Multiobjective optimization of the cooling system of a marine diesel engine

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Abstract
An intelligent cooling system directly influences the thermal load of high-temperature components, heat distribution, and fuel economy of a diesel engine. An optimal coolant pump rotational speed map is a key factor in intelligent cooling control strategies. In this study, we designed an experimental variable coolant flow system for a maritime diesel engine. Experiment design and D-optimal designs were used to optimize the parameters of the diesel engine cooling system. The diesel engine speed, load, and freshwater rotational pump speed were selected as variables. The temperature of the high-thermal-load zone of the combustion chamber components, fuel consumption rate, effective power, and peak cylinder pressure were selected as response variables, and the D-optimal method was used to sample the experimental points. Polynomial response surface models were obtained using a stepwise algorithm. A multiobjective optimization problem was converted into a simple-objective optimization problem using the ideal point method. A genetic algorithm was used to optimize the single-objective function globally to obtain the optimal freshwater pump speed map for a diesel engine under all conditions. On average, the optimized cooling system decreased the fuel consumption by 1.901%. Six typical propulsive conditions were selected to confirm the validity of the optimization results. The experimental results indicate that the fuel consumption decreased by 2.35%, the effective power increased by 2.26%, and the power consumption of the water pump decreased by 17.83%. The combination of experiment design and D-optimal designs offers the advantages of low cost, high efficiency, and high precision in solving multiobjective optimization problems involving strong coupling and nonlinear systems. The results of this research provide data support and a theoretical basis for intelligent cooling control strategies.

KEYWORDS
design of experiments, diesel engine cooling system, D-optimal designs, fuel consumption rate, multiobjective optimization
1 | INTRODUCTION

Diesel engines are widely used in ships as their main power plant. Considerable research has been focused on improving the thermal efficiency and fuel consumption savings of diesel engines. Numerical studies have presented various approaches to optimizing the heat distribution of diesel engines to decrease their fuel consumption and pollutant emissions. New control strategies and combustion models, such as homogeneous-charge compression ignition (HCCI) are used in internal combustion engines (ICE) to improve the effective thermal efficiency and reduce NOx and particle matter emissions. During the compression stroke, the homogeneous fuel charge reaches the auto-ignition temperature near the top dead center and auto-ignites simultaneously at several locations in the cylinder. To create a homogeneous charge, fuel is injected into the preheated air, either by port fuel injection (PFI) or direct injection (DI) into the cylinder in an HCCI engine. In this engine type, certain advanced techniques and control strategies, such as fuel management, homogeneous-charge preparation, and exhaust gas recirculation are widely used to achieve an optimal combustion phase and heat release rate for the entire operating cycle of HCCI engine. A modification of fuel was achieved through vaporization of liquid petrol coupled with water electrolysis of gas and air, to address the problem of a delayed heating rate under the low-load conditions of the HCCI engine. The results indicated that fuel consumption greatly decreased in comparison to the case in which liquid fuel was used. Nomura et al. tested the effects of a methane additive and the compression ratio on an HCCI engine fueled by dimethyl ether (DME). The results indicated that optimizing the DME/methane ratio was beneficial in achieving a higher output power in comparison to DME–HCCI combustion. Lu et al. combined petrol and diesel in an HCCI engine and found that the homogeneous mixture formation timescale and ignition timing could be controlled both macroscopically and locally in the combustion chamber by the stratification of fuels, which was beneficial in achieving high thermal efficiency and low emissions. The homogeneous mixture was composed of vaporized gases and fresh air from outside the cylinder. And the results indicated that the NOx and smoke emissions were significantly reduced in comparison with those resulting from the conventional mode of operation. Hot exhaust gas recirculation (EGR) can heat the mixture in the cylinder and promote fuel evaporation. Because of the thermal, diluent, and chemical effects of EGR, it can decrease the maximum combustion temperature, thereby reducing the NOx emissions of HCCI engines. Spark-assisted compression ignition (SACI) in an HCCI engine can not only make it operate under lean mixture conditions and provide control of the combustion phase but also maintain high thermal efficiency and low NOx and particulate matter (PM) emissions simultaneously. NOx generation occurs mainly in high-temperature and oxygen-rich environments. Certain measures to reduce the combustion temperature can decrease nitrogen oxide (NOx) emissions, but this also lowers soot oxidation, causing the serious deterioration of PM emission. Conventional diesel engines operate in diffusion combustion mode, wherein the chemical reaction rate of fuel is much higher than the mixing and diffusion rates of fuel and air. The distributions of the fuel-air mixture and temperature in the cylinder are uneven. Therefore, NOx emissions occur in the high-temperature region of the outer-field diffusion flame, and PM emissions are produced in the inner high-temperature region that is low in O2. Under the low speed and load conditions of a diesel engine, the cooling intensity of a traditional cooling system is too high, which leads to a low temperature in the combustion chamber and at the combustion chamber wall. Therefore, the atomization quality of fuel injection becomes inadequate and aggravates, the nonuniformity of the fuel–air mixture. In the process of fuel combustion, the combustion chamber has both local high-temperature regions that are oxygen-rich and local high-temperature regions that lack oxygen, which together result in increased NOx and PM emissions. Therefore, the combustion control for an HCCI engine involves controlling the cooling system, which is influential in improving fuel economy and reducing NOx and PM emissions. Furthermore, the outer coolant temperature of the ICE has a considerable effect on hydrocarbon (HC) emissions. The evaporation of fuels is affected by the in-cylinder temperature of the engine: a low in-cylinder temperature results in a poor homogeneous mixture. Therefore, the fuel is traditionally enriched in the engine to produce a relatively richer mixture, which leads to the ignition of a small amount of vaporized fuel, thus significantly sacrificing fuel economy and increasing hydrocarbon (HC) and carbon monoxide (CO) emissions. Guo et al. tested the effects of different outer coolant temperatures on the HC emissions of a turbocharged gasoline direct-injection (TGDI) engine. The results indicated that HC emissions decreased with increasing outlet coolant temperature, for the following reasons: First, the wall temperature of the combustion chamber increased with the coolant temperature, which is conducive to fuel evaporation. With increasing coolant water temperature, the fuel–air mixing rate increased; thus, a homogenous mixture could easily form in the cylinder. Second, the oil film thickness naturally decreased after the blended fuel impinged on the wall, and the HC emissions decreased significantly because of absorption by the lubricating oil. Third, the propagation of the turbulent flame could approach the wall of the combustion chamber because of the decreased heat loss through the wall, thereby reducing the flame quenching distance.
Thus, the flow of coolant water recirculation in the engine should be precisely controlled to maintain a suitable temperature of the coolant during engine operation, which apparently decreases HC formation, especially during cold starts and cold weather.

In many of these studies, changes in engine cooling and thermal needs were considered. The analysis of heat balance includes the study of the processes of energy conversion and heat transfer in an engine from the perspective of system integration, which can provide a basis for improving the cooling system and thermal efficiency of engines. The cooling system plays an important role in controlling the thermal load, heat balance, and performance of a diesel engine. A traditional water pump shaft is connected directly to a crankshaft. The pump rotational speed depends entirely on the engine’s rotational speed and is not related to the cooling demand. With the development of electronic control technology, intelligent cooling has become the main development trend in modern diesel engine cooling systems. Intelligent cooling can provide accurate coolant flow in accordance with cooling demand, achieve intelligent control of heat balance, and improve thermal efficiency.

For example, researchers have attempted to optimize engine performance by using an electronic cooling water pump and regulating the pump rotational speed in accordance with the cooling demand. In comparison to conventional water pumps, electrical water pumps have been demonstrated to improve engine function. The modification coolant system (MCS) enables a reduction in the coolant flow rate by more than 15.2% under steady-state conditions in comparison to the original system flow rate under new European driving cycle (NEDC) operating conditions. A control model for the cooling system was developed using an electric water pump, an electric fan, and a heated thermostat. The results show that fuel consumption decreases by 1.1% under the NEDC cycle operation in comparison to a conventional cooling system. For example, the Hyundai Motor Company of Korea produces a car with a radiator and condenser-cooling fans that regulate the air volume depending on the cooling demand, and the temperature of the coolant and air-conditioning condenser are jointly regulated in multiple stages. Fan power consumption is decreased by 90%, and fuel consumption is decreased by 10%.

Properly reducing coolant flow can optimize heat distribution, improve thermal efficiency, and reduce toxic exhaust emissions, but it may also cause excessive thermal loads on high-temperature parts, such as the piston and cylinder head. The optimal coolant flow map (or coolant pump speed map) is a key factor in an intelligent cooling control strategy. In a recent study, a simulation model of an engine cooling system was established using the GT-Cool software. Based on the premise of meeting in-cylinder heat dissipation requirements and taking the minimum power consumption of the cooling system as the goal, an optimal control strategy for matching the coolant pump speed with the electric fan speed was obtained by numerical simulation. The goal of “regulating the cooling intensity on demand” and reducing the power consumption of the cooling system was thereby achieved.

The thermal management intelligent system (THEMIS) and Cool Master, which regulate the pump rotational speed based on the water temperature, have been developed by the firm Valeo. Research shows that the heat distribution can be optimized based on the cooling intensity, reducing the cooling losses and improving the thermal efficiency of a diesel engine. Optimization of the cooling intensity also keeps the lubricants at the optimal working temperature, which reduces friction losses between the friction pairs of the diesel engine and improves the mechanical efficiency of the diesel engine.

Considerable research on intelligent cooling control strategies has been based on simulation modeling, taking the outlet coolant temperature as a safety index of thermal load and the minimization of power consumption of the cooling system as an optimization objective to obtain the optimal pump (fan) rotational speed map. Numerical simulations can improve the efficiency of problem-solving. However, a diesel engine is a strong coupling and nonlinear system, which can lead to errors between the simulation results and the actual state. To a certain extent, the outlet coolant temperature can reflect the thermal load state of the cooling water jacket of the cylinder head, but it cannot accurately reflect the thermal load of the high-temperature areas of the cylinder head. Taking the optimization objective to be minimizing the power consumption of the cooling system only reduces the fuel consumption of the cooling components; it does not fully consider the improvement of the power and fuel economy of the engine from the perspective of heat balance.

In this study, we designed an experimental system for coolant flow control for a marine diesel engine. The principles of design of experiments (DoE) were used to develop an approach to multiobjective optimization of the characteristic field of the cooling system. D-optimal designs were adopted for use in sampling experiment points, stepwise algorithm fitting response surface model, and a global genetic algorithm (GA) for obtaining optimal solutions. The power and fuel economy of the diesel engine were taken to be the optimization objectives for the purpose of calculating the optimal

![Figure 1](image.png)
cooling water pump rotational speed map. The accuracy of the calculation outcome and the effectiveness of the algorithms were verified experimentally.

2 | DIESEL ENGINE EXPERIMENT SYSTEM

2.1 | Heat balance theory

Most marine diesel engines are equipped with exhaust-driven turbochargers and intercoolers. Figure 1 shows the distribution of the total heat released by the fuel combustion in the cylinder.

Equation (1) shows the total heat distribution.

\[ Q_t = Q_e + Q_w + Q_f + Q_{res}, \]  

where \( Q_t \) is the total heat released by fuel combustion, kJ/h; \( Q_e \) is the heat converted into effective power, kJ/h; \( Q_w \) is the heat removed by the coolant (including the heat removed by the freshwater \( Q_{we} \) and the intercooler \( Q_{wi} \)), kJ/h; \( Q_f \) is the heat removed by the exhaust gas, kJ/h; and \( Q_{res} \) is the lost heat, kJ/h.

\[ Q_t = M_{fu} \cdot H_u, \]  

where \( M_{fu} \) is the mass flow rate of fuel (kg/h) and \( H_u \) is the low calorific value of fuel (kJ/kg).

\[ Q_e = 3.6 \times 10^3 P_e, \]  

where \( P_e \) is the effective power, kW.

In fact, the combustion efficiency of the fuel cannot reach 100% because some unburnt fuel is directly introduced into the exhaust gas. Therefore, the exhaust heat of the diesel engine consists of two components: the exhaust gas heat from the complete combustion of fuel and the heat from the incomplete combustion of fuel. The sum of the two heat components can be calculated empirically. The total heat of the exhaust gas can be calculated using Equation (4):

\[ Q_f = M_f (T_f - T_a) C_{paf}, \]  

where \( M_f \) is the mass flow rate of the exhaust gas (kg/h), \( T_a \) is the ambient temperature (K), \( T_f \) is the temperature of the exhaust gas at the turbine outlet (K), and \( C_{paf} \) is the specific heat at a constant exhaust gas pressure of 1.09924 kJ/(kg·K).

Because of the high temperature of the exhaust gas, it is difficult to measure its mass flow rate. The sum of the mass flow rate of the fuel and inlet air indicates the flow rate of the exhaust gas:

FIGURE 2 Schematic illustration of the experiment system. A, Site layout of the experiment system. B, Electronic control unit
\[ M_f = M_{fu} + M_{ru} \tag{5} \]

where \( M_{fu} \) is the mass flow rate of the fuel (kg/h) and \( M_{ru} \) is the mass flow rate of the intake air (kg/h).

\[ Q_w = M_w (T_{wout} - T_{win}) \cdot C_{pw}, \tag{6} \]

where \( M_w \) is the mass flow rate of the coolant (kg/h), \( T_{wout} \) is the coolant temperature at the engine outlet (K), \( T_{win} \) is the coolant temperature at the engine inlet (K), and \( C_{pw} \) is the specific heat at a constant coolant pressure, kJ/(kg·K).

\[ Q_{res} = Q_t - (Q_e + Q_w + Q_f). \tag{7} \]

### 2.2 Overall design of experiment system

The research object was a six-cylinder in-line, supercharged, water-cooled, four-stroke, marine high-speed diesel engine. The rated speed was 1800 r/min, the rated power was 690 kW, and the maximum cylinder pressure was 17.0 MPa. The cooling system consisted of a high-temperature freshwater system and a low-temperature seawater system. The seawater pump and freshwater pump of the original engine cooling system were of the traditional type (diesel engine-driven pump). In the controllable coolant flow experiment system, the seawater and freshwater pumps were driven by electric motors. An electronic control unit transmitted the control signals to the frequency converter, which controlled the rotational speed of the water pump.

Figure 1 is a schematic illustration of the experimental system. T, P, and Q represent the temperature sensor, pressure sensor, and flow sensor, respectively, of the corresponding positions in Figure 2. The experimental system was primarily composed of a motor-driven freshwater pump, a motor-driven seawater pump, an electronic proportional valve, a heater, an electronic control unit (ECU), a data acquisition system, frequency converters, flow sensors, temperature sensors, and sensing wires. The spool position of the electronic valve regulated the freshwater flow into the heater and radiator to maintain the engine inlet freshwater temperature at approximately 343 ± 2 K. The ECU was used to send control signals. The flow sensor sends the freshwater flow signal to the ECU, and the ECU judges the difference between it and the target flow. When the freshwater flow is higher (lower) than the target value, the frequency converter reduces (increases) the rotational speed of the motor-driven freshwater pump to reduce (increase) the water flow. The motor-driven seawater pump was regulated by the ECU to control the temperature of the charged air after the intercooler. The layout of the experimental system is shown in Figure 3. The experimental apparatus is summarized in Table 1.

### 2.3 Temperature measurement of cylinder head and piston

The cylinder heads of the diesel engine were unit type; therefore, one of the cylinder heads was selected for research. According to the experimental results, the flow rate (Q) and flow rate nonuniformity (Φ) of each cylinder under rated conditions are listed in Table 2. The water flow through the first and third cylinders was the lowest, which indicated that the thermal load of the first cylinder was the highest under the same thermal boundary conditions, so the piston and cylinder head of the first cylinder were selected as the research objects.

The piston temperature was measured using a storage steady-state temperature-measuring device. The device consisted of an integrated circuit, which was sealed in a block and fixed in the piston pin boss, moving with the piston. A real-time clock chip, data memory, and batteries were integrated into the sealing block. Omega TT-K-40 thermocouples were used to measure temperature. The temperature signals were transmitted and stored in the memory during the experimental process. After the experiment, the experimental data were exported using special software. The device did not require an external signal line, which is convenient for measurement. In the transmission process, signal interference was avoided; thus, the measurement accuracy was higher. As shown in Figure 4, there are five representative measurement points. Point 1 is located in the middle of the top piston ring land, point 2 is located in the inlet valve relief, point 3 is located in the central protrusion area of the piston top, point 4 is located in the piston bowl, and point 5 is located in the exhaust valve relief.

Armored thermocouples were used to measure the cylinder head temperature. Mounting holes were machined on the cylinder head to install the thermocouples, and C2 glue was injected into the hole to fix the thermocouples. The installation holes were machined on the cylinder head, and the sheathed thermocouples were fixed in holes with C2 glue. As shown in Figure 5, five representative measurement points are arranged on the piston. As shown in Figure 4, there were five representative measuring points. Owing to the complex internal structure of the cylinder head, the uneven distribution of the heat source leads to severe thermal stress. Too many holes drilled in the same cylinder head easily lead to thermal fracture failure. Therefore, the third cylinder head was selected to install points A and B, and the first to install points C, D, and E. Point A is located in the bridge area between the two intake valves, point B is located in the bridge area between the intake valve and the exhaust valve, point C is located in the bridge area between the two exhaust valves, point D is located at the edge of the exhaust valve, and point E is located near the fuel injector mounting hole.
point was divided into the upper and lower layers. The upper and lower points were 16 and 9 mm away from the cylinder head fire-face, respectively.

### 3 | ORIGINAL ENGINE HEAT BALANCE EXPERIMENT

The original engine heat balance experiment was carried out to study the matching characteristics between the engine and the cooling system, and the high heat-load areas of diesel engine high-temperature components under propulsive conditions. The experimental propulsive conditions of the diesel engine were 1134 r/min (172 kW), 1429 r/min (344 kW), 1637 r/min (517 kW), 1740 r/min (619 kW), 1800 r/min (691 kW) and 1860 r/min (762 kW), corresponding to 25%, 50%, 75%, 90%, 100% and 110%, respectively. During the experiment, the electric seawater pump and electric freshwater pump were simulated as a traditional type through ECU control. The ratio of the freshwater pump rotational speed to engine speed was 1.47, and the ratio of seawater pump rotational speed to engine speed was 1.67. The ambient temperature was 27°C, the inlet temperature of the coolant water was 343 ± 2 K, and the inlet temperature after the intercooler was

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**FIGURE 3** Site layout of the experimental system and electronic control unit

**TABLE 1** Experiment apparatus

| Parameter                                      | Apparatus                          | Apparatus model | Range       | Accuracy   |
|------------------------------------------------|------------------------------------|-----------------|-------------|------------|
| Effective power (kW)                          | Hydraulic dynamometer              | LiantongYP1900  | 0-1900      | 0.2%       |
| Diesel engine speed (r/min)                    |                                    |                 | 0-3000      | 0.5%       |
| Diesel engine torque (N·m)                     |                                    |                 | 0-17200     | 0.2%       |
| Pressure of the intake and exhaust system (MPa) | Pressure transmitter               | RZS-3351GP     | 0-10        | (0.0029r+0.071)% |
| Pressure of the cooling system (MPa)           |                                    |                 |             |            |
| Pressure of the lubricating oil system (MPa)   |                                    |                 |             |            |
| Inlet gas temperature (°C) before / after intercooler | Temperature sensor | PT100   | 0-200       | 0.2%       |
| Sea / freshwater system temperature (°C)      |                                    |                 |             |            |
| Gas temperature before the turbocharger (°C)   |                                    |                 |             |            |
| Lubricant system temperature (°C)              |                                    |                 |             |            |
| Fuel consumption rate (kg/h)                   | Fuel consumption meter             | ToCeil-CMFD020  | 0-200       | ±0.12%     |
| Gas flow rate (kg/h)                           | Thermal mass flowmeters            | ToCeil20N200    | 0-4800      | 1%R        |
| Exhaust temperature of inlet and outlet turbine (°C) | Temperature sensor | K-Thermal couple | 0-1000      | 0.4%       |
| Sea / freshwater system flow rate (L/min)      | Electromagnetic flowmeter          | LDG             | 15-900      | 0.3%R      |
| Smoke number (100%) (BOSCH)                    | Smoke meter                        | AVL415          |             | 0.1        |
| Cylinder pressure curve (Bar)                  | Combustion analyzer                | AVL631          | 0-300       | 0.01       |

**TABLE 2** Water flow rate and flow rate nonuniformity

| Q/Φ   | Cylinder number |
|-------|-----------------|
|       | #1  | #2   | #3   | #4   | #5   | #6   |
| Q (L/min) | 94.11  | 94.95  | 94.60  | 97.26  | 99.85  | 102.58  |
| Φ (%)   | -3.20  | -2.34  | -2.70  | 0.04  | 2.70  | 5.51  |
313 ± 2 K. The atmospheric pressure was 100 kPa, and the air humidity was 60%.

### 3.1 Variation characteristic of the fuel consumption rate under propulsive conditions

As shown in Figure 6, the fuel consumption rate decreases slowly at first and then increases rapidly with the decrease in diesel load under the propulsive characteristics of the diesel engine. The fuel consumption rate was up to 249 g/(kW·h) under a 25% rated load, which indicates that the diesel economy is poor under medium and low-load conditions and that there is room for improvement. Under high-load conditions (above 75% rated load), the fuel consumption rate was stable at 206-210 g/(kW·h), with a better fuel economy.

### 3.2 Variation characteristics of heat balance under propulsive conditions

As shown in Figure 7, when the total heat generation rate of the fuel is 100% in internal combustion engines, the effective power, exhaust losses, and freshwater cooling losses
account for approximately 85% of the total heat generation rate of the fuel. With a decrease in the engine load, the freshwater cooling losses increased gradually, and the effective power decreased gradually. At a rated load of 25%, the effective power decreased to 30.54%, and the freshwater cooling losses increased to 29.17%.

Because the freshwater pump is of a traditional type, the water flow rate cannot be regulated according to the cooling demand. Under low-load conditions, the engine was over-cooled by the cooling system, and the cooling loss was high. The fuel injection quantity and air intake flow per cycle in the cylinder were low at the low speed of the diesel engine. Excessive cooling results in a lower combustion chamber temperature; therefore, the quality of fuel injection atomization is poor, and the air–fuel mixture is not uniform. This resulted in insufficient fuel combustion and decreased thermal efficiency. The engine power and fuel economy worsened. With an increase in engine speed and load, the injection quantity and air intake flow increased, the injection atomization quality was improved, the air-fuel mixture was more uniform, and the combustion chamber temperature increased. Therefore, the fuel combustion quality was improved, the thermal efficiency was improved, and the fuel consumption was reduced.

3.3 Variation characteristics of cylinder head and piston temperature under propulsive conditions

The temperatures of each measuring point of the cylinder head are listed in Table 3. The temperatures of the lower measuring points were tens of degrees centigrade higher than those of the upper measuring points. Because the lower points are closer to the cylinder head fire surface, more combustion heat is transferred to the lower points.

The lower measurement point temperatures are shown in Figure 8. With an increase in engine load, the combustion pressure and temperature in the cylinder increased, and all measuring point temperatures increased gradually. The temperature at point C was significantly higher than that at other measuring points. Point C is located in the bridge area between the two exhaust valves. High-temperature exhaust gas is discharged through the exhaust valves, which makes the area continuously subject to the scouring effect of high-temperature and high-pressure gases. The cooling water flow through the bridge area is very low. The temperature in this area is higher due to comprehensive factors, reaching 563.9 K and 569.9 K at 100% and 110% of the rated load, respectively. Owing to the high concentration of the fuel–gas mixture near the injector, the in-cylinder combustion releases more heat. Therefore, the temperature at point E is second to that at point C. Point B is located in the bridge area between the intake and exhaust valves, whose temperature is lower than that at point E. Point D, located at the edge of the exhaust valve, is far from the combustion center. Point A, located at the bridge area between intake valves, is cooled by intake air; therefore, the temperatures of the two points are the lowest.

Each temperature measurement point of the piston is shown in Figure 9. Point 3 had the highest temperature. This is because of the amount of heat released and the significant explosion pressure produced at the moment of fuel combustion. The distance between the central protrusion area of the piston top and the flame center was small, and there was less piston material in this area. All these factors cause the temperature of this area to increase greatly. Point 3 reaches 511.6 K and 516.5 K at 100% and 110% of the rated load, respectively. Owing to the scouring effect of the combustion flame, the temperatures at points 5 and 2 were both higher and very close. The point 4 temperature was relatively low, owing to the cooling effect of the cooling oil–antrum of the piston. Point 1, far from the piston top and cooled by the cooling water jacket of the cylinder, had the lowest temperature.

The experimental results show that point C of the cylinder head and point 3 of the piston had the highest heat loads.

| Measuring point | Propulsive conditions (% rated load) |
|-----------------|-------------------------------------|
|                 | 25%  | 50%  | 75%  | 90%  | 100% | 110% |
| Lower point A/K | 449.4 | 470.2 | 481.6 | 489.9 | 499.2 | 502.6 |
| Upper point A/K | 412.0 | 424.9 | 429.5 | 433.8 | 439.0 | 441.7 |
| Lower point B/K | 456.3 | 480.3 | 495.2 | 504.1 | 514.8 | 518.8 |
| Upper point B/K | 416.4 | 432.9 | 441.1 | 446.7 | 453.3 | 456.7 |
| Lower point C/K | 485.7 | 516.1 | 534.4 | 550.0 | 563.9 | 569.9 |
| Upper point C/K | 446.1 | 468.6 | 480.8 | 490.9 | 499.9 | 504.0 |
| Lower point D/K | 447.3 | 474.9 | 487.0 | 496.2 | 504.8 | 508.3 |
| Upper point D/K | 416.4 | 435.2 | 443.3 | 450.2 | 455.5 | 458.0 |
| Lower point E/K | 469.5 | 489.0 | 504.7 | 514.6 | 522.4 | 524.2 |
| Upper point E/K | 420.5 | 433.7 | 442.5 | 448.9 | 453.4 | 454.2 |

**TABLE 3** Temperatures of cylinder head measuring points
When optimizing the cooling system, one should focus on two high-temperature areas.

4 | OPTIMIZATION PROCEDURE

DoE is a relatively new approach to optimization based on the response surface model. It includes mathematical modeling, statistical theory, and computer-aided modeling. DoE is useful in estimating the effects of independent variables on responses by establishing a relationship between inputs and outputs. The DoE optimization workflow is illustrated in Figure 10. It consists mainly of four steps: sampling experiment points, fitting the response surface model, multiobjective optimization design, and verification of the optimization results.

4.1 | D-Optimal designs

4.1.1 | Regression matrix

DoE optimal design theory is concerned with a variety of algorithms, of which the D-optimal algorithm is considered to be most suitable for engine experiments. The D-optimal design is relatively simple and practical and is a model-specific design. A D-optimal design is generated by an iterative search algorithm and seeks to minimize the covariance of the parameter estimates for a specified model. This is equivalent to maximizing the determinant $D = |X^T X|$, where $X$ is the design matrix of the model terms (the columns) evaluated at specific treatments in the design space (rows).

D-optimal designs and response surface fitting require listing the matrix of candidate points, that is, the design matrix. The design of experiment (DoE) was based on an intelligent control cooling system, in which the freshwater pump was driven by an electric motor, such that the rotational speed of the freshwater pump was decoupled from that of the diesel engine, the diesel engine speed ($x_1$), diesel load rate ($x_2$), and electric motor-driven freshwater pump speed ($x_3$) were considered as variables. The value range of $x_2$ was 20%–100% of the maximum load, which is the maximum power of the diesel engine at different engine speeds. The range of the electric motor-driven freshwater pump speed ($x_3$) was 1.47 times that of 70%–110% of the diesel engine speed ($x_1$), and the step length was 5%. The diesel engine speed ranged from 900 to 1800 r/min, and the step length was 100 r/min. Therefore, $x_3$ ranged from 70%·1.47·$x_1$ to 110%·1.47·$x_1$ ($x_3$∈[1.029·$x_1$, 1.617·$x_1$]), and the step length was 5% (0.0735·$x_1$). The levels and values of the variables are listed in Table 4.

A second-order polynomial regression model with quadratic cross-terms was used in this study, based on DoE...
modeling experience for relevant engines. The regression matrix was as follows:

\[ FX(i,j) = \Pi_{k=1}^{\text{model}(j,k)}, \]  

where \( FX(i,j) \) is an element of the regression matrix, \( X(I, k) \) is the element of the design matrix, and \( \text{model}(j, k) \) is the optional order of \( X(I, k) \).

### 4.1.2 Sampling experiment points

If the design matrix (full factor experiment) were adopted, 729 experimental points would be needed, which would result in high costs in terms of both funds and time. The number of experiments determined by DoE is closely related to the accuracy of the experimental design. The parameter that reflects the accuracy of the experimental design is the standardized variance.

\[ \delta(\xi) = \frac{\text{var}(Y)}{\sigma^2} \]  

where \( \delta \) is the standardized variance, \( \text{var}(Y) \) is the distribution variance of the design error, and \( \sigma^2 \) is the measurement error.

Figure 11 shows the relationship between the standardized variance and the number of experiments. Considering the cost of time and funds, 150 experimental points were selected. It was possible to have an average standardized variance of approximately 0.15, and the design variance was reduced to 3/20 of the measurement error. The key to DoE is to distribute the sample points evenly and reasonably. There are 729 experimental points in the design matrix, and 150 experimental points have three \( C_{150}^{729} \) possibilities. According to Fedorov’s algorithm, the D-optimal design program is compiled, as shown in Figure 12, and the D-optimal design scheme is shown in Figure 13. The values of the statistical parameters \( D \) and \( A \) of the D-optimal design results are 3.98 and 9.98, respectively.

The statistical parameters \( D \) and \( A \) are the standards for judging the error of the polynomial coefficient vector. The covariance matrix of coefficient vector \( b \) is

\[ \text{var}(b) = \frac{\sigma^2}{(FX^T FX)^{-1}} \]

where \( FX \) is the regression matrix, \( A^* \) is the adjoint matrix of \( FX^T \), and \( n \) is the number of D-optimal design points.

\[ \sigma^2 \] cannot be avoided, and the goal of the D-optimal design is to reduce the ratio of \( \text{var}(b) \) to \( \sigma^2 \). Thus, increasing the denominator \( |FX^T FX| \) can effectively reduce the error. According to Equations (11) and (12), the larger the \( D \)-value, and the smaller the \( A \)-value, the smaller is the polynomial coefficient vector error. The D-optimal sampling points were uniform and reasonable, as shown in Figure 13.

### 4.2 Response surface model

The response surface model expresses the functional relationship between variables and responses. Generally, the higher the order of the polynomial model, the stronger is the data-matching ability. However, it may also cause overfitting, which reduces the accuracy of the response surface. A second-order quadratic cross-polynomial model was adopted in this study. We used the standard regression algorithm, PRESS (Minimize PRESS), and stepwise algorithm to calculate the coefficients of the polynomial model based on linear regression theory. Three indices were used to evaluate the fitting accuracy of the response surface: \( R^2 \)-Sqr, \( \text{Adj.-Sqr} \) (adjusted-\( R^2 \)), and \( Q \)-Sqr. \( \text{Adj.R-Sqr} \) denotes the modified \( R^2 \). Compared with \( R^2 \), \( \text{Adj.R-sqr} \) can eliminate overfitting, and may more accurately reflect the fitting quality of the response surface. \( \text{Adj.R-sqr} \) ranges between 0 and 1; the closer it is to 1, the higher the fitting accuracy is.

\[ R^2 = 1 - \left( \frac{n - 1}{n - k - 1} \right) \left( 1 - R^2 \right), \]

\[ \text{adjusted } R^2 = 1 - \left( \frac{n - 1}{n - k - 1} \right) \left( 1 - R^2 \right), \]

where \( Y_{p,i} \) is the predicted response value, \( Y_{o,i} \) is the measured response value, \( n \) is the number of experiments, and \( k \) is the number of items in the polynomial model.

To improve the fitting accuracy of the response surface, different regression algorithms were used. We compare the accuracy of the third-order quadratic cross-term model and the third-order cubic cross-term model. The results are shown in Table 5. Table 5 shows that the adjusted-\( R^2 \) of the second-order quadratic cross-term model was the largest, indicating that the fitting accuracy was the highest, and the stepwise algorithm has a higher accuracy than the others.

In the D-optimal design experiment, the motor-driven seawater pump was regulated to stabilize the charged air at 35°C after the intercooler. A stepwise algorithm was used to fit the response surface model:
4.3 | Response surface model

4.3.1 | Single-objective optimization design

In most engineering practices, there is a trade-off between different objective functions. In our research, the ultimate goals are to maximize the effective power \( f_1 \) and to minimize the fuel consumption rate \( f_2 \), peak cylinder pressure \( f_3 \), temperature of point C in the cylinder head fire-face \( f_4 \), and the temperature of point 3 in the piston top \( f_5 \). However, there was a restrictive relationship between the different parameters. The main purpose of multiobjective optimization is to determine the global optimal cooling pump speed.

In this study, the ideal point method based on multiple-goal decision-making was used to transform the scattered, different dimensional, and extrema multioptimization problem into a single-objective optimization problem for overall evaluation. The problem of calculating the maximum of \( f_1, f_2 \), and the minimums of \( f_3, f_4, f_5 \) were transformed into calculating the ideal optimal value of a single function. The weights of the multiobjective functions were the same. The ideal point method was used to construct a single-objective optimization function:

\[
\begin{align*}
    f_{all} &= \sqrt{(f_1 - f_{1\text{min}})^2 + (f_2 - f_{2\text{min}})^2 + (f_3 - f_{3\text{max}})^2 + (f_4 - f_{4\text{max}})^2 + (f_5 - f_{5\text{max}})^2}.
\end{align*}
\]

where \( f_{1\text{min}}, f_{2\text{min}}, f_{3\text{max}}, f_{4\text{max}}, f_{5\text{max}} \) represent the minimum and maximum values in the design space of variables, \( f_{all} \) is the single-objective function, \( f_{1\text{min}} \) is the minimum fuel consumption rate, \( f_{3\text{max}} \) is the maximum effective power, \( f_{5\text{min}} \) is the minimum peak cylinder pressure, \( f_{5\text{min}} \) is the minimum of point C in the cylinder head fire-face, and \( f_5 \) is the minimum temperature of point 3 in the piston top.

The values of \( f_1, f_5 \) meeting \( f_{all} \) equal to the global minimal value are the optimal solutions for comprehensively evaluating the improvement of diesel power, economy, and reliability, and the corresponding freshwater pump speed which is the global optimal solution.

\[
\begin{align*}
    f_1 &= -2.7877x_1 + 1.891x_2 - 1.4504E^{-2}x_3 + 1.3540E^{-3}x_1^2 + 1.4197E^{-3}x_2^2 - 5.0272E^{-7}x_3^2 - 2.3198E^{-3}x_1 \cdot x_2 + 2.4411E^{-3}x_1 \cdot x_3 - 4.0577E^{-5}x_2 \cdot x_3 + 1.7427E^3 \\
    f_2 &= 4.4094x_1 - 3.2404x_2 + 7.6962E^{-2}x_3 - 2.2611E^{-3}x_1^2 - 1.2194E^{-1}x_2^2 - 4.9981E^{-1}x_3^2 + 4.8475E^{-3}x_1 \cdot x_2 + 2.3623E^{-1}x_1 \cdot x_3 + 1.9812E^{-4}x_2 \cdot x_3 - 2.1660E^3 \\
    f_3 &= 1.5008x_1 + 2.1981E^{-1}x_2 + 7.5575E^{-3}x_3 - 1.3251E^{-1}x_1^2 + 1.6825E^{-1}x_2^2 - 1.5247E^{-1}x_3^2 + 6.8686E^{-4}x_1 \cdot x_2 - 4.5783E^{-1}x_1 \cdot x_3 + 1.4299E^{-5}x_2 \cdot x_3 - 2.5683E^2 \\
    f_4 &= 1.4772E^1x_1 - 8.9014x_2 + 9.1018E^{-2}x_3 - 8.0578E^{-3}x_1^2 - 9.0773E^{-3}x_2^2 + 2.0908E^{-5}x_3^2 + 1.4052E^{-2}x_1 \cdot x_2 - 2.3720E^{-8}x_1 \cdot x_3 + 2.8194E^{-4}x_2 \cdot x_3 - 6.6733E^3 \\
    f_5 &= -8.5891E^{-1}x_1 - 4.1321E^{-1}x_2 - 4.1699E^{-1}x_3 + 8.1643E^{-4}x_1^2 + 1.1276E^{-1}x_2^2 + 4.1518E^{-1}x_3^2 - 1.9169E^{-3}x_1 \cdot x_2 + 1.4994E^{-1}x_1 \cdot x_3 - 4.4470E^{-1}x_2 \cdot x_3 + 5.8078E^2,
\end{align*}
\]

where \( x_1, x_2, x_3 \) are the variables; \( x_1 \) is the diesel speed, \( x_2 \) is the diesel load rate, \( x_3 \) is the freshwater pump rotational speed. \( f_1 \) represents the responses; \( f_2 \) is the fuel consumption rate, \( f_3 \) is the effective power, \( f_4 \) is the peak cylinder pressure, \( f_5 \) is the temperature of point C in the cylinder head fire-face, and \( f_5 \) is the temperature of point 3 in the piston top.

The RMSE of \( f_1, f_5 \) is 1.108, 4.230, 0.219, 6.704 and 4.377, respectively. For \( f_1 \) example, 50 experimental values were randomly selected for verification. The residual map between the calculated values and the experimental values is shown in Figure 14, and the absolute error range is [2.5, -2.8]. All indexes indicate that the fitting accuracy of the response surface is high and can be used for subsequent multiobjective optimization calculations.
To ensure the safety and reliability of the engine, some constraints should be satisfied for the present optimization. According to the technical specifications of this type of diesel engine, $x_3$ was set to be no lower than 900 r/min and no higher than 3200 r/min; $f_3$ was set to be no higher than 17.6 MPa.

The creep temperature of the cast-iron cylinder head was approximately 660 K. A cylinder head plate was proposed as a homogeneous heat conduction plate. Considering the safety margin of the thermal load on the cylinder head fire surface, Fourier's law is introduced:

$$ dq \propto dA \cdot dT/dl, $$

(18)

$$ dq = -\lambda \cdot dA \cdot dT/dl $$

(19)

where $q$ is the heat conduction rate, W; $A$ is the heat conduction area, m$^2$; $dT/dl$ is the temperature gradient, K/m; and $\lambda$ is the thermal conductivity, W/(m·K).

When the homogeneous plate is in a stable heat conduction state, $q$ does not change with time, $A$ and $\lambda$ are constants; then, Fourier's law may be simplified as follows:

$$ q = -\lambda \cdot A \frac{dT}{dl}, $$

(20)

Equation (20) is integrated as follows:

$$ q = \lambda \cdot A \cdot \frac{T_1 - T_2}{l}, $$

(21)

$$ q = \left(\frac{T_1 - T_2}{\frac{l}{\lambda \cdot A}}\right) = \Delta T / R, $$

(22)
where \( l \) is the thickness of the flat plate, \( m \); \( \Delta T \) is the temperature difference between the two sides of the flat plate, labeled as the heat conduction thermal driving force, \( K \); and \( I/\lambda \cdot A \) is the heat conduction thermal resistance \( (R) \), \( kW/\)W.

In the process of steady heat conduction in the multilayer plate, the heat conduction rate of each layer is the same:

\[
q_1 = \cdots = q_N = q, \quad (23)
\]

\[
q = \Delta T_1/R_1 = \cdots = \Delta T_N/R_N, \quad (24)
\]

\[
\Delta T_1: \cdots : \Delta T_N = R_1: \cdots : R_N, \quad (25)
\]

Equation (25) shows that the temperature drop of each layer in the multilayer plate is proportional to the thermal resistance. The temperature of the cylinder head fire-face is \( T_1 \), the temperature of the lower-layer point \( C \) (9 mm from the cylinder head fire-face) is \( T_2 \), and the temperature of the upper-layer point \( C \) (16 mm) is \( T_3 \). \( T_2 \) is 563.8 K, \( T_3 \) is 499.85 K, \( l_1 \) is 9 mm, \( l_2 \) is 7 mm. According to Equation (25), \( \Delta T_2 \) is 64 K, \( \Delta T_1 \) is 82.29 K, and \( T_1 \) is 646.14 K. When the temperature of the cylinder head fire surface \( (T_1) \) is 660 K, the heat conduction rate is considered unchanged during the steady heat transfer process:

\[
\frac{\left( T_3 - \hat{T}_2 \right)}{\left( T_3 - T_2 \right)} = 1, \quad (26)
\]

According to Equation (26), the temperature at 9 mm from the cylinder head fire-face \( (\hat{T}_2) \) is 577.71 K. The thermal conductivity of gray cast iron decreased with an increase in temperature. Furthermore, considering the safety margin of the thermal load of the cylinder head fire face, \( f_d \) is set to be no more than 575 K. The maximum creep temperature of the aluminum alloy piston was approximately 635 K, and \( f_s \) was set to be no more than 620 K.

In the process of numerical calculation, the diesel engine speed ranged from 900 to 1800 r/min \( (x_1 \in [900, 1800]) \) r/min, and the calculation step \( (s_1) \) was 50 r/min. The diesel engine load rate ranged from 10\% to 100\% \( (x_2 \in [10\%, 100\%]) \), and the calculation step \( (s_2) \) was 5\%. The process of refining the calculation steps of diesel engine speed and diesel engine load rate consisted of increasing the dimensions of optimal solutions of freshwater pump rotational speed to improve the accuracy of fitting the optimal rotational speed map of the freshwater pump under all operating conditions of the diesel engine. In the numerical calculation software, Equation (27) was programmed to calculate the minimum value of the single-objective function \( f_{all} \) under different working conditions of the diesel engine, and the corresponding freshwater pump rotational speed was the global optimal solution to harmonize the sub-objective functions \( f_1-f_3 \).

\[
\begin{align*}
3200 \geq x_3 \geq 900 \\
1800 \geq x_1 \geq 900, s_1 = 50 \\
100\% \geq x_2 \geq 10\%, s_2 = 5\% \\
f_3 \leq 17.6 \\
f_4 \leq 575 \\
f_5 \leq 620 \\
\end{align*}
\]

\[
\begin{align*}
\text{min} f_1 &= -2.7877x_1 + 1.891x_2 - 1.4504E^{-3}x_3^2 + 1.3540E^{-3}x_1^2 + 1.4197E^{-3}x_2^2 - 5.0272E^{-7}x_3^3 - 2.3198E^{-3}x_1 \cdot x_2 + 2.4411E^{-5}x_1 \cdot x_3 - 4.0577E^{-5}x_2 \cdot x_3 + 1.7427E^3 \\
\text{max} f_2 &= 4.4049x_1 - 3.2404x_2 + 7.6962E^{-2}x_3 - 2.2611E^{-3}x_1^2 - 1.2194E^{-1}x_2^2 - 4.9981E^{-1}x_3^3 + 4.8475E^{-3}x_1 \cdot x_2 + 2.3623E^{-1}x_1 \cdot x_3 + 1.9812E^{-4}x_2 \cdot x_3 - 2.166E^3 \\
\text{min} f_3 &= 1.5008x_1 + 2.1981E^{-1}x_2 + 7.5575E^{-3}x_3 - 1.3251E^{-1}x_1^2 + 1.6825E^{-1}x_2^2 - 1.5247E^{-1}x_3^3 + 6.8686E^{-4}x_1 \cdot x_2 - 4.5783E^{-1}x_1 \cdot x_3 + 1.4299E^{-5}x_2 \cdot x_3 - 2.5683E^2 \\
\text{min} f_4 &= 1.4772E^3x_1 - 8.9014x_2 + 9.1018E^{-2}x_3 - 8.0578E^{-3}x_1^2 - 9.0773E^{-3}x_2^2 + 2.0908E^{-5}x_3^3 + 1.4052E^{-2}x_1 \cdot x_2 - 2.3720E^{-4}x_1 \cdot x_3 + 2.8194E^{-4}x_2 \cdot x_3 - 6.6733E^3 \\
\text{min} f_5 &= -8.5891E^{-1}x_1 - 4.1321E^{-1}x_2 - 4.1699E^{-1}x_3 + 8.1643E^{-4}x_1^2 + 1.1276E^{-1}x_2^2 + 4.1518E^{-1}x_3^2 - 1.9169E^{-3}x_1 \cdot x_2 + 1.4994E^{-4}x_1 \cdot x_3 - 4.4470E^{-1}x_2 \cdot x_3 + 5.8078E^2 \\
\text{min} f_{all} &= \sqrt{(f_1 - f_{1\text{min}})^2 + (f_2 - f_{2\text{max}})^2 + (f_3 - f_{3\text{min}})^2 + (f_4 - f_{4\text{min}})^2 + (f_5 - f_{5\text{min}})^2}
\end{align*}
\]
4.3.3 | Multiobjective optimization evolutionary algorithm

To handle the nonlinear constraints of the multiobjective problem described in Section 4.3.2, Equation (27), a nonlinear constraint function was enabled. Taking the ideal minimum value of $f_{all}$ as the optimization goal, a genetic algorithm\textsuperscript{64,65} was invoked to seek global optimal solutions for the freshwater pump rotational speed. Individual samples were sampled at a depth of 40. The mutation rate and maximum number of iterations were set to 10% and 100, respectively. The convergence was judged when there was no change after 20 iterations, a single-point intersection was used for hybridization, the optimal elimination was used for selection, and the unified mutation mode was used for mutation. The iterative process of global optimization of the effective power-freshwater pump rotational speed under rated conditions of engine is shown in Figure 15. The green dots represent the sampling points, the blue dots represent the optimal points in each generation, and the red dots represent the global optimal points. Owing to space limitations, the iterative optimization process of other objective functions is not shown.

The data form of the optimal solutions of the freshwater pump rotational speed is a three-dimensional array, which consists of the diesel engine speed, diesel engine load rate and optimal freshwater pump rotational speed. The range of diesel engine speed ($x_1$) was 900 r/min to 1800 r/min, the calculation step ($s_1$) was 50 r/min, and the level of $x_1$ was 19. The range of diesel engine load rate ($x_2$) was 10% to 100%, the calculation step ($s_2$) was 5%, and the level of $x_2$ was 19. Accordingly, there are 361 diesel engine operating points, and the optimal solutions of the freshwater pump rotational speed were fitted, as shown in Figure 16. The figure is used as a feedforward control map for the intelligent cooling control strategy. Figure 17 shows the rotational speed map of the freshwater pump before optimization.

Figures 18-19 show the reduction map and reduction rate map of the freshwater pump speed after optimization. Figure 18 shows that the decrease in the freshwater pump speed after optimization is more obvious with an increase in diesel engine speed under the same engine load. The decrease in the freshwater pump speed after optimization is more obvious with a decrease in engine load under the same diesel engine speed. The water pump speed decreases the most during diesel engine high-speed and low-load conditions. The results show that the average speed of the freshwater pump is reduced by 636 rpm in the entire working condition of the diesel engine. The freshwater pump rotational speed was reduced by 636 r/min on average under all engine conditions. Figure 19 shows that the reduction rate of the freshwater pump was 636 r/min.

![FIGURE 14](image1.png) Residual of the fuel consumption rate

![FIGURE 15](image2.png) Iterative process of global optimization of effective power–freshwater pump rotational speed

| Order-cross items | Adj-Sqr | Regression algorithm |
|-------------------|---------|----------------------|
|                   |         | Standard regression | Minimize PRESS | Stepwise  |
| 2-2               | adjusted-$R^2$ | 0.991           | 0.993           | 0.996     |
| 3-2               | adjusted-$R^2$ | 0.932           | 0.956           | 0.961     |
| 3-3               | adjusted-$R^2$ | 0.911           | 0.943           | 0.953     |

**TABLE 5** Comparison of different model quadric cross-terms
pump rotational speed is more obvious with a decrease in engine load under the same diesel engine speed, and the maximum reduction rate is close to 50%. The pump rotational speed was reduced by 31.81% on average under all engine conditions.

Figures 20 and 21 show fuel consumption rate maps before and after optimization. The fuel consumption rate of the diesel engine increases with a decrease in the engine load, and the changeable rule of the fuel consumption rate is similar before and after optimization.

Figures 22 and 23 show the fuel-saving map and fuel-saving rate map after optimization. The economy of the engine is improved significantly. With the reduction in engine load, the fuel-saving rate increases, which is in line with the real situation. Figure 22 shows that the fuel consumption rate is reduced by 4.73 g/kW·h an average under all engine conditions. Figure 23 shows that the optimized cooling system
decreased fuel consumption by 1.901% on average compared with the traditional cooling system under all engine conditions.

4.4 | Experimental verification

To verify whether the DoE method is both effective and accurate in diesel engine optimization, six typical propulsive conditions (25%–110% diesel rated load) were selected to perform experiments on diesel engine performance. The rotational speed of the motor-driven cooling water pump was regulated according to the control map (Figure 17) in the verification experiment. Figures 24–25 show the comparison between the optimized predicted and experimental values of the fuel consumption rate and effective power of the diesel engine. The experimental and predicted values are very close, and the error is less than 5% which is within 10% of the engineering allowable error. This proves the accuracy of the results and the effectiveness of the DoE method. This is because a series of effective methods and algorithms were adopted in the DoE optimization process.

Table 6 lists a comparison of diesel engine parameters before and after the optimization of the cooling system, including freshwater pump rotational speed ($x_3$), freshwater pump power consumption ($P_{fw}$), fuel consumption rate ($f_1$), and effective power ($f_2$). Table 6 shows that the optimized fuel consumption rate was significantly reduced, with an average reduction of approximately 4.4% under diesel medium and low-load conditions. The average reduction is 2.35% under all engine conditions. When the cycle fuel injection quantity is kept constant, the effective power increases significantly at medium and low-load conditions with an average increment of approximately 4.3%. The average increment is 2.26% under all engine conditions. The rotational speed and power consumption of the cooling pump evidently decrease in medium and low-load conditions. The cooling pump speed decreases by 40.01%, and there are power consumption savings of 78.41%. At a 110% rated load, the cooling pump speed increases by 4.24%. The average pump power consumption saves 17.83% under all engine conditions.

In the controllable coolant flow experiment system, the seawater pump and the freshwater pump were driven by electric motors, which obviously did not consume effective power.
Considering the above influence, we analyze the net increase in the effective power. We carried out bench tests to study the working characteristics of electric motor-driven coolant pumps. The seawater and freshwater pumps are centrifugal pumps. In theory, the relationship between the shaft power and the pump rotational speed is approximately cubic, and so is the relationship between the electric motor input power and the pump speed. Therefore, a regression model between the shaft power, electric motor input power, and electric motor-driven coolant pump rotational speed was established. Equation (28) shows the following regression model:

$$ P = p_1 + p_2 \cdot n_w + p_3 \cdot n_w^2 + p_4 \cdot n_w^3, $$

where $P$ represents the coolant pump shaft power/electric motor input power, $p_1$-$p_4$ are the estimated parameters, and $n_w$ is the coolant pump rotational speed.

Based on the test data, the least squares method was employed to estimate the parameters ($p_1$-$p_4$) which are listed in Table 7. In the table, $P_{fs}$ is the shaft power of the freshwater pump, $P_{fm}$ is the electric motor input power of the freshwater pump, $P_{ss}$ is the shaft power of the seawater pump, and $P_{sm}$ is the electric motor input power of the seawater pump.

The shaft power and electric motor input power of the electric motor-driven coolant pump were calculated using Equation (28). The comparison between the regressive and experimental values is shown in Tables 8 and 9. In the table, $n_{fw}$ is the rotational speed of the freshwater pump, $\varepsilon_{fs}$ is the relative error of the shaft power of the freshwater pump, $\varepsilon_{fm}$ is the relative error of the electric motor input power of the freshwater pump, $n_{sw}$ is the speed of the seawater pump, $\varepsilon_{ss}$ is the relative error of the shaft power of the seawater pump, $\varepsilon_{sm}$ is the relative error of the electric motor input power of the seawater pump.

As shown in Tables 8 and 9, the relative errors are <5%, which indicates that the regression model has a high fitting accuracy and can be used to analyze the power consumption of electric motor-driven coolant pumps. The above six typical propulsion operating points were selected to carry out the test to analyze the net increase in effective power after the optimization of the cooling system.

### Table 6

| Performance parameter | Before and after optimization | Propulsive conditions (% diesel rated load) |
|-----------------------|-------------------------------|-------------------------------------------|
|                       |                               | 25% | 50% | 75% | 90% | 100% | 110% |
| $x_3$ (r/min)         | Original                      | 1667 | 2100 | 2406 | 2557 | 2646 | 2734 |
|                       | Optimized                     | 1000 | 1600 | 2000 | 2130 | 2280 | 2850 |
| Reduction / %         |                               | 40.01 | 23.81 | 16.87 | 16.70 | 13.83 | 4.24 |
| $P_{fw}$ (kW)         | Original                      | 1.59 | 2.7 | 3.75 | 4.37 | 4.77 | 5.2 |
|                       | Optimized                     | 0.50 | 1.45 | 2.40 | 2.79 | 3.29 | 5.79 |
| Reduction / %         |                               | 68.70 | 46.07 | 35.9 | 36.23 | 31.17 | -11.53 |
| $f_1$ (g/kW.h)        | Original                      | 250 | 214.8 | 209.9 | 212.4 | 213.5 | 217.3 |
|                       | Prediction                    | 236.45 | 207.74 | 202.81 | 206.32 | 212.45 | 214.33 |
|                       | Experiment                    | 237.4 | 205.6 | 201.9 | 207.2 | 211.3 | 214.2 |
| Reduction / %         |                               | 5.04 | 4.26 | 3.82 | 2.43 | 1.01 | 1.42 |
| $f_2$ (kW)            | Original                      | 172 | 344 | 517 | 619 | 691 | 762 |
|                       | Prediction                    | 183.25 | 359.33 | 538.46 | 635.27 | 696.45 | 779.37 |
|                       | Experiment                    | 181.8 | 360.2 | 535.1 | 632.3 | 698.5 | 777.4 |
| Increment / %         |                               | 5.69 | 4.72 | 3.51 | 2.15 | 1.09 | 2.02 |

### Table 7

| Response variable | Parameter | Estimated value |
|-------------------|-----------|----------------|
| $P_{fs}$          | $p_1$     | -7.2119E-01    |
|                   | $p_2$     | 1.4250E-03     |
|                   | $p_3$     | -4.8009E-07    |
|                   | $p_4$     | 2.7446E-10     |
| $P_{fm}$          | $p_1$     | -8.4845E-01    |
|                   | $p_2$     | 1.6765E-03     |
|                   | $p_3$     | -5.6482E-07    |
|                   | $p_4$     | 3.2289E-10     |
| $P_{ss}$          | $p_1$     | -5.4224E-01    |
|                   | $p_2$     | 1.2265E-03     |
|                   | $p_3$     | -7.1615E-07    |
|                   | $p_4$     | 2.4210E-10     |
| $P_{sm}$          | $p_1$     | -6.3793E-01    |
|                   | $p_2$     | 1.4430E-03     |
|                   | $p_3$     | -8.4253E-07    |
|                   | $p_4$     | 2.8482E-10     |
In the traditional cooling system, prior to optimization of the freshwater pump speed, the effective power \( P_e \) values were 172 kW, 344 kW, 517 kW, 619 kW, 691 kW, and 762 kW, respectively; the power consumption values of the freshwater pump \( P_{fs} \) were 1.59 kW, 2.70 kW, 3.75 kW, 4.37 kW, 4.77 kW, and 5.20 kW, respectively; the power consumption values of the seawater pump \( P_{ss} \) were 0.86 kW, 1.59 kW, 2.40 kW, 2.91 kW, 3.61 kW, and 5.24 kW, respectively; and the total power consumption values of the cooling system \( P_{tc} = P_{fs} + P_{ss} \) were 2.45 kW, 4.29 kW, 6.15 kW, 7.28 kW, 8.01 kW, and 8.81 kW, respectively. Therefore, the effective output power \( P_{be} = P_e - P_{tc} \) values were 174.45 kW, 348.29 kW, 523.15 kW, 626.28 kW, 698.01 kW, and 770.81 kW, respectively, without the cooling system consuming effective power.

In the controllable coolant flow experiment system, the seawater pump and the freshwater pump are driven by electric motors, and the output effective power \( P_{oe} \) values were 181.8 kW, 360.2 kW, 535.1 kW, 632.3 kW, 698.5 kW, and 777.4 kW, respectively, after optimization of the freshwater pump speed. Therefore, excluding the influence of the cooling system consuming effective power, the values of the net increase of the effective output power \( P_{ne} = P_{oe} - P_{be} \) were 7.35 kW, 11.91 kW, 11.95 kW, 6.02 kW, 0.49 kW, and 6.59 kW, respectively, after optimization of the cooling system. Meanwhile, the rates of net increase in the effective output power \( \psi_{ne} \) were 4.27%, 3.46%, 2.31%, 0.97%, 0.07%, and 0.86%, respectively. Table 10 summarizes the detailed data.
5 | CONCLUSIONS

In this study, an original engine heat balance and D-optimal design experiment were carried out, and the design of experiment approach was applied to the multiobjective optimization design of the cooling system. The conclusions are as follows:

1. Under medium and low-load conditions, the diesel engine was seriously overcooled, resulting in poor power and economy.
2. The thermal load of the bridge area between the two exhaust valves of the cylinder head and the center protruding area of the piston top was the highest. Therefore, attention should be paid when optimizing cooling systems.
3. The multiobjective optimization design based on the design of experiments can save experimental time and economic costs, and the accuracy of the optimization results is high. The optimized cooling system decreases fuel consumption by 1.901% on average (approximately 4.73 g/kW·h) under all engine conditions. The pump rotational speed was reduced by 31.81% on average (approximately 636 r/min) under all engine conditions.
4. The optimal freshwater pump rotational speed map was obtained under all conditions, providing data support for an intelligent cooling control strategy. After optimization, the power and fuel economy of the diesel engine were significantly improved. This optimization method provides a new technical approach for the improvement of relevant strong coupling and nonlinear systems.

6 | LIMITATIONS OF THE STUDY AND SUGGESTIONS FOR FURTHER RESEARCH

1. The calculation results indicate that the cooling system consumes a negligible part of the effective power. Thus, the following are considered to be the main reasons for the net increase in the effective output power after the optimization of the freshwater pump speed: The fuel combusts completely owing to the increased uniformity of the fuel-air mixture in the cylinder, which is beneficial in improving the thermal efficiency: As the temperature of the lubricating oil increases, the friction loss between the friction pairs of the diesel engine decreases, and as the cooling loss is reduced, the thermal efficiency of the diesel engine improves. Nevertheless, the contribution rate of each of the above factors to the net increase in the effective output power needs to be further analyzed.
2. The reason for the change in the flow field characteristics of the cooling water jacket after optimization of the freshwater pump speed is unclear. Therefore, to further correct the optimal freshwater pump speed, it is necessary to perform a three-dimensional visualization simulation of the flow field of the cooling water jacket.
3. This study focused on single-parameter multiobjective optimization of the cooling system. Based on our research experience, however, it is of great importance that multi-parameter multiobjective optimization is carried out in future research.

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CONFLICTS OF INTEREST
The authors declare no conflict of interest.

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