Heat transfer analysis of piston cooling using nanofluids in the gallery

Hao Liu, Minli Bai, Jizu Lv, Peng Wang, Yuyan Wang, Chengzhi Hu

School of Energy and Power Engineering, Dalian University of Technology, Dailan, 116024, People's Republic of China
E-mail: liuyuhao1985@vip.qq.com

Published in Micro & Nano Letters; Received on 1st April 2014; Revised on 21st October 2014; Accepted on 27th November 2014

Attaining proper temperature in engines is always a challenge for engine manufacturers primarily because temperature has a significant role in engine performance and emissions. With the development of new technology in the field of nanofluids, it seems very promising to use nanofluids as the coolant in internal combustion engines. In this reported work, Cu and diamond nanoparticles with diameters of 50 nm were dispersed in conventional engine oil to create nanofluids (volume fractions of 1, 2 and 3%) to serve as the cooling medium for a piston-cooling gallery. The piston thermal loading calculations were carried out based on numerical hydrodynamic simulation through a liquid–solid-coupled thermodynamic method. The obtained results indicate that using different volume fractions of nanofluids can effectively reduce the thermal loading of the piston.

1. Introduction: Continuous technological development in engine manufacturing has increased the demand for high-efficiency engines. The thermal loading in internal combustion (IC) engines and its effect on the durability and reliability of engines have received much attention from researchers. The piston cooling system is one of the critical components affecting the engine thermal management, and it has been a source of concern for engine manufacturers. The piston suffers periodic thermal and mechanical loading because of long-term exposure to the combustion chamber [1]. Prolonged exposure to high temperatures might result in carbon deposits on the piston, which eventually leads to poor performance of the IC engine, such as blow-by and high oil consumption [2]. Therefore, it has become vitally important to maximise the efficiency of the piston cooling system. However, with the cylinder pressure in IC engines increasing, simply modifying or extending the cooling gallery structure might be insufficient until the power rating reaches a given level [3]. Therefore, selecting a cooling medium with a better heat transfer capacity has become necessary as an additional cooling measure.

Choi and Eastman [4] proposed the concept of 'nanofluids', which is a new class of heat transfer fluid engineered by dispersing nanometre-sized particles in a base fluid to increase the thermal conductivity and the heat transfer performance of the engine. They showed that adding a small amount of nanoparticles (<1 vol%) to a base fluid would approximately double the thermal conductivity of the fluid. One of the key purposes of preparing nanofluids is to enhance the fluid heat transfer. Yu *et al.* [5] found that an increase in heat transfer of ∼15–40% can be achieved by using different types of nanofluids.

There is a significant need in many industrial fields for oil-based heat transfer fluids for energy-efficient heat exchangers. Thus, much effort has been focused on the development of oil-based nanofluids. In recent years, with the development of numerical techniques and computer technology, it has become possible to obtain detailed knowledge by modelling the piston cooling system using computational fluid dynamics (CFD) software.

In 2003, Kajiwara *et al.* [6] performed a basic two-dimensional (2D) CFD analysis on simplified gallery geometry, predicting the variation trend of the heat transfer coefficient with respect to the amount of oil in a gallery. In 2007, Yi *et al.* [7] employed a multiphase approach, using the volume of the fluid and the dynamic mesh model to evaluate the heat transfer in a piston cooling gallery. In 2010, Zhang Weizheng *et al.* [8] conducted numerical research on transient heat transfer in an oscillating piston and obtained parameters of oil fill ratio and wall heat transfer coefficient against crank angle at different engine speeds. In 2013, Wang *et al.* [9] calculated the heat transfer coefficients of Cu–oil and diamond–oil nanofluids in the piston cooling gallery directly using a coupled volume of fluid (VOF)/level-set method. As compared with traditional engine oil, the total averaged heat transfer coefficients of Cu–oil nanofluids with volume fractions of 1, 2 and 3% increased by 7.77, 17.08 and 29.33%, respectively, whereas those of diamond–oil nanofluids increased by 12.01, 29.14 and 44.33%, respectively. Nanofluid coolants thus improve the heat exchanger performance and increase the cooling effectiveness and the engine power.

Research in the field of nanofluids has shown that there are two methods for analysing the hydrodynamic and the heat transfer loading of nanofluids [10, 11], the single-phase method and the two-phase method. In the single-phase method, it is supposed that the nanoparticles have the same velocity as the base fluid molecules and that the particles are in thermal equilibrium. Thus, in the single-phase model, the nanofluids are considered similar to the base fluids with different thermophysical properties. However, in the two-phase model, the nanoparticles are considered a separate phase from the base fluid, with a different velocity and temperature. Both single-phase and two-phase models can be used to explain the mechanism of the increased heat transfer rates. From the microscopic point of view, the two-phase method is introduced to describe the effects of interactions between the suspended nanoparticles and the base liquid particles, as well as those between the solid nanoparticles. Many studies have shown that nanofluids with low concentrations can be analysed with a single-phase method [12, 13].

The modelling of piston temperature distribution is extremely important for keeping thermal loads within acceptable levels at the interfaces (top dead centre and bottom dead centre). There are many reports on the calculation of thermal loads; however, most of the literature on piston cooling is concerned with the effect on piston materials, coatings and configurations [14, 15]. The influence of thermal loads on new types of cooling media such as nanofluids is less well covered. For piston thermal management, the boundary conditions are usually introduced by a fixed value or semi-empirical expressions [16, 17]. A more accurate description of the boundary conditions is required when using different nanofluids as the cooling medium in the gallery. In the work reported in this Letter, by introducing more accurate boundary conditions [9], numerical simulations were calculated from a macroperspective to investigate the piston thermal loading when using nanofluids in the cooling

doi: 10.1049/mnl.2014.0432

Table 1 Engine specifications

Item	Specification	Accuracy
engine	BF6M1013-30E3	
number of cylinders	6	
number of strokes	4	
bore × stroke	130 × 108 mm	
maximum power	220 kW	
rated power speed	2300 rpm	
maximum torque	1100 N m	
pressure transducer		±1%
temperature sensor	K-type thermocouple	±0.75% T
fuel consumption meter	AVL FCM05	0.1–110 kg/h
dynamometer	AVL electric dynamometer (WE-42)	±0.5% F.S.

gallery. The nanofluids were assumed to be single phase and thus the thermophysical properties of the nanofluids were calculated by a single method. Cu–oil and diamond–oil nanofluids with volume fractions of 1, 2 and 3% were utilised as the piston cooling gallery medium for a diesel engine. The piston thermal loading calculation was carried out based on a numerical hydrodynamic simulation through a liquid–solid-coupled thermodynamic method.

2. Physical models and mesh generation: The specifics of the engine are given in Table 1. According to the manufacturer's specifications, the error of the K-type thermocouples is ±0.75% of the measured temperature value. The physical properties of the piston set and the cylinder line are summarised in Table 2. The hexagonal structured mesh generated for the piston set and the cylinder liner is shown in Fig. 1.

3. Thermal analysis models

3.1. Basic equations for thermal analysis: Stable thermal loading means that the temperature field of the piston remains unchanged during the working process of the piston and that the heat flowing from the combustion gas through to the piston top equals that discharged from the cooling gallery, oil film and cylinder. Based on the fundamental of heat transfer and abiding by the law of conservation of energy, we obtain a differential equation for solid heat conduction which can be simplified as

$$\frac{\partial T}{\partial \tau} = \alpha\left[\frac{\delta T^2}{\delta x^2} + \frac{\delta T^2}{\delta y^2} + \frac{\delta T^2}{\delta z^2}\right] + \frac{\omega}{c\rho} \tag{1}$$

where α is the thermal diffusivity of the material, $\alpha = k/c\rho$ in m^2/s, c is the specific heat at constant pressure of the material, ρ is the density of the material and ω is the internal heat source intensity of the material.

Although the heat conduction equation establishes the relationship between the temperature and space and time, to understand

Table 2 Physical properties of the piston set and cylinder liner

Item	Material	Thermal conductivity, W/(m·K)	Density, kg/m^3	Specific heat, J/kg·K
piston	aluminium alloy	155	2680	864
piston rings	cast alloy	39	7570	470
cylinder liner	phosphorous cast iron	52	7270	418

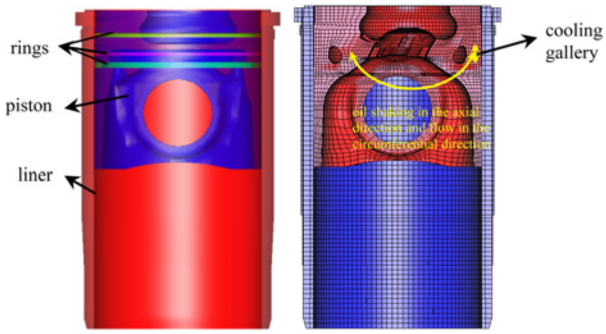

Figure 1 *Geometric model and structured mesh for the piston set and cylinder liner*

the actual temperature distribution within the piston, the boundary condition and the initial condition, which are called definite conditions, must be determined. We can then obtain the coupling solutions for the different equations. In this Letter, the third boundary condition is employed for analysing and solving the temperature distribution of the piston, which means that the heat exchange coefficient h and the temperature T_f of the cooling medium in contact with the objects are treated as variables. The equation can be expressed as follows

$$-k\frac{\partial T}{\partial n} = h(T - T_f) \tag{2}$$

where $\boldsymbol{n}$ is the exterior normal vector of the object boundary, T_f the temperature of the surrounding medium and h is the convection heat exchange coefficient.

3.2. Energy equation of the oil film: The heat transfer mode in the oil film is assumed to be one-dimensional (1D) heat conduction because the film is a thin layer. On the basis of this assumption, the heat transfer relationship between the piston and the cylinder can be linked. Assuming that the circumferential fluid flow can be neglected and the flow in the oil film is laminar, the energy equation in the oil film can be expressed as follows

$$\rho c_p u\frac{\partial T}{\partial x} = k\frac{\partial^2 T}{\partial z^2} \tag{3}$$

Considering the heat produced as a result of straight asperity contact and integrating the energy equation from 0 to h in the z direction, the energy equation is expressed as

$$\rho c_p\int_0^h u\frac{\partial T}{\partial x}\,dz = \int_0^h k\frac{\partial^2 T}{\partial z^2}\,dz + Q_2 \tag{4}$$

where ρ is the density of the lubricant, u the piston velocity, k the thermal conductivity of lubricant, c_p the specific heat capacity of the lubricant and Q_2 is the friction heat of the oil film.

3.3. Determination of the boundary conditions: The heat flux on the piston top surface is not symmetric, which causes the temperature profiles on the piston crown to vary significantly, especially at high engine speeds and loads. When calculating the stable temperature field, it is necessary to calculate the average heat transfer coefficient and the average temperature of the combustion gas with a working cycle. The average heat transfer coefficient and the average temperature of the gas are obtained by the GT-power 0D model.

The heat flux removed by the piston cooling gallery can occupy as much as 60–70% of the total heat passed to the piston from the combustion gases, which effectively decreases the piston thermal

Table 3 Average convective heat transfer coefficients of the piston cooling gallery using nanofluids [W/(m^2·K)]

Cooling medium	Top wall/increase data, %	Bottom wall/increase data, %
engine oil	2140	1858
1% Cu	2318/8.32%	1990/7.10%
2% Cu	2487/16.21%	2196/18.19%
3% Cu	2824/31.96%	2326/25.19%
1% diamond	2458/14.86%	2039/9.74%
2% diamond	2810/31.31%	2351/26.53%
3% diamond	3175/48.36%	2598/39.83%

loading [18]. Wang *et al.* [9] recently simulated the detailed oil flow and convective heat transfer process of Cu–oil and diamond–oil nanofluids with volume fractions of 1, 2 and 3%. However, the influence of piston thermal load on the heat transfer coefficients of the cooling media was not studied in that work. On the basis of the previous work, the average heat transfer coefficients can be further used for heat transfer simulation of piston cooling. The numerical simulation results employed in this Letter are shown in Table 3.

The convective heat transfer coefficient of the re-circulating cooling water in the cylinder liner was obtained from the simulation results of MInli *et al.* [19].

4. Prediction models of thermophysical properties for nanofluids

4.1. Density and specific heat capacity: The nanofluid density is the average of the base fluid and the nanoparticles density [20]

$$\rho_{\rm nf} = (1-\phi)\rho_{\rm f} + \phi\rho_{\rm p} \quad (5)$$

Assuming that the base fluid and the nanoparticles are in thermal equilibrium, the nanofluid specific heat capacity can be calculated using the following equation [21]

$$c_{\rm nf} = (1-\phi)c_{\rm f} + \phi c_{\rm p} \quad (6)$$

where the subscripts nf, p and f refer to the nanofluids, particles and base fluid, respectively, and ϕ is the nanoparticle volume fraction.

4.2. Viscosity model: The viscosity of nanofluids can be correlated with the following equation proposed by Xiaofei [22] based on the Einstein viscosity formula

$$\mu_{\rm nf} = \mu_{\rm f}\left[1 + 2.5\phi\left(1 + \frac{8.868}{r}\right)\right] \quad (7)$$

where $\mu_{\rm nf}$ is the nanofluid viscosity, $\mu_{\rm f}$ is the base fluid viscosity, ϕ the nanoparticle volume fraction and r is the nanoparticle radius.

4.3. Thermal conductivity model: The thermal conductivity of nanofluids was calculated by the prediction model of Xuan *et al.* [23]

$$\frac{k_{\rm nf}}{k_{\rm f}} = \frac{k_{\rm p} + 2k_{\rm f} + 2\phi(k_{\rm p} - k_{\rm f})}{k_{\rm p} + 2k_{\rm f} - \phi(k_{\rm p} - k_{\rm f})} + \frac{18\phi H A k_B T}{\pi^2 \rho_{\rm p} d_{\rm p}^6}\tau \quad (8)$$

where the subscripts nf, p and f refer to the nanofluids, particles and base fluid, respectively, ϕ is the nanoparticle volume fraction, H is the overall heat exchange coefficient, A is the outer surface area of the nanoparticle, k_B is the Boltzmann constant, T is the temperature and τ is the comprehensive relaxation time constant. The properties of nanofluids estimated by the above formulas are listed in Table 4.

5. Experimental validation of model: A series of specially manufactured metal plug specimens were installed on the surface of the piston to measure the temperature. The metal plugs employed in the tests were made of Gcr6 steel and were covered to maintain a constant temperature (820 ± 10°C) until they were well annealed before each test. The properties of the Gcr6 metal plugs were calibrated to have a measurement resolution of ± 5°C. All test conditions were controlled by the measurement professionals in the research and development department of the DEUTZ (Dalian) Engine Corporation. The experiment was conducted on an AVL bench test system (Fig. 2) at an engine speed of 2300 rpm and a power of 220 kW. The metal plug locations in the piston are shown in Fig. 3. The controlled test conditions are shown in Table 5. To eliminate the potential impact of ambient variations, all tests were conducted using the same conditions as much as possible. However, in engine bench tests, it is difficult to maintain a consistent temperature for the outlet water, intercooler and oil, which can be affected by the actual working conditions of the diesel engine. We repeated the tests several times and took the average of the values as the final value. Therefore, these measurements could be considered the average values of several tests. For experimental validation, other cooling conditions (oscillation cooling in the gallery and jet

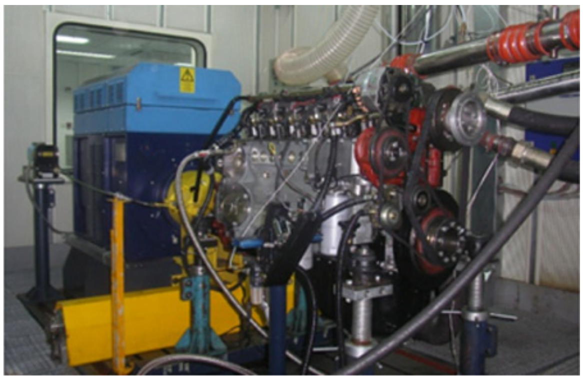

Figure 2 *AVL diesel engine bench test system*

Table 4 Properties of nanofluids with different volume fractions (373 K)

Nanoparticles	Volume fraction, %	Thermal conductivity, W/(m.K)	Viscosity, kg/(m.s)	Density, kg/m^3	Specific heat, J/(kg.K)
Cu	1	0.1616	0.00247	812	1901
Cu	2	0.1743	0.00254	895	1747
Cu	3	0.1871	0.00261	977	1619
diamond	1	0.1935	0.00247	758	2016
diamond	2	0.2381	0.00254	786	1948
diamond	3	0.2828	0.00261	814	1884

doi: 10.1049/mnl.2014.0432

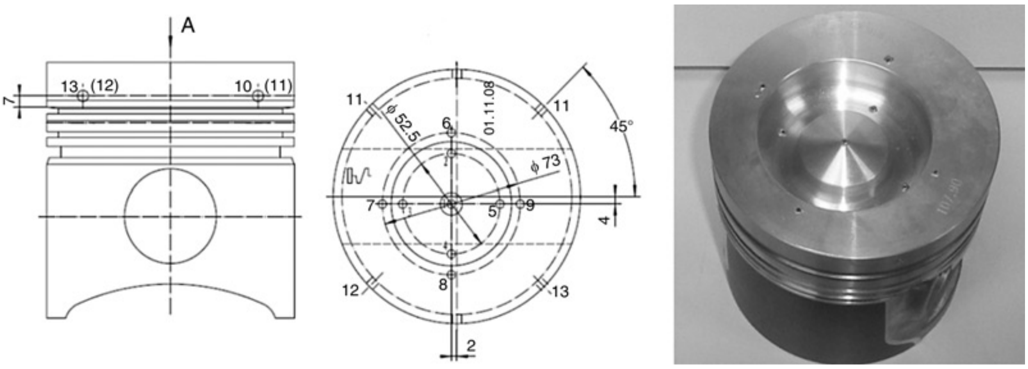

Figure 3 *Metal plug locations in the piston*

Table 5 Controlled test conditions

Item	Controlling value
temperature of outlet water	85 ± 5°C
temperature of the intercooler	45 ± 5°C
pressure drop of the intercooler	8 – 10 kPa
oil temperature	38 ± 2°C
inlet pressure	−2.8 to −3.5 kPa
intake temperature	10–40°C
discharge pressure	15 ± 0.5 kPa

cooling) must also be conducted, otherwise there might be an accident. Since it was difficult to measure the temperature of the inner surface of the cooling gallery, the numerical simulation considered independently the oscillation cooling in the gallery. The base oil and diamond–oil nanofluids were used in the validation tests. Table 6 shows the test results and the simulation results. It is observed that the simulation results match reasonably well with the experimental data. These experiments validate the model and confirm that the model is able to predict the temperature distribution of pistons with reasonable accuracy under various boundary conditions. Furthermore, the addition of nanoparticles can lead to variations in the thermophysical properties of nanofluids, which is inevitable when there are variations in thermal loads. Hence, this simulation can be used for further analysis of piston conditions by using nanofluids with parametric variations**.**

6. Results and discussion: On the basis of the established model, the definite boundary conditions with a stable thermal analysis module of finite element analysis software, and the predicted convective heat transfer coefficients and thermophysical properties for nanofluids, we carried out the calculations and obtained the results shown in Fig. 4. It is observed that the piston temperatures are generally reduced by using nanofluids as the cooling media in the piston gallery. The piston temperature

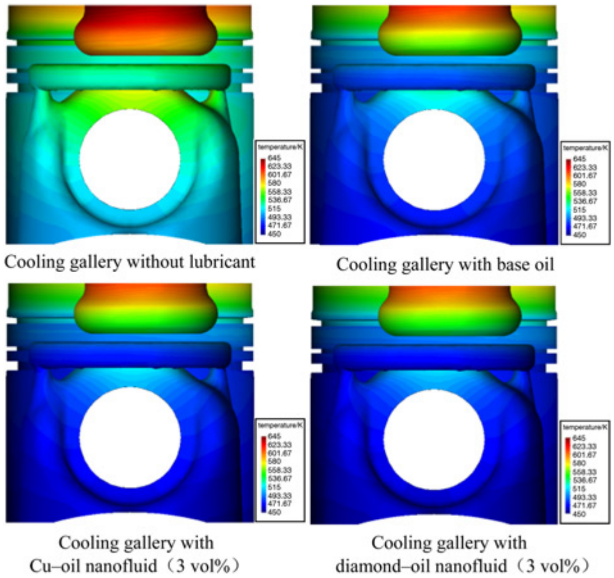

Figure 4 *Stable temperature distribution of the piston with different cooling media*

changed uniformly from the piston top surface to the bottom, with the maximum temperature occurring at the piston top surface. To evaluate the thermal conditions of the piston, we should pay attention to the maximum temperature. Without a lubricant in the cooling gallery, the maximum temperature was ∼644.07 K. In the case of aluminium pistons, long-term operation at high temperatures can cause local degradation of material properties, resulting in severe piston failure. The most effective cooling is achieved by using lubricants in the cooling gallery because a significant portion of the piston surface is in direct contact with the lubricant, and short heat flow paths from temperature severe piston zones to the gallery surface are characteristic. By using base oil, 3 vol% Cu–oil nanofluid and 3

Table 6 Temperature results of tests and simulations

Locations	1	2	3	4	5	6	7	8	9	10	11	12	13
base oil test	564	552	557	554	553	617	623	621	617	554	560	557	551
base oil simulation	561.3	549.6	552.5	552.8	549.8	614.9	617.9	616.4	614.5	544.4	560.2	553.9	544.3
error, %	0.48	0.44	0.81	0.22	0.58	0.34	0.83	0.75	0.41	1.76	−0.04	0.56	1.23
Diamond, 1 vol% test	553	548	551	549	550	611	617	615	610	545	554	551	540
Diamond, 1 vol% simulation	550.2	545.4	546.3	544.2	546.7	608.5	612.1	610.4	607.8	537.8	553.9	547.5	532.6
error, %	0.51	0.48	0.86	0.88	0.60	0.41	0.80	0.75	0.36	1.34	0.02	0.64	1.39

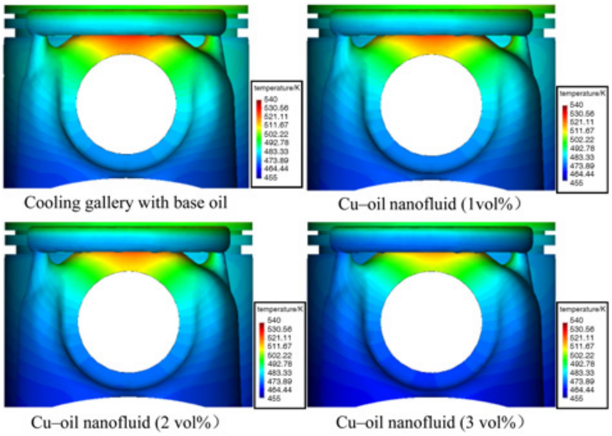

Figure 5 *Stable temperature distribution of the piston with Cu–oil nanofluid*

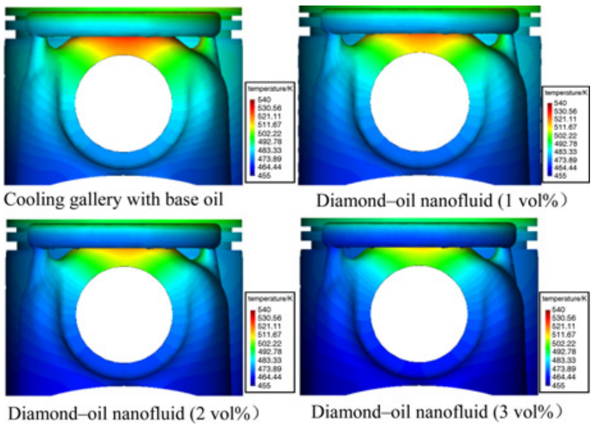

Figure 6 *Stable temperature distribution of the piston with diamond–oil nanofluid*

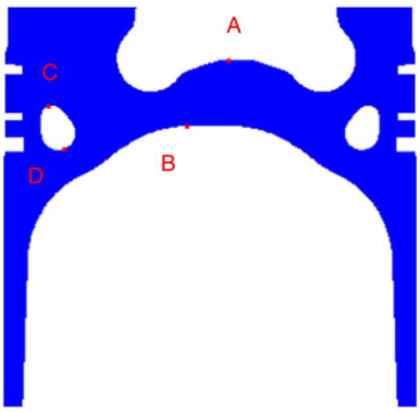

Figure 7 *Key nodes of stable piston temperature*

vol% diamond–oil nanofluid in the piston cooling gallery, the maximum temperatures were 623.95, 611.93 and 609.3 K, respectively. The maximum temperature decreased with increasing distance to the cooling gallery. Visual comparison is difficult because of the small changes of the maximum value. For better visual comparison, partial views of the model are shown in Figs. 5 and 6; the legends use the same upper and lower limits. The Figures show the stable temperature distribution of the piston using nanofluids with volume fractions of 1, 2 and 3% as the cooling media. They show clear temperature differences as a result of using nanofluids with different volume fractions. It is thus possible to significantly decrease the piston temperature by the use of nanofluids in the cooling gallery.

Some specific nodes in the piston are selected for comparison. Fig. 7 shows the node locations and Table 7 shows the simulation results. At the centre of the top surface of the piston, the centre of the bottom surface of the piston, the top surface of the gallery and the bottom surface of the gallery, the maximum temperature was reduced by ~40°C using base oil as the cooling medium in the piston gallery. These results show that the piston thermal load can be significantly decreased in the existence of an optimally designed cooling gallery. When the cooling medium was replaced with nanofluids, the temperatures were further reduced. Moreover, enhancement of the cooling effect increases with an increasing nanoparticle volume fraction in the nanofluids, as shown in Table 7. The maximum temperature was reduced by 53°C using the 3 vol% diamond–oil nanofluid, thus effectively improving the working condition of the piston. The thermal load of the temperature-severe zone decreased and the thermal stress also decreased, thus extending the service life of the piston. Suspended nanoparticles in base fluids can alter the fluid flow and heat transfer characteristics of the base fluids. Application of nanofluids as the cooling medium improves the heat transfer coefficients on all direct-contact surfaces, and the enhancement of thermal conductivity plays an effective role in the cooling process.

7. Conclusion: A model for the cooling gallery of the piston from a diesel engine was developed using the finite element analysis method and this model was validated by experiments. The numerical investigation of heat transfer conditions in the cooling gallery of a piston produced quantitative results of piston temperature. The heat transfer coefficients and physical properties of traditional engine oil were significantly improved by the addition of Cu/diamond nanoparticles. Two types of nanofluids, Cu–oil nanofluids and diamond–oil nanofluids with volume fractions of 1, 2 and 3% effectively reduced the piston thermal loading when used as the heat transfer medium for the piston cooling gallery. This numerical investigation confirmed that effective reduction of piston temperature can be achieved by using nanofluids as the cooling medium.

8. Acknowledgments: This research was supported by the National Natural Science Foundation of China (grant nos. 51276031, 51476019 and 51376002). The authors acknowledge the support of the DEUTZ (Dalian) Engine Co. for the bench tests reported in this Letter.

Table 7 Stable temperature of piston with different cooling media in gallery (K)

Locations	Without lubricant	Base oil	Cu, 1 vol%	Cu, 2 vol%	Cu, 3 vol%	Diamond, 1 vol%	Diamond, 2 vol%	Diamond, 3 vol%
A	633.2	592.9	590.5	588.3	584.2	588.7	584.4	580.2
B	607.2	564.6	562.0	559.7	555.3	560.1	555.5	551.1
C	545.8	504.9	502.4	500.1	495.9	500.5	496.0	491.7
D	540.8	488.8	485.7	482.9	477.6	483.4	477.8	472.6

doi: 10.1049/mnl.2014.0432

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