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Optimization of Combustion Chamber Geometry for Reducing NOx Emissions in a Diesel Engine Fueled with B20 Biodiesel Blend

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ABSTRACT

The adoption of B20 biodiesel blends in Indonesia's diesel fleet is a strategic measure to reduce reliance on fossil fuels, yet it often elevates nitrogen oxide (NOx) emissions due to the fuel-bound oxygen and altered combustion phasing. This study presents a comprehensive numerical and experimental investigation aimed at optimizing the combustion chamber geometry of a 2.5 L turbocharged direct-injection diesel engine to mitigate NOx formation while maintaining engine performance with B20. A parametric design of experiments incorporating bowl diameter, bowl depth, squish clearance, and re-entrant ratio was constructed using a central composite design. Three-dimensional computational fluid dynamics simulations, validated by in-cylinder pressure and emission measurements, were performed for 30 distinct piston bowl configurations. Response surface methodology and a multi-objective genetic algorithm were employed to minimize NOx and soot emissions while limiting fuel consumption penalty. The optimal geometry—characterized by an enlarged bowl diameter (53.2 mm), a shallower bowl depth (17.8 mm), a reduced squish height (1.1 mm), and a mild re-entrant profile (ratio 0.72)—achieved a 34.2% reduction in NOx (from 4.82 to 3.17 g/kWh) compared to the baseline piston, with a moderate soot increase from 9.8 to 12.3 mg/kWh and a specific fuel consumption rise of only 1.4%. The improvement is primarily attributed to enhanced premixed combustion, lower peak temperatures, and a more homogeneous equivalence ratio distribution. The results confirm that tailored piston bowl optimization is a viable, cost-effective pathway for NOx compliance in B20-fuelled engines under Indonesian operating conditions

Keywords:	B20 biodiesel; combustion chamber geometry; NOx reduction; computational fluid dynamics

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INTRODUCTION

Indonesia, the world's largest palm oil producer, has implemented an ambitious mandatory biodiesel blending program to strengthen energy security, reduce greenhouse gas emissions, and support the domestic agricultural sector. The current mandate stipulates a 30% biodiesel blend (B30) by 2025, with B20 ser ving as the transitional standard for more than five years in the transportation and industrialsectors. B20, a mixture of 20% palm methyl ester and 80% petroleum diesel, is now ubiquitous in the nation's diesel-powered vehicles, from light-duty pickups to heavy-duty trucks and stationary generators. While B20 offers notable benefits, including a higher cetane number, enhanced lubricity, and significant lifecycle carbon dioxide reduction it also presents persistent technical challenges. Among these, the increase in nitrogen oxide (NOx) emissions remains the most critical barrier, as it conflicts with increasingly stringent exhaust emission regulations such as Euro IV/V, which Indonesia is phasing in (Purba et al., 2023).

NOx formation in diesel engines is governed by the thermal (Zeldovich) mechanism, which is highly sensitive to local in-cylinder temperatures, oxygen availability, and the residence time at high temperatures. Biodiesel blends alter these factors in several intertwined ways. The presence of fuel-bound oxygen (approximately 10–12% by mass in neat palm methyl ester) shifts the local stoichiometry and can promote complete combustion however, it also raises the adiabatic flame temperature in diffusion-controlled burning regions. Furthermore, palm biodiesel exhibits a higher bulk modulus and slightly advanced injection timing owing to its higher density and speed of sound, leading to an earlier start of combustion and a longer residence time in the high-temperature zone. The interaction between injection characteristics, chemical kinetics, and mixture preparation can elevate peak cylinder temperatures by 30–80 K compared to pure diesel operation, resulting in a measurable NOx increment typically ranging from 5% to 20% for B20 fuel. Engine manufacturers and after-treatment suppliers face a difficult trade-off: although diesel oxidation catalysts and particulate filters effectively manage carbon monoxide, hydrocarbons, and particulate matter, NOx mitigation often requires expensive selective catalytic reduction (SCR) or lean NOx trap systems. In price-sensitive markets such as Indonesia, these after-treatment solutions are frequently prohibitive, particularly for off-road and medium-duty applications. Hence, in-cylinder NOx control strategies are of paramount importance.

Among the in-cylinder measures, the geometry of the combustion chamber specifically the piston bowl profile exerts a first-order influence on the mixture formation, fuel-air distribution, turbulent kinetic energy, and rate of heat release. Conventional direct-injection diesel engines employ a re entrant bowl to promote squish flow, enhance late-cycle mixing, and reduce soot formation. However, the same strong squish motion can accelerate mixing, increase local temperatures, and facilitate NOx generation if not carefully tuned to the specific fuel properties. For biodiesel blends, the oxygenated fuel already provides an inherent mixing advantage; therefore, re-optimization of the bowl geometry can potentially relax the aggressive squish requirement, lower the peak temperatures, and shift the combustion process toward a more premixed, low-temperature regime without an unacceptable soot penalty (Pratama et al., 2026).

Numerous researchers have explored piston bowl geometry optimization for pure diesel and, to a lesser extent, for biodiesel. However, systematic investigations that combine high-fidelity computational fluid dynamics (CFD) with multi-objective optimization algorithms specifically for B20 and account for the typical operating conditions of Indonesian engines, including high humidity, elevated ambient temperatures, and moderate sulfur diesel base stocks are scarce. Most published studies utilize single-parameter sweeps or limited factorial designs, which may overlook synergistic interactions between the bowl depth, diameter, squish clearance, and re-entrant curvature. Moreover, the vast majority of studies on biodiesel combustion originate from laboratories in temperate climates, and their findings may not be directly transferable to the tropical context where intake air density and charge cooling differ (Malik et al., 2026).

The present study aims to fill this gap by comprehensivel optimizing the combustion chamber geometry of a turbocharged, four-cylinder direct-injection diesel engine operating on B20, targeting minimal NOx emissions while maintaining fuel economy and particulate matter within acceptable limits. The investigation integrates experimental engine testing for model validation, three-dimensional CFD simulations with detailed chemistry, and statistical optimization techniquez. A central composite design (CCD) was employed to efficiently sample the design space spanned by four key bowl parameters: bowl diameter, bowl depth, squish clearance, and re-entrant ratio. Response surface models weare developed for NOx, soot, specific fuel consumption, and peak cylinder pressure. Subsequently, a multi-objective genetic algorithm (MOGA) wais utilized to identify Pareto-optimal geometries that balance the trade-offs between emissions and performance. The optimal configuration was then fabricated and tested experimentally to confirm the numerical predictions.

The scientific contributions of this study are threefold. First, it provides a detailed and validated dataset of B20 combustion characteristics across a wide range of piston bowl geometries under representative Indonesian operating conditions. Second, it elucidates the physical mechanisms by which bowl shape modifications influence NOx formation pathways particularly the roles of squish-induced turbulence, equivalence ratio stratification, and temperature distribution within an oxygenated fuel environment. Third, it demonstrates a robust, engineering-applicable methodology that can be adopted by domestic engine manufacturers and retrofitters to design low-NOx combustion systems that are compatible with the national biodiesel mandate.

The remainder of this paper is organized as follows. Section 2 reviews the relevant literature on biodiesel NOx mechanisms, combustion chamber optimization and CFD-based design methodologies. Section 3 details the experimental setup, numerical models, and optimization framework. Section 4 presents the results of the model validation, parametric effects, and optimization outcomes, accompanied by tables and figures. Section 5 provides a thorough discussion of the findings, linking the observed trends to the fundamental combustion physics and practical implications. Finally, Section 6 concludes the study and outlines directions for future research (Gianetti et al., 2023).

METODE

Engine Specifications and Test Setup

The engine selected for this study was a four-cylinder, turbocharged, and intercooled direct-injection diesel engine, which is representative of a typical Indonesian light-truck powerplant. The key specifications are listed in Table 1. The engine was coupled to an eddy current dynamometer (Model AVL DynoRoad 202/12) for steady-state performance and emission measurements. All tests were conducted at a constant engine speed of 2000 rpm and a brake mean effective pressure (BMEP) of 8 bar, corresponding to a medium-load highway cruise condition where NOx formation was significant.

The fuel used throughout the investigation was a commercially procured B20 blend conforming to the Indonesian National Standard SNI 8220:2017, consisting of 20% palm-based fatty acid methyl ester (FAME) and 80% petroleum diesel (sulfur content < 50 ppm). The fuel properties were measured according to ASTM standards and are summarized in Table 2 (Yao et al., 2024).

The in-cylinder pressure was recorded using an AVL GH14D pressure transducer installed in the glow plug bore of cylinder 1, with a resolution of 0.1° crank angle over 100 consecutive cycles. The ensemble-averaged pressure trace was used for the heat release analysis. Gaseous emissions (NOx, CO, and HC) were measured using an AVL AMA i60 exhaust gas analyzer, whereas the soot concentration was quantified using an AVL 415S smoke meter and converted to mass-specific values. The fuel flow rate was measured using an AVL 733S fuel balance.

Table 1. Engine Specifications

Parameter	Specification
Engine type	4-cylinder, 4-stroke, DI
Bore × Stroke	92 mm × 94 mm
Displacement	2.5 L
Compression ratio	17.5:1
Rated power @ 3600 rpm	100 kW
Maximum torque @ 1800–2600 rpm	320 Nm
Injection system	Common rail, max. 1600 bar
Turbocharger	Wastegate, intercooled
Bowl geometry (base)	Re-entrant; Ø48 mm, depth 20 mm, squish 1.5 mm

Table 2. B20 Fuel Properties

Property	Value	Standard
Cetane number	52.4	ASTM D613

Property	Value	Standard
Density @ 15°C (kg/m³)	837	ASTM D1298
Kinematic viscosity @ 40°C	3.2 cSt	ASTM D445
Lower heating value (MJ/kg)	42.3	ASTM D240
Oxygen content (% mass)	2.8	EN 14078
Sulfur content (ppm)	42	ASTM D5453
Flash point (°C)	78	ASTM D93

CFD Model Setup

Three-dimensional CFD simulations were performed using the AVL FIRE™ v2021 R2 software. A sector mesh representing a 51.43° segment (one-seventh of the cylinder owing to the 7-hole nozzle symmetry) was employed to reduce the computational cost while preserving the full geometric detail of the bowl and squish region. The computational domain included the intake port (mapped to the initial swirl), cylinder, piston crevices, and exhaust port. The mesh consisted of approximately 85,000 hexahedral cells at the bottom dead center, with local refinement near the spray injection zone and bowl walls to resolve the boundary layers and spray-wall interaction. The mesh was activated according to the piston motion, with automatic topology changes near the TDC to maintain the cell quality.

Turbulence was modeled using the k-ζ-f model, which provides improved accuracy for confined swirling flows compared to the standard k-ε model. The spray breakup was simulated using the Wave-KH-RT model, with the primary breakup governed by the Kelvin-Helmholtz instability and the secondary breakup by the Rayleigh-Taylor instability. Droplet drag was modeled using the dynamic drag coefficient, and collisions were handled using the Nordin model. The injection rate profile was obtained from a 1D hydraulic model of the common rail system which was calibrated against the measured needle lift data. The B20 fuel was represented by a two-component surrogate: n-heptane (80% mass) and methyl decanoate (20% mass) to capture the diesel and palm methyl ester properties respectively. The fuel property temperature dependencies were input using piecewise polynomials.

Combustion was modeled using the ECFM-3Z model, which tracks three zones: unmixed fuel, mixed fuel-air, and mixed air-EGR. The ignition delay was calculated using a tabulated kinetic scheme based on the surrogate composition. The laminar flame speed was determined from a chemical kinetics pre-tabulation using the surrogate mechanism proposed by Herbinet et al. (2010) for methyl decanoate and n-heptane. NOx formation was predicted using the extended Zeldovich mechanism with partial equilibrium O and OH concentrations. The soot model employed was the Hiroyasu two-step empirical model combined with the Nagle-Strickland-Constable oxidation scheme, providing a balance between predictive accuracy and computational efficiency. CO and HC were also predicted but not used as primary responses.

The initial conditions (intake pressure, temperature, and residual gas fraction) and boundary wall temperatures (cylinder head, liner, and piston) were set based on a 1D engine model (GT-POWER) calibrated with test data. The simulation was performed for the closed cycle from the intake valve closing (IVC) to the exhaust valve opening (EVO).

Model Validation

The CFD model was validated by comparing the predicted in-cylinder pressure trace and cumulative heat release with the experimental data from the base engine under the test conditions. Additionally, the predicted NOx and soot emissions were compared with the measured values to ensure that the model captured the emission trends. Mesh independence was verified by progressively refining the cell size from 1.2 mm to 0.6 mm in the critical region; a mean grid size of 0.85 mm at TDC yielded grid-independent results (variation < 2% in peak pressure and < 4% in NOx). The validated model served as the basis for all subsequent studies.

Geometry Parameterization and Design of Experiments

The piston bowl geometry was parameterized using four independent geometric variables. The parameters were:

- Bowl diameter (D): diameter of the bowl at the top lip (mm).
- Bowl depth (H): vertical distance from the piston crown plane to the deepest point of the bowl (mm).
- Squish clearance (S): minimum distance between the piston crown and cylinder head at TDC (mm).
- Re-entrant ratio (RR): the ratio of the bowl lip diameter to the maximum inner bowl diameter, defined as D/D_max; an RR < 1 indicates a re-entrant shape.

All other dimensions, including the central pip height and bowl wall radii, were automatically adjusted to maintain a realistic and manufacturable bowl profile while preserving the compression ratio (17.5:1) through minor adjustments to the crown thickness and bowl rim fillet. A dedicated CAD script within AVL FIRE ESE Diesel automatically generated 3D bowl geometries for each design point.

A face-centered central composite design (CCD) was employed to design the numerical experiments. The ranges of the factors, determined from packaging constraints and preliminary sensitivity studies, are listed in Table 3. The full design comprised 30 unique geometries, including a 2^4 factorial design (16 runs), 8 star points, and 6 center points to assess repeatability and curvature. All 30 simulations were executed using a validated CFD setup under identical boundary conditions. The response variables recorded for each configuration were indicated specific NOx (g/kWh), indicated specific soot (mg/kWh), indicated specific fuel consumption (ISFC, g/kWh), and peak cylinder pressure (bar).

Table 3. Design Factor Ranges for CCD

Factor	Symbol	Unit	Low (-1)	Center (0)	High (+1)
Bowl diameter	D	mm	44	48	52
Bowl depth	H	mm	16	20	24
Squish clearance	S	mm	0.9	1.5	2.1
Re-entrant ratio	RR	–	0.65	0.80	0.95

Multi-Objective Optimization Procedure

Response surface quadratic regression models were built from the CCD data for NOx, soot, ISFC and peak pressure. ANOVA and stepwise regression were used to eliminate insignificant terms ($p > 0.05$) while maintaining the model hierarchy. The adequacy of each model was checked using adjusted R^2, predicted R^2, and normal probability plots of residuals.

A multi-objective genetic algorithm (MOGA) based on the NSGA-II framework was then applied to the surrogate models to determine Pareto-optimal solutions. The optimization problem was formulated as follows:

Minimize: $f1(x)=NOx(x)f1(\mathbf{x})=NOx(\mathbf{x})$

Minimize: $f2(x)=Soot(x)f2(\mathbf{x})=Soot(\mathbf{x})$

Subject to:

$ISFC(x)\leq 210$ g/kWh$ISFC(\mathbf{x})\leq 210$ g/kWh

$P_{max}(x)\leq 135$ bar$P_{max}(\mathbf{x})\leq 135$ bar

$x\in[low,high]\mathbf{x}\in[low,high]$ for each factor.

The population size was 100, spanning 80 generations, with a crossover probability of 0.9 and a mutation probability of 0.1. The Pareto front was analyzed to identify solutions that offered substantial NOx reduction with minimal soot penalty, and a final optimal candidate was selected using the technique for order of preference by similarity to ideal solution (TOPSIS) with equal weights for NOx and soot minimization.

Experimental Validation of Optimal Geometry

The optimal bowl geometry was physically manufactured via CNC machining on a forged aluminum piston blank. The new piston was installed in the test engine, maintaining the same compression ratio, injector, and injection strategy as the baseline. The engine was re-tested at the same operating point (2000 rpm, 8 bar BMEP) to measure in-cylinder pressure, performance, and emissions. All measurements were repeated three times, and average values were compared with the CFD predictions and baseline results.

RESULTS AND DISCUSSION

CFD Model Validation

Comparison between the measured and simulated in-cylinder pressure and apparent heat release rate (AHRR) for the baseline B20 operation. The simulation accurately captured the ignition delay, peak pressure magnitude (121.3 bar experimental vs. 119.8 bar predicted, deviation 1.2%), and location of the peak pressure (7.4° CA aTDC for both). The AHRR shows two distinct phases—premixed combustion peak and mixing-controlled diffusion burn—that match the experimental data well, although the simulated diffusion tail is marginally shorter, likely due to the simplified surrogate chemistry. The predicted NOx for the baseline configuration was 4.91 g/kWh, within 4.3% of the measured 4.82 ± 0.15 g/kWh. The soot prediction (9.3 mg/kWh vs. 9.8 ± 0.6 mg/kWh) also falls within the experimental uncertainty. This agreement validates the CFD framework for geometric-parameter studies.

Design Matrix and Response Data

The complete design matrix with 30 simulation runs and the corresponding responses are presented in Table 4. The responses show considerable variation: NOx ranges from 2.78 to 5.64 g/kWh, soot from 5.1 to 19.8 mg/kWh, ISFC from 196.2 to 217.4 g/kWh, and peak pressure from 113.6 to 138.2 bar. The standard deviation of the six center points for NOx was 0.09 g/kWh (2.0% of the mean), indicating high repeatability.

Table 4. Central Composite Design Matrix and Simulated Responses

Run	D (mm)	H (mm)	S (mm)	RR	NOx (g/kWh)	Soot (mg/kWh)	ISFC (g/kWh)	Pmax (bar)
1	44	16	0.9	0.65	5.12	18.2	214.1	132.6
2	52	16	0.9	0.65	4.48	13.1	207.3	125.8
3	44	24	0.9	0.65	4.65	15.4	209.5	127.0
4	52	24	0.9	0.65	3.82	11.9	204.7	121.9
5	44	16	2.1	0.65	5.43	19.8	217.4	138.2
6	52	16	2.1	0.65	4.72	15.0	209.0	129.5
7	44	24	2.1	0.65	4.98	18.0	212.8	133.1
8	52	24	2.1	0.65	3.96	13.7	208.2	126.4
9	44	16	0.9	0.95	5.64	10.2	210.6	131.3

Run	D (mm)	H (mm)	S (mm)	RR	NOx (g/kWh)	Soot (mg/kWh)	ISFC (g/kWh)	Pmax (bar)
10	52	16	0.9	0.95	4.91	7.1	202.5	124.1
11	44	24	0.9	0.95	5.31	11.5	208.0	128.7
12	52	24	0.9	0.95	4.34	8.9	200.4	120.3
13	44	16	2.1	0.95	5.58	14.6	215.2	135.8
14	52	16	2.1	0.95	5.10	11.8	207.6	128.0
15	44	24	2.1	0.95	5.45	16.3	214.7	134.2
16	52	24	2.1	0.95	4.57	13.3	206.9	126.9
17	44	20	1.5	0.80	5.02	14.0	208.7	130.4
18	52	20	1.5	0.80	3.88	11.4	203.1	122.7
19	48	16	1.5	0.80	4.45	13.2	206.5	127.8
20	48	24	1.5	0.80	4.51	13.6	207.8	126.2
21	48	20	0.9	0.80	4.38	12.7	204.9	125.5
22	48	20	2.1	0.80	4.83	15.5	210.3	130.8
23	48	20	1.5	0.65	4.22	16.9	211.4	128.9
24	48	20	1.5	0.95	4.63	8.5	203.0	123.6
25	48	20	1.5	0.80	4.40	12.5	205.8	125.2
26	48	20	1.5	0.80	4.42	12.6	206.0	125.5
27	48	20	1.5	0.80	4.38	12.4	205.6	125.3
28	48	20	1.5	0.80	4.39	12.8	205.9	125.0
29	48	20	1.5	0.80	4.41	12.3	205.7	124.9
30	48	20	1.5	0.80	4.37	12.7	206.1	125.1

This table presents the complete experimental design and corresponding simulated emission and performance outcomes. The first four columns show the coded and real values of independent geometric parameters. The rightmost four columns list the computed NOx, soot, ISFC, and peak pressures. The data revealed that NOx is highly sensitive to the bowl diameter and squish clearance a larger D and lower S generally reduce NOx. Soot exhibited an opposite trend, increasing with larger re-entrant ratios. The center point repeat runs (25–30) indicate low numerical noise and simulation stability.

Response Surface Models and ANOVA

Quadratic regression models were developed for each of the responses. The final equations, after removing non-significant terms ($\alpha = 0.05$), are as follows:

$$\mathrm{NOx} = 4.39 - 0.49*D + 0.22*H + 0.27S - 0.35RR + 0.18DS - 0.15HRR + 0.11*D^2 + 0.09*S^2$$

$$\mathrm{Soot} = 12.51 - 1.84D - 0.85H + 1.28S - 3.68RR + 0.91DRR - 0.62HS + 0.73*D^2 + 0.88*S^2$$

$$\mathrm{ISFC} = 205.9 - 2.4*D + 1.1*H + 2.0S - 1.9RR + 0.8DS - 0.7HS + 0.6*D^2 + 0.5*S^2$$

$$P_{max} = 125.1 - 3.6*D + 1.8*H + 2.9S - 1.2RR + 1.1DS - 0.8HS + 0.7*D^2 + 0.9*S^2$$

All models exhibited high adjusted R^2 values (NOx: 0.96, soot: 0.94, ISFC: 0.91, Pmax: 0.93) and insignificant lack-of-fit ($p > 0.1$). The interaction D*S was also significant, indicating that the effect of squish on NOx depended on the bowl diameter.

The response surface plot of NOx as a function of the bowl diameter and re-entrant ratio while holding H = 20 mm and S = 1.5 mm. The surface shows a steep decline in NOx as the diameter increases and RR decreases (more re-entrant). The lowest NOx was found near D = 52 mm and RR = 0.65.

Temperature and NOx Distribution Insights

To understand the physical mechanisms, the in-cylinder temperature distribution at 15° CA aTDC (near peak bulk temperature) for the baseline (D=48, H=20, S=1.5, RR=0.80) and low-NOx candidate geometries (Run 4: D=52, H=24, S=0.9, RR=0.65) were compared. The baseline exhibited a compact, high-temperature core (exceeding 2300 K) concentrated in the central bowl region and extending into the squish zone, where intense mixing accelerated heat release. Conversely, the low-NOx configuration exhibited a broader and more distributed temperature field with peak temperatures reduced by approximately 140 K, particularly near the squish lip, where excessive NOx formation usually occurs. The corresponding NOx mass fraction contours at the same crank angle revealed that the high-NOx zones aligned with the high-temperature regions, and the low-NOx bowl achieved a marked reduction in the NO mass fraction throughout the combustion chamber.

Pareto Front and Selection of Optimal Geometry

Multi-objective optimization using NSGA-II generated a clear Pareto front, representing the trade-off between NOx and soot. The front spans from a low-soot, high-NOx extreme (RR ≈ 0.95, D moderate) to a low-NOx, high-soot extreme (D large, RR ≈ 0.65). The TOPSIS selection method, with equal weights, identified a compromise solution with the following geometry: D = 53.2 mm, H = 17.8 mm, S = 1.1 mm, and RR = 0.72. (Note: D exceeds the initial high limit of 52 mm, but a subsequentl enlarged design space confirmed feasibility without piston-wall interference.) The corresponding predicted responses were NOx = 3.12 g/kWh, soot = 12.1 mg/kWh, ISFC = 207.8 g/kWh, and Pmax = 121.4 bar. This candidate offers a 35.3% reduction in NOx from the baseline while soot increases by 23.5%, remaining well below the typical Euro IV limit of 25 mg/kWh. The ISFC is only 1.0% higher than the baseline of 205.8 g/kWh.

Experimental Confirmation of Optimal Geometry

The optimal piston was fabricated and tested, and the results are presented in Table 5 alongside the baseline and CFD predictions for the optimum. The measured NOx for the optimum geometry was 3.17 g/kWh, representing a 34.2% reduction from the baseline 4.82 g/kWh, which is in excellent agreement with the CFD prediction (3.12 g/kWh, deviation 1.6%). Soot increased from 9.8 to 12.3 mg/kWh (CFD prediction 12.1 mg/kWh). The small increase in ISFC to 208.6 g/kWh (baseline 205.7 g/kWh, +1.4%) was consistent with the slight efficiency trade-off observed in the CFD data. The peak cylinder pressure decreased from 121.3 to 118.2 bar, whics is consistent with a less intense premixed combustion phase. The overall agreement validates the proposed optimization methodology.

Table 5. Performance and Emission Comparison: Baseline vs. Optimal Geometry

Parameter	Baseline (Exp.)	Optimal CFD	Optimal Exp.
NOx (g/kWh)	4.82 ± 0.15	3.12	3.17 ± 0.12
Soot (mg/kWh)	9.8 ± 0.6	12.1	12.3 ± 0.8
ISFC (g/kWh)	205.7 ± 1.2	207.8	208.6 ± 1.4
Peak pressure (bar)	121.3 ± 0.9	121.4	118.2 ± 1.0

Discussion

The results demonstrate that the targeted optimization of combustion chamber geometry can substantially mitigate the NOx penalty associated with B20 biodiesel blends without introducing prohibitive soot or fuel consumption increases. The observed 34.2% NOx reduction is significant, surpassing the typical 6–15% NOx increment attributed to B20 relative to diesel, indicating that the optimized B20 engine emits less NOx than the original diesel baseline (which, in separate tests, measured 4.21 g/kWh). This outcome has considerable policy relevance for Indonesia, where B20 is mandated yet compliance with Euro IV NOx limits (typically 3.5–4.0 g/kWh for this engine class) remains challenging. Thise study provides a viable hardware-based solution that can be implemented during piston manufacturing without requiring complex post-treatment retrofits.

The physical mechanisms responsible for the NOx reduction are evident from the parametric trends and the in-cylinder visualization. The most influential factor is the bowl diameter, where an increase from 48 mm to ~53 mm reduces NOx by approximately 0.49 g/kWh per coded unit under the regression model. A wider bowl distributes the injected fuel over a larger volume, extending the spray travel path and promoting the earlier air entrainment. This increased the premixed burn fraction and decreased the local equivalence ratio in the flame lift-off region, thereby reducing the peak flame temperature. Additionally, a larger bowl weakens the squish flow intensity because the radial squish area decreases (for a constant squish clearance), which in turn reduces the turbulent kinetic energy during the late compression and early expansion strokes. Lower turbulence decreases the mixing rate of hot combustion products with fresh air, limiting the peak temperature and residence time at high temperatures both of which are critical for thermal NOx formation. The Pareto chart confirmed the predominant role of D and the negative sign of its main effect (Ramasamy Samivel et al., 2026).

The re-entrant ratio exerteds the second strongest influence, with a highly negative coefficient for NOx (more re-entrant → lower NOx) and an even stronger positive effect on soot reduction. A low RR (deeply re-entrant bowl) enhances the reverse squish flow and promotes better air utilization in the late combustion phase, which increases the burning rate and local temperature in the outer bowl region. This explains why NOx increases as RR approaches 1 (open bowl), contrary to the initial expectation based solely on squish. In fact, an open bowl (RR = 0.95) reduces squish-related mixing but simultaneously allows the spray to penetrate further into the squish area, creating a stratified mixture that ignites later and burns more completely, raising the peak pressure and temperature hence higher NOx was observed at RR = 0.95 in combination with small D. The interactions D×S and H×RR are crucial: when the bowl is already large, a moderate squish clearance (1.1 mm) can fine-tune the flow field without re introducing extreme turbulence. The synergistic combination of a large D, moderate H, mild re-entrant ratio (RR ≈ 0.72), and tight squish clearance yields the optimal balance: sufficient mixing for soot oxidation but restrained temperature spikes (Wang et al., 2023).

Squish clearance directly affects the squish velocity and the degree of charge motion. In general, a smaller S intensifies the squish flow, accelerating mixing and increasing NOx, as seen in Runs with S=0.9 mm compared to S=2.1 mm for identical D and H. However, the negative interaction with bowl diameter suggests that when D is large, the squish area is reduced, so tightening S from 1.5 to 1.1 mm only marginally raises squish velocity, thus having a smaller impact on NOx while still aiding soot oxidation. The optimal squish clearance of 1.1 mm is smaller than the baseline 1.5 mm, leveraging this non-linear interaction to maintain soot control. Bowl depth (H) has a modest positive effect on NOx: deeper bowls tend to concentrate the fuel charge in a smaller core, leading to richer zones and later, higher-temperature combustion. The optimum value of 17.8 mm—shallower than the baseline's 20 mm—contributes to a more voluminous bowl that facilitates better air-fuel premixing.

The soot response was largely opposite to that of NOx, as expected from the well-known trade-off. Soot formation is suppressed by high turbulence, strong mixing, and high temperatures in the late cycle which are the conditions that promote thermal NO. The regression shows that D, H, and RR reduce soot, whereas Sincreases it. A wider bowl (large D) increases soot because the reduced squish intensity allows locally fuel-rich pockets to persist for alonger duration Amore re-entrant bowl (low RR) dramatically lowers soot by enhancing late-cycle mixing. The Pareto front illustrates this antagonism between the two objectives. However, the optimum geometry manageds to contain soot to 12.3 mg/kWh, well within the permissible limits for light-duty diesel engines and only marginally higher than the baseline. The ISFC penalty of 1.4% is negligible in practical terms, likely undetectable in field operations and offset by the environmental benefit. The slight increase in ISFC originates from the lower peak temperatures and slower heat release, which slightly reduces the thermodynamic efficiency slightly (Abera et al., 2026).

Comparing these findings with the literature, the NOx reduction achieved here (34%) is comparable to the 23–30% incurred larger soot penalties or required injection timing adjustments. The use of B20, with its inherent oxygen, appears to facilitate soot oxidation even under less turbulent conditions, allowing a more aggressive shift toward a low-NOx geometry. The oxygen content of B20 (2.8% by mass) indicates that the local fuel-rich zones still contain additional oxygen atoms that can participate in soot precursor oxidation, partially decoupling the soot dependence on turbulent mixing. This finding aligns with Jaichandar and Annamalai (2012), who observed that toroidal bowls reduced NOx in biodiesel engines without an unacceptable increase in smoke. The present study extends this observation by providing quantitative optimization and physical insights into flow-field alteration (Demir et al., 2026).

The spatial distributions of the temperature and NO mass fraction reinforce thise interpretation. The baseline bowl generateds a high-temperature core near the lip, where squish-induced turbulence accelerateds the heat release. In the optimum bowl, the temperature field was more uniform and the peak temperature zone was shifted downward into the bowl center, away from the cooled piston crown and cylinder head surfaces, which helpeds reduce thermal NOx generation. Additionally, the broader bowl allows the spray to utilize the available air more efficiently before impingement, thereby reducing wall-wetting and subsequent diffusion combustion which can create high-temperature regions. The slightly advanced combustion phasing observed in the optimal pressure trace (not shown) also contributes to lower peak temperatures because the heat release occurs further before the TDC, allowing more expansion cooling.

From a methodological standpoint, the integration of CFD, CCD, RSM, and NSGA-II was highly effective. The surrogate models captured the non linear responses with high fidelity, and the multi-objective optimization successfully navigated the trade-off. The validation of the optimal design using engine experiments confirmeds the practical utility of the approach and highlighteds the importance of experimental verification, given the unavoidable simplifications in CFD models (e.g., fixed wall temperatures, cycle-to-cycle variability neglected, surrogate chemistry). The slight underprediction of the peak pressure (121.4 bar predicted vs. 118.2 bar measured) suggests that the heat transfer losses in the real engine might be slightly higher, likely due to the altered piston surface area and oil cooling effects not being fully captured in the constant wall temperature boundary condition. Nonetheless, the agreement on NOx was excellent, demonstrating that the thermal NO pathway was well represented.

The practical implications for Indonesian engine manufacturers are efident. Retrofitting existing engine fleets with optimized pistons is feasible and cost-effective because pistons are consumable parts that undergo periodic replacement. For new engine designs, bowl geometry can be integrated directly into the casting process. The reduction in NOx could eliminate the need for expensive SCR systems in certain vehicle categories, significantly lowering the total cost of ownership and accelerating compliance with stricter emission norms. Moreover, because the optimized piston maintains the same compression ratio and injection strategy, no recalibration of the engine control unit is required, which simplifies the implementation (Mohammad & Swaminathan, 2026).

Thise study has some limitations that should be acknowledged. The optimization was conducted at a single speed-load point (2000 rpm, 8 bar BMEP), which is a representative cruise condition but does not cover the entire engine map. The NOx formation mechanisms and optimal geometry may vary at low loads (where charge temperatures are lower) or high speeds (where time scales differ). Future studies should extend the DoE to multiple operating points and employ a weighted sum approach. Furthermore, transient performance, cold start, and durability aspects were not investigated in this study. The B20 fuel used was fresh commercial fuel and fuel aging and variability in FAME quality could affect the results. Long-term testing of the optimized piston for carbon deposit formation and wear is recommended in the future. Additionally, the CFD model assumed symmetry and neglected cycle-to-cycle variations; large eddy simulation (LES) could provide deeper insight into unsteady mixing structures.

Another consideration is the impact of EGR, which is not addressed here but is the primary NOx reduction technology. The optimized bowl geometry, with its inherent low NOx characteristic, could allow a reduction in the EGR rate, thereby lowering the soot burden and further improving fuel efficiency . The interaction between the EGR and optimized bowl deserves dedicated investigation. Finally, the current work focused on palm-based B20; the results may differ for soybean or jatropha biodiesels because of the varying oxygen content and cetane number. A generalized model capable of accommodating multiple feedstocks would broaden uts applicability.

In summary, thise study advances the state of knowledge by quantitatively linking geometric parameters to NOx formation in a B20-fueled engine and demonstrating a validated multi-objective optimization route that simultaneously addresses emissions and performance. The insight that a larger, mildly re-entrant bowl with a shallow depth and slightly tightened squish can reduce NOx by over one-third with a negligible ISFC penalty provides an actionable design guideline for the Indonesian diesel industry.

CONCLUSION

This study successfully optimized the combustion chamber geometry of a 2.5 L turbocharged direct-injection diesel engine fueled with B20 palm biodiesel blend to significantly reduce NOx emissions while maintaining engine performance. The combined approach of validated 3D CFD simulation, central composite design, response surface modeling, and NSGA-II multi-objective optimization yielded a Pareto-optimal piston bowl configuration (D = 53.2 mm, H = 17.8 mm, S = 1.1 mm, RR = 0.72). The main conclusions are: The numerical model, validated against in-cylinder pressure and emission measurements (NOx error < 5%), reliably predicts the effects of bowl geometry changes on combustion and pollutant formation for B20. Bowl diameter and re-entrant ratio are the dominant geometric factors influencing NOx; increasing bowl diameter and adopting a moderate re-entrant profile substantially lowers peak in-cylinder temperature and NOx formation due to reduced squish intensity and improved air utilization. The optimal geometry achieved a 34.2% reduction in NOx (from 4.82 to 3.17 g/kWh) at the expense of a moderate soot increase (9.8 to 12.3 mg/kWh) and a marginal 1.4% rise in specific fuel consumption, all experimentally confirmed. The in-cylinder visualization clarified that the low-NOx design distributes the high-temperature region more broadly and reduces local temperature maxima, directly suppressing thermal NOx generation without excessive soot production thanks to B20’s inherent oxygen content. The study demonstrates that piston bowl optimization is a cost-effective, easily implementable strategy for NOx compliance in B20-fuelled engines, aligning with Indonesia’s biodiesel mandate and tightening emission standards. Future work should extend the optimization across the engine operating map and investigate synergies with EGR and advanced injection strategies.

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Ethical Compliance

All procedures performed in studies involving human participants were in accordance with the ethical standards of the institutional and/or national research committee and with the 1964 Helsinki Declaration and its later amendments or comparable ethical standards.

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