

Numerical Investigation of a 4-Stroke, 8-Cylinder Diesel Engine Using a 1-D Code

Emmanuel Anye*

University of Strathclyde, 100 montrose street, Glasgow, G4 0LZ, Scotland, UK

Email: emmanuel.anye-ngang@strath.ac.uk

Abstract

Diesel engines are important for the movement of people and goods. But they are also essential for generating electricity. While it is desirable that engines become, more efficient, reliable, durable and also environmentally-friendly, achieving this requires a better understanding of the complexities surrounding engine combustion processes. Recent technological advancements, including computer simulation have facilitated research and development of new diesel engines. Using a 1-D model of an 8-Cylinder, 4-Stroke diesel engine and applying a vib combustion model, this study presents how in-cylinder processes affect diesel engine performance. The results reveal close proximity of the developed model to test bed measurements of the engine under investigation. Additionally, the findings demonstrate the capability of AVL BOOST to provide advanced models that allow for adequate prediction of engine performance.

Keywords: diesel; engine; 4-stroke; high-speed; simulation.

1. Introduction

Engineering science faces the difficult task of being able to exactly describe technical processes in order to comprehend the behavior of complex systems towards predicting future behavior of such the same in a reliable manner. For internal combustion engines in particular, this difficulty is more evident, making engine simulation and modelling (in an attempt to describe and analyze these complex behaviors) an important topic. This is due to the fact that there is a wide range of engine configurations alongside a variety of analytic techniques that can be used to study engines. Because of the necessity and usefulness of engine modelling, many research laboratories have developed thermodynamic models with varying degrees of scope, complexity and ease of use [1].

* Corresponding author.

Thus engine simulation tools could be either transfer function models, mean value models, zero or one-dimensional (0-D/1-D) models or three-dimensional models (3-D). There is a lot of literature pertaining to the wide usage of engine simulation codes during the design, development and optimization of the reciprocating internal combustion engine [2-4]. 1-D and 3-D codes can be coupled together while carrying out engine simulations. This is advantageous in saving computational time as only the complex processes would be simulated using 3-D while other processes can be done using 1-D. Several studies [5-7] have explored coupling methodology between the 1-D and 3-D interfaces. Essentially, internal combustion engines are specialized plants wherein energy conversion processes take place. The chemical energy that the fuel contains is first converted into thermal energy (within the engine's combustion chamber) and then into mechanical energy.

2. Theoretical basis of engine simulation

For the purpose of this study, a detailed simulation programme (BOOST) developed for the modelling of a complete engine was used to simulate the entire engine cycle, including combustion [9]. The flexibility of the code allows for simulation of a variety of engine configurations, including four-stroke, two-stroke, diesel, gasoline, dual fuel, natural gas etc. This paper presents a complete engine model for a 4-stroke, 8-cylinder direct injection engine to which the BOOST code has been applied in order to study engine performance.

2.1. Equations used in model

The first law of thermodynamics when applied to high pressure cycle of the model suggests that the change of internal energy in the cylinder is equivalent to the sum of piston work, the fuel heat supplied; the wall heat losses and the enthalpy flow due to blow-by. For the model set-up in this study, the cylinder configuration in Figure 1 is used. Applying the first law of thermodynamics to determine the thermodynamic state of the cylinder, the following equation is used:

$$\frac{d(m_c * u)}{d\alpha} = -p_c * \frac{dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - h_{BB} * \frac{dm_{BB}}{d\alpha} + \sum \frac{dm_i}{d\alpha} * h_i - \sum \frac{dm_e}{d\alpha} * h_e - q_{ev} * f * \frac{dm_{ev}}{dt} \quad (1)$$

In addition to equation (1), the mass balance within the cylinder is determined from the algebraic sum of the in-flowing and out-flowing masses i.e.

$$\frac{dm_c}{d\alpha} = \sum \frac{dm_i}{d\alpha} - \sum \frac{dm_e}{d\alpha} - \frac{dm_{BB}}{d\alpha} + \frac{dm_{ev}}{dt} \quad (2)$$

In equations (1) and (2) above, m_c is the cylinder mass, u is the specific internal energy, p_c is the cylinder pressure, V is the cylinder volume, Q_F is the fuel energy, Q_w is the wall heat loss, α is the crank angle, h_{BB} is the enthalpy of blow-by, d_{mi} is the mass element flowing into cylinder, d_{me} is the mass element flowing out of the cylinder, h_i is the enthalpy of the in-flowing mass, h_e is enthalpy of mass leaving the cylinder, q_{ev} is evaporation heat of fuel, f is fraction of evaporation heat from cylinder charge, m_{ev} is evaporating fuel, Q is total fuel heat input.

The model developed in this study assumes internal mixture preparation in order to analyze the gas composition during combustion. It is reliant on the following assumptions: The fuel added to the cylinder charge is immediately combusted; the combustion products mix instantaneously with the rest of the cylinder charge and form a uniform mixture; the A/F ratio of the charge diminishes continuously from a high value at the start of combustion to the final value at the end of combustion.

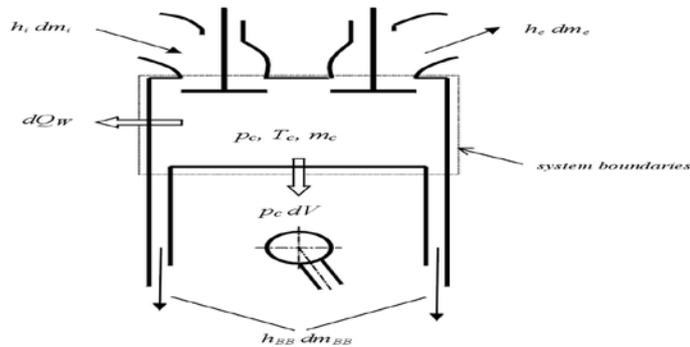


Figure 1: Energy balance in cylinder

For the present study, a simple model for the combustion is used and it involves specifying the rate of heat release. The heat release for any particular engine is determined from the measured cylinder pressure history. For the direct input of the heat release curve, a vibe function is used [10]. The single vibe function is given by:

$$\frac{dx}{d\alpha} = \frac{a}{\Delta\alpha_c} * (m + 1) * y^m * \exp^{-a*y^{(m+1)}} \quad (3)$$

Where a is vibe parameter, m is the shape parameter, $\Delta\alpha_c$ is combustion duration, x is the mass fractioned burned and y is defined by equation (5)

$$dx = \frac{dQ}{Q} \quad (4)$$

$$y = \frac{\alpha - \alpha_0}{\Delta\alpha_c} \quad (5)$$

In equation (4) and (5), Q is total fuel heat input, while α is the crank angle, α_0 is the crank angle at which combustion begins. Integrating the vibe function provides the fuel mass that was burned since the start of combustion i.e.

$$x = \int \frac{dx}{d\alpha} * d\alpha = 1 - \exp^{-a*y^{(m+1)}} \quad (6)$$

Where x is the mass fractioned burned

3. Engine model layout

A 1-D model is created within BOOST by choosing relevant elements and connecting them appropriately as shown in Figure 2. A summary of the elements used in building model is provided in Table1. The model creation and pre-processing is a vital stage in the whole cycle of carrying out an effective simulation. A great deal of input data is meticulously defined in order to set-up the engine configuration. However, compared to setting up a 3-D model, the input data required for 1-D models are generally less. For the 1-D model shown, an initial 1-cylinder model is built and calibrated against measured experimental results from test bed. Subsequently, the entire 8 cylinder is set up and run in identical mode (meaning each cylinder is assumed to have the same details/configuration). Then the Turbocharger, Air cooler, Air cleaner and Catalyst are all added to model. The entire model is then calibrated prior to running the simulation at every time step. Post-processing is done in which cycle dependent results as well as crank angle dependent results are analysed.

Table 1: Elements used in model creation

Element Name	Number Used
Pipe	29
System Boundary	2
Plenum	2
Cylinder	8
Measuring Point	17
Air Cooler	1
Air Cleaner	1
Catalyst	1
Turbocharger	1
Junction	6
Engine	1
Pipe End	64
Assembled	3
All Pipes	32
All Plenums	8
All Boundaries	2

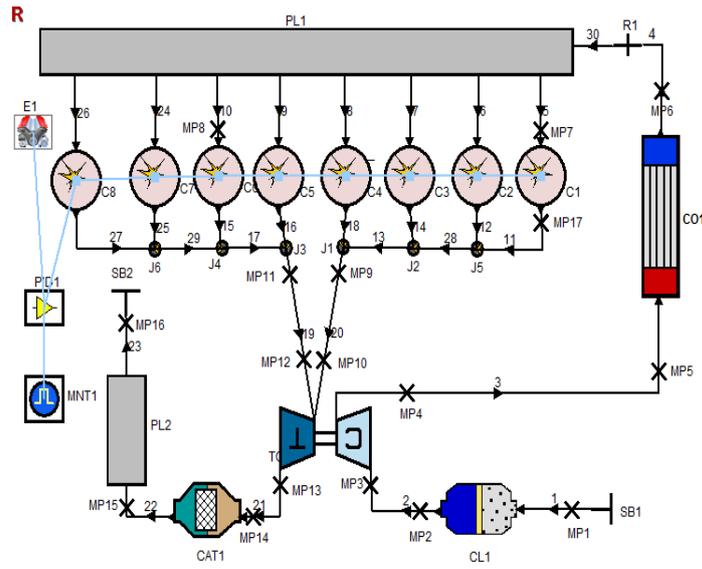


Figure 2: Model engine configuration and layout

The engine modelled in this study is a 4-stroke, high-speed diesel engine. The main engine parameters are given in Table 2 and a very apt description of the engine under study is provided in [11].

Table 2: Main specifications of engine used for study

Engine parameter	Value/Specification
Number of cylinders	8
Cylinder arrangement	60°V
Displacement	34.5L
Bore	170 mm
Stroke	190 mm
Connecting rod length	380 mm
Compression ratio	13
Turbocharger	1 ABB RR-151
Operating speed range	1200 – 1800 rpm
Power output	785 KW @ 1800
BMEP	15.2 bar
Engine total polar moment of inertia	25.9 kgm ²

4. Results and discussion

Some of the results derived from steady state runs were for the engine under consideration are presented in this section. By virtue of the available measured experimental data and for the purpose of comparison, the model was studied for an operating point of 1500 rpm while varying the engine loads.. The results from series calculations for steady-state simulation are then compared against available experimentally measured engine data.

4.1. Mass flow rate through the compressor

The power consumption of the turbo compressor depends on the mass flow rates in the compressor and the enthalpy difference over the compressor (see equation 7). The enthalpy difference is influenced by the pressure ratio, the inlet air temperature and the isentropic efficiency of the compressor. Therefore, to calibrate models, a good starting point is the mass flow rate through the compressor. To achieve the simulation results shown in Figures 3, 4, 5 and 6, the desired pressure ratio at the turbo compressor is specified while the flow resistance of the turbine is automatically adjusted till a satisfactory energy balance is attained across the turbocharger

$$P_c = \dot{m}_c * (h_2 - h_1) \quad (7)$$

In Equation (7), P_c denotes power consumption of the turbo compressor, \dot{m}_c is the mass flow rate through the compressor and $h_2 - h_1$ is the enthalpy difference between the compressor inlet and outlet, respectively.

A full model for turbocharger is generally used in unsteady engine operation simulations. However, for steady state engine performance as carried out, the simplified turbocharger model (which does not consider the variation of turbocharger speed) was used. This model is reported to provide good accuracy for steady state engine operations – a claim supported by several test calculations [13]. The advantage enjoyed with utilizing this simplified model over the full model is that, there is considerable reduction in required input. Only the mean values for the compressor, turbine and mechanical efficiencies have to be specified. This is particularly vital, considering that turbine performance maps are often not provided by turbocharger manufacturers. As shown in Figure 3, there is acceptable agreement between the experimentally measured results and the calculated results from simulation. The results tend to be much more similar for full load engine operation. However, at low load engine operation, the comparison between simulation and experimental results reveals a more noticeable difference.

In Figure 3 (as well as in subsequent Figures), the various load conditions at which the simulation runs were carried out are shown on the horizontal axis. Ranging from low, through medium and eventually full load conditions, point 1 denotes engine operation at very low load (circa 0.002% engine load) while point 6 corresponds to full load (100%) operation. In between the low and full load scenarios, points 2, 3, 4 and 5 were also simulated and they correspond to 23%, 46%, 72% and 85% respectively.

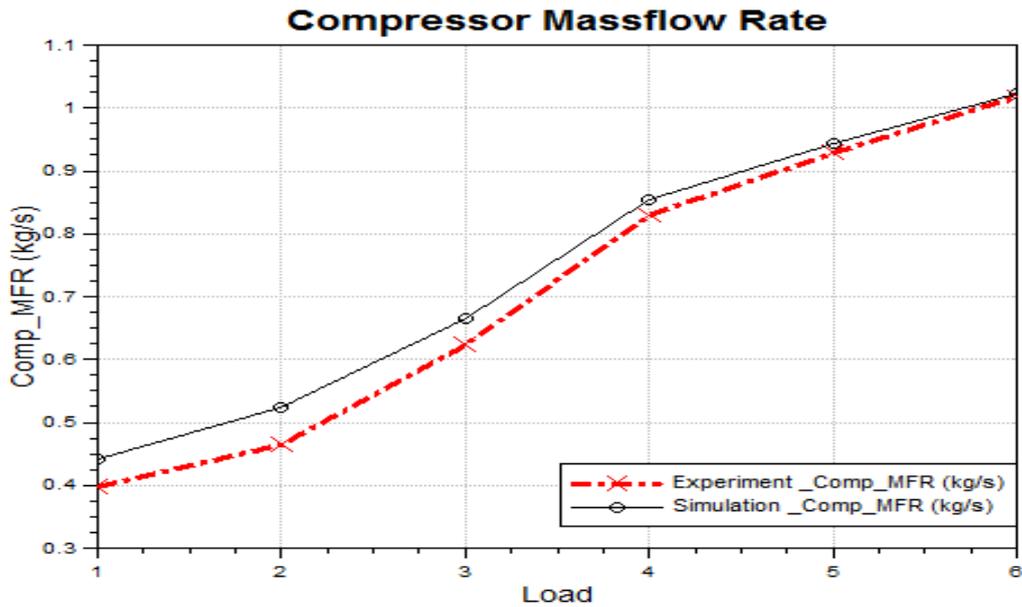


Figure 3: Comparison of compressor mass flow rate

4.2. Gas exchange pressure and temperature

In the simulated model, the air receiver is simulated using the plenum. Using the plenum to model the intake receiver, predicts equal air distribution whereas in reality, this is often a critical issue especially for long receivers with small cross sectional areas. Also in the model developed, it is assumed that the total volume in the receiver is always constant. The plenum model used in this case does not account for pressure waves in the intake receiver. Based on these assumptions and as can be seen from Figure 4, there is good agreement between the measured and the predicted results across the entire engine operating range.

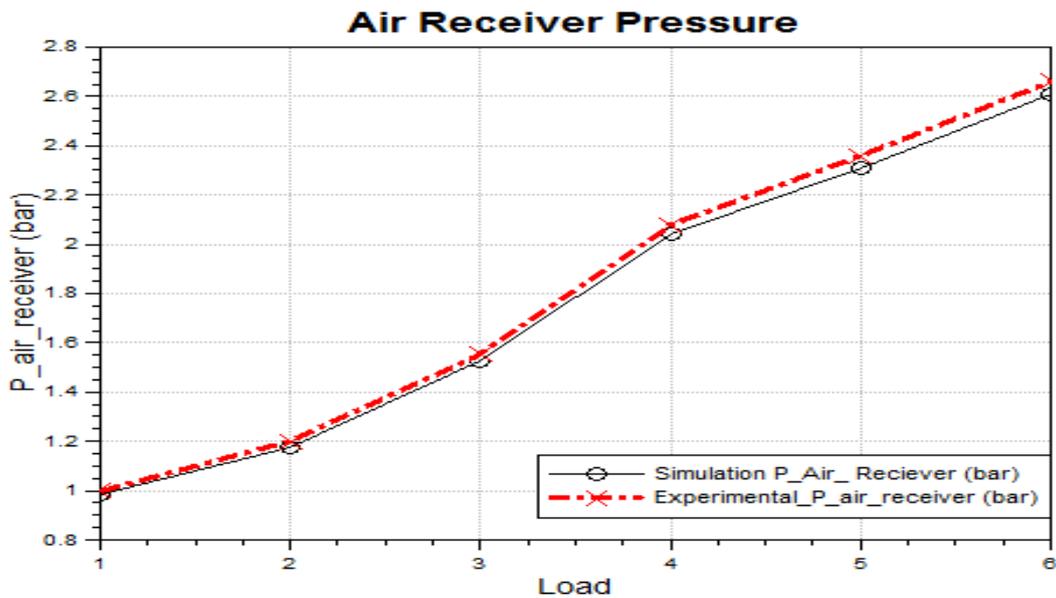


Figure 4: Comparison of pressure in air receiver

The power delivered by the turbine was determined by the turbine mass flow rate and the change in enthalpy within the turbine. The simplified turbine model used in this study acted a pure power supplier for the mechanically connected element.

$$P_T = \dot{m}_T * \eta_m * (h_3 - h_4) \tag{8}$$

Where P_T is the power delivered to the turbine, \dot{m}_T is the turbine mass flow rate, η_m is the mechanical efficiency of the turbocharger and $h_3 - h_4$ is the enthalpy difference between the turbine inlet and outlet respectively.

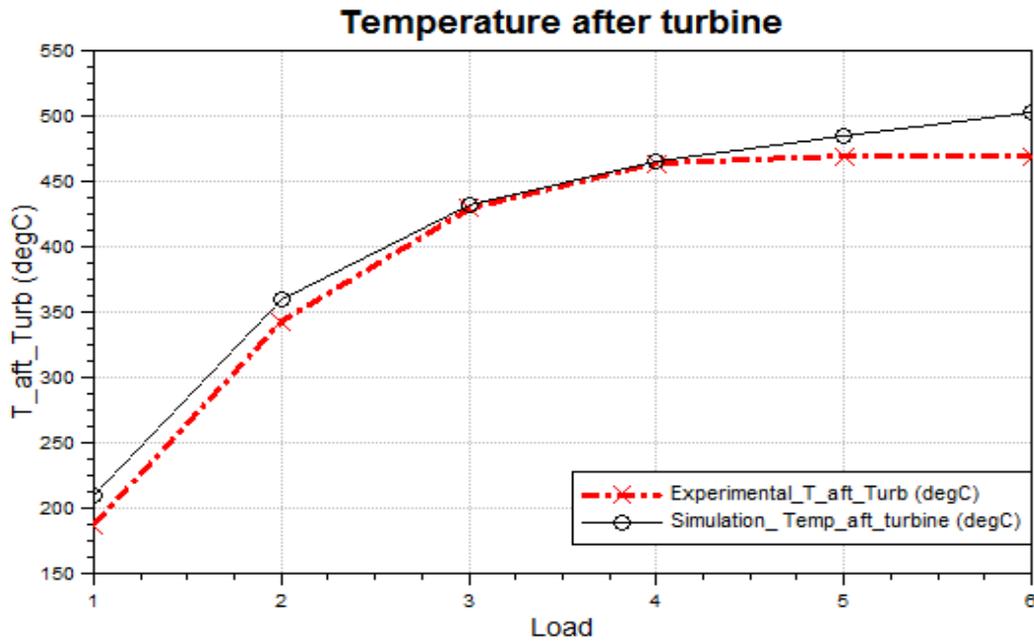


Figure 5: Comparison of exhaust temperature

The predicted results from simulation reveal slightly higher exhaust temperatures than those obtained from experiments. While the comparison between experimental and simulated results suggests acceptable agreement, there is a uniform trend of the exhaust temperatures over the entire engine operating range. At full load operating conditions, the difference between measured and simulated results is somewhat more conspicuous as shown in Figure 5. This slightly higher exhaust temperature can be attributed to the fact that in the simulated configuration, a ‘simplified model’ for the turbocharger is used which does not exactly match the one used when carrying out experimental measurements.

4.3. Brake mean effective pressure and engine power

Several factors are important to an engine user, one of which is the engine’s performance over its entire operating range. This factor, along with several others, controls the total engine costs and also determines whether or not the engine in operation can satisfy environmental legislation. In considering engine performance, the maximum power (or torque) available at each speed within the useful engine operating range is of particular

importance. Additionally, the range of speed and power over which the engine operation is satisfactory are vital for defining engine performance accurately. In the present study, predicted engine power was compared to experimentally measured power for several engine operating points. The power in this case is usable power delivered by the engine to the load.

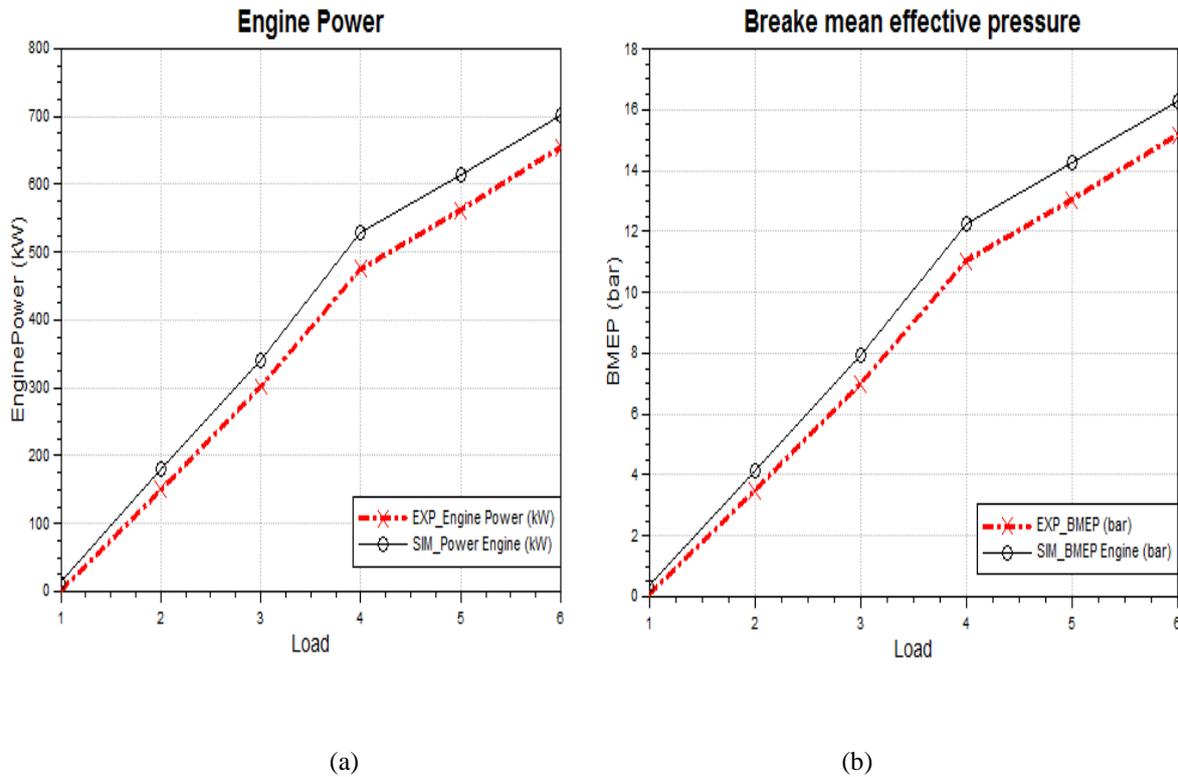


Figure 6: Engine power (6a) and brake mean effective pressure (6b)

The study also compared the predicted brake mean effective pressure (BMEP) with that obtained experimentally. A similar pattern to that of the power is observed. The BMEP is a more useful relative engine performance measure and is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. For the steady-state operating points of the modelled engine, the combustion constants were adjusted during the calibration to ensure that maximum cylinder pressure along with engine power are predicted with sufficient adequacy. Figure 6 reveals that, over the entire engine operating points, there is uniformity of trends for the engine power and brake mean effective pressure. Unlike at high engine load conditions, at low loads, the simulated results are much more close to the measured ones.

4.4. Cylinder pressure characteristics

Pressure information for the air in the cylinder over the entire working cycle of the engine can be used to compute the work transfer from the gas to the piston. The cylinder pressure along with corresponding cylinder volume throughout the engine cycle is plotted on a p-V diagram as shown in Figure 7. From experimentally obtained results, the maximum cylinder pressure for the engine under study was 137 bar. The predicted maximum cylinder pressure as seen in the Figures 7 and 8 provides acceptable agreement. In addition, over the

entire engine operating range, when predicted maximum cylinder pressure results are compared against experimentally obtained ones, significant levels of agreement are observed.

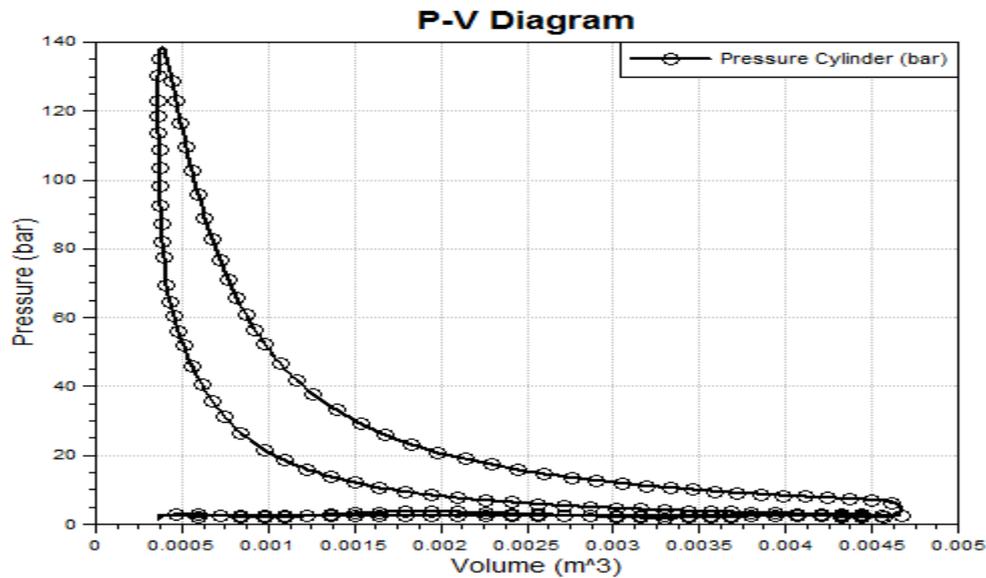


Figure 7: Pressure-volume diagram for engine model at 1500 rpm

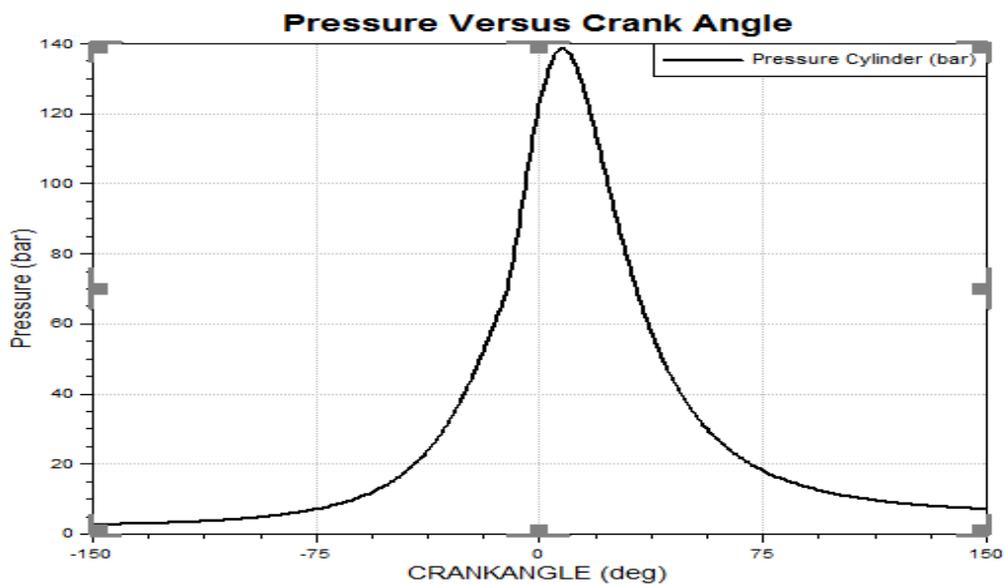


Figure 8: Peak cylinder pressure for engine operating at 1500 rpm and full load

In agreement with expectations from reviewed literature [12], maintaining the engine speed at 1500 rpm while decreasing the engine load, was found to reduce cylinder pressure, leading to eventual loss in engine power (Figure 9). Additionally, the air-fuel equivalence ratios for the different simulation cases are within acceptable limits with the lower ratio observed during the simulation of the highest load scenario.

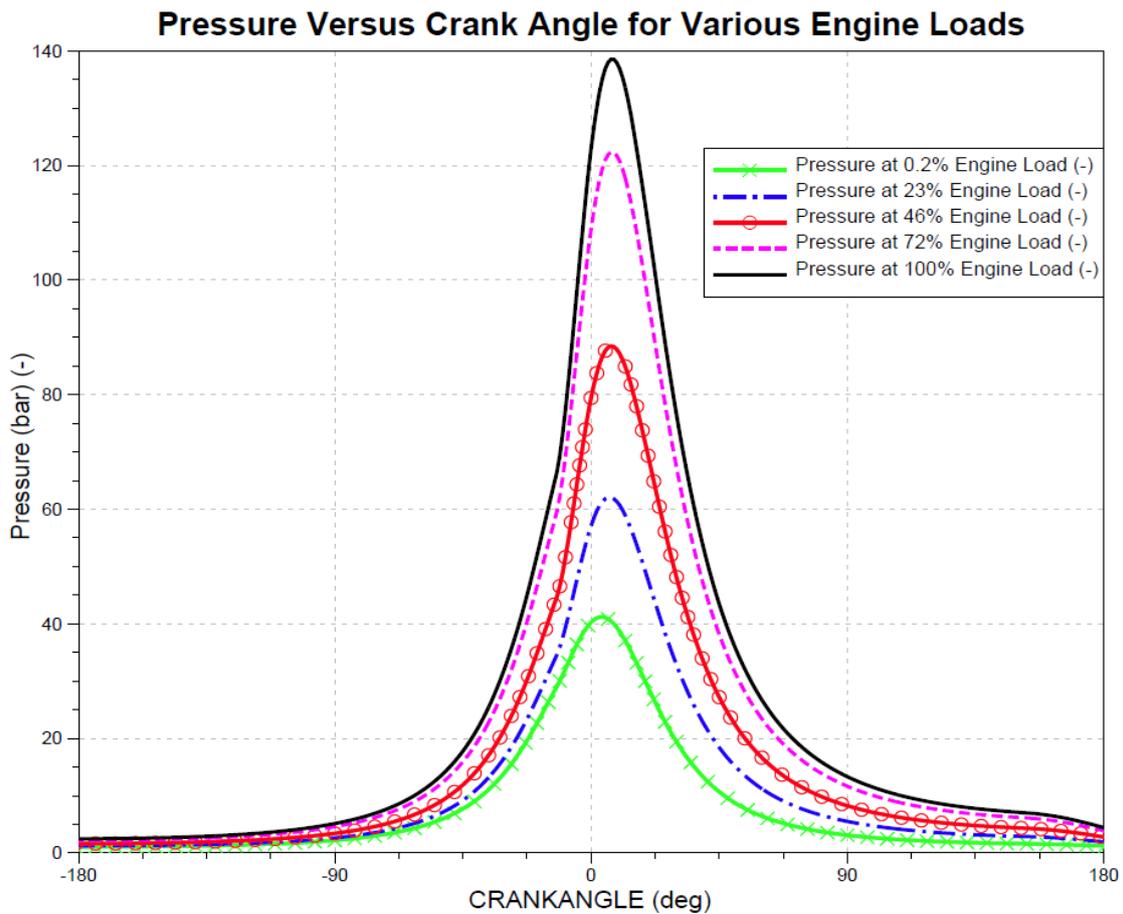


Figure 9: Variation of cylinder pressure for different engine load conditions

5. Conclusion

This study has focused on numerical simulation and analysis of a 4-stroke, high-speed diesel engine. It is relevant in the current context of climate change and stricter environmental legislation pertaining to reducing tail pipe emissions from engines. Understanding the intricate combustion taking place in the diesel engine and how it affects the engine’s performance informs compliance with stringent environmental requirements. Therefore, while providing insight into the fundamentals of the engine combustion, this study predicts the performance of a diesel engine using a fully integrated internal combustion engine simulation tool. This tool, developed within AVL’s applied thermodynamic department, is capable of providing advanced models that allow for adequate prediction of engine performance, tailpipe emissions and acoustics. As an engine cycle and gas exchange simulation software, which also simulates combustion, AVL BOOST allows users to construct a model of the complete engine. Initially, a single cylinder model was built and calibrated with adjustments made to combustion constants (vibe parameters) so as to adequately define the rate of heat release. After calibration of the single cylinder model, the entire engine configuration is set up as shown in Figure 2. Subsequently, a series calculation is then performed to analyse the steady-state engine operation at a constant speed of 1500 rpm while varying both the engine load and fuel mass per cylinder per cycle. Additionally, post-processing the results ensures that comparison is made between experimentally obtained measurements and predicted simulation

results. The foremost conclusion resulting from the current study is that the engine simulation code along with the characteristic combustion sub models can be used to adequately predict behavior and performance even when an unsophisticated turbocharger model is used. Furthermore, the study also supports the fact that, reducing the amount of fuel mass in the cylinder per cycle, leads to a consequent reduction in maximum combustion pressure and an eventual loss in power produced. It is recommended that future studies investigate what happens when a diesel engine is retrofitted to operate as a dual fuel engine. The author equally recommends further investigation of such an engine using more advanced, yet more time consuming 3-D codes.

References

- [1] A. Shamekhi, N. Khatibzadeh and A.H. Shamekhi. "Performance and emissions characteristics investigation of a bi-fuel SI engine fueled by CNG and gasoline." In: ASME 2006 Internal Combustion Engine Division Spring Technical Conference (ICES2006), Aachen, Germany, 2006
- [2] N. Watson and K. Banisoleiman. "Computers in diesel engine turbocharging systems design." In: IMechE, CO5/87, 1987
- [3] T. Sams, G. Regner and F. Chemla. "Integration of simulation tools to optimize engine concepts." In: Motortech. Z., (9). pp.601-608, 2000
- [4] P. Bartsch, P. Prenninger and I. Allmer. "Transient performance optimization of turbocharged engines by means of gas exchange simulations." In: IMechE International conference on turbocharging and air management systems, C554/016/98, pp.237-251, 1998
- [5] A. Pourkhesalian, A. Shamekhi, and F. Salimi. "Alternative fuel and gasoline in an SI engine: A comparative study of performance and emissions characteristics." Fuel, 2010, 89(5), 1056-1063.
- [6] H. Chum and R. Overend. "Biomass and renewable fuels." Fuel Processing Technology, 2001, 71(1-3), 187-195.
- [7] K. Egeback, H. Henke, B. Rehnlund, M. Wallin and R. Westerholm. "Blending of ethanol in gasoline for spark ignition engines – Problem inventory and evaporative measurements." Rapport MTC 5407, 2005
- [8] G. Merker, C. Schwarz., G. Stiesch and F. Otto. "Simulating combustion: Simulation of combustion and pollutant formation for engine-development." Berlin: Springer, 2006
- [9] L. Grabowski, K. Pietrykowski and M. Wendeker. "AVL simulation tools." Graz: Politechnika Lubelska, 2012
- [10] I.I Vibe. "Brennverlauf und Kreisprozess von Verbrennungsmotoren", Verlag Technik, Berlin, 1970
- [11] G.Theotokatos and N. Kyrtatos. "Investigation of a large high- speed diesel engine transient behavior including compressor surging and emergency shutdown." J. Eng. Gas Turbines Power, 2003, 125(2), 580.
- [12] J. Heywood. "Internal combustion engine fundamentals." McGraw-Hill, 1988