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CFD, Performance, and Emission Analysis of a Compression Ignition Engine Using Compressed Biogas (CBG) Produced from Municipal Organic Solid Waste

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Abstract: ***Rapid urbanization has substantially increased municipal solid waste (MSW) generation, necessitating sustainable waste management and renewable energy solutions. Municipal organic solid waste (MOSW), constituting 60–70% of MSW, represents a valuable feedstock for biogas production through anaerobic digestion. This study investigates the production of compressed biogas (CBG) from biodegradable MOSW and its utilization in a compression ignition (CI) engine operating in dual-fuel mode. Vegetable wastes comprising brinjal, cabbage, cucumber, carrot, and potato were characterized using total solids, moisture content, total organic carbon, and volatile solids. Mechanical, hydrothermal, chemical, and biological pretreatments were evaluated to enhance biomethanation, followed by anaerobic co-digestion with garden leaf waste using NaOH-treated substrates and biogas slurry inoculum. Raw biogas was upgraded by removing moisture, CO_2, and H_2S using gas purification techniques and supplied to a single-cylinder CI engine with diesel as the pilot fuel. Computational Fluid Dynamics (CFD) was employed to optimize a Venturi gas mixer by analyzing different flow-pass configurations and gas inlet angles. The optimized four-pass mixer with a 45° gas inlet was fabricated and experimentally validated. The dual-fuel engine achieved up to 85% diesel substitution while maintaining satisfactory brake thermal efficiency, improved combustion characteristics, and reduced CO and CO_2 emissions. The proposed integrated approach demonstrates the feasibility of utilizing MOSW-derived CBG as a sustainable alternative fuel for compression ignition engines, promoting renewable energy generation, reduced fossil fuel dependence, lower emissions, and effective municipal waste management.***

Keywords: Compressed Biogas (CBG), Municipal Organic Solid Waste (MOSW), Computational Fluid Dynamics (CFD), Engine Performance and Emissions.

1. Introduction

Rapid urbanization, industrialization, and population growth have significantly increased the generation of Municipal Solid Waste (MSW), creating major environmental and public health challenges, particularly in developing countries. In India, Municipal Organic Solid Waste (MOSW), comprising vegetable waste, fruit waste, food waste, garden residues, flowers, and paper, accounts for nearly 60- 70% of the total MSW. Effective utilization of this biodegradable fraction through renewable energy production offers a sustainable solution for waste management while reducing landfill disposal and associated environmental impacts.

India generates over 100,000 metric tonnes of MSW daily, with biodegradable waste constituting approximately 70–80% in many urban areas. Conventional disposal practices such as open dumping and landfilling contribute to greenhouse gas emissions, groundwater contamination, odour problems, and loss of valuable resources. Consequently, sustainable Municipal Solid Waste Management (MSWM) emphasizing waste segregation, recycling, resource recovery, and energy generation has become an essential component of circular economy initiatives.

Anaerobic digestion (AD) is a well-established biological process for converting biodegradable organic waste into biogas, primarily composed of methane (CH_4) and carbon dioxide (CO_2). The process involves hydrolysis, acidogenesis, acetogenesis, and methanogenesis, with hydrolysis often acting as the rate-limiting step, particularly for lignocellulosic biomass. Appropriate pre-treatment methods such as mechanical, chemical, thermal, and biological treatments improve substrate biodegradability and enhance methane production.

Vegetable market waste is an attractive feedstock for anaerobic digestion because of its high moisture content, favorable carbon-to-nitrogen ratio, and rapid bio-degradability. However, its fast degradation may result in excessive volatile fatty acid accumulation and pH reduction, inhibiting methanogenic activity. Co-digestion with garden leaf waste and suitable pre-treatment techniques can improve process stability and biogas yield. In the present study, vegetable wastes including brinjal, cabbage, cucumber, carrot, and potato were characterized based on moisture content, total solids, volatile solids, and total organic carbon prior to anaerobic co-digestion using biogas slurry as inoculum.

Raw biogas contains impurities such as CO_2, H_2S, and moisture, which reduce its calorific value and limit its direct application in internal combustion engines. Upgrading biogas to increase methane concentration above 90% produces Compressed Biogas (CBG), a renewable fuel with properties comparable to compressed natural gas (CNG). CBG has emerged as a promising alternative transportation fuel due to its potential to reduce fossil fuel consumption and greenhouse gas emissions.

Compression Ignition (CI) engines operating in dual-fuel mode, with diesel as the pilot fuel and CBG as the primary fuel, offer an effective approach for improving fuel sustainability while maintaining engine performance. The efficiency of dual-fuel combustion largely depends on air–fuel mixing and combustion characteristics. Computational Fluid Dynamics (CFD) has become an effective tool for optimizing gas mixer geometry, intake flow, and combustion processes, thereby reducing development time and experimental cost.

In this study, CFD was employed to design and optimize single-pass, two-pass, and four-pass gas mixing chambers at different inclination angles. The optimized mixer was fabricated and integrated into a single-cylinder CI engine operating in diesel–CBG dual-fuel mode. Experimental investigations were conducted under various engine loads and CBG substitution ratios to evaluate brake thermal efficiency, brake specific fuel consumption, combustion characteristics, and exhaust emissions. CFD predictions were validated through experimental results to assess the effectiveness of the optimized gas mixer.

The present work integrates municipal organic waste management, anaerobic digestion, biogas upgrading, CFD-based gas mixer optimization, and dual-fuel engine experimentation into a comprehensive framework for sustainable energy production. The study demonstrates the feasibility of utilizing CBG derived from Municipal Organic Solid Waste as an alternative fuel for CI engines, contributing to renewable energy utilization, reduced fossil fuel dependence, lower exhaust emissions, and environmentally sustainable waste management.

2. Literature Review

The increasing demand for sustainable energy and the depletion of fossil fuel resources have intensified research on renewable fuels for compression ignition (CI) engines. Among the available alternatives, compressed biogas (CBG) has gained considerable attention owing to its renewable nature, environmental benefits, and compatibility with existing CI engines. Biogas produced from municipal organic solid waste (MOSW) offers a dual advantage by reducing landfill waste while generating clean and sustainable energy [1, 2].

Biogas is produced through anaerobic digestion of biodegradable organic materials such as municipal organic waste, food waste, agricultural residues, and animal manure. Raw biogas typically contains 50–70% methane (CH_4), 30–45% carbon dioxide (CO_2), and trace amounts of hydrogen sulphide (H_2S), water vapour, and other impurities, which reduce its calorific value and engine performance. Previous studies have demonstrated that feedstock characteristics and process optimization significantly influence methane yield and biogas quality [13,15–17].

Biogas upgrading is essential to increase methane concentration and improve fuel properties. Methane-enriched CBG exhibits combustion characteristics comparable to compressed natural gas (CNG), resulting in higher calorific value, improved combustion stability, and enhanced engine performance. Studies have reported that biogas containing more than 85% methane is well suited for CI engine applications due to its superior combustion characteristics and lower environmental impact [12, 14].

Dual-fuel operation, where CBG is inducted through the intake manifold and diesel is injected as the pilot fuel, is an effective approach for utilizing gaseous fuels in conventional CI engines with minimal modifications. Previous investigations have shown that engine performance is strongly influenced by methane concentration, pilot diesel quantity, injection timing, compression ratio, and operating conditions. Proper optimization enables stable combustion while substantially reducing diesel consumption [1–5].

Brake thermal efficiency (BTE) generally decreases with increasing biogas substitution because of the lower heating value of biogas compared with diesel. However, methane enrichment and optimization of pilot diesel injection considerably improve combustion efficiency and minimize the reduction in BTE. Experimental studies have demonstrated that high diesel replacement ratios can be achieved while maintaining satisfactory engine performance [1, 4, 11].

Similarly, brake-specific fuel consumption (BSFC) generally increases with increasing biogas substitution owing to the lower energy density of gaseous fuels. Nevertheless, upgrading biogas to increase methane concentration improves fuel economy and reduces the increase in BSFC compared with raw biogas [3, 5, 11, 12].

Combustion characteristics, including ignition delay, heat release rate, combustion duration, and peak cylinder pressure, govern engine efficiency and emissions. In dual-fuel engines, the methane–air mixture is ignited by pilot diesel injection, producing predominantly premixed combustion. Optimization of pilot diesel injection has been shown to improve combustion stability and heat release characteristics, particularly at high diesel substitution levels [2,4,19].

Emission characteristics have been widely investigated because of stringent environmental regulations. Methane-rich CBG significantly reduces carbon monoxide (CO), smoke, and particulate matter emissions owing to improved air–fuel mixing and cleaner combustion. However, hydrocarbon (HC) emissions generally increase under lean operating conditions because of incomplete oxidation of methane. Optimization of pilot fuel quantity, injection timing, and intake turbulence effectively mitigates HC emissions while maintaining high diesel substitution ratios [1–5].

Nitrogen oxide (NO_x) emissions remain a major challenge in methane-based dual-fuel engines due to elevated combustion temperatures at higher engine loads. Previous studies have shown that exhaust gas recirculation (EGR), delayed pilot injection, and optimized combustion strategies effectively reduce NO_x formation without significantly compromising engine performance [1, 3, 4].

Computational Fluid Dynamics (CFD) has become an important tool for optimizing gaseous fuel induction systems by predicting pressure distribution, velocity fields, turbulence

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intensity, and fuel concentration. CFD-based investigations have demonstrated that optimized intake system geometry significantly enhances air–fuel mixing while reducing pressure losses [6–8].

The Venturi gas mixer plays a crucial role in achieving homogeneous air–fuel mixing in dual-fuel engines. Numerical and experimental studies have reported that appropriate selection of throat diameter, converging and diverging angles, and gas nozzle orientation substantially improves mixing efficiency, combustion characteristics, and overall engine performance while reducing emissions [8,10,18].

Although extensive studies have been conducted on biogas production, upgrading technologies, dual-fuel combustion, emission control, and CFD-based intake system optimization, limited research has integrated MOSW-derived compressed biogas production, CFD optimization of the Venturi gas mixer, and comprehensive experimental evaluation of engine performance, combustion, and emissions within a single framework. Moreover, studies on high diesel substitution using methane-enriched CBG produced specifically from municipal organic solid waste remain limited. Therefore, the present study addresses this research gap by integrating CBG production from MOSW, CFD-assisted Venturi mixer optimization, and experimental investigation of the performance, combustion, and emission characteristics of a CI engine operating in diesel–CBG dual-fuel mode.

3. Objectives

The primary objective of this study is to evaluate the feasibility of utilizing Compressed Biogas (CBG) produced from Municipal Organic Solid Waste (MOSW) as an alternative fuel for a Compression Ignition (CI) engine through biogas production, Computational Fluid Dynamics (CFD)-based gas mixer optimization, and experimental performance and emission analyses.

The specific objectives are:

- To produce and upgrade biogas from biodegradable Municipal Organic Solid Waste (MOSW) through anaerobic digestion by optimizing mechanical, hydrothermal, chemical, and biological pretreatment methods, characterizing feedstock and digestion process parameters, and generating high-quality Compressed Biogas (CBG) suitable for Compression Ignition (CI) engine applications.
- To design and optimize a Venturi-type gas mixer using CFD by evaluating different geometrical configurations and gas injection angles for enhanced air–fuel mixing.
- To fabricate the optimized gas mixer and integrate it with a single-cylinder CI engine operating in diesel–CBG dual-fuel mode.
- To experimentally evaluate the performance, combustion, and exhaust emission characteristics of a Compression Ignition (CI) engine under varying engine loads and Compressed Biogas (CBG) substitution ratios, and compare the results with conventional diesel operation.
- To validate the CFD predictions through experimental investigations and evaluate the effectiveness of the optimized as mixer in improving air–fuel mixing, combustion, and engine performance.

4. Methods and Methodology

4.1 Biogas Production from Municipal Organic Solid Waste

Municipal organic solid waste (MOSW), comprising vegetable, fruit, and garden waste, was manually segregated to remove non-biodegradable materials. The biodegradable fraction was subjected to mechanical, thermal (≤120 °C), chemical (dilute acid/alkali), and biological pretreatments to improve biodegradability and methane yield. The pretreated substrate was mixed with water and inoculum to form a homogeneous slurry and digested anaerobically under controlled pH, temperature, and hydraulic retention time. Biogas production occurred through hydrolysis, acidogenesis, acetogenesis, and methanogenesis under mesophilic (35 ± 2 °C) and thermophilic (50 ± 2 °C) conditions.

4.2 Feedstock and Biogas Characterization

Feedstock and digester slurry were characterized for total solids (TS), volatile solids (VS), moisture content (MC), total organic carbon (TOC), chemical oxygen demand (COD), total dissolved solids (TDS), total suspended solids (TSS), pH, and carbon-to-nitrogen (C/N) ratio following standard methods. Biogas composition (CH_4, CO_2, H_2S, and trace gases) was analyzed prior to upgrading.

4.3 Biogas Upgrading and Compression

Raw biogas was upgraded by removing moisture, CO_2, and H_2S using gas purification techniques. The purified gas, containing >85% methane, was compressed into high-pressure cylinders as compressed biogas (CBG) for engine experiments.

4.4 CFD-Based Venturi Gas Mixer Design

A Venturi gas mixer was designed and optimized using ANSYS Fluent. The design considered CBG calorific value, intake air velocity, air–fuel flow rates, stoichiometric air–fuel ratio, and Venturi geometry. CFD simulations evaluated pressure, velocity, turbulence, and air–CBG mixing characteristics for different configurations. The optimized mixer comprised a 20° converging angle, 5° diverging angle, 45° gas nozzle inclination, 30 mm inlet/outlet diameter, and 18 mm throat diameter, ensuring uniform mixing with minimal pressure loss.

4.5 Fabrication of the Optimized Gas Mixer

The optimized Venturi mixer was fabricated from steel and installed in the intake manifold of a single-cylinder, four-stroke, water-cooled compression ignition engine to provide homogeneous air–CBG mixing during dual-fuel operation.

4.6 Engine Experimental Investigation

Engine tests were conducted at 1500 rpm, 210 bar injection pressure, and 23° bTDC injection timing under steady-state

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conditions. The engine was operated in diesel mode and diesel–CBG dual-fuel mode using the optimized mixer. Brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), and brake power (BP) were measured. Combustion characteristics were assessed from in-cylinder pressure, while exhaust emissions (CO, HC, CO_2, and NO_x) were quantified using calibrated gas analyzers.

4.7 Validation and Data Analysis

CFD predictions were validated against experimental performance and combustion data. Comparative analyses between diesel and diesel–CBG dual-fuel operation were performed to evaluate the effectiveness of the optimized Venturi mixer. The effects of CBG substitution on engine performance, combustion, and emissions were analyzed to assess the feasibility of utilizing MOSW-derived CBG as a sustainable fuel for compression ignition engines.

5. Computational Fluid Dynamics (CFD) Analysis of the Venturi Gas Mixer

The three-dimensional (3D) models of the Venturi gas mixer developed for the two gas inlet configurations (90° and 45°**)** are shown in Figure 5(a)**.** These models were imported into the Computational Fluid Dynamics (CFD) software FLOWVISION to investigate the air–fuel mixing characteristics of compressed biogas (CBG) and atmospheric air. The primary objective of the CFD analysis was to evaluate the influence of gas inlet inclination on the flow behavior and mixing performance within the Venturi mixer, with the aim of achieving a homogeneous air–fuel mixture before entering the engine intake manifold. Efficient mixing is essential for improving combustion efficiency, enhancing engine performance, and minimizing exhaust emissions.

The CFD simulations were performed under identical boundary conditions for both inlet configurations. The distributions of velocity, pressure, turbulence, and methane concentration were analyzed to assess the mixing characteristics inside the Venturi mixer. As illustrated in Figure 5(b)**,** the flow field was examined at four successive cross-sectional planes along the length of the mixer. These four analysis planes represent the progressive development of the air–methane mixture from the gas entry region to the outlet, enabling a detailed comparison of the mixing process for the two inlet orientations.

The simulation results obtained from the four analysis planes were used to compare the performance of the 90° and 45° gas inlet configurations**.** The 45° gas inlet inclination produced smoother flow patterns, reduced flow separation and recirculation, and more uniform methane concentration throughout the mixing chamber than the 90° configuration. Consequently, the 45° configuration achieved superior air–fuel homogeneity, making it the preferred design for supplying a well-mixed combustible mixture to the engine intake manifold.

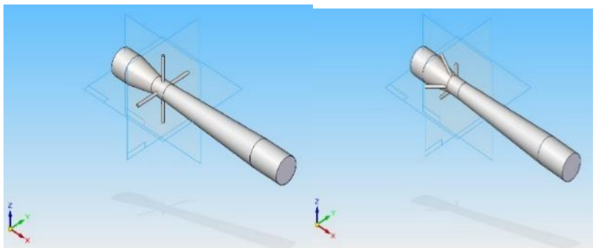

Figure 5(a): Three-dimensional models of the Venturi gas mixer developed for CFD analysis with 90° and 45° gas inlet configurations

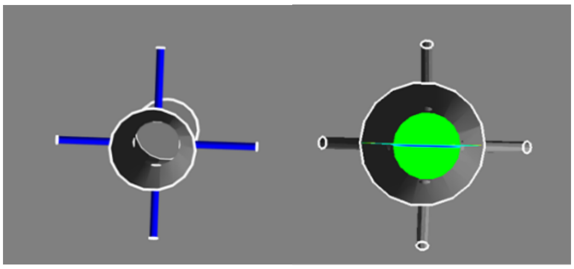

Figure 5(b): CFD flow analysis in FLOWVISION showing the four analysis planes for the 90° and 45° gas inlet inclination configurations

5.1 Boundary conditions

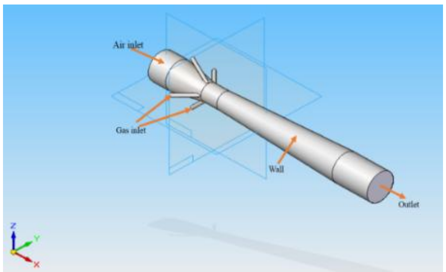

Figure 5.1: Boundary conditions applied to the CFD model

The boundary conditions for the CFD model were assigned as shown in Figure 5.1. Four boundary conditions were specified in FLOWVISION to accurately simulate the mixing of air and methane gas within the Venturi-shaped gas mixer. These include two inlet boundaries, one wall boundary, and one outlet boundary.

The CFD model was defined using four boundary conditions. Atmospheric air entered the mixing chamber through the air inlet at a pressure of 1.0 bar, while compressed methane was supplied through the gas inlet at 1.5 bar to ensure effective injection and mixing with the incoming air. The Venturi mixer walls were modeled as stationary, impermeable surfaces with a no-slip boundary condition**,** confining the flow within the mixing chamber. At the outlet, a suction pressure of 0.7 bar (negative gauge pressure) was applied to represent the air–methane mixture entering the engine intake manifold and to simulate the intake suction generated by the Compression Ignition (CI) engine during the induction stroke.

5.2 CFD Analysis of Air–Gas Mixing Contours

The mixing contours of the air–gas flow at the first circular plane for two gas inlet configurations: 90° and 45°. In the 90° gas inlet configuration, the methane jet enters perpendicular to the airflow, resulting in non-uniform mixing and localized high-concentration regions due to flow separation and recirculation. In contrast, the 45° gas inlet configuration promotes smoother gas entry along the airflow direction, producing a more uniform concentration distribution and enhanced air–fuel mixing. The improved mixing characteristics of the 45° inlet reduce flow disturbances and contribute to better combustion performance, making it the preferred configuration for the Venturi gas mixer design.

5.2.1. Pressure Contours

The pressure contours obtained from the CFD simulation are shown in Figure 5.2.1. The contour plot illustrates the pressure distribution and mixing characteristics of atmospheric air and methane gas inside the Venturi-shaped gas mixer. In the contour, red represents the atmospheric air region, while blue represents the methane gas entering through the fuel inlet.

It can be clearly observed from the pressure contours that the volume fraction of atmospheric air is significantly higher than that of methane gas, which is consistent with the air–fuel ratio required for efficient combustion in a compression ignition engine operating in dual-fuel mode. As the flow progresses through the mixing chamber, the methane gas gradually disperses into the air stream, resulting in a more homogeneous air–fuel mixture before entering the engine intake manifold.

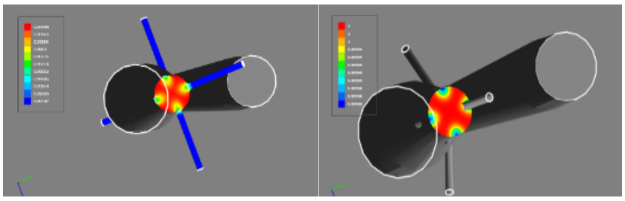

Figure 5.2.1: Gas flow entering the first circular plane with 90° and 45° gas inlet configurations

5.2.2 Flow Distribution at the First Circular Plane

The flow distribution of air and methane gas at the first circular plane is shown in Figure 5.2.2. As the gases enter the mixing chamber, the negative pressure applied at the outlet creates a suction effect that draws both air and methane toward the outlet.

It can be observed that the methane gas does not immediately reach the center of the first circular plane because it enters the mixing chamber through a side inlet oriented at 90° to the main airflow direction. Instead, the methane gas initially flows along the side of the chamber and gradually moves toward the center under the influence of the outlet suction and the surrounding airflow. This flow pattern promotes progressive mixing of methane with atmospheric air before the mixture enters the engine intake manifold, thereby improving the homogeneity of the air–fuel mixture.

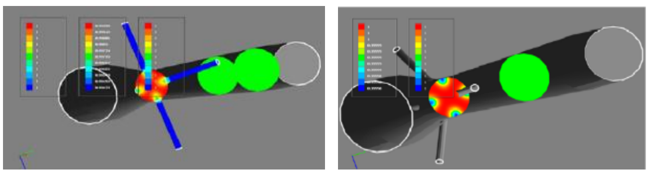

Figure 5.2.2: Mixing contours at the first circular plane for 90° and 45° gas inlet configurations.

5.2.3 Flow Distribution at the Second Circular Plane

The flow distribution at the second circular plane is shown in Figure 5.2.3. Compared to the first circular plane, a significantly improved mixing pattern of atmospheric air and methane gas is observed at this section of the mixing chamber. As the flow progresses downstream, the interaction between the two gases increases, resulting in a more uniform air–fuel mixture.

The CFD results indicate that the methane gas injected at an angle of 45° exhibits superior mixing characteristics compared to the 90° injection configuration. The 45° inlet provides a smoother flow path, allowing the methane gas to penetrate the main air stream more effectively and promoting enhanced turbulence and mixing. In contrast, the 90° inlet causes a more abrupt change in flow direction, leading to comparatively lower mixing efficiency.

Therefore, the 45° methane inlet configuration offers better air–fuel mixing and is expected to improve combustion performance in the compression ignition engine.

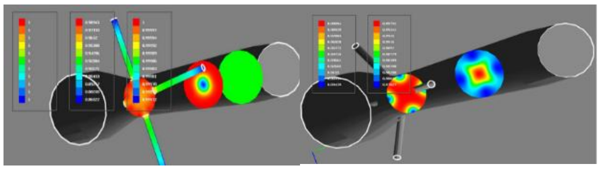

Figure 5.2.3: Mixing contours at the second circular plane for 90° and 45° gas inlet configurations

5.2.4. Flow Distribution at the Third Circular Plane

The flow distribution at the third circular plane is shown in Figure 5.2.4. The CFD pressure contours indicate that the mixing of atmospheric air and methane gas is nearly complete at this section of the mixing chamber. Compared with the first and second circular planes, the third plane exhibits a much more uniform distribution of the air–fuel mixture.

From the pressure contour analysis of all three circular planes, it is evident that the methane gas gradually disperses into the air stream as it travels along the flow path within the Venturi-shaped mixing chamber. The interaction between the gases, together with the chamber geometry and outlet suction, enhances turbulence and promotes progressive mixing. By the time the flow reaches the third circular plane, the methane gas is uniformly distributed throughout the air stream, indicating the formation of a homogeneous air–fuel mixture before entering the engine intake manifold.

The results confirm that the designed mixing chamber provides effective air–methane mixing, which is expected to improve combustion efficiency and reduce engine emissions.

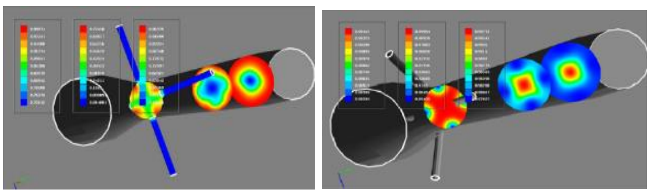

Figure 5.2.4: Mixing contours at the third circular plane for 90° and 45° gas inlet configurations

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5.2.5 Comparison of 90° and 45° Gas Inlet Configurations

From Figure 5.2.5. it can be observed that when methane gas is introduced into the Venturi gas mixer through a 90° inlet, a localized backflow or recirculation zone is created near the Venturi section. This backflow opposes the incoming air stream, resulting in increased flow resistance and a reduction in the engine's volumetric efficiency. The abrupt change in the flow direction also reduces the effectiveness of air–fuel mixing.

To overcome this limitation, a four-port gas mixer with 45° inclined methane inlets was adopted. The 45° inlet configuration allows the methane gas to enter the main airflow more smoothly, minimizing flow separation and backflow while promoting better turbulence and homogeneous mixing. Consequently, the 45° gas mixer provides improved air–fuel mixing and enhances the volumetric efficiency of the engine compared with the conventional 90° gas inlet design.

Furthermore, the CFD analysis indicates that increasing the length of the gas mixer increases the residence time of the gases within the mixing chamber, thereby improving the mixing process. The extended flow path allows methane and air to interact more effectively, resulting in a more homogeneous air–fuel mixture before entering the engine intake manifold.

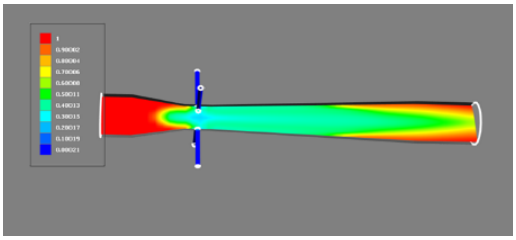

Figure 5.2.5: Mixing contours at the fourth pass on the flat plane with a 90° gas inlet inclination

5.2.6 Velocity Vector Contours

The velocity vector contours obtained from the CFD simulation are presented in Figure 5.2.6, illustrating the flow characteristics of the air–methane mixture within the Venturi gas mixer equipped with a 45° gas inlet inclination.

The velocity distribution indicates that the flow velocity is lowest near the walls of the mixing chamber due to the no-slip boundary condition, where the fluid velocity approaches zero at the wall surface. As the flow moves away from the walls toward the center of the passage, the velocity gradually increases, reaching its maximum value along the central axis. A significant increase in velocity is observed at the Venturi throat, where the cross-sectional area is minimum. This acceleration is attributed to the Venturi effect, which converts pressure energy into kinetic energy as the fluid passes through the converging section. The resulting high-velocity region generates stronger turbulence and enhances the entrainment and mixing of methane with the incoming air. Consequently, a more uniform and homogeneous air–fuel mixture is achieved before entering the engine intake manifold, leading to improved combustion characteristics and overall engine performance.

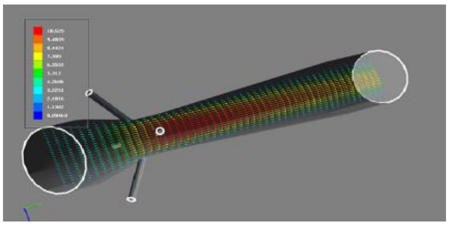

Figure 5.2.6: Velocity vector contours at the fourth plane for the Venturi gas mixer with a 45° gas inlet inclination

5.3 Design of the Venturi Gas Mixer

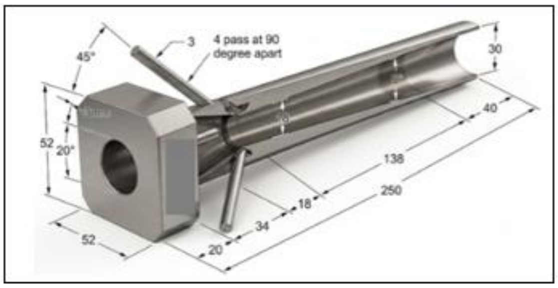

Figure 5.3: Model of the Venturi mixer

The design of the Venturi gas mixer was carried out through a systematic procedure based on the engine operating conditions and the fuel properties of compressed biogas (CBG). Initially, the actual calorific value of the biogas fuel was determined to evaluate its energy content. Based on the selected engine operating conditions, the intake air velocity was calculated. Subsequently, the mass flow rate of biogas required to achieve the desired diesel replacement ratio was determined, followed by the calculation of the corresponding intake air mass flow rate. Using these values, the stoichiometric air–fuel ratio for the biogas fuel was calculated. Finally, the Venturi section was designed using the calculated flow parameters together with established Venturi design guidelines reported in the literature.

The geometric parameters of the Venturi mixer were selected based on recommendations available in the literature. The design criteria included a converging angle of 15°–20°, a diverging angle of 5°–10°, a gas nozzle inclination angle ranging from 10° to 90°, and a diameter ratio (β), defined as the ratio of throat diameter to inlet diameter, of 0.6. Based on CFD analysis and design optimization, the final dimensions selected for the present study consisted of a converging angle of 20°, a diverging angle of 5°, and a gas nozzle inclination angle of 45°. The mixer was designed to achieve a maximum diesel replacement of 80% using biogas. The inlet and outlet diameters were both maintained at 30 mm, while the throat diameter was selected as 18 mm, corresponding to a diameter ratio (β) of 0.6. The length of the converging section was fixed at 54 mm, the diverging section length at 178 mm, and the overall length of the Venturi gas mixer at 250 mm.

The 45° gas nozzle inclination was selected because the CFD results demonstrated superior air–fuel mixing with reduced flow recirculation compared with the conventional 90° nozzle configuration. The throat diameter of 18 mm provided sufficient acceleration of the intake air through the Venturi throat, thereby generating the pressure differential required for effective biogas induction. The optimized geometric dimensions ensured homogeneous mixing of biogas and air

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while minimizing pressure losses, resulting in improved volumetric efficiency and enhanced combustion characteristics of the dual-fuel compression ignition engine.

6. Working of the Experimental Setup

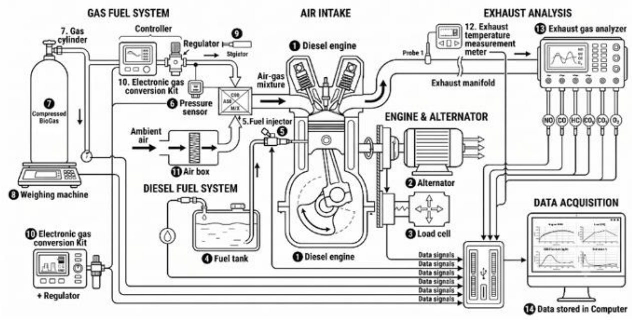

Figure 6: Schematic diagram of the experimental setup

The experimental setup operates on a dual-fuel combustion principle where Compressed Bio-Gas (CBG) acts as the primary gaseous energy source to substitute for diesel, while a small quantity of diesel fuel acts as a pilot ignition source. This integrated system features precise fuel delivery, power absorption mechanisms, and real-time electronic data acquisition.

Regarding the air and gaseous fuel induction system, the compressed bio-gas is stored at high pressure in a gas cylinder which is placed directly on a digital weighing machine to precisely monitor the gravimetric fuel consumption rate over time by measuring the continuous weight reduction. The gas then flows through an electronic gas conversion kit equipped with a pressure regulator to step down the high cylinder pressure to a safe operational level. Simultaneously, atmospheric air enters through an air box that dampens intake pressure pulsations and ensures clean air delivery. The regulated gas subsequently passes a pressure sensor and enters a gas mixer or mixing chamber where the air and gaseous CBG are thoroughly blended to form a homogeneous air-fuel mixture before entering the engine cylinder through the intake manifold.

For the liquid fuel injection system, conventional diesel is drawn from a dedicated fuel tank and passes through the fuel lines to the fuel injector. Unlike standard diesel engines where diesel provides 100% of the energy, this dual-fuel mode injects a reduced pilot quantity of diesel near the end of the compression stroke. Because CBG has a high self-ignition temperature, the highly atomized pilot diesel auto-ignites first under compression, creating multiple flame fronts that subsequently ignite the surrounding air-CBG mixture.

In terms of power absorption and load measurement, the combustion of this dual-fuel mixture drives the piston inside the diesel engine to convert chemical energy into mechanical rotation at the crankshaft. The engine's crankshaft is rigidly coupled to an alternator or dynamometer that applies a controlled electrical or mechanical load to the engine to simulate real-world operating conditions. A precision load cell is mounted to the alternator assembly to measure the reaction torque, allowing the brake power of the engine to be accurately computed by combining this force data with the engine speed.

For emission and exhaust analysis, the high-temperature combustion products exit via the exhaust manifold where an exhaust temperature measurement meter utilizing a thermocouple probe monitors the exhaust gas temperature as a critical indicator of combustion efficiency and thermal loading. Concurrently, a sample of the exhaust gas is routed into an exhaust gas analyzer that quantifies the volumetric concentration of emissions, such as nitrogen oxides, carbon monoxide, unburnt hydrocarbons, carbon dioxide, and residual oxygen, to evaluate environmental performance.

Finally, the data acquisition and automation system continuously route raw electronic signals from the load cell, pressure sensor, weighing machine, thermocouple, and exhaust gas analyzer through a centralized data acquisition system interface unit. These processed signals are fed directly into a computer running dedicated engine performance analysis software that logs, processes, and displays real-time graphical variations of vital parameters—such as brake thermal efficiency, specific fuel consumption, cylinder pressure versus crank angle, and emission trends- to enable systematic analysis.

7. Results and Discussions

The engine experiments were conducted under the optimized operating conditions of 210 bar diesel injection pressure**,** 1500 rpm engine speed**,** and 23° before top dead center (bTDC) injection timing**.** These parameters were selected based on preliminary optimization studies to achieve stable combustion, improved brake thermal efficiency, and reduced exhaust emissions during diesel–compressed biogas (CBG) dual-fuel operation. Under these optimized conditions, the engine performance, combustion characteristics, and exhaust emissions were evaluated at different brake power levels and varying diesel substitution ratios using compressed biogas produced from municipal organic solid waste.

7.1 Engine Performance, Combustion Characteristics

7.1.1 Brake Thermal Efficiency

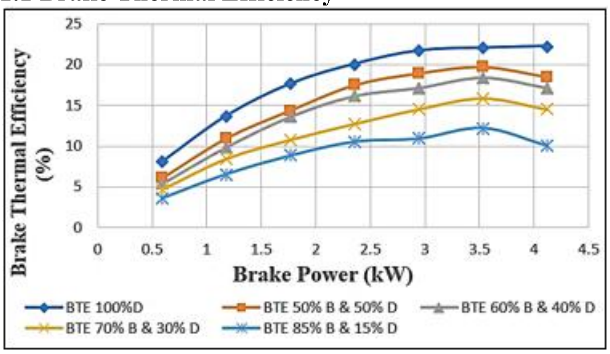

Figure 7.1.1: Variation of Brake Thermal Efficiency (BTE) with Brake Power (BP) for different biogas substitution percentages

The variation of brake thermal efficiency (BTE) with brake power (BP) for different biogas substitution levels is shown in Figure 7.1.1. The engine operated smoothly with diesel replacement by biogas up to 85%. It was observed that the BTE increased with an increase in brake power up to the part-

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load condition due to improved combustion and better utilization of the supplied fuel. However, at full-load operation, the BTE showed a slight reduction because of the higher fuel consumption required to meet the increased power demand. Furthermore, the BTE decreased with increasing biogas substitution, primarily due to the lower calorific value of biogas (approximately 37,000 kJ/kg) compared to diesel fuel. Nevertheless, the engine exhibited stable dual-fuel operation even at high biogas substitution levels. The maximum BTE recorded at an 85% diesel replacement, the feasibility of operating the compression ignition engine with a high proportion of biogas while maintaining satisfactory thermal performance.

7.1.2 Specific Fuel Consumption

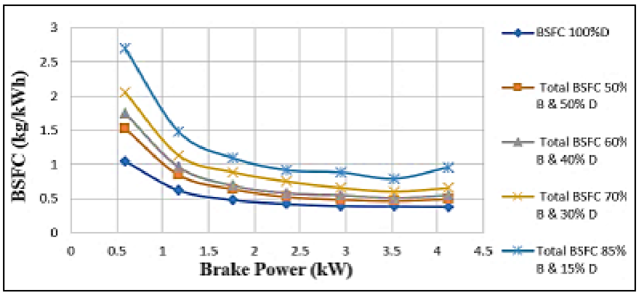

Figure 7.1.2: Variation of Brake Specific Fuel Consumption (BSFC) with Brake Power (BP) for different biogas substitution percentages

The variation of brake specific fuel consumption (BSFC) with brake power (BP) for different biogas substitution levels is shown in Figure 7.1.2. It was observed that the BSFC increased with increasing biogas substitution across the entire range of brake power. This increase is primarily attributed to the lower calorific value of biogas compared to diesel, which requires a greater quantity of fuel to produce the same brake power output. In general, BSFC decreased with increasing brake power up to the optimum operating condition due to improved combustion efficiency; however, higher biogas substitution resulted in comparatively higher BSFC values. The maximum BSFC of 2.73 kg/kWh was recorded at a brake power of 0.588 kW**,** corresponding to the highest biogas substitution level investigated.

7.1.3 Pressures curves under the conditions at 210bar pressure, 23^0 bTDC

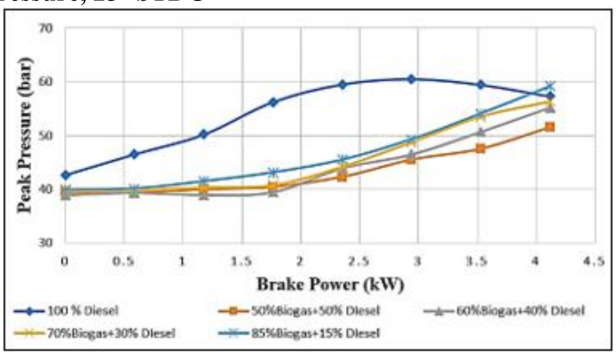

Figure 7.1.3: Variation of peak cylinder pressure with Brake Power (BP) for different biogas substitution percentages

The variation of in-cylinder peak pressure for different diesel–biogas substitution ratios is shown in Figure 7.1.3. The peak pressure characteristics were analyzed for five operating conditions: 100% diesel, 50% biogas with 50% diesel, 60% biogas with 40% diesel, 70% biogas with 30% diesel, and 85% biogas with 15% diesel. It was observed that the peak cylinder pressure varied with the percentage of biogas substitution due to changes in the combustion characteristics of the dual-fuel mixture. The presence of methane-rich biogas influenced the ignition delay and heat release rate, thereby affecting the maximum cylinder pressure. Despite the increase in biogas substitution up to 85%, stable combustion was achieved under all operating conditions, demonstrating the feasibility of high biogas substitution in the dual-fuel compression ignition engine while maintaining satisfactory combustion performance.

7.2 Emission characteristics

7.2.1 Carbon Monoxide (CO) emission

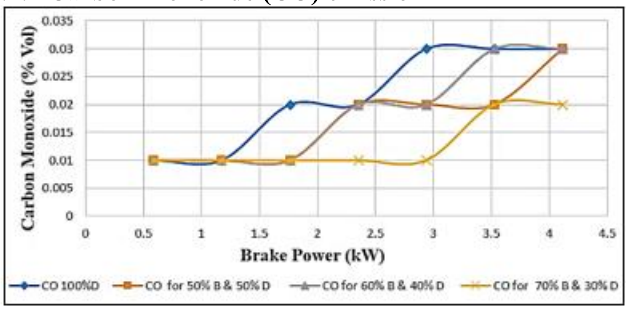

Figure 7.2.1: Variation of CO emission with Brake Power (BP) for different biogas substitution percentages

The variation of carbon monoxide (CO) emissions with brake power (BP) for different biogas substitution levels is shown in Figure 7.2.1. It was observed that CO emissions decreased with increasing replacement of diesel by biogas. This reduction is mainly attributed to the higher methane content of the biogas, which promotes more complete combustion. The compressed biogas (CBG) used in the present study contained approximately 86% methane, which possesses superior combustion characteristics compared to conventional diesel fuel. The enhanced combustion efficiency and the availability of a cleaner-burning gaseous fuel resulted in lower CO emissions throughout the operating range. Therefore, increasing the proportion of biogas in the dual-fuel operation effectively reduced incomplete combustion and improved the overall emission characteristics of the compression ignition engine.

7.2.2. Hydrocarbon (HC) emission

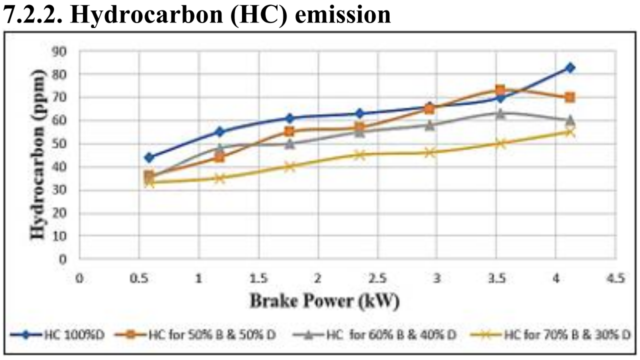

Figure 7.2.2: Variation of HC emission with Brake Power (BP) for different biogas substitution percentages

The variation of hydrocarbon (HC) emissions with brake power (BP) for different biogas substitution levels is shown in Figure 7.2.2. It was observed that HC emissions increased with increasing brake power. This increase is primarily attributed to incomplete combustion occurring at higher engine loads, where a larger quantity of fuel is supplied to meet the increased power demand. At higher brake power, the shorter combustion duration and locally rich fuel–air mixtures can lead to incomplete oxidation of the fuel, resulting in increased HC emissions. Although biogas is a cleaner-burning fuel, HC emissions tend to increase under high-load operating conditions due to combustion inefficiencies associated with dual-fuel operation.

7.2.3 Carbon Dioxide (CO_2) emission

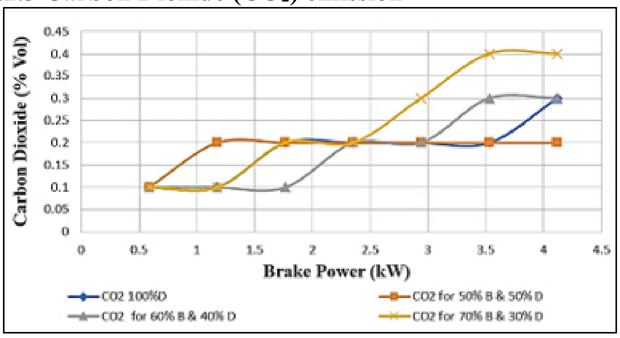

Figure 7.2.3: Variation of CO_2 emission with Brake Power (BP) for different biogas substitution percentages

The variation of carbon dioxide (CO_2) emissions with brake power (BP) for different biogas substitution levels is shown in Figure 7.2.3. It was observed that CO_2 emissions increased with increasing brake power for both diesel and biogas–diesel dual-fuel operation due to the higher quantity of fuel consumed at increased engine loads. At part-load conditions, the CO_2 emissions for biogas operation were found to be nearly the same as those of conventional diesel operation. However, at full-load conditions, the biogas-fueled engine produced comparatively higher CO_2 emissions because of the more complete combustion of methane-rich biogas. The compressed biogas used in the present study contained a high methane content, which promoted efficient combustion and greater conversion of carbon to CO_2 rather than carbon monoxide (CO). The increase in CO_2 emissions at higher biogas substitution levels indicates improved combustion efficiency and more complete oxidation of the fuel in the dual-fuel compression ignition engine.

7.2.4 Nitrogen Oxides (NO_X) emission

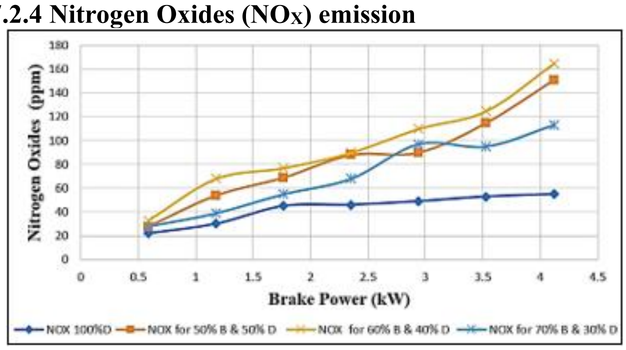

Figure 7.2.4: Variation of NO_x emission with Brake Power (BP) for different biogas substitution percentages

The variation of nitrogen oxides (NO_x) emissions with brake power (BP) for different biogas substitution levels is shown in Figure 7.2.4. It was observed that NO_x emissions increased with increasing brake power for both diesel and biogas–diesel dual-fuel operation. Compared to conventional diesel operation, the engine fueled with biogas exhibited relatively higher NO_x emissions. This increase is primarily attributed to the faster combustion characteristics of methane-rich biogas, which result in higher in-cylinder combustion temperatures and increased oxygen availability during combustion. These conditions promote the thermal formation of NO_x. As the engine load increased, the higher combustion temperature further enhanced NO_x formation. Therefore, although biogas offers advantages in reducing carbon monoxide and particulate emissions, it tends to produce comparatively higher NO_x emissions because of its rapid and more complete combustion characteristics.

8. Conclusion

The present study demonstrated that the production and utilization of Compressed Biogas (CBG) from Municipal Organic Solid Waste (MOSW) is a technically feasible and sustainable approach for renewable energy generation and dual-fuel Compression Ignition (CI) engine applications. The major conclusions are as follows:

- Mechanical pretreatment is an essential primary processing step for all types of organic waste. Although it partially disrupts the cellulose structure through size reduction, it is insufficient to achieve complete degradation of cellulose and hemicellulose.
- Hydrothermal pretreatment significantly enhances the biodegradability of the substrate by effectively breaking down cellulose into smaller fractions. It is an economical pretreatment technique and provides higher biogas yields compared to mechanical pretreatment.
- Chemical pretreatment, particularly alkali treatment, effectively degrades cellulose, hemicellulose, and lignin through chemical reactions. It is especially suitable for lignocellulosic yard waste, where strong lignin–cellulose–hemicellulose bonds cannot be sufficiently disrupted by mechanical or hydrothermal pretreatments alone.
- Biological pretreatment is the most economical method among the investigated techniques; however, its major limitation is the longer retention time required for biogas production.
- For large-scale biogas plants, mechanical and hydrothermal pretreatments are the most economical and practically viable options.
- The combination of pretreatment and co-digestion substantially improved biogas production. Co-digestion of 80% vegetable waste and 20% garden leaf waste produced approximately four times higher biogas yield than individual digestion of the respective substrates.
- Thermo-mechanical pretreatment was identified as a cost-effective method for improving biogas generation while maintaining process simplicity.
- CFD analysis proved to be an effective tool for optimizing the air–CBG mixing characteristics of the Venturi mixer. Among the investigated configurations, the four-pass, 45° gas inlet inclination model exhibited superior mixing performance and flow uniformity.

- The optimized four-pass, 45° Venturi mixer was successfully fabricated and integrated into the CI engine operating in dual-fuel mode for experimental validation.
- Engine performance and emission studies confirmed that the optimized dual-fuel system achieved up to 85% diesel substitution with CBG, demonstrating the potential of MOSW-derived CBG as an effective alternative fuel for compression ignition engines while significantly reducing diesel consumption.

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