Design and experimental validation for nonlinear control of internal combustion engines with EGR and VVT

Haoyun Shi and Tielong Shen

Department of Engineering and Applied Sciences, Sophia University, Tokyo, Japan

1. Introduction

For combustion engines, advanced technologies such as variable valve timing (VVT) and exhaust gas recirculation (EGR) have been developed to satisfy the regulation on fuel consumption and emission. In the past researches, it has been shown that CO₂ can be reduced by increasing the intake manifold pressure and reducing the exhaust manifold pressure (reduce the pumping loss) [1]. VVT and EGR are regarded as the key actuators which determine the pressure of the intake and the exhaust manifolds [2–4]. However, as is well-known, the relationship between VVT, EGR position and the pressure of manifolds is nonlinear and coupled [5,6]. It motivates to develop a coordinated control strategy for VVT and EGR which render the pressure of the manifolds to reach the set-point. However, to control the pressure is not an easy task, since the dynamics of intake and exhaust manifold is a complicated system involving mechanical, aerodynamics and thermal dynamics, etc. In the past two decades, the control problem of combustion engines with VVT or EGR loop has been investigated with the technology of dynamic system modelling and feedback control. For example, a linearized model of EGR and VGT is discussed in [7,8], and the model is used to design a model predictive control strategy to improve the transient performance. And the papers [9,10] studied the control Lyapunov function-based nonlinear control design method which possesses a guaranteed robustness property equivalent to gain and phase margins. Meanwhile, an experimental comparison between linear-quadratic regulator and model predictive control are presented by Kang and Shen [11]. In the paper [12], the problem of EGR control is solved by using nonlinear model and Lyapunov design to find the nonlinear feedback control law.

However, it is still a challenging issue to design a coordinative control scheme for the combustion engines with VVT and EGR loop. As is well-known, there is a feedback gas flow from the exhaust path to the intake path in the combustion engines. Therefore, one big challenging is how to handle this coupling between the intake and the exhaust path, especially during the transient operation mode. Furthermore, the intake VVT installed for improving the efficiency will affect to the mass of air charge, so that the gas mass balance in the intake manifold is also affected by VVT operation. As a result, for the combustion engines with EGR loop and VVT, manipulation of the actuator EGR and VVT, including transient mode, becomes a challenging task. Concerning the engines with EGR loop only, several challenging results have been reported such as [7–10]. A natural idea to handle the dynamics of the air paths, the state variable of the exploited dynamic model involves the pressure of the intake and the exhaust manifolds, since the gas mass flow coupling between the intake and the exhaust paths. However, different control purpose leads to different dynamic model. For example, to control the oxygen-fuel ratio and the EGR fraction, the state space model is of eight dimension [8] or seven dimension as shown in [7], in the latter minimizing pumping loss is targeted as control objective. Meanwhile, a lower order model with three dimension, involving the pressure of the intake and exhaust manifolds, is developed in [9,10] to control...
the air-fuel ratio and the EGR fraction. Different from these results, we will show a second-order dynamic model by focusing on the pressure of intake and exhaust manifolds. Benefit from this simple structure of the model, a feedback control scheme is designed by feedback nonlinear compensation that regulates the state of intake and exhaust manifolds at an optimized operating point in the sense of minimizing fuel consumption.

Therefore, the main purpose of this paper is to provide a design approach to the control problem of gasoline engines with actuators of VVT and EGR. Along the research line shown in [12], we will challenge this problem with dynamical model-based feedback control approach combined with experiment-based static maps that enable us to decide the optimal set-point. The contribution of this paper might be summarized as follows: First, a simple dynamic model with two dimension is proposed and with the model, it is shown that a linear coordinate transformation with feedback compensation will lead this model to a decoupled structure so that the convergence can be easily obtained by adding error feedback. Second, it is newly proposed that from the view of fuel consumption optimization the state of the proposed model, the pressure of intake and exhaust manifolds can be used to manage the static operation mode of the engine. Based on this fact, a pre-calibrated map that provides optimal set-point of engine operation is designed by using sequential quadratic programming algorithm. Finally, experiment validation results are provided that conducted on a full-scaled industrial production engine.

2. Physical background and problem

We start in this section with a brief explanation of physical background of combustion engines and the set-up of experiment which enables us to understand the engine properties and the control design problem.

A systematic configuration of the engine control system is sketched in Figure 1, where only a single cylinder is indicated for sake of simplicity, and the attention is focused on the control actuation signals and the measured signals for feedback control. As well-known, combustion engine is a device for transforming the fuel energy to mechanical power by managing the four strokes. More precisely, it depends on the ignition timing and the state of in-cylinder gas mixture, and the combustion process which is usually represented by the profile of in-cylinder pressure, and one more important factor that determines the energy efficiency and the mount of generated thermal energy during one cycle is the mount of mass of the fuel and the fresh gas charged to cylinder, if we suppose the ignition timing and the fuel mass injection are decided for keeping the minimum advance for the best torque and the air-fuel ratio to a desired value [13]. In this case, the total mass of the fuel, the fresh gas and the burnt gas will be determined by the operation of the throttle angle, the valve timing and the opening of EGR valve.

Meanwhile, the efficiency from the fuel to the mechanical power can be evaluated by the fuel mass injected during a cycle at a given operation point with a certain torque and rotational speed of the crankshaft. Hence, at the given engine operating point the efficiency can be improved through minimizing the injected fuel mass per cycle, and a feasible way to achieve this goal is tuning the actuations such as the throttle, VVT and EGR valves. Motivated by this observation, these three operating variables are chosen as control inputs and the fuel mass injected per cycle is control objective to be reduced as small as possible under the constraint of required torque generation and the speed. Obviously, based on the conservation law, the output power is balanced with the fuel energy and the losses during the energy conversion. However, this energy conversion process involves complex dynamics such as air fluid dynamics in the intake and the exhaust path, wall-wetting fuelling dynamics, the thermal dynamics in-cylinder and mechanical rotation dynamics [14]. The control inputs contribute to the objective of reducing the fuel mass through the dynamics. Therefore, it is a challenging how to design a control law that not only balancing the system with minimum fuel mass but also manage the dynamics to converge to the balanced point.

A lot of researches have been reported to manage the dynamics of a combustion engine. Many attentions focused on the modelling of the dynamics; however, it should be noted that there is no uniformed model to represent the dynamic behaviour of combustion engines. Most approach of modelling is control-oriented and mean-valued according to the control objectives. Motivated by the previous papers [15,16], these papers also choose the pressure of the intake manifold and the exhaust manifold to coordinate the
dynamics of the engine system. The causality of these internal variables (the pressure of intake manifold $p_m$ and the pressure of exhaust manifold $p_e$) to the fuel mass, and to the control input are shown in the experiment data in Figure 2, where a gasoline engine with four cylinders is used for conducting the experiments. It can be seen from the experiment that under the constraint of required torque and speed, the fuel mass is changed according to different values of the pressures which shows the potential in minimizing the fuel. At the same time, the delivered value of the control inputs will force the pressure $p_m, p_e$ to a static value but with transient phenomenon as shown in Figure 3.

Therefore, by handling the internal variables of the engine, we investigate the following two problems in this paper: (1) Decide the optimal values $p_m^*$ and $p_e^*$ that minimize the fuel consumption under constraint of indicated mean effective pressure (IMEP) variation. (2) Design a feedback control law that renders the internal dynamics of the engine to the given state. Hence, the proposed control structure is as shown in Figure 4. In the next section, we will begin with the modelling.

3. Modelling

In this section, a model in the sense of mean-value for the gasoline engine focused on the air path system is presented. As mentioned in Section 3.2, the mean-value model describe the physical process of air path which observe the conservation of energy and conservation of mass. The parameter of the model are identified by recursive least square algorithm according to the engine specification, the environment or the set-point changes.

3.1. Experiment set-up

In order to collect the data for modelling, the experiment is implemented on a real engine test bench. Figures 5 and 6 show the engine photo and rapid prototype system. In this test bench, a 1.8L 4-cylinders engine is equipped and it is coupled with an low-inertia alternating-current electrical dynamometer. The specification of the gasoline engine is listed in Table 1. The dynamometer can be used for emulating the external load disturbance in real time. The real-time control system dSPACE 1006 is equipped to fully control the engine via the bypass connection with the engine prototype ECU. The engine ECU can control the engine itself and provide the whole engine sensor signals and actuator signals back to the dSPACE, while the control
algorithm can be programmed and downloaded in dSPACE to control engine through enabling the control channel of the prototype ECU.

### 3.2. Dynamic model

The intake manifold can be viewed as a thermodynamic control volume with a fixed volume $V$ that stores mass and energy. According to mass conservation law, the rate of mass change in the manifold is determined by the difference between the inlet and the outlet mass flow as follows:

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out} \tag{1}$$

where $\dot{m}_{in}$ is the mass flow passing through the throttle valve and $\dot{m}_{out}$ the mass flow leaving the manifold and sucked to the cylinders. Moreover, under the assumption that there is no temperature change in the intake system, i.e. $T$ is constant, the internal energy in the manifold $c_v \Delta T$ changes only by the difference of the inlet and outlet mass flow rate $\dot{m}_{in} - \dot{m}_{out}$. Combining this with the ideal gas law $p_m V = R m T$ directly obtains the following dynamical equation

$$\frac{dp_m}{dt} = \frac{RT}{V} (\dot{m}_{in} - \dot{m}_{out}) \tag{2}$$

where $R$ is the gas constant, $c_v$ is the specific heat.

As an energy conversion device, the energy involved in the in-cylinder gas mixture, the total mass of gas flow $\dot{m}_{out}$ during one inlet stroke with injected fuel, is released during the combustion stroke to the heat energy which is further transformed to mechanical work by forcing the piston, and the torque acting on the crankshaft.

Based on the mass conversation law and the ideal gas equation, the main relevant dynamic models adopt the mean-value model, including the dynamic behaviour in the intake manifold pressure $p_m$ and the exhaust manifold pressure $p_e$ as follows [17].

$$\dot{p}_m = \alpha_m (\dot{m}_{th} + \dot{m}_{egr} - \dot{m}_{cyl}) \tag{3}$$

$$\dot{p}_e = \alpha_e (\dot{m}_{cyl} + \dot{m}_f - \dot{m}_{egr} - \dot{m}_{t wc})$$

Here, $\alpha_m = \frac{RT}{V_m}, \alpha_e = \frac{RT}{V_e}, \dot{m}_{th}, \dot{m}_{egr}, \dot{m}_{cyl}, \dot{m}_{t wc}$ are the mass flow through the throttle, EGR, VVT, and the three-way catalyst, $\dot{m}_f$ denotes the fuel flow.

In this study, the fluid is assumed as incompressible, so $\dot{m}_{th}, \dot{m}_{egr}$ and $\dot{m}_{t wc}$ can be calculated by Bernoulli’s law.

$$\dot{m}_{th} = \psi_{th} [1 - \cos(u_{th})] \sqrt{p_a - p_m} \tag{4}$$

$$\dot{m}_{egr} = \psi_{egr} M_1(u_{egr}) \sqrt{p_e - p_m}$$

$$\dot{m}_{t wc} = \psi_{t wc} \sqrt{p_e - p_a}$$

where $p_a$ denotes the atmospheric pressure and $\psi_{th}$, $\psi_{egr}$, $\psi_{t wc}$ denote the flow coefficient of throttle valve, EGR, and three-way catalyst, respectively. Polynomial $M_1(u_{egr})$ describes the relationship of $u_{egr}$ and EGR opening area.

Since the feature of engine gas exchange, $\dot{m}_{cyl}$ cannot be defined in continuous time domain. So $\dot{m}_{cyl}$ is described by a mean-value model which equals to the product of volumetric efficiency and the total charge volume. The volumetric efficiency is modeled by a Polynomial $M_2(p_m, \omega, u_{vvt})$ corresponding to $p_m$, engine speed $\omega$ and $u_{vvt}$.

$$\dot{m}_{cyl} = \psi_{vvt} M_2(p_m, \omega, u_{vvt}) p_m \omega \tag{5}$$

where $\psi_{vvt}$ denotes the flow coefficient of VVT.

The $\dot{m}_f$ is calculated by a map $M_3(p_m, \omega)$ which is calibrated by steady experimental data.

$$\dot{m}_f = \psi_f M_3(p_m, \omega) \tag{6}$$

where $\psi_f$ denotes the flow coefficient of fuel flow.

To simply the identification, we define the following parameters: $a = \alpha_m \psi_{th}, b = \alpha_m \psi_{egr}, c = \alpha_m \psi_{vvt}, d = \alpha_e \psi_{vvt}, e = \alpha_e \psi_{f}, f = \alpha_e \psi_{egr}, g = \alpha_e \psi_{t wc}$.

Finally, the dynamic model can be summarized as follows:

$$\dot{p}_m = a [1 - \cos(u_{th})] \sqrt{p_a - p_m} + bM_1(u_{egr}) \times \sqrt{p_e - p_m} - cM_2(p_m, \omega, u_{vvt}) \sqrt{p_m \omega}$$

$$\dot{p}_e = dM_2(p_m, \omega, u_{vvt}) \sqrt{p_m \omega} + eM_3(p_m, \omega) - fM_1(u_{egr}) \sqrt{p_e - p_m} - g \sqrt{p_e - p_a} \tag{7}$$

### 4. Control design

Suppose a set-point of engine operation is given by $p_{m \text{set}}, p_{e \text{set}}$ which is determined in the next section. A feedback control law that renders the engine system converge to the set-point is designed as follows.

![Figure 6. Rapid prototype system.](Image)

**Table 1.** Specification of the engine test bench.

| Cylinder type | L-type 4 cylinders |
|---------------|--------------------|
| Compression ratio | 13.1 |
| Fuel injection | Port injection |
| Displacement | 1.797 L |
| Max output power | 72 kw @ 5200 rpm |
| Max torque | 142 Nm @ 5200 rpm |
The feedback control law is constructed for \( \dot{m}_{th} \) and \( \dot{m}_{egr} \), since once obtain the feedback control law, the valve opening of \( u_{th}, u_{egr} \) and \( u_{vvt} \) can be determined by taking inverse of the following model:

\[
\begin{align*}
\dot{m}_{th} &= \psi_{th}(1 - \cos(u_{th})) \sqrt{p_e - p_m} \\
\dot{m}_{egr} &= \psi_{egr} M_1(u_{egr}) \sqrt{p_e - p_m} \\
\dot{m}_{cyl} &= \psi_{vvt} M_2(p_m, \omega, u_{vvt}) p_m \omega
\end{align*}
\]  

(8)

Note that (3) is equivalent to

\[
\begin{align*}
\alpha_k \dot{p}_m &= \alpha_m \dot{a}_e (\dot{m}_{egr} - \dot{m}_{cyl}) \\
\alpha_m \dot{p}_e &= -\alpha_m \dot{a}_e (\dot{m}_{egr} - \dot{m}_{cyl}) + \alpha_m \dot{a}_e (\dot{m}_{cyl} - \dot{m}_{egr})
\end{align*}
\]  

(9)

Let \( \dot{p}_\Sigma \) replaces \( \dot{p}_m + \frac{a_{\psi}}{\dot{a}_e} \dot{p}_e \), the Equation (3) can be transformed as follows:

\[
\begin{align*}
\dot{p}_\Sigma &= \alpha_m (\dot{m}_{th} + \dot{m}_{f} - \dot{m}_{lwc}) \\
\dot{p}_e &= \alpha_e (\dot{m}_{f} + \dot{m}_{egr} - \dot{m}_{cyl} - \dot{m}_{lwc})
\end{align*}
\]

(10)

where the following coordinate transformation is applied

\[
(p_m, p_e) \rightarrow (p_\Sigma, p_e)
\]

(11)

Define the error as \( \varepsilon_\Sigma = p_\Sigma - p_\Sigma^* \) and \( \varepsilon_e = p_e - p_e^* \). Then, we have from (11) the following tracking error dynamics

\[
\begin{align*}
\dot{\varepsilon}_\Sigma &= \alpha_m (\dot{m}_{th} + \dot{m}_{f} - \dot{m}_{lwc}) \\
\dot{\varepsilon}_e &= \alpha_e (\dot{m}_{f} + \dot{m}_{egr} - \dot{m}_{cyl} - \dot{m}_{lwc})
\end{align*}
\]

(12)

Proposition 4.1: For the given \( p_m^*, p_e^* \), let the feedback control law be as follows:

\[
\begin{align*}
\dot{m}_{th} &= \dot{m}_{lwc} - \dot{m}_{f} - k_1 \varepsilon_\Sigma \\
\dot{m}_{egr} - \dot{m}_{cyl} &= \dot{m}_{lwc} - \dot{m}_{f} - k_2 \varepsilon_e
\end{align*}
\]

(13)

then, the closed-loop system (12) with control law (13) is asymptotically stable at the origin, if the control gains satisfy \( k_1 > 0, k_2 > 0 \).

Proof: It is easy to show the following control loop system by substituting the control law (13) to (14),

\[
\begin{align*}
\dot{\varepsilon}_\Sigma &= -\alpha_m k_1 \varepsilon_\Sigma \\
\dot{\varepsilon}_e &= -\alpha_e k_2 \varepsilon_e
\end{align*}
\]

(14)

Clearly, this is a decoupled first-order linear differential equation. Thus, the convergence of the error \( \dot{\varepsilon}_\Sigma, \dot{\varepsilon}_e \) can be directly concluded by noting the positive value of the parameter \( k_1 > 0, k_2 > 0 \).

It should be noted that since we use three actuators to control two state variables, the system has one more degree of freedom. From the physics, the pressure of intake manifold is greatly affected by the throttle opening. To force the state variables \((p_m, p_e)\) to the given point, which is decided from the view of optimizing the fuel efficiency, the effect of cooperative transient operation of EGR and VVT should be compensated by the throttle opening. From the dynamic model, it is obvious that operation of EGR and VVT is not completely free, cooperative manipulation is necessary to satisfy the relation of the control law (13). In practice, VVT set-value can be decided with priority, then EGR is determined by the constraint of the control law, since the response of VVT is much slower than EGR loop.

Finally, the nonlinear feedback control law can be designed as:

\[
\begin{align*}
\dot{u}_{th} &= \phi_{th}^{-1}((\dot{m}_{lwc} - \dot{m}_{f} - k_1 (p_\Sigma - p_\Sigma^*)) / \sqrt{p_e - p_m}) \\
\dot{u}_{egr} &= \phi_{egr}^{-1}((\dot{m}_{cyl} + \dot{m}_{lwc} - \dot{m}_{f} - k_2 (p_e - p_e^*)) / \sqrt{p_e - p_m})
\end{align*}
\]

(15)

where \( \dot{m}_{cyl} = \psi_{vvt} M_2(p_m, \omega, u_{vvt}) p_m \omega \).

5. Experiment validation

5.1. Set-point optimization

The set-point must be constraint by the operating point, i.e. for a given operating point, the desired set-point must be chosen under the constraint that the generated torque equivalent to a given value. Moreover, the variation of IMEP is lower than a tolerable value.

The fuel consumption rate \( \dot{m}_f \) is chosen as the cost function of static set-point optimization. Then, the optimization problem can be formulated as follows

\[
\min \begin{bmatrix} \dot{m}_f(p_m, p_e) \end{bmatrix}
\]

(16)

subject to \( \tau_d - \tau_e(p_m, p_e) = 0 \) and \( \text{IMEP}_{cov} \leq \text{IMEP}_{cov}^* \), where \( \tau_d \) is torque demand and \( \tau_e(p_m, p_e) \) is engine output torque. Denote the solution as

\[
\begin{bmatrix} p_m^* \\ p_e^* \end{bmatrix} = \arg \min_{p_m, p_e} m_f(p_m, p_e)
\]

(17)

The fuel consumption rate \( \dot{m}_f \) depends on the engine. In this research, an experimental data-based fitting model is used. For example, at the operating condition with a constant engine speed 1600 r/min and output torque 40 Nm. The water temperature \( T_w = 80^\circ\text{C} \), equivalent ratio \( \lambda = 1 \) were also be fixed. Here, the constraint configuration are: \( u_{egr} \in [0, 20], u_{vvt} \in [0, 40] \). \( m_f(p_m, p_e) \) is a second-order polynomial model corresponding to \( p_m \) and \( p_e \), where \( a_1, a_2, a_3, a_4, a_5, a_6 \) are the fitting parameters of model. Here, we use root mean square error (RMSE) to evaluate the fitting accuracy which a value closer to 1 indicates that the fit has a satisfactory accuracy.

\[
\text{RMSE} = \sqrt{\frac{1}{n} \sum_{i=1}^{n} w_i (y_i - \hat{y}_i)^2}
\]

(18)
\[ m_f(p_m, p_e) = a_1 + a_2 p_m + a_3 p_e + a_4 p_m^2 + a_5 p_m p_e + a_6 p_e^2 \]

\[ a_1 = 43.76; \quad a_2 = -0.326; \quad a_3 = 0.8639; \]

\[ a_4 = 0.07045; \quad a_5 = -0.1395; \quad a_6 = 0.3669; \]

\[ \text{RMSE} = 1.033 \]  \hspace{1cm} (19)

Figure 7 shows the residual error of fitting (19).

Equation (20) shows that the \( \text{IMEP}_{\text{cov}} \) fitting result.

\[ \text{IMEP}_{\text{cov}}(p_m, p_e) = b_1 + b_2 p_m + b_3 p_e + b_4 p_m^2 + b_5 p_m p_e + b_6 p_e^2 \]

\[ b_1 = 1.422; \quad b_2 = 0.4745; \quad b_3 = 0.5453; \]

\[ b_4 = -0.2111; \quad b_5 = 0.2058; \quad b_6 = 0.07301; \]

\[ \text{RMSE} = 0.5033 \]  \hspace{1cm} (20)

With this model, the Sequential Quadratic Programming (SQP) [18] is used to solve the optimization problem with inequality constraint. Table 2 shows the set-point optimization validation of 6 different experiment conditions. Fuel by calibration means the minimum fuel consumption rate obtained by calibration experiment. Fuel by SQP means the minimum fuel consumption rate which is calculated by model. The result shows the difference between calibration and model calculation are quite similar, and all \( \text{IMEP}_{\text{cov}} \) are under 2%.

From the solution of all the operating condition, the map of \( p_m^*, p_e^*, u_{egr}^*, u_{vvt}^* \) can be obtained. Figure 9 shows the set-point map.

5.2. Dynamic model validation

The dynamic model of the intake manifold pressure and exhaust manifold pressure is identified which the validation conditions are conducted under the constant speed mode. The \( u_{th}, u_{egr}, u_{vvt} \) are given incentives by square wave signals. Figure 10 shows the validation result of dynamic \( P_m \) and \( P_e \) modelling. The result shows that the error of \( P_m \) and \( P_e \) are under 10%. The result indicates the identified model parameters can be used in the design of error feedback controller.

5.3. Experimental analysis

The control experiment applying the nonlinear state feedback controller (15) by directly using the measured \( p_m, p_e \) signals as the feedback variables of the control scheme, which is shown in Figure 4, is conducted. In these experiments, the throttle valve, EGR, VVT control based on the proposed control scheme are focused. The other engine conditions are controlled as follows: the engine speed is controlled by the dynamometer at a fixed speed mode, the water and oil temperatures are controlled at a constant condition which are controlled by the AND temperature regulator, while the spark advance and the fuel injection are controlled

| Engine Speed [rpm] | Torque Demand [Nm] | Fuel by calibration [mg/cycle] | Fuel by SQP [mg/cycle] | IMEPcov by SQP [%] |
|--------------------|---------------------|-------------------------------|------------------------|--------------------|
| 1200               | 40                  | 42.03                         | 42.75                  | 1.39               |
| 1200               | 60                  | 55.56                         | 56.34                  | 1.28               |
| 1600               | 40                  | 42.21                         | 42.9                   | 1.39               |
| 1600               | 60                  | 58.48                         | 58.72                  | 1.30               |
| 2400               | 40                  | 43.71                         | 43.65                  | 1.31               |
| 2400               | 60                  | 58.03                         | 57.77                  | 1.14               |

Figure 9. Set-point map by solving \( P_{ss} \). (a) \( P_m \) map. (b) \( P_e \) map. (c) \( u_{egr} \) map and (d) \( u_{vvt} \) map.
Figure 10. The validation of dynamic $P_m$ and $P_e$ modelling.

Figure 11. Experimental validation of the proposed controller.

by the traditional controller to the optimal conditions according to the engine conditions.

Figure 11 shows the control performance of the nonlinear feedback controller (15) during the torque transitions 30–70–30 Nm and fixed engine speed 2400 r/min. The blue line called set-point is the reference of the torque which is required by the driver, while the corresponding reference of $p_m$ and $p_e$ are generated by the optimal set-point MAP according to the demand torque reference. The green line is the result of open-loop controller. The red line (Case 1) and dot black line (Case 2) shown in the figure are the nonlinear feedback controller based full state control effect. Here, the controller gains of Cases 1 and Cases 2 are designed as $k_1 = 0.005$, $k_2 = 0.08$ and $k_1 = 0.01$, $k_2 = 0.1$, respectively, to achieve the stability of the closed-loop system (13) according to Proposition 1. Both cases 1 and 2 in Figure 11 illustrate that the states $p_m$, $p_e$ converge to the desired value, which are calculated by the stationary optimal problem (16) and (17). The transient response of both the intake manifold pressure and the exhaust manifold pressure can be improved with the increase in the controller gain, and the corresponding torque will be regulated to the reference faster. However, it will also result in the overshoot and oscillation of the system states when the controller gain is increased too large, in these overshoot and oscillation cases, the engine knock and misfire maybe occur which should be avoided. In the next study, a dynamic combustion constraint will be discussed, which gives a limit to the actuators respectively.

6. Conclusion

This paper proposed a control design scheme for internal combustion engines. The design process involves two phase, set-point optimization and dynamic transient control. The former is formulated as a static optimization problem with the cost function of fuel consumption rate with the constraint of engine operating point and variation of IMEP, and the latter is a model-based Lyapunov design that guarantee the error dynamics under the provided feedback control converge to the origin. Effectiveness of the presented control scheme can be claimed with the demonstrated experiment results.

However, it should be noted that the experiment validation is only performed for a single operating mode. For industrial application, the optimal set-point must be performed for covering much wilder engine operating rage. Moreover, to implement the proposed feedback control law, the pressure of exhaust manifold needs to be online measured, however, it will increase the production cost. A feasible way is to introduce a state observer of the proposed model. These two issues will be continued as further research of the next stage.
Disclosure statement
No potential conflict of interest was reported by the author(s).

Notes on contributors

Haoyun Shi received the B.S. Degree in process equipment and control engineering from the East China University of Science and Technology, Shanghai, China, in 2012, and the M.S. degree in mechanical engineering from Sophia University, Tokyo, Japan, in 2014. His M.S. dissertation was about stochastic control theory with applications in gasoline engine systems. After graduation, He worked for Hitachi Group, Tokyo, as a Researcher for four years. During that period, his research interests include control theory, engine plant modelling techniques, and the development of engine management systems. Now he is a Ph.D student in Sophia University. He is currently a member of the IEEE and JSME.

Tielong Shen received the Ph.D. degree in mechanical engineering from Sophia University, Tokyo, Japan. He has been a Faculty Member of the Chair of Control Engineering with the Department of Mechanical Engineering, Sophia University, since 1992, where he currently serves as a Full Professor. Since 2005, he has also served concurrently as a Luoja Xuezhe Chair Professor with Wuhan University, Wuhan, China, and as a Tangaoqing Xuezhe Chair Professor with Jilin University, Changchun, China. His current research interests include control theory and applications in mechanical systems, powertrain systems, and automotive systems. He is currently a member of the IEEE Automotive Control Technical Committee and the IFAC Automotive System Control Technical Committee.

References

[1] Heywood JB. Internal combustion engine fundamentals. New York (NY): McGraw-Hill; 1988.
[2] Matsuo S, Ikeda E, Ito Y, et al. The new toyota inline 4 cylinder 1.8L ESTEC 2ZR-FXE gasoline engine for hybrid car. SAE Technical Paper 2016-01-0684; 2016. doi:10.4271/2016-01-0684
[3] Fontana G, Galloni E. Variable valve timing for fuel economy improvement in a small spark-ignition engine. Appl Energy. 2009;86:96–105.
[4] Alger T, Gingrich J, Roberts C, et al. Cooled exhaust-gas recirculation for fuel economy and emissions improvement in gasoline engines. Int J Engine Res. 2011;12(3):C252–C264.
[5] Hong S, Park I, Shin J, et al. Simplified decoupler-based multivariable controller with a gain scheduling strategy for the exhaust gas recirculation and variable geometry turbocharger systems in diesel engines. J Dyn Syst Meas Control. 2017 May;139(5):051006.
[6] Leroy T, Chauvin J. Control-oriented aspirated masses model for variable-valve-actuation engines. IFP Energies nouvelles; 1 et 4 avenue de Bois-Preau, 92852 Rueil-Malmaison, France; 2010 Jul.
[7] Wahlstrom J, Eriksson L, Nielsen L. EGR-VGT control and tuning for pumping work minimization and emission control. IEEE Trans Control Syst Technol. 2010 Jul;18(4):993–1003.
[8] Wahlstrom J, Eriksson L. Output selection and its implications for MPC of EGR and VGT in diesel engines. IEEE Trans Control Syst Technol. 2013 Dec;21(3):932–940.
[9] Nieuwstadt M, Kolmanovsky I, Moraal P, et al. EGR-VGT control schemes: experimental comparison for a high-speed diesel engine. IEEE Control Syst Mag. 2000 Jun;20(3):63–79.
[10] Jankovic M, Jankovic M, Kolmanovsky I. Constructive Lyapunov control design for turbocharged diesel engine. IEEE Trans Control Syst Technol. 2000 Mar;8(2):288–299.
[11] Kang X, Shen T. Experimental comparisons between LQR and MPC for spark-ignition engine control problem. In: The 36th Chinese Control Conference; 2017 Jul 26–28.
[12] Jang W, Shen T. Lyapunov-based nonlinear feedback control design for exhaust gas recirculation loop of gasoline engines. ASME J Dyn Syst Meas Control. 2017 Jan;141(5):051005. Paper No: DS-18-1320.
[13] Teodosio L, De Bellis V, Bozza F. Fuel economy improvement and knock tendency reduction of a downsized turbocharged engine at full load operations through a low-pressure EGR system. SAE Paper No. 2015-01-1244; 2015.
[14] Guzzella L, Onder C. Introduction to modeling and control of internal combustion engine systems. Springer; 2009. ISBN 978-3-642-10774-0.
[15] Eriksson L, Nielsen L. Modeling and control of engines and drivelines. Chichester: John Wiley Sons; 2014.
[16] Shen T, Zhang J, Jiao X, et al. Transient control of gasoline engines. Boca Raton: CRC Press; 2015.
[17] Kim S, Choi S, Jin H. Pressure model based coordinated control of VGT and dual-loop EGR in a diesel engine air-path system. Int J Autom Technol. 2016 Mar;17(16):193–203.
[18] Powell M. A fast algorithm for nonlinearly constrained optimization calculations. In: Numerical analysis. Vol. 139(5). Berlin, Heidelberg: Springer; 1978. p. 144–157.