Two Loop based Dynamical Feedback Stabilization Control of a Diesel Engine with EGR & VGT

H.P. Wang, J. Bosche, Y. Tian and A. El Hajjaji

Abstract—In this paper, a fourth-order dimensional non-linear model is proposed for a diesel engine equipped with a Variable Geometry Turbocharger (VGT) and an Exhaust Gas Recirculation (EGR) valve. The referred model which takes into account the engine crankshaft speed dynamics and the air-path dynamics is a non-minimum phase unstable system. The fuel flow rate W_f which is considered as input for the engine crankshaft subsystem acts as an external perturbation for the three-order dimensional non-minimum phase air-path subsystem. The global control objectives are to track desired values of engine speed, intake manifold pressure and compressor flow mass rate which can be suitably chosen according to low emission criterions. For the considered objectives two loop based dynamical feedback stabilization control is proposed: an inner loop and an outer loop. The inner loop considers a control based on Lyapunov function which realizes the desired engine speed trajectory tracking. The outer loop concerns EGR & VGT control. It is developed from a particular extended nonlinear air-path subsystem with its modified outputs. This outer loop ensures both the desired intake manifold pressure and the desired compressor mass flow rate trajectory tracking. The outer loop dynamical feedback stabilization control provides also the external fuel mass flow rate perturbation rejection.

The robustness and efficiency of the proposed method are demonstrated by simulation results.

I. INTRODUCTION

The diesel engine has been booming mainly for economic reasons (low fuel consumption, low maintenance costs) and environmental concerns (low emissions of greenhouse gases: CO_2). Furthermore, with the development of particle filters, the diesel engine has won an image of clean engine. However, a key challenge remains the reduction of emissions of nitrogen oxides (NO_x) which are produced during combustion procedures. In order to fulfill with more constraining Europeans emissions standards, automakers innovate in new technologies: such like the post-treatment of Exhaust Particulate Filters (DPF), multiple injection system, common rail injection system [10], Exhaust Gas Recirculation (EGR) technique [6], [4], [15], [1], [19] and Variable Geometry Turbocharger (VGT) system [6], [4], [15], [1], [19].

The diesel engine design objectives are to obtain the best possible performance in terms of speed regulation (such as maintaining good transient and steady-state performance and ensuring good external friction and load disturbance rejection capabilities), fuel consumption economy and exhaust harmful gas minimization emissions.

In literature, there exists many separate researches on engine speed regulations (such like, optimal gain scheduling controller [7], H_∞ control based on Linear Parameter Varying (LPV) approach [17], adaptive methods [8], sliding mode control [11], Lyapunov function based control [14] or PID control [9]) and on engine air-path dynamics equipped with an EGR and a VGT valve (such like, constructive Lyapunov control [6], nonlinear model predictive control [4], flatness based control [15], dynamical feedback stabilization control [1], LPV decoupling based control [12] or PID based EGR-VGT control [19]). The problem is that these are conflicting and coupling objectives, e.g. engine speed control has significant impact on engine performance, fuel consumption economy and exhaust gas emission [3]. The combinaison study of these two part dynamics is necessary to obtain a better engine performance.

Thus in this paper firstly, we propose a fourth-order dimensional non-linear Multi-Input Multi-Output (MIMO) diesel engine model which takes into account not only the engine crankshaft speed dynamics but also the air-path dynamics. The fuel flow rate W_f is considered as an input for the engine crankshaft subsystem and acts as an external perturbation for the non-minimum phase air-path subsystem. Then for the proposed nonlinear MIMO system, a two loop based dynamical feedback stabilization control is proposed to realize the control objectives of the desired engine speed, intake manifold pressure and compressor flow mass rate trajectories tracking (suitably chosen according to low emission criterions). The proposed entire control has two loops: the inner loop which considers a control based on Lyapunov function realizes the desired engine speed trajectory tracking, while the outer loop EGR & VGT control is developed from a particular extended nonlinear air-path subsystem. Based on this subsystem and its corresponding modified outputs, the EGR & VGT controller ensures both the desired intake manifold pressure and the desired compressor mass flow rate trajectory tracking. Meanwhile the outer loop control provides also the external fuel mass flow rate perturbation rejection. Compared with other linearization based methods, such as PID method [18], the proposed entire two loop based dynamical feedback stabilization control considers a more important functioning domain.

The paper is organized as follows: the referred fourth-order nonlinear MIMO engine model which includes the crankshaft and air-path dynamics is proposed in Section

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II and in Section III for the considered nonlinear non-minimum phase system, a two loop based dynamical feedback stabilization controller is developed to realize the control objectives. Then, some numerical simulation results are shown in Section IV to demonstrate the entire controller efficiency and robustness. Finally, some concluding remarks and discussions are given in Section V.

II. TURBOCHARGED DIESEL ENGINE: FUNCTIONING, MODELING AND SELECTED CONTROL OBJECTIVES

A. TDE Functioning

A TDE which includes the EGR, VGT and Crankshaft is illustrated in Fig. 1. At the end of this schematic diagram
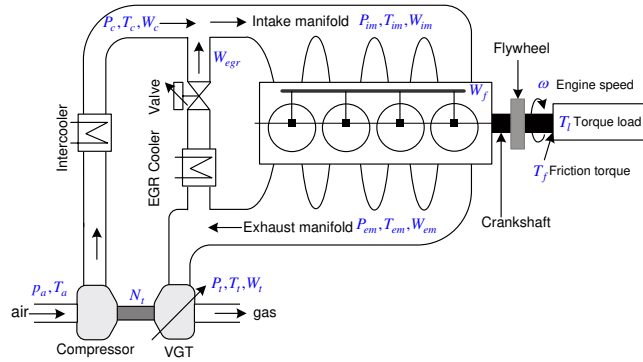

Fig. 1. TDE schematic diagram with EGR, VGT and Crankshaft

is the turbocharger composed of the turbine with variable geometry and the compressor. During the TDE functioning, the turbine takes the energy from the exhaust gas to power the compressor. As the turbine is linked to the compressor via a common shaft, the compressor starts consequently rotating and bringing in fresh air via the intercooler. Then the fresh air and a part of exhaust gas coming through EGR cooler and EGR valve is pumped from the intake manifold into the four cylinders. The fuel is injected directly into cylinders and burned, producing the torque on the crankshaft. The hot exhaust gas is pumped out into the exhaust manifold. Part of the exhaust gas comes from the exhaust manifold through the turbine out of the engine and the other part is recirculated back into the intake manifold.

The VGT provides the required fresh air under different load and engine speed, especially under high load and low crankshaft angular speed [13] where the exhaust gas kinetic energy is not powerful enough to speedup the turbocharger axis and deliver the sufficient power to compress the fresh air. And the variable geometry allows also to limit the power of the turbine at hight crankshaft angular speed in order to prevent the turbocharger destruction due to thermal or mechanical restriction. The diagram also shows the intercooler and the EGR-cooler that are used to reduce the intake manifold temperature. Noted that a part of the exhaust gas is recirculated into the combustion cylinders to reduce No_x formation refer to [1], [6].

B. Diesel Engine Model

1) Air-path dynamics: A third order nonlinear model of the air-path for the referred TDE can be derived by 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 time constant τ for modeling the power transfer dynamics of the VGT. For a more detailed derivation of the mean-value model, one can refer to [6], [19]. Under the assumption that the intake manifold temperature T_{im} (K), exhaust manifold temperature T_{em} (K), the compressor efficiency η_c and the time constant τ (s) of the turbocharger are constant, this modeling approach results in the following nonlinear model

$$\dot{P}_{im} = \frac{RT_{im}}{V_{im}}(W_c - W_{ie} + W_{egr}) \tag{1}$$

$$\dot{P}_{em} = \frac{RT_{em}}{V_{em}}(W_{ie} + W_f - W_{egr} - W_t) \tag{2}$$

$$\dot{P}_c = \frac{1}{\tau}(\eta_m P_t - P_c) \tag{3}$$

with P_{im} (Pa) the intake manifold pressure, P_{em} (Pa) the exhaust manifold pressure, P_c (Pa) the power transferred by the turbocharger, η_m the mechanical efficiency, and V_{im}, V_{em} (m³) the volumes of the intake and exhaust manifold. In the above TDE model, W_c (Kg/s) which describes the relation between the flow rate through the compressor and the power is modeled as

$$W_c = k_c \frac{P_c}{(\frac{P_{im}}{p_a})^\mu - 1} \tag{4}$$

with $k_c = \frac{\eta_c}{c_p T_a}$, and η_c the compressor efficiency, T_a (K) the ambient temperature, c_p, c_v heat at constant pressure and volume, μ a constant, and p_a (Pa) the ambient pressure.

The flow rate W_{ie} (Kg/s) from the intake manifold into the engine cylinders is modeled by the speed-density equation [4]

$$W_{ie} = k_{ie} P_{im} \tag{5}$$

where $k_{ie} = \frac{\eta_v \omega V_d}{4\pi T_{im} R}$, η_v describes the volumetric efficiency and can be represented approximately as

$$\eta_v = \alpha_0 + \alpha_1 \omega + \alpha_2 \omega^2 \tag{6}$$

with $\alpha_0, \alpha_1 > 0$ and $\alpha_2 < 0$, ω (rad/s) is the engine speed, V_d (m³) denotes the displacement volume. Although the real-time inputs are the EGR and VGT valves opening, W_{egr} (Kg/s) and W_t (Kg/s) which describe respectively the flow rate through the EGR and VGT valve are considered as inputs here (for aim of simplicity and not loss of generalities). It is noted that the fuel mass flow rate W_f (Kg/s) and the engine speed ω which are input and output of the crankshaft dynamics are considered as known external parameters for the above air-path models.

The turbine power P_t is modeled by the equation

$$P_t = k_t \left(1 - \left(\frac{P_{em}}{p_a}\right)^{-\mu}\right) W_t \tag{7}$$

where $k_t = c_p T_{em} \eta_t$, with η_t the turbine efficiency.

2) Engine crankshaft dynamics: The engine crankshaft dynamics which is obtained by fundamental Newton second law is described by the following power mechanical equation

$$\dot{\omega} = \frac{1}{J\omega}(P_e - P_{fl}) \qquad (8)$$

where J [Kgm^2] is the inertia of the engine assumed to be constant, $\omega(t)$ (rad/s) is the engine speed, $P_e(t)$ (W) is the effective engine power and $P_{fl}(t)$ (W) are the friction and load power which are produced respectively between the parts of the engine, and external load depending on driving conditions. The referred engine power $P_e(t)$ is defined as

$$P_e(t) = H_f \eta_e W_f \qquad (9)$$

where H_f (J/Kg) denotes the fuel heating value and η_e is the engine effective conversion efficiency. The referred η_e can be generally approximated by different expressions which is a product of two functions where the first function is described as a polynomial of engine speed and the second is a second-order polynomial or an exponential form of the air-fuel ratio [13], [16]. According to [2], [14], [16], the first function can be neglected in the control design, here η_e is modeled as

$$\eta_e = a_\lambda + b_\lambda \lambda + c_\lambda \lambda^2, \quad \text{with} \quad \lambda = \frac{W_{ie}}{W_f}. \qquad (10)$$

Thus from the above description, the considered engine dynamics is

$$\dot{\omega} = q_1(\omega, P_{im})u_1 + q_2(\omega, P_{im}) + \frac{q_3(\omega, P_{im})}{u_1} \qquad (11)$$

$$\dot{x} = f(x, u_1) + g_1(x)u_2 + g_2(x)u_3 \qquad (12)$$

where

$q_1(\omega, P_{im}) = H_f \frac{a_\lambda}{J\omega}, \quad u_1 = W_f, \quad u_2 = W_{egr}, \quad u_3 = W_t,$

$q_2(\omega, P_{im}) = \frac{1}{J\omega}\left(\frac{H_f b_\lambda V_d \omega(\alpha_0 + \alpha_1 \omega + \alpha_2 \omega^2)P_{im}}{4RT_{im}\pi} - P_{fl}\right),$

$q_3(\omega, P_{im}) = \frac{H_f c_\lambda V_d^2(\alpha_0 + \alpha_1 \omega + \alpha_2 \omega^2)^2 P_{im}^2 \omega}{16JR^2T_{im}^2\pi^2},$

$$f(x, u_1) = \begin{bmatrix} k_{im}k_c \frac{P_c}{(\frac{P_{im}}{P_a})^\mu - 1} - k_{im}k_{ie}P_{im} \\ k_{em}k_{ie}P_{im} + k_{em}u_1 \\ -\frac{1}{\tau}P_c \end{bmatrix}, \, g_1(x) = \begin{bmatrix} k_{im} \\ k_{em} \\ 0 \end{bmatrix},$$

$$g_2(x) = \begin{bmatrix} 0 \\ -k_{em} \\ K_0\left(1 - (\frac{P_a}{P_{em}})^\mu\right) \end{bmatrix}, \quad x = [P_{im} \; P_{em} \; P_c]^T \text{ with}$$

$k_{im} = \frac{RT_{im}}{V_{im}}, \, k_{em} = \frac{RT_{em}}{V_{em}}, \, K_0 = \frac{\eta_m k_t}{\tau}.$

C. Selected control objectives

The controlled outputs consists of the engine speed ω and the input manifold pressure P_{im} and the compressor mass flow rate W_c (instead of the AFR and EGR fraction because the latter are not accessible for measurments in a vehicle [6]).

Thus considering the entire nonlinear system (11-12), the vector output is selected as:

$$y = [\omega \; P_{im} \; W_c]^T. \qquad (13)$$

The first control objective is to design the injection input flow rate u_1 to track the desired engine speed reference ω_d of ω and the second control objective is to design the control laws of u_2 and u_3 in the air-path to track $P_{im,d}$ of P_{im} and $W_{c,d}$ of W_c in rejection of the perturbation of first input signal.

III. Two Loop Dynamical Stabilization Control

To realize the referred control objectives, a two loop based dynamical feedback stabilization control whose architecture is illustrated in Fig. 2 is proposed for the considered non-linear non-minimum phase engine system (11-13). This
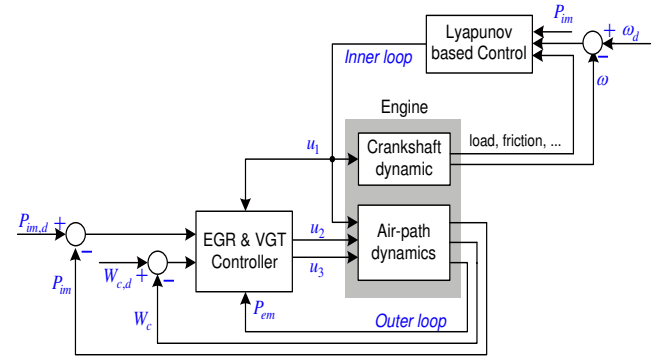

Fig. 2. Two loop based dynamic stabilization control architecture

dynamical feedback stabilization control consists of an inner loop of a Lyapunov function based controller (u_1) and an outer loop of EGR & VGT controller (u_2 & u_3). The inner loop is used to realize the desired engine speed tracking despite of external friction and load perturbations. And the outer loop is adopted to ensure the desired intake manifold pressure and the desired compressor mass flow rate tracking in the aim of fulfilling the emission requirements and the elimination of the coupling effect from the inner loop of u_1.

A. Inner loop of Lyapunov function based controller

The inner loop objective, considering the subsystem defined in (11), is to find a continuous positive control law $u_1(t)$ to realize the engine speed trajectory tracking of its desired reference speed $\omega \to \omega_d$.

According to [21], [22], [20], the proposed Lyapunov function is defined as

$$V(\omega) = \frac{1}{2}e_\omega^2, \quad \text{with} \quad e_\omega = \omega_d - \omega. \qquad (14)$$

In order to ensure its derivation negativity

$$\dot{V}(\omega) = -e_\omega \dot{\omega} < 0. \qquad (15)$$

we choose a particular function which stabilizes the crankshaft dynamic as follows

$$\dot{\omega} = K_\omega S(e_\omega), \quad \text{with} \quad K_\omega > 0, \; S(e_\omega)e_\omega > 0. \qquad (16)$$

where $S(e_\omega)$ is a smooth function with $S(0) = 0$. Therefore, the negativity of the chosen Lyapunov function is ensured, and from relations defined in (11) and (16), one obtains

$$q_1(\omega, P_{im})u_1 + q_2(\omega, P_{im}) + \frac{q_3(\omega, P_{im})}{u_1} = K_\omega S(e_\omega) \qquad (17)$$

with $q_1(\omega, P_{im}) > 0$ and $q_3(\omega, P_{im}) < 0$ (known from [14]).

Then according to [14], the injection flow rate $u_1(t)$ can be calculated from the following second order differential equation

$$q_1(\omega, P_{im})u_1^2 + [q_2(\omega, P_{im}) - K_\omega S(e_\omega)]u_1 + q_3(\omega, P_{im}) = 0. \tag{18}$$

Then, the resolution gives

$$u_1 = \frac{-[q_2(\omega, P_{im}) - K_\omega S(e_\omega)] + \sqrt{\delta}}{2q_1(\omega, P_{im})} \tag{19}$$

where $\delta = [q_2(\omega, P_{im}) - K_\omega S(e_\omega)]^2 - 4q_1(\omega, P_{im})q_3(\omega, P_{im})$. Because of δ always positive derived from the fact of $q_1(\omega, P_{im}) > 0$ and $q_3(\omega, P_{im}) < 0$, the proposed Lyapunov function based controller defined in (19) is always existing and positive.

B. Outer loop of EGR & VGT controller

The outer loop objective, which takes into account the injection fuel flow rate u_1 as an external disturbance for the air-path sub-system, is to propose two control signals for EGR & VGT control (u_2 & u_3).

1) zero dynamics of the air-path subsystem: Considering (12) and its corresponding output $y_a = [P_{im} \ W_c]^T$, the entire third-order air-path subsystem relative degree vector $(\rho_2, \ \rho_3) = (1,1)$ can be calculated from the following equation (see [5] for geometric nonlinear control method)

$$\dot{y}_a = \beta_{y_a}(x) + \alpha_{y_a}(x)\begin{bmatrix} u_2 \\ u_3 \end{bmatrix}, \tag{20}$$

where

$$\alpha_{y_a}(x) = \begin{bmatrix} k_{im} & 0 \\ -\frac{\mu k_c k_{im} P_c \frac{P_{im}^{\mu-1}}{p_a^\mu}}{\left(\left(\frac{P_{im}}{p_a}\right)^\mu - 1\right)^2} & -\frac{k_c K_0\left(1-\left(\frac{p_a}{P_{em}}\right)^\mu\right)}{\left(\frac{P_{im}}{p_a}\right)^\mu - 1} \end{bmatrix} \quad \text{and}$$

$$\beta_{y_a}(x) = \begin{bmatrix} k_{im}\left(k_c\frac{P_c}{\left(\frac{P_{im}}{p_a}\right)^\mu - 1} - k_{ie}P_{im}\right) \\ -\frac{\frac{1}{\tau}k_c P_c}{\left(\frac{P_{im}}{p_a}\right)^\mu - 1} - \frac{\mu k_c^2 k_{im} P_c^2 \frac{P_{im}^{\mu-1}}{p_a^\mu}}{\left(\left(\frac{P_{im}}{p_a}\right)^\mu - 1\right)^3} + \frac{\mu k_c k_{im} k_{ie} P_c\left(\frac{P_{im}}{p_a}\right)^\mu}{\left(\left(\frac{P_{im}}{p_a}\right)^\mu - 1\right)^2} \end{bmatrix}.$$

Because the sum of the vector relative degree components equals to 2 which is less than the order 3 of the air-path system, there exits an one-dimensional zero dynamics. The examination of this zero dynamics stability is necessary before deriving the following vector control law

$$\begin{bmatrix} u_2 \\ u_3 \end{bmatrix} = \alpha^{-1}(x)(-\beta(x) + v(x)) \tag{21}$$

where $v(x) = [v_2, \ v_3]^T = [K_2(P_{im} - P_{im,d}) \ K_3(W_c - W_{c,d})]^T$ with $K_2 > 0, K_3 > 0$ ensuring the trajectory tracking of the desired output values. Supposing that with the proposed control law (21), the zero dynamics is given by

$$\dot{P}_{em} = k_{em}W_{c,d}\left[1 - \frac{\left(\frac{P_{im,d}}{p_a}\right)^\mu - 1}{\eta_m k_t k_c\left(1-\left(\frac{p_a}{P_{em}}\right)^\mu\right)}\right] \tag{22}$$

which is unstable. Its corresponding single equilibrium is $P_{em,e} = p_a\left(1 - \frac{\left(\frac{P_{im,d}}{p_a}\right)^\mu - 1}{\eta_m k_t k_c}\right)^{-\frac{1}{\mu}}$.

2) EGR & VGT Controller:

To overcome the unstable zero dynamics (22), a technique of changing the output $y_a = [P_{im} \ W_c]^T$ is adopted such that the modified air-path subsystem has trivial zero dynamics , that is consisting of a single equilibrium point. According to [1], this new output $\tilde{y}_a$ can be selected as

$$\tilde{y}_a = \begin{bmatrix} h_2(x) \\ h_3(x) \end{bmatrix} = \begin{bmatrix} P_{im} \\ P_c + \frac{K_0}{k_{em}}\left(P_{em} - \frac{p_a}{1-\mu}\left(\frac{P_{em}}{p_a}\right)^{1-\mu}\right) \end{bmatrix} \tag{23}$$

Unfortunately considering (12) with its new output (23), the decoupling matrix

$$\alpha(x) = \begin{bmatrix} k_{im} & 0 \\ -K_0\left(1-\left(\frac{p_a}{P_{em}}\right)^\mu\right) & 0 \end{bmatrix} \tag{24}$$

which is not invertible and thus has not dynamical stabilization control law as defined in (21).

In order to realize a feasible dynamical stabilization control law as defined in (21), one considers $[v_2, \ v_3]^T$ as the new input signals of the following defined extended air-path nonlinear system

$$\dot{P}_{im} = k_{im}\left(k_c\frac{P_c}{\left(\frac{P_{im}}{p_a}\right)^\mu - 1} - k_{ie}P_{im} + z\right) \tag{25}$$

$$\dot{P}_{em} = k_{em}(k_{ie}P_{im} + u_1 - z - v_3) \tag{26}$$

$$\dot{P}_c = -\frac{1}{\tau}P_c + K_0\left(1-\left(\frac{p_a}{P_{em}}\right)^\mu\right) \tag{27}$$

$$\dot{z} = v_2 \tag{28}$$

with $z = u_2 = W_{egr}$, $v_3 = u_3 = W_t$.

Then considering the nonlinear system (25-28) and the output (23) as an entire nonlinear dynamical system to be controlled, one obtains the following relations with geometric nonlinear control [5]

$$\ddot{\tilde{y}}_a = \beta_{\tilde{y}_a}(x_e, u_1) + \alpha_{\tilde{y}_a}(x_e, u_1)\begin{bmatrix} v_2 \\ v_3 \end{bmatrix}, \tag{29}$$

where $x_e = [P_{im}, P_{em}, P_c, z]$ is the state of considered system (25-28), u_1 which is calculated from the inner loop (19) for the crankshaft is considered as external perturbation for the EGR & VGT control and $\alpha_{\tilde{y}_a}(x_e, u_1)$ and $\beta_{\tilde{y}_a}(x_e, u_1)$ are defined as follows

$$\alpha_{\tilde{y}_a} = \begin{bmatrix} k_{im} & \frac{k_{im}k_c K_0\left(1-\left(\frac{p_a}{P_{em}}\right)^\mu\right)}{\left(\frac{P_{im}}{p_a}\right)^\mu - 1} \\ K_0\left(\left(\frac{p_a}{P_{em}}\right)^\mu - 1\right) & \frac{\mu K_0 k_{em}}{p_a}(z - u_1 - k_{ie}P_{im})\left(\frac{p_a}{P_{em}}\right)^{\mu+1} \\ & + \frac{K_0}{\tau}\left(\left(\frac{p_a}{p_{em}}\right)^\mu - 1\right) \end{bmatrix}, \tag{30}$$

$$\beta_{\tilde{y}_a} = \begin{bmatrix} -k_{im}^2\left(k_{ie} + \frac{\mu k_c P_c\left(\frac{P_{im}}{p_a}\right)^{\mu-1}}{p_a\left(\left(\frac{P_{im}}{p_a}\right)^\mu - 1\right)^2}\right)\left(\frac{k_c P_c}{\left(\frac{P_{im}}{p_a}\right)^\mu - 1} + z - k_{ie}P_{im}\right) \\ -k_{im}\dot{k}_{ie}P_{im} - \frac{k_{im}k_c P_c}{\tau\left(\left(\frac{P_{im}}{p_a}\right)^\mu - 1\right)} \\ \frac{1}{\tau^2}P_c + \frac{\mu k_{em}K_0}{p_g}\left(k_{ie}P_{im} + u_1 - z\right)^2\left(\frac{p_a}{P_{em}}\right)^{\mu+1} + \\ K_0\left(1-\left(\frac{p_a}{P_{em}}\right)^\mu\right)\left(k_{ie}k_{im}\left(\frac{k_c P_c}{\left(\frac{P_{im}}{p_a}\right)^\mu - k_{ie}P_{im} + z}\right) + \dot{k}_{ie}P_{im} + \dot{u}_1\right) \end{bmatrix}. \tag{31}$$

Because the relative degree of the extended nonlinear system (25-28) with (23) equals to $\tilde{\rho}_2 + \tilde{\rho}_3 = 4$ the dimension of

the state of the extended system, a new feedback stabilization based control which is defined as follows

$$\begin{bmatrix} v_2 \\ v_3 \end{bmatrix} = \alpha_{\tilde{y}_a}^{-1}(x_e)(-\beta_{\tilde{y}_a}(x_e) + \varphi) \qquad (32)$$

can be applied to (29). The referred new control φ in (32) is defined as

$$\varphi = \begin{bmatrix} -K_2^1 \dot{P}_{im} - K_2^2(P_{im} - P_{im,d}) \\ -K_3^1 \dot{h}_3 - K_3^2(h_3 - h_{3,d}) \end{bmatrix} \qquad (33)$$

where $K_i^j \in \Re$ are chosen so that, for $i = 2, 3$, the characteristic polynomials $s^2 + K_i^1 s + K_i^2$ are Hurwitz, and $h_{3,d}$, which is a new reference defined from ($P_{im,d}$ and $W_{c,d}$) for h_3 (the second component of the new output $\tilde{y}_a$ in (23)) is defined as follow

$$h_{3,d} = \frac{W_{c,d}\left(\left(\frac{P_{im,d}}{p_a}\right)^\mu - 1\right)}{k_c} + \frac{K_0}{k_{em}}\left(P_{em,e} - \frac{p_a}{1-\mu}\left(\frac{P_{em,e}}{p_a}\right)^{1-\mu}\right). \qquad (34)$$

The tracking fact of this control (33) ensures the trajectories tracking of P_{im} of $P_{im,d}$ and $W_{c,d}$ of W_c (via the Intermediate output h_3).

IV. NUMERICAL RESULTS

In order to validate and illustrate the proposed entire two loop based dynamical stabilization control performances, the following case is considered. The engine crankshaft works under a variable external friction and load power which is illustrated in Fig. 3 and a speed significant measurement noise (simulated by a limited-band white noise whose noise power is [0.01]).
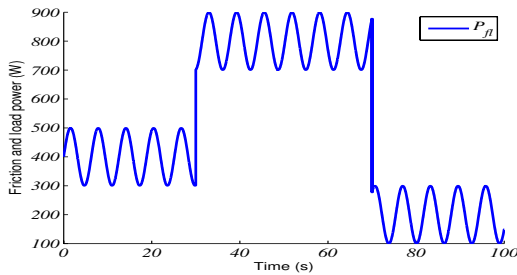

Fig. 3.   External friction and load torque generated power disturbance

The engine speed trajectory tracking response and the fuel mass flow rate control are illustrated respectively in Fig. 4 and Fig. 5. It can be observed that the inner loop stabilization control based on Lyapunov function can ensure an excellent engine speed trajectory tracking in spite of variable external friction and load disturbance and significant engine speed measurement noises.

The outer loop control, under the external perturbated inner loop control ($u_1 = W_f$) for the air-path dynamics, ensures the trajectory tracking of the intake manifold pressure P_{im}, the compressor mass flow rate W_c and its intermediate reference h_3. These variables and their corresponding desired references ($P_{im,d}$, $W_{c,d}$ and $h_{3,d}$) are illustrated respectively in Fig. 6, Fig. 7 and Fig. 8.
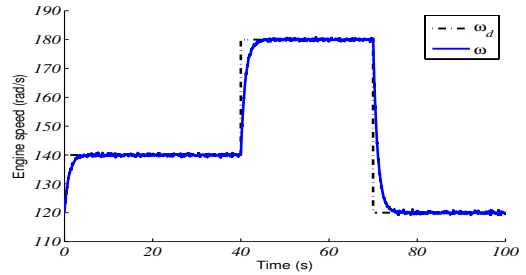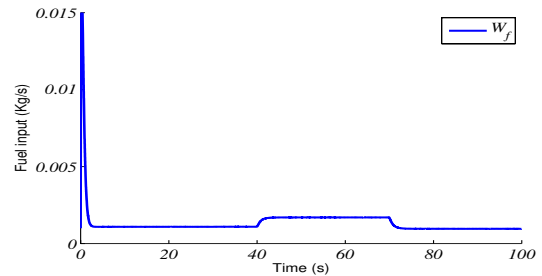

Fig. 4.   Engine speed regulation
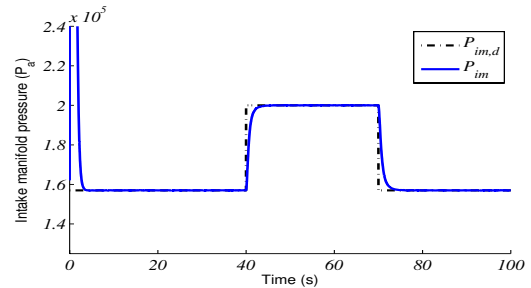

Fig. 5.   Inner loop fuel mass flow rate input

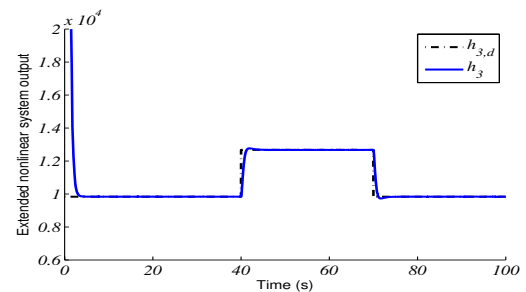Fig. 6.   Intake manifold pressure regulation

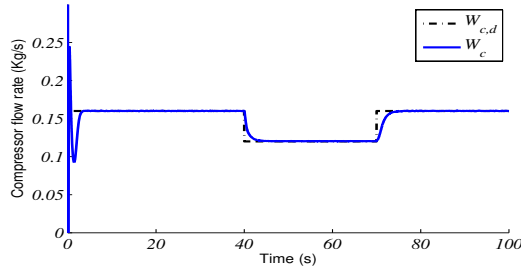Fig. 7.   Intermediate output h_3 regulation for W_c

Fig. 8.   Compressor mass flow rate regulation

Moreover according to [13] and [14], an opacity can be used to quantify the soot and particule mass emission. The opacity model is defined as

$$O_p = (k_0 + k_1\omega)P_{im}^{k_2+k_3\omega}W_f^{k_4+k_5\omega}, \qquad (35)$$

where $k_i, i = 0,...,5$ are constants [14]. Under the proposed entire control strategies, it can be noted that from Fig. 9, the soot and particule mass emission trends to zero.
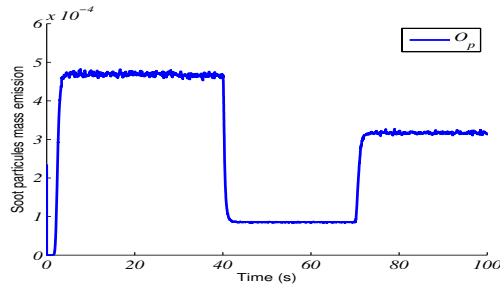

Fig. 9.   Opacity under the proposed two loop based stabilization control

V. CONCLUSIONS AND FUTURE WORKS

In this paper, a fourth-order dimensional non-linear diesel engine model (with EGR & VGT) which takes into account the engine crankshaft dynamics and the air-path dynamics is considered. The fuel flow rate W_f which is considered as input for the engine crankshaft subsystem acts as an external perturbation for the three-order dimensional non-minimum phase air-path subsystem. A two loop based dynamical feedback stabilization control is proposed to deal with this nonlinear unstable system. The inner loop which is based on a classical Lyapunov function based control realizes the desired engine speed trajectory tracking. And the outer loop EGR & VGT control which is developed from an extended fourth-order air-path subsystem with its modified outputs ensure not only the desired intake manifold pressure and the compressor mass flow rate trajectories tracking, but also the external fuel mass flow rate perturbation rejection.

Future works involve the implementation of the entire two loop based stabilization control on an experimental diesel engine and the formulation of an optimization criterion to minimize the fuel consumption, polluted air emissions and soot and particule emissions.

VI. ACKNOWLEDGMENTS

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