

Air Leak Detection for a Turbocharged SI Engine using Robust Estimation of the Turbocharger Dynamics

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ABSTRACT

Proper operation of an internal combustion engine is required by demands of a vehicle driver and governmental legislations. Therefore it is necessary to monitor, within an online technique, the engine and detect any fault which disrupts its normal operation. In this paper, the air-charge path, as a key element in a turbocharged engine, is monitored for an air leakage fault. At first, a robust algorithm to estimate unmeasured turbocharger rotational speed is presented. The sliding mode methodology is used to design the estimator which is shown to be robust to the compressor modeling uncertainties. The estimation error from the sliding mode observer (SMO) is then used to detect abnormal behavior of the turbocharger along with the engine due to a leakage fault in the air-charge path. Experimental results from a modern turbocharged SI engine indicate the designed monitoring technique is able to detect a leakage fault, of 7 mm or higher sizes, in the air-charge path.

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INTRODUCTION

Safe and reliable operation of spark ignition (SI) engines have gained high importance inspired by customer demands and legislative requirements. In modern engine control units (ECUs), in some cases, 40% of the software implemented in the ECU is dedicated to algorithms which monitor the engine operation. These algorithms benefit from different techniques such as range check [1], model based fault detection [2-3] and plausibility check [4].

Model-based fault detection (FD) has seen an increasing interest by researchers owing to its ability to operate in a large range of conditions and to reduce the calibration efforts [5-6]. A major concern in model-based FD techniques is accurate modeling or, if that is not possible, to increase the robustness of the designed FD algorithm to modeling uncertainties. Many techniques for robust fault detection have been proposed for both linear and nonlinear systems [7,8,9,10,11,12,13,14,15,16]. For nonlinear systems, in a way of looking, these techniques are divided into two major groups. In the first group of techniques, the model uncertainty is included either in the structure of an estimation algorithm used to generate the residual or, in the threshold which evaluates the residual [7,8,9]. Examples of this technique are interval constraint problem [7], interval observers [8] and adaptive threshold residual evaluation [9]. The main target in this technique is to design the algorithm (in the form of an estimator) which its error stays bounded with/without error. When a fault occurs in the system, the absolute of the estimation error (which is

bounded) deviates from normal values it has at fault-free operations and this deviation is the major criteria for the FD algorithm. In the second group of robust FD techniques, the main concept is to transform the system into a form which is decoupled from uncertainty disturbances (thus it is insensitive to disturbances while at the same time responsive to faults) [10,11,12,13,14]. This group has many approaches such as logic dynamic approach [10], unknown input observer (UIO) [11], algebraic UIO [12], sliding mode observer (SMO) [13-14] and geometrical approach [15]. In all these methodologies, using diffeomorphisms/transformation, the uncertainty disturbance is moved into an un-observable subspace while keeping the faults in the observable subspace. A required condition is that the fault and the disturbance subspaces are not overlapped and their intersection is empty [15-16]. Despite the ability to remove the effect of disturbances completely, there are two requirements (in addition to the overlap condition) associated to the disturbance decoupling approach which limit their application to all nonlinear systems, 1) a nonlinear diffeomorphism/transformation is required to transform the nonlinear system into a desired form (this is usually a complex task to find it for highly nonlinear systems) and 2) there is a requirement for the number of sensors as: $N_s > N_f + N_d$ where N_s is the number of measured outputs (sensors), N_f is the number of faults to be detected and N_d is the number of disturbances which are to be decoupled. This means that the number of sensors should be greater than the amount of the faults and disturbances.

In this paper a nonlinear observer is designed to estimate the dynamics of a turbocharger coupled to a SI engine.

Turbocharging a SI engine (with the aim of improving fuel economy) increases the risk of air leakage in the boost path located between the compressor and the throttle due to high pressure and temperature in the path. For SI engines equipped with a hot-film flow meter (HFM) before the compressor, this fault is detected by comparing the modeled throttle flow with the measured air flow before the compressor [9,10,11,12,13,14,15,16,17]. When the SI engine is not equipped with an HFM sensor, as some of modern commercial vehicles, this fault can be detected using an observer for the turbocharger dynamics as outlined in this paper. The sliding mode technique is used to design the observer, which maintains its stability even when uncertainty increases due to a leakage fault in the boost path before the air throttle. The estimation error from the SMO is then used as the residual signal since its absolute value increases when a leakage fault is introduced after the compressor. The designed FD is tested experimentally for detection of a leakage fault introduced to a turbocharged SI engine running in a test cell.

The paper is outlined in 6 sections. After this introduction, the modeling for the turbocharger dynamics is described in [section 2](#). This model is used in [section 3](#) for designing a sliding mode observer which is shown to converge to the measured turbocharger rotor speed. [Section 4](#) describes how the SMO estimation error is used as a residual to detect a leakage fault in the boost path. Results of the SMO and the detection strategy are validated with experimental data measured from a turbocharged engine in [section 5](#). Finally, the conclusion of this paper is presented in [section 6](#).

TURBOCHARGER DYNAMICS MODELING

The turbocharger rotational dynamics is modeled using the coupling between two engine subsystems: a) the turbocharger rotor dynamics, b) the pressure dynamics inside the intercooler. As shown in [Figure 1](#), inputs to the intercooler subsystem are the compressor flow, $\dot{m}_c$, and temperature after the compressor ($T_{ds,c}$) while its output to the turbocharger rotor subsystem is the boost pressure, P_b . Therefore the coupling variables between the two subsystems are flow over the compressor, pressure inside the control volume (i.e. the intercooler) and the temperature downstream the compressor. Here at first the turbocharger rotor dynamics is formulated, in a mean value modeling concept, and then a mathematical modeling for the intercooler pressure dynamics is presented. Finally the overall state space formulation used for the sliding observer, which is the main tool for fault detection in this work, is shaped.

Turbocharger Rotor Dynamics

The differential equation expressing the rotational dynamics for the turbocharger's wheel is extracted based on the enthalpy balance between the turbine and the compressor [18]:

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

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.

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

where $\dot{m}_t$ is the turbine gas flow in "kg/s", η_t is the turbine isentropic efficiency 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.

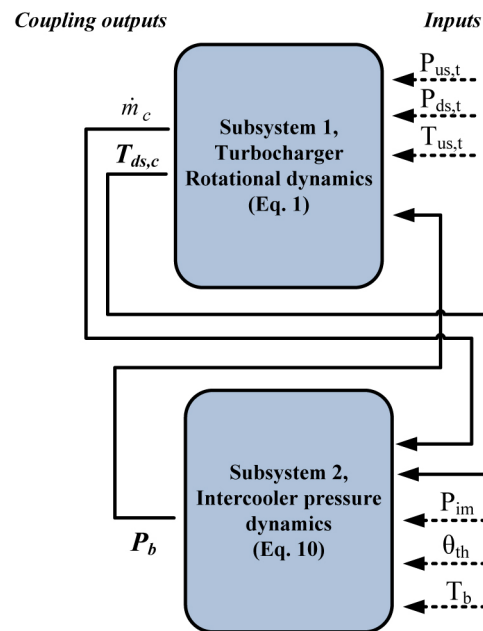


Figure 1. Schematic of the coupled turbocharger dynamics and intercooler pressure dynamics

The turbine flow is estimated using a single nozzle model modified by including an effective pressure ratio to avoid early choking prediction [19]:

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

$$f(Pr_t^*) = \sqrt{\frac{2\gamma_{em}}{\gamma_{em} - 1} (Pr_t^{*2/\gamma_{em}} - Pr_t^{*\gamma_{em} + 1/\gamma_{em}})} \quad (3-b)$$

$$Pr_t^* = \max(Pr_{t,eff}, \left(\frac{2}{\gamma_{em} + 1}\right)^{\frac{\gamma_{em}}{\gamma_{em} - 1}}) \quad (3-c)$$

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

$$Pr_{t,eff} = (P_{ds,t} / P_{us,t})^\alpha \quad (4)$$

where α and $A_{t,eff}$ are estimated using an optimization technique detailed in [19].

The turbine efficiency together with mechanical efficiency is estimated using the definition of the thermo-mechanical efficiency reported in [20] as the following:

$$\eta_m \eta_t = \frac{\dot{m}_c \cdot C_{p,a} \cdot (T_{ds,c} - T_{us,c})}{\dot{m}_t C_{p,e} T_{us,t} \left[1 - (P_{us,t} / P_{ds,t})^{(\gamma_e - 1) / \gamma_e} \right]} \quad (5)$$

If the turbine thermo-mechanical efficiency from equation (5) is modified by a temperature factor as: $\eta_t \cdot (T_{ds,c} - T_{us,c}) / T_{amb})^{0.39}$, it would get correlated to the turbine flow as shown in Figure 2. Thus, once the turbine flow is estimated from equation (3), by estimating the compressor downstream temperature (which is discussed later in this section) one can estimate the turbine and mechanical efficiency using Figure 2.

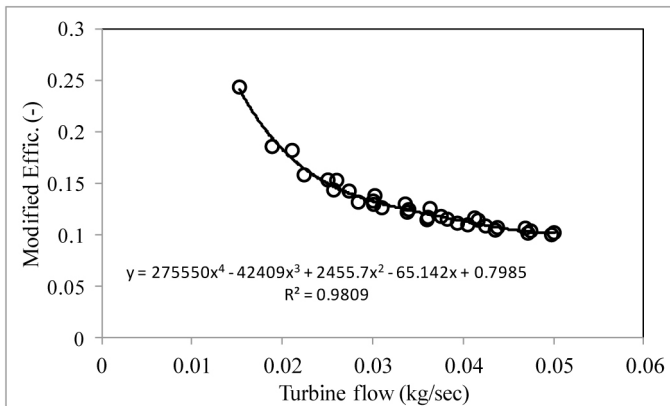


Figure 2. The turbine modified efficiency correlated to the turbine flow.

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

$$PW_c = \frac{1}{\eta_c} \dot{m}_c \cdot C_{p,a} T_{us,c} \left((P_{ds,c} / P_{us,c})^{\gamma_a - 1} / \gamma_a - 1 \right) \quad (6)$$

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 the compressor map (Figure 3) once ω and $P_{ds,c} / P_{us,c}$ are known. Avoiding difficulties associated with using this characteristic map, transformation techniques are developed to replace the compressor discontinuous characteristic map by a 1-D continuous curve [18]. To do these transformations, two non-dimensional parameters known as compressor head parameter, Ψ and normalized compressor flow rate, Φ are defined as follows.

$$\Psi = C_{pa} T_{us,c} \left(\left(\frac{P_{ds,c}}{P_{us,c}} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right) / (0.5 U^2) \quad (7-a)$$

$$U = \pi / 60 \cdot d \cdot \omega \quad (7-b)$$

$$\Phi = \frac{\dot{m}_c}{\rho_a d^3 \omega} \quad (7-c)$$

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".

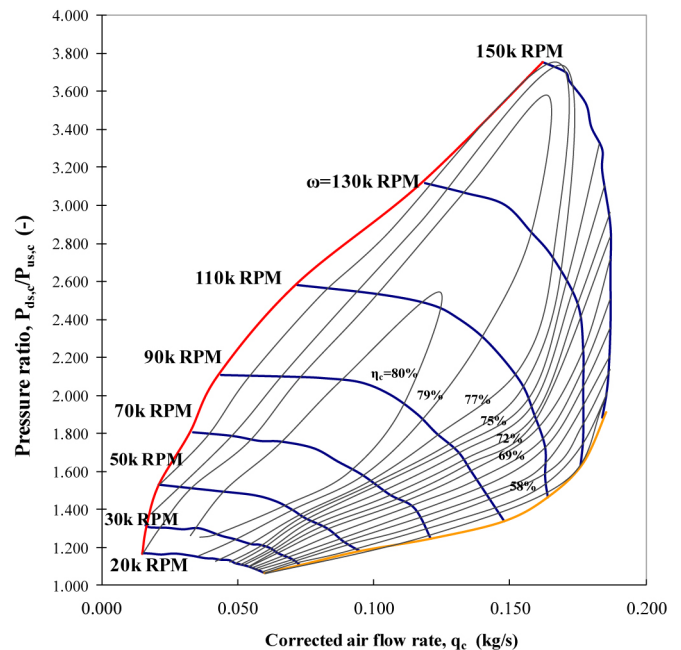


Figure 3. Characteristic map of the compressor in this study.

Results of the transformations are plotted in Figure 4. As observed, the compressor characteristic map is transformed into a relatively narrow band by the Φ - Ψ transformation. The relation between Φ and Ψ for any turbo-speed constant curve can be represented by:

$$\Phi = \alpha_1 \Psi^3 + \alpha_2 \quad (8)$$