Calculation and validation of stirring resistance in cam-shaft rotation using the moving particle semi-implicit method

Nobuhiro YUHASHI*, Ichiro MATSUDA* and Seiichi KOSHIZUKA**

*Maruyama Mfg. Co., Inc.
1554-3 Konumata, Togane-city, Chiba 283-0044, Japan
E-mail: yuhashin@maruyama.co.jp

**Department of System Innovation, The University of Tokyo
7-3-1 Hongo, Bunkyo-ku, Tokyo 113-8656, Japan

Received 25 May 2016

Abstract

The moving particle semi-implicit (MPS) method has been used in a wide range of industrial fields with the free surface. Many machines use oil for lubrication as well as cooling. In this study, stirred fluid flow of various oil types with rotating cam-shafts was calculated by using the MPS method. Moreover, experiments were conducted in order to validate the calculation results by comparing the torque values. When the oil viscosity is 30 cs or lower than 30 cs, the calculated torques agree well with those of the experiment. When the oil viscosity is larger than 30 cs, the calculated torques are lower than those of the experiment. This is due to the involvement of small air bubbles. The air bubbles are retained for a long time when the oil viscosity is high, which increases the oil surface and eventually causes a higher torque. This phenomenon is not calculated because the air bubbles are not modeled in this study. In the case when the oil viscosity is 30 cs, the calculation results of the torque values were in good agreement with those of the experimental results. The calculation results show convergence with respect to the particle size.

Key words: MPS method, Cam-shaft, Fluid resistance, Contact angle, Sensitivity analysis, GCI

1. Introduction

Many machines use oil to provide lubrication as well as cooling. In a slider-crank mechanism, which is typified by engines and reciprocating pumps, large deformation on free surfaces, including splitting and merging takes place. A shaft and a case that can sufficiently satisfy lubrication and cooling performances need to be designed. In the situation wherein the optimal shapes of the shaft and case were considered with experimental data, experiments under various conditions were required. For this reason, development costs have increased. However, if the qualitative and/or quantitative evaluation of the oil behavior based on numerical simulation is realized, a drastic cost-reduction in development processes can be expected. Furthermore, if the shapes of the shaft and case are already designed, deciding on the optimal oil level or the oil’s physical properties for the given shapes become important issues.

Methods used for calculating complex phenomena seen in free surfaces include the grid method and the particle method. In a discretization method of the fundamental equation, such as the Volume of Fluid (VOF) method (Hirt and Nichols, 1981) in which Euler-type fixed computational mesh is used, numerical diffusion resulting from discretization of advective terms is inevitable, and blurs in the phase are common problems in this regard. To ease these problems, it is necessary to use a different scheme of a higher-order or a special algorithm that uses the VOF method and Level Set method (Osher and Fedkiw, 2003), which leads to a more complicated discretized equation. In contrast, the MPS method applied to an issue of incompressible flow is a Lagrange-type technique that is conducted by moving calculation points. Since the advective term is calculated by tracing the particle movement, it is possible to avoid the numerical diffusion caused by the difference between the advective terms without complicated procedures. In the MPS method, calculations...
are performed by dividing the fluid into particles instead of meshing, and by tracing the behavior of the particles. Therefore, it is easy to treat free surface flow or splash with large deformation (Koshizuka et al., 1995; Koshizuka and Oka, 1996). For this reason, calculations are applied to various areas including breaking waves (Koshizuka et al., 1998), split of droplets (Nomura et al., 2001; Shirakawa et al., 2001; Shibata et al., 2004a), sloshing of tanks (Chikazawa et al., 1999, 2001; Sueyoshi and Naito, 2001; Naito and Sueyoshi, 2001), the issue of deck wetness of a vessel in a wave (Koshizuka et al., 2004; Shibata et al., 2004b; Shibata and Koshizuka, 2007), mixing tanks in the chemical process (Koshizuka et al., 2015), and tsunami simulation (Murotani et al., 2014; Nannichi et al., 2016). Furthermore, a case study of the MPS method in industrial machines has been performed involving dynamic simulation of water splashing by an outboard engine, and soil cultivation (Yamamoto et al., 2011). Therefore, we focus on the MPS method (Koshizuka and Oka, 1996), which is a particle method that calculates complicated flow-field behaviors including deformation, split, and merge on free surfaces.

In the numerical simulation of rotating objects, Al-Shibl et al. (2007) reported that the windage power loss in air was calculated as a two-dimensional problem, and the calculations were compared with experimental data. They modeled two gear teeth as a calculation area and reduced the calculation time by applying the periodic boundary conditions in the circumferential direction. In their paper, the actual windage power loss was estimated by applying a correction to the two-dimensional calculation results. Hill et al. (2008) reported that the windage power loss in air was calculated as a three-dimensional problem. However, they modeled a half gear to reduce the calculation time by applying the symmetric boundary conditions in the rotation axis direction. Furthermore, it was noted that the turbulence model is important in the calculation of the windage power loss for higher revolving speeds. Additionally, they revealed that three-dimensional dependency appears on the pressure and velocity distributions of the tooth surface. It was evident that the three-dimensional calculation was necessary for rotating objects. However, these calculations targeted only air, and the influence of the liquid phase was not considered. In regard to the calculation of the two-phase flow of oil and air, Tan et al. (2011) reported that the stirring loss and oil behavior were calculated using the VOF method when part of a gear was immersed in oil. In their report, the qualitative evaluation of the calculation results was conducted using the parameters of tooth height and tooth pitch angles of the gear to study the impact that the gear shape has on the stirring resistance. There were differences between the results of the experiments and the oil behavior along with stirring resistance at a revolution range of 500 rpm or higher, and was concluded that the difference due to the grid spacing of the mesh influenced the results. However, when a high-resolution calculation was conducted, the simulation cost increased. In industrial product applications, such a calculation cost problem must be resolved. However, the influence of gear pairs is not considered in their study. On the other hand, Li et al. (2008), Imai et al. (2009), and Arisawa et al. (2009, 2011, and 2013) conducted numerical simulation on the gear pair. Especially, Arisawa et al. (2013) conducted numerical simulation on several gear pairs as a three-dimensional problem. They chose the VOF method for the study of the shroud shape. The purpose of their study was the calculation and reduction of stirring loss in the transmission of an aircraft engine lubricated by oil. Although it was reported that the calculation results for the reduction rate of stirring loss because of the difference of the shroud shape were in good agreement with experimental results, a quantitative evaluation has not been conducted. As a case study of the rotating objects in the MPS method, Muto et al. (2010) reported the prediction of the fluid resistance in stirred fluid using the particle method. Yuhashi et al. (2015) chose the MPS method for cam-shaft torque calculation and validation, and evaluated the convergence of the calculation results using the GCI method (Grid Convergence Index method) (ASME, 2009).

As for the stirring resistance force of fluid in a slider-crank mechanism, both the case and the shaft shapes, and the oil’s physical properties are important design parameters. However, little has been reported on the relation between the oil’s physical properties and the stirring resistance. In this study, the stirring resistance, which means resistance that a rotating cam-shaft encounters when the oil is churned up, is calculated by numerical simulation. To perform this task, first, the oil behavior and the stirring resistance are calculated by the MPS method under the condition in which part of the cam-shaft is immersed in oil. Second, an experiment is conducted using a model equivalent to the numerical simulation to validate the results by comparing the torque obtained by the numerical simulation and the torque from the experiment. In this case, the simulation software Particleworks Ver. 4.5 (Prometech Software, Inc.) is used. Since the air bubble model is not supported in Particleworks, the calculations are conducted at a revolving speed of less than 4000 rpm where an aeration effect is not developed, as demonstrated by Leprince et al. (Leprince et al., 2009).
2. Numerical simulation based on MPS method

2.1. Governing equations

The MPS method developed by Koshizuka and Oka (1996) is a Lagrange-type numerical calculation technique for incompressible flow with free surfaces. The governing equations of incompressible flow are the following continuity equation expressed in Lagrangian notation and the Navier-Stokes equation:

\[
\frac{D\rho}{Dt} = 0 \quad (1)
\]

\[
\frac{D\mathbf{u}}{Dt} = -\frac{1}{\rho} \nabla P + \nu \nabla^2 \mathbf{u} + \mathbf{g} + \mathbf{F}_s \quad (2)
\]

where \( \rho, t, \mathbf{u}, P, \nu, \mathbf{g}, \) and \( \mathbf{F}_s \) represent density, time, velocity vector, pressure, kinematic viscosity, gravitational acceleration vector, and external force vector, respectively. Numerical simulation is conducted after discretizing the governing equations. Spatial differentiation operators such as gradient and Laplacian in the governing equations are replaced by the corresponding inter-particle interaction models (Koshizuka and Oka, 1996).

2.2. Surface tension and wettability model (Kondo et al., 2007)

Surface tension and wettability can be calculated by the inter-particle potential force \( \mathbf{F}_s \) in the MPS method, and the inter-particle potential force is expressed as follows:

\[
\mathbf{F}_{sij} = \begin{cases} 
\nabla \phi_{ffij} & \text{if } j \text{ is fluid} \\
\nabla \phi_{fsij} & \text{if } j \text{ is solid}
\end{cases} \quad (3)
\]

\[
\begin{align*}
\phi_{ff} &= C_{ff} p(r) \\
\phi_{fs} &= C_{fs} p(r)
\end{align*} \quad (4)
\]

where \( C_{fs} \) and \( C_{ff} \) represent the potential coefficient between fluid and solid, and that between fluid and fluid, respectively. Moreover, \( p(r) \) is the potential function and is defined as follows:

\[
p(r) = \frac{1}{3} \left( r - \frac{3}{2} r_{\text{min}} + \frac{1}{2} r_e \right) (r - r_e)^2 \quad (5)
\]

where \( r, r_{\text{min}}, \) and \( r_e \) represent the particle distance, the initial minimum particle distance, and the influence radius of potential force, respectively. In this study, \( r_e = 3.1 r_{\text{min}} \) is used.

The wettability on the solid wall is represented by a contact angle, and the relationship between the surface tension coefficient and the contact angle is expressed by the following Young’s equation:

\[
\sigma_s - \sigma_{fs} - \sigma_f \cos \theta = 0 \quad (6)
\]

where \( \theta, \sigma_s, \sigma_{fs}, \) and \( \sigma_f \) represent the contact angle, surface energy of the solid wall, interfacial energy between solid and fluid, and surface energy, respectively. Because the potential energy is proportional to the potential coefficient, the relation shown below is obtained:

\[
\frac{C_{fs}}{C_{ff}} = \frac{1}{2} (1 + \cos \theta) \quad (7)
\]
2.3. Torque calculation

If the fluid is churned up by the rotating object around the fixed axis, the torque \( T \) exerted on the rotating object is expressed as follows:

\[
T = T_{\text{pressure}} + T_{\text{viscosity}} + T_{\text{surface tension}} + T_{\text{inertia}}
\]  

(8)

where \( T_{\text{pressure}} \), \( T_{\text{viscosity}} \), \( T_{\text{surface tension}} \), and \( T_{\text{inertia}} \) represent pressure, viscosity, surface tension, and inertial contribution to the torque, respectively. Although the inertial torque cannot be ignored in general, it is treated as \( T_{\text{inertia}} = 0 \) for the steady state.

Figure 1 shows the schematic of the torque calculation. Navier-Stokes equation, including the surface tension term, is expressed as follows:

\[
\frac{Du}{Dt} = -\frac{1}{\rho} \nabla P + \nu \nabla^2 u + g + \nabla \phi
\]  

(9)

The external force vector term in Eq. (2) represents the surface tension term. In Eq. (9), \( \nabla \phi \) represents surface tension. When the cam-shaft revolves, the fluid particle of neighborhood \( i \) moves according to the Navier-Stokes equation. The torques from the pressure, viscosity, and surface tension are considered to be the resistance force that the cam-shaft receives, which is expressed as follows:

\[
f_i = -m_i \left( -\frac{1}{\rho} \nabla P + \nu \nabla^2 u + \frac{1}{\rho} \nabla \phi \right)
\]  

(10)

where \( f_i \) represents a force vector that the cam-shaft gave to the particle \( i \), and \( m_i \) is the mass of an individual particle. Therefore, the torque is calculated from the following equation, as the reaction to the force given to all the particles interacting with the cam-shaft:

\[
T = - \sum_{i \in \text{fluid particle}} \mathbf{r}_i \times f_i
\]  

(11)

where \( \mathbf{r}_i \) represents the distance vector from the revolution center to fluid particle \( i \).

2.4. Calculation conditions

Figure 2 shows the computational domain. The cam-shaft is placed at the center of the cylindrical case and immersed in lubricating oil up to the revolution center. The initial fluid particles are arranged at regular intervals at a fixed distance \( d \). In the particle method, it is common to arrange wall particles as the wall boundary conditions. However, it is difficult
to express smooth surfaces accurately. In addition, as the number of wall particles increase in a three-dimensional
calculation, the calculation time and memory requirements increase. Hence, in this study, numerical simulation has been
conducted using a polygon model (Harada et al., 2008) for the wall boundary condition. In the polygon model, the wall
particles are not necessary, which reduces the calculation time and memory. The non-slip condition between the wall
surface and the fluid is set.

The static contact angle of each oil type is measured by the $\theta/2$ method using DropMaster (Kyowa Interface Science
Co., Ltd.). The measurement results are listed in Table 1. Figure 3 shows the appearance of each oil type for the
measurement. In the $\theta/2$ method, the measurement is conducted using the ratio of the droplet height $h$ and width $w$. The
relation with the contact angle is shown below:

$$\theta = 2\tan^{-1}\left(\frac{2h}{w}\right)$$

(12)

To calculate the wettability with high accuracy, the use of a very fine particle diameter as well as a large amount of
calculation time and memory are required. In particular, when the contact angle is small, sufficient accuracy in
calculations cannot be secured even if very fine particle diameter is used. For this reason, the contact angle is treated as
0°. The dynamic contact angle is assumed to be equivalent to the static contact angle. We confirmed that the torque is
underestimated, in the case of not using the LES model. Therefore, in this study we are using the LES model. LES (Large
Eddy Simulation) (Arai et al., 2013) is used as the turbulent flow model. The calculation conditions are listed in Table 2.

| Oil type | (a) 1 cs | (b) 5 cs | (c) 30 cs | (d) 50 cs | (e) 100 cs |
|----------|---------|---------|----------|----------|----------|
| Contact angle : $\theta$ [°] | 0 | 0 | 4.5 | 6.7 | 7.5 |

Fig. 2  Configuration of the computational domain.

Fig. 3  Measurement of static contact angle.
3. Torque measurement and visualization experiment

Figure 4 shows the experimental rig. The cam-shaft driven by an AC dynamometer is placed in a transparent acrylic case with a cylindrical shape for the purpose of visualizing the oil behavior. The stirring resistance force and the revolving speed are measured by a torque sensor positioned between the dynamometer and the cam-shaft. The oil behavior is recorded by a high-speed camera (Phantom, Vision Research, Inc.). Torque values are measured 10 times when the oil temperature reaches 25 °C. The torque values are considered as the average, and the difference between the max/min values is expressed as an error bar. The mechanical loss generated by the bearings is measured for each revolving speed, and the stirring resistance force of the oil is determined by subtracting the mechanical loss from the total stirring resistance force. The oil used for this experiment is commercially available silicone oil. The physical properties at the temperature of 25 °C are listed in Table 3.

Table 2 Calculation conditions.

| Oil type | (a) 1 cs | (b) 5 cs | (c) 30 cs | (d) 50 cs | (e) 100 cs |
|----------|----------|----------|----------|----------|------------|
| Density : $\rho$ [kg/m$^3$] | 818 | 915 | 955 | 960 | 965 |
| Kinematic viscosity : $\nu$ [m$^2$/s] | $1 \times 10^{-6}$ | $5 \times 10^{-6}$ | $30 \times 10^{-6}$ | $50 \times 10^{-6}$ | $100 \times 10^{-6}$ |
| Surface tension coefficient : $\sigma$ [N/m] | $16.9 \times 10^{-3}$ | $19.7 \times 10^{-3}$ | $20.7 \times 10^{-3}$ | $20.8 \times 10^{-3}$ | $20.9 \times 10^{-3}$ |
| Contact angle : $\theta$ [°] | 0 | 0 | 0 | 0 | 0 |
| Particle distance : $d$ [mm] | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |
| Number of particles : $N$ | 58858 | 58858 | 58858 | 58858 | 58858 |
| Influence radius : $r_e$ [mm] | $3.1d$ | $3.1d$ | $3.1d$ | $3.1d$ | $3.1d$ |

Table 3 Physical properties of the oil at 25 °C.

| Oil type | (a) 1 cs | (b) 5 cs | (c) 30 cs | (d) 50 cs | (e) 100 cs |
|----------|----------|----------|----------|----------|------------|
| Density : $\rho$ [kg/m$^3$] | 818 | 915 | 955 | 960 | 965 |
| Kinematic viscosity : $\nu$ [m$^2$/s] | $1 \times 10^{-6}$ | $5 \times 10^{-6}$ | $30 \times 10^{-6}$ | $50 \times 10^{-6}$ | $100 \times 10^{-6}$ |
| Surface tension coefficient : $\sigma$ [N/m] | $16.9 \times 10^{-3}$ | $19.7 \times 10^{-3}$ | $20.7 \times 10^{-3}$ | $20.8 \times 10^{-3}$ | $20.9 \times 10^{-3}$ |
4. Numerical calculation results and discussion

4.1 Comparison of oil behavior

The specifications of the computer used for this study and the calculation time are listed in Table 4. Figures 5, 6, and 7 show comparisons of oil behavior at revolving speeds of 250, 1000, and 1750 rpm, respectively. In Fig. 5, the case of 1 cs is not churned, whereas the cases of 5 cs, 30 cs, 50 cs, and 100 cs adhere to the cam-shaft and are churned owing to their viscosities. The inhomogeneous adherence of the cam-shaft surface is generated by the effect of surface tension, and this behavior can be reproduced. Figures 6 and 7 show that a larger amount of oil is churned up because of the revolution of the cam-shaft at 1000 and 1750 rpm, respectively. Also, we observe that the oil flow is irregular on the inner wall of the case owing to the surface tension. The typical streaky patterns of churned-up oil are observed on the case inner wall at 1000 rpm. As shown in Figs. 5, 6, and 7, the numerical simulation results are in good agreement with the experimental results with respect to the remaining oil level and the shape of the oil surface at the respective revolving speeds for the 30 cs, 50 cs, and 100 cs cases. In contrast, the amount of churned-up oil is smaller than that of the experiment when the revolving speed is increased for the 1 cs and 5 cs cases. In particular, the amount of oil churned up by the cam-shaft revolving at 1750 rpm is substantially smaller compared with the experimental results. In this calculation, the roughness of the cam-shaft surface is not taken into consideration. Therefore, one reason for the substantially smaller quantity of the oil churned up in the 1 cs and 5 cs cases is thought to be the effect of the roughness of the cam-shaft surface.

Table 4 Specifications of the computer and computation time.

| Specifications of the computer and computation time. |
|-----------------------------------------------------|
| OS | Windows 8 (64 bit) |
| CPU | Intel (R) Core i7 4700HQ 2.40 GHz (4 core) |
| RAM | 16.0 [GB] |
| Computation time | Approximately 8 ~ 12 [hours] |

(a) Experimental results

(b) Calculation results

Fig. 5 Comparison of oil behavior at 250 rpm.
Fig. 6  Comparison of oil behavior at 1000 rpm.

Fig. 7  Comparison of oil behavior at 1750 rpm.
4.2 Comparison of torque

Figure 8 shows the comparison of the torque at each revolving speed. The torque is calculated as the average of 5 revolutions after reaching a steady state. The figure shows that the calculation results are in good agreement with the experimental results in the 1 cs, 5 cs, and 30 cs cases. However, the torque at the revolving speed of 1000 rpm or higher is lower than that of the experiments, and the difference between the calculation and the experiment increases as the revolving speed increases. One of the reasons for this difference is the insufficient particle size in the numerical simulation to analyze the atomization of intensely churned oil droplets as the revolving speed increases. If the number of particles is insufficient, the velocity distribution of the churned-up oil cannot be accurately calculated near the cam-shaft, which affects the viscosity term. Therefore, the torque caused by viscosity is considered to be lower at the revolving speed of 1000 rpm or higher. In addition, the calculation results of the 50 cs and 100 cs cases greatly deviate from the experimental results. Hence, it is found that the difference between the calculation results and the experimental results increases with increase in the viscosity and revolving speed. Figure 9 shows the distribution of the particle number density on the cam-shaft at 1750 rpm. In Fig. 9, red indicates that the particle number density is high. Oil is kept in contact with the cam-shaft where the color is red. It can be clearly seen from Fig. 9 that in the 50 cs case, the contact area between the cam-shaft and the oil is less than that of the 5 cs case. The resistance force acting on the cam-shaft can be thought to decrease where the contact is lost. In high-viscosity oil, the oil level decreases when the revolving speed increases because a large amount of oil adheres to the cam-shaft and is churned up. Accordingly, the contact area between the cam-shaft and the oil is reduced. Thus, one reason for the difference is that the cam-shaft is idling in the high-viscosity calculation. Moreover, as shown in Figs. 5, 6, and 7, the generation of micro air bubbles can be observed in the experiment. If the viscosity is low, the effect of the micro air bubbles can be ignored because the revolving speed is sufficiently lower than the threshold value at which the micro air bubbles stay in the oil, and the stirring resistance force is enhanced, as demonstrated by Leprince et al. (2009). However, when the viscosity of the oil increases, removing the micro air bubbles from the oil is difficult. Moreover, the contact area with the cam-shaft is increased because of an increase in the oil level resulting from the effect of the micro air bubbles. Hence, it is necessary to consider the effect of the micro air bubbles. The torque increases because of an increase in the stirring resistance force, resulting from the effect of micro air bubbles in the 50 cs and 100 cs cases.

Fig. 8  Comparison of torque in each revolving speed.

Fig. 9  Distribution of particle number density on cam-shaft at 1750 rpm.
Figure 10 shows the respective contributions of viscosity, pressure, and surface tension to the torque at 1750 rpm. The figure clearly shows that the torque generated by viscosity is dominant. Although the effects of the pressure and surface tension are small, it can be seen that the influence of pressure tends to increase as the oil viscosity decreases. The torque oscillation is shown in accordance with each revolving cycle. In addition to this oscillation, it can be noted that an oscillation with a higher frequency is superimposed on the torque. In the moving particle semi-implicit (MPS) method, the particle number density is used as the source term of the pressure Poisson equation. Since it greatly changes spatially and/or temporally owing to the Lagrangian motion of the particles, a pressure oscillation is generated. The pressure oscillation is non-physical and a problem specific to the particle method. In this study, the calculation is performed using an improved formulation of the pressure Poisson equation (Kondo and Koshizuka, 2008) for suppressing the oscillation in the MPS method. However, further improvement is required; in particular, when the numerical simulation is conducted using polygon walls (Harada et al., 2008), suppressing the pressure oscillation is difficult because of the density errors near the wall boundary with a large curvature. Therefore, further stabilization of the pressure calculation in the MPS method is a future issue.

5. Sensitivity analysis

In the numerical simulation of the torque generated by the cam-shaft revolution, Yuhashi et al. (2015) reported the sensitivity analysis with respect to the contact angle, kinematic viscosity coefficient, and surface tension coefficient. The report demonstrates that the sensitivity to the contact angle and surface tension is small, and the influence of viscosity is large.

In general, if the grid increment is insufficient in the numerical simulation, a phenomenon cannot be properly reproduced. Hence, to obtain a calculation result with a quantified accuracy, sufficient spatial resolution must be set. For example, in FDM (finite difference method) or FVM (finite volume method), which use a mesh, the mesh dependency has been eliminated as the convergence solution. In the particle method, the particle size should be considered for the spatial resolution. Thus, sensitivity analysis is conducted by focusing the number of particles as an index that represents the spatial resolution. Here the revolving speed is set at 1750 rpm, in which the difference between the calculation results and the experimental results was the largest. At this setting, the oil level is represented by an input variable $V'$. 

Fig.10 Contribution of viscosity, pressure gradient, surface tension and total torque at 1750 rpm.
Figure 11 shows the comparison of the torque with different particle numbers at 1750 rpm. As shown in the figure, although the torque monotonically increases with an increase in the number of particles in the 1 cs and 5 cs cases, it can be seen that further fine particles are required because the convergence is insufficient. On the other hand, in the 30 cs case, the torque increases with an increase in the number of particles, and in this case, the result converges to the experimental result. However, in the 50 cs and 100 cs cases, it can be seen that the torques indicate decreasing tendencies with an increase in the number of particles. From the above results, a rigorous quantitative evaluation cannot be performed regarding the convergence of numerical calculations using the GCI method (ASME, 2009) except for the case of 30 cs.

Figure 12 shows comparison of the torque for each kinematic viscosity at 1750 rpm. From this figure, it can be seen that the calculation and experimental results are in good agreement in the 1 cs, 5 cs, and 30 cs cases. From this result, the accuracy of the spatial discretization of the viscosity term has been verified. In contrast, the calculation results greatly deviate from the experimental results in the 50 cs and 100 cs cases. This difference is due to idling the cam-shaft, and there is not the influence of the accuracy of the spatial discretization of the viscosity term. Although the viscosity of the oil is increased, the torque value is kept constant because of idling the cam-shaft. The torque for 30 cs is highly reliable since the calculation result shows the convergence, and it can be determined that a sufficient number of particles are ensured. Furthermore, the convergence of the calculation results was estimated using the GCI method. The difference between the convergence value and the experimental result is approximately 6% in this study.

Figure 13 shows the relation between the ratios of the lubricating oil and torque. In this figure, we observe that the torque increases monotonically in line with an increase in the oil level, and the higher the viscosity becomes, the larger the increase rate. If micro air bubbles are mixed in the oil, the apparent viscosity decreases, and it is considered that the contact area between the oil and the cam-shaft increases. However, the torque in the 100 cs case is approximately 70%
of the experimental result, even if the amount of oil increases by 20%.

Figure 14 shows the comparison of oil behavior with different number of particles at 1750 rpm. As shown in Fig. 14, the calculation result using fine particles shows that the oil is churned up more in the 5 cs case, and the oil behavior is close to the experimental result. However, the difference caused by the particle size change is not observed in the 50 cs case. Figure 15 shows the comparison of the torque with different numbers of particles. As shown in Fig. 15, the contribution of viscosity increases when the number of particles increases in the case of 5 cs. However, it decreases when the number of particles increases in the 50 cs case. If a calculation of higher resolution is conducted, the churning of the oil increases in the case of 50 cs. Therefore, the cam-shaft is idling since particles acting on the cam-shaft are decreased.

The above results indicate that in the 1 cs, 5 cs, and 30 cs cases, the calculation of higher resolution is required in a high-revolving range. Moreover, the torque is underestimated due to the idling of the cam-shaft, because churning of the oil increases when the revolutions increase and the resistance force on the cam-shaft decreases in the 50 cs and 100 cs cases.

![Comparison of oil behavior with different number of particles at 1750 rpm.](image)

![Comparison of the torque with different number of particles at 1750 rpm.](image)

Based on the above discussion, it is noted that the effect of the micro air bubbles can be ignored in the 1 cs, 5 cs, and 30 cs cases because the major impact is from the spatial resolution. On the other hand, the increase of stirring resistance force resulting from the micro air bubble generation cannot be ignored in the 50 cs and 100 cs cases, which makes the effect of micro air bubbles an important factor.

6. Conclusion

In this paper, the oil behavior and the torque in a case were calculated using the MPS method. Additionally, the validation of the calculation results was conducted to take advantage of the numerical simulation in the product development. The conclusions are as follows:

1. In the case of the oil of 1 cs, 5 cs, and 30 cs, the calculation results of the torque values were in good agreement with those of the experimental results. However, in the case of the oil of 50 cs and 100 cs, the calculated torques values were lower than those of the experiment.
Although the calculated torque values showed good correspondences in the 1 cs and 5 cs cases, the oil behavior was not reproduced. From the sensitivity analysis results, the calculated torque values show an increasing tendency by higher resolution. Moreover, it was confirmed that maintaining sufficient resolution for calculations in a high-revolving range is necessary because oil droplets are atomized. If viscosity is very low, it is thought that not only the resolution, but also the roughness of the cam-shaft surface need to be taken into consideration.

In the 30 cs case, the calculation results were in good agreement with the experimental results. That is, both the oil behavior and the torque showed good results, and the phenomena were reproduced by the MPS method. From the sensitivity analysis results, it can be concluded that the calculation with high-resolution secures sufficient spatial resolution and the calculation result was highly reliable. Furthermore, the convergence of the calculation results was estimated using the Grid Convergence Index (GCI) method. The difference between the convergence value and the experimental result was approximately 6% in this study.

The oil behavior was properly reproduced in the 50 cs and 100 cs cases, however, the calculated torque values showed great deviations from the experimental results, and the difference increased with increase in the revolving speed. Also, the torque value showed a decreasing tendency in high-resolution, and the viscosity torque decreased. One reason for this is that for the calculation with high-resolution, more oil is churned up and the resistance force on the cam-shaft decreases because of the cam-shaft idling.

In the 50 cs and 100 cs cases, the reason why the calculated torque value estimates are smaller than the experimental results is not considered to be the effect of an apparent increase of the oil level due to the micro air bubbles generated in the oil. The main reason for the results is that the stirring resistance force increases due to the generated micro air bubbles.

The quantitative evaluation method suggested in ASME V&V 20 regarding the numerical solution convergence could not be applied to the cases other than 30 cs.

In future, for industrial application in various fields, the prediction and evaluation at a higher revolving range are considered to be significant. In particular, since the effects of micro air bubbles cannot be ignored at 4000 rpm or higher revolving range, or for viscous oils, the development of an air bubble model is expected.

References

Al-Shibl, K., Simmons, K., Eastwick, C. N., Modelling windage power loss from an enclosed spur gear, Journal of Power and Energy, Proceedings of the Institution of Mechanical Engineers (2007), pp.331–341.

Arai, J., Koshizuka, S. and Murozono, K., Large eddy simulation and a simple wall model for turbulent flow calculation by a particle method, International Journal for Numerical Methods in Fluids, Vol.71 (2013), pp.772–787.

Arisawa, H., Nishimura, M., Imai, H., Tanaka, K. and Goi, T., CFD simulations for reduction of oil churning loss and windage loss on aeroengine transmission gears, Proceedings of ASME Turbo Expo (2009), pp.62–72, Paper No. GT2009–59226.

Arisawa, H., Nishimura, M., Imai, H., Tanaka, K. and Goi, T., CFD simulations and experiments for reduction of oil churning loss and windage loss on aeroengine transmission gears, Transaction of the JSME (in Japanese), Series C, Vol.79, No.800 (2013), pp.1213–1225.

Arisawa, H., Nishimura, M., Tanaka, K., Imai, H. and Goi, T., Oil scavenging CFD simulations and experiments in the accessory gearbox of an aircraft engine, Proceedings of IGTC (2011).

Chikazawa, Y., Koshizuka, S. and Oka, Y., Numerical analysis of sloshing with large deformation of elastic walls and free surfaces using MPS method, Transaction of the JSME (in Japanese), Series B, Vol.65, No.637 (1999), pp.2954–2960.

Chikazawa, Y., Koshizuka, S. and Oka, Y., Numerical analysis of three-dimensional sloshing in elastic cylindrical tank using moving-particle semi-implicit method, Computational Fluid Dynamics Journal, Vol.9 (2001), pp.376–383.

Harada, T., Koshizuka, S. and Shimazaki, K., Improvement of wall boundary calculation model for MPS method, Transaction of the JSCE (in Japanese) (2008), Paper No.20080006.

Hill, M. J., Kunz, R. F., Noack, R. W., Long, L. N., Morris, P. J. and Handschuch, R. F., Application and validation of unstructured overset CFD technology for rotorcraft gearbox windage aerodynamics simulation, Proceedings of American Helicopter Society 64th Annual Forum (2008).
Hirt, C. and Nichols, B. D., Volume of fluid method for the dynamics of free boundaries, Journal of Computational Physics, Vol. 39 (1981), pp. 201–225.

Imai, H., Goi, T., Arisawa, H. and Nishimura, M., Reduction in gear windage and churning loss by optimum shroud design with aid of CFD, Proceedings of JSME International Conference on Motion and Power Transmissions (2009).

Kondo, M., Koshizuka, S. and Takimoto, M., Surface tension using inter-particle potential force in moving particle semi-implicit model, Transaction of the JSCSES (in Japanese) (2007), Paper No.20070021.

Kondo, M. and Koshizuka, S., Improvement of stability in moving particle semi-implicit method, International Journal for Numerical Methods in Fluid, Vol.65 (2011), pp.638–654.

Koshizuka, S., Go, S., Oka, Y. and Tanizawa, K., Numerical analysis of two-dimensional experiment of shipping water on deck using a particle method, Proceedings of ASME Heat Transfer/Fluid Engineering Summer Conference (2004).

Koshizuka, S., Kaito, S., Kikuchi, Y., Kujime, M., Ishiba, Y. and Horiguchi, A., Development of a deaeration model for stirred tank analysis in moving particle simulation method, 4th International Conference on Particle-Based Methods. Fundamentals and Applications (PARTICLES 2015), a416.

Koshizuka, S., Nobe, A. and Oka, Y., Numerical analysis of breaking waves using the moving particle semi-implicit method, International Journal for Numerical Methods in Fluids, Vol.26 (1998), pp.751–769.

Koshizuka, S., Tamako, H. and Oka, Y., A particle method for incompressible viscous flow with fluid fragmentation, Computational Fluid Dynamics Journal, Vol.4 (1995), pp.29–46.

Koshizuka, S. and Oka, Y., Moving particle semi-implicit method for fragmentation of incompressible fluid, Nuclear Science and Engineering, Vol.123 (1996), pp.421–434.

Leprince, G., Changent, F., Ville, P. and Jarnias, F., Influence of oil aeration on churning losses, Proceedings of Japanese Society of Mechanical Engineers International Conference on Motion and Power Transmissions (2009), pp.463–468.

Li, L., Henk, K. V., Graham, K. H., Theo, P. and Chris, H., Numerical investigation on fluid flow of gear lubrication, SAE Technical paper (2008), No.08SFL-0432.

Murotani, K., Koshizuka, S., Tamai, T., Shibata, K., Mitsune, N., Yoshimura, S., Tanaka, S., Hasegawa, K., Nagai, E. and Fujisawa, T., Development of hierarchical domain decomposition explicit MPS method and application to large-scale tsunami analysis with floating objects, Journal of Advanced Simulation in Science and Engineering, Vol.1, No.1 (2014), pp.16–35.

Muto, K., Sakai, I., Ozaki, N., Prediction of the fluid resistance in stirred fluid using the particle method, Proceedings of JSAE Annual Congress (2010), No.23–10 , pp.1–5.

Naito, S. and Sueyoshi, M., A numerical analysis of violent free surface flow flood car deck using particle method, Proceedings of 5th International Workshop Stability and Operation Safety of Ships (2001), pp.12–13.

Nannichi, Y., Murotani, K., Koshizuka, S., Nagai, E., Fujisawa, T. and Anju, A., Three-dimensional flooding analysis in the turbine building of Fukushima Dai-ichi nuclear power station by the tsunami of great east japan earthquake using particle method, Annual Meeting of the Atomic Energy Society of Japan (2016) IB15.

Nomura, K., Koshizuka, S., Oka, Y. and Obata, H., Numerical analysis of droplet breakup behavior using particle method, Journal of Nuclear Science and Technology, Vol.38 (2001), pp.1057–1064.

Osher, S. and Fedkiw, R., Level set methods and dynamic implicit surfaces, Springer (2003).

Shibata, K., Koshizuka, S., Go, S., Oka, Y. and Tanizawa, K., A three-dimensional numerical analysis code for shipping water on deck using a particle method, Proceedings of ASME Heat Transfer/Fluid Engineering Summer Conference (2004a).

Shibata, K., Koshizuka, S. and Oka, Y., Numerical analysis of jet breakup behavior using particle method, Journal of Nuclear Science and Technology, Vol.41 (2004b), pp.715–722.

Shibata, K. and Koshizuka, S., Numerical analysis of shipping water impact on a deck using a particle method, Ocean Engineering, Vol.34 (2007), pp.585–593.

Shirakawa, N., Horie, H., Yamamoto, Y., Okano, Y. and Yamaguchi, A., Analysis of jet flows with the two-fluid particle interaction method, Journal of Nuclear Science and Technology, Vol.38 (2001), pp.729–738.

Sueyoshi, M. and Naito, S., A study of nonlinear fluid phenomena with particle method (Part1)-Two dimensional problems, Journal of the Kansai Society of Naval Architects, No.236 (2001), pp.191–198 (in Japanese).

Tan, N., Aoki, T., Inoue, K. and Yoshitani, K., Numerical simulation of two-phase flow driven by rotating object, Transaction of the JSME (in Japanese), Series C, Vol.77, No.781 (2011), pp.1699–1714.
The American Society of Mechanical Engineers (ASME), Standard for Verification and Validation in Computational Fluid Dynamics and Heat Transfer (2009), ASME V&V 20-2009.

Yamamoto, Y., Sato, T. and Anraku, G., Dynamic simulation of water and soil using particle method, Transaction of the JSAE (2011), Paper No.20119563.

Yuhashi, N., Matsuda, I. and Koshizuka, S., Calculation and evaluation of torque generated by the rotation flow using moving particle semi-implicit method, Transaction of the JSCSES (in Japanese) (2015), Paper No.20150007.