Lubrication mechanism in gearbox of high-speed railway trains

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Abstract
In this study, the moving particle semi-implicit (MPS) method is employed for numerical simulation of flow field in gearbox of high-speed railway trains so as to understand the lubrication mechanism and mode in the gearbox during high-speed operation. A high-fidelity 3D model for a high-speed train gearbox under actual working conditions is created for the first time. RecurDyn and ParticleWorks co-simulation is conducted to acquire the flow field distribution in the gearbox under the coupling of multi-fields and to calculate the churning losses. Effects of key parameters including rotation speed, viscosity, and immersion depth on the churning power losses of the gearbox are investigated to determine their influences on the lubrication performance of the gearbox. Those results form a theoretical base for futuristic optimization of the high-speed train gearboxes.

Keywords: High-speed railway train gearboxes, Mesh-free moving particle semi-implicit method, Immersion depth, Rotation speed, Lubricant viscosity, Churning power loss

1. Introduction

Lubrication of gearbox has been studied by many investigators using experimental, analytical, and numerical methods. Changenet and Velex (2006) derived a series of formulae that can accurately predict the churning losses for one pinion characteristic of automotive transmission geometry based on experimental analysis over a wide range of speeds, gear geometries, lubrications, and immersion depths. Laruelle et al. (2017) provided a wide variety of experimental tests on churning losses to find out the impact of speed, lubricant, temperature, and gear geometry on splash lubrication power losses. An extended equation was then developed from the experimental results to estimate churning losses of bevel gears. Leprince et al. (2012) set up a test rig to measure the quantity of lubricant splashed at different locations on the casing walls and deduced a set of formulae based on the experimental results to predict the lubricant flow rate generated by one spur gear at various places on the casing. Chen and Matsumoto (2016) performed oil churning loss experiments to clarify the influence of the relative position of gears and casing wall shape of gearbox under splash lubrication condition. They concluded that in addition to the well-known factors, relative position, casing wall shape, and the steady-state oil surface also had significant effects on the churning loss. Liu et al. (2012) suggested a new lubricant and measured the transmission efficiency, bulk temperature, and gear tooth scuffing at four different immersion depths. A range of reasonable immersion depths was obtained from that research. Neurouth et al. (2017) also followed an experimental approach to decide if splash lubrication technique is worth considering for high-speed gears (i.e. for tangential gear speed up to 60 m/s). The researchers used a specific test rig to operate a single spur or helical gear in various operation conditions...
conditions and eventually concluded that the clearance between the tested gear and the flanges significantly affected the churning losses and oil aeration.

However, due to the limitations of present experimental techniques, it is still extremely difficult to observe the instantaneous speed and direction of splash of the lubricant oil in the gearbox. Experimental analysis of internal flow field in gearboxes with complicated configurations or in high-speed stages is almost impossible.

Inadequacies and limitations in our understanding of gearbox lubrication under the conditions of the experiments can be overcome by using numerical simulation methods. Nowadays, there are five popular numerical methods that are being used for studying the lubrication mode of gearbox. Those methods include the finite volume method (FVM), the finite differential method (FDM), the finite element method (FEM), the particle methods, and the lattice Boltzmann method (LBM). In particular, common particle methods include the MPS method, the smoothed particle hydrodynamics (SPH) method, and the finite volume point (FVP) method. Liu et al. (2018) applied FVM to develop a computational fluid dynamics (CFD) model for a dip-lubricated planetary test gearbox and used that model to obtain a precise description of oil distribution within the gearbox through simulations. In order to further reduce the computational effort required for CFD simulations, Concli and Gorla (2017) presented an automated mesh-partitioning strategy and used it to calculate the churning power losses of a planetary gearbox with a very complicated configuration and kinematic arrangements through an efficient CFD simulation. Hu et al. (2019) developed a CFD model for a gearbox with spiral bevel geared transmission and used that model to find out how the rotation speed and the oil fill level of gears, the dynamic viscosity and density of the oil, and the helicopter tilt angle impact the churning power losses. The developed numerical model was validated by experimental results. Renjith (2015) employed the FVM method to study the lubrication of the differential gear system of vehicles through multiphase CFD analysis. The numerical results showed the total power consumption by the differential system and the affecting factors, as well as the lubrication flow characteristics. Fernandes et al. (2018) put forward a hybrid polymer gear concept that integrated metal inserts into its polymer gear teeth to achieve an enhanced thermal behavior. Materials and geometries of the inserts were selected based on their influences on the weight of the inserts and the maximum operation temperature of the surface of the polymer gear teeth through FEM simulations. Rama and Ganasekar (2019) used a validated 3D FEM model to explore the magnitude and location of stress intensity factors under four different load cases, and calculate the load distribution ratio of a contact point along the contact line and load sharing ratio between the pairs in contact. The numerical results provided valuable guidelines for predicting the crack propagation and lifetime of spur gear drives. Ji et al. (2018) applied a mesh-free SPH method for modeling and simulations of oil flow inside a gearbox. A multiphase SPF formulation was used to resolve the complex multiphase flow and obtain the flow field behavior, the quantity and size of bubbles generated due to the rotation of the gears, and the velocity field and profile beneath the oil surface. The numerical results were in comparatively good agreement with the experimental particle image velocimetry results. Imin and Geni (2014) also established SPH discrete equations and carried out numerical simulations to capture the stress variation, distribution, and propagation on the tooth profile surface during a gear meshing impact process. The numerical model and the SPH method were validated by comparing the numerical results with the experimental data. Pichandi and Anbalagan (2018) proposed an effective numerical approach for solving the natural convection in a 2D square enclosure based on LBM. The simulation results depicted the natural convection heat transfer and fluid flow behavior in a 2D square enclosure mounted with various sizes of blockage ratios and imposed sinusoidal wave on both vertical walls. Authors of this paper also followed a combined experimental-computational approach to design roller enveloping hourglass worm gears with high precision of transmission and reduced backlash (Deng et al., 2019a, 2019b, 2019c, 2019d). FEM was used to create computational models for the developed worm gears and simulations were performed to find out how key parameters affect meshing characteristics of the worm gears.

From the review of previous studies, it can be seen that those numerical methods are effective in design and analysis of gear system while each method has its own applications. FVM has been mainly used for simulation of the flow field in the gearbox; FEM is a powerful tool for problems on structure and mechanics of gears; while compared with FEM and FVM, LBM and FDM are seldom used for gear simulations.

Different from above methods, the particle methods do not need explicit surface tracking by a mesh or a scalar quantity. In those methods a continuum is discretized by a discrete number of particles without mesh constraints. Each particle moves accordingly with its own mass, density, velocity, and the external/internal forces applied to it. Compared with the traditional numerical methods (FEM, etc.), the particle methods can solve more complex geometry and physics problems including large deformation and damage problems. Therefore, those methods are powerful tools for solving

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incompressible flows and can be used for simulation of the flow field in gearboxes. Accuracy of the particle methods can be easily controlled by altering the number of particles. Among the typical particle methods (SPH, MPS, and FVP), the SPH method approximates function derivatives based on a kernel derivative, it is more efficient by following an explicit prediction-correction process but its accuracy and stability are lower than the other two methods (Koshizuka and Oka, 1996; Khayyer and Gotoh, 2011; Liu et al., 2018). Both the MPS and FVP method approximate the function derivatives by weight average based on the kernel function. The fundamental difference between these two methods is in the FVP method, the governing equations are discretized following the approach used in the FVM (Liu et al., 2017, 2019).

The MPS method was developed for incompressible flow with free surfaces (Koshizuka et al., 1995) and is chosen for this study to model and simulate the flow of lubricant inside the gearbox of high-speed railway trains. Li et al. (2018) applied the MPS method to calculate the churning losses of a single helical gear in a transmission system. The simulation results agreed very well with the experimental results. In their study, the influence of rotation of bearings on the internal flow field was not considered. Other than that, little research appears to have been done to analyze the churning losses of the gearbox using the MPS method.

Gearbox is a core component of the high-speed railway trains; its performance directly determines the safety and quality of those trains. For the purpose of optimizing and designing this component to achieve higher performance, it is imperative to understand its lubrication mechanism. In this paper, a 3D model for the high-speed train gearbox is first presented; next, computational analysis on its churning losses use the MPS method is described and the lubrication mechanism of the gearbox is then unraveled. This paper is organized in the following manner: In Section 2, we give a brief review of the MPS method. In Section 3, the procedure of establishing and simplifying the 3D gearbox model is described. In Section 4, we present simulation results of the lubrication performance of our model and in Section 5 we discuss the influence of different parameters on this performance. Finally, in Section 6, we conclude and give some remarks on our future work.

2. MPS theory

2.1 Governing equations

The MPS method was proposed by Koshizuka and Oka [20] in 1996. This method is a mesh-free particle method for analyzing incompressible free surface flow, which discretized the incompressible Navier-Stokes equation using a Taylor series. The governing equations of that method are the equation of continuity and the incompressible Navier-Stokes equation, which can be expressed as:

Equation of continuity (law of mass conservation)

$$\frac{\partial \rho}{\partial t} = 0$$

(1)

Navier-Stokes equation (law of momentum conservation)

$$\frac{D u}{D t} = -\frac{\nabla P}{\rho} + \nu \nabla^2 u + g$$

(2)

where $\frac{D}{D t}$ denotes the Lagrangian time derivative, $\rho$ is the density, $u$ is the velocity, $P$ is the pressure, $\nu$ is the kinetic viscosity, and $g$ is the gravity.

2.2 Algorithm

In the Navier-Stokes equation, the pressure term is implicitly calculated and other terms are explicitly calculated by the Laplacian model.

Implicit calculation of the pressure term

$$\nabla^2 P_{n+1} = \frac{\rho}{\Delta t^2} \frac{n^*-n^0}{n^0}$$

(3)

Explicit calculation of other terms

$$\frac{u^*-u^k}{\Delta t} = \nu \nabla^2 u^k + g$$

(4)
In the correction step, velocity and position correction are calculated by considering the pressure gradient

\[
\frac{u_{k+1} - u^*}{\Delta t} = -\frac{\nabla p_{k+1}}{\rho}
\]  

(5)

In above equations \( n \) is the particle number density, \( n_0 \) represents its initial value, superscript “k” is the designation of timestep, and superscript “*” indicates a physical quantity at a stage where the explicit calculation has been completed.

### 2.3 Effective radius and kernel function

In the MPS method, a particle interacts with neighboring particles in an interaction zone covered with a kernel function (Koshizuka and Oka, 1996). If the distance between two particles is larger than the interaction zone radius, the kernel function is set to zero. That radius is called the effective radius and all the particles within that radius are adjacent particles (Fig. 1). The effective radius \( (r_e) \) of a particle is taken to be 2 to 4 times of the particle diameter so that the number of particles within \( r_e \) is kept constant.

Fig. 1 Adjacent particles (red) around the particle (olive green) within the effective radius

The kernel function (or weight function) commonly used in the MPS method is (Fig. 2)

\[
w(r_{ij}) = \begin{cases} 
\frac{r_e}{r_{ij}} - 1 & (r_{ij} \leq r_e) \\
0 & (r_{ij} > r_e)
\end{cases}
\]  

(6)

where \( i \) and \( j \) are particle numbers, \( r_{ij} \) is the distance between particle \( i \) and \( j \) \( (r_{ij} = |r_j - r_i|) \), and \( r \) represents the particle position vector.

Fig. 2 Kernel function

### 2.4 Particle number density and collision

Particle number density at the position of the particle \( i \) is defined as the sum of the kernel function

\[
n_i = \sum_{j \neq i} w(r_{ij})
\]  

(7)

In an incompressible state, the initial particle number density \( n_0 \) is a constant. Fig. 3 shows a particle arrangement in
initial state, the particles are arranged in an orthogonal lattice pattern with an initial inter-particle distance.

The force acting on particles resulting from particle-particle collision is divided into a normal force ($F_n$), a shear (tangential) force ($F_s$), and a drag force ($F_d$) as $F = F_n + F_s + F_d$. Fig. 4 is a force diagram when two particles collide.

Fig. 3 Initial arrangement of uniform particles

Fig. 4 Schematic showing the force between two colliding particles

A flowchart is created based on the MPS theory and specific requirements of MPS-based simulations of flow field and lubrication inside gearboxes. All the simulations involved in this study are conducted using RecurDyn/ParticleWorks, a software suite for liquid flow simulation. Readers can refer to (Prometech Software, 2016; FunctionBay Technical Support, 2016) for a full explanation of MPS algorithms.

3. Modeling and Simulation

3.1 Model of a high-speed train gearbox and simplification

A high-fidelity 3D model for the high-speed train gearbox is established using Unigraphics NX (Fig. 6). The high-fidelity model is then simplified based on the configuration and working principle of high-speed train gearbox to enhance computing efficiency while maintaining accuracy. As a mesh-free particle method, the MPS method does not require connection between nodes of the simulation domain (mesh), therefore all complex structural details inside the gearbox can be retained in the simplified model in order to achieve the best overall accuracy for the simulation. It should be mentioned that it may be especially difficult to create a useful mesh from the geometry of those details if using a mesh-based method such as FEM. As shown in Figs. 7 and 8, compared with the original models, only trivial structures and parts (chamfers, fillets, keyways, nuts and bolts, and parts of the shafts that extend outside the gearbox) are removed in the simplified models. Removal of those removed trivial structures and parts will not affect the simulation results. Fig. 9 illustrates the geometry and dimensions of the gearbox and key parameters of the gear pair are listed in Table 1.
Fig. 5 Flowchart of MPS-based simulations

Fig. 6 Exploded view of gearbox components: 1) Upper end cover of casing 2) casing 3) output shaft 4) output gear 5) tapered roller bearing of output shaft 6) output shaft seal ring 7) output shaft bushing 8) output shaft end cover 9) input shaft end cover 10) input shaft bushing 11) input shaft seal ring 12) tapered roller bearing of input shaft 13) input shaft 14) input gear (driving gear or pinion)
From Fig. 9 and Table 1, following parameters can be calculated:
Gear ratio = 69/29 = 2.37931
Pitch circle diameter (PCD) for pinion: 29 × 7 = 203 mm; PCD for gear: 69 × 7 = 483 mm
Distance between the center of input shaft and output shaft: D = (203 + 483)/2 = 343 mm
Tip circle diameter (TCD) for pinion: 203 + 2.0 × 7 = 217 mm; TCD for gear: 483 + 2.0 × 7 = 497 mm
Bottom circle diameter (BCD) for pinion: 203 – 2.5 × 7 = 185.5 mm; BCD for gear: 483 – 2.5 × 7 = 465.5 mm
Tooth height (TH) for pinion = (TCD(P) – BCD(P))/2 = (217 – 185.5)/2 = 15.75 mm; TH for gear = 15.75 mm

Maximum dimension of the gearbox (B) is 763 mm from Fig. 9
Maximum dimension of gear tips (L): L = (TCD(P) + BCD(G))/2 = (217 + 497)/2 = 363 mm
Clearance between the tooth tip of the gear and the gearbox (C): C = (B – L)/2 = (763 – 720) = 21.5 mm

3.2 Numerical simulation

The simplified gearbox assembly model is first read into Recurdyn, in which dynamic parameters are set. The overall simulation time is set as 5 s to eliminate the impact of sudden change of gear speed on the simulation results. In the simulation, 0 ~ 1s is an acceleration phase that represents the starting of the gear system and 1s ~ 5s is a uniform rotation phase. The addition of the acceleration phase will alleviate the influence of the drastic change of liquid on the simulation.

In particular, attempting to reflect the behavior of the gearbox in greater detail, the bearings are also considered in this simulation. We assume that during rotation of the shafts, the rollers of each bearing undergo pure rolling without sliding. For the bearings on both input and output shafts, their outer rings are fixed to the casing with zero rotation velocity, and the inner rings of the bearings rotate with their corresponding shafts at same velocities. Rollers geometrically contact with the raceway surfaces of the outer and inner rings at outer contact points and inner contact points, respectively. Due to the same reason, the tangential velocity of an individual roller at its outer contact point is zero and the velocity at its inner contact point is the same as the tangential velocity of the inner ring. Thus, the rotation and tangential velocities of each roller and the separator can be calculated from the rotation velocities of the shafts. The calculated rotation velocities are then applied to different components of the bearings as kinematic pairs (Fig. 10).

Next, fluid properties are set in ParticleWorks. In that software environment, an internal flow field in the simplified casing model is created; the simplified casing, input and output gears are represented by polygons; and the lubricant is modeled as a fluid represented as a collection of particles. Considering the limitation of computing capacity, the diameter of each lubricant particle is set as 2 mm, and an air density of 1.097 kg/m³ is used for this simulation.

A proper timestep is critical for running liquid flow simulations with ParticleWorks. The timestep should be set smaller than a certain critical value to make the calculation stable while a timestep much smaller than necessary will lead to a considerable waste of computation time. Eqn. (8) is employed to determine the initial timestep

\[ \Delta t = \min \left( \Delta t_{in}, \frac{C_{max}}{u_{max}} \frac{u_{max} l_0}{d \nu} \right) \]

where \( l_0 \) is the diameter of particle, \( u_{max} \) is the maximum velocity of the particles, \( d \) is the diffusion coefficient, \( v \) is the kinematic viscosity of the fluid (represented as particles) and \( C_{max} \) is its maximum value, \( C_{max} \) is the Courant coefficient (de Moura and Kubrusly, 2013) and is set as 0.2 for this simulation. \( \Delta t_{in} \) is the initial timestep, \( \frac{C_{max} l_0}{u_{max}} \) is the calculated step size based on the Courant-Friedrichs-Lewy (CFL) condition, and \( \frac{1}{2} \frac{d l_0}{\nu + \nu_{max}} \) is from the stability condition of viscosity calculation. That stability condition is only applied for explicit calculation.

Seven MPS models for the high-speed train gearbox are generated and simulated to investigate how different rotation
speeds, viscosities, and immersion depths affect and determine the lubrication performance of the gearbox. Parameters of the developed MPS models are displayed in Table 2. Model 1~3 are simulated to study the influence of the input gear’s rotation speed on the flow field inside the gearbox, where 2000 rpm, 4000 rpm, and 6000 rpm belong to low, medium, and high speed range for the gear, respectively. Models 1, 4, and 5 focus on the effect of the immersion depth on the internal flow field. Three depths, 1 h, 1.5 h, and 2 h are selected from the standard range of 0.5 h to 3 h. Model 1, 6, and 7 are used to compare the viscosity of lubricant on the churning losses. According to JB/T 8831-1999 (Chinese Industry Standard), the viscosity of the lubricant for high-speed train gearboxes should be above 100 cst at 40°C, therefore we select 100 cst, 120 cst, and 140 cst for comparison.

In order to verify that the results yielded from the MPS models with the 2 mm diameter particles are reliable, we reran the model 1 by reducing the particle diameter to 1 mm and the simulation results almost remained the same. However, if the particle diameter is 1 mm, the total amount of particles becomes 1,756,891, which almost reaches the capacity of our computer workstation. Furthermore, an experimental validation of the applied modeling and simulation approach is presented by Deng et al. (2020).

| Model | Viscosity of lubricant at 40°C [cst] | Rotation speed of input gear [rpm] | Immersion depth [h] | Initial timestep [s] | Particle diameter [mm] | Particle number |
|-------|---------------------------------|---------------------------------|--------------------|---------------------|------------------------|----------------|
| 1     | 120                             | 6000                            | 1.5                | 1.46                | 2                      | 213,322        |
| 2     | 120                             | 2000                            | 1.5                | 4.38                | 2                      | 213,322        |
| 3     | 120                             | 4000                            | 1.5                | 2.19                | 2                      | 213,322        |
| 4     | 120                             | 6000                            | 1                  | 1.46                | 2                      | 163,893        |
| 5     | 120                             | 6000                            | 2                  | 1.46                | 2                      | 263,024        |
| 6     | 100                             | 6000                            | 1.5                | 1.46                | 2                      | 213,322        |
| 7     | 140                             | 6000                            | 1.5                | 1.46                | 2                      | 213,322        |

Note: h is the tooth height and h = 15.75 mm.

4. Analysis of results

The impacts of various rotation speeds, viscosities, and immersion depths on the internal flow field and churning losses of the gearbox are discussed through analyzing the velocity distributions of the lubricant flow inside the gearbox, velocity vectograms, and distributions of particle number density of the lubricant. The lubricant particles adhered to the gear surfaces rotate with the gears, whose velocities can be calculated based on the rotation speed of the gears and the distances between the particles to the gear axes. However, most lubricant particles inside the gearbox move slowly, whose velocities are far lower than those of the adhered particles. If those high velocities were included in velocity field distribution plots, differences in the velocities of most lubricant particles that did not attach to the gear surfaces would be too small to be differentiated, so that the distribution of the entire velocity field could not be observed. In calibrating the velocity field distribution plots, we found that when the maximum plot scale limit was set as 1 m/s (Figs. 11, 13, 15), the differences in those velocities could be best plotted.

4.1 Influence of rotation speed

As can be seen from Figs. 11 and 12, at 2000 rpm, the lubricant is mainly concentrated on the bottom part of the output gear and only a small amount of the lubricant is splashed by that gear and reaches the meshing zone along the top wall of the casing. When the speed increases to 4000 rpm, the lubricant is forcedly diffused all over the casing by means of the gear rotation and the amount of the lubricant that is carried to the meshing zone obviously increases. In the high speed range (6000 rpm), the lubricant is entirely distributed inside the casing.
4.2 Influence of immersion depth

Figs. 13 and 14 show that the velocity field distributions at different immersion depths are very close but the amount of the splashed lubrication oil when the immersion depth is 2 h is clearly higher than the splashed oil amount when the immersion depth is 1 h and 1.5 h.
4.3 Influence of lubricant viscosity

The viscosity of the lubricant has a direct effect on its velocity. As shown in Fig. 15, as the viscosity increases, the velocity of the lubricant and the rate of its adhesion on internal walls of the casing decrease. On the other hand, the amount of the lubrication oil splashed by the output gear is less affected by its viscosity.

4.4 Analysis of churning power losses

Through postprocessing of the software, we calculated friction torques of the lubricated gears and shafts. By adding those torques together we could have the churning loss torque. The churning power losses then can be calculated based on that loss torque following an approach employed by Liu et al. (2017). Eqn. (9) explains how to calculate the churning power loss $P$. In that equation $T$ denotes the churning loss torque of gears, $n$ denotes the rotation speed, $T'$ is the churning loss torque of gear bearings, $n'$ is the estimated rotation speed of bearing. Subscript “1” means the driving gear, and “2” means the output gear. The unit of the torque is N·m and of the speed is rpm.
P = \frac{T_1 n_1 T_1' n_1' + T_2 n_2 T_2' n_2'}{9550} \quad (9)

Table 3 lists the calculated churning loss torques of different rotating parts and the overall churning power losses for all seven MPS models.

| Model | T_1 [N\cdot m] | T_2 [N\cdot m] | T_1' [N\cdot m] | T_2' [N\cdot m] | P [kW] |
|-------|----------------|----------------|----------------|----------------|-------|
| 1     | 0.055          | 1.430          | 0.00232        | 0.0225         | 0.4159|
| 2     | 0.001          | 0.642          | 0.000123       | 0.0138         | 0.0573|
| 3     | 0.026          | 1.090          | 0.00114        | 0.0195         | 0.2048|
| 4     | 0.024          | 0.848          | 0.00130        | 0.0117         | 0.2410|
| 5     | 0.239          | 2.091          | 0.00690        | 0.0435         | 0.7103|
| 6     | 0.085          | 1.339          | 0.00299        | 0.0239         | 0.4111|
| 7     | 0.068          | 1.460          | 0.00397        | 0.0308         | 0.4336|

Those results are also displayed in Figs. 17 to 20. In those figures T_1 is represented in blue, T_2 is represented in orange, T_1' is represented in gray, and T_2' is represented in red.

From Fig. 17 it can be found that the overall churning power loss increases with the increasing rotation speed and the proportions of the churning power loss of the output gear in the entire power loss remain consistent, which are 98.55%, 93.73%, and 90.80% for the rotation speed of 2000 rpm, 4000 rpm, and 6000 rpm, respectively. Fig. 18 shows that the overall churning power loss increases as the immersion depth increases, and the proportion of the power loss of the output gear slightly reduces. From the results in Table 3, it can be calculated that when the immersion depth is 1 h, 1.5 h, and 2 h, this proportion becomes 92.93%, 90.80%, and 77.75%, respectively. It can be seen from Fig. 19 that as the lubricant viscosity increases, both the total churning loss and the proportion of the power loss of the output gear only slightly increase. When the viscosity is 100 cst, 120 cst, and 140 cst, this proportion is calculated as 86.01%, 90.08%, and 88.92%, respectively.

Fig. 17 Churning power losses at different rotation speeds
Fig. 18 Churning power losses at different immersion depths

Fig. 19 Churning power losses from different lubricant viscosities
Fig. 20 Comparison of churning power losses of the high-speed train gearbox

Fig. 20 is a histogram that clearly shows the effects of different parameters on the churning losses. From that figure, it can be seen that the rotation speed and immersion depth evidently affect the churning losses while the influence of the lubricant viscosity is little. Weight percentage of effect of individual parameter on the churning power losses cannot be determined using the present method and ParticleWorks. In order to precisely identify the contribution of each parameter on the change of the churning losses, new numerical method is needed.

From Figs. 17 to 20, it can also be deduced that the most part of lubricant churning loss comes from the losses of the input and output gears and the losses of the bearings are very small. It needs to be mentioned that the results listed in Table 3 and plotted in Figs. 17 to 20 are average values calculated based on instantaneous values. Therefore, in following Figs. 21 to 23, we present the churning loss torques for the input and output gears under different conditions to visually show the influence of the three factors on the churning losses. From those figures, instantaneous values for the churning loss torques can be extracted. From Fig. 21, it can be seen that the churning loss torque of the input gear is very low when the rotation speed is 2000 rpm, while as the speed increases to 4000 rpm and 6000 rpm that torque increases obviously with serious oscillations. The same phenomenon can be observed from the output gear.

Fig. 22 shows that when the immersion depth is 1 h and 1.5, the churning loss torques of both the input and output gears remain low with slight oscillations, while as the depth increases to 2 h, the amplitude of the oscillations drastically increase. The deeper the gears are submerged into the oil the higher the loss torques. It is deduced from Fig. 23 that the impact of the lubricant viscosity on the churning loss torques of the gears is not obvious. Some dominant frequency components can be found from Figs. 21 to 23, the relation between those dominant frequencies of the rotation speed deserves further investigation.
Fig. 21 Churning power loss torques at different rotation speeds
Fig. 22 Churning power loss torques at different immersion depths
5. Conclusion

A high-fidelity model for a high-speed train gearbox is created and used for MPS-based simulations to inspect the lubrication mechanism inside high-speed train gearboxes. Influences of different rotation speeds, immersion depths, and lubricant viscosities on the internal flow field of the gearbox are inspected through solving the seven MPS-based models. Flow and splashing of the oil agitated by the rotating gears are explicitly animated through the numerical simulations. Simulation results have confirmed that the proportion of the churning losses of the bearings in the entire churning loss is very small (from 0.6% to 0.9%) and negligible. Most of these losses come from the gears, especially the output gear. The churning loss of the output gear accounts for 80% ~ 90% of all losses. The influences of different factors on the proportion of output gear’s churning loss are found as: 1) the rotation speed has little effect on that proportion; 2) that proportion slightly declines as we raise the immersion depth; and 3) as the lubricant viscosity increases, that proportion also increases. Even though the input gear is not submerged into the lubricant oil and does not directly participate in lubrication, the oil

![Graphs showing churning power loss torques corresponding to different lubricant viscosities](image-url)

Fig. 23 Churning power loss torques corresponding to different lubricant viscosities
splashed by the output gear will reach the input gear and cause considerable churning losses. In summary, in designing such gearboxes, the churning losses of bearings can be ignored but the losses of both gears have to be considered. Also, a new numerical model needs to be developed in the future to precisely identify the contribution of each parameter on the change of the churning losses.

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