SAE INTERNATIONAL

An Evaluation of Knock Determination Techniques for Diesel-Natural Gas Dual Fuel Engines

2014-01-2695
Published 10/13/2014

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CITATION: Singh, A., Anderson, D., Hoffman, M., Filipi, Z. et al., "An Evaluation of Knock Determination Techniques for Diesel-Natural Gas Dual Fuel Engines," SAE Technical Paper 2014-01-2695, 2014, doi:10.4271/2014-01-2695.

Abstract

The recent advent of highly effective drilling and extraction technologies has decreased the price of natural gas and renewed interest in its use for transportation. Of particular interest is the conversion of dedicated diesel engines to operate on dual-fuel with natural gas injected into the intake manifold. Dual-fuel systems with natural gas injected into the intake manifold replace a significant portion of diesel fuel energy with natural gas (generally 50% or more by energy content), and produce lower operating costs than diesel-only operation. Diesel-natural gas engines have a high compression ratio and a homogeneous mixture of natural gas and air in the cylinder end gases. These conditions are very favorable for knock at high loads. In the present study, knock prediction concepts that utilize a single step Arrhenius function for diesel-natural gas dual-fuel engines are evaluated. A heavy duty diesel engine with the capability of running both natural gas and diesel is operated at points where knock occurs and the cylinder pressure traces are recorded. Constants of the Arrhenius function for dual-fuel operation are determined empirically based on the cylinder pressure and temperature histories. Several methods for knock prediction and coefficient fitting are evaluated, and their accuracy is compared. Results indicate that Arrhenius integral methods can determine if a cycle will knock with accuracy up to 72% using coefficients similar to those for methane.

Introduction

The relative importance of natural gas over other fuels has gained momentum recently with the discoveries of substantial new supplies of natural gas worldwide. The development of shale extraction techniques has lowered natural gas prices by a sizable margin, providing an advantage over conventional fuels. Stringent emission norms have led to the use of expensive after-treatment technologies which increases both the capital and operating costs of the vehicles [1]. It also has benefits in terms of carbon dioxide emissions over diesel [2, 3] due to a more favorable carbon to hydrogen ratio. The relatively low CO_2 emissions (over diesel and gasoline) and economic benefits of natural gas motivate its use in internal combustion engines [4].

This research focuses on diesel-natural gas dual fuel engines. Natural gas has high research octane number (>120), it varies with the composition of natural gas [5], which enables its use in high compression ratio applications without knock over a majority of operating points [6]. A high compression ratio provides fuel efficiency improvement, but can be challenging for spark-ignition system to provide sufficient energy within durability expectations [7]. This motivates the use of an alternate energy source for ignition, which in this research is diesel fuel.

The choice of natural gas injection strategy is the primary categorization method for dual fuel engines. There are two common architectures; (1) direct injection of natural gas and diesel into the combustion chamber [8], and (2) natural gas injection into the intake manifold combined with direct injection of diesel [9]. In both the configurations, diesel is the source of ignition, as it has high cetane number; however, this investigation utilizes the latter injection strategy.

In natural gas-diesel dual fuel engines with natural gas injected into the intake manifold, a portion of the diesel energy content (around 50% or more) is replaced by natural gas. The natural gas charge enters the cylinder in a near homogeneous mixture, populating the end gases[1] far from the initial ignition source with fuel. The combination of high compression ratio and the presence of methane within the end gas makes this injection configuration susceptible to knock, and limits NG substitution at high loads. It has been observed that dual fuel knock characteristics are similar to spark-ignition engines when the diesel injected is small portion of the charge energy. In that case, most of the energy is released by the flame propagation

1. End gas is defined as the stagnant unburned fuel-air mixture located farthest from the ignition source near the walls in the cylinder chamber. [11]

into the natural gas-air mixture and knock occurs in the end gas region away from the pilot ignition centers, the traditional SI manner. In contrast, when diesel pilot is large in quantity, the reaction rates increase abnormally near the pilot ignition centers causing knock, which differs substantially from traditional SI knock in the spatial context. [10].

The research investigates several auto-ignition models for diesel-natural gas dual fuel engines which can be incorporated into engine simulations for developing dual fuel calibrations.

The experimental setup and data collections techniques are first described. Then, the auto-ignition modeling approach utilized is presented, followed by a description of experimental knock characterization. Following this section, multiple approaches to knock prediction are implemented, evaluated and compared and conclusions are drawn.

Experimental Setup and Data Collection

This research is carried out using a 15L Cummins ISX550 with cooled EGR that was modified for dual fuel operation. Engine specifications are shown in Table 1. Natural gas is injected downstream of the intercooler and upstream of the EGR mixer. Cylinders five and six were instrumented with passage mounted AVL GH15DK piezoelectric cylinder pressure sensors. The sensors were installed keeping in mind the location of valve seats, water jackets and injector nozzle because drilling the hole for the sensor through water jacket or injector path can prove damaging for the engine. To characterize the engine at different operating points, pressure sensors, lambda sensors, thermocouples, a turbocharger speed sensor, mass air-flow meter, AVL KMA Mobile diesel flow meter and natural gas flow meter were installed. Figure 1 shows a schematic diagram of engine sensor locations.

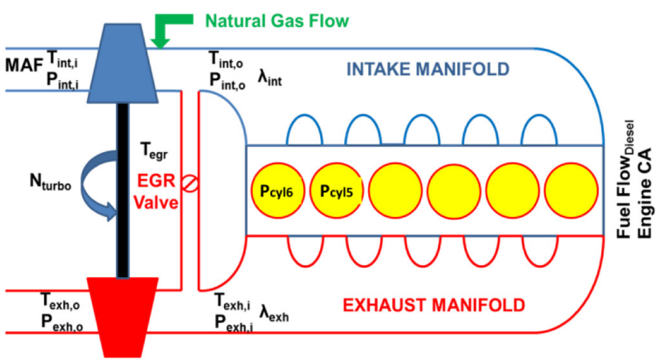

Figure 1. Engine schematic showing locations of all sensors and flow meters

All signals are recorded using an AVL IndiSmart data acquisition system at 0.1° CA resolution. The data at different operating points was acquired by running the truck equipped with Cummins ISX550 engine on water brake chassis dynamometer.

Table 1. Engine specifications

Engine:	Cummins ISX550
No of Cylinders	Inline 6-cylinder
Displacement:	15 liters
Compression ratio:	17:1
Valve configuration:	Overhead valves
Valves per cylinder:	4 (2-Intake, 2-Exhaust)
Aspiration	Turbocharged
Bore x stroke:	137 x 169 mm
Fuel system:	High Pressure Direct Injection – Time Pressure Fuel System (HPI – TP)
Horsepower:	550 hp @ 2000 RPM
Torque:	2508Nm @ 1150-1600 RPM
Turbocharger	Variable Geometry
After-treatment	Cummins Particulate Filter-DPF - DOC

Knock Modeling and Prediction

Developing simulation models for knock onset is a research focus as it facilitates knock prediction and engine performance analysis in an inexpensive way. Models for knock onset vary in complexity - from using chemical kinetics packages [13] to global single step Arrhenius functions representing hydrocarbon oxidation reactions. From the controls point of view, Arrhenius functions are considered to be an efficient way to predict knock due to their simplicity and good physical and chemical representation [14]. The application of this induction-time based correlation has been widely studied and used in rapid compression machines (RCM), constant volume bombs and internal combustion engines.

Detailed chemical kinetics models capture the reactions of all fuel species present leading up to auto-ignition. However, but the biggest disadvantage of kinetics-based models is their high computational cost. The knock integral approach using an Arrhenius function is computationally inexpensive and has successfully been incorporated into simulation models of the engines to predict knock satisfactorily [11].

Typically, the Arrhenius function is calibrated to experimental data for a variety of equivalence ratios. The inputs to this function are the relevant pressure and temperature histories. While, different forms of Arrhenius equations have been developed in the past for different fuel types, the general form of the Arrhenius equation is shown in Equation 1. Where A, n and B are the fitted parameters depending on the fuel [6].

$$\tau = A\, p^{-n} \exp\left(\frac{B}{T}\right) \quad (1)$$

In a RCM, Eq. 1 is used to represent the ignition delay where coefficients A, n and B are determined empirically. A is the pre-exponential constant; n is exponent of cylinder pressure and B represents the activation energy of the auto-ignition reaction of the fuel air mixture. Combustion is considered to be at constant pressure in a RCM, as compared to in IC engines where pressure is changing during compression and combustion. It is important to note that the end gases are compressed by burned gases from combustion. The end gas compression process is assumed to be polytropic and the temperature rises accordingly. According to Livengood and Wu [15], the end gas auto ignition chemistry is considered to be cumulative and can be predicted by integrating the pressure and temperature at different time steps, as shown in Fig. 2,. It is assumed that auto-ignition occurs when:

$$\int_{t_{IVC}}^{t_k} \frac{dt}{\tau} = 1 \tag{2}$$

Where t_{IVC} is the time when intake valve closes and the end gas compression begins and t_k is the time when auto-ignition occurs.

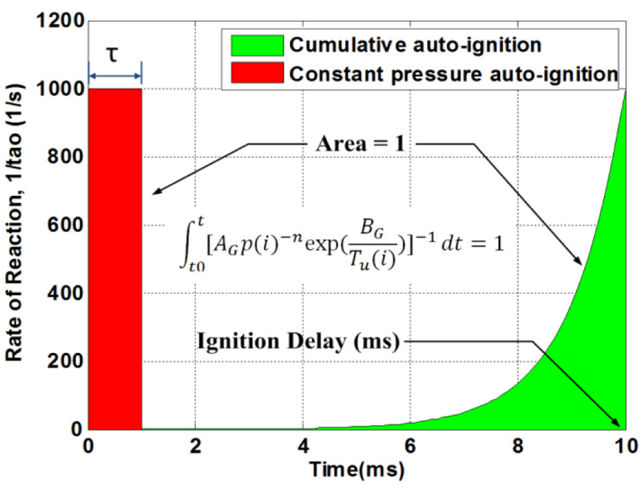

Figure 2. Illustration of the Livengood-Wu Integral for predicting auto-ignition in a changing pressure and temperature environment [14]

In the current study, the Arrhenius function is used to predict auto-ignition in a natural gas - diesel dual fuel engine. The inputs to this equation are pressure and temperature. Cylinder pressure data is collected by running the engine at different operating points with knock. The end gases are considered to be compressed adiabatically and their temperature values are determined by the following adiabatic relation expressed in Equation 3.

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\left(1-\frac{1}{\gamma}\right)} \tag{3}$$

Where T_2 and T_1 are temperatures at two different points, p_1 and p_2 represent cylinder pressures and Υ is the ratio of constant pressure specific heat to the constant volume specific heat of the mixture in the cylinder.

Using this foundation, this work analyzes different forms of the Arrhenius function to determine which form provides the best knock prediction for this particular dual fuel case.

Additionally, Lefebvre et al [16] developed an alternative ignition delay correlation for methane - air mixtures, represented by Equation 4 and having coefficients as shown in Table 2.

$$\tau = A.\varphi^m.p^n.e^{\frac{E}{RT}} \tag{4}$$

where, τ is the ignition delay, A is constant, φ is methane-air equivalence ratio, p is pressure, m & n are exponents, E is the activation energy of the methane-air mixture, R is gas constant, T is temperature of the mixture. This correlation is also analyzed and compared to the Arrhenius expressions.

In general, natural gas contains about 95% or more of methane content [17] but it varies from source to source, so the coefficients in Table 2 are used as the initial values in a regression analysis to find new factors for the dual fuel engine experimental data used in this study.

Table 2. Ignition Delay coefficients for Methane-Air mixture

A	m	N	E/R
2.4e-3	-0.19	-0.99	12500

Prediction of Knock Onset: Comparison of Methods and Validation

For this research, a knocking cycle is defined by the threshold of the high frequency (> 3 kHz) within an experimental pressure trace exceeding 1.5 bar [18]. The criterion used here to define knock onset is the crank angle corresponding to the first crest of the pressure trace which meets both the high frequency threshold and exceeds 1.5 bar in amplitude. For example, the cycle shown in Fig.3 is classified as knocking because the high frequency portion of cylinder pressure exceeds 1.5 bar (the defined threshold value) and knock onset occurs at 11 CAD ATDC.

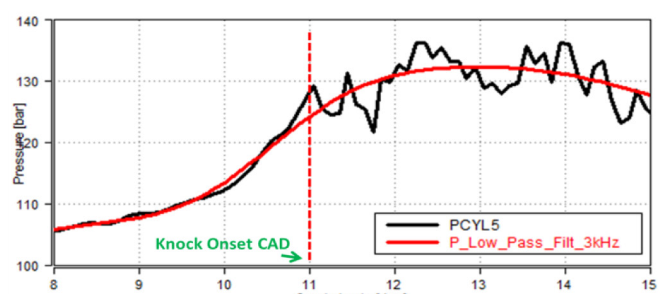

Figure 3. Knocking cycle with the explanation of finding the knock onset crank angle degrees

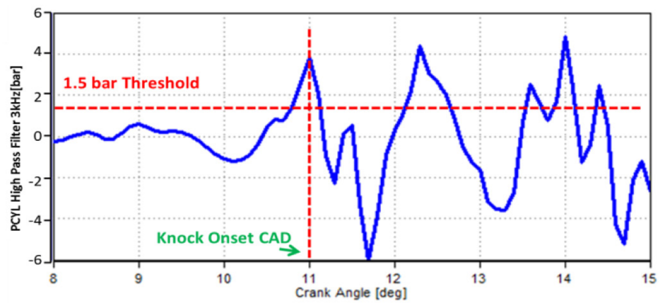

Figure 3. (cont.) Knocking cycle with the explanation of finding the knock onset crank angle degrees

Before getting into the methodology for knock prediction, the following terms require definition:

Knock Integral Value (KIV) - Is defined as the Arrhenius function value, with empirically determined coefficients, integrated from the time the end gas starts being compressed to the time indicated by the user. The window used for integrating the ignition delay correlation is from IVC to 20 CAD ATDC.

Pressure Intensity (PI) - Is defined as the area under the high frequency portion (>3 kHz) of cylinder pressure [19] integrated in terms of crank angle degrees. The integration window used for this research is −30 CAD to 20 CAD ATDC. Pressure intensity is expressed as follows:

$$\mathrm{PI} = \frac{1}{t_2 - t_1}\int_{t_1 = t(\theta_1)}^{t_2 = t(\theta_2)} |p_{hpf}(t)|\,dt \tag{5}$$

Where, t_1 & t_2 represent time events, p_{hpf} is the cylinder pressure passed through high pass filter

The following sections describe the performance of three different models for knock prediction when applied to a natural gas-diesel dual fuel engine.

Knock Prediction Method One: Arrhenius Without Equivalence Ratio

Eq.6 represents the auto-ignition model based on the Arrhenius equation. The constants (C1, C2, and C3 in Eq. 6) in this equation are found empirically by fitting experimental data. Initial values used in the regression analysis for finding constants were taken from those of methane [15], Table 2, as it is assumed that most end gases are a mixture of only natural gas and air. At the engine operating conditions shown in Table 3, the set of constants for 60 individual knocking cycles were determined and then averaged. These constants are shown in Table 4.

$$\frac{1}{C_1}\int_{\theta_{IVC}}^{\theta_k} p(\theta)^{C_2} \exp\left(-\frac{C_3}{T(\theta)}\right) d\theta = 1 \tag{6}$$

Where, C_1,C_2 and C_3 are constants, Θ_{IVC} is the crank angle degree corresponding to the intake valve close event, Θ_k is the crank angle degree corresponding to knock onset, p is cylinder pressure and t is temperature.

Table 3. Engine Operating Conditions used for experiment

Engine Speed	1200 RPM
IMEP	13 bar
Inlet air temperature	43˚ C
Intake manifold pressure	1.8 bar
Natural Gas - Air Equivalence ratio	0.6

The constants shown in Table 4 were then used to predict the knock onset angle for different set of 40 knocking cycles. Fig 4 shows the ability of the model to predict knock and how it behaves for training and the validation cycles. The root mean square error between the predicted and measured knock onset is 5.61CAD. In general, the model predicts an earlier knock onset than actually occurs experimentally.

Knock Prediction Method Two: C3 as a Function of Equivalence Ratio

Seref Soylu [11] developed a knock model for natural gas SI engines with coefficient C3 dependent on equivalence ratio of the natural gas-air mixture (φ) in the cylinder. It is assumed that the end gas produces knock and that the end gas is composed of natural gas and air only. As C3 represents the activation energy of the mixture being auto-ignited, it must depend on the equivalence ratio of the intake mixture. To investigate this, C1 and C2 were kept constant from Method One and another regression analysis was performed to find C3 for each of the knocking cycle. For this test, the engine was operated at the same conditions shown in Table 3 but with φ varied from 0.59 to 0.65. It is evident from the Fig. 5 that C3 shows a linear trend (albeit not very well correlated) with the natural gas-air equivalence ratio. Arrhenius constants with C3 dependent on equivalence ratio are shown in Table 5 and were then used to predict the knock onset again for the knocking cycles. The prediction improved by a large margin relative to method one as the RMSE reduced to 2.28 CAD and the points cloud shifted nearer to the ideal knock onset prediction line as shown in Figure 6.

Table 4. Coefficients for Eq 6 determined empirically

C1	C2	C3
0.00274	0.799	13740

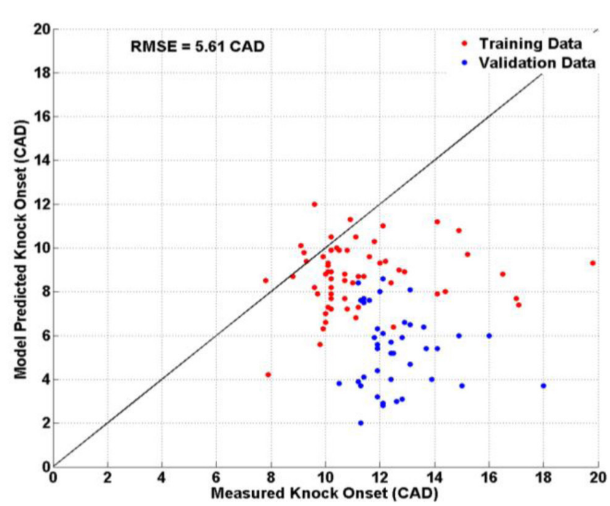

Figure 4. Knock onset prediction based on the coefficients shown in Table 4.

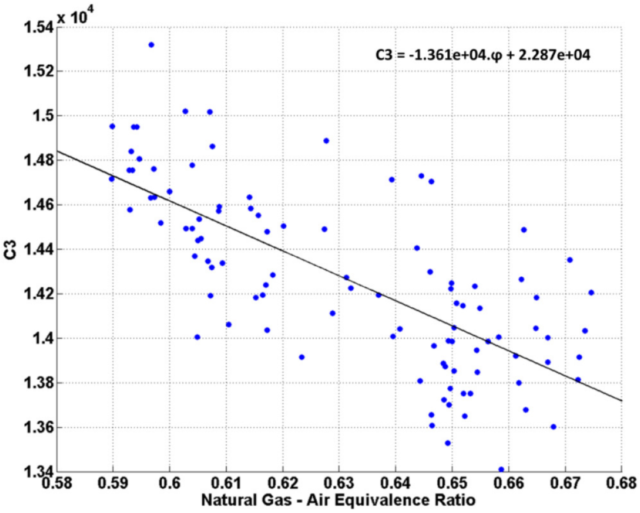

Figure 5. C3 dependence on Natural Gas-Air Equivalence Ratio (φ)

Table 5. Coefficients for Eq. 6 determined empirically with C3 dependent on natural gas-air equivalence ratio

C1	C2	C3
0.00274	0.799	-13610*φ + 22870

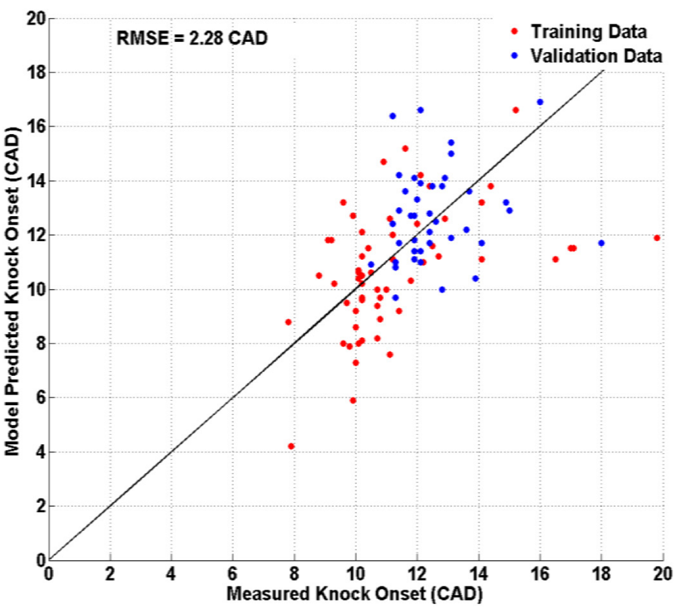

Figure 6. Knock onset prediction based on the coefficients shown in Table 5

Knock Prediction Method Three: Lefebvre Method

Lefebvre et al [15] proposed Eq. 7 as an alternative auto-ignition model.

$$\frac{1}{C_1}\int_{\theta_{IVC}}^{\theta_k} p(\theta)^{C_2}\varphi^{C_4}\exp\left(-\frac{C_3}{T(\theta)}\right)d\theta = 1 \tag{7}$$

Where, C_1,C_2,C_3 and C_4 are constants, Θ_{IVC} is the crank angle degree corresponding to the intake valve close event, Θ_k is the crank angle degree corresponding to knock onset, p is cylinder pressure and t is temperature.

Using regression analysis, all four constants were determined for dual fuel engine data and are shown in the Table 6. Although the knock onset angle prediction improved as compared to the method one, the Lefebvre method was not better than method two in terms of RMSE (3.14 CAD) as shown in Figure 7.

Table 6. Coefficients for Eq 7 determined empirically with φ as a separate entity in the model

C1	C2	C3	C4
0.00238	0.793	13593.14	2.08

In comparison to the coefficients for a methane-air mixture (Table 2), all the coefficients for dual fuel operation are very close except for the exponent of the φ. The C2 and C3 constants are of opposite sign of exponents m and n (Table 2) because the ignition delay expression (Eq 4) was used in numerator to form Eq 7. The equivalence ratio of the natural gas-air mixture for the knocking end gas depends on the location where it occurs in the cylinder chamber. In a dual-fuel, diesel ignited engine, non-traditional 'end gas-like' zones may occur between diesel injector jets. Additionally, the local φ value of these 'end gas-like' zones may differ substantially from the original homogeneous charge thanks in part to oxygen depletion due to the diesel combustion. These factors could be responsible for the altered exponential behavior of φ. Mathematically, the C4 value in Eq. 4 will be lesser if other coefficients are kept constant, which will make the coefficient values closer to those determined for methane burning gas turbines by Lefebvre et al [15].

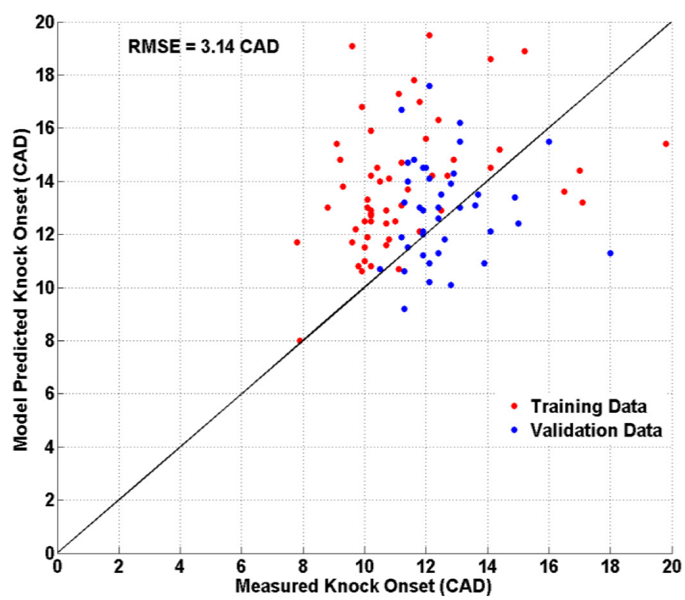

Figure 7. Knock onset prediction based on the coefficients shown in Table 5

Predicting Knock/Non-Knock Cycles: Comparison of Methods

While the prior section revealed that Method Two, the Arrhenius expression including equivalence ratio dependence, shows the strongest propensity to accurately predict the crank angle onset of knock within a natural gas-diesel, dual fuel engine. However, a practical engine merely necessitates the accurate prediction of whether or not the current cycle will knock, regardless of absolute phasing accuracy. From an engine control point of view, predicting whether the current cycle is a knock or non-knock cycle is very important as it will lead to the decision whether to adjust combustion phasing (with diesel injection timing in a dual fuel engine). For testing the knock prediction capabilities of all three Methods, 1000 cycles containing intermittent knock were recorded. For these 1000 transient cycles, engine operating conditions are shown in Figure 8 & 9. The inlet air temperature was 43.6°C and the natural gas-air equivalence ratio was 0.6 for the experiment. During the experiment, knock occurs intermittently as conditions vary in the combustion chamber.

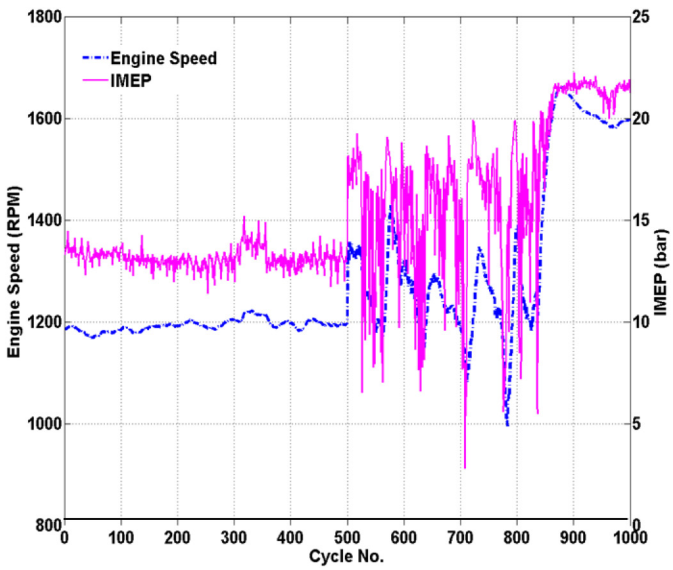

Figure 8. Engine RPM and IMEP for 1000 transient cycles

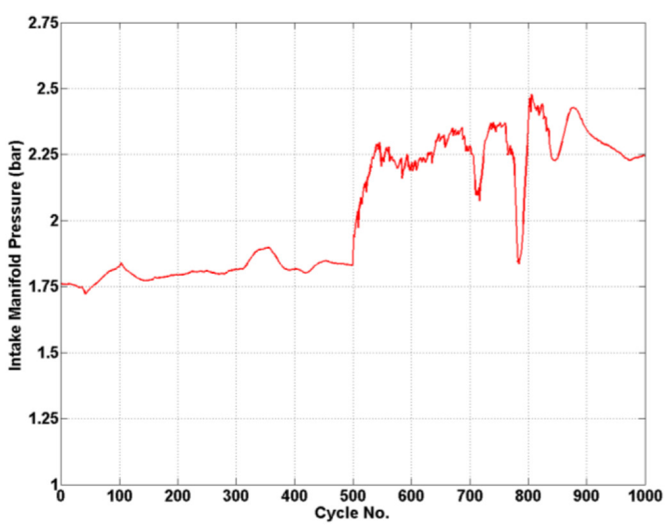

Figure 9. Intake Manifold Pressure for 1000 transient cycles

In Figure 10, the 1000 experimental cycles were subjected to knock prediction via Method One. Classification of knocking cycles (squares in Fig. 10) and non-knocking cycles (triangles in Fig.10) is determined by the high frequency portion of cylinder pressure with a threshold exceeding 1.5 bar (discussed earlier). According to the auto-ignition model, knock onset takes place when the knock integral value (KIV) reaches one. This means all the cycles which lie in Zones 2 & 3 of Fig. 10 are knocking according to Method One, but this is not the case for the experimental data. The Arrhenius Method One knock prediction accuracy is very poor in Zone 3 (PI<20 & KI>1). There can be two reasons for poor accuracy in Zone 3 which are as follows:

a. When there is knocking, the turbocharger speed fluctuates leading to frequent rise and fall in intake boost pressure. This leads to varying amount of natural gas-air mixture in the cylinder each cycle. If the boost pressure is higher than the referenced cycle used for finding the coefficients of the Arrhenius function, it leads to higher cylinder pressures during the compression stroke.
b. The cycle has very late combustion phasing.

Either or both of the above reasons can lead to an Arrhenius KIV>1 even for a non-knocking cycle. KIV essentially depends on the area under the referenced knocking cycle from IVC to knock onset. The reason for this can be seen when a knocking cycle and higher load non-knocking cycle are integrated until 30CAD ATDC, as shown in Fig. 11. Here the area under the non-knocking cycle curve will be higher due to regions 1 and 3 being greater than region 2. In some cases either of the areas of regions 1 & 2 will be sufficient enough to compensate the area of region 3. This leads to a higher KIV value prediction, even for the non-knocking cycles even when pressure intensity is very low.

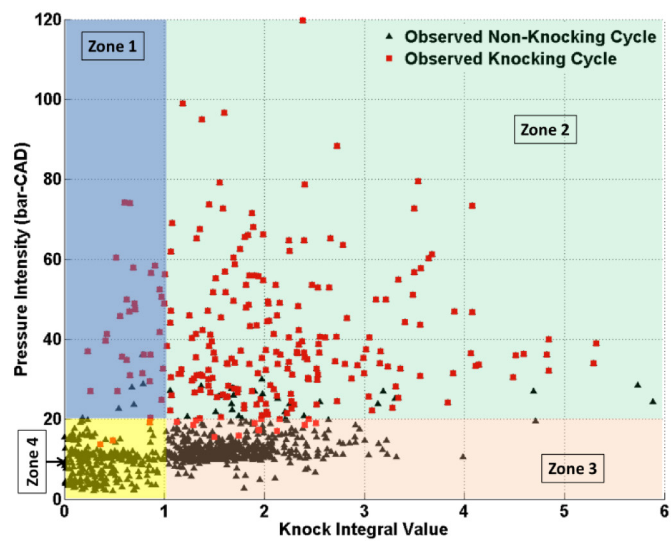

Figure 10. Knock Integral Value vs Pressure Intensity for 1000 transient cycles with intermittent knock occurrence with coefficients from method one

As evident from Fig.10, if a threshold of pressure intensity is also utilized with the KIV to predict knock, Zone 2 (PI>20 & KIV>1), the knock integral model prediction accuracy is very high as most of the model-predicted knocking cycles (KIV>1) are also the observed knocking cycles. Similarly, model accuracy is high in Zone 4 (PI<20 & KIV<1) where observed non-knocking cycles are predicted not to knock by the model (KIV<1).

The prediction accuracy for Method 1 is 86.8% in case of knocking cycles whereas it is only 35.6 % for non-knocking cycles. The prediction accuracy for knocking cycles is calculated by dividing the number of knocking cycles predicted by the model (KIV>1) with the number of observed knocking cycles; similarly it is calculated for non-knocking cycles. The model inaccuracy in Zone 1 is due to knocking cycles where boost pressure is lower than the referenced knocking cycle, leading to KIV <1 but still PI is higher. Auto-ignition model accuracy in Zone 1 worsens if the Arrhenius equation takes into account the equivalence ratio of the natural gas-air mixture as shown in Fig. 12 & Fig. 13, representing Method two and Method three, respectively. This is likely due to the changes in φ which depend on the varying oxygen consumed by the diesel every cycle.

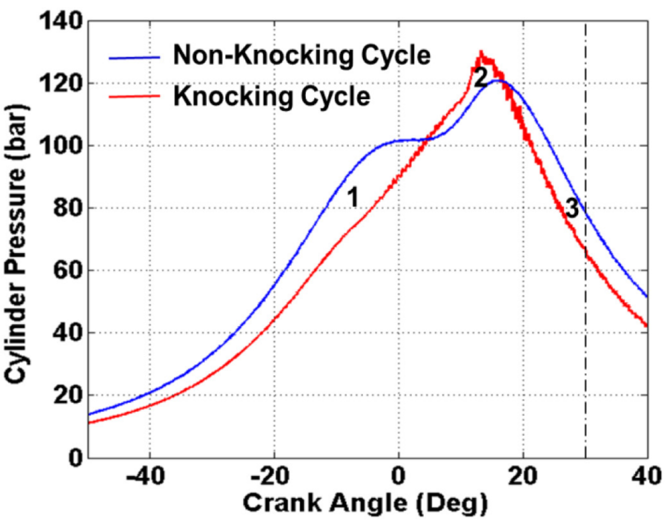

Figure 11. Cylinder Pressure Cycle comparison between high load and late combustion cycle and the referenced knocking cycle

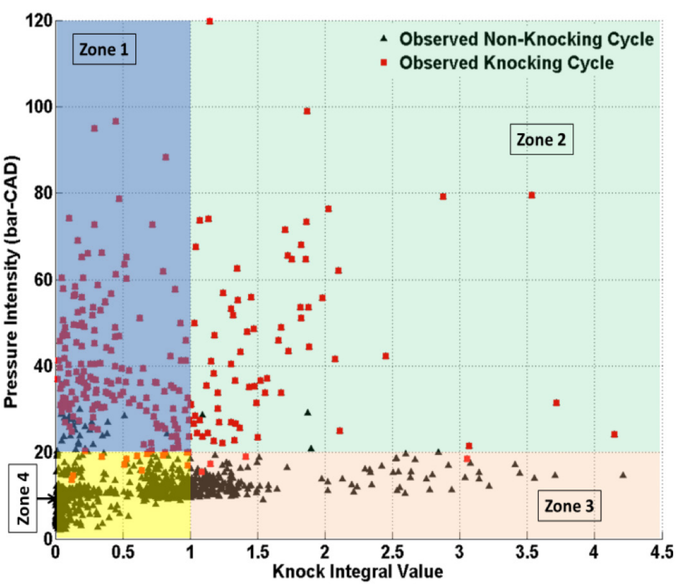

Figure 12. Knock Integral Value vs Pressure Intensity for 1000 transient cycles with intermittent knock occurrence with coefficients from method two

This approach has shown that, for classifying cycles as either knocking or non-knocking, both Pressure Intensity and Knock Integral Value have to be utilized as the Arrhenius function does not have sufficient accuracy to predict knocking cycles with high confidence by itself.

Sensitivity Analysis of Coefficients

To investigate if further improvements in prediction accuracy can be achieved, each of the coefficients C1, C2 and C3 are varied. The prediction accuracy for knocking and non-knocking cycles is defined as earlier. The prediction accuracies are added to find the combination of coefficients where the model prediction accuracy is at a maximum for knocking as well as non-knocking cycles.

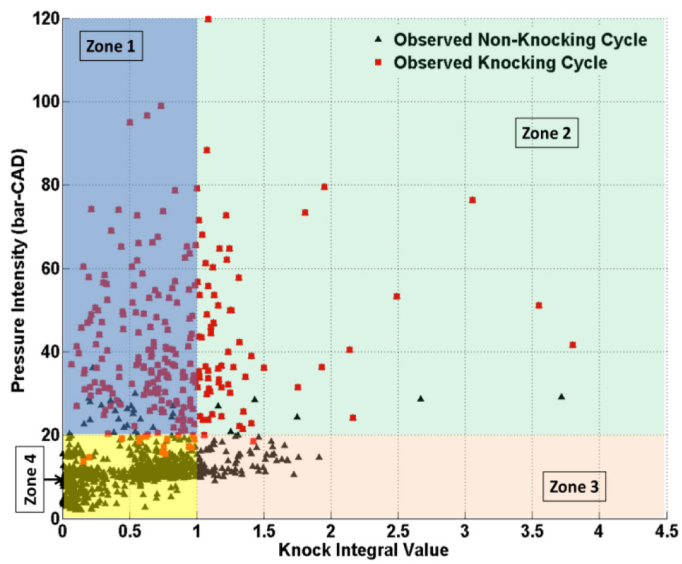

Figure 13. Knock Integral Value vs Pressure Intensity for 1000 transient cycles with intermittent knock occurrence with coefficients from method three

It is evident from the Fig. 14, as C3 changes there exists a trade-off between knocking and non-knocking prediction accuracy. If C3 is increased by 20%, the prediction of the model increases. In this case, the auto-ignition model is able to predict 80% of the observed knocking cycles as knocking cycles and for non-knocking cycles, prediction accuracy is 49%. The overall model prediction accuracy, which is calculated by dividing the sum of correctly predicted knocking and non-knocking cycles with the total number of cycles, is 56.7%. A similar sensitivity analysis corresponding to C1 and C2 can be found in the Appendix.

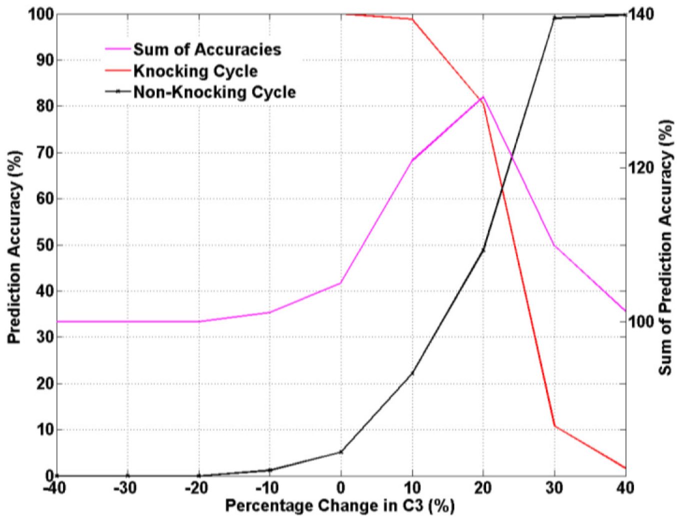

Figure 14. Sensitivity Analysis of C3 towards the knock and non-knock prediction accuracy

Optimization of Arrhenius Coefficients

Results of the coefficient sensitivity analysis suggest that higher overall dual-fuel knock prediction accuracy could be obtained if constants were fit outside of those for natural gas only combustion. In all previous sections, the regression analyses for finding Arrhenius coefficients were done keeping in consideration of the physics and chemistry involved for methane. This requires that the coefficient C3 is held very close to the activation energy of a methane and air mixture. In an attempt to improve accuracy, a wider range of coefficients were considered to see if knock prediction accuracy can be enhanced.

In the case of method one, with the coefficients listed in Table 7, the prediction accuracy for knocking cycles can be increased to 72% and for non-knocking cycles to 77.3% as compared to the coefficients listed in Table 4. While this is a significant increase in prediction accuracy, the idealized constants are well outside of the original values for methane.

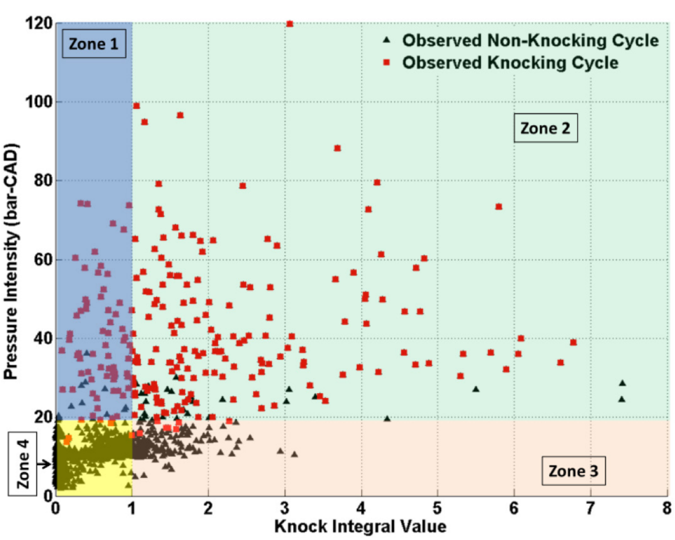

Figure 15. Knock Integral Value vs Pressure Intensity for 1000 transient cycle with intermittent knock using mathematically optimized coefficients shown in Table 7

Table 7. Coefficients for Method 1 with increased prediction accuracy

C1	C2	C3
0.0024	2.1	21200

Conclusions

In this study, an auto-ignition integral method using different forms of the Arrhenius function was implemented to predict both knock and its onset for a diesel-natural gas dual fuel engine. The conclusions for this study are as follows:

1. The knock onset prediction capability of the auto ignition model improves with the consideration of natural gas-air mixture equivalence ratio in the model.
2. The coefficient in the Arrhenius function representing activation energy of the fuel mixture shows dependency on the natural gas air-fuel ratio and it decreases with higher equivalence ratios.
3. Utilization of a pressure intensity threshold can substantially increase the accuracy of the Arrhenius knock onset prediction.
4. The Arrhenius coefficients for the dual fuel engine case are very close to those found in methane-only studies, suggesting the knock mechanisms are not greatly influence by the presence of diesel fuel in this dual fuel application.
5. The knock prediction capability of the Arrhenius model can be increased to over 72% by optimizing the coefficients, but this may remove some of the physical and chemical relevance of the model by allowing the activation energy constant to vary substantially from that of Methane.

Abbreviations

IVC - Intake Valve Closing

CAD - Crank Angle Degrees

KIV - Knock integral Value

PI - Pressure Intensity

P_{hpf} - cylinder pressure passed through high pass filter

RMSE - Root Mean Square Error

ATDC - After top Dead Center

C1, C2, C3 - Coefficients of Arrhenius Equation

RPM - Revolutions per minute

RCM - Rapid Compression Machine

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APPENDIX

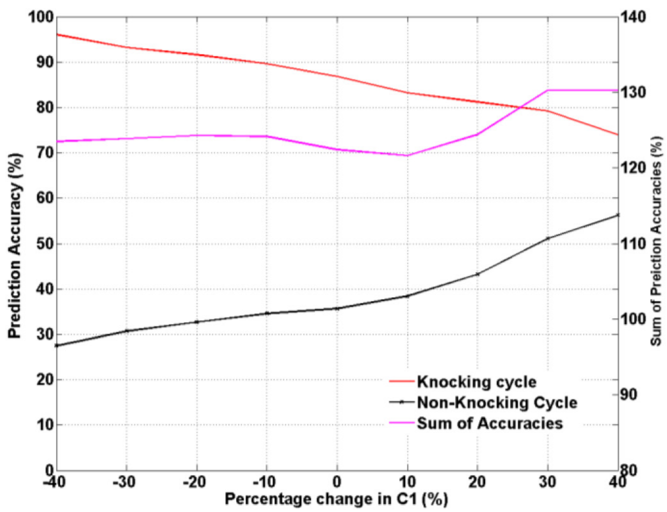

Figure 16. Sensitivity Analysis of C1 towards the knock and non-knock prediction accuracy.

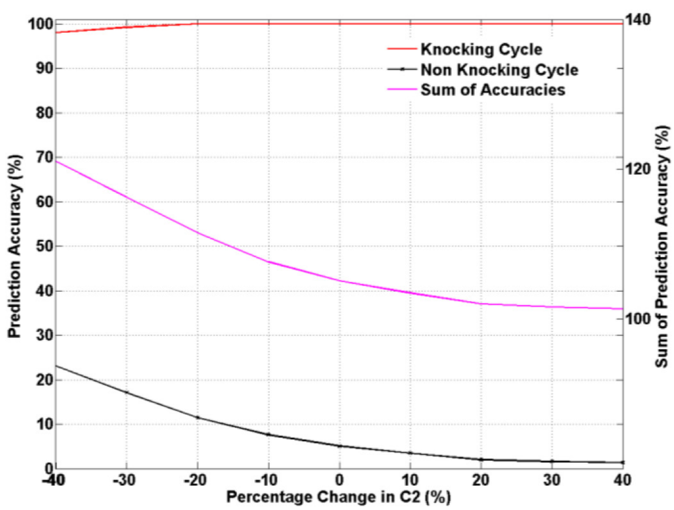

Figure 17. Sensitivity Analysis of C2 towards the knock and non-knock prediction accuracy.