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Study on Dual-loop Cooling System for Non-road Mobile Machinery

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Abstract. To enhance heat transfer efficiency of cooling system of non-road mobile machinery, we have a modification on classic cooling system. Specifically, according to different hot sources, we partition classic cooling system into two separate coolant loops, i.e., coolant loop working at high temperature and working at low temperature, respectively. Through simulation via Flowmaster and verification on open area test sites, we have a systematic research on dual-loop cooling system, and observe that heat dissipation effect of dual-loop cooling system is superior to that of single-loop cooling system. More specifically, compared with classic cooling system, our system has gained 49.3% on volume coefficient, 24.5% on power coefficient, 5.8% on effective resistance coefficient, and shortened the length of pressurized air circulation loop up to 54.5%. In a nutshell, dual-loop cooling system can largely reduce the temperature of pressurized air and prevent the pollutant discharge, and the wheel loader, which carries our system, can satisfy 97/68/EC Stage C discharge standard.

Key words. Heat transfer, Cooling system, Heat exchanger, Non-road mobile machinery
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1. Introduction

Non-road mobile machineries are equipped with higher and higher power engine. As a result, new-type technologies such as hybrid power and turbo charger have been introduced into their dynamical system development. Nevertheless, the introduction of these new-type technologies will also bring the burden to classic cooling systems, which will pose a threat to safe operation of non-road mobile machinery if they cannot meet requirements. Recently, innovative engine cooling systems have made a great progress on reducing weight and packaging, and fuel consumption and pollutants. For instance, Nucleate boiling engine cooling system [1, 2] can make coolant liquid of the cooling system uniformly distributed under the high-load cir-

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cumstance, by adding a membrane in the expansion tank. Additionally, they can increase the temperature of cylinder head up to 5 to $10°C$, while lowering the pressure and flow of cooling loop. THEMIS [3] and CoolMaster [4] are able to conduct a precise flow control on the cooling system by introducing an electric water pump and an electric valve. As a result, the boiling problem that easily happens when engine stops at middle-low speed can be effectively avoided. Ultimate Cooling [5, 6] classifies the radiator of coolant into high-temperature and low-temperature parts, and offers flexibility to assign the flow of coolant according to its load. We note that aforementioned cooling systems make sufficient use of water-cooled heat exchanger, which not only possesses superior heat dissipation performance, but also has a much smaller volume compared with air-cooled heat exchanger. Without considering the flow direction of cooled air, water-cooled heat exchanger can be flexibly arranged in the power cabin, and brings great convenience [7] for the design of cooling system. In this paper, according to different heat productivities from hot source, we divide classic single cooling loop into two separated, i.e., high-temperature and low-temperature, cooling loops on the basis of classic cooling system of non-road mobile machinery. High-temperature loop cools the cylinder and low-temperature loop uses the water-cooled transmission oil cooler, water-cooled charge air cooler, and water-cooled hydraulic oil cooler. Meanwhile, air side only has the High Temperature Radiator and Low Temperature Radiator with a tandem arrangement. With certain brand wheel loader as a carrier, we leverage Flowmaster to construct its dual-loop cooling system and simulate via Heat Transfer Steady State to estimate heat exchange amount. Moreover, we verify Flowmaster simulations on open area test sites. According to the experimental results, we observe that dual-loop cooling system is superior to classic cooling system in terms of heat transfer efficiency and pollutant prevention. It can reduce the mutual interference of radiators, make heat dissipation evenly, and decrease pressure loss.

2. Model of Dual-loop cooling system

2.1. Governing equations

The motion and heat transfer of fluids follows three conservative laws. Mathematically, it can be expressed in Eqn. 1. Moreover, heat exchange capacity of cooling system obeys Newton cooling formula, which is defined in Eqn. 2. In this paper, we adopt Efficiency-Number of Heat-transfer Unit in Eqn. 3 to calculate Heat Transfer Rate.

$$\frac{\delta\left(\rho\phi\right)}{\delta t} + div\left(\rho u\phi\right) = div\left(\Gamma grad\phi\right) + s\,,$$

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0\,. \tag{1}$$

Where s is momentum source, Γ is diffusion coefficient, u is velocity vector

$$Q = \varepsilon \cdot C_{min} \left(T_{hi} - T_{ci}\right) , \tag{2}$$

$$\varepsilon = 1 - \frac{1}{exp\left(\frac{NTU^{0.22}}{C_r}\left(1 - exp\left(-C_r \cdot NTU^{0.22}\right)\right)\right)} , \tag{3}$$

where C_r is constant-pressure specific heat, NTU is Heat-transfer Unit.

The Turbulent Nusselt number in Flowmaster can be computed using Eqn. 4. Additionally, pressure loss generated by the air that flows to the radiator module and power cabin is calculated in Eqn. 5.

$$Nu = 0.028Re^{0.8}Pr^{0.4} , \tag{4}$$

$$\begin{cases} \Delta P = \frac{1}{2}\rho f U_{Amin}^2 , \\ f = \left(k + 1 - \sigma^2\right) - \left(1 - \sigma^2 - k_e\right)\frac{v_e}{v_i} + 2\left(\frac{v_\varepsilon}{v_i} - 1\right) + \alpha R_e^b \frac{A}{A_c}\frac{v_u}{v_i} . \end{cases} \tag{5}$$

Where f is streamwise pressure loss coefficient, ρ is mean gas density, U_{Amin} gas velocity at the minimum flow area, α is core friction coefficient, b is core friction exponent.

Regarding the evaluation of cooling system, there is still no unified industry standard. However, a reasonable cooling system should first satisfy the requirement of heat exchange capacity in the whole vehicle, and then pursue small volume, low-power consumption, and high efficiency. In this paper, we compare dual-loop cooling system with classic cooling system by considering heat exchange capacity, volume, driving power, and pressure loss[8].

Specifically, we leverage the following three factors as the quantitative indicators to evaluate degree completion of a cooling system. That is, volume coefficient as shown in Eqn. 6, where V_c is volume of a cooling system; Power coefficient as shown in Eqn. 7, where P_c is driving power and P_e is rated power—It reflects to what degree that cooling fan matches radiator and the smaller its value is, the better the match is; Effective resistance coefficient as shown in Eqn. 8, where ΔP_r is air side pressure loss of Radiator module—It is the ratio of resistance of air side and the entire resistance of the cooling path ΔP and it reflects to what degree that the cooling path matches the design of radiator—the bigger its value is, the more efficient Radiator module is.

$$\xi_v = \frac{V_c}{P_e} , \tag{6}$$

$$\xi_p = \frac{P_c}{P_e} , \tag{7}$$

$$\xi_{\Delta p} = \frac{\Delta P_r}{\Delta P} , \tag{8}$$

2.2. *Simulation of thermal balance of dual-loop cooling system*

There are big temperature differences among different hot sources of non-road Mobile Machineries. Classic cooling systems are not only inefficiency but also easily to be undercooling or overheating. Dual-loop cooling system (See Figure 1) cascades all hot sources, except the engine, into a low- temperature loop and uses independent radiator, makes only the engine cooled by high-temperature loop. Low-temperature and high-temperature loops are independent are each has a water pump [9, 10]. Section view of the air side of Radiator module is shown in Figure 3, where left is the baseline cooling module and right is the dual-loop cooling module.

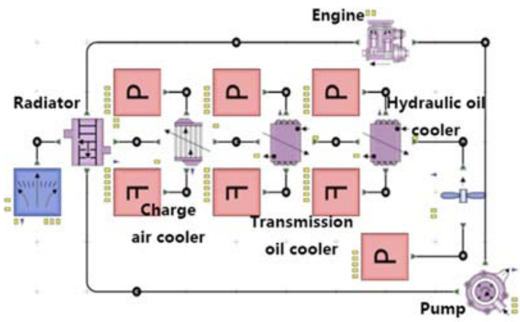

Fig. 1. Configuration for classic cooling system

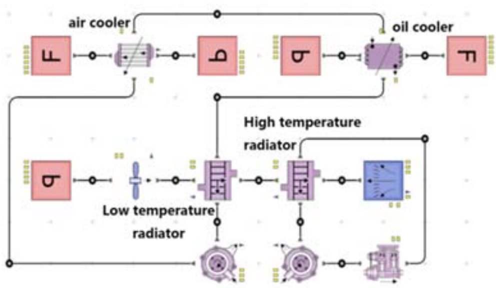

Fig. 2. Configuration for Dual-loop cooling system

We leverage Flowmaster to construct the dual-loop cooling system model, and use it to simulate Heat Transfer Steady State. Moreover, we simulate the peak load of two cooling systems by referring to the 97/68/EC standard. We set the boundary conditions as follows: ambient temperature is 35°C; engine power is 162kW; mass flow rate of pressurized air is 0.33 kg/s with temperature 120°C; volume flow rate of radiator is 280 L/min; volume flow rate of hydraulic oil is 100L/min with temperature 85°C; volume flow rate of transmission oil is 90L/min with temperature 95°C. The steady simulation result is shown in Table 1. From it, we observe that the heat exchange effect of high temperature radiator of dual-loop cooling system is comparable with that of classic cooling system, and can satisfy the heat dissipation requirement of the engine. However, the heat exchange capacity of transmission oil, hydraulic oil, and intercooler of dual-loop cooling system is superior to the classic cooling system.

Table 1. Comparison of heat transfer steady state values

	Inlet temperature of radiator°C	Outlet temperature of radiator°C	Outlet temperature of transmission oil°C	Outlet temperature of hydraulic oil°C	Outlet temperature of pressurized air°C
Classic system	80	72.5	76.1	77.4	61.1
Dual-loop system	80	72.8	75.5	74.9	54.2

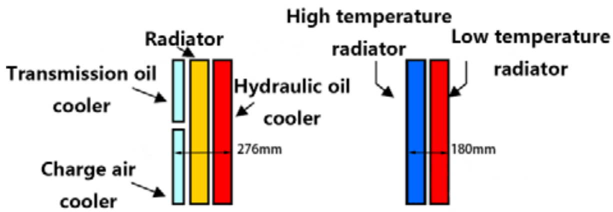

Fig. 3. Baseline cooling module and new cooling module for Dual-loop cooling system

3. Experimental analysis of Dual-loop cooling system

3.1. Thermal balance analysis of Dual-loop cooling system

The wheel loader of certain brand in the experiment has a rated power 162kW. Its intake air temperature cannot exceed 85°C and its mass flow rate is about 0.33kg/s under the rated power. We compare its test data of high-temperature and low-temperature radiators with the simulated data of Heat Transfer Transient generated by Flowmaster and show the results in Figures 4 and 5.

We observe that, the initial temperature of high-temperature circulation increases rapidly. When it reaches a certain temperature the thermostat is fully open so that the temperature dramatically decreases. Finally, the inlet and outlet temperature of high-temperature radiators stabilize at around 80°C and 73°C, respectively. The result of low-temperature circulation is similar. However, due to its small coolant flow, the initial temperature increases relatively smoothly and the inlet and outlet temperature of low-temperature radiators finally stabilize at around 77°C and 72°C, respectively. The thermal equilibrium temperature of high-temperature and low-temperature circulations are almost in line with above simulation results. Nevertheless, there is a big gap between the curve of its temperature increasing stage and the curve of simulation results of Flowmaster. It is because Flowmaster uses

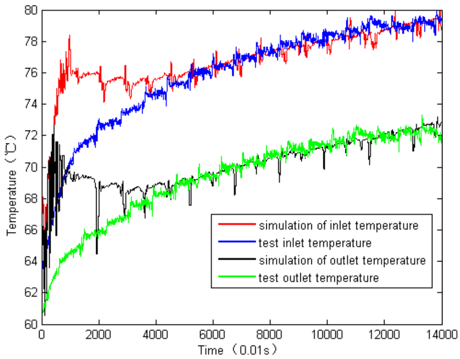

Fig. 4. Comparison of high temperature loop simulation and experiment values

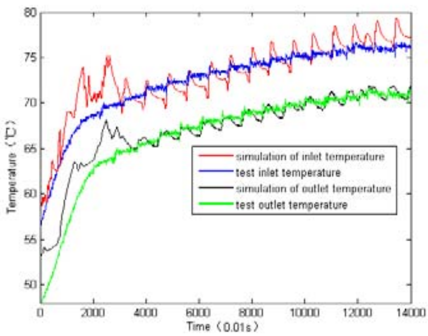

Fig. 5. Comparison of low temperature loop simulation and experiment values

one-dimensional simulation based on a perfect condition and thus ignores several details that would affect the experimental results. For instance, the effect of vent of power cabin to the flow field of inner cabin, response time of thermostat and water pump, etc. Although Flowmaster can accurately calculate all nodes' final thermal equilibrium temperature, it is inaccurate to describe the temperature-rise period.

3.2. Analysis of low- temperature loop

We design the open area test site as in [14], shown in Figure 6. Specifically, we have a 35°C ambient temperature and an asphalt pavement; we add erasure cotton in the power cabin; the flow pattern of cooled air in the power cabin is that top and bottom inflow and back outflows (See Figure 6).

The experiment is conducted in two cases, i.e., high-speed running test and shovel cycling test. High speed running test stays in the highest gear and maximum accelerator state and has a vehicle speed at 20-30km/h. We preheat the wheel loader before the test and use data acquisition machine from DEWEsoft to collect the data. We execute a shovel cycling test and high-speed running test and show the compared results of their cooling system in Figures 7-10.

The compared results in the two cases are illustrated in Table 2. In the high

Fig. 6. Air flow direction and data acquisition instrument

speed running test, the average outlet temperature of transmission oil cooler and average outlet temperature of hydraulic oil cooler of dual-loop cooling system are around 3°C lower than that of classic cooling system, while in the shovel cycling test, they are 2°C lower. Under the same working condition heat transfer rate of the heat source is basically the same it only related to the temperature difference. Moreover, the average heat exchange capacity of transmission oil and hydraulic oil cooler can increase up to 16.1% and 13.4%, and 15.2% and 19.2% in two cases, respectively.

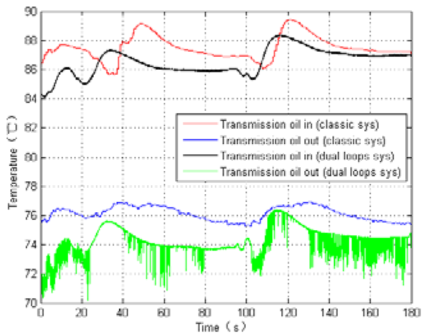

Fig. 7. Comparison of transmission oil temperature in high speed running test

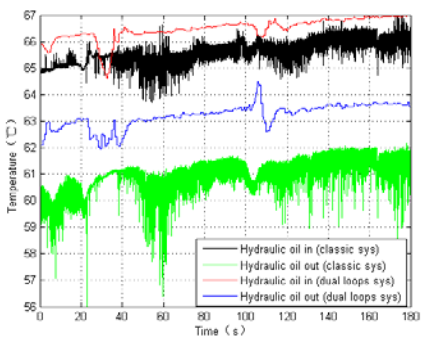

Fig. 8. Comparison of hydraulic oil temperature in high speed running test

In summary, the heat exchange efficiency of low-temperature hot source can be

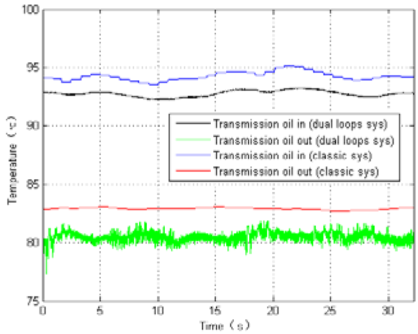

Fig. 9. Comparison of transmission oil temperature in shovel cycling test

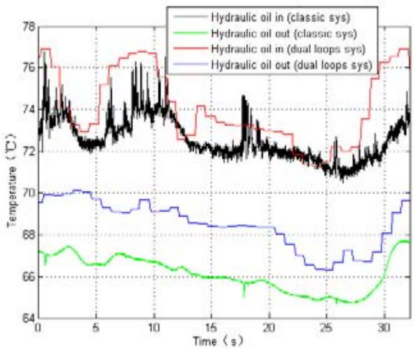

Fig. 10. Comparison of hydraulic oil temperature in shovel cycling test

enhanced by separately radiating the low-temperature and high-temperature hot sources of the cooling system, as it can ignore the effect of high-temperature hot source and adjusts heat exchange capacity freely. Although both two cooling systems can satisfy the heat exchange capacity requirement of low-temperature hot source of the wheel loader, dual-loop circulating cooling owns a higher heat exchange efficiency and largely reduces the temperature of low-temperature hot source.

Table 2. Comparison of test results in different working conditions

	Transmission oil inlet °C	Transmission oil outlet °C	Hydraulic oil inlet°C	Hydraulic oil outlet°C
high speed running (classic system)	87.5	77.1	66.5	63.2
high speed running (Dual-loop system)	86.5	74.1	65.6	61.8
shovel cycling (classic system)	94.3	83.1	74.1	68.5
shovel cycling (Dual-loop system)	92.8	80.1	72.6	66.1

3.3. Effect of Dual-loop cooling system on dynamic performance and emission

(1) Effect of Dual-loop Cooling System on dynamic performance

Dynamic performance of engine is mainly affected by volume flow rate and temperature of intake air. Dual-loop cooling system adopts water-cooled charge air cooler (WCAC) to cooler pressurized air. Compared with charge air cooler (CAC), its resistance is much smaller and heat exchange efficiency is much higher. As shown in Figure 11, the average temperature of intake air of engine of WCAC is around 10°C smaller than that of CAC. Moreover, WCAC can largely reduce the circulation loop length of pressurized air, and thus effectively reducing frictional drag, increasing the relative inlet pressure of engine, and improving atomization effect of diesel. Additionally, we can observe from Figure 12 that the average torque of wheel loader is 558.7N·m by using WCAC, while it is 594.1Nm by using CAC, which demonstrates that the decrease of inlet temperature and the increase of inlet pressure can promote the dynamic output of engine.

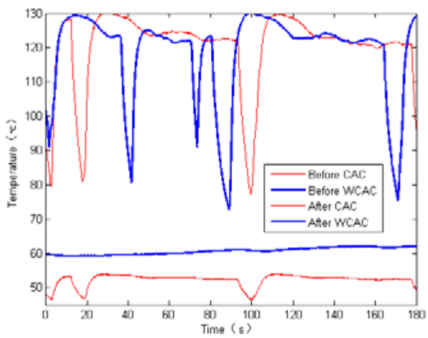

Fig. 11. Intake temperature comparison of high speed running test

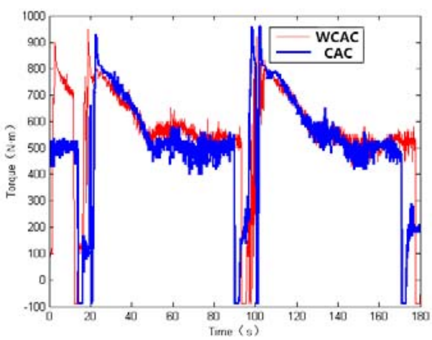

Fig. 12. Torque comparison of high speed running test

(2) Effect of Dual-loop Cooling System on emission

The main combustible pollutants of diesel engine are NOX and soot. By leveraging dual-loop cooling system, these pollutants can be largely reduced. According to the test, there is no significant difference between the discharge of classic cooling system and dual-loop cooling system when engine is in idle state. Especially, the two systems have almost the same discharge when the inlet temperature is 60°C. When the revolving speed of engine exceeds 1000rpm, the revolving speed of compressor increases and the inlet pressure dramatically increases. In this case, the discharge capacity of classic cooling system tends to be accelerated increasing, which is especially obvious on the discharge of NOX (See Figure 13)—discharge capacity of NOX in high-speed region of classic cooling system is far more surpassing than that of dual-loop cooling system when the inlet temperature is 60°C.

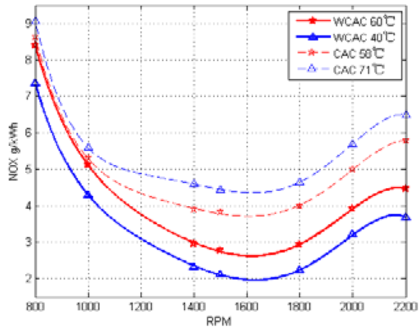

Fig. 13. Comparison of NOX emission in WCAC and CAC

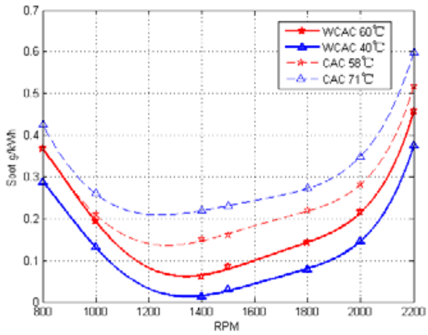

Fig. 14. Comparison of soot emission in WCAC and CAC

The generation of NOX and soot is closely related to the combustion quality. CAC used by classic cooling system has a limited heat dissipation performance. It is far worse than WCAC in terms of cooling effect of pressurized air. The decrease of inlet temperature can enhance charging efficiency, which has a positive effect on the combustion quality in cylinder. However, from Figures 13 and 14, there still exists some difference when the inlet temperatures are same. It is because the length in the hot side flow direction and the pressure loss coefficient of CAC are both larger than those of WCAC, and the overlong channel of CAC can make Frictional Drag coefficient bigger than that of WCAC. As a result, the pressure loses too much when the pressurized air flows through CAC. However, compared with the pollutant reduction effect caused by reducing inlet temperature, the pollutant effect of increasing inlet pressure on non-road mobile machinery is much smaller [12] [13].

From the test data in Table 3, we note that the inlet pressure and inlet rate of flow of dual-loop cooling system are higher than those of classic cooling system. When the inlet temperature is 60°C, the discharge of NOX and soot of engine that uses CAC just satisfies EC Stage B2 Standard. However, when the inlet temperature is higher than 60°C, then we need some after-treatment system to meet the demand. Nevertheless, the discharge of dual-loop cooling system that uses WCAC is much superior to classic cooling system. Especially, its soot discharge has satisfied EC Stage C Standard and its pollutant discharge can also satisfy EC Stage C Standard when using some after-treatment system. Based on the bench test, the pollutant discharge can be further reduced when the inlet temperature decreases to 40°C, which approaches direct discharge. It can be shown that there is a huge potential for dual-loop cooling system to control the combustible pollutant.

Table 3. Engine state comparison

	Mass flow rate of air (kg/s)	Pressure loss (kPa)	Inlet pressure (bar)	Power (kw)
WCAC	0.270574	3.08	2.54375	136.6
CAC	0.270194	3.39	2.54069	135.1

3.4. Comparison of Dual-loops cooling system with classic cooling system

Table 4 shows the evaluation indicators calculated via Eqns 6, 7, and 8 of the two cooling systems. Due to the usage of water-cooled oil cooler and WCAC, the volume of radiator module of dual-loop cooling system is decreased. Moreover, the design of dual-loop cooling system that only high-temperature radiator and low-temperature radiator are left in the air side makes the pressure loss of radiator module is far smaller than classic cooling system. Therefore, its Effective resistance coefficient is higher than classic cooling system. Regarding power coefficient, although the low-temperature loop adds an independent water dump, the decrease of air side pressure loss makes cooling fan power lower than that of classic cooling system, which causes that the whole driving power of dual-loop cooling system is lower than that of classic cooling system. As WCAC can be installed in any place of power cabin, the engine's intake hose of pressurized air of dual-loop cooling system can be shortened up to 54% when compared with classic cooling system.

Table 4. Cooling System Evaluation

	Volume coefficient	Power coefficient	Effective resistance coefficient	Intake hose length (m)
Classic system	1.901	0.0677	0.553	1.1
Dual-loop system	0.963	0.0511	0.585	0.5
promotion	49.3%	24.5%	5.8%	54.5%

4. Conclusions

(1) We leverage Flowmaster to construct a Thermal Management Model of Dual-Loop Cooling System. Through the steady thermal equilibrium of simulated system, we validate that dual-loop cooling system can satisfy the heat exchange requirement in the entire vehicle. Moreover, we validate the correctness of the simulated model based on open area test site. Compared with classic cooling system, dual-loop system is advantageous on the cooling effect of hydraulic system, drive system, and air inlet system.

(2) Dual-loop cooling system can control the inlet temperature of engine to be around 50°C, and show the potential to further reduce the inlet temperature. The dynamic output can be enhanced by appropriately reducing the inlet temperature or increasing the inlet pressure. Compared with classic cooling system, dual-loop cooling system has a huge potential to reduce pollutant discharge, which has the ability to satisfy EC stage C Standard.

(3) Compared with classic cooling system, dual-loop cooling system has gained 49.3% on volume coefficient, 24.5% on power coefficient, 5.8% on effective resistance coefficient, and shortened the length of pressurized air circulation loop up to 54.5%. Due to the decrease of the number of air side heat exchanger, the air side pressure loss decreases, and the radiator module efficiency increases, and the power consumption decreases.

Acknowledgement

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