Simulation and Analysis of the Impact of Cylinder Deactivation on Fuel Saving and Emissions of a Medium-Speed High-Power Diesel Engine

Ying Liu, Alexandr Kuznetsov and Bowen Sa

1. Introduction

According to the European legislation for heavy-duty vehicles (Regulation (EU) 2019/1242), the CO₂ emission level compared with the EU average in the reference period should be reduced by 15% after 2025 and by 30% after 2030 [1]. Due to increasingly stringent emission regulations, the manufacturers of internal combustion engines (ICEs) strive to reduce fuel consumption and greenhouse gas emissions by improving engine performance. Different technologies have been developed to improve engine thermal efficiency to achieve this goal, including start-stop [2,3], rotation speed reduction [4,5], proper selection of technological operations [6], and cylinder deactivation [7–9]. Cylinder deactivation (CDA) has received significant interest and in-depth investigations for gasoline and light-duty diesel engines, considering its convenience of implementation on existing machines. The principle of CDA is to change effective engine displacement to impel engine running with higher effective efficiency at light-load conditions without compromising output power [10].

The principle of CDA application for reduced fuel consumption in spark ignition (SI) engines is that after some cylinders are deactivated, the rest of the active cylinders
require more fresh charge to maintain the brake mean effective pressure (BMEP); as a result, the throttle valve opens more widely, intake manifold pressure increases, and pumping losses decrease [11]. The application of CDA with variable valve actuation (VVA) in SI engines contributes to reducing the brake-specific fuel consumption (BSFC) up to 17–25% depending on engine configuration, operating points, and engine control [12–15]. CDA operation combined with dynamic skip fire strategies can reduce engine vibration and bring an additional reduction in BSFC [16,17]. The maximum BSFC improvement achieved for those engines with large displacement and multi-cylinders is attributed to pumping loss reduction under CDA operation. Currently, SI engines employing CDA technology are produced by many manufacturers, e.g., Mercedes-Benz, Honda, General Motors, Volkswagen, Ford, Mazda, and Fiat Chrysler Automobiles. In accordance with data in [18], in the 2020 model year, about 14% of light-duty vehicles sold in the United States used CDA.

Even though the production of compression ignition (CI) engines with CDA has not yet been realized, much research has been dedicated to CI engines’ thermal and emission performance under CDA operation. For CI engines, the advantages of CDA to improve fuel economy by reducing pumping losses are not as evident as for throttled SI engines because CI engines have no throttle but an extensive use of turbocharging. However, combustion processes in CI engines at idle or small loads are worse than those at medium and high loads due to slower evaporation and mixing rate [19]. CDA technology improves combustion quality and reduces emissions by transferring active cylinders from a low-efficiency to a high-efficiency operation range. The more significant the difference in efficiency and power from low load to medium load, the greater the possibility of CDA application on such engines [20,21]. Therefore, CDA has application potential for high-power engines, but runs at a light load and remains idle for a long time.

In CI engines, the most common way to realize CDA is to shut off the fuel injection to deactivate cylinders. This method was investigated in [22] on a 4-cylinder 2.66 L high-speed diesel engine for off-road excavators. Besides cutting off the fuel supply to deactivated cylinders, it is necessary to cooperate with different valve operation strategies to minimize engine losses, such as pumping losses and heat transfer losses. Jean-Paul Zammit et al. [23] conducted an experiment to deactivate two out of four cylinders on a 2.2 L turbocharged diesel engine with two strategies for deactivating cylinders: deactivating fuel supply and deactivating both fuel supply and valve motions. Alternating half of the cylinders’ deactivation was experimentally realized in a 6-cylinder diesel engine [24]. The most fuel consumption reduction was achieved when alternating CDA was employed.

Similarly, the BSFC reduction change with engine load growth is not monotonous at a constant speed. A simulation and experiments of CDA with disabled valves were implemented by S. Pillai et al. [24] in a high-speed diesel engine. The results demonstrate a 9% reduced BSFC at 10% full load and a 30% reduced BSFC at 20% full load for 1500 rpm. However, the same operation of CDA applied to different engines does not always achieve the BSFC reduction at a light load and when idle, since the indicated efficiency of active cylinders is associated with the conventional engine performance (the dependence of the indicated efficiency and load), which determines the available operation range of engine for CDA.

The indicated diagram in deactivated cylinders depends on different strategies of valve operation. When the intake and exhaust valves are entirely closed in deactivated cylinders, combustion does not occur, and the charge in cylinders changes periodically with piston movement. Due to gas leakage into the crankcase, the maximum gas pressure in the combustion chamber changes until it reaches a stable value. This process lasts only about 1 s for a light-duty diesel engine at a speed of 800 rpm so that this period can be neglected compared with the reactivation duration for cylinder work [25]. However, the minimum speed of heavy-duty diesel engines is as low as 300 rpm when idle. The change in the maximum gas pressure in deactivated cylinders interacts with the heat transfer in the combustion chamber, affecting the indicated efficiency. Therefore,
it is necessary to investigate the factors impacting gas or air state change in deactivated cylinders at low speeds. The results obtained can be used for further investigation of fuel consumption in transient processes with CDA operation.

In addition to the indicated efficiency, the other factor influencing fuel consumption is mechanical losses. At a light load and when idle, more than 20% of fuel energy is expended on overcoming friction and mechanical actuation [26]. After some cylinders are deactivated, gas pressure and temperature play an opposite role in mechanical friction for the active and deactivated cylinders, considering their impact on lubricated oil thickness and oil viscosity. The inconsistent effect of CDA on overall frictional losses of compression rings and big-end bearings has been noted for engines with different configurations. S. Bewsher et al. [27] found that the total frictional loss under CDA is 9.53% higher than that under normal operation. Similar results have been achieved in [28–30]. However, a decrease in the overall frictional loss under CDA operation has been reported in [31,32]. These studies demonstrate the necessity of considering total system effects with the application of CDA.

The CDA strategy with all valves closed in deactivated cylinders has a greater advantage for large-scale diesel engines since they have large size components and high inertia force from the perspective of reducing mechanical losses for valve trains.

The fuel consumption reduction caused by CDA implies emission reduction, especially CO₂ emissions that rely primarily on the amount of burnt fuel. NOx emissions are strongly dependent on combustion temperature, load, and oxygen concentration [9]. Under CDA operation, about the doubled mass of cycle fuel delivery for the normal operation is supplied to the active cylinder, enhancing the combustion temperature in the active cylinder and consequently increasing the formation of NOx emission. The increase in NOx emissions under CDA operation has been experimentally found for a 6-cylinder 11 L diesel engine with an asymmetric exhaust structure [33]. However, the reduction in NOx emission has also been reported for CI engines under CDA operation. About 19–25% reduction in NOx emission was obtained for a 6-cylinder 7.3 L turbocharged diesel engine without aftertreatment under CDA operation [32]. This is attributed to the fact that the decreased boost pressure in the compressor, caused by the increased backpressure before the turbo inlet under CDA operation, results in a reduction in the amount of fresh air charged and accordingly less oxygen in the active cylinder. Moreover, the increased emissions can be compensated by EGR and aftertreatment [23,34]. CDA improves the performance of the aftertreatment system by increasing the exhaust gas temperature [31,35–37]. In order to ensure the efficient operation of the aftertreatment system, its temperature must reach approximately 250 °C [23]. The increment of exhaust gas temperature under CDA operation ranges from 62 to 150 °C [24]. A transient test of a 6-cylinder Cummins diesel engine with CDA, implemented through a camless variable valve actuation system, has shown that during extended idle operation with CDA, the emissions of soot and NOx were reduced by 72% and 52%, respectively, associated with fuel savings up to 40% [38].

From the above literature analysis, it can be concluded that the effect of CDA introduction to a large extent depends on engine configurations, operation points, and CDA strategies, and most of the performed research relates to SI engines and high-speed CI engines for vehicles and road transport. However, there is insufficient data about investigating and applying CDA in medium-speed high-power CI engines until now. The medium-speed CI engines will maintain their important role in a long term in maritime business and railway transports [39–41]. It is also highly meaningful to improve the performance of the existing engines. Medium-speed high-power diesel engines of rail traction locomotives with series 8, 12, 16 cylinders have a wide output power range, whose rated power might exceed 2000 kW. According to locomotive operation characteristics, traction locomotives may run at multi-start/stop status or stay idle for a long time and only in a limited percentage of the time (about 10%) at full power mode [41], which leads to fuel energy losses. In addition, idle and start/stop processes are characterized by low engine efficiency, high specific fuel consumption, and exhaust emissions deterioration [42]. In such cases, CDA can be of great practical value.
The objective of the study is to evaluate the effectiveness of introducing CDA to medium-speed high-power CI engines by BSFC and emission level analysis. A thermodynamic engine model coupled with a dynamic mechanical system based on a turbocharged 16-cylinder diesel engine D-49 with CDA operations has been developed to investigate the effect of CDA on engine performance (including the indicated efficiency and mechanical losses in the active and deactivated cylinders). Although the technology of Rolling CDA or Dynamic Skip Fire could bring more fuel saving benefits and improve NVH characteristics under CDA operation [9], the fixed CDA is a more appropriate method for the large-scale diesel engine D-49, because most electric-hydraulic valve systems have a slow response, but the high-response electromagnetic valve system is impossible to be used for large inertia systems [9,43]. From this point of view, in the present study, fixed CDA is applied to the investigated diesel engine. Two CDA modes were considered here: only cutting off the fuel supply; cutting off the fuel supply and disabling valves in deactivated cylinders. In order to achieve an accurate assessment of the impact of CDA on engine emission characteristics, a 3D CFD combustion model has been created and used.

2. Numerical Models

The investigated engine D-49 is produced by Kolomensky Zavod (Moscow, Russia) and is used on mainline and shunting locomotives, mobile power plants, stationary drilling, and ship installations. The main parameters of the engine are listed in Table 1.

Simulation models developed for the diesel engine D-49 are described in the section. The geometric parameters of the investigated engine subsystems used were taken from an open technical manual of the diesel engine [44] attributed to the popularity of the diesel engine in Russia. Fuel injection profiles in different operating conditions were obtained by thermal-hydraulic modeling of the fuel injection system of the investigated engine in the software package Injection developed by professor Grekhov L.V. in Bauman Moscow State Technical University.

| Parameters               | Value                             |
|--------------------------|-----------------------------------|
| Engine type              | V16                               |
| Rated power              | 2250 kW                           |
| Rated speed              | 1000 rpm                          |
| Stoke/Bore               | 260 mm/260 mm                     |
| Compression ratio        | 13.5                              |
| Total displacement       | 220.8 L                           |
| Valve train              | Four valves per cylinder (two intake; two exhaust) |
| Fuel supply system       | Direct-acting plunger pump         |

2.1. Engine Model

A simulation model for the investigated engine was developed with GT-SUITE to study the influence of CDA on mechanical losses and fuel consumption. The developed engine model combines a thermodynamic system (combustion and gas exchange) and a dynamic mechanical system (piston-connecting rod-crankshaft mechanism) and is shown in Figure 1. The working parameters of the combustion process (pressure, temperature, etc.) as boundary conditions affect the calculation of mechanical losses in each cylinder; the mass flow rate in the thermodynamic system is determined by crankshaft speed calculated by the conservation of angular momentum in the dynamic system. Based on this model, the overall impact of CDA on the BSFC and engine performance can be analyzed, as well as the specific parameters between the active and deactivated cylinders.
The predictive diesel combustion model ‘EngCylCombDIJet’ was chosen to include the effect of fuel injection and atomization on the combustion process [45]. In this model, the fuel spray is divided into hundreds of zones to calculate its development and the evaporation rate of fuel droplets based on momentum conservation. Given the specifics of the fuel injection system of the diesel engine D-49, increasing injection mass means higher injection pressure at low loads, accurate injection profiles for different cycle deliveries are settled to guarantee the accuracy of calculation results. The combustion process is calculated in the block ‘EngCylinder’, and the simulated gas pressure is transmitted as an input force to the ‘Piston’ model.

The mechanical model consists of several parts, including the crankshaft, reciprocating components, valvetrain, and accessories. Therein, mechanical losses from valvetrain friction and accessories friction are calculated using semi-empirical formulas [46]. There are three compression rings and one oil ring between the piston and the cylinder wall (Figure 2). The piston-connecting rod-crankshaft model allows a convenient investigation for analyzing friction losses in the active and deactivated cylinders with CDA operation.

The gas pressure on the top and radial outer surfaces of the piston ring is considered as the pressure acting on the top surface [47]. The pressure applying on the bottom surface is calculated using a blow-by model proposed in [48]. Hydrodynamic friction is modeled using the Reynolds hydraulic equation, and the contact friction between mental surfaces is calculated using the Greenwood-Tripp model [49]. The result analysis is conducted on steady state conditions. Therefore, the mechanical system is set to ‘Rigid’ in the block ‘Cranktrain Analysis’, not considering the fluctuation of crankshaft speed resulting from torsional deformation. The ‘Rigid Engine Block’ has defined space coordinates of the crankshaft support. The physical interaction between the supports and ground is described by the relationship between damping and resilience.

The prototype engine D-49 has a V-shaped arrangement of cylinders with two camshafts; therefore, it is most convenient to deactivate one side of eight cylinders from the point of view of practical implementation. Fuel supply is cut off to deactivate cylinders in all cases of the CDA modes studied in this work. In order to control air/gas mass left in deactivated cylinders after the closure of the intake and exhaust valves, the valve closing timing needs to be adjusted with the crankshaft position signal.

![Figure 1. Constructed engine model of the diesel engine D-49.](image)
In consideration of cams with the pushrod configured in the diesel engine D-49, cylinder deactivation can be conveniently implemented by controlling the hydraulic lash adjuster to disconnect the camshaft and pushrod [50]. Therefore, the friction losses resulting from relative movements of valvetrain components are considered zero for deactivated cylinders in simulations. The temperature of lubricating oil in the piston group is taken as the average value of wall temperature in deactivated cylinders and the temperature of the piston ring [51].

In order to control NOx emissions in diesel engines with CDA operations, exhaust gas recirculation (EGR) systems are used. Burned gases are recirculated into the intake manifold to displace excess oxygen, decreasing flame temperatures, and NOx production. The EGR mass friction is control by the opening angle of the EGR valve.

2.2. 3D Engine Combustion Model

In order to get an accurate assessment of the changes in emission levels with CDA operation, a three-dimensional CFD model for the combustion chamber of the diesel engine D-49 was constructed in software FIRE ESE DIESEL, allowing to perform model settings, injection, and combustion analysis in diesel engines reliably and accurately [52]. In the diesel engine D-49, the injector with ten nozzles is placed centrally over the top of the piston. Given the symmetrical location of the injector at the combustion chamber, the 36° sector is selected as a computational domain (Figure 3). The total number of cells is 144,436 at TDC (top dead center). The calculation time step is 0.2 °CA. The calculation starts from 541 °CA (the beginning of compression) and ends at 841 °CA (the end of combustion). The boundary conditions and initial conditions are taken from the calculated results in GT-SUITE. The development of fuel spray was modeled with a Wave breakup model based on the instability of the surface wave [53]. In this model, breakup size-constant C1 decides the size of fuel droplets after crushing, and breakup time-constant C2 influences the characteristic breakup time, whose range is 5.0–60 in view of the different nozzle. The Dukowicz model [54] was used to calculate the evaporation of fuel droplets by applying semi-empirical functions for the heat transfer combined with a heat-mass transfer analogy. In this model, heat transfer-constant E1 is multiplied by the heat transfer coefficient calculated by the Ranz and Marshall correlation [55]. The k-ε-f model was employed for turbulence modeling [56]. The extended coherent flame model-3 zone (ECFM-3Z) [57]
was chosen to calculate the combustion process. ECFM-3Z is a combustion model based on a flame surface density transport equation for the premixed combustion and a mixing model that can describe inhomogeneous turbulent diffusion combustion. Mixing model parameters influence the transfer of fuel from the pure fuel zone to the mixed zone. Auto-ignition time is calculated by Two-Stag model, in which the Auto-ignition delay time for cool flame ignition and the main ignition is interpolated from tabulated values, which was created based on a n-heptane mechanism with 159 species and 770 reactions [58]. The Extended Zeldovich model is used to predict NOx emission and calculate the formation of nitrogen, oxygen, and hydrocarbon radicals, and this mechanism is solved in a sequential method for computational effectiveness. The kinetic model [59] was used to describe soot formation and oxidation. The model parameters have been tuned to ensure good consistency between the calculated results and experimental data.

![3D computational mesh of the combustion chamber at TDC.](image)

Figure 3. The 3D computational mesh of the combustion chamber at TDC.

In this work, the 1D simulation results (initial air pressure and temperature in cylinder, chamber wall temperature, etc.) and the injection profile are substituted into the 3D CFD model as boundary conditions to calculate the combustion heat release rate and cylinder pressure curve, which are used to validate 1D combustion modeling in turn. The parameters of engine performance are obtained from the 1D simulation, and the emission characteristics are converted according to the 3D calculation results.

2.3. Model Validation

The BSFC is a superimposed effect of the indicated efficiency and mechanical losses. For the combined model, the verification of the BSFC between calculation and experiment data is not enough to guarantee simulation accuracy. In addition, to analyze the impact of these indicators on engine performance with CDA operation, both the indicated efficiency and mechanical losses need to be validated separately.

The comparisons between the simulated and experimental indicated efficiency for four different engine speeds are depicted in Figure 4. A good agreement between simulation and experimental results is observed with the differences, which are less than 4%. Mechanical losses in engines are estimated by mechanical mean effective pressure (MMEP). The predicted mean mechanical loss pressure is extremely close to the experimental values. Mechanical losses generally increase with increasing engine speed; at the constant speed, the change in mechanical losses is not obvious as the load grows from part load to full load. However, the BMEP is small at light-load conditions or idle, so the change in mean mechanical loss pressure on mechanical efficiency cannot be ignored.
The accuracy of the 3D CFD model was evaluated by comparing the indicated diagram and engine emissions between simulation and experimental data. The initial and boundary conditions of the simulated operating condition obtained from GT-SUITE are shown in Table 2. Based on the experimental cylinder pressure curve, the parameter values of the breakup model and combustion model were successfully set and are shown in Table 3 after numerous corrections. The validation of the indicated diagram for the combustion model at a speed of 300 rpm is shown in Figure 5. The maximum relative errors of cylinder pressure between simulation and experiment do not exceed 5%. The validation of the emission formation models was performed by comparing the brake-specific carbon monoxide (BSCO) and the brake-specific nitrogen oxides (BSNOx) with experimental data at a speed of 450 rpm and under different engine loads. As shown in Table 4, the relative errors for the BSNOx prediction are less than 4%, and the maximum relative error for the BSCO prediction is 5.3% at a load of 4000 N·m. These comparisons confirm that the 3D CFD model can guarantee high accuracy and be used for following investigations.

Figure 4. Comparisons of simulations and experiments at engine speed: (a) 450 rpm; (b) 556 rpm; (c) 742 rpm; (d) 1000 rpm.
Table 2. The initial and boundary conditions of the simulated engine operating condition from GT-SUITE.

| Parameters                        | Value     |
|----------------------------------|-----------|
| Initial air pressure             | 1.18 bar  |
| Initial air temperature          | 322 K     |
| Cylinder wall temperature        | 345 K     |
| Cylinder head temperature        | 354 K     |
| Piston top temperature           | 358 K     |
| Start and end of injection       | $-8 \, ^\circ$CA/$-7.5 \, ^\circ$CA |
| The cycle injection mass         | 80 mg/cycle |

Table 3. Some parameters of submodels settled in AVL ESE DIESEL.

| Model Constant                     | Value |
|------------------------------------|-------|
| Dukowicz evaporation model E1      | 1.2   |
| Wave breakup model C1              | 0.61  |
| Wave breakup model C2              | 15    |
| ECFM-3Z combustion model Mixing model constant | 1 |
| Auto ignition parameter            | 1     |

Figure 5. The comparison of the simulated in-cylinder pressure based on the crank angle when idle with experiment data.

Table 4. The comparison of the BSNOx and BSCO emissions between simulation and experiment.

| Engine Load N·m | BSNOx, g/(kW·h) | Relative Error (%) | BSCO, g/(kW·h) | Relative Error (%) |
|-----------------|-----------------|--------------------|----------------|--------------------|
| 4000            | Simulation      | 12.17              | 4.40           | 5.0                |
|                 | Experiment      | 12.66              | 4.64           |                    |
| 5940            | Simulation      | 12.60              | 3.60           | 5.3                |
|                 | Experiment      | 12.91              | 3.80           |                    |
| 8210            | Simulation      | 13.28              | 2.27           | 3.0                |
|                 | Experiment      | 13.16              | 2.21           |                    |
3. Results and Discussion

Three CDA modes were employed and to be compared with the normal engine work (N—all cylinders work normally) at similar steady state conditions:

1. CDAo—cutting off the fuel supply in the deactivated cylinders;
2. CDAc—cutting off the fuel supply and disabling intake and exhaust valves in the deactivated cylinders;
3. CDAc + EGR—cutting off the fuel supply and disabling intake and exhaust valves in the deactivated cylinders incorporated with controlling the EGR rate.

It is worth noting that for the investigated diesel engine D-49, the injection timing and rail pressure vary with engine load and engine speed. These change characteristics are maintained under CDA operations. As an example, Figure 6 shows the injection profile under the normal operation and the injection profiles in the active cylinder under two CDA modes at a speed of 450 rpm and a load of 1000 N·m.

![Figure 6](image_url)

**Figure 6.** Injection profile under the normal operation and injection profiles in the active cylinder under two CDA modes at a speed of 450 rpm and a load of 1000 N·m.

3.1. Fuel Consumption

The above-mentioned CDA modes are evaluated by the brake-specific fuel consumption (BSFC) at engine speeds of 450 rpm and 556 rpm, and engine loads from 1000 to 4000 N·m (Figure 7).

![Figure 7](image_url)

**Figure 7.** Comparison of BSFC for two CDA modes and the normal operation at engine torques from 1000 to 4000 N·m and engine speeds: (a) 450 rpm; (b) 556 rpm (the calculated results were obtained from 1D simulation of GT-SUITE).
Figure 7 demonstrates that the BSFC reduction is only obtained at the CDAc mode by cutting off the fuel supply and closing the intake and exhaust valves in deactivated cylinders at the engine load from 1000 to 3000 N·m. At the engine torque of 1000 N·m, the BSFC at CDAc mode compared with the normal operation is decreased by 61.4 g/(kW·h) (11%) at 450 rpm and by 82.02 g/(kW·h) (14%) at 556 rpm, respectively. At a low torque range, the fuel-saving benefit from cutting off the fuel supply and closing valves to disable cylinders diminished with the load increase due to the limited amount of fresh air charged in the active cylinders, deteriorating the quality of the combustion process. The CDAo mode obviously leads to a BSFC increase, owing to extra energy losses and inadequate energy released. The effect of two CDA modes on the BSFC is a complex result of changes in the indicated efficiency and other energy losses. A detailed analysis of the impact of two CDA modes on these factors is given as follows:

3.2. Energy Analysis for the Engine

The effect trends of CDA on the BSFC at engine speeds of 450 rpm and 556 rpm are similar (Figure 7), so energy analysis is performed at 450 rpm and 1000 N·m.

The change in the engine indicated efficiency $\eta_i$ depends on the indicated efficiency $\eta_{i,act}$ in the active cylinder and the motoring losses in the deactivated cylinder. Figure 8 shows the indicated efficiency of the engine and the active cylinder at an engine load of 1000 N·m and an engine speed of 450 rpm.

![Figure 8](image_url)

*Figure 8. Comparison of the indicated efficiency for the engine and the active cylinder at an engine speed of 450 rpm and an engine load of 1000 N·m (the calculated results were obtained from 1D simulation of GT-SUITE).*

The indicated efficiency $\eta_{i,act}$ of the active cylinder is increased by 3.28% for the CDAo mode and by 4% for the CDAc mode. This can be explained by the fact that, when half of the cylinders are deactivated, more fuel mass is injected into the active cylinder with higher injection pressure. As shown in Figure 6, the maximum injection pressure increases from 198.7 bar at the normal mode to 373.1 bar at the CDAc mode and to 376.1 bar at the CDAo mode. The increased injection pressure accelerates fuel atomization and evaporation and promotes the mixture formation process [68]. More mixture is formed during the ignition delay period, which leads to an increase in combustion heat release rate at the rapid combustion period (Figure 9). The proportion of tail combustion heat release in the whole combustion decreases, thus improving the effective efficiency of the active cylinder. However, this benefit for the BSFC reduction in the active cylinder is offset by the motoring losses in the deactivated cylinder, which is expressed as a closed curve area in Figure 10 ($V/V_a$—the ratio of the above piston space to the total work volume). For the CDAo mode,
trapped air in the deactivated cylinder is compressed to a higher pressure and temperature in the compression stroke, $p_i$ (the average indicated pressure—reflect the motoring losses) in the deactivated cylinder $-0.3$ bar, but for the CDAc mode, this value is decreased to $-0.04$ bar due to less mass compressed in the deactivated cylinder. Combining the changes in the motoring losses and the cylinder-indicated efficiency, the engine-indicated efficiency for the CDAO mode is reduced by $0.7\%$ and is increased by $2.8\%$ for the CDAc mode.

![Figure 9](image1)

**Figure 9.** The normalized P-V diagram and combustion heat release at different modes (the calculated results were obtained from 1D simulation of GT-SUITE).

![Figure 10](image2)

**Figure 10.** The normalized P-V diagram in the deactivated cylinder under the CDAO mode and CDAc mode (the calculated results were obtained from 1D simulation of GT-SUITE).

The obtained calculation results prove the possibility of fuel consumption improvement using CDA by transforming cylinder work operations from low efficiency to high efficiency. Closing all valves in the deactivated cylinder reduces the motoring losses to reduce fuel consumption.

Figure 11 shows the comparison of the energy distribution diagram at the normal mode (N) and two CDA modes at an engine speed of 450 rpm and a load of 1000 N·m. Ensuring the same brake engine power $N_e$, the friction losses $N_f$, pumping losses $N_p$, and heat transfer losses $N_h$ are analyzed and compared. The heat transfer losses $N_h$ are significant in the CDAO and CDAc modes, with a slight increase in the CDAc mode compared to the CDAO mode.

![Figure 11](image3)

**Figure 11.** The comparison of the energy distribution diagram at the normal mode (N) and two CDA modes (the calculated results were obtained from 1D simulation of GT-SUITE).
heat transfer losses $N_h$, and exhaust energy $N_{ex}$ are analyzed and compared. The heat transfer loss is increased at the CDAo mode due to the higher combustion temperature in the active cylinder. At the CDAc mode, disabling all valves in the deactivated cylinders, the heat transfer losses to cylinder walls are decreased. This can be explained that the exhaust-intake strokes are replaced by the expansion-compression strokes, during which the irreversible pressure change produces the motoring losses, which are slight compared with the brake engine power. The average pressure of the motoring losses is $-0.04$ bar, equivalent to half the pumping losses of a normally worked cylinder at the same engine load. The pumping losses have a $37\%$ decline at the CDAc mode due to no pumping losses in the deactivated cylinder.

The change in exhaust energy $N_{ex}$ is related to the temperature and the flow rate of exhaust gas from the active cylinders and the deactivated cylinders. At an engine speed of 450 rpm and a load of 1000 N·m, the flow rate of exhaust gas for the normal, CDAo, and CDAc models is 2877.1, 2910.9, and 1520.6 kg/h, respectively. In comparison with the normal mode, at CDAo mode, the temperature of exhaust gas from the active cylinders rises by 80.4 K, but the temperature of non-combustion air from the deactivated cylinder decreases by 98.6 K, which leads to an 8.76 kW decrease in exhaust power $N_{ex}$ for the whole engine. In comparison with the normal mode, at CDAc mode, the temperature of exhaust gas from the active cylinders rises by 53.2 K, but the flow rate of exhaust gas is reduced by $47.1\%$, due to disabling the motion of intake and exhaust valves for the deactivated cylinder. As a result, the exhaust power $N_{ex}$ is reduced by 15.08 kW at the CDAc mode compared with the normal mode.

The mechanical loss changes are comprehensive results of the work change in active cylinders and deactivated cylinders. As shown in Figure 9, the analysis of mechanical losses was carried out for the prototype engine and two CDA modes. Engine mechanical losses consist of friction losses of piston group, friction losses of crankshaft bearings, friction losses of the valvetrain, and auxiliary mechanism losses. In comparison with the normal operation, the mechanical losses for the CDAc mode are reduced from 66.3 kW to 65.5 kW, which are attributed to the reduction in friction losses of the valvetrain. However, with all valves closed in the deactivated cylinders, the frictional losses of piston rings are increased due to the raised gas pressure and the lubricating oil temperature decrease. Friction losses
of the main bearings and big-end connecting rod bearings are decreased due to the overall load drop. Moreover, at a constant engine speed and low loads, the auxiliary mechanism losses almost do not change practically.

3.3. Emission Performance

The calculated results on the brake-specific pollutant emissions of the investigated engine under the normal mode and CDA modes at an engine speed of 450 rpm were given in Figures 12 and 13.

**Figure 12.** The brake-specific carbon monoxide (BSCO) (a), the brake-specific nitrogen oxides (BSNOx) (b), and the brake-specific soot emission (BSsoot) (c) for the normal operation and two CDA modes at an engine speed of 450 rpm and an engine load of 1000 N·m (the emission results were transformed from 3D simulation of ESE DIESEL).

**Figure 13.** Comparison of soot formation in the active cylinder between the normal operation and two CDA modes at an engine speed of 450 rpm and an engine load of 1000 N·m (the calculated results were obtained from 3D simulation of ESE DIESEL).

According to the result in Figure 12, in comparison with the normal engine operation, all CDA modes lead to a decrease in CO emission. At the engine load of 1000 N·m, the CDAo mode results in a reduction in CO emission from 42.98 to 10.63 g/(kW·h) (by 75.26%). This reduction is mainly attributed to the higher injection pressure (see Figure 6) in the active cylinder, promoting the quality of fuel-air mixing, and the increased
combustion temperature (see Figure 14), which improved the evaporation of fuel on wetted in-cylinder surfaces. Enhanced fuel-air mixing and heat release due to higher injection pressure have also been reported in [61]. The relationship of CO emission with the fuel injection and temperature in the cylinder is in accordance with the results in [62].

As can be seen from Figure 12, the soot emission can almost be neglected at the load of 1000 N·m and 450 rpm. Both CDA modes reduce soot emission from 0.0070 g/(kW·h) to 0.0026 g/(kW·h) (by 62.9%). By tracking the change in the soot emission with the crankshaft angle (Figure 13), it can be found that in comparison with the normal operation, the increased fuel injection mass in the active cylinder at both CDA modes leads to a richer mixture (under the premise of the same intake air mass), and more intensive formation of soot happens near the TDC. However, the formed soot is mainly oxidized on the expansion cycle. At two CDA modes, soot oxidation is promoted attributed to the increased temperature on the expansion cycle (Figure 14). A weak influence of the valve closure on soot emissions is observed. The NOx emission is obviously higher under CDA modes compared with normal operation due to a higher combustion temperature. In comparison with the CDAc model, the BSNOx emission for the CDAo model increases by 7.1 g/(kW·h) due to a higher combustion temperature. This can be explained that the pumping losses and mechanical losses for the CDAo model are higher due to the normal valve motions; in addition, due to a higher combustion temperature. This can be explained that the pumping losses and mechanical losses for the CDAo model are higher due to the normal valve motions; in comparison with normal operation due to a higher combustion temperature. In comparison with normal operation due to a higher combustion temperature. This can be explained that the pumping losses and mechanical losses for the CDAo model are higher due to the normal valve motions; in comparison with normal operation due to a higher combustion temperature. In comparison with normal operation due to a higher combustion temperature. This can be explained that the pumping losses and mechanical losses for the CDAo model are higher due to the normal valve motions; in comparison with normal operation due to a higher combustion temperature.

The reduction in NOx emissions was achieved by changing the EGR rate (adjusting the EGR valve opening angle). Figure 15 shows the effect of the change in the EGR rate (from 0 to 50%) at the CDAc + EGR mode at a speed of 450 rpm and a load of 1000 N·m on the BSFC and NOx emissions. When the EGR rate is gradually increased from 0 to 50%, BSNOx emission is reduced by 18 g/(kW·h), and the BSFC is increased by 12 g/(kW·h). This can be explained by the fact that the higher mass fraction of exhaust gas in the intake charge leads to a lower excess air coefficient in the active cylinder, which leads to a decline in combustion efficiency [60,63]. The effect of EGR in suppressing BSNOx emissions becomes less obvious as the EGR rate increases. This is consistent with experimental results.
of the relationship between the EGR rate and BSNOx in [64] and results of NOx formation suppression by high excess air ratio in [65].

![Figure 15](image1.png)

**Figure 15.** BSFC–BSNOx trade-off for the CDAc + EGR mode with the EGR rate from 0 to 50% at an engine speed of 450 rpm and an engine load of 1000 N·m (the emission results were transformed from 3D simulation of ESE DIESEL).

Figure 16 shows the effect of the different EGR rates on CO and Soot emissions under the CDAc + EGR mode at a speed of 450 rpm and a load of 1000 N·m. Increasing the EGR rate from 0% to 50%, the BSCO emission is increased by 8 g/(kW·h), which is close to double the CO emission level at the 0% EGR rate. This is because EGR effectively suppresses the reaction rate of CO oxidation in combustion and reduces the combustion temperature, retarding the evaporation of liquid fuel drops on the surface of the cylinder wall. With the different EGR rates, the BSsoot emission is relatively stable and remains at the level of 0.0025 g/(kW·h), which is close to being negligible. Although the raised EGR rate leads to a longer combustion delay period (see Figure 17) and decreases the generation rate and oxidation rate of soot, however, the soot oxidation process occurs more completely due to sufficient combustion time. The results are consistent with the conclusions reported in [63,64,66]. Although the increased EGR rate decreases BSNOx emissions effectively, the BSFC and the BSCO emissions are increased.

![Figure 16](image2.png)

**Figure 16.** BSCO–BSNOx trend and BSsoot–BSNOx trend for the CDAc + EGR mode with the EGR rate from 0% to 50% at an engine speed of 450 rpm and an engine load of 1000 N·m (the emission results were transformed from 3D simulation of ESE DIESEL).
Figure 17. Soot formation in the active cylinder under the CDAc + EGR mode with the EGR rate from 0% to 50% at an engine speed of 450 rpm and an engine load of 1000 N·m (the calculated results were obtained from 1D simulation of ESE DIESEL).

According to the comparison above, the optimal option is that the EGR rate is 30%, in which the BSNOx emission is effectively reduced to 4.2 g/(kW·h) at the cost of a 0.8% increase in the BSFC and without the increase in the BSCO emission. Although EGR causes a slight BSFC increase, it is far less than the BSFC reduction brought by CDA.

The post-turbine exhaust gas temperatures are illustrated in Figure 18 for the load of 1000 N·m. At the CDAo mode, the exhaust gas temperature marginally declines since air with a lower temperature in deactivated cylinders flows into the exhaust pipe, which has a cooling effect on the exhaust gases from the active cylinder. Compared with the normal mode, the exhaust gas temperature is increased by 50 K at the CDAc mode and by 70 K at the CDAc + EGR mode. This can be explained that the deactivated cylinder is isolated from the exhaust pipe, so the charge in the cylinder does not enter the exhaust pipe, which leads to due to nearly twice more heat release from fuel combustion, which exceeds the stable working temperature for diesel engine aftertreatment, ensuring high catalytic efficiency to improve emissions. Higher exhaust gas temperatures are reached that reflect the reduced gas-side heat transfer (Figure 9), which is attributed to the fact that the increased heat transfer losses in the firing cylinders are more than offset by the lower heat transfer losses in the disabled cylinders, and the recovery of exhaust gas thermal energy heating the intake air.

Figure 18. The exhaust gas temperature for the normal operation and CDA modes at an engine speed of 450 rpm and engine loads of 1000 N·m and 4000 N·m (the calculated results were obtained from 1D simulation of GT-SUITE).
4. Conclusions

The effect of three CDA modes on the improvement of the BSFC and emission levels was analyzed on a medium-speed high-power diesel engine D-49. A numerical study was performed by associating the 1D simulations in the GT-SUITE and 3D CFD simulations in the AVL ESE DIESEL. A predicted friction model was established to estimate friction losses. The impact of CDA operations on the work parameters of the diesel engine was investigated with 1D simulations. The emission formation in the combustion chamber under different operations has been evaluated with 3D CFD simulations. The main conclusions are summarized as follows:

(1) The BSFC reduction is obtained in the engine operating range from 1000 to 3000 N·m at 450 rpm and 556 rpm. CDA with intake and exhaust valves closed reduces the BSFC by 11 and 14% at engine speeds of 450 rpm and 556 rpm at a load of 1000 N·m.

(2) The BSFC reductions are attributed to the increase in the indicated efficiency in the active cylinder and the reduction in the motoring losses, the pumping losses, and friction losses of the valvetrain in the deactivated cylinder. Fixed cylinder deactivation will increase the hydraulic friction loss in the deactivated cylinders.

(3) At low speeds and light engine loads, CDA with all valves closed without EGR reduces CO and soot emissions attributed to the improved fuel-air mixing formation and increased combustion temperature in the active cylinder, which in turn results in an increase in NOx emission.

(4) At 450 rpm and 1000 N·m, CDA combined with EGR control dramatically reduces the BSNOx emission to 4.2 g/(kW·h) at the cost of only a 0.8% increase in the BSFC, while the BSCO emission is maintained in low levels.

(5) The investigation confirms the benefit of combining CDA with all valves closed and EGR as an effective method to reduce the BSFC and engine emission levels simultaneously for a medium-speed high-power engine, without the significant modification of engine construction.

Steady states were considered in the study. In the next research stage, it is meaningful to study the engine transient response process and vibration characteristics during cylinder deactivating and reactivating for the medium-speed high-power engine with high inertia components by using the simulation models developed and validated in the present study.

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Abbreviations

| Abbreviation | Description                      |
|--------------|----------------------------------|
| BSFC         | Brake-specific fuel consumption  |
| BSCO         | Brake-specific carbon monoxide   |
| BSsoot       | Brake-specific soot              |
| BSNOx        | Brake-specific nitrogen oxides   |
| CDA          | Cylinder deactivation            |
| CDAo         | Cylinder deactivation with valve open |
| CDAc         | Cylinder deactivation with valve closed |
| EGR          | Exhaust gas recirculation        |
| MMEP         | Mechanical mean effective pressure |
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