ABSTRACT
An experimental study is performed to investigate the combustion characteristics of an ethanol-gasoline, dual fueled, single cylinder spark ignition (SI) engine. A dual fuel injection system with both Direct-Injection (DI) and Port-Fuel-Injection (PFI) is used in this work. The performance of PFI-E85 and DI-gasoline, and PFI-gasoline and DI-E85 systems is presented. E85 is a blend of 85% ethanol and 15% gasoline by volume. In each test, the percentage of E85 is varied from 100 (0% gasoline) to 0 (100% gasoline) to compare the various cases. PFI-gasoline and DI-gasoline (PFI & DI-gasoline) results are also presented to provide a baseline for comparison. The cycle-to-cycle variability is presented using coefficient of variation (COV) of indicated mean effective pressure (IMEP). Mass fraction burned (MFB) and burn duration are determined from the analysis of measured in-cylinder pressure data. The well known Rassweiler and Withrow method (Model 1), with a new linear model for the polytropic index, is used to obtain the MFB curves. The differences are presented for the net pressure method (Model 2) to evaluate the burn rates. It is found that combustion is faster with the increase in PFI percentage for all the three setups with dual fuel injection. The PFI-E85 and DI-gasoline system showed that the burn duration decreases significantly with the increase in PFI percentage; however, the PFI-gasoline and DI-E85 system showed only slight differences with the increase in PFI percentage. Model 2 showed good agreement with Model 1 at high load conditions; however, it predicts slower combustion at light load conditions.

1. INTRODUCTION
Improvement in fuel efficiency and reduction in exhaust emissions are the two main driving forces behind the new developments in internal combustion (IC) engines. In addition, the increasing interest in using renewable energy sources as alternative fuels has prompted research into the nature of combustion when these fuels are utilized. Ethanol is one of these fuels. Ethanol has a higher octane number than gasoline and thus the potential for higher efficiency, by increasing compression ratio, exists. However, its lower heating value leads to fewer driving miles per gallon when it is used as a direct replacement in current gasoline engines. Also, vehicle modifications are required to operate the gasoline vehicles on high ethanol content blended fuels due to its poor startability in cold weather and the requirement of ethanol corrosion resistant materials for the fuel system. Therefore, the blend of ethanol (in gasoline) is more widely used in the vehicles that are designed to operate on gasoline fuel. This reduces the consumption of fossil fuels, and due to the high latent heat of evaporation of ethanol, combustion temperature decreases compared to the vehicles operating on pure gasoline. This enables higher torque and higher thermal efficiency due to reduced cooling heat loss [1]. In United States, the blend of 10 percent ethanol in gasoline is widely used in many fuel blends. In Brazil, the ethanol percentage is even higher and this is already in use for several years [2]. Ethanol can be produced from renewable energy sources such as corn, wheat, and sugar cane.
Several studies have been performed to investigate the effect of ethanol fuel in IC engines. Starkman et al. [3] provided the comparison between various alcohol and hydrocarbon fuels. Brinkman [2] studied the effect of equivalence ratio and compression ratio (CR) on efficiency and exhaust emissions of an IC engine operating on ethanol fuel. Salih and Andrews [4] studied the influence of gasoline-ethanol blends on emissions and fuel economy in a spark ignition engine. Their results showed significant reduction in nitrogen oxides (NOx) and CO emissions using ethanol blended fuels. The unburned hydrocarbons emissions with ethanol blends were found to be higher than for 100 % gasoline. Nakata et al. [1] also investigated the influence of ethanol fuel on SI engine performance, thermal efficiency, and emissions. Bromberg et al. [5] studied the case with DI ethanol and PFI gasoline where ethanol was only used to reduce the engine knock; however, the study of ethanol fuel in IC engines was not extended to optimize the combination of both PFI and DI fuel injection systems.

The objective of this paper is to investigate the combustion characteristics of an ethanol-gasoline, dual fueled, single cylinder SI engine. DI and PFI injections are implemented to study the engine performance with various percentages of injected fuels. Experiments are performed at engine speeds of 1500 and 2500 rpm with different load conditions. In the following sections, a brief description of the experimental setup is first provided, followed by the measurement procedure, and then the results of various test runs are presented. Finally concluding remarks are summarized from this work.

2. EXPERIMENTAL SETUP

![Fig. 1: Experimental rig](image)

Figure 1 shows the experimental setup of the dual fueled spark ignition engine used for this study. The engine used for the experiments is a single cylinder Ford 5.4 L 3-valve engine with two intake and one exhaust valves. It has a bore diameter of 90.17 mm and a stroke length of 105.66 mm. Both Port-Fuel-Injection (PFI) and Direct-Injection (DI) are used for each test measurement. Details of seven different tests performed for PFI-E85 & DI-gasoline, PFI-gasoline & DI-E85, and PFI & DI-gasoline systems are shown in Table 1. E85 in this work refers to a blend of 85 % ethanol and 15 % gasoline by volume. Experiments are performed at different loads, 3.3 bar IMEP and wide open throttle (WOT), and engine speeds, 1500 and 2500 rpm. For each test measurement, five different cases are considered with PFI percentages of 100, 70, 50, 30 and 0. The percentage of DI is correspondingly increased to maintain a constant relative air-to-fuel ratio (λ), inverse of fuel-to-air equivalence ratio (φ). For each case, the experiments started at zero DI fuel injection, and then the DI fuel injection was increased to the desired percentage while maintaining the air-to-fuel ratio (λ). Note that when certain percentage of E85 is used, the air-to-fuel ratio (λ), which is to be constant, is measured by the oxygen sensor. Pressure measurements inside the engine cylinder are taken for 300 consecutive cycles with one degree of crank angle resolution. The measurement range for the pressure transducer was from 0 to 250 bar. The uncertainties due to environmental variations are minimized by taking measurements for different tests in a single day with constant engine coolant temperature.

<table>
<thead>
<tr>
<th>A. PFI-E85 &amp; DI-gasoline</th>
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<tr>
<td>Test No.</td>
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<td>A1.</td>
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<td>A2.</td>
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<th>B. PFI-gasoline &amp; DI-E85</th>
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<tr>
<td>B1.</td>
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<td>B2.</td>
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<th>C. PFI &amp; DI-gasoline</th>
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<tr>
<td>C1.</td>
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<tr>
<td>C2.</td>
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Tab. 1: Test matrix

3. MEASUREMENT PROCEDURE

The primary purpose of combustion in a piston engine is to generate the pressure for shifting the expansion process away from the compression process and produce the work cycle. IMEP is an important parameter to measure the performance of this work producing cycle that transforms heat from the combustion process into mechanical work. The measured pressure profile is used to calculate the IMEP for each cycle, Equation 1, and thus the mean is obtained over N consecutive cycles (300 cycles in this work) for each case.

\[
IMEP = \frac{\int pdV}{V_o}
\]
where \( V_d \) is the displaced volume. Work delivered to the piston over the entire cycle is obtained using the trapezoid rule. Cyclic variability is presented using coefficient of variation (COV) in indicated mean effective pressure, \( COV_{\text{imep}} \). It defines the cyclic variability in terms of indicated work per cycle. This is obtained using Equation 2, [6], where \( \sigma \) is the standard deviation.

\[
COV_{\text{imep}} = \frac{\sigma_{\text{imep}}}{\text{IMEP}} \times 100
\]

### 3.1 MASS FRACTION BURNED (MFB) ANALYSIS

MFB is a measure of fraction of energy released, due to combustion of fuel inside an engine cylinder, with respect to the total energy released at the end of combustion during a cycle. Several methods have been suggested to evaluate the MFB in gasoline engines from the measured pressure data. Brunt and Emtage [7] compared the performance of five alternative MFB models using simulated and experimental data. They preferred the Rassweiler and Withrow model to produce the best results to determine the mass fraction burned in their comparative tests. Shayler et al. [8] investigated the best form of signal noise are important parameters to calculate the mass fraction burned using Rassweiler and Withrow method.

This work utilizes the Rassweiler and Withrow method, denoted as Model 1 in this paper, with a new linear model for polytropic index to evaluate the MFB curves. Start of combustion (SOC) and end of combustion (EOC) are determined from the logarithmic pressure indicator diagram using a least-square fit algorithm. Comparison is presented for another analysis model, denoted as Model 2, which utilizes the net pressure to evaluate the MFB curves as discussed in Section 3.1.2. Model 2 has an advantage that the data processing time is short enough to allow for on-line processing.

#### 3.1.1 Model 1, Rassweiler and Withrow method

The implementation details of this method by Rassweiler and Withrow [9] vary in literature. The data depicted in Figure 2 are provided for the measured pressure data in Test C2 with 100 % PFI-gasoline. In this figure, zero crank angle degree (CAD) represents the piston position at top dead center (TDC). The initial state, \( i \), and the final state, \( f \), are the essential singularities of the combustion process which represent the start and end of combustion, respectively. The transition from \( i \) to \( f \) is referred to as the dynamic stage of combustion, Bitar et al. [10]. In this model, it is assumed that for any crank angle interval, \( \Delta \theta \), the actual pressure change, \( \Delta p \), inside an engine cylinder is composed of a pressure rise due to combustion, \( \Delta p_c \), and a pressure change due to the volume change, \( \Delta p_v \), therefore \( \Delta p = \Delta p_c + \Delta p_v \). The pressure change due to volume change can be obtained using a polytropic index. A constant index value was chosen originally by Rassweiler-Withrow where the magnitude was the average of the polytropic index before and after the combustion process. Brunt and Emtage [7] used the compression index, \( n_c \), up to TDC and the expansion index, \( n_e \), thereafter. A new linear model for polytropic index, \( n_p \), during the combustion process is introduced in this work. Therefore the pressure change due to volume change in any interval, \( \Delta \theta \), in the absence of combustion, can be obtained using Equation (3).

\[
\Delta p_v = p_e - p_{e-1} = p_{e-1}\left(\frac{V_{e-1}}{V_e}\right)^{n_p} - 1
\]

An indicator diagram in logarithmic scales, Figure 3, is used to evaluate the polytropic indexes, \( n_c \) and \( n_e \). \( V_i \) in the figure represents the normalized volume with respect to the clearance volume. The two states, \( i \) and \( f \), are identified by the departure of the curve from the state lines representing the compression and expansion processes using a least-square fit algorithm. Now, with the known \( \Delta p_p = \Delta p - \Delta p_v \), and assuming that the mass of charge burned in the interval, \( \Delta \theta \), is proportional to the pressure rise due to combustion, the MFB at the end of the \( k \)th interval can be evaluated using Equation 4,
\[
\sum_{k=p}^{k} \frac{m_{b_k}}{m_{b_{ref}}} \approx \sum_{k=0}^{N} \Delta p
\]

where N is the total number of crank angle interval during the combustion process.

### 3.1.2 Model 2, Net Pressure method

This model evaluates the mass fraction burned by normalizing the net pressure with respect to the overall net pressure increase at the end of combustion, Zhu et al. [11]. The net pressure change, \( \Delta p(k) \), between the two crank angles is:

\[
\Delta p(k) = \left\{ p_k + p_1 \left( \frac{V(k)}{V(k+1)} \right)^{1/n} \right\} \frac{V(k)}{V_{Ik}}
\]

and the net pressure at each crank angle is

\[
p_{NET}(k) = p_{NET}(k-1) + \Delta p(k)
\]

where \( p \) is the pressure, \( V \) is the volume, and \( V_{Ik} \) is the chamber volume at the ignition point. The simplicity of Model 2 is apparent from Equations 5 and 6. It is computationally efficient and due to the short data processing time, it can be used for online processing.

**Figure 4:** MFBs for PFI & DI-gasoline system (PFI-gasoline 100 %) at WOT (top) and 3.3 bar IMEP (bottom)

Model 2 is also in good agreement with Model 1 when MFB is evaluated at higher load condition; however, it predicts slower combustion at light load condition. It is found that the accurate identification of EOC is an important measure to minimize the uncertainties associated with the MFB calculations. EOC in Model 1 is determined accurately using least-square fit algorithm; however, in Model 2, it is determined by locating the peak of the net pressure. At high load condition, Model 2 determines EOC with a reasonable accuracy and therefore the results are in close agreement with Model 1. At part load condition, this method is less than ideal to determine the EOC in Model 2. It predicts EOC significantly later than the actual CAD value and therefore Model 2 predicts slower combustion compared to Model 1. Also, the constant polytropic index, \( n = 1.3 \) (in Model 2), is in better agreement at WOT condition \((n_c = 1.34 \) and \( n_e = 1.37)\) compared to the part load condition \((n_c = 1.17 \) and \( n_e = 1.20)\). In order to distinguish the effect of polytropic index variation and EOC determination, Figure 4 (bottom) with pink dots shows the MFB curve calculated using Model 2 with index \( n = n_e = 1.20 \) instead of constant polytropic index, \( n = 1.3 \), used in Equation 5. As expected, the difference reduces with respect to Model 1. We believe that the rest of the difference is due to the unique feature of evaluating EOC in Model 1 which is an important factor for MFB calculations.

### 4. EXPERIMENTAL RESULTS

Combustion characteristics of ethanol-gasoline dual fueled injection system in a single cylinder SI engine are presented. Figure 5 shows the mean IMEP (top) and COV\(_{\text{imep}}\) (bottom) for the various test data. As shown in the figure, the mean IMEP is almost constant for DI-E85 & PFI-gasoline system for different cases with the decrease in DI-E85 (or increase in PFI-gasoline); however, it increases with the increase in PFI-E85 for PFI-E85 & DI-gasoline system at similar load conditions. The mean...
IMEP is about 7.5 bar for PFI-E85 (100 %) & DI-gasoline (0 %) system when engine is operated at 1500 rpm with WOT. At similar operating conditions, the mean IMEP for DI-E85 & PFI-gasoline is about 9 bar. At 2500 rpm engine speed with WOT, the mean IMEP increases from about 10 to 10.92 bar with the increase in E85 percentage (0 to 100) for PFI-E85 & DI- gasoline system. The COV of IMEP decreases with the increase in PFI percent (beyond 50 %) for each test point (Tests A and B in Table 1). The cyclic variability has the peak either at 30 or 50 % of PFI. 100 % PFI shows the minimum cyclic variability for each test, and the COV of IMEP is even less than 1 for PFI-E85 & DI-gasoline system when the engine is operated at 1500 rpm with WOT condition.

4.1 PFI-E85 and DI-Gasoline System

This section describes the combustion characteristics of ethanol-gasoline dual fueled injection system in a single cylinder SI engine where PFI system is used for E85 injection and DI system is used for gasoline injection, (PFI-E85 & DI-gasoline). Tests A1, A2 & A3 in Table 1 represent these measurements at 3.3 bar IMEP and WOT load conditions with engine speeds of 1500 and 2500 rpm. Five different cases are considered for each test with PFI E85 percentages of 100, 70, 50, 30 and 0. The percentage of DI gasoline is correspondingly increased to maintain a constant relative air-to-fuel ratio (λ) for each case.

![Figure 6](http://example.com/fig6.png)

Fig. 6: Averaged Pressure (top) and MFB curves (bottom) for PFI-E85 & DI-gasoline system at 3.3 bar IMEP and 1500 rpm

Figure 6 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test A1 (3.3 bar IMEP, 1500 rpm). Three different cases are shown with PFI E85 percentages of 0, 70 and 100. As shown in the figure, the peak pressure increases with the increase in PFI-E85. This is expected due to lower combustion gas temperature with ethanol than gasoline. Based on this, it can be assumed that the cooling heat loss from the combustion chamber decreases with ethanol. Similarly, the burn duration decreases with the increase in PFI-E85. Ethanol has an advantage of high combustion speed for SI engines [2]. Combustion process starts at about -20 CADs for all the cases. There is no significant difference in 10 % burn durations; however, 50 % and 90 % burn durations are much faster with 100 % PFI-E85 when compared to the other cases. The burn duration is about 51 CADs for 100 % PFI-E85 and increases with the decrease in PFI-E85. The difference of Model 2 results when compared to Model 1 is presented using vertical bars in the figure. The vertical bars in the downward direction indicate that Model 2 predicts slower combustion when compared to Model 1 and the difference increases with the decrease in PFI percentage. Model 2 determination of EOC is significantly later than the actual value and therefore it predicts slower combustion compared to Model 1.

![Figure 7](http://example.com/fig7.png)

Fig. 7: Averaged Pressure (top) and MFB curves (bottom) for PFI-E85 & DI-gasoline system at WOT and 1500 rpm

Figure 7 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test A2 (WOT, 1500 rpm) with PFI E85 percentages of 0, 70 and 100. Similar to Test A1 results, the peak pressure increases with the increase in PFI E85. It is interesting to note that the peak pressure is about 44.5 bar with 100 % PFI-E85 and then decreases sharply with the decrease in PFI-E85. Also, there is some phase shift in the peak pressure locations (CADs) due to fixed spark timing considered for the test data. As expected, the burn duration increases with the decrease in PFI-E85. At this load condition, combustion starts at about -12 CADs for all the cases and significant difference can be observed even at earlier stages (10 % burn) with much faster combustion for 100 % PFI-E85. Similarly, 50 % and 90 % burn durations are also less for 100 % PFI-E85 when compared to the other cases. The burn duration is about 39

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CADs for 100 % PFI-E85 and increases with the decrease in PFI E85. Model 2, at 100 % PFI-E85, predicts slightly faster combustion than Model 1. It is in good agreement when PFI-E85 is 70 % and assumes slightly slower combustion when PFI-E85 is 0 %. Overall, Model 2 is in good agreement with Model 1 at this higher load condition.

Figure 8 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test A3 (WOT, 2500 rpm) with PFI E85 percentages of 0, 70 and 100. Similar to Tests A1 and A2, the peak pressure increases with the increase in PFI, and it is as high as 55 bar with 100 % PFI-E85. The burn duration is about 37 CADs for 100 % PFI. It increases with the decrease in PFI-E85; however, no significant difference can be observed when PFI-E85 is 70 and 0 %. Also, Model 2 shows good agreement with Model 1 at this high load and higher engine speed condition.

4.2 DI-E85 and PFI-Gasoline System

This section describes the combustion characteristics of ethanol-gasoline dual fueled injection system in a single cylinder SI engine where DI system is used for E85 injection and PFI system is used for gasoline injection, (DI-E85 & PFI-gasoline). Tests B1 and B2 represent these measurements at 3.3 bar IMEP and WOT load conditions, respectively. Five different cases are considered for each test with PFI gasoline percentages of 100, 70, 50, 30 and 0. The percentage of DI E85 is correspondingly increased to maintain a constant relative air-to-fuel ratio for each case.

Figure 9 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test B1 (3.3 bar IMEP, 1500 rpm). Only two cases with DI E85 percentages of 0 and 100 are shown due to less variation in measured pressure and MFB curves. As shown in the figure, there is no significant difference in measured pressure profiles at this part load condition; however, it is interesting to note that the peak pressure increases slightly with the decrease in DI-E85. Similar to the pressure profiles, no significant difference is observed in burn durations. The burn duration is about 58 CADs for 0 % DI-E85 and increases only slightly with the increase in DI-E85. Similar to the results shown in Figure 6 (PFI-E85 & DI-gasoline system with part load condition) Model 2 predicts slower combustion and the difference with respect to Model 1 increases with the decrease in PFI percentage.

Fig. 8: Averaged Pressure (top) and MFB curves (bottom) for PFI-E85 & DI-gasoline system at WOT and 2500 rpm

Fig. 9: Averaged Pressure (top) and MFB curves (bottom) for DI-E85 & PFI-gasoline system at 3.3 bar IMEP and 1500 rpm

Fig. 10: Averaged Pressure (top) and MFB curves (bottom) for DI-E85 & PFI-gasoline system at WOT and 1500 rpm
Figure 10 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test B2 (WOT, 1500 rpm) with DI-E85 percentages of 0 and 100. This figure can be compared with Figure 7 (PFI-E85 & DI-gasoline system) with similar load conditions. The peak pressure is slightly higher for 0 % DI-E85, and it is as high as 46 bar. Similar to the part load condition, Figure 9, there is no significant difference in burn durations, and it is about 40 CADs for 0 % DI-E85. Model 2 results at this high load condition are in good agreement with Model 1 due to accurate determination of EOC.

4.3 PFI-Gasoline and DI-Gasoline System

This section describes the combustion characteristics of dual injection system where gasoline is used for both PFI and DI. Tests C1 and C2 represent these measurements at 3.3 bar IMEP and WOT load conditions, respectively. Five different cases are considered for each test with PFI gasoline percentages of 100, 70, 50, 30 and 0. The percentage of DI gasoline is correspondingly increased to maintain a constant relative air-to-fuel ratio for each case.

Figure 11 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test C1 (3.3 bar IMEP, 1500 rpm). Three different cases with PFI gasoline percentages of 0, 50 and 100 are shown in the figure. The peak pressure increases with the increase in PFI percentage, and it is about 15 bar for 100 % PFI; however, no significant difference can be observed in burn durations. Combustion process starts at about -18 CADs (all cases) and the burn durations are 61, 63 and 65 CADs for 100, 50 and 0 % PFI-gasoline, respectively. As seen earlier, Model 2 has slower combustion, compared to Model 1, when engine is operated at 3.3 bar IMEP.

Figure 12 shows the averaged pressure profiles (top) and MFB curves (bottom) for Test C2 (WOT, 1500 rpm) with PFI gasoline percentages of 0, 50 and 100. Similar to Test C1, the peak pressure increases with the increase in PFI, and it is as high as 48 bar with 100 % PFI. The burn duration is slightly shorter, and about 45 CADs for 100 % PFI. No significant difference is observed in further cases with the decrease in PFI percentage. Again, Model 2 shows good agreement with Model 1 when engine is operated at wide open throttle.

5. CONCLUSION

An experimental study is performed to investigate the combustion characteristics of a dual fuel injection system in a single cylinder SI engine. Variations including PFI-E85 & DI-gasoline, PFI-gasoline & DI-E85, and PFI & DI-gasoline systems are compared at different load conditions. In each test study, the percentage of DI system is varied from 0 to 100 %. The cycle-to-cycle variability is presented using COV in IMEP. MFB and burn angles are determined from the analysis of measured in-cylinder pressure data.

- PFI-E85 & DI-gasoline: Results show that mean IMEP increases with the increase in PFI-E85 due to reduced charge cooling effect and fast ethanol combustion. Similarly, the combustion is faster with the increase in ethanol percentage (reduced DI charge cooling). The burn durations are 51, 39 and 37 CADs with 100 % PFI-E85 when engine is operated at 3.3 bar IMEP (with 1500 rpm), WOT (1500 rpm) and WOT (2500 rpm) load conditions, respectively.
• PFI-gasoline & DI-E85: Results show that there is no significant difference in the mean IMEP with the increase in DI-E85. The effects of fast ethanol combustion due to increase in DI-E85 are compensated by the increase in DI charge cooling effects. Similarly, combustion duration decreases only slightly with the decrease in DI-E85. The burn durations with 0 % DI-E85 are 58 and 40 CADs when engine is operated at 3.3 bar IMEP and WOT load conditions, respectively.

• PFI & DI-gasoline: At part load condition, there is no significant difference in burn durations with the variation in PFI; however, at WOT it decreases only slightly with the increase in PFI. The burn durations with 100 % PFI are 61 and 45 CADs when engine is operated at 3.3 bar IMEP and WOT load conditions, respectively.

The comparison to evaluate the mass fraction burned from Model 2 (Net Pressure method) is presented with respect to Model 1 (Rassweiler and Withrow method with a new linear model for polytropic index). Model 2 is of interest due to an advantage that the data processing time is short enough to allow for on-line processing. Results show that Model 2 is in good agreement with Model 1 at high load conditions; however, it predicts slower combustion at light load conditions and the difference increases with the decrease in PFI percentage. It is found that Model 2 determines the EOC with reasonable accuracy at WOT; however, its prediction of EOC at part load condition is significantly later than the actual value.

ACKNOWLEDGMENTS

Financial support from Michigan Economic Development Corporation (MEDC) is gratefully acknowledged.

REFERENCES


NOMENCLATURE

SYMBOLS

\( f \): Final State of Combustion
\( \phi \): Fuel-to-Air Equivalence Ratio
\( i \): Initial State of Combustion
\( \lambda \): Relative Air-to-Fuel Ratio, Inverse of \( \phi \)
\( n_c \): Polytropic Index, Compression
\( n_e \): Polytropic Index, Expansion
\( p \): In-Cylinder Pressure
\( \Theta \): Crank Angle Degree
\( V \): Cylinder Volume

ABBREVIATIONS

CR: Compression Ratio
CAD: Crank Angle Degree
COV: Coefficient of Variation
\( \text{COV}_{\text{imep}} \): Coefficient of Variation in Indicated Mean Effective Pressure
DI: Direct-Injection
E85: Blend of 85 % ethanol and 15 % gasoline by volume
EOC: End of combustion
IC: Internal Combustion
IMEP: Indicated Mean Effective Pressure
MFB: Mass Fraction Burned
PFI: Port-Fuel-Injection
RPM: Revolutions per minute
SI: Spark Ignition
SOC: Start of Combustion
TDC: Top Dead Center
WOT: Wide Open Throttle