Combustion Phasing and Work Output Modeling for Homogeneous Charge Compression Ignition (HCCI) Engines

Song Chen and Fengjun Yan*

Abstract—Combustion phasing and work output are the two critical indicators for achieving and maintaining homogeneous combustion for homogeneous charge compression ignition (HCCI) engines. However, the complicated thermodynamic process keeps the combustion phasing model from being explicit. The current models for CA₅₀ (crank angle when 50% fuel consumed) requires information that is hardly available for commercial engines. In this paper, polynomial models for combustion phasing and work output are proposed. The intake conditions, including intake manifold temperature, pressure and burned gas fraction as well as the fuel injection are chosen as the four variables by which the combustion phasing and work output are determined. The impacts of various compression ratios on the two indicators (represented by CA₅₀ and IMEP) were investigated and have been classified into two groups: vertical shift and curve deformation and both can be well modeled and therefore compensated. The models are validated through high-fidelity simulations.

I. INTRODUCTION

Homogeneous charge compression ignition (HCCI) offers a low pollutant emissions and high thermodynamic efficiency solution for the stricter and stricter constraint regulations on vehicle emissions[1][2][3][4]. From this point of view, the HCCI engines is the combination of the best properties of spark ignition (SI) and compression ignition (CI) for the reason that the SI engines have cleaner emissions, while the CI engines have a better efficiency[5]. However, unlike SI engines and CI engines, in which the combustion is ignited by spark and injection(to a great extend) respectively, the combustion of HCCI is rely more on the thermochemical states of the mixture in the cylinder. Thus, to obtain the combustion phasing and further to construct a relationship between it and the intake conditions are both difficult and necessary for achieving, maintaining, and also expanding operating range of HCCI engines.

The Arrhenius integral, as shown in equation (1), is the dominant model for the start of combustion (SOC)[6][7]:

\[ k_{th} = \int_{\text{SOC}}^{\text{IVC}} AP_{cyl}^m(t) \exp\left( \frac{E_a}{RT_{cyl}(t)} \right) [F]^n(t) [O_2]^b(t) dt, \]  

where, \( P_{cyl} \) and \( T_{cyl} \) are the cylinder pressure and temperature, respectively; \([F]\) and \([O_2]\) are the fuel and oxygen fraction in the cylinder; \( R \) is the gas constant; \( A, E_a, a \) and \( b \) are all empirical terms. The SOC is determined by taking the integral of equation (1) from the timing of intake valve closing (IVC) to an instant, where the integral reaches to a threshold, \( k_{th} \), then this instant is considered as the SOC.

Although under some conditions, \( P_{cyl} \), \( T_{cyl} \), \([F]\) and \([O_2]\) can be approximately viewed as constant, it is still very hard to get an analytical solution for equation (1), which is one of the important reason prevent it from utilized in practice.

Since the combustion efficiency is closely tied with the CA₅₀ (crank angle when 50% fuel consumed), people care more about CA₅₀ in some cases than SOC. To get the CA₅₀, it requires the combustion duration as well, besides the timing of SOC. The combustion duration can be modeled as[8][9]:

\[ \text{Dq} = n(T_{cyl}^2 - T_m^2) \left( \frac{T_{cyl}}{T_m} \right)^{5/3} \exp\left( \frac{E_a}{3RT_{cyl}} \right), \]  

where, \( n \) is an empirical parameter; \( T_{SOC} \) is the cylinder temperature at SOC and \( T_m \) is the mean temperature during combustion. Only empirical models of \( T_m \) exist.

From the above equations one can see that it is really hard to find the CA₅₀ in an analytical way. One easier method to handle it is simplifying the Arrhenius integral and the combustion duration in maps where only major parameters considered.

In [8], three parameters: the fuel mass injected, the in-cylinder temperature at IVC and the auto ignition level were considered and their relationship with SOC was condensed in a map instead of taking the integral. The combustion duration was simplified to a constant value in the operating range.

In [5][10], the integral was condensed into a map where the SOC is a function of fuel and oxygen concentration and the in-cylinder temperature at the defined point. The duration is approximated as an affine function of the SOC.

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The intake pressure, temperature and oxygen fraction were mapped with the SOC in [11]. In [12] the SOC was fitted with injection timing while fixing other conditions.

To sum up, the CA50 can hardly be estimated in an analytical fashion, and the existing alternative methods require great effort on calibration.

A novel model for the combustion phasing (CA50) is chosen and work output (represented by indicated mean effective pressure, IMEP) are proposed in this paper. The CA50 is fitted by a polynomial model with parameters of intake temperature, pressure and burned gas fraction and the mass of injection; and the IMEP is modeled by a linear equation of injection. The value of the coefficients of the parameters were calibrated in steady cases and validated in transient cycles in a turbocharged, compression ignited, gasoline engine with EGR (exhaust gas recirculation) system in GT-Power software environment. The impacts of various compression ratios on the two indicators (CA50 and IMEP) were investigated and have been classified into two groups: vertical shift and curve deformation and both can be well modeled and, therefore compensated. The simulations show that: 1) the proposed models can well predict the CA50 and IMEP; 2) it only requires about ten parameters calibration; 3) the hardly obtained information in commercial engines, such as the in-cylinder temperature and oxygen fraction, can be avoided. The simple models can be potentially utilized in HCCI controlling.

The arrangement of the rest of this paper is as follows. In section II, the combustion phasing and work output models are demonstrated and calibrated. The proposed models are validated in section III. The impacts of various compression ratios are investigated in section IV. Conclusive remarks are given in section V.

II. COMBUSTION PHASING AND WORK OUT MODELING

A. Parameters selecting and modeling

Variable geometry turbocharged (VGT) engine with EGR system (as shown in Figure 1) provides a flexible control architecture which has been commonly utilized in gasoline engines to achieve HCCI and also has the potentiality to expand its operating range.

It is claimed that by tuning the EGR valve and the rack position of the VGT, the intake pressure, \( P_{\text{int}} \), intake temperature, \( T_{\text{int}} \), and burned gas fraction, \( F_{\text{int}} \), can all be well controlled in a specific range [13][14][15][16], which facilitates the controlling of HCCI to a great extent.

The sensitivity of the Arrhenius integral in equation (1) with respect to variables including \( P_{\text{cyl}} \), \( T_{\text{cyl}} \), \( [F] \), \( \text{[O}_2\text{]} \), etc, were analyzed in [9]. It’s said that comparing to other parameters, the SOC is more sensitive to \( T_{\text{cyl}} \), \( q_{\text{inj}} \) (crank angle at the intake valve closing) and the compression ratio, CR.

Figure 1. Engine architecture with variable geometry turbocharger and EGR loop

Considering the controllability and sensitivity of the parameters as analyzed above, the parameters for the modeling of CA50 are chose as: \( T_{\text{int}} \), \( P_{\text{int}} \), \( F_{\text{int}} \) and the mass of injection, \( m_f \). Fixing the \( q_{\text{inj}} \), the CA50 can be modeled in a quadratic polynomial equation:

\[
CA_{50} = aT_{\text{int}} + bT_{\text{int}}^2 + cP_{\text{int}} + dP_{\text{int}}^2 + eF_{\text{int}} + fR_{\text{int}}^2 + gm_f + hm_f^2 + i,
\]

where, \( a \), \( b \), \( c \), \( d \), \( e \), \( f \), \( g \), \( h \), \( i \) are constant coefficients.

Since IMEP has a strong linear relationship with the mass of injection [5], the IMEP is modeled as a linear equation:

\[
\text{IMEP} = qm_f + w,
\]

where, \( q \) and \( w \) are constant coefficients.

B. Parameters calibrating

For the purpose of calibration, a wide range of operating conditions should be covered. Considering the constraints on the actuators, the ranges of the intake conditions were selected as:

\( T_{\text{int}}, 325 \sim 370K; P_{\text{int}}, 1 \sim 1.75\text{bar}; F_{\text{int}}, 10\% \sim 35\% \).

A gasoline engine with VGT and EGR system as shown in Figure 1. was constructed in GT-Power. 155 cases, composed by different combinations of intake conditions for each, were conducted in the simulation, as shown in the following 2 figures.

For the homogeneous combustion, the mass of injection was calculated based on desired air fuel ratio (AFR) as shown in Figure 3, and the injection timing was set as -10 CA (10 degree before top dead center), the CR was 20.
Figure 2. Intake temperature and pressure of the calibration

Figure 3. The burned gas fraction and mass of injection of the calibration

The real CA50 and IMEP were computed when the states reached steady for each case, as shown in Figure 4 and Figure 5. By applying the least-squares method, the parameters of proposed models of CA50 and IMEP were estimated as:

\[
CA_{50} = 0.386T_{int} - 0.00044T_{int}^2 - 69.53P_{int} + 12.86P_{int}^2 + 5.68F_{int} + 96.61F_{int}^2 + 2.87m_f - 0.028m_f^2 - 74.80
\]

(5)

\[
0.325m_f + 1.07 = IMEP.
\]

(6)

The modeled CA50 and IMEP, calculated by equation (5) and (6), are shown in Figure 4 and Figure 5.

The errors of CA50 and IMEP are plotted in Figure 6 and Figure 7.

One can see from Figure 6 and Figure 7, the errors of the proposed models are within a small range and both have standard deviation less than 0.16. Therefore the CA50 and IMEP have been well modeled by polynomial equations.
C. Model validation

In order to validate the above models, different intake conditions and simulating strategy were applied. Rather than the case-based steady simulation during calibration, the validation was conducted in a cycle-based transient way and the corresponding intake conditions and injection are shown in Figure 8. and Figure 9. By applying the calibrated parameters in equation (5) and equation (6), the modeled and real (got from GT-Power directly) CA and IMEP are compared in Figure 10. and Error! Reference source not found.

Although the intake conditions are different and the simulation is transient, the CA and IMEP can be well predicted, and thus the proposed models validated.

III. THE IMPACTS OF VARIOUS COMPRESSION RATIOS

As mentioned in section II, the CA is very sensitive to CR. Compression ratios can not only change the increasing rates of cylinder temperature and pressure according to the ideal gas law, but also have influence on the amount of residual gas for every single cycle. Both of the two factors introduced by various compression ratios have significantly impacts on CA and IMEP. Therefore, to further investigate the impacts of various compression ratios on the models, a series of simulations with different CRs were conducted.

A. vertical shift

When the CR decreases, the increasing rates of the in-cylinder temperature and pressure will decrease in the compression process, thus, it requires more time for the Arrhenius integral, as demonstrated in equation (1), to reach the \( k_a \), and vice versa. Therefore, decreasing CR will retard CA, and increasing CR can advance CA. This phenomenon is clearly shown in the Figure 12, where the simulated CA for different CR are plotted. The simulation settings are the same with the models validation in section II except for the compression ratios.
Taking the CA$_{50}$ curve whose CR is 20 (the red solid line) as a standard (the models’ parameters were calibrated based on this case), when the CR decreased from 19 to 15 every CA$_{50}$ shifts upward roughly identical distance, as shown in Figure 12. This kind of impact brought by various compression ratios is called vertical shift and it is the direct results of the changes of the in-cylinder conditions.

Figure 12. The calculated CA$_{50}$ for various CR

### B. deformation

Besides the distinct shift of the CA$_{50}$ curve, their shapes are also changed, especially when the CR decreased to 15. By applying the CA$_{50}$ model to all cases, the differences between the real and the modeled CA$_{50}$ for each CR are plotted in Figure 13. Figure 13. tells us that the deformation of the CA$_{50}$ errors curve for each CR tend to be more severe when the CR decreasing. It is can be seen as the impact of the residual gas. Smaller CR means larger amount of residual gas, and thus, stronger coupling effect of adjacent cycles. This impact on the curve from the residual gas is vividly called deformation effect.

Figure 13. The errors of modeled CA$_{50}$ for various CR

### C. modeling and compensating for the impacts

The two impacts brought by various compression ratios, named vertical shift and deformation, significantly damage the performance of the proposed model. However, the strong regularity of the impacts on the curves inspires us that they can be modeled and then compensated. To compensate it, a compensating model with two parameters, the offset factor, off$_{CA_{50}}$, and deformation factor defm which indicates the impact of the residual gas on the CA$_{50}$, are introduced:

$$R_{CA_{50}}(k) = M_{CA_{50}}(k) + \text{off}_{CA_{50}} + \text{defm} \cdot R_{CA_{50}}(k)$$

where, $M_{CA_{50}}$ is the CA$_{50}$ calculated by the model and $R_{CA_{50}}$ is the compensated CA$_{50}$; $k$ is the cycle index, $k = 1, 2, L$. For each CR, off$_{CA_{50}}$ and defm are calibrated and tabulated in TABLE I.

#### TABLE I. VALUES OF off$_{CA_{50}} AND defm$

<table>
<thead>
<tr>
<th>CR</th>
<th>20</th>
<th>19</th>
<th>18</th>
<th>17</th>
<th>16</th>
<th>15</th>
</tr>
</thead>
<tbody>
<tr>
<td>off$<em>{CA</em>{50}}$</td>
<td>0</td>
<td>0.6</td>
<td>1.2</td>
<td>1.8</td>
<td>2.5</td>
<td>3.1</td>
</tr>
<tr>
<td>defm</td>
<td>0</td>
<td>0.1</td>
<td>0.2</td>
<td>0.3</td>
<td>0.6</td>
<td>1.6</td>
</tr>
</tbody>
</table>

Generally, the shift of each CR is roughly proportional with respect to the difference between them and the standard CR, 20. The deformation factor, defm, increased as the CR decreased which indicates that the coupling effect is getting severe: the compensated CA$_{50}$ in next cycle ($R_{CA_{50}}(k+1)$) is determined not only by the next modeled CA$_{50}$ ($M_{CA_{50}}(k+1)$) but also by the previous one ($M_{CA_{50}}(k)$), and the proportion of previous one is bigger as $d$ is getting larger and larger than 0.

By applying the compensating model, equation (7), the errors of compensated CA$_{50}$ is plotted in Figure 14. Comparing to Figure 13., the errors are significantly reduced after compensating.

Figure 14. The errors of compensated CA$_{50}$ for various CR
To be noted, the values of the two factors tabulated in TABLE I. are calculated only for this situation. The strong regularity of the residual effect as shown in Figure 12. and Figure 13. suggests us by recalibrating them the compensating model can be conveniently applied to other situations.

For the IMEP, comparing to CA_{50}, the curve deformation introduced by residual is very weak, as shown in Figure 15.

Figure 15. The calculated IMEP for various CR

The compensating model for the vertical shift is very similar to CA_{50} as equation (7). The errors of modeled IMEP before and after compensating for the vertical shift are plotted in the figure 16.

Figure 16. The modeled IMEP errors before and after compensating for the offset for various CR

Figure 16. shows that after compensating for the shift, the IMEP can be modeled very accurately.

IV. CONCLUSION

Novel models of CA_{50} and IMEP for HCCI engines is proposed in this paper. The impacts of various compression ratios on the models show strong regularity and can be well modeled. Therefore, CA_{50} and IMEP can be predicted in high accuracy. Benefitting from the simple polynomial and high-accurate characteristics, these two models can be potentially utilized in the HCCI controlling field.

REFERENCES