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Numerical investigations on the performance and emissions of a turbocharged engine using an ethanol-gasoline blend

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A R T I C L E I N F O

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A B S T R A C T

Due to a scarcity of fossil fuel supplies and concerns about pollution, the use of ethanol in gasoline has become a priority in the automobile industry. This paper aims to investigate the effect of different ethanol-gasoline fuel blend ratios, namely E20 (% ethanol + % gasoline), E50 (% ethanol + % gasoline), and E75 (75% ethanol + 25% gasoline) on a 1.6 L turbocharged, 4-cylinder, 2017 Proton Preve Premium CFE CVT engine, where E0 (pure gasoline) is taken as reference fuel. In addition, different speed intervals, which include 1000 RPM, 2000 RPM, and 5000 RPM, are employed for each fuel blend. The production of four major emissions, NOx, CO, CO2, and HC, and performance parameters such as thermal efficiency, volumetric efficiency, and brake-specific fuel consumption, are evaluated using SolidWorks for CAD modelling. This then is transferred to ANSYS for emission and performance analysis. According to the findings, increasing ethanol concentration and engine speed increases volumetric efficiency and brake-specific fuel consumption by up to 12.89% and 6.59%, respectively. It was also discovered that ethanol and increasing engine speed had an 11.39% reduction in thermal efficiency. Furthermore, the addition of ethanol occurs, along with an increase in speed, exhaust gas emissions are reduced by up to 21.74% compared to pure gasoline.

Nomenclature

$\dot{m}_a$	Mass flow rate of air
$\dot{m}_f$	Mass flow rate of fuel
AFR	Air-fuel ratio
BSFC	Brake-specific fuel consumption
BP	Brake power
P_{b-max}	Maximum engine brake power
$N_{@Pb-max}$	Speed at maximum brake power
N	Speed

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η_v	Volumetric efficiency
V_a	Actual volume air intake into cylinder
V_b	Theoretical volume air intake into cylinder
n_r	Number of crankshaft rotations per engine cycle
ρ_a	Density of air
P_a	Air pressure intake of the engine
R_a	Specific gas constant for dry air
T_a	Intake air temperature
η_{th}	Thermal efficiency
LCV	Calorific value of fuel
HC	Hydrocarbons
CO	Carbon monoxide
CO_2	Carbon dioxide
NO_x	Nitrogen oxide

1. Introduction

Due to a shortage of fossil fuel supplies and concerns about pollution, the use of ethanol in gasoline has become a priority in the automotive sector. It was employed in an internal combustion engine (ICE) to lessen the negative impacts of toxic pollutants on the environment [1].

The necessity to preserve fossil resources and the tightening of pollution regulations compel the adoption of some practical solutions for engines that run on alternative fuels. Continuous research in recent years has resulted in the creation of flexible-fuel engines that can run on any combination of ethanol and gasoline. Furthermore, numerous research discovered that a greater compression ratio improves the density of the fuel-air mixture and flow turbulence in the combustion chamber, resulting in high cylinder pressure and fast burning speed. Furthermore, turbocharging internal combustion engines is one of the strategies for improving their performance. Furthermore, turbocharging is becoming more critical in the development of gasoline engines in order to minimize fuel consumption and pollutants. Turbocharging may boost an engine's power density while also lowering the vehicle's fuel consumption. The findings show that turbocharging is a low-cost and effective technique to reduce fuel consumption and greenhouse gas emissions. Furthermore, NO_x (nitrogen oxide), CO (carbon monoxide), CO_2 (carbon dioxide), and HC (hydrocarbons) are the principal exhaust gas emissions of engines, which are created in very high-temperature in-cylinder conditions in the presence of nitrogen and oxygen from the air and fuel. Acid rain is caused by nitrogen oxides, which harm human health, degrade buildings, pollute rivers and marine habitats, and destroy trees. Furthermore, they can penetrate deeply into lung tissues, causing irritations, causing or worsening respiratory disorders and deteriorating cardiac problems.

The primary result of hydrocarbon combustion processes is CO_2. Anthropogenic carbon dioxide is the primary cause of the greenhouse effect and global warming. Carbon monoxide is a colourless, odourless, and tasteless gas with a density somewhat lower than that of air. When insufficient oxygen is available for burning, it occurs because of the incomplete oxidation of carbohydrates. CO produces CO_2 when there is enough oxygen present. CO has a 220-fold higher capacity to bind with haemoglobin in the blood, reducing oxygen transport to tissues and causing asphyxia. CO also has a role in the creation of tropospheric ozone. Hydrocarbons are the fuel molecules that are burnt in a cylinder during combustion.

This paper aims to provide a more in-depth look at ethanol-gasoline blends in turbocharged engines, analyzing the unique properties of an ethanol fuel blend to determine the most efficient proportions or quantities that could be used in a turbocharged engine, and examine why future cars should focus more on fuel blends rather than pure-gasoline vehicles.

The following is a breakdown of the paper's structure: The prior study on the influence of fuel blends is discussed in Section 2. The approach is then explained in Section 3. In section 4, the verified model's findings are provided, along with discussion and analysis. Finally, in Section 5, the paper's conclusions are given.

2. Previous work on fuel blend impact

Three key elements will be examined in this section. Starting with the primary goal of this investigation, the influence of ethanol-gasoline on turbocharged engines. As a result, past studies must be examined to properly comprehend the primary characteristics that must be addressed and how the fuel blend affects turbocharged engines. Second, more study is needed on the effects of different fuel mixes on SI engines. The objective is to understand how different gasoline mixes with varying amounts influence the engine and identify which fuel blends are the most efficient. Finally, the influence of fuel mixes utilising CFD simulation analysis is the final subtopic in this section. The significance of that subtopic stems from future research into using various CFD software with various fuel mixes, which provides insight into which software may be used for this article.

2.1. Effect of ethanol-gasoline on turbocharged engines

Previous research on the influence of the fuel blend, namely ethanol-gasoline, on turbocharged engines is briefly covered in this subtopic. The impact of a turbocharged engine on ethanol-gasoline blends has been studied extensively. Using 1-D GT-Power code simulation software, Mahmoudi et al. [2] examined methods to increase performance and emissions on gasoline turbocharged Nissan Maxima 1994. A turbocharger is said to be an efficient approach to minimize fuel consumption and greenhouse gas emissions. The article covered NOx, CO, CO2, and unburned hydrocarbons, based on spark-ignition (SI) engines (UHC), among the major pollutants covered in the article. After obtaining the data, the researcher discovered that the turbocharger enhanced the engine's torque by nearly double, from 275 Nm to 543 Nm at 4000 RPM (revolutions per minute). As the speed increased till it reached 6000 RPM, power surged to 371 HP, which was the most horsepower ever achieved. Moving on to the turbocharged engine's emissions, NOx and CO2 both showed a change in values as engine speed fluctuated, with maximum NOx emissions occurring at 4000 RPM, around 490 Nm of brake torque, and highest CO2 emissions occurring at 4500 RPM, around 520 Nm of brake torque. Carbon monoxide (CO) and hydrocarbon (HC) emissions, on the other hand, increased as torque increased. The researcher has concluded that increasing electricity will increase emissions. Research done by Balla et al. [3] had undergone an examination of different ratios of ethanol-gasoline on performance, such as power, BSFC, TE, VE, as well as exhaust gases. The ratios were E10, E20, E30, and ethanol 40% gasoline 60% (E40) on a one-cylinder, four-stroke engine. The experiment was done on different speed intervals between 1500 and 2500 RPM with 250 RPM increments. Mathematical calculations were done for engine performance parameters such as fuel consumption, brake power, VE, and TE. It was found from the results that as engine speed increases, brake power and VE tend to decrease too. The reason behind that was the time for induction stroke, which implies that less air enters the cylinder, which decreases the VE. BSFC was found to drop with the increment of engine speed. Moreover, as the ethanol content increased, engine power output was shown to increase due to the improvement of the air-fuel equivalence ratio, which, therefore, caused an increase in BTE. Regarding emissions, it was seen that as the ethanol ratio increased, a significant reduction in exhaust gas emissions was observed. Scala and Galloni [4] analysed the performance of a "downsized" spark-ignition engine fueled by gasoline and bio-butanol blends. Experimental tests have been carried out at operating points ranging from low to medium engine speed and load. In another work, Rosdi et al. [1] advocated for using ethanol-gasoline blends in engines, claiming that biofuels, specifical ethanol, reduce pollution and greenhouse emissions due to several ethanol characteristics, including high octane number, low sulphur content, and production from plant biomass. The researcher discovered that up to 30% ethanol improves engine performance, with properties such as lower energy content and higher density of ethanol-gasoline blend causing an increase in the brake-specific fuel consumption (BSFC) of the engine, as well as a significant reduction in emissions due to the oxygen level in ethanol. Ngo and Nguyen [5] investigated the influence of ethanol on combustion characteristics under gasoline compression ignition (GCI) mode. The conventional gasoline with a RON of 95 and gasoline-ethanol blend (20% ethanol in volume [E20]) was experimentally compared in terms of combustion characteristics and performance.

Furthermore, Lee et al. [6] reported on the effect of a turbocharger mounted on the exhaust manifold with many engine changes on the performance and thermal efficiency of an H2 SI-engine in contrast to natural aspiration (N/A) circumstances on the high load potential. At intervals of 1000 RPM, records were set in the range of 2000-6000 RPM engine speed. The turbocharger only enhanced power and torque until 5000 RPM, with a peak gain of 41% at 2000 RPM, but it did not affect increasing brake thermal efficiency (BTE). However, it was found that the turbocharger significantly improved the overall air mixture, which was noticeably thinner at high speeds. As a result, after studying the effect of ethanol-gasoline on turbocharged engines, it was discovered that more research into the influence of various ethanol-gasoline ratios on turbocharged engines is necessary.

2.2. Effect of fuel blends on SI engines

This subtopic focuses on prior research on the influence of fuel mixes on SI engines, which looked at various fuel blends. Hawas et al. [7] investigated the use of bioethanol to reduce emissions, as well as braking torque, power, thermal efficiency (TE), and specific fuel consumptions (SFC) throughout a range of 1500-300 RPM, using an E20 ethanol-gasoline blending ratio. Different equipment was utilised to measure the pressures, temperatures, and flows within the engine, as well as different air-fuel ratios with values of 0.9, 1, and 1.2. The software used for the experiment was AVL software. Because of the bioethanol characteristics, E20 has been demonstrated to increase combustion and reduce brake-specific energy consumption at all air-fuel combination dosages. It was discovered that engine output increases with a lambda value of 0.1 while NO_x levels decrease. Compared to a lambda value of 0.9, a lambda value of 1 resulted in a 50% reduction in NO_x at the same engine power. Bioethanol, according to the study, is an excellent fuel alternative. Costa and Sodré [8] conducted another type of experiment in which different compression ratios (CRs), specifically 10:1, 11:1, and 12:1, were tested and evaluated based on the highest performance and lowest emissions with a fuel blend ratio of ethanol of 22% - gasoline 78% (E22) and hydrous ethanol (E100). It had been observed that as the compression ratio increased at high speeds, performance improved significantly for both E22 and hydrous ethanol, E100. Hydrous ethanol, however, had a greater specific fuel consumption than E22. Changing the compression ratio did not affect the specific fuel consumption (SFC) or the thermal efficiency (TE) of E22. In the case of E100, however, when the compression ratio increased, SFC significantly decreased while TE increased. According to the researcher, future studies should use a half-opened throttle to simulate real-world driving situations better. Hasan et al. [9] utilised several methodologies in their investigation of E10 and E20 blends in SI engines under five different CRs in trials of analysing the effect of modifying the CR on tailpipe emissions. The researcher stated that when the CR increased, pollutants such as HC, CO, and NO_x decreased, whereas CO_2 increased owing to the better TE during combustion. Regarding performance metrics, the results showed that as ethanol concentration increased, engine torque, brake power, BTE, and BSFC improved. The experiment was carried out on a two-wheel motorbike with a huge displacement four-stroke engine. After completing the experiment, it was discovered that ethanol did reduce cold emissions, with a significant reduction compared to pure gasoline (E0). This is due to the additional ethanol content, which produced a lower flame temperature, resulting in a lower exhaust temperature and thus producing lower emissions. Baek et al. [10] investigated the impact of engine control parameters on combustion behaviors and particle number emissions with a spark ignition direct injection (SIDI) engine using various gasoline-ethanol blended fuels. Ilhak et al. [11] conducted a separate experiment in which three fuels, acetylene, ethanol, and gasoline, were tested under 25% and 50% loads, as well as varied surplus air ratios. In all the studies, it was discovered that the acetylene operation had the greatest BTE. However, with the provided loads, gasoline was shown to have the greatest emissions. Furthermore, when the load increases, so do the emissions. Acetylene was found to have the lowest levels of UHC and NO emissions. However, due to the low octane number of acetylene, ethanol is still considered the best fuel to utilise. Mourad [12] conducted another study in which he intended to combine butanol 50:50 with ethanol and then mix it with gasoline in quantities of 2%, 5%, 10%, 15%, and 20% at engine speeds of 1000-3400 RPM. The study's findings revealed a significant boost in performance and smooth operations. Furthermore, compared to normal gasoline fuel, which was utilised as a reference, emissions were greatly reduced owing to the oxygen concentration in fuel mixtures. There was a significant reduction in HC and CO emissions. However, when utilising alternative fuel blends, a minor drop in engine power was seen. Oxenham and Wang [13] investigated and optimized the Miller cycle, methanol, ethanol and turbocharging when applied to a high-performance gasoline engine. The engine's capability to operate when exclusively powered by biofuels was assessed numerically using the 1D gas dynamics tool 'WAVE', a 1D Navier–Stokes equation solver. Iodice et al. [14] discussed the feasibility of ethanol as an alternative transportation fuel, including world production and ethanol production processes. The physicochemical properties of ethanol and gasoline are then compared to analyze their effects on combustion efficiency and exhaust emissions. Then, the pathways of NOX formation inside the cylinder of SI engines are discussed in depth. Zhang et al. [15] investigated the effects of different blending ratios of methanol in gasoline-methanol blends on the characteristics of fuel consumption, emissions and acceleration of the vehicle, providing a reference for the application of methanol on passenger vehicles. Chedthawut et al. [16] wanted to see how the results were affected by two variables, diesel-water plastic oil (WPO) blending ratios and engine speeds. The value of the data from a huge diesel engine, revealed throughout a wide range of engine speeds, is the motivation for this study. The researcher used a full-factorial experimental design (FFD) to plan this experiment, which helps to demonstrate the impacts of fuel mixing ratios and engine speeds. WPO can be a viable alternative fuel like biodiesel, according to the study, if a suitable operating cost-benefit analysis or optimization is done in future research. Furthermore, further study on fuel mixes is needed to assess their influence on engine performance and emissions.

2.3. *Effect of fuel blends using CFD simulation analysis*

Previous research on the influence of fuel blends is mentioned in this subtopic, all of which were analysed using CFD software. Zhao and Wang [17] appraised the effect of biobutanol addition on an ethanol-gasoline (E10) engine, evaluating the potential benefits of EGR technology from the performance improvement and exhaust emissions reduction point of view in conjunction with a butanol/ethanol-gasoline blend in an SI engine. Another study was conducted by Mashkour et al. [18], who looked at the various phenomena within a cylinder. The study discovered that ANSYS ICE CODE combined with the dynamic mesh approach might be utilised to construct internal combustion engines. Because friction losses-driven interactions of engine components are not included in the basic CFD simulation of the experiment, the temperature and pressure values for firing simulation were somewhat higher than in the experimental investigation. Zareei et al. [19] studied the functional properties and exhaust emissions regarding compression ratio at different speeds. The numerical solution of the governing equations on the fluid flow inside the combustion chamber and the numerical solution of one-dimensional computational fluid dynamics with the GT -Power software was carried out. Lliev [20] developed a one-dimensional model of a four-cylinder, four-stroke, multi-point injection system SI engine and a direct injection system SI engine for predicting the effect of various fuel types on engine performances, specific fuel consumption, and emissions. Finally, Rajesh Govindan [21] used ANSYS FLUENT R14.5 software to investigate the influence of blending ratio on combustion characteristics in a compression ignition (CI) engine, where parameters such as in-cylinder pressure, temperature, heat release rate, and other variables were determined. The results revealed that the blends released less heat during the premixed combustion phase than diesel, with diesel releasing 35.03% more heat during the premixed phase and 1.09% more heat during the mixing stage than a 20% blend ratio. Moving on to in-cylinder pressure and temperature, the researcher discovered that as the blending ratio increases, so does the temperature and in-cylinder pressure, with a 30% blend ratio having a peak in-cylinder pressure almost 1.05% greater than the 10% blend ratio. However, engine functioning was smooth and consistent for diesel and thumba biodiesel blends. After studying the influence of fuel blends using CFD simulation analysis, it was discovered that there is a lack of research on the effect of ethanol-gasoline on various ratios using CFD software.

The literature above investigated the impacts of different fuel mixtures on engine performance and emissions, providing a clear picture of the criteria used and the influence of different blends on engine performance and emissions. However, the impact of turbochargers on the performance and emissions of gasoline engines has not been investigated yet and needs further consideration of more ratios to comprehend the performance fluctuation properly; hence the aim of this paper was established.

3. Methodology

This section describes the fuel mix qualities and ratios utilised in the experiment, engine and turbocharger specs and mathematical formulae applied. In addition, all experimental circumstances, such as engine rpm, must be indicated. A brief on the combustion and controlling calculation methodologies and equations employed in this paper is also required. When the simulation has been

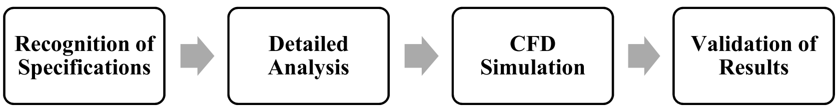

Fig. 1. Work execution steps.

successfully finished, the findings are validated by comparing the performance results with the produced simulation results. Fig. 1 depicts the work execution steps.

In Fig. 1, Phase 1 entails identifying specifications, including data preparation and arrangement, where specifications and materials to be utilised in this study are stated, with some of them coming from prior articles indicated in the background research. Phase 2 is dedicated to extensive analysis, with many analyses to be done during this time. To begin, a combustion analysis, a standard procedure for determining the chemical formula of a compound containing hydrogen and carbon, must be conducted. Second, a theoretical analysis will be carried out, which will entail theoretical calculations to get the essential parameters. Furthermore, phase 3 includes CFD simulation, which requires simulation software to attain the parameters established in the theoretical portion. As a starting point, a numerical model setup and a description of the simulation programme must be completed. Second, simulation must be carried out in accordance with the specified criteria for the desired outcomes. Finally, step 4 entails confirming outcomes before going on to the final phase. Validation of simulation results is part of the study. The analysis's general goal is to determine which fuel ratio is the most efficient while producing the least emissions. The validation process aims to ensure that outcomes are correct, with a percentage error rate of 5% or less.

3.1. Phase 1: recognition of specifications

The precise identification of vital key components is critical for further investigation. Engine specifications, fuel mix qualities, and engine speed variation & experimental circumstances are the three aspects of specification recognition. In order to acquire the desired performance and emission outcomes, specifications are defined to act as inputs for mathematical calculations and ANSYS. After reviewing and analysing past research, it was determined that ethanol affected engine performance and emissions. As a result, ethanol will be employed for future research into its engine effects.

A range of ethanol-to-gasoline ratios was investigated once the baseline research was completed. When ratios changed, most often with an increment value, performance attributes altered as well. Except for E100, preceding publications cited in the literature study did not conduct enough testing for high content blends; thus, a high blend ratio will be studied in this work. As a result, the blend ratios used in this study are E20 (20% ethanol, 80% gasoline), E50 (50% ethanol, 50% gasoline) and E75 (75% ethanol, 25% gasoline). For comparison, E0 (pure gasoline) should also be utilised.

The model inputs that were utilised in this investigation provide all relevant information. Technical details of the engine used for this experiment are shown in Table 1. The experiment was conducted on a Malaysian-made 2017 Proton Preve Premium CFE CVT turbocharged 1.6 L 4-inline cylinders.

The properties of various fuel blends used are listed in Table 2.

Table 2 displays the calorific values and stoichiometric AFR for the necessary fuels, which range from E0 to E75. The numbers above show that the calorific value increases slightly when the fuel blend ratio increases. Due to the qualities of ethanol relative to gasoline, the stoichiometric AFR drops as the fuel blend ratio increases. The heating value of ethanol is lower than that of gasoline. Furthermore, ethanol's stoichiometric air-fuel ratio is around two-thirds that of pure gasoline. As a result, the amount of air required for full combustion is lower for ethanol.

The atmospheric characteristics of the working environment are described in Table 3, and air enters the intake system.

The engine speeds utilised in this experiment were determined based on the engine's ability to attain its maximum power and torque values, as stated in the Table above, with all trials conducted at WOT. Intake air temperature, pressure, and coolant temperature are shown.

Table 4 shows the turbocharger specifications taken from a prior study [9].

The turbocharger utilised in this experiment was built by BorgWarner, one of the world's major automotive suppliers.

The critical parameters for the cylinder are described in Table 5, with most of the data coming from prior work [21].

The construction of a computational domain, a reduced version of geometry and boundary conditions, is the first stage in the CFD process. In this study, the computational domain is combustion chamber geometry. The table above lists all the required parameters.

3.2. Phase 2: detailed analysis

The second phase of this project entails a careful investigation of both the combustion and theoretical aspects. To begin, a combustion analysis, a standard procedure for determining the chemical formula of a compound containing hydrogen and carbon, must be

Table 1
Engine specifications [25].

Number of cylinders	4 cylinders (inline)
Displacement	1.6 L
Total displacement	1561 cc
Boost pressure maximum	75 kPa
Fuel Tank capacity	50 L
Bore	7.6 cm
Stroke	8.6 cm
Compression ratio	8.9:1
Clearance Volume	197.47 cm^3
Power Maximum	138 hp @ 5000 RPM
Torque Maximum	205 Nm @ 2000 RPM

Table 2
Fuel blend properties [22].

Fuel	Lower Calorific Value (kJ/kg)	H/C Ratio	O/C Ratio	Stoichiometric AFR
E0	2.968	0.48	0	14.7
E20	2.976	2.0	0.07	13.27
E50	2.988	2.3	0.19	11.61
E75	2.995	2.6	0.33	10.26

Table 3
Experimental conditions.

Parameters	Values
Engine speed	1000, 2000, & 5000 RPM
Intake Air Temperature	18.2 °C
Coolant Temperature	95 °C
Intake Air Pressure	75 kPa
Intake System	WOT (Wide-open throttle)

Table 4
Turbocharger specifications.

Items	Specifications
Model [−]	BorgWarner, K03-2075
Comp. Wheel Outside Diameter (O.D.) [mm]	2.0
Comp. Wheel Inducer Diameter [mm]	38
Turbine Wheel O.D. [mm]	1.8
Turbine Wheel Exducer [mm]	42
Turbine Area/Radius Ratio	5

Table 5
In-cylinder parameters [21].

Parameter	Value
Connecting Rod Length	10 cm
Crank Radius	4.6 cm
Intake Valve Closing (IVC)	570° CA ABDC
Exhaust Valve Open (EVO)	833° CA BBDC
Fuel Injection Timing	23° CA BTDC
Injector Orifice Diameter	0.15 mm
Injector Nozzle Opening	203 Bar

conducted. Second, a theoretical analysis will be carried out, which will entail theoretical calculations to get the essential parameters.

Combustion analysis is a typical method of evaluating the chemical composition of a substance containing hydrogen and carbon. Thermal engines create energy by combusting fuel and oxygen (from the air). In order to ensure the combustion phase, a specified amount of fuel and air must be delivered to the combustion chamber. The combustion process is complete when all the fuel has been consumed, and the exhaust gas includes no unburned fuel. Eq. (1) represents the whole combustion reaction that occurs when the burning of various components occurs when O_2 is present, which is the fuel ethanol (C_2H_5OH) that burns in oxygen (O_2) to give/produce carbon dioxide (CO_2) and water (H_2O), according to Dixit's derivation [24].

$$C_2H_5OH + 3O_2 \rightarrow 2CO_2 + 3H_2O \tag{1}$$

The stoichiometric air-fuel ratio is the best (theoretical) for complete combustion. The stoichiometric air-fuel ratio for a gasoline (petrol) engine is 14.7:1. This indicates that 1 kg of fuel requires 14.7 kg of air to burn completely. Even if the AFR is not stoichiometric, combustion can occur. The air-fuel combination is deemed lean if the air-fuel ratio approaches the stoichiometric ratio. The mixture is described as rich when the air-fuel ratio is smaller than the stoichiometric ratio. For a gasoline engine, an AFR of 16.5:1 is lean, and 13.7:1 is rich.

The air-fuel ratio (AFR) is the ratio of the mass flow rate of air ($\dot{m}a$) to the mass flow rate of fuel ($\dot{m}f$) when the engine is running. In this experiment, however, stoichiometric AFRs will be employed for all fuels. Eq. (2) depicts the AFR formula, which will be utilised for subsequent calculations. In the theoretical study, several engine performance characteristics will be explored and analysed in this work. A handful of the formulas that will be utilised are included.

$$\dot{m}_a = \dot{m}_f \times AFR \tag{2}$$

The brake-specific fuel consumption (BSFC), which is a measure of the vehicle's fuel efficiency [6], is calculated using Eq. (3).

$$\mathrm{BSFC} = \frac{\dot{m}_f}{\mathrm{BP}} \tag{3}$$

where the braking power (BP) is the power available at the crankshaft and is written:

$$\mathrm{BP} = \mathrm{P}_{b-max} \times \frac{\mathrm{N}}{\mathrm{N}_{@Pb-max}} \times \left[0.87 + \left(1.13 \times \frac{\mathrm{N}}{\mathrm{N}_{@Pb-max}}\right) - \left(\frac{\mathrm{N}}{\mathrm{N}_{@Pb-max}}\right)^2\right] \tag{4}$$

where P_{b-max} stands for the maximum engine brake power. N is the speed of the engine crankshaft. Finally, $N_{@Pb-max}$ is the speed at the maximum brake power produced.

The volumetric efficiency, $\boldsymbol{\eta}_v$, is expressed as follows [6]:

$$\eta_v = \frac{V_a}{V_b} = \frac{\dot{m}_a \times n_r}{N \times \rho_a \times V_b} \times 100\% \tag{5}$$

where V_a denotes the actual volume of intake air taken into the cylinder/engine, and V_b denotes the engine/theoretical cylinder's volume during the intake engine. The number of crankshaft revolutions in a complete engine cycle is denoted by n_r. Finally, ρ_a denotes air density, which may be computed as follows:

$$\rho_a = \frac{P_a}{R_a \times T_a} \tag{6}$$

where P_a denotes the engine's intake air pressure. Second, R_a is the dry air-specific gas constant. Finally, T_a denotes the temperature of the intake air.

The thermal efficiency, η_{th}, is defined as [6]:

$$\eta_{th} = \frac{\mathrm{BP}}{\dot{m}_f \times \mathrm{LCV}} \times 100\% \tag{7}$$

where LCV denotes the fuel's calorific value, which varies depending on the kind of fuel utilised.

3.3. Phase 3: CFD simulation

Moving on to the third execution stage, which entails the setup and application of the CFD simulation, the simulation programme that will be utilised in this article is ANSYS, which will be used for the analysis necessary to discover the appropriate parameters. However, SolidWorks will be used to create the CAD model first.

3.3.1. Geometry & meshing

The development of the computational domain is the first step in mathematical modelling in CFD. The simulation in this study begins with the inlet valve closed at 570° CA ABDC and finishes with the exhaust valve open at 833° CA BBDC. The model has three bodies, two of which are valve bodies. The engine's fluid domain is the third body. Inlet and outflow make up the fluid domain. The lower half of the fluid domain is shaped like the piston head's combustion chamber. Fig. 2 depicts the model's intake and output valves from a clear viewpoint.

3.3.2. Combustion modelling

This study looks at fuels: pure gasoline and ethanol-gasoline mixtures. E0, E20, E50, and E75 are considered ratios. Table 2 lists the liquid fuel properties that are needed for the simulation. If necessary, more properties will be secured. Because it is the standard combustion model, the b-Weller combustion model was utilised for the simulation in this work. The dimensionless variable b, which defines the species concentration of the combustion reactants in each computational cell, is evaluated in this model. Variable b has a value of 0–1, with 0 indicating that the combustion process has been completed in the entire cell and 1 indicating that no combustion process has occurred. To characterise the development of b in time and space states, the transport equation states [20]:

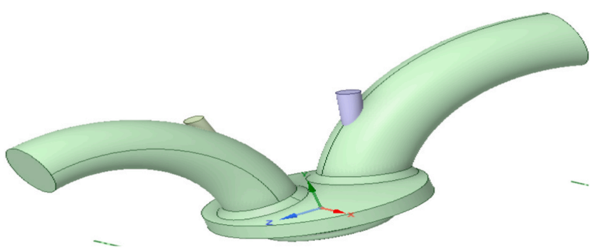

Fig. 2. Inlet & outlet model valve.

$$\frac{\partial(\overline{\rho}\widetilde{b})}{\partial t} + \nabla\cdot(\overline{\rho}\widetilde{\mathbf{u}}\widetilde{b}) - \nabla\cdot\Gamma_b\nabla\widetilde{b} = -\left[\overline{\rho}\Xi + (\overline{\rho}_u - \overline{\rho})\min\left(\Xi, \Xi_{eq}\right)\right]S_u|\nabla\widetilde{b}| \tag{8}$$

where $\overline{\rho}$ is the density. $\overline{\rho}_u$ is the density in the unburnt mixture. $\widetilde{u}$ stands for the fluid flow vector. Γ_b is equal to the turbulent viscosity divided by the turbulent Prandtl coefficient. Finally, S_u is the laminar flame speed.

3.3.3. Governing equations

The simulation is built using a set of fluid dynamics equations. The momentum, continuity, and energy equations are among these equations. The following is the continuity equation Hamid et al. [20]:

$$\partial t + \nabla \times (\rho U) = 0 \tag{9}$$

Where U denotes the three-dimensional flow velocities (x, y, & z-axis), ρ is the fluid's density in kilogrammes per cubic metre, t stands for time and ∇ is the gradient operator.

Second, the following momentum equation is based on Newton's second law, with the surface force on the control volume and the body forces expressed as:

$$\frac{\partial(\rho U)}{\partial t} \times \nabla \times (\rho U \otimes \ U) = -\nabla p \ + \nabla \times \tau + \ S_M \tag{10}$$

where τ stands for the strain rate, which is the rate of deformation with respect to time and S_M stands for the momentum source.

Finally, the energy equation, commonly known as the Navier-Stokes equation, explains the energy change rate inside the fluid element and is stated in Eq. (11).

$$\frac{\partial(\rho h_{tot})}{\partial t} - \frac{\partial \rho}{\partial t} + \nabla \times (U h_{tot}) = \nabla \times (\lambda \nabla T) + \nabla \cdot (U \ \times \tau) + \ U \times \ S_M \tag{11}$$

where h_{tot} is the total enthalpy, and λ stands for the thermal conductivity.

3.3.4. Boundary conditions

The simulation's boundary conditions are as follows (data is derived from various papers) [21,24]:

- Limitation: Maximum engine speed @ 5000 RPM & CR @ 8.9
- Cylinder head temperature: 293.2 K; Piston Bowl: 400 K
- Wall temperature value is maintained at 293.2 K (assuming it is equal to in-cylinder mixture temperature)
- Constant pressure boundary conditions were used at the intake and exhaust manifolds.
- Attach boundaries were specified on the coincident cell face near the cells above/below the valve.
- No-slip wall boundary condition in conjunction with logarithmic law of wall is used.
- Cylinder walls are adiabatic.
- Mass flow rates of fuel are based on values calculated in the results
- Fuel Injectors (Start & End): 23^o Crank Angle (CA) BTDC & 7^o CA After Top Dead Centre (ATDC)
- Injector Nozzle Diameter: 0.15 mm
- Initial injection velocity: 181 m/s – 192 m/s

3.4. Phase 4: validation of results

After successfully acquiring the results, a validity technique will be applied, which will be accomplished by giving an error rate between the theoretical and simulated outcomes. Based on our findings, a simple technique will be employed, in which a percentage of error, or error rate, will be computed by comparing the estimated theoretical outcomes to the simulation results, with an error rate of no more than 5% being required. The validation formula, which is as follows, is used to calculate the % error rate between theoretical

Table 6
Mass numerical values.

Parameter	Value @ E0	Value @ E20	Value @ E50	Value @ E75
$\dot{m}_f$ (kg/s) @ 1000 RPM	0.00246	0.00247	0.00249	0.00262
$\dot{m}_f$ (kg/s) @ 2000 RPM	0.00517	0.00532	0.00548	0.00578
$\dot{m}_f$ (kg/s) @ 5000 RPM	0.014	0.0147	0.01522	0.01582
$\dot{m}_a$ (kg/s) @ 1000 RPM	0.0362	0.0327	0.0289	0.0269
$\dot{m}_a$ (kg/s) @ 2000 RPM	0.076	0.0704	0.0637	0.0593
$\dot{m}_a$ (kg/s) @ 5000 RPM	0.2058	0.1945	0.1767	0.1623

and simulation results:

$$\text{Percentage Error\%} = \frac{\text{Approximate Value} - \text{Exact Value}}{\text{Exact Value}} \times 100\% \tag{12}$$

4. Results analysis & discussion

4.1. Required outcomes

The results of the vehicle's performance and emissions are to be displayed in this subsection. Theoretical findings, which are acquired using the equations indicated in the methodology section, and simulation results, which are generated using ANSYS, are among the results obtained. As previously indicated, performance factors include brake-specific fuel consumption, volumetric and thermal efficiency. Second, for all fuel blends indicated at different defined speed intervals, hydrocarbons, carbon monoxide, carbon dioxide, and nitrogen oxides are measured. After that, both findings are validated, and a % error rate is calculated.

Factors are calculated for each of the fuel blend, namely E0 to E75, as well as different engine speeds, which are 1000, 2000, and 5000 RPM. Table 6 below shows the results obtained for each factor.

Based on the above obtained mass values, it could be concluded that there is a significant relation between the mass flow rate values, fuel blend ratio and speed intervals. Starting off with the mass flow rate of fuel, it is taken as average values from various articles, due to the experiment being simulation-based. It is seen that the mass flow rate of fuel slightly increases with the increase of fuel blend ratio as well as the increase in speed intervals. As for the mass flow rate of air, the fuel blend ratio had increased, the mass flow rate slightly decreased, which is due to the change in AFR value. However, it was observed that for each fuel blend ratio, the speed interval increased from 1000 to 5000 RPM, mass flow rate of air slightly increased. By considering all factors mentioned regarding the mass flow rate of air, one can safely highlight that the highest flow rate of air is achieved for E0 at 5000 RPM, compared to the E75 at 1000 RPM.

The theoretical and simulation values for BSFC at the selected fuel mix percentages and engine speeds are shown in Table 7.

Based on the results presented in Table 7, the fuel mixing ratio affects the BSFC value. When looking at the BSFC values at 5000 RPM, the BSFC value, which starts at 0.4691 kg/kWh for pure gasoline, grew marginally with the addition of ethanol, reaching a value of 0.5301 kg/kWh at an E75 fuel blend ratio. The same may be said for increasing the speed interval. Several reasons contribute to this shift. The first is braking power, which is affected by the change in speed interval. Second, the modification in fuel mass flow rate increased the BSFC value. Finally, it can be shown that both the fuel mix ratio and engine speed positively influence the BSFC value, resulting in an increased value. Furthermore, the increase in BSFC is highly connected to fuel attributes such as reduced heating value, high kinematic viscosity, density, and air-fuel ratio, with the conclusion that the greater density and octane number of ethanol-gasoline blends are responsible for the rise in BSFC Rosdi et al. [1]. Table 8 presents the volumetric efficiency values.

Starting with the fuel mix ratio at 5000 RPM, a reduction is visible as the ratio climbs from E0 to E20, with the value falling from 54.6908% to 51.6828%. Volumetric efficiency improved as the fuel blend was increased to E50 and E75, attaining values of 59.4567% and 61.8006%, respectively. That is because the air mass flow rate slightly decreases when the blend ratio increases. In other words, more air is introduced into the cylinder, enhancing its efficiency. Regarding speed, it had a favourable influence on volumetric efficiency, with the best results coming at 5000 RPM. Hence, volumetric efficiency is the highest using the E75 fuel ratio at 5000 RPM. When compared to pure gasoline, volumetric efficiency for mixed fuel was found to be 0.6% to 4.5% greater.

Furthermore, the charge cooling of the intake manifold while employing ethanol-gasoline blends would explain the rise in volumetric efficiency. In addition, the amount of water in ethanol corresponds to the octane number in gasoline. The volumetric efficiency increases as the octane number rise. Rosdi et al. [1] observed similar results for other ethanol blends, claiming that the higher latent heat of alcohol fuels resulted in lower intake manifold temperature and improved volumetric efficiency. Table 9 shows the thermal efficiency effects of various fuel blends and engine speeds.

Based on the findings, it can be concluded that both engine speed and fuel ratio influence thermal efficiency. The thermal efficiency of the various fuel blends decreased at 1000 RPM, starting at 84.5335% for pure gasoline and decreasing to 78.5956% for E75. Furthermore, the thermal efficiency rating fell as the fuel blend ratio increased. The calorific value of different fuel blends and the mass of fuel augmentation are the reasons for the rise in efficiency. It could also be put into consideration that lean-burn combustion does cause a significant impact on the values as well. Finally, after completing our theoretical calculations, E0 shows the highest thermal efficiency at 1000 RPM, and E75 at 5000 RPM is the least efficient. It should also be considered that lean-burn combustion significantly influences the values. Finally, based on our theoretical calculations, E0 has the maximum thermal efficiency at 1000 RPM, while E75 has the lowest at 5000 RPM. Theoretically, as the speed increases, the thermal efficiency falls.

The use of an ethanol-gasoline blend can result in cleaner combustion due to the increased oxygen content in the chemical structure and superior volatility of ethanol compared to pure gasoline. The most significant CO emissions for the E0 test gasoline were recorded for each engine speed based on the data obtained. The E75 blend has the lowest CO emissions. As a result, CO emissions were reduced

Table 7
BSFC theoretical values.

Engine speed (rpm)	@ E0	@ E20	@ E50	@ E75
1000	0.3986	0.4007	0.4039	0.4248
2000	0.4488	0.4616	0.4758	0.5018
5000	0.4691	0.4926	0.5100	0.5301

Table 8
Volumetric efficiency values.

Engine speed (rpm)	@ E0	@ E20	@ E50	@ E75
1000	48.0498	43.4733	48.6943	51.214
2000	50.5109	46.7606	53.5579	56.478
5000	54.6908	51.6828	59.4567	61.8006

Table 9
Thermal efficiency values.

Engine speed (rpm)	@ E0	@ E20	@ E50	@ E75
1000	84.5335	83.863	82.8561	78.5956
2000	75.0776	72.7928	70.3323	66.54
5000	71.816	68.2124	65.6173	62.9811

as the fuel blend ratio increased.

CO_2 is a product of incomplete combustion, which occurs when there is insufficient air in the air-fuel combination. When ethanol, which has enough oxygen, is blended with gasoline, the engine's combustion improves, and CO_2 emissions are lowered, as seen in the graph. Because ethanol has less carbon than gasoline, it produces less CO_2, which contributes significantly to global warming during combustion. For the same engine speed, CO_2 emissions produced by fuel blends were found to be lower. CO_2 emissions were reduced when blending fuel was used instead of gasoline. The hydroxyl radical (-OH) in ethanol facilitates full combustion in the engine cylinder, which reduces CO_2. In addition, because ethanol has a lower carbon content than gasoline, it produces less CO_2. CO_2 emissions are significantly reduced because of quick evaporation and improved mixing of air and fuel mixtures. Because of the additional oxygen in gasoline fuel blends, good combustion was achieved. CO was altered into CO_2 because of the full combustion of bioethanol in fuel.

The effect of employing ethanol-gasoline blends indicates a decreasing trend in NO_x emission as compared to gasoline, as seen in the graph. The temperature of the cylinder, the oxygen content, and the residence time for the reaction of gas temperature are all factors in the creation of NOx emissions Rosdi et al. [1]. The temperature of the cylinder rises as the cylinder pressure is increased. Because of the high latent heat of vaporisation, lower heating value, and oxygen concentration in alcohol, the combustion temperature in ethanol-gasoline blends is considerably lowered, resulting in fewer NO_x emissions. According to certain studies, the reaction time of each engine cycle was lowered. The temperature of the gas in the cylinder was lowered. NO_x emissions were also reduced due to that condition.

According to the graph, HC emissions drop as ethanol concentration rises. As the amount of ethanol used rises, the mixture becomes more homogeneous, reducing HC emissions and enhancing combustion. Fuel mixing aids full combustion and consequently improves wall quenching by lowering hydrocarbon emissions. In comparison to other fuel blends, gasoline emits more HC. HC emissions usually are present when incomplete combustion occurs. When an engine runs on ethanol-gasoline mixtures, HC emissions decrease. Previous research has found that adding ethanol to gasoline reduces HC emissions because non-polar molecules cannot absorb water molecules in the lubricating oil layer as efficiently as polar ones (Rosdi et al. [1]. As a result, the probability of emitting HC can be minimised. Lean combustion and water in gasoline-ethanol blends improve full combustion, resulting in lower HC levels.

The validation step begins when the findings have been achieved effectively.

Table 10 displays the validation of BSFC values, in which theoretical and simulation results are compared, and a % error rate is calculated.

The validation for volumetric efficiency is shown in Table 11. The initials "Theo." and "Sim." are used in the table to denote theoretical and simulation findings, respectively, with theoretical results acquired using mathematical equations and simulation findings obtained using the specified software. After comparing BSFC theoretical values to simulation data, such as E0 at 1000 RPM, and obtaining BSFC values for both, theoretical with a value of 0.3986 kg/kWh, and simulation with a value of 0.4125 kg/kWh, it is discovered that the percentage error rate for all fuels at different speed intervals is less than 3.38%, indicating that BSFC values are accurate.

The volumetric efficiency achieved for both theoretical and simulation is 48.3% and 47.94%, respectively, using the fuel mix E20 at 1000 RPM as a reference. As a result, after calculating a percentage error rate between these two values, it was discovered that the largest error rate attained is less than 0.86%, supporting the volumetric efficiency acquired.

Finally, Table 12 illustrates the thermal efficiency validation. With E20 at 1000 RPM and theoretical and simulation values of 83.86% and 84.57%, the error rate was determined to be less than 4.87%, verifying the results.

5. Conclusion and future work

This study employed mathematical equations and ANSYS software to analyze significant parameters in terms of performance and emissions of a fuel blend, namely ethanol-gasoline, in various ratios, as well as to test such ratios at various speed intervals. The engine utilised in this experiment is a 2017 Proton Preve Premium CFE CVT with a turbocharged 1.6 L, 4-inline cylinders, and a four-stroke engine. E0, the reference fuel, E20, E50, and E75 were utilised fuel blend ratios. Second, speed intervals of 1000, 2000, and 5000 RPM

Table 10
BSFC validation. (Theo. [25]).

Engine speed (rpm)	@ E0		@ E20		@ E50		@ E75	
	Theo.	Sim.	Theo.	Sim.	Theo.	Sim.	Theo.	Sim.
1000	0.3986	0.4125	0.4007	0.4147	0.4039	0.4181	0.4248	0.4397
2000	0.4488	0.4337	0.4616	0.4461	0.4758	0.4598	0.5018	0.4849
5000	0.4691	0.4695	0.4926	0.493	0.5100	0.5105	0.5301	0.5306
Error rate (%)	<3.38		<3.38		<3.39		<3.38	

Table 11
Volumetric efficiency validation (Theo. [25]).

Engine speed (rpm)	@ E0		@ E20		@ E50		@ E75	
	Theo.	Sim.	Theo.	Sim.	Theo.	Sim.	Theo.	Sim.
1000	48.05	47.65	48.3	47.94	48.69	48.34	51.21	50.83
2000	50.51	50.14	51.96	51.58	53.56	53.17	56.48	56.06
5000	54.69	54.36	57.42	56.93	59.46	59.02	61.8	61.37
Error rate (%)	<0.83		<0.86		<0.73		<0.75	

Table 12
Thermal efficiency validation. (Theo. [25]).

Engine speed (rpm)	@ E0		@ E20		@ E50		@ E75	
	Theo.	Sim.	Theo.	Sim.	Theo.	Sim.	Theo.	Sim.
1000	84.533	84.53	83.86	84.57	82.86	83.52	78.59	79.45
2000	77.1	80.44	74.79	78.62	73.33	75.94	68.54	72.04
5000	71.82	74.3	68.21	71.13	65.62	68.4	62.98	65.84
Error rate (%)	<4.36		<4.87		<4.07		<4.86	

were employed at all fuel blend percentages. Firstly, it was found that adding ethanol to gasoline, as well as increasing engine speed, improves BSFC value and volumetric efficiency, with E75 achieving values of 0.5301 kg/kWh and 61.8% at 5000 RPM, proving to be the most efficient with increases of 6.59% in BSFC and 4.5% in volumetric efficiency when compared to pure gasoline. In addition, the optimal mass numerical value for E75 at the speed 5000 RPM was found as 0.1603, which is Kg/s. Secondly, due to the calorific value at different fuel blends, thermal efficiency decreases with increasing fuel blend ratio as well as engine speed increment, with E0 at 1000 RPM having the highest thermal efficiency with a value of 84.53%, which is 6% higher than that of E75 fuel ratio at the same speed.

Moreover, the higher ethanol percentage resulted in a considerable reduction in emissions. Furthermore, engine speed has a beneficial influence on emissions by decreasing them, with E75 emitting 21.74% less at 5000 RPM than pure gasoline at the same speed. Finally, the error rate is within an acceptable range for all parameters, which is less than 4.86%. This result attributes to the ethanol for the same engine speed holding less carbon than gasoline, and less CO_2 is produced, which plays a significant role in global warming during combustion. CO_2 emissions produced for fuel mixtures were lower for the same engine speed.

Several ethanol-gasoline blend ratios are to be tested to determine the effect of fuel on the vehicle's performance and emissions. Second, using fuel-blending, investigate the features of in-cylinder airflow in a turbocharged engine. Finally, the influence of altering the combustion chamber shape with fuel-blending is to be investigated.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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