

Input Adaptation for Control Oriented Physics-Based SI Engine Combustion Models Based on Cylinder Pressure Feedback

Shu Wang
ICAR-Clemson Univ.

Qilun Zhu
Clemson Univ.

Robert Prucka
ICAR-Clemson Univ.

Michael Prucka and Hussein Dourra
FCA US LLC

ABSTRACT

As engines are equipped with an increased number of control actuators to meet fuel economy targets, they become more difficult to control and calibrate. The additional complexity created by a larger number of control actuators motivates the use of physics-based control strategies to reduce calibration time and complexity. Combustion phasing, as one of the most important engine combustion metrics, has a significant influence on engine efficiency, emissions, vibration and durability. To realize physics-based engine combustion phasing control, an accurate prediction model is required. This research introduces physics-based control-oriented laminar flame speed and turbulence intensity models that can be used in a quasi-dimensional turbulent entrainment combustion model. The influence of laminar flame speed and turbulence intensity on predicted mass fraction burned (MFB) profile during combustion is analyzed. Then a rule based methodology for laminar flame speed and turbulence intensity correction is proposed. The combustion model input adaptation algorithm can automatically generate laminar flame speed and turbulence intensity correction multipliers based on cylinder pressure feedback for different engine operating conditions. The correction multipliers can be stored into maps or regression equations that then feed into the main combustion model to improve overall prediction accuracy. Research results from this paper show the combustion model prediction error (RMSE) decreased by 88% (from 0.8 CAD to 0.1CAD) for fitting data and 58% (from 1.2 CAD to 0.5 CAD) for validation data when this method was utilized.

CITATION: Wang, S., Zhu, Q., Prucka, R., Prucka, M. et al., "Input Adaptation for Control Oriented Physics-Based SI Engine Combustion Models Based on Cylinder Pressure Feedback," *SAE Int. J. Engines* 8(4):2015, doi:10.4271/2015-01-0877.

INTRODUCTION

Fuel efficiency and emissions regulations result in spark-ignition (SI) engines that incorporate a large number of control actuators, like variable valve timing, external EGR, charge motion control valves, etc. Calibration and control of modern SI engines with an increasing number of actuators becomes more difficult due to significantly increased actuator set point combinations for a given engine operating condition. For these applications map-based calibration and control strategies become cumbersome, since their complexity increases significantly when control actuators are added to the system. Additionally, more actuators make transient calibration and control more difficult. All of these considerations pave the way for model-based engine control algorithm development.

For SI engines, combustion phasing has a significant influence on fuel efficiency [1, 2, 3], emissions [4, 5, 6, 7], engine durability (knock and exhaust temperature) [8] and combustion variation [9]. The SI combustion process can be modeled with differing levels of complexity, from a 0-D Wiebe function [10] to 3-D CFD models with chemical kinetics [11, 12, 13, 14, 15]. For physics-based feed-forward combustion phasing control it is desirable to incorporate a model design that is robust to a wide range of inputs, so model complexity should be selected with the trade-off of predictive capability and computational power in mind. The Wiebe function approach is capable of running in real-time, but it is based on a large set of empirically determined coefficients to capture combustion characteristics over a wide range of operating conditions. 3-D CFD combustion models can be highly accurate with minimal empirical tuning, but require too much computational effort for real-time operation. Quasi-dimensional turbulent flame entrainment models

have been shown to accurately predict combustion characteristics with reduced empirical fitting (as compared to a Wiebe approach) while having lower computational complexity than 3-D CFD [16, 17, 18, 19]. Different from more complex multi-dimensional combustion models, quasi-D models do not capture flow structure and flame propagation detail, lowering computational complexity. This makes them good candidates for consideration in model-based engine combustion control applications.

SI engine combustion rate is mainly decided by turbulent flame entrainment rate, mixture laminar flame speed and engine cylinder chamber geometry [20]. Knowledge of laminar flame speed is critical for both flame entrainment and burn-up [21]. For laminar flame speed (S_L), the key factors are air-to-fuel ratio (AFR), pressure, temperature, fuel type, and residual gas fraction (RGF) [22]. S_L is very sensitive to RGF, but accurate residual gas mass (RGM)/RGF measurement and modeling in real-time is challenging [1].

Turbulent flame entrainment models generally rely on turbulence intensity, u' , to describe and approximate in-cylinder charge motion. Turbulence intensity is defined by the root-mean-squared velocity fluctuation in-cylinder, and has a primary effect on the turbulent flame entrainment process. The predicted magnitude of u' , therefore, has a significant influence on combustion phasing prediction accuracy. Direct experimental turbulence intensity measurement during combustion is difficult due to the extreme atmosphere in-cylinder [23, 24, 25, 26]. Without easily accessible empirical data to validate models, turbulence intensity predictions intended for real-time control can have limited prediction accuracy over a wide range of engine operating conditions.

This research describes models for quasi-dimensional turbulent flame entrainment combustion, semi-physical laminar flame speed and piston motion based turbulence intensity [21]. Then, a cylinder pressure based adaptation algorithm is proposed to correct the combustion calculation input models for laminar flame speed and turbulence intensity. Results of the proposed adaptation method are quantified with experimental data and conclusions are drawn.

EXPERIMENTAL SETUP AND DATA COLLECTION

A naturally-aspirated 3.6L V-6 port fuel injected test engine is utilized for this research. The engine is equipped with oil-driven dual-independent valve phasing systems on both banks. The camshaft phasers shift a fixed valve lift profile relative to the crankshaft (i.e. the valves have a fixed duration and lift). A pent-roof shaped combustion chamber contains two intake and two exhaust valves per cylinder. A summary of basic engine geometry is provided in [Table 1](#).

An AC engine dynamometer is used for the experimental portion of this research. Crank angle resolved data acquisition is performed using an AVL-671 system. Cylinder pressures are measured using AVL GH12D piezoelectric sensors. Piezoresistive Kulite sensors are used for dynamic pressure measurements in both the intake and exhaust ports of the test engine. The data are sampled in 0.5 crank angle degree intervals.

Table 1. Engine parameters

Fuel	Gasoline (87 Pump Octane)
Compression Ratio	10.2
Bore	96 mm
Stroke	83 mm
Connecting Rod Length	156.5 mm
Intake Valve Diameter	39 mm
Exhaust Valve Diameter	30 mm

COMBUSTION AND INPUT MODEL DESCRIPTIONS

A quasi-dimensional turbulent entrainment combustion model is used for this research, which was originally proposed by Blizard and Keck [16] and then refined by Tabaczynski et al. [27]. This turbulent flame entrainment based combustion model assumes that the fresh mixture at the flame front is (1) entrained into small eddies and then (2) burned up in a characteristic time. Based on these assumptions, the flame entrainment and burned up processes are shown below as [Equation 1](#) and [2](#) respectively.

$$\frac{dm_e}{dt} = \rho_{unburned} A_{flame} (u' + S_L) \quad (1)$$

$$\frac{dm_b}{dt} = \frac{m_e - m_b}{\tau} + \rho_{unburned} A_{flame} S_L \quad (2)$$

[Equation 1](#) describes the unburned mass entrainment rate at the flame front. It is assumed the flame propagates through unburned charge along Kolmogorov scale vortices entraining turbulent eddies. The unburned mass entrainment rate is determined by unburned mixture density, flame front area, laminar flame speed and turbulence intensity. After unburned mixture entrainment, mass burn-up occurs at a rate described by [Equation 2](#). Burn-up occurs at a characteristic time, τ , which is defined as the time to burn up an eddy at laminar flame speed. The eddy size is assumed to be Taylor microscale (λ) [28] and can be calculated by [Equation 3](#).

$$\frac{\lambda}{L} = \sqrt{\frac{15}{\xi}} \left(\frac{u' L}{\nu} \right)^{-\frac{1}{2}} \quad (3)$$

The integral length scale, L , is utilized during the burn-up process. Prior to start of combustion, L is assumed to be the instantaneous chamber height [29]. After start of combustion the unburned mixture is compressed at a rate based on the rapid distortion theory [30]. The rapid distortion theory is applied when the timescale of turbulence distortion is much shorter than large eddy turnover or decay.

timescales [31]. The characteristic length scale calculation is shown as Equation 4. L_o and ρ_{uo} are defined as the instantaneous chamber height and unburned gas density at start of combustion.

$$L = L_o \left(\frac{\rho_{uo}}{\rho_{unburned}} \right)^{\frac{1}{3}} \quad (4)$$

In Equation 3 ν is defined as kinematic viscosity and can be calculated by dividing dynamic viscosity, μ , by gas density. Dynamic viscosity of the unburned mixture is assumed equal to that of air [21] and is described as a function of unburned mixture temperature, as shown in Equation 5.

$$\mu = (3.3 \times 10^{-7}) T_{unburned}^{0.7} \quad (5)$$

Unburned gas density

After start of combustion, the mixture in-cylinder can be separated into burned and unburned zones. According to Heywood, unburned gas density is approximately four times that of the burned gas [21]. Based on this assumption, unburned gas density can be calculated by Equation 6.

$$\rho_{unburned} = \frac{m_{total} + 3m_b}{V} \quad (6)$$

Flame Front Area

For this research the flame is assumed to propagate spherically from the spark plug gap through the unburned mixture. The flame will contact the cylinder head, piston and cylinder walls with increasing burned gas radius. Based on geometric calculations, the flame front area is defined by Equation 7.

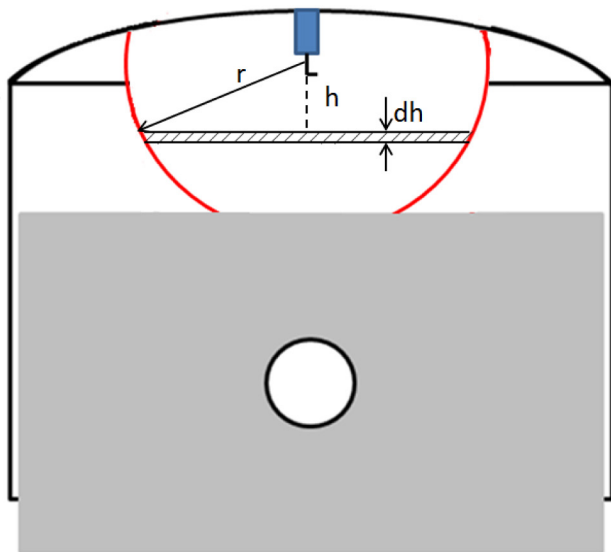


Figure 1. Flame geometry and contact with cylinder

$$A_{flame} = \int 2\pi\sqrt{r^2 - h^2} \cdot dh \quad (7)$$

As shown in Figure 1, r is the flame radius and h is the distance to the flame center. Flame front area can be calculated by integrating circular band areas with height dh . In this research, flame thickness is assumed 3mm and the flame radius r can be calculated from burned gas radius r_b plus flame thickness. Burned gas radius has a relationship with burned gas volume as shown in Equation 8. Burned gas volume is calculated from burned mass and burned mixture density.

$$V_b = \int \pi(r_b^2 - h^2) \cdot dh \quad (8)$$

LAMINAR FLAME SPEED MODEL

Laminar flame speed is an important intrinsic property of a combustible fuel, air, and burned gas mixture. It is defined as the velocity, relative to and normal to the flame front, with which unburned gas moves into the front and is transformed to the products under laminar flow conditions [21]. Under pressures and temperatures that are the common at engine operation conditions, laminar flame speed is measured in spherical closed vessels by propagating a flame radially outward from the vessel center. The laminar burning velocity is then given by Equation 9.

$$S_L = \frac{dm_b/dt}{A_{flame}\rho_{unburned}} \quad (9)$$

Laminar flame speed is defined as a function of unburned mixture thermodynamic properties and composition [21]. Equation 10 shows the semi-physics-based laminar flame speed power law model. In the model, the reference laminar flame speed $S_{L,0}$ is defined by the Equation 11. It is related to fuel properties and air-to-fuel ratio. Residual gas effects are accounted for in the final step of flame speed calculation, as in Equation 12. Table 2 provides fuel property parameters for gasoline.

$$S_L = S_{L,0} \left(\frac{T_{unburned}}{T_0} \right)^\alpha \left(\frac{p}{p_0} \right)^\beta \quad (10)$$

Where $\alpha = 2.4 - 0.271\phi^{3.51}$, $\beta = -0.357 + 0.14\phi^{2.77}$

$$S_{L,0} = B_m + B_\phi(\phi - \phi_m)^2 \quad (11)$$

$$S_L(x_b) = S_L(x_b = 0) (1 - 2.06x_b^{0.77}) \quad (12)$$