Numerical and experimental investigation of splashing oil flow in a hypoid gearbox

Qianlei Peng, Liangjin Gui, and Zijie Fan

Abstract
The oil distribution and churning loss are the main concerns in the design of a lubrication system for a gear transmission. This article describes a numerical method of simulating the splashing oil flow in a hypoid gearbox, obtaining the oil distribution, and calculating the churning loss. A three-dimensional numerical model is built to simulate the complex oil flow when hypoid gears splash oil inside the box. The volume of fluid (VOF) model is applied to describe the oil-air two-phase flow characteristics. In addition, the flank-moving technique is introduced to overcome the difficulty of generating cells in the tooth-meshing zone. To validate the numerical approach, a test rig is designed to observe the oil distribution in the gearbox and measure the churning loss of the hypoid gears. The comprehensive experimental investigation illustrates that the numerical results correspond well with the measured results. The results of this article show that the numerical method can accurately predict the complex oil flow in a hypoid gearbox and provide a numerical approach for investigating the oil flow and churning loss inside real-vehicle axle housing. This model can serve as a design optimization tool in the future.

Keywords: CFD; hypoid gear; splash lubrication; oil distribution; churning loss

1. Introduction
Splash lubrication is widely used in engineering machinery, particularly in low-speed and heavy-loading transmission systems, such as vehicle transmissions and axles. It is difficult and inefficient to predict splashing oil flow using a theoretical or empirical method due to its unsteady nature and multiphase oil-air flow state. This engineering problem is more noteworthy for the hypoid gearbox in consideration of the compact space and the cross layout.

With the development of computational fluid dynamics (CFD) and related commercial software, several researchers have used numerical methods to simulate the splash oil flow inside a gearbox in recent years. Moshammer, Mayr, Kargl, and Honeger (2006) built a numerical model of a cylindrical gearbox and investigated the oil distribution when gears splashed oil at different rotational speeds. Lemfeld, Frana, and Unger (2007) studied the influence of the viscosity and sway angle on the oil distribution in a gearbox based on a simplified numerical model. Li, Versteeg, Hargrave, Potter, and Halse (2009) built a two-dimensional numerical model of a cylindrical gearbox to investigate the influences of the rotation speed, oil level and oil viscosity on the oil flow. Lin et al. (2013) evaluated the lubrication state of a vehicle transmission using a simulation model and numerical results. Gorla et al. (2013) calculated the hydraulic losses of a gearbox using a numerical method and attempted to determine the relationship between the gear geometrical parameters and hydraulic losses. To estimate the churning loss of a manual transmission, Kodela, Kraetschmer, and Basa (2015) tried to sum up the individual contributions of each gear and the percentage of the intermeshing loss. This investigation provided a new and simplified CFD method, but it had obvious limitations. Liu, Jurkschat, Lohner, and Stahl (2017) investigated the oil distribution and the churning loss with almost no restrictions on the house shape and operating conditions, and the numerical results were compared with the test results. Concil and Gorla (2017) introduced an innovative meshing methodology to reduce the computational effort when calculating the churning loss in a planetary gearbox. These researchers mainly focused on the oil flow in a cylindrical gearbox, whereas hypoid gears have a different layout. Cylindrical gears are installed on the parallel axes, whereas hypoid gears are installed between the cross axes (Litvin & Fuentes, 2004). Thus, the oil flow is more complex in a hypoid gearbox. There is little...
information available in the literature on the prediction of the oil flow in such a crosswise installed hypoid gearbox, which is common in vehicle axles. Furthermore, these previous investigations adopted non-teeth methods or gear-separation methods to cope with the narrow gap at the meshing zone, which significantly alters the oil flow state and is not applicable in a hypoid gearbox.

This article aims to predict the oil flow when hypoid gears splash oil inside the box. Using numerical methods and models, the oil distribution is determined from simulation results, and the churning loss of the hypoid gears is calculated. Experiments are conducted on a designed test rig to validate the numerical model. A comparison of the results indicates good agreement between the numerical model and experimental results.

2. Theory

2.1. Volume of fluid (VOF) multiphase model

Splash lubrication in a gearbox is a complex process with a two-phase oil-air flow. The VOF model (Hirt & Nichols, 1981) has been used to describe the oil flow. The volume ratio of the air or oil phase in each volume cell is calculated by the VOF method, and the sum of the air phase and oil phase volume ratios in each cell is unity. The phase interface between the oil and air can be calculated with the transport equation of the phase volume ratio shown below. The geometric reconstruction approach is applied for phase interface detection and the computation of the fluxes through the control volume faces.

\[ \alpha_{oil} + \alpha_{air} = 1 \quad (1) \]

where \( \alpha_{oil} \) is the oil volume ratio in the cell and \( \alpha_{air} \) is the air volume ratio in the cell.

When \( \alpha_{oil} = 0 \), the volume cell is filled with air; when \( \alpha_{oil} = 1 \), the volume cell is filled with oil; and when \( 0 < \alpha_{oil} < 1 \), the volume cell has an oil-air phase interface.

The phase interface between the oil and air can be calculated with the transport equation of the phase volume ratio shown below. The geometric reconstruction approach is applied for phase interface detection and the computation of the fluxes through the control volume faces.

\[ \frac{\partial \alpha_{oil}}{\partial t} + \nabla \cdot (\alpha_{oil} \vec{u}) = \frac{S_{oil}}{\rho_{oil}} \quad (2) \]

where \( t \) is time, \( \rho_{oil} \) is the density of oil, \( \vec{u} \) is the velocity vector, and the source term \( S_{oil} \) is typically zero.

The fluid properties of the mixture in each volume cell are calculated based on the volume ratio.

\[ \rho = \alpha_{oil} \rho_{oil} + (1 - \alpha_{oil}) \rho_{air} \quad (3) \]

\[ \mu = \alpha_{oil} \mu_{oil} + (1 - \alpha_{oil}) \mu_{air} \quad (4) \]

In the equations, \( \rho \) is the density of the mixture in the volume cell, \( \mu \) is the dynamic viscosity of the mixture in the volume cell, \( \mu_{oil} \) is the dynamic viscosity of the oil and \( \mu_{air} \) is the dynamic viscosity of the air.

Using the VOF method, one continuity equation (5) and one momentum equation (6) are solved for the oil-air mixture. Thus, the two phases have the same pressure and velocity fields. The accuracy of the solution at the phase interface is affected when there is a significant difference in the velocities of the two phases.

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0 \quad (5) \]

\[ \frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \nabla \cdot (\vec{t}) + \vec{F} \quad (6) \]

where \( p \) is pressure, \( \vec{t} \) is the stress tensor and \( \vec{F} \) includes contributions due to body forces.

\[ \vec{t} = \mu \left[ \nabla \vec{u} + \nabla \vec{u}^{T} - \frac{2}{3} \nabla \cdot \vec{u} \right] \quad (7) \]

\[ \vec{F} = \rho \vec{g} + \vec{f} \quad (8) \]

where \( I \) is the unit tensor, \( \rho \vec{g} \) is the gravitational body force, and \( \vec{f} \) is the external body force, which includes model-dependent source terms or user-defined sources.

The control equations use the weighted average properties from equations (3) and (4); therefore, the continuity equation (5) and the momentum equation (6) are associated with the transport equation of the phase volume ratio (equation (2)).

2.2. Turbulence model

To describe the turbulence characteristics, the \( k-\varepsilon \) turbulence model was applied to model the effect of turbulent fluctuations on the mean flow. The rotating motion of the gears results in a high strain rate of the fluid and highly bending streamlines. The renormalization group (RNG) \( k-\varepsilon \) turbulence model (Yakhot & Orszag, 1986) was selected because it has certain advantages when modeling rotating and swirling flow. This model is a semi-empirical model based on transport equations of the turbulence kinetic energy \( k \) and dissipation rate \( \varepsilon \).

\[ \frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho \vec{u} k)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_k \mu_{eff} \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon \quad (9) \]

\[ \frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon \vec{u})}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_\varepsilon \mu_{eff} \frac{\partial \varepsilon}{\partial x_j} \right] + \frac{C_{1\varepsilon} \varepsilon}{k} G_k - C_{2\varepsilon} \rho \varepsilon^2 / k \quad (10) \]

In the equations, \( \mu_{eff} \) is the modified viscosity given by \( \mu_{eff} = \mu + \mu_t \); \( \mu_t \) is the eddy viscosity given by...
326 Q. PENG ET AL.

\[ \mu_t = \rho C_\mu (k^2/\varepsilon) \]

is the generation of turbulent kinetic energy due to the mean velocity gradients; \( C_{1e}^* \) is the correction term of \( C_{1e} \) given by \( C_{1e}^* = C_{1e} - (\eta(1 - \eta/\eta_0)/1 + \beta \eta^3) \), where \( \eta = (k/\varepsilon) \sqrt{2S_{ij} \cdot S_{ij}} \) and \( S_{ij} = (1/2)((\partial u_i/\partial x_j) + (\partial u_j/\partial x_i)) \); \( C_\mu, C_{1e}, C_{2e}, \alpha_k \) and \( \alpha_\varepsilon \) are constants given by \( C_\mu = 0.0845, C_{1e} = 1.42, C_{2e} = 1.68, \alpha_k = \alpha_\varepsilon \approx 1.393; \eta_0 = 4.38 \); and \( \beta = 0.012 \).

Relative to the standard \( k-\varepsilon \) model (Launder & Spalding, 1974), the RNG \( k-\varepsilon \) model corrects the turbulent viscosity and improves the \( \varepsilon \) equation such that it is more applicable for describing the turbulent flow inside the gearbox.

The RNG \( k-\varepsilon \) model is used to describe the fully developed turbulence region that has a high Reynolds (Re) number. To model the near-wall region, semi-empirical wall functions are used to bridge the viscosity-affected region between the fully developed turbulence region and the wall. The model of standard wall functions based on the work of Launder and Spalding (1974) was selected because it has been widely used in industrial flow studies and provides an efficient calculation method (Rapley, Eastwick, & Simmons, 2007; Webb, Eastwick, & Morvan, 2010). Notably, the value in the critical region around the gear boundary varies from 0.2 to 80.2, which means that the wall function is not entirely valid. This issue is a flaw when applying the wall function in this simulation. However, the global quality of the simulation is still acceptable for obtaining the splashing flow field in the gearbox.

### 2.3. Torque and power loss calculations

The rotating gears generate pressure and viscous forces and torque when they interact with the surrounding air-oil fluid. The total torque is calculated as follows:

\[ \vec{M} = \vec{r} \times \vec{F}_p + \vec{r} \times \vec{F}_v \]  

(11)

where \( \vec{M} \) is the torque vector, \( \vec{F}_p \) is the pressure force vector, \( \vec{F}_v \) is the viscous force vector, and \( \vec{r} \) is the center distance vector.

Then, the churning loss of the gear can be calculated as follows:

\[ P = \vec{M} \cdot \vec{\omega} \]  

(12)

where \( P \) is the churning loss of the gear and \( \vec{\omega} \) is the angular velocity vector.

### 3. Numerical models

#### 3.1. Geometry

The geometry of the hypoid gears was modeled using a parametric design procedure, as shown in Figure 1. The geometric parameters of the hypoid gears are listed in Table 1. The size of the gearbox is 1,500 mm x 1,150 mm x 1,000 mm.

#### 3.2. Gear operation

The average normal backlash of the hypoid wheel and pinion was designed to be less than 330 \( \mu \)m. It is difficult to generate computational cells in such a narrow fluid zone at this scale. As shown in Figure 2, different methods have been applied in previous research to overcome this difficulty. Such methods include the non-teeth method (Lemfeld et al., 2007; Lemfeld & Frana, 2009), cutting-teeth method (Concli, Conrado, & Gorla, 2010).

| Parameter | Wheel | Pinion |
|-----------|-------|-------|
| Hand | Right | Left |
| Number of teeth \( z \) | 37 | 9 |
| Transmission ratio \( i \) | 4.11 |
| Module \( m/\text{mm} \) | 11.49 |
| Average pressure angle \( \alpha/(^\circ) \) | 22.5 |
| Desired pinion spiral angle \( \beta/(^\circ) \) | 43.65 |
| Face width \( b/\text{mm} \) | 61 | 64.75 |
| Pitch diameter \( d/\text{mm} \) | 425.13 | 117.65 |
| Shift angle \( \Sigma/(^\circ) \) | 90 |
| Offset \( E/(\text{mm}) \) | 26 |
| Depth factor \( k \) | 3.8 |
| Wheel mean addendum factor \( c1 \) | 0.17 |
| Mean clearance factor \( c2 \) | 0.15 |

Table 1. Geometric parameters of the hypoid gears.

![Figure 1. Geometric model.](image-url)
(a) Non-teeth method    (b) Cutting-teeth method (single  
(c) Cutting-teeth method (both) (d) Gear-separation method

Figure 2. Gear modeling methods in previous research.

2014; Lin et al., 2013; Moshammer et al., 2006) and gear-separation method (Dong, Lin, & He, 2012; Gorla et al., 2013; Li et al., 2009). The non-teeth method and cutting-teeth method eliminate all or part of the gear teeth and considerably change the gear geometry. In fact, the velocity of the gear teeth is higher than that of the other parts of the gear, so the teeth have a significant influence on the oil flow state when the gear churns the oil. The gear-separation method increases the backlash of the mesh zone by increasing the center distance between the wheel and pinion. For real-engineering machinery, the installation location of the wheel or pinion is related to other structures, such as the bearing, housing or oil passage. Changing the center distance may lead to intrusive problems between the gear and these structures.

Considering the limitations of the above methods, this article introduces the flank-moving method to consider the backlash of the mesh zone. The flank-moving method increases the backlash by slightly decreasing the tooth thickness and keeps the original installation location of the wheel and pinion. As shown in Figure 3, the flanks of the gear tooth are rotated by a predetermined angle around the gear center point O, which causes the flanks to move to the tooth inside. The predetermined angle should be adjusted based on an actual model.

With the same operation for all tooth flanks of the wheel and pinion, the gear profile remained nearly constant, and the tooth thickness was marginally reduced. The backlash between the tooth flanks near the mesh zone was increased such that computational cell generation in the mesh zone became feasible, as shown in Figure 4.

Unlike the none-tooth method and cutting-tooth method, the flank-moving method requires no elimination of teeth. This advantage ensures that the simulation geometry is similar to the actual gear profile. Compared to the gear-separation method, the flank-moving method requires no increase in the center distance between the wheel and pinion and maintains the original installation location. Therefore, the proposed approach is more applicable and avoids intervening problems when modeling real-engineering machinery. A method similar to the flank-moving method is the scaling method used by Liu et al. (2017). In the scaling method, the gap between the tooth flanks was increased by scaling the pinion and the wheel to 99% of their actual size. However, the scaling process might lead intrusive problems between the new tooth flanks in the mesh zone when applied to hypoid gears, especially considering the complicated geometrical relationship between the hypoid wheel and the hypoid pinion. Research (Fan, Peng, & Gui, 2015; Peng, Gui, & Fan, 2015) has shown that the flank-moving method has an insignificant influence on the simulation result of splash oil flow in a gearbox.

3.3. Mesh

An unstructured tetrahedral element has been used and local refinement has been applied near the gear boundary to discretize the numerical model of the fluid domain in a hypoid gearbox. The element size has a remarkable influence on the results of numerical simulation. It is necessary to find a compromise between the quality
of the numerical results and the calculation time. Thus, the average element size in the narrow fluid zone around gears was varied from 3.0 mm to 0.5 mm. No significant improvement in the numerical churning loss was observed for an element size smaller than 1.25 mm. However, the calculation time notably increases when a small element size is applied. Hence, the average size in the fluid zone around gears is set to 1.25 mm for the numerical model. The initial number of cells is 1,117,628, as shown in Figure 5.

The boundary surfaces of the fluid domain were extracted and divided into different groups based on geometry. Figure 6 shows the boundary surfaces of the wheel and pinion.

### 3.4. Physical properties and boundary conditions

The gearbox was filled with lubrication oil and air. As in actual work conditions, 85W/90 oil for the axle was selected in the simulation. The physical properties of oil and air are listed in the Table 2.

No-slip boundary conditions were established at the walls of the gearbox, and the wall function method was applied in the near-wall region. The gear motion was described via a user-defined function (UDF), and the rotation speeds of the pinion and wheel were set according to different work conditions. The dynamic mesh method was applied to update the computational cells in the vicinity of the gear pair because the rotation of boundaries can result in cell quality degradation and the risk of divergence. Therefore, the spring-based smoothing method and the local cell remeshing method were adopted by agglomerating the computational cells when certain skewness or size criteria were violated; thus, the cells automatically adapted to fit the boundary rotation and were sufficiently smooth for computational stability.

The static oil level in the gearbox was 70.5 mm below the axis of the wheel. To initialize the numerical model, the volume fraction of the oil phase for the cell below the oil level was set to 1, and the volume fraction of the oil phase for the cell above the oil level was set to 0, as shown in Figure 7. A geometric reconstruction scheme was used for the cell containing the oil-air interface.

### 4. Experimental apparatus

A specialized test rig was designed to validate the simulation model and results. The function of the test system includes (1) observing and recording the oil distribution when the hypoid gears splash the oil and (2) measuring the power loss when the gears churn the oil.

Figure 8 is a schematic diagram of the test rig, where B1–B6 are bearings. Figure 9 is a photograph of the whole

---

**Table 2.** The physical properties of oil and air.

| Property                              | Oil              | Air              |
|---------------------------------------|------------------|------------------|
| Kinematic viscosity (mm²/s)           | 153 (40°C)       | 16.6 (40°C)      |
|                                       | 15.3 (100°C)     | 22.9 (100°C)     |
| Density (kg/m³)                       | 900 (15°C)       | 1.225 (15°C)     |

---

**Figure 5.** Mesh of the fluid domain.

**Figure 6.** Mesh of the hypoid gear boundary.

**Figure 7.** Initialization of the fluid domain.
The wheel and pinion were installed on the gear base through the bearings. The observing box was made of transparent material. The gear base and observing box constituted an enclosed box with a size of $1,500 \text{ mm} \times 1,150 \text{ mm} \times 1,000 \text{ mm}$, the same size as in the numerical model. A certain amount of 85W/90 oil was added to the box. The electrical motor drove the pinion and wheel through the shaft, and the oil splashed inside the box due to the rotating motion of the wheel and pinion. The entire process was recorded by the camera.

The torque sensor measured the input torque and rotation speed and computed the input power of the gearbox. The torque sensor had a 0.05 accuracy class with a nominal range of 500 N.m. The input power kept balance with the power loss inside the gearbox under a constant rotation speed.

The power loss $P_{\text{in,oil}}$ was first measured when the wheel and pinion were immersed in oil and rotated at a certain speed. Then, the power loss $P_{\text{in,no_oil}}$ was measured when the wheel and pinion were not immersed in oil and rotated at the same rotation speed. For the non-immersed condition, a small quantity of oil was supplied to the gear mesh region and the bearing location, assuming that the values of friction power loss for the two conditions were approximately equal (Ariura, Ueno, Sunaga, & Sunamoto, 1973).

\[
P_{\text{in,oil}} = P_{G,f} + P_{G,c} + P_{B,f} + P_{B,c} + P_{S} \tag{13}
\]
\[
P_{\text{in,no_oil}} = P_{G,f} + P_{B,f} + P_{S} \tag{14}
\]

where $P_{G,f}$ is the friction power loss of the hypoid wheel and pinion, $P_{B,f}$ is the friction power loss of the bearings, is the churning loss of the hypoid wheel and pinion, $P_{B,c}$ is the churning loss of the bearings, and $P_{S}$ is the power loss of the seals.

The difference in the power loss under the two lubrication conditions was due to the churning loss of the gears and bearings.

\[
P_{\text{churning}} = P_{\text{in,oil}} - P_{\text{in,no_oil}} \tag{15}
\]

where $P_{\text{churning}}$ is the churning loss in the gearbox.

In this test, the churning loss of the bearings $P_{B,c}$ was rather small relative to the churning loss of the hypoid gears $P_{G,c}$. Thus, relationship shown in equation (16) can be considered.

\[
P_{\text{churning}} \approx P_{G,c} \tag{16}
\]

The oil temperature was also monitored throughout the test.

![Figure 9. Picture of the test rig.](image)

![Figure 10. Detailed photograph of the test hypoid gears.](image)
5. Numerical results and experimental validation

All numerical runs were performed with the commercial software package ANSYS FLUENT 13.0. The momentum and turbulent quantities were discretised using the second-order upwind scheme. The PRESTO! scheme was chosen for the pressure interpolation. The least squares cell-based evaluation was used to determine the gradient. In addition, the first-order implicit scheme was adopted for the temporal discretization. The cells were remeshed every five time steps to fit the boundary rotation. The number of cells changed to 1,097,845 at the end of the simulation.

The average calculation time needed for each numerical run was approximately 75 hours based on the available computational resource of two Intel Xeon E5 2690 processors operated at 2.90 GHz.

To study the mesh independence, a refined model with 2,332,193 cells was built. This number of cells was almost twice that originally used (1,117,628). The improvement in the numerical results was not significant using the refined model. However, the refined model required a calculation time of 246 hours, which was 328% longer than that required by the original model in this article. In consideration of the calculation efficiency, the original model proposed in this article was considered more appropriate for engineering applications.

5.1. Oil distribution

Figures 11–13 show the contours of the oil volume fraction in the hypoid gearbox and the velocity streamlines around the hypoid gears. The contours of the oil volume fraction indicate the oil distribution conditions, and the velocity streamlines describe the oil flow trend around the gears. The wheel plays the dominant role in the splashing oil flow because the immersion depth of the wheel is considerably greater than that of the pinion. Additionally, the oil distribution results observed in the experiment were compared with the numerical results.

Figure 11 shows the oil distribution and streamline results when the input rotation speed of the pinion is 415 rpm. The rotation speed of the wheel is only 100.9 rpm because there is a transmission ratio of 4.11 between the pinion and wheel. This work condition could be regarded as a low-speed work condition. Under this
condition, the oil churned by the gear was nearly entirely coated around the gear surface, and only a small amount of oil was splashed out. The experimental observation was consistent with the numerical result.

Figure 12 shows the oil distribution and streamline results when the input rotation speed of the pinion is 1,037 rpm. When the rotation speed increased, the oil churned by the gear was splashed out in the radial direction. The oil flow reached the top and right surfaces of the box, and the oil distribution in the box exhibited a fan shape. The fan shape was also observed in the experiment, which corresponded well with the simulation result.

Figure 13 shows the oil distribution and streamline results when the input rotation speed of the pinion was increased to 1,866 rpm. Under this work condition, oil was fully splashed out and reached both side surfaces and the top surface of the box. The oil was distributed on nearly the entire section of the box. Additionally, the oil flow observed in the experiment was consistent with the numerical result.

5.2. Churning loss

Figure 14 shows the numerical results of the splashing torque on the wheel and pinion at different rotation speeds. The horizontal axis indicates the input rotation speed of the pinion.

The churning loss of the gears during the splashing process was of particular interest in this research. The churning loss could be computed based on the above splashing torque. As shown in Figure 15, the red dotted line indicates the churning loss of the wheel, and the blue dotted line indicates the churning loss of the pinion. The dark solid line indicates the total churning loss of the wheel and pinion.

Overall, the churning loss of the wheel and pinion increased with the input rotation speed. The wheel contributed more to the total churning loss because it had a larger diameter and therefore interacted more with the oil.
Figure 16 shows the torque test results at different input rotation speeds. The red dotted line indicates the torque result when the wheel and pinion are immersed in oil, and the blue dotted line indicates the torque result when the wheel and pinion are not immersed in oil.

As described above, the difference between the two lines in Figure 16 is the splashing torque of the wheel and the pinion, as shown in Figure 17.

The churning loss of the gears was obtained from the test results of the torque at different rotation speeds. As shown in Figure 18, the red dotted line indicates the total power loss when the gears are immersed in oil, and the blue dotted line indicates the power loss when the gears are not immersed in oil. The dark solid line indicates the total churning loss of the wheel and pinion, which is the difference between the above two lines.

Figure 19 shows the churning loss obtained from the simulated and measured values. The oil properties for the different speed conditions were adjusted according to the measured oil temperatures.

The comparison shows that the churning loss increased with increasing speed. The maximal error between the simulated and measured values was 15.3%, which occurred at a rotation speed of 831 rpm. The error at the other points was below 10%, which is acceptable for most engineering applications. The diagram also validates the numerical model and method.

6. Conclusion

This article conducted a numerical investigation of hypoid gear lubrication. A CFD model was built considering multiphase and turbulent flow. The flank-moving method was introduced to overcome the difficulty of generating cells in the teeth-meshing zone, and the actual tooth profile and the original installation location for the hypoid gears were kept the same. The numerical results
provided a direct visualization of the oil flow inside the hypoid gearbox, and the oil distribution and churning loss were determined from the numerical results. The oil flow was significantly influenced by the hypoid wheel due to its large size and deep immersion depth.

The predictions for the oil distribution and churning loss were compared with the experimental results on a hypoid gear test rig designed to observe the oil distribution and measure the churning loss under different input rotation speeds. The comparison confirmed that the numerical model and method are reliable. The oil distribution results from the numerical calculations correspond well with the experimental observations. In addition, the error in the predictions of churning loss was lower than 15.3%, which is acceptable for most engineering applications.

An important contribution of this work is establishing a feasible approach to understand the oil flow inside a hypoid gearbox for engineering applications, such as vehicle drive axles. In future work, this modeling method will be applied to a complex drive axle, and an optimization strategy for the oil distribution and churning loss will be explored.

**Disclosure statement**

No potential conflict of interest was reported by the authors.

**References**

Ariura, Y., Ueno, T., Sunaga, T., & Sunamoto, S. (1973). The lubricant churning loss in spur gear systems. *Bulletin of JSME, 16*(95), 881–892.

Concil, F., & Gorla, C. (2017). Numerical modeling of the churning power losses in planetary gearboxes: An innovative partitioning-based meshing methodology for the application of a computational effort reduction strategy to complex gearbox configurations. *Lubrication Science*. doi:10.1002/ls.1380

Concil, F., Conrado, E., & Gorla, C. (2014). Analysis of power losses in an industrial planetary speed reducer: Measurements and computational fluid dynamics calculations. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 228*, 11–21.

Dong, C. F., Lin, T. J., & He, Z. Y. (2012). Numerical simulation of flow field in the gearbox based on moving mesh. *Mechanical Research & Application, 2*, 17–19.

Fan, Z. J., Peng, Q. L., & Gui, L. J. (2015). *China Patent No. 201510155076.6*. Beijing: State Intellectual Property Office of the P.R.C.

Gorla, C., Concil, F., Stahl, K., Höhn, B. R., Michaelis, K., Schultheiß, H., & Stemplinger, J. R. (2013). Hydraulic losses of a gearbox: CFD analysis and experiments. *Tribology International, 66*(7), 337–344.

Hirt, C. W., & Nichols, B. D. (1981). Volume of fluid (VOF) method for the dynamics of free boundaries. *Journal of Computational Physics, 39*(1), 201–225.

Kodela, C., Kraetschmer, M., & Basa, S. (2015). Churning loss estimation for manual transmission gear box using CFD. *SAE International Journal of Passenger Cars - Mechanical Systems, 8*(1), 391–396.

Launer, B. E., & Spalding, D. B. (1974). The numerical computation of turbulent flows. *Computer Methods in Applied Mechanics and Engineering, 3*(2), 269–289.

Lemfeld, F., & Frana, K. (2009). Study of the geometrical model parameters for simplification of tooth system. *Journal of Applied Science in the Thermodynamics and Fluid Mechanics, 3*(2), 1–4.

Lemfeld, F., Frana, K., & Unger, J. (2007). Numerical simulation of unsteady oil flows in the gear-boxes. *Journal of Applied Science in the Thermodynamics and Fluid Mechanics, 1*(1), 1–5.

Li, L., Versteeg, H. K., Hargrave, G. K., Potter, T., & Halse, C. (2009). Numerical investigation on fluid flow of gear lubrication. *SAE International Journal of Fuels and Lubricants, 1*(1), 1056–1062.

Lin, Y. H., Hu, Z. H., Xiong, C. Q., Zang, M. Y., Jia, Y., Chen, Y., . . . Zhao, F. Q. (2013). Research of flow field simulation for lubrication system and effect evaluation on a 7-speed dual clutch transmission. *Proceedings of the FISITA 2012 World Automotive Congress* (pp. 285–298). Berlin: Springer-Verlag.

Litvin, F. L., & Fuentez, A. (2004). *Gear geometry and applied theory* (2nd ed.). Cambridge: Cambridge University Press.

Liu, H., Jurkschat, T., Lohner, T., & Stahl, K. (2017). Determination of oil distribution and churning power loss of gearboxes by finite volume CFD method. *Tribology International, 109*, 346–354.

Moshahmer, T., Mayr, F., Kargl, K., & Honeger, C. (2006, April). Simulation of oil flow in gear box housing in 2006 SAE World Congress (2006-1-1574). Michigan, U.S.A.: SAE International.

Peng, Q. L., Gui, L. J., & Fan, Z. J. (2015). Gear splash lubrication numerical simulation and validation based on tooth-face-moving method. *Transactions of the Chinese Society of Agricultural Engineering, 31*(10), 51–56.

Rapley, S., Eastwick, C., & Simmons, K. (2007, May). The application of CFD to model windage power loss from a spiral bevel gear. *Proceedings of ASME Turbo Expo 2007: Power for Land, Sea and Air* (GT2007-27879). Montreal, Canada: International Gas Turbine Institute.

Webb, T., Eastwick, C., & Morvan, H. (2010, June). Parametric modelling of a spiral bevel gear using CFD. *Proceedings of ASME Turbo Expo 2010: Power for Land, Sea, and Air* (GT2010-22632). Glasgow, UK: International Gas Turbine Institute.

Yakhot, V., & Orszag, S. A. (1986). Renormalization group analysis of turbulence. I. Basic theory. *Journal of Scientific Computing, 1*(1), 3–51.