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Modelling the effects of ethanol fumigation on engine performance and emissions in a six-cylinder, common rail diesel engine

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Abstract

This paper describes a one-dimensional thermodynamic model developed using AVL BOOST with the objective of analysing the performance, combustion parameters and NOx emissions of a Euro III, six-cylinder turbocharged Cummins diesel engine. The model was validated against experimental data obtained from the same engine run at a constant speed of 2000 rpm at varying load conditions (full, three quarter (3Q) and half load) using low sulphur diesel fuel (D100), as well as fumigated ethanol at 10% (D90), 20% (D80) and 30% (D70) substitutions (by energy). The results for D100, D90, D80 and D70 were found to be in good agreement with the experimental data. The percentage variation for engine performance parameters such as: brake power (BP), indicated power (IP), indicated torque (IT) and mean effective pressure (MEP) for D100 have been found to be approximately in the range of -5% to 1.5% for all loads, whereas, the fuel energy variation was only 0.33% for all loads. With increasing ethanol fumigation, a rise in peak pressure of the cycle, more rapid initial heat release rate and a reduction in the NOx emissions were observed in this study.

1 Introduction

Alcohol is a form of renewable energy which can be produced from carbon-based agricultural feed stocks, locally grown crops and even waste products [1]. Sugarcane residue is another renewable energy source of alcohol production [2]. In the last decade, an increasing trend of alcohol fuel production from renewable sources has been noted globally [3]. Ethanol is considered to be one of the more promising alternative fuels. Compared to methanol, ethanol has a lower enthalpy of vaporisation and auto-ignition temperature, hence better ignition characteristics [4, 5].

Broukhian et al. [6] found that knock at high loads limited the proportion of ethanol which could be fumigated. At nearly all engine operating conditions, modest gains in thermal efficiency were observed, while NOx decreased at all load conditions. In fact, with the use of fumigated alcohol, if efficient air utilisation takes place, then it can result in even greater power output. Although displacement of diesel by ethanol can be done to about 50%, it has limitations, which have been discussed in detail by [4, 7, 8]. Furthermore, as the evaporation of alcohol in the intake manifold lowers the temperature of the incoming air, which results in an increase in density, more air, and hence oxygen, could be made available for combustion [8, 9]. Ghadikolaei et al. [10] found that fumigating alcohols in diesel engines leads to a reduction of NOx and CO_2 in most tests, and PM in all cases.

Fumigated ethanol has been shown to cause a significant increase in the rate of pressure rise, relative to operation using diesel fuel alone [11]. The increased rate of pressure rise has been attributed to the auto-ignition of the homogeneous fumigated charge after initial combustion reactions had already raised the cylinder temperature and pressure [10]. It has been reported that this effect is worsened by an increase in ignition delay, associated with the cooling effect of the fumigated alcohol [10]. Similar findings associated with an increase in ignition delay with ethanol fumigation have also been reported [4, 5, 8, 9]. Bodisco et al. [12] utilised combustion resonance as a means of determining the onset of ignition. The experiments comprised of fumigated ethanol substitutions up to 50% at full load, 3Q load and half load. The results have demonstrated that at full load there is a decrease in ignition delay with increasing ethanol substitutions, whereas at half load there is an increase in ignition delay with increasing ethanol substitutions. Bodisco et al. [12] suggested that this conflicting result is a consequence of the auto-ignition of ethanol. This has also been reported in other literature sources [11].

As ethanol is fumigated into the engine, higher values of peak pressure are obtained owing to more rapid combustion [13]. Other combustion parameters, which are affected by ethanol fumigation, are ignition delay and combustion duration [13]. Fuels having a low cetane number (such as ethanol) will typically have a longer ignition delay, as ignition occurs sufficiently late in the expansion process, owing to the high latent heat of vaporisation of ethanol [13]. This has been reported in the literature by various authors [12, 13]. With an increase in load, a decrease in ignition delay is observed, attributable to a richer fuel supply [13]. Ignition timing will affect most of the in-cylinder parameters including in-cylinder pressure, maximum rate of pressure rise, peak cylinder pressure

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and temperature, hence affecting NOx emissions. Combustion duration has also been observed to decrease by increasing the quantity of ethanol fumigated. This has been attributed to enhanced mixing of fuel and air and an increased availability of oxygen in the cylinder, due to the changes in physico-chemical properties of the air-fuel mixture, namely; combustion temperature, oxygen concentration, latent heat of vaporisation, fuel distribution, ignition delay and cetane number [13]. The peak heat release rate (HRR) is also observed to increase as the quantity of injected ethanol is increased. In comparison to diesel fuel only (with no ethanol), the crank angle at which the peak heat release rate occurs is also retarded by 1-4° with ethanol fumigation [13]. This is due to ethanol being an oxygenated fuel, the HRR increases due to the availability of more oxygen and thus the combustion is improved [13].

Owing to the complex combustion issues associated with ethanol fumigation and the potential gains in terms of reducing global dependence on non-renewable fuels, this paper will detail an AVL BOOST engine model for investigating the effects of ethanol fumigation in terms of performance and emissions. This model can be utilised in future studies to understand the behaviour of ethanol during cold-start conditions, improve our fundamental understanding of the effects of ethanol fumigation and to develop strategies for the control of engines utilising ethanol as a secondary fumigated fuel.

2 Development of the AVL BOOST Model

2.1 Engine Model

A six-cylinder engine model has been built using AVL BOOST software, replicating the geometry and dimensions of the engine for generating and connecting different elements used for model design, such as: cylinders, engine, turbocharger, intercooler, ECU (engine control unit) and the engine interface for the ECU to communicate to the engine [14]. The intake and exhaust manifolds, turbocharger and intercooler are connected in the workspace through pipes, having measuring points at the ends, wherever required, and the pipe flow model has been developed as a one-dimensional thermodynamic model based on the first law of thermodynamics. The elements are defined according to the physical parameters of the engine and the physical geometries such as pipes, manifolds, etc. have been defined after being measured accurately on the engine.

The model is based on the fuel consumption data and start of injection data from the engine running at 2000 rpm (at different load conditions) from the experiments conducted by Bodisco et al. [5].

2.2 Combustion Model Setup

The combustion model used in the engine is a zero-dimensional (two-zone spatial discretisation), with the energy conservation between the burned and unburned zones being maintained. The two-zone model allows for the calculation of emissions such as NOx, CO and soot. The double vibe function, used in the combustion model, is defined by two single vibe functions, i.e. the start of combustion and the combustion duration, defined by a parameter 'm' (denoted as the shape parameter). The following form of equations can be used to describe this single vibe function and the rate of heat release approach [14].

$$\frac{dQ_B}{d\alpha} = Q_B \frac{\alpha}{\Delta\alpha_c}.(m+1).y^m.e^{-a.y^{(m+1)}} \qquad (1)$$

where $y = \frac{\alpha - \alpha_0}{\Delta\ \alpha_c}$, Q_B is fuel heat input, α is the crank angle in degrees, α_0 is the angle of the start of combustion in degrees, $\Delta\alpha_c$ is the angle of combustion duration in degrees, m is the shape parameter and a is the vibe parameter [14]. Here, the value of the vibe parameter, a, is set to 6.9, in order to obtain complete combustion. The values of the shape parameter m, the vibe parameter a, combustion start and combustion duration are specified as an input to the combustion model. These values can be specified either as constant values or as a function of engine speed (rpm) and engine load. To obtain an estimate of the required combustion duration and selection of the variables, a standard table can be referred to in the manual. [14]. The choice of these parameters in the model for dual fuel (diesel/ethanol) was possible because of the availability of experimental fuel consumption and combustion start data for this paper.

2.3 Engine Model Simulation

The model was run at 2000 rpm (full, 3Q and half load) for diesel only fuel. An engine operating speed of 2000 rpm was selected as rated power is achieved at this speed, for this engine. The model is further reformed to include a secondary ethanol injector to inject ethanol in varying proportions in fumigation mode. The location of the ethanol injector is selected to be after the intercooler (before the intake manifold). In the cases of ethanol injection, a portion of the diesel fuel, on an energy basis, is replaced by ethanol fuel, as 10% (D90), 20% (D80) and 30% (D70), with the fuel data used from the experiments conducted by [5].

3 Results and Discussion

3.1 D100 (Diesel only)

The results are presented in tabulated form for engine performance parameters. Rows 3 and 4 of Table 2 show the results for D100 compared with experimental results. Figure 1 shows the comparison between the pressure vs crank angle curve obtained from the experiment and the model for full load, with very good agreement reached between the experiment and the model.

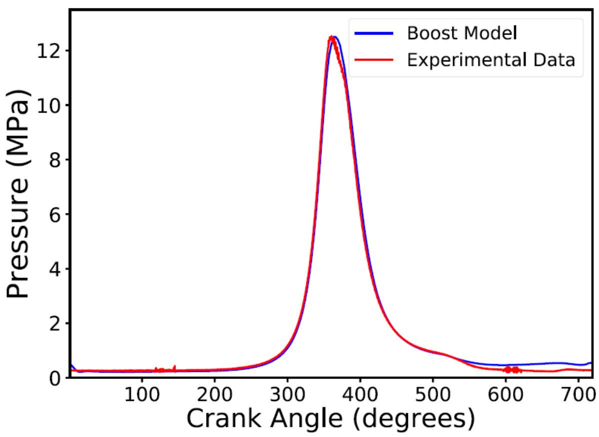

Figure 1: Comparison of in-cylinder pressure for Cylinder 1 for D100, full load, 2000 rpm between the BOOST model and the experimental data

3.2 D90, D80 and D70 (ethanol fumigation)

Rows 6 and 7 of Table 2 show the model results for engine performance parameters for D90 (10% ethanol fumigation), for 2000 RPM, full load, compared with experimental results. Tables for D80 and D70 have been omitted owing to space issues. However, the D90 results shown in Table 2 are reflective of the overall fit. Owing to space constraints, tabulated results 3Q and half load for D100 and D90 have been omitted. Similarly, the D80 and D70, 2000 RPM at full, 3Q and half load conditions have not been shown. However, the discussion of the results will be all-encompassing.

3.3 Discussion

As the ethanol fuel is injected before the intake manifold, it gets pre-mixed with the incoming charge of air and enters as a homogeneous mixture of ethanol and air. As diesel is injected, it initiates the combustion, igniting the homogeneous mixture of ethanol and air. This results in faster combustion, with a higher rate of heat release for the same volume of fuel, especially in the early stages of combustion, consequently resulting in higher peak pressure.

At full load, more rapid pre-mixed combustion and higher peak pressures are observed, with increasing ethanol fumigation, as also reported in the literature [12, 13], which would result in better combustion, with the peak pressure occurring closer to TDC. An important factor that would contribute to the high peak pressure would be the auto-ignition temperature and the lower cetane number of ethanol, compared to diesel. This pattern is seen in the present model, which is the same as that observed in the experiments at full load (refer to Table 2 for full load readings and Fig 2). At half load, there is an increase in the ignition delay and decreased combustion duration, as also reported in the literature by [12, 13]. This suggests that a decrease in combustion duration leads to higher peak pressures, but this effect will be compensated by the increase in ignition delay, and hence the values of peak pressure only see a limited increase with increased ethanol fumigation. However, as the quantity of fumigated ethanol is further increased, this effect becomes more significant and there is a decrease in the peak pressure values, but due to more oxygen being available for combustion with increasing ethanol, the combustion is improved and hence, we see an improvement in the performance parameters of the engine. This is the pattern observed in the present model and the same is observed from the experimental data.

At 3Q load, it is observed in the model that there is hardly any change, or very minimal change, in the performance parameters of the engine. At the other loads, with an increase in ethanol fumigation, the performance parameters are enhanced. This pattern can be observed in the model and in the experiments. This could be due to no change in the ignition delay and combustion duration at the 3Q load condition, and the cooling effects of ethanol would cause a decrease in the temperature of the inlet manifold and then an overall temperature decrease in the combustion chamber, leading to lower peak pressure.

There is a slight improvement in the performance of the engine with ethanol fumigation in the present model; with the brake thermal efficiency increasing from 36% with D100 to 38% with D90 (10% ethanol fumigation), at full load. The increase in the brake thermal efficiency can be attributed to the high laminar flame speed of ethanol, which would result in increased pre-mixed combustion—which is evident from the higher rate of pressure rise and increased in-cylinder pressure. The increased pre-mixed combustion, due to the availability of more homogeneous air-ethanol mixture at higher loads, is also verified by the heat release diagram, Fig 2, with ethanol fumigation of 30% (D70) having the highest HRR. In the D100 (diesel only fuel) case, the model performance parameters for 3Q and half load are very close to the experimental performance parameters, with the variations approximately around 1.5% (approximately 7% for full load). However, the variations slightly increase in all cases of ethanol fumigation for lower loads, but decrease at full load. In the D90 (10% ethanol) case, the variations in the performance parameter are in the range of approximately 4% to 5% for full load and approximately 5% to 6% for 3Q and half load. For D80 (20% ethanol), the variations improve with increasing ethanol fumigation and are in the range of 0.5% for full and 3Q load, and 2% for half load conditions. For D70 (30% ethanol), the variations are in the range of approximately 2% to 3% for full and 3Q load, while it is approximately 8% to 9% for half load. This could be due to the limitations of the combustion model in AVL BOOST when it handles the complexity of the incoming charge, as a fumigated homogeneous mixture of ethanol-air and then pockets of heterogeneous diesel fuel, as diesel is injected into the cylinder to initiate combustion. However, as the ethanol fumigation further increased and a more homogeneous mixture is available for the engine, the combustion (in the model) seems to become more stable, with the variations in the performance parameters coming down for higher ethanol fumigation cases, as compared to D90 (10% ethanol).

Figure 2 shows the heat release rate in the model for different fuel setups (D100, D90, D80 and D70) for full load. D70 shows the highest rate of heat release with the highest in-cylinder peak pressure. The limitations with the HRR diagram can be attributed to two main causes. Firstly, the combustion model used in AVL BOOST has limited capability to handle the complexity of the charge when ethanol is fumigated. Secondly, the combustion duration and other combustion parameters such as the 'shape parameter' – 'm' (m = 6.9 to have complete combustion), with ethanol fumigated cases have been kept the same as D100 in the model. The combustion start timings were slightly advanced for ethanol fumigation cases in the model, according to the experimental data, however, the combustion duration was not changed, as compared to D100. The trend shown in Fig 2 is quite satisfactory, showing the increase in heat release (more rapid pre-mixed combustion phase) with increasing ethanol fumigation. With the availability of a faster burning homogeneous mixture of ethanol-air, the combustion is faster and earlier. Cycle-by-cycle variations and ignition delay could be two very important parameters to investigate the combustion process and its limitations observed in this paper.

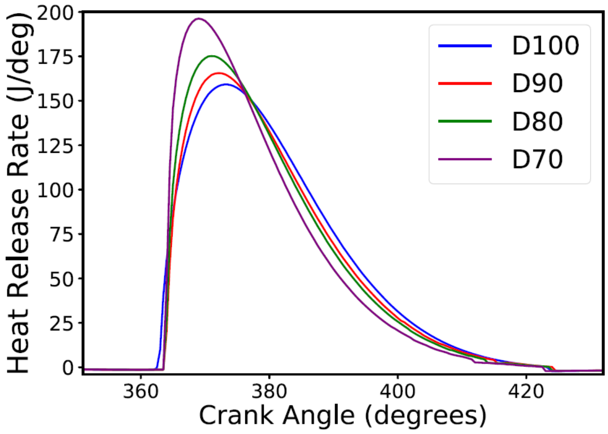

Figure 2: Heat release rate for increasing ethanol fumigation (D100, D90, D80 and D70) for full load, 2000 RPM, as determined by the BOOST model for Cylinder 1.

As the model is developed according to the rate of flow of fuel in the experiments, with ethanol replacing the diesel fuel on the basis of substitution by energy, the fuel energy variation for D100, as well as for all ethanol fumigation cases, between the model and experiments, comes out to be approximately 0.33%.

The combustion model used in the engine model developed in AVL BOOST has limited capability to accurately calculate the NOx emissions. Rather, it gives the trend of the emissions at various operating points, which is also influenced by the model. The multipliers used in the model to influence the emissions have been tuned up for the D100 case, and then maintained to be approximately the same for ethanol fumigation cases to estimate the trend of the NOx emissions. NOx results are shown in Table 1, the NOx trend in the model has been found to be similar to the experiments.

Load	D100 Model	D100 Exp	D90 Model	D90 Exp	D80 Model	D80 Exp	D70 Model	D70 Exp
	(ppm)	(ppm) *	(ppm)	(ppm)	(ppm)	(ppm) *	(ppm)	(ppm) *
Full	72.7		61.5	60.9	62.9	59.8	59.5	59.4
3Q	53.5		47.9	50.9	43.8		45.1	48.6
Half	26.3		32.3	36.5	23.8		23.1	

Table 1: Boost model and experimental NOx results
*some experimental data was not available

4 Conclusions

This modelling work was undertaken to understand the effects of ethanol fumigation in a Euro III, six-cylinder diesel engine running under steady state conditions, at 2000 RPM, under varying load conditions. The model discussed in this paper has been found to be in good agreement with the experimental results in terms of engine performance, combustion parameters, and NOx emissions. It has provided insight into the behaviour of ethanol, when supplied in the fumigated mode, for the engine running under steady state, at various load conditions. The variation of engine performance parameters, combustion parameters and NOx in the model have been well contained within 5% as compared to the experimental data, except for the case of D70, at half load, where the engine performance parameters are more than 5%. The model can serve as a useful tool for further potential research involving more complex transient running conditions of the engine, as well as to investigate the behaviour of fumigating other secondary fuels with diesel. The limitations of the combustion model can be a subject for further study, as there are changes in the combustion process with the use of different fuels, owing to chemical and physical differences.

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Parameter	Engine Speed	NOx	Brake Power	Indicated Power	Indicated Torque	Brake Mean Effective Pressure	Indicated Mean Effective Pressure	Brake Thermal Efficiency	Peak Firing Pressure	Fuel Energy (Diesel)	Fuel Energy (Ethanol)	Total Fuel Energy
Unit	rpm	ppm	kW	kW	Nm	kPa	kPa	%	kPa	kJ	kJ	kJ
D100 Experiment	2006	N/A	157.2	161.6	771.6	1600.4	1648.1	39.03	12418	24.2	0.0	24.2
D100 Model	2000	72	146.9	152	725.8	1498	1550.3	36.27	12656	24.3	0.0	24.3
% variation	-0.3	-	-6.9	-6.3	-6.3	-6.8	-6.3	-7.5	1.9	0.3	0.0	0.3
D90 Experiment	2003	61	158.5	162.2	774.5	1617.5	1641.7	40.20	12471	21.7	1.9	23.6
D90 Model	2000	61.5	151.6	156.6	747.7	1547	1597.1	38.57	12697	21.7	1.9	23.6
% variation	-0.2	1.0	-4.5	-3.5	-3.5	-4.5	-2.7	-4.2	1.7	0.0	0.3	0.1

Table 2: Comparison of engine performance data for D100 and D90 between the model and the experiment at 2000 RPM, full load

References

[1] M. Balat, Energ. Source. 27 (6) (2005) 569-577.

[2] E. Bocci, A. Carlo, and D. Marcelo, Energ. 34 (2009) 689-698.

[3] R. Gilbert, and A. Perl, Energy and Transport Futures, University of Calgary, (2005) 1-96.

[4] M. Qudais, O. Haddad, and M. Qudaisat, Energ. Convers. Manage., 41 (2000) 389-399.

[5] T. Bodisco, S. Low Choy, and R.J. Brown, Appl. Ther. Eng. 60 (1-2) (2013) 79-87.

[6] E. Broukhiyan, and S. Lestz, SAE Technical Paper 811209 (1981).

[7] B. Chauhan, N. Kumar, S. Pal, and J. Du, Energ. 36 (2) (2011) 1030-1038.

[8] G. Prakash, A. Ramesh, and A. Shaik, SAE Technical Paper 1999-01-0232 (1999).

[9] Z. Sahin, and O. Durgun, Energ. Convers. Manage., 48 (7) (2007) 1952-1964.

[10] M. Ghadikolaei, Renew. Sust. Energ. Rev., 57 (2016) 1440-1495.

[11] T. Hayes, L. Savage, R. White, and S. Sorenson, SAE Technical Paper 880497 (1988).

[12] T. Bodisco, P. Trondle, and R.J. Brown, Energ., 84 (2015) 186-195.

[13] D. Jamuwa, D. Sharma, and S. Soni, Energ. Convers. Manage., 115 (2016) 221-231.

[14] AVL BOOST, User guide, Edition, 2010.