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A Design of Cooling Water Jacket Structure and an Analysis of Its Coolant Flow Characteristics for a Horizontal Diesel Engine

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ABSTRACT

In order to fulfill the technical requirements of a high-efficiency low-emissions off-road horizontal diesel engine, a unique design is proposed and optimized in this paper for the cooling water jacket structure with a forced-cooling closed-loop cooling system. The cooling water flow rate, temperature, and pressure at the inlet and several other critical locations of the cooling water jacket were measured and analyzed at different engine operating conditions for the water jacket designs. A numerical simulation model of the coolant flow and the cooling system was built and used to analyze the thermal/fluid characteristics of the coolant flow in the water jacket. The impact of different structural and packaging design parameters on coolant flow and heat transfer was investigated. The design deficiency of an original (earlier) design of the water jacket was pointed out and an improved design was proposed. The cooling water jacket structure and the water pump flow rate were finally optimized.

INTRODUCTION

The internal combustion engine components exposed to the high-temperature gas inside the engine are subject to high thermal loading during engine operations. If the engine is under-cooled, engine performance (e.g., power characteristics, fuel economy, and emissions), durability, and reliability are deteriorated. If the engine is over-cooled, excessive cooling may result in rough running of the engine, excessively high heat rejections, and an increase in engine friction, hence also lead to an overall degradation in engine performance (French, 1999, [1]). Therefore, a well-designed engine cooling system and cooling components are not only required for the normal operation of the engine, but also important for overall engine performance enhancement (Liu *et al.*, 2003, [2]).

Advanced numerical simulation with computational fluid dynamics (CFD) plays a critical role in modern diesel engine cooling system design because it can provide an in-depth understanding of the coolant flow fields inside the water jacket around complex cooling components, which are difficult to measure. Moreover, CFD simulations can also help screen various cooling system/component designs in an efficient and effective way that was not feasible using primitive empirical formulas (Kruger *et al.*, 2008, [3]; Antonelli *et al.*, 2009, [4]).

Small horizontal diesel engines have been widely used in many countries in off-road (e.g., agricultural) applications for almost a century due to their advantages in compactness, easy installation and maintenance, simple structure, and low cost.

The demand for single-cylinder or double-cylinder horizontal diesel engines has exceeded eight million units in China. Simple evaporative or condensation cooling has been largely used in many horizontal diesel engines (Xu, 1994, [5]) due to their design requirement for a simple cooling system structure and low cost. These primitive cooling system options directly affect the overall structural compactness of the engine, and usually lead to relatively poor specific weight (i.e., the ratio of engine weight to power), durability, reliability, fuel economy, and emissions. For example, there have been many durability problems or design challenges in regard to cylinder head cooling with these cooling options when the engine is upgraded to a higher power rating. In this case, the cooling system is often insufficient or incapable of handling the increased thermal load at critical locations inside the engine. As the market demand for higher power density increases for horizontal engines, the forced-cooling system has to be considered when upgrading the engine. The forced-cooling system design for horizontal diesel engines is different from the design for vertical (or upright) diesel engines due to their unique design packaging requirements and different orientation of the cylinders. As a result, the coolant flow characteristics of horizontal diesel engines are different from those of vertical diesel engines, from both a system and a component design point of view.

Controlling the three-dimensional flow and heat transfer of cooling water in the engine water jacket is one of the most important design issues in diesel engine water jacket design. The design directly affects the efficiency of the cooling system, the thermal load acting on the high-temperature components, and the energy distribution and utilization of the entire engine. The cylinder-to-cylinder distribution or evenness of the cooling water and achieving a uniform flow distribution of the cooling water from front cylinders to rear cylinders are also very important for multi-cylinder diesel engines (Bai *et al.*, 2004, [6]). In vertical (or upright) diesel engines, the piston moves up and down along the vertical direction, and the cylinder head is located above the engine block. The cold coolant is fed into the cylinder block water jacket usually from the side of the engine block and circulates before it enters the cylinder head water jacket. The hot coolant finally leaves out of the cylinder head after it exchanges heat with the walls of the water jackets in the cylinder block and the cylinder head. In vertical diesel engines, water jackets can be designed relatively easily in terms of location, orientation, and packaging without many concerns on vapor lock in the cooling circuit (Xu *et al.*, 2010, [7]). The situation for horizontal diesel engines is different. In horizontal diesel engines, the piston moves along the horizontal direction, and the cylinder head and the engine block are located at the same horizontal level. An oil pan is usually placed below the cylinder head and the cylinder block. This configuration makes the packaging in cooling water jacket design difficult. If the water jacket and coolant circuit are not designed properly, it is difficult to circulate the coolant and vapor lock can easily happen (Xu, 1994, [5]).

Forced-cooling closed-loop cooling systems have been extensively used in almost all vertical inline or Vee engines for on-highway vehicular applications, and their cooling systems and cooling water flow characteristics have been analyzed by many previous researchers (Kruger *et al.*, 2008, [3]; Fontanesi and McAssey, 2009, [8]; Antonio *et al.*, 2010, [9]). On the contrary, although the market demand for horizontal diesel engines is high, the forced-cooling closed-loop system has not been widely applied to horizontal diesel engines until recently due to their special design considerations and difficulties when modern high-power-density engines become highly desirable. To date, there has been no published literature to present the cooling water flow characteristics, the cooling design, and advanced optimization via CFD for the forced-cooling closed-loop cooling system in this important category of off-road engines - the horizontal diesel engines.

In this paper, a unique design of cooling water jacket structure with a forced-cooling closed-loop system is presented for a new generation of off-road horizontal diesel engine, which has been designed and developed independently by the authors and their affiliations. The cooling water flow rate, temperature, and pressure in the water jacket were measured at different engine speeds in order to reveal the overall cooling system characteristics of the engine. A three-dimensional coolant flow model of the water jacket was built with AVL's CFD software, FIRE, and used to conduct a cooling water CFD simulation. Finally, the cooling water jacket structure was optimized to achieve an improved design, compared to the original design.

The paper presents a comprehensive process of advanced cooling system/component design, testing (at high altitude conditions), numerical simulation, and design optimization.

COOLING WATER JACKET STRUCTURE DESIGN

The new generation off-road horizontal diesel engine has the following design and operating characteristics: a forced-cooling closed-loop cooling system; four-stroke; 115 mm cylinder bore; 120 mm engine stroke; 2.493 liters engine displacement; a compression ratio of 17.5; naturally aspirated; 37 kW rated power; 2400 rpm (revolutions per minute or r/min) rated speed; 167 N.m peak torque; 1600 rpm peak torque speed; the minimum fuel consumption at full load being less than 225 g/(kW.h); and meeting the Chinese Stage-II off-road emissions regulation. Figure 1 shows the new generation off-road horizontal diesel engine.

In general, it is very challenging to achieve effective and efficient circulation patterns of the cooling water in the forced-cooling closed-loop system in horizontal diesel engines because the cylinder head and the water jacket are located at the same horizontal level. In order to improve cooling water circulation and prevent vapor lock in the cooling circuit, a shared cooling-water-feed chamber (called shared water chamber) was designed to be located between the pushrod chamber and the oil pan underneath the cylinders, as shown in Fig. 2. A water pump was installed on the left side of the shared water chamber. Some water passages (i.e., the water inlet holes in the cylinder block) that connect the cylinder water jacket were designed to be located besides the pushrod holes for each cylinder. The water passages in the cylinder block and in the cylinder head were connected by the water inlet holes in the cylinder head. In this paper, the water inlet hole in the cylinder block refers to the flow passage that connects the shared water chamber and the water jacket of the cylinder block/liner. The water inlet hole in the cylinder head refers to the flow passage that connects the water jacket of the cylinder block and the water jacket of the cylinder head. The water inlet port refers to the entry location in the shared water chamber where the cooling water is fed into the engine by the water pump. The water outlet port refers to the exit location in the cylinder head where the cooling water flows out of the engine after it circulates through the shared water chamber, the engine block water jacket, and the cylinder head water jacket. The cooling water is fed from the inlet port to the shared water chamber, and flows into the water jacket of the cylinder block via the cylinder block's water inlet holes, and then flows into the water jacket of the cylinder head via the cylinder head's water inlet holes, and finally flows out of the engine through the cylinder head's water outlet port. This design effectively and intelligently utilizes the room between the cylinder water jacket and the oil pan, improves cooling water circulation, and avoids vapor lock. Vapor lock was encountered in the upper side of the water jacket of the cylinder block and in the shared water chamber in an experimental prototype engine of an earlier design where the shared water chamber was located at the upper side the horizontal engine, resulting in coolant flow problems. The vapor lock was found to be caused by the accumulated water vapor moving up to the shared water chamber. Therefore, relocating the shared water chamber to the lower side of the engine may reduce the tendency of vapor lock. Note that packaging constraints on the geartrain side and the flywheel side of this engine do not allow the shared water chamber to be located at these locations. Therefore, a good choice of the location of the share water chamber is at the lower side of the engine. Moreover, the design enhances the strength and the stiffness of the lower part of the cylinder block. Figure 3 shows the three-dimensional model of the water jacket of the horizontal diesel engine (Shen, *et al.*, 2008a, **[**10**]**; Shen, *et al.*, 2008b, **[**11**]**).

COOLING WATER FLOW EXPERIMENT

Experimental setup and measurement locations

The test bed was composed of a hydrodynamic dynamometer of the WE31 type and a fuel consumption measurement device of the 50-200 volumetric type. The measurement was conducted at an altitude of 1920 m, an atmospheric absolute pressure of 81.5 kPa, environmental temperatures of 20-25 °C, and relative humidity of 20-35%. The sensors and flow meters were calibrated and tested before the experiment. The water chambers and passages of the prototype engine were cleaned by using compressed air. During the test, the water pump was driven by an electric motor to pump the coolant into the engine water jacket to circulate. The engine outlet coolant temperature was controlled at approximately 75 °C in order to reflect realistic engine applications in real world. The schematic diagram of the coolant flow experimental devices is shown in Fig. 4.

There were fourteen measurement locations selected as shown in Fig. 5. The symbol T represents temperature and P represents pressure. Each measurement location and its numbering are explained in details below:

1. The coolant flow rate was measured at the outlet of the water pump.

2. There were six coolant temperature measurement locations as follows: at the water pump outlet (T0); underneath the intake port of cylinder 2 (T1); underneath the intake port of cylinder 1 (T2); underneath the exhaust port of cylinder 2 (T3); underneath the exhaust port of cylinder 1 (T4); and at the outlet of the thermostat (T5).

3. There were seven pressure measurement locations at the following: the water pump outlet (P0); the last water inlet hole of cylinder 2 (P1); the water inlet hole at the exhaust side of cylinder 2 near the central baffle plate (P2); the water inlet hole of cylinder 1 near the central baffle plate (P3); the water inlet hole near the exhaust port of cylinder 2 (P4); the water inlet hole near the intake port of cylinder 1 (P5); and the outlet of the thermostat (P6).

The measurements were taken every 200 rpm at full load from 1000 to 2400 rpm. The coolant flow rate was measured by using the LWGY-40 turbo fluid flow meter. The coolant pressure was measured by using the PPM-WX-JC miniature pressure sensor and the PPM-KL22 amplifier. The temperatures were measured by using a platinum resistor temperature sensor, which was calibrated with a constant-temperature thermo-tank at 0, 20, 60, and 100 °C. The experimental data were collected and processed by using the engine testing system that has been developed in-house at KUST (Kunming University of Science and Technology).

TEST RESULT AND ANALYSIS

The relationship between water pump flow rate and engine speed is shown in Fig. 6. It shows that the water pump flow rate increases from 0.85 kg/s to 3.46 kg/s within the engine speed range of 1000-2000 rpm, and the coolant flow rate basically linearly increases with engine speed. However, in the engine speed range of 2000-2400 rpm, the water pump flow rate only slightly increases from 3.46 kg/s to 3.65 kg/s, and the flow rate does not increase much when the engine speed increases.

Figures 7 and 8 illustrate the coolant pressures at different measurement locations and the total coolant pressure drop (P0-P6) within the water jacket, respectively, as functions of engine speed. It is observed that when the engine speed increases the coolant pressure at each measurement location increases, and the total coolant pressure drop also increases.

Figure 9 shows the measured coolant temperatures as a function of engine speed. As the engine speed increases, the thermal load and the coolant temperatures at different measurement locations increase. The coolant temperature reaches the maximum level at 2400 rpm full load (rated power), as shown in Fig. 9. Generally, the measured coolant temperatures in the water jacket of the cylinder head are higher than those of the cylinder block. It was noted that the coolant temperatures underneath the exhaust ports were always high during the test but with large variations. Moreover, Fig. 9 shows that the coolant temperature underneath the exhaust port of cylinder 2 (T3) is higher than that of cylinder 1 (T4).

MODELING AND EXPERIMENTAL VALIDATION OF COOLANT FLOW

The water jacket of the engine has a complicated geometry and structure with many irregular surfaces. A simulation model of the water jacket was created by simplifying the real geometry (e.g., some sharp corners) without losing the validity of modeling the coolant flow. The hexahedral element was used to mesh the model with AVL's pre-processing software FAME. Fine grids were used in the finite volume mesh in the areas around the cylinder head gasket. The total number of elements in the meshed grids was approximately 775,000, with 684,000 being hexahedral grid elements, which occupy 88.3% of the total grids. The meshed model of the water jacket is shown in Fig. 10.

Engine rated power is a typical operating condition that has high thermal loading. Therefore, rated power was selected in the simulation analysis. The coolant flow was assumed to be incompressible and in steady state at the given engine operating condition. The *k-ε* turbulence model was used in the CFD simulation.

1. Inlet boundary conditions Experimental results indicated that the inlet coolant flow rate of the water jacket at rated speed (2400 rpm) was 3.65 kg/s, the inlet coolant temperature was 343 K, and the turbulence kinetic energy was 1 m^2/s^2 (a boundary condition determined by empirical data of AVL and experimental results). The turbulence length scale was taken as 0.001 m in the model.

2. Outlet boundary conditions Because the outlet port of the water jacket was connected to a long hose, the coolant pressure gradient was set zero according to the fully developed boundary condition of the flow field.

3. Wall boundary conditions The metal surface temperatures of the cylinder head and the cylinder liner were assumed 393 K and 373 K, respectively, in the simulation. The wall function was applied to the region near the wall because the *k-ε* turbulence model is only applicable to the turbulent region that has a certain distance away from the wall ([12]; Liang, 2007, [13]; Robinson *et al.*, 2007, [14]).

The analysis in this paper is not conjugate heat transfer CFD analysis. A thermal boundary condition on the water jacket wall is applied with constant wall temperatures. In the analysis of water jacket, conjugate heat transfer analysis is important for metal temperature predictions in critical engine parts, such as the valve bridge, the gasket, and the cylinder liner for thermo-mechanical fatigue analysis (Etemad *et al.*, 2005, [15]; Fontanesi *et al.*, 2010, [16]). Coolant boiling models are also important for the accuracy of metal wall temperature predictions (Fontanesi and McAssey, 2009, [8]; Fontanesi *et al.*, 2010, [16]; Fontanesi *et al.*, 2009, [17]; Dong *et al.*, 2010, [18]). The conjugate modeling and boiling modeling increase the complexity and computational time of water jacket CFD simulation. The main focus of this paper is to design a unique water jacket structure for a horizontal engine from an overall macro flow pattern perspective, especially for a relative comparison between an earlier design and an improved design. Both experimental means and traditional CFD simulation (relatively simpler without conjugate and boiling modeling) were used to compare different design proposals. The primary focus is to explore the overall "macro" coolant flow distribution patterns rather than the more accurate details that are enabled by certain boiling models and conjugate modeling. The experimental validation showed that the relatively simpler approach without including a boiling model and conjugate modeling could fulfill the design purpose in this study. However, it should be pointed out that more accurate predictions of heat transfer coefficient and metal wall temperature would require advanced conjugate and boiling modeling, for the purpose of highly accurate heat transfer and durability analysis, and hence the importance of conjugate and boiling modeling should not be overlooked.

The comparison between the measurement data and the CFD simulation data of the coolant pressures at six measurement locations is shown in Table 1. The result shows that the trends of the simulation and the measurement are consistent, with a maximum discrepancy of 11.1% occurring for P6. The discrepancies at other measurement locations are all within 10%. This model validation confirms that the computational results match well with the measurement results, and the simulation model can be used for the following analysis.

CFD SIMULATION RESULTS AND ANALYSIS

The performance parameters of velocity, temperature, and pressure distributions of the coolant flow in the water jacket are inter-related and influenced each other. Figures 11, 12, 13, 14 illustrate the distributions of flow velocity, heat transfer coefficient (HTC, the overall heat transfer coefficient from outer metal wall to the coolant), pressure, and temperature of the coolant flow in the water jacket. The overall spatial average velocity of the coolant flow is 1.00 m/s; the spatial average velocities of the coolant flow in the water jackets of the cylinder block and the cylinder head are 1.11 m/s and 0.85 m/s, respectively; and the spatial average flow velocity in the valve bridge area in the cylinder head, which has high thermal loading, is 0.94 m/s, as shown in Fig. 11. According to AVL's computational experience and related literature (Liang, 2007, [13]; Gavali *et al.*, 2007, [19]; Chen, 2003, [20]; Matsutani *et al.*, 2005, [21]), satisfactory engine cooling usually requires a coolant flow velocity greater than 0.5 m/s for overall spatial average and greater than 1 m/s for the valve bridge area. Therefore, the spatial average coolant flow velocity basically satisfies engine design requirements in this original design.

The spatial distribution of heat transfer coefficient is directly related to the distribution of coolant flow velocity. Both of them exhibit similar or common characteristics. For example, the HTC in the high-flow-velocity region is large, and the HTC in the low-flow-velocity region, the vortex region, and the dead-flow region (i.e., "dead corner") is small. A high HTC and a uniform cylinder-to-cylinder distribution of HTC are required in the regions subject to high thermal loading (Liang, 2007, [13]; Gavali *et al.*, 2007, [19]; Chen, 2003, [20]). Usually, a HTC greater than 5000 W/(m^2.K) is required in engine design for the areas on the cylinder liner having a high thermal load (Matsutani *et al.*, 2005, [21]). Figure 12 shows that the overall spatial average HTC of the water jacket is 7767 W/(m^2.K) in this original design, and the spatial average heat transfer coefficients of the water jackets in the cylinder block and in the cylinder head are 9285 W/(m^2.K) and 5865 W/(m^2.K), respectively. However, the heat transfer coefficients of the two cylinders are uneven, and the heat transfer coefficients for the exhaust ports and some local areas on the water jacket of the cylinder block do not reach 5000 W/(m^2.K) in this original design.

The engineering design requirements for engine cooling system usually include the following. The heat transfer area needs to be sufficient and the metal wall thickness needs to be appropriate. Proper cooling in critical areas (e.g., cylinder head) needs to be maintained by controlling the spatial average and local metal temperatures and coolant temperatures. The coolant should flow smoothly with well organized motions and sufficiently high velocity in the water jacket with minimum pressure loss and water pump power consumption, while satisfying heat transfer requirements. Moreover, it is important to avoid abrupt changes in coolant pressure or sudden pressure drops along the route of the coolant flow in the water jacket, especially at the inlets and outlets of the cooling circuit, in order to prevent cavitation in the coolant and the associated damage (Liang, 2007, [13]; Gavali *et al.*, 2007, [19]). Figure 13 shows that the overall coolant pressure drop (loss) in the water jacket is 0.027 MPa, and large pressure losses mainly occur at the inlet and outlet ports of the water jacket, as well as the water inlet holes in the cylinder block and in the cylinder head (especially the one in the cylinder head near the water jacket outlet port). The pressure drop from the inlet port to the outlet port of the water jacket is 0.036 MPa.

It is usually relatively easier to measure temperature than velocity in engine coolant flow experiments. Therefore, coolant temperature distributions are often investigated as a focus in experimental studies. The situation is generally opposite in CFD simulations of engine coolant flow, where accurate coolant temperature distribution is difficult to obtain due to many inaccurate or uncertain boundary conditions used in the simulation and the complex dependency of the temperature distribution upon velocity distribution and other factors such as fluid state (turbulent or laminar) (Liang, 2007, [13]; Gavali *et al.*, 2007, [19]). Figure 14 shows that the overall spatial average coolant temperature in the water jacket is 345.3 K. The coolant temperature at the inlet port is lower than the temperatures at other locations. The coolant temperature in the water jacket at the engine intake port side is relatively low due to high coolant flow velocities. The coolant temperature in the upper portion of the cylinder liner water jacket is higher than the temperature in the lower portion due to the higher thermal load and heat rejection in the upper portion. It is also observed that the coolant temperature in the water jacket of the exhaust port in the cylinder head is high because of high thermal loading and relatively worse distributions of coolant flow velocity and heat transfer coefficient.

Figure 15(a) shows the coolant flow velocity distribution at the cross-sectional area of the shared water chamber. Z direction refers to the direction along the cylinder centerline, pointing toward the cylinder head. The coordinate position of Z = 0 refers to the bottom plane of the installed cylinder head. Two cross-sectional diagrams are taken at Z = −84 mm and Z = −95 mm, respectively. It is observed in Fig. 15(a) that the coolant flow velocity gradually increases in the four water inlet holes from cylinder 1 (close to the water jacket inlet port) to cylinder 2. The coolant flow rate going through No.1 water inlet hole is low, and small vortices are formed in the region of sudden expansion due to the flow inertia of the fluid and the fact that a high-speed coolant flow enters a region of sudden expansion at No.1 water inlet hole of cylinder 1 near the water jacket inlet port. The front ends of No. 2 and No. 3

water inlet holes were connected in the original engine design to form a large cross-sectional area, but the rear end was split into two small ϕ14 holes/passages. Due to the flow inertia caused by fully developed flows, the coolant flow rate at the front ends of the two water inlet holes was high. However, coolant flow collision occurred at the rear ends of the passage due to sudden changes in the cross-sectional area, and the kinetic energy of the coolant flow was converted to pressure-related energy, forming a pressure differential which caused reverse flows and a large vortex in the middle of the shared water chamber in this original design. Figure 15(b) illustrates the coolant pressure distribution in the shared water chamber. Figure 15(b) shows that due to the effect of coolant flow inertia, the coolant pressure at No. 1 water inlet hole is low, the pressures at No. 2 and No. 3 water inlet holes are high, and pressure at No. 4 water inlet hole is low owning to the sudden changes in the cross-sectional areas in the flow passage of No. 4 water inlet hole. Because the coolant pressures at the four water inlet holes of the shared water chamber are uneven, the coolant pressure distributions inside the water jackets of the two cylinder blocks become uneven, too.

Figure 16 illustrates the coolant velocity vector distribution on the inner surface of the cylinder block water jacket, as well as the coolant velocity distribution on a cross-sectional area along the Z-direction of the water jacket. As shown in Fig. 16, the coolant flow velocity distributions exhibit complicated flow patterns in the water jacket with uneven distributions caused by the original designs of the four water inlet holes in the cylinder block. Particularly, there is a great difference in flow distribution between the two cylinders. In this original design, the coolant coming from the two water inlet holes located at both sides of the cylinder entered the water jacket of the cylinder block, and merged in the upper region of each cylinder at the intake port side, accompanying with flow collisions and disturbance, and then a part of the flow went to the cylinder head and the other part flowed to the lower exhaust port side. Because the coolant flow rate and velocity of No. 1 water inlet hole were less than those of No. 4 water inlet hole, the flow disturbance occurring in the upper region of cylinder 1 was significantly less than that in cylinder 2. Moreover, a tangential arrangement on the cylinder liner for the water inlet holes in each cylinder was used in the original design, and it leads to very low flow velocities at the exhaust port side.

Figure 17 shows the distribution of heat transfer coefficient on the inner surface of the cylinder water jacket. It is observed that the distribution is uneven, with high heat transfer coefficients occurring at the two sides of the plane perpendicular to the thrust plane. The heat transfer coefficients at the exhaust port side are low due to insufficient coolant flow, resulting in under-cooling. The heat transfer coefficients at the intake port side have some low values locally because of undesirable collisions of the coolant flows coming from both sides, the interference of the coolant flow caused by the bolt shoulder in the cylinder head, and the associated energy losses of the coolant flow.

Figure 18 shows the coolant flow velocity and heat transfer coefficient distribution on the bottom plane of the cylinder head. The calculated coolant average flow velocity on the bottom plane of the cylinder head is 0.66 m/s, and the calculated average heat transfer coefficient is 4810.4 W/(m^2.K), which are apparently low. It is observed that the coolant flow velocity at the intake port side is higher than that at the exhaust port side in the original design, and the coolant flow velocity at the water inlet hole of cylinder 1 is higher than that of cylinder 2. Because the location of the water outlet port was near the exhaust port of cylinder 1, it is believed that the water inlet holes in the cylinder block were unreasonably located in the original design, and it resulted in a poor design of the water inlet holes in the cylinder head in terms of passage locations. Because of the difference between the two cylinders in coolant flow velocity at the bottom plane of the cylinder head, coolant heat transfer coefficients of the two cylinders are also uneven. The heat transfer coefficient of cylinder 1 is significantly greater than that of cylinder 2, as seen from the heat transfer coefficient distribution in Fig. 18(b).

Figure 19(a) shows the coolant flow distribution on the cross-sectional plane in the valve bridge area in the cylinder head. It is observed that the coolant flows well in the valve bridge area in cylinder 1 with a spatial average flow velocity 1.03 m/s. The flow velocities in the upper portion in Fig. 19 are high with a smooth distribution, and the flow velocities in the lower portion are low with vortices. As a comparison, the coolant of cylinder 2 flows worse than cylinder 1, with a spatial average velocity 0.87 m/s and low flow velocities in both upper and lower portions. All of these phenomena revealed by the CFD simulation are directly related to the design of the shared water chamber and the water inlet holes in the cylinder block and in the cylinder head, and the original design needs to be optimized in order to improve cooling performance. As shown in the coolant heat transfer coefficient distribution at the valve bridge area in Fig. 19(a), the heat transfer coefficients of the two cylinders differ greatly due to the unreasonable design of the original water jacket structure. The heat transfer coefficient of cylinder 1 is 4625 W/m^2.K and the coefficient of cylinder 2 is 3173 W/m^2.K in the valve bridge area. Both of them do not satisfy the cooling requirements.

There were ten water inlet holes (No. 1-10) and three head holes (No. 11-13) in the original water jacket design, as shown in Fig. 20. The calculated coolant mass flow rates of the water inlet holes in the cylinder head are shown in Fig. 21. It is observed that the flow rate distribution is uneven among different water inlet holes, with the sum of the flow rates of No. 2-5 holes at the intake port side occupying 46% of the total flow rate, and the sum of the flow rates of No. 2-5 holes at the exhaust port side occupying only 32% of the total flow rate. Because No. 1 water inlet hole is close to the water jacket outlet port, 13% of the total coolant flow goes through No. 1 water inlet hole and then directly flows out of the water jacket without being fully utilized. Moreover, 9% of the total coolant flow enters the water jacket from No. 6 water inlet hole.

OPTIMIZATION OF COOLING WATER JACKET STRUCTURE DESIGN AND WATER PUMP FLOW RATE

In a horizontal diesel engine, the cylinders are arranged horizontally and the pistons also move horizontally. The coolant flow pattern in the cooling water jacket of a horizontal engine is very different from that of a vertical engine. Therefore, reasonably designing the water jacket structure for a horizontal diesel engine is very important, especially the structures of the shared water chamber, the water inlet holes of the cylinder block, and the water inlet holes of the cylinder head. The quality of the design of these parts directly affects the cooling evenness between the cylinders in the cylinder block and the cylinder head, and also affects the cooling evenness between the upper and lower parts of each cylinder.

The above CFD analysis shows that the original water jacket design basically met the requirements of engine design. However, the coolant flow velocity and heat transfer coefficient distributions were uneven, with large vortices occurring in the middle of the shared water chamber and in the top portion of the water jacket of cylinder 2. There were also some dead zones in the valve bridge area in the cylinder head of cylinder 2 and in some local areas under the two exhaust ports. Therefore, the water jacket structure needs to be optimized.

After an investigation on the impacts of different structural and packaging design parameters on coolant flow and heat transfer by using Design of Experiments (DOE) with orthogonal arrays, a new design was proposed to revise and optimize the water jacket structure as follows. (The details of the DOE study are not included in this paper due to space limit.) First, the cross section of the shared water chamber has been revised from the previous square shape to a circular shape with a radius of 16 mm. Only two water inlet holes/passages in the cylinder block, B1 and B4, which connect the shared water chamber to the cylinder water jacket, have been retained in the new design. The diameter of the water inlet hole B1 has been increased from the previous 12 mm to 22 mm. The water inlet hole B4 has been revised from a circular section of diameter 18 mm to a semi-circular section of diameter 28 mm. Secondly, only five water inlet holes/passages in the cylinder head, H2, H4, H6, H7, and H9, which connect the cylinder block water jacket to the cylinder head water jacket, have been retained. The schematic diagram of the optimized cooling water jacket is shown in Fig. 22.

The coolant flow velocity distribution of the entire optimized water jacket is shown in Fig. 23. It is observed that the overall spatial average velocity of the coolant in the water jacket has reached 1.40 m/s, increased by 40% compared to the original design. The spatial average coolant velocities of the water jackets in the cylinder block and in the cylinder head have reached 1.59 m/s and 1.20 m/s, respectively, increased by 43.2% and 41.2%. There are no large vortices any more in the shared water chamber, as shown in Fig. 23(b). The coolant flow tangentially entering the water jacket of the cylinder block via the water inlet holes B1 and B4 can flow smoothly and efficiently without the flow collisions and large vortices that occurred in the top area of the cylinder block water jacket as shown in the original design. Figure 23(c) shows that the coolant flow rate at the bottom plane of the cylinder head water jacket has increased significantly, reaching 1.02 m/s, increased by 54.5% compared to the original design. The distributions of coolant flow velocity in various areas of each cylinder and at the intake and exhaust port side are uniform. The coolant flow pattern of the optimized design in the valve bridge area has been significantly improved, reaching 1.13 m/s, raised by 18.9% compared to the original design. Moreover, the cylinder-to-cylinder variation of coolant flow patterns has also been greatly improved.

The heat transfer coefficient distribution of the optimized water jacket design is shown in Fig. 24. It is observed that the overall spatial average heat transfer coefficient has reached 11009 W/(m^2.K), increased by 41.7% compared to the original design. The average heat transfer coefficients of the water jackets in the cylinder block and in the cylinder head have reached 11975 W/(m^2.K) and 9843 W/(m^2.K), respectively, increased by 30.0% and 67.8%. Therefore, the overall heat transfer coefficient of the water jacket has increased substantially after the design optimization, especially in the cylinder head, which is subject to high thermal loading, with a large increase of 67.8%. The heat transfer coefficients of the optimized water jacket design on the bottom plane of the cylinder head and in the valve bridge area in the cylinder head have increased, reaching 9555.5 W/(m^2.K) and 5641.9 W/(m^2.K) respectively, and corresponding to 98.6% and 44.7% increases respectively compared to the original design.

Figure 25 illustrates the pressure distribution of the coolant for the optimized water jacket design. It is observed that the coolant pressure in the shared water chamber has greatly increased compared to the original design because the cross-sectional area of the shared water chamber is significantly reduced from the original area of 1520 mm^2 to the optimized area of 803.8 mm^2. The overall coolant pressure drop of the optimized water jacket is 0.042 MPa, increased by 16.7% compared to the original design, under the same boundary condition of water pump flow. It should be noted that the optimized water jacket design can actually reduce the required water pump flow rate and the associated parasitic power loss.

Figure 26 presents a comparison of the coolant mass flow rate at the water inlet holes in the cylinder head between the original design and the optimized design. It is observed that the coolant flow rate has significantly increased after the optimization, with the largest gain of an increase by 2.15 times occurring at the water inlet hole H9. The coolant flow rate at the intake port side has slightly decreased by 5.8% after the optimization, and the coolant flow rate at the exhaust port side has greatly increased by 33.9%. The ratio of the coolant flow rate between the intake port side and the exhaust port side in the original design was 1.51. After the design optimization, the ratio has changed to 1.06, demonstrating a great improvement in reducing the gap or unevenness in cooling capability between the two sides.

The water pump has been also optimized by changing the pulley ratio to reduce pump speed and power according to the changes in the optimized water jacket design. Based on the optimized calculation results, a water pump flow rate of 3.1 kg/s is sufficient for the engine cooling requirement at rated power. Table 2 shows a comparison of water pump and cooling system performance before and after the design optimization. It is observed that the optimized water jacket design results in a reduction in water pump flow rate from 3.65 kg/s in the original design to 3.1 kg/s in the optimized design, with a corresponding reduction of 0.32 kW in water pump power from 2.10 kW to 1.78 kW. As a result, the engine brake power loss can be reduced by 0.88% at the rated power condition. Assuming the brake specific fuel consumption at rated power is 240 g/(kW.h), such a saving in water pump power can result in a fuel consumption saving of 76.8 g/h. All the engine and cooling performance parameters can still meet the engine requirements after the optimization or reduction of water pump flow rate.

CONCLUSIONS

A cooling water jacket structure with forced-cooling closed-loop system was designed and optimized to fit the required structural characteristics of a new generation modern horizontal diesel engine. The coolant flow field, pressure distribution, heat transfer coefficient distribution, and cylinder-to-cylinder uniformity were investigated with both experimental and CFD numerical simulation techniques. Complete performance characteristics of the cooling system/component design are illustrated in this paper by advanced CFD modeling for the horizontal diesel engine. Both water jacket structure and water pump flow rate have been optimized. The following conclusions can be drawn from the design, testing, and analysis.

1. When the forced-cooling closed-loop system is used in horizontal diesel engines, the coolant flow characteristics in the water jacket are complex, and it requires careful design optimization to solve the challenges. Using a unique and patented shared water chamber as coolant inlet located in the space between the pushrod chamber and the oil pan below the cylinders may improve the coolant flow, achieve effective coolant circulation, and avoid vapor lock in the cooling circuit.

2. The cross-sectional area of the water inlet holes/passages in the cylinder block and their layout positions greatly affect the coolant flow characteristics in the water jacket. Using one tangential water inlet hole for each cylinder can greatly improve the coolant flow conditions in the water jacket, increase coolant flow velocity, and reduce vortices and flow losses.

3. The location, amount, and cross-sectional area of the water inlet holes/passages in the cylinder head of horizontal diesel engines have great influence on coolant flow characteristics in the water jacket. A reasonable design and layout of the water inlet holes in the cylinder head can significantly increase the overall coolant flow velocity in the water jacket of the cylinder head, reduce cylinder-to-cylinder variation in cooling, improve the cooling in the critical valve bridge area in the cylinder head to sustain high thermal loading, and enhance the effectiveness of cooling at the exhaust port side.

4. Optimizing water jacket structure design can reduce the required water pump flow rate while still satisfying engine cooling requirements. The reduced water pump flow rate can lead to a significant reduction in water pump power consumption and engine fuel consumption. As a result, the entire engine cooling system can be optimized to enhance the effectiveness of cooling, and engine efficiency can be improved.

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Figure 1. The new generation off-road horizontal diesel engine.

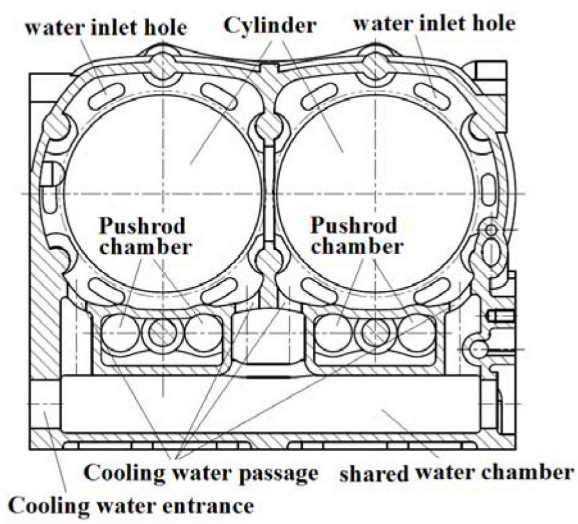


Figure 2. Cylinder block water jacket structure.

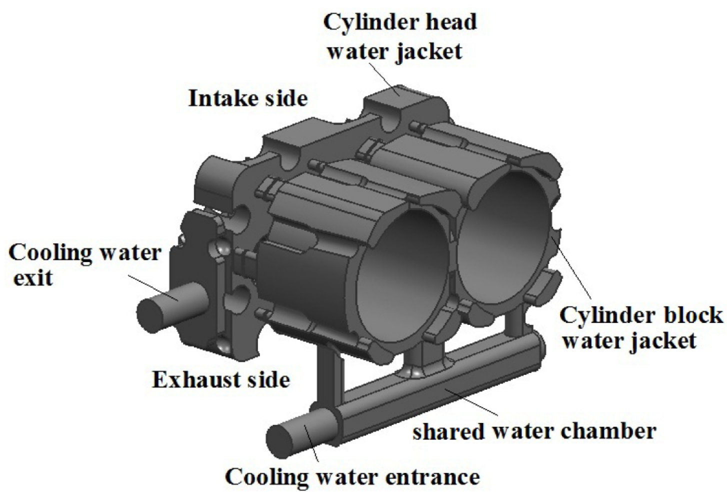


Figure 3. Three-dimensional model of cooling jacket.

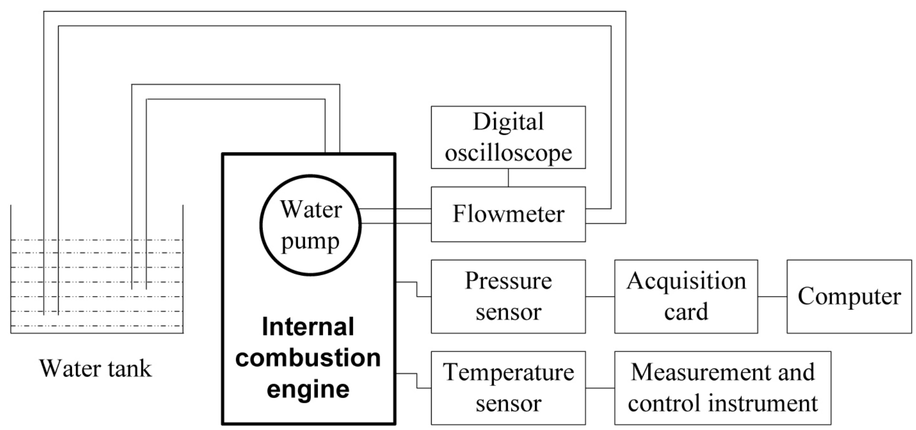
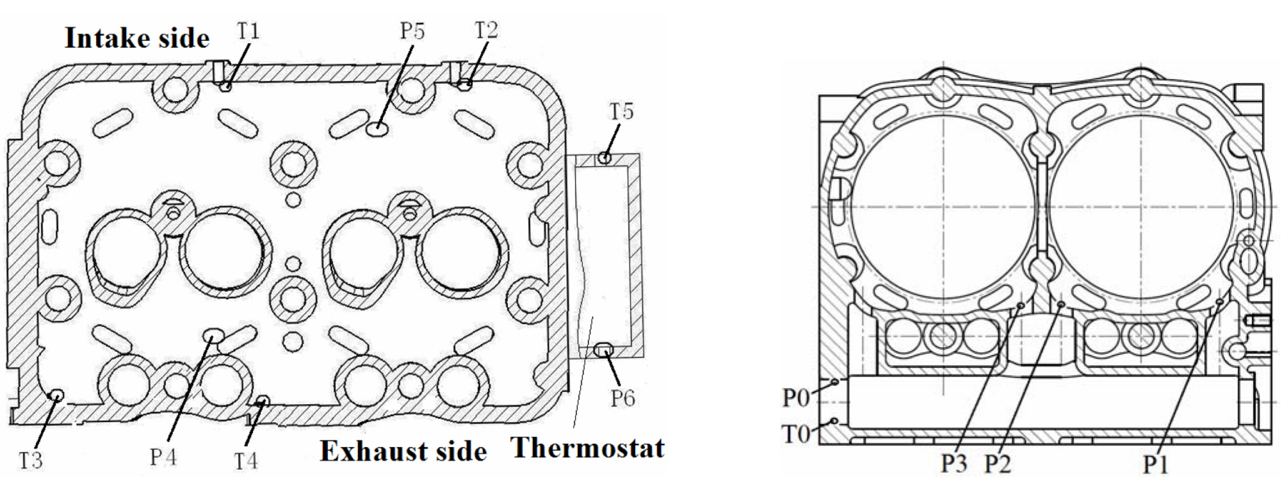


Figure 4. Illustration of testing instrument.



(a) Cylinder head *(b) Cylinder block*

Figure 5. Arrangement of measurement locations in the water jackets of the cylinder head and the cylinder block.

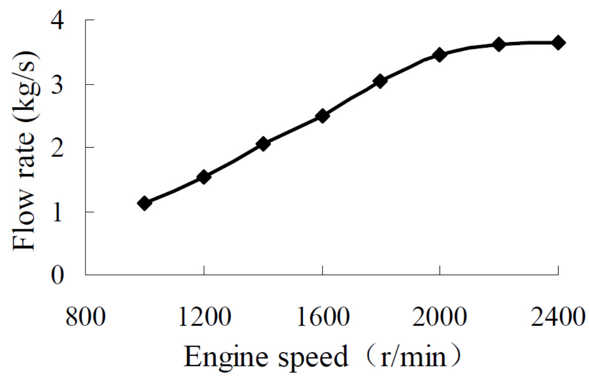


Figure 6. Variation of water pump outlet flow rate.

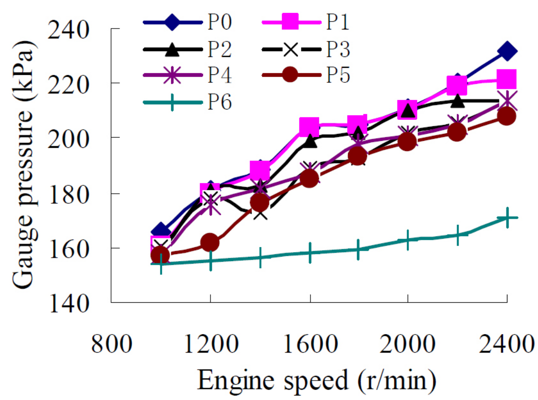


Figure 7. Variation of coolant pressures.

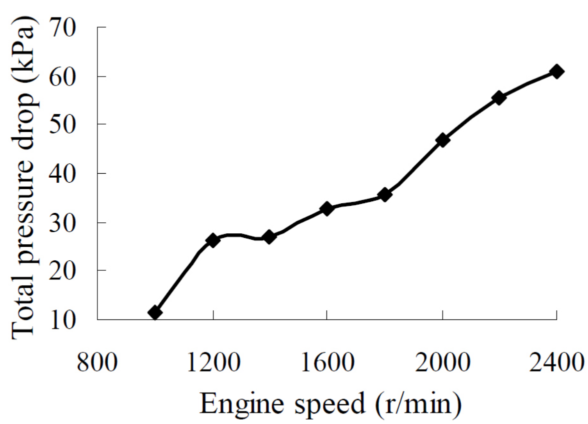


Figure 8. Variation of the total coolant pressure drop.

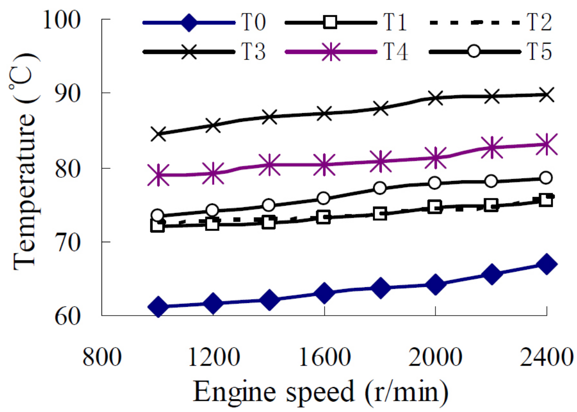


Figure 9. Variation of coolant temperatures.

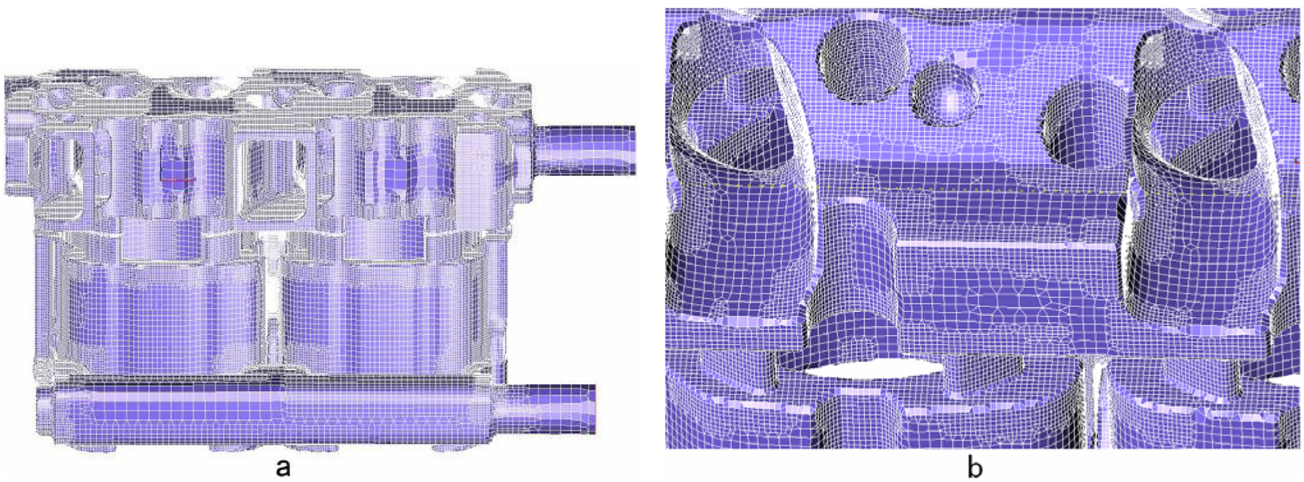


Figure 10. The computational model of the water jacket (a - overall computational grid; b - local grid of the intake port side).

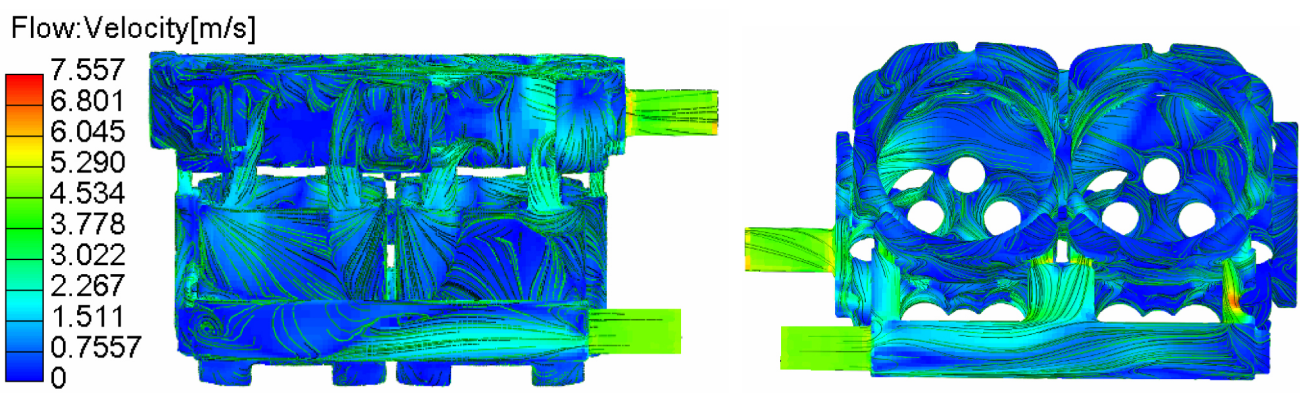


Figure 11. Coolant flow velocity distribution in the cooling water jacket.

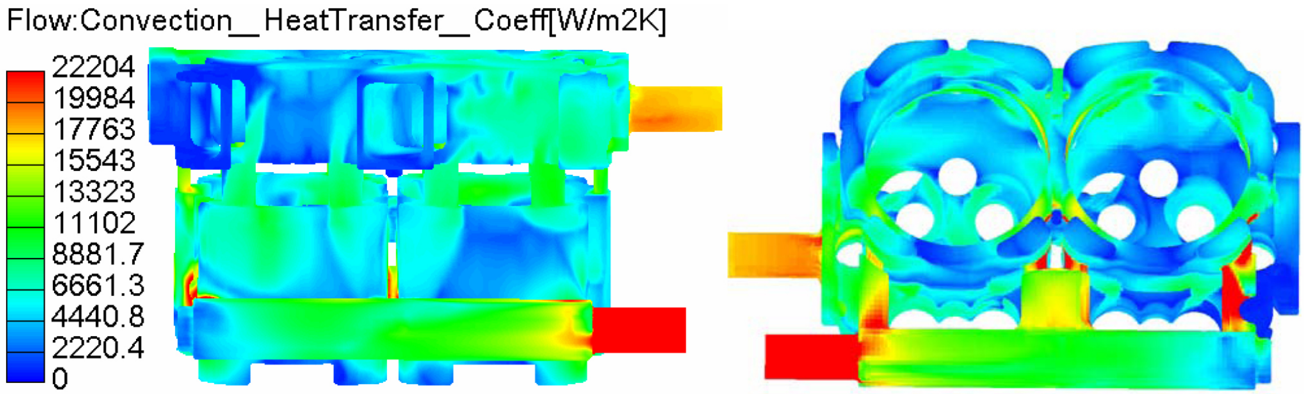


Figure 12. Heat transfer coefficient distribution in the cooling water jacket.

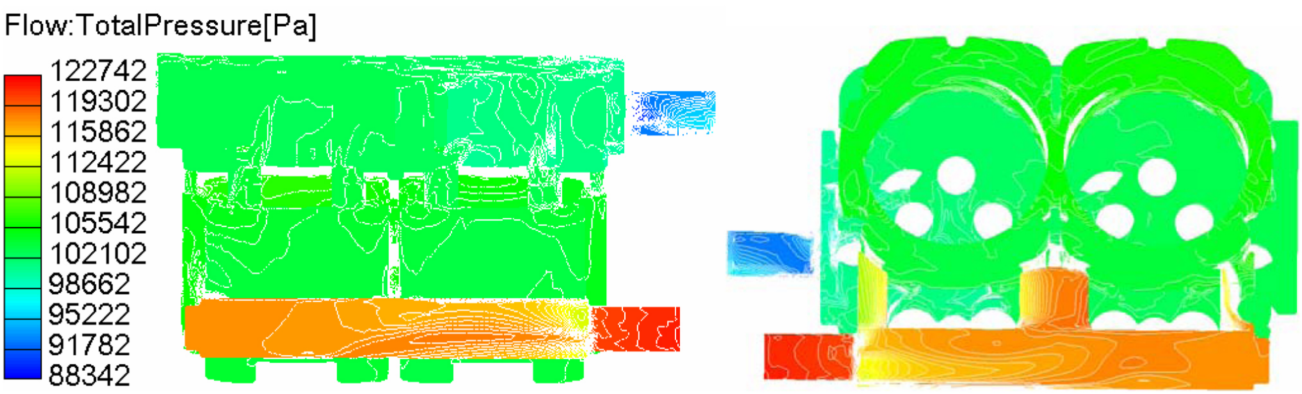


Figure 13. Coolant pressure field in the cooling water jacket

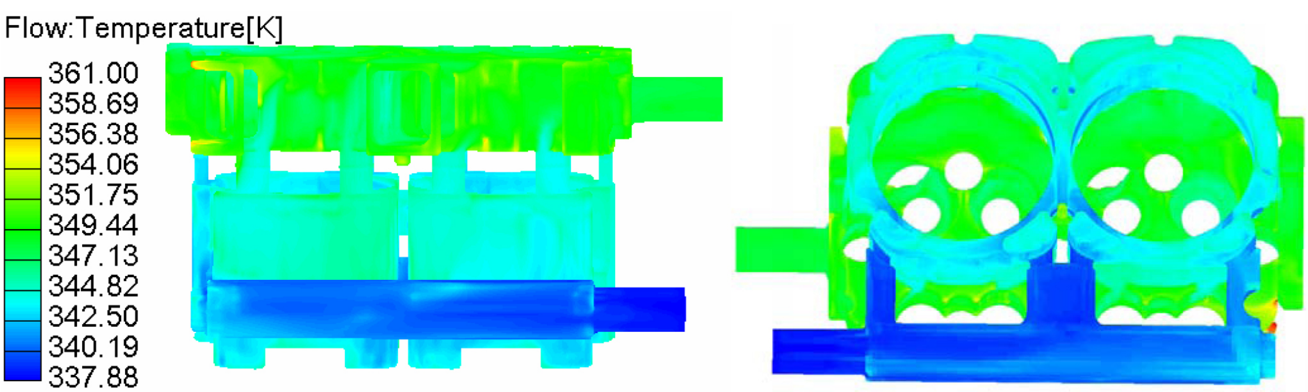
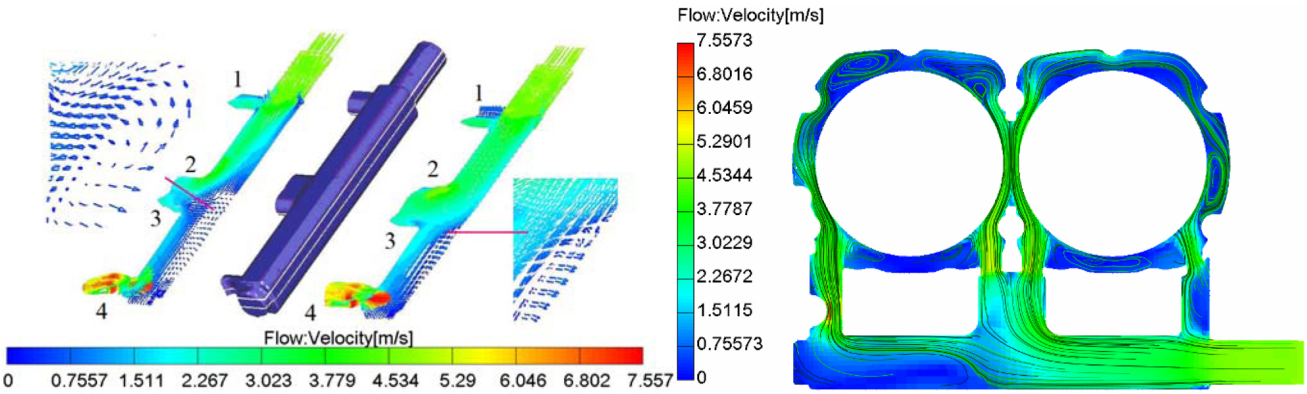
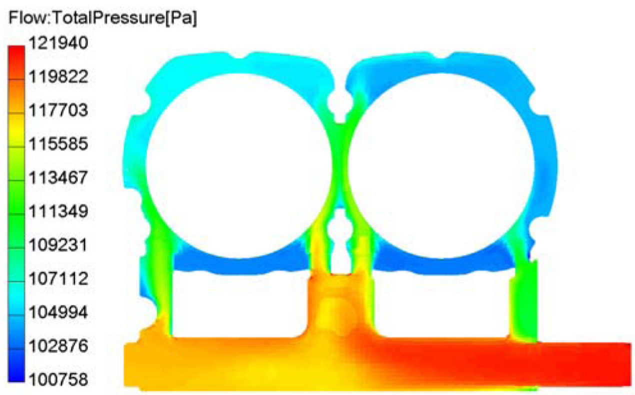


Figure 14. Coolant temperature field in the cooling water jacket.



(a) Coolant flow velocity distribution



(b) Coolant pressure distribution

Figure 15. Coolant flow velocity and pressure distributions in the shared water chamber.

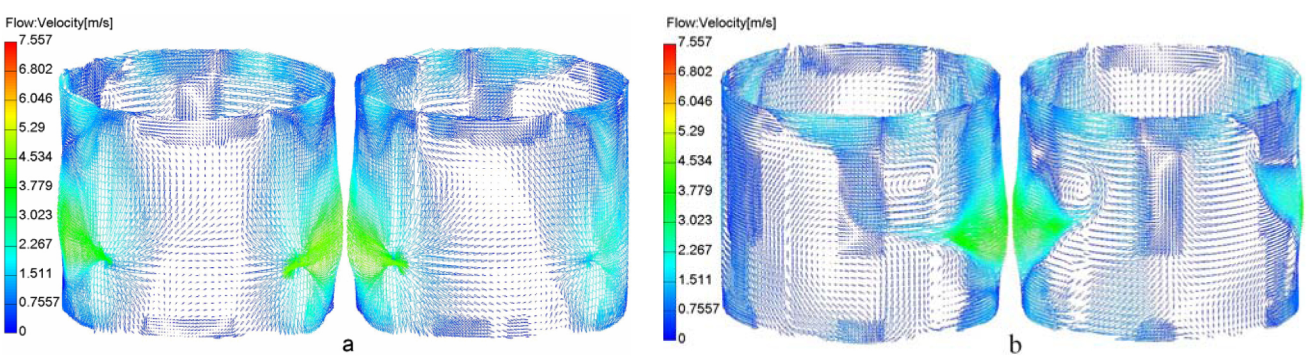


Figure 16. Flow velocity vector distribution on the cylinder inner surface (a - exhaust side; b - intake side).

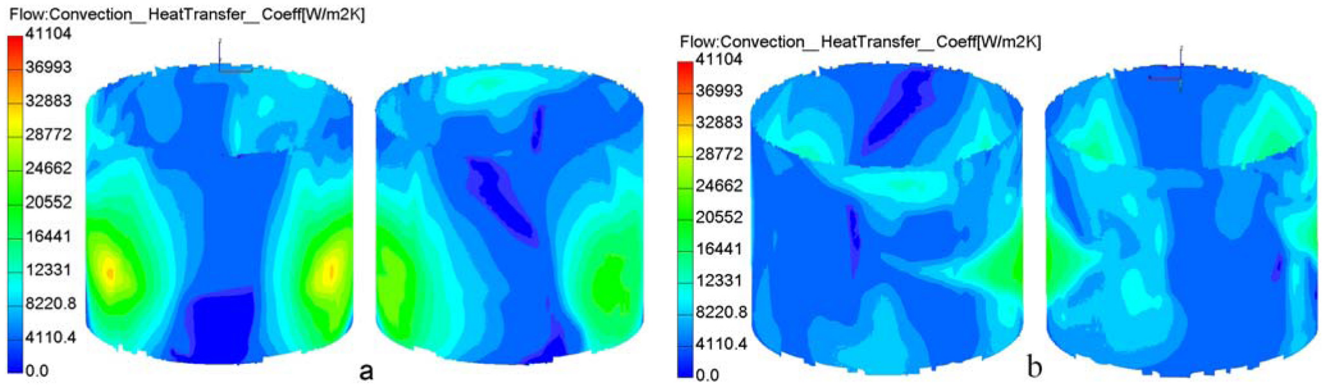


Figure 17. Heat transfer coefficient distribution on the cylinder inner surface (a - exhaust side; b - intake side).

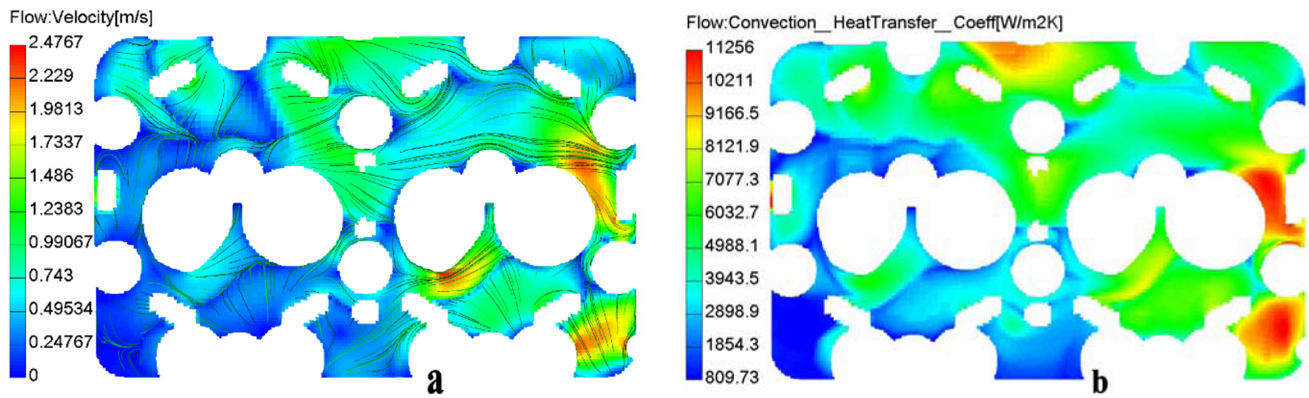


Figure 18. Coolant flow velocity and heat transfer coefficient distributions on the cylinder head mounting surface (a - coolant flow velocity, b - heat transfer coefficient).

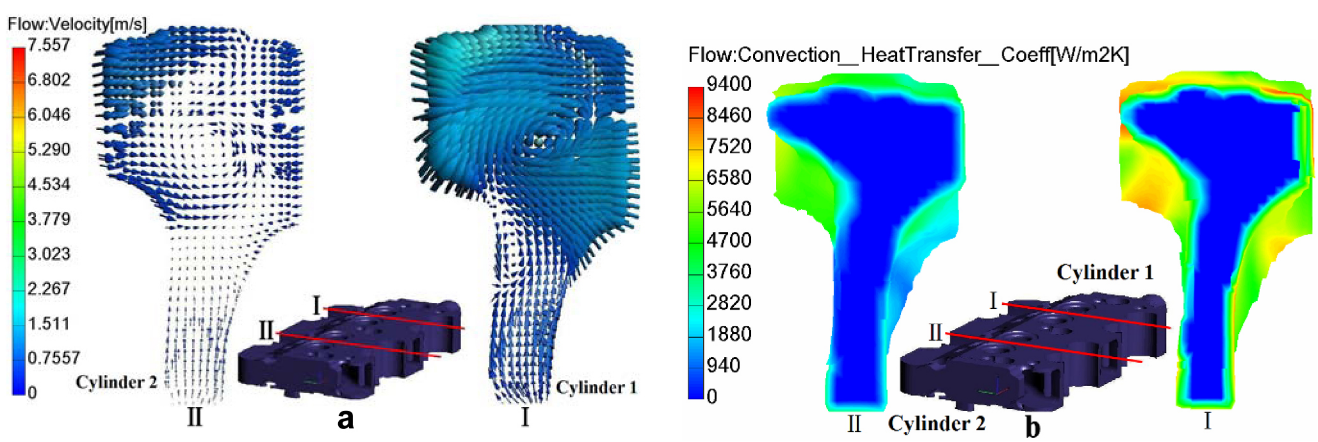


Figure 19. Coolant flow velocity and eat transfer coefficient distributions in the valve bridge area in the cylinder head (cross-sectional view; a - coolant flow velocity, b - heat transfer coefficient).

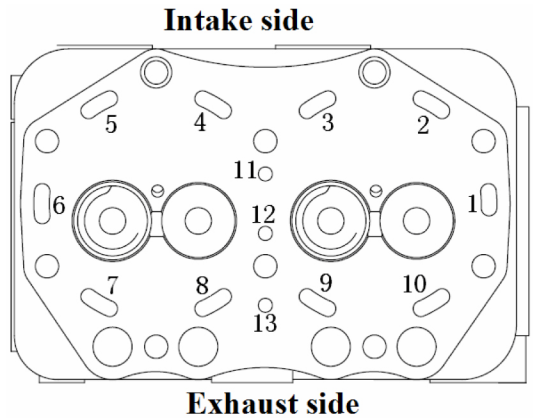


Figure 20. The layout of water inlet holes/passages of the cylinder head in the original design.

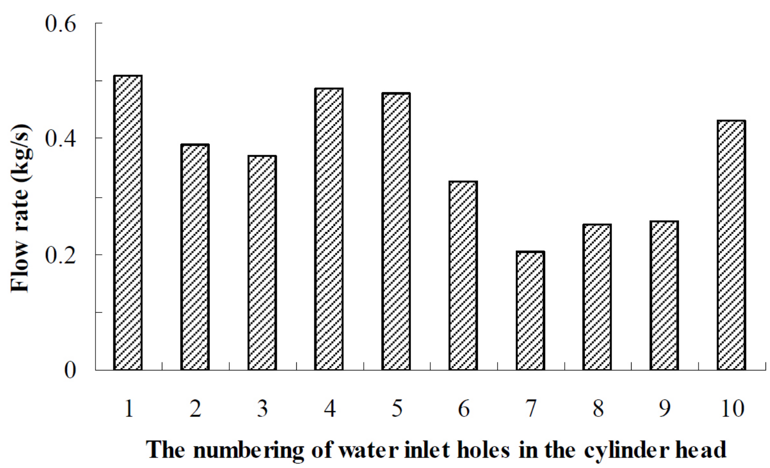
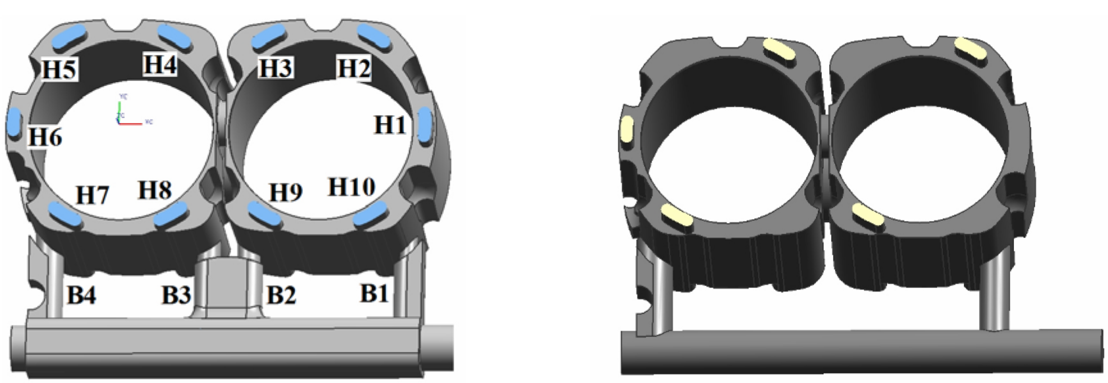
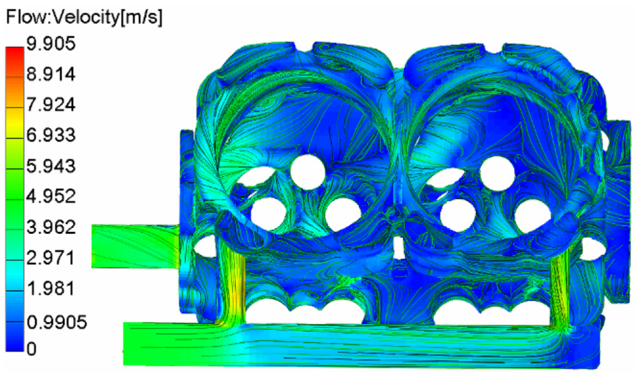


Figure 21. The coolant flow rates of cooling water passages of the cylinder head in the original design.

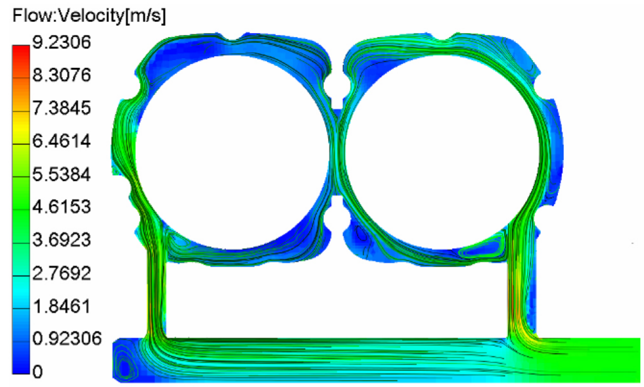


(a) The original design before optimization *(b) The revised design after optimization*

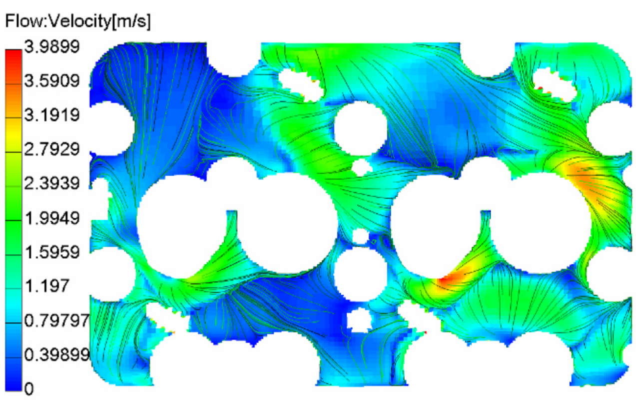
Figure 22. Different structure designs of the cylinder block water jacket.



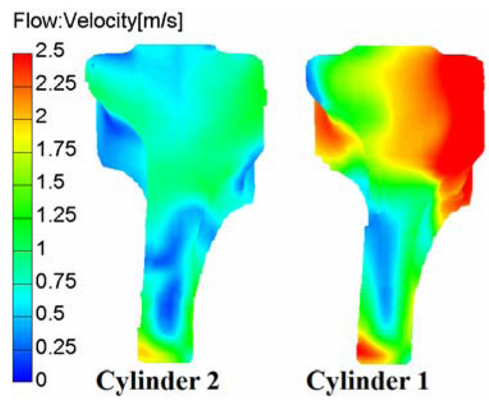
(a) Overall distribution



(b) Distribution in the shared water chamber

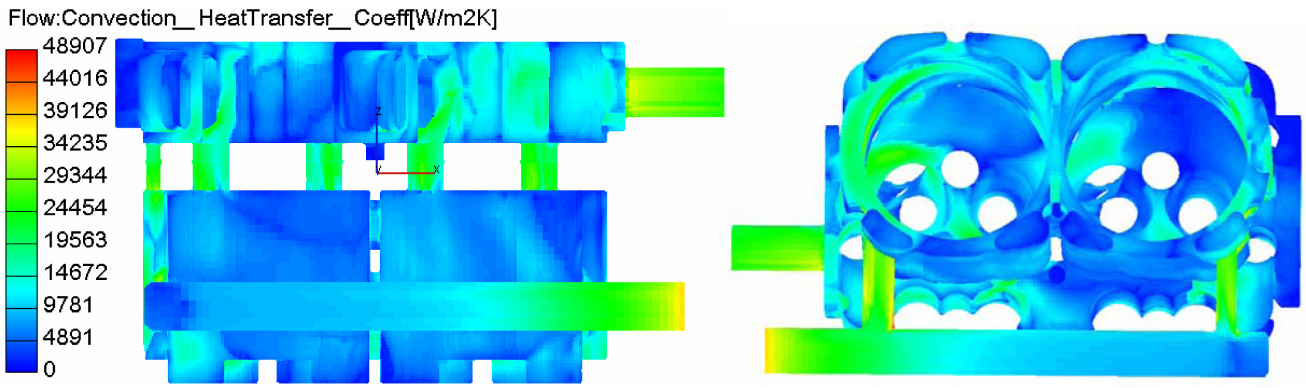


(c) Distribution on the cylinder head mounting surface

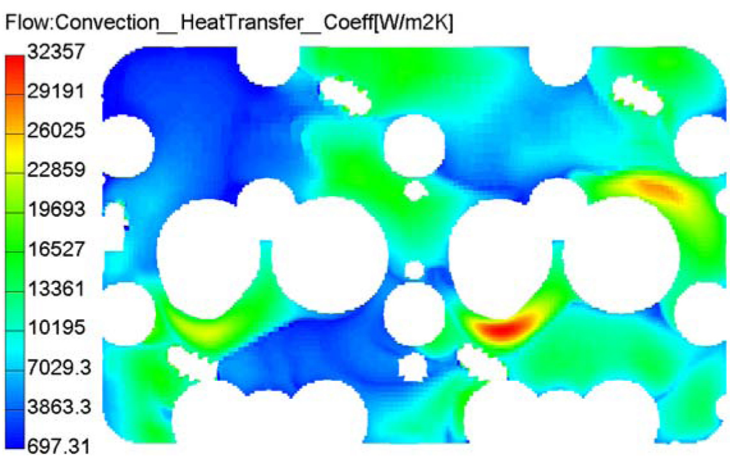


(d) Distribution in the valve bridge area in the cylinder head (cross-sectional view)

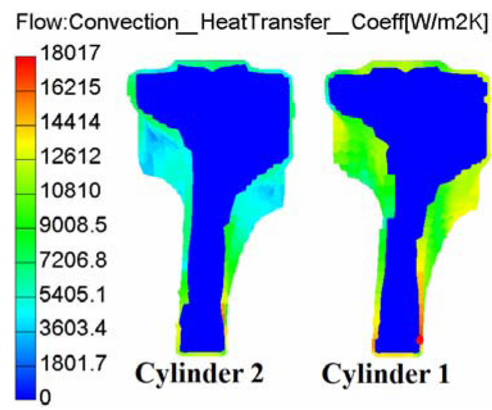
Figure 23. Coolant flow velocity distributions in the cooling water jacket after design optimization.



(a) Overall distribution

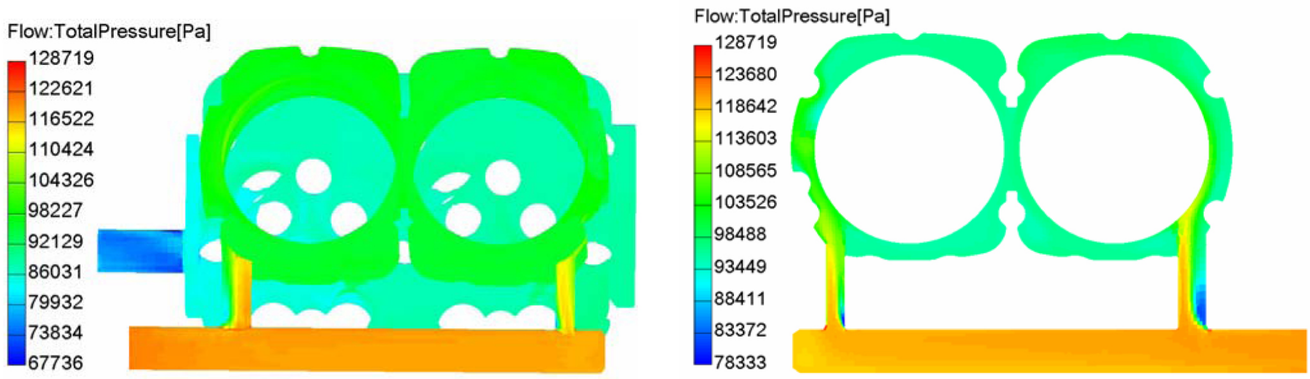


(b) Distribution on the cylinder head mounting surface



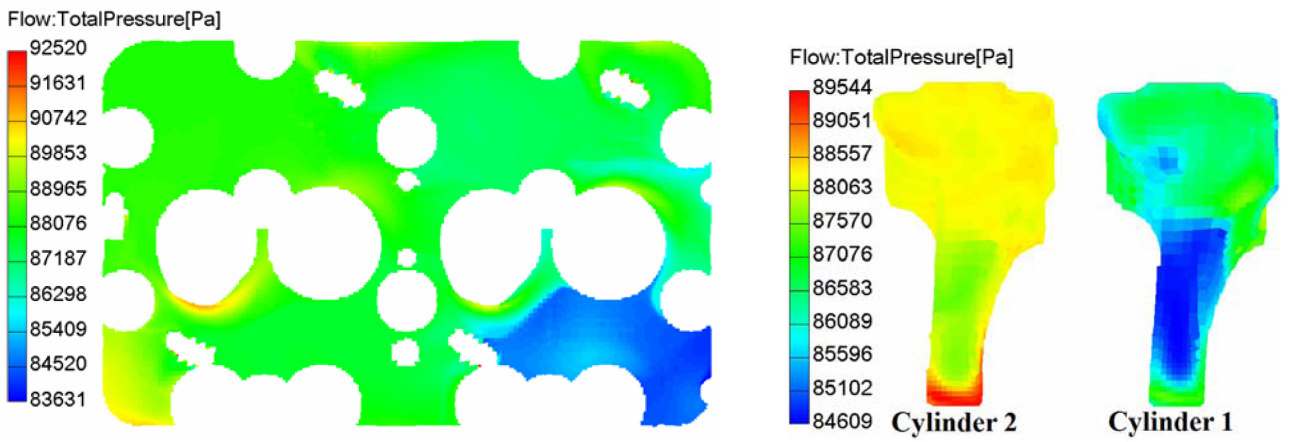
(c) Distribution in the valve bridge area in the cylinder head (cross-sectional view)

Figure 24. Heat transfer coefficient distributions in the cooling water jacket after design optimization.



(a) Overall distribution

(b) Distribution in the shared water chamber



(c) Distribution on the cylinder head mounting surface

(d) Distribution in the valve bridge area in the cylinder head (cross-sectional view)

Figure 25. Coolant pressure distributions in the cooling water jacket after design optimization.

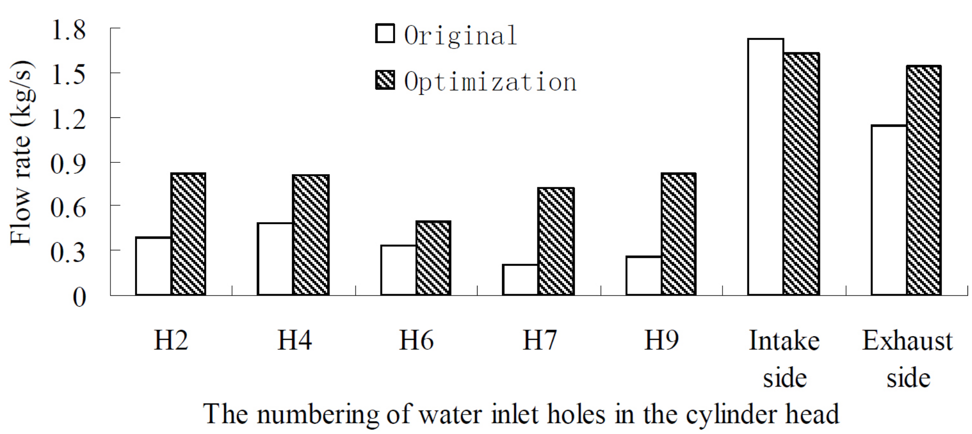


Figure 26. Coolant flow rate comparison at the cooling water passages in the cylinder head between the original design and the optimized design.

Table 1. Coolant pressure comparison between calculation and experiment

Measurement location	P_1	P_2	P_3	P_4	P_5	P_6
Experimental data	140.0 kPa	132.2 kPa	132.0 kPa	132.2 kPa	126.1 kPa	103.4 kPa
Computational data	146.9 kPa	144.9 kPa	144.3 kPa	140.2 kPa	138.5 kPa	114.8 kPa
Error or discrepancy	4.9%	9.6%	9.3%	6.1%	9.8%	11.1%

Table 2. Engine coolant flow characteristics comparison of the water jacket between the original design and the optimized design of the water pump

		Original water pump		Optimized water pump	
		Cylinder 1	Cylinder 2	Cylinder 1	Cylinder 2
Water pump flow rate		3.65 kg/s		3.1 kg/s	
Spatial average coolant flow velocity	Valve bridge area	1.58 m/s	0.98 m/s	1.41 m/s	0.76 m/s
	Entire water jacket	1.4 m/s		1.2 m/s	
	Cylinder head water jacket	1.2 m/s		1.03 m/s	
	Cylinder block water jacket	1.59 m/s		1.36 m/s	
Spatial average heat transfer coefficient	Valve bridge area	10817 W/(m^2.K)	8869 W/(m^2.K)	8217 W/(m^2.K)	6370 W/(m^2.K)
	Entire water jacket	11009 W/(m^2.K)		9798 W/(m^2.K)	
	Cylinder head water jacket	9843 W/(m^2.K)		8730 W/(m^2.K)	
	Cylinder block water jacket	11975 W/(m^2.K)		10672 W/(m^2.K)	
Water pump power		2.10 kW		1.78 kW	

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

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