

# Heat Radiation Analysis Based on Internal Flows of Radiator

Takashi WATANABE\*

Masayuki IWATA\*

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## ABSTRACT

Rectangular-shaped radiators for motorcycles have been used previously in many cases. With the latest high-powered engines, the use of any shaped radiators is increasing to get high heat radiation within a limited space. This type of radiator has different cooling tube lengths. It is more difficult to predict the heat radiation amount for this type as compared to rectangular types, as velocity distribution is uneven. Moreover, a sufficient analysis regarding heat radiation characteristics has not been done and dependency of flow amount on heat radiation has been unclear. By combining calculations of a three-dimensional flow field and one-dimensional heat radiation, a method has been constructed that can predict the heat radiation amount for any shaped radiators easily and precisely.

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## 1. Introduction

The ordinary construction of motorcycle radiators contain a large number of flattened cross section cooling tubes running vertically within a rectangular radiator frame, with all the tubes of the same length. Recently, it has become necessary to design radiators capable of large amounts of heat radiation, as engines are producing increasing power. Meanwhile, more and more radiators are now designed with any shapes due to layout space restrictions. Many such any shaped radiators show considerable deviations from their predicted heat radiation characteristics when these are calculated with the methods developed for rectangular radiators.

There have been limited analysis of the heat radiation value from radiators, mainly because most of the radiators have been rectangular, lending themselves to empirical formulas developed from previous models. With the increasing need to use any shaped radiators, however, there has come an increasing need to develop new analytical techniques for quantitative heat radiation characteristics of radiators and for accurate predictions of heat radiation

performance in radiators of any shapes. Oshima reported on one method for predicting the radiator heat radiation value<sup>(1)</sup>, but there have been few discussions of specific methods for the influence of coolant flow rate or the characteristics of heat transfer from the cooling tubes to the air. Furthermore, there has not necessarily been enough research into the influence of flow rate and radiator specifications on heat radiation characteristics even for the rectangular radiators conventionally used.

The authors have developed a system capable of predicting the heat radiation characteristics of a radiator of any shapes quickly and with high accuracy, by combining a three-dimensional flow field calculation with one-dimensional heat transfer calculation.

## 2. Investigation of Analysis Methods

The basis of the cooling performed by a radiator is typically a forced convection heat transfer. Analysis can be divided into treatment of the flow field and treatment of the heat transfer, i.e., first, the flow field alone is analyzed, and

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\* Asaka R&D Center

then, those results are used to examine the heat transfer. Actual analyses are carried out in one of the following three methods.

Method A: The flow of radiator coolant in the tubes is considered as a one-dimensional flow with a uniform profile across the cross section. The flow field and heat transfer analyses are both done in one dimension.

Method B: Three-dimensional analyses are performed for both the flow field and the heat transfer.

Method C: The flow field is solved in three dimensions and after that, the heat analysis is done in one dimension.

Since the fundamental theory of Method A is simplified, it is easy to write in-house code; since the calculation time is short, it is easy to observe the effects of many parameters. On the other hand, the speed distribution of the coolant flow in the tubes is strongly influenced by the geometry of the radiator tank and other components, so this has the drawback of limited accuracy because of the one-dimensional assumption for speed distribution.

Heat transfer analysis in Method B is based on the speed distribution with full consideration of system geometry. Figure 1 shows the predicted temperature distribution on the radiator surface, according to this method. Figure 2 is the image of the actual radiator from a thermo-viewer. The results of temperature distribution in these two analyses match qualitatively. However, the calculation time for a single case is long in this method, and extremely long if the coolant flow rate is a parameter of the calculation.

In Method C, a three-dimensional analysis is carried out only for one case in any single shape for the time-consuming flow field calculation; all heat transfer parameters are analyzed in one dimension. This method offers the advantages of speedy results and practical accuracy levels when flow rate is a parameter. Procedure C was used in this study due to these advantages. A comparison of all three methods is shown in Table 1.

### 3. Formulation of One-Dimensional Analysis of Cooling Tube

#### 3.1. Modeling of Cooling Phenomena

The first task in the analysis of heat transfer in cooling tubes is the creation of the model. Figure 3 shows the state of heat flow in a tube used in a radiator. As water passes through the tube, it loses heat through the tube walls and its temperature drops. In order to analyze heat exchange, the overall heat resistance  $R$  between the heat exchange medium (water), and the destination of the heat flow, (air), must be found. This value is the sum of the three resistances given below<sup>(2), (3)</sup>.

$$R = 1/H = 1/hw + t/\lambda + 1/ha \quad (1)$$

$R$  : Overall heat resistance

$H$  : Overall heat-transfer coefficient

$hw$  : Coefficient of heat transfer from water to cooling tube wall

$ha$  : Coefficient of heat transfer from cooling tube wall to air

$t$  : Cooling tube thickness

$\lambda$  : Thermal conductivity of cooling tube

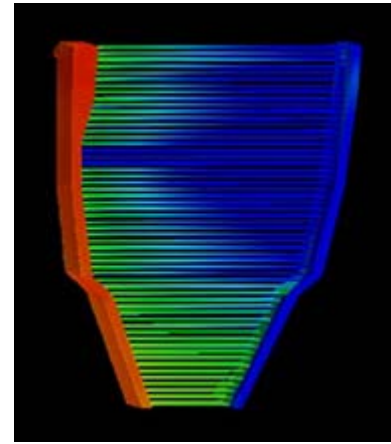


Fig. 1 Temperature distribution by simulation



Fig. 2 Temperature distribution by thermo-viewer

Table 1 Comparison among three methods for heat radiation of radiators

| Methods | Flow field calculation | Heat radiation calculation | Calculation time | Calculation accuracy |
|---------|------------------------|----------------------------|------------------|----------------------|
| A       | One dimension          | One dimension              | Acceptable       | Unacceptable         |
| B       | Three dimension        | Three dimension            | Unacceptable     | Acceptable           |
| C       | Three dimension        | One dimension              | Acceptable       | Acceptable           |

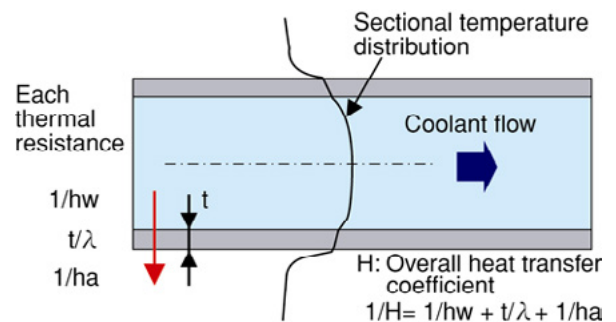


Fig. 3 Cooling model of radiator tube

Both  $h_w$  and  $h_a$  represent heat transfer coefficients between fluids and solids, but they are deeply influenced by the flow conditions at the wall. In other words, they are influenced not only by the flow velocity but also by whether the flow is laminar or turbulent. Therefore, it is necessary to pre-determine, in the usual flow zone used for practical uses, whether the flow is laminar or turbulent in order to select the experimental formula and the theoretical values for heat transfer. The formulas for Reynolds' number and coefficient of heat conduction given below were borrowed from References (2) and (3).

(1) Water flow in cooling tube

$$Re_d = ud / \nu \quad (2)$$

$$d = 4A / s \quad (3)$$

$Re_d$  : Reynolds number

$u$  : Flow velocity

$\nu$  : Kinematic viscosity

$d$  : Hydraulic diameter

$A$  : Cross sectional area of tube

$s$  : Cross section circumferential length of tube

When  $Re_d < 2300$ , flow is laminar, and the Nusselt number  $Nu_d$  is determined as follows:

$$Nu_d = 3.66 + 0.063 (Gz + 0.039Gz^{0.33}) \quad (4)$$

$$Gz = (x / d) / (Re_d Pr) \quad (5)$$

$Pr$ : Prandtl number

The flow is turbulent at  $Re_d > 2300$ ;  $Nu_d$  is then defined by this equation:

$$Nu_d = 0.023 Re_d^{0.8} Pr^{0.3} \quad (6)$$

From the definition of the Nusselt number, the coefficient of heat transfer  $h_w$  is:

$$h_w = \frac{Nu_d \lambda}{x} \quad (7)$$

(2) Air flow over cooling tube surface

When the flat plate boundary layer theory is applied, the laminar flow condition is:

$$Re_x = Ux / \nu < 3.2E5 \quad (8)$$

$U$  : Flow velocity

$x$  : Distance from leading edge of flat plate

In laminar flow, the mean Nusselt number  $Nu_m$  and mean coefficient of heat transfer  $h_a$  are given by the following equations:

$$Nu_m = (0.458Pr^{0.33}Re_x^{0.5}) \times 2.0 \quad (9)$$

$$h_a = \frac{Nu_m \lambda}{x} \quad (10)$$

The results of the investigations using Eq. (2) indicated that both laminar and turbulent flows would be found in the cooling tubes under ordinary operating conditions of the motorcycle. On the other hand, when the cooling tube surface contacts air, the flow can be judged as laminar using Eq. (8) with actual operating conditions considered. Therefore, it was found that Eq. (9) is applicable.

### 3.2. Formulation of Numerical Procedure for One-Dimensional Heat Radiation Value

Heat balance in a single cooling tube is considered (see Fig. 4). If the temperature changes by the amount  $dT$  as the coolant passes through an infinitesimally thin zone of thickness  $dx$ , Eq. (11) can be obtained. Heat conduction in the direction of coolant flow is negligible in the cooling tube because its influence is very small.

$$C_p \rho u A dT = -Hs(T - T_o)dx - s\varepsilon\sigma(T^4 - T_o^4)dx \quad (11)$$

$C_p$  : Coolant specific heat

$\rho$  : Coolant density

$A$  : Cross-sectional area of tube

$s$  : Cross section circumferential length of tube

$u$  : Mean velocity on cross section of tube

$H$  : Overall heat-transfer coefficient from cooling water to air

$T_o$  : Ambient air temperature

$T$  : Coolant temperature at distance  $x$  from inlet

$\varepsilon$  : Thermal emissivity

$\sigma$  : Stefan-Boltzmann constant

Equation (11) is an ordinary differential equation showing temperature  $T(x)$  in coordinate  $x$ . The first term on the right side indicates heat radiation due to heat transfer, and the second term, heat radiation due to thermal radiation. The temperature distribution can be found by solving this equation. Mathematical solution of this equation is difficult due to the second term, but it is easily solved by numerical integration. Figure 5 shows an example of results for Eq. (11) solved with the fourth-order Runge-Kutta method. The figure shows water temperature distribution in the cooling tube assuming an inlet temperature of 353 K and ambient temperature of 283 K. It indicates the temperature gradually drops as it flows through the tube from the inlet to the outlet. The three lines represent flow velocities of 0.5 m/s, 1.75 m/s and 3.0 m/s in the tube; the higher the flow velocity, the higher the exit temperature.

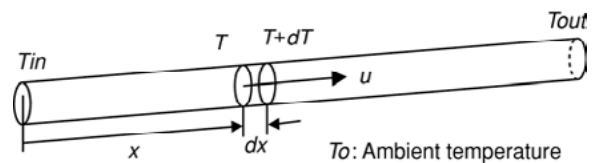


Fig. 4 Cooling tube model to introduce one dimensional heat radiation theory

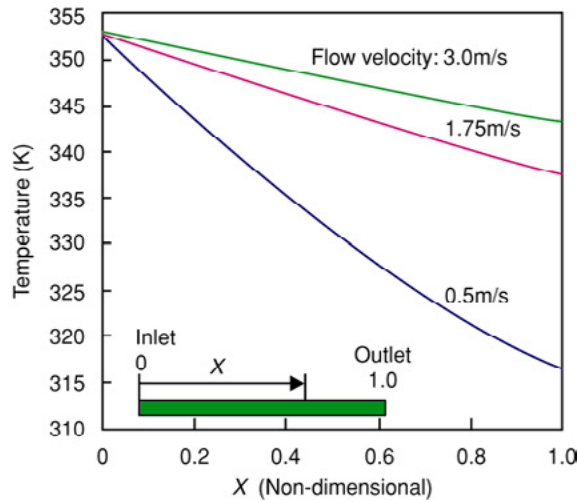


Fig. 5 Temperature distribution on cooling tube obtained from one dimension theory

If the inlet temperature is  $T_{in}$  and the outlet is  $T_{out}$ , the heat radiation value in a single tube  $Q_i$  is given by Eq. (12).

$$Q_i = C_p \rho u A (T_{in} - T_{out}) \quad (12)$$

$T_{in}$  : Inlet temperature

$T_{out}$  : Outlet temperature

#### 4. Correlation between Measured Values and Theoretical Predictions

##### 4.1. Overview of Experiment

Equation (11) showed that the cooling tube outlet temperature and the heat radiation value could be calculated if the geometrical specifications of the tube, tube inlet temperature and ambient air temperature were known. In order to confirm the validity of this equation, a heat radiation test was carried out with a single tube to obtain the heat radiation value for comparison with the results of Eq. (11). Figures 6 and 7 show a schematic and a photograph of the test apparatus.

Single aluminum tubes of the type installed in a complete radiator were used in the following specifications:

Specification A: Cross-section 24 mm × 2.0 mm, length  $L = 600$  mm, with fins

Specification B: Cross-section 24 mm × 2.0 mm, length  $L = 800$  mm, with fins

Specification C: Cross-section 24 mm × 2.0 mm, length  $L = 800$  mm, no fins

The fins were attached to the top and bottom of each tube; the heat radiation area with fins attached was 15.6 times the area with no fins. The test conditions were the following:

- (1) Wind velocity was constant and water velocity in the tube was adjusted to observe coolant speed-dependent changes in the heat radiation value.
- (2) Water velocity was constant and cooling wind velocity was adjusted to observe wind speed-dependent changes in the heat radiation value.

The inlet temperature  $T_{in}$  and ambient temperature  $T_{out}$

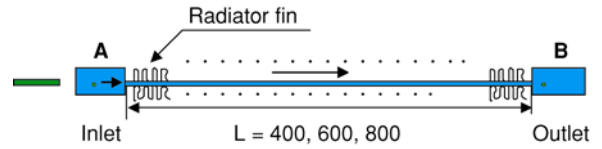


Fig. 6 Cooling tube for testing

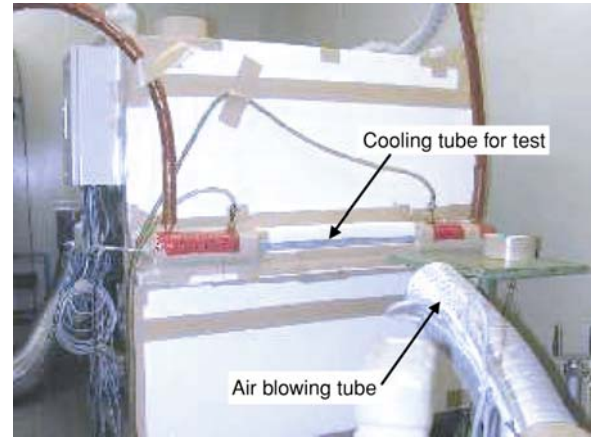


Fig. 7 Outside view of experimental device for cooling tube

were set at 353 K and 283 K, respectively. The heat radiation value for the entire tube  $Q$  was calculated by measuring the outlet temperature  $T_{out}$  in accordance with Eq. (13).

$$Q = C_p \rho q (T_{in} - T_{out}) \quad (13)$$

$C_p$  : Water specific heat

$\rho$  : Water density

$q$  : Volume flow rate

##### 4.2. Comparison of Experimental Measured and Theoretical Predictions

The following discusses the results of the test and compares them with the results predicted by Eq. (11). The thermal emissivity  $\varepsilon$ , used in Eq. (11), was set at 0.08 for the aluminum surface.

###### (1) Without fins

Figure 8 compares the calculated values and measured values for heat radiation with a Spec. C tube under constant wind and varying water velocity in the tube. This figure shows wide disparities between the two measured values at low coolant speeds. This was probably due to measurement errors because of the low absolute values for heat radiation when fins are not attached to the tube. The theoretically predicted value was about halfway between the two measured values and represented the average of the two measurements quite well.

Figure 9 also compares calculated and measured values from a Spec. C tube, but here, the water velocity was constant while the wind velocity was varied. There was some divergence of the measured values from the predictions at the highest wind velocities, but overall, the values matched well.

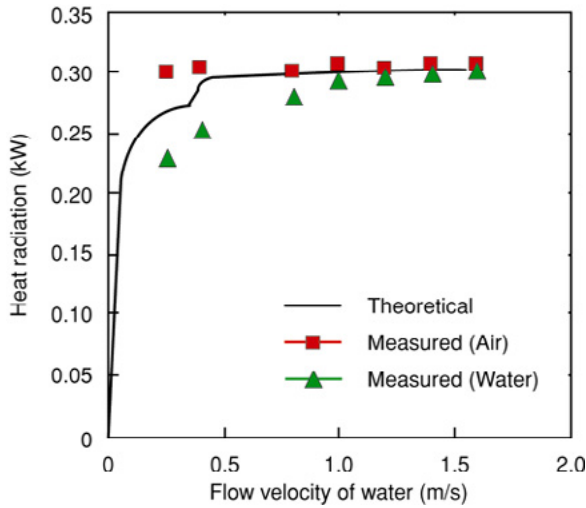


Fig. 8 Heat radiation characteristics of single tube (Wind velocity: Const. Without Fins)

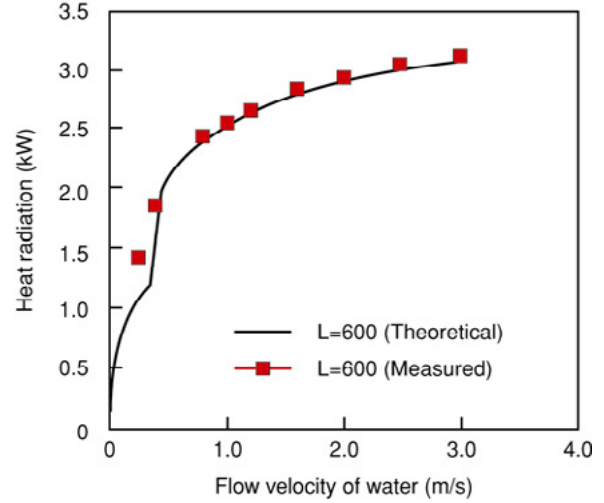


Fig. 10 Heat radiation characteristics of single tube (Wind velocity: Const. With Fins)

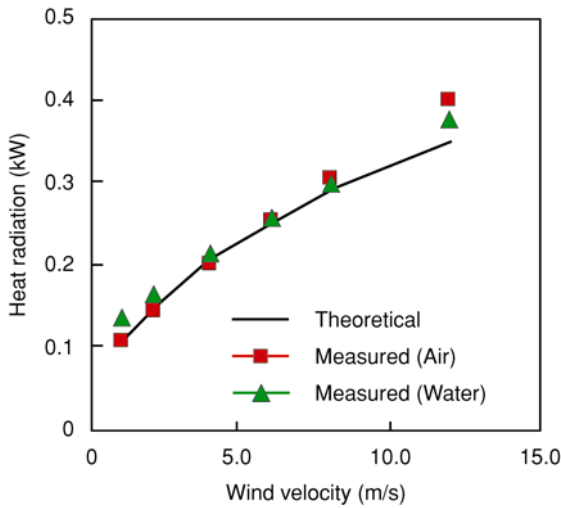


Fig. 9 Heat radiation characteristics of single tube (Water velocity: Const. Without Fins)

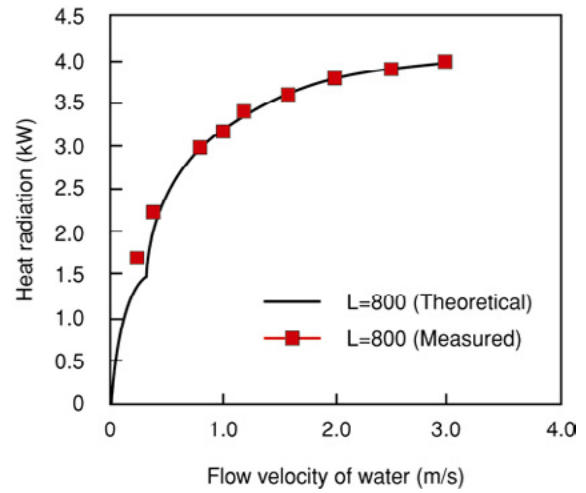


Fig. 11 Heat radiation characteristics of single tube (Water velocity: Const. With Fins)

These results indicate that Eq. (11) accurately predicted the heat radiation value when the fins are not attached.

(2) With fins

Since the heat-radiating area was increased 15.6 times by the addition of fins, the value of  $ha$  in Eq. (1) was modified to show the influence of this change.

Figures 10 and 11 show the predicted and measured values for heat radiation from single tubes 600 mm and 800 mm long. In both cases, the wind velocity was constant and the results were observed while the water velocity was varied. These findings again confirmed the validity of Eq. (11).

Figures 12 and 13 show predicted and measured heat radiation values from tubes 600 and 800 mm long under constant water velocity and varying wind velocity. These indicate some deviation of the actual values from the predictions by the laminar flow theory at the low and high wind velocity range, but there is a good match in the center of the range.

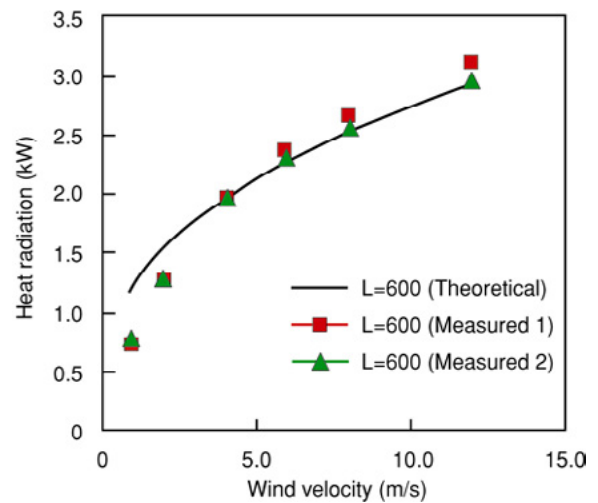


Fig. 12 Heat radiation characteristics of single tube (Water velocity: Const. With Fins)

## (3) Comparison of heat radiation with and without fins

Figure 14 shows the ratio between heat radiation values when the tubes were and were not equipped with fins for both theoretical and measured values. There is a good quantitative match between the theoretical and measured values nearly everywhere. These results show that the ratio of heat radiation is not constant but dependent on the coolant flow velocity in the tube. The reason is that, according to Eq. (6), the higher the flow velocity, the higher the Nusselt number on the water side; this changes the overall heat transfer coefficient  $H$ . Since this behavior was reproduced by Eq. (11), the present theory seems to have been confirmed.

## (4) Effect of thermal transfer

Since Eq. (11) allows accounting for emissivity  $\varepsilon$ , a measure of the influence of thermal radiation, the calculation was re-performed by varying  $\varepsilon$ .

Figure 15 shows the characteristics of heat radiation at emissivity of 0.08<sup>(3)</sup> for aluminum used for a cooling tube this time and at  $\varepsilon = 1.0$ , the upper limit for this parameter. Heat radiation was increased about 7% by raising  $\varepsilon$  from 0.08 to 1.

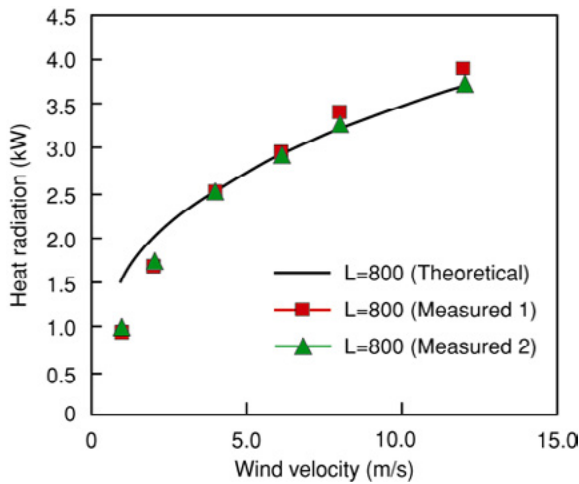


Fig. 13 Heat radiation characteristics of single tube (Water velocity: Const. With Fins)

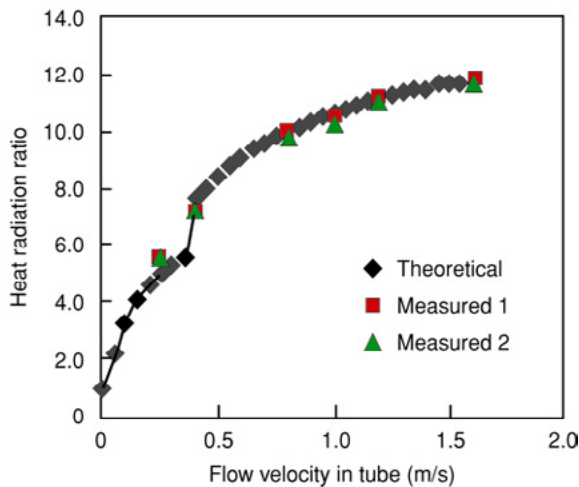


Fig. 14 Comparison between experimental results and theoretical results of heat radiation ratio ( $r$ ): (heat radiation with fins)/(heat radiation without fins)

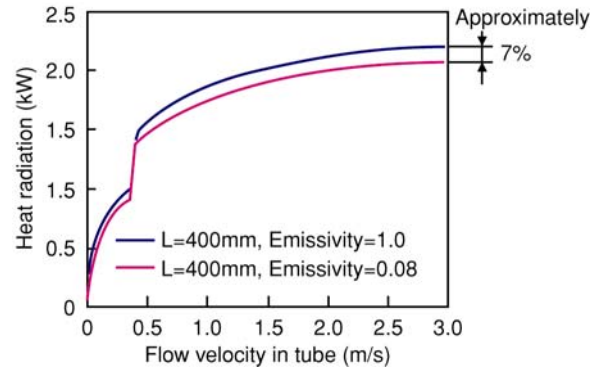


Fig. 15 Influence of emissivity in thermal radiation

Since the thermal emissivity of a real material is always less than 1, the difference due to increased  $\varepsilon$  would be less than 7%, but this result shows the necessity of thermal radiation in the performance of radiators.

## 5. Extension of Results to a Complete Radiator

It has been shown that the theory predicts experimental results reasonably well for single radiator tubes. An actual radiator is made up of multiple tubes; the theory for single tube can be extended to multiple tubes as follows. The heat radiation value  $Q_{tot}$  from a complex radiator consisting of  $n$  tubes with different lengths (of lengths  $L_1, L_2, \dots, L_n$ ) is the sum of the heat radiation  $Q_i$  from each cooling tube.

From Eq. (12),

$$Q_{tot} = \sum_{i=1}^n Q_i \quad (14)$$

The temperature reduction of mixed coolant water flowing from different tubes,  $T_m$ , is given by:

$$T_m = \frac{\sum Q_i}{C_p \sum u_i A_i} \quad (15)$$

$u_i$ : Flow velocity in  $i^{\text{th}}$  tube

$A_i$ : Cross-sectional area of  $i^{\text{th}}$  tube

However, the area ratio for heat radiation due to the addition of fins is different from that of single tubes. Since each pair of neighboring tubes shares a fin, the area ratio is one-half the value calculated for single tubes.

Thus, if  $u_i$  is already established, it is possible to find the temperature reduction of the coolant due to heat radiation and cooling in a complete radiator consisting of multiple tubes. Here  $u_i$  can be found from three-dimensional flow field calculations. Figures 16 and 17 show the results of the three-dimensional fluid flow calculations for the flow field of a complete radiator. They show the pressure distribution and the distribution of mean flow velocities in each cooling tube, respectively. Figure 18 presents a comparison of the predicted heat radiation characteristics using the preceding flow velocity distribution and the actual measured values. The vertical axis represents the dimensionless heat radiation value, where the value 1 represents the heat radiation attained when the flow rate is 100 l/m.

This graph indicates that the calculations showed an acceptable match with actual results in a complete radiator.

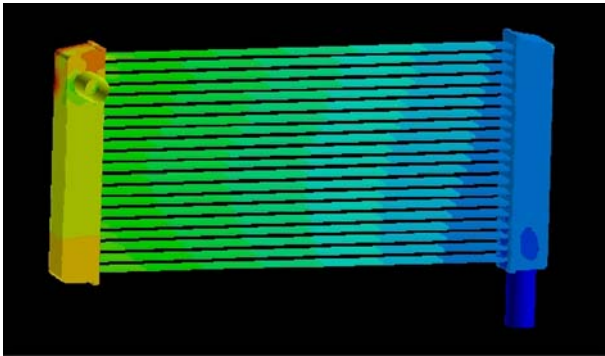


Fig. 16 Pressure distribution obtained from three-dimensional flow field simulation

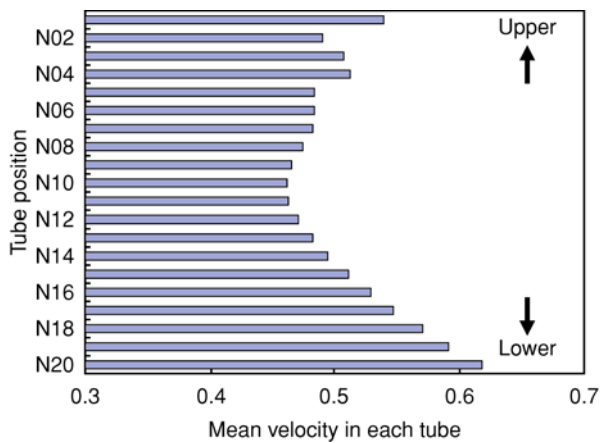


Fig. 17 Mean velocity distribution over whole tubes

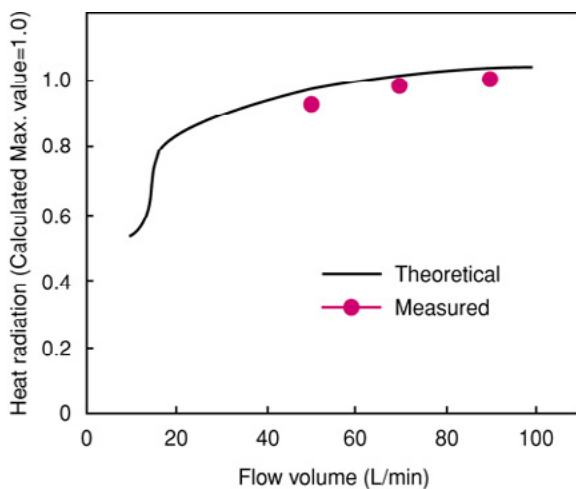


Fig. 18 Comparison of heat radiation characteristics between theoretical values and experimental values

## 6. Conclusion

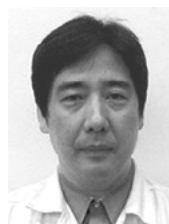
Combining calculations for a three-dimensional flow field and a one-dimensional heat transfer model of a motorcycle radiator of any shapes enabled creation of a method for quickly and accurately predicting heat radiation characteristics under a wide variety of conditions.

This method allows rapid estimation of heat radiation characteristics for radiators of various face shapes, so it is useful for refinement of radiator specifications in early stages of development.

## References

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■ Authors ■



Takashi WATANABE



Masayuki IWATA