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Research Article

Exploring the feasibility of dimethyl ether (DME) and LPG fuel blend for small diesel engine: A simulation perspective

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Abstract. There is a looming global crisis owing to the increase in greenhouse gases and the escalating fossil fuel process. The issue is further compounded by the ongoing conflicts in different places in the world. Hence, there is an urgent need for a bouquet of alternative fuels suitable to power the incumbent internal combustion engine. Among various options available Dimethyl Ether (DME) is a friendly environment fuel, easy to liquefy, and suitable for use in diesel engines, while Liquefied Petroleum Gas (LPG) is another potential alternative fuel suitable for internal combustion engines. The present study is an endeavor to investigate the characteristics of a diesel engine powered with DME-diesel blends as pilot fuel while LPG was used as the main fuel. During engine testing, different blends of diesel-DME were used containing 0%, 25%, 50%, and 75% DME. The AVL Boost software was employed for modeling the engine performance and tailpipe emission. The test fuel combination was successful in running the engine sans any abnormality in sound or performance. The results showed carbon monoxide (CO) and hydrocarbon (HC) emissions were reduced using the test fuel combination while there was a marginal increase in the oxides of nitrogen (NOx) levels. In general, the combination of DME and LPG could be considered as a potential and promising solution to reducing pollutant emissions.

Keywords: Dimethyl Ether; Alternative fuel; Emission characteristics; Liquefied Petroleum Gas; AVL-Boost; Engine performance

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1. Introduction

As the population and economic growth of emerging nations continue to rise, the majority of these countries are confronted with a growing need for energy, particularly in the transportation sector (Changxiong *et al.*, 2023; Minchev *et al.*, 2023). The use of energy for transportation has been restricted, and new alternatives have been introduced to replace the fossil fuels that are now in use (Said *et al.*, 2023; Sharma and Sharma, 2021; Sun *et al.*, 2021). Indeed, internal combustion engines use fuel such as gasoline and diesel fuel. However, means of transport often use compression ignition engines with diesel fuel, which is a highly efficient heat engine (Bhowmik *et al.*, 2018; V. G. Nguyen *et al.*, 2023; Tran *et al.*, 2023). As reported, the use of fossil fuel for engines has released a huge of pollutant emissions into the environment (Bakır *et al.*, 2022; Menon and Krishnasamy, 2018). Thus, there has been a significant amount of research carried out on alternative fuels as a result of the growing need for reduced levels of energy consumption and exhaust emissions in recent years (V. N. Nguyen *et al.*, 2023; Sharma and Sharma, 2022; Wei *et al.*, 2023). Alternative fuels to diesel engines have received increasing attention in recent years due to concerns about environmental pollution and the limited nature of fossil fuels (Geng *et al.*, 2017; Serbin *et al.*, 2022; Zhao *et al.*, 2020).

In the literature, many works are studying the use of alternative fuels for internal combustion engines. Indeed, alternative fuels such as compressed natural gas (CNG) (Hoang *et al.*, 2023a; Wategave *et al.*, 2021), biogas (Bui *et al.*, 2022; Sharma *et al.*, 2022b), liquefied natural gas (LNG) (Fosudo *et al.*, 2024; Olszewski *et al.*, 2023; Pan *et al.*, 2019), and liquefied petroleum gas (LPG) (Ashok *et al.*, 2015; Organ *et al.*, 2020; Ortega *et al.*, 2021), biodiesel (Jit Sarma *et al.*, 2023; Tuan Hoang *et al.*, 2021; Wei *et al.*, 2024), hydrogen (Hoang *et al.*, 2023b; Murugesan *et al.*, 2022), ethanol/methanol/butanol (Liu *et al.*, 2017; Sun *et al.*, 2019; Veza *et al.*, 2022), and ether (Kamei *et al.*, 2021; Zhang, 2023) are becoming more popular among researchers as the targets of their fuel study. These alternative fuels offer promising alternatives that can help reduce emissions of harmful pollutants and reduce the impact of transportation on climate change (Barid and Hadiyanto, 2024; Moorthi *et al.*, 2022). Among the above-mentioned alternative fuels, biodiesel is found as the primary alternative one for diesel engines (Hoang, 2021; Hoang *et al.*, 2022b), which is primarily derived from renewable sources such as vegetable oils, animal fats, algae, or recycled cooking grease/oil (Dong and Sharma, 2023; Le *et al.*, 2024a, 2024b; V. G. Nguyen *et al.*, 2023a; V. N. Nguyen *et al.*, 2024). Biodiesel can be blended with conventional diesel fuel in use or has been used as a clean fuel to reduce greenhouse gas emissions and rely on fossil fuels (Hoang *et al.*, 2021; Jain *et al.*, 2023; Nayak *et al.*, 2022; Wang *et al.*, 2023). In addition, biomass-originated biofuels are also considered as potential

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alternatives to fossil fuels since biomass could be used to produce liquid biofuel (Hoang *et al*., 2022; Hoang and Pham, 2021; Nguyen *et al*., 2024; Rahman *et al*., 2023) or gaseous biofuels (Fetriyuna *et al*., 2024; Hasibuan *et al*., 2020; Utami *et al*., 2019). Another promising option is bio-oil, which is produced through the process of converting biomass, agricultural residues, or wastes, into high-quality alternatives (Aysu *et al*., 2017; Nguyen and Le, 2023; Varma and Mondal, 2017). Bio-oil boasts better combustion properties, gas low emissions, and is compatible with existing diesel systems, making it an attractive option for reducing the carbon footprint of diesel fuel-powered vehicles (Rajamohan *et al*., 2022; Yusuf *et al*., 2023).

Among the aforementioned alternative fuels, dimethyl ether (DME) has been emerging as a promising one since the large-scale manufacture of DME has been shown to have potential (Li *et al*., 2020; Vinkeloe *et al*., 2022). Due to the fact that it can be derived from a diverse range of sources and has exceptional physical, chemical, and storage qualities, DME is garnering a lot of interest as a potential source of energy for the 21st century (de Jong *et al*., 2023). As a result of the rapidly growing need for fuel in Asia, both for families and for transportation, DME is showing a great deal of promise as an alternative fuel (Gao *et al*., 2020; Işık *et al*., 2020). The toxicity of DME is minimal, and its formation of ozone via photochemical reactions is comparable to that of LPG (Lee *et al*., 2011, 2009). This is because DME is a suitable propellant. At first look, DME seems to be a fantastic and effective alternative fuel for diesel engines, with combustion that is virtually completely smoke-free (Arcoumanis *et al*., 2008; Ga and Thai, 2020). The reason for this is not only because it has a low auto-ignition temperature and that it vaporizes fairly instantly, but it is also since the molecular structure does not have a straight carbon-carbon link and that the fuel contains oxygen at a percentage of around 35 percent by mass (Arcoumanis *et al*., 2008). In addition, emissions from using DME were shown to be lower than diesel emissions when the fuel delivery system was constructed appropriately (Hagos *et al*., 2019; Sharon *et al*., 2012; Zhao *et al*., 2014). The use of DME as an additive or an ignition enhancer even makes it possible to employ a variety of alternative fuels for normal diesel combustion, dual fuel operation, and HCCI operation (Mohan *et al*., 2017; Wang and Yao, 2020; Wang *et al*., 2016a).

The literature review shows that researchers have been exploring the use of DME in IC engines as a potential substitute for diesel. Li *et al.* (Li *et al*., 2020) investigated the use of DME in a gasoline engine using the concept of micro-flame ignited combustion. The influence of both the single as well as double DI DME methods on combustion parameters and flame formations was explored using lambda 2.0. The findings demonstrate for a DI DME, the heat release mechanism comprises DME oxidation processes (stage I) and DME-gasoline hybrid combustion (stage II). Regardless of DI techniques, ignition timing lags and heat release decreases in stage I. In another study, Mathan Raj *et al.* (Mathan Raj *et al*., 2021) employed the blends of di-methyl carbonate and DEE in a common rail diesel injection-based engine. At higher load circumstances, the BTE of the engine improved. It produced superior findings in terms of CO and hydrocarbon emissions, but the DEE specimen at 600 bars pressure produced less smoke emissions. Olsen *et al.* (Olsen *et al*., 2007) employed DME as pilot fuel, and varying flow rates of natural gas were employed as the main fuel in a diesel engine. The BTE of the engine improved with a higher flow rate of natural gas. In another study by Chen *et al.* (Chen *et al*., 2021), two different types of blends i.e., diesel/polyoxymethylene DME/methanol and diesel-methanol. The engine was tested at different operational settings and flow rates. According to the findings, the test blend showed inferior HRR and greater peak cylinder pressure. Following the switch from diesel to P50 fuel injection for the pilot mode, the dual-fuel engine saw a reduction concerning the ignition delay and the length of the combustion process. In addition, the NOx as well as PM that were generated were lower compared to the ones that were released by the diesel/methanol engine for engine load. Gao *et al.* (Gao *et al*., 2020) conducted a numerical simulation of an engine powered with polyoxymethylene dimethyl ether-diesel. Special emphasis was on the formation of soot and nitrogen oxide (NO) formation. The findings indicate that the temperature within the cylinder as well as the percentage of oxygen atoms had a significant contribution to the emissions that are produced by fuels that are a combination of PODE and diesel. Within the cylinder, PODE contributes to an increase in the average combustion temperature as well as the oxygen atom distribution range, which ultimately results in a reduction in the amount of lean combustion zone with inconsistent mixture (Marković *et al*., 2024). The ignition delay time of PODE-diesel mix fuel is decreased as a result of the inclusion of PODE, which increases the rate at which the combustion process occurs overall (H. Chen *et al*., 2019). The blending proportion of PODE is raised, which results in an increase in the production of NO and the distribution of NO concentrations. At the same time, the emissions of soot are greatly reduced, and the final soot production from P30 fuel under calibrated operating conditions is reduced by approximately 80 percent in comparison to diesel.

In the domain of LPG and its use in diesel engines in dual-fuel mode, diesel or biodiesel or their blends is used as pilot fuel while LPG is used as a main fuel (Guan *et al*., 2017). Mohsen *et al.* (Mohsen *et al*., 2023) investigated the effects of carrying the compression ratio on the engine's emission and performance. With the assistance of the Diesel-RK simulation program, the numerical analysis was carried out for this engine operation using the multizone combustion model. It was reported that when the rate of LPG was raised a reduction in BTE and EGT values was observed. Up to 18.8% reduction in CO and an 8% reduction in HC was observed, at 20 lpm and 15 lpm flow of LPG, respectively. It was also noted the LPG induction helped in NOx emissions reduction. At the higher CR ratio, a positive impact on BSFC was observed. This was attributed to a higher cylinder pressure as well as combustion temperature, which has the effect of decreasing the delay period, igniting fuel quickly, and producing more power in a shorter amount of time. When the compression ratio was raised, there was a significant reduction in CO and HC, but there was an increase in NOx. The effects of adding an oxygenated additive to a dual-fuel engine with biodiesel as pilot fuel and LPG as the main fuel were investigated by Shaor *et al.* (Shoar *et al*., 2021). At the dual-fuel engine operating in NG70 mode, the optimal circumstances were achieved by including a 0.2 vol% TGME addition into the diesel fuel to provide the best possible results. Both the braking power and the brake thermal efficiency rose by 10.54 with 12.77%, respectively, as compared to the typical diesel combustion method under similar circumstances. Nevertheless, there was a 25.16 percent drop in the cost of the quantity of electricity that was produced. Furthermore, there was a decrease in emissions of carbon dioxide (CO2), carbon monoxide (CO), and nitrogen oxides (NOx) by 76.77, 40.9, as well as 1.31%, respectively.

The research study ascribed here covers critical issues with diesel engine emissions while also investigating the use of alternate fuels. The emissions from diesel engines include toxic substances such as CO, HC, NOx, and particulate matter, which contribute to environmental pollution and health concerns. As a result, the current project investigates the ability of alternate fuels to alleviate these concerns. The emphasis is on DME and LPG to power an existing diesel engine. DME generated from a range of raw materials, including biomass and natural gas, is a potential alternative with plentiful resources in Vietnam. Similarly, LPG (propane and butane) is an alternate fuel generated using the liquid method. The study seeks to evaluate the feasibility and efficiency of DME-LPG fuel blends at various parameters by simulating a small diesel engine with AVL-Boost software.

2. Theoretical basis of AVL boost software

The AVL Boost program is a specialized software application that makes it possible to simulate the duty cycles and thermodynamic processes that are inherent in internal combustion (IC) engines that are powered by a variety of fuel sources. For this investigation, the software is utilized exclusively for simulating engines that are powered by DME, which enables a full investigation of the performance characteristics of these engines.

2.1. Thermodynamics equation

The process of combustion is an irreversible change that takes place within the context of an internal combustion engine. It is responsible for transforming chemical energy through thermal energy (Kirkpatrick, 2020; Semin *et al.*, 2020). It is necessary to have a sophisticated understanding of the intermediary reactions that orchestrate the shift from the original fuel mixture to the eventual combustion byproduct to have complete knowledge of the medium's condition during this process. Throughout human history, the majority of examples of such reactions have been explained for fundamental fuels such as hydrogen and methane. Nevertheless, regardless of the complexity of the fuel, the fundamental laws of thermodynamics, in particular the first law, offer a framework that may be utilized to determine the link between the starting and final states of the combustion process (Awad *et al.*, 2013; Heywood, 2018; Prah and Katrašnik, 2009).

$$\frac{d(m_c u)}{d\alpha} = -p_c \frac{dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_W}{d\alpha} - h_{BB} \frac{dm_{BB}}{d\alpha} + \sum \frac{dm_i}{d\alpha} h_i - \sum \frac{dm_e}{d\alpha} h_e - q_{ev\,.f} \cdot f \cdot \frac{dm_{ev}}{dt} \qquad (1)$$

Herein, $\frac{d(m_c u)}{d\alpha}$ denotes internal energy change, while $-p_c \frac{dV}{d\alpha}$ cycle work done, (J/degree) $\frac{dQ_F}{d\alpha}$ represents the heat input (J/degree), $\sum \frac{dQ_W}{d\alpha}$ heat loss through the wall (J/degree), $h_{BB} \frac{dm_{BB}}{d\alpha}$ denotes the enthalpy loss due to air leakage (J/degree), MC represents the mass of the medium inside cylinder (kg), u here is internal energy (J/kg), pc denotes pressure inside the cylinder in the bar, V denotes the volume of cylinder volume in m^3, QF denotes the heat of fuel supplied in Joules, Qw is loss of heat to the wall in Joules, α is the crankshaft rotation angle in degrees, hBB denotes the enthalpy value of trapped air in J/kg.

The difference in mass that occurs within the cylinder may be determined by calculating the whole mass volume that enters and leaves the cylinder for each individual (Nugroho *et al.*, 2023; Park *et al.*, 2019). Eq. (1) may be utilized for both internal and exterior mixture formation engines of the same application. On the other hand, the two examples described above each have a different change in the composition of the combination. In the case of the process of mixture formation inside the cylinder, there are a few assumptions that can be made: The fuel that is supplied into the cylinder is burned immediately; The combustion mixture is mixed immediately with the residual gas that is present in the cylinder; The A/F ratio reduces continually from a high value starting point of the combustion process to a low number at the end of the process. Following the change, Eq. (1) may be transformed into Eq. (2) as follows (Chan Nguyen *et al.*, 2019; Qadiri, 2023):

$$\frac{dT_c}{d\alpha} = \frac{1}{m_c\left(\frac{\partial u}{\partial T} + \frac{\partial u}{\partial p} \cdot \frac{p_c}{T_c}\right)} \left[\frac{dQ_F}{d\alpha}\left(1 - \frac{u_e + \frac{\partial u}{\partial p} p_c}{H_u}\right) - \frac{dQ_w}{d\alpha} - \frac{dm_{BB}}{d\alpha}\left(h_{BB} - u - \frac{\partial u}{\partial p} p_c\right) - m_c \frac{\partial u}{\partial \lambda} \frac{\partial u}{\partial p} - p_c \frac{dV_a}{d\alpha}\left(1 - \frac{\partial u}{\partial p} \frac{m_c}{V_c}\right)\right] \qquad (2)$$

Herein, T_c *denotes cylinder temperature in K,* u_c *is the* specific internal energy of the mass of medium inside the cylinder (J/kg), also H_{uh} is low calorific value (J/kg), λ *denotes coefficient of* air residue.

Several factors, including pressure, temperature, and the composition of the gas mixture, are required to solve the equation shown above. These factors include the process of the combustion model, the heat dissipation law, and the heat transfer process via the cylinder wall. The link between pressure, temperature, and density may be established by using the equation of state in conjunction with the following (Bellér *et al.*, 2021; Ling and Abas, 2018):

$$p_c = \frac{1}{V} m_c R_c T_c \qquad (3)$$

The solution of Eq. (3) offers the estimation of pressure through the equation of state.

2.2. Combustion model

Many alternative models exist that explain the combustion process in diesel engines. AVL created the AVL MCC fire model to generate fire rules using a mix of the Vibe combustion model and a fire model that takes into account the turbulent kinetic energy of the fuel jet. A diesel engine's combustion process is divided into four stages: the ignition delay time frame, premixed combustion phase, mixed-controlled combustion phase, and late combustion phase. Though, the heat generated by the burning fuel is mostly in the fast and main combustion phases; the falling combustion stage emits nearly no heat, and the rapid burning stage emits very little heat; this heat does not generate work. Consequently, the combustion process may be characterized in Eq. (4) (Bellér *et al.*, 2021; Chan Nguyen *et al.*, 2019; Qadiri, 2023):

$$\frac{dQ_{total}}{d\alpha} = \frac{dQ_{MCC}}{d\alpha} + \frac{dQ_{PMC}}{d\alpha} \qquad (4)$$

Herein, $\frac{dQ_{total}}{d\alpha}$ denotes the total heat in the cylinder, $\frac{dQ_{MCC}}{d\alpha}$ denotes the change in temperature during the main combustion phase, $\frac{dQ_{PMC}}{d\alpha}$ represents the change in temperature during the rapid burring phase of combustion.

This stage occurs immediately after the delayed combustion. The air mixture created in the previous stage burns rapidly, causing the pressure and temperature in the cylinder to surge. Because the heat release rate is high while the cylinder capacity remains constant, the quick combustion phase is similar to the isovolumetric heat supply. The heat release rate, given in Eq. (5) for this level is estimated using the Vibe formula (Park *et al.*, 2019; Prah and Katrašnik, 2009):

$$\frac{\frac{dQ_{PMC}}{Q_{PMC}}}{d\alpha} = \frac{a}{\Delta\alpha_c} \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{(m+1)}} \quad (5)$$

Herein, $y = \frac{(\alpha - \alpha_{id})}{\Delta\alpha_c}$

The main combustion stage occurs after the quick burning stage and involves mixing the gas while preparing and burning, resulting in a slow-burning process known as diffusion (Nour *et al.*, 2022)(Z. Sun *et al.*, 2021). Burning speed is determined by the speed at which fuel and air are mixed or prepared, resulting in a smoother combustion process (Zare *et al.*, 2019). This step is analogous to the isobaric heating process, with the burning speed reducing as the oxygen content falls. Thus, although this process runs smoothly, the efficiency of turning heat into work is low. In actuality, around 50-60% of the fuel cycle burns at this stage. In order to provide accurate predictions regarding the parameters of combustion in direct injection as well as self-ignition engines, the AVL_BOOST program makes use of the AVL Mean value combustion (MCC) model. As stated in Eq. (6), the process of heat release is adjusted by modifying the fuel grade and the density of the turbulence (Gürbüz, 2020; Wang *et al.*, 2020).

$$\frac{dQ}{d\emptyset} = C_{Mod} \cdot f_1 \cdot (M_F Q) \cdot f_2 \cdot (k, V) \quad (6)$$

Herein, $f_1(M_F Q) = M_F - \frac{Q}{LCV}$ and $f_2(k,V) = \exp(C_{rate}\frac{\sqrt{k}}{\sqrt[3]{V}})$, while in this case, C_{Mod} denotes model constant [kJ/kg.0TK], constant of mixing rate is denoted with C_{rate} [s], k represents the density of local kinetic energy [m^2/s^2], M_F denotes mass of fuel supplied [kg], LCV represents the lower calorific value [kJ/kg], Q denotes heat release (accumulated) in [kJ], V is the instantaneous cylinder volume [m^3], $\emptyset$ denotes the crankshaft angle.

The function f1 is dependent on the quantity of fuel that is injected so the amount of fuel that is burnt. The function f2 assesses the mixing of air and fuel during this interval, and as a result, it is heavily impacted by kinetic energy. Kinetic energy may be produced via vortices, jets, and vortices, with jets being the primary source of kinetic energy generation. Two components make up the spray's kinetic energy: the kinetic turbulence energy and the dissipated energy. The usable energy that contributes to the improvement of air-fuel mixing is known as kinetic turbulence energy. The energy that is lost as a result of the exchange of energy between the surface of the fuel particle and the air that surrounds it is referred to as dissipated energy (Köten *et al.*, 2020; Vasudev *et al.*, 2022; Villenave *et al.*, 2024). Using Eq. (7) can determine the total amount of kinetic energy that is produced by the spray:

$$\frac{dE_u}{d\alpha} = \frac{dE_i}{d\alpha} - \frac{dE_{diss}}{d\alpha} \quad (7)$$

Herein, E_u denotes Turbulent kinetic energy created by the spray jet at a crankshaft rotation angle of α. E_{diss} refers to kinetic energy dissipation and E_i represents total kinetic energy, as given in Eq. (8) and Eq. (9):

$$dE_i = C_{turb} \cdot \frac{1}{2} \cdot dm_f \cdot v^2 \quad (8)$$

$$\frac{dE_i}{d\alpha} = C_{turb} \cdot 18 \cdot \rho_f \cdot \left(\frac{n}{C_d A_n}\right)^2 \left[\frac{1}{\rho_f}\frac{dm_f}{d\alpha}\right]^3 \quad (9)$$

In the above, the following variables are included as $\frac{dE_i}{d\alpha}$ denotes the kinetic energy of the fuel jet (J/degree), C_{turb} denotes the spray's kinetic turbulence energy, n represents the engine speed, ρ_f denotes the fuel density, C_d denotes the flow coefficient, $A_n$represents the area of the nozzle's cross-sectional area, and $\frac{1}{\rho_f}\frac{dm_f}{d\alpha}$ denotes the quantity of injected fuel in the cylinder concerning the crank angle.

The combustion delay stage's usual factors are the combustion delay time (s) or the combustion delay angle (degrees). These rely on the fuel's features and make-up, such as its cetane number, viscosity, and so on. The ignition delay time is additionally influenced by things like the pressure and temperature in the cylinder during the time of input, the fuel spray characteristics, fuel properties, how turbulent the medium is, and so on (Zhai *et al.,* 2021). As a result, it is hard to look at combustion delay using all the variables and factors that are used to run the program are too hard to understand. To put it more simply, it may alter the delay in the ignition, calibration factor, and combustion parameter in case different fuels are tested. In general, the above models' factors need to be changed based on the results of experiments.

2.3. Heat transfer model

The heat transmission process through the combustion chamber to the cylinder wall and back is estimated using the heat transfer Eq. (10) (Miao and Milton, 2005).

$$Q_{wi} = A_i \cdot \alpha_i \cdot (T_c - T_{wi}) \quad (10)$$

Herin, Q_{wi} denotes the heat transferred towards the wall of the cylinder, cover, and piston, A_i denotes the piston's heat transfer area, heat transfer area of cylinder, heat transfer area in the case of the engine cover,α_i being coefficient of heat transfer, T_c is medium's temperature, T_{wi} is temperature the wall temperature (Mikalsen and Roskilly, 2009; Yu *et al.*, 2020). To determine the heat transfer coefficient in Eq. (5) for diesel engines, the Woschni model is suggested as appropriate in Eq. (11) (Mikalsen and Roskilly, 2009; Rizwanul Fattah *et al.*, 2014; Yu *et al.*, 2020).

$$\alpha_w = 130\, D^{-0.2} p_c^{0.8} T_c^{-0.53} \left[C_1 . c_m + C_2 \frac{v_D . T_{c1}}{p_{c,1} V_{c,1}} (p_c - p_{c,0})\right]^{0.8} \quad (11)$$

Herein, C_1 = 2.28 + 0. 308.cu/cm, C_2 = 0.00324 in the cases of direct injection engines, D denotes the diameter of the cylinder, c_m denotes the mean speed of the piston, p_c is the pressure at the time of inlet valve closing while $p_{c,0}$ denotes inverted cylinder pressure.

2.4. Suction and discharge model

The throttling flow equation across the small gap is used to compute the quantity of intake and exhaust air. The flow-suppressing coefficients are taken into consideration in the following Eq. (12):

$$\frac{dm}{dt} = A_{eff} . p_{01} . \sqrt{\frac{2}{R_0 . T_{01}}} . \psi \quad (12)$$

Herein, the $\frac{dm}{dt}$ denotes incoming and exhaust airflow, A_{eff} denotes area through, p_{01} denotes the pressure of the medium just before the throttling process, R_0 represents the gas constant, ψ denotes the coefficient of current, which can be expressed in the case of negative lines, as given in Eq. (13):

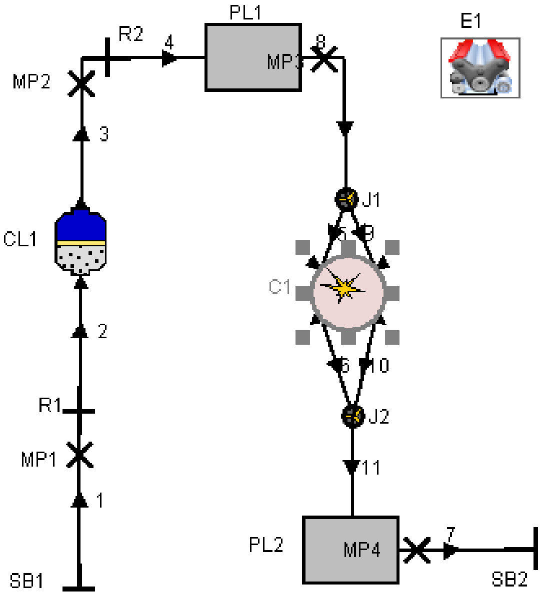

Fig. 1. Kubota RT140 engine model

$$\psi = \sqrt{\frac{\kappa}{\kappa-1}\cdot\left[\left(\frac{p_2}{p_{01}}\right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_{01}}\right)^{\frac{\kappa+1}{\kappa}}\right]} \quad (13)$$

2.5. Model for air leakage

The quantity of air leakage is computed based on the piston-cylinder clearance and the mean pressure in the crankcase. The air leak area is computed utilizing the following Eq. (14):

$$A_{eff} = D \cdot \pi \cdot \delta \quad (14)$$

Herein, the A_{eff} denotes the area of air leakage, while D is the diameter of the cylinder, δ denotes the clearance between Piston – cylinder.

One may choose either the AVL MCC fire model or the heat transfer model. Woschni 1978 in simulation calculations since these models are appropriate for the Kubota RT140 engine, which is a unitary combustion chamber CI engine with direct injection, and the simulation mode is set to full load. For this investigation, the AVL Boost program is used to compute and model the technical characteristics of the Kubota RT140 engine, as shown in Figure 1, which is powered by a combination of DME and LPG engines.

Fig. 2. Kubota RT140 engine

Table 1
Specifications of test Kubota RT140 engine

Parameters/symbols	Value
Dia. of cylinder (D)	97 mm
Working volume (Vh)	709 ml
Compression ratio (ε)	18:1
Piston stroke (S)	96 mm
Rated capacity	11 kW @ 2400 rpm
Maximum torque (Me-max)	42 Nm @ 1400 rpm
Early injection angle (⊠s)	25 °
Injection pressure	240 bars
Fuel supply system	Mechanical type

2.6. Kubota RT140 engine specifications and parameters

The Kubota RT140 engine, as shown in Figure 2, is a reliable and powerful diesel engine that offers dependable performance and good torque. It is manufactured by Kubota Corporation, a renowned Japanese company with a reputation for agricultural and industrial machinery. This engine features a single-cylinder configuration, making it compact and versatile. With its robust construction and reliable performance, the Kubota RT140 engine has been popular and widely used in Vietnam, especially in the agricultural sector, to power small tractors, water pumps, and other farm machinery. In addition, it is commonly used in generators to provide reliable backup power in rural and remote areas. The engine's versatility, durability, and fuel efficiency make it a popular choice for farmers, mine operators, and businesses throughout Vietnam. The main specification of the test engine is listed in Table 1.

2.7. Build the model in AVL_BOOST software

To compute the volume of fuel that is provided to a cycle, experimental observations, and the following expression are employed to estimate the quantity of fuel that is given to the cycle, as shown in Eq. (15):

$$V_{ct} = \frac{N_e \cdot g_e \cdot \tau}{120 \cdot n \cdot i \cdot \rho_{nl}} \text{ (ml/cycle)} \quad (15)$$

Herein, the N_e denotes the usable power of the test engine in kW, g_e represents the rate of fuel consumption in g/kW.h, n denotes the engine speed in rpm, τ denotes the number of periods, ρ_{nl} denotes the fuel's density in g/cm^3. Also, the term i represents the cylinder number.

The engine was rigorously tested on a test bench under load conditions, especially at 100% load. Engine speeds from 1400 to 2100 rpm were carefully monitored to capture complete performance data. However, to ensure accurate analysis across the speed range, volume per cycle values were calculated using

Table 2
Diesel fuel volume/cycle determined from the experiment

n (rpm)	Vct (ml/cycle)
1100	0.0454
1400	0.0484
1700	0.0512
1900	0.0521
2100	0.0543
2400	0.0584

Table 3
Elements in the model

TT	Element name	Symbol
1	Boundary condition element	SB
2	Voltage stabilizer element	PL
3	Cylinder element	C
4	Barrier element	R
5	Measuring point element	MP
6	Air filter element	CL

Table 4
Element of choice for Kubota RT140 engine model

No	Element	Quantity
1	Pipe	11
2	System boundary	2
3	Plenum	2
4	Cylinder	1
5	Restriction	2
6	Measuring point	4
7	Air cleaner	1
8	Junction	2
9	Restrictions	2

interpolation at intermediate speeds based on measured data points of these predictions, in conjunction with the calculated velocities, which were carefully recorded and presented in detail in Table 2 for reference experiments. This careful approach ensures a thorough understanding of engine behavior and performance under a wide range of operating conditions, facilitating informed decision-making and optimization efforts.

The Kubota RT140 engine model in AVL Boost software is carefully designed to include detailed visualizations of complex engine assemblies and components. Leveraging AVL Boost's robust capabilities, every element of the model is carefully designed to capture the nuances of engine construction and use it carefully to ensure accuracy and integrity for real-world operation. Table 3 and Table 4 provide a detailed description, showing the materials, symbols, and numerical parameters used in the Kubota RT140 engine model. These tables are invaluable resources for engineers and researchers, allowing them to easily analyze and manipulate the model to conduct in-depth investigations and enhance engine performance. By integrating simulations and simulations in AVL Boost, the Kubota RT140 engine model allows them as users can analyze and adjust the engine configuration, increasing performance, reliability, and overall performance.

Input parameters for the engine model include geometric dimensions of parts, working conditions, and process control parameters. The amount of fuel supplied to a cycle is kept constant, gct = 0.0249 g/cycle.

3. Results and discussion

3.1. Model reliability evaluation

The main outcomes of the simulation results comparing diesel and DME-powered engine operation in terms of torque and fuel consumption have been listed in Table 5. It is important to note that the torque output of the Kubota RT140 engine decreases consistently when it is fuelled with DME as opposed to diesel throughout all of the speeds that were tested, with an average drop of 50.5% being seen across the full-speed range,

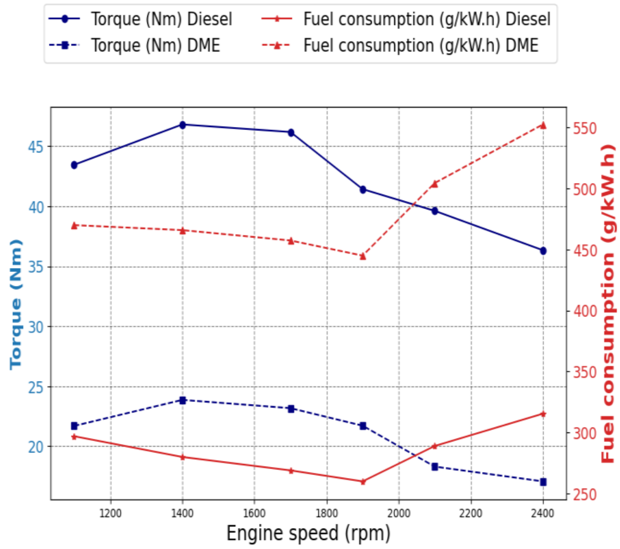

Fig. 3. Engine torque and fuel consumption during experiment and simulation

as shown in Figure 3. This decrease may be due to the lower calorific value of DME, which is 33.1% lower than that of diesel, as well as the much lower liquid density of DME, which is 19.73% lower than diesel.

3.2. Power characteristics and fuel consumption of the Kubota RT140 engine when using DME and LPG

Figure 4 provides insights into the power characteristics of the engine fueled with DME and LPG. The data reveal a notable difference in efficiency between the two fuel types, with DME consistently providing lower efficiency compared to LPG at different engine speeds. The average 43.13% reduction in engine efficiency when using DME can be attributed to several factors. First, compared to LPG, DME has a lower calorific value, which means it releases less energy per unit mass during combustion. This inherent difference in efficiency translates directly into a reduction in the power output of the engine. In addition, the combustion characteristics of DME may differ from those of LPG, resulting in differences in thermal efficiency and heat transfer characteristics These factors combine to contribute to the observed reduction in engine efficiency when fueled with DME (IKEMOTO *et al.*, 2005; Lee *et al.*, 2011).

The large power anomaly of 44.8% at 1800 rpm highlights the sensitivity of engine performance to operating conditions and fuel characteristics. Engine operating efficiency at this speed may interact with DME and LPG combustion characteristics in ways that increase the difference in output Factors such as combustion efficiency, air and fuel mixing efficiency, ignition timing can affect power output, and variations in these parameters between DME and LPG combustion processes. Similarly, the significant difference in fuel consumption, with DME exhibiting 48.7% higher consumption compared to LPG at 1400 rpm, confirms the complex relationship between fuel characteristics and engine work well between the emphasis (Yeom and Bae, 2009). Factors such as fuel quantity, thermal stability, and combustion efficiency may affect fuel consumption and contribute to the observed differences between DME and LPG (Anggarani *et al.*, 2020; Y. Chen *et al.*, 2019).

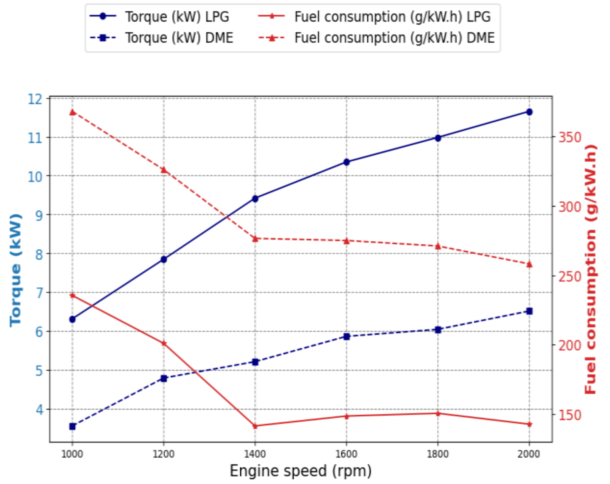

Fig. 4. Engine torque and fuel consumption using LPG and DME

3.3. Simulate the Kubota RT140 engine when using a mixture of DME and LPG

Simulation results provide valuable insight into the complex relationship between fuel content and engine performance. Considering the different blend ratios of LPG and DME, it is clear that the power output of the engine is remarkably related to the calorific value of the fuel blend, as shown in Figure 5.

At 0% DME, where the engine operates on LPG alone, the energy output is very high obtained. These results are consistent with the known fact that LPG boasts a significantly higher caloric content compared to DME. However, as DME is added to the fuel blend, the power output of the engine gradually decreases, this phenomenon results from the lower calorific value of DME compared to LPG (Jamsran and Lim, 2016). The generating capacity of an engine is directly affected by the energy content of the fuel, and the addition of low-energy DME naturally reduces energy output (de Jong *et al.*, 2023; Fabiś and Flekiewicz, 2022a). The reduction in power output is most pronounced when the DME ratio is the highest, up to 75%. This composition results in a significant reduction in engine power, indicating that DME has a significant effect on engine performance. Similarly, at engine speeds as low as 1000 rpm, where the engine operates at relatively low speeds, the reduction in efficiency due to the addition of DME is small compared to speeds as high as 1400 rpm.

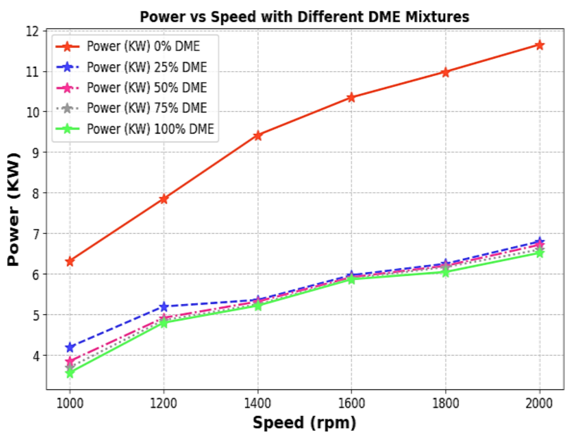

Fig. 5. Engine power variation when fueled with a mixture of LPG and DME

3.4. Emissions when using a mixture of DME and LPG

In particular, the characteristics of nitrogen oxides (NOx), carbon monoxide (CO), and soot emissions play an important role in assessing the environmental impacts of different fuel mixtures (Kim and Park, 2016; Wang *et al.*, 2013). In our study, the observed trend indicates that NOx emissions decrease with decreasing concentration of DME in the fuel mixture. As depicted in Figure 6a, at 1000 rpm, NOx emissions ranged from 392.4 ppm with 0% DME to 690.7 ppm with 100% DME. As the engine speed increased to 2000 rpm, NOx emissions increased accordingly, with values ranging from 1244.5 ppm with 0% DME to 1706.1 ppm with 100% DME. These data highlight the influence of both engine speed and DME concentration on NOx emissions. As reported, NOx concentration in the combustion

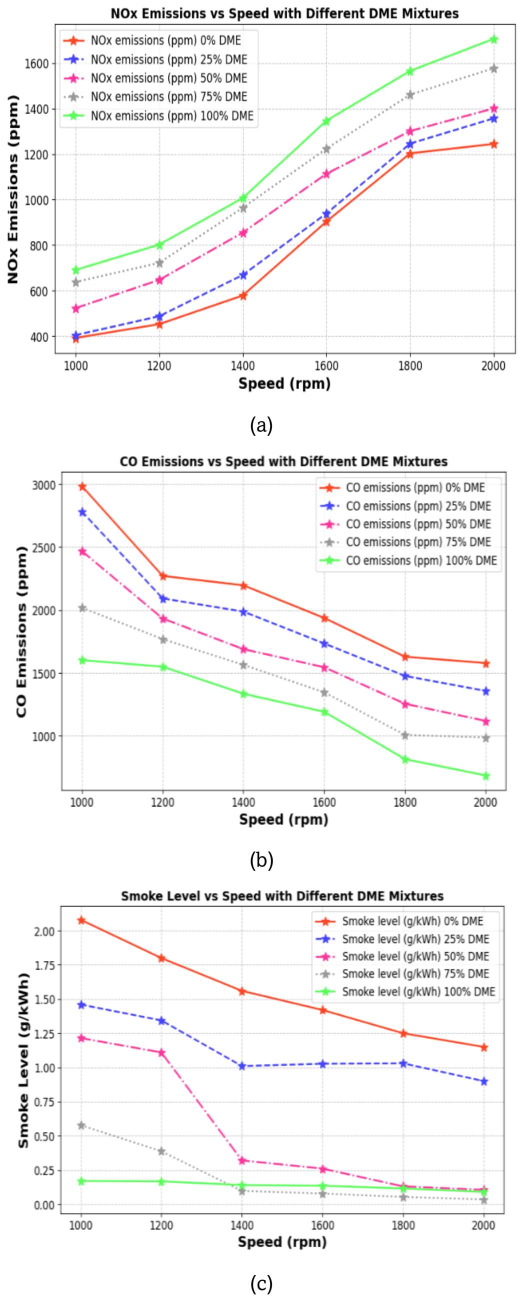

Fig. 6. Emission characteristics (a) NOx values; (b) CO; (c) Smoke from the engine when using a mixture of LPG and DME

depends much on temperature, the higher the combustion temperature is, the higher the NOx concentration is (Asghar *et al.*, 2021; Cao and Johnson, 2024). Notably, increasing DME concentrations increased NOx concentrations, emphasizing the importance of optimizing fuel blends to reduce emissions in DME-LPG-powered diesel engines has reduced emphasis (Wang *et al.*, 2016b). Furthermore, the observed trend highlights the need for comprehensive emission control strategies to address environmental concerns (Fabiś and Flekiewicz, 2022a; Huang *et al.*, 2022; Zhang *et al.*, 2020).

The data show CO emissions at different engine speeds for diesel engines fueled by different blends of dimethyl ether (DME) and liquefied petroleum gas (LPG). At an engine speed of 1000 rpm, CO emissions still exhibit a gradual decrease as the DME content in the fuel blend increases from 0% to 100%, as depicted in Figure 6b. This trend continues at all tested engine speeds, indicating the effect of fuel composition on CO emissions. Notably, at higher engine speeds such as 1800 rpm and 2000 rpm, the decrease in CO emissions becomes more pronounced with increasing DME concentration. This can be attributed to the rich oxygen of DME, which facilitates complete combustion and reduces CO emissions, especially at high engine speeds where thermal efficiency is critical (Fabiś and Flekiewicz, 2022b; Park *et al.*, 2010). These findings highlight the potential of DME-LPG blends as viable alternatives to mitigate CO emissions in diesel engines, and offer promising prospects for cleaner and more environmentally friendly technologies (Anggarani *et al.*, 2020; Arcoumanis *et al.*, 2008; Olsen *et al.*, 2007).

Smoke volume data for various blends of dimethyl ether (DME) and liquefied petroleum gas (LPG) in diesel engines reveal interesting insights into the combustion properties and effects of different fuel compositions on the properties of the environment. In the low engine speed range of 1000 to 1200 rpm, a decrease in smoke concentration is observed with increasing concentration of DME in the fuel blend is depicted in Figure 6c. For example, at 1000 rpm, the smoke density decreases from 2.08 g/kWh with 0% DME to 0.17 g/kWh with 100% DME. This decrease can be attributed to the presence of DME with more complete thermal oxygen molecules, thereby reducing particle emissions and cleaner combustion. However, in the higher engine speed range of 1400 to 2000 rpm, the smoke rate tends to be stable or increases slightly with increasing DME concentration. This trend may be due to increased thermal instability at higher velocities or to better thermal properties obtained with some DME-LPG blends (Devaraj *et al.*, 2021; Surendrababu *et al.*, 2023).

3.5. Burned volume fraction in cylinder of DME and LPG at 1400 rpm

Figure 7 shows the burned volume fraction profile of DME and LPG at 1400 rpm. This study shows that there are obvious changes in the combustion behavior of the two types of fossil fuels. The measurement shows that DME burns a smaller and lesser amount of fuel compared to LPG. This discrepancy stems from fundamental differences in the combustion mechanisms of the two fuels. DME, which has a lower density than LPG, requires a lighter fuel-air combination in the combustion chamber. As a result, DME combustion takes longer. This lighter combination, combined with a lighter burn, allows for more flawed burns in many places throughout the mix (A Kakoee *et al.*, 2019). In contrast, LPG has a higher fuel-air combination, allowing faster and more complete combustion. The increased LPG density guarantees a more uniform distribution of fuel throughout the combustion zone, resulting in more efficient combustion and more efficient combustion volume

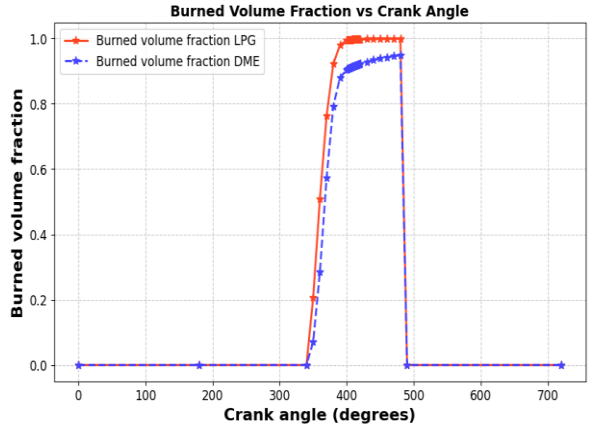

Fig. 7. Burned volume fraction of DME and LPG at 1400 rpm

fractionation. Overall, the observed differences highlight the need to consider fuel parameters such as density and combustion kinetics when optimizing the combustion process for internal combustion engines (Devaraj *et al.*, 2021; Joy *et al.*, 2019; Zannis *et al.*, 2019).

3.6. Heat release rate in the cylinder when using DME and LPG at 1400 rpm

The HRR during combustion plays an important role in the combustion efficiency and performance of the engine. In our study, the combustion efficiency at an engine speed of 1400 rpm was compared for two alternative fuel options: DME and LPG. Figure 8 shows that the heat output is higher than that of DME when LPG is used. This observation may be attributed to the higher solubility of LPG compared to DME. LPG has higher energy density per unit mass (Dinesh *et al.*, 2022; Sulaiman *et al.*, 2013), resulting in significantly higher energy density at the same time compared to DME. Consequently, the combustion produced by LPG exhibits a more pronounced combustion inside the cylinder. Normally, the difference in heat output highlights the influence of fuel characteristics on engine performance (Sharma *et al.*, 2022a). The higher efficiency of LPG translates into higher combustion characteristics, contributing to higher combustion and potentially improving engine performance. Conversely, DME,

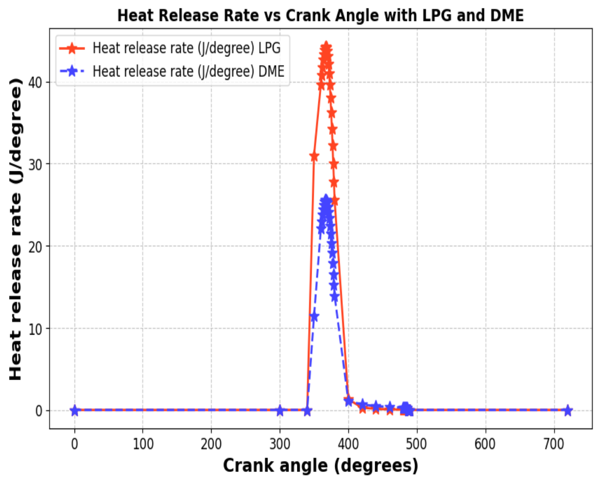

Fig. 8. Heat release rate in the cylinder when using DME and LPG at 1400 rpm

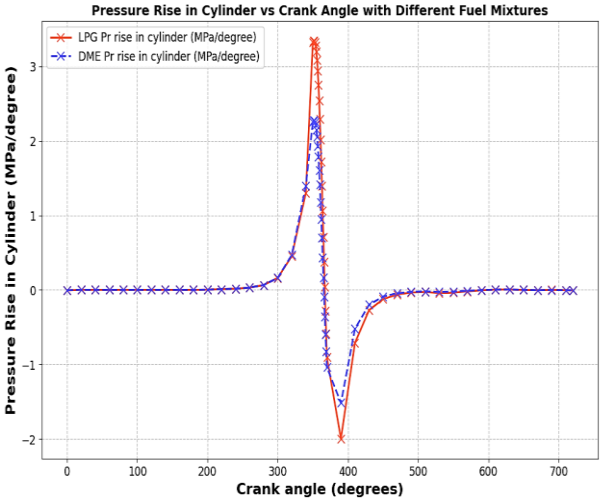

Fig. 9. Rate of cylinder pressure rise using DME and LPG at 1400 rpm

even as a functional alternative fuel, has shown comparatively low calorific values. Understanding the differences in heat release between different fuel choices is important to optimize engine performance and achieve desired performance outcomes (Handayani *et al.*, 2021; Kamei *et al.*, 2021).

3.7. Rate of pressure rise in the engine cylinder

The pressure gradient is an important parameter for evaluating the efficiency of fuel combustion in the cylinder. At specific engine speeds such as 1400 rpm, understanding how different fuels such as DME and LPG affect this parameter provides valuable insight into their combustion properties. Figure 9 depicts the variation of the pressure gradient in the cylinder along the crank angle at 1400 rpm for DME and LPG.

Inspection of the graph reveals specific pathways in the combustion process. Initially, the pressure rise during the early ignition phase is relatively small, indicating a gradual increase in combustion pressure. As combustion proceeds to the rapid combustion phase, characterized by combustion faster, the pressure in the cylinder increases significantly, indicating a more efficient energy conversion (A. Kakoee *et al.*, 2019; Semelsberger *et al.*, 2006). Thereafter, in the main combustion phase, the combustion process remains stable, but the pressure gradient slows down as the combustion pressure begins to depress the piston (Ferrari *et al.*, 2022). This depression reduces the pressure effect and causes a sudden drop in pressure gradient at cylinder compression. Comparing the pressure rise curves of DME and LPG, LPG exhibits a significant pressure rise in the cylinder throughout the combustion process This observation shows that LPG combustion produces instantaneous combustion, and heating pressure occurs relative to DME at 1400 rpm. Understanding these combustion kinetics is important for optimizing engine performance and emission control strategies (Killol *et al.*, 2019; Surendrababu *et al.*, 2023).

4. Conclusion

The article applied advanced simulation techniques using AVL-Boost software to thoroughly investigate the technological behavior of the Kubota RT140 diesel engine under different fuel conditions, in particular Dimethyl Ether (DME) and Liquefied Petroleum Gas (LPG) combination. It could be concluded from the obtained results that the engine exhibits its peak performance when fuelled with 100% LPG, which means that LPG is suitable to maximize engine power. In addition, fuel consumption shows a decreasing trend as the proportion of DME in fuel blends decreases. This finding indicates that reducing the amount of DME in fuel blends improves fuel efficiency. Moreover, NOx emissions exhibit an increase with increasing concentration of DME in the fuel blend. This finding confirms the higher NOx emissions associated with DME compared to LPG. Conversely, CO emissions decrease with increasing DME fraction in fuel blends. This result confirms the lower CO content of DME compared to LPG. Also, smoke emissions rose at higher DME concentrations in the fuel blend, indicating a more efficient combustion process with increasing DME concentration. Finaly, DME exhibits slower and more controlled combustion kinetics compared to LPG, resulting in reduced fuel burn volume and rate. When using LPG instead of DME, the heat release rate inside the cylinder is much greater. Similarly, the cylinder pressure gradient is greater in LPG fuel compared to DME. In sum, these findings provide valuable insights into the performance and emissions of the Kubota RT140 engine concerning different fuel compositions and shed light on possible strategies for improving engine performance and emission control well in real-world applications. In the future, it should conduct experimental studies on the use of DME for diesel engines to have comprehensive evaluations.

Acknowledgments

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