Development of Particle-Based Impingement Jet-Flow Heat Transfer Coefficient Model for Thermal-Fluid Analysis of Oil Jet-Cooling Electric Vehicle (EV) Motor

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Keywords Rotary machine, Oil-cooling, Thermal-fluid simulation, Heat transfer coefficient

Abstract

When developing rotating machines that aim to achieve high output, high rotation speed, and compact size, an oil cooling system with high cooling performance is often adopted. The oil jet local cooling method, which sprays oil onto high-temperature areas, is attracting attention for its efficiency. When developing products that use oil jet local cooling, thermal-fluid analysis is essential to improve work efficiency and acquire new expert knowledge. Thermal-fluid analysis of oil jet local cooling has many issues, however, with an accuracy and calculation time due to complex liquid surface deformation and difficulty in evaluating heat transfer coefficient.

We used the particle-based method, one of the thermal-fluid analysis methods, to develop a particle – based method impingement jet heat transfer coefficient model for the design study of oil jet local cooling. The model was constructed through a comparison with the correlation equation of the impinging jet experiment, and the validity of the model was confirmed by comparing it with the oil cooling rotor experiment. It was revealed that the heat transfer coefficient model can roughly reproduce the heat transfer coefficient on the rotor surface.

1 Preface

In recent years, rotating machines such as Electric Vehicle (EV) motors have been required to be more compact with high output and high rotation speed. Local cooling technology that injects or sprays oil to high-temperature areas is actively used to improve cooling performance. Thermal-fluid analysis technology is essential for the optimization and advanced design of local cooling technology. Analysis of localized cooling by injection or spraying, however, is extremely difficult due to the complexity of free-surface flow and heat transfer.

The particle-based method is a fluid analysis method that can easily handle large deformations of the liquid surface and movement of the wall surface. It is, therefore, suitable for oil flow analysis inside the rotating machine. The particle-based method, however, cannot accurately calculate heat transfer between a fluid and a solid. It was, therefore, difficult to apply it to temperature analysis inside a rotating machine.

We developed a particle-based impingement jet-flow heat transfer coefficient model that is applicable to oil impingement jet-flow local cooling based on our idea for application to cooling system design for oil-cooling EV motors[1]. This paper introduces an overview of this model, verification experiments, and results.

2 Building a Particle-Based Impingement Jet-Flow Heat Transfer Coefficient Model

2.1 Overview

Fig. 1 shows the impingement jet flow[1]. According to the papers of Hirata and Morioka[2], an experimental correlation expression formula is proposed in relation to the heat transmission coefficient in the case when a water jet impinges vertically against the wall at the velocity V_{inj} from a round nozzle with a diameter D apart from Distance H. Our

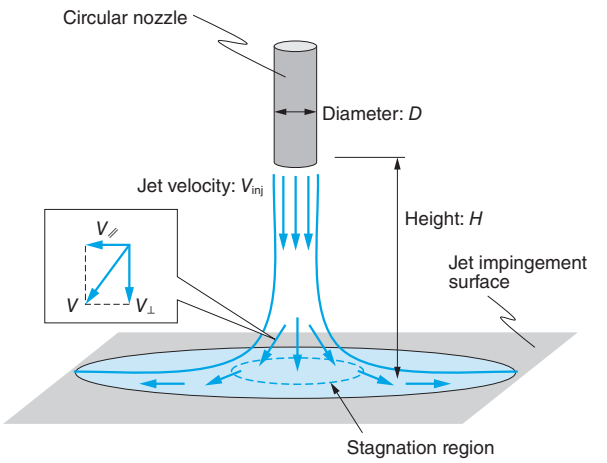

Fig. 1 **Impingement Jet Flow**[1]

The situation of impingement jet flow is shown.

model was created to reproduce this correlation formula.

In this process, we employed a commercially available fluid simulation software, Particleworks 6.2 (by Prometech Software, Inc.), based on the Moving Particle Simulation Method (MPS Method)[3], a particle-based method. The developed heat transfer coefficient model was implemented with the aid of a Software Development Kit by Particleworks.

2.2 Modeling for Local Heat Transfer Coefficient

First, the local heat transfer coefficient h_x is calculated from the local physical quantity in the vicinity of wall surface based on the heat transfer coefficient correlation formula. Based on the mean Nusselt number correlation formula[4] of the isothermal cylinder that crosses at right angles with the fluid flow, the following expression is established:

$$Nu_x = \frac{h_x x_{Nu}}{\lambda} = C_1 Re_x^n Pr^{0.37} \quad \cdots\cdots (1)$$

Where, Nu_x is the local Nusselt number, and λ and Pr are the heat transfer coefficient of the fluid and the Prandtl number, respectively. Value x_{Nu} is the characteristic length of the local Nusselt number (explained in Section **2.4** below). Value Rl_x is the local Reylolds number that is defined below.

$$Re_x = \frac{V_{Re} x_{Re}}{\nu} \quad \cdots\cdots (2)$$

Where, Value ν is the kinematic viscosity of fluid. Values V_{Re} and x_{Re} are the characteristic velocity and the characteristic length of the local Reynolds number, respectively. These will also be explained in Section **2.4** below. **Table 1** shows the mutual dependency of Re_x, C_1, and n in Expression (1). Values C_1 and n are the parameters dependent on the local Reynolds number as defined in **Table 1**. Values β_{Re2} and β_{Re3} are the tuning parameters intended for the fine tuning so that the distribution of heat transfer coefficients by simulation can fit the experimental correlation formula of impingement jet flow by Hirata and Morioka[2].

Table 1 **Mutual Dependency of Re_x, C_1, and n in Expression (1)**

Relationship of dependency among Re_x, C_1, and n in Expression (1) is shown. Values of β_{Re2} and β_{Re3} are the tuning parameters.

Re_x	C_1	n
1～40	0.75	0.4
40～1000	$0.51 \times \beta_{Re2}$	0.5
1000～2×10^5	$0.26 \times \beta_{Re3}$	0.6
2×10^5～1×10^6	0.076	0.7

2.3 Determination of Stagnation Region

In this model, the aforementioned characteristic velocity and characteristic length are modified according to the case whether the fluid particles are staying in the stagnation region or not. Moving forward, it is assumed that the fluid particles stay in the stagnation region when $V_{\perp}$ is in the following state while the fluid particle velocity V is divided into Component $V_{\perp}$ vertical to the wall surface and Component $V_{//}$ parallel to the wall surface as shown in **Fig. 1**.

$$V_{\perp} \geqq \alpha_{ST} V_{//} \quad \cdots\cdots (3)$$

Where, Value α_{ST} is the tuning parameter that assumes a value between 0 and 1. This value is determined so that distribution of heat transfer coefficients from simulation coincides with the experimental correlation formula of impingement jet flow.

2.4 Modeling for Characteristic Length and Characteristic Velocity in Stagnation Region

First, outside the stagnation region, the characteristic lengths x_{Nu} and x_{Re} are assumed to be the traveling distance x_{TL} of fluid particles moving along the wall surface in the vicinity of the wall. The characteristic velocity V_{Re} is assumed to be $V_{//}$.

Next, the modeling process in the stagnation region is explained. In consideration of the distance that affects the heat flow in the direction perpendicular to the wall surface, the characteristic length x_{Nu}

of the local Nusselt number is assumed to be the effective radius of $3.1l_0$ conforming to the Moving Particle Simulation (MPS) method, where Value l_0 is the initial distance between particles. The characteristic length x_{Re} of the local Reynolds number is basically assumed to be x_{TL} similarly as for the case of outside the stagnation region. In the stagnation region, however, the situation is assessed shortly after the collision of fluid flow against the wall, and it can be considered that Value x_{TL} becomes extremely small. For this reason, when the traveling distance is smaller than l_0 that corresponds to the resolution of the MPS method, a condition of $x_{Re} = l_0$ holds.

For the characteristic velocity V_{Re} of the local Reynolds number in the stagnation region, it is preferable to adopt the jet flow velocity V_{inj}. Nevertheless, this value cannot be adopted because it is impossible to know the accurate V_{inj} value from the local physical quantity under general conditions. Accordingly, it is presumed that the Bernoulli law is holding for the fluid flow from the nozzle outlet to the stagnation region so that the jet flow is evaluated by the expression below.

$$\tilde{V}_{inj} = \sqrt{\frac{2}{\rho} P_s + V^2} \quad \cdots\cdots (4)$$

Where, ρ is the fluid density, and P_s and V are the static pressure and velocity of fluid particles in the vicinity of the stagnation region, respectively. In other words, $V_{Re} = \tilde{V}_{inj}$ in the stagnation region.

2.5 Determination of Tuning Parameters and Verification of Reproducibility by Comparison with Experimental Correlation Expression Formula of Impingement Jet Flow

The system shown in **Fig. 1** was analyzed and the result of implemented model analysis was compared with the experimental correlation expression formula of impingement jet flow by Hirata and Morioka[2]. The fluid used was water at a room temperature. **Fig. 2** shows the behavior of impingement jet flow[1].

Fig. 3 shows heat transfer coefficient distribution[1]. On the left side of the drawing, distribution of heat transfer coefficients obtained from impingement jet flow analysis, processed by mapping on the wall surface, is shown. On the right side of the drawing, the graph of heat transfer coefficient distribution is shown, obtained by circular average for each distance from the center point of impingement jet flow. The black line shows the correlation formula by

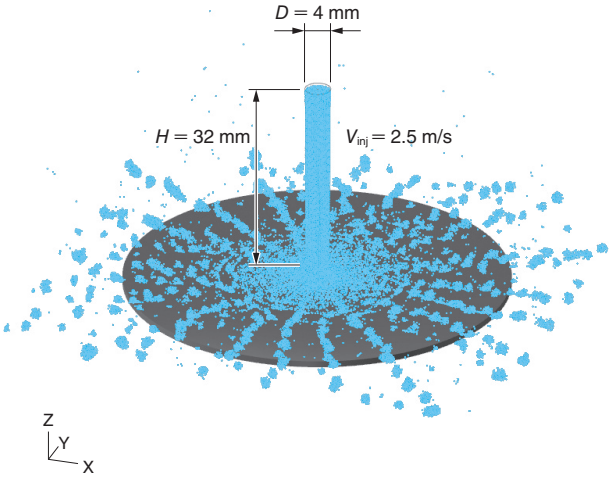

Fig. 2 **Behavior of Impingement Jet Flow[1]**

Conditions for behavior of impingement jet flow and the result of steady-state flow are shown.

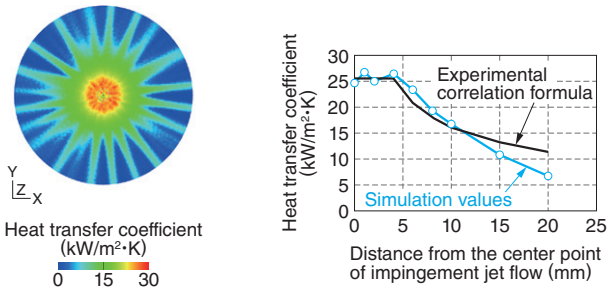

Fig. 3 **Heat Transfer Coefficient Distribution[1]**

The result of heat transfer coefficient mapping on the wall surface by impingement jet flow simulation and distribution of heat transfer coefficients in relation to the distance from the center point of impingement jet flow are shown.

Hirata and Morioka[2] and the blue dotted line shows the result of simulation with the use of the relevant model. The set of the tuning parameter is expressed by $(\alpha_{ST}, \beta_{Re2}, \beta_{Re3}) = (0.5, 1.6, 1.3)$. The analytical error was 5% or less in the stagnation region and tens of percent outside the region. It can be recognized that experiments can be roughly reproduced with this model. The reason why errors are increased in a distant area from the stagnation region can be explained by the fact that fluid particles are radially spread and this worsens the reproducibility of jet flows.

3 Verification of Model Validity by Rotor Oil-Cooling Experiment

Fig. 4 shows the configuration of rotor oil-cooling experiment and the situation of the experiment[1]. The adequacy of the implemented model

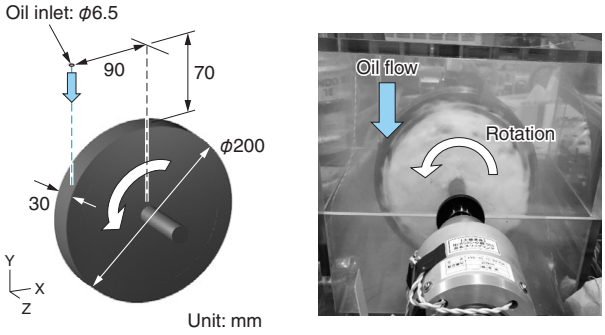

Fig. 4 Configuration of Rotor Oil-Cooling Experiment and Situation of Experiment[1]

The configuration of rotor oil-cooling experiment (drawing on the left) and the situation of the experiment (picture on the right) are shown.

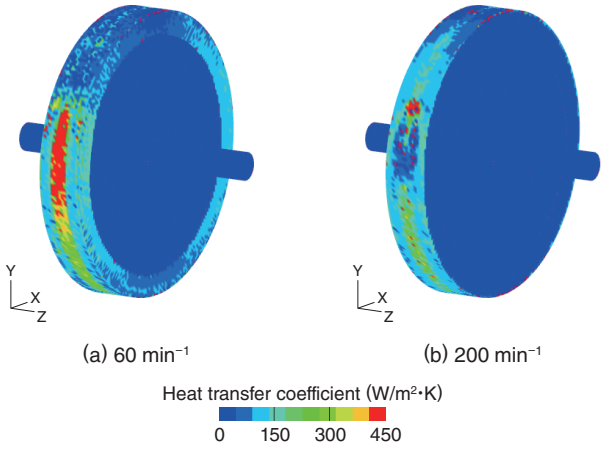

Fig. 5 Distribution of Heat Transfer Coefficients Based on Result of Analysis of Rotor Oil-Cooling Experiment[1]

The distribution of heat transfer coefficients obtained from the result of analysis of rotor oil-cooling experiment is shown. The result of the analysis was obtained in two cases of rotations: (a) 60 min^{-1} and (b) 200 min^{-1}.

was confirmed through the comparison of the model simulation result with the heat transfer coefficient on the rotor surface acquired from rotor oil-cooling experiment. An aluminum rotor with an embedded heater was put in a polycarbonate case and a coolant was made to flow down at the rate of 1.5 L/min indicated by an arrow mark atop the rotation axis. The coolant used was an industrial oil named Highlandwide15 (by ENEOS Corporation). Based on the coolant temperature on the rotor outer circumference and the quantity of heat removed by the coolant, the mean heat transfer coefficient on the rotor outer circumference was determined.

Fig. 5 shows the distribution of heat transfer coefficients on the rotor outer circumference when

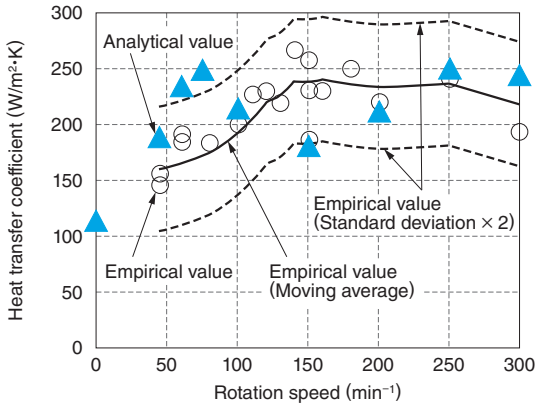

Fig. 6 Average Heat Transfer Coefficient around Rotor Circumference Versus Rotation Speed[1]

The average heat transfer coefficient around the rotor circumference for each rotation speed in the experiment and analysis are shown.

the test was analyzed with the use of the implemented model. Fig. 6 shows the mean heat transfer coefficient on the rotor outer circumference per rotation based on the result of experiment and analysis. The solid line shows the mean value obtained from the averaged 3-section movement and the dotted line shows the twice value (2σ) of standard deviation obtained from the data of 140 min^{-1} to 160 min^{-1}. The mean heat transfer coefficient by simulation was almost the same as the value by experiments and kept within 2σ of empirical values. Trends of dependency on rotations were also roughly grasped. Consequently, we confirmed that phenomena can be almost simulated with the use of implemented model.

4 Postscript

We have developed a heat transfer coefficient model that can calculate the local heat transfer coefficient on the surface of a cooled component, made possible by flow analysis using the particle-based method. This aims the creation of a thermal-fluid analysis method that can be used in the design of local cooling of high-temperature parts using jet flow in the development of oil-cooled motors.

Going forward, we plan to further optimize the model by comparing correlation equations between oblique collisions, wall movement, and oil jet experiments, and aim to improve the accuracy of our heat transfer coefficient models.

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《**References**》

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