

Thermal Management of Air-cooled Motorcycle Engines Using Forced Oil-cooling System

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ABSTRACT

Engine cooling substantially affects power output, fuel economy, noise reduction and durability. Estimating each engine part's temperature in the early stages of design is important in the development of air-cooled motorcycle engines. In this study, engine temperature distributions were obtained through heat transfer coefficient calculations based on Computational Fluid Dynamics (CFD). From the analysis, it was found that by using air-cooling with a forced oil-cooling system, the cylinder head temperature sufficiently decreased, and the resultant temperature distribution was almost the same as that of water-cooled engines. It was also found that the temperature gradient among cylinders was manipulated by controlling the cooling-oil flow. Furthermore, heat loads were controlled as well, and the knocking limit was also enhanced when the engine was tested in a prototype motorcycle.

1. Introduction

Although customer needs with regard to motorcycles have diversified in recent years, motorcycles equipped with simple air-cooled engines are still popular. There is also high demand in the markets of developing countries for air-cooled engines, which have a comparatively lower number of parts and therefore lower costs.

An important issue with air-cooled engines is that the cooling depends on the air-flow generated by the moving motorcycle. Determining the total heat dissipation and correcting the temperature difference among cylinders in multi-cylinder engines are important factors that contribute to cooling performance. To pursue optimal engine cooling performance also means to pursue functional beauty in an air-cooled engine, which is also important from a design perspective. Conventionally, the development of air-cooled engines was based on the results of visualizing the cooling-air flow in wind tunnel tests and measuring the temperature in actual tests. Quantitative analysis of flow behavior and temperature forecasts at an early stage of engine design was considered too challenging. Controlling exhaust emissions and enhancing fuel economy have continued to become more important even for motorcycles, and controlling the engine temperature to

help achieve these objectives has become an important issue.

In this study, CFD analysis of the cooling-air behavior was performed to determine the cooling performance for an air-cooled engine. Some useful knowledge such as effective cooling-fin shapes and optimal positions for the oil cooler and air inlets was discovered⁽¹⁾. Secondly, the effectiveness of the oil-cooling system was studied to additionally enhance the cooling performance of air-cooled engines. To verify the effectiveness of the oil-cooling system, CFD analysis was used to obtain the local heat transfer coefficient for the cooling-air and the average heat transfer coefficient for the cooling-oil. These values were then used to calculate engine heat conduction and obtain a temperature distribution for each engine part. Note that the partial cell Cartesian Coordinate method^{(2), (3)} was used for the CFD analysis in this study. This method can be used in flow fields of various shapes and doesn't require many man-hours for mesh generation.

This paper describes the results of a theoretical study of an in-line four-cylinder, air/oil-cooled engine with a displacement of 900 cm³. These theoretical results were then utilized on a prototype motorcycle to test the effectiveness of using the forced oil-cooling system.

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2. Engine Outline

2.1. Concept of Air/Oil-cooled Engines

In the design concept stage of the new air-cooled engine, the development goal was to achieve the temperature distribution comparable to that of water-cooled engines, in order to meet the objectives for power output, fuel economy, noise reduction, exhaust emissions, durability and reliability.

The targets related to cooling and the technical methods (concepts) for achieving them are as follows:

- (1) To study cooling-fin shapes that increase the cooling-air flow rates and produce more effective cooling.
- (2) To determine an oil cooler position and internal structure that help achieve good cooling performance.
- (3) To reduce heat transmission and increase heat dissipation by shortening the length of the exhaust ports and modifying the thickness of the port walls, in order to reduce the temperature around the cylinder head combustion chamber which is the heat source.
- (4) To increase heat dissipation by providing a special oil-cooling system oil line to counteract temperature rises due to heat transmission around the combustion chamber. Also, to provide an orifice in the cooling system oil line and adjust the amount of oil supplied to each cylinder oil jacket in order to correct the temperature difference among the cylinders.
- (5) To add an oil temperature control function with oil temperature sensor information to the FI (Fuel Ignition) control, in order to adjust to changes in the riding conditions and environment.

Cooling-air flow analyses have been performed in the past for items (1) to (3) above, focusing on the cooling-fin shape of the air-cooled engine and the oil cooler position. Heat conduction was calculated for the entire engine to additionally develop these items and to study item (4), which enabled the temperature forecasts on each part theoretically. In particular, for multi-cylinder engines, it is important not only to reduce the maximum temperatures but to also correct the temperature difference among cylinders, in order to increase durability and reliability and stabilize quality. Consequently, fluid and heat analysis was used to quantitatively calculate the optimal amount of oil flow.

For item (5), the thermostat and temperature sensor layout positions were studied to determine an efficient temperature control method.

Note that although studies^{(4), (5)} on oil-cooling systems for engine cooling have already been reported, not many studies have been conducted using quantitative or visualized data.

In line with the study described above, it was decided to design and produce a new engine. It was also decided to combine both CFD analyses and actual tests to verify the accuracy and validity of the calculation results.

2.2. Forced Oil-cooling System

Figure 1 (a) shows the oil line systems within the structure of the forced oil-cooling system.

The green and red areas in the figure are the oil lines for lubrication and cooling respectively. It is a forced oil-cooling structure, where oil supplied from an oil pump with two rotors is divided into two oil lines. One oil line is for lubrication and the other is exclusively for cooling. A temperature sensor for the FI system was provided in the cooling system oil line to additionally enhance fuel economy and comply with emission regulations. A thermostat was positioned in front of the oil cooler to enhance the warming-up performance.

As a cooling medium, oil has a lower specific heat than cooling water, so the cooling performance of the oil jacket shape with the same amount of flow is lower than that of the water-cooling system. The oil viscosity coefficient changes according to the oil temperature. This characteristic is utilized in the system and shapes were

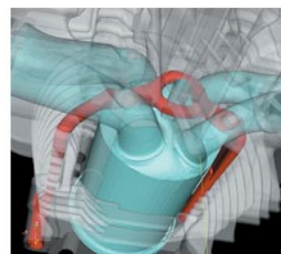
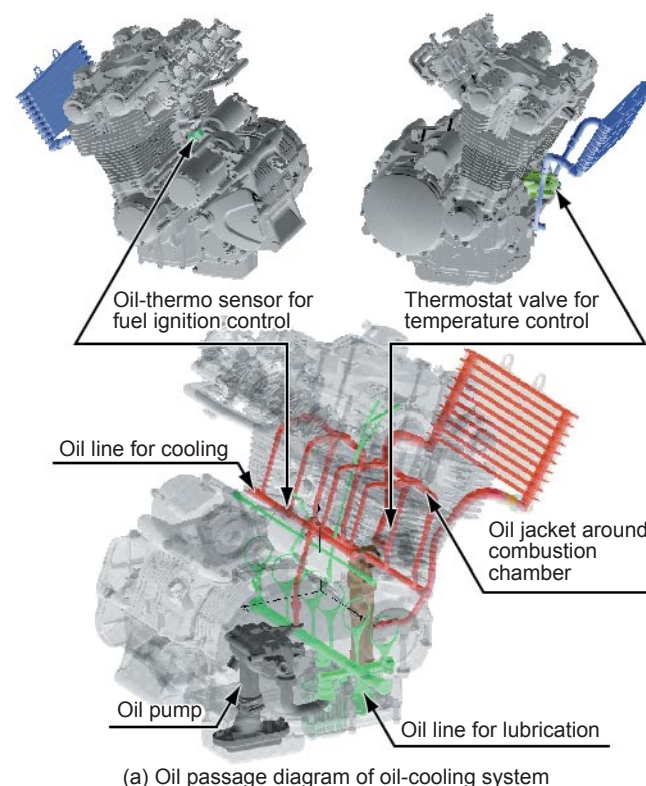


Fig. 1 Outline of engine oil-cooling system

studied to intensify the flow rate of the cooling-oil to increase the heat transfer coefficient. In order to achieve efficient cooling around the combustion chamber which is a heat source, the oil jacket was positioned near the cylinder head combustion chamber. In the case of the forced oil-cooling system, stagnations of oil-flow generate localized oil temperature spikes, resulting in the degradation of the oil. Another concern is when the surface area inside the oil line is too large, the heat receiving area will increase and the cooling-oil temperature will rise too much. Specifically, the shape that was used in the oil lines is different from the water jacket shape that covers the entire combustion chamber, which is similar to the design used in water-cooled engines. A line was provided between the respective intake and exhaust ports as shown in Fig. 1 (b) and items of concern in production (such as making the jacket core and casting of cylinder head) were also taken into account, to produce an oil jacket shape with less stagnation and the optimal heat receiving surface area.

3. Calculation Outline

3.1. CFD Analysis Properties

In the case of designing a new air-cooled engine, CFD analysis is effective as an examination item in an initial stage of design.

To effectively utilize the air flow generated by the moving motorcycle, the following properties were analyzed by the CFD: cooling-air flow, optimal shape for the cooling-fin, efficient position for the oil cooler, etc. In a wind-tunnel test, only visualization of air-flow is available. On the other hand, with the CFD analysis, it was possible to study the flow field under actual riding conditions before making the prototype, quantitatively study the cooling performance, and calculate the heat transfer coefficients for each engine area. By developing these results, it was also possible to calculate the temperature distribution of the engine and to link it to heat strain analysis.

In addition, when using an oil-cooling system, the study of the oil jacket shape, flow method of cooling-oil, optimization of the amount of cooling-oil recirculation, as well as the study of the exhaust port shape in order to minimize the heat source, was treated as analysis properties using CFD.

3.2. Calculation Procedure

Figure 2 shows the calculation procedure. With a numerical wind tunnel, shape data is created using 3D-CAD systems (Fig. 2 (a)) such as CATIA, and then written into a file in STL (Stereo Lithography) data format. Commercially-available software⁽⁶⁾ was then used to correct STL data and perform parts conversion. Figure 2 (b) shows some of the STL data after conversion.

To analyze the flow in a way equivalent to actual

riding conditions, a solid model of the completed motorcycle was placed in a numerical wind tunnel as shown in Fig. 2 (c), the Cartesian Coordinates (Fig. 2 (d)) were set, and the boundary conditions were created. There are approximately 2,370,000 cells of 133 x 128 x 139 in the calculation grid. After the calculation, the local heat transfer coefficient (Fig. 2 (e)) was obtained from the equation⁽⁷⁾ derived from Karman's analogy. The CFD analysis of the cooling-oil flow was also performed on the oil jacket.

The heat transfer coefficient obtained above and the heat flux in areas such as the combustion chamber and exhaust port were used to calculate engine heat conduction and obtain the engine temperature distribution.

Here, the heat conduction was calculated by developing the program of the Partial cell Cartesian Coordinate method in the CFD calculation. The same information is used for the Cartesian Coordinates as is used for the cooling-air flow CFD calculation, so interpolation calculations between coordinates for the heat transfer coefficient are not necessary.

3.3. Boundary Conditions

For the boundary conditions in the CFD calculation of cooling-air behavior analysis, a flow rate equivalent to that

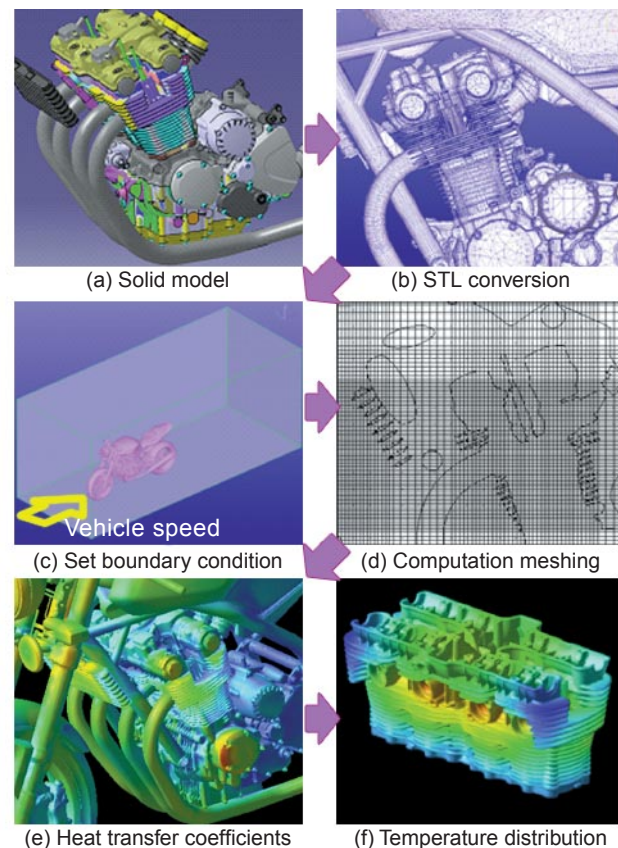


Fig. 2 Procedure of air-cooled engine analysis using CFD code

of motorcycle speed was applied to the entrance of the numerical wind tunnel as shown in Fig. 2 (c), with a turbulence intensity of 5% and length scale of 0.05m, and taking the ground and side walls as fixed walls.

For the boundary conditions in the calculation of the engine heat conduction, the local heat transfer coefficient for cooling-air obtained from the CFD calculation and the average heat transfer coefficient for the oil jacket were used. However, in addition to these, it was also necessary to take the following items into account even though these values were not very clear; cooling due to lubricating oil and the heat flux around the combustion chamber, in the cylinder walls and the intake and exhaust port inner walls⁽⁸⁾.

The objective of this study was to examine the cooling performance of air/oil-cooled engines, and in particular to study the effect of the oil jacket. Measured data from a test engine was substituted for the temperature at the bottom surface of the simulated cylinder barrel (upper crankcase deck surface), and this value was adjusted to approximately match the measurement points of the prototype engine.

The calculation grid in the heat conduction calculation was made uniform with CFD calculation of the cooling-air flow, so the grid of the heat conductive calculation is relatively rough. On the other hand, the minimum thickness of the cylinder head is thinner than the grid, so as Fig. 3 shows, in some areas it becomes necessary to determine two heat transfer coefficient settings for cooling-air and cooling-oil inside one cell. Furthermore, in some areas, boundary conditions are necessary for heat fluxes that make contact with the combustion chamber. To enable calculations even in cases such as those described above, the shape data for the calculation of heat conduction (STL data) is divided into groups by setting boundary conditions, and heat transfer coefficients, heat fluxes, and actual measured temperatures for each calculation cell to enable the setting of multiple boundary conditions.

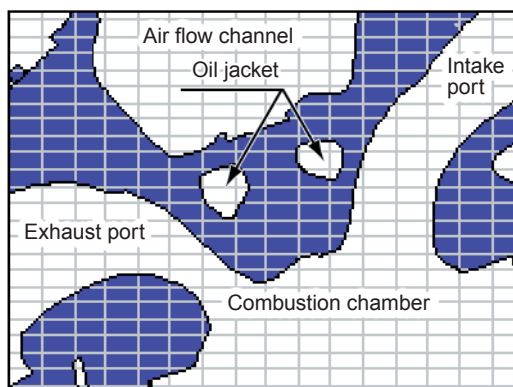


Fig. 3 Computation grids for heat conductive calculation

4. Calculation Results and Discussion

4.1. Heat Transfer Coefficient of Oil Jacket

The heat transfer coefficient of the cooling-oil (α : W/m^2K) was calculated in order to calculate the amount of heat received by the oil jacket. Figure 4 shows an example of these calculation results.

By decreasing the line cross section area between intake and exhaust ports, the flow rate rises and the heat transfer coefficient increases. On the other hand, the heat receiving area does not change much when the cross section shape is varied, so the cooling efficiency is enhanced. The wide area on the exhaust port side is a stand (base) that enables the oil jacket core to be stood up when casting the cylinder head. The optimal stand from a variety of shapes was selected using the CFD, to make sure it did not generate stagnation of cooling-oil.

In contrast to cooling-water, the viscosity of oil varies according to the oil temperature. Since the heat transfer coefficient varies according to the oil temperature, the relationship between the oil flow, oil temperature, and the heat transfer coefficient was obtained with CFD. However,

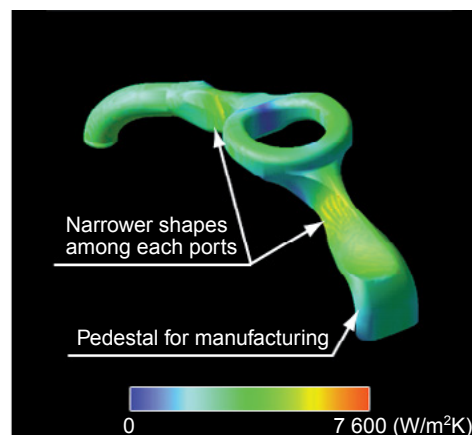


Fig. 4 Heat transfer coefficients of oil jacket

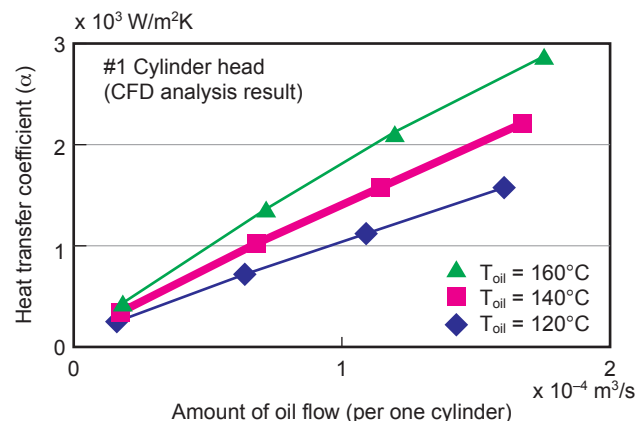


Fig. 5 Relationship between oil flow amount and heat transfer coefficients

the average heat transfer coefficient for the oil jacket was treated as the average values of only the oil jacket in the cylinder head area. Figure 5 shows these results. From these results it can be understood that the heat transfer coefficient is 70 to 80% higher when the oil temperature is 160 °C than when it is 120 °C.

With the water-cooling system, when the water temperature rises, the difference in temperature (ΔT : °C) between the coolant and the coolant line inner wall surface becomes smaller, and so the amount of heat dissipation drops.

However, with an oil-cooling system, when the oil temperature rises, α increase is larger than the reduction due to ΔT , so sometimes the amount of heat dissipation rises when the oil temperature is high. For example, if it is assumed that the oil jacket inner wall surface temperature is 250 °C, when the oil temperature rises from 120 °C to 160 °C, ΔT changes by 0.7 from 130 °C to 90 °C, while α changes by 1.7 and the heat transfer coefficient ($\alpha\Delta T$) changes by 1.19.

Consequently, while the water jacket in a water-cooling system covers the entire combustion chamber, for the oil jacket shape it is desirable to reduce the heat receiving area and design a simple shape that increases the oil flow rate.

4.2. Comparison with Surface Temperature Measurement

Figure 6 shows a comparison of the engine heat calculation results and the surface temperature measurement results using an infrared thermometer (thermoviewer) in a bench test. For the calculation and measurement conditions, the air-flow velocity and engine load were equivalent to 80 km/h.

In the calculation, α was obtained using the equation derived from Karman's analogy, and the heat was calculated based on the assumption that the cooling-air temperature is 25 °C. In the calculation results, the temperature of the cooling-fin on the exhaust port side is lower than the cooling-fin on the intake port side, but in the image shown by the thermoviewer, the cooling-fin temperatures on the exhaust port side and the intake port side are approximately the same. This is probably because the air temperature around the exhaust pipe is high in actual engines, and so the temperature difference is lower than in the calculation.

In summary, while there is still room to modify a number of items, it can be said that approximately appropriate calculation results were obtained for the engine temperature distribution.

4.3. Effect of the Oil-cooling System

In order to study the effectiveness of oil-cooling, heat conduction was calculated with and without oil-cooling. The calculation conditions were set as an engine load equivalent to 230 km/h with no oil-cooling, and so the

heat transfer coefficient for the oil jacket was set as zero. Figure 7 shows the temperature distribution of these calculation results.

From the figure it can be understood that the ends of the cooling –fins are cool but that the temperature is high on exhaust port inner walls in the intermediate cylinders (No.2, No. 3 cylinders) and that the cooling in the central area of the engine is insufficient. Part (a) in Fig. 8 shows the temperature (T_{plug} : °C) around the cylinder head ignition plug seat in the calculation. It is estimated that the difference of the T_{plug} temperature between cylinders is approximately 40 °C, and that the maximum temperature of T_{plug} for the intermediate cylinders is close to 300 °C.

Next, to verify the effectiveness of oil-cooling, the heat transfer coefficient of the oil jacket was recalculated with the equivalent cooling-oil amount for each cylinder of $1 \times 10^{-4} \text{ m}^3/\text{s}$ (6.0 l/min) and an oil temperature of 160 °C. Figure 9 shows these results. The temperature over the entire engine is lower, so it can be interpreted that oil-cooling is effective. As shown by (b) in Fig. 8, it is estimated that there is a temperature reduction of approximately 70 °C for T_{plug} with the intermediate cylinders having the highest temperature. Then, the calculation and measured values were compared for

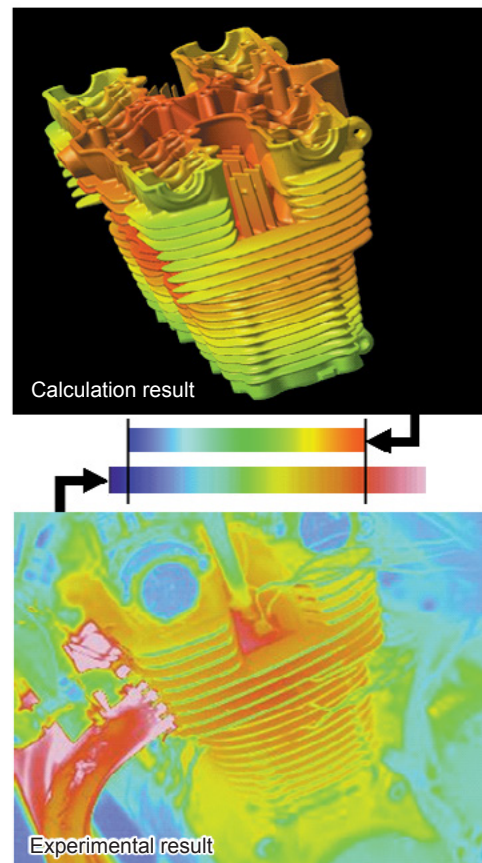


Fig. 6 Comparison of temperature (at 80 km/h)

verification. Part (c) in Fig. 8 shows the measured values.

Although the precision of the calculation values is insufficient, the temperature difference among the cylinders (difference in T_{plug} between each outer side (Nos. 1 and 4) and intermediate (Nos. 2 and 3) cylinders) is approximately the same for the calculated values and the measured values, and so experimentation using calculations is possible in order to correct the temperature difference among the cylinders.

When looking closely at Fig. 9 (a), first it can be seen that the corner areas of the cylinder head cooling-fin on the exhaust port side are well cooled. In contrast, the overall temperature around the combustion chambers in the central area of the engine is high. In particular, the temperature of the inner wall around the exhaust port is high. This is probably because the air flow generated by the moving motorcycle is blocked by the front tire and front suspension so that the cooling-air on the front center of the engine is insufficient.

Figure 9 (b) shows temperature distribution around the cylinder head combustion chamber. It can be understood that the temperature of the combustion chamber wall is higher on the exhaust port side than that

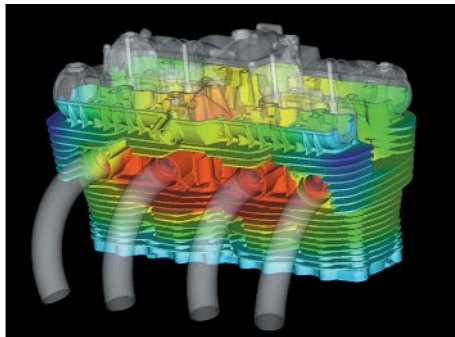


Fig. 7 Temperature distribution without oil-cooling

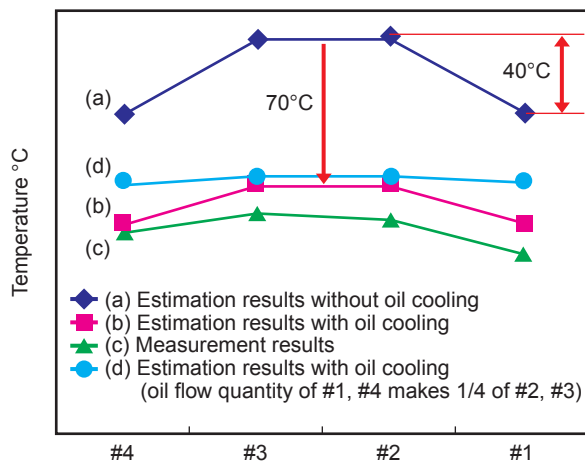
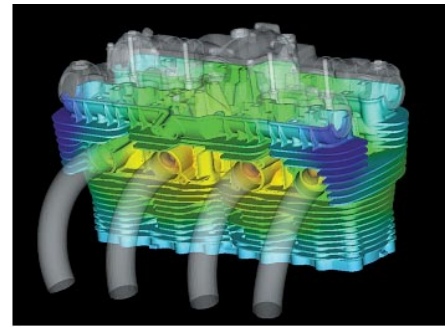
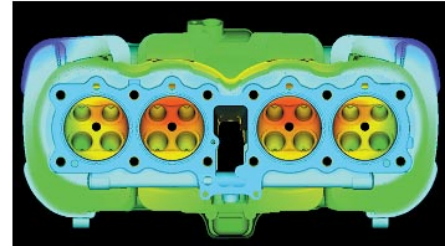


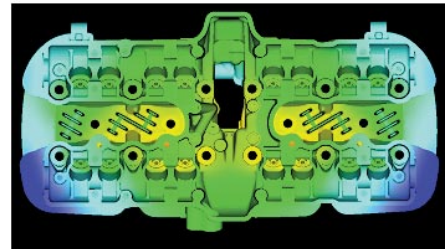
Fig. 8 Calculation of cylinder head temperature (at 230 km/h)



(a) Oblique perspective view

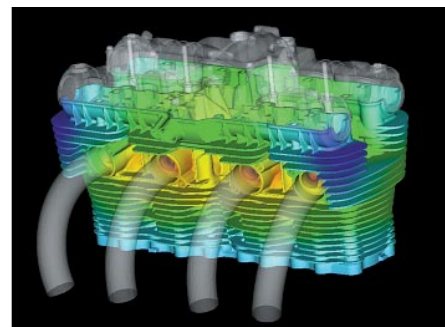


(b) Bottom view

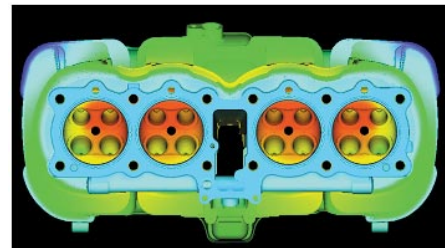


(b) Top view

Fig. 9 Temperature distribution with oil cooling ($T_{\text{oil}} = 160^{\circ}\text{C}$)



(a) Oblique perspective view



(b) Bottom view

Fig. 10 Temperature distribution of improving uneven temperature among cylinders ($T_{\text{oil}} = 160^{\circ}\text{C}$)

of the intake port, and that the combustion chamber temperature is particularly high between the two branching exhaust ports. When comparing the cylinders, it can be understood that the outer side cylinders are well cooled.

4.4. Correction of the Temperature Difference among Cylinders

In order to study the correction of the temperature difference among cylinders, calculations were made with many different amounts of cooling-oil for each cylinder. The calculation results shown in Figs. 10 and 8 (d) were obtained by leaving the amount of cooling-oil flow per cylinder to the intermediate cylinders (cylinder Nos. 2 and 3) unchanged and controlling the amount of cooling-oil flow to the outer side cylinders (cylinder Nos. 1 and 4) with an orifice to a flow rate of approximately 1/4 ($0.25 \times 10^{-4} \text{ m}^3/\text{s}$). As such, it was understood that the temperature difference among cylinders could be corrected.

5. Confirmation of Oil-cooling Effectiveness with an Actual Riding Test

5.1. Decrease of Cylinder Head Temperature around Ignition-Plug Seat

The cylinder head temperature around the ignition-plug seat (T_{plug}) was used as an indicator to evaluate the cooling performance of the engine.

Figure 11 shows a graph of the relationship between T_{plug} and the amount of cooling-oil per cylinder supplied to the oil jacket (for each cylinder), with the data obtained from the test. The results showed that when the engine speed was 7 000 rpm, the amount of cooling-oil that should be supplied to each cylinder is approximately $1 \times 10^{-4} \text{ m}^3/\text{s}$. As a result, the discharging volume of oil pump could be minimized from viewpoint of decreasing friction.

Figures 12 (a) and (b) show the temperature reduction effect in an actual riding test when using this oil-cooling system. Figure 12 (a) shows the relationship between T_{plug} and the indicated mean effective pressure (IMEP: MPa). The points indicated with round marks in the figure are the temperatures of representative mass-production models and the points indicated with square marks are the temperatures for this engine with and without oil-cooling. The overall temperature of the combustion chamber inner walls in water-cooled engines is lower than that in air-cooled engine, and consequently T_{plug} is lower. This is because the combustion chamber area in water-cooled engines is covered by cooling medium (coolant liquid) with good heat exchange properties. On the other hand, air-cooled engines do not have a cooling medium and have to depend on the heat mass, so there is a tendency for T_{plug} to sharply rise in proportion to the IMEP.

In the case of the air/oil-cooled engine that was

produced in this study, for the same IMEP, a temperature reduction effect of approximately 70°C is obtained as shown in Fig. 12 (a). The data without oil-cooling in the figure are the results with the same engine when only air-cooling was performed, without circulating the cooling-oil.

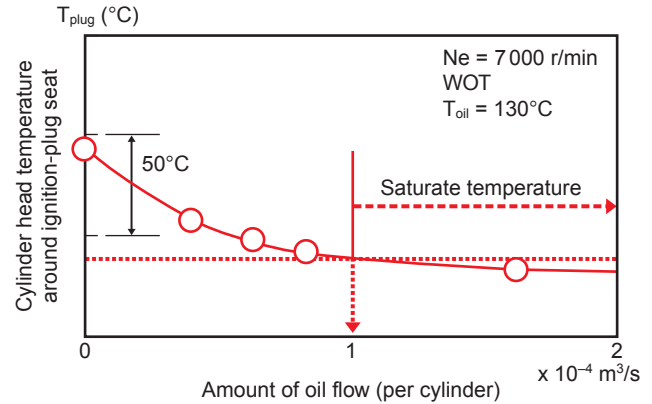
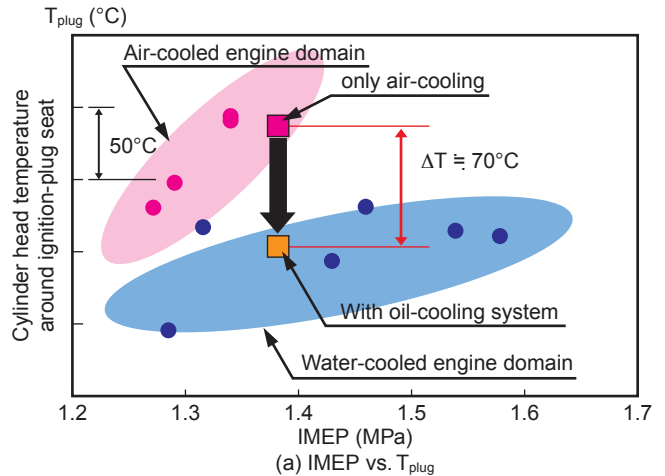
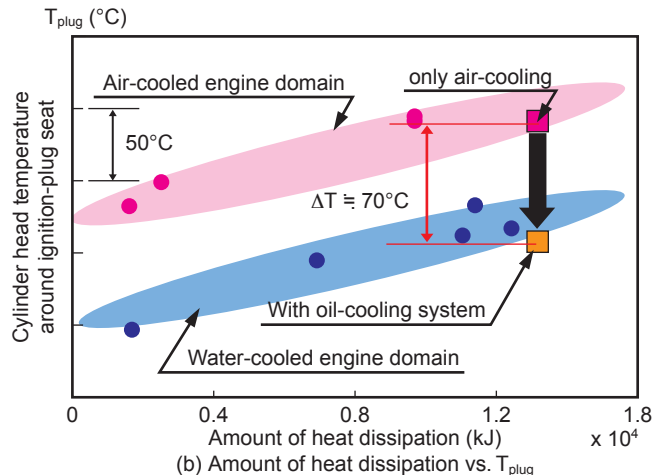


Fig. 11 Optimization of oil flow amount (per cylinder)



(a) IMEP vs. T_{plug}



(b) Amount of heat dissipation vs. T_{plug}

Fig. 12 Effect of oil-cooling

Figure 12 (b) shows the relationship between T_{plug} and the amount of heat dissipation (Q_t) (kJ) after correction of Fig. 12 (a). In the same way as Fig. 12 (a), it shows a temperature reduction effect of about 70 °C when compared with engines that have approximately the same amount of engine heat generation. Note, however, that in Fig. 12 (b) temperatures corresponding to representative mass-production models were used for each point. Q_t (kJ) is the amount of heat dissipation and was calculated from equations (1)⁽⁸⁾.

$$Q_t = Q_{t1} + Q_{t2} + Q_{t3} \quad (1)$$

Here, Q_{t1} (kJ) is the heat dissipation conducted from around the combustion chamber, Q_{t2} (kJ) is the heat dissipation radiated from around the combustion chamber, and Q_{t3} (kJ) is the heat dissipation from the friction heat in the reciprocating part of the piston.

Figure 13 shows the relationship between the IMEP and compression ratio (ϵ) by motorcycle category for representative Honda mass-production models. It can be understood from the figure that when comparing engines with the same output (same IMEP), it is possible to increase the compression ratio of an air-cooled engine to

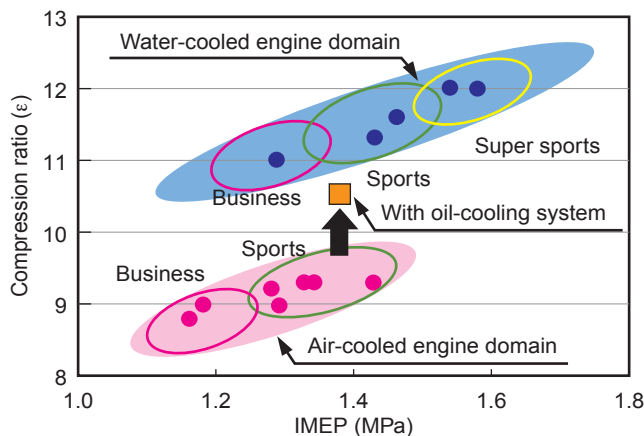


Fig. 13 Effect of oil-cooling for compression ratio

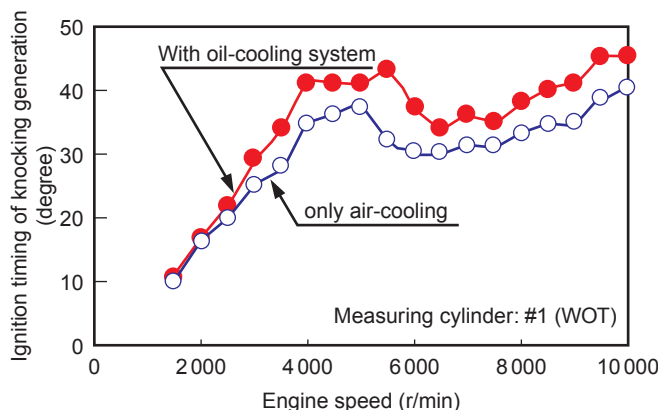


Fig. 14 Effect of oil-cooling for engine knocking limit

near that of a water-cooled engine for sports category of motorcycles produced in this study.

5.2. Increasing the Knocking Limits

Figure 14 shows the effect of increasing the knocking limits when using this forced oil-cooling system in an air-cooled engine.

In all engine speed ranges of 2 000 rpm or higher, an enhancement of about +5° or more was seen in the knocking generation ignition timing. It was assumed that the reason for this was because the temperature of the unburned mixed gas was reduced by lowering the wall surface temperature around the combustion chamber and making it more uniform.

In addition, from the results of the durability test, it was confirmed that the valve seat deformation and deterioration and blow-by gas discharge were corrected to levels comparable to those in a water-cooled engine.

6. Conclusion

- (1) The use of the forced oil-cooling system in an air-cooled engine has the potential to match the power output, environmental performance, durability and reliability of a water-cooled engine.
- (2) It was understood that the temperature difference among cylinders could be corrected by controlling the amount of cooling-oil.
- (3) It was confirmed with an actual test that, with the combined use of the forced oil-cooling system, temperature distribution comparable to that of a water-cooled engine was available even with an air-cooled engine.
- (4) It was also confirmed that the forced oil-cooling was effective in other areas, such as enhancing the knocking limits.

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