

Sliding Mode Observers to Detect and Isolate Faults in a Turbocharged Gasoline Engine

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ABSTRACT

This paper presents a novel model-based algorithm which is able to detect and isolate major faults assigned to the gas exchange path of a gasoline engine both in the intake and exhaust sides. The diagnostics system is developed for detection and isolation of these faults: air leakage fault between the compressor and the air throttle, exhaust manifold pressure sensor fault, wastegate stuck-closed fault and wastegate stuck-open fault. Sliding mode observers (SMOs) are the core detection algorithms utilized in this work. A first order SMO is designed to estimate the turbocharger rotational dynamics. The wastegate displacement dynamics coupled to the exhaust manifold pressure dynamics is estimated using a second order SMO. Verified with experimental data from a modern TC gasoline engine running in a test cell, the two sliding mode observers are then used in a strategy to detect the faults in the gas exchange path. The proposed diagnostics system shows successful results in isolation of the four faults for gasoline engines without requiring a hot-film air-flow meter (HFM).

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INTRODUCTION

For modern vehicle powertrains, it is essential to monitor, within an online technique, the engine and detect any fault which disrupts its operation. As an online technique, model-based fault detection and isolation (FDI) has been proposed for different engine components with observers being utilized as the core of detection strategies [1, 2, 3]. Nonlinear modeling of engines, which is necessary due to complex engine behavior, implies using more advanced observer techniques to guarantee stability and low calibration effort compared to linear observer designing. The sliding mode theory, being introduced in 1960s, has found an outstanding place in nonlinear observer design possessing robustness to disturbances, having some well developed systematic design approaches and being in lower orders compared to the original dynamic system [4,5].

The gas exchange system in a turbocharged (TC) engine (including the intake and exhaust paths) faces many failures since the turbocharger exacerbates the harsh working conditions for components involved therein. Air leakage after the compressor [6,7], turbine failure [8], pressure sensor fault [9], leakage in the exhaust manifold [10] and wastegate (WG) failures [11] are among reported faults for the gas exchange system. The leakage after the compressor is commonly detected using measurements from a HFM sensor installed upstream the compressor [11,12]. Hence, for modern gasoline engines without a HFM, new detection strategies are required. In the exhaust path, on the other hand, the WG faults are detected using parity relations formulated by comparing the measured

exhaust manifold pressure to pressure estimated by the turbine map [11]. The correct operation of the exhaust manifold pressure sensor is required for detection of WG faults for the work presented in [11].

In this paper two nonlinear observers are designed to capture dynamics in the intake and exhaust path of a turbocharged gasoline engine. A first order sliding mode observer (SMO) is proposed to estimate the rotational dynamics of the turbocharger coupled to pressure dynamics of a control volume between the compressor and the air throttle. Moreover, a second order sliding mode observer (SOSMO) is designed to estimate the pressure dynamics inside the exhaust manifold and also the displacement dynamics of the WG actuator. The two observers are shown to retain their stability even when uncertainty increases due to faults occurring in the engine. The estimation error from the first order SMO and a virtual force estimated by the SOSMO are then used as the residual signals which their values deviates when a fault occurs. The designed FDI is tested experimentally for detection of four different faults introduced to a TC gasoline engine running in a test cell including: air leakage after the compressor, WG stuck-open and WG stuck-closed faults and a gain error fault for the pressure sensor upstream the turbine (Figure 1).

The paper has the following structure. Section 2 describes design of a first order SMO for estimation of the turbocharger rotational dynamics. In section 3, a second order SMO is designed for estimation of the WG displacement and a virtual force disrupting the WG position. Performances of the two designed observers during operation of an engine in fault-free and with faults are presented in section 4. Section 5 concludes the paper.

THE TURBOCHARGER FIRST ORDER SMO

The TC engine structure studied here is shown in Figure 1. For this engine, three major types of dynamics are considered for modeling the gas exchange system of a turbocharged engine including: the turbocharger rotational dynamics, displacement dynamics of the wastegate actuator and pressure dynamics inside three control volumes (1- between the compressor and throttle 2- the intake manifold 3- the exhaust manifold). This section describes modeling the turbocharger rotational speed and pressure inside the control volume between the compressor and the air throttle which are then used in a first order SMO design that which estimates the turbocharger speed.

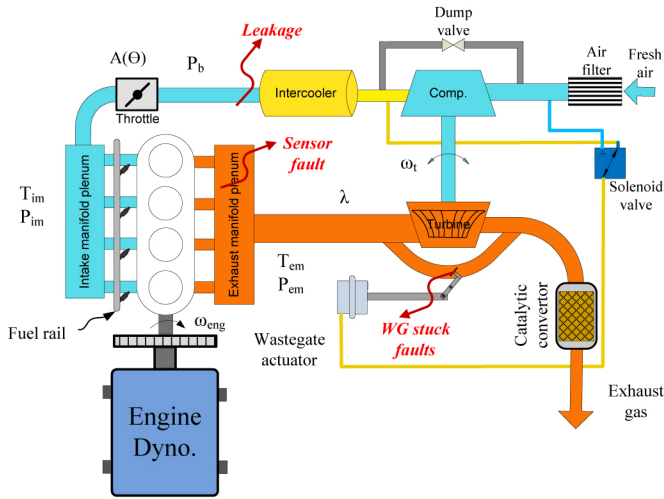


Figure 1. Schematic of the engine test setup and considered faults.

Turbocharger Rotational Dynamics

The turbocharger rotational dynamics is modeled using the coupling between the turbocharger rotor dynamics and the pressure dynamics inside the intercooler. The coupling variables between the two subsystems are flow over the compressor, $\dot{m}_c$, pressure inside the control volume (i.e. the intercooler and connecting pipes), P_b , and the temperature downstream the compressor, $T_{ds,c}$ [13].

The differential equation expressing the rotational dynamics for the turbocharger's wheel is extracted based on the energy balance between the turbine and the compressor:

$$\frac{d}{dt}(0.5I.\omega^2) = I.\omega.\dot{\omega} = \eta_m PW_t - PW_c \quad (1)$$

where ω is the rotor speed in "rad/s", I is the rotor rotational inertia in "kg.m²", η_m is the turbocharger mechanical efficiency, PW_t is the power obtained from exhaust gases in the turbine in "W" and PW_c is the power consumed by the compressor in "W" to increase the air-charge enthalpy. In expanded form PW_t is written as follows.

$$PW_t = \eta_t \cdot \dot{m}_t \cdot C_{p,e} T_{us,t} \left(1 - (P_{ds,t} / P_{us,t})^{(\gamma_e - 1) / \gamma_e} \right) \quad (2)$$

where $\dot{m}_t$ is the turbine gas flow in "kg/s", η_t is the turbine isentropic efficiency (estimated as described in [15]) and $P_{ds,t} / P_{us,t}$ is the ratio of pressures downstream and upstream the turbine. $T_{us,t}$, $C_{p,e}$ and γ_e are measured temperature in "K", specific heat at constant pressure in "kJ/(kg.K)" and the ratio of the specific heats for exhaust gas upstream the turbine.

The turbine flow is estimated using a single nozzle model modified by including an effective pressure ratio to avoid early choking prediction [14]:

$$\dot{m}_t = \frac{P_{us,t}}{\sqrt{R_{em} T_{us,t}}} \cdot A_{t,eff} \cdot f(Pr_t^*) \quad (3)$$

$$f(Pr_t^*) = \sqrt{\frac{2\gamma_{em}}{\gamma_{em} - 1} (Pr_t^{*2/\gamma_{em}} - Pr_t^{*\gamma_{em} + 1/\gamma_{em}})} \quad (4)$$

$$Pr_t^* = \max(Pr_{t,eff}, \left(\frac{2}{\gamma_{em} + 1} \right)^{\frac{\gamma_{em}}{\gamma_{em} - 1}}) \quad (5)$$

The effective pressure ratio ($Pr_{t,eff}$) is calculated from the turbine actual pressure ratio using

$$Pr_{t,eff} = (P_{ds,t} / P_{us,t})^\alpha \quad (6)$$

where α and $A_{t,eff}$ are estimated using an optimization technique detailed in [15].

Similarly to the turbine, consumed power in the compressor is formulated as follows.

$$PW_c = \frac{1}{\eta_c} \dot{m}_c \cdot C_{p,a} T_{us,c} \left((P_{ds,c} / P_{us,c})^{\gamma_a - 1} / \gamma_a - 1 \right) \quad (7)$$

where $C_{p,a}$ is the air specific heat at constant pressure in "kJ/(kg.K)", γ_a is the ratio of the specific heats for the intake air, η_c is the compressor isentropic efficiency, $\dot{m}_c$ is the flow over the compressor in "kg/s", $P_{ds,c} / P_{us,c}$ is the ratio of pressures downstream and upstream the compressor and $T_{us,c}$ is the air temperature before the compressor in "K". The air flow over the compressor can be predicted by using transformation techniques which are developed to replace the compressor discontinuous characteristic map by a 1-D continuous curve [13]. Two non-dimensional parameters known as compressor head parameter, Ψ and normalized compressor flow rate, Φ are defined as follows.

$$\Psi = C_{pa} T_{us,c} \left((P_{ds,c} / P_{us,c})^{\gamma_a - 1} / \gamma_a - 1 \right) / (0.5 U^2) \quad (8)$$

$$\Phi = \frac{\dot{m}_c}{\rho_a d^3 \omega} \quad (10)$$

where d is the compressor impeller diameter in “m”, U is the impeller tip velocity in “m/s” and ρ_a is the air density in “kg/m³”. Results of transforming a compressor map using the Φ - Ψ definition are shown in Figure 2. As shown, the map can be reduced to a band with a lower and an upper limit (Fit1 and Fit2 in Figure 2). The average of this band can be represented by a polynomial as:

$$\Phi = \alpha_1 \Psi^3 + \alpha_2 \quad (11)$$

with $\alpha_1 < 0$. Assuming the compressor rotor speed and boost pressure are known, one can calculate Ψ from equation (8). Then by using equation (11) in equation (10), $\dot{m}_c$ can be computed. However, there would be variations in α_i if one uses equation (11) for all points inside the compressor map. This variation is treated as uncertainty during the state-space representation explained later in this section.

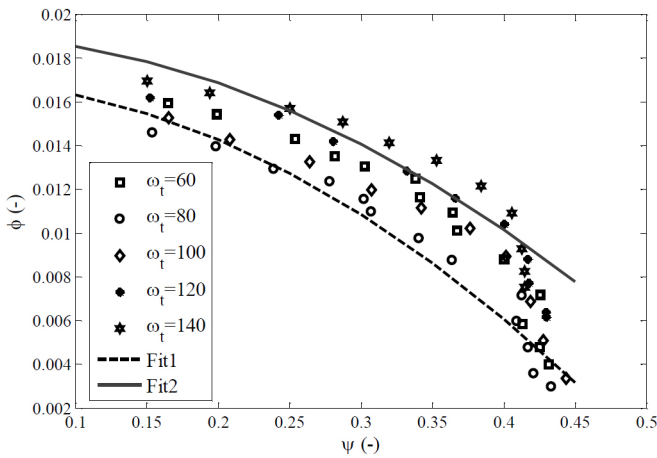


Figure 2. Reduced compressor characteristic map. Fit1 and Fit2 are lower and upper level curves fitted to the data.

The compressor isentropic efficiency, defined as the ratio of ideal isentropic work to the actual work done by the compressor, is predicted with $\eta_c = \eta_c(\Psi)$ as outlined in [16]. Therefore, $T_{ds,c}$ is computed using the following formulation.

$$T_{ds,c} = \left(\frac{(P_{ds,c}/P_{us,c})^{(\gamma_a-1)/\gamma_a}}{\eta_c(\Psi)} + 1 \right) * T_{us,c} \quad (12)$$

Once $T_{ds,c}$ is calculated from equation (12), it is used to calculate the temperature inside the intercooler (T_{ave}), which is considered as the average of boost temperature (T_b) and $T_{ds,c}$.

Intercooler Pressure Dynamics

Mass and energy balance equations are coupled to isentropic flow equation through the throttle to determine pressure dynamics inside the intercooler and its connecting pipes. The pressure loss across the intercooler is assumed negligible which means $P_b \approx P_{ds,c}$ [16]. Therefore, differential equation of the pressure dynamics is:

$$\frac{dP_{ds,c}}{dt} = \frac{RT_{ave}}{V} (\dot{m}_c - \dot{m}_{th}) \quad (13)$$

where, V is the volume between the compressor and the throttle in m³, $\dot{m}_{th}$ is the flow over air throttle in “kg/s” and R is the ideal gas constant for the air in “kJ/(kg.K)”. In equation (13), $\dot{m}_{th}$ is calculated by the equation of isentropic flow over a restriction:

$$\dot{m}_{th} = \frac{P_{ds,c}}{\sqrt{RT_{ave}}} A(\theta_{th}) \sqrt{\frac{2\gamma_a}{\gamma_a-1} ((Pr_{th}^*)^{\frac{2}{\gamma_a}} - (Pr_{th}^*)^{\frac{\gamma_a+1}{\gamma_a}})} \quad (14)$$

$$Pr_{th}^* = \max \left(\frac{P_{im}}{P_{ds,c}}, \left(\frac{2}{\gamma_a+1} \right)^{\frac{\gamma_a}{\gamma_a-1}} \right) \quad (15)$$

where $A(\theta_{th})$ is the throttle effective area as a function of the throttle angle, θ_{th} .

State Space Equations

The rotor speed and boost pressure are selected as the state variables for the air-charge system. Replacing $P_{ds,c}$ and ω in equation (1) and equation (13) with x_1 and x_2 respectively, results in the following state space form for the overall turbocharger system dynamic equation

$$\frac{dx_1}{dt} = \xi_1 \frac{\left(\left(\frac{x_1}{P_{us,c}} \right)^{\beta_a} - 1 \right)^3}{x_2^5} + \xi_2 x_2 - \xi_3 x_1 \cdot f\left(\frac{P_{im}}{x_1}\right) + \Delta\Theta_1 \quad (16)$$

$$\frac{dx_2}{dt} = \frac{u}{x_2} - C_1 \frac{\left(\left(\frac{x_1}{P_{us,c}} \right)^{\beta_a} - 1 \right)^4}{x_2^6} - C_2 \left(\left(\frac{x_1}{P_{us,c}} \right)^{\beta_a} - 1 \right) + \Delta\Theta_2, \quad \beta_a = (\gamma_a - 1) / \gamma_a \quad (17)$$

where $x_1 = P_{ds,c}$ and $x_2 = \omega$ are the states and $u = \eta_m PW_t$ is the model input. $\Delta\Theta_i$ in each of the above equations are bounded uncertainties cause by using single values for α_i in equation (11) regardless of the compressor operating point. ξ_i and C_i are system parameters which are listed in the Appendix I. These parameters are either constant or functions of the engine variables which are easily available from measurements in a production ECU.

Robust SMO Design

The purpose here is to give an estimate ($\hat{x}_2$) of the turbocharger rotor speed x_2 using measurements of conventional parameters and states and having feedback from the boost pressure, x_1 . This is done here by designing a sliding mode observer which converges to the states x_1 and x_2 in a step by step manner. SMOs use the concept of equivalent injection signal [5] to create a sliding motion on the estimation errors and produce state estimations. Considering state-space equation of the charge system from equations (16)-(17) one can design a nonlinear sliding mode observer of the following form:

$$\dot{\hat{x}}_1 = \xi_1 \frac{\left(\left(\frac{\hat{x}_1}{P_{us,c}} \right)^{\beta_a} - 1 \right)^3}{\hat{x}_2^5} + \xi_2 \hat{x}_2 - \xi_3 \hat{x}_1 \cdot f \left(\frac{P_{im}}{\hat{x}_1} \right) + \lambda_1 E_1 \operatorname{sgn}(e_1) \quad (18)$$

$$\dot{\hat{x}}_2 = \frac{u}{\hat{x}_2} + C_1 \frac{\left(\left(\frac{\hat{x}_1}{P_{us,c}} \right)^{\beta_a} - 1 \right)^4}{\hat{x}_2^6} - C_2 \left(\left(\frac{\hat{x}_1}{P_{us,c}} \right)^{\beta_a} - 1 \right) + \lambda_2 E_2 \operatorname{sgn}(\tilde{e}_2) \quad (19)$$

where $[\hat{x}_1, \hat{x}_2]$ are estimation of states $[x_1, x_2]$, $\lambda_i s'$ are the observer gains, and:

$$e_1 = x_1 - \hat{x}_1 \quad (20)$$

$$\tilde{e}_2 = \tilde{x}_2 - \hat{x}_2. \quad (21)$$

The complementary state, $\tilde{x}_2$, is used based on the equivalent injection concept since the real error for x_2 ($e_2 = x_2 - \hat{x}_2$) is not available during the observer computation. Definition of the complementary states can be found within the following proposition.

Proposition 1: Consider the observer in equations (18) and (19) with

$$E_1 = 1, \quad E_2 = \begin{cases} 1 & \text{if } (e_1 = 0) \\ 0 & \text{else} \end{cases} \quad (22)$$

If;

$$\tilde{x}_1 = \hat{x}_1 \quad (23)$$

$$\tilde{x}_2 = \hat{x}_2 + \lambda_1 E_1 \operatorname{sgn}(e_1) / \xi_2 \quad (24)$$

Then the observer is stable.

Proof 1: The proof for stability is addressed in [13].

THE EXHAUST MANIFOLD SECOND ORDER SMO

Two main dynamics are considered for the exhaust manifold; 1) the pressure dynamics inside a control volume between the exhaust valves and the turbine inlet 2) the WG actuator displacement dynamics. The following describes modeling of these two main dynamics and designing a second order SMO based on the modeling results.

Exhaust Manifold Pressure Dynamics

The exhaust manifold pressure dynamics is determined based on the ideal gas law and the mass flow conservation:

$$\frac{dP_{us,t}}{dt} = \frac{R_{em} T_{us,t}}{V_{em}} (\dot{m}_{in,em} - \dot{m}_{out,em}) \quad (25)$$

where, R_{em} is the specific gas constant for the exhaust flow and V_{em} is the volume between the turbine inlet and the exhaust valves. In equation (25), $\dot{m}_{in,em}$ is the gas flow into the exhaust manifold and $\dot{m}_{out,em}$ is the gas flow that leaves the exhaust manifold:

$$\dot{m}_{in,em}(t) = [\dot{m}_a + \dot{m}_f]_{t-\tau} \quad (26)$$

$$\dot{m}_{out,em}(t) = [\dot{m}_t + \dot{m}_{WG}]_t \quad (27)$$

where $\dot{m}_a$ and $\dot{m}_f$ are the engine air-charge and fuel flow and, τ is a time delay between closing of the intake valves and opening of the exhaust valves which is about 270 crank angle degrees. The exhaust manifold output flow from equation (27) is the sum of the turbine flow, $\dot{m}_t$, and the gas flow which is bypassed through the WG, $\dot{m}_{WG}$.

Modeling the Exhaust Manifold Input Flow

The following speed-density relation is a common way to calculate the engine cylinder input air mass flow (air-charge) using the intake manifold pressure P_{im} , and the engine speed ω_{eng} [17]. The speed-density relation is corrected by the volumetric efficiency η_{vol} as;

$$\dot{m}_a = \eta_{vol} (P_{im}, T_{im}, \omega_{eng}) \frac{P_{im} V_{eng} \omega_{eng}}{2 R_{im} T_{im}} \quad (28)$$

where T_{im} is the intake manifold temperature, R_{im} is the specific gas constant for the air and V_{eng} is the displacement volume of the engine. The volumetric efficiency is calibrated during the engine base calibration process for *fault-free* operations. An estimation error occurs in equation (28) if the engine does not follow its normal behavior caused by a fault in the exhaust manifold which changes the engine back pressure, leading to a change in the volumetric efficiency. To accurately estimate the engine air-charge in *faulty* and *fault-free* working conditions, the intake manifold dynamics is considered here to calculate the deviation in the volumetric efficiency $\Delta \eta_{vol}$ from its calibrated values.