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# Control of Homogeneous Charge Compression Ignition (HCCI) Engine Through Stochastic Variables

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**Abstract:** A stable and efficient operation of homogeneous charge compression ignition (HCCI) engine requires the combustion phasing to be tightly controlled at a proper set-point which can be done by controlling mixture reactivity. Combustion phasing is important in minimizing the hazardous emission, ensuring complete combustion and optimizing the brake specific fuel consumption. One of methods to control mixture reactivity is by actuating the injected air-fuel ratio (AFR). Simulation has been done using MATLAB® tool on the lean mixture of air and fuel (equivalence ratio = 0.4). Firstly, the engine is run under 1000 revolution per minute for 20 cycles. The equivalence ratio in the cylinder fluctuates during six first cycles. This phenomenon happens as the equivalence ratio at the intake manifold is kept constant. The remaining residual gases of previous cycle and the performance of mixing process have left significant effect on the current cycle. At intake temperature of 373K, equivalence ratio in the cylinder is increased up to 39% and is decreased down to 27%. At intake temperature of 473K, maximum increased and decreased in cylinder equivalence ratio is 7.3% and 37.8%. However, at $T_{in}$ 573K, there is no increment in cylinder equivalence ratio instead of decrement by maximum of 41%. Thus, a change in intake temperature has significantly varied the equivalence ratio in the cylinder and consequently has affected the pressure inside the cylinder.

**Key words:** HCCI engine · Air-fuel ratio · Intake temperature · Mixture reactivity · Equivalence ratio

## INTRODUCTION

Nowadays according to [1], lot of research works done to improve thermal efficiency and to control and reduce emission trade-off on such engines. This is to cope with future emission legislation requirements and reduce fuel consumption while maintaining high engine efficiency. One of most important advancements is to implement Homogeneous Charge Compression Ignition (HCCI) combustion in engine. HCCI is governed by the fact that the fuel and air are mixed before combustion starts and mixture auto-ignites as a result of the temperature increase at the end of compression stroke [2]. The auto-ignition is determined solely by the chemical kinetics of the mixture in the cylinder which consists of fresh mixture of air and fuel and trapped exhaust gas from previous cycle.

In HCCI engine, the fuel and air should be mixed homogeneously before the combustion starts and the mixture would be ignited spontaneously due to temperature increasing at the end of the compression stroke [3]. There is no direct combustion trigger in HCCI engine. This auto-ignition behaviour makes the combustion being very sensitive to its initial conditions like inlet pressure, temperature, fuel composition and homogeneity of the mixture as mentioned in [4-6].

One of the possible ways to control the combustion phasing is to control the mixture reactivity which is actuated by the equivalence ratio. Equivalence ratio control is done by the fuelling system of the engine. The fuelling rate, mf is the fuel delivered every cycle by fuel injector. This fuel delivered will vary due to the injector inaccuracies. This inaccuracy has produce uncertainty in fuelling rate which leads to the modelling of stochastic engine model.

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Another actuator involved is the inlet temperature which is usually controlled by a plenum heater to keep the temperature constant at 573K. However, this temperature changes due to changing speed and ambient temperature. Therefore, uncertainty is needed to be included in the equations involving inlet temperature. These instabilities are resulted from random movement of gas mixture in the combustion cylinder which is affected by the engine initial conditions. Random movement of gas mixture has caused the in some parameters inside the cylinder produced along with uncertainty. In-cylinder flow conditions in internal combustion engines are inherently stochastic. A reliable model of HCCI engine performance must include the description of the processes that captures cycle-to-cycle variations which can be regarded as uncertainty or random noise.

**Theory of Hcci Combustion:** Mean value models (MVM) are typical in control oriented models for engine. For HCCI engine, the MVM HCCI is modelled by separating the combustion process into three phases: In cylinder mixing process and compression stroke (from intake valve closing (IVC) until start of combustion (SOC), from start of combustion (SOC) to Crank Angle Degree 50 (CAD50), expansion stroke (from expansion of gases until exhaust valve opening (EVO). The result is an algebraic relationship between the gas properties at IVC and the in cylinder gas properties at exhaust valve closing (EVC) as stated in this section was derived by [7].

**In-cylinder Mixing Process:** At intake manifold, AFR is affected by percentage of EGR from exhaust manifold. The AFR from intake manifold, entering the combustion chamber is given by equation [7]:

$$AFR_{cyl}(k) = (1 - x_{cyl})(1 + AFR_{man}(k))$$

$$\frac{m_{cyl}(k)}{m_{cyl}(k)-m_{egr}(k)} - 1 \quad (1)$$

$AFR_{cyl}$ is considering the additional air contained in the residual gas. $AFR_{man}$ is the air to fuel ratio of the intake charge. The k indicates variables as function of the current cycle. The $x_{cyl}$ is the residual gas fraction in the cylinder which is represented by:

$$m_{cyl} = m_{EVC}(k-1)\frac{m_{egr}(k)}{m_{cyl}(k)} \quad (2)$$

The total mass of gas mixing in the cylinder $m_{cyl}$ found using the ideal gas law at IVC is the sum of mass of fresh charge, $m_f$ and mass of trapped residual gas $m_{egr}$,

$$m_{cyl}(k) = m_f(k) + m_{egr}(k) \quad (3)$$

R denotes the specific gas constant defined by the universal gas constant divided by the molecular weight,

$$R = \frac{R_u}{MW} \quad (4)$$

The mass of residual gas trapped in the cylinder is determined using the ideal gas law at (exhaust valve close) EVC,

$$m_{egr}(k) = \frac{P_{EVC}(k-1)V_{EVC}}{RT_{EVC}(k-1)} \quad (5)$$

The initial $P_{IVC}$ and $T_{IVC}$ are set as [7],

$$P_{IVC}(k) = p_{man} \quad (6)$$

$$P_{IVC}(k)\frac{m_{cyl}(k) - m_{egr}(k)}{m_{cyl}(k)} T_{man}(k) + \frac{m_{egr}(k)}{m_{cyl}(k)} T_{EVC}(k-1) \quad (7)$$

**Compression Stroke:** The in-cylinder gases undergo a polytropic compression from Input Valve Close (IVC) until Start of Combustion (SOC). During this phase, $T_{cyl}$ and $p_{cyl}$ are calculated as [7],

$$T_{cyl}(k,\theta) = T_{IVC}(k)\left(\frac{V(\theta_{IVC})}{V\theta}\right)^{n_n - 1} \quad (8)$$

$$p_{cyl}(k,\theta) = p_{IVC}(k)\left(\frac{V(\theta_{IVC})}{V\theta}\right)^{n_c} \quad (9)$$

where as $n_c$ is the polytropic exponent. The cylinder volume is found from the slider crank equation,

$$V(\theta) = V_c + \frac{\pi B^2}{4}\left(1 + a - a\cos\theta - \sqrt{1^2 + (a\sin\theta)^2}\right) \quad (10)$$

**SOC until CA50:** The start of combustion occurs when the in-cylinder temperature, $T_{cyl}$ and pressure $p_{cyl}$ are at $\theta_{SOC}$ which then are called as $T_{SOC}$ and $p_{SOC}$ and calculated using eq. (8) and (9) with (10) evaluated at $\theta_{SOC}$. The in-cylinder pressure and temperature adhere to eq. (8) and (9) until 50% of the crank angle (CA50) [7],

$$\theta_{SO50} = \theta_{SOC} + \Delta\theta \quad (11)$$

at which,

$$T_{CA50} = T_{OC} + \Delta\theta \quad (12)$$

whereas $T_{oc}$ is the in-cylinder gas temperature before combustion found from Equation (14) evaluated at $\theta_{SOC}$. The temperature rise due to combustion, $\Delta T$ is calculated from [7],

$$\Delta T(k) = \frac{\square_c LHV}{c_v} \frac{1\text{-}x_{cyl}(k)}{1+AFR_{cyl}(k)} \quad (13)$$

where the combustion efficiency $\square_c$ accounts for incomplete combustion and is assumed to be constant, LHV is the lower heating value of the fuel and $c_v$ is the specific heat at constant volume of the in-cylinder gas. The pressure at $\theta_{CA50}$ can be found using the ideal gas law.

$$p_{CA50} = p_{oc}\, T_{CA50}/T_{oc} \quad (14)$$

where $p_{oc}$ is the pressure before the combustion and calculated using eq. (9) at $\theta_{CA50}$.

**Expansion Stroke:** The gas undergoes a polytropic expansion until the exhaust valve opens (EVO). The in-cylinder pressure and temperature at EVO are calculated as [7, 8],

$$T_{EVO}(k) = T_{CA50}(k) \left( \frac{V(\theta_{CA50})}{V(\theta_{EVO})} \right)^{n_e\text{-}1} \quad (15)$$

$$p_{EVO}(k) = p_{CA50}(k) \left( \frac{V(\theta_{CA50})}{V(\theta_{EVO})} \right)^{n_e} \quad (16)$$

At exhaust valve closing (EVC), it is assumed that the gas undergoes an adiabatic expansion from the pressure at EVO down to the exhaust manifold pressure and there is a drop in temperature $\Delta T_{ex}$ due to heat loss,

$$T_{EVC}(k) = T_{EVO}(k) \left( \frac{V_{EVC}(k)}{p_{EVO}(k)} \right)^{(n_e-1)/n_e} + \Delta T_{ex} \quad (17)$$

$$p_{EVC}(k) = p_{EVO}(k) \left( \frac{V(\theta_{EVO})}{V(\theta_{EVC})} \right)^{n_e} \quad (18)$$

The residual gas fraction, $X_{EVC}(k)$ after the combustion complete assuming complete combustion is,

$$x_{EVC}(k) = \left( \frac{1+AFR_{stoich}}{1 + AFR_{cyl}(k)} \right) \left(1\text{-}\, x_{cyl}(k)\right) + x_{cyl}(k) \quad (19)$$

**Equivalence Ratio Relationship:** The equivalence ratio (Ø) is defined as the ratio of the stoichiometric air-to-fuel ratio ($AFR_{stoich}$) to the actual air-to-fuel ratio ($AFR_{actual}$) that that takes place in the combustion chamber [9]. It is regarded as $AFR_{cyl}$

$$\varnothing = \frac{AFR_{stoich}}{AFR_{cyl}} \quad (20)$$

The AFR is expressed in the ratios of corresponding masses of gases which react with the fuel composition. In internal combustion engines, the combustion usually takes place in the presence of air and not pure oxygen. Air contains oxygen ($O_2$), nitrogen ($N_2$), argon and other vapours and inert gases. Since neither nitrogen nor argon and inert gases involves in chemical reaction, it is sufficiently to assume that 21% of air is $O_2$ and 79% of air is $N_2$. The stoichiometric equation for complete combustion of hydrocarbon fuel is given as [10],

$$C_mH_nO_p + \left( m + \frac{n}{2}\text{-}\frac{p}{2} \right) O_p + 3.76 \left( m + \frac{n}{2}\text{-}\frac{p}{2} \right) N_2$$

$$\rightarrow CO_2 + \frac{n}{2} H_2O + 3.76 \left( m + \frac{n}{2}\text{-}\frac{p}{2} \right) N_2 \quad (21)$$

The equation above is for ideal combustion where all hydrocarbons are oxidized using all oxygen. In actual practice, this is not possible. More oxygen than the theoretical amount is needed to achieve complete combustion. The excess air is usually expressed as the percentage of the theoretical air. For example, if the complete combustion is achieved by 20% of excess air, the equation becomes as [9],

$$C_mH_nO_p + (1.2) \left( m + \frac{n}{2}\text{-}\frac{p}{2} \right) O_2 + (1.2)3.76$$

$$\left( m + \frac{n}{2}\text{-}\frac{p}{2} \right) N_2 \rightarrow mCO_2 + \frac{p}{2}\, H_2O + 3.76 \left( m + \frac{n}{2}\text{-}\frac{p}{2} \right) N_2 \quad (22)$$

Then, the $AFR_{soich}$ can be obtained from eq. (21) and expressed as,

$$AFR_{stoich} = \frac{m_{air}}{m_{fuel}} \tag{23}$$

$m_{air}$ is the mass of air per mole fuel, which consists of mass of $O_2$ and mass of $N_2$. $m_{fuel}$ is the mass of fuel per mole fuel.The value of $AFR_{cyl}$ is depending on the burnt mass fraction, $x_{cyl}$ which is consequently depending on the previous cycle of burnt mass fraction after EVC, $x_{EVC}$ and also the $AFR_{man}$. In order to sustain the equivalence ratio, Ø within 0.3-0.55, these two parameters must be controlled. $x_{EVC}$ is obtained during the combustion in the cylinder which is hardly controlled while $AFR_{man}$ is depending on the mass of fuel and air at the intake.

The initial residual gas mass fraction, $X_{EVC}(k-1)$ is found using a model proposed by [10],

$$X_{EVC}(k\text{-}1) = \alpha + \beta \tag{24}$$

$$\alpha = \sqrt{\frac{1}{C}}\frac{\pi\sqrt{2}}{360}\frac{r_c\text{-}1}{r_c}\frac{OF}{N}\sqrt{\frac{RT_{man}|P_{exh}\text{-}P_{man}|}{p_{exh}}}\left(\frac{p_{exh}}{p_{man}}\right)^{(k+2)/2k} \tag{25}$$

$$\beta = \frac{1}{C}\frac{r_c\text{-}1}{r_c}\varnothing\frac{V_{IVO}}{V_{dis}}\left(\frac{p_{exh}}{p_{man}}\right)^{1/k} \tag{26}$$

where as $r_c$ indicates the compression ratio, $p_{exh}$ is the exhaust pressure, R is the gas constant and k is the ratio of specific heats, $V_{IVO}$ is the cylinder volume at IVO moment and $V_{dis}$ is the displacement volume. OF is the valve overlap factor that is calculated using geometry and the timing both intake and exhaust valve. The parameter C is calculated as [10],

$$C = \left[1 + \frac{LHV}{C_v T_{man}\left(\frac{m_{cyl}}{m_f}\right)r_c^{k-1}}\right]^{1/k} \tag{27}$$

whereas LHV represents the lower heat value of the fuel and $C_v$ is the constant-volume specific heat capacity of the in-cylinder gas at the IVC moment.

The first term $\alpha$ is attributed to the portion of the residual gas caused by the back-flow during valve overlap period and the second term $\beta$ accounts for the trapped exhaust gas before Input Valve Open (IVO). The value of $X_{EVC}(k-1)$ is used as starting value in an iterative process of HCCI combustion in order to control $AFR_{man}$.

Table 1: Engine parameters

| | |
|---|---|
| Displaced Volume | 1600 $cm^3$ |
| Bore size | 120.5 mm |
| Crank radius | 70 mm |
| Stroke length | 140 mm |
| Connecting Rod | 260 mm |
| Exhaust Valve Open | -175 CAD |
| Exhaust Valve Closed | 55 CAD |
| Inlet Valve Open | -70 CAD |
| Inlet Valve Close | -175 CAD |

Table 2: Initial values for simulations of engine

| Ø | CR | $T_{in}$ [K] | $P_{in}$ [k Pa] | Speed, N (rpm) |
|---|---|---|---|---|
| 0.4 | 10 | 373, 473, 573 | 40 | 1000 |

**Methodology:** Simulation was done using MATLAB® tools for engine specifications as in Table 1 and 2. A simple structure of a dynamic full cycle simulation model has been formed that includes the effect of thermodynamic properties of residual gasses from previous cycle on the current cycle. Figure 1 shows the flowchart of overall model programmed in MATLAB computing environment.

## RESULTS AND DISCUSSION

**Model Validation:** Guo *et al*. [11] have carried out a series of experiments on HCCI combustion to study the effects of AFR to the combustion phase. Traditionally, diluting the mixture is done by reducing the injected fuel into the intake air. However, the same effect can be achieved by throttling the intake air while keeping the fuel flow rate constant. Their experiment data has been used for validation of numerical method by comparing the pressure profile for various intake pressures at a compression ratio of 10, exhaust temperature of 104 kPa and engine speed of 900 rpm. As shown in this Figure 2, fair levels of agreement are obtained for pressure traces.

The results of simulation also were compared with experimental data from Maurya and Agarwal [3]. Measurements are taken at a constant speed of 1500 rpm and intake temperature of 423 K. Good agreements between the simulated and experimental results were obtained. The maximum errors are 5.4% and 7.4% for the maximum temperature and maximum pressure in the cylinder respectively as shown in Figure 3.

**Simulation Results:** The simulation has been taken to study the effect of air-to-fuel ratio (AFR) and initial temperature ($T_{in}$) to the combustion process in the cylinder for 20 combustion cycles. At first both AFR and intake temperature are treated as deterministic variables

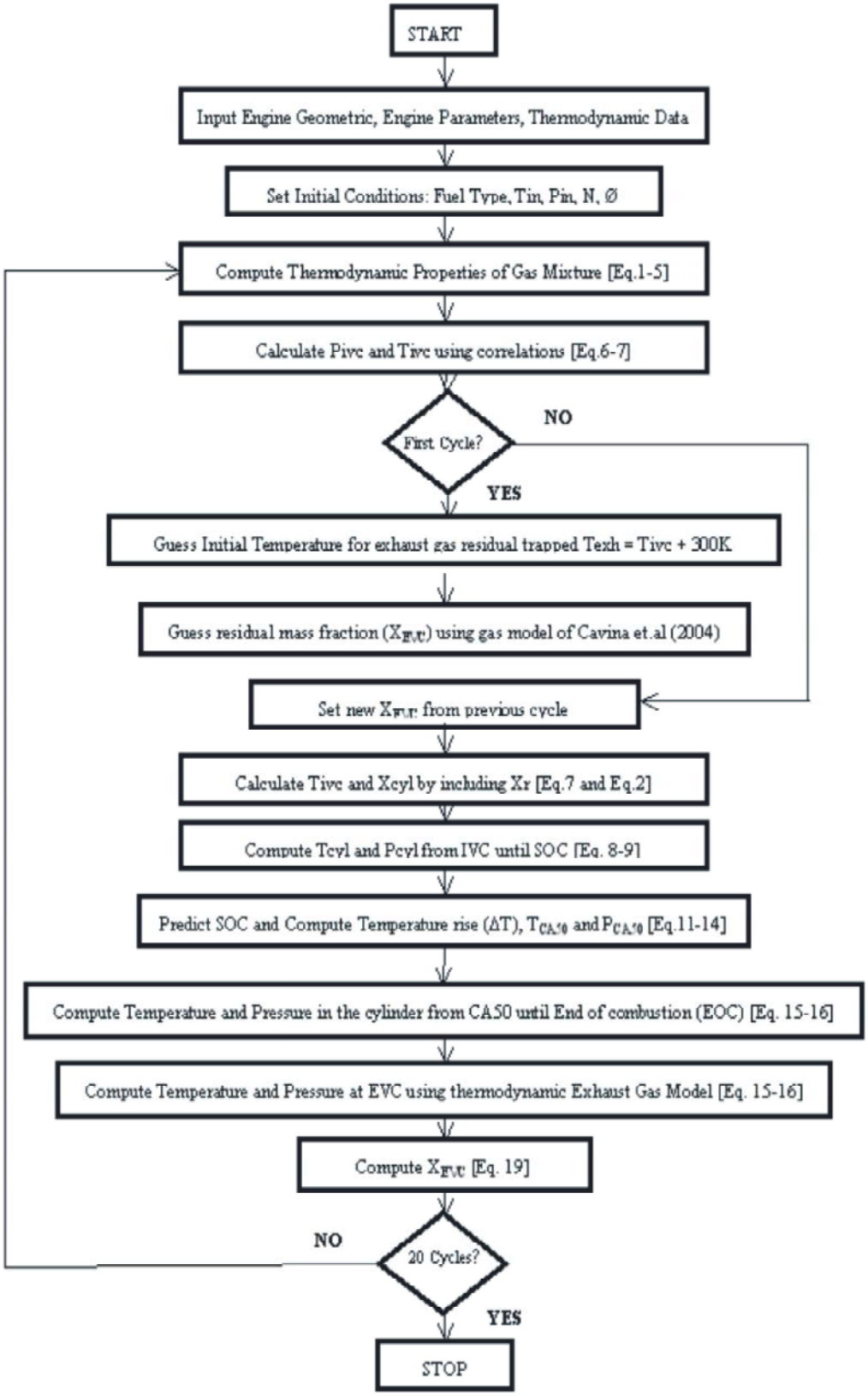


Fig. 1: Schematic Flowchart of Dynamic Full Cycle HCCI Model

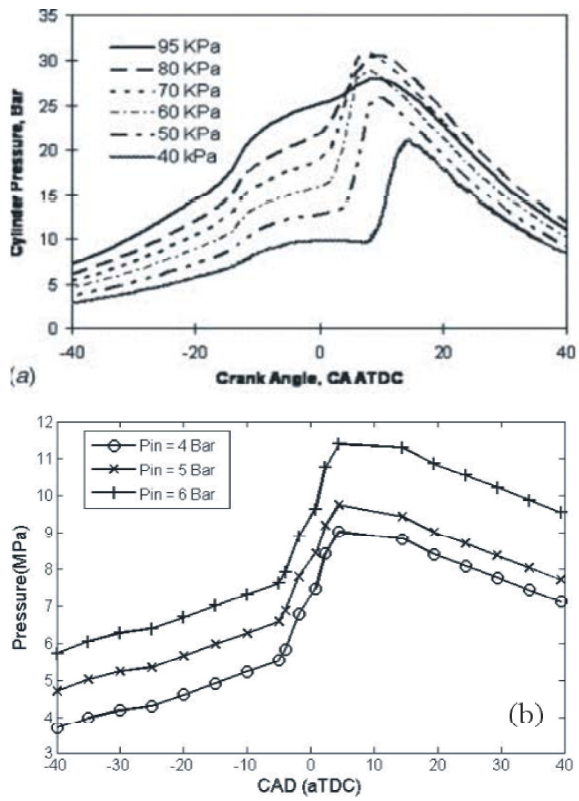


Fig. 2: Pressure traces results for (a) Experimental [11] and (b) Numerical

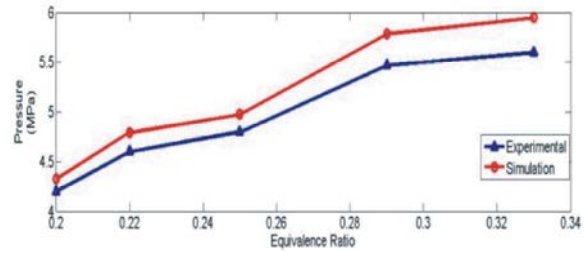


Fig. 3: Peak pressure comparison for Experimental [3] and Numerical.

and simulation is run for 20 cycles. Pressure changes of different cycles are plotted in Figure 4 shows the in cylinder pressure changes vary largely resulted from changing in-cylinder equivalence ratio.The equivalence ratio in the cylinder is fluctuating before it becomes more stable after 6 cycles. This phenomenon happens as the equivalence ratio at the intake manifold is kept constant. The remaining residual gases of previous cycle and the behaviour of mixing process have left significant effect on the current cycle. At very high speed of 3000 rpm, the combustion has stopped at cycle 6 because of very dilute AFR in the cylinder. Misfiring cannot be avoided.

Figure 4 also shows the peak pressure varying for different cycles. Peak pressure is an indication of rate of expansion after top dead centre (ATDC). Lower peak

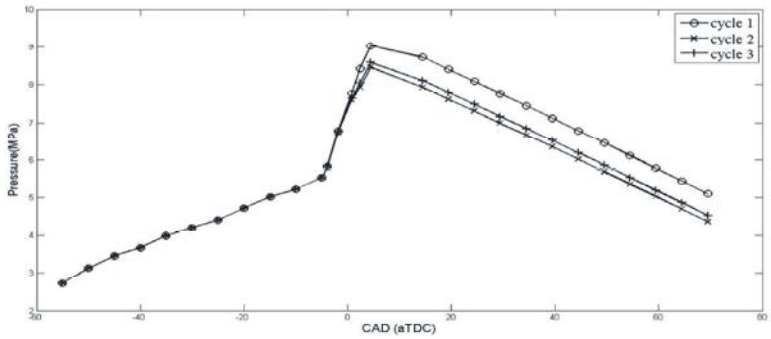


Fig. 4: In cylinder pressure changes with respect to engine cycles

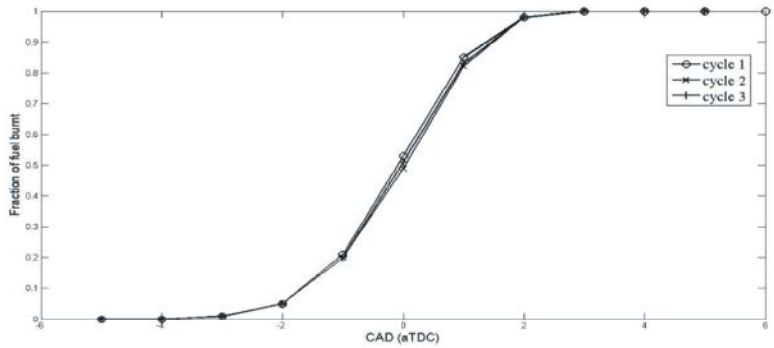


Fig. 5: Fuel Burnt Rate changes with respect to engine cycles

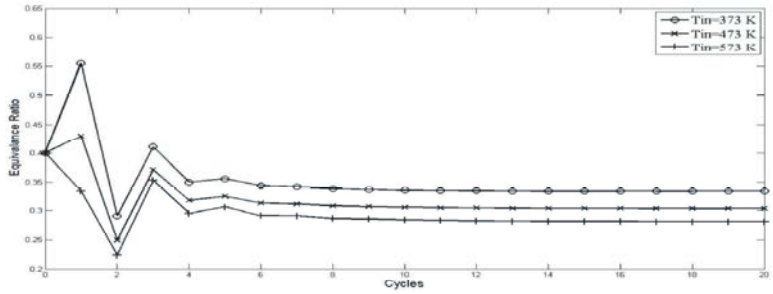


Fig. 6: In cylinder equivalence ratio changes with respect to engine cycles

pressure will result in slower expansion after TDC, slower combustion and less efficient combustion. Figure 5 presented the fraction of fuel burnt for first three cycles. The fraction fuel burnt varies slightly with respect to combustion cycle and it determines the combustion duration. Combustion duration in first cycle is faster than in the second cycle and third cycle. However, the second cycle is slower than the third cycle. It shows of instability for each cycle which has affected the in cylinder peak pressure and the equivalence ratio of in the cylinder for each cycle.

Figure 6 shows in cylinder equivalence ratio changing with respect to engine cycles for different intake temperature. For all intake temperature, the fluctuations are significant for first six cycles and become more stable after that. However, as the intake temperature is increased without increasing the fuel injected, the mixture of AFR becomes leaner. At $T_{in}$ 373K, maximum equivalence ratio in the cylinder is 0.554 which is increased 39% from its intake equivalence ratio and maximum decrease is 0.291 which is decreased by 27%. At $T_{in}$ 473K, maximum increased and decreased in cylinder Ø is 7.3% and 37.8%. While at $T_{in}$ 573K, there is no increment in cylinder Ø instead the maximum decrement by 41%. When the temperature is increased more than 573K, misfiring happens at cycle 2 and the combustion has stopped.

## CONCLUSION

It is crucial to have uncertainties variables in the model in order to minimize the fluctuating during the transient process that is before the combustion stabilized. In engineering programming, the data that is subjected to significant uncertainty can be modelled using stochastic programming. Most of the time, this uncertainty is treated as negligible and the model is commonly called as deterministic. In cylinder flow conditions in internal combustion engines are inherently stochastic.

A reliable model of HCCI engine performance must include the description of the processes that captures cycle-to-cycle variations which can be regarded as uncertainty or random noise. Results show that in-cylinder pressure and equivalence ratio vary every cycle when run under simple discrete engine model which observes only the deterministic features. A change in intake temperature has significantly varied the equivalence ratio in the cylinder and consequently has affected the pressure inside the cylinder. Equivalence ratio variation seems becoming stable after six cycles, but having much lower value than at the first cycle. Intake temperature and equivalence ratio are related to each other in terms of intake air throttle and injectied fuel mass. Thus, either intake air throttle or injected fuel mass should be modeled including its uncertainty parts to control the stability during first six cycles of engine operations.

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