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Cyclic moving average control approach to cylinder pressure and its experimental validation

Po LI [1,2], Tielong SHEN [2], Junichi KAKO [3], Kaipei LIU [1]

(1.School of electrical engineering, Wuhan Unversity, Wuhan Hubei 430072, China;

2.Department of Mechanical Engineering, Sophia University, Tokyo 102-8554, Japan;

3.Higashifuji Technical Center, Toyota Motor Corporation, Shizuoka 410-1193, Japan)

Abstract: Cyclic variability is a factor adversely affecting engine performance. In this paper a cyclic moving average regulation approach to cylinder pressure at top dead center (TDC) is proposed, where the ignition time is adopted as the control input. The dynamics from ignition time to the moving average index is described by ARMA model. With this model, a one-step ahead prediction-based minimum variance controller (MVC) is developed for regulation. The performance of the proposed controller is illustrated by experiments with a commercial car engine and experimental results show that the controller has a reliable effect on index regulation when the engine works under different fuel injection strategies, load changing and throttle opening disturbance.

Keywords: In-cylinder pressure balancing; Cyclic moving average modeling; ARMA model; MVC

1 Introduction

In internal combustion engines, the cyclic variation of in-cylinder pressure profile is one of the important factors that affect torque generation precision and fuel efficiency. For example, the speed fluctuation caused by cyclic torque variation is not ignorable even in the steady operation mode, and more than 10% fuel efficiency is damaged by the cyclic variability [1]. Furthermore, the cyclic imbalance is also a cause damaging exhaust emission performance. As shown in [2], when an engine runs in lean situation, 30% of the emission can be reduced if the engine runs without cyclic variability. However, the cyclic variation of in-cylinder pressure profile, as a result of the combustion process in each cycle, is an unavoidable natural phenomenon with chaotic property [3]. Therefore, cyclic balancing is still a challenging issue in automotive engine management.

From the control point of view, a feasible way to deal with the cyclic imbalance is to develop a model of the cyclic variation and to establish a model-based control approach with the model. Indeed, in the past two decades, a number of studies have been reported along this research line. A combustion event-based discrete model for the cyclic variation of heat release was proposed in [4, 5] where the main attention is focused on the effect of residual gas, and furthermore, the model is utilized in on-line heat release balancing control by introducing neural networks for learning the relationship from fuel charge to heat release [6]. However, the air and fuel inlet dynamics are not taken into account in the physical event-based model. In contrast to the physical model, most of the papers addressed the cyclic variation modeling using the statistic approach and in-cylinder pressure-related indexes are explicitly used to evaluate cyclic balance performance. For instance, balancing the location of peak pressure (LPP) was investigated in [7] where the ARMA model is introduced to describe the dynamics from the ignition time to the LPP and the generalized minimum variance controller was proposed based on the time series model. The cyclic balancing scheme with the LPP index was also investigated in [8, 9]; however, a neural network is used to predict the LPP and a conventional PID control law is exploited for the LPP regulation. The mass fraction burned (MFB) is an alternative in-cylinder pressure-related index for the cyclic balancing control. In [10] the MFB balancing control was presented with PI control strategy. For the heavy burden in calculation, the indicated mean effective pressure (IMEP), as another index for cyclic variability, is usually used for off-line evaluation, and a feedback control with a predictive model for IMEP is proposed in [11].

In this paper, we will present a moving-average control approach to the cylinder pressure at the top dead center (TDC). Four-cycle moving average of the cylinder pressure at TDC is chosen for evaluating the cyclic variability and the spark advance (SA) is used as the control input. First, we will introduce ARMA model to describe the dynamics from the actuating signal to the control output, the moving average of the TDC pressure. With the model, a one-step ahead prediction-based minimum variance controller will be provided to regulate the control output at the nominal value according to the steady operation mode. The second contribution of this paper is to provide an experimental validation of the proposed control approach with a commercial car engine. It will be shown that the modeling and the control approach are effective for rejecting the variation of the moving average index even when the engine works under different fuel injection strategies, additional load changing and throttle opening disturbance.

2 Preliminaries

In the spark ignition engines, combustion of the air and fuel mixture trapped in cylinders produces heat to force the piston to move down, and the linear motion of the piston is transferred by the connection rod into the rotation of the crankshaft. However, due to the uncertainties in the air and

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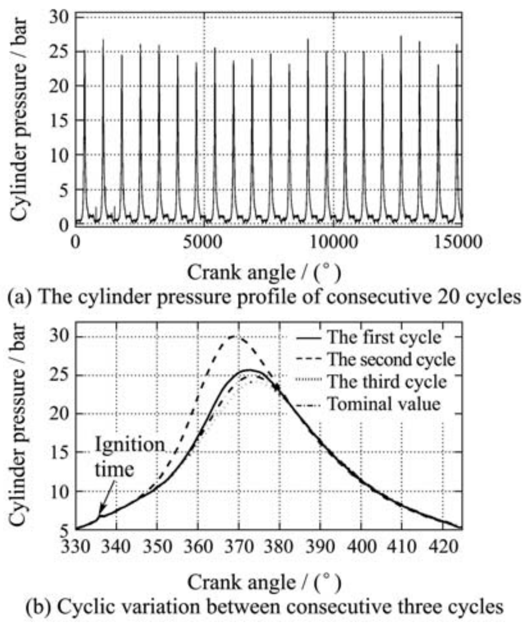
fuel intake paths, statistical combustion efficiency, etc., the combustion profile varies cyclically, and this cyclic variation will immediately cause fluctuation in torque generation. As a result, it damages the engine performance.

For example, this cyclic variation can be observed from the cylinder pressure profile. Fig.1 shows an experimental result on a six-cylinder engine test bench, where actuating signals such as throttle angle, spark advance and fuel injection are set to be constants so that the engine works at 2000 rpm under the constant load 86.5 Nm. It is obvious from Fig.1 (a) that the maximum value of the cylinder pressure of each cycle is different and the variation range is about 4 bar. Furthermore, comparing the cylinder pressure profiles in each cycle, it is also easy to observe the cyclic imbalance. Fig.1 (b) shows three consecutive cycles and the average cylinder pressure profile of 1000 cycles, from which remarkable changes of cylinder pressure value at the same location in crankshaft angle can be observed during the combustion, especially, around the TDC in crankshaft angle.



(a) The cylinder pressure profile of consecutive 20 cycles

(b) Cyclic variation between consecutive three cycles

Fig. 1 In-cylinder pressure profile.

To evaluate the performance of torque generation, IMEP is popularly accepted and $\mathrm{IMEP}(k)$ of the k-th cycle is defined as:

$$\mathrm{IMEP}(k) = \frac{1}{V_0}\int_0^{4\pi} P(\theta)\mathrm{d}V(\theta), \tag{1}$$

where V_0 is the cylinder volume from TDC to BDC, $P(\theta)$ and $V(\theta)$ are in-cylinder pressure and cylinder volume from TDC to the piston at crank angle θ, respectively. However, in practice, the $\mathrm{IMEP}(k)$ is not feasible for performing on-line feedback balancing control, due to the heavy burden in calculation.

As shown in [3, 12], the in-cylinder pressure-related indexes are usually employed to evaluate the combustion variation. In this paper, we choose the cylinder pressure $P(\theta)$ at TDC as the cyclic balancing performance index, and actually, in the controller design, we will use the four cycles moving average value as the control target to avoid undesirable sensitivity on the stochastic feature in the combustion process, i.e., the control output of the cyclic balancing control system which will be designed later is defined by

$$y(k) = \frac{1}{4}\sum_{i=0}^{3} P_{\mathrm{TDC}}(k-i), \tag{2}$$

where $P_{\mathrm{TDC}}(k)$ is the cylinder pressure at TDC of the k-th cycle. Fig.2 illustrates the response $y(k)$ of the experiments shown in Fig.1.

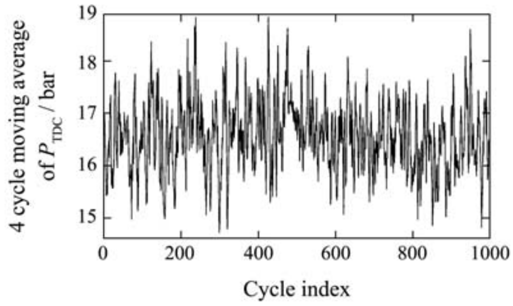


Fig. 2 Four-cycle moving average of P_{TDC}.

It should be noted that the in-cylinder pressure at TDC is not the only index for indicating combustion variation. For example, in-cylinder pressure at crankshaft angle 370° may also be a candidate. However, in most steady operation modes, crankshaft angle 370° is near LPP and rapid or slow combustion may lead the peak pressure position to appear before or after this position, respectively (Fig.1 (b)). In this situation, the measurement of in-cylinder pressure at crankshaft angle 370° cannot indicate the combustion variation effectively.

Many actuating factors might affect the control output $y(k)$. However, due to the evident causality between the starting time of the combustion and P_{TDC}, we choose the spark advance as the control actuating signal. The effect of spark advance on P_{TDC} can be observed more clearly from Fig.3. In the experiments for Fig.3, besides the spark advance, all other actuating signals are kept constant the same as the experiments of Fig.1. In Fig.3, each of the six curves is the average profile of 1000 consecutive combustion cycles under different spark advance. It is noticeable that the average of P_{TDC} increases with spark advance enlarging.

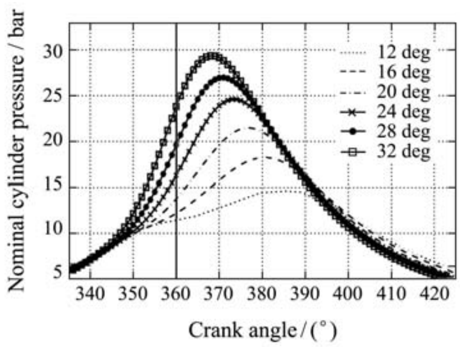


Fig. 3 Cylinder pressure profile with different spark timing.

3 Modeling and control law

As mentioned above, the aim of this paper is to provide a control law for regulating the chosen index $y(k)$, and consequently, for improving the cyclic balance. To this end, with the selected actuating signal, we will first establish a dynamical model for the controlled plant, and based on the model a feedback control law will be derived subsequently.

Indeed, the mechanism determining the physical dynamics from the chosen control input and the output is not simple, and moreover, in practice this dynamics is inter-coupled with other paths such as emission control loop, throttle operation, etc. For example, the combustion with variation disturbs the rotational motion of the crankshaft, and the variation in speed during induction stroke will result in the fluctuation of intake air mass. Moreover, the combustion of the following cycle is affected by the recycled residual gas from its previous one. Based on a physical hypothesis, a model emphasizing residual gas effect was proposed in [4] to produce dynamical cyclic variation pattern. However, this model is not feasible in the stoichiometric situation. Moreover, the calculation of heat release is a heavy burden for online application. In fact, since the heat release process for combustion is not clear, it is difficult to establish the dynamics of combustion between cycles from aerodynamics and thermodynamics. In other words, it is difficult to model the dynamics of the moving average index between cycles from the view of physics.

As an alternative, system identification methods can be used from the view of statistics. In this paper, we employ the following ARMA model to represent the dynamics from ignition time u to y:

$$A(q^{-1})y(k) = B(q^{-1})u(k-m) + C(q^{-1})\varepsilon(k), \quad (3)$$

where

$$A(q^{-1}) = 1 + a_1q^{-1} + a_2q^{-2} + \cdots + a_iq^{-i},$$
$$B(q^{-1}) = b_0 + b_1q^{-1} + b_2q^{-2} + \cdots + b_jq^{-j},$$
$$C(q^{-1}) = c_0 + c_1q^{-1} + c_2q^{-2} + \cdots + c_rq^{-r},$$

and m is the system delay.

To identify the model for the engine test bench, which will be explained in detail later, the recursive least squares algorithm is applied with the selection of the orders, and the model order that determines the system structure is decided by a trial-and-error approach with experimental results. As a result, the following 2nd-order model is determined.

$$(1 + a_1q^{-1} + a_2q^{-2})y(k) = b_0u(k-1) + \varepsilon(k) \quad (4)$$

with identified parameters as

$$a_1 = -0.88587,\ a_2 = 0.16736,\ b_0 = 0.13226.$$

The experimental result used in the parameter identification is performed under the steady operation mode as shown in Table 1. In the experiment, the input signal u is generated to be pseudo random sequence both in amplitude and time duration. Data of 600 cycles is used for parameter identification, which is illustrated in Fig.4. The effectiveness of the model (4) is validated with experimental result. Fig.5 shows the modeling error, the average and the standard deviation is -0.0587 bar and 0.594 bar, respectively.

From the view of practical application, the error of the second-order model with determined parameters is allowable. It should be noted that increasing the order of the model does not necessarily lead to improving the model precision. Fig.6 shows the response of the 3rd-order model with the error between the model and the measured data from $a_1 = -0.8669, a_2 = 0.0037, a_3 = 0.1780, b_0 = 0.1788$ and $b_1 = 0.0310$. The average and the standard deviation of the error is -0.0633 bar and 0.5905 bar, respectively.

Table 1 Specifications of the steady mode for model structure determination.

Throttle open ($^\circ$)	6.9
Load (Nm)	60
Speed (rpm)	1800
Fuel injection for cylinder No.1, 3, 5 (mm^3/cycle)	13.1
Fuel injection for cylinder No.2, 4, 6 (mm^3/cycle)	13.6
Nominal value of SA of No.2 BTDC ($^\circ$)	26.8
Nominal value of P_{TDC} of No.2 (bar)	12.6

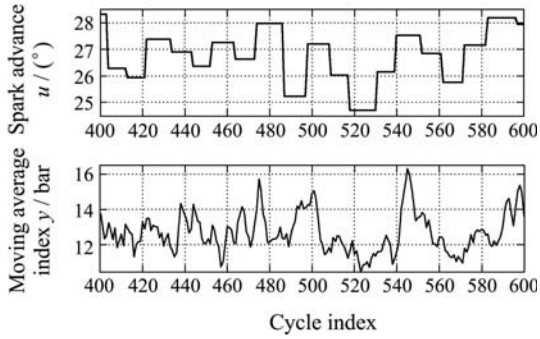


Fig. 4 Experiment data for system identification.

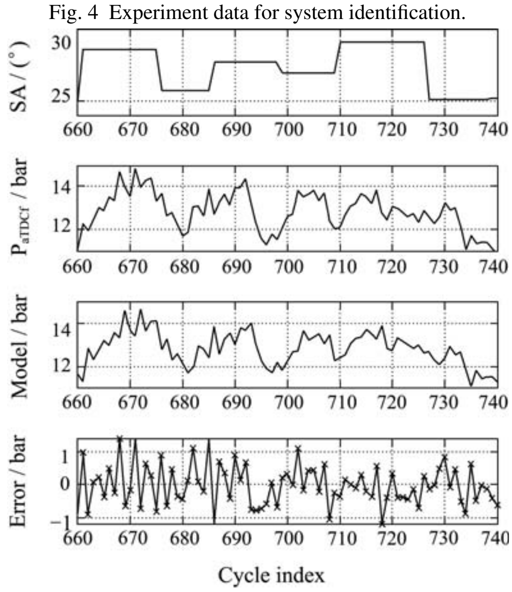


Fig. 5 Identified 2nd-order model output.

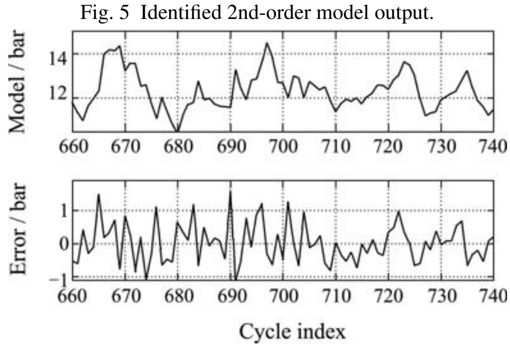


Fig. 6 Identified 3rd-order model output.

The control subject considered in this paper is to keep $y(k)$ to the nominal value that is determined by the corresponding operation mode. Thus, a natural choice for the cost function is the variance of the error

$$J = \min_{u(k)}(E[(y(k+1) - y_0)^2]), \quad (5)$$

where y_0 is the given nominal value, which is decided by the steady operation mode.

As is well-known, a routine with the identified model (4) along the design procedure according to [13] obtains the minimum variance control law as follows:

$$u(k) = \frac{1}{b_0}y_0 + \frac{a_1}{b_0}y(k) + \frac{a_2}{b_0}y(k-1). \tag{6}$$

4 Experimental validation

4.1 Experimental set-up

The hardware system of the test bench comprises the SI engine, engine electronic control unit (ECU), dSPACE and a dynamometer. The schematic diagram of the system is shown in Fig.7. The experimental engine is a Toyota 3-liter, 6-cylinder and direct injection production and the six cylinders are denoted by No.1 to No.6 according to the ignition sequence. In cylinder No.2, a pressure transducer (KISTLER Type 6118AFD35Q01) is equipped at the position of the spark plug and the pressure is used for on-line control. We use the dynamometer to simulate additional load effect. The control algorithm is built in Matlab/Simulink software platform and downloaded to dSPACE from a computer. Meanwhile, this computer is also employed for parameter resetting and data saving during experiments. The generated spark advance command for cylinder No.2 is sent from dSPACE to ECU every cycle and in other cylinders the ignition time are fixed to the nominal value. The sampling period of dSPACE is set as 0.25 ms and the synchronization between the dSPACE and ECU is guaranteed by the signal of crank angle encoder (KISLTER Type 2613B1), which generates alternate high and low TTL format voltage every 6° in crank angle.

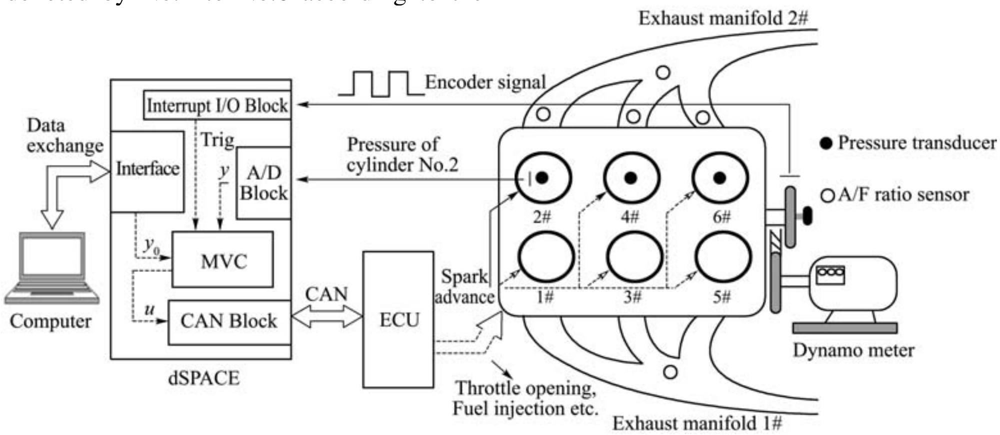
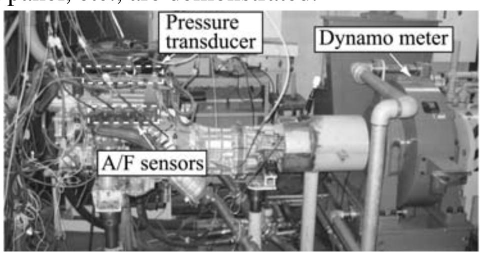


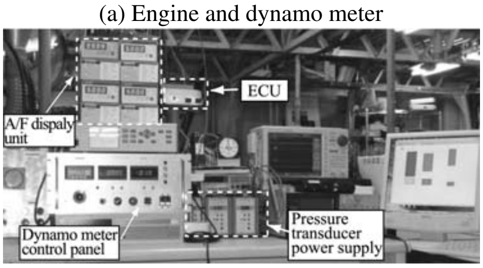
Fig. 7 Schematic diagram for test bench system.

Other commands for engine operation, such as fuel injection and throttle opening etc., are controlled by the ECU. During experiments the dSPACE can reset these commands to simulate the disturbance effect.

Fig.8 shows the real image of the test bench system. The engine, dynamometer system, individual A/F sensors and cylinder pressure transducers are illustrated in Fig.8 (a). In Fig.8 (b) the apparatus for power supply, display unit and control panel, etc., are demonstrated.



(a) Engine and dynamo meter



(b) Experiment instrument and appatatus

Fig. 8 Test bench system.

4.2 Experimental validation for control scheme

In practice, an on-line recursive least-squares identification is taken for consecutive 2500 cycles with a forgotten factor 0.998 when the engine works under the same operation mode of Table 1. The identified parameters are $a_1 = -0.833, b_1 = 0.125, c_1 = 0.138$. During experiments a limit of spark advance deviation is set as $\pm 2°$ for engine protection.

To validate the proposed model and model-based control scheme, experiments are implemented when the engine works under different fuel injection strategies, load changing and throttle opening disturbance. In all experiments, the same model-based control law are adopted with respect to the on-line identified parameters as mentioned above. The experiments are carried out for two cases according to fuel injection strategies. The first case is fixed fuel supply and the other is fuel injection adjusting under an air/fuel ratio (A/F) balancing controller.

4.2.1 Case 1: Engine works with fixed fuel injection

With fuel injection fixed to the nominal value, Fig.9 and Fig.10 show results of regulating the moving average index. The experiment for Fig.9 is with fixed throttle opening and additional load. As shown in Fig.9, when the controller switches off at the cycle index 108, the standard deviation changes from 0.8928 bar to 1.0160 bar with about 20% improvement in performance. After the controller operates again from cycle index 236, the standard deviation is then reduced to 0.5880 bar.

Fig.10 demonstrates the regulation results for load disturbance which changes from the nominal value 60 Nm to 45 Nm during experiments. It is clear P_{TDC} decreases when load changes without spark advance adjustment and this trend is effectively counteracted by the proposed control scheme.

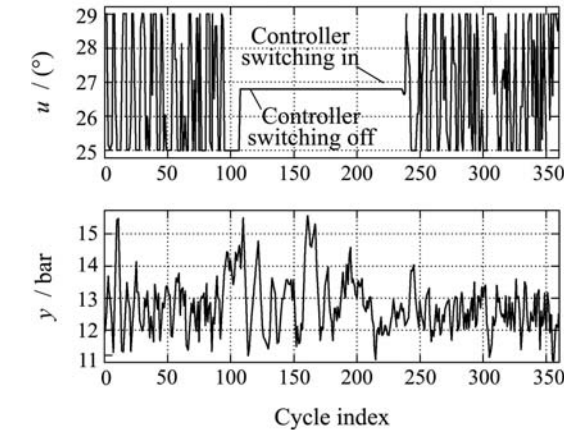


Fig. 9 Control with fixed fuel injection.

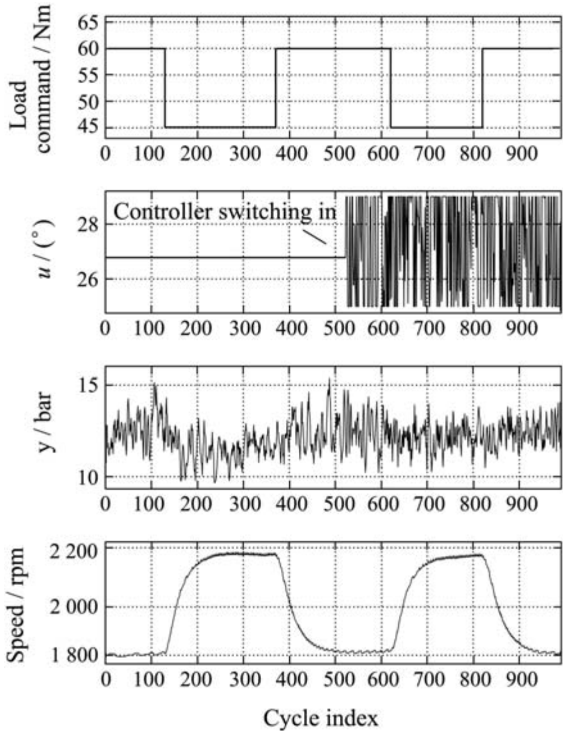


Fig. 10 Control with fixed fuel injection under load disturbance.

4.2.2 Case 2: Engine works with individual A/F balancing

In practice, the fixed fuel injection is not a flexible condition for A/F balancing. By using the individual A/F sensors (see in Fig.7), a PI controller is employed for fuel injection adjustment to keep A/F ratio to the stoichiometric value. The experimental results for moving average index regulation under this situation are illustrated in Fig.11, Fig.12, and Fig.13, respectively.

Fig.11 shows the experimental result when the engine works under fixed throttle opening and additional load. The controller switches off at the cycle index 133 and standard deviation changes from 0.6845 bar to 1.2718 bar. After the controller operates again from cycle index 259, the standard deviation reduces to 0.7154 bar, which is more than 40% improvement.

The load disturbance experimental results under individual A/F balancing is illustrated in Fig.12 and the trend for decreasing in P_{TDC} is counteracted by the adjustment of spark advance. However, due to the coupling effect between the proposed MVC controller and the A/F balancing one, the regulating effect shown in Fig.12 is not as clear as that shown in Fig.10.

Fig.13 shows the experimental results when the engine works with throttle opening disturbance and in this situation the additional load is fixed. Fig.13 (a) illustrates the performance when the throttle opening increases from 6.9° to 7.5°. Similarly, in Fig.13 (b) the experimental result when the throttle opening decreases from 6.9° to 6.3° is demonstrated. In both situations, the proposed control scheme shows reliable effect for regulation.

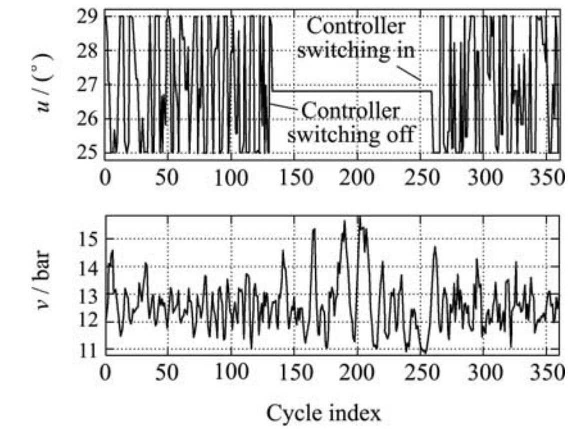


Fig. 11 Control with A/F balancing.

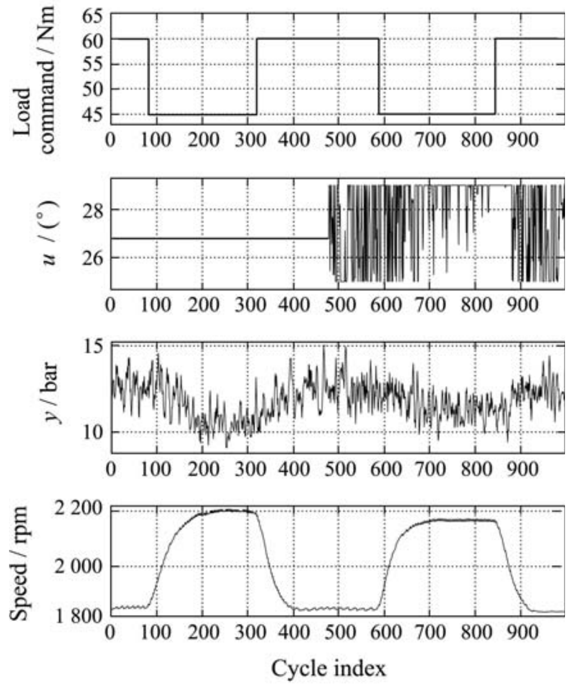


Fig. 12 Control with A/F balancing under load disturbance.

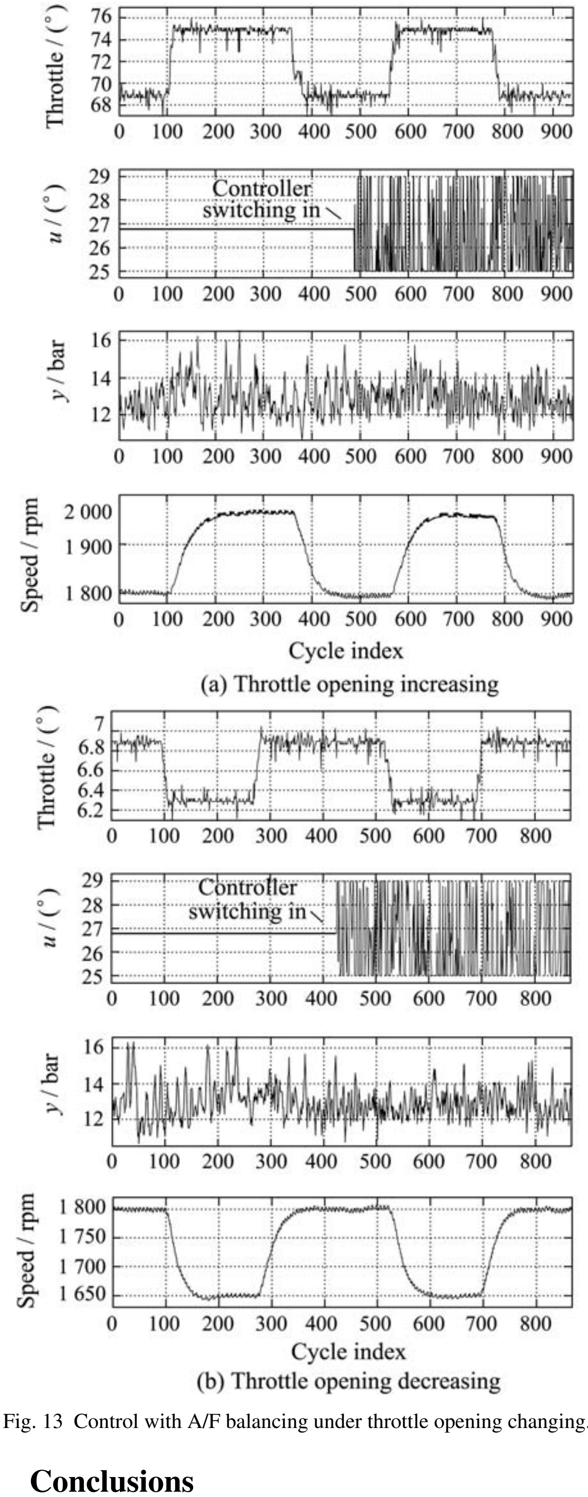


Fig. 13 Control with A/F balancing under throttle opening changing.

5 Conclusions

This paper presented a four-cycle moving average control approach to regulating the in-cylinder pressure at TDC. ARMA model was introduced for the dynamics from spark advance to the moving average index and a model-based MVC controller was developed. Furthermore, the proposed model and model-based control law were validated by experiments. Experimental results show that the proposed control scheme can obtain reliable effect for index regulation when engine works under different fuel injection strategies, additional load changing and throttle opening disturbance. It should be noted that in this paper, the proposed SA adjusting scheme is developed only for $P_{\rm TDC}$ balancing. In fact, the ignition signal is also a traditional actuator of control loops for engine performance in other aspects, such as emission control, etc., which are not considered in this paper. One of the future works will focus on the development of the SA control strategies by considering both the combustion balancing and other aspects.

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References

[1] J. Vance, P. He, J. Sarangapani, et al. Neural network-based output feedback controller for lean operation of spark ignition engines[C]//*Proceedings of American Control Conference*. New York: IEEE Press, 2006: 1898 – 1905.

[2] T. Inoue, S. Matsushita, K. Nakanishi, et al. Toyota lean combustion system: the third generation system[M]//*International Congress & Exposition Technical Papers*. Warrendale: SAE Press, 1993: No.930873.

[3] O. Nir, D. Mark, S. Eran. Cyclic variability in spark ignition engines: A literature survey[M]//*International Congress & Exposition Technical Papers*. Warrendale: SAE Press, 1994: No.940987.

[4] C. S. Daw, C. E. Finney, J. B. Green Jr., et al. A simple model for cyclic variations in a spark ignition engine[M]//*SAE International Fall Fuels and Lubricants Meeting and Exposition*. Warrendale: SAE Press, 1996: No.962086.

[5] C. E. A. Finney, J. B. Green Jr., C. S. Daw. Symbolic time-series analysis of engine combustion measurements[J]. *SAE Transactions*, 1998, 106(3): 888 – 897.

[6] P. He, S. Jagannathan. Neuro-controller for reducing cyclic variation in lean combustion spark ignition engines[J]. *Automatica*, 2005, 41(7): 1133 – 1142.

[7] G. Triantos, A. T. Shenton, S. D. Carroll. Minimum variance control of cylinder peak pressure position[C]//*Proceedings of IFAC Symposium on Advances in Automative Control*. Oxford: Elsevier Science Ltd., 2005: 143 – 148.

[8] S. Park, P. Yoon, M. Sunwoo. Feedback error learning neural networks for spark advance control using cylinder pressure[J]. *Proceedings of the Institution of Mechanical Engineers-Part D*, 2001, 215(2): 625 – 636.

[9] S. Park, P. Yoon, M. Sunwoo. Cylinder pressure-based spark advance control for SI engines[J]. *JSME International Journal-Series B*, 2001, 44(2): 305 – 312.

[10] N. Muller, O. Nelles, R. Isermann. Closed-loop ignition control using on-line learning of locally-tuned radial basis function networks[C]//*Proceedings of American Control Conference*. New York: IEEE Press, 1999: 1356 – 1360.

[11] S. P. Stevens, P. J. Shayler. A basis for the predictive control of cyclic dispersion in a spark ignition engine[C]//*Proceeding of IMechE International Conference on Combustion in Engines Technology, Applications and the Environment*. 1992: 175 – 179.

[12] J. D. Powell. Engine control using cylinder pressure: past, present, and future[J]. *Journal of Dynamic Systems, Measurement, Control*, 1993, 115(2B): 343 – 350.

[13] K. J. Astrom. *Introduction to Stochastic Control Theory*[M]. New York: Academic Press, 1970.

Po LI received his B.E. degree in Electronic Information Engineering from Wuhan University, Wuhan, China, in 2004, where he is currently pursuing his Ph.D. degree. He has been a collaborator of the Department of Engineering and Applied Science, Sophia University, Tokyo, Japan, since 2006, and joined the Cooperative Research of Powertrain System Control, supported by Toyota Motor Corporation, Tokyo. His current research interests include nonsmooth dynamics and engine balancing control. E-mail: lipo@me.sophia.ac.jp.

Tielong SHEN received his Ph.D. degree in Mechanical Engineering from Sophia University, Tokyo, Japan. From April 1992, he has been a faculty member of the Chair of Control Engineering in Department of Mechanical Engineering, Sophia University, where he currently serves as an associate professor. His research interests include control theory and application in mechanical systems, power systems, and automotive powertrain. E-mail: tetusin@hoffman.cc.sophia.ac.jp.

Junichi KAKO received his B.E. degree from Nagoya Institute of Technology, Nagoya, Japan. From 1989, he has been a faculty member of Toyota Motor Corporation, Tokyo, Japan. Since 2002, he has been with Future Project Division in which he was responsible for the R&D of model-based engine control system. Currently, he is developing engine control systems in the Powertrain Management Engineering Division, Toyota Motor Corporation. E-mail: jun@kako.tec.toyota.co.jp.

Kaipei LIU received his Ph.D. degree in Application Engineering of Computer Technology from Wuhan University, Wuhan, China. During 1994-2000, he has been a faculty member of Wuhan University of Hydraulic and Electric Engineering, Wuhan, China. He was a visiting research fellow at Saarland University, Germany, from 1994 to 1995. Since August 2000, he served as a professor of School of Electrical Engineering, Wuhan University, Wuhan, China. His research interests include digital signal process, adaptive signal process, harmonic detection, and control of power system. E-mail: pliu@whu.edu.cn.