A SIMULATION STUDY ON THE EFFECT OF BOOST PRESSURE ON COMBUSTION CHARATERISTICS OF BIOGAS ENGINE

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DOI: https://doi.org/10.57001/huih5804.2026.131

ABSTRACT

This study investigates the influence of boost pressure ratio on the combustion characteristics and overall performance of a biogas-fueled internal combustion engine. The simulation was carried out using the Diesel-RK software to evaluate variations in engine power, thermal efficiency, in-cylinder pressure, and pollutant emissions under different boost pressure ratios ranging from 1.0 to 2.0. The engine model was operated at full load with a constant excess air ratio (λ = 1.2) over a speed range of 1000 - 2400rpm. Simulation results show that increasing boost pressure significantly improves combustion efficiency and engine output due to enhanced air–fuel mixing and higher intake density. The indicated mean effective pressure and brake power increased by up to 90% at the highest boost ratio, while the specific energy consumption decreased by approximately 5%. However, NO_x emissions were found to increase with boost pressure as a result of higher combustion temperature. The study identifies an optimal boost pressure ratio between 1.6 and 1.8 for biogas engines, achieving the best balance between performance enhancement, fuel economy, and emission control.

Keywords: *Biogas engine, boost pressure, Diesel-RK simulation, combustion characteristics, NO_x emission.*

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Received: 02/3/2026
Revised: 08/5/2026
Accepted: 29/5/2026

1. INTRODUCTION

Biogas is a renewable energy source derived from the anaerobic breakdown of organic waste, primarily composed of CH_4 and CO_2. A significant benefit of biogas is its capacity for on-site production from agricultural waste, contributing to the reduction of greenhouse gas emissions and operational expenses. Nevertheless, biogas possesses several limitations, including a low calorific value, the risk for component corrosion due to contaminants such as H_2S and CO_2, and challenges in storage and delivery.

Since the early 1980s, numerous scientists have been engaged in investigating the feasibility of utilizing biogas for internal combustion engines. Hickson evaluated a diesel engine modified for spark ignition, revealing a power decrease of roughly 35% relative to diesel and 40% in comparison to gasoline [1]. Neyeloff and Cunkel established the ideal compression ratio to be between 8.5:1 and 15:1 for biogas applications [2]. Thring noted that biogas is only viable for utilization near its production site due to transportation constraints and advocated for its conversion into liquid gasoline. By 1985, Jenbacher Werke AG had successfully commercialized lean-burn biogas engines, incorporating enhancements to the cylinder head and intake system.

In Vietnam, Professor Bui Van Ga and his associates are trailblazers in this domain. Since 2007, the group has implemented various small and medium-scale biogas-powered generating systems. Experimental results on a 110cc motorcycle engine indicate that enriching the fuel by absorbing H_2S and CO_2 led to an approximate 30% increase in calorific value, while hydrocarbon and carbon monoxide emissions remained within permissible limits [3, 4]. The biogas-diesel dual-fuel engine system functions stably, utilizing approximately 1 m^3 of biogas per kWh of electricity, and does not generate soot [5-7]. In 2009, the research team forecasted a possible decrease of up to 4 million tons of CO_2 annually, or 6.5% of the nation's carbon emissions. Tran Cong Minh and colleagues utilized AVL Boost software to simulate a diesel engine modified for biogas consumption. The findings indicated a reduction in power alongside enhanced fuel efficiency; NO_x emissions were markedly diminished, however CO levels rose; the ideal spark advance angle was determined to be between 11 - 19° before top dead center [8].

In conjunction with investigations into alternative fuels, turbocharging technology is regarded as a viable method for diminishing fuel use and emissions. Since 2012, more than 50% of vehicles in Europe have transitioned to turbocharged engines, attributable to their potential for "downsizing" - decreasing engine displacement while preserving power and torque, in accordance with fuel efficiency trends and emissions taxation regulations. Silva and associates indicated that a 20% reduction in capacity, coupled with a 20 - 50% rise in charging pressure, results in a 6 - 14% drop in fuel consumption, and a reduction in emissions of HC, CO, and NO by 2 - 4%, 7 - 20%, and 8 - 23%, respectively [9]. Petitjean and his associates have validated the enhancement impact on both gasoline and diesel engines. During their research in Vietnam, author Nguyen Trung Kien and colleagues observed that the supercharged biogas engine enhanced power output by 2kW at 1500rpm, while NO_x emissions were reduced by as much as 85.8%, however CO emissions rose by 37.5% [10].

The biogas-air mixing ratio is determined by the combustion properties of biogas, characterized by a slow combustion rate owing to its elevated CO_2 concentration, and the inclination to function in a lean mixture mode to mitigate NO_x emissions. Research conducted by Bui Van Ga and associates [11] indicates that CFD simulations of the combustion chamber structure and ignition parameters for an engine converted from diesel to biogas reveal that the optimal operating region is within the λ range of 1.1 to 1.3, with λ = 1.2 deemed ideal for maximizing efficiency and minimizing emissions. Research by Jung and colleagues [12] demonstrated that forced ignition engines utilizing biogas attained steady power output and minimal emissions within the λ range of 1.2 - 1.3.

Consequently, both research avenues utilizing biogas as a substitute for conventional fuels and implementing boosting technology demonstrate considerable potential for enhancing efficiency, diminishing emissions, and fostering sustainable development within the transportation-energy sector. Further study is necessary to optimize the ignition system, air-fuel ratio, and boost management to address the constraints in power and stability of biogas engines. This study utilized Diesel-RK modeling software to assess the impact of the turbocharger ratio on the energy efficiency, technical and economic attributes, and emissions of a biogas-powered engine. The study team established the ideal compression ratio for turning a conventional diesel engine to operate only on biogas fuel.

2. METHODS

(1) Research Protocol

The study process comprised four primary steps, integrating theoretical analysis with numerical simulation with Diesel-RK software. The process encompasses:

(a) Collection of input data for the reference engine.

- The chosen engine is a six-cylinder diesel model S6D108, adapted to function on 100% biogas fuel. The compression ratio was decreased to 12:0, appropriate for biogas engines [2].

- The input parameters comprise cylinder displacement, piston diameter, stroke, compression ratio, cam profile, and intake-exhaust flow coefficients.

(b) Development of the simulation model with Diesel-RK

- A one-dimensional simulation model was developed, comprising the primary functional components: intake, compression, combustion, and exhaust.

The combustion process was characterized by a two-stage Vibe model. Boundary conditions: ambient pressure of 1bar and intake air temperature of 300K. The fuel utilized is biogas, including 60% CH_4 and 40% CO_2, with a lower heating value (LHV) of 16.31MJ/kg.

(c) Configuration of simulation scenarios

- The engine was presumed to function in full-load circumstances (100%) with a constant air excess ratio (λ) of 1.2 [11, 12].

- Six boost pressure ratios were examined, ranging from 1.0 (naturally aspirated) to 2.0, with increments of 0.2.

- The engine speed range was established from 1000 to 2400rpm, with increments of 200rpm.

- The retrieved metrics are engine power, specific energy consumption, in-cylinder pressure and temperature, heat release rate, and NO_x emissions.

(d) Analyze and appraise the simulation outcomes.

- Analyze the power output and energy consumption rate for each boost ratio.

- Examine the progression of heat release rate, pressure, and peak temperature within the cylinder.

- Assess the trend of NO_x emissions in relation to the thermal mechanism.

- Ascertain the ideal boost ratio for performance and emissions.

(2) Constructing the simulation model

The simulation model was constructed in the Diesel-RK software with a one-dimensional structure, as illustrated in Figure 1, comprising the following components:

- Cylinder block: replicates operational volume and the operations of compression and expansion.

- Assess the pressure drop, flow coefficient, and inlet/exhaust gas temperature of the intake and exhaust manifolds.

- Combustion model: employing the Wiebe dual-stage model, defined by the burning rate function and the combustion completion coefficient.

- Turbocharging model: replicates the turbocharging process with associated pressure levels, guaranteeing a linear increase in intake air volume corresponding to intake pressure.

- Fuel characteristics: biogas is characterized as a composition of CH_4 and CO_2 in a ratio of 60:40, possessing a density of $\rho = 1.15 kg/m^3$, and a lower heating value of LHV = 16.31MJ/kg.

- Boundary conditions: Ambient pressure of 1bar, ambient temperature of 300K - Engine speed range: 1000 to 2400rpm

- Air excess ratio $\lambda = 1.2$

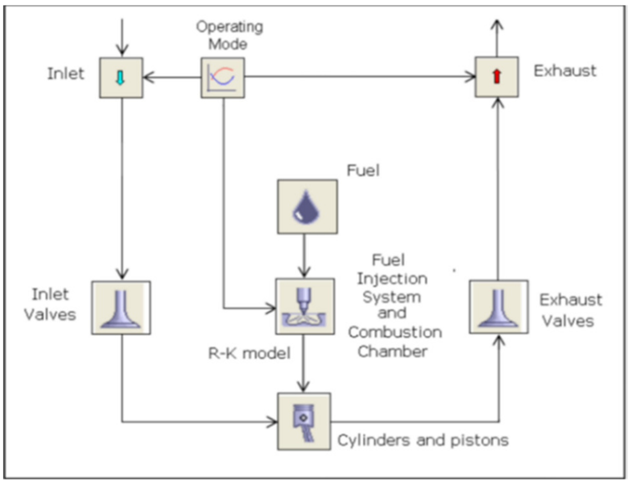

Figure 1. Schematic diagram of the hot air drying system utilizing exhaust heat for the feed inlet

The model was calibrated by juxtaposing simulation outcomes with the original engine's empirical data, guaranteeing a power deviation of under 5%. Subsequently, the model was expanded to various supercharging pressure values to assess performance, combustion processes, and pollutants. The fundamental specifications of the engine are presented in Table 1. Figure 2 illustrates the original diesel engine piston alongside the biogas engine piston following the modification of the compression ratio through the truncation of the piston crown.

Table 1. Fundamental parameters of the research engine

Parameter	Original Diesel Engine	Converted Biogas Engine
Fuel	Diesel	Biogas (65% CH_4)
Diameter x Stroke	108mm	130mm
ratio	17.5: 1	12.0:1
Maximum power	125kVA at 1500rpm	85kVA at 1500rpm
Specific power	10,41kVA/liter	7,08kVA/liter

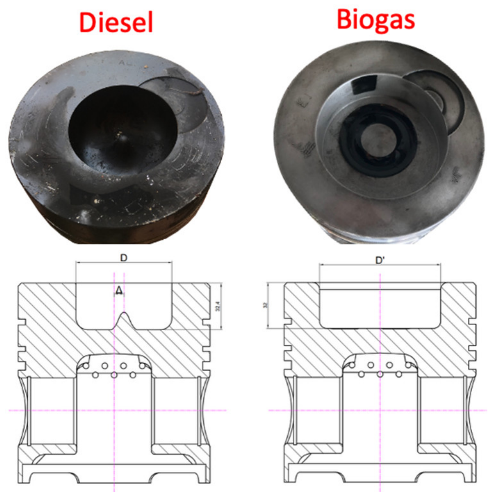

Figure 2. Shape of diesel and biogas engine piston crown after conversion

The combustion chamber of the original diesel engine is asymmetrical, engineered to generate a vortex that facilitates the mixing of directly injected fuel with air, resulting in a non-homogeneous mixture that auto-ignites at elevated compression ratios. During the transition to biogas utilization, the combustion chamber is modified and reconfigured to a central geometry, while the combustion chamber volume (V_c) is augmented to lower the compression ratio, aligning with the superior anti-knock characteristics of biogas. This axially symmetric configuration facilitates the creation of a uniform gas-biogas mixture prior to ignition, transitioning the combustion process from the diffusion-dominated characteristics typical of diesel engines to homogeneous forced combustion akin to that in spark-ignition engines, thus enhancing combustion stability and diminishing emissions.

3. RESULTS AND DISCUSSION

(1) Economic and Technical Characteristics of the Engine

Figure 3 illustrates the variation in biogas engine power relative to speed across several boost ratios. The findings indicate that power escalates approximately linearly with speed, and an elevated boost ratio correlates with increased power. At a velocity of 1000rpm, the power escalated from 28.9kW at a ratio of 1.0 to 53.1kW at a ratio of 2.0; conversely, at 2380rpm, the power surged from 63.2kW to 121.0kW, indicating an increase exceeding 90%.

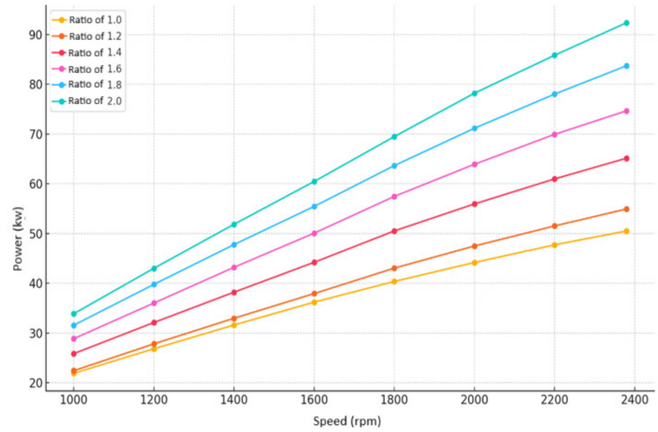

Figure 3. Compare effective power across varying boost ratios

This increase is chiefly because to the heightened pressure, which elevates the amount of air entering the cylinder, hence enhancing charging efficiency and permitting the combustion of additional fuel during the same cycle. Moreover, supercharging augments the effectiveness of the combustion process, so enhancing both the engine's technical performance and fuel efficiency.

Energy consumption is utilized to assess the economic performance of the engine, determined utilizing the subsequent formula:

$$E_{dnl} = G_{nltt}.Q_t$$

where

E_{dnl}: Energy consumed

G_{nltt}: Fuel consumption mass

Q_t: Lower heating value of the fuel, 42.5MJ/kg.

Figure 4 illustrates the variation in energy consumption (MJ/kWh) of the biogas engine relative to rotational speed and various boost ratio levels. The results indicate that in the low-speed range (below 1600rpm), the energy consumption rate exhibits a small increase with rising pressure ratios. The explanation is because at low velocities, the intake air volume rises, while the combustion rate and fuel-air mixing capability have not markedly enhanced, leading to constrained combustion efficiency. As the engine speed surpasses 1600rpm, the efficacy of the turbocharger becomes increasingly evident. Energy usage diminishes as the compression ratio rises, particularly between 1.4 and 1.8, signifying a more efficient fuel-air mixture intake and combustion process. At rotational speeds of 2000 - 2400rpm, energy consumption at a boost ratio of 1.8 is approximately 4 - 5% lower than in the absence of boosting, signifying a notable enhancement in the engine's overall thermal efficiency.

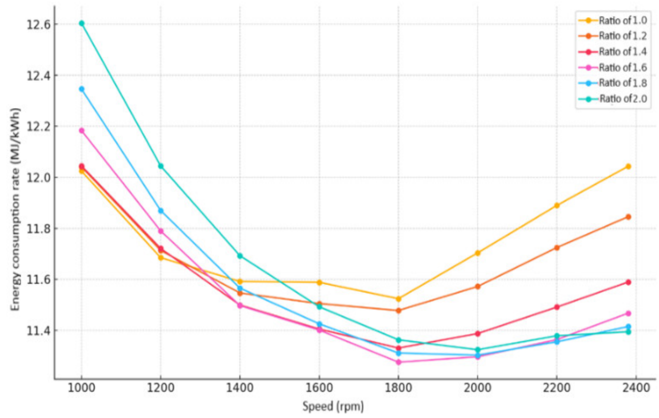

Figure 4. Compare energy consumption rates at different boost ratios

Nevertheless, as the ratio escalates to 2.0, the energy consumption rate tends to either plateau or experience a modest rise, attributable to heightened mechanical and thermodynamic losses at elevated input pressures. The findings indicate that the ideal boost ratio for the biogas engine lies between 1.6 and 1.8, enhancing energy conversion efficiency while ensuring acceptable fuel consumption over the engine's operational speed spectrum. Increasing the intake air compression ratio is an efficient method to enhance the power and overall efficiency of biogas engines.

(2) Attributes of the Combustion Process

Figure 5 illustrates the heat release rate of the biogas engine at 1500rpm across various boost ratio values. Simulation results indicate that with an increase in the boost ratio, the maximum heat release rate (dQ/dα) dramatically escalates, and the heat release peak shifts nearer to the top dead center. This demonstrates that the combustion process is expedited and more intense, as the augmented air intake facilitates a more complete and stable combustion mixture. This enhancement in combustion velocity concurrently improves combustion efficiency while also resulting in elevated peak pressure within the combustion chamber. The simulation data presented in Figures 4 and 5 demonstrate that intake pressure significantly influences the maximum temperature and pressure within the cylinder. As the compression ratio escalates from 1.0 to 2.0, the peak temperature within the cylinder experiences a modest rise from roughly 1767K to 1800K, however the peak pressure undergoes a substantial increase from approximately 63.7bar to above 120bar. The concurrent rise in maximum pressure and temperature signifies enhanced combustion efficiency because to elevated pressure, although it also cautions against an increased likelihood of overheating and mechanical strain on the components of the combustion chamber. Consequently, in the design and optimization of turbocharged biogas engines, it is imperative to calibrate the spark advance angle and restrict the maximum pressure suitably to guarantee longevity and operational safety.

The rise in maximum cylinder pressure and temperature (Figures 6 and 7) signifies a more rapid and thorough combustion process, which directly influences the indicated mean effective pressure (IMEP) and indicated thermal efficiency (η_i) of the engine. Increasing the compression ratio from 1.0 to 1.6 - 1.8 results in a substantial rise in the stated mean effective pressure, indicating a proportional enhancement in work output energy within the cylinder. This aligns with the findings of enhanced capacity and lower energy use demonstrated in prior figures.

Nevertheless, as the boost ratio persists above 1.8, despite the ongoing rise in maximum pressure, the enhancement in IMEP and η_i appears to plateau, partially attributable to energy dissipated as heat via the cylinder walls and mechanical losses. Moreover, excessively elevated peak pressure might augment mechanical stresses on the piston, connecting rod, and bearings, so impacting the engine's overall durability.

Consequently, the ideal boost ratio for a biogas engine must provide elevated combustion efficiency and output while simultaneously restricting the maximum pressure to a safe threshold. The simulation outcomes in this research reveal that a boost ratio of roughly 1.6 to 1.8 is optimal, enhancing stated performance while maintaining maximum pressure and temperature within permissible design parameters.

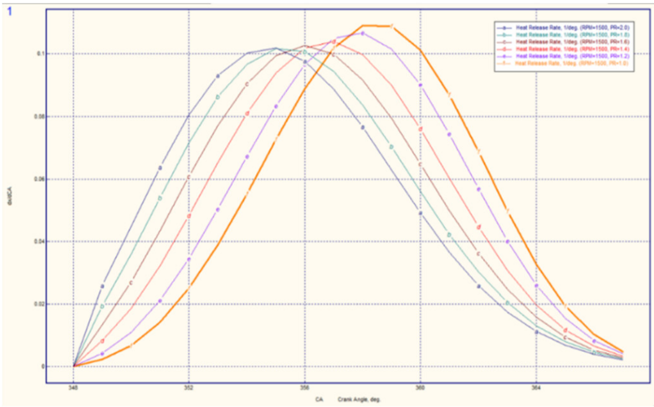

Figure 5. Heat release rate at 1500rpm for different boost ratios

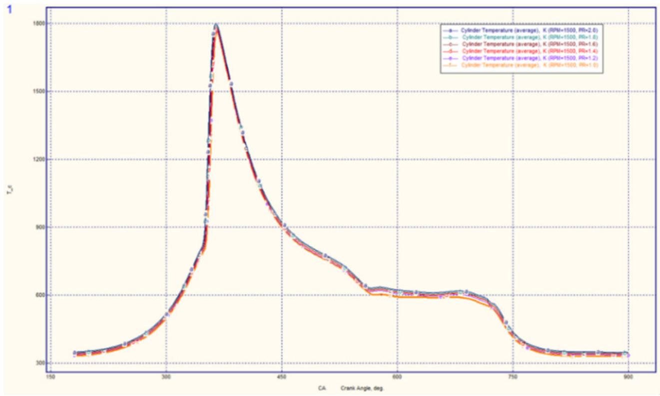

Figure 6. Temperature variation in the cylinder at 1500rpm for different boost ratios

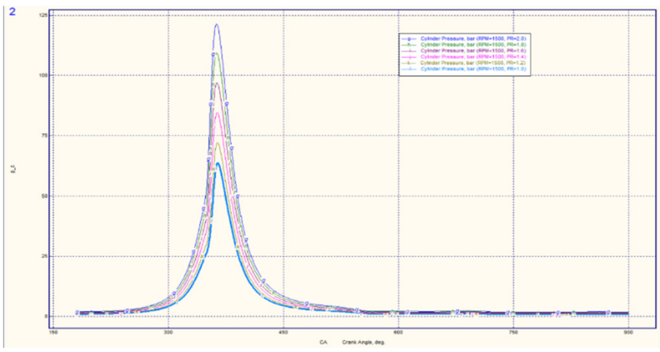

Figure 7. Pressure development in the cylinder at 1500rpm for different boost ratios

(3) Composition of Hazardous Emissions

Figure 8 illustrates the trends of NO_x emissions (g/kWh) from the biogas engine in relation to engine speed and varying degrees of turbocharging. Simulation results indicate a substantial rise in NOx emissions with an increase in boost ratio, although the impact of engine speed is rather minimal. At low speed (1000rpm), NO_x emissions rose from 0.0077g/kWh at a boost ratio of 1.0 to 0.022g/kWh at 2.0; conversely, at high speed (2380rpm), this figure escalated to around 0.0296g/kWh, reflecting an increase of nearly 3.8 times. This increase arises from the elevated boost ratio, which enhances the intake air volume, resulting in increased combustion pressure and temperature, hence promoting NO_x generation through the thermal process, the principal mechanism in spark-ignition engines [13].

While supercharging promotes combustion efficiency and engine output, it concurrently increases nitrogen oxidation at elevated temperatures (> 1800K), resulting in a substantial rise in NO_x with the supercharging ratio. Nonetheless, as the primary constituent of biogas is CH_4, which exhibits a reduced combustion velocity and calorific value compared to diesel, the maximum combustion temperature remains considerably lower, leading to overall NO_x emissions that are 50 - 70% less than those of a diesel engine operating under identical load conditions [14]. The modeling results align with prior experimental investigations on dual-fuel biogas-diesel engines, indicating that NOx emissions rise linearly with intake pressure and increasing biogas ratios [15]. In the high-speed domain, despite the increase in power, the reduced combustion duration constrains the period of elevated temperatures, resulting in a more gradual rise in NOx compared to the medium-speed domain.

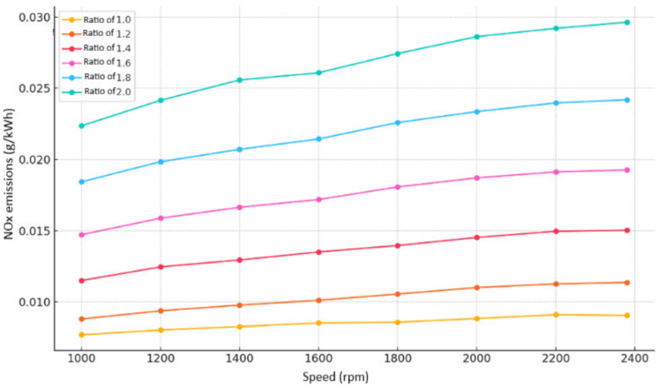

Figure 8. Compare the composition of toxic emissions from the engine

Turbocharged biogas engines have markedly lower NOx emissions than diesel engines, while simultaneously attaining superior energy efficiency compared to naturally aspirated engines. This validates the feasibility of implementing turbocharging technology in engines that utilize just biogas fuel, enhancing energy efficiency while preserving environmentally sustainable emission standards.

4. CONCLUSIONS

The research modeled the impact of the boost ratio on the combustion process, energy efficiency, and emissions of a biogas engine utilizing Diesel-RK software. The simulation model is based on a modified S6D108 diesel engine optimized for exclusive operation on biogas fuel, featuring turbocharger ratios between 1.0 and 2.0 spanning a speed range of 1000 - 2400rpm, with a constant air-fuel ratio of $\lambda = 1.2$. The simulation results indicate that engine power markedly rises with the boost ratio, particularly within the mid to high-speed range. At a velocity of 2380rpm, power surged by over 90% as the boost ratio escalated from 1.0 to 2.0, attributable to an augmented air intake and enhanced combustion efficiency.

Energy consumption (MJ/kWh) diminishes with a rising pressure ratio, attaining a minimum value between 1.6 and 1.8, which aligns with the peak thermal efficiency. At a boost ratio of 2.0, the enhancement in efficiency is negligible owing to heightened mechanical and thermodynamic losses.

The peak heat release rate and peak combustion pressure markedly escalate with the compression ratio. The peak temperature in the cylinder escalated from roughly 1767K to 1800K, while the peak pressure surged from 63.7bar to beyond 120bar, indicating a more rapid and intensified combustion process.

NO_x emissions rise roughly linearly with the compression ratio due to elevated combustion temperatures, although remain considerably lower than those in diesel engines, attributable to the clean combustion properties and reduced flame propagation velocity of biogas.

The analysis indicates that the best boost ratio for the biogas engine lies between 1.6 and 1.8, where the engine attains an effective equilibrium of power, efficiency, and emissions. Implementing turbocharging in biogas engines is a viable method to enhance energy efficiency and decrease fuel consumption, while maintaining environmentally sustainable emission levels, thereby supporting sustainable development objectives and mitigating greenhouse gas emissions in the energy and transportation sectors.

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