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Control of a turbocharged Diesel engine with EGR system using Takagi-Sugeno's approach

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***Abstract*— Environmental issues and respect for the European Union's objectives for green house emissions reduction, has required an improved in the sophistication of control strategies. This article focuses on the control of the air path diesel engine. Most studies consider the following control setup: intake pressure and intake air flow are controlled using EGR and VGT valves. A simple state feddback control law strategy based on a nonlinear control Lyapunov function and $\mathcal{D}$-stabilisation is proposed. $\mathcal{LMI}$ terms have been developed. The goal is to achieve the reference tracking of desired values of intake and exhaust pressures. First at all, a description of the airpath system is given, then, the mean value model of the diesel engine is presented and converted into a Takagi-Sugeno'model. The model was validated with the AMEsim software. Finally, to ensure the performance of the proposed approach, simulation results are presented .**

I. INTRODUCTION

To respect the novel and strict environmental legislative regulations, a new type technology of Homogeneous Charge Compression Ignition (HCCI) combustion with Variable Geometry Turbocharger (VGT) and Exhaust Gas Recirculation (EGR) systems can be used in diesel engines to reduce drastically the Nitrogen oxides emissions.

Since July 1992, date of entry into force of the emission standard EURO 1, the automakers are facing to a problem: designing vehicles rejecting a minimum of pollutants. This standard limitation of the amount of pollutants released by motor vehicles, is defined every 4 years by the European Union and becomes more and more stringent. It is therefore a visible challenge, with the only objective decreasing pollutant emissions. Diesel engines are now equipped with controllable actuators VGT and EGR valves. These actuators have reached a level of sophistication for better control of combustion. The principle of EGR is to inject burned gases into the intake collector to reduce the formation of certain pollutants (particularly nitrogen oxides). The diesel engine is not anymore the seat of a continuous combustion, but the succession of combustion synchronized with the movements of the piston. The exhaust gases are evacuated between the different times of burning and fresh air is bring into the cylinders through the valves. The basic period of this sequence is called a cycle. A cycle consists of four phases: intake, compression, expansion and exhaust.

The purpose of controlling a diesel engine is to provide the torque required by the user while minimizing emissions and noise.

A primal concern for these new technological integrations in diesel engines lies in the control of the air intake system gas pressure and mass air flow. Accurate references tracking allow the diesel engine to avoid stall and keep pollutant air emissions and noise below acceptable level. Under this context, a multi-input and multi-output controller is proposed to guarantee fast and accurate intake system dynamics trajectory tracking. This referred controller uses a normalized VGT and EGR actuators positions respectively A_{vgt} and A_{egr} as control input signals. In this article, gas flow rates from the EGR and turbine are directly controlled. The fueling mass rate W_f and the engine crankshaft speed N are considered as external inputs. The entire proposed diesel engine model and the controller are extended to be tested and validated on a four-cylinder diesel engine model running on AMEsim ®platform in co-simulation with Matlab ®Simulink. The AMEsim model used for simulation has been validated on an engine tested developed by IFP (Institut Français de Pétrole). The efficiency of the proposed method is demonstrated by co-simulation results.

Many propositions of mathematical models of diesel engine have been proposed since the 60's [1], [2], [9], [12]. The model should be chosen carefully to ensure the best description of the system. In this paper, a third order nonlinear model is considered [10] using the conservation of mass and energy, the ideal gas law for modeling the intake and exhaust manifold pressure dynamics, and a first order differential equation with the time constant τ for modeling the power transfer dynamics of the VGT.

The paper is structured as follows. In section II, the air path diesel engine system is described. A detailed description of the nonlinear mean value engine model to be controlled and the validation of this model with AMEsim (LMS) simulator is also proposed in section II. In section III, the modeling of the air path system of turbocharged diesel engine by a T-S fuzzy model is presented. Finally, the development of the control tools is given in section IV. Numerical illustration are proposed in section V whereas conclusion and future works are summarized in section VI.

Notations : We denote by M', the transpose conjugate of M,

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by $\mathcal{H}(M)$ the Hermitian expression $M+M'$. The Kronecker product is denoted by $\otimes$. $\mathbb{I}_n$ is the identity matrix of order n, $\mathbb{O}$ is the null matice of suitable dimension. At last, $\tilde{z}$ is the conjugate of the scalar number z.

II. System descriptions

A. Preliminaries

A schematic diagram of the considered turbocharged diesel engine including the EGR and VGT is shown in Figure 1. At the bottom of the diagram is the turbocharger consisting of a variable geometry turbine and a compressor connected to the same shaft. The turbine takes the energy from the exhaust gas to power the compressor. Frech air and the Exhaust Gas Recirculation (EGR) flows reach the intake manifold and are aspirated into the cylinders. The fuel is injected directly into the cylinders and burned, producing the torque on the crank shaft. The hot exhaust gas is pumped out into the exhaust manifold. Part of the exhaust gas flows from the exhaust manifold are ejcted from the engine through the turbine. The other part is recirculated back into the intake manifold. The diagram also shows the intercooler and the EGR-cooler that are used to reduce the intake manifold temperature. Coolers are not incuded in the model. The nomenclature of the considered variables is summarized into the Table I.

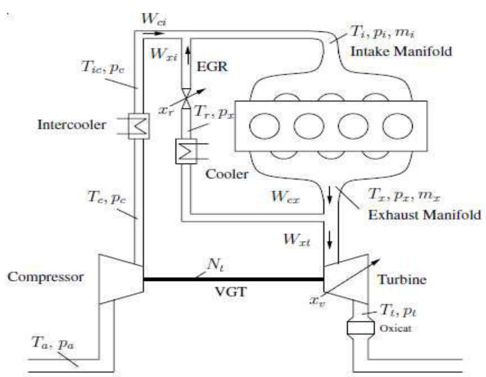

Fig. 1. Schematic diagram of a turbocharged diesel engine with EGR

B. Modeling of Turbocharged Diesel Engines

Many references of the literature consider mean value engine modeling approaches [10]. This model uses temporal and spatial averages of relevant temperatures, pressures and mass flow rates. It leads to a seven state reference model which described the gas mass changes, pressures and burnt gas fraction dynamics in the intake and exhaust manifolds. The seventh parameter is for the dynamic of the turbocharger.

This seventh-order model can be reduced to a third-order one under specific hypotheses: intake and exhaust manifolds burned gas fractions are difficult to measure. It is the case in this paper and these parameters are not considered in the model. For the same reasons the intake and exhaust burned gas mass fractions are ignored. The turbocharger dynamics are modeled as a first order lag power transfer with a time constant τ. Therefore, the third-order model is described by the equation (1):

TABLE I
Nomenclature. i.m and e.m refers to the intake manifold and exhaust manifold

Variable	Description	Value.Unit
EGR	Exhaust Gas Recirculation	-
AFR	Air Fuel Ratio	-
N	Engine speed	1500 tr/min
p_i	Gas pressure in the intake manifold	Pa
p_x	Gas pressure in the exhaust manifold	Pa
P_c	Compressor power	-
P_t	Turbine power	-
W_{xi}	EGR mass flow rate	$Kg.s^{-1}$
W_{xt}	turbine mass flow rate	$Kg.s^{-1}$
W_{ci}	Compressor mass flow rate	$Kg.s^{-1}$
W_f	Fuel mass flow rate	6 Kg/h
W_{ie}	Total mass flow rate into the engine	-
V_i	Volume of the i.m	$0.006m^3$
V_x	Volume of the e.m	$0.001m^3$
T_a	ambient temperature	300
T_i	temperature in the i.m	313
T_x	temperature in the e.m	519
p_a	ambient pressure	$101.3Pa$
c_p	heat at constant pressure	$1014.4\frac{J}{KgK}$
c_ν	heat at constant volume	$727.4\frac{J}{KgK}$
η_c	compressor efficiency	0.61
η_t	turbine efficiency	0.76

$$\begin{aligned} \dot{p}_i &= \frac{RT_i}{V_i}(W_{ci}+W_{xi}-W_{ie}) \\ \dot{p}_x &= \frac{RT_x}{V_x}(W_{ie}-W_{xi}-W_{xt}+W_f) \\ \dot{P}_c &= \frac{1}{\tau}(-P_c+\eta_m P_t) \end{aligned} \tag{1}$$

Only intake and exhaust manifold pressures and turbocharger power dynamics are considered. τ, η_m, V_i and V_x represent respectively the time constant, the mechanical efficiency, and the volumes of the intake and exhaust manifolds. P_t is the turbine power modeled by

$$P_t = W_{xt}c_pT_x\eta_t(1-(\frac{p_a}{p_x})^{\mu}) \tag{2}$$

η_t is the turbine efficiency, whereas η_c, T_a, are respectively the compressor efficiency and the ambient temperature.
c_p, c_ν, are the heat at constant pressure and volume, $\mu = \frac{c_p-c_\nu}{c_p}$ is a constant and p_a is the ambiant pressure. Numerical values of these parameters are known (see Table I).
W_{ci} describes the relation between the flow through the compressor and the power. This relation is modeled as

$$W_{ci} = \frac{\eta_c}{c_pT_a}\frac{P_c}{(\frac{p_i}{p_a})^{\mu}-1} \tag{3}$$

As shown in figure 1, W_{xi} describes the flow through the EGR valve which is modeled by the standard orifice flow equation.

$$W_{xi} = \begin{cases} \frac{A_{egr}(x_{egr})p_x}{\sqrt{RT_x}} \\ \times [\gamma^{\frac{1}{2}} \frac{2}{\gamma+1}^{\frac{\gamma+1}{2(\gamma-1)}}], \quad \frac{p_x}{p_i} \leq \frac{2}{\gamma+1}^{\frac{\gamma+1}{2(\gamma-1)}} \\ \frac{A_{egr}(x_{egr})p_x}{\sqrt{RT_x}} \sqrt{\frac{2\gamma}{\gamma-1}} \\ \times [\frac{p_x}{p_i}^{\frac{2}{\gamma}} - \frac{p_x}{p_i}^{\frac{\gamma+1}{\gamma}}], \quad \frac{p_x}{p_i} > \frac{2}{\gamma+1}^{\frac{\gamma+1}{2(\gamma-1)}} \end{cases} \quad (4)$$

$A_{egr}(x_{egr})$ is the effective area of the EGR valve, T_x is the exhaust manifold temperature, and R is the gas constant (T_x and R are known).
γ is the specific heat ratio ($= \frac{c_p}{c_v}$). The γ value of air is 1.394.
The flow W_{ie} from the intake manifold into the cylinders is modeled by the speed-density equation

$$W_{ie} = \eta_\nu \frac{p_i N V_d}{120 T_i R} \quad (5)$$

With $\eta_\nu = 0.87$, the volume efficiency and $V_d = 0.002m^2$ the displacement volume.
The turbine flow W_{xt} is given by

$$W_{xt} = \frac{A_{vgt}(x_{vgt})p_x}{\sqrt{T_x}} \sqrt{\frac{2p_a}{p_x}(1 - \frac{p_a}{p_x})} \quad (6)$$

Combining the equations of the simplified nonlinear model (1) and the flow equations above (2), (3), (4), (5) and (6), the following nonlinear differential equations are obtained

$$\begin{aligned} \dot{p}_i &= \frac{RT_i}{V_i}(\frac{\eta_c}{c_p T_a} \frac{P_c}{(\frac{p_i}{p_a})^\mu - 1} + \frac{A_{egr}(x_{egr})p_x}{\sqrt{RT_x}} \sqrt{\frac{2p_i}{p_x}(1 - \frac{p_i}{p_x})} \\ &- \eta_\nu \frac{p_i N V_d}{120 T_i R}) \\ \dot{p}_x &= \frac{RT_x}{V_x}(\eta_\nu \frac{p_i N V_d}{120 T_i R} - \frac{p_x}{\sqrt{RT_x}} \sqrt{\frac{2p_i}{p_x}(1 - \frac{p_i}{p_x})} A_{egr}(x_{egr}) \\ &- \frac{A_{vgt}(x_{vgt})p_x}{\sqrt{T_x}} \sqrt{\frac{2p_a}{p_x}(1 - \frac{p_a}{p_x})} + W_f) \\ \dot{P}_c &= \frac{1}{\tau}[-P_c + \eta_m \eta_t c_p T_x (1 - (\frac{p_a}{p_x})^\mu) \times \\ &\frac{p_x}{\sqrt{T_x}} \sqrt{\frac{2p_a}{p_x}(1 - \frac{p_a}{p_x})}] A_{vgt}(x_{vgt}) \end{aligned} \quad (7)$$

Remark 1

1) The engine speed N and the fueling rate W_f are considered as known external parameters.
2) The EGR position and the VGT position range between 0 and 1.

C. Model validation using AMESIM

In this part, the model (7) is validated with AMEsim®platform. AMESim software is a virtual simulator for the modeling and analysis of systems, that allows to reproduce the dynamic behavior of engines in various scenarios. AMEsim software was used in order to run the engine model in co-simulation with SIMULINK. That's mean running the two separate models with the two different software at the same time. Fig. 2 shows a brief overview of co-simulation.

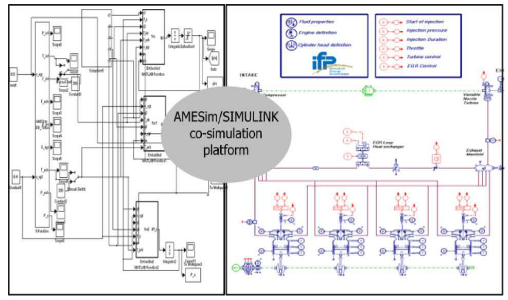

Fig. 2. Co-simulation AMEsim/SIMULINK schematic

The four-cylinder diesel engine model from the AMEsim®platform and the one from SIMULINK were both simulated with the same effective areas, A_{egr} and A_{vgt}. Figures 3 and 4 provides a comparison of the intake and exhaust pressure responses obtained by the two models.

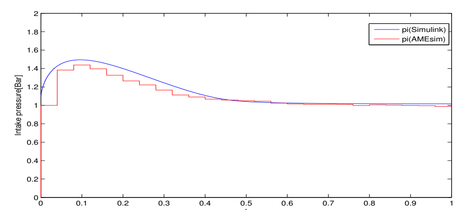

Fig. 3. Intake manifold pressure response

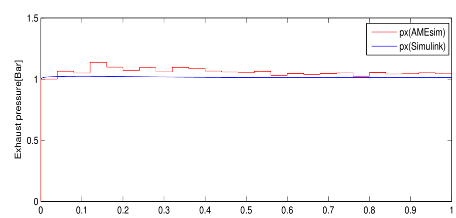

Fig. 4. Exhaust manifold pressure response

III. Takagi-Sugeno's fuzzy representation

In this part, a TS fuzzy representation of the nonlinear model given by (7) is proposed. For this TS representation, it is supposed that flows through the turbine and EGR are directly controlled.

The nonlinear system (1), with relations (2), (3) and (5), can be represented in the state space such as:

$$\begin{bmatrix} \dot{p}_i \\ \dot{p}_x \\ \dot{P}_c \end{bmatrix} = A(\rho(t)) \begin{bmatrix} p_i \\ p_x \\ P_c \end{bmatrix} + B(\rho(t)) \begin{bmatrix} W_{xi} \\ W_{xt} \end{bmatrix} + D W_f$$

$$y(t) = C\,x(t) \;=\; \begin{bmatrix} \mathbb{I}_2 & \mathbb{O} \end{bmatrix} x(t) \;=\; \begin{bmatrix} p_i \\ p_x \end{bmatrix}.$$

with

$$\begin{cases} A(\rho(t)) = \begin{bmatrix} -a_k a_2 & 0 & a_k a_1 \rho_1 \\ b_k a_2 & 0 & 0 \\ 0 & 0 & -c_k \end{bmatrix} \\ \\ B(\rho(t)) = \begin{bmatrix} a_k & 0 \\ -b_k & -b_k \\ 0 & c_k c_1 \rho_2 \end{bmatrix} \;;\; D = \begin{bmatrix} 0 \\ b_k \\ 0 \end{bmatrix} \\ \\ a_k = \dfrac{RT_i}{V_i},\; b_k = \dfrac{RT_x}{V_x} \text{ and } c_k = \dfrac{1}{\tau} \end{cases}$$

$\rho = [\rho_1\ \rho_2]$ is the time varying parameters vector with

$$\rho_1(p_i) = \frac{1}{(\frac{p_i}{p_a})^{\mu} - 1}, \quad \rho_2(p_x) \quad = 1 - (\frac{p_a}{p_x})^{\mu} \tag{8}$$

This T-S model representation was designed to be employed for control design purposes.

Assuming that p_i and p_x are bounded ($p_i \in [\underline{p_i}, \overline{p_i}]$ and $p_x \in [\underline{p_x}, \overline{p_x}]$), which means that

$$\forall i \in 1,..,n = 2, \quad \underline{\rho_i} \leq \rho_i \leq \overline{\rho_i} \tag{9}$$

There are n nonlinearities resulting in the TS model with $r = 2^n = 4$ linear model number.

The T-S fuzzy dynamic model is described by fuzzy IF-THEN rules [8], which locally represent linear input-output relations of nonlinear systems. A continuous fuzzy model can be described by

IF ρ_1 is M_{i1} and ρ_2 is M_{i2} Then

$\dot{x}(t) = A_i x(t) + B_i\ u \quad i = 1, 2, ..., 4.$

Let's:

$x(t) = \begin{bmatrix} p_i & p_x & P_c \end{bmatrix}^T$ and $u(t) = \begin{bmatrix} W_{xi} & W_{xt} \end{bmatrix}^T$

$$\dot{x}(t) = \sum_{i=1}^{r} h_i(\rho(t)) \Big\{ A_i x(t) + B_i u(t) \Big\} \tag{10}$$

Where $x(t) \in \mathbb{R}^n$ is the state vector, $u(t) \in \mathbb{R}^m$ is the control input, $\rho(t)$ is the premise vector and $h_i(\rho(t)) \geqslant 0$ are the nonlinear scalar functions and assumed to verify the convex sum property:

$$\sum_{i=1}^{r} h_i(\rho(t)) = 1 \tag{11}$$

The parameters $h_i(\rho(t))$ represent the nonlinear aspect of the system. Knowing the sub-models for each measure of $\rho(t)$, it is possible to reconstruct the multi-model. This work was completed with AMESim and Fig. 5 compares the signals generated by the multi model and those generated by the nonlinear model. Basing on the results of Fig. 5, it is assumed that the satisfaction of the multi-model representation is approved.

IV. Closed-loop Control of the air-path diesel engine

This section is dedicated to design a state feedback fuzzy controller using a fuzzy observer.

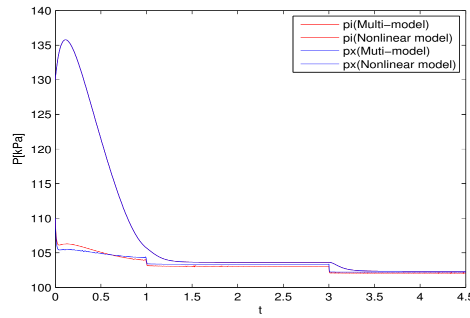

Fig. 5. Intake and exhaust manifold pressure responses of the nonlinear model and the TS model

A. Strategy of control

Turbocharged diesel engine control is a challenging task due to system nonlinearities and constraints on the inputs and process variables. Various control methods have been applied to the control of diesel engines [3], [7], [11]. The air feedback loop purpose is to control the air and burned gas masses in the cylinder. In practical applications, those variables can not be measured. Yet, equivalent variables can be considered. In this article, a nonlinear control Lyapunov function is applied. The idea is to minimize the error between the original output y and a suitable constructed reference input $\tilde{y}$ which the values to be tracked are specifically chosen. The closed-loop configuration is shown in Fig. 6. The integrator is added to eliminate the static error.

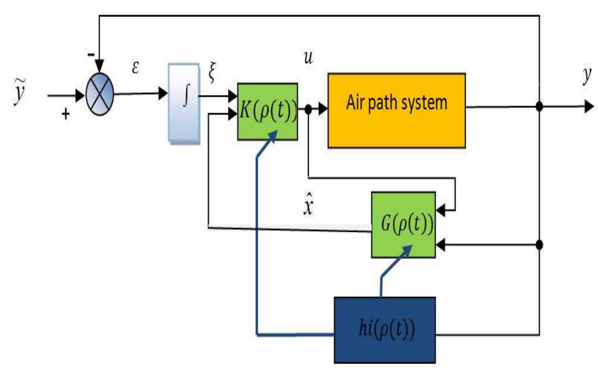

Fig. 6. Block diagram of the closed-loop of the air system

The observer dynamics are

$$\begin{aligned} \dot{\hat{x}}(t) &= \sum_{i=1}^{r} h_i(\rho(t)) \Big\{ A_i \hat{x}(t) + B_i u(t) + G_i(y(t) - \hat{y}(t)) \Big\} \\ \hat{y}(t) &= C\hat{x}(t) \end{aligned} \tag{12}$$

where G_i, $i = 1, .., r$ are the observers gains.

The state feedback control law inferred is

$$u(t) = -\sum_{i=1}^{r} h_i(\rho(t)) \Big\{ K_i \hat{X}(t) \Big\} \tag{13}$$

with

$$\hat{X}(t) = \begin{bmatrix} \hat{x}(t) \\ \xi(t) \end{bmatrix} \text{ and } K_i = \begin{bmatrix} K_{xi} & \vdots & K_{\xi i} \end{bmatrix}$$

The observer error is then defined by:

$$e(t) = x(t) - \hat{x}(t) \tag{14}$$

The tracking error is

$$\dot{\xi}(t) = \tilde{y}(t) - y(t) \tag{15}$$

Using (10), (12) and (14), it comes

$$\begin{aligned} \dot{x}(t) &= \sum_{i=1}^{r}\sum_{j=1}^{r} h_i(\rho(t))h_j(\rho(t)) \\ &\quad \Big\{(\overline{A_i} - \overline{B_i}K_j)x(t) + \overline{B_i}K_{xj}e(t)\Big\} \\ \dot{e}(t) &= \sum_{i=1}^{r}\sum_{j=1}^{r} h_i(\rho(t))h_j(\rho(t))\Big\{(A_i - G_jC)\Big\}e(t) \end{aligned} \tag{16}$$

with

$$\overline{A_i} = \begin{bmatrix} A_i & \mathbb{O} \\ -C & \mathbb{O} \end{bmatrix} \text{ and } \overline{B_i} = \begin{bmatrix} B_i \\ \mathbb{O} \end{bmatrix}$$

Then we can write the augmented system

$$\begin{aligned} \begin{bmatrix} \dot{x} \\ \dot{e} \end{bmatrix} &= \sum_{i=1}^{r}\sum_{j=1}^{r} h_i(\rho(t))h_j(\rho(t)) \times \\ &\quad \begin{bmatrix} \overline{A_i} - \overline{B_i}K_j & \overline{B_i}K_{xj} \\ \mathbb{O} & A_i - G_jC \end{bmatrix} \begin{bmatrix} x \\ e \end{bmatrix} \end{aligned} \tag{17}$$

The state matrix of the augmented system is triangular. This means that the problem can be treated as two completely independent steps:

1) stabilization pairs $(\overline{A_i}; \overline{B_i})$: controller gains;
2) stabilization pairs $(A_i; C)$: observer gains.

To ensure the convergence of the observer faster than the controller, we propose to make a pole placement. Thus, the main objective is not only to stabilize the pairs above but to $\mathcal{D}$-stabilize them. Here, $\mathcal{D}$ corresponds to a shifted vertical half plane or a disc centered on the real axis of the complex plane. When the whole spectrum of the state matrix is strictly clustered in the specified region $\mathcal{D}$, the system (or the state matrix) is said $\mathcal{D}$-stable. It means that the corresponding eigenvalues have to be placed in a region, here noted $\mathcal{LMI}$-region [4], of the complex plane. $\mathcal{D}$ is described by:

$$\mathcal{D} = \{z \in \mathbb{C} \mid \alpha + \beta z + \beta^T \tilde{z} \prec 0\} \tag{18}$$

B. Main result

In order to compute the gains K_i and G_i, the theorem of [6] is extended to the root clustering approach and more precisely to the $\mathcal{D}$-stabilization.

Theorem: Let consider $\mathcal{D}_k$ and $\mathcal{D}_g$ two $\mathcal{LMI}$-regions defined by (18). One region is for the controller gains, the other for observer gains. If there exist symmetric and positive definite matrices Q and P, some matrices M_i and N_i such that the following LMIs are satisfied $\forall (i,j) = 1,..,r$, $i < j$, then TS fuzzy system (17) is globally asymptotically $\mathcal{D}$-stable via TS fuzzy controller (13) based on fuzzy observers (12):

$$\phi_{ii} = \alpha_k \otimes Q + \boldsymbol{\mathcal{H}}\left(\beta_k \otimes (\overline{A_i}\, Q - \overline{B_i}\, M_i)\right) < 0 \tag{19}$$

$$\begin{aligned} \psi_{ij} &= \alpha_k \otimes Q + \boldsymbol{\mathcal{H}}\left(\beta_k \otimes (\overline{A_i}\, Q - \overline{B_i}\, M_j)\right) \\ &\quad + \boldsymbol{\mathcal{H}}\left(\beta_k \otimes (\overline{A_j}\, Q - \overline{B_j}\, M_i)\right) < 0 \end{aligned} \tag{20}$$

$$T_{ii} = \alpha_g \otimes Q + \boldsymbol{\mathcal{H}}\left(\beta_g \otimes (P\, A_i - N_i\, C)\right) < 0 \tag{21}$$

$$\begin{aligned} \Theta_{ij} &= \alpha_g \otimes Q + \boldsymbol{\mathcal{H}}\left(\beta_g \otimes (P\, A_i - N_j\, C)\right) \\ &\quad + \boldsymbol{\mathcal{H}}\left(\beta_g \otimes (P\, A_j - N_i\, C)\right) < 0 \end{aligned} \tag{22}$$

$$\zeta_{Mi} = \begin{bmatrix} \gamma_{M_i}\mathbb{I}_n & M_i^T \\ M_i & \mathbb{I}_m \end{bmatrix} > 0, \tag{23}$$

$$\zeta_{Ni} = \begin{bmatrix} \gamma_{N_i}\mathbb{I}_n & N_i^T \\ N_i & \mathbb{I}_p \end{bmatrix} > 0 \tag{24}$$

The controller and the observer are defined as follows:

$$K_i = M_i Q^{-1} \quad \text{and} \quad G_i = P^{-1} N_i \tag{25}$$

Proof : The proof can be directly inspired from [6].

Remark 2: When computing the controller and observer gains, any constraint on the flows values (system inputs) is considered. However, when solving the LMIs constraints (19), (20), (21), (22), (24) and (25), it is about minimizing the values of these gains. Indeed, the inequalities ζ_{Mi} and ζ_{Ni} minimize the norms of matrices M_i and N_i.

V. Numerical illustration

The numerical illustration considers the nonlinear diesel engine model defined in section III with the numerical values given by the Table I. The computations are performed with MATLAB 7.1 and its LMITOOLBOX [5] on a PC Pentium 2930 Mhz.

The clustering region $\mathcal{D}_k$ considered in this example is a disc with center $c = -5$ and radius $r = 4.5$ whereas $\mathcal{D}_g$ is a vertical half-plane defined by $x < -10$. The pressure range of variation is such as $p_i \in [130\text{kPa}, 180\text{kPa}]$ and $p_x \in [85\text{kPa}, 125\text{kPa}]$.

Numerical values of the controller and observer gains are not given here but it is interesting that $\forall i \in \{1,..,4\}$:

$$\max(||K_i||) = 9.645\, e^{-6} \text{ and } \max(||G_i||) = 9.971\, e^{3}.$$

Fig. 7 shows the system states p_i and p_x and the corresponding desired values. The turbine and EGR gas flow rates generated via the TS fuzzy controller are illustrated by Fig. 8. It can be noticed that the two pressures are completely controlled. This is also verified by the plotting of the responses errors, given by Fig 9.

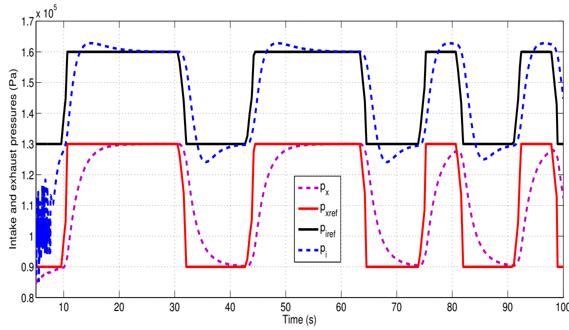

Fig. 7. Intake and exhaust manifold pressures

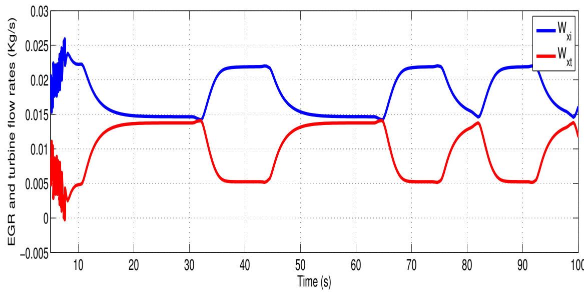

Fig. 8. EGR and turbine flow rates

VI. CONCLUSIONS AND FUTURE WORKS

A. Conclusions

Emissions of an engine change according to multiple parameters. The complexity of the diesel engine system and the presence of nonlinearities in the model make the control task arduous. A way to take into account the nonlinearities of the system is to use a multi model approach such as the Takagi-Sugeno's representation. In this article, the mean value model of the diesel engine is presented and converted into a Takagi-Sugeno'model, a simple control strategy based on a nonlinear control Lyapunov function is proposed.

B. Future works

Although the control tools proposed above are used to model and control some engine parameters in a certain range of operation, this work must evolve in two directions:

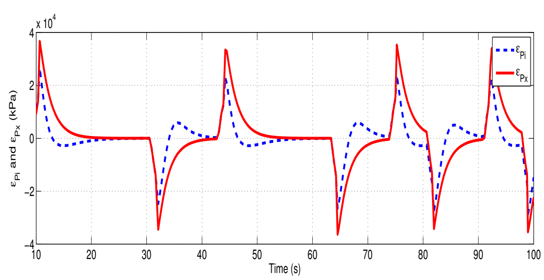

Fig. 9. Intake and exhaust manifold responses errors

1) Initially, it is obvious that it is essential to consider the loop fuel. One consequence is that the model of the engine must be more sophisticated, which limits the scope of solutions of optimization problems for control;
2) On the other hand, it should be defined several engine operating regimes and combustion modes. In fact, it is not possible to compute a single control law for the all diesel engine mode operations. Different controllers should be designed for different combustion mode : the conventional diesel combustion and the low temperature combustion mode [13].

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REFERENCES

[1] A.Ejiri, J.Sasaki, Y.Kinoshita, K.Shimotani, and R.Iizawa. Transient control of air intake system in diesel engines. *SICE Annual Conference*, 6:503–508, 2010.

[2] A.Plianos and R.Stobart. Dynamic feedback linearization of diesel engines with intake variable valve actuation. *IEEE International Conference on Control*, pages 455–460, 2007.

[3] A.Stefanopoulou, I.Kolmanovsky, and J.Freudenberg. Control of variable geometry turbocharged diesel engines for reduced emissions. *IEEE Transactions on Control Systems Technology*, 8, 2000.

[4] M. Chilali and P. Gahinet. H_∞ Design with Pole Placement Constraints : An LMI Approach. *IEEE Transactions on Automatic Control*, vol. 41(3):pp 358–367, 1996.

[5] P. Gahinet, A. Nemirovski, A.J. Laub, and M. Chilali. LMI Control Toolbox. *The MathWorks*, 1995.

[6] A.El Hajjaji, A.Chadli, M.Oudghiri, and M.Pages. Observer-based robust fuzzy control for lateral dynamic's. *Proceedings of American Control conference*, pages 4664–4669, 2006.

[7] H. Kerkeni, J. Lauber, and T.M. Guerra. Estimation of individual incynlinder air mass flow via periodic observer in takagi-sugeno form. *IEEE Vehicle Power and Propulsion Conference (VPPC)*, 10.1109:1–6, 2010.

[8] K.Tanaka, T.Ikeda, and H.O.Wang. Fuzzy regulators and fuzzy observers: relaxed stability conditions and lmi-based designs. *IEEE Transactions on Fuzzy systems*, 6:1–16, 1998.

[9] M.Herceg, T.Raff, R.Findeisen, and F.Allgower. Nonlinear model predictive control of a turbocharged diesel engine. *IEEE International Conference on Control*, pages 2766–2771, 2006.

[10] M.Jankovic and I.Kolmanovsky. Constructive lyapunov control design for turbocharged diesel engines. *IEEE Transactions on Control Systems Technology*, 8:288–299, 2000.

[11] M. Van Nieuwstadt, P. Moraal, I. Kolmanovsky, A. Stefanopoulou, P.Wood, and M.Criddle. Decentralized and multivariable designs for egr vgt control of diesel engine. *IFAC Workshop on Advances in Automotive Control*, 1998.

[12] Junmin Wang. Robust nonlinear control with singularity avoidance for diesel engines having multiple combustion modes. *American Control Conference*, pages 4401–4406, 2007.

[13] Junmin Wang. Hybrid robust air-path control for diesel engines operating conventional and low temperature combustion modes. *IEEE Transactions on Control, Systems technology*, 16:1138–1151, 2008.