Nonlinear Reliable Control based Super-Twisting Sliding Mode Algorithm with the Diesel Engine Air Path

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Abstract: In this paper we developed a passive fault tolerant control strategy for the diesel engine air path. This strategy is carried out under the concept of Higher Order Sliding Mode Control (HOSMC). The proposed fault tolerant strategy incorporates a super twisting controller which handles parametric uncertainties and actuator failures. In this paper we consider two types of actuator failures, additive and loss-of-effectiveness faults. The simulations of the proposed controller on a recently validated experimental air path diesel engine model, show good results for actuator failures conditions even in the presence of uncertainties on model parameters.

Keywords: Diesel engine air path, Nonlinear control, Robust Nonlinear Control, Super-Twisting Algorithm (STA), Sliding Mode Control(SMC)

1. INTRODUCTION

One of the main motivations for developing Diesel engine control algorithms is the reduction of exhaust gas to meet the requirements of emission standards EURO V and VI. These requirements could be fulfilled by using new and sophisticated control algorithms in order to provide the required engine torque which lead to a compromise between the optimal fuel consumption and a given exhaust gas emission level. The most harmful exhaust gas is the Oxide of Nitrogen (NO_x) coming from the oxidation of the nitrogen monoxide in the combustion chamber.

For modern engines such as Turcbocharged Diesel (TDE), controlling the NO_x emissions primarily depends on two feedback variables namely the Exhaust Gas Recirculation (EGR) rate and the Air Fuel Ratio (AFR) in the intake manifold. These two feedback variables depends on the EGR and the Variable Geometry Turbine (VGT) actuators whose position determine the amount of the EGR flow in the intake manifold and thus, controls the AFR and the EGR ratios variables. The challenge is thus, to design an efficient controller that manages the two actuators signals.

Controlling the diesel engine emissions requires a description model of the diesel engine air path. In the past decade many TDE air path models were developed and presented in (Guzzella and Amstutz (1998), Kao and Moskwa (1995), Kolmanovsky et al. (1997), Moraal et al. (1997), Jankovic and Kolmanovsky (2000)). The particular model used in this paper was outlined and validated experimentally in (Jankovic and Kolmanovsky (2000)).

Air path Diesel engine control faces a major difficultly which is characterized by the phenomenon of coupling for both the VGT and the EGR actuators. Conventional calibration/mapping approaches such as PI controllers requires a highly time consuming calibration task, therefore, the attention of researchers has been focused on multivariable controllers in order to reduce the calibration task, produce a satisfactory robust results in terms of torque response and emission levels.

Multivariables controllers design can be divided into linear and nonlinear ones. In the past decade, considerable efforts have been dedicated to the control of modern diesel engines. Several classical control methods were proposed in the literature, e,g., robust gain-scheduled controller based on a Linear Parameter Varying (LPV) model for turbocharged diesel engine in (Jung and Glover (2006), Lihua et al. (2007), Xiukun and del Re (2007)), indirect passivation in Larsen et al. (2000), predictive control in Ferreau et al. (2007), feedback linearization (Plianos et al. (2007), Dabo et al. (2009)), backstepping based control (Fredriksson and Egart (2001)) and H_∞ control (van Nieuwstadt et al. (1998)). In Jankovic and Kolmanovsky (2000) the authors developed a nonlinear multivariable controller based on a Control Lyapunov Function (CLF) for controlling the exhaust manifold pressure and the compressor mass flow rate. Most of these algorithms are control-oriented models i.e the control laws computed by these algorithms are based upon a model of the diesel engine air path. These controllers work badly because of discrepancies between the description model and the real system due to natural model parametric uncertainties or when faults occur in the engine. This is the case where robust nonlinear control methods are suitable in order to enhance the robustness of the controller making it less sensitive to parametric uncertainties and fault tolerant to sudden failure which can occur on the diesel engine air path.

In this paper, we deal with both model parametric uncertainties and engine actuator faults. In the TDE air path, model parametric uncertainties arise from variations in the engine cartographies, sources such as temperature change, and external or internal unmodeled disturbances. Engine actuator faults affect the TDE air path actuators. In this paper, we consider two types of actuator faults. The first one, model the faults as bounded additive periodic unknown signals that are superposed onto the control signal. The second one, consider the case of the loss of actuator effectiveness, modeled by a multiplicative factor that, when multiplied to the control signal, will reduce its effectiveness depending on the value of this factor.

The work presented in this paper, aims to achieve a Fault-tolerant-Controller(FTC) which handles the considered actuator fault types and the model parametric uncertainties which affects the TDE air path. Many FTC schemes were proposed in the literature. The readers are referred to the survey in (Patton (1997), Zhang and Jiang (2003)). Generally speaking, the FTC controllers can be classified into two types, namely, passive and active FTC (Zhang and Jiang (2003)). In this paper, we choose the passive approach to design our FTC controller.

The active FTC guarantee stability and performance for the faulty model and this by reconfiguring the controller online, based on the fault-detection-and-diagnosis (FDD) block that detects, isolates, and estimates the current fault. The passive approach uses a unique robust controller which deals with all the expected faults, without using an FDD block or needing a control reconfiguration. This approach has the advantage to avoid the time delay caused by the online diagnosis of the faults and reconfiguration of the controller. In this paper, we choose the passive approach for the proposed FTC controller. Indeed, nowadays the Electronic Control Units (ECU's) are characterized by limited computational capacities, which makes the available time window for control very short. Moreover the use of a unique robust controller to deals with modelling uncertainties and faults, allows us to simplify the control structure in order to enhance its robustness facing model parametric uncertainties.

Sliding Mode Control (SMC) which is known to perform well under parametric uncertainties and external disturbances (Utkin (1977), Pisano and Usai (2011)) has become widespread and one of the most popular robust nonlinear control method. In (Utkin et al. (2000), Upadhyay et al. (2002)) the authors proposed a sliding mode based controllers which coordinates the EGR and the VGT actuator signals for regulation control of modern diesel engine. SMC Hybrid air path controllers for multiples combustion modes were also proposed in (Wang (2008)).

The main disadvantage of classic sliding mode control is characterized by small oscillations at the output of the system whose effects can be harmful to motion control systems. This phenomenon well known under the name of chattering can appear due to fast dynamics which have been omitted from the model, fast switching discontinuous control and digital implementation issues. Thus, in (Levant (1993)), the Higher-Order Sliding-Mode control (HOSMC) was used in order to reduce or to eliminate the chattering phenomenon at high frequencies. Several algorithms to carry out HOSMC have been developed in the literature (see Pisano and Usai (2011) for a complete review). Among them, the Super-Twisting control algorithms (STA) require that the sliding variable be relative degree 1 with no need of the derivative of the sliding surface S . With its simplicity of implementation and its power to eliminate the chattering phenomenon, STA algorithms are preferable over the classic sliding mode (Pisano and Usai (2011), Levant (1993), Gonzalez et al. (2011)).

Comparing to the work (Utkin et al. (2000), Upadhyay et al. (2002), Wang (2008)), the contribution of this paper consists in developing an STA controller centred on achieving passive fault tolerance for the TDE air path . The proposed controller handles both parametric uncertainties and actuator failures. Two types of actuator faults are treated in this paper, additive actuator faults and loss-of-effectiveness.

This paper is organized as follows. Section 2 introduces the TDE air path modeling. Section 3 introduces the systems that we are dealing with together with the assumptions required. The passive fault-tolerant ST based controller will be described in section 4 . Simulation results are given in section 5. Section 6 summarizes conclusions and describes the future work.

2. MATHEMATICAL MODEL OF THE DIESEL ENGINE AIR PATH

The schematic diagram of the diesel engine is shown in Fig. 1. At the top of the diagram we can see the turbocharger and the compressor mounted on the same shaft. The turbine delivers power to the compressor by transferring the energy from the exhaust gas to the intake manifold. Together, the mixture of air from the compressor and the exhaust gas from the EGR valve with the injected fuel burns, and produces the torque on the crank shaft.

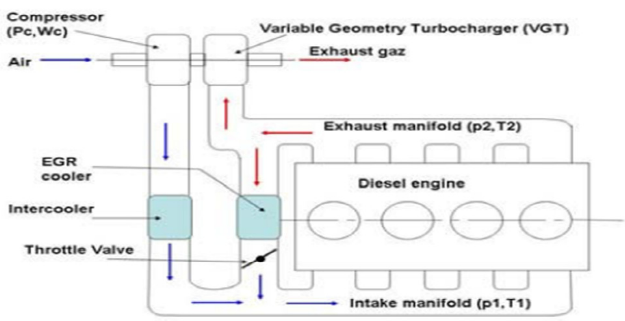

Fig. 1. Turbocharged Diesel Engine.

The full-order TDE model is a seventh-order one which contains seven states: intake and exhaust manifold pressure (p_1 and p_2), oxygen mass fractions in the intake and exhaust manifolds (F_1 and F_2), turbocharger speed (ω_{tc}) and the two states describing the actuator dynamics for the two control signals (u_1 and u_2).

In order to obtain a simple control law, and due to the fact that the oxygen mass fraction variables are difficult to measure, the seventh-order model is reduced to a third-order one (Jankovic and Kolmanovsky (2000)).

$$\begin{cases} \dot{p}_1 = k_1(W_c + W_{egr} - k_e p_1) + \dfrac{\dot{T}_1}{T_1} p_1 \\ \dot{p}_2 = k_2(k_e p_1 - W_{egr} - W_t + W_f) + \dfrac{\dot{T}_2}{T_2} p_2 \\ \dot{P}_c = \dfrac{1}{\tau}(\eta_m P_t - P_c) \end{cases} \quad (1)$$

where the compressor and the turbine mass flow rate (W_c and W_t) is related to the compressor and the turbine power (P_c and P_t) as follows:

$$W_c = P_c \frac{k_c}{p_1^{\mu} - 1} \quad (2)$$

and:

$$P_t = k_t(1 - p_2^{-\mu}) W_t \quad (3)$$

Where:

$k_c = \frac{\eta_c}{c_p T_a}$, $k_t = c_p \eta_t T_2$, $k_1 = \frac{R_a T_1}{V_1}$, $k_e = \frac{\eta_v N V_d}{R_a T_1}$ $k_2 = \frac{R_a T_2}{V_2}$

Table 1. Nomenclature of the diesel engine variables

Variables	Name	Units
p_1	Intake Manifold Pressure	Pa
p_2	Exhaust Manifold Pressure	Pa
P_c	Compressor power	W
P_t	Turbine power	W
W_c	Compressor mass flow	Kg/s
W_t	Turbine mass flow	Kg/s
W_f	Fueling mass flow rate	Kg/s
η_v	Engine volumetric efficiency	-
η_c	Compressor isentropic efficiency	-
η_t	Turbine isentropic efficiency	-
η_m	Turbocharger mechanical efficiency	-
V_1	Intake manifold volume	m^3
V_2	Exhaust manifold volume	m^3
V_d	Engine Volume cylinder	m^3
T_a	Ambient temperature	K
T_1	Intake manifold temperature	K
T_2	Exhaust manifold temperature	K
R_a	specific gas constant	$J/Kg/K$

Notice that the real inputs are the EGR valve and the VGT valve openings. The considered inputs, in this case for the sake of simplicity, are $u_1 = W_{egr}$ and $u_2 = W_t$, which are respectively the air flow through the EGR and the VGT valves.

Since $\dot{T}_1$ and $\dot{T}_2$ have very slow variations Jankovic and Kolmanovsky (2000), their dynamic can be neglected. This yields the following simplified model:

$$\begin{cases} \dot{p}_1 = k_1(W_c + W_{egr} - k_e p_1) \\ \dot{p}_2 = k_2(k_e p_1 + W_f - W_{egr} - W_t) \\ \dot{P}_c = \dfrac{1}{\tau}(\eta_m P_t - P_c) \end{cases} \quad (4)$$

When replacing W_c and P_t by their expressions in (2) and (3), the simplified model can be expressed under the following control- affine form:

$$\dot{x} = f(x) + g_1(x)u_1 + g_2(x)u_2 \quad (5)$$

where $x = (p_1, p_2, P_c)^T$ and

$$f(x) = \begin{bmatrix} k_1 k_c \dfrac{P_c}{p_1^{\mu} - 1} - k_1 k_e p_1 \\ k_2(k_e p_1 + W_f) \\ \dfrac{-P_c}{\tau} \end{bmatrix} \quad (6)$$

$$g_1(x) = \begin{bmatrix} k_1 \\ -k_2 \\ 0 \end{bmatrix} \quad g_2(x) = \begin{bmatrix} 0 \\ -k_2 \\ K_o(1 - p_2^{-\mu}) \end{bmatrix} \quad (7)$$

with $K_o = \frac{\eta_m}{\tau} k_t$

We notice that the TDE model parameters (k_1, k_2, k_c, k_e, k_t, τ, η_m) have been identified under steady state conditions (i.e constant engine speed and constant fueling rate) and extensive mapping. The nomenclature of the TDE parameters can be found in Table 1.

Note that in Jankovic and Kolmanovsky (2000), the authors proved that the set Ω, defined by:

$$\{\Omega = (p_1, p_2, P_c) : 1 < p_1 < P_1^{max}, \\ 1 < p_2 < p_2^{max}, 0 < P_c < P_c^{max}\}.$$

is an invariant set.

3. SYSTEM FAULT DESCRIPTION

Here, we consider the class of nonlinear system of the form:

$$\dot{x} = f(x) + g(x)u \quad (8)$$

Where $x \in \mathbf{R}^n$,$u \in \mathbf{R}^m$ represent the state and the input vector respectively. The vector fields f and columns g are supposed to satisfy the classical smoothness assumptions with $f(0) = 0$. Adding to the previous classical assumptions, we assume that system (8) is affected by the following types of actuator faults.

Assumption 1. In this paper we assume two types actuator faults.

- An additive actuator fault enters the system in such a way that the faulty model can be written:

$$\dot{x} = f(x) + g(x)(u + F(x,t)) \quad (9)$$

  where $F(x,t)$ is bounded by an unknown positive constant D_m i.e

$$\|F(x,t)\| < D_m. \quad (10)$$

- The actuator loss-of-effectiveness is represented by a multiplicative matrix α which affects the performance of each actuator in such a way that:

$$\dot{x} = f(x) + g(x)\alpha u \quad (11)$$

Where $\alpha \in \mathbf{R}^{m\times m}$ is a diagonal continuous time varying matrix whose diagonal elements α_{ii}, $i = 1,...,m$ are **unknown** and defined as: $0 < \epsilon \leq \alpha_{ii} \leq 1$

Combining (9) and (11) the global fault model of system (8) can be written as follows:

$$\dot{x} = f(x) + g(x)(\alpha u + F(x,t)) \quad (12)$$

4. FAULT-TOLERANT SUPER TWISTING CONTROLLER DESIGN FOR DIESEL ENGINE AIR PATH

4.1 Background

Consider the nonlinear SISO uncertain system:

$$\begin{cases} \dot{x} = f(x) + g(x)u \\ S(x,t) \end{cases} \quad (13)$$

The control objective is to steer the sliding variable S to 0 in finite time. Suppose that S admits a relative degree equal to 1, differentiating S with respect to time leads to:

$$\dot{S} = \frac{\delta S}{\delta t} + \frac{\delta S}{\delta x}\dot{x} = \frac{\delta S}{\delta t} + \frac{\delta S}{\delta x}f(x) + \frac{\delta S}{\delta x}g(x)u \quad (14)$$

which leads to:

$$\dot{S} = \Psi(x,t) + \varphi(x,t)u \quad (15)$$

with:

$\Psi(x,t) = \frac{\delta S}{\delta t} + \frac{\delta S}{\delta x}f(x)$, $\varphi(x,t) = \frac{\delta S}{\delta x}g(x)$ where $\Psi(x,t)$, $\varphi(x,t)$ are uncertain and bounded functions that satisfy:

$$\begin{cases} \Phi > 0, |\Psi(x,t)| < \Phi \\ 0 < \Gamma_1 < \varphi < \Gamma_2 \end{cases} \quad (16)$$

The following control law $u = u_1 - \beta_1|S|^\rho sign(S)$ with:

$$\dot{u}_1 = \begin{cases} -\alpha_1 sign(S) & if \quad |u| \leq U_m \\ -Um & if \quad |u| > U_m \end{cases} \quad (17)$$

Where U_m is a real constant and α_1, β_1 are chosen as follows Fridman and Levant (2002):

$$\begin{cases} \alpha_1 > \dfrac{\Phi}{\Gamma_1} \\ \beta_1^2 \geq \dfrac{4\Phi}{\Gamma_1^2}\dfrac{\Gamma_2(\alpha_1+\Phi)}{\Gamma_1(\alpha_1-\Phi)} \quad 0 \leq \rho \leq 0.5 \end{cases} \quad (18)$$

ensures the convergence of the sliding variable S toward zero in finite time.

4.2 Diesel engine air path control strategy

In this paper, the proposed air path control strategy operates under the diesel conventional combustion mode conditions. This particular mode was characterized by the author in Wang (2008). In this work the author suggested that for an optimal control performance, compressor mass flow W_c and Exhaust pressure manifold P_2 are suitable choice for key output variable to be controlled. By a suitable change of coordinates, the authors in Ali et al. (2012) proposed to replace the compressor mass flow set-point (W_{cd}) into an intake manifold pressure set-point (P_{1d}). This transformation simplifies the control structure by defining new vector set-point(P_{1d}, P_{2d}). The goal now, is to find a closed-loop controls which tracks these two variables.

4.3 Problem statement

consider system (1-7)and define the following two sliding variables S_1, S_2

$$\begin{cases} S_1 = p_1 - p_{1d} \\ S_2 = p_2 - p_{2d} \end{cases} \quad (19)$$

The time derivative of S_1, S_2 along the trajectories of system (1-7) leads to:

$$\begin{cases} \dot{S}_1 = k_1 W_c + k_1 u_1 - k_1 k_e p_1 - \dot{p}_{1d} \\ \dot{S}_2 = k_2 k_e p_1 + k_2 W_f - k_2 u_1 - k_2 u_2 - \dot{p}_{2d} \end{cases} \quad (20)$$

Now assume that the actuator faults described in (9) and (11) affects the VGT and the EGR valves. Following (12) the faulty diesel engine air path model is rewritten as follows:

$$\begin{cases} \dot{S}_1 = k_1 W_c + k_1\alpha_1 u_1 - k_1 k_e p_1 + k_1 F_1 - \dot{p}_{1d} \\ \dot{S}_2 = k_2 k_e p_1 + k_2 W_f - k_2\alpha_1 u_1 - k_2\alpha_2 u_2 - k_2(F_1+F_2) - \dot{p}_{2d} \end{cases} \quad (21)$$

Here (α_1, α_2), characterize the amount of the loss-of-effectiveness model which affects the EGR and the VGT actuators. (F_1, F_2) characterize the additive faults which also affects the same actuators.

Remarks 1. In (21) the additive faults terms F_1, F_2 characterize physically a leakage which can occurs on both EGR and VGT valves. So, it is quite realistic to consider such a type of faults in the faulty diesel air path system.

Remarks 2. The actuator loss-of-effectiveness $\alpha_{1,2}$ in (21) characterize the actuator capability to achieve the control requirement. For example if $\alpha_i = 1$ we have a healthy actuator, if $\alpha_i < 1$, the actuator is working partially. This physically happen when the opening of the EGR or the VGT valves is restricted for some reason

Problem statement:
Consider the faulty system (21) and the vector of sliding surface $S = (S_1, S_2)^T$, find a stabilizing closed-loop control which guarantee finite time convergence of S toward zero.

4.4 Control design

In this section a super-twisting controller is designed for system (21) based on the following assumptions.

Assumption 2. All the states of system (21) are available for measurements at every instant.

Assumption 3. The uncertainties on the set of model parameters P=(k_1 , k_2 , k_c , k_e , k_t , τ , η_m , μ) enters system (21) in additive way, i.e for a model parameters P_i

$$P_i \subset P, P_i = P_{oi} + \delta P_{oi}$$

where $P_{oi}(1 \leq i \leq 7)$ is the nominal value of the parameter, δP_{oi} the uncertainty on the model parameter.

Assumption 4. The additive faults F_1, F_2 are uniquely time-dependent.

Assumption 5. We assume that α_1, α_2 are successfully diagnosed by FDM (Fault Detection Mechanism)

Consider now the faulty system (21) with the sliding manifold $S = [S_1 S_2]^T$. It is clear that the relative degree of S with respect to control inputs $u = [u_1 u_2]^T$ is equal to 1. The dynamics of S takes the following form:

$$\dot{S} = A(X, F, P_1, X_d) + B(\alpha, P_2)u \tag{22}$$

Where $X = [p_1, p_2, W_c]^T$, $F = [F_1, F_2]^T$, $P_1 = [k_1, k_2, k_e]$, $X_d = [p_{1d}, p_{2d}]^T$, $\alpha = [\alpha_1, \alpha_2]^T$, $P_2 = [k_1, k_2]^T$ and

$$A = \begin{pmatrix} k_1 W_c - k_1 k_e p_1 + k_1 F_1 - \dot{p}_{1d} \\ k_2 k_e p_1 + k_2 W_f - k_2(F_1 + F_2) - \dot{p}_{2d} \end{pmatrix}$$

$$B = \begin{pmatrix} \alpha_1 k_1 & 0 \\ -\alpha_1 k_2 & -\alpha_2 k_2 \end{pmatrix}$$

However system (22) is coupled with respect to the control inputs $u = [u_1 u_2]^T$. To decouple the system (22) with respect to the control input, we apply the following transformation which defines a new sliding manifold S^*.

$$S^* = B(\alpha, P_2)^{-1} S \tag{23}$$

Differentiating S^* with respect to time yields to:

$$\dot{S}^* = \dot{B}(\alpha, P_2)^{-1} S + B(\alpha, P_2)^{-1} \dot{S} \tag{24}$$

For a given operating point and assume small variations of α around constant value α_0, $B(\alpha, P_2)^{-1}$ is constant, we can easily derive:

$$\dot{S}^* = B(\alpha, P_2)^{-1} A + u \tag{25}$$

Which can be written as follows:

$$\begin{cases} \dot{S}_1^* = -\dfrac{1}{\Delta(B)} \alpha_2 k_2 A_{11} + u_1 \\ \\ \dot{S}_2^* = \dfrac{1}{\Delta(B)} (\alpha_1 k_2 A_{11} + \alpha_1 k_1 A_{21}) + u_2 \end{cases} \tag{26}$$

Where $\Delta(B) = -\alpha_1 \alpha_2 k_1 k_2$, $A_{11} = k_1 W_c - k_1 k_e p_1 + k_1 F_1 - \dot{p}_{1d}$, $A_{21} = k_2 k_e p_1 + k_2 W_f - k_2(F_1 + F_2) - \dot{p}_{2d}$.
Taking $\Psi_1(x, t) = -\frac{1}{\Delta(B)} \alpha_2 k_2 A_{11}$ and $\Psi_2(x, t) = \frac{1}{\Delta(B)} (\alpha_1 k_2 A_{11} + \alpha_1 k_1 A_{21})$, we derive:

$$\begin{cases} \dot{S}_1^* = \Psi_1(x, t) + u_1 \\ \\ \dot{S}_2^* = \Psi_2(x, t) + u_2 \end{cases} \tag{27}$$

Obviously a passive fault-tolerant control based on the super twisting algorithm (15-18) can be developed for system (27). The proposed STA controller for system (27) takes the following form:

$$\begin{cases} u_1 = U_1 - \beta_1 |S_1^*|^{1/2} sign(S_1^*) \\ \dot{U}_1 = -\alpha_1 sign(S_1^*) \\ u_2 = U_2 - \beta_2 |S_2^*|^{1/2} sign(S_2^*) \\ \dot{U}_2 = -\alpha_2 sign(S_2^*) \end{cases} \tag{28}$$

with $\lambda_1, \lambda_2, \beta 1, \beta 2$ satisfying the following conditions:

$$\begin{cases} \lambda_1 > C_1 \\ \beta_1^2 \geq \dfrac{4C_1(\alpha_1 + C_1)}{(\alpha_1 - C_1)} \\ \lambda_2 > C_2 \\ \beta_2^2 \geq \dfrac{4C_2(\alpha_2 + C_2)}{(\alpha_2 - C_2)} \end{cases} \tag{29}$$

Where C_1, C_2 will be defined below. In the next section we state our main results.

4.5 Main result

Theorem 1. Consider the uncertain faulty system (21). The passive fault tolerant STA controller (28) under conditions (29) ensure that the sliding manifolds $[S_1, S_2]$ converges asymptotically to zero in finite time.

Proof. The stability and the convergence of the proposed controller (28) is based on the convergence of the super twisting algorithm (17-18). First we notice that to prove the convergence of the sliding manifolds $[S_1, S_2]$, we need to prove the convergence of the sliding manifolds $[S_1^*, S_2^*]$. indeed from (23) we can see that S^* results from a linear combination of S (B^{-1} is a constant matrix) thus, if $S^* \longrightarrow 0$ then $S \longrightarrow 0$. Since System (27) is decoupled with respect to the control inputs, we can prove the convergence of the sliding manifolds S_1^* and S_2^* separately. Consider now the sliding manifold dynamic of S_1^* which is described in (27)(the proof is exactly the same for the sliding manifold S_2^*). $\Psi_1(x, t)$ (res $\Psi_2(x, t)$) involves the parametric uncertainties and the actuator faults described in assumptions 1 and 2, recall that the set $\Omega = (p_1, p_2, P_c)$ is invariant and physically bounded, there exists a constant C_1 (resp C_2) such that the following inequalities holds:

$$\begin{cases} |\Psi_1(x, t)| \leq C_1 \\ |\Psi_2(x, t)| \leq C_2 \end{cases} \tag{30}$$

As a consequence equation (30) implies the following differential inclusions which is understood in the Filippov sense Filippov (1988) :

$$\begin{cases} \dot{S}_1^* \in [-C_1, C_1] + u_1 \\ \dot{S}_2^* \in [-C_2, C_2] + u_2 \end{cases} \tag{31}$$

Thus a super twisting controller can be designed for system (31) following the control structure described in (17) under conditions (18). This complete the proof.

5. SIMULATION AND EVALUATION

In this section, we report numerical results obtained from the simulation of controller (28) on the reduced third order model developed in (1-7). Numerical simulations were performed in real-time Software In the Loop (SIL) using the dSpace modular simulator. This real-time platform is based on the DS-1006 board interfaced with Matlab/Simulink software. The engine used is a common rail direct-injection in-line-4-cylinder provided by a French manufacturer. Numerical values of (η_t,η_c,η_v,η_m) cartographies in the TDE model were provided by the manufacturer. The values of the model parameters k_1, k_2, k_c, k_e, k_t, τ, η_m and μ are usually identified around some given operating points. In these simulations, the parameters of the model (1-7) were taken from (Larsen et al. (2000)) i.e (k_1=143.91, k_2=1715.5, k_c=0.0025, k_e=0.028, k_t=391.365, τ=0.15, η_m=0.95, μ=0.285). To avoid the chattering associated with sliding motion, a well-known continuous approximation of the function $sign(S)$ is given by:

$$Sign(S) = \frac{s}{|s| + \xi} \tag{32}$$

This approximation is used to ensure that the sliding motion will be in the vicinity of the line ($S = 0$). In this simulation the approximation of the $sign$ function has been implemented with $\xi = 0.01$. Moreover the the sampling step time for all the simulations is the same, 10^{-4}. In what follows, the robustness and fault tolerance of the STA controller are discussed.

5.1 Case 1: 20 % parametric uncertainties only:

In this case, we consider a healthy actuators. The purpose here is to show the performance of controller (28) facing only parametric uncertainties in the TDE air path model. The nominal parameters values for this model were varied with a maximum of 20 % increase in each. In this simulation, the controller parameters were chosen such as: $\lambda_1, \beta_1, \lambda_2, \beta_2$=[0.3,0.3,0.3,0.3].

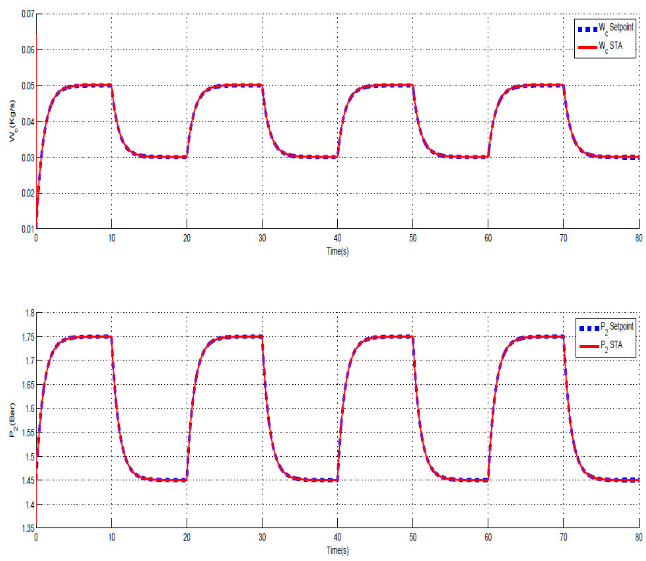

Fig. 2. W_c and P_2 tracking performance for the STA with only 20 % parametric uncertainties in the TDE air path model.

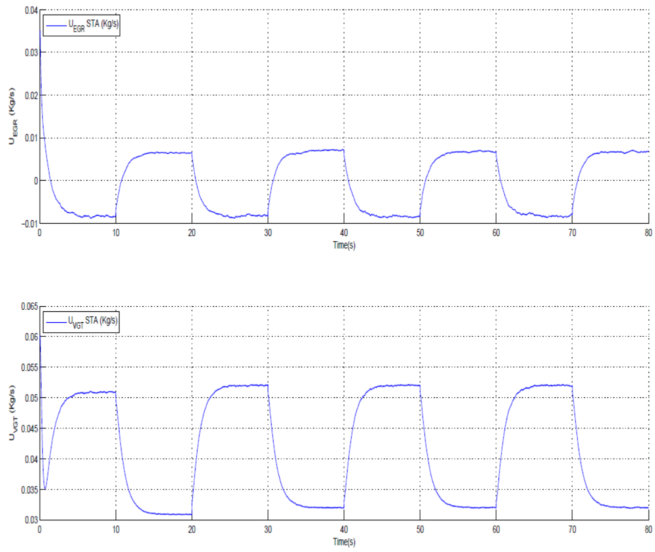

Fig. 3. Control inputs U_{EGR} and U_{VGT} for the STA with only 20 % parametric uncertainties in the TDE air path model.

Fig. 2. shows the tracking performance of the controlled outputs W_c and P_2. We can observe that the desired W_c and P_2 trajectories (blue dashed lines), are being tracked with good performances and small tracking error. Remembering that Assumption (3) assume that parametric uncertainties enters system (21) in additive way, Fig. 3. shows the consistency of control efforts ($u_1 = U_{egr}, u_2 = U_{vgt}$) produced by controller (28). It is clear that the structure of controller (28) is robust facing parametric uncertainties since the control gains ($\lambda_1, \beta_1, \lambda_2, \beta_2$) does not depends on these parameters. RMS of Control Effort (CE) and Tracking Error (TE) of this case will serve as basis of comparison of the next cases (see Tables 2.).

5.2 Case 2: Additive leakage time varying faults and multiplicative loss-of-effectiveness time varying faults in EGR and VGT actuators with 20% parametric uncertainties:

In this case, we evaluate the performance of controller (28) by considering both of additive and multiplicative time varying faults (leakage and loss-of-effectiveness) added to 20% parametric uncertainties. We consider a fault scenario where:

- At t=25 s, the additive leakage time variying fault $F(t)$ which affect the EGR and the VGT actuators, takes the following form:

$$F(t) = \begin{cases} 0 \times [1,1]^T & if \quad t < 25s \\ -0.05 + 0.02\sin(0.2\pi t) \times [1,1]^T & if \quad t \geq 25s \end{cases}$$

- At t=55 s, the multiplicative loss-of-effectiveness time varying fault occurs in the EGR and the VGT actuators following this model:

$$\alpha = \begin{cases} I_{2\times2} & if \quad t < 55s \\ 0.2 + 0.05\sin(0.2\pi t) \times I_{2\times2} & if \quad t \geq 55s \end{cases}$$

In this simulation, the controller parameters were chosen such as: ($\lambda_1, \beta_1, \lambda_2, \beta_2$)=[0.3,0.3,0.3,0.3].

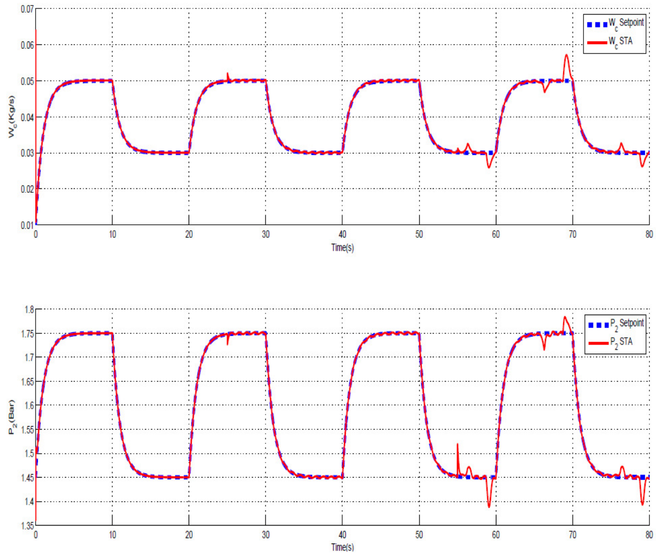

Fig. 4. W_c and P_2 tracking performance for the STA with Additive leakage time varying faults and multiplicative loss-of-effectiveness time varying faults in EGR and VGT actuators.

We can observe from the tracking performances of the controlled outputs W_c and P_2 shown in Fig. 4. that the STA controller exhibits an oscillatory behavior starting from t = 25s with a small magnitude. After t = 55s, we also observe a random behavior with some peaks of divergence that the controller try to reduce but still unable to completely eliminate.
Fig. 5. shows the control efforts ($u_1 = Uegr, u_2 = Uvgt$) produced by controller (28). We can see that the STA controller (28) tried to compensate the leakage at t=25s and the loss-of-effectiveness at t=55s.

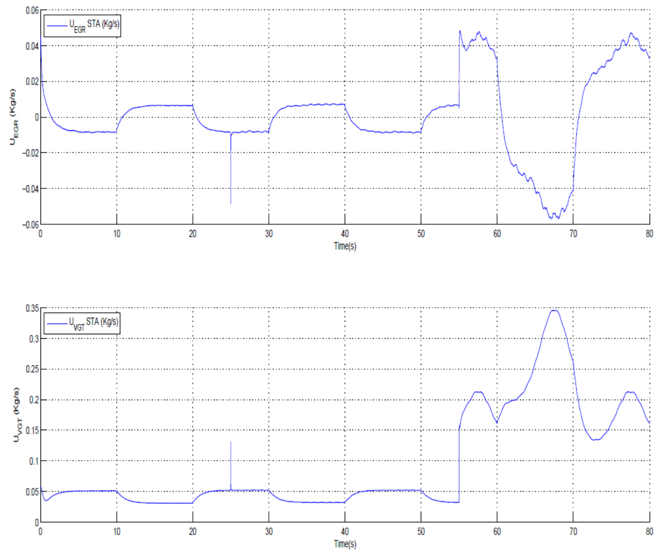

Fig. 5. Control inputs U_{EGR} and U_{VGT} for the STA with Additive leakage time varying faults and multiplicative loss-of-effectiveness time varying faults in EGR and VGT actuators.

Looking to Table2., we can see an increase of the CE since the controller tries to reduce the additive and loss-of-effectiveness faults considered in this case. The RMS of TE reflect the impotence of the STA to reject completely the faults with similar gains as it was in case 1.

5.3 Case 3: Additive leakage time varying faults and multiplicative loss-of-effectiveness time varying faults in EGR and VGT actuators with 20% parametric uncertainties:

To remedy the problems of robustness and fault tolerance found in case 2, we propose to increase the gains of the STA controller. So we consider the same fault scenario in case 2 by changing the controller gains to $\lambda_1, \beta_1, \lambda_2, \beta_2$=[3,3,3,3].

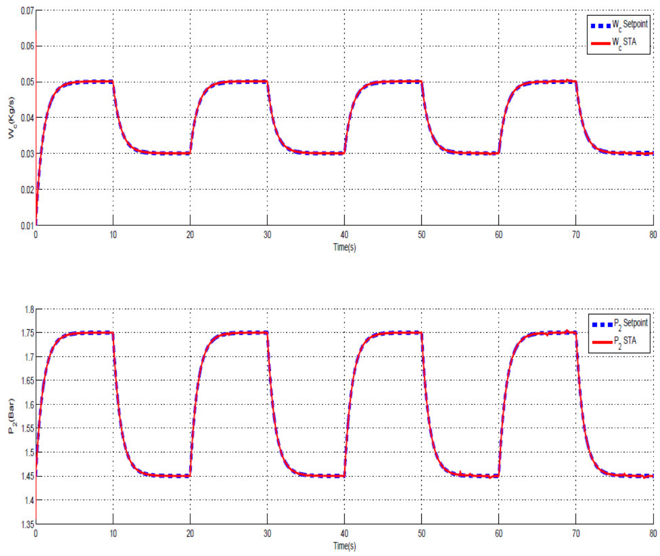

Fig. 6. W_c and P_2 tracking performance for the STA with Additive leakage time varying faults and multiplicative loss-of-effectiveness time varying faults in EGR and VGT actuators [improved gains].

We can observe from Fig. 6. that the desired W_c and P_2 trajectories (blue dashed lines), are being tracked with good performances and small tracking error. We can see that the oscillatory behavior starting from t = 25s and the peaks of divergence after t = 55s have been considerably reduced.
The leakage (additive time varying faults) and the actuator loss-of-effectiveness (multiplicative time varying faults) are completely rejected from the states trajectories by the STA Controller.

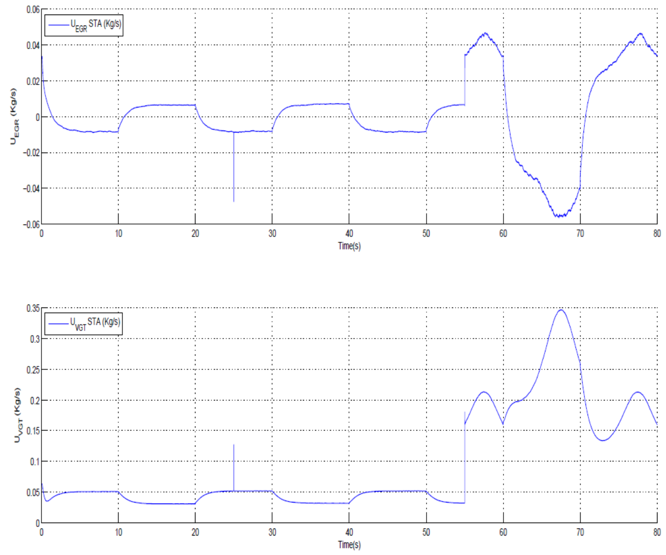

Fig. 7. Control inputs U_{EGR} and U_{VGT} for the STA with Additive leakage time varying faults and multiplicative loss-of-effectiveness time varying faults in EGR and VGT actuators [improved gains].

Fig. 7. shows the control efforts $(u_1 = U_{egr}, u_2 = U_{vgt})$ produced by controller (28). We can see that the STA controller (28) managed to compensate the leakage at t=25s and the loss-of-effectiveness at t=55s. We can observe that the RMS of TE in Table 2. is considerably reduced comparing to the case 2, where the CE is slightly increased. This demonstrate the success of the STA controller to completely reject the actuators faults.

Table 2. RMS of Tracking Error (TE) and Control Effort (CE)

	RMS CE EGR	RMS CE VGT	RMS TE (W_c)	RMS TE (P_2)
Case 1	0.0133	0.0439	3.0965e-004	0.0036
Case 2	0.1143	0.1033	1.6044e-003	0.0309
Case 3	0.1243	0.1099	1.3424e-004	5.9371e-004

6. CONCLUSIONS AND FUTURE WORK

In this paper, a fault-tolerant STA controller is designed for controlling the diesel engine air path. Numerical simulations show that the proposed controller is fault-tolerant when a leakage and loss-of effectiveness affects both the EGR and the VGT actuator valves. In our future work we plan to propose an active fault-tolerant control scheme for the TDE, combining a fault estimator and the proposed STA controller.

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