An HCCI Control Oriented Model that Includes Combustion Efficiency

Khashayar Ebrahimi, Masoud Aliramezani
Charles Robert Koch

Mechanical Engineering Department, University of Alberta, Edmonton, Alberta, Canada (e-mail: ebrahimi@ualberta.ca)

Abstract: A control oriented model that includes combustion timing, engine load and combustion efficiency for Homogeneous Charge Compression Ignition (HCCI) engines is developed. In HCCI engines, a lean homogeneous air-fuel mixture auto-ignites due to compression and combustion occurs at lower temperatures compared to spark ignition and diesel engines. The low HCCI combustion temperature results in low NO_X level, however unburnt HC and CO levels are high. Higher thermal efficiencies are realized for higher combustion efficiencies when combustion timings is appropriate. First, the effects of valve timing and fueling rate on combustion efficiency are investigated experimentally. Then, the influence of combustion efficiency on HC and CO emissions is studied. A physics based control oriented model of HCCI engine combustion efficiency and emissions for future control design is developed. This model includes the effect of trapped residual gas and fueling rate on combustion timing and output power. The developed model has acceptable accuracy for combustion timing, load and combustion efficiency prediction when compared to experimental data. This model is useful for combustion timing and load control in HCCI engines while simultaneously considering the constraints of combustion efficiency and emission.

Keywords: HCCI, combustion efficiency, combustion timing, load, control oriented model, emissions

1. INTRODUCTION

HCCI engines present an opportunity for fuel consumption and emission reduction in automotive industry (Zhao, 2006). HCCI operating range is limited by misfire at low loads and knock at high loads. The major HCCI engine emissions are CO and unburnt HC (Zhao, 2006). These emissions have been controlled using different actuators and strategies, for example, Variable Valve Timing (VVT) (Schramm, 2014), oxidation catalyst (Williams et al., 2008), Exhaust Gas Recirculation (EGR) and fuel octane number (Lu et al., 2005). In addition, HCCI combustion timing is difficult to control since there is no direct initiator of combustion. Combustion timing in HCCI engines is highly sensitive to trapped mixture temperature, pressure and composition at Intake Valve Closing (IVC). Several techniques have been developed and implemented for combustion timing control in HCCI engines including variable compression ratio (Haraldsson et al., 2003), intake air heating (Najt and Foster, 1983), dual fuels (Audet and Koch, 2009), pilot injection (Ravi et al., 2012), VVT (Bengtsson et al., 2006a; Shaver et al., 2005) and EGR (Kang and Druzhinina, 2010). Among these strategies, VVT is interesting since it reduces residual gas heat loss and achieves fast cycle-by-cycle control response (Ebrahimi and Koch, 2015).

Control oriented models for HCCI combustion timing and load control using different actuators are being actively developed. A short summary of some HCCI control oriented models is given to provide context for the model developed in this work. A mean value model is developed in (Rausen et al., 2005) for HCCI combustion timing control. Exhaust rebreathing lift is used as the valve timing strategy and the model inputs are valve timing and fueling rate. The model has five continuous states: intake manifold mass and burned gas fraction and exhaust manifold mass, pressure and burned gas fraction. The discontinuous states of the model are in-cylinder temperature and pressure at IVC, in-cylinder mixture composition and combustion timing. Multi-variable output-error state space model is used for system identification in (Strandh et al., 2004) and the model order is defined based on singular values of the Hankel matrix. The model output is combustion timing and the model inputs are the fuel ratio of dual fuels, injected fuel energy, engine speed and the intake manifold pressure and temperature. The model is used for LQG controller design. A two-input, two-state model is developed in (Shaver et al., 2006) for HCCI combustion timing control. The model states are peak in-cylinder pressure and combustion timing and the model inputs are IVC and Exhaust Valve Closing (EVC) timings. The control oriented model is validated against a detailed physical model and is used to design a H_2 controller. A four-state linear model is developed for combustion timing and load control in (Ravi et al., 2010). The model states are oxygen and fuel concentration, temperature and in-cylinder volume at IVC and the model inputs are fueling rate, IVC and EVC timings. A five-state model is developed in (Bidarvatan et al., 2014) for combustion timing and load control using Discrete Sliding Mode Controller (DSMC). The model states are combustion timing, temperature and pressure at start of combustion and residual gas mass fraction and temperature. The model inputs are fuel octane number and fuel equivalence ratio. A five-state model is developed for HCCI combustion timing and load control in (Ravi et al., 2012) using Model Predictive Controller (MPC) with valve timing and split fuel injection as the main actuators. The model used in (Ravi et al., 2012) is from (Ravi et al., 2010) with split injection combustion threshold as a new state. A six-order Multi-Input Multi-Output (MIMO) model is developed using system identification in (Bengtsson et al., 2006b) for HCCI combustion timing and load control using MPC. The model inputs are IVC timing, injected fuel energy, intake temperature and engine speed and the outputs are combustion timing, output power and maximum rate of pressure rise. In (Ebrahimi and Koch, 2015), a five-state model is developed for combustion timing and load control using MPC. The model inputs are injected fuel energy and Symmetric Negative Valve Overlap (SNVO) duration while the model states are IVC temperature, residual gas fraction, fuel equivalence ratio, injected fuel energy and combustion timing.

A four-state control oriented model is developed in this work for HCCI combustion timing and load control considering effects of fueling rate and valve timing on combustion efficiency. The model inputs are SNVO duration and the fueling rate. The model states are IVC temperature, fuel equivalence ratio, combustion efficiency and combustion timing. The EVC timing is set to before Top Dead Center (TDC) in the exhaust stroke and the Intake Valve Opening (IVO) timing is set to the same amount, or symmetric, after TDC in SNVO strategy (Ebrahimi and Koch, 2015). The model is then validated against the measured steady-state experimental data. The physical model developed in (Ebrahimi et al., 2013) is used as virtual engine for transient model validation. To the authors' knowledge, this is the first time that combustion efficiency is included in HCCI control oriented modeling.

2. EXPERIENTIAL SETUP

Experiments are conducted on a single cylinder Ricardo Hydra Mark III engine equipped with EVVT system (Stolk and Gaisberg, 2001). Details of the single cylinder engine and the EVVT mechanism are found in (Schramm, 2014). N-heptane is used as fuel for this experiment. The experimental conditions of the steady-state points measured for this study are summarized in Table 1. The amount of injected fuel energy and the SNVO duration are varied while all other engine variables are kept constant. Pressure traces from 100 consecutive engine cycles are recorded every 0.1 CAD for each engine test and other engine variables are measured at a constant sample rate of 100 Hz during each test.

3. DATA ANALYSIS

On the engine, SNVO duration is varied for several injected fuel energies while other parameters are kept constant as listed in Table 1. The measurements are used to develop and validate the control oriented model. Combustion timing, burn duration and engine output power

Table 1. Engine Operating Conditions

Parameter	Values
Engine Speed [rpm]	725 - 825
T_{Intake} [$°C$]	80
P_{Intake}[kPa]	88 - 90
Injected Fuel Energy	0.356 - 0.455
$T_{Coolant}$ [$°C$]	50
Oil temperature [$°C$]	50
ON [PRF]	0
EVC [bTDC]	$-350°$ - $-300°$
IVO [bTDC]	$300°$ - $350°$
EVO [bTDC]	$-180°$
IVC [bTDC]	$180°$

are from (Schramm, 2014) while combustion efficiency is calculated using a modified single zone model as

$$\eta_{Comb} = \frac{c_1 Q_{HR}}{m_f LHV_f} + c_2 \tag{1}$$

where Q_{HR}, m_f and LHV_f are the net energy released during combustion, the injected fuel mass and n-heptane low heating value respectively. c_1 and c_2 are constants and are listed in Table 2. Details of Eqn. 1 are given in Appendix A.

3.1 SNVO duration and fueling rate effects on HCCI combustion

First, SNVO effects on combustion efficiency and combustion timing at constant injected fuel energies are shown in Fig. 1. The IVC temperature increases at higher NVOs (Ebrahimi et al., 2013) and the combustion timing advances as a result (see Fig. 1(a)). With advanced combustion timing, the trapped mixture has enough time to completely burn and the reactions are quenched later which improves combustion efficiency (see Fig. 1(b)). Combustion efficiency is improved considerably at low loads with increasing SNVO duration compared to high loads. Next, the effects of combustion efficiency on unburnt HC, CO and CO_2 at constant injected fuel energies are shown in Fig. 2 for engine operation without ringing or misfire. The CO and HC are reduced at higher combustion efficiencies while CO_2 is increased as shown in Fig. 2(a) through (c). Engine emissions are related to combustion efficiency and can be influenced by valve timing and fueling rate as actuators.

4. CONTROL ORIENTED MODEL

A Control Oriented Model (COM) of a single cylinder HCCI engine is developed and parameterized using measured experimental data. The COM that is based on physics is developed to consider the effects of valve timing and fueling rate on combustion timing, engine output power and combustion efficiency. At the start of the cycle, fresh charge is inducted into the cylinder and intake charge is assumed to instantaneously mix with the trapped residual gas at IVC. In-cylinder gas temperature at IVC, T_{IVC} is calculated as

$$T_{IVC,k} = \frac{x_{r,k} c_{p,r} T_{exh,k-1} + (1 - x_{r,k}) c_{p,i} T_{int,k}}{c_{p,IVC}} \tag{2}$$

where $c_{p,r}$, $c_{p,i}$ and $c_{p,IVC}$ are residual gas, intake charge and IVC trapped charge specific heat values at constant

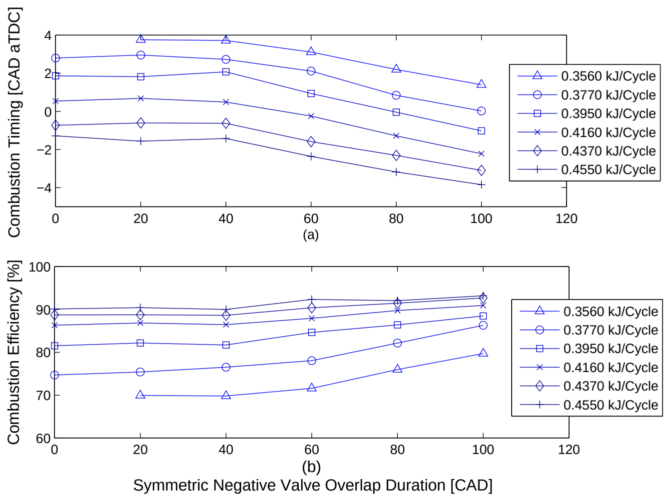

Fig. 1. Effects of SNVO duration on (a) combustion timing (b) combustion efficiency at constant injected fuel energies

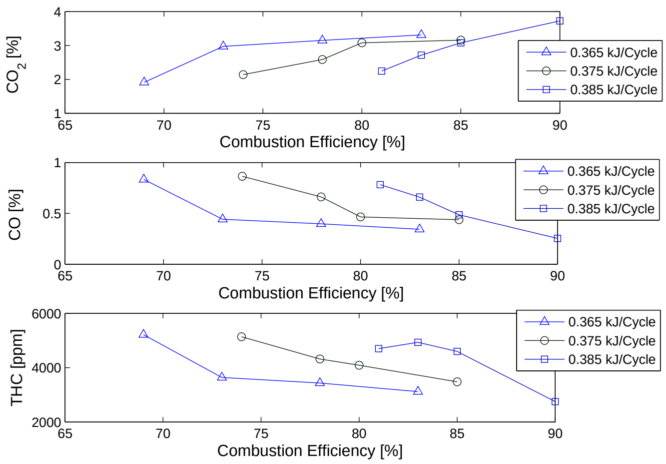

Fig. 2. (a) CO_2 (b) CO and (c) Unburnt HC versus combustion efficiency at constant injected fuel energies

pressure. x_r and k are residual mass gas fraction and cycle number respectively. All model constant values are listed in Table 2. Residual gas mass fraction, x_r is calculated as (Heywood, 1988)

$$x_{r,k} = \frac{M_r y_{r,k}}{M_i - M_i y_{r,k} + M_r y_{r,k}} \tag{3}$$

where M_i and M_r are the exhaust and intake gas mean molecular weights. Residual gas mole fraction, y_r is calculated as

$$y_{r,k} = \frac{P_{exh,k-1}}{P_{IVC,k}} \frac{V_{EVC,k-1}}{V_{IVC,k}} \frac{T_{IVC,k}}{T_{exh,k-1}} \tag{4}$$

where V_{EVC} and V_{IVC} are the in-cylinder volume at EVC and IVC respectively and are calculated form slider-crank mechanism equation (Heywood, 1988). The intake process is assumed to take place at atmospheric pressure

$$P_{IVC,k} = P_{int,k} \tag{5}$$

as our HCCI engine usually operates at wide-open throttle with the variable valve timing used to control the amount of trapped charge.

The crank angle of fifty percent fuel mass fraction burned, θ_{50} is calculated as

$$\theta_{50,k} = \theta_{soc,k} + 0.5\Delta\theta_k \tag{6}$$

where θ_{soc} and $\Delta\theta$ are the start of combustion and burn duration respectively. Sensitivity analysis is performed to define the parameters which have important effect on start of combustion. The start of combustion, θ_{soc} is calculated as

$$\theta_{soc,k} = c_3 T_{IVC,k}\phi_k + c_4\phi_k + c_5 T_{IVC,k} + c_6 \tag{7}$$

where ϕ is fuel equivalence ratio and c_i are model constants determined using Matlab Model-Based Calibration toolbox and are listed in Table 2. The other sub-models including burn duration, fuel equivalence ratio, combustion efficiency and IMEP are parameterized with the same method.

Burn duration, $\Delta\theta$ is calculated as

$$\Delta\theta_k = c_7\theta_{soc,k} + c_8 \tag{8}$$

The temperature and pressure at the crank angle of fifty percent fuel mass fraction burned, with the assumption of isentropic compression, are calculated as

$$P_{50,k} = P_{int,k}\left(\frac{V_{IVC,k}}{V_{50,k}}\right)^{\gamma} \tag{9}$$

$$T_{50,k} = T_{IVC,k}\left(\frac{V_{IVC,k}}{V_{50,k}}\right)^{\gamma-1} \tag{10}$$

where V_{50} is the in-cylinder volume at the crank angle fifty of mass fraction burned. γ is assumed constant and is parameterized using the experimental data.

The first law of thermodynamics is applied to the system to calculate in-cylinder gas temperature after combustion. Temperature after combustion, T_{AC} is calculated as

$$T_{AC,k} = T_{50,k} + \frac{c_9\phi_k\eta_{c,k}LHV_f}{c_p AF_{stoich}} \tag{11}$$

where η_c and AF_{stoich} are combustion efficiency and stoichiometric air fuel ratio respectively and c_p is the specific heat ratio at θ_{50}.

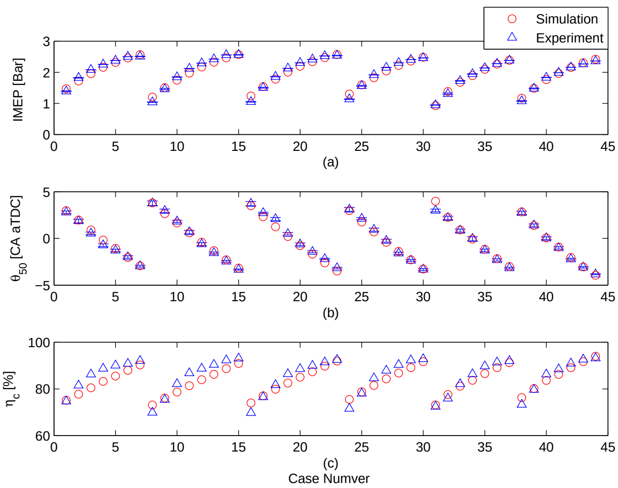

Fig. 3. Steady state validation of COM (a) engine output power (b) combustion timing and (c) combustion efficiency

The fuel equivalence ratio, ϕ is calculated as

$$\phi_k = c_{10}T_{IVC,k}y_k + c_{11}T_{IVC,k} + c_{12}y_k + c_{13}m_{f,k}LHV_f + c_{14} \quad (12)$$

Combustion efficiency, η_c in Eqn. 11, is calculated as

$$\eta_{c,k} = c_{15}\theta_{50,k} + c_{16}T_{IVC,k} + c_{17}y_{r,k} + c_{18}\phi_k^{c_{19}} + c_{20} \quad (13)$$

By applying the ideal gas law before and after combustion, the in-cylinder pressure after combustion is calculated as

$$P_{AC,k} = \frac{P_{50,k}T_{AC,k}}{T_{50,k}} \quad (14)$$

The expansion process is assumed to be isentropic and the in-cylinder gas temperature and pressure at Exhaust Valve Opening (EVO) is calculated as

$$T_{EVO,k} = (\frac{V_{50,k}}{V_{EVO,k}})^{\gamma_e - 1}T_{AC,k} \quad (15)$$

$$P_{EVO,k} = (\frac{V_{50,k}}{V_{EVO,k}})^{\gamma_e}P_{AC,k} \quad (16)$$

where γ_e is determined using experimental data and V_{EVO} is the in-cylinder volume at EVO.

At EVO, blowdown to the exhaust manifold pressure is assumed to occur as an isentropic process. The residual gas temperature after blowdown is calculated as (Heywood, 1988)

$$T_{exh,k} = \left(\frac{P_{exh,k}}{P_{EVO,k}}\right)^{\frac{\gamma_e - 1}{\gamma_e}} T_{EVO,k} \quad (17)$$

where P_{exh} is the exhaust manifold pressure.

Finally, engine output work is calculated from a correlation obtained from experimental data:

$$\text{IMEP}_k = c_{21}\eta_k m_{f,k}LHV_f + c_{22}\eta_k + c_{23}m_{f,k}LHV_f + c_{24}\theta_{50,k} + c_{25} \quad (18)$$

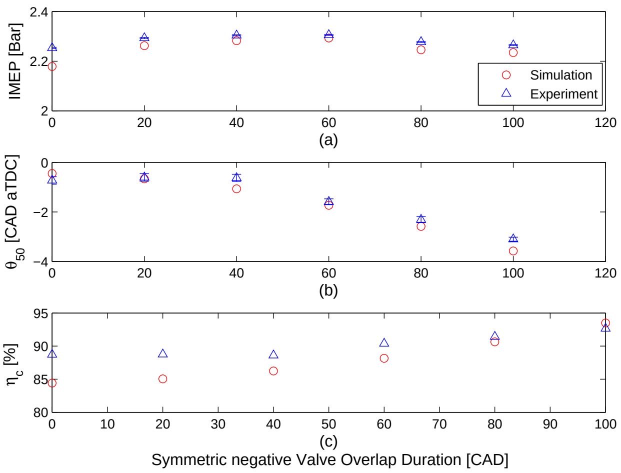

Fig. 4. Steady state validation for SNVO sweep at $m_f LHV_f$= 0.4374 $\frac{kJ}{Cycle}$, ω=825 RPM, P_{int}=88.4 kPa, T_{int}=80^oC (a) engine output power (b) combustion timing and (c) combustion efficiency

The control oriented model, in standard state-space form where the states can be written as a function of the inputs and state variables of the previous cycle, is

$$x_{k+1} = f(x_k, u_k, w_k) \quad (19)$$
$$y_k = g(x_k, u_k, w_k)$$

where x is the state of the system, u is the control input and w is a disturbance. The states, the inputs and the outputs for this system are

$$x = [T_{ivc}\ \phi\ \eta_c\ \theta_{50}]^T \quad (20)$$
$$u = [m_f Q_{LHV}\ \theta_{EVC}]^T$$
$$y = [\theta_{50}\ \text{IMEP}]^T$$

Table 2. Model parameters and constants

c_1	55.271	c_2	44.176	c_3	0.00788
c_4	-4.043	c_5	-0.0036	c_6	1.493
c_7	1.7365	c_8	0.581	c_9	0.76
c_{10}	0.0012	c_{11}	-7.7 $\times$ 10^{-5}	c_{12}	-0.3582
c_{13}	0.6734	c_{14}	0.071	c_{15}	-0.044
c_{16}	0.039	c_{17}	2.86	c_{18}	17.47
c_{19}	3.99	c_{20}	67.79	c_{21}	-0.2764
c_{22}	0.1385	c_{23}	29.90	c_{24}	0.075
c_{25}	-12.36	$c_{p,r}$	1.079	$c_{p,i}$	1.15
$c_{p,IVC}$	1.046	c_p	1.113	M_i	29.39
M_r	28.74	γ_e	1.3	γ	1.29

5. CONTROL ORIENTED MODEL VALIDATION

For the steady state experimental data range listed in Table 1, the COM is compared to experiment in Fig. 3. Fig. 3 shows that the COM captures combustion timing, combustion efficiency and engine output power with average errors of 0.65 CAD, 7.54% and 0.09 Bar respectively. The COM is then compared for SNVO duration variations at a constant fueling rate in Fig. 4. The amount of trapped residual gas increases with an increase in SNVO and the in-cylinder gas temperature at IVC increases. Combustion timing advances with increased IVC temperature and the combustion efficiency increases as shown in Fig. 4. IMEP increases with increasing SNVO until SNVO reaches 60 CAD then decreases due to increased compression work. The predicted values from the COM match the experiments well. Then, fueling rate is varied and SNVO duration is kept constant and the simulation is compared to experiment in Fig. 5. The effects of fueling rate on combustion timing, combustion efficiency and output power at constant SNVO are shown in Fig. 5. Combustion timing advances with increase in fueling rate as the fuel reactivity tends to increase from lean to rich conditions (Shahbahkti, 2009). IMEP increases as fueling rate is increased and combustion efficiency increases with advanced combustion timing. These results indicate that COM captures the fueling rate and the trapped residual gas effects on combustion timing well.

Transient validation is performed using the Detailed Physical Model (DPM) (Ebrahimi et al., 2013) and the results are shown in Fig. 6. The COM captures the trends of the DPM when step changes are applied to the fueling rate and SNVO duration. The COM does not match the

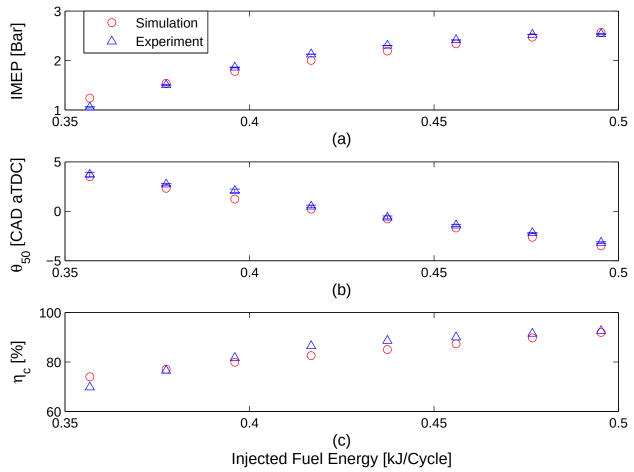

Fig. 5. Steady state validation for fuel sweep at SNVO= 40 CAD, ω=825 RPM, P_{int}=88.4 kPa, T_{int}=80^oC (a) engine output power (b) combustion timing and (c) combustion efficiency

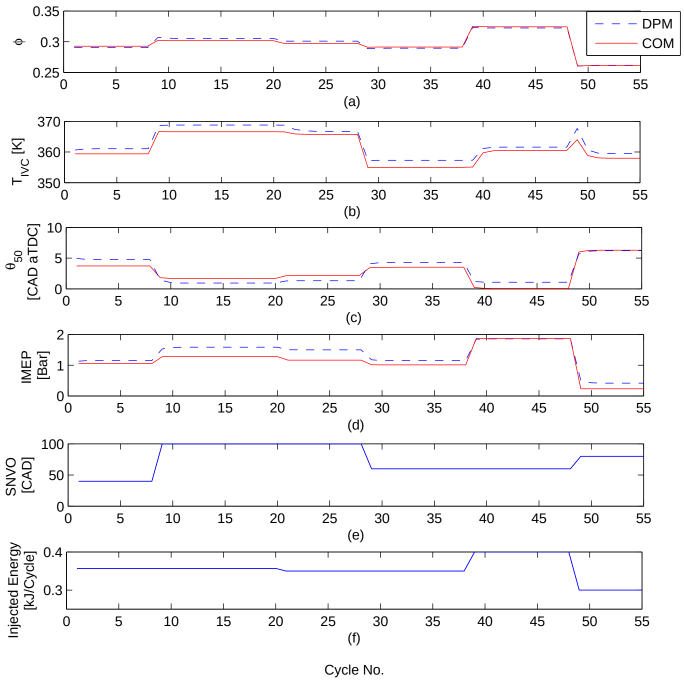

Fig. 6. Transient COM validation against DPM [ω=825 RPM, P_{int}=88.4 kPa and T_{int}=80^{o}C]

DPM values exactly and this is attributed to the much simpler structure of the COM. However, the COM ability to predict DPM state values make the COM useful for model-based controller design that includes combustion efficiency.

6. CONCLUSIONS

A physics based 4-state nonlinear control oriented model is developed for HCCI combustion timing and output work control considering combustion efficiency. The model includes the effect of trapped residual gas and fueling rate on HCCI combustion. The model is parameterized using the measured 44 steady state engine points and then is validated against the experimental data. A detailed physical model is used to check the COM performance for transient operation to step changes in fueling and SNVO. The COM shows acceptable accuracy in predicting DPM states. Since the model is based on physics, the parametrization for different fuels and engines should be straightforward. The model can be used as the basis for feedback control design of HCCI engine ignition timing and load while also considering constraints on combustion efficiency and emission.

REFERENCES

Audet, A. and Koch, C.R. (2009). Actuator Comparison for Closed Loop Control of HCCI Combustion Timing. SAE 2009-01-1135.

Bengtsson, J., Strandh, P., Johansson, R., Tunestål, P., and Johansson, B. (2006a). Model predictive control of Homogeneous Charge Compression Ignition (HCCI) engine dynamics. IEEE International Symposium on Intelligent Control.

Bengtsson, J., Strandh, P., Johansson, R., Tunestål, P., and Johansson, B. (2006b). Multi-Output Control of a Heavy Duty HCCI Engine Using Variable Valve Actuation and Model Predictive Control. SAE 2006-01-0873.

Bidarvatan, M., Shahbakhti, M., Jazayeri, S., and Koch, C. (2014). Cycle-to-cycle modeling and sliding mode control of blended-fuel HCCI engine. *Cont. Eng. Prac.*, 24(0), 79 – 91.

Dernotte, J., Dec, J.E., and Ji, C. (2015). Energy Distribution Analysis in Boosted HCCI-like/LTGC Engines - Understanding the Trade-Offs to Maximize the Thermal Efficiency. *SAE 2015-01-0824*.

Ebrahimi, K. and Koch, C. (2015). Model Predictive Control for Combustion Timing and Load Control in HCCI Engines. SAE 2015-01-0822.

Ebrahimi, K., Schramm, A., and Koch, C.R. (2013). A Control Oriented Model with Variable Valve Timing for HCCI Combustion Timing Control. SAE 2013-01-0588.

Haraldsson, G., Tunestål, P., Johansson, B., and Hyvönen, J. (2003). HCCI Combustion Phasing with Closed-Loop Combustion Control Using Variable Compression Ratio in a Multi Cylinder Engine. SAE 2003-01-1830.

Heywood, J.B. (1988). *Internal combustion engine fundamentals*. New York : McGraw-Hill, 1988.

Jun, D. and Iida, N. (2004). A Study of High Combustion Efficiency and Low CO Emission in a Natural Gas HCCI Engine. SAE 2004-01-1974.

Jung, D. and Assanis, D.N. (2001). Multi-Zone DI Diesel Spray Combustion Model for Cycle Simulation Studies of Engine Performance and Emissions. SAE 2001-01-1246.

Kang, J.M. and Druzhinina, M. (2010). HCCI engine control strategy with external EGR. In *Proceedings of the 2010 American Cont. Conf.*, 3783–3790.

Lu, X.C., Chen, W., and Huang, Z. (2005). A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR. part 2. effect of operating conditions and EGR on HCCI combustion. *Fuel*, 84(9), 1084 – 1092.

Najt, P.M. and Foster, D.E. (1983). Compression-ignited homogeneous charge combustion. *SAE 830264*.

Rausen, D., Stefanopoulou, A., Kang, J.M., Eng, J., and Kuo, T.W. (2005). A mean-value model for control of homogeneous charge compression ignition (HCCI) engines. *Trans. of the ASME. Jour. of Dyn. Sys., Meas. and Cont.*, 3(3), 355–362.

Ravi, N., Roelle, M., Liao, H.H., Jungkunz, A., Chang, C.F., Park, S., and Gerdes, J. (2010). Model-Based Control of HCCI Engines Using Exhaust Recompression. *IEEE Trans. on Cont. Sys. Tech.*, 18(6), 1289–1302.

Ravi, N., Liao, H.H., Jungkunz, A.F., Widd, A., and Gerdes, J.C. (2012). Model predictive control of HCCI using variable valve actuation and fuel injection. *Cont. Eng. Prac.*

Schramm, A. (2014). *Effects of Negative Valve Overlap on HCCI combustion and its use in the control of HCCI combustion timing.* Master's thesis, Univ. of Alberta.

Shahbahkti, M. (2009). *Modeling and Experimental Study of an HCCI Engine for Combustion Timing Control.* Ph.D. thesis, Univ. of Alberta.

Shahbahkti, M., Ghazimirsaied, A., and Koch, C.R. (2010). Experimental study of exhaust temperature variation in an HCCI engine. *Jour. of Auto. Eng. - Part D*, 224, 1177–1197.

Shaver, G., Roelle, M., and Gerdes, J. (2005). Decoupled control of combustion timing and work output in residual-affected HCCI engines. In *Proceedings of the 2005 American Cont. Conf.*

Shaver, G., Roelle, M., and Gerdes, J. (2006). A two-input two-output control model of HCCI engines. In *Proceedings of the 2006 American Cont. Conf.*

Stolk, T. and Gaisberg, A.V. (2001). Elektromagnetischer aktuator, German Patent DE10025491A1.

Strandh, P., Bengtsson, J., Johansson, R., Tunestål, P., and Johansson, B. (2004). Cycle-to-cycle control of a dual-fuel HCCI engine. SAE 2004-01-0941.

Williams, S., Hu, L.R., Nakazono, T., Ohtsubo, H., and Uchida, M. (2008). Oxidation Catalysts for Natural Gas Engine Operating under HCCI or SI Conditions. *SAE 2008-01-0807.*

Xu, H., Liu, M., Gharahbaghi, S., Richardson, S., Wyszynski, M., and Megaritis, T. (2005). Modelling of HCCI Engines: Comparison of Single-zone, Multi-zone and Test Data. SAE 2005-01-2123.

Zhao, H. (2006). *HCCI and CAI engines for the automotive industry.* CRC Press.

Appendix A. MODIFIED SINGLE ZONE MODEL FOR COMBUSTION EFFICIENCY CALCULATION

The exhaust gas temperature in HCCI engines is low as the engine runs with lean mixture (Zhao, 2006; Shahbahkti et al., 2010). Due to low combustion temperature in HCCI engines, the oxidation does not continue in exhaust manifold and the combustion efficiency is mainly dependent on the concentration of unburnt HC and CO in the exhaust gases (Jun and Iida, 2004; Dernotte et al., 2015). The combustion efficiency is calculated using measured engine emission as (Dernotte et al., 2015)

$$\eta_c = 100 - 100 \times \frac{m_{HC} LHV_f + m_{CO} LHV_{CO}}{m_f LHV_f} \qquad \text{(A.1)}$$

where m_{HC} and m_{CO} are the measured unburnt HC and CO masses respectively in the exhaust. The parameter LHV_{CO} is the lower heating value of CO. Accurate emission measurement for whole engine operating range is time consuming and difficult. To avoid emission measurement, a model used in (Schramm, 2014) is further improved. The combustion efficiency in (Schramm, 2014) is calculated as

$$\eta_c = 100 \times \frac{Q_{HR}}{m_f LHV_f} \qquad \text{(A.2)}$$

where Q_{HR} is the net heat release and is calculated from

$$Q_{HR} = \int_{SOC}^{EOC} \left(\frac{dQ_{HR}}{d\theta} \right) d\theta \qquad \text{(A.3)}$$

where SOC and EOC are the Start of Combustion and End of Combustion respectively. Rate of heat release, $\frac{dQ_{HR}}{d\theta}$ is calculated from a single zone model(Jung and Assanis, 2001).

The combustion efficiency calculated from Eqn. A.2 is compared to the combustion efficiency calculated based on measured emission (Eqn. A.1) in Fig. A.1(a). The combustion efficiency values calculated from Eqn. A.2 is less than the values calculated from the measured emission (Eqn. A.1) with the average error of 10.13%. The reason is single zone models are not accurate in predicting burn rate, mixture composition and temperature at IVC (Xu et al., 2005). The single zone model accuracy is improved using MATLAB Model-Based Calibration Toolbox as

$$\eta_c = \frac{c_1 Q_{HR}}{m_f LHV_f} + c_2 \qquad \text{(A.4)}$$

The improved single zone model is parameterized and validated against the combustion efficiency values calculated based on measured emission in Fig. A.1(b). The new developed single zone model shows acceptable accuracy with the average error of 2.17% for one SNVO and fuel sweep.

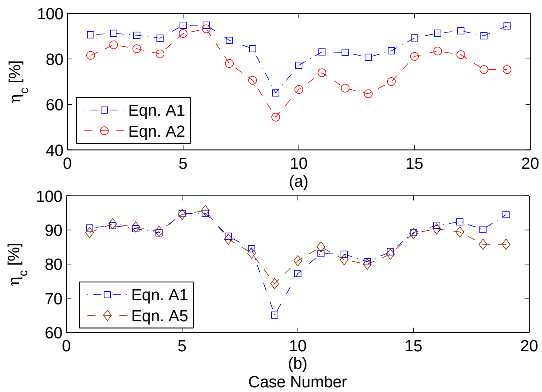

Fig. A.1. (a) Combustion efficiency calculated based on measured emission vs combustion efficiency calculated from single zone model, (b) Combustion efficiency calculated based on measured emission vs combustion efficiency calculated from modified single zone model [$m_f LHV_f$= 0.33-0.39$\frac{\text{kJ}}{\text{Cycle}}$, SNVO=40-120 CAD, ω=825 RPM, P_{int}=88.4 kPa and T_{int}=80^{o}C]