controller are considered in order to determine the control structure and insignificant terms are dropped off to form minimum phase *PI* controllers or lead/lag filters. The parameters of the controller selected are gain scheduled and an anti-windup scheme is applied to the integrators.

The resulting reduced order controller has a *PI* structure from intake pressure error, P_i^{err}, to throttle, θ, and a simple integrator from boost pressure error, P_b^{err}, to θ. From P_b^{err} to wastegate, β, the controller has a *PI* structure with a lag term, and a simple integrator from P_i^{err} to β. The nature of this cross coupling term stems from the equal effect of wastegate on intake and boost pressures at low frequency.

The dynamic response of the reduced order controller is compared with the decentralized controller and the $fsfb$ controller in Figure 8. It is seen that the torque response is similar to the other controllers, while the settling times of the pressures and the throttle command are improved compared to the decentralized controller. The wastegate command, however, is as large as the decentralized system and has a significantly longer settling time. Additional dynamical complexity may be required to fully alleviate the tradeoff that exists within the decentralized controller.

VII. Conclusion and Future scope

This paper presented a nonlinear model of a turbocharged gasoline engine equipped with wastegate and electronic throttle. A model-based approach was taken to system analysis and control development. A TITO decentralized control was developed to meet steady state and transient specifications. This system exhibited a tradeoff between actuator response and settling time of boost pressure. In an effort to alleviate this problem, a low order, easy to calibrate output feedback controller was evaluated. This controller was developed using the analysis of a full state feedback controller simulated on the plant. This approach improved the throttle response, as well as the settling time of boost pressure, but the wastegate response remained undesirable.

Future work should focus on implementation of the controller on a vehicle and evaluation of robustness properties. In addition, nonlinear control strategies can be considered in order to improve the wastegate response. Finally, additional system actuators such as external exhaust gas recirculation and variable cam timing should be considered.

VIII. Acknowledgments

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References

[1] N. Watson and M. S. Janota, *Turbocharging the Internal Combustion Engine*. Wiley-Interscience Division, 1982.

[2] M. Muller, E. Hendricks, and S. C. Sorenson, "Mean value modeling of turbocharged spark ignition engine," *SAE Technical Paper*, no. 980784, 1998.

[3] L. Eriksson, L. Nielsen, J. Brugard, J. Bergström, F. Pettersson, and P. Andersson, "Modeling of a turbocharged S.I. engine," *Annual Reviews in Control*, vol. 26, p. 129, 2002.

[4] P. Moraal and I. Kolmonovsky, "Turbocharger modeling for automotive control applications," *SAE Technical Paper*, no. 1999-01-0908, 1999.

[5] R. J. Wakeman and D. Wright, "Closed loop turbocharger control with transient wastegate functions," *SAE Technical Paper*, no. 860487, pp. 131–135, Feb. 1986.

[6] J. R. Zurlo, E. O. Reinbold, and J. Mueller, "The Waukesha turbocharger control module: A tool for improved engine efficiency and response," *ASME, Fall Technical Conference*, vol. 27, no. 4, pp. 35–40, October 1996.

[7] S. Rohde and M. Philipp, "Combined boost pressure and knock control system for S.I. engines including 3-D maps for control parameters," *SAE Technical Paper*, no. 890459, p. 81, 1989.

[8] J. Sun, I. Kolmanovsky, Brehob, J. Cook, J. Buckland, and M. Haghgooie, "Modeling and control of a gasoline direct injection stratified charge (DISC) engine," *Conference on Control Applications*, vol. 1, pp. 471–477, 1999.

[9] L. Lezhnev, I. Kolmanovsky, and J. Buckland, "Boosted gasoline direct injection engines: Comparison of throttle and VGT controllers for homogenous charge operation," *SAE Technical Paper*, no. 2002-01-0709, 2002.