Experimental Investigation of Close-Loop Control of HCCI Engine Using Dual Fuel Approach

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ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) offers great promise for excellent fuel economy and extremely low emissions of NO_x and PM. HCCI combustion lacks direct control on the 'start of combustion' such as spark timing in SI engines and fuel injection timing in CI engines. Auto ignition of a homogeneous mixture is very sensitive to operating conditions of the engine. Even small variations of the load can change the timing from 'too early' to 'too late' combustion. Thus a fast combustion phasing control is required since it sets the performance limitation of the load control. Crank angle position for 50% heat release is used as combustion phasing feedback parameter.

In this study, a dual-fuel approach is used to control combustion in a HCCI engine. This approach involves controlling the combustion heat release rate by adjusting fuel reactivity according to the conditions inside the cylinder. Two different octane fuels (methanol and n-heptane) are used for the study. Port fuel injection technique is used for preparing homogeneous mixture of methanol, heptane and air using two separate injectors for methanol and heptane. Close loop control of combustion phasing is attained by instantaneous variation of fuel ratio of methanol and n-heptane while maintaining the injected fuel energy constant. Total fuel energy injected is used to control the IMEP of the engine. It is found that controller is able to keep close track of the reference combustion phasing using PID control by changing the fuel ratio. PID control of combustion phasing and IMEP using dual fuel is achieved and is successfully demonstrated in the HCCI engine in the study.

INTRODUCTION

HCCI is an alternative engine concept for the future, which has potential to provide significantly improved efficiency and emissions characteristics vis-a-vis current engine technologies. Stable and efficient operation of HCCI combustion engines need tightly controlled combustion phasing. One of the main challenges in HCCI engine is combustion control since the onset of combustion depends on temperature, pressure, and mixture formation in the combustion chamber. In HCCI engine, combustion starts automatically in combustible mixture at auto-ignition temperature during the compression stroke. However there is no direct actuator to initiate the combustion. Without robust combustion control, excessively advanced or retarded combustion (too late combustion) can take place in the engine. Early/ advanced combustion can yield unacceptable pressure rise rates or unacceptable peak cylinder pressure, causing excessive noise, which may potentially damage the engine. Additionally, NO_x emission from the engine tends to increase with ignition advance [1-2]. Another driver to have an effective close-loop combustion control is the fact that late combustion phasing leads to incomplete combustion and increasing emissions of carbon monoxide (CO) and unburned hydrocarbons (UHC). The worst case of 'too late combustion' is a complete misfire, which if repeated, can cause engine to stall. Another possibility is that the engine enters a vicious cycle with one misfiring followed by a cycle with very high combustion rate due to the fuel-rich residuals, followed by another misfire and so on [3-4]. Thus the proper choice of the combustion phasing at each operating point is very crucial for HCCI engines. In HCCI engine, some operating points are inherently unstable. This means that it is impossible to map an HCCI engine reliably. Therefore close loop combustion control is necessary to guarantee correct combustion phasing [3, 5].

HCCI engines do not have direct actuator for combustion phasing control. Therefore feedback is essential to control the combustion phasing of the HCCI engine. The combustion phasing can be defined based on several different criterions such as crank angle position for10% or 50% heat release, crank angle position for peak pressure, crank angle position for maximum pressure rise rate etc. [6]. For close-loop control of the combustion phasing, it is necessary to measure the combustion phasing. There are several sensors used for combustion phase sensing. The most commonly used method for combustion phasing determination is by employing the heat release analysis of the in-cylinder pressure data. In a laboratory environment, piezoelectric pressure transducers are generally used for the in-cylinder pressure measurements. However, due to higher cost and shorter life span of piezoelectric pressure transducers, they are not used in production grade engines. However, several low-cost transducers exist for pressure sensing [7,8,9]. The accuracy of these low-cost sensors is sufficient for feedback control of HCCI combustion phasing [10]. Optical pressure transducers are also used for combustion phasing determination [3]. One of the most promising technologies in combustion phasing is ion current sensors, which are relatively inexpensive and use the electronic conductive properties in the reaction zone [11-12]. The drawback with ion current sensing is that it gives only localized information. However, if the charge is reasonably homogeneous as in case of HCCI engine, a localized measurement is sufficient. The ion current signal is dependent on the fuel properties and engine operating conditions [10]. Another means to obtain the combustion phasing is to use knock sensors or microphones on the engine block [13-14].

Several means to actuate the combustion phasing in HCCI engine control have been suggested by various researchers [10, 15,16,17,18] such as dual fuels, variable valve actuation, variable compression ratio and thermal management. They all fulfill the requirement of fast actuation, which is essential to control the combustion phasing, however all methods have their own merits and drawbacks.

In dual fuel type control methodology, two fuels with different auto-ignition properties are used. The system has main fuel of a high octane number rating and secondary fuel with low octane number rating [16]. Different auto-ignition properties of dual fuel system are used to control the combustion phasing in HCCI as blending the two fuels in different fuel ratios changes the auto-ignition properties. The use of commercial fuels or mixtures of single-component fuels and commercial fuels and primary reference fuels have been investigated in previous studies [18-19]. Test fuels considered include mixtures of ethanol and n-heptane [20]; n-heptane and diesel [21]; and gasoline and diesel [22-23]. These studies successfully demonstrated the dual fuel HCCI concept with varying levels of feasibility.

Variable valve actuation (VVA) provides a very fast means of affecting the breathing of the engine. For HCCI control, there are two major methods; residual gas control and effective compression ratio control [3]. VVA can be used to control the initial charge temperature by retaining residual gas or re-breathing hot exhaust gas through the exhaust valve. Another way to control the charge temperature is by advancing or retarding the intake valve closing (IVC). This reduces the effective compression ratio of the engine. Use of VVA for control of HCCI engine is of growing interest and researchers are exploring it by modeling as well as experiments [24,25,26]. Variable compression ratio (VCR) can be used to control combustion phasing by increasing the compression ratio and the charge temperature after compression. VCR can be achieved by several different methods. The drawbacks are that VCR system currently does not allow individual cylinder control, which is necessary to obtain good combustion phasing control, and VCR systems are complex and expensive [10].

Electrical preheating of air (used in this study) has slow response for cycle-by-cycle control of HCCI engine. Therefore intake temperature control is usually thought as 'too slow' approach for close-loop control of transient HCCI combustion. A more sophisticated solution is fast thermal management (FTM) introduced by Haraldsson et al. which makes use of two temperature sources; one hot and the other cold [27]. With fast thermal management, the intake air is a mixture of air from the cold source and the hot source. Two inversely coordinated throttles select the correct ratio of hot and cold air flow in order to obtain appropriate intake air temperature [3].

Several different control methods have been attempted for HCCI control such as physical model based control, experimentally derived model based control, and manually tuned controllers [28]. Several researchers have used physical model based controllers for HCCI combustion control. Close loop combustion control of HCCI engine using PID controllers and model based controllers are described using an active valve train (AVT) hydraulic valve timing system [24,29]. The development of a physical model for non-linear control of HCCI was done [30]. Researchers used a simplified non-linear feedback controller to regulate the CA_{50} during load transients. Control was done by modifying the lift of the secondary exhaust valve opening to control internal EGR. Shaver et al. developed a MIMO controller, which was used to decouple the control of combustion phasing and peak cylinder pressure [31]. Kulzer et al. developed both data driven and physics based models that were used to design controllers to track load, while regulating the pressure rise and CA_{50} [32].

Several investigations also used empirically-derived model based controllers. The use of system identification to design model based controllers and the implementation of these controllers on various engine test setups is also attempted [33,34,35,36]. Strandh et al. used a model based Linear-Quadratic-Gaussian (LQG) controller, which was developed using system identification techniques [34]. The model based LQG controller was shown to perform slightly better than the manually tuned PID controller. Fast Thermal Management was used in yet another study to control the CA_{50} timing [27]. In this study, control was done with a manually tuned PID loop. The time constant was found to be 8 engine cycles, which was noted as being relatively slow. Bengtsson et al. examined potential future HCCI control strategies [35]. They suggested that more detailed physical based control models are required and it was also indicated that cycle-to-cycle control of trapped residuals would be extremely beneficial.

Some non-model based controllers are also used in HCCI combustion investigations. Ohmura et al. implemented PI controller for HCCI combustion by modulating the fraction of external residuals, as well as the temperature of the intake air/ residual mixture [37]. They used a combination of two PI controllers, a slow one for IMEP, and a faster one for CA_{50}. Souder et al. showed that microphones can be used to provide feedback signals for combustion phasing control [13]. An exhaust back-pressure valve was used to regulate the amount of residuals, which were used to control the combustion phasing.

In this paper, close loop control is achieved using dual fuel (methanol and n-heptane) method by using in-cylinder pressure signal feedback at constant engine speed of 1200 rpm. Most dual fuel studies on HCCI engines are conducted using primary reference fuels. However there are few studies [6, 10] conducted using ethanol blends (ethanol-heptane) and heptane. In the present study, 100% methanol and n-heptane is used for the experimental evaluation.

EXPERIMENTAL SETUP

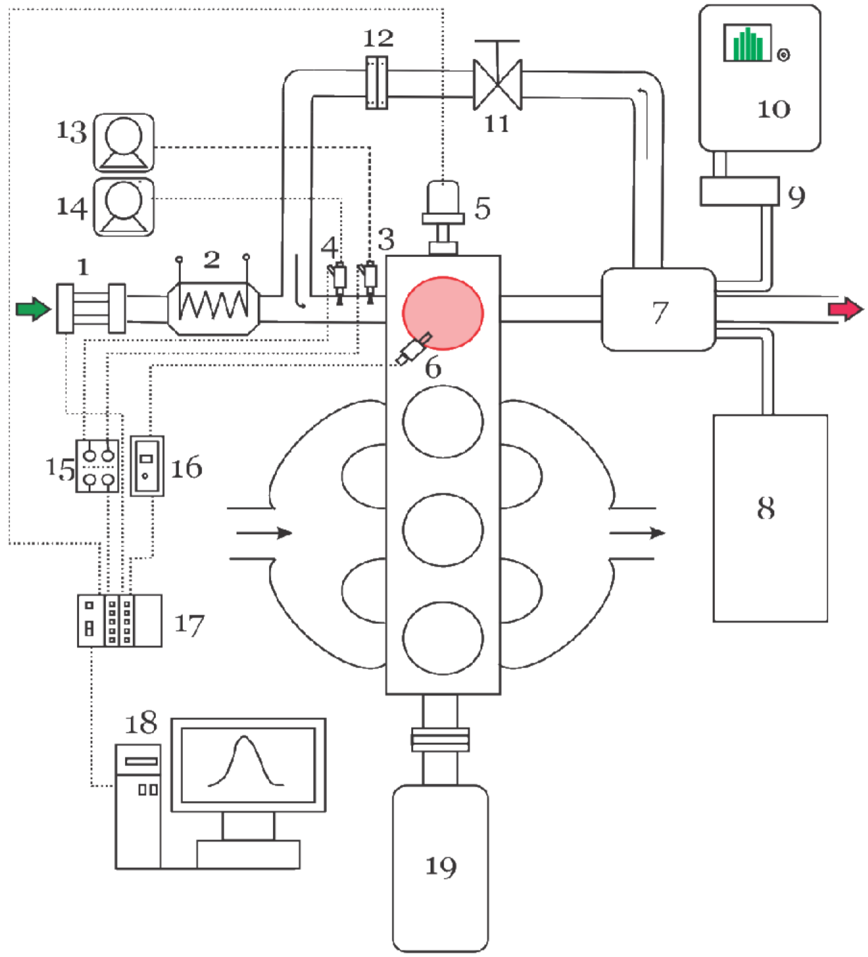
A four cylinder, four-stroke, water-cooled, naturally aspirated, direct injection diesel engine is modified for undertaking proposed investigations. Test engine specifications are given in Table 1. The engine is coupled with an eddy current dynamometer. One of the four cylinders of the engine is modified to operate in HCCI combustion mode, while the other three cylinders operated like an ordinary diesel engine, thus motoring the first cylinder for achieving HCCI combustion. The intake and exhaust manifolds of HCCI cylinder are separated from the remaining three cylinders. Schematic of the experimental setup is shown in Figure 1. The fuel is premixed with air using port fuel injector installed in the intake manifold. Electronic fuel injector has 4 nozzle holes and the fuel is injected in the manifold at 3 bar fuel injection pressure. The quantity of fuel injected in every cycle and injection timings are controlled using Compact-RIO micro-controller (cRIO-9014, National Instruments) and a customized injection driver circuit. Compact-RIO (Reconfigurable Input-Output) combines an embedded real-time processor, a high-performance FPGA (field-programmable gate array), and hot-swappable I/O (input-output) modules. Each I/O module is connected directly to the FPGA, providing low-level customization of timing and I/O signal processing. The FPGA is connected to the embedded real-time processor via a high-speed PCI bus. Compact-RIO is programmed by LabVIEW FPGA and LabVIEW Real-Time module software.

Table 1 Test engine specifications.

Make/ Model	Mahindra/ Load-king
No. of cylinders	Four
Displaced volume	652 cc/ cylinder
Stroke/Bore	94/ 94 mm
Connecting Rod Length	158 mm
Compression ratio	17.5:1
Number of Valves	2/ cylinder
Exhaust Valve Open/ Close	56° BBDC/ 5° ATDC
Inlet Valve Open/ Close	10° BTDC/ 18° ABDC

The Compact-RIO receives signals from precision shaft encoder (H25D-SS-2160-ABZC, BEI, USA), air mass flow meter (HFM5, Bosch, Germany) and an in-cylinder piezoelectric pressure transducer (6013, Kistler, Switzerland). Compact RIO generates an output pulse to trigger the fuel injector after processing the acquired signals according to the user defined operating conditions. Based on the output pulse, fuel injector injects the required quantity of fuel in the intake manifold, at an appropriate time.

Air supplied to HCCI cylinder is measured by a hot-film air mass flow meter, which precisely measures the actual intake air-mass flow rate. To achieve HCCI combustion with proper combustion phasing of gasoline like fuels, the air-fuel mixture must be preheated to a required temperature before entry into the cylinder. Fresh air entering the engine is preheated using an electric air preheater upstream of the intake manifold. The intake air preheater is controlled by a close loop controller, which maintains constant intake air temperature as defined by the user. The heater controller takes feedback from a thermocouple installed in the intake manifold, immediately upstream of the fuel injector. A thermocouple in conjunction with a digital temperature indicator is used for measuring the intake and exhaust gas temperatures. Provision for EGR (exhaust gas recirculation) is made for the HCCI cylinder so that some exhaust gas can be re-circulated using EGR valve for controlling the combustion phasings.



1. Hot-film Air Mass Meter 2. Air Preheater 3. Solenoid Fuel Injector 1
4. Solenoid Fuel Injector 2 5. Precision Shaft Encoder
6. Peizoelectric Pressure Transducer 7. Exhaust Plenum
8. Raw Exhaust Emission Analyzer 9. Thermo-diluter 10. EEPS
11. EGR Valve 12. Orifice Plate 13. Fuel Tank 1 with Fuel Pump
14. Fuel Tank 2 with Fuel Pump 15. Injector Driver Circuit
16. Charge Amplifier 17. Compact RIO 18. Data Logging Computer
19. Eddy Current Dynamometer

Figure 1. Schematic of the experimental setup.

The in-cylinder pressure is measured using a piezoelectric pressure transducer, which is mounted flush with the cylinder head. To measure the crank angle degree (CAD) position, precision optical shaft encoder is coupled with the crankshaft using a flexible helical coupling. The in-cylinder pressure history data acquisition and combustion analysis is done using a LabVIEW based program developed for this study.

RESULTS AND DISCUSSION

In this section, control method and results of combustion control of HCCI engine using dual fuel method are discussed.

Control Method

One of the main challenges in HCCI engine is structuring the system to control combustion phasing for maintaining high thermal efficiency and avoiding excessive rate of pressure rise (ROPR). Higher ROPR due to high combustion rate leads to engine ringing. Controlling CA_{50} in the expansion stroke leads to high thermal efficiency and also helps in avoidance of an excessive rate of pressure rise. It is therefore required to control the engine operating conditions so that the combustion phasing occurs at a certain desired CAD. The combustion phasing can be constant even if the combustion duration varies. Chemical kinetics in the combustion chamber evolves over time but the combustion phasing is expressed in CAD, which is not dependent on engine speed. The choice of crank angle position where the combustion should occur mostly depends on the load. When the load increases, the combustion phasing is required to be shifted to a later CAD because early combustion phasing close to TDC leads to a higher rate of pressure rise.

In this study, dual fuel injection method is used to control the combustion phasing and IMEP of engines. Figure 2 shows the concept of HCCI combustion control in this investigation. IMEP is strongly dependent on input energy per cycle therefore it is controlled by amount of fuel energy supplied. The combustion phasing controller uses the ratio between the two fuels, methanol and n-heptane, as the process input. CA_{50} is used as an index of combustion phasing and used as a process output, which is controlled using fuel ratio. In addition, maximum rate of pressure rise and misfire conditions are avoided. Currently to avoid these two limits, fuel ratio and fuel energy limits are provided in the controller.

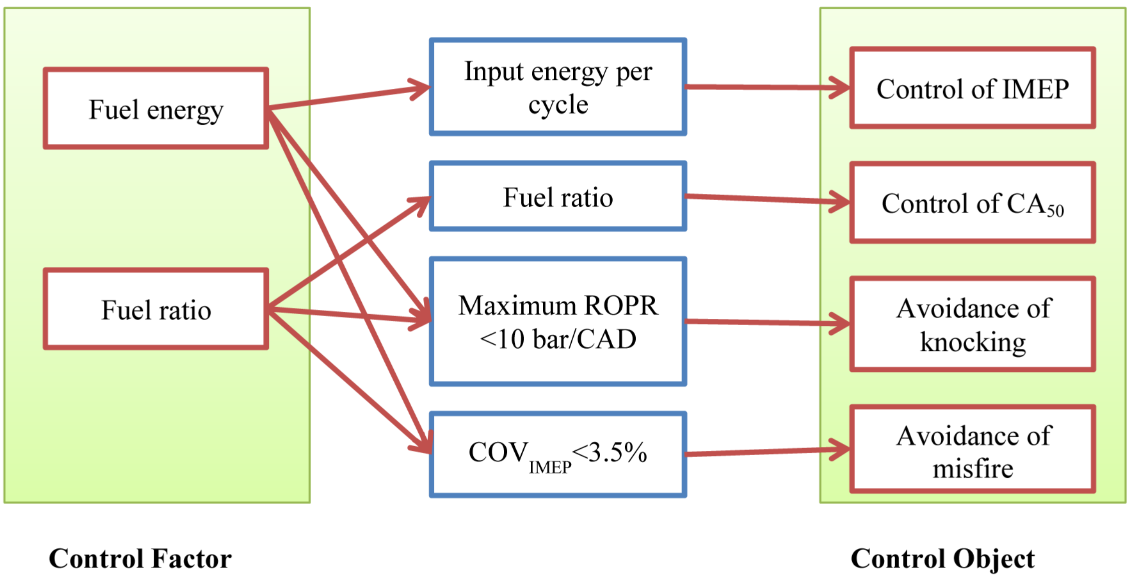


Figure 2. HCCI control concept used in present study.

A schematic of the control structure is given in Figure 3. Speed and intake air temperature controller (shaded region) work independently to maintain constant engine speed and intake air temperature. Close loop control experiments are conducted in this study at fixed engine speed of 1200 rpm and constant intake air temperature. Fixed engine speed is maintained by using engine dynamometer controller. Intake air temperature is maintained by PI controller of the electrical preheater.

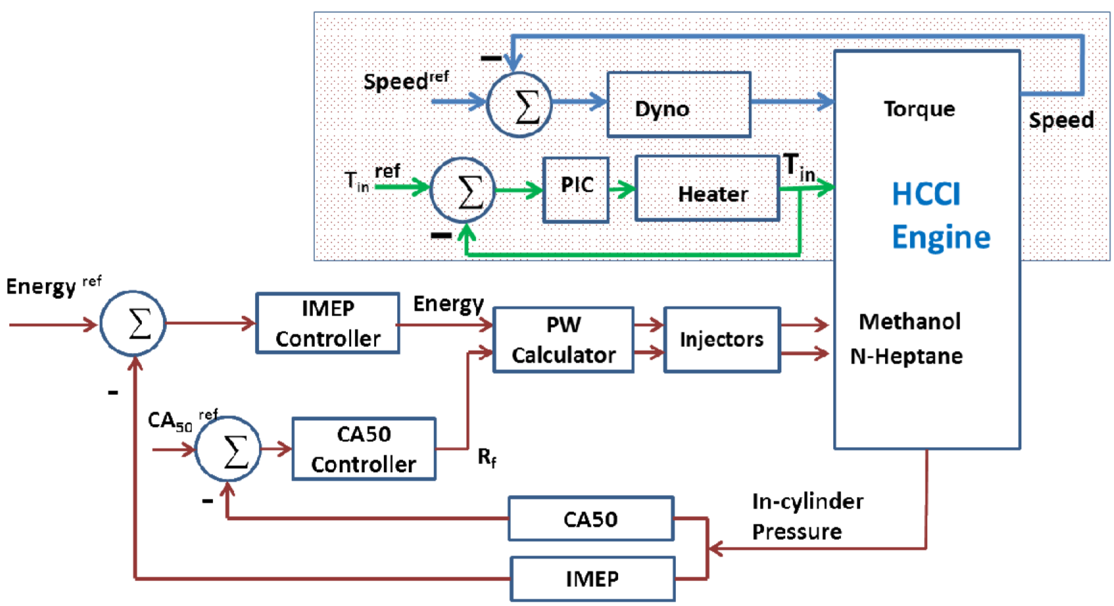


Figure 3. Schematic of the control structure.

The combustion phasing and IMEP controller take input of CA_{50} and IMEP, which are calculated from in-cylinder pressure data. Based on the reference value of CA_{50}, combustion phasing controller decides the fuel ratio (R_f), which is given to the pulse width (PW) calculator. PW calculator calculates the PW of fuel injection for both fuel injectors based on the given fuel ratio and energy to be injected to a given combustion cycle. Based on the PW, injection driver injects the required quantity of methanol and n-heptane in the intake manifold. In this study, experiments are conducted to test both the controllers separately. During testing of CA_{50} controller, the reference combustion phasing is given and energy is kept constant. The CA_{50} controller maintains the combustion phasing provided by changing the fuel ratio. Similarly IMEP controller maintains the reference IMEP by changing the energy supplied to the engine cycle.

The calculation of volume of both fuels depending on the energy injected per cycle and the fuel ratio of methanol and n-heptane is shown below.

Total energy of fuel injected and fuel ratio is related to the volume of fuel injection and calorific value of fuels with the following equations:

$$W = V_1 Y_1 + V_2 Y_2 \tag{1}$$

$$R_f = \frac{V_1}{V_1 + V_2} \tag{2}$$

Where, W = Total Energy injected per cycle (J)
R_f = Fuel ratio
Y_1= Calorific value n-heptane (J/ ml)
Y_2= Calorific value methanol (J/ ml)
V_1= Volume per cycle of n-heptane (ml)
V_2= Volume per cycle of methanol (ml)

The volume of both fuels can be calculated in terms of energy and fuel ratio given as

$$V_1 = \frac{W \times R_f}{\left[Y_2 (1 - R_f) + Y_1 R_f \right]} \tag{3}$$

$$V_2 = \frac{W \times (1 - R_f)}{\left[Y_2 (1 - R_f) + Y_1 R_f \right]} \tag{4}$$

Pulse width of electrical pulse for both injectors is calculated using injector calibration chart. IMEP controller and CA_{50} controller are based on PID controller. The PID controller compares the set point (SP) to the process variable (PV) to obtain the error (e) by e(t) = SP-PV. Then the PID controller calculates the controller action, u(t), where K_p is controller gain. The control law is stated in equation below.

$$u(t) = K_p \left\{ e(t) + \frac{1}{T_i} \int_0^t e(\tau) d\tau + T_d \frac{de(t)}{dt} \right\} \tag{5}$$

Where, u(t): control signal
e(t): Error
K_p: Proportional gain
T_i: Integral time
T_d: Derivative time

This control law is implemented using LabVIEW software in injection driver program to actuate the fuel injectors for methanol and n-heptane. Depending on the output of controller, different quantities of fuels are injected in the intake manifold.

Combustion Phasing Control

HCCI combustion phasing is not only dependent on the temperature and pressure conditions of the compression stroke, but also on the combustion chemistry of the fuels [38]. Different fuels will auto-ignite differently therefore taking two fuels of different auto-ignition quality and blending them in real time, the combustion phasing can be changed [16]. To find range of combustion phasing by varying the fuel ratio, engine is operated in the steady state with constant fuel energy input and varying the fuel ratio till the engine starts ringing (very high ROPR). The changes in HCCI combustion phasing (CA_{50}) is associated with changes in fuel ratio and corresponding ROPR values are shown in Figure 4.

It can be observed from the figure that combustion phasing advances as the fuel ratio increases due to higher amount of n-heptane fueling. The n-heptane has lower auto-ignition temperature therefore on increasing the proportion of n-heptane, combustion phasing advances. Figure 4 also shows the corresponding maximum rate of pressure rise ($ROPR_{max}$) with fuel ratio of n-heptane and methanol. It is also observed that maximum rate of pressure rise increases as fuel-ratio increases because combustion phasing advances. Higher amount of total energy injected leads to higher $ROPR_{max}$ at same fuel ratio due to higher amount of n-heptane. It is found that for the fuel ratio change from 0 to 0.5, variation of 10 CAD in combustion phasing is obtained for both fueling in steady state operating conditions (Figure 4). This is an acceptable range for controlling the combustion phasing by varying fuel ratio in dual fuel mode.

Combustion phasing is controlled by CA_{50} controller by varying the fuel ratio using PID control. Gains of the controller are set manually by observing the behavior of the control signal. The step response of controller is shown in the figure 5. It is found PID controller was able to accomplish close tracking of the reference value by changing R_f value automatically.

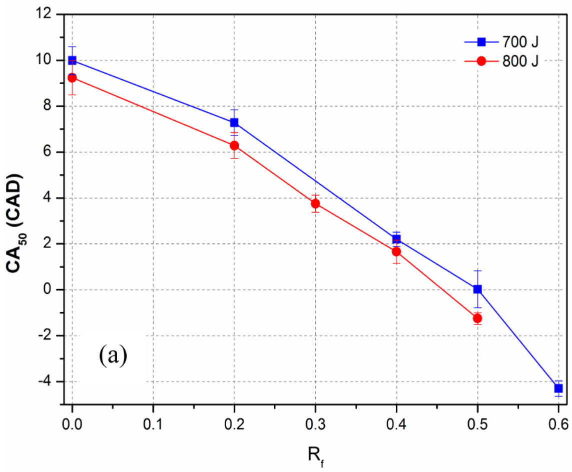


Figure 4. Variation of (a) CA_{50} and (b) $ROPR_{max}$ with fuel ratio of n-heptane and methanol at 1200 rpm.

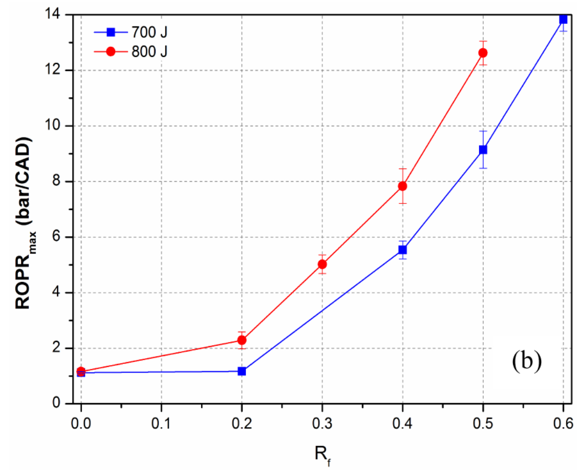


Figure 4. (cont.) Variation of (a) CA_{50} and (b) $ROPR_{max}$ with fuel ratio of n-heptane and methanol at 1200 rpm.

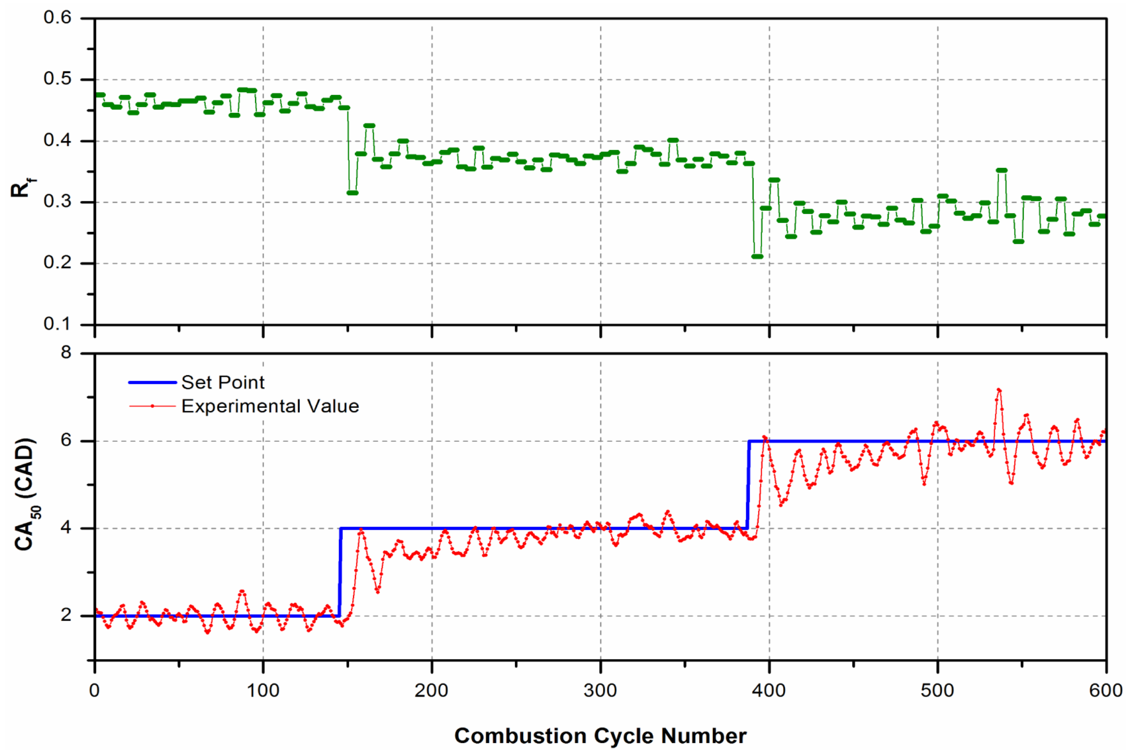


Figure 5. Step response of controller (closer view) for 700 J injected fuel energy.

Figures 6 and 7 show the PID control of combustion phasing using dual fuel ratio for 700J injected energy for two different gain settings of controller. It can be observed from figure 6 that controller was able to closely track CA_{50}. The oscillations slightly increase for reference value of 0 CAD. The reason for oscillations for reference setting at TDC for the first case is due to high rate of pressure rise in the combustion chamber. Due to higher rate of pressure rise, there are oscillations in actual pressure signal, which also leads to cycle-to-cycle variations. To decrease the oscillations at 0 CAD reference value, integral time of controller is increased (Figure 7), which leads to more deviation from the reference value. The residuals of combustion phasing (difference of reference value and process value) for both setting are shown in Figure 8. It can be observed that controller with gain setting of integral time 0.08 min is able to closely track (residual mean = 0.061 CAD) combustion phasing and standard deviation of residuals is 0.534 CAD.

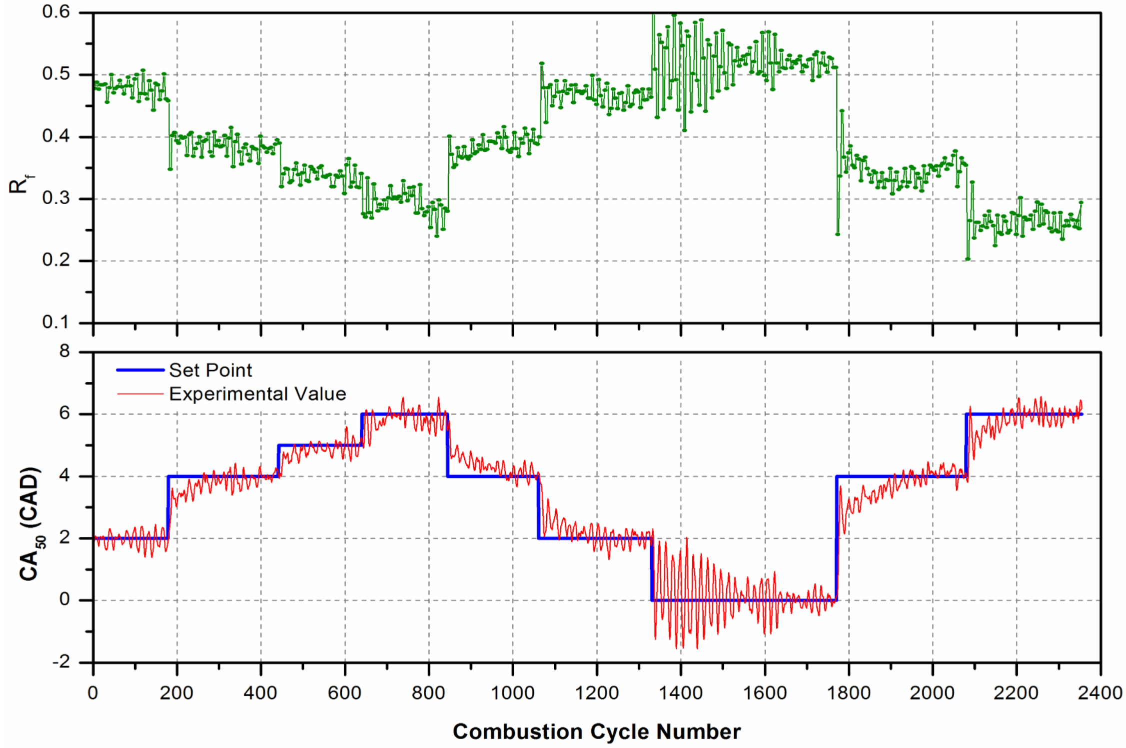


Figure 6. PID control of combustion phasing at 700 J energy injected per cycle (K_p=17; T_i=0.08).

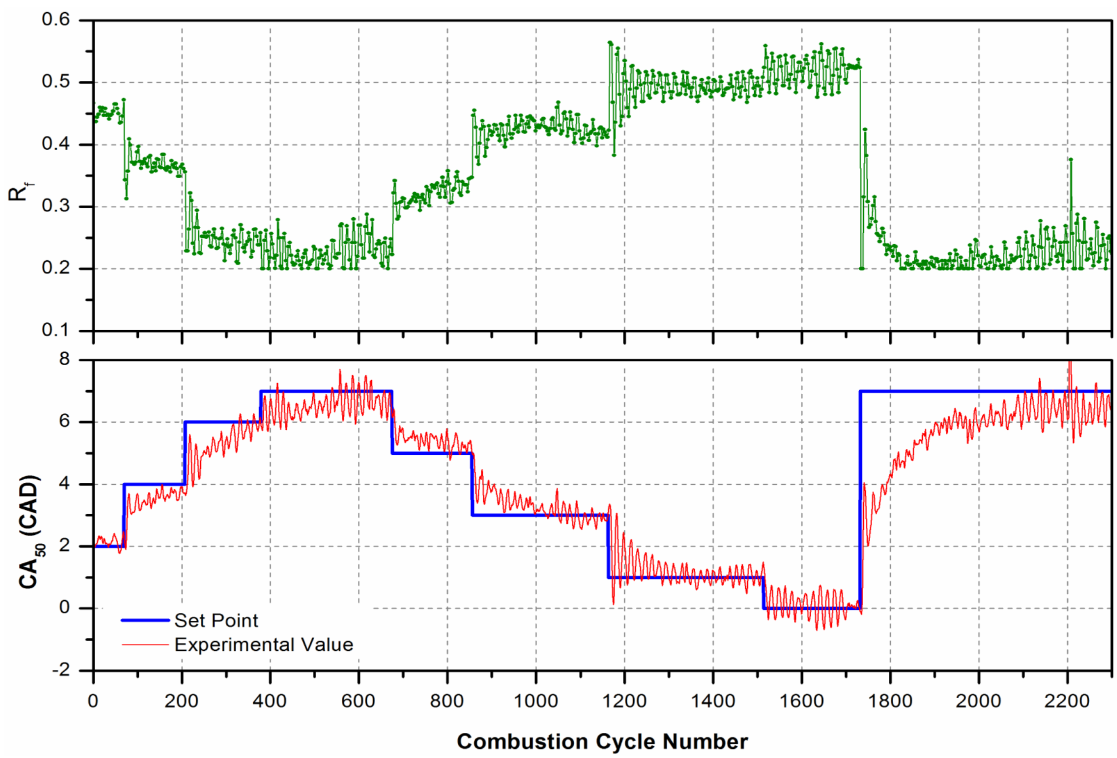


Figure 7. PID control of combustion phasing at 700 J energy injected per cycle (K_p=17; T_i=0.11).

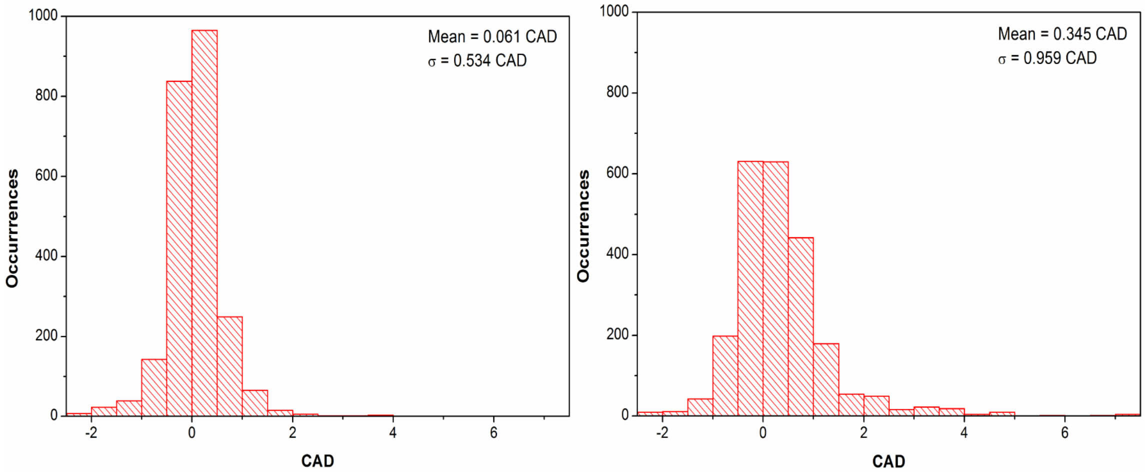


Figure 8. Residuals of combustion phasing for PID control shown in Figure 6 (left) and 7 (right).

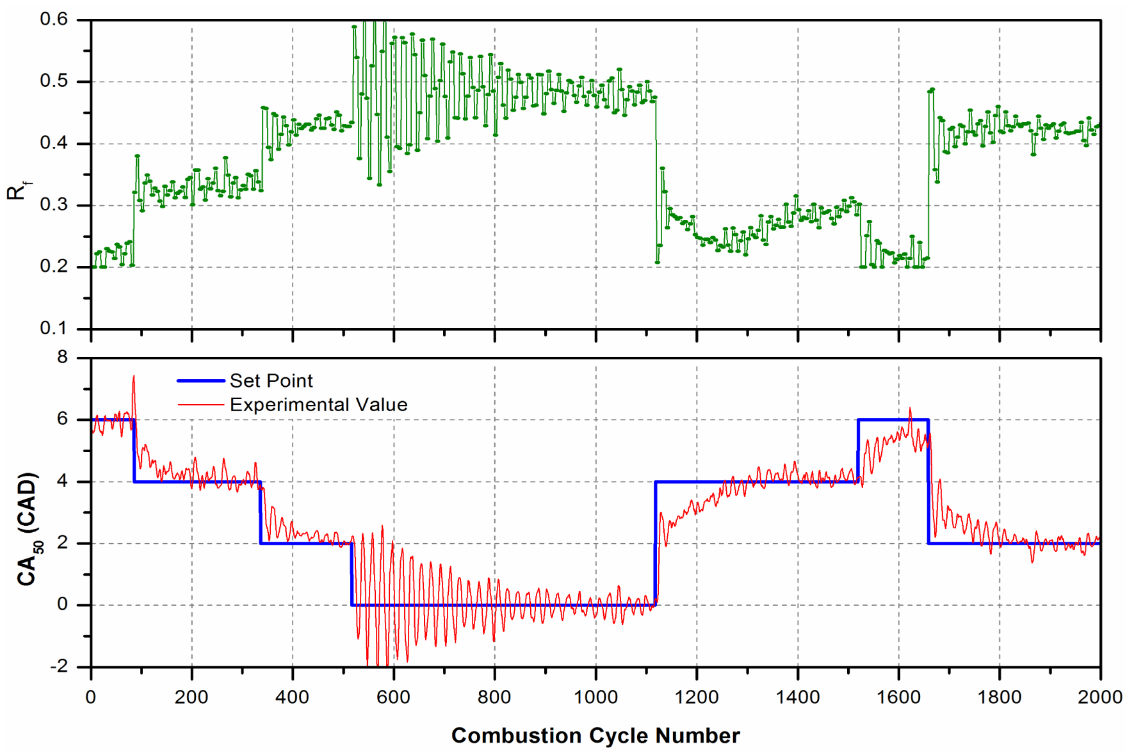


Figure 9. PID control of combustion phasing at 800 J energy injected per cycle (K_p= 17; T_i = 0.08).

On increasing integral time of controller, residual mean increases, which indicate deteriorating controller performance. These results show that there is a dynamic relationship between fuel ratio and the combustion phasing. The close-loop control introduces a non-causal coupling between input and output due to the feedback of the output through the controller.

Figures 9-10 show the PID control of combustion phasing with 800 J and 600 J injected energy in the manifold with the same best gain setting determined from Figure 6. It can be observed that controller was able to track the reference value of combustion phasing for both energy values. In figure 9, the oscillations increase for reference setting at TDC due to higher energy and higher amount of n-heptane fueling (higher R_f), which increases rate of pressure rise. Controller adjusts the R_f values and after 100 cycles, the oscillations reduce. Thus it can be concluded that controller requires different gain setting for different ranges of combustion phasing and fuel energy injected per cycle. The response of the system to changes in the set point is rather slow (up to 150 cycles). This response is slow for practical HCCI engine. Therefore advanced controller such as adaptive controller and model based predictive control may be more effective in controlling combustion phasing in wide range of operating conditions. PID control of combustion phasing is successfully demonstrated however application of advanced controller is beyond the scope of this research.

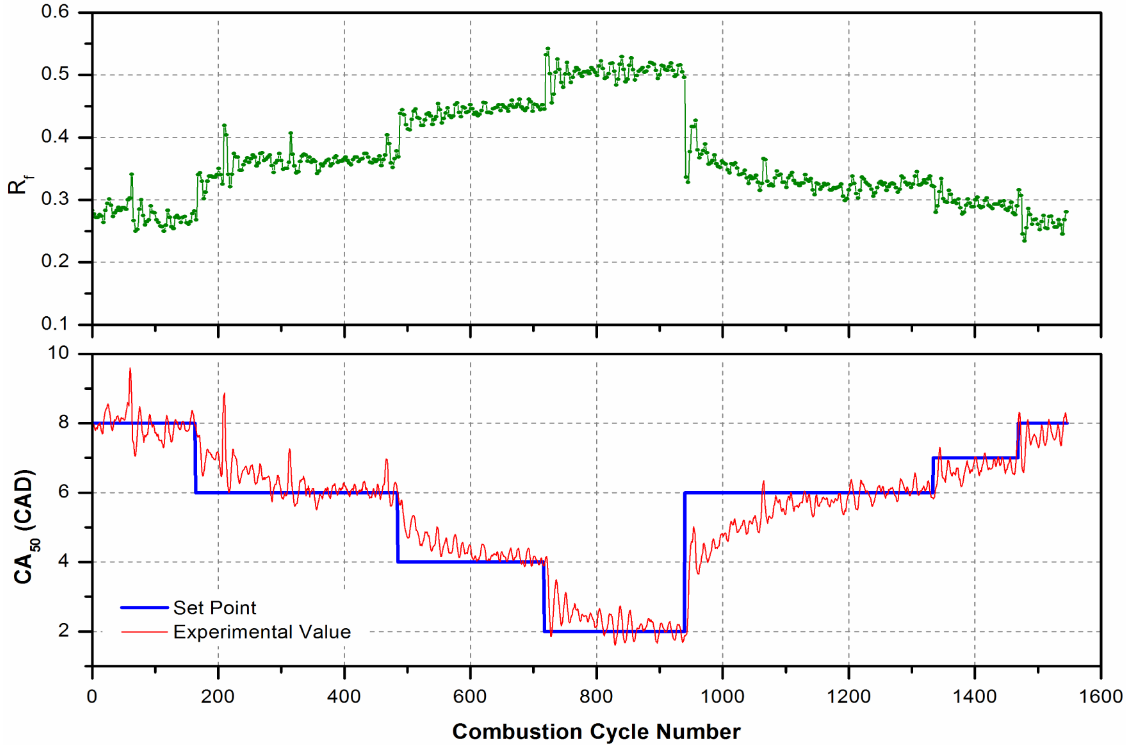


Figure 10. PID control of combustion phasing at 600 J energy injected per cycle (K_p=17; T_i=0.08).

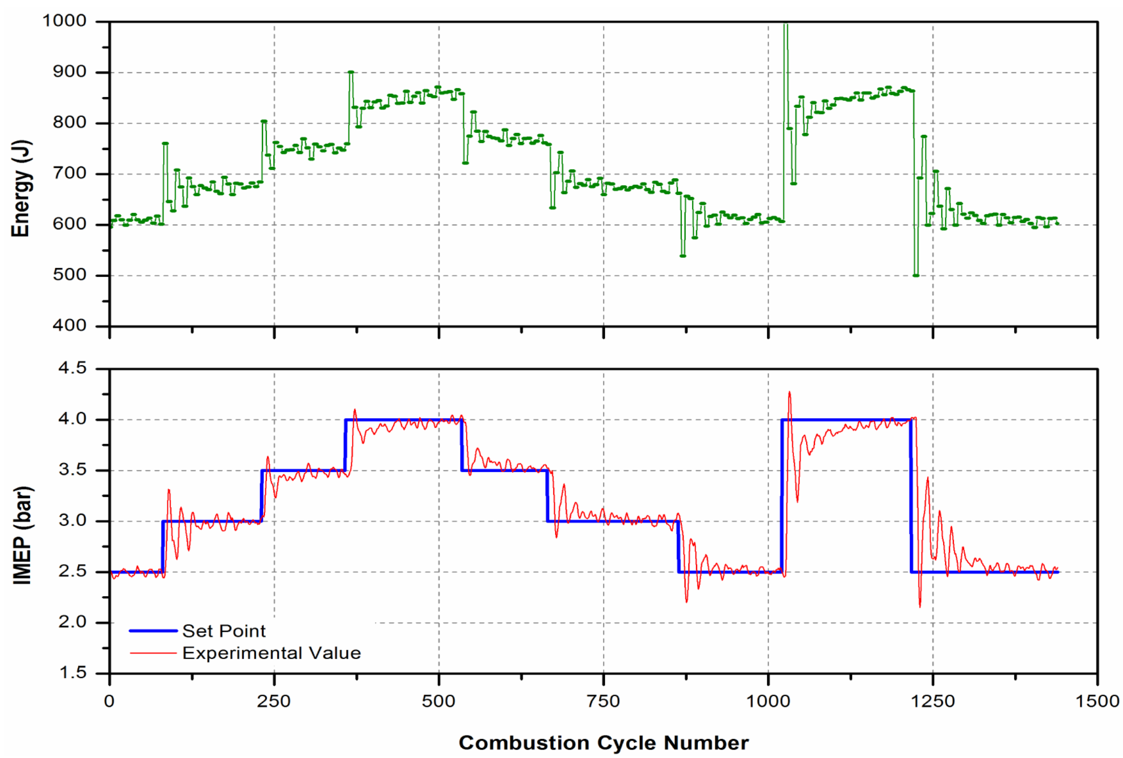


Figure 11. PID control of IMEP at R_f = 0.3 at gain setting (K_p=50 T_i= 0.05).

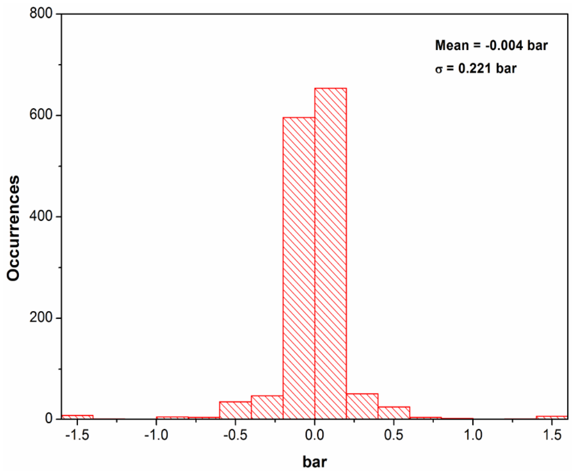


Figure 12. Residual of IMEP for PID control of Figure 11.

IMEP Control

IMEP is a measure of engine's capacity to perform work and meet engine load. IMEP is strongly correlated with injected fuel energy per cycle [39-40]. Therefore fuel energy per cycle is chosen as a control factor. Figure 11 shows the PID control of IMEP at fixed fuel ratio (0.3) by changing the energy injected per cycle using proportional gain of 50 and integral time 0.05 minutes. This gain setting is found to be suitable for close tracking of reference value of IMEP. It is very clear from residuals variation (Figure 12) that controller was able to accomplish close tracking of reference IMEP. Residual mean is 0.004 bar with standard deviation 0.22 bar, which indicates that process value of IMEP is very close to the reference value.

Figures 13-14 show the PID control of IMEP using variation of fuel energy for different fuel ratios (R_f = 0.4 and 0.5) with same optimized gain setting of the controller. It can be observed from these figures that controller is able to accomplish close tracking of IMEP for these fuel ratios also. Thus PID control of combustion phasing and IMEP using dual fuel injection is achieved and successfully demonstrated for HCCI engines.

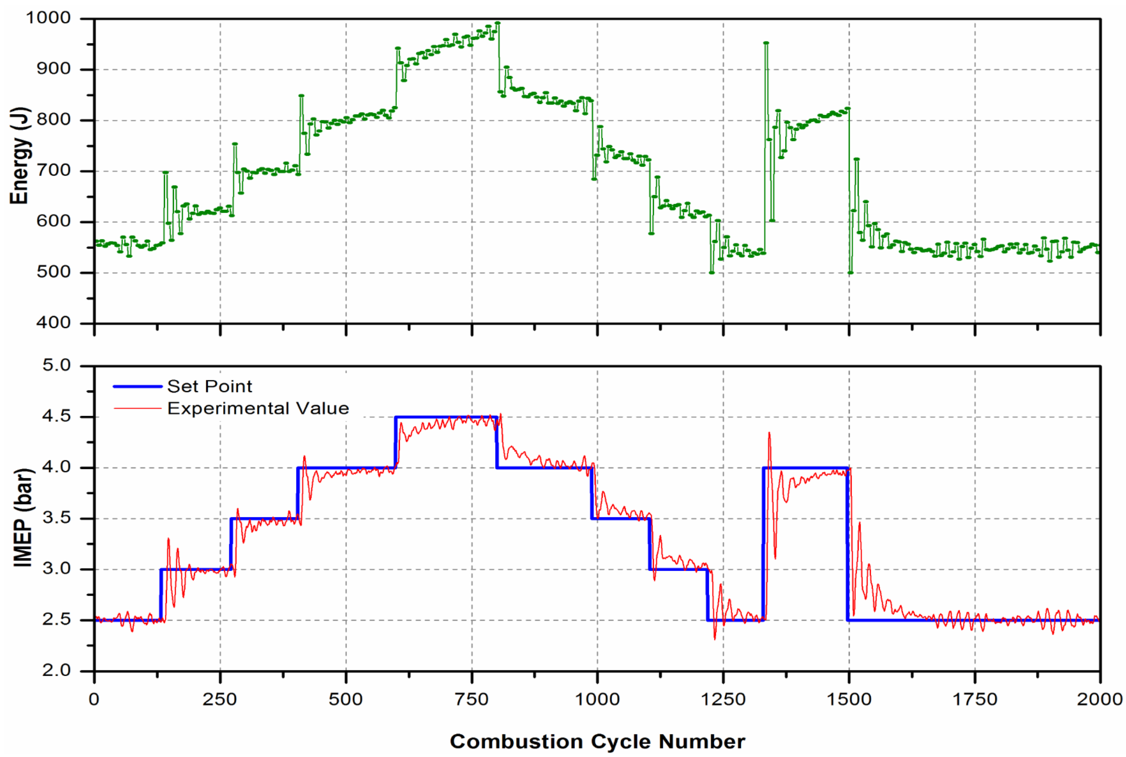


Figure 13. PID control of IMEP at R_f = 0.4 at gain setting (K_p= 50 T_i = 0.05).

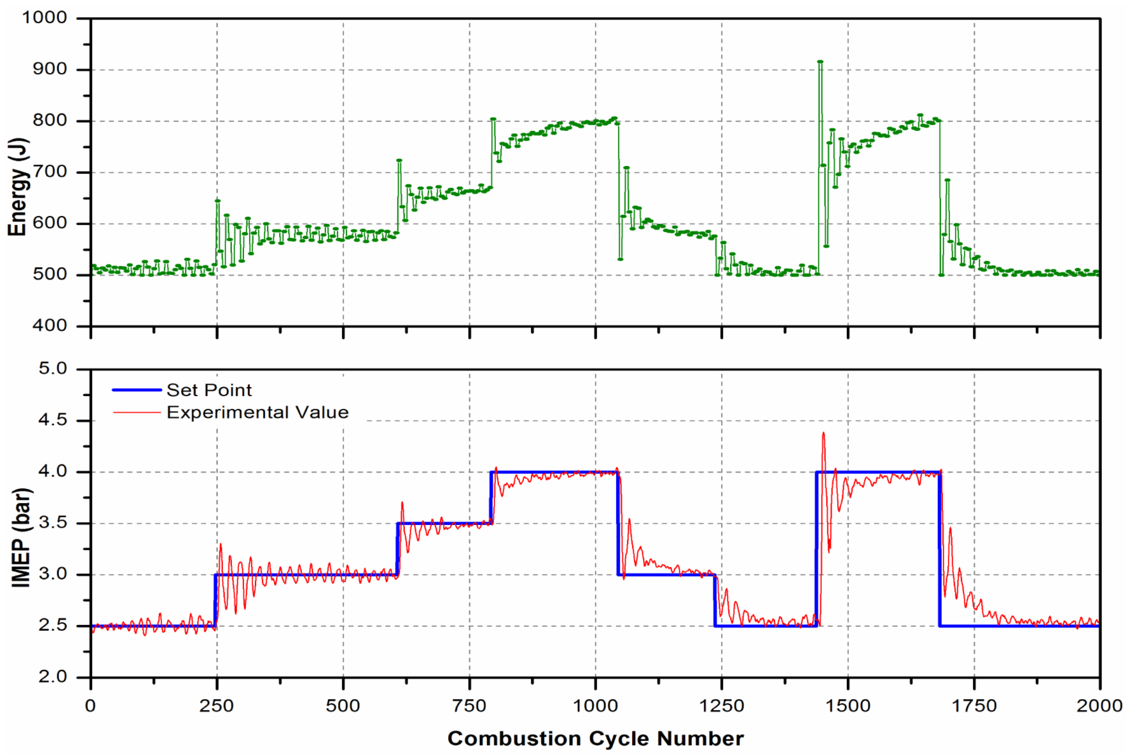


Figure 14. PID control of IMEP at R_f=0.5 at gain setting (K_p=50 T_i= 0.05).

CONCLUSIONS

Experimental investigations of close loop control of HCCI engine is conducted at an engine speed of 1200 rpm. Close loop control of combustion phasing and IMEP using dual fuel (methanol and n-heptane) injection method is designed and tested by PID control method. During close loop control test, it is found that the controller requires different gain setting for different range of combustion phasing and fuel energy injected per cycle. Therefore advanced controllers such as adaptive controller and model based predictive control may be more effective in controlling combustion phasing in wide range of operating conditions. PID control of combustion phasing and IMEP using dual fuel is achieved and is successfully demonstrated in HCCI engines.

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DEFINITIONS/ABBREVIATIONS

CAD - Crank angle degree
CI - Compression Ignition
FTM - Fast thermal management
FPGA - field-programmable gate array
HCCI - Homogeneous charge compression ignition
IMEP - Indicated mean effective pressure
LQG - Linear-quadratic-Gaussian
MIMO - Multiple input multiple output
PID - Proportional integral differential
PI - Proportional integral
PW - Pulse width
ROPR - Rate of pressure rise
SI - Spark ignition
TDC - Top dead center
VVA - Variable valve actuation
VCR - Variable compression ratio
CA_{50} - Crank angle position for 50% heat release
R_f - Fuel ratio
K_p - Proportional gain
T_i - Integral time

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