

A Real-Time Model for Spark Ignition Engine Combustion Phasing Prediction

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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 large number of control actuators encourages the investigation of physics-based control strategies to reduce calibration time and complexity. Of particular interest is spark timing control and calibration since it has a significant influence on engine efficiency, emissions, vibration and durability. Spark timing determination to achieve a desired combustion phasing is currently an empirical process that occurs during the calibration phase of engine development. This process utilizes a large number of stored surfaces and corrections to account for the wide range of operating environments and conditions that a given engine will experience. An obstacle to realizing feedforward physics-based combustion phasing control is the requirement for an accurate and fast combustion model. In this research, a quasi-dimensional turbulent flame entrainment combustion model for the purpose of real-time combustion phasing prediction is proposed. The proposed algorithm utilizes existing engine sensors and calculates combustion phasing from a physics based model in real-time. Transient engine dynamometer results using a rapid-prototype engine controller demonstrate a prediction accuracy for the location of fifty percent mass fraction burned (CA50) within 3.6 crank angle degrees.

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INTRODUCTION

Fuel economy and emissions regulations result in spark-ignition (SI) engines that incorporate a large number of control actuators. Calibration and control of these engines with becomes more difficult due to a significantly increased number of actuator set point combinations at a given engine operating condition. Map-based calibration and control strategies become cumbersome for these applications, since their size increases exponentially with degrees of control freedom.

Spark timing has a significant influence on fuel efficiency, torque and emissions so its accurate calibration is of critical importance to overall system performance. However, the calibration of spark timing is particularly difficult using a map-based control strategy since it is the final actuator scheduled prior to combustion. Therefore, it must incorporate compensations for all other actuator set points and environmental conditions. These considerations necessitate model-based engine control strategies that can generate desired spark timing in real-time with decreased calibration effort.

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]. Combustion phasing is generally defined by the crank angle location at which fifty percent of the in-cylinder charge is burned which is

commonly referred to as CA50. Spark timing is the actuator used to control combustion phasing in SI engines. In order to select a proper spark timing to meet the desired combustion phasing, an accurate combustion model is needed. The combustion model is required to estimate the crank angle duration between a given spark timing and the desired combustion phasing (CA50).

The combustion process of SI engines 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]. The combustion model constructed for control purposes needs to balance between model fidelity and computational complexity. Combustion modeling using the Wiebe function approach is capable of real-time operation, but it requires a large set of empirically determined coefficients to function in a wide range of operating conditions. 3-D CFD combustion models can be highly accurate with minimal empirical calibration, but they require excessive computational effort for control applications. Previous papers demonstrated that the quasi-dimensional turbulent flame entrainment models can accurately predict combustion characteristics with reduced empirical fitting (as compared to a Wiebe approach) while having lower computational complexity than 3-D CFD methods [16, 17, 18, 19]. As opposed to more complex multidimensional combustion models, quasi-D models do not capture flow structure and flame propagation detail, reducing computational complexity which makes them good candidates for model-based engine control applications.

Quasi-dimensional combustion models for SI engines generally utilize inputs for in-cylinder turbulence levels, laminar flame speed and engine cylinder chamber geometry [20]. This research incorporates a quasi-dimensional turbulent entrainment combustion model originally developed by *Blizard* and *Keck* [16], and later refined by *Tabaczynski et al.* [29]. This turbulent flame entrainment based combustion model requires several inputs, the most critical of which are turbulence intensity and laminar flame speed.

Estimating and/or modeling laminar flame speed and turbulence intensity is important for accurate burn rate prediction. Power-law models for laminar flame speed (S_L) that incorporate air-to-fuel ratio (AFR), pressure, temperature, fuel type, and residual gas fraction (RGF) [22] are commonly used. Direct experimental turbulence intensity measurement during combustion is difficult due to the extreme in-cylinder atmosphere [23, 24, 25, 26]. Simplified turbulence intensity prediction models have been developed that are capable of real-time prediction, with the primary input being mean piston speed [37]. Without easily accessible empirical data to calibrate these models they rely on estimations from experimental data for simulations.

In this research, a quasi-dimensional turbulent flame entrainment combustion model for the purpose of real-time combustion phasing prediction is proposed. The proposed algorithm utilizes existing engine sensors and calculates combustion phasing from a physics based model in real-time. The experimental setup is first explained, along with the rapid-prototype control system. The base combustion model is then introduced, followed by a discussion of input models. Input models for laminar flame speed, turbulence intensity, cylinder pressure prediction, and a “Grey-Box” flame kernel development model are described. Real-time steady-state and transient results are then used to quantify the accuracy of the proposed algorithm and conclusions are drawn.

EXPERIMENTAL SETUP AND DATA COLLECTION

A naturally-aspirated 3.6L V-6 port fuel injected engine is utilized for this research. The engine is equipped with oil-driven dual-independent valve phasing systems on both cylinder 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 valves and two exhaust valves per cylinder. A summary of basic engine geometry is provided in [Table 1](#).

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

An AC engine dynamometer is used for the experimental portion of this research. Crank angle resolved data acquisition is performed using an AVL-671 IndiMaster system. Cylinder pressures are measured with passage-mounted 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. Measured cylinder pressures are also sent to a Cylinder Pressure Development Controller (CPDC) unit to compute CA50 [28] in real-time. Control models are tested in real-time using a rapid-prototype ETAS ES910 system overriding the stock ECU.

COMBUSTION MODEL AND ALGORITHM DESCRIPTION

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.* [29]. 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 in [Equations \(1\) and \(2\)](#).

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

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

[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.

[Figure 1](#) summarizes the information flow of the proposed combustion phasing model. The flame kernel development and residual gas models need to be run once per engine cycle. The combustion model generates burned mass, which is sent to the cylinder pressure model to calculate heat release. Finally, the computed cylinder pressure is sent back to combustion model. The cylinder pressure and combustion models formulate an ODE system that needs to be solved numerically in the crank angle domain.

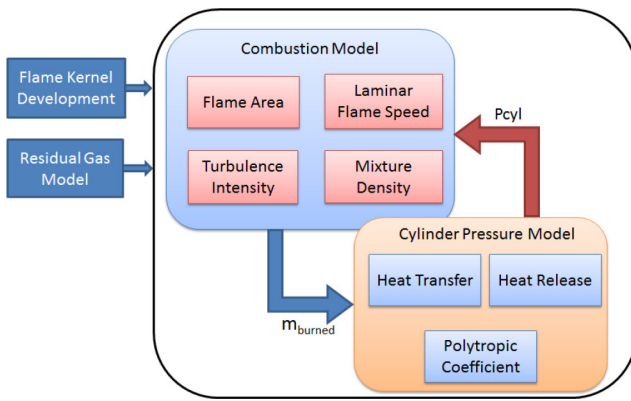


Figure 1. Layout of the combustion phasing prediction algorithm

COMBUSTION INPUT MODELS

This combustion model requires several inputs, all of which require separate models. Input models for laminar flame speed, turbulence intensity, cylinder pressure prediction, and a “Grey-Box” flame kernel development are of primary importance to the proposed algorithm. The following sections discuss the modeling process of all combustion model inputs in detail.

Taylor Microscale λ and Integral Length Scale L

Turbulence length scales are used to quantify the physical size of key structures within the flame. The eddy size is assumed to be Taylor microscale (λ) [30] 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 used to describe the largest possible eddy size in the combustion chamber, and is initiated as the instantaneous chamber height [31]. After start of combustion the unburned mixture is compressed at a rate based on the rapid distortion theory [32]. The rapid distortion theory is applied when the timescale of turbulence distortion is much shorter than large eddy turnover or decay timescales [33]. The characteristic length scale calculation based on the rapid distortion assumption is shown in Equation (4).

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

Where:

L_o is instantaneous chamber height

ρ_{uo} is unburned gas density at start of combustion.

In Equation (3), ν is defined as kinematic viscosity, which can be calculated by dividing dynamic viscosity, μ , by gas density [37]. 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 [21], unburned gas density is approximately four times that of the burned gas. 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 consequently with increasing burned gas radius. Figure 2 shows the concept of this flame propagation process.

Based on geometric calculations, the flame front area can be computed by integrating circular band areas with height dh , as shown in Equation (7).

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

In this research, flame thickness is assumed to be 3mm which yields the best combustion phasing prediction accuracy for the test engine. 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 described in Equation (8). Burned gas volume is calculated from burned mass and burned mixture density.

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

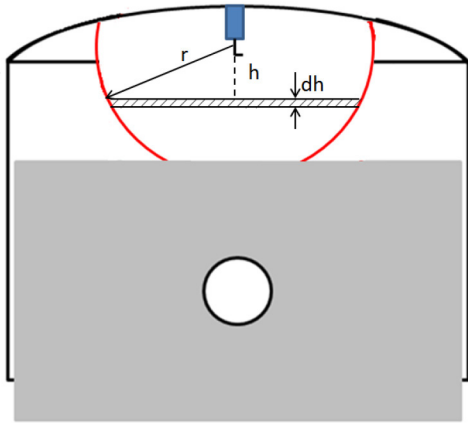


Figure 2. Flame geometry and contact with cylinder

Laminar Flame Speed S_L

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]. Laminar flame speed is measured in spherical closed vessels by propagating a flame radially outward from the vessel center. The laminar burning velocity can be calculated with Equation (9).

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

Laminar flame speed can be estimated with unburned mixture thermodynamic properties and composition [21]. Equation (10) shows the semi-empirical laminar flame speed power law model.

$$S_L = S_{L,0} \left(\frac{T_{unburned}}{T_0} \right)^\alpha \left(\frac{P_{cyl}}{P_0} \right)^\beta \quad (10)$$

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

In this model, the reference laminar flame speed, $S_{L,0}$, is defined by Equation (11), and 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 shown in Equation (12).

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

$$S_L = S_{L,0}(1 - 2.06RGF^{0.77}) \quad (12)$$

Table 2 provides fuel property parameters for gasoline.

Table 2. Laminar flame speed, S_L , property parameters for gasoline [21]

Fuel	ϕ_m	B_m	B_ϕ
Gasoline	1.21	30.5	-54.9

Residual gas fraction (RGF) is a critical input to the laminar flame speed calculation in Equation (12). RGF is calculated from engine air mass flow and residual gas mass (RGM). While most modern IC engines control systems have accurate cylinder charge estimation for fuel injection control, RGM estimation is not commonly available.

This research utilizes a semi-physical RGM model based on previous research from Wang et al [35]. Their model is a modified version of the semi-empirical residual fraction mass (RGM) model originally developed by Fox [43]. These models separate RGM into two parts; 1) from trapped residual at exhaust valve closing (EVC) due to unswept cylinder volume, and 2) exhaust gas backflow into the cylinder and intake runner during the valve overlap period. Terms $\Delta P_{e_{OL}}$ and $\Delta P_{i_{OL}}$ were added to account for wave tuning dynamics to the original Fox model, as shown in Equation (13).

$$\begin{aligned} RGM &= C_1 \cdot \sqrt{\left(\frac{P_e}{RT_e} \right) \cdot \left((P_e + \Delta P_{e_{OL}}) - (P_i + \Delta P_{i_{OL}}) \right)} \\ &\cdot A_{flow} \cdot \frac{OLV}{N} + C_2 \cdot \left(\frac{P_e}{RT_e} \right) \cdot V_c \end{aligned} \quad (13)$$

Where:

C_1 and C_2 are experimental calibrated constants.

P_e and P_i are average exhaust and intake pressure

$\Delta P_{e_{OL}}$ and $\Delta P_{i_{OL}}$ are exhaust and intake pressure difference caused by wave tuning dynamic.

A_{flow} is valve flow area

OLV is valve overlap volume

N is engine speed

T_e is exhaust temperature

Unburned gas temperature is another critical input for laminar flame speed. In this research, the unburned mixture temperature is calculated based on the ideal gas law. The equation is shown below:

$$T_{unburned} = \frac{p \cdot V_{unburned}}{m_{unburned} \cdot R_{unburned}} \quad (14)$$

Where:

$R_{unburned} = 287 \text{ J/(kg} \cdot \text{K)}$ is the specific gas constant of unburned gas mixture.

Cylinder Pressure Calculation

Cylinder pressure is a critical input for combustion phasing prediction since it is used to calculate laminar flame speed and cylinder temperature. Since this research develops a feed-forward prediction of combustion phasing, cylinder pressure sensors are not utilized.