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SUSTAINABLE DEVELOPMENT OF ENERGY, WATER AND ENVIRONMENT SYSTEMS

Combustion and emission characteristics for a marine low-speed diesel engine with high-pressure SCR system

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Abstract

In order to avoid the production of sulfates and nitrates in marine diesel engines that burn sulfur-containing fuels, the operating temperature of their high-pressure selective catalytic reduction (HP-SCR) systems should be higher than 320 °C. For marine low-speed diesel engines, only the pre-turbine exhaust gas temperature can meet this requirement under specific conditions, with the main engine modulation method helping to increase the exhaust gas temperature. However, the main engine modulation method brings down the power output and fuel economy of the main engine and causes the matching problem of the turbine and the other devices with the main engine. The original engine model of the marine low-speed diesel engine and the high-pressure SCR system configuration model have been constructed using one-dimensional simulation software. In addition, the performance of the high-pressure SCR system under the conditions of low-sulfur and high-sulfur exhaust gas was thoroughly analyzed. Moreover, the two main engine modulation schemes of the scavenging bypass and the turbine exhaust bypass of the original engine matching with the high-pressure SCR system were studied. The study found that the weighted average value of the NO*x* under the condition of low-sulfur exhaust gas met with the requirement of the IMO Tier III regulations when the low-speed diesel engine was matched with the high-pressure SCR system. However, the weighted average value of the NO*x* under the condition of high-sulfur exhaust gas was slightly higher than that required by the IMO Tier III regulation. In addition, the optimal main engine modulation scheme for this low-speed diesel engine was clarified by comparing the effects of the scavenging bypass and the turbine exhaust bypass modulation on the exhaust performance, and the working performance of the original engine. With an opening of 0.4 of the CBV valve under 25% engine load, the weighted average NO*x* of the original exhaust gas was 3.38 g/(kW·h), the power had decreased by 0.7%, and the fuel consumption had increased by 1.0%. Furthermore, when the EGB valve opening was 0.3, the weighted average value of NO*x* was 3.31 g/(kW·h), the power had reduced by 2.4% and the fuel consumption had increased by 2.5%. Both modulation scheme methods made the exhaust performance of the original engine meet the requirements of the IMO Tier III emission regulations, but the scavenging bypass modulation scheme had less impact on the original engine's performance.

Keywords High-pressure SCR system · Low-speed diesel engine · Main engine modulation · Matching performance · Simulation calculation

Introduction

In recent years, ship exhaust emissions have become a major source of atmospheric pollution in the countries with developed ports and sea areas dealing with intensive shipping routes (CIMAC 2008; Fung et al. 2014). In order to reduce the pollution of ship exhaust emissions in the atmosphere, the International Maritime Organization (IMO), the European Union, and the USA have formulated strict regulations (MAN 2018), which affected the relevant ports and routes of Europe and North America. China also plans to extend the existing Emission Control Area to all sea areas within 12 nautical miles of the coasts and ports (MTPRC 2018) that will surely have an impact on the related sea routes.

With the development of global laws and regulations that limit the air pollution from ships, green shipping and efficient

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transportation has become an important subject for the shipping community so as to achieve "near zero emission" or even "zero emission" of pollutants from the marine diesel engines. Currently, selective catalytic reduction is the only technology recognized by the International Maritime Organization (IMO) that is used to reduce NOx emissions for various types of marine engines and ships (IMO 2013), which theoretically meets both the IMO Tier III regulation and the other stricter emission standards.

Compared with vehicle diesel engines, the marine diesel engine's SCR system is divided into a high-pressure SCR system (HP-SCR) in front of the turbine and a low-pressure SCR system (LP-SCR) in the back of the turbine (Ayodhya and Narayanappa 2018; Döring et al. 2016; MAN 2018; Ryu et al. 2016; Sandelin and Peitz 2016) Since the exhaust gas in the exhaust pipe drives the turbine to work, the temperature of the exhaust gas before and after the turbine differs by 50~175 °C—so the high-pressure SCR system in front of the turbine can make full use of the higher exhaust gas temperature and also improve its activity. At the same time, the high-pressure SCR system has a high working pressure, and its absolute pressure is several times higher than that of the low-pressure SCR system, which is equivalent to reducing the volumetric flow rate or line speed of the treated exhaust gas and increasing the SCR reaction time. Therefore, under the same design requirements, the high-pressure SCR system has a more compact structure and higher energy efficiency utilization and is more suitable for the low-speed diesel engines having higher power (Fig. 1).

Nevertheless, when the marine low-speed diesel engines use inferior high-sulfur fuels, there will inevitably be a large number of SOx in the exhaust gas (CIMAC 2008; Fung et al. 2014). In such cases, the lowest operating temperature of the SCR system or the denitration catalyst must be determined to avoid the formation of deposits, such as sulfates (Cai et al. 2016; Xie et al. 2017; Xue and Wang 2018). Generally, under high-sulfur exhaust gas conditions (fuel sulfur content 3.5%), the operating temperature of the high-pressure SCR system is required to be higher than 320 °C in order to avoid the production of sulfate and nitrate (Christensen et al. 2011). Only the pre-turbine exhaust gas temperature can meet this requirement under some operating conditions, especially for the low-speed marine diesel engines, which have high thermal efficiency. Yet, when the engine loads are low less than 15% MCR, the exhaust gas temperature before the turbine will not meet this requirement and the injection of urea solution must be stopped to prevent the formation of deposits inside the SCR system (MAN 2018).

At present, there are two main engine modulation methods: the cylinder bypass valve (CBV) and the turbine exhaust gas bypass valve (EGB) (Christensen et al. 2011; MAN 2018). Both methods depend on the increment of the exhaust gas temperature at the expense of the fuel economy by reducing the fresh air quality entering the combustion chamber and the increment of the fuel injection rate. Nevertheless, this will decrease the power output and the fuel economy of the main engine and will bring the matching problem between the turbine and the main engine.

Under some cases when the working condition of the marine diesel engine changes (during acceleration or deceleration) and because of the thermal inertia of the catalyst in the SCR system, the exhaust gas temperature is observed to be different or delayed for a short time, both before and after the SCR reactor (between the exhaust gas receiver outlet and the turbine inlet). This causes energy imbalance before and after the turbine, which affects the normal operation of the turbine. Obviously, the degree of thermal delay mainly depends on the size of the SCR reactor and the specific heat capacity of the catalyst. Therefore, it is necessary to study the matching performance between the high-pressure SCR system and the main engine.

High-pressure SCR system and its components

The high-pressure SCR system is installed before the turbocharger of the diesel engine to improve the SCR reaction temperature. The SCR system has a compact layout and high exhaust gas-energy utilization, but it has a great influence on

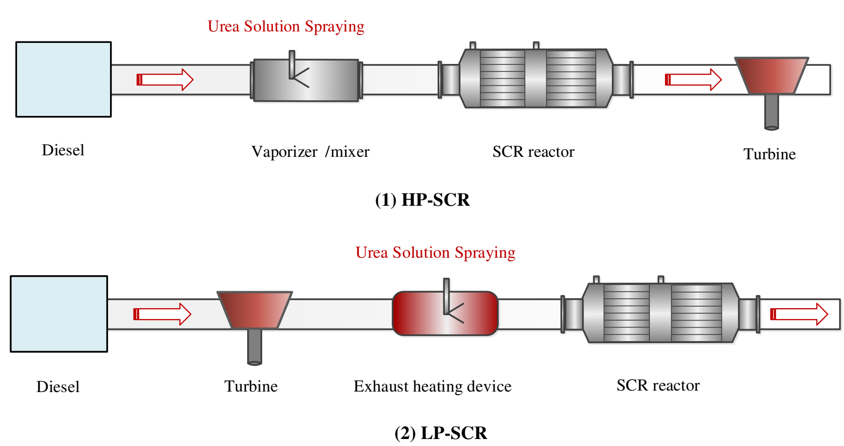


Fig. 1 Schematic diagram of the high-pressure SCR system and the low-pressure SCR system for the marine diesel engine

the performance of the diesel engine and the turbocharger. It is mainly used in the low-speed two-stroke diesel engines, which burn the high-sulfur fuels.

Figure 2 shows the high-pressure SCR system developed by MAN B&W, which mainly includes four parts: the evaporative/mixer, the SCR reactor, the auxiliary fan, and the bypass valves (MAN 2018). The evaporation/mixer is mainly to ensure complete evaporation and decomposition of the injected urea solution and to ensure uniform mixing of the reducing agent NH_3 with the exhaust gas. Since the exhaust gas in the exhaust pipe drives the turbine to work, the temperature of the exhaust gas before and after the turbine differs by 50 to 175 °C, so the SCR system in front of the turbine can make full use of the higher exhaust gas temperature to improve the denitration efficiency.

In the high-pressure SCR system, the bypass valves consist of the reactor sealing valve (RSV), the reactor throttle valve (RTV), the reactor bypass valve (RBV), the EGB, and the CBV (MAN 2018). Combinations of five pneumatic valves are used to control the treated exhaust gas flows through the SCR reactor. For diesel engines that meet the IMO Tier II standard, the switch of the bypass valve group determines the operating mode of the diesel engine: Tier II mode and Tier III mode. However, the two working modes of the diesel engine do meet the different emission standards.

Simulation model of marine high-pressure SCR system

Using the GT-Power simulation software to model a certain type of low-speed marine diesel engine and its matched high-pressure SCR system, this chapter deliberates on a cylinder model, an intake/exhaust system model, a turbocharger model, an intercooler model, and a high-pressure SCR system model, illustrated in terms of five independent sub-model systems combined with their respective mathematical descriptions.

Cylinder model

The cylinder system is modeled as an enclosed space composed of components such as the cylinder head, the cylinder liner, and the piston crown. The fresh charge enters the cylinder through the intake system and the fuel is injected into the cylinder through the injection system and is mixed with fresh air to start the burning process near the top dead center. The energy released by the fuel combustion is converted into kinetic energy through the crank linkage mechanism and the burned exhaust gas is discharged through the exhaust system.

The cylinder model is illustrated in terms of the stroke, the cylinder diameter, the connecting rod length, the compression ratio, and the head clearance height of the diesel engine. The heat transfer model used the Woschni's empirical formula for calculations (Gamma 2016) and the combustion model used the quasi-dimensional combustion model based on droplet evaporation (Gamma 2016), which divided fuel injection into several sub-regions, each of which is independent of the other. Lastly, the evaporation, combustion, and exothermic processes are calculated separately.

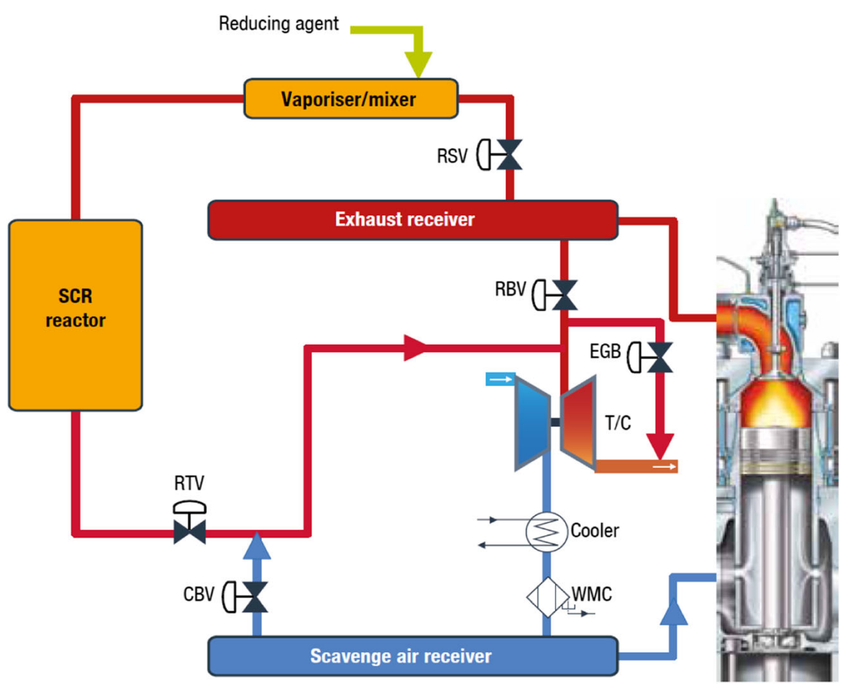


Fig. 2 High-pressure SCR system developed by MAN

Intake and exhaust system model

Since the axial dimension of the diesel engine intake and exhaust pipes is much larger than the radial dimension, the axial flow effect in the duct will be greater than the radial flow effect; and the gas flow in the intake and exhaust pipes might be regarded as one-dimensional unsteady flow. According to the Naiver-Stokes equation of the intake and exhaust system, the fluid state in the intake and exhaust pipes can be solved by the continuity conservation equation, the momentum conservation equation, and the energy conservation equation (Gamma 2016) respectively.

In the process of calculation, according to the set discrete length, the intake and exhaust systems are discretized into several grids, which are connected with each other through the boundary. In each independent grid, it is assumed that the scalar values of temperature, pressure, internal energy, density, and helium of each grid center are fixed while the mass flow rate, the flow rate, the composition concentration, and so on are calculated at the grid boundaries. The parameters of the intake and exhaust system are calculated by considering the friction loss, the pressure loss, and the heat transfer loss in the pipeline.

Turbocharger model

The turbocharger model consists of the compressor sub-model and the turbine sub-model. Between them, the performance MAP of the compressor consists of a series of data points, which mainly describes the characteristics of the compressor's rotation speed, the pressure ratio, the mass flow rate, and the thermodynamic efficiency under different working conditions. The MAP can be used to obtain the thermodynamic efficiency and the boost in the pressure ratio, under the current working conditions as shown in Fig. 3.

Throughout the calculation process, the turbocharger must meet three equilibrium conditions during the steady-state operation: (1) the output power supplied by the turbine must be equal to the power consumed by the compressor in each cycle; (2) the exhaust gas flowing through the turbine must sum up the fresh air flow rate and the circulating fuel injection value through the compressor; and (3) the compressor and the turbine must rotate at the same speed and must maintain this speed in one cycle (Gamma 2016).

The outlet temperature of the turbocharger is calculated from the enthalpy of the turbine and the compressor system. Moreover, the power supplied by the turbine and the power consumed by the compressor can be calculated from the efficiency obtained by the enthalpy and the MAP curves. The main formula of the turbocharger model is as follows (Gamma 2016):

Turbine:

$$h_{\mathrm{T,out}} = h_{\mathrm{T,in}} - \Delta h_{\mathrm{T}} \eta_{\mathrm{T}} \tag{1}$$

$$P_{\mathrm{T}} = \dot{m}_{\mathrm{T}} \left(h_{\mathrm{T,in}} - h_{\mathrm{T,out}} \right) \tag{2}$$

$$\Delta h_{\mathrm{T}} = c_{\mathrm{p}} T_{\mathrm{total,in}} \left[1 - \pi_{\mathrm{T}}^{(1-\kappa)/\kappa} \right] \tag{3}$$

Compressor:

$$h_{\mathrm{C,out}} = h_{\mathrm{C,in}} - \Delta h_{\mathrm{C}} \frac{1}{\eta_{\mathrm{C}}} \tag{4}$$

$$P_{\mathrm{C}} = \dot{m}_{\mathrm{C}} \left(h_{\mathrm{C,in}} - h_{\mathrm{C,out}} \right) \tag{5}$$

$$\Delta h_{\mathrm{C}} = c_{\mathrm{p}} T_{\mathrm{total,in}} \left[\pi_{\mathrm{C}}^{(\kappa-1)/\kappa} - 1 \right] \tag{6}$$

$$T_{\mathrm{total,in}} = T_{\mathrm{in}} + \frac{u_{\mathrm{in}}^2}{2 c_{\mathrm{p}}} \tag{7}$$

In the formula, $h_{\mathrm{C,in}}$ and $h_{\mathrm{T,in}}$ are the enthalpy values for the compressor and the turbine inlet; $h_{\mathrm{C,out}}$ and $h_{\mathrm{T,out}}$ are the enthalpy values of the compressor and the turbine outlet; h_{C} and h_{T} are the enthalpy changes of the compressor and the turbine; η_{C} and η_{T} are the efficiency of the compressor and the turbine; $\dot{m}_{\mathrm{C}}$ and $\dot{m}_{\mathrm{T}}$ are the gas mass flow through the compressor and the turbine; π_{C} and π_{T} are the pressure ratio of the compressor and the expansion ratio of the turbine; P_{C} and P_{T} are the power of the compressor and the turbine; T_{in} is the inlet gas temperature; T_{total} is the total inlet gas temperature; u_{in} is the inlet velocity of the gas; and c_{p} is the heat capacity of the gas at constant pressure ratio.

Intercooler model

According to the heat balance, the total heat released by the charged air through the intercooler is:

$$Q_{\mathrm{C}} = c_{\mathrm{p}} \dot{m}_{\mathrm{C}} \left(T_{\mathrm{C,out}} - T_{\mathrm{C,in}} \right) \tag{8}$$

The total heat carried by the cooling medium through the intercooler is (Gamma 2016):

$$Q_{\mathrm{W}} = c_{\mathrm{W}} \dot{m}_{\mathrm{W}} (T_{\mathrm{W2}} - T_{\mathrm{W1}}) \tag{9}$$

where c_{w} represents the specific heat of the cooling medium, m_{w} represents the flow rate of the cooling medium, $T_{\mathrm{C,in}}$ represents the temperature of the charge air at the compressor outlet, $T_{\mathrm{C,out}}$ represents the air temperature at the outlet of the intercooler, T_{W1} represents the temperature of the cooling medium at the inlet of the intercooler, and T_{W2} represents the temperature of the cooling medium at the outlet of the intercooler.

The charge air delivered to the cooling medium is (Gamma 2016):

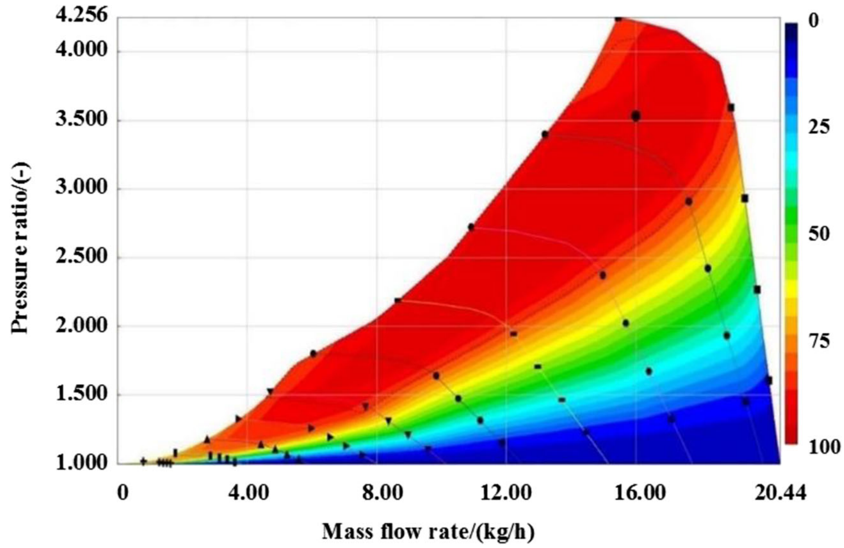


Fig. 3 Performance map of the compressor

$$Q_{CW} = K \int_0^A \Delta T dT = KA\delta T \tag{10}$$

where K represents the heat transfer coefficient, A represents the surface area of the intercooler flow pipe, and T represents the average logarithmic temperature difference determined by the intercooler compressed air, cooling medium channel form, and flow characteristics. The calculation method is (Gamma 2016):

$$\delta T = \frac{(T_{C,in} - T_{W2}) - (T_{C,out} - T_{W1})}{\ln \frac{T_{C,in} - T_{W2}}{T_{C,out} - T_{W1}}} \tag{11}$$

The resistance loss in the intercooler is calculated as follows:

$$\Delta p_C = P_{C,in} - p_{C,out} = \eta_r \frac{\dot{m}_C^2}{\rho_C} \tag{12}$$

where η_r represents the resistance coefficient, usually the resistance coefficient of the intercooler that can be selected by referring to the same type of test data, and the general range of values is 3~5 kPa.

SCR system model

There are two kinds of reversible reactions and irreversible reactions in the SCR reaction, and are divided into a gas phase reaction and a surface reaction. During the calculation, the forward reaction rate constant is solved according to Arrhenius's law, which is determined by the pre-exponential factor, the temperature coefficient, and the activation energy. For reversible reactions, the reverse reaction rate is determined by the forward reaction rate constant and the chemical equilibrium constant (Gamma 2016). For surface reactions, it is assumed that the reactants are adsorbed on the surface of the solid catalyst and then are formed on an activated intermediate compound with another reactant in the gas phase to form the final product. Since the surface reaction is a chemical reaction control step, the overall reaction rate is proportional to the coverage of the adsorbed material on the surface of the catalyst and the partial pressure of the gas phase material.

The SCR system model constructed in this paper is divided into two parts: the reaction kinetic model and the catalyst model. SCR reaction pathway, the reaction kinetics model including the urea decomposition reaction, the surface denitration reaction, and the ammonium salt passivation reaction (Chen et al. 2018; Ciardelli et al. 2007a, b; Ebrahimian et al. 2012; Li et al. 2017a, b; Ma et al. 2010; Nova et al. 2006; Nova et al. 2007; Thogersen et al. 2010; Topsoe et al. 1995; Topsoe et al. 1998; Wang et al. 2018; Xu et al. 2017; Zhu et al. 2018a, b; Zhu et al. 2019) which are to simulate urea's decomposition, NOx catalytic reduction, and low-temperature ammonium salt formation. The catalyst model uses a porous medium model (Gamma 2016), which ignores the radial diffusion of the components in the catalytic reactor. Amongst them, the SCR reaction kinetics model contains chemical reaction pathways that are shown in Table 1.

Simulation model and verification of the original engine

Table 2 shows the main parameters data of a certain type of the low-speed diesel engine. According to the characteristics of the low-speed diesel engine of a certain type of ship and by using the GT-Power simulation software, this section constitutes the prototype model, as shown in Fig. 4.

In order to ensure the accuracy of the original engine simulation model, this section compares and analyzes the

Table 1 SCR reaction pathway and reaction rate

Serial number	Chemical reaction	Pre-exponential factor	Activation energy Ea (J/kmol)
Urea decomposition reaction			
R01	$CO(NH_2)_2 \rightarrow HNCO+NH_3$	4.9×10^3	2.3×10^7
R02	$HNCO+H_2O \rightarrow NH_3+CO_2$	2.3×10^5	6.2×10^7
Surface denitration reaction			
R03	$NH_3+V \rightarrow NH_3 \cdot V$	6.7×10^8	0
R04	$NH_3 \cdot V \rightarrow NH_3+V$	1.0×10^{13}	8.4×10^4
R05	$4NH_3 \cdot V+4NO+O_2 \rightarrow 4N_2+6H_2O+4V$	5.0×10^{12}	7.5×10^4
R06	$4NH_3 \cdot V+2NO+2NO_2 \rightarrow 4N_2+6H_2O+4V$	8.0×10^3	6.5×10^4
R07	$8\ NH_3 \cdot V+6NO_2 \rightarrow 7N_2+12H_2O+8V$	3.0×10^3	7.1×10^4
R08	$4NH_3 \cdot V+3O_2 \rightarrow 2N_2+6H_2O+4V$	1.7×10^{13}	2.0×10^5
Nitrate formation reaction			
R09	$2NH_3 \cdot V+2NO_2 \rightarrow N_2+NH_4NO_3 \cdot V+H_2O+V$	3.4×10^{-2}	0.0
R10	$NH_4NO_3 \cdot V \rightarrow NH_3+HNO_3+V$	10.0	26.0
R11	$NH_3 \cdot V+HNO_3 \rightarrow NH_4NO_3 \cdot V$	1.1×10^{-6}	0.0
R12	$NH_4NO_3 \cdot V \rightarrow N_2O+2H_2O+V$	7.5×10^9	1.1×10^3
Sulfate formation reaction			
R13	$V+SO_2 \rightarrow SO_2 \cdot V$	1.3×10^{-4}	0.0
R14	$SO_2+0.5O_2 \rightarrow SO_3$	1.2	40.0
R15	$SO_3+H_2O+2NH_3 \cdot V \rightarrow (NH_4)_2SO_4 \cdot V+V$	0.5	0.0
R16	$(NH_4)_2SO_4 \cdot V \rightarrow NH_4HSO_4 \cdot V+NH_3$	2.3×10^{10}	1.1×10^2
R17	$2NH_4HSO_4 \cdot V \rightarrow (NH_4)_2S_2O_7 \cdot V+H_2O+V$	7.9×10^9	1.3×10^2
R18	$3(NH_4)_2S_2O_7 \cdot V \rightarrow 2NH_3+2N_2+6SO_2+9H_2O+3V$	7.2×10^{12}	1.6×10^2

numerical simulation and experimental test values of the original engine cylinder pressure under 100% engine load and the results are shown in Fig. 5. Under 100% engine load, the simulated value of the original cylinder pressure agrees well with the test value, and the calculation precision meets the actual work's requirements.

Table 2 Main technical parameters of a low-speed diesel engine

Project	Data
Type	Two stroke, turbocharging, and intercooling
Cylinder diameter	460 mm
Piston stroke	1932 mm
Connecting rod length	1972 mm
Compression ratio	21
Number of cylinders	6
Rated speed	129 r/min
Rated power	8282 kW
Rated fuel consumption rate	172.75 g/(kW·h)
Crankshaft steering	Anti-clockwise
Firing order	1-5-3-4-2-6

Table 3 shows the simulation data and bench test data of the low-speed diesel engine performance parameters under different engine loads. As shown in Table 3, the relative errors of the relevant parameters of the diesel engine are less than 5% at 25%, 50%, 75%, and 100% engine loads, respectively. Amid them, the speed corresponding to the four loads is 81 r/min, 102 r/min, 117 r/min, and 129 r/min, respectively.

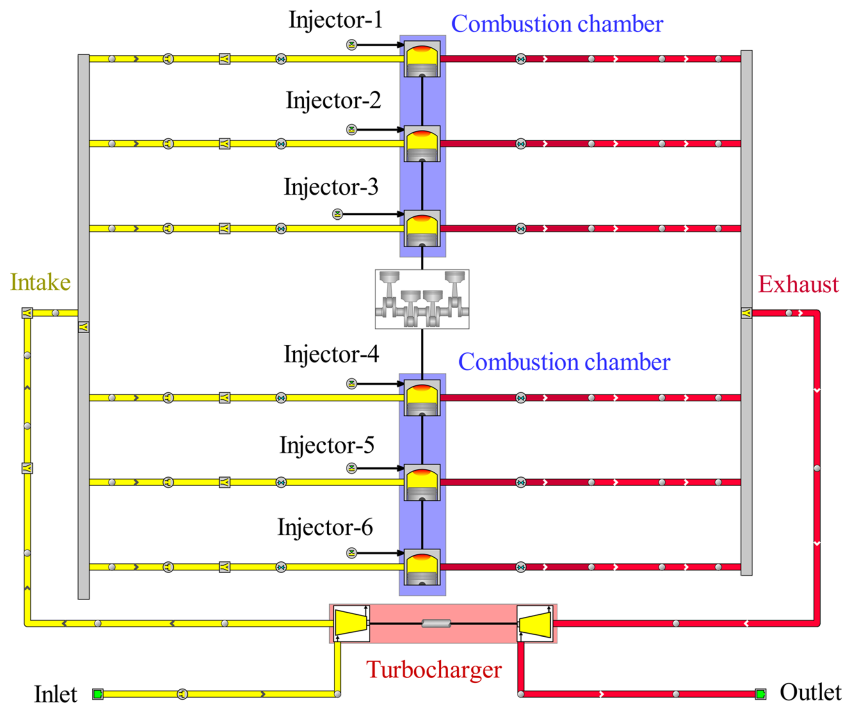
Table 4 shows the comparison of the simulation data and the bench test data of the low-speed diesel engine exhaust parameters under different engine loads. From Table 4, we can see that the relative error between the experimental value and the simulated value of the concentration of each component in the exhaust of diesel engine is within 2% and the calculation accuracy of this model can meet the design requirements of the high-pressure SCR system.

Building model and performance simulation of high-pressure SCR system

Building model of high-pressure SCR system

Based on the original model of the marine low-speed diesel engine, combined with the operating parameters of the high-

Fig. 4 Simulation model of diesel engine



pressure SCR system, the high-pressure SCR system model is embedded in the exhaust system of original model to analyze the performance of the high-pressure SCR system. The SCR reactor is arranged in front of the diesel engine turbine and is connected with the diesel engine model by two Flow Circuit Splitter modules. The two different working processes are also calculated. The configuration model is constructed as shown in Fig. 6.

Performance analysis of the high-pressure SCR system

At present, the marine low-speed diesel engine fuels can be divided into the low-sulfur fuel and the high-sulfur fuel. Because of the difference in fuel types, the composition and concentration of the exhaust gases would be slightly different.

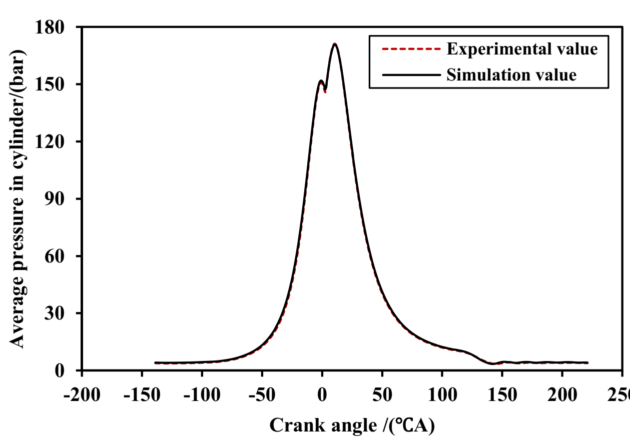


Fig. 5 Comparison diagram of indicator diagram under rated working condition

Based on these differences in the fuel, this section evaluated the performance of the high-pressure SCR system under the following two conditions:

1. High-sulfur exhaust gas condition: the diesel uses heavy fuel oil (HFO) with sulfur content of 3.5% (mass ratio).
2. Low-sulfur exhaust gas conditions: the diesel uses marine diesel oil (MDO) with no sulfur.

According to the IMO emission regulations, the diesel engine test cycle distribution was based on the operation of the ship main propulsion characteristic test mode (E3 test cycle) and the constant speed operation of the marine auxiliary test mode (D2 mode). This report selected the low-speed diesel engine as the main propulsion power and calculated the weighted average of the relevant emissions using the ISO 8178 test cycle E3 test cycle. NO*x* emission value under each load was multiplied by its corresponding weighted factor, and the sum was its weighted average value. Figure 7 shows the NO*x* emission rate of the original exhaust gas with the high-pressure SCR system under low-sulfur and high-sulfur exhaust gas conditions.

As shown in Fig. 7, the NO*x* emission rate of the original engine under 25% engine load was much higher compared with the other loads, whereas the NO*x* emission under the remaining loads was lower than that required by the IMO Tier III regulation. For the marine diesel in the paper, the required value is 3.4 g/kW·h. Furthermore, the weighted average value of NO*x* for the original engine under the condition of low-sulfur exhaust gas was 3.08 g/(kW·h), which met the requirements of the IMO Tier III regulations, while the

Table 3 Simulation data and test data of a low-speed diesel engine performance parameter

Performance parameter		Load			
		25%	50%	75%	100%
Power (kW)	Test value	2074	4141	6201	8282
	Simulation value	2093	4171	6182	8204
	Relative error	0.92%	0.73%	0.31%	0.94%
Fuel consumption (g/(kW·h))	Test value	180.03	174.61	171.84	172.75
	Simulation value	178.33	173.42	170.35	172.86
	Relative error	0.94%	0.68%	0.87%	0.06%
Exhaust flow (g/s)	Test value	19,500	15,900	11,400	7200
	Simulation value	18,999	15,403	11,547	7237
	Relative error	2.57%	3.12%	1.29%	0.51%
Exhaust temperature in front of turbocharger (°C)	Test value	282	335	365	420
	Simulation value	270	324	349	401
	Relative error	4.35%	3.32%	4.26%	4.58%
Pressure before turbocharger (bar)	Test value	1.557	2.457	3.467	4.236
	Simulation value	1.575	2.393	3.351	4.138
	Relative error	1.16%	2.59%	3.34%	2.32%

weighted average value of NO*x* for the original engine under the condition of high-sulfur exhaust gas was 3.47 g/(kW·h), being slightly higher than the requirements of the IMO Tier III regulations. Regarding the E3 cycle, the relevant calculation results of the original engine are shown in Table 5.

Therefore, this section compares the SCR denitration efficiency of the marine diesel engine under low-sulfur and high-sulfur exhaust conditions at different loads. The results are shown in Fig. 8.

From Fig. 8, under low-sulfur exhaust gas conditions, the denitration efficiency of the high-pressure SCR system was higher than 70% and the highest denitration efficiency was 84.5% at 50% engine load. In addition, under the condition of high-sulfur exhaust gas, the denitration efficiency of the high-pressure SCR system under different loads decreased and the smallest denitration efficiency was 5.5% at 25% engine load. This will happen mainly due to the exhaust gas temperature being relatively low at 25% engine load, the

Table 4 Simulation data and test data of a diesel engine exhaust parameter

Exhaust parameter		Load			
		25%	50%	75%	100%
NO (ppm)	Test value	1035	960.8	895.9	955.2
	Simulation value	1038.16	960.43	901.99	957.04
	Relative error	0.31%	0.04%	0.68%	0.19%
NO_2 (ppm)	Test value	182.6	169.6	158.1	168.6
	Simulation value	182.13	168.96	159.58	169.91
	Relative error	0.26%	0.37%	0.94%	0.78%
SO_2 (ppm)	Test value	740	830	1070	1200
	Simulation value	746.75	830	1082.39	1213.64
	Relative error	0.91%	0	1.16%	1.14%
O_2 (%)	Test value	18.02	16.73	16.15	15.16
	Simulation value	18.15	16.99	16.44	15.32
	Relative error	0.70%	1.57%	1.79%	1.05%
H_2O (%)	Test value	3.13	3.12	3.11	3.11
	Simulation value	3.13	3.14	3.12	3.12
	Relative error	0.09%	0.71%	0.39%	0.35%

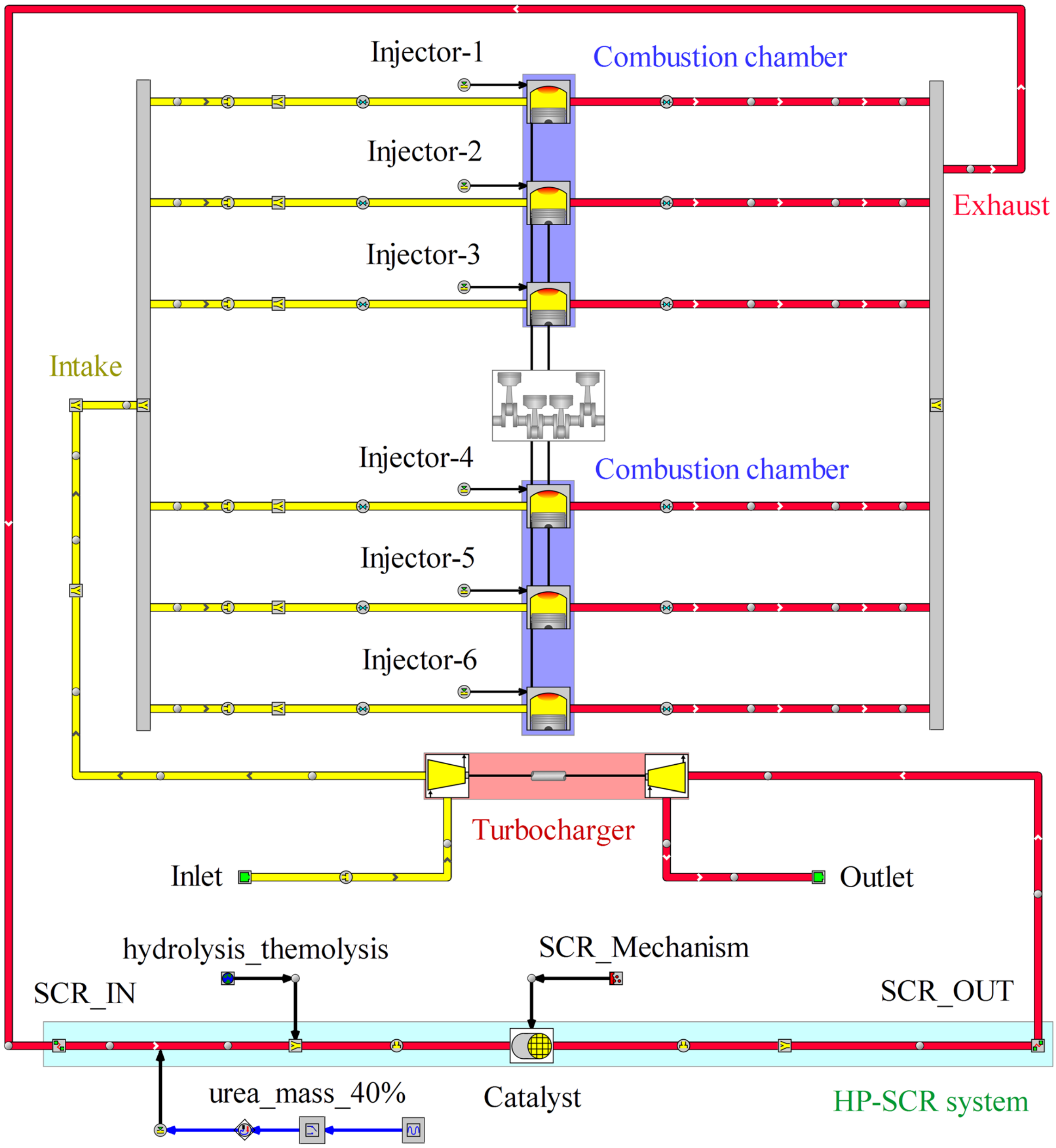


Fig. 6 The building model of the high-pressure SCR system

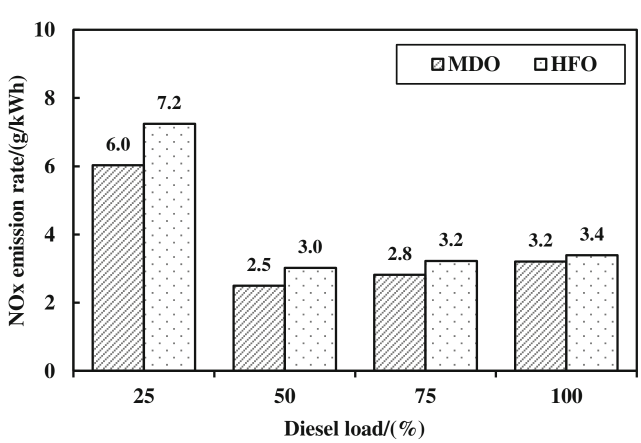


Fig. 7 The emission ratio of the SCR system under different loads

formation of some ammonium salts and other deposits, the low activity of the catalyst, and the relatively low denitration efficiency.

Analysis of the original engine modulation scheme

Original engine modulation scheme and model

Based on the calculation, the low-speed diesel engine of this type could not meet the IMO Tier III emission requirements after matching the high-pressure SCR system with the high-

Table 5 Emission parameters of the original engine with the E3 cycle

Load (%)	25	50	75	100
E3 cycle weighting factor	0.15	0.15	0.50	0.20
Power/(kW)	2074	4141	6201	8282
NOx emission under the low-sulfur exhaust gas/(g/kW·h)	6.03	2.50	2.82	3.21
NOx emission under the high-sulfur exhaust gas/(g/kW·h)	7.24	3.02	3.22	3.39

sulfur fuel. Therefore, it was necessary to adjust the working parameters of the main engine in order to improve the exhaust temperature under low engine loads. At present, the main engine matching high-pressure SCR system is modulated by the cylinder bypass valve and the exhaust gas bypass valve, which means adding the CBV and the bypass pipe in the exhaust pipe of the scavenging air receiver and the turbine, and adding the EGB and the bypass pipe in the exhaust pipe of the exhaust receiver and the turbine.

For the scavenging bypass, by opening the CBV valve, part of the scavenging gas in the scavenging air receiver is discharged directly from the upstream of the turbocharger end, which reduces the total amount of the fresh air entering the main engine cylinder and the air-fuel ratio, resulting in insufficient fuel combustion. To ensure that the output power of the main engine is constant, the fuel injection must be increased. With the increase of the fuel consumption, the exhaust gas temperature of the diesel engine will inevitably rise. For the turbine exhaust bypass, by opening the EGB valve, part of the exhaust gas after denitration is discharged directly through the turbine, reducing the amount of exhaust gas entering the turbine, the inlet pressure, and the total amount of the fresh air into the cylinder in order to raise the exhaust temperature.

In this section, CBV and EGB modulation models were constructed and embedded into the high-pressure SCR system configuration model to analyze the influence of the different modulation schemes on the original engine's performance. For the model, the opening diameter of the valve was used

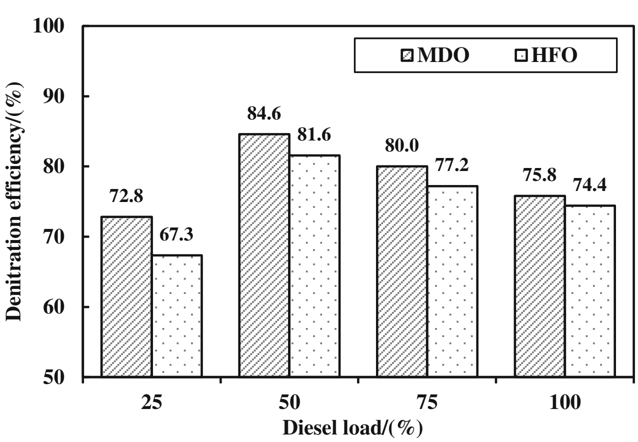


Fig. 8 Denitration efficiency of high-pressure SCR system under different loads

to change the bypass exhaust flow rate. Therefore, the valve opening was defined as the diameter ratio of the valve opening section and the exhaust pipe section. Figure 9 shows the original modulation model of the high-pressure SCR system.

Influence of the modulation scheme on the original engine exhaust parameters

Figure 10 shows the variation of the inlet exhaust temperature of the SCR system with the valve opening under different modulation schemes at 25% engine load. From Fig. 10, when the CBV and the EGB valve opening increases, the inlet exhaust gas temperature of the SCR system is also increased. However, compared with the CBV scavenging bypass modulation, the influence of the EGB turbine exhaust bypass modulation on the exhaust gas temperature was more obvious. For vanadium-based catalysts, SO_2 had less impact on the activity of the catalyst when the exhaust gas temperature was higher than 325 °C, and the denitration efficiency of SCR system was higher. Therefore, only when the CBV valve opening is 0.4 or the EGB valve opening is 0.3, the inlet exhaust gas temperature of the SCR system would be higher than 325 °C, thereby meeting the high operating temperature requirements of this system under the high-sulfur exhaust gas conditions.

Figure 11 shows the variation of the inlet exhaust pressure of the SCR system with the valve opening under different modulation schemes at 25% engine load. As can be seen from Fig. 11, when the CBV and EGB valves are opened, the inlet exhaust gas pressure of the SCR system decreases with the increase of the valve opening and the change in the exhaust gas pressure is small when the CBV valve is opened. Besides, when the CBV valve opening is 0.4, the pre-turbine pressure decreases from 155 to 153 kPa, concluding a reduction by 1.3% and when the EGB valve opening is 0.3, the pre-turbine pressure decreases from 155 to 139 kPa—a drop of 10.3%.

The influence of the modulation scheme on the emission performance of the original engine

After the original engine was modulated, the exhaust gas temperature of the pre-turbine increased, and the denitration efficiency of the SCR system also changed. Figure 12 shows the variation of the inlet exhaust gas temperature with the valve opening for different modulation schemes at 25% engine load. It can be seen from Fig. 12 that when the CBV and the EGB

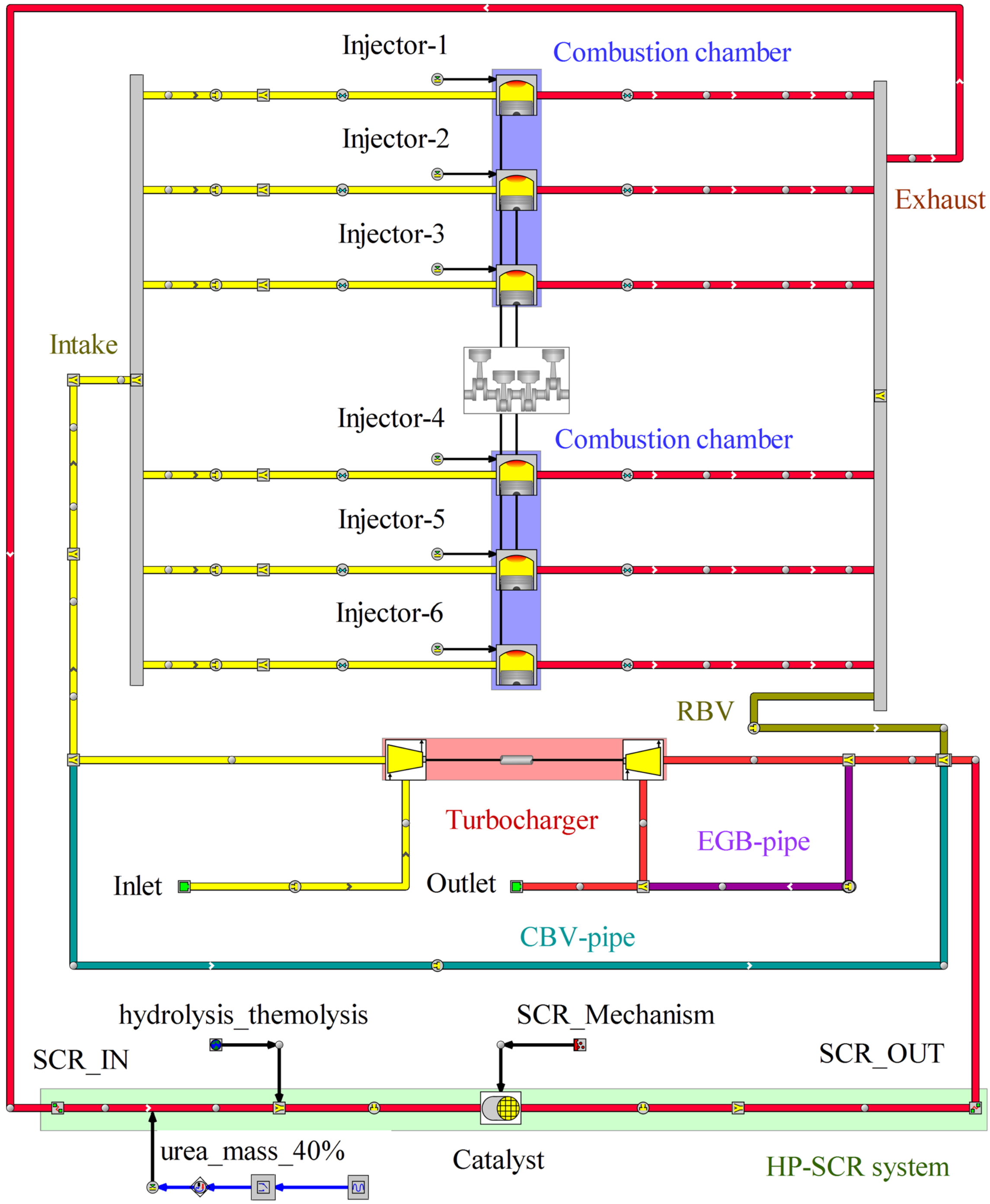


Fig. 9 Model diagram of the original engine modulation for the high-pressure SCR system

valves are opened, the denitration efficiency of the high-pressure SCR system increases as the valve opening increases. Since the change in denitration efficiency is mainly caused by the change in the exhaust gas temperature, the two trends are the same. In addition, when the CBV valve opening is 0.4, the denitration efficiency of the SCR system rises to 77% and when the EGB valve opening is 0.3, the denitration efficiency of the SCR system increases up to 82.6%.

Figure 13 shows the variation of the NO*x* emission rate of the original modulation scheme with the valve opening for different modulation schemes at 25% engine load. It can be seen from Fig. 13 that when the CBV and the EGB valve are opened, the NO*x* emission rate of the original exhaust gas decreases as the valve opening increases. When the CBV valve opening is 0.4, the specific emission ratio of the original exhaust gas is 5.61 g/(kW·h) at 25% engine load, and the NO*x*

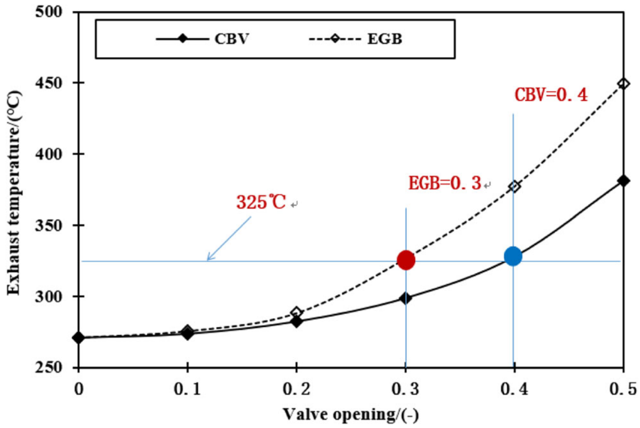


Fig. 10 The variation of the inlet exhaust gas temperature of the SCR system with the valve opening

weighted average of the original exhaust gas is 3.38 g/(kW·h) when using the E3 test cycle. Moreover, When the EGB valve opening is 0.3, the specific emission ratio of the original exhaust gas is 4.22 g/(kW·h) at 25% engine load and the NO*x* weighted average of the original exhaust gas is 3.31 g/(kW·h) when using the E3 test cycle. In both cases, the NO*x* weighted average of the original exhaust gas meets the IMO Tier III regulations. Therefore, both modulation schemes can meet the requirements of the marine low-speed diesel engine when the CBV valve opening is 0.4 and the EGB valve opening is 0.3.

The influence of the modulation scheme on the working performance of the original engine

Both modulation schemes affected the power and economy of the original engine. Figure 14 shows the variation of the original engine power with the valve opening for different modulation schemes at 25% engine load. When the CBV and the EGB valves are opened, the original exhaust gas power slowly decreases as the valve opening increases. When the CBV

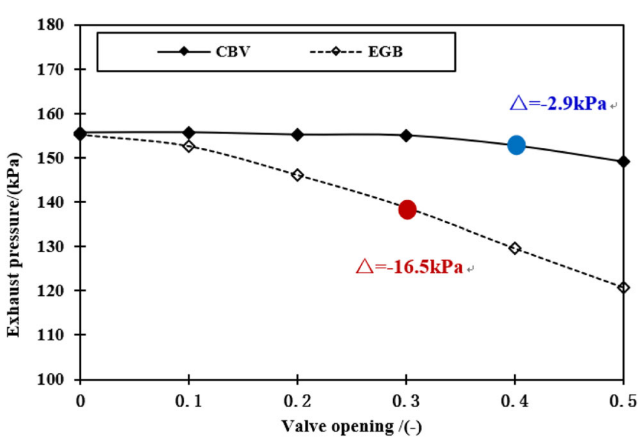


Fig. 11 The variation of the inlet exhaust gas pressure of the SCR system with the valve opening

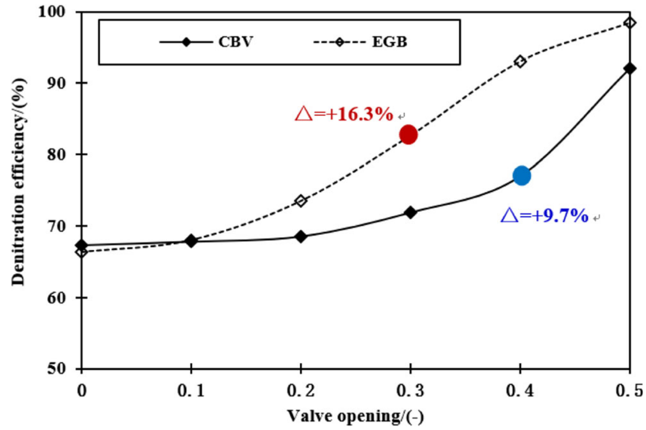


Fig. 12 Variation of the inlet exhaust gas temperature of SCR system with valve opening

valve opening is 0.4, the original engine power decreases from 2093 to 2043 kW, which means a reduction by 2.4%. When the EGB valve opening is 0.3, the original engine power decreases from 2093 to 2078 kW, which means a reduction by 0.7%.

Figure 15 shows the variation of the original engine oil consumption with the valve opening for different modulation schemes at 25% engine load. As can be seen from Fig. 15, when the CBV and the EGB valves are opened, the original engine oil consumption also increases. In addition, when the CBV valve opening is 0.4, the original engine fuel consumption increases from 178.1 to 179.8 g/(kW·h), which means an increase by 1.0%. Moreover, when the EGB valve opening is 0.3, the original engine fuel consumption is 178.1 g/(kW·h) that rises to 182.6 g/(kW·h), showing an increase of 2.5%.

Optimization scheme of the high-pressure SCR system matching

In order to improve the efficiency of the high-pressure SCR system under low engine loads, the performance of the original engine is modulated by the scavenging bypass and the

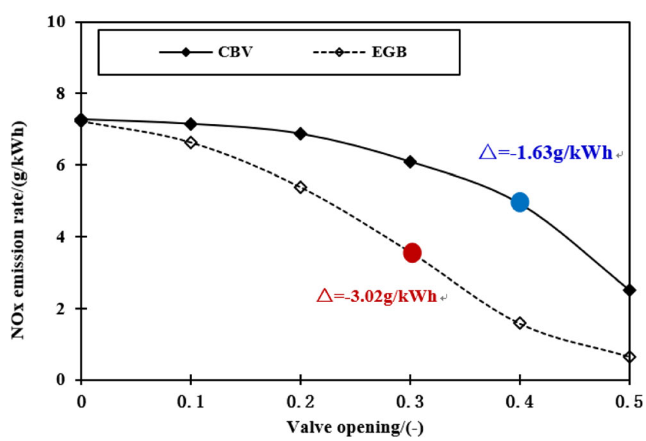


Fig. 13 The variation of NOx emission rate of the original engine exhaust with valve opening

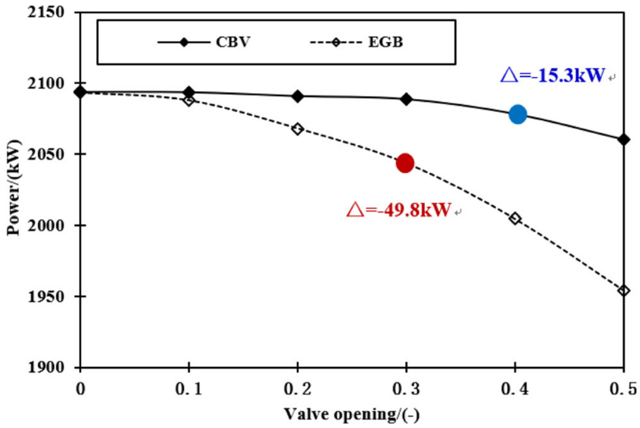


Fig. 14 The variation of the original engine power with the valve opening

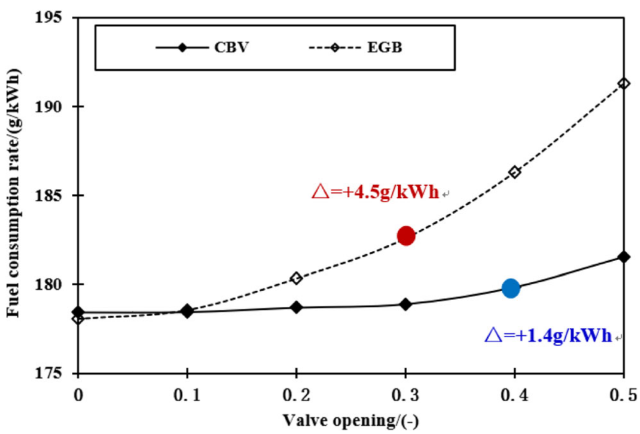


Fig. 15 The change of the original engine oil consumption with the valve opening

turbine exhaust bypass, respectively. The modulation scheme is shown in Table 6. Since the exhaust temperature and denitration efficiency of the original engine can meet the working requirements under high load, the bypass modulation is not applied. Under low engine load (25%), the exhaust gas temperature of the original pre-turbine is too low, which is easy to produce sulfate and nitrate blocking the catalyst pore, and this decreases the efficient denitration of the SCR system. The NO*x* emission value of the original turbine cannot meet the requirements of the IMO Tier III regulations, so it is necessary to make bypass modulation.

At 25% engine load, when the CBV valve opening is 0.4, the original NO*x* weighted average is 3.31 g/(kW·h), the power reduces by 0.7%, and the fuel consumption increases by 1.0%. When the EGB valve opening is 0.3, the NO*x* weighted average of the original exhaust gas is 3.31 g/(kW·h), the power decreases by 2.4%, and the fuel consumption increases by 2.5%. Therefore, both modulation schemes can achieve the original emission performance of the IMO Tier III emission regulations, but the scavenging bypass modulation has less impact on the performance of the original engine.

Conclusions

Using one-dimensional simulation software, this paper constructed the original model of the marine low-speed diesel engine and the matching model of the high-pressure SCR system, analyzed the matching performance of the high-pressure SCR system under low-sulfur and high-sulfur exhaust gas conditions, and studied the main engine modulation scheme of the original engine matching with the high-pressure SCR system. The main work and conclusions are as follows.

Based on the original model of the marine low-speed diesel engine, the high-pressure SCR system model was embedded in the exhaust system of the original model. The engine's performance of the high-pressure SCR system with light fuel (low-sulfur exhaust gas, no sulfur) and heavy fuel (high-sulfur exhaust gas, 3.5% S) was analyzed. The results showed that the NO*x* emission rate of the original engine under 25% engine load was much higher than that of the other loads, and the NO*x* emission under the remaining loads was lower than that required by the IMO Tier III regulation. Furthermore, the weighted average of NO*x* under the condition of low-sulfur exhaust gas was 3.08 g/(kW·h) when using the ISO8717E3 test cycle, which met the requirements of the IMO Tier III regulation; and the NO*x* emission under the condition of high-sulfur exhaust gas was lower than that required by the IMO Tier III regulation. The NO*x* weighted average value of the original engine

Table 6 NOx emission value of the marine diesel with HP-SCR system by CBV and EGB modulation (g/kW·h)

Load(%)	25	50	75	100
Original engine	7.24	3.02	3.22	3.39
CBV modulation	5.61	-	-	-
EGB modulation	4.22	-	-	-

was 3.47 g/(kW·h), slightly higher than that of the IMO Tier III regulations.

Based on the configuration model of the high-pressure SCR system, the modulation models of the cylinder bypass valve (CBV) and the exhaust gas bypass valve (EGB) were constructed. The effects of the scavenging bypass and the turbine exhaust bypass modulation schemes on the emission performance and performance of the original engine were analyzed, and the modulation schemes suitable for the marine low-speed diesel engine were obtained. It was found that at 25% engine load when the CBV valve opening was 0.4, the weighted average NO*x* of the original exhaust gas was 3.38 g/(kW·h), the power decreased by 0.7%, and the fuel consumption increased by 1.0%, and when the EGB valve opening was 0.3, the weighted average value of NO*x* was 3.31 g/(kW·h), the power was reduced by 2.4%, and the fuel consumption was increased by 2.5%. Both modulation schemes met with the requirements of the IMO Tier III emission regulation, but the scavenging bypass modulation scheme had less impact on the original engine performance.

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