Numerical Study on the Performance and NOx Emission Characteristics of an 800cc MPI Turbocharged SI Engine

Seungmin Kim 1, Jaesam Sim 2, Youngsoo Cho 3, Back-Sub Sung 4 and Jungsoo Park 1,*

Abstract: The main purpose of this study is to optimize engine performance and emission characteristics of off-road engines with retarded spark timing compared to MBT by repurposing the existing passenger engine. This study uses a one-dimensional (1D)-simulation to develop a non-road gasoline MPI turbo engine. The SI turbulent flame model of the GT-suite, an operational performance predictable program, presents turbocharger matching and optimal operation design points. To optimize the engine performance, the SI turbulent model uses three operation parameters: spark timing, intake valve overlap, and boost pressure. Spark timing determines the initial state of combustion and thermal efficiency, and is the main variable of the engine. The maximum brake torque (MBT) point can be identified for spark timing, and abnormal combustion phenomena, such as knocking, can be identified. Spark timing is related to engine performance, and emissions of exhaust pollutants are predictable. If the spark timing is set to variables, the engine performance and emissions can be confirmed and predicted. The intake valve overlap can predict the performance and exhaust gas by controlling the airflow and combustion chamber flow, and can control the performance of the engine by controlling the flow in the cylinder. In addition, a criterion can be set to consider the optimum operating point of the non-road vehicle while investigating the performance and exhaust gas emissions accompanying changes in boost pressure. With these parameters, the design of experiment (DoE) of the 1D-simulation is performed, and the driving performance and knocking phenomenon for each RPM are predicted during the wide open throttle (WOT) of the gasoline MPI Turbo SI engine. The multi-objective Pareto technique is also used to optimize engine performance and exhaust gas emissions, and to present optimized design points for the target engine, the downsized gasoline MPI Turbo SI engine. The results of the Pareto optimal solution showed a maximum torque increase of 12.78% and a NOx decrease of 54.31%.

Keywords: gasoline MPI turbo SI engine; wide open throttle (WOT); multi-objective pareto; design of experiment (DoE); knocking

1. Introduction

The world is focused on abnormal climate phenomena caused by air pollution, such as global warming, and regulations are being tightened in an effort to reduce air pollution. In particular, the commercial automobile industry, which has a direct impact on air pollution, is facing strict regulations that reduce emissions, such as EURO 6 and Tier-5, and is studying the development of new combustion technologies or technologies that can reduce pollutant emissions, such as aftertreatment systems. Non-road vehicles, such as tractors and utility terrain vehicles (UTVs), which are used in harsh environments, however, mainly use
diesel engines owing to their high power and fuel economy. Diesel engines installed in non-road vehicles emit air pollution-causing substances such as nitrogen oxides (NOx), carbon monoxide (CO), and particulate materials (PM). However, non-road vehicles are subject to lower regulations compared to the commercial automobile industry. Owing to the increasing severity of air pollution, regulations for non-road vehicles are also being tightened and are at similar levels as those for the passenger/commercial automobile industry. To reduce diesel engine emissions of exhaust gas pollutants, aftertreatment systems that utilize diesel oxidation catalysts (DOCs), diesel particulate filters (DPFs), and selective catalyst reduction (SCR) are being installed. However, it is expensive to fit non-road vehicles, which are used in harsh environments and have fast consumable replacement cycles, with such systems. As a result, gasoline engines for non-road vehicles are drawing attention as a replacement for diesel engines, owing to their lower emissions due to the strong emission reduction capability of a three-way catalytic converter (TWC). However, the gasoline engines in non-road vehicles have low power and torque at low loads. To compensate for this, it is necessary to develop a high-performance gasoline engine equipped with turbochargers.

Heywood introduced turbochargers as one of several ways to boost engine power. The turbine in a turbocharger uses the engine exhaust flow energy to power the compressor, which compresses air, increasing its density, and supplies it to the combustion chamber. The engine power increases as the high density air is combusted [1]. Arbab et al. used biodiesel to compare the results of turbocharged engines with those of non-turbocharged engines. Studies have shown that turbocharger engines have a higher thermal efficiency than non-turbocharger engines [2]. In addition, Fenely et al. introduced the variable geometry turbocharger (VGT) into a low-cost, high-efficiency booster and observed exhaust regulations, which are expected to play an important role in energy recovery from reciprocating engines [3]. To satisfy the corporate average fuel economy (CAFE) program by utilizing turbochargers, which can improve power and fuel efficiency, Palowski and Splitter compared the SI natural aspirated engine and turbocharger SI engine with an increased research octane number (RON). They confirmed the high efficiency of the turbocharger engine, which they concluded would become commonplace [4]. Turbochargers are the most common technology in diesel engines. However, consumers of gasoline engines tend to prefer nature-absorption gasoline engines owing to their high exhaust temperatures. It is difficult to enhance the durability of such engines; furthermore, they have a lower intake air volume and turbo lag at low RPM compared to diesel engines. However, to satisfy stricter regulations and the needs of consumers who desire high torque responsiveness even at low RPM, the installation of turbochargers is inevitable. Downsizing of the installation of an effective turbocharger and confirmation of the efficiency of various turbochargers was shown in the twin scroll turbine study of Yokoyama et al. [5] and the mono- and multi-scroll turbine comparative study by Walkingshaw et al. [6]. In both these studies, a high boost pressure, increased engine torque, and high turbocharger efficiency were confirmed at a low RPM. In addition, Zi et al. used a 1.5 L 4-cylinder gasoline turbo engine to separate the turbine and compressor, and studied an electric-booster and turbo-generator (EBTG) system driven by a motor to improve fuel efficiency and performance at low speeds and light loads. The combination of a gasoline engine and a turbocharger has been shown to be effective combination through various methods [7].

Gasoline engines equipped with turbochargers are steadily increasing in number. Gasoline engines are classified according to their fuel injection methods as port fuel injection (PFI) types that injects fuel into the port, direct injection (DI) types that inject fuel directly into the cylinder, such as a diesel engine, and dual port injection (DPI) types that combine both methods. A multi-point injection (MPI) engine that is categorized as a PFI type engine was utilized in this study; we also studied the Turbo MPI SI engine. Wang et al. conducted a 1D-simulation study for turbocharger matching using valve timing and an air-to-fuel ratio with spark timing to optimize the 1.4 L MPI turbo gasoline engine. Their results revealed that, with their proposed approach, parameters such as knocking and exhaust
temperature were optimized, the performance of the engine improved, and the accuracy of the conceptual design was improved [8]. Ducahaussoy and Barbier studied a 2 L MPI gasoline turbocharged engine and confirmed that as RON increased, knock resistance, emission reduction, and fuel consumption rate improved [9]. Zhang et al. used cooled EGR to comply with EURO 6 and confirmed a reduction in the nitrogen oxide emissions of an 11.6 L 6-cylinder CNG MPI turbo engine [10]. Then, in order to meet the Tailpipe CH4 and NH3 regulatory standards that change according to the engine load, they determined that an excess air ratio correction and ASC additional installation are necessary. In addition, Kim et al. researched liquid phase LPG injection (LPLI) of an LPG MPI engines using a CFD simulation for an 11 L single-cylinder engine. As a result, the turbulent intensity due to the squish area, the knocking tendency due to the change in the piston cavity, and the lean burn limit were confirmed [11]. These studies indicate that research on MPI turbocharged engines is being actively conducted.

However, of the engines that have been studied previously, the passenger/commercial car MPI engine has the disadvantage of being larger in size than other engines. For this reason, downsizing it is necessary to tighten regulation compliance and the same type of engine contrast lower displacement and higher power. The key to achieving an MPI turbocharged engine is downsizing. Nozawa et al., who conducted numerical simulations to evaluate friction loss and changes in fuel efficiency and performance through engine downsizing, confirmed that the fuel efficiency improved as the engine was downsized [12]. Shibata et al. developed a 1.0 L 3-cylinder turbo GDI engine that replaces the naturally aspirated 1.8 L engine of the Honda Civic compact car through engine downsizing, and confirmed a higher power than before and a fuel efficiency improvement of 20% or more [13].

In developing Fiat’s TwinAir engine, Mastrangelo et al. selected a 2-cylinder engine as the best method for fuel economy and low CO2 emissions. They compared the thermodynamic efficiency of different cylinder engines downsized below 1.0 L. They found that a high level of downsizing is possible with a low overall displacement when two cylinders are applied. Moreover, as a 2-cylinder engine has a relatively large single-cylinder volume, it was confirmed to have a high thermodynamic efficiency. Furthermore, they found that the performance of a downsized 2-cylinder engine with a turbocharger was equivalent to that of the NA engine with a displacement of more than 50%, and the fuel efficiency improved by 25% [14]. As described above, with a downsized turbocharged engine, the same or a higher output can be obtained even with a lower displacement. In addition, it is an environmentally friendly technology as it results in lower emissions of carbon dioxide and exhaust gases such as nitrogen oxides, and improves fuel efficiency.

Many operating parameters should be optimized and considered when downsizing an engine. Using a numerical approach, Eriksson et al. studied the downsizing modeling of a turbocharged SI engine for carbon dioxide reduction [15]. Utilizing a SAAB 2.3 L turbocharged gasoline engine to investigate engine components such as the throttle valve, turbine, and catalyst based on the mean value model, they performed carbon dioxide reduction experiments, verified the downsized model, and explained the engine component variables. Ahmadi optimized the variable valve timing (VVT) of a 4-cylinder MPI engine via a 1D simulation and genetic algorithms (GA) in relation to valve timing, which is one of the constraints and operating variables in engine design [16]. In addition, Su et al. considered spark timing as the main variable while conducting research related to the hydrogen–gasoline rotary engine (HGRE), which is a dual-fuel engine [17].

With the hydrogen concentration and settings fixed, timing changes and proceeded with research on engine combustion and emission characteristics with changes in hydrogen concentration at a fixed spark timing. For diesel engines, Panda and Ramesh considered the injection strategy as the primary variable, and studied the effects of methanol–diesel dual-fuel combustion by single pulse injection (SPI), pilot and main injection (PMI), and post injection along with pilot and main injection (PMPI) injection strategies [18]. It was confirmed that the application of pilot injection improves the combustion rate of double fuel, and PMPI reduces combustion stability and emission of exhaust pollutants owing
to the influence of pilot injection and post injection. To optimize engine combustion and operation, engine downsizing was performed while considering various parameters such as the spark timing, valve timing or valve overlap, injection strategy, compression ratio, and engine components.

Previous studies have mainly utilized field experimental methods for engine downsizing and turbocharger matching. However, owing to the characteristics of an engine, there are many variables with a wide range of operating conditions to consider, and there are limits in field experiments. To complement this, numerical analysis through one-dimensional (1D) and three-dimensional (3D) simulations was used. Park et al. predicted the performance of a diesel–natural gas dual-fuel engine through a 1D simulation. Injection timing, EGR rate, and variable compression ratio (VCR) were selected as predictors, and the maximum torque point, minimum BSFC, and minimum NOx emission design point were presented using the Latin hypercube sampling (LHS) and Pareto methods [19]. Park et al. used a 1D simulation-based design of experiment (DoE) to evaluate NOx emissions for diesel–methane dual-fuel engine performance under various load conditions. They also presented an engine control strategy optimized using Pareto technology that can improve both the BSFC and NOx emissions [20]. Liu and Dumitrescu used 3D simulations to study the combustion phenomenon of natural gas caused by the piston crown shape of existing diesel engines and converted conventional diesel engines to SI-type natural gas engines. Using the G-equation-based RANs model, they predicted the spark timing, equivalence ratio, engine speed-related combustion phenomena, and natural gas operating conditions, and the predicted performance and emissions effects were shown to be in agreement with the experimental results [21]. Numerical studies employing 1D and 3D simulations have been performed as they are more effective in terms of time and cost than experiments, and can improve the accuracy of experiments by providing predictions.

The target engine of this study is for non-road vehicle system. According to the current emission regulations targeting non-road vehicles, the NOx regulation is not hard, but it is necessary to look at the NOx tendency in preparation for the upcoming post stage 5 regulation. Currently, the non-road gasoline engine can satisfy the non-road vehicle emission regulation without installing a three-way catalytic converter (TWC) while passenger vehicle should install TWC. This study is a prior study to prepare for post stage 5 in the process of converting an existing automobile engine to a non-road engine.

The main purpose of the present study is to derive trends according to industrial support during conceptual design in the process of off-road engine production manufactured by OEMs. The combustion of the conventional MPI SI engine adopts the MBT optimal method, but the target engine of this study uses the retarded spark timing compared to the MBT as a design guide, not the MBT optimal method. The reason is that the knocking problem becomes severe in the MBT area as it is based on the turbocharged under MPI layout. When a turbocharger is used for a gasoline engine, direct injection is usually adopted, owing its hard auto-ignition. Nevertheless, the reason for using the turbocharger in the MPI method in this study is to recycle the engine by re-purposing it to an off-road engine through retrofit of an existing passenger engine.

In this study, a 1D simulation was used to determine the optimum operating point of the downsized 800CC gasoline MPI turbo engine of the target engine. The SI turbulent flame model, which can predict the driving performance and exhaust emissions, was used as the 1D simulation model. After finding the MBT point using the DoE method, optimization was performed using the selected operation parameter. The operation parameters we selected were the spark timing, intake valve overlap, and boost pressure, which affect performance. Then, the optimum design points of the target engine were determined by predicting the driving performance of each engine speed (RPM) and the point at which the Knock phenomenon occurred. In addition, by using the multi-objective Pareto technique, the optimization proceeds against the performance and exhaust gas emissions that can comply with environmental regulations in non-road vehicles.
2. Methodologies

The process for optimizing the MPI turbo SI engine of a non-road vehicle is shown in Figure 1. The engine used in this study had two intake valves and two exhaust valves in one cylinder. The study was conducted with a gasoline-based 2-cylinder MPI Turbo SI engine composed of an intercooler, an intake manifold, a WGT, etc. The detailed engine specifications are listed in Table 1. The optimization process is described in detail in this section.

![Overall process of the numerical study for non-road MPI Turbo SI engine.](image)

**Figure 1.** Overall process of the numerical study for non-road MPI Turbo SI engine.

| Item                        | Specification                  |
|-----------------------------|--------------------------------|
| Engine volume (cc)          | 798                            |
| Cylinder arrangement        | 2cyl., I-type                  |
| Bore -Stroke (mm)           | 77.4–84.8                      |
| Compression ratio           | 9.2                            |
| Connecting rod length (mm)  | 128.4                          |
| Wrist pin to crank offset (mm)| 10                             |
| Firing intervals (° CA)     | 360                            |
| Injection type              | MPI                            |
| Max. torque@ RPM            | 124.5 N-m@2500 RPM             |
| Max. power@ RPM             | 56.8 kW@5500 RPM               |

2.1. Experimental Set-Up

The experiments were conducted to obtain input data before proceeding with the 1D modeling. Figure 2 shows a schematic of the device used in the experiment. The experimental equipment consisted of two main parts: the engine control part and the engine performance and exhaust gas analysis parts. The engine control system consisted of an AC dynamometer that was enclosed by the rest of the components, i.e., the intercooler (AVL ConsysBoost 3000), fuel (AVL FuelExact PLU 500), oil (AVL ConsysLube 50), and coolant (AVL ConsysCool 450) control system. The experiments were conducted using a...
combustion indicator (AVL Indiset Advanced Gigabit) and an exhaust gas analyzer (AVL AMA i60) installed for exhaust gas analysis and performance analysis. The experiments were carried out with a wide-open throttle (WOT) state in a total of 10 sections at engine speeds varying from 1600 RPM to 5500 RPM, and data for the 1D modeling was obtained. The testing was performed by the Korea Automotive Technology Institute, KATECH.

![Figure 2. Schematic of the engine testing device from KATECH, South Korea.](image)

Table 2 summarizes the operating conditions used in the experiments.

| Case No. | RPM  | IMEP (Bar) | Boost Pressure (Bar) | Spark Timing (deg) | Maximum Intake Valve Opening Position (Deg) | Peak Cylinder Pressure (Bar) | Brake Torque (N-m) | Pareto Solution |
|----------|------|------------|----------------------|--------------------|---------------------------------------------|----------------------------|-------------------|-----------------|
| 1        | 1600 | 17.7       | 1.648                | 13.5               | 469                                         | 53.8                       | 101.7             | ✓               |
| 2        | 1800 | 19.9       | 1.814                | 12.8               | 469                                         | 59.4                       | 113.4             | ✓               |
| 3        | 2000 | 20.1       | 1.796                | 9                  | 469                                         | 60.5                       | 115.4             | ✓               |
| 4        | 2500 | 21.8       | 1.819                | 5.4                | 459                                         | 69.2                       | 124.5             | ✓               |
| 5        | 3000 | 20.6       | 1.759                | 2.3                | 459                                         | 66.8                       | 121.3             |                 |
| 6        | 3500 | 20        | 1.792                | −6                 | 459                                         | 72.6                       | 114.4             |                 |
| 7        | 4000 | 19.6       | 1.846                | −8.6               | 459                                         | 75.5                       | 110.8             |                 |
| 8        | 4500 | 19.5       | 1.851                | −11.3              | 459                                         | 76.3                       | 109.0             |                 |
| 9        | 5000 | 18.9       | 1.844                | −14.3              | 459                                         | 78.9                       | 103.9             |                 |
| 10       | 5500 | 18.5       | 1.850                | −17.3              | 459                                         | 81.2                       | 98.7              |                 |

(−) advance.
2.2. Detailed 1D Modeling

1D modeling was performed using data obtained from the engine experiment. The program used for 1D modeling was the GT-suite, which is a conceptual design commercial program.

Figure 3 shows a 1D-model map depicting the MPI turbo SI engine using the GT-suite. The experiment used engine speed (RPM), spark timing, boost pressure, and intake valve overlap as the engine operating parameters. The specifications required for modeling, such as the piston head, spark location, connecting rod length, valve lift and diameter, compression ratio, and coefficient of friction that make up the engine, which were provided by the engine manufacturer, were applied to the cycle simulation. Based on the constructed 1D model, thermodynamic properties in each duct and pipe, combustion behaviors and mechanical behaviors are matched to find model accuracies under each operating conditions shown in Table 2. Especially for FMEP (Friction mean effective pressure), well-known Chen-Flynn model was used to calculate engine friction.

![1D-model map](image)

Figure 3. 1D-model map.

Optimization was performed by selecting engine speeds of 1600 RPM, 1800 RPM, and 2500 RPM, which are the operating conditions utilized in practice. Representative engine performances are calculated by governing equations provided from GT-suite and the well-known equations related to torque, power, and BSFC, etc. can be found from Ref. [1].

Numerical Study

The combustion model used in this study did not use the Wiebe model, which is a non-predictive model, but the SI turbulence model, which is a predictive model. As the Wiebe model is interpreted in a state where 10–90% burn duration and 50% burn point are fixed,
the input Wiebe parameters required for numerical analysis must be measured according to the spark timing. However, as the purpose of this study is to predict performance and emission by tracing the retarded spark timing that can avoid knocking in reverse, the spark timing itself should be used as a variable.

The SI Turb model predicts the in-cylinder combustion rate, exhaust gas, and knocking phenomenon of a spark-ignition engine. Furthermore, the SI Turb model divides the cylinder volume into two zones, a burned zone and an unburned zone, and at each time step, the fuel and air mixture moves to the combustion zone from the unburned zone. The following process was used to perform the combustion calculation in the GT-suite [22]. Detailed calculation processes can be found in our previous work [23].

In the two-zone model, governing equations for energy balance are solved for each time step as follows:

\[
\frac{d(m_u e_u)}{dt} = -p \frac{dV_u}{dt} - Q_a - \left( \frac{dm_f}{dt} h_f + \frac{dm_a}{dt} h_a \right) + \frac{dm_f}{dt} h_f, \text{for the unburned zone; (1)}
\]

\[
\frac{d(m_b e_b)}{dt} = -p \frac{dV_b}{dt} - Q_b + \left( \frac{dm_f}{dt} h_f + \frac{dm_a}{dt} h_a \right) \text{for the burned zone. (2)}
\]

If you look to the right of Equation (1), the energy equation for the unburned zone, four terms can be seen. Each term represents pressure work, heat transfer, combustion, and the addition of empty from injected fuel, respectively. The third term includes the instantaneous rate of fuel combustion or the burn rate. In the combustion of SI engines, the main characteristic of combustion gas mixtures composed of fuel and air is the laminar flame speed. The stratification velocity refers to the relative velocity at which unburned gas enters the front of the flame and changes to the product under the stratification flow condition. The speed of the laminar flame is given by Equation (3).

\[
S_L = \frac{dm_b}{dt} A_f \rho_u
\]

The laminar flame speeds of methanol, propane, isooctane, methanol, ethanol, gasoline, and hydrogen were measured using the pressure and temperature generated inside the engine and at the equivalent ratio. Equation (4) can then be derived using the power law.

\[
S_L = S_{L,0} \left( \frac{T_u}{T_{ref}} \right)^a \left( \frac{P}{P_{ref}} \right)^\beta
\]

The GT-suite program was derived from Equation (4).

\[
S_L = \left( B_m + B_\Theta (\varphi - \varphi_m)^2 \right) \left( \frac{T_u}{T_{ref}} \right)^a \left( \frac{P}{P_{ref}} \right)^\beta f(\text{dilution})
\]

In this study, \( f(\text{dilution}) = 1 - 0.75 \times DEM \left( 1 - \left(0.75 \ DEM \ Dilution \right)^7 \right) \).

The exhaust gas analysis was also performed during fuel combustion. The mechanism of NO generation is as follows:

\[\begin{align*}
N_2 + O &= NO + N \text{ (for the } N_2 \text{ oxidation rate)} \\
N + O_2 &= NO + O \text{ (for the } N \text{ oxidation rate)} \\
N + OH &= NO + H \text{ (for the } OH \text{ reduction rate)}
\end{align*}\]

This mechanism was used to predict NO formation in the SI Turb model as an extended Zeldovic mechanism [23].
2.3. Knock Analysis

The knock phenomenon is the most important consideration when optimizing Turbo MPI engines for non-road vehicles, as it is the main cause of damage to the spark ignition engine. Empirical formulations based on an Arrhenius function were used to predict the knock phenomenon [24]. Based on this Arrhenius function, Equation (9) was obtained by matching the data measured over the pressure and temperature range of a given fuel–air mixture.

\[ \int_{t=0}^{t_1} \frac{1}{\tau} dt = 1 \]  

Equation (9) is referred to as the induction time integral or Livengood and Wu’s auto-ignition integral, and is expressed as Figure 4 when knocking occurs [25]. At this time, \( \tau \) is the induction time at the instantaneous temperature and pressure of the mixture, \( t \) is the end-gas compression start \((t = 0)\), and \( t_i \) is the auto-ignition time. Equation (9) shows that, in a given mixture, the induced period chemical reaction is possible only in a gaseous state with the total production rate of important chemical species. It is obtained under the assumption that the concentration of important chemical species is constant at the start of auto-ignition. Equation (10) is obtained through various experimental formulas for hydrocarbon or mixed fuels for a specific induction time.

\[ \tau = A p^{-n} \exp \left( \frac{B}{T} \right) \]  

Figure 4. Two combustion zones and the induction time integral equation.

\( A, n, \) and \( B \) are the fuel-dependent parameters. The GT-Suite, a simulation program used for engine optimization in this study, uses three induction times to capture and predict the different chemistries of auto-ignition at a wide range of temperatures based on a detailed kinetics simulation. We also utilized the gasoline kinetic reaction mechanism
described by Equation (11) that was devised by Ra and Reitz and which focuses on gasoline combustion [26].

\[
\tau_i = M_1 a_i \left( \frac{\text{ON}}{100} \right)^{b_i} \left[ \text{Fuel} \right]^{c_i} \left[ \text{O}_2 \right]^{d_i} \left[ \text{Diluent} \right]^{e_i} \text{Exp} \left( \frac{f_i}{M_2 T} \right) \quad i = 1, 2, \text{ and } 3
\]  

(11)

\( M_1 \) is the multiplier of knock induction time, \( \text{ON} \) is the octane number of fuel, and \( M_2 \) is the multiplier of activation energy. [Diluent] is the concentration of the diluent, which could be \( \text{N}_2, \text{CO}_2, \text{ or } \text{H}_2\text{O} \), and is expressed in mol/m\(^3\), as are [Fuel] and [O\(_2\)].

Table 3 shows the model constants \( a_i \) to \( f_i \) used to obtain the induction time. The overall induction time \( \tau \) is given by Equation (12):

\[
\frac{1}{\tau} = \frac{1}{\tau_1} + \frac{1}{\tau_2} + \frac{1}{\tau_3}
\]

(12)

Table 3. \( a_i \) to \( f_i \) F model constants.

|   | A           | b       | c       | d       | e       | f       |
|---|-------------|---------|---------|---------|---------|---------|
| 1 | 4.445 \times 10^{-7} | 3.613   | -0.64   | -0.564  | 0.3978  | 12,920  |
| 2 | 11,941,423  | 3.613   | -0.64   | -1.4596 | 0.4867  | -1957   |
| 3 | 8.905 \times 10^{-7} | 0       | -0.25   | -0.547  | 0       | 166,856 |

When the induction time is obtained through Equation (12), and the induction–time relationship is calculated by Equation (9), it can be confirmed whether auto-ignition occurs at the point where the result value is 1. The prediction of the knock phenomenon through this approach was confirmed to be almost similar to the data obtained by experiments, avoiding the range of the spark timing and the point of abnormal combustion where knocking occurred. It is possible to confirm a high prediction for knocking.

2.4. Design of Experiment and Multi-Objective Pareto Optimization

Design of experiment (DoE) is a statistical technique used to study the effects of several parameters simultaneously. Owing to time and cost limitations, it is not possible to study all the characteristics of an engine under various conditions in a field experiment. However, if research is conducted using the DoE technique, the experimental constraints can be minimized and predicted under wider conditions. In this study, the relationship between the operational parameters was verified and optimized using a full factorial design. This is the theoretical basis for all experimental planning methods, and is frequently used for optimization and parameter selection. In addition, in the reference model, the knocking area, which is an area that experiences an abnormal combustion phenomenon, was confirmed by sweeping the relationships of the parameters. Subsequently, the optimization ranges of the operation parameters, i.e., the spark timing, boost pressure, and intake valve overlap, were selected, and the final DoE was performed.

Table 4 lists the range of the dominant variables for optimization. When a response model, such as brake torque and BSFC, is obtained through DoE, multi-objective optimization may be performed. Multi-objectives (hereinafter referred to as MO), which require optimization, may be formulated as follows: [27–29]
Table 4. Input variables and ranges.

| Input Variables            | 1600 RPM | 1800 RPM | 2500 RPM |
|----------------------------|----------|----------|----------|
| Spark Timing [CA ATDC]     | 12–15    | 15.5–17.5| 8.5–10.5 |
| Intake valve overlap [CA]  | 460–474  | 461–476  | 449–471  |
| Boost pressure [bar]       | 1.65–1.85| 1.8–2.0  | 1.8–2.0  |

DoE

Minimize

\[
\vec{y} = \vec{F}(\vec{x}) = \begin{bmatrix} f_1(\vec{x}), f_2(\vec{x}), \ldots, f_n(\vec{x}) \end{bmatrix}^T
\]

(13)

\[
g_j(\vec{x}) \leq 0, \ j = 1, 2, \ldots, M
\]

(14)

\[
\vec{x} = [x_1, x_2, \ldots, x_p]^T \in \Omega
\]

(15)

Here, \(\vec{y}\) represents the objective vector, \(g_j\) represents the constraints, and the \(P\)-dimensional vector represents the determinants within the \(\vec{x}\) parameter space \(\Omega\). The ideal optimization for the overall objective is expressed in Equation (13):

\[
\vec{x}_0^* \in \Omega : \forall \vec{x} \in \Omega, f_i(\vec{x}_0^*) \leq f_i(\vec{x}), \text{for } i \in \{1, 2, \ldots, N\}
\]

(16)

However, in the case of engines, results that satisfy all the desired conditions cannot be obtained owing to the very many variables that need to be considered, and there are trade-off relationships. Therefore, the Pareto optimization technique devised by Francis Ysidro and generalized by Vilfredo Pareto was used in this study. The decision vector \(\vec{u} = [u_1, u_2, \ldots, u_p]^T\) is referred to as Pareto-dominant when decision vector \(\vec{v} = [v_1, v_2, \ldots, v_p]^T\) is a minimization context expressed as follows:

\[
\forall i = \{1, \ldots, N\}, f_i(\vec{u}) \leq f_i(\vec{v}), \text{and } \exists j \in \{1, \ldots, N\} : f_j(\vec{u}) < f_j(\vec{v})
\]

(17)

The Pareto front shown in Figure 1 can be checked, and the most suitable optimization result value can be selected. A scalarization method such as the weighted linear sum or \(\epsilon\)-constraints can be used to optimize the MOs. Recently, there has been a trend towards using the multi-objective genetic algorithm (MOGA) as an optimization method. Inspired by evolutionary biology, this algorithm finds approximations for the Pareto optimal sets using a multi-point search method. The MOGA is applied in various fields, and methods that combine it with simulations are frequently used in the engineering field [29,30]. In this study, Pareto multi-objective optimization was performed based on the MOGA. Detailed optimization processes using multi-objective Pareto optimization based on design of experiment can be found in our previous research [19,20].

3. Results and Discussion

3.1. Comparison of the 1D Simulation and Experimental Results

A reliable reference 1D model is required to optimize the 800cc gasoline MPI turbo SI engine for non-road vehicles. The reference 1D models for use in the study were modeled using specifications and engine test results provided by the original equipment management (OEM). In the WOT state, 10 engine speeds from 1600 to 5500 RPM were simulated with an error range of \(\pm 5\%\). Figure 5 shows a comparison of the simulation data and engine test data. The items on the left are the torque, power, and IMEP, which represent the main performance metrics of the engine. It was confirmed that the main performance simulation data matched the engine experimental data within \(\pm 5\%\). The items on the right
are the boost pressure, cylinder pressure, and turbine inlet temperature (TIT) from the thermodynamic perspective of the engine. It was confirmed that all the comparison items had engine experimental data and results within ±5% of the error range. Furthermore, all the comparison items had simulation results within ±5% of the engine experimental data error range. Based on these results, the reliability of the reference model was determined. Here, the error bar graph was used to show that the simulation data, not the error range of the experimental data, was approximately ±5%. The operation section was selected using the validation results. As for the selection conditions, the driving section to be optimized was selected based on the multi-frequency driving area of the non-road vehicle and the driving section with a comparable A/F ratio for each RPM. Unlike commercial/passenger vehicles running on the road, non-road vehicles operating in rough terrain, such as soil and mountainous terrain, are more likely to run at low RPM than at high RPM; therefore, the optimal driving sections were selected as 1600, 1800, and 2500 RPM. The experimental conditions for the selected RPM in Table 2 were confirmed.

3.1.1. Effect of the Spark Timing

In SI engines, the spark timing is the main variable of the engine and has a significant influence on the initial combustion state, thermal efficiency, and abnormal combination. The crank angle at which the performance, pollutant discharge, and knocking occurred was confirmed at various spark timings. The spark timing of the selected sections, i.e., 1600, 1800, and 2500 RPM, was a DoE sweep in 66 sections corresponding to −40° to 25° CA (aTDC) at intervals of 1° CA.

Figure 6 represents the effect of spark timing on engine performance are introduced at each rpm with fixed boost pressure and valve overlap shown in Table 2 which are given for base operating conditions.
Figure 6. (a) Knocking occurrence, torque change, and NOx emission amounts according to changes in spark timing at 1600 RPM. (b) Knocking occurrence, torque change, and NOx emission amounts according to changes in spark timing at 1800 RPM. (c) Knocking occurrence, torque change, and NOx emission amounts according to changes in spark timing at 2500 RPM.

Figure 6 shows the variations in the maximum brake torque (MBT) point and torque change tendency, NOx emission, and knock onset prediction tendency for each RPM at various spark timings.

Figure 6a shows the result of sweeping the spark timing at 1600 RPM. The effect of the change in the spark timing on the brake torque was confirmed. The MBT point timing at 1600 RPM was $-1^\circ$ CA (aTDC), and the torque was 115.2 N-m. The MBT points were determined as it is the standard technique for finding the optimal spark timing with the SI engine’s traditional combustion optimization method. After finding the MBT point, the
experimental results, which are marked with green asterisks, and DoE results obtained by simulation were compared. For 1600 RPM, a spark timing of 13.5° CA (aTDC) and a torque of 101.7 N·m were obtained experimentally. The experimental torque value matched the sweep DoE result. This indicates that the 1D simulation map calibration is a reliable model and is similar to the engine used in the experiment. In addition, the experimental spark timing was very slow compared to the MBT point timing. Unlike existing natural aspiration (NA) engines, engines equipped with turbochargers can realize a high performance by achieving a considerable boost pressure in WOT operation by retarding the spark timing. Another reason for the retarded spark timing is the fast catalyst light-off during cold engine start-up; however, the most important reason is that knocking can be avoided. As mentioned earlier, the GT-suite program can predict the knock phenomenon using Equation (9) for the induction time integral. The knock-onset result in Figure 6a is the result of predicting the knocking timing that occurs according to the change in spark timing. In the case of 1600 RPM, knocking occurs from −24° CA to 6° CA (aTDC) and is highlighted by red shading. Furthermore, the knock onset crank angle mainly occurs during flame propagation, the main combustion section of the SI engine combustion process. Based on the prediction results, acceptable spark timing in which knocking did not occur is indicated by the blue shading. The timing ranges from 7° CA to 21° CA (aTDC). To select the spark timing for engine optimization, the cylinder pressure was set as a constraint. The motoring combustion pressure of the 1D simulation was 39 bar, and spark timings after 21° CA (aTDC) with a pressure lower than the motoring combustion pressure were excluded from the optimization range. The effect of the change in spark timing on the amount of NOx was also confirmed. The minimum NOx emission tendency was predicted at 3° CA advanced timing compared to the MBT point. In addition, the amount of NOx was unstable after 21° CA, which is believed to be due to excessive retard timing and incomplete combustion as the piston in the cylinder approaches the BDC. In this way, in Figure 6a, the ranges of the spark timings for the other RPMs were selected by applying the criteria for finding spark timings that can be optimized. Figure 6b shows the effect of changes in spark timing at 1800 RPM on torque, NOx, and knocking. As shown in Figure 6a, the tendency of the torque is consistent with the experimental data. The MBT timing at 1800 RPM is 1° CA more advanced −2° CA compared to that at 1600 RPM, and torque has a value of 130.7 N·m, an increase of 13.45%. The range of timing values at which knocking occurs is from −28° CA to 13° CA (aTDC), which is predicted to occur between −36° CA and 6° CA (aTDC), which is 35.48% wider range. The spark timing applicable to engine optimization at 1800 RPM ranges from 7° CA to 21° CA (aTDC) and is 40% less than that at 1600 RPM. The minimum amount of NOx was obtained at −6° CA, a point 4° more advanced than the MBT point. It rises from 18° CA and shows an unstable appearance, which confirms that incomplete combustion occurred due to excessive retardation of spark timing, as shown in Figure 6a. The sweep result of the spark timing at 2500 RPM is shown in Figure 6c. The MBT point has a value of −8° CA and a torque of 144.7 N·m. Comparing the experimental data with the DoE data, it was confirmed again that the simulation data, as shown in Figure 6a,b, were consistent with the experimental data. In Figure 6c, the MBT point advanced 7° CA compared to that in Figure 6a, and the torque increased by 25.6%. The knocking area was predicted to occur between −36° CA and 6° CA (aTDC), which confirmed that knocking occurred in a 2.38% wider range than the 38.7% wider range in Figure 6a. It was confirmed that the knocking area increased as the RPM increased. The available spark timing for optimization ranged from 7° CA to 13° CA (aTDC). A timing that is 7° CA is less than that of Figure 6a and 2° CA less than that of Figure 6b can be used. It was predicted that the minimum amount of NOx was obtained at −14° CA with a 6° advance at the MBT point. In addition, it was confirmed that the timing section judged to be incomplete combustion, which could be inferred from the results of NOx, has a wider range of 8° CA compared to those in Figure 6a,b. It may be seen that as the RPM increases, the retarded spark timing does not help complete combustion in the cylinder. As shown in
3.1.2. Effect of the Boost Pressure

The engine used in this study was a gasoline SI engine equipped with a turbocharger. The turbo gasoline engine provides a boost pressure higher than the atmospheric pressure, increasing the amount of incoming air and therefore increasing the amount of fuel that can be burned. This has the advantage of increasing the maximum torque of the engine and enables a maximum output to be realized at a low RPM. However, under these conditions, knocking occurs. In this section, the effect of boost pressures and spark timing on engine performance are introduced at each rpm with valve overlap as shown in Table 2, which are given for base operating conditions.

To select the optimization range, the effect of variations in the spark timing and boost pressure on the occurrence of knocking was determined. To predict the occurrence of knocking, DoE was applied, together with the spark timing obtained through a numerical simulation and boost pressure selected based on experimental data. Subsequently, the phenomena associated with changes in each variable were predicted.

Figure 7 shows a contour of knock onset at a speed of 1600 RPM. The spark timing range was $7^\circ$ CA to $21^\circ$ CA (aTDC), and the boost pressure was DoE at 0.5 bar intervals from 1.65 bar to 2.2 bar.
As shown in Figure 7a, it is possible to confirm the range of the allowed boost pressure and knock onset crank angle for the spark timing. In the GT-suite, the crank angle at which knocking does not occur is expressed as 0. The left area is the knock onset section based on 0, and the purple area is the area where knocking does not occur. For a spark timing of 7° CA, it was predicted that knocking occurred when the boost pressure exceeded 1.65 bar. It was confirmed that no knocking occurred at a spark timing of 17° CA, which is 10° CA retarded, at all the boost pressures. Figure 7a confirms, that when the spark timing was retarded by 1° CA, knocking did not occur, and the allowable boost pressure increased by 0.5 bar. However, the optimal spark timing for each of the boost pressures cannot be selected when selecting the range of optimal variables for spark timing and boost pressure. The spark timing and boost pressure ranges were selected based on NOx so that a good torque response and low emissions could be obtained even at low RPM for turbo gasoline engines that are applied to non-road vehicles. Figure 7b shows a contour of the variation in the NOx emissions with variations in the spark timing and boost pressure at 1600 RPM. When checking the amount of NOx generated after a spark timing of 19° CA (aTDC), it can be seen that the change width exhibits a rapidly changing trend. It can be determined that, irrespective of the effect of boost pressure, the combustion in the cylinder is incomplete owing to the excessively retarded spark timing. Therefore, the spark timing from 19° CA to 21° CA (aTDC) was excluded from the optimization range. Figure 7c shows the variation in the torque with changes in the two variables. In selecting the section, the lowest torque limit was set to 96.62 N-m or higher. This was −5% of the experimental value, which was 101.7 N-m. It was confirmed that, from 11° CA to 15° CA (aTDC), generating a high torque at the lowest boost pressure was possible. In addition, as it is possible to confirm that the higher the boost pressure, the higher the torque, it was confirmed that the spark timing of 12° CA (aTDC) is allowed by checking the section allowing the boost pressure to be higher than the experimental value of 1.65 bar. The optimization range at 1600 RPM was selected based on the results shown in Figure 7a–c. The optima spark timing range was from 12° to 15° CA (aTDC), and the optimal boost pressure range was from 1.65 to 1.85 bar.
Figure 8 shows a contour that predicts the knock onset that appears according to variations in the spark timing and boost pressure. The range of the sweep spark timing was 13.5° to 21° CA (aTDC), and the boost pressure proceeded with DoE at 0.5 intervals in the range of 1.8 bar to 2.2 bar.

Figure 8. (a) Predicting knocking phenomenon according to changes in spark timing and boost pressure at 1800 RPM. (b) Predicting NOx amounts according to changes in spark timing and boost pressure at 1800 RPM. (c) Predicting torque tendency according to changes in spark timing and boost pressure at 1800 RPM.
In Figure 8a, the knock onset area is on the left side of the purple area. The allowable boost pressure increased by 0.5 bar each time the spark timing was retarded by 1° CA from a spark timing of 14° CA, at which knocking does not occur at the experiment boost pressure value of 1.8 bar. A similar tendency was observed at 1600 RPM. The spark timing range at which all the pressures in the range 1.8–2.2 bar set in the DoE range are allowed is from 17.5° CA to 21° CA (aTDC). However, based on Figure 8b, combustion is unstable after a spark timing of 18° CA (aTDC). Based on this, the spark timings after 18° CA are not suitable as optimization variables. The results of the torque to be confirmed by other considerations are shown in Figure 8c. The lowest torque limit criterion was set as 107.73 N-m, which is −5% of the error range of the experimental torque value, 113.4 N-m. In Figure 8b, the range of the optimization variables was confirmed, except for the timing after a spark timing of 18° CA, which was confirmed by the NOx amount. It can be seen that a suitable spark timing range is 15–16° CA (aTDC). In the range of spark timing with the limit torque value set in Figure 8c, the range of the acceptable boost pressure is narrow. Therefore, although it is lower than 107.73 N-m, which was set as the experimental minimum torque limit standard, the timing value of 18° CA was allowed, as it can be supplemented by an increase in boost pressure. By synthesizing this, an optimization range for 1800 RPM was selected. The optimal spark timing was selected from 15.5° CA to 17.5° CA (aTDC), and the optimal boost pressure was selected from 1.8 bar to 2 bar as the range of variables for optimization.

Figure 9 shows the contour for 2500 RPM. The DoE section was set to a spark timing of 6° to 13° CA (aTDC) and boost pressure of 1.8 to 2.2 bar.

Figure 9 shows the knock onset contour that appears according to changes in the spark timing and boost pressure at 2500 rpm. The purple area represents the area where knocking does not occur, while the left area represents the area where knocking occurs at various spark timings and boost pressures.
Similar to other RPMs, the allowed pressure tended to increase by 0.5 bar as the spark timing was 1° CA retarded. Figure 9b shows an unstable NOx emission after a spark timing of 11° CA (aTDC), confirming that combustion in the cylinder was not performed properly. This spark timing range can be allowed up to boost pressure of 2.2 bar, but it can be confirmed that the range after 11° CA is not suitable for selecting an optimized range. The optimization range was selected from the remaining spark timings, as confirmed in Figure 9b. Figure 9c is a contour that confirms the tendency of the torque to change. At this time, the lowest torque criterion was set to 118.28 N-m, which is −5% of the error range of the 2500 RPM experimental torque value of 124.5 N-m. As a result, the range of spark timing between 8.5° and 9.5° CA (aTDC) could be selected. However, both the timing and boost pressure were narrow enough to be selected as the range of optimization variables; for example, at 1800 RPM, a spark timing of 10.5° CA could be selected, even though it exceeds the limit based on the boost pressure. The ranges 8.5–10.5° CA (aTDC) and 1.8–2 bar were selected as the optimization variable ranges for spark timing and boost pressure, respectively, at 2500 RPM.

3.1.3. Effect of Intake Valve Opening Position

In this study, the intake valve opening position was also selected as an optimization variable for the MPI turbo gasoline engine. The valve overlap may control the combustion stability and pollutant discharge by controlling the gas flow and the amount of residual combustion gas in the cylinder. In addition, valve overlap has a significant impact on combustion efficiency and engine performance, such as increasing the effective expansion ratio, reducing pumping loss, and increasing torque at low RPM. Such a valve overlap is important in downsized engines. The engine in this study was a downsized MPI turbo gasoline engine that adopted a variable valve.

In this section, the effect of intake valve opening positions on engine performance are introduced at each rpm with fixed spark timing and boost pressure shown in Table 2 which are given for base operating conditions.

To vary the intake valve opening and closing event, maximum intake valve opening positions were treated as a parameter. For optimization, the timing of the intake valve was adjusted to determine the changes in torque and NOx through overlap with the exhaust.
valve, and then the optimization range of the valve overlap was selected. The expression, “feasible region” in Figure 10, represents the allowable intake valve opening positions where similar torque can be obtained compared to torque level and lower NOx emission can be found compared to NOx from base engine operating conditions.

![Figure 10](image)

**Figure 10.** (a) Torque and NOx tendencies according to changes in IVO at 1600 RPM. (b) Torque and NOx tendencies according to changes in IVO at 1800 RPM. (c) Torque and NOx tendencies according to changes in IVO at 2500 RPM.
After fixing the base conditions, valve lift, and exhaust valve closed (EVC) of the reference model that was used to carry out the validation, DoE proceeded by adjusting only the intake valve opening (IVO) timing. Based on the experimental value of the maximum intake valve opening position (MOP) of the selected RPM, 80 sampling results were confirmed by advancing and retarding 40° CA before and after.

The results for 1600 RPM are shown in Figure 10a. DOE was performed from MOP 429° to 509° CA, and torque changes were confirmed. The torque of the reference MOP 469° CA indicated with a green asterisk is 101.7 N-m, and the overlap interval is 64° CA. As a result of the DoE simulation, it was confirmed that the tendency of the sweep torque and that of the experimental values were consistent. The MBT point was identified as a reference point for selecting the valve range. The MBT point was 455° CA, which was 14° CA more advanced than the reference point, and the torque was 110.69 N-m. It can be confirmed that the torque increases as the overlap section becomes wider than the reference, but the torque tends to decrease after the MBT point. It is judged that if the IVO timing advances more than the MBT point, combustion in the cylinder is unstable owing to the effect of the wider overlap, resulting in a decrease in torque. In the case of a non-road vehicle, the responsiveness of the torque should be good, even at low speeds. Therefore, the maximum overlap limit was set as 455° CA, which is the MOP of the MBT point. Based on the experimental torque, the limit of the minimum overlap was determined to be 474° CA with a torque of 96.62 N-m or more, which is a 5% error range. The amount of NOx generated had a minimum value at MOP 479° CA, which is 10° CA retarded compared to the reference value, MOP 469° CA. The overlap section is a 54° CA interval, which is 10° CA lower than the existing 64° CA, which acts as a diluent such as EGR due to the increase in the residual gas friction in the cylinder as the interval of the overlap decreases. As a result, during combustion, the temperature of the cylinder is lowered, and the NOx emission is reduced. However, MOP 479° CA was excluded as it did not reach the minimum overlap limit of 96.62 N-m. Therefore, MOP 474° CA, which satisfies the torque limit standard, was selected as the minimum overlap. Conversely, the overlap interval widens, which is considered to increase the combustion temperature and increase the emissions of NOx as fresh air pushes out the residual gas in the cylinder, which acts as a diluent as the IVO timing is advanced. Torque is also important; however, NOx emissions must be minimized. Therefore, MOP 460° CA, where the amount of NOx increased rapidly, was set as the maximum overlap limit. Taken together, the optimization range of 1600 RPM is a total of 14 sections from MOP 460° CA with a maximum overlap interval of 72° CA shown in shades of blue to MOP 474° CA with a minimum overlap interval of 54° CA. Figure 10b shows the intake valve overlap DoE result of 1800 RPM. 1800 RPM uses MOP 469° CA with the same overlap interval of 64° CA such as 1600 RPM and the torque is 113.4 N-m. As shown in Figure 10a, it was confirmed that the prediction of the simulation results was accurate when comparing the DoE data with the experimental data. In addition, when the overlap limit is selected using only torque, the MOP is 476° CA with torque above 107.73 N-m, which is the error range of −5% of the experimental data, is the limit of minimum overlap. The maximum overlap limit was up to MOP 455° CA, the MBT point with a maximum torque of 119.3 N-m. However, as the amount of NOx tends to increase rapidly after MOP 461° CA, MOP 461° CA with an overlap of 70° CA interval was set as the maximum overlap limit. The minimum overlap limit was selected as MOP 476° CA with a NOx minimum value and a minimum torque reference value of 107.73 N-m or more. The intake valve overlap optimization range of 1800 RPM is 15 sections from MOP 461° CA with a maximum overlap interval of 70° CA, to MOP 476° CA with a minimum overlap of 56° CA interval. Figure 10c shows the results obtained at 2500 RPM. DoE was performed in a total of 80 sections from MOP 414° to 494° CA. Figure 10c shows that the tendency of torque and NOx is gentle, unlike in Figure 10a,b. The data value MOP 454° CA (interval 78° CA) used in the experiment shows a torque of 124.5 N-m, which is not consistent with the DoE simulation results. However, as the DoE simulation result value was 126.49 N-m, which is the result within the error range of 5%, it was judged that
the experimental value and DoE tendency were consistent. The MBT point is MOP 442° CA with a value of 130.45 N-m, but MOP 449° CA with a steep increase in NOx amount was set as the maximum overlap limit. The minimum overlap limit can be set to the MOP 479° CA, where the NOx amount is minimum; however, considering the torque, the torque within the error range of 5% of the experimental torque value was set as the lowest limit. An MOP of 471° CA with a torque value of 118.3 N-m or more was set as the minimum limit. The maximum overlap of 2500 RPM was selected as an optimization range of 22 ranges, from MOP 449° CA with an 84° CA interval to a minimum overlap to an MOP of 471° CA with a 62° CA interval. To achieve Pareto optimization from spark timing to IVO, a range was selected for each RPM. Table 4 summarizes the range of each variable and the number of DoE samplings. Based on these variables, the non-road gasoline MPI turbo SI engine was optimized.

3.2. Multi Objective Pareto Solution

In Section 3.2, the ranges of the variables for optimization at each RPM were selected. The DoE simulation was conducted at spark timing intervals of 0.5° CA, boost pressure intervals of 0.5 bar, and 2° CA intervals for the MOPs that enable confirmation of IVO. A total of 240–280 DoE sampling data points were obtained. Pareto optimization was performed based on the DoE results, and the torque and NOx were determined. In this study, we focused on the torque and NOx emissions so that a comparison can be made between the proposed engine and diesel engines, the main internal combustion engine for non-road vehicles. An engine for non-road vehicles must have a high torque even at a low RPM, and in the case of NOx emissions, they directly contribute to environmental pollution, which is currently a major issue. Figure 11a,b show the sensitivity of the variables for torque and NOx. Sensitivity analysis was performed based on the polynomial based regression coefficient conducted from DoE results.

![Figure 11. (a) Torque sensitivities of the operation parameters. (b) NOx sensitivities of the operation parameters.](image-url)

Figure 11a can confirm the effect of spark timing, boost pressure, and IVO on torque at each RPM. At 1600 RPM, it can be observed that the factor with the greatest effect on torque is boost pressure, followed by the IVO. Similarly, at 1800 RPM and 2500 RPM, the boost pressure is the factor that greatly affects torque, followed by IVO. However, as the RPM increases from 1600 to 2500 RPM, the impact of spark timing and boost pressure increases, while the impact of the IVO decreases. At high RPM, the effects of the boost pressure and spark timing are more magnified than the effect of IVO on the performance torque. For example, at a spark timing of 10° CA (aTDC) in Figure 9c, the torque shows a tendency to increase as the allowable boost pressure increases. Figure 11b shows the relationship between NOx and these factors. It was found that the effect of the IVO was...
the greatest at all the RPMs. For a 2500 RPM, IVO is more absolute than at other RPMs, and the boost pressure can be seen to be unaffected in predicting NOx emissions with a sensitivity of 0.0004.

The effects of different variables on torque and NOx were confirmed, and multi-objective Pareto optimization was performed to find maximum torque and minimum NOx emission levels.

Figure 12a shows the Pareto front and solution for each RPM for the targeted torque and NOx. The Pareto front was confirmed through 200 samples optimized from Pareto optimization. When checking the Pareto front by RPM in Figure 12a, it can be seen that the Pareto front rises sharply as the distribution interval at 1800 RPM and 2500 RPM widens, while at 1600 RPM, it is densely distributed and forms a gentle curve. It can be inferred that the relationship between the operation parameters changes as the RPM increases. Furthermore, Figure 12b shows a design parameter. The design parameter was obtained in a 2500 RPM WOT state, and the maximum torque value and maximum NOx generation can be checked at design point 1. With respect to the Pareto front, both torque and NOx can be seen as proportional curves that rise. The increase in performance can be confirmed by the increase in torque, which is the target, but the minimum NOx value, which is another target, has not been achieved, indicating that this is a trade-off relationship, not a proportional curve. Design point 2 has a torque similar to that of point 1, but it can be seen that the amount of NOx decreases. Design point 4 is a solution that has a minimum NOx amount, but a minimum torque is also a solution with a minimum value. Point 3 was a divergent solution. In such a Pareto front, engineers can select the appropriate design parameters to obtain an optimized solution.

Figure 12. (a) Pareto optimum solution according to the Brake torque and BSNOx. (b) Design points of the pareto optimum solution.

Figure 13a–c are graphs that summarize the Pareto solution for variables in design parameters 1 and 4.

Figure 13a shows the change in spark timing, and it can be confirmed that there is no significant change in the process of changing from point 1 to 4. Figure 13b shows that the boost pressure showed a slight change at 1800 RPM and 2500 RPM, but it was confirmed that the boost pressure reduced when the minimum NOx amount and minimum torque performance were observed at 1600 RPM. For this reason, it can be seen that the lower the RPM, the greater the performance and the amount of exhaust gas are affected by the boost pressure. Figure 13c shows the effect of IVO, confirming that the overlap interval decreased toward point 4, where the amount of NOx decreased to a minimum at all the RPMs. In addition, it was confirmed once again that the emission of exhaust gas was greatly affected.
by valve timing, as shown in Figure 11b. Through the multi-objective Pareto optimization, it was confirmed that IVO control is an effective way to optimize torque and NOx emissions in a non-road gasoline MPI turbo SI engine.

4. Conclusions

The purpose of this study was to optimize a down-sized 800cc gasoline MPI turbo SI engine for non-road vehicles through a 1D simulation. Spark timing, boost pressure, and intake valve overlap were selected as the optimization variables. In addition, by selecting a section where knocking, an abnormal type of combustion, does not occur, the optimal solution that maximized torque and minimized NOx was identified after conducting Pareto front multi-objective optimization. The relationships and results of each variable are summarized below.

- Through DoE, it was confirmed whether torque and NOx tendency and knocking occurred according to the change in spark timing. An increase in RPM expanded the occurrence area of knocking, and it was confirmed that excessively retarded spark timing caused incomplete combustion, resulting in an unstable NOx tendency.
• Through the relationship between spark timing and boost pressure, it was possible to confirm the range of boost pressures that could be allowed at each spark timing without knocking. In addition, the optimization range could be selected to predict the occurrence of NOx.

• The tendencies of torque and NOx were confirmed through the DoE of the intake valve overlap. It was confirmed that the valve overlap affects the performance of the engine and exhaust gas emissions by controlling the residual gas and fresh air in the cylinder. Additionally, it was confirmed that the more minimal the overlap interval, the higher the residual gas fraction, which reduces the amount of NOx. It can be seen that the IVO is inversely proportional to performance and exhaust emissions, and it was confirmed that, if it exceeds the appropriate overlap, it adversely affects the performance and exhaust gas.

• The solution was presented using the Pareto technique. The trade-off relationship was confirmed through the Pareto front for each RPM, and through each design point, the area for maximizing torque and minimizing NOx emissions was determined. In addition, by studying the sensitivities of the variables, it was possible to confirm their relationships and effects on engine performance, and their effects on the exhaust gas. In the optimal solution, it was confirmed that IVO had the greatest impact.

• Figure 14a is a graph comparing the optimal Pareto solution selected at 1600 RPM with the results of the DoE simulation parameters. (1) shows a 6.98% increase from the base torque and a 5.57% decrease in NOx as a result of MOP 460° CA, which is 8° CA wider than the baseline overlap interval. (2) can confirm the effect of spark timing and boost pressure, and as a result of the 1° CA true timing and 0.15 bar increased boost pressure compared to the baseline value, the torque increased by 13.86% and NOx decreased by 48.21%. (3) is an optimization solution that uses the Pareto technique by synthesizing all the parameters. It has a 1.37° CA more advanced spark timing than the baseline value, a 0.19 bar greater boost pressure, and 4° CA reduced overlap interval. Although it has 1.1 N-m less torque than (2), it shows a 12.78% increase compared to the baseline value and a 54.31% decrease in NOx.

• Figure 14b is a comparative graph of 1800 RPM. (1) shows that the overlap section was widened by 7° CA, the torque increased by 4.11% compared to the baseline, and NOx decreased by 28.01%. (2) had a 4.7° retard spark timing compared to the baseline, but the boost pressure increased by 0.25 bar, resulting in a 3.42% increase in torque and a 42.82% decrease in NOx. (3) confirmed that the 2.93° CA retarded spark timing, 0.19 bar increase in boost pressure, and an overlap interval that was widened by 1° CA resulted in a 5.88% increase in torque compared to the baseline and a 43.75% decrease in NOx. Furthermore, at 1800 RPM, we found that even when spark timing, which has a significant impact on performance control was somewhat retarded, the relationship between the other variables, such as the increase in boost pressure, could compensate for the torque and reduce NOx.

• Figure 14c shows a comparison of the 2500 RPM results. (1) can confirm a 37.52% NOx reduction and a 1.91% torque increase as the overlap intervals widens by 6° CA. (2) had a spark timing that was retarded by 4.1° CA compared to the baseline, but through an increase in boost pressure, the torque increased by 3.58% and NOx decreased by 42.64%. It can be seen that spark timing and boost pressure may offset one another. (3) offset the effect of the retarded spark timing through an increase in boost pressure and the decreased overlap interval; as a result, a 2.38% torque increase and 49.74% NOx decrease compared to the baseline were obtained.
Figure 14c shows a comparison of the 2500 RPM results. (1) can confirm a 37.52% NOx reduction and a 1.91% torque increase as the overlap intervals widens by 6° CA. (2) had a spark timing that was retarded by 4.1° CA compared to the baseline, but through an increase in boost pressure, the torque increased by 3.58% and NOx decreased by 42.64%. It can be seen that spark timing and boost pressure may offset one another. (3) offset the effect of the retarded spark timing through an increase in boost pressure and the decreased overlap interval; as a result, a 2.38% torque increase and 49.74% NOx decrease compared to the baseline were obtained.
Figure 14. (a) Comparison of the Pareto solution and DoE simulation results at 1600 RPM. (b) Comparison of the Pareto solution and DoE simulation results at 1800 RPM. (c) Comparison of the Pareto solution and DoE simulation results at 2500 RPM.

Author Contributions: Formal analysis, investigation, methodology, validation, writing—original draft, S.K.; supervision, writing—review & editing, J.S.; resources, writing—review & editing, Y.C.; funding acquisition, B.-S.S.; conceptualization, funding acquisition, resources, supervision, writing—review & editing, J.P. All authors have read and agreed to the published version of the manuscript.

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Abbreviations

1D One-dimensional
3D Three-dimensional
aTDC After top dead center
CA Crank angle
CAFE Corporate Average Fuel Economy
CO Carbon monoxide
DI Direct injection
DOC Diesel oxidation catalyst
DoE Design of experiment
DPF Diesel particulate filter
DPI Dual port injection
EVC Exhaust valve closed timing
GA Genetic algorithms
IVO Intake valve opening timing
MO Multi-objective
MOGA Multi-objective genetic algorithm
MOP Maximum intake valve opening position
MPI Multi point injection
NA Naturally aspirated
NOx Nitric oxides
PFI Port fuel injection
PM Particulate matter
RON Research octane number
SCR Selective catalytic reduction
SI Spark ignition
SITurb SI Turbulent Flame combustion model
UTV Utility terrain vehicles
VCR Variable compression ratio
VVT Variable valve timing
WGT Waste gate turbocharger
WOT Wide-open throttle

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