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NUMERICAL AND EXPERIMENTAL RESEARCH ON THE DI-CI ENGINE PERFORMANCE CHARACTERISTICS UNDER LPG - DIESEL DUAL FUEL COMBUSTION MODE

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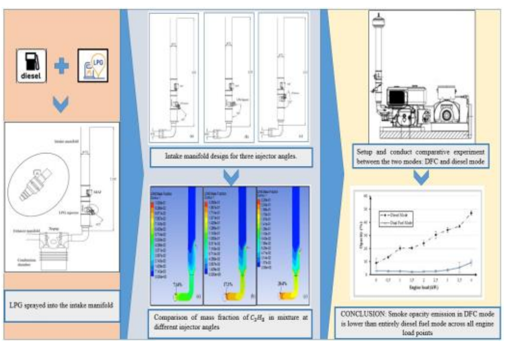
Graphical abstract



Abstract

Properties of Liquefied Petroleum Gas (LPG) are suitable as an alternative fuel for internal combustion engines. This work aims to combine both numerical optimization analysis of LPG injector placement angle and subsequent experiment with selected mounting injector angle to investigate the effects of using LPG as a parallel-partial substitute fuel with diesel fuel under dual-fuel combustion mode. An electronic injection fuel control system was applied to the modified intake manifold to generate the appropriate LPG injection. Three mounting angle injector positions including 45⁰, 90⁰ and 135⁰ in the upstream direction of intake air have been defined by Ansys Fluent to take into account efficient LPG-air mixing. Then, the experiment was conducted with LPG-diesel dual-fuel combustion mode (DFC mode) and compared to entirely diesel combustion mode as baseline data. Different load conditions ranging from idle to 4.0 kW were imposed at a constant engine speed of 1700rpm. The obtained results revealed that the injector's mounting angle position by 45° opposite to the intake air flow showed the correlation with the calculated LPG-air ratio in dual fuel combustion reaction. In DFC mode, the brake thermal efficiency (BTE) decreased on average by 4.6% and the LPG substitution rate gradually decreased while brake-specific fuel consumption (BSFC) increased for all engine loads. The exhaust temperature in the dual-fuel combustion mode was found to be higher than that of full diesel fuel mode at low loads (less than 2.5 kW) and began to decrease at higher loads. Exhaust smoke opacity was significantly reduced when operating in DFC mode.

Keywords: Diesel engine, LPG-Diesel dual fuel, ANSYS fluent, Engine performance, Emissions

1.0 INTRODUCTION

The utilization of alternative fuels on diesel engines represents an effective solution for mitigating harmful emissions and enhancing the diversity of fuel resources. Regarding this aspect, many fuels such as ethanol, methanol, natural gas, LPG, biodiesel, etc are currently being considered for eco-sustainable fuel on diesel engines [1]. Notably, LPG stands out among these alternatives due to its elevated calorific value in comparison to conventional fuels like gasoline and diesel. Moreover, the combustion by-products of LPG are devoid of sulfur and lead compounds, resulting in reduced particulate matter levels and contributing to its eco-friendly nature [2]. Furthermore, the versatility of this fuel type extends to transportation and storage, as it can be conveniently transformed from a gaseous to a liquid state through pressure elevation or controlled temperature reduction [3]. Hence, the adoption of LPG as an alternative fuel showcases great potential in terms of both economic and environmental advantages.

In addition to the aforementioned benefits, the differences in the physical and chemical properties of LPG fuel require the implementation of new technical methods on diesel engines. Instead of converting diesel engines to run entirely on LPG, the option of using a DFC (LPG-diesel dual combustion) mode is applied. LPG is normally introduced into the intake manifold through mechanisms like throttle valves or injectors. However, for enhanced adaptability in regulating fuel delivery, adopting LPG injectors is deemed more suitable, without necessitating modifications to the engine's structural integrity. An investigation by Mohanad M. Al-kaabi *et al*. [4], utilized LPG/diesel fuel in the LOMBARDINI 15LD315 engine. In this case, LPG undergoes vaporization and is subsequently sprayed into the intake manifold prior to entering the engine cylinder, the outcomes demonstrated that the emissions of NOx, HC, CO and CO2 are diminished in comparison to the exclusive use of diesel fuel. An LPG-diesel engine must consider elements such as injection timing and the proportion of LPG substitution to achieve diverse operating modes. Goldsworthy's research [5] unveiled that enhancing the propane ratio under high load conditions instigated a pronounced surge in the rate of pressure rise, subsequently triggering a knocking phenomenon. At a substitution of the ratio by 20% propane, the rate of pressure rise increases by about 17 bar/crank angle (CA), while for substitution ratios of 26% and 35% propane, the rate of pressure rise corresponds to 19 bar/CA and 58 bar/CA, accompanied knocking effect. Other factors, including injection timing, LPG injection pressure, and the constituent ratios within the LPG fuel, were also carefully examined [6].

Furthermore, in the case of engines employing the LPG injection technique through the intake manifold, an investigation on the impact of fuel injector orientation (location and angle) and intake manifold geometry has been explored across multiple studies. The work by Anas Ansari *et al*. [7], demonstrated the correlation between injector mounting and mixing characteristics through computational simulations ANSYS Fluent. A suitable mounting angle injector position facilitates the creation of the homogeneous mixture owing to sufficient spacial and temporal combustible formation. Depending on the injector placement, if positioned near the combustion chamber, the LPG-air mixture might encounter inadequate mixing due to a short diffusion time, while locating injectors far from the combustion chamber could lead to poor fuel mixtures, resulting in diminished power output due to ineffective combustion. Similarly, other studies have indicated that the positioning of gaseous fuel injectors influences vaporization and fuel-air mixing processes, subsequently impacting combustion, brake power, exhaust emissions, and fuel consumption [8], [9]. The intake manifold with LPG fuel injectors must be designed to ensure high mixing efficiency and sufficient airflow-LPG volume. The aerodynamic factors within the intake manifold generate turbulent swirls that aid in diffusion and the mixing of the air-fuel blend, as examined in various studies [10], [11], [12], [13]. In terms of injector angles on the intake manifold, Thomas and Somasundaram [14] have used CFD to investigate the effect of LPG gas injection at angles of 180^0, 30^0, 45^0, and 90^0 on the performance and emissions characteristics of a dual-fuel (LPG and diesel) engine. Their findings highlighted a roughly 5% increase in engine thermal efficiency (with a 40% diesel-60% LPG ratio) and an approximately 35% reduction in NOx emissions concentration when injected at a 30° angle. Another study carried out by Venkateswarlu Chintala and K.A. Subramanian [15] also yielded similar outcomes when injecting CNG gas at angles of 0^0, 45^0, 90^0, and 225^0.

Overall, the existing research discussed in-depth the impacts of factors influencing the efficiency of the dual-fuel LPG-diesel combustion mode process, yet a coherent link between the simulated process and then continuing to carry out experimental evaluations on the engine performance has been lacking. Therefore, it is essential to seamlessly integrate a series of consecutive studies, starting from considering the modified design of the intake manifold structure to suit the LPG fuel-providing approach, investigating the LPG injector mounting orientation, and then experimenting with the selected mounting injector angle to collect data on the diesel engine performance characteristics in the DFC mode. This work aims to explore quantitatively the impact of injector mounting angles, namely 45^0, 90^0, and 135^0 along the intake manifold of the DI-CI diesel engine type Vikyno RV125 by CFD method in order to analyze variations in the gas density distribution of LPG-air at the mentioned injector angles. Then, a comparative experiment was conducted on the diesel engine, comparing the DFC mode with entirely diesel fuel mode (diesel mode) under varying engine loads and speed conditions.

2.0 METHODOLOGY

2.1 The Analysis of Intake Manifold Structure for LPG Providing Approach

In this study, the LPG fuel is depressurized and directly injected into the engine intake manifold. Consequently, the intake manifold is required a larger diameter and relatively greater length to enhance the vaporization and mixing capability of LPG with air. Additionally, the positioning of the mass Air-flow (MAF) categorized in Denso 197-6020 sensor for determining the air flow rate and the location of the LPG injector (Bosch) along the intake manifold are considered when establishing the intake manifold design structural parameters. Furthermore, the shape of the manifold must avoid sharp bends to minimize localized airflow losses and enhance the engine's volumetric efficiency. Based on the original RV125 engine's intake system, a proposed intake manifold configuration is fabricated, as depicted in Figure 1. From this figure, the LPG injector is installed above the intake port in the incoming air direction to take advantage of turbulence created as the airflow passes through the intake port, facilitating more effective dispersion of LPG. For the injector mounting angles along the intake manifold, three angles of 45^0, 90^0 and 135^0, are employed based on inquisition from previous studies [14]. The intake manifold model with the best-simulated mixing results will be selected for fabrication and testing within the actual system.

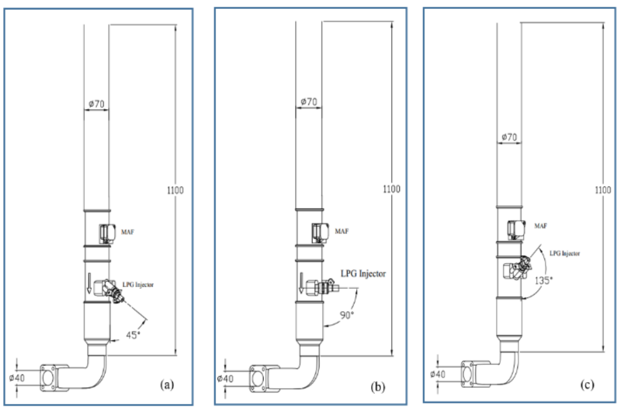


Figure 1 Intake manifold design for three injector angles

2.2 Equations and Simulating Conditions for LPG Mixing in the Intake Manifold

The simulation of intake airflow is carried out under assumptions that include incompressible flow and no chemical reactions between air and LPG gas (C_3H_8). The Species Transport model is utilized as a tool for mixing specific fluids, namely O_2, N_2, and C_3H_8, within the Ansys Fluent application [16]. The pressure-based solver is chosen for fluid flows ranging from low to high velocities [17]. The standard k-epsilon turbulence model, developed by Launder and Spalding is also employed in this study case. Additionally, the continuous flow of the fluid is determined by the conservation equations of physics [18]:

Continuity Equation:

$$\frac{\partial \rho}{\partial_t} + div(\rho u) = 0 \quad (1)$$

Where ρ represents the fluid density and **u** denotes the three-dimensional flow velocity along the x, y, and z directions.

The equation of momentum conservation (Navier-Stokes equation[19]):

$$\frac{\partial(\rho u)}{\partial_t} + div(\rho u * \mathrm{u}) = -div(p) + div * \tau + S_M \quad (2)$$

Where p, τ and S_M respectively represent the fluid pressure, strain-rate tensor, and body force.

The equation of energy conservation:

$$\frac{\partial(\rho i)}{\partial_t} + div(\rho i u) = -\rho\, div\, u + div(k\, grad\, T) + \Phi + \mathrm{u} * S_M \quad (3)$$

Where i, k, T, Φ respectively denote internal energy, thermal conductivity, temperature, and dissipation rate.

The required inlet conditions for setting up the two fluid streams mixing problem mentioned above include the air velocity at the inlet of the intake manifold (v_1) and the outlet velocity of the LPG injector (v_{out}), as illustrated in Figure 2. Due to the closed geometry of the intake manifold, v_1 is dependent on the suction velocity at the valve throat (v_2) of the engine and is determined by the continuity equation:

$$Q = v_1 A_1 = v_2 A_2 \quad (4)$$

Where A_1, A_2 respectively are the cross-sectional areas at the inlet and outlet of the intake manifold.

Additionally, v_2 is dependent on the engine speed through equation (5) [20]:

$$v_2 * i * f_2 * \gamma_2 = v_p * F_p * \gamma_p \quad (5)$$

Where i is the number of valves in one cylinder; f_2 is the flow area of the valve seat throat; γ_2 and γ_p are the air mass density at the throat and in the cylinder, respectively; v_p is the piston velocity; F_p is the piston crown area on.

Based on the pressure difference phenomenon between the two inlet and outlet ends of the injector, the injector outlet velocity v_{out} is determined by Bernoulli's principle:

$$P_{in} + \frac{\rho v_{in}^2}{2} + \rho g h_{in} = P_{out} + \frac{\rho v_{out}^2}{2} + \rho g h_{out} \quad (6)$$

Where: P_{in}, P_{out} and v_{in}, v_{out} represent the pressure and velocity at the inlet and outlet of the injector, respectively; ρ is the density; g is the acceleration due to gravity.

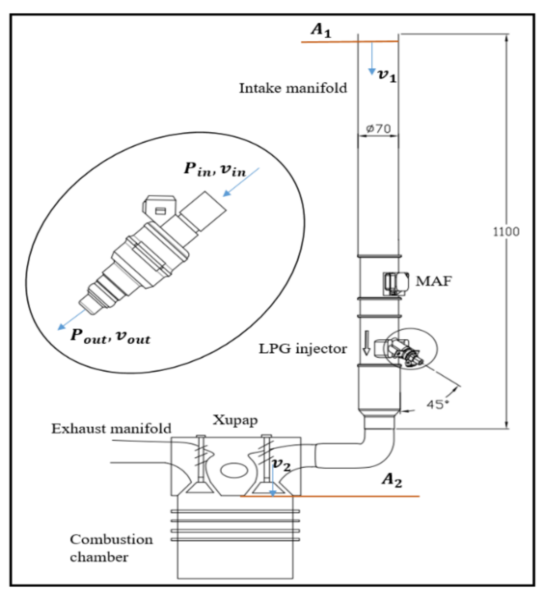


Figure 2 Illustration of intake manifold layout and airflow direction

Table 1 Technical specifications for the diesel engine and LPG injector [21]

Description	**Parameters**
Engine type	4 strokes, NA, DI
Number of intake valves/cylinder $\boldsymbol{i}$	1, horizontal type
Compression ratio	18:1
Diameter $\boldsymbol{d_1}$	70 mm
Diameter $\boldsymbol{d_2}$	40 mm
Piston stroke length	90 mm
Engine speed	1700 rpm
Piston crown diameter	94 mm
Injector inlet pressure $\boldsymbol{P_{in}}$	1.3 bar
Injector outlet pressure $\boldsymbol{P_{out}}$	1 bar
Injector inlet velocity $\boldsymbol{v_{in}}$	0 m/s
LPG density $\boldsymbol{\rho}$	2.222 kg/m^3

The test engine in this work is usually used for applications of generators, farm agriculture irrigation diesel engine water pump. Table 1 displays the technical specifications for the diesel Engine type Vikyno RV-125 and LPG injector. From equations (4), (5), and (6), the simulation input conditions for intake air on Ansys are determined at 1700 rpm of engine speed and presented in Table 2.

Table 2 Boundary conditions for intake air inflow in Ansys

	Inlet air	**Inlet LPG**
Velocity	9.3m/s	164m/s
Temperature	30℃	15℃
Mass fraction of species.	81% N_2, 19% O_2	100% C_3H_8

3.0 EXPERIMENTAL SYSTEM SETUP AND RESEARCH CONDITIONS

3.1 Experimental Setup

The schematic arrangement of experimental apparatus for conducting the performance and emission characteristics of the converted Diesel engine type Vikyno RV125 into Dual Fuel Combustion mode (DFC) is depicted in Figure 3. The diesel engine was connected to a single-phase 5.0kW, 220V, 50Hz AC generator through a belt drive, the alternating current was generated by the generator to supply the load consumption system, which includes adjustable thermistors ranging from 0.5 kW to 10 kW. A load cell was applied to the generator to determine the generator stator's torque. Diesel and LPG fuel consumption were collected by using an electronic weight scale system. The engine speed was determined by an encoder coaxially mounted with the flywheel, which is then converted into a digital signal sent to the ECU for calculating LPG injection timing based on load and speed variations (LPG injection-based speed regulator PID control), displayed on the Labview programmer [22]. Temperature sensor signals from the intake air, exhaust, lubrication, and cooling were gathered to observe and analyze the engine's operational status as well as mass air flow rate was noted at approximately 27.3 g/s at 1700 rpm during the experiment. The KOENG OP-201 smoke meter was utilized to assess the engine's opacity emission levels in different test modes. The fuel filters are replaced, and the MAF intake air flow sensor is calibrated on the intake manifold before testing on the engine [22]

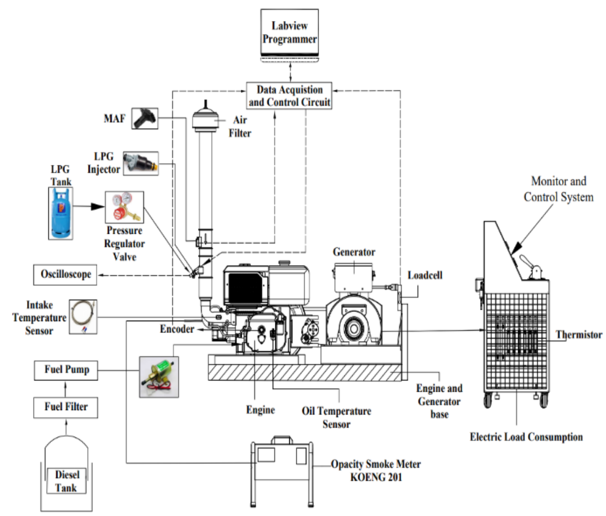


Figure 3 Experimental Setup Diagram of the Engine

3.2 Research Conditions

Table 3 and Table 4 present the fuel characteristics and test conditions. Differences in fuel characteristic values will impact the engine's performance and emissions. The engine was tested across the load range from idle to 4.0 kW, under a constant speed of 1700 rpm in both test scenarios: DFC mode and diesel fuel mode serving as a comparative sample. In the DFC mode, the engine was initially operated stably with diesel fuel at 1200 RPM under idle conditions. Subsequently, the electronic LPG injector control system adjusted the amount of LPG required to achieve the desired engine speed of 1700 RPM. At each test load, the LPG injection quantity was controlled by a PID algorithm to provide appropriate injection rates [22]. Furthermore, due to the specific characteristics of LPG fuel and its compatibility under certain load and speed conditions of the diesel engine, adjusting the additional portion of diesel fuel is necessary for some operating conditions [19].

Table 3 Characteristics of LPG and Diesel Fuels [23]

Fuel properties	Diesel	LPG
Density (g/cm3)	0.79 ÷ 0.87	0.51 ÷ 0.58
Lower heating value (kJ/kg)	42,500	45,908
Self – ignition temperature (∘C)	240	454
Carbon-to-hydrogen ratio (C/H)	~0.47	0.39

Table 4 Experimental conditions

Fuel	Diesel, LPG-Diesel
Load variation (kW)	Idle to 4.0
Load step (kW)	0.5
Measurement time/condition (Minutes per cycle)	10
Engine speed (rpm)	1700

4.0 RESULTS AND DISCUSSIONS

4.1 Effect of Injector Angle on Mixture Formation

Figure 4 illustrates the diffusion of LPG fuel compositions in the intake manifold. In general, all three injector-angle mounting positions lead to a broad dispersion of gaseous fuel around the injector. However, the LPG gas distribution density in the intake manifold with a 90^0 tilted angle and a 135^0 tilted angle is denser than that of a 45^0 tilted angle. The cause was determined to be that LPG fuel injected with a 45-degree tilted angle design, creates better aerodynamic flow resistance with the incoming air stream, thereby enhancing diffusion. As a result, the density of the LPG-air mixture is improved. The analysis of the results indicated that the mass ratio of LPG in the LPG-air mixture at the end of the intake manifold is 7.14%, 17.1%, and 20.4% corresponding to the 45^0, 90^0, and 135^0 tilted angle injector installation, respectively. Particularly, the LPG rate in the mixture under the 45-degree tilted angle condition illustrates the correlation with the calculated LPG-air ratio in dual fuel combustion reaction. Regarding these analyses, the results tend to align with the research of M.A. Jemni, demonstrating similarity in the LPG-air ratio reaching 6.5% [11]

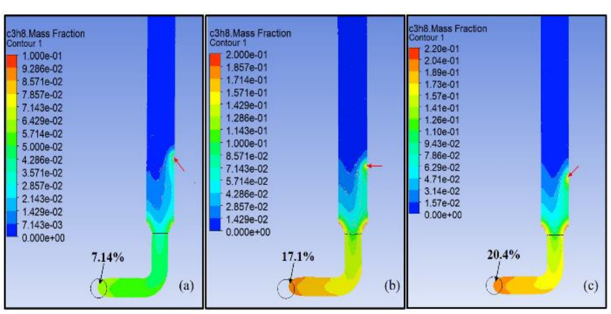


Figure 4 Comparison of mass fraction of C_3H_8 in mixture at different injector angles {(a) 450, (b) 900, (c) 1350}

4.2 Brake Thermal Efficiency and Brake-Specific Fuel/Energy Consumption Characteristics in DFC Mode

Figure 5 presents the Brake Specific Fuel Consumption (BSFC) and Brake specific energy consumption. (BSEC) characteristics of the engine operating in both diesel mode and DFC mode with the actual intake manifold fabrication of a 45-degree injector mounting angle. Generally, both BSFC and BSEC decrease as the engine load increases. When comparing the engine operation modes under low load conditions, a significant disparity in BSFC and BSEC between the two modes is evident. The cause of this outcome is attributed to the inefficient combustion of the LPG substitution in the DFC mode, resulting in its substantial presence during the late burning phase and leading to a phenomenon of missing flames in the exhaust manifold. This caused fuel wastage and elevated exhaust gas temperature. This confirmation is consistent with the studies conducted by Kumaraswamy & Prasad [24] which demonstrate similar findings at low loads. In line with the previously discussed pattern, the BSEC graph reflects a comparable trend.

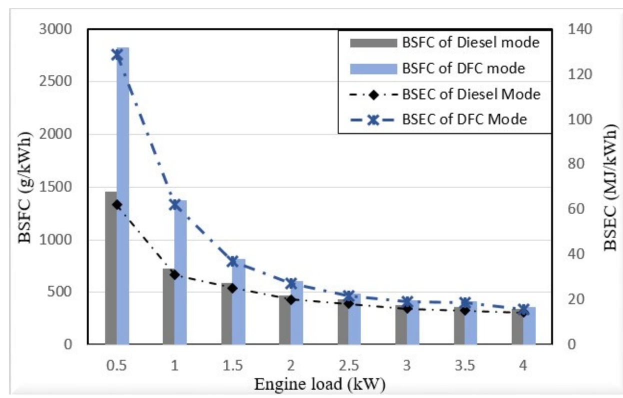


Figure 5 Brake-specific fuel consumption and brake-specific energy consumption characteristics at different load points and a constant engine speed of 1700 rpm

Brake Thermal Efficiency (BTE) is a crucial metric for evaluating engine performance in terms of energy utilization. Figure 6 illustrates the engine's brake thermal efficiency during testing in the two operating modes. The graph indicates that the Vikyno RV125 engine operates with lower thermal efficiency in the DFC mode compared to the diesel mode across all tested points. Specifically, the average BTE difference between the two operation modes across load points is 4.6%. The rationale behind this result is elucidated by Lata and Misra [25]. Due to higher LPG flame speed and diffusion capability compared to diesel fuel, a significant portion of oxygen from the air is initially mixed with LPG. Consequently, when diesel fuel is injected, it struggles to adequately mix with the remaining oxygen in the combustion chamber, leading to reduced participation in the combustion process and being pushed along the exhaust flow. Furthermore, the distinct physical properties of diesel and LPG could affect their heat absorption and mixing ability within the engine cylinder, consequently diminishing fuel energy utilization efficiency.

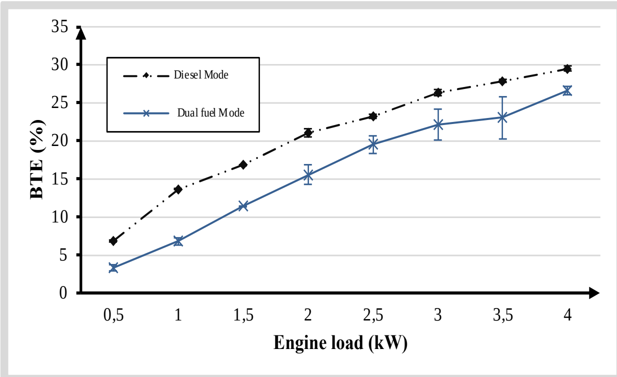


Figure 6 Thermal efficiency of RV125 engine with different load points and a constant engine speed of 1700 rpm

4.3 Opacity Emission Concentration

Smoke opacity emission is one of the parameters used to assess soot emissions in engine exhaust during its operation. Figure 7 compares the smoke opacity emission concentrations of two testing modes as varying engine loads. As shown in the graph, smoke opacity increases progressively with higher loads. From the comparison of the Vikyno diesel engine's operation modes, the soot emission concentration in the DFC mode remains relatively unchanged across load points and decreases significantly when compared to the diesel mode. This could be attributed to the inherently lower C/H ratio of LPG than that of diesel fuel, meaning that the mass of carbon in LPG is lower, resulting in reduced soot emissions and smaller opacity emission compared to traditional diesel fuel [19]. Additionally, under high load conditions, the proportion of diesel in the fuel mixture increases as seen in Figure 8, resulting in an increase of opacity emissions at 4.0 kW.

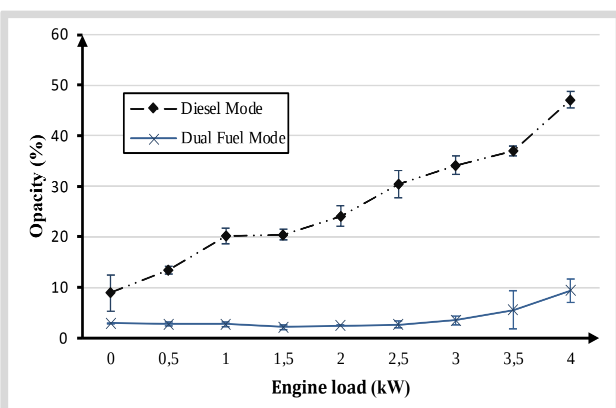


Figure 7 Smoke opacity emission at various load points and a constant engine speed of 1700 rpm

4.4 Engine Exhaust Gas Temperature

The exhaust gas temperature contributes to the assessment of combustion efficiency in engines. Figure 8 presents exhaust gas temperature in both Diesel Mode and Dual Fuel Combustion Mode at a constant engine speed of 1700 rpm with varying engine loads. Overall, the exhaust gas temperature increases as the engine load consumption increases. When comparing the two fuel modes, from 0kW to 2kW in load points, the exhaust gas temperature in the Dual Fuel Combustion Mode is higher than in the Diesel Mode. This may be explained by the higher proportion of LPG substitution in the fuel mixture (approximately 91%) as well as the non-homogeneous mixing density in the combustion chamber, which causes incomplete combustion of LPG fuel. This phenomenon leads to flame occurrences along the exhaust pipe (as mentioned in 4.1). From the load point of 2.5kW to 3.5kW, the exhaust gas temperatures in both modes do not differ significantly, but at the 4kW load point, the exhaust gas temperature in the Diesel Mode tends to be higher than in the Dual Fuel Combustion Mode. The reason is that from the load point of 2.0kW and onwards, the fuel is mixed and burned more efficiently in the combustion chamber. Concurrently, the proportion of LPG in the fuel mixture decreases compared to lower load points (specifically around 80% at 4.0kW). Similar trends in exhaust gas temperature have also been observed in other studies by the author Miqdam Tariq Chaichan [26] and the author Deo Raj Tiwari [27].

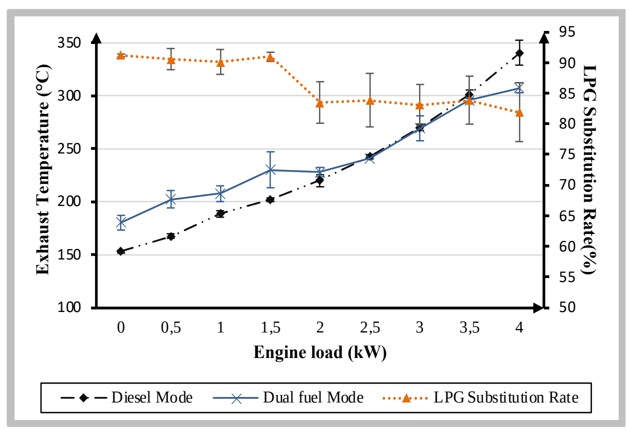


Figure 8 Exhaust gas temperature of RV125 engine with different load points and a constant engine speed of 1700 rpm

5.0 CONCLUSION

The conclusions in this study are drawn based on the LPG-air mixing on the intake manifold (CFD) simulation and experimental testing on the modified Vikyno-RV125 engine using dual-fuel LPG-diesel with LPG injection through the intake manifold. The injector mounting angle on the intake manifold affects the mixing process by the molecular density distribution of LPG fuel on the incoming air stream. The injector positioned at a 45-degree angle opposite direction to the intake airflow yields the most favorable results, with a mass ratio of LPG in the manifold at 7.14%, closely aligned with the calculated value from the combustion equation. The experimental brake-specific fuel consumption shows significant variation at low loads among two fuel testing combustion modes and gradually decreases as engine load increases. The average difference of 4% in Brake Thermal Efficiency (BTE) between diesel mode and DFC mode was noted across various load points while the thermal efficiency of the DFC mode was recognized as lower than diesel mode. Smoke opacity emission in DFC mode is noticeably lower than that of entirely diesel fuel mode across all engine load points. It was detected less than 9.38% of tested points under operating DFC mode. There is a slightly increase trend in smoke opacity emission at high loads in DFC mode although decreasing the LPG substitution rate in the mixture.

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Conflicts of Interest

The author(s) declare(s) that there is no conflict of interest regarding the publication of this paper.

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