

## Article

# Effect of Stratified Charge Combustion Chamber Design on Natural Gas Engine Performance

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**Abstract:** This study investigates the performance and combustion behavior of a spark ignition engine retrofitted to operate on compressed natural gas (CNG), with a focus on a newly developed stratified charge pre-chamber design. The engine was modified to include an auxiliary intake valve that enables partial enrichment of the pre-chamber mixture without the need for a dedicated fuel injector. This hybrid approach combines the mechanical simplicity of passive systems with the enhanced combustion control of active pre-chambers. Both experimental tests and computational fluid dynamics (CFD) analyses were carried out under partial load conditions (8 Nm) and engine speeds ranging from 900 to 1700 rpm. The results demonstrate improvements in indicated mean effective pressure (IMEP), combustion stability, and flame propagation speed—particularly at lower engine speeds where stratified combustion effects are more pronounced. However, increasing engine speed resulted in reduced volumetric efficiency and elevated exhaust temperatures, indicating potential for further optimization via turbocharging or advanced scavenging techniques. Overall, the findings validate the effectiveness of the proposed design in enhancing thermal efficiency and ignition stability in CNG-fueled engines, especially under urban driving conditions.

**Keywords:** pre-chamber design; natural gas combustion; engine performance analysis



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## 1. Introduction

The growing demand for energy, rising fuel prices, and increasingly strict environmental regulations have intensified the pursuit for cleaner and more efficient alternative fuels. Among these, natural gas stands out due to its high octane number, relatively low carbon content, and proven ability to reduce emissions compared to conventional gasoline or diesel fuels. As such, compressed natural gas (CNG) has gained traction, particularly in the transportation sector.

Numerous studies have examined the performance of gasoline engines retrofitted to operate on CNG. Kato et al. [1] presented the development of a 2.2 L gasoline engine adapted for natural gas operation, highlighting reductions in power and torque, which were partially recovered by adjusting valve timing and compression ratio. Similarly, Aslam et al. [2] observed a 16% reduction in brake mean effective pressure (BMEP) and an 18% improvement in fuel consumption when switching from gasoline to CNG, emphasizing the trade-off between engine output and fuel economy. Varma and Mittal [3] analyzed bowl-in-piston versus flat-piston geometries in a CI engine retrofitted for CNG SI operation, showing a 2.58% higher gross indicated efficiency for the bowl-in-piston geometry along with significantly different NO<sub>x</sub> and CO emissions profiles.

Pre-chamber ignition systems have been investigated extensively as a means to improve combustion stability, especially under lean burn conditions prevalent in natural gas engines. Shapiro et al. [4] and Adams [5] explored how pre-chambers generate high-energy flame jets that facilitate faster combustion. Alvarez et al. [6] provided a comprehensive review of pre-chamber ignition strategies, outlining their potential for improved efficiency and emissions reduction. Khan et al. [7] examined swirl-induced flow in the main chamber and its role in enhancing methane combustion dynamics with a passive pre-chamber setup, identifying its impact on flame propagation, turbulence, and combustion stability.

The effectiveness of fuel–air stratification and in-cylinder turbulence in enhancing lean combustion has been well documented. Garg and Ravikrishna [8] modeled stratified charge combustion in a CNG engine, showing how careful control of air–fuel layering affects flame propagation. Studies by Liu and Dumitrescu [9] and Korb et al. [10] emphasized the role of tumble and swirl motion in promoting homogeneous combustion and stabilizing ignition. Krieger Filho et al. [11] explored combustion chamber design with various piston geometries, finding that re-entrant bowls enhanced turbulence kinetic energy and combustion stability for methane-based fuels.

Design parameters such as nozzle angle, throat diameter, and pre-chamber volume significantly impact combustion behavior. Benajes et al. [12] demonstrated how geometric variation alters ignition delay and pressure rise. Li et al. [13] introduced a plug-and-play passive pre-chamber design and examined the effects of nozzle geometry. Zhao et al. [14] confirmed that structural differences in pre-chambers affect combustion efficiency, especially in turbocharged engines. Jialong Li et al. [15] further examined the influence of orifice taper and volume ratio on combustion, showing that straight orifices provided superior flame propagation compared to tapered variants.

CNG-fueled engines operating under lean conditions benefit substantially from pre-chamber strategies. Yousefi and Madjid [16] and Roso et al. [17] evaluated thermal efficiency and emission reductions in dual-fuel and stratified applications. Lu et al. [18] analyzed passive pre-chamber performance under low-load lean burn operation, while Krajnović et al. [19] compared pre-chamber-assisted homogeneous charge compression ignition (HCCI) with conventional SI methods. Also, Song et al. [20] compared CNG direct injection (DI) and port fuel injection (PFI) systems, showing that DI significantly improves volumetric efficiency and combustion characteristics, especially under lean burn conditions. This supports the importance of precise fuel delivery strategies in enhancing performance and reducing emissions in natural gas engines.

Custom and experimental applications of pre-chamber systems have also gained attention. Sementa et al. [21] evaluated passive and active modes in small spark ignition engines, and Roubaud and Favrat [22] studied their impact in biogas-fueled cogeneration systems. The findings, while focused on biogas, offer valuable insights for natural gas applications due to the fuels' similar combustion characteristics.

While extensive work has been conducted on pre-chamber ignition systems, most studies have focused on full-load, steady-state, or high-speed engine conditions. However, there remains a significant gap in understanding the influence of pre-chamber design on engine performance and combustion behavior under partial load and low-to-medium speed conditions, which are particularly relevant to urban driving. The pre-chamber configuration developed in this study addresses this gap by incorporating a stratified charge combustion strategy in a single-cylinder natural gas engine, tested across various engine speeds and under partial load, using both experimental data and CFD-supported analysis.

## 2. Materials and Methods

### 2.1. Pre-Chamber Design and Test System

The experiments were carried out on a four-stroke, single-cylinder, spark ignition engine which was fueled by natural gas. The engine employed for the tests conducted during this study has a cylinder head which is designed and retrofitted as seen Figure 1. To investigate the performance, combustion, and fuel consumption characteristics of different structured pre-chamber designs, the cylinder head of the engine was redesigned and the specific parameters of the engine are shown in Table 1.

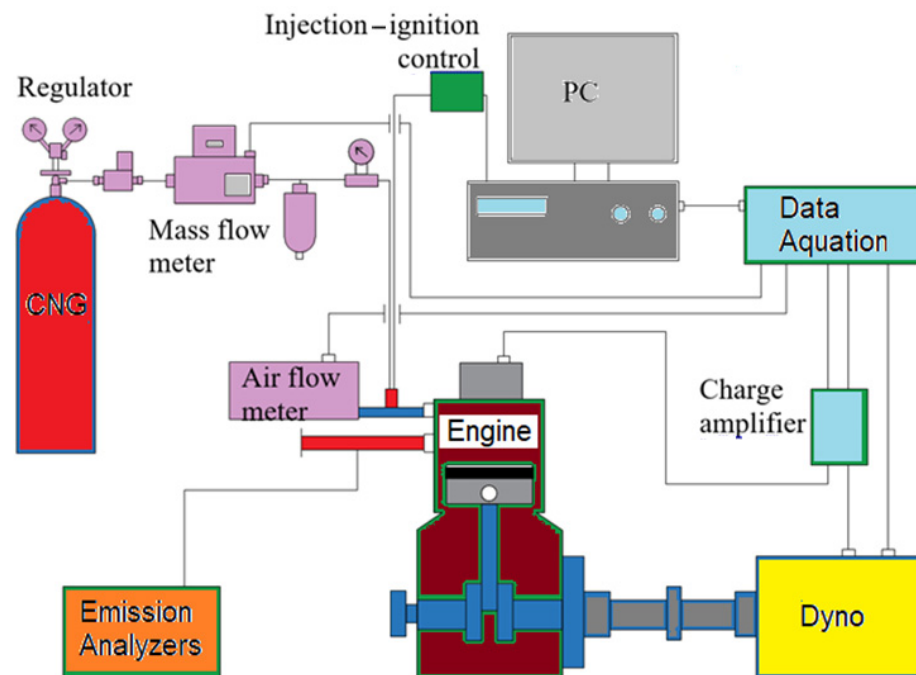


Figure 1. The engine performance test system.

Table 1. Engine specifications.

| ENGINE                  | KIRLOSKAR-TV1       |
|-------------------------|---------------------|
| Number of cylinders     | Single              |
| Bore                    | 87.5 mm             |
| Stroke                  | 110 mm              |
| Displaced volume        | 661 cm <sup>3</sup> |
| Compression ratio       | 10:1                |
| Engine speed            | 2000 rpm (max.)     |
| Connecting rod length   | 234 mm              |
| Inlet valve open        | 4.5 °CA             |
| Inlet valve closed      | 35.5 °CA            |
| Exhaust valve open      | 35.5 °CA            |
| Exhaust valve closed    | 4.5 °CA             |
| Injection type          | Port ignition       |
| Ignition type           | Electronical        |
| Inlet valve clearance   | 0.18 mm             |
| Exhaust valve clearance | 0.20 mm             |

In-cylinder pressure data were recorded using a PCB Piezotronics 111A22 piezoelectric pressure transducer, manufactured by PCB Piezotronics, Inc. (Depew, NY, USA). To determine effective engine performance parameters, an electric dynamometer was used, namely the Sensortronics 60001 model, S type, with a 0–50 kg capacity universal load cell.

A thermal mass flow meter (New-Flow brand TSF series) with a 0–500 L/min measuring range was used to measure fuel consumption. A crankshaft encoder was employed to capture trigger and crank position information. The experimental engine is equipped with a variable compression ratio feature, allowing the compression ratio to be adjusted according to changes in the combustion chamber design. This capability enables the compression ratio to be modified as desired, ensuring that all experiments can be conducted under consistent and controlled conditions.

In the study, a second intake valve was added to the engine design to allow a rich mixture to be directed into the pre-chamber, as shown in Figure 2. The original engine had only two valves: one for intake and one for exhaust. The new design introduced an auxiliary intake valve, which enables the rich mixture to be delivered specifically to the pre-chamber.

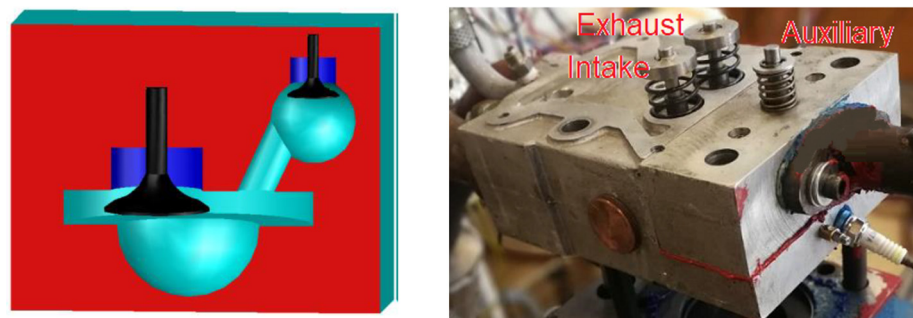


**Figure 2.** Auxiliary valve system for intake.

The pre-chamber configuration developed in this study exhibits several key design distinctions compared to those commonly reported in the literature. Most notably, the proposed system incorporates an auxiliary intake strategy to partially enrich the pre-chamber mixture without the use of a dedicated fuel injector, thus maintaining a simplified mechanical layout akin to passive systems while achieving performance closer to active configurations. This hybrid approach enables controlled stratification without the complexity of high-speed direct injection systems typically seen in active pre-chamber designs. So, it offers an optimized solution that combines the simplicity of passive designs with the high-performance potential of stratified active systems while ensuring ease of integration into existing engines.

The natural gas has better performance in lean mixtures compared to gasoline fuel in terms of thermal efficiency and fuel consumption. However, classical gasoline engines generally work on the principle of homogeneous charge that limits the lean mixing ratio. The stratified charge method can be applied to improve and decrease the amount of fuel in the mixture.

The stroke volume of the original engine is 661 cm<sup>3</sup>, and the clearance volume is 58 cm<sup>3</sup>. Based on the reference literature, the ratio of the pre-chamber volume to the total clearance volume is typically in the range of 25–30% [23,24]. In this study, the pre-chamber volume was set to approximately 25% of the clearance volume.

The pre-chamber has a spherical shape with a volume of 14.5 cm<sup>3</sup> and features an auxiliary intake valve and a throat. To ensure smooth combustion flow between the pre-chamber and the piston bowl, the throat's angle and diameter were designed to be 25 degrees and 10 mm, respectively, as depicted in Figure 3.

The performance calculations for analyzing the experimental results were performed by referring to [24,25]. The natural gas used in the experiments contained a methane ratio of over 93%, as confirmed by the production certificate.

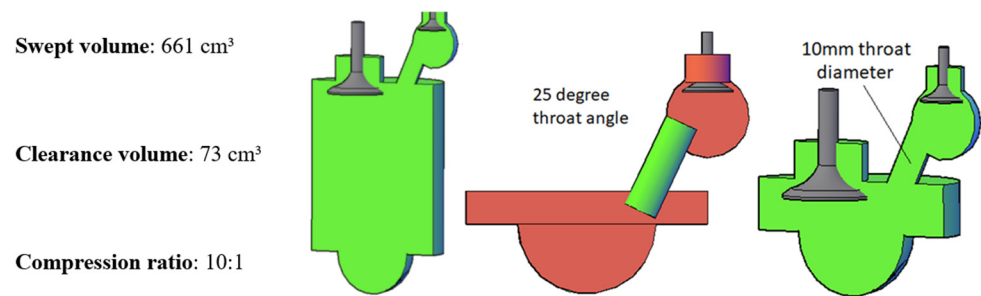


**Figure 3.** Pre-chamber combustion chamber design.

## 2.2. Flame Model for Partial Premixed Combustion

Computational fluid dynamics (CFD) was used to simulate the combustion process of a stratified charge spark ignition engine, along with the flame propagation resulting from all of the physical and chemical phenomena involved. In the CFD simulations, a detailed three-dimensional model of the combustion chamber including the pre-chamber, main chamber, valves, and piston geometry was developed to ensure physical accuracy. The computational domain was discretized using a structured hexahedral mesh, with localized refinement in critical regions such as the spark plug, throat nozzle, and pre-chamber walls. Boundary conditions were defined based on motoring (unfired) cycle data and supporting experimental measurements. The inlet and outlet ports were modeled as pressure boundaries, with pressure and temperature values corresponding to the intake and exhaust phases. Wall boundary conditions were set to constant temperatures derived from experimental thermocouple data, and wall motion was captured using dynamic mesh techniques to account for piston and valve movement. A no-slip condition was applied to all solid surfaces.

The combustion model utilized was the Partially Premixed Combustion (PPC) model with G-equation-based flame propagation, suitable for turbulent jet-ignited systems. Turbulence effects were modeled using the RNG  $k$ - $\epsilon$  turbulence model, known for its improved accuracy in recirculating and high-shear flows. Chemical kinetics were simplified to a single-step methane–air reaction for computational efficiency, in line with the approach used in the current study.

Since the chemical time scale is much smaller than the turbulence time scale, there exists a thin reaction layer where chemical reactions are predominant. Reference [26] shows that this region would consist of various counter-diffusion flames, known as small flamelets. Peters also introduced a new coordinate system, called the Z-coordinate system, by transforming the Cartesian coordinate system. This system allows for the formulation of conservation equations for enthalpy, which depend on temperature and species, in a one-dimensional form based solely on the mixture ratio. The Lewis number ( $Le$ ) is a dimensionless number:

$$Le = \frac{\alpha}{D} \quad (1)$$

Here,  $\alpha$  and  $D$  denote thermal diffusion and mass diffusion, respectively. Assuming  $Le = 1$ , the flame equations can be expressed as follows. Molecular species:

$$\rho \frac{\partial Y_i}{\partial t} - \rho \frac{\chi}{2} \frac{\partial^2 Y_i}{\partial Z^2} - m_i = 0 \quad (2)$$

Temperature:

$$\rho \frac{\partial T}{\partial t} - \rho \frac{\chi}{2} \frac{\partial^2 T}{\partial Z^2} + \sum_i^N \frac{h_i}{c_p} m_i - \frac{1}{c_p} \left( \frac{\partial P}{\partial t} + q_R \right) = 0 \quad (3)$$

Here,  $\chi$  is the distribution rate and is defined below.

$$\chi = 2D \left( \frac{\partial Z}{\partial \chi_\alpha} \right)^2 \quad (4)$$

Here,  $q_R$  represents the radiation losses and  $Y_i$  represents the mass fraction of species. The production rate  $m_i$  of the chemical reaction can be calculated as follows:

$$m_i = W_i \sum_k X_i \nu_{ik} w_k \quad (5)$$

Here,  $W_i$  is the molecular weight of the species,  $\nu_{ik}$  is the matrix of stoichiometric coefficients, and  $w_{ik}$  is the reaction rate of reaction  $k$ . The flame model solves the composition of the species depending on the mixing ratio with the CFD code. The boundary conditions are air temperature, fuel temperature, and stoichiometric distribution. The laminar  $G$ -Equation according to the kinematic balance at the flame front is as follows [27]:

$$\frac{\partial G}{\partial t} + v_{\alpha,u} \frac{\partial G}{\partial \chi_\alpha} = s_L(Z) |\nabla G| \quad (6)$$

The  $G$ -Equation for turbulent combustion is derived by separating the scalar  $G$  and flow velocity  $v$  into two components, mean and unsteady:

$$\begin{aligned} G(x, t) &= \bar{G}(x, t) + G'(x, t) \\ v(x, t) &= \bar{v}(x, t) + v'(x, t) \end{aligned} \quad (7)$$

If Equation (7) is written in place of Equation (6) and the cumulative average is applied, the  $G$ -Equality will be as follows:

$$\frac{\partial \bar{G}}{\partial t} + \bar{v}_{\alpha,u} \frac{\partial}{\partial \chi_\alpha} (\overline{v'_{\alpha,u} G'}) = \bar{s}_L \sigma - \overline{D_L \kappa |\nabla G|} + L n_\alpha \overline{\frac{\partial v_\alpha}{\partial \chi_\beta} n_\beta |\nabla G|} \quad (8)$$

Here,  $\sigma$  is the absolute value of the scalar gradient,  $\kappa$  is the flame strain,  $L$  is the boundary length, and  $D_L$  is the boundary diffusivity. A solution has been obtained using the equations explained above while developing the partial premixed combustion model.

### 3. Results and Discussion

#### 3.1. CFD Analysis of Pre-Chamber Design

In the CFD analyses, methane properties were used as fuel, and simulation values were recorded for a  $\Phi = 0.95$  equivalence ratio. In Figures 4 and 5, the flame propagation combustion simulation and temperature change zones in the cylinder depending on the crankshaft angle are shown according to average contours, respectively.

Figure 4 illustrates the combustion progress within the cylinder, showing average combustion contours at various crank angles. After spark ignition, the pre-ignition phase ( $-12$  to  $-4$  °CA) occurs. At  $-12$  °CA, the combustion chamber shows minimal combustion activity, with low values observed across the domain. The mixture in this region is rich. As the crank angle advances ( $-8$  and  $-4$  °CA), the initial combustion region near the spark plug ignites, and the flame front begins propagating outward. The second phase for the pre-combustion chamber is peak combustion ( $0$  to  $12$  °CA). Flame propagation within the cylinder indicates that by  $-4$  °CA, combustion is completed within the throat and begins transitioning into the main combustion chamber. At  $0$  °CA, the pressure effect of the newly developing flame in the main chamber on the piston surface is expected to be relatively weak. However, by  $+12$  °CA, the pressure exerted on the piston is anticipated

to increase significantly due to the acceleration of the combustion process. This suggests that a stratified combustion engine design in which the maximum pressure effect occurs around or after +12 °CA can be achieved through the influence of the throat geometry. It is further shown that the throat design contributes not only to the effective combustion of a leaner mixture in the main chamber but also to achieving higher combustion speeds in a shorter time.

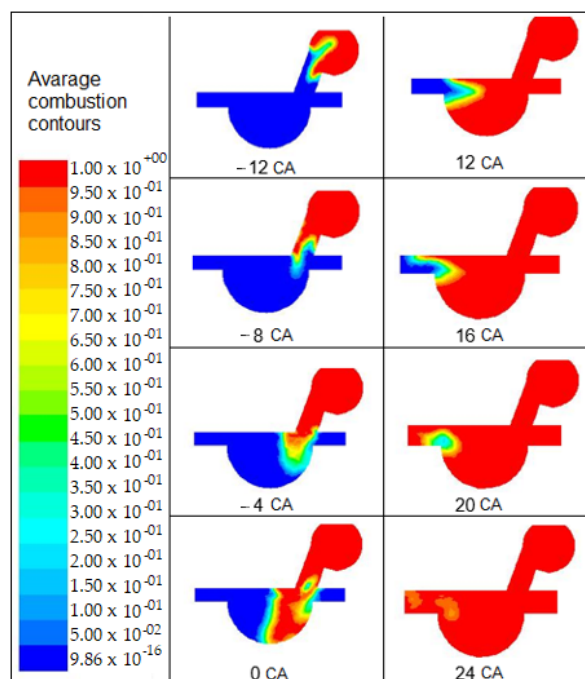


**Figure 4.** Simulation of combustion and flame development in combustion chamber depending on crank angle.

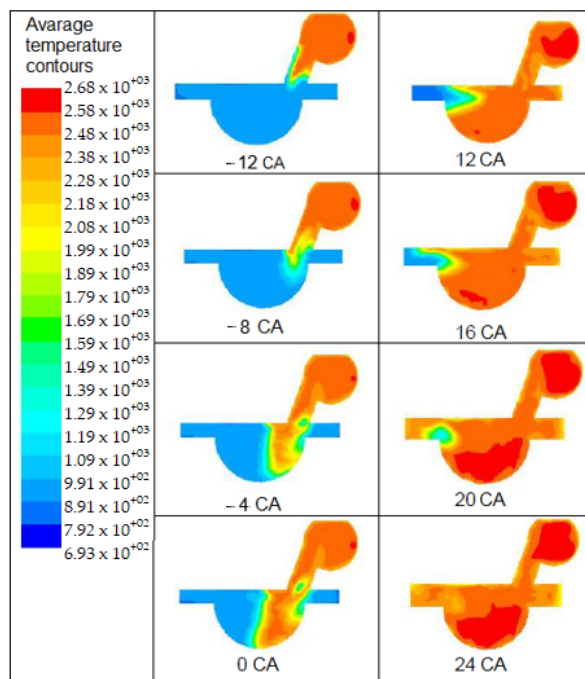


**Figure 5.** Simulation of average temperature change in combustion chamber depending on crank angle.

At 0 °CA (TDC), the combustion zone is more developed, with maximum combustion activity near the spark plug and the center of the chamber which includes the bowl. By +12 °CA, the combustion process covers nearly the entire cylinder, with higher combustion intensities shown in red. The third phase is the post-combustion phase (16 to 24 °CA). At +16 °CA, the combustion activity becomes uniform, indicating that most of the fuel has burned. At +20 and 24 °CA, the cylinder appears fully combusted, with minimal new activity. The red areas represent the highest combustion intensity (close to complete burning), while the blue areas indicate minimal combustion. The gradient demonstrates the progression of the flame front and combustion efficiency over time. This visualization of the pre-chamber design is crucial for understanding flame propagation, optimizing ignition timing, and improving engine efficiency.

Figure 5 illustrates the average temperature contours in the combustion chamber at various crank angles. The stratification of the mixture, rich in the pre-chamber and lean in the main chamber, creates a gradient in reactivity and temperature. When the flame jets from the pre-chamber enter the lean main chamber, they interact with a more resistive environment, but the high-temperature core of the jet and momentum maintain flame propagation. Moreover, stratification reduces the probability of flame quenching by preventing the flame from having to grow entirely from a weak kernel in a low-reactivity zone.

Similar to flame propagation, at −12 °CA, the temperature is relatively low, indicating the compression phase before ignition. As the crank angle approaches the TDC, the temperature begins to increase near the spark plug location, indicating the start of combustion. At 0 °CA (TDC), the temperature increases significantly as the combustion process is fully underway. At +12 °CA, the high-temperature region expands across the chamber, showing the flame front propagation. By +16 °CA, the combustion chamber exhibits a uniform temperature distribution as the burned gases expand. At +20 °CA and beyond, the temperatures start to stabilize, indicating the expansion stroke where heat is converted to mechanical energy. The color gradient indicates that the highest temperature zones are near the spark plug and during the early combustion phase. The maximum temperature reaches up to 2600 K. This contour visualization helps analyze flame propagation, thermal efficiency, and possible heat losses during the combustion process for combustion chamber designed engine.

### 3.2. Experimental Engine Performance Analysis

The experimental studies were conducted after the engine reached its operating temperature. The engine used in the experiments was water-cooled, and a circulation pump was driven using a 500 L water reservoir. In all of the experiments, natural gas was injected at a constant pressure of 3 bar. Additionally, the injection timing was designed to be 5° before intake valve opening (IVO). The fuel amount was controlled by extending the injector's open time. At a fuel pressure of 3 bar, the desired fuel amount could be injected into the cylinder within a time range of 6 ms and 20 ms. Experimental measurements were made when the engine speed was stabilized, and each measurement was repeated three times for verification. The natural gas and air mixture was created in the intake manifold and then introduced into the cylinder. In order to observe the effect of a stratified charged pre-chamber under partial load (8 Nm), the performance measurements were performed at different speeds. Figures 6 and 7 show the in-cylinder pressure versus the indicated mean effective pressure (IMEP) at 1500 RPM and the maximum pressure changes, respectively.

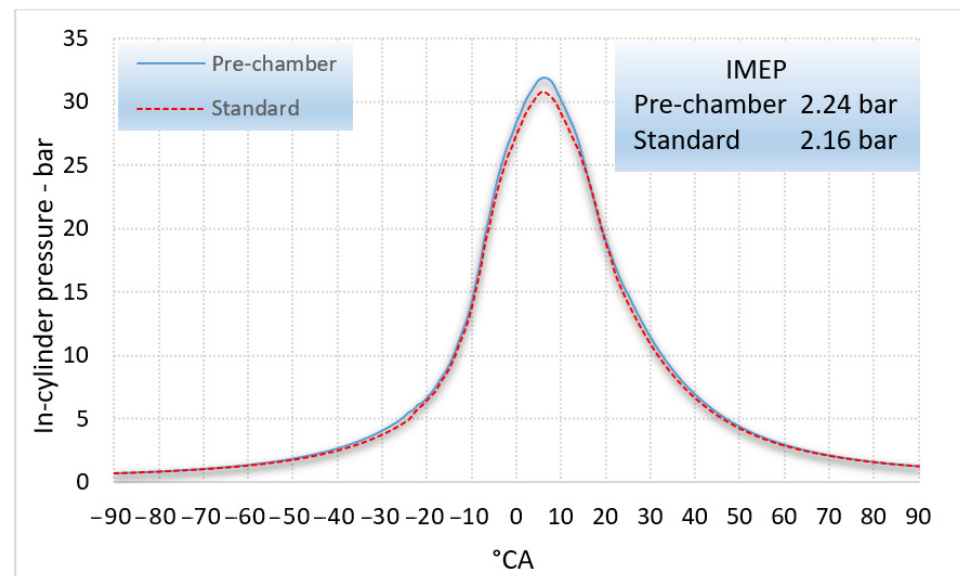


**Figure 6.** In-cylinder pressure curve at 1500 rpm and 8 Nm load.

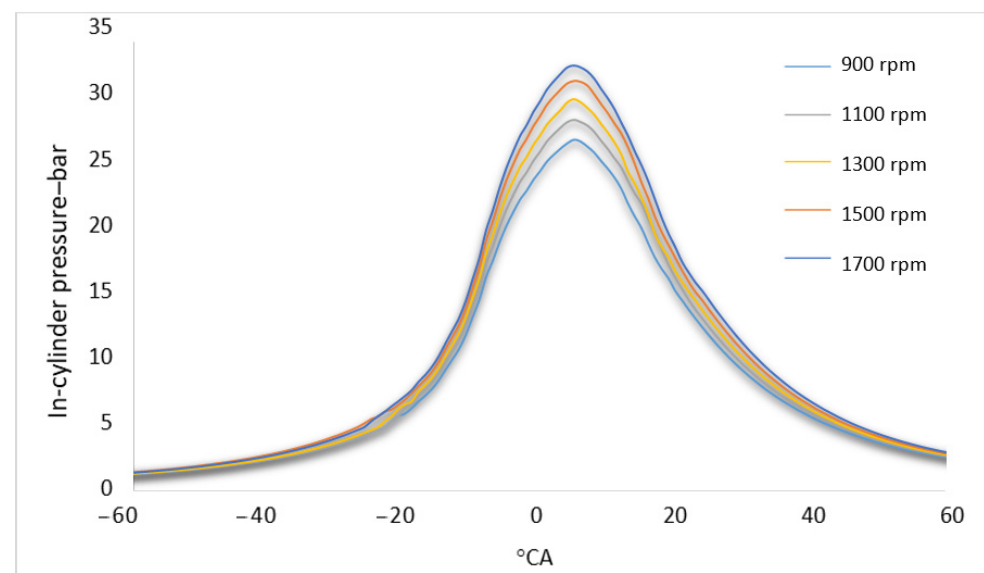


**Figure 7.** In-cylinder pressure curves for engine speeds.

Figure 6 illustrates the variation of in-cylinder pressure with crank angle for two different engine configurations: the baseline homogeneous combustion engine and the newly developed pre-chamber stratified charge design. The measurements were conducted under the same operating condition (1500 rpm and 8 Nm load) to ensure direct comparability. The peak pressure reaches approximately 33 bar for the pre-chamber engine, which is slightly higher than that of the standard configuration. More importantly, the IMEP for the pre-chamber engine is measured as 2.24 bar, compared to 2.16 bar for the standard design. This corresponds to a 3.7% increase in indicated performance, highlighting the effectiveness of the stratified combustion concept.

Figure 7 depicts the in-cylinder pressure (bar) versus crank angle (°CA) for different engine speeds (rpm). The in-cylinder pressure peaks around the top dead center (TDC) for all engine speeds, which is expected during the combustion process. As the engine speed increases (from 900 rpm to 1700 rpm), the peak pressure slightly increases too. The maximum pressure points shift slightly to the right, indicating changes in the timing and duration of the combustion process as engine speed increases. The peak pressure remains

relatively consistent across speeds, suggesting efficient combustion. The net area under the cylinder pressure curve represents the indicated work, and the ratio of this value to the stroke volume is known as the IMEP value. For a more precise comparison, the IMEP values calculated from the in-cylinder combustion pressure curves are presented in Figure 8.

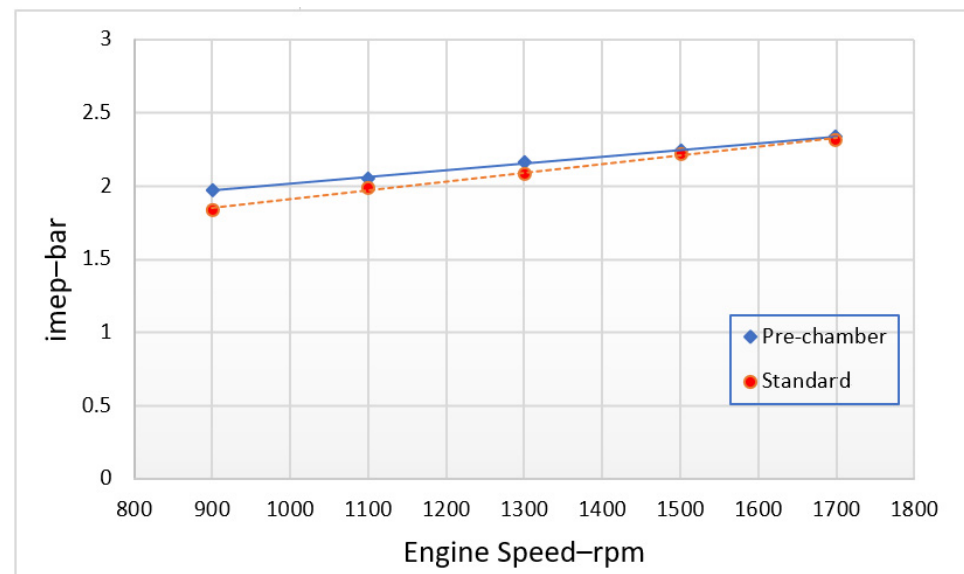


**Figure 8.** Variation in IMEP with engine speed for pre-chamber and baseline configurations.

Figure 8 compares the indicated mean effective pressure (IMEP) of the pre-chamber and standard combustion chamber designs across a range of engine speeds from 900 rpm to 1700 rpm under identical load conditions.

As illustrated in Figure 8, the IMEP values for both engine configurations increase with rising engine speed; however, the performance advantage offered by the pre-chamber design becomes less pronounced at higher speeds. For instance, at 900 rpm, the IMEP of the pre-chamber engine reaches 1.90 bar, while the baseline engine records 1.80 bar. This indicates that the staged combustion process initiated by the pre-chamber contributes more effectively to combustion efficiency at low engine speeds, particularly under lean mixture conditions.

As engine speed increases, the IMEP values of both configurations begin to converge. At 1700 rpm, the pre-chamber and baseline engines achieve IMEP values of 2.34 bar and 2.31 bar, respectively. This narrowing gap indicates that the benefits of stratified combustion are less prominent at higher speeds, where reduced residence time and enhanced turbulence promote faster and more homogeneous combustion. As a result, the staged combustion effect becomes less influential, and the performance characteristics of both engine configurations tend to align. The power values calculated with the load cell and the IMEP are given in Figure 9.

Figure 9 shows the relationship between indicated power and brake power across different engine speeds. The indicated power increases steadily with engine speed, reaching a peak of approximately 2.5 kW at 1700 rpm. This trend indicates a rise in the energy produced within the engine cylinders as the speed increases. Similarly, brake power also increases with engine speed but remains consistently lower than the indicated power. At 1700 rpm, brake power peaks at around 1.8 kW. The difference between the indicated power and brake power arises from mechanical losses, including friction and heat dissipation within the engine. While both power curves follow a similar trend, the gap between the two widens at higher speeds, reflecting a decrease in mechanical efficiency as the engine speed increases. The decrease in mechanical efficiency indicates an increase in friction losses. To

better understand this, it is important to examine the exhaust temperature and changes in volumetric efficiency. Figures 10 and 11 illustrate the graphs of exhaust temperature and volumetric efficiency, respectively, providing insights into how these factors vary with engine speed and contribute to the overall performance.

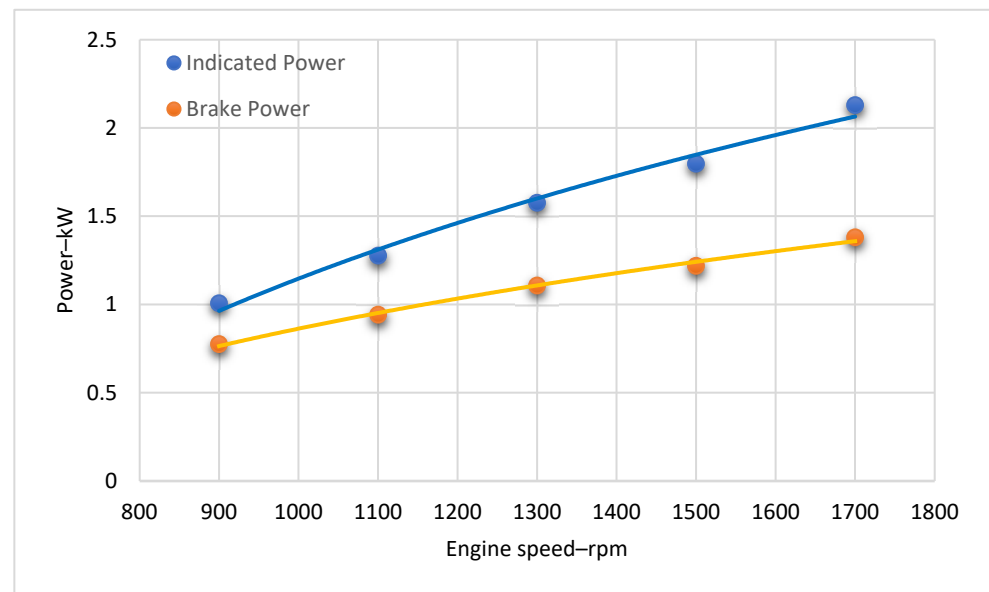


**Figure 9.** Comparison of indicated power and brake power at various engine speeds (rpm).

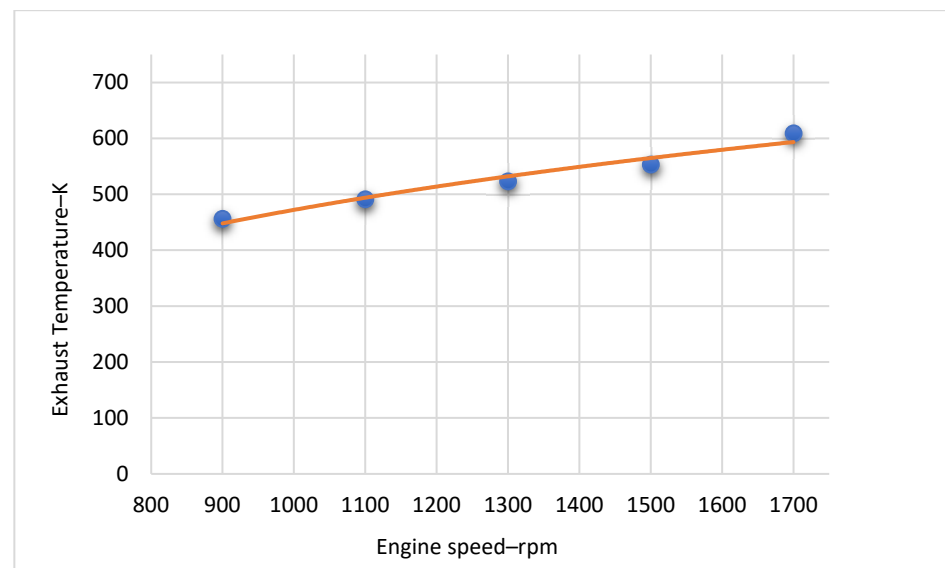


**Figure 10.** The exhaust temperatures for engine speeds.

Figure 10 illustrates the exhaust temperature (in Kelvin) as a function of engine speed (rpm). The exhaust temperature rises steadily with engine speed, ranging from approximately 500 K at 900 rpm to around 600 K at 1700 rpm. This increase reflects higher combustion temperatures as the engine operates at higher speeds. Higher exhaust temperatures at increased engine speeds suggest improved energy release during combustion, although they also indicate greater heat losses through the exhaust system. The consistent rise in exhaust temperature aligns with the expected behavior of engines under load, as more fuel and air are combusted to meet the demands of higher speeds.

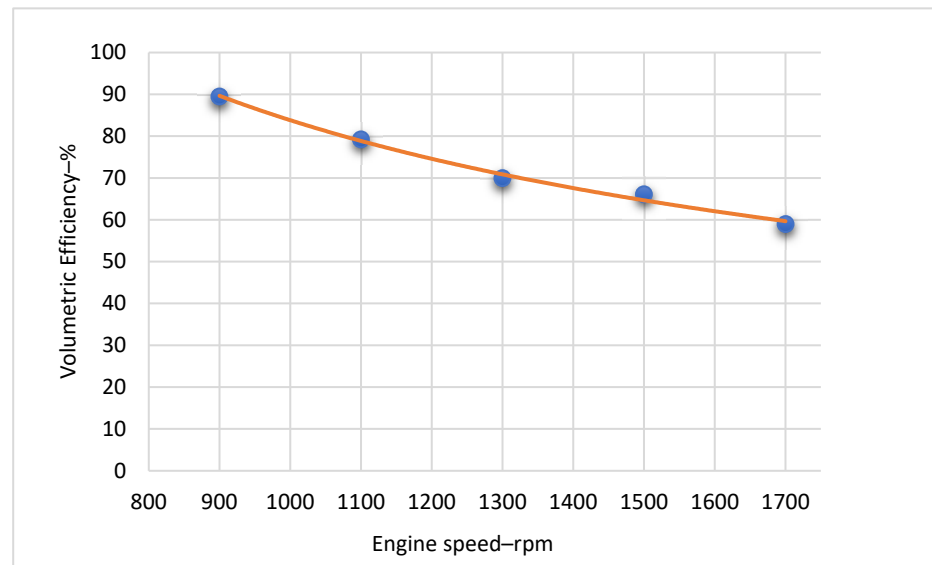


**Figure 11.** The volumetric efficiency variation for engine speeds.

Figure 11 represents the volumetric efficiency of the engine as a function of engine speed (rpm). Volumetric efficiency decreases as engine speed increases. It starts at approximately 90% at 900 rpm and drops to around 70% at 1700 rpm. This decline is typical for naturally aspirated engines, where the ability to fill the cylinder with the air–fuel mixture diminishes at higher speeds due to reduced intake time. At higher engine speeds, the shorter intake duration limits the complete filling of the cylinder, leading to a reduction in volumetric efficiency. This can affect overall engine performance, particularly at high speeds. The decreasing trend highlights the potential benefits of forced induction systems, such as turbochargers, which can maintain or improve volumetric efficiency at elevated engine speeds.

#### 4. Conclusions

This study has demonstrated the effectiveness of a novel stratified charge pre-chamber design in enhancing the performance and combustion characteristics of a natural gas-fueled spark ignition engine operating under partial load. By integrating an auxiliary intake valve to enrich the pre-chamber mixture without a dedicated injector, the system achieves a simplified yet effective stratification strategy that bridges the advantages of both passive and active pre-chamber concepts. Experimental and CFD analyses confirm that the proposed design improves the indicated mean effective pressure (IMEP), flame propagation speed, and combustion stability, particularly at lower engine speeds where the benefits of stratified combustion are most pronounced. At 900 rpm, the pre-chamber configuration showed an IMEP of 1.90 bar compared to 1.80 bar in the standard design. This performance advantage gradually diminished with increasing speed, aligning with the expected behavior due to shorter residence time and enhanced turbulence at high rpm. The engine also exhibited improved lean burn capability. However, mechanical losses and exhaust heat increased with engine speed, as indicated by rising exhaust temperatures (up to 600 K) and declining volumetric efficiency (from 90% to 70%). These findings suggest that further optimization such as turbocharging or improved scavenging may be required for high-speed applications. Overall, the results validate that the proposed pre-chamber system offers significant advantages in thermal efficiency, ignition control, and part-load operation. It is especially suited for urban driving conditions, where low-to-medium speeds dominate. This design provides a practical and cost-effective solution for advancing natural gas engine technologies in line with future efficiency and emission targets.

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**Data Availability Statement:** The data presented in this study are available on request from the corresponding author.

**Conflicts of Interest:** The author declares that he has no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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