Research Article

Numerical Calculation Analysis and Characteristic Research on Windage Loss of Oil-Jet Lubricated Aviation Gear Pair

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The VOF (volume of fluid) multiphase transient simulation model of the windage loss of the gear pair under oil-jet lubrication was carried out by using the dynamic mesh technology with the powerful parallel computing capabilities of the Super Cloud Computing Center. Firstly, a two-phase (oil-gas) turbulence numerical model was established in the process of oil-jet lubrication. The numerical simulation test was designed by orthogonal experiment. The influence of the oil-jet lubrication parameters and their interaction on the windage power loss was studied by means of variance analysis. The results showed that the influence of injection speed on the windage power loss was the largest and proportional, followed by injection temperature and injection pressure, and the latter two factors were inversely proportional. Then, the fitting calculation formula of windage loss related to each influencing factor is obtained based on the numerical simulation results. Furthermore, by observing the velocity vector distribution of the internal flow field of the gearbox with different time, the formation mechanism of windage loss is understood intuitively, and the measures to reduce windage loss are put forward. Finally, the mechanical and energy characteristics of the windage loss under different oil injection parameters are proposed, by analyzing and calculating the differential pressure force, viscous force, turbulent kinetic energy, and turbulent dissipation rate around the gear pair. This paper provides a method guidance for the calculation of windage power loss and efficiency of aviation gear pair under elastohydrodynamic lubrication in engineering application.

1. Introduction

Oil supply for lubrication and cooling is necessary for aero-engine transmission gears, and it is because the gears have heavy-load and high-speed conditions (pitch velocity up to 100 m/s). During oil injection, oil and air mix around the gear to form oil-gas two-phase flow. Then, the gear rotation drives the two-phase flow to produce a strong rotating flow field. Since the fluid is viscous, it interacts with the rotating gear to cause windage loss, which then consumes the gear transmission power, thus reducing the transmission efficiency of the gear system [1]. The windage power losses generate a lot of heat. At this time, if the lubrication and cooling effect of gear teeth is poor, it will lead to instantaneous sharp temperature rise, which is very prone to tooth surface gluing, gear wear, gear thermal deformation, gear metal phase transformation, lubrication seal failure, etc., resulting in serious

degradation of transmission system performance and even accidents [2].

At present, many experimental studies have been carried out on gear windage power loss. Anderson and Loewenthal [3] established a calculation formula to estimate the windage power loss of a spur gear pair and considered the influence factors such as gear structure size, working conditions, and fluid characteristics by experiment. However, the maximum pitch velocity of the gear is 40 m/s. Heingartner et al. [4] conducted a large number of tests and proposed a calculation formula of windage losses for helical gear. Diab and Ville [5, 6] experimentally studied the windage loss of rotating spur gears in the air and obtained the analytic formula of the dimensionless torque coefficient of the windage loss by dimensionless analysis and quasianalytical method, respectively. Ruzek [7] studied the windage power loss of disc, large and small spur gear, and helical gears under five

different combinations through a series of experiments by a special test rig. The results show that although there is coupling interaction during both gear engagement, the deputy total windage loss of the gear pair is approximately equal to the sum of the two gears when they are not engaged. Simmons [8, 9] studied the windage power loss of a single spiral bevel gear with or without shroud and meshing gear pairs through experiments and numerical simulation. In addition, the effects of pressure and oil mist in the working chamber on the windage power consumption were also studied by experiments. Lord [10] studied the effects of shroud, tooth profile, and lubricant flow on windage power loss of single-tooth and meshed cylindrical gears through a large number of experiments. Delgado [11] and Handschuh [12] obtained a relatively comprehensive gearbox windage data set through the gear windage test facility at NASA Glenn Research Center. The effects of spur gear geometry, shroud geometry, different lubrication system configuration, system pressure, and temperature and gear meshing on windage loss were studied.

Different approaches are available to calculate the windage losses of the oil-jet lubricated aviation pinion-gear pairs. These are as follows: ① measurements on test rigs, ② calculation with empirical equations, and ③ simulation with computational fluid dynamics (CFD). Continuous increases in computational capacity have brought CFD methods into the spotlight as a new way of investigating windage power losses [13, 14]. Massini [15] experimentally measured the parameters related to the windage loss of a single spur gear under free lubrication. He measured the velocity and vector diagram of the flowing fluid by using particle image velocimetry (PIV) technology. The results verified that CFD numerical simulation is an effective tool for predicting the windage power losses. It is one of the experimental works of realizing the visualization of the flow field near the high-speed gear by PIV technology for the first time. Ruzek [16] and Concli [17] both studied the windage loss of rotating gears using the sliding grid method and multireference frame model. The results obtained were in very good agreement with the measured results for the investigated operating condition. Concli [18] carried out experiments and simulations on the measurement of churning loss of the gear pair in the gearbox. VOF model and dynamic grid technology were used in the analysis and calculation. The difference between the final simulation results and the experiments was 8%, proving the correctness of fluid simulation. Al-shibl et al. [19] used FLUENT to study the windage power loss of a single spur gear rotating in the air. In order to simplify the calculation, a two-dimensional plane model was established for calculation and solution. The windage loss of gears at different speeds was analyzed and compared with the published experimental data, as well as the windage power loss of gears with shroud. Tommaso et al. [20] developed an adaptive mesh method that allowed automatic refinement and/or coarsening of the mesh at the air-oil interface. He conducted a detailed numerical study on the radial impact of an oil injection jet on a single spur gear using the fluid volume (VOF) model and obtained the influence law of the oil-jet angle on windage torque and lubrication performance.

In summary, since experimental studies and existing empirical formulas are limited under certain constraints and operating conditions [21, 22], the applicability limitations and lack of accuracy of the empirically have promoted the development of the finite volume (FV) CFD to predict the windage power losses of gear transmissions [23]. CFD simulation is useful for visualizing the flow field and understanding the phenomena. On the other hand, independently from the complexity of the system and the operating conditions, CFD is possible to simulate and predict the behavior of the system in each condition [24, 25].

Due to the two-phase flow of the oil and air during oil-jet lubrication, the gear rotates at least 2 revolutions to achieve stable flow field, while the computer operating time of general laboratory is about 2 weeks. In order to solve the problem of computing resource, this paper uses the powerful parallel computing capacity of the Super Cloud Computing Center to conduct multiphase transient emulation, with an average of 3 days to complete a result.

2. Theoretical Equation of Numerical Calculation

2.1. Fluid Control Equation. The oil-jet lubrication process of gear pair is an oil-gas two-phase flow problem, which requires multiphase numerical calculation analysis; so, the homogeneous model of Euler-Euler VOF two-phase flow modeling is necessary to capture oil flow behavior [26]. In the VOF model, it is assumed that each phase fluid in each computational unit has the same velocity and pressure, and the mixture composed of each phase fluid is regarded as homogeneous flow. The same momentum equations are solved for the insoluble fluid, and the volume fraction of each fluid is calculated in the whole computational domain. Model control equation is [27] as follows:

(1) Volume conservation equation is as follows:

$$\sum_{q=1}^2 \alpha_q = 1, \quad (1)$$

where the lowercase letter q is the fluid phase, q , oil, and air here. α_q is the volume fraction of q phase. N is the number of fluid phases; here, $N = 2$.

(2) Quality conservation equation is as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = \Gamma_{q1q2} - \Gamma_{q2q1}, \quad (2)$$

where $\rho = \sum_{q=1}^N \alpha_q \rho_q$ is the mixed fluid density, and ρ_q is the density of q phase. t is the time. u , v , and w are the components of the fluid velocity along the x , y , and z axes, respectively. Γ_{q1q2} is the positive interphase mass flow rate per unit volume from phase $q1$ to $q2$.

(3) Momentum conservation equation is as follows:

$$\frac{\partial(\rho U)}{\partial t} + \nabla \cdot (\rho U U) = -\nabla p + \nabla \cdot \tau + \rho g + F, \quad (3)$$

where $U = ui + vj + wk$ is the flow velocity. P is the mixed fluid pressure. F represents external body forces. ∇ is the divergence. τ is the viscous stress tensor, and the calculation formula is

$$\tau = \mu \left[\nabla U + (\nabla U)^T - \frac{2}{3} \delta \nabla \cdot U \right], \quad (4)$$

where μ is the hydrodynamic viscosity.

2.2. Turbulent Model Equation. Two important dimensionless numbers in the flow field: Reynolds number and Mach number are the main parameters affecting fluid motion.

(1) The Reynolds number is expressed as

$$Re = \frac{\rho v l}{\mu}, \quad (5)$$

where ρ , v , l , and μ are the density, flow velocity, characteristic length, and dynamic viscosity coefficient of the two-phase mixed fluid, respectively.

(2) The Mach number can be expressed as

$$Ma = \frac{v}{a}, \quad (6)$$

where v and a are the velocity of two-phase mixed fluid and local sound velocity, respectively.

The gear speed of aviation high-speed engine is up to tens of thousands of revolutions per minute. When the gear rotates at high speed and stirs the oil and air two-phase flow, the fluid Reynolds number is up to 10^6 orders of magnitude. Due to the complex geometrical structure of gears, it is easy to have a strong turbulent flow state when rotating at high Reynolds number. So, the two-equation turbulence model SST $k-\omega$ based on the assumption of eddy viscosity is used to solve the flow field. The SST $k-\omega$ model can be converted into $k-\varepsilon$ model characteristics in free flow, thus avoiding the common problem of $k-\omega$ being too sensitive to inlet free flow turbulence [28].

k and ω of the SST $k-\omega$ model are turbulent kinetic energy and specific dissipation rate, respectively, where $\omega = \varepsilon C/k$ is the ratio of ε to k . The turbulent kinetic energy equation (k equation) and turbulent dissipation rate equation of two-phase mixed fluid (ω Equation) are

$$\begin{aligned} \frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) &= \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{k3}} \right) \frac{\partial k}{\partial x_j} \right] \\ &+ P_k - \beta^* \rho k \omega + S_k, \end{aligned} \quad (7)$$

$$\begin{aligned} \frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_i}(\rho \omega u_i) &= \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 3}} \right) \frac{\partial \omega}{\partial x_j} \right] + \alpha_3 \frac{\omega}{k} P_k - \beta_3 \rho \omega^2 \\ &+ 2(1 - F_1) \rho \frac{1}{\omega \sigma_{\omega 2}} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} + S_\omega. \end{aligned} \quad (8)$$

Here,

$$\begin{aligned} \mu_t &= \rho \frac{a_1 k}{\max(a_1 \omega, SF_2)}, \\ P_k &= \mu_t \left(\frac{\partial w_i}{\partial x_j} + \frac{\partial w_j}{\partial x_i} \right) \frac{\partial w_i}{\partial x_j} - \frac{2}{3} \frac{\partial w_k}{\partial x_k} \left(3\mu_t \frac{\partial w_k}{\partial x_k} + \rho k \right), \end{aligned} \quad (9)$$

where σ_{k3} , $\sigma_{\varepsilon 2}$, $\sigma_{\varepsilon 3}$, β_3 , α_3 , β^* , and a_1 are the model correlation constant, ε is the dissipation rate, x_i and x_j are the cartesian coordinate displacement component, u_i is the velocity component, P_k is the turbulence generation term caused by viscous forces, F_1 and F_2 are mixed functions, S is the strain rate tensor value, and S_k and S_ω are user-defined source items [29].

2.3. Dynamic Mesh Model. Dynamic mesh technology is used to simulate the problem of the shape change of the computational domain caused by the rigid motion or deformation of the boundary [30].

The computational domain is divided into n grids in total. The volume of grid i is $\Delta(\text{vol})_i$. If each grid has m surfaces, grid surface j is represented by $S_{i,j}$. For each grid control volume, there is an integral mass conservation equation [31].

$$\frac{d}{dt} \int_{\Delta(\text{vol})_i} \rho d(\text{vol}) + \sum_{j=1}^m \int_{S_{i,j}} \rho (V - W_{R_i}) \times dR_i = 0, \quad (10)$$

where R_i is the perimeter of the i -th grid; R_i is the circumferential boundary with direction; V is the flow velocity vector; W_{R_i} is the velocity vector of the boundary around the solution domain.

When the mesh volume changes, there are geometric conservation equations in integral form for each mesh

$$\frac{d}{dt} \int_{\Delta(\text{vol})_i} d(\text{vol})_i = \sum_{j=1}^m \int_{S_{i,j}} W_{R_i} \cdot dR_i. \quad (11)$$

W_{R_i} on the common boundary of adjacent grids is the same. When solving the flow field density, advancing one time step Δt , then

$$\Delta \int_{\Delta(\text{vol})_i} d(\text{vol})_i = \Delta t \cdot \sum_{j=1}^m \int_{S_{i,j}} W_{R_i} \cdot dR_i, \quad (12)$$

$$\Delta \int_{\Delta(\text{vol})_i} \rho d(\text{vol})_i = -\Delta t \cdot \sum_{j=1}^m \int_{S_{ij}} \rho (V - W_{R_i}) \cdot dR_i. \quad (13)$$

The average mesh density of this time step ρ_{equi} is

$$\rho_{\text{equi}} = \frac{\int_{\Delta(\text{vol})_i} \rho d(\text{vol})_i}{\int_{\Delta(\text{vol})_i} d(\text{vol})_i}. \quad (14)$$

3. Numerical Calculation Model of Windage Loss of Gear Pair

3.1. Solid Model. The basic parameters of spur gear pair of a reduction gearbox in aviation is shown in Table 1.

The rotating motion of gear pair will cause the boundary movement or boundary deformation of the fluid domain. In order to make the numerical calculation model of windage loss closer to the actual physical model and the calculation results more accurate and reliable, the dynamic mesh algorithm is used to reconstruct the mesh near the meshing area of gear pair in real time. However, due to the extremely small meshing clearance of gear pair, high-quality meshing area cannot be obtained in the flow field. Therefore, the thickness of tooth is appropriately reduced for standard assembled gear pair to ensure that the minimum tooth-side clearance of meshing area has 2-3 layers of boundary layer thickness, which is convenient for generation reconstruction of the mesh.

The oil-jet nozzle is installed on the central plane of the tooth width. The gearbox model with the oil-jet nozzle is created in UG. The distance between the end face of the gear and the wall of the gearbox is 2 mm, the diameter of the nozzle is 2 mm, and the diameter of the oil outlet is 22.5 mm. In order to reduce the simulation difficulty, reduce unnecessary computing resources, and improve the simulation efficiency, the gearbox model is properly simplified. The gear shaft, bearing, pipe, small fillet, and various chamfers are ignored. The final three-dimensional model of gearbox and the coordinate system is shown in Figure 1.

Extract the internal flow field of gearbox and do not consider the tiny structure of gearbox. Boolean operation in UG is used to obtain the flow field model. The cross-section model of the fluid domain through the tooth width center plane (i.e., X-Y plane) is shown in Figure 2.

3.2. Meshing. ANSYS ICEM-CFD has been adopted to generate unstructured tetrahedral element meshes based on the finite volume method. In the numerical calculation of windage loss, it is necessary to calculate the force of fluid on gear tooth surface, tooth top surface, tooth root surface, and end face. So, it is necessary to set boundary layer mesh on these gear surfaces. A size function is used to create a refinement around the gear teeth, oil-jet nozzle, and oil outlet. This choice of meshing allows the minimization of the total number of elements in order to have a reasonable computing time. The number of mesh cells in the whole fluid domain is about 9.31 million.

TABLE 1: Basic parameters of spur gear pair.

Parameters	Pinion	Gear
Number of teeth	56	92
Module/mm		2.54
Pressure angle/°		22.5
Tooth width/mm	6	6
Tip diameter/mm	147.32	238.76
Rotation direction	Clockwise	Anti-clockwise

FIGURE 1: 3D model of gearbox.

FIGURE 2: Fluid field model of gearbox.

3.3. CFD Methodology. The commercial CFD code FLUENT v16.0 has been used to solve the flow field. All simulations use a three-dimensional solver and are solved transiently with a dynamic mesh algorithm. A CFD code that applies the VOF method to model free surface flow was utilized to simulate the two-phase flow with gear meshing. Parameter settings of the CFD model are shown in Table 2.

4. Simulation Computation of the Windage Power Loss

The configuration of the Super Cloud Computing Center is given in Table 3.

4.1. Calculation Method. The windage moment exerted on the gear pair by the fluid can be extracted by the module of force on boundary in CFD-POST. Since the viscous force and pressure of the surface of each element are different, the moment on a wall surface (with respect to the specified axis) can be obtained by calculating the vector sum of the moments exerted on the specified axis. The formula is

$$M = \sum_{i=1}^n (r \times F_p) + \sum_{i=1}^n (r \times F_v), \quad (15)$$

TABLE 2: Parameters of the numerical calculation model.

Description	Parameter value
Dimension	3D
Equation control calculation method	Pressure-based separation solver
Time	Transient
Gravity	Exist (-Y direction)
Velocity formulation	Absolute
Multiphase model	VOF model (implicit body force activated)
	Primary phase: air
	Secondary phase: oil
Viscous model	SST $k-\omega$
Energy model	On
Inlet boundary	Velocity-inlet
	Oil-jet velocity: 15 m/s
	Inlet pressure: 0.31 MPa
Outlet boundary	Pressure-outlet
	Relative pressure: 0 MPa
Wall condition	Nonslip
Dynamic mesh	Smoothing and local remeshing
	Pressure velocity coupling: coupled
	Gradient: least squares cell-based
	Pressure: PRESTO
	N-S formulation: finite volume method
Discretization schema	Spatial discretization: second order upwind
	Volume fraction: CICSAM
	Courant number: 10
	Momentum relaxation factor: 0.3
Solution controls	Pressure relaxation factor: 0.2
	Standard initialization
	Patch volume fraction of phase 2: 0
Initialization	
Time step	2e-6 s

where M is the windage moment of the gear pair (N.m); F_p is the pressure vector on the element surface (N); F_v is the viscous force vector on the element surface (N); r the setting axis vector; and n is the number of cells on the selected surface.

The formula for calculating the windage power loss of a single gear is

$$P_w = P_{wf} + P_{wt} = (M_f + M_t)\omega, \quad (16)$$

where P_{wf} and P_{wt} are windage power loss (W) on the end face and tooth surface of the gear; ω is the angular velocity of gear (rad/s).

The formula of the windage power loss of gear pair is

$$P = P_{w1} + P_{w2}, \quad (17)$$

where P_{w1} and P_{w2} are the windage power loss of the pinion and gear, respectively.

4.2. Design of Numerical Simulation Orthogonal Test Scheme. The windage loss is not only related to the gear structure and design parameters such as the number of teeth, module, tooth width, transmission ratio, and rotational speed but also related to the volume, density, temperature, viscosity, flow field shape, and the sealing property of the gearbox. The oil-jet temperature and pressure mainly affect the fluid density and viscosity, while injection velocity and internal structure of gearbox mainly affect the shape of flow field. In given working conditions and gear parameters, the layout parameters of the oil-jet nozzle are important factors affecting the oil flow into the meshing area of gear pair. The definition of nozzle layout parameters is shown in Figure 3.

In Figure 3, the inclination angle β represents the acute angle between the oil-jet line and the common tangent of the pitch circle of the gear pair. The oil-jet height H is the vertical distance between the oil-jet nozzle and the center line of the gear pair. The deviation distance L is the straight-line distance between the nozzle and the intersection point of the common tangent of the indexing circle to the gear pair. The selection of nozzle layout parameters is explained in detail in another paper by the author "Influence of Windage Effect on Impingement Depth and Multi-objective Optimization Design Method of Oil Jet Nozzle Layout Parameters for Aviation Gear Pair." The parameter value is shown in Table 4.

In order to find the effect of each oil-jet parameter, orthogonal test is used to study the influence of these variables on the test results. In the simulation test design, the windage moment is the test index, which is obtained in CFD-POST: pinion speed (A), oil-jet temperature (B), oil-jet pressure (C), and oil-jet velocity (D) are the test factors. The level number of each test factor is set as 4, as shown in Table 5. The standard orthogonal table $L_{16}(4^5)$ is selected for the 16 groups of numerical simulation analysis of the factors combination. The results are shown in Table 6. Then, add the windage moment values of each factor at each level, denoted as Y_i . Then, calculate its average value and record as $\bar{Y}_i$. Finally, calculate the range of each factor, recorded as R_j . Arrange the range values in order, as shown in Table 7.

According to the results of variance analysis in Table 7, the influence order of various factors on the windage moment is $A > D > B > C$. That is, the pinion speed has the greatest influence, followed by the oil-jet velocity, and the oil-jet pressure has the least influence.

4.3. Mathematical Analysis of Simulation Test Results. In order to achieve the comparability of the calculation results, the windage moment obtained from the numerical simulation test is dimensionless, and the dimensionless coefficient C_m is obtained from equation (18).

$$C_m = M / (0.5\rho\omega^2 R^5). \quad (18)$$

The least square method was used to fit the relationship between the dimensionless coefficient of gear windage

TABLE 3: Super Cloud Computing Center.

Resources	Model	Configuration
HPC cloud	BSCC-A-1	CPU: 2 channel * 32 cores AMD EPYC 7452 @2.35 GHz Memory: 256G Network: IB high-speed network
	BSCC-A-2	CPU: 2 channel * 32cores, EPYC 7452 @2.35 GHz Memory: 512G Network: IB high-speed network, DDR4
	BSCC-A-3	CPU: 2 channels * 32 cores, EPYC 7452 @2.9 GHz Memory: 256G Network: IB high-speed network, DDR4
	CSTC13-V6	CPU: 2 channels * 20 cores Intel XeonGold 6230 CPU @ 2.10 GHz Memory: 192 GB Network: IB high-speed network

FIGURE 3: Layout parameters of the nozzle.

TABLE 4: Layout parameters of the nozzle.

Parameters	Inclination angle β (°)	Oil-jet height H (mm)	Deviation distance L (mm)
Value	5	69.8	40

moment and pinion speed (N), oil-jet temperature (T), oil-jet pressure (P), and oil-jet velocity (V). The nonlinear regression method based on the least square method was used to fit the function curve, and the power function of C_m is

$$C_m = 1.6843 \cdot N^{2.921} \cdot T^{-0.286} \cdot P^{-0.0965} \cdot V^{2.586} \quad (19)$$

According to equation (19), the windage moment of aviation gear pair under oil-jet lubrication is approximately proportional to the 2.92 power of pinion speed, inversely proportional to the 0.29 power of oil-jet temperature, inversely proportional to the 0.10 power of oil-jet pressure, and directly proportional to the 2.59 power of oil-jet velocity.

Hands Schuh [32] pointed out that when the gear linear velocity reaches 125 m/s, the windage loss of the gearbox accounts for about half of the power loss and can dominate other loss mechanisms. Therefore, the accuracy of the calcu-

lation of the windage power loss will seriously affect the efficiency of the gear system. At present, in engineering application, the calculation of windage loss of the aviation gear system under oil injection lubrication mainly refers to the recommended values in some international and national standards without specific calculation formulas. The fitting formula of windage loss obtained based on the calculation results of oil-gas two-phase turbulence numerical model in this section is more suitable for practical engineering application, so as to ensure more accurate efficiency of gear system.

5. Results and Discussion

5.1. Analysis of Internal Flow Field Characteristics of Gearbox. The velocity distribution of the fluid inside the gearbox can reflect the characteristics of the flow field. The plane with coordinate $Z = 2$ mm was selected to observe the velocity vector distribution of the fluid in the gearbox at different times, as shown in Figure 4.

It can be seen from Figure 4 that the high-speed rotation of the gear pair drives the high-speed movement of the surrounding fluid, resulting in a large flow rate of the fluid. There are rotating separation vortices in the upper part of the into-meshing area and the lower part of the out-of-meshing area. The rotating vortices are becoming larger and larger. As shown in Figure 4(d), the flow field tends to be substantially stable when approximately $t = 0.022$ s. At this time, there is a larger self-spin separation vortex in the upper part of the into-meshing area. That is, there is a portion of the fluid having a certain speed without entering the meshing area. The accumulation of fluid causes the temperature of the gearbox to rise here. At the same time, around the large vortex in the out-of-engagement area, fluid is thrown out from the engagement, and most of the fluid is brought into the teeth of the pinion, resulting in the reflow of the lubricating fluid. So, this high-speed rotating lubricating oil exerts a resistance moment on the gear, thus consuming power.

Figure 5 shows velocity streamlines of the gearbox with the pinion rotation speed of 10000 rpm and at the time of $t = 0.02$ s. The fluid rotates along with gears in the direction

TABLE 5: Factor level.

Level value	Factors			
	A Pinion speed/(r/min)	B Oil-jet temperature/K	C Oil-jet pressure/MPa	D Oil-jet velocity/(m/s)
1	11806	313	1	20
2	9000	333	3	10
3	7000	353	5	30
4	5000	373	7	40

TABLE 6: Experiment scheme and result.

Test number	Column				Windage moment (N/m)
	A	B	C	D	
1	11806	313.15	1	20	1.913
2	11806	333.15	3	10	1.852
3	11806	353.15	5	30	1.907
4	11806	373.15	7	40	2.032
5	9000	313.15	3	30	1.672
6	9000	333.15	1	40	1.738
7	9000	353.15	7	20	1.352
8	9000	373.15	5	10	1.414
9	7000	313.15	5	40	1.381
10	7000	333.15	7	30	1.357
11	7000	353.15	1	10	0.954
12	7000	373.15	3	20	1.182
13	5000	313.15	7	10	0.857
14	5000	333.15	5	20	1.246
15	5000	353.15	3	40	0.971
16	5000	373.15	1	30	0.753

TABLE 7: Analysis of variance.

Item	Column			
	A	B	C	D
Y_1	7.704	5.184	5.358	5.693
Y_2	6.176	6.193	5.677	5.077
Y_3	4.874	5.823	5.948	5.689
Y_4	3.827	5.381	5.598	6.122
$\bar{Y}_1$	1.926	1.296	1.339	1.423
$\bar{Y}_2$	1.544	1.548	1.419	1.269
$\bar{Y}_3$	1.218	1.455	1.487	1.422
$\bar{Y}_4$	0.956	1.345	1.399	1.530
R_j	0.969	0.252	0.147	0.261
Influence order: $A > D > B > C$				
Optimal levels: A_4, D_2, B_1, C_1				
Optimal combination: $A_4 D_2 B_1 C_1$				

of gear rotation and converges at the top of the gearbox, after which most of the fluid enters the mainstream of the pinion. There is no steady area in the entire gearbox due to the turbulent state of the fluid.

The formation of the windage moment is mainly due to the resistance of the oil and air to gears. Gears stir the oil because of effect of friction, and the oil will rub and collide with each other. Finally, part of the input power of the gear-box is dissipated as heat. Figure 6 shows velocity volume renderings of the internal flow field at the time of $t = 0.02$ s. At this time, the windage power loss can be considered as the sum of three main terms, single-phase air alone, oil acceleration, and oil recirculation [33]. In general, the oil acceleration loss is mainly the acceleration work (momentum transfer) for which low-velocity oil is accelerated to the peripheral velocity of gears. This depends on the mass and velocity change of oil cluster. The oil recirculation term depends on the amount of excess lubricating oil that is reentered by eddy currents around the rotating gear teeth. The windage loss is flow drag on the face of gear teeth which is induced by the generation of small eddies.

This windage moment not only reduces the transmission efficiency but also dissipates in the form of heat energy, resulting in too high temperature rise of gearbox. Therefore, it is necessary to set a baffle on the upper part of the gear pair into-meshing area to make the fluid enter the meshing area smoothly. At the same time, a deflector needs to be set at the lower part of the out-of-meshing area to make the fluid move to the oil outlet port at the lower part of the gearbox. So, reducing windage power loss is conducive to improving gear transmission efficiency and reducing temperature rise.

5.2. Mechanical Characteristics of Windage Loss. The windage power loss of the gear pair is mainly due to the high-speed flow of the fluid between the pinion and gear. The two gears accelerate the fluid around their teeth, respectively, and carry part of the fluid to collide and squeeze at their meshing points. At this time, the interaction force between the oil-gas mixture and the gear pair is the mechanical characteristic of the windage loss.

Ignoring the instability data at the initial stage of simulation calculation, the windage moment on tooth surface and end surface of gear was monitored from 0.006 s, as shown in Figure 7 when the oil-jet velocity is 20 m/s.

It can be seen from Figure 7 that the windage moment of each characteristic surface of the gear is greater than that of the pinion. That is, the more the number of teeth, the greater the windage moment. This is because the module of gear pair is the same, the more the teeth number, the larger the outer diameter of the gear, and the surface area of the gear side increases. This fact is explained by the increase of the

FIGURE 4: Velocity vector diagram of flow field inside gearbox at different times: (a) $t = 0.0074$ s, (b) $t = 0.0116$ s, (c) $t = 0.0167$ s, and (d) $t = 0.0224$ s.

FIGURE 5: Velocity streamlines.

FIGURE 6: Velocity volume rendering of the internal flow field.

active surfaces being opposed to the air flow, resulting in the increase of viscous resistance, and consequently amplification of the power loss. On the other hand, increasing the teeth number will cause a significant change of the static pressure starting from the root diameter of the gear to the outside diameter, which will induce a significant variation of pressure gradients at the tip of the tooth gear.

Comparing Figures 7(a) and 7(b), it can be seen that the tooth surface moment is one order of magnitude higher than

the end face moment. That is, the tooth surface windage power loss accounts for a high proportion in the total windage power loss. The fluid is affected by the pressure difference between the suction surface and the pressure surface of the gear, which will produce a torque opposite to the direction of the gear rotation. At this time, in order to make the gear rotate at a constant speed, a certain amount of power must be consumed. In addition, the surface and end face of the gear will be affected by the viscous force in the

FIGURE 7: Windage moment curves of gear pair with time (a) tooth surface, (b) end surface, and (c) the gear pairs.

FIGURE 8: Pressure distribution nephogram of gear pair (a) the gear pairs, (b) the pinion, and (c) the gear.

flow field, which is tangent to the contact surface. The viscous force and differential pressure force will cause reverse torque, thus consuming system power. The distribution clouds of pressure and viscous force are shown in Figures 8 and 9.

It is known from Figure 8 that the maximum pressure appears on the tooth surface and end face near the gear into-meshing area, because there is just injected lubricating oil here, and the gear teeth about to engage squeeze the oil, resulting in greater pressure here. The pressure on the tooth surface and end face on the out-of-meshing side is small and negative, because the oil-air mixture around the teeth to be separated has been thrown away or squeezed out. The space

between the gear teeth increases rapidly, resulting in a smaller pressure value here.

Figure 9 shows that the viscous force of gear pair is two orders of magnitude smaller than the differential pressure force, and the extreme value also appears in the into-meshing area of gear teeth. The windage power loss is mainly caused by the differential pressure moment, and the proportion of viscous moment is small.

5.3. Energy Characteristics of Windage Loss. From the mechanical point of view, the windage power loss of gear pair is caused by viscous force and differential pressure force. From the perspective of energy, part of the energy lost by

FIGURE 9: Viscous force distribution nephogram of gear pair (a) the gear pairs, (b) the pinion, and (c) the gear.

windage is converted into heat energy in the friction between oil-gas mixture and the gear. And the other part is dissipated in the turbulent movement of the fluid. That is, the kinetic energy loss caused by the windage moment is transformed into heat energy through the friction between fluid and solid wall surface and the viscous force between fluid microclusters. Because the friction force is small at this time, most of the energy is converted into heat energy, resulting in the increase of fluid turbulent kinetic energy.

The turbulent kinetic energy is estimated by turbulence intensity and average velocity. The calculation method is

$$k = \frac{3}{2}(U \cdot I)^2, \quad (20)$$

$$I = 0.16(\text{Re})^{-0.125},$$

where k is the turbulent kinetic energy, I is the turbulence intensity, U is the average velocity, and Re is the Reynolds number.

The calculation method of turbulent dissipation rate is

$$\varepsilon = C_\mu^{0.75} \frac{k^{1.5}}{l}, \quad (21)$$

$$l = 0.07L,$$

where ε is the turbulent dissipation rate, l is the turbulent scale, L is the characteristic length, and C_μ is the empirical constant.

The turbulent kinetic energy and turbulent dissipation rate of mass weighted integral under different oil-jet velocities can be calculated by CFD-POST, as shown in Table 8.

The time-varying distribution of turbulent kinetic energy in the internal flow field of the gearbox is shown in Figure 10.

Figure 10 shows that with the increase of time, the turbulent movement of fluid in the meshing area becomes more and more intense due to the rotation of the gear pair. At this time, the oil-gas two-phase flow that is not fully accelerated will directly obtain energy through mass and energy exchange in the fluid domain, thus increasing the windage

TABLE 8: Turbulent kinetic energy and dissipation rate with different oil jet velocities.

Oil-jet velocities (m/s)	Turbulent kinetic energy ($\text{kg m}^2/\text{s}^2$)	Turbulent dissipation rate ($\text{kg m}^2/\text{s}^3$)
10	0.0493	315.1422
20	0.0537	346.2673
30	0.0865	547.9328
40	0.1055	705.6306
50	0.1274	829.7635

power loss, as can also be seen from Figure 10(d) that after the full development of fluid turbulent motion, the extreme value of turbulent kinetic energy becomes smaller. This is because the kinetic energy difference between the gear pair and the oil-gas mixture is larger at the beginning. When the fluid domain reaches the equilibrium state, the windage power loss will gradually decrease with time.

It can be seen from Table 7 that turbulent kinetic energy and turbulent dissipation rate both increase with the increase of oil-jet velocity. This indicates that the turbulent motion of oil-gas mixture in computational fluid domain becomes more and more intense.

5.4. Windage Loss Characteristics at Different Pinion Speeds.

It can be seen from Section 3.2 that the pinion speed is the greatest factor affecting the windage moment. Figure 11 shows the windage moment and power loss with the pinion speed when the oil-jet temperature is 20°C , the oil-jet pressure is 0.31 MPa, and the oil-jet velocity is 20 m/s. The nozzle layout parameters are the same as Section 3.2.

As shown in Figure 11, the windage moment of the gear pair increases with the increase of the pinion speed. When the speed of the pinion exceeds 8000 r/min (linear speed is about 60 m/s), the windage loss of the gear pair increases almost linearly, and the windage phenomenon becomes particularly severe. This is because that with the increase of rotating speed, the Reynolds number increases, and the fluid turbulence becomes more intense. So, the fluid force on the

FIGURE 10: Turbulent kinetic energy of flow field in the gearbox: (a) $t = 0.0074$ s, (b) $t = 0.0116$ s, (c) $t = 0.0167$ s, and (d) $t = 0.0224$ s.

FIGURE 11: Windage moment and power loss at different speed of pinions.

gear becomes greater, and the differential pressure torque and viscosity torque both increase.

5.5. Windage Loss Characteristics at Different Oil-Jet Velocities. According to the Section 3.2, oil-jet velocity is a significant factor affecting the windage moment. The oil-jet velocity represents the flow rate of lubricating oil injected into the gearbox, which mainly affects the pumping/pocketing power loss in the meshing area of the gear pair. Pumping losses are commonly described as the squeezing of the

incompressible lubricating oil in the interspatial regions between meshing teeth.

According to the principle of gear meshing, the meshing process of gear pair can be divided into three continuous stages: double-tooth meshing, single-tooth meshing, and double-tooth meshing. So, the volume fraction of lubricating oil near each meshing point is constantly changing. In order to obtain the variation of lubricating oil in the meshing area of gear pair with time at different oil-jet velocity, a bounded plane is made with coordinate $Y = 0.1$ mm from each

FIGURE 12: Oil volume fraction versus oil-jet velocities.

meshing point. The instability data at the beginning of simulation calculation is ignored, and the volume fraction of lubricating oil on this plane at 0.006 s to 0.007 s is obtained, as shown in Figure 12.

Figure 12 shows that the lubricating oil flow near the meshing trace changes periodically with the meshing of the gear teeth. The greater the oil-jet velocity, the greater the volume fraction value of the lubricating oil in the meshing area of the gear pair, and the shorter the fluctuation time from zero to the stable value. This is because at a lower oil-jet velocity, part of the lubricating oil is easily taken away by the rotation of the gear pair flow field. Increasing the oil-jet velocity is advantageous to the lubricating oil passing through the high-speed airflow field at the edges of the gear pair and falling onto the tooth surfaces. Therefore, oil can better enter the meshing area to ensure the sufficient lubrication and cooling of the gear pair.

However, with the increase of oil-jet velocity, the volume fraction of lubricating oil in the meshing area will not always increase. Although the oil spot area on the tooth surface will increase, the oil splash phenomenon at the impact point will become more and more serious, resulting in a decrease in the amount of oil entering the meshing area. Meanwhile, when the pressure reaches the vaporization value due to the rotation of the gears, the lubricant changes its state from liquid to vapor due to the cavitation [25], which will result in an increase in the gas fraction, thereby reducing the volume fraction of the oil. When the flow field is stable, the volume fraction of lubricating oil in the meshing area of gear pair will tend to a stable value and change slightly periodically.

The layout parameters of the oil-jet nozzle are the same as those in Section 3.2. The oil-jet temperature is set to 20°C, the oil-jet pressure is set to 0.31 MPa, and the pinion speed is 11000 r/min. Figure 13 shows the influence of oil-jet velocity on the windage moment. When the oil-jet pressure and nozzle diameter are the same, the different oil-jet velocity means different oil flow rate.

Figure 13 shows the effect of oil-jet velocity on windage losses. It can be seen that the windage losses slightly increase with the oil-jet velocity. This is because when the oil-jet velocity is low, the oil flow injected into the gearbox is less. The oil quantity on the gear surface and meshing area is also less, and the energy to drive the fluid to accelerate rotation is

FIGURE 13: Windage moment against different oil-jet velocities.

FIGURE 14: Windage torque of gear pair with pinion speed under different oil jet velocities.

also less. With the increase of oil-jet velocity, the oil spot area and oil volume fraction on the tooth surface will increase. According to the analysis of the energy characteristics of the windage loss, the windage loss increases.

When the gear geometry and working conditions are determined, the windage moment is closely related to the relative size of oil-jet velocity and gear pair speed. Figure 14 shows the variation trend of windage moment with pinion speed at different oil-jet velocity.

According to the gear parameters and working conditions, when the pitch circle velocity is 10 m/s, 20 m/s, 30 m/s, and 40 m/s, the corresponding pinion speed is 1400 r/min, 2700 r/min, 4000 r/min, and 5400 r/min, respectively. Figure 14 shows that when the rotational speed of the pinion is less than 5000 r/min, the windage torque values at the four oil-jet velocities are relatively small. As can be seen from Figure 14, an increase in the rotational speed of the gears means an increase of the velocity of the oil-air two-phase fluid, while higher oil-jet velocity can lead to more lubricant oil to be stirred. With the increase of fluid velocity and the amount of lubricating oil in the gearbox, the possibility of friction and collision between the solid and the oil phases per unit time increases; thus, more energy is consumed, and windage power losses.

TABLE 9: Density and kinetic viscosity.

Temperature t /(°C)	Density ρ /(kg/ m ³)	Kinetic viscosity ν /(mm ² / s)
35	986.9	30.578
100	938.6	4.97

5.6. *Windage Loss Characteristics at Different Oil-Jet Temperature and Oil-Jet Pressure.* The compressibility and expansibility of fluid reflect the properties of fluid volume varying with pressure and temperature. Peng-Robinson (PR) state equation (22) reflects the state characteristics of oil-gas two-phase flow:

$$P = \frac{RT}{V-b} - \frac{a}{V(V+b) + b(V-b)}, \quad (22)$$

where P is the absolute pressure of fluid, V is the fluid volume; R , a , and b are constant, and T is the thermodynamic temperature of fluid.

Temperature and pressure mainly affect the density and dynamic viscosity of the fluid. The experimental results show that the influence of pressure change on dynamic viscosity is small, and the influence of pressure change can be ignored in the variation range of less than 10 atmospheres. With the increase of temperature, the density of gas and liquid decreases, but the dynamic viscosity of gas increases and the viscosity of liquid decreases.

A lubricant oil Mobil Jet Oil II is used in this paper, with its density and kinetic viscosity shown in Table 9.

According to the known data in Table 8, the formulas for the Mobil Jet Oil II's density, viscosity, and temperature in combination with formulas [34] for the physical properties of aviation lubricants are

$$\begin{aligned} \rho &= 0.998 - 7.44 \times 10^{-4}(t - 20), \\ \ln[\ln(\nu + 0.8)] &= 21.479 - 3.532 \ln(t + 273.15). \end{aligned} \quad (23)$$

The relationship between kinematic viscosity and dynamic viscosity of lubricating oil is

$$\nu = \frac{\mu}{\rho} \times 10^6. \quad (24)$$

Given oil temperatures, the corresponding dynamic viscosity and density of oil can be obtained, as shown in Table 10.

According to the volume fraction of the monitored lubricating oil, the density and viscosity of the mixture of oil-air two-phase flow under oil injection lubrication can be calculated as follows:

$$\begin{aligned} \rho_{\text{mix}} &= \frac{\rho_1 \rho_2}{\rho_2(1 - f_m) + \rho_1 f_m}, \\ \mu_{\text{mix}} &= \frac{\mu_1 \mu_2}{\mu_2(1 - f_m) + \mu_1 f_m}, \end{aligned} \quad (25)$$

where ρ_{mix} , ρ_1 , and ρ_2 and μ_{mix} , μ_1 , and μ_2 are the den-

TABLE 10: Dynamic viscosity and density for air and the lubricating oil.

Temperature t /(°C)	Air		Oil	
	Density ρ /(kg/ m ³)	$10^6 \mu$ /pa·s	Density ρ /(kg/ m ³)	μ /pa·s
20	1.205	18.1	998	0.0604
40	1.128	19.1	983.1	0.0248
60	1.06	20.1	968.2	0.0125
80	1.0	21.1	953.4	0.0728
100	0.946	21.9	938.6	0.0047

FIGURE 15: Effect of oil-jet temperature on windage moment of gear pair.

sity and dynamic viscosity of the mixture, the first phase, and the second phase, respectively. f_m is the volume fraction of the second phase.

Keep the parameter values of other variables unchanged, only change the oil-jet temperature, oil-jet density, and dynamic viscosity in FLUENT, and the windage torque of gear pair at different oil-jet temperatures is calculated, as shown in Figure 15.

Figure 15 shows that the windage moment decreases with the increase of oil-jet temperature, which is a direct result of the influence of temperature on both viscosity and density of the fluid surrounding the gears as well as inside the meshing zone. As the oil-jet temperature increases, the air-oil mixture inside the gearbox, being compressible in nature, sees an increase in the operating kinematic viscosity value and a decrease in operating density. The density of air and lubricating oil decreases with the increase of flow field temperature, which means that the mass of two-phase fluid involved in turbulent motion decreases when the volume inside the gearbox remains unchanged. The increase of gas viscosity with the increase of temperature means that the viscous force between the gas clusters participating in the turbulent motion increases. The data on decreasing the viscosity of lubricating oil with the increase of temperature is suggestive of decreased viscous shear force between the liquid microclusters involved in turbulent motion. This combined effect in turn causes a decrease in the windage power loss, as evinced by the curves in Figure 15, which agrees with observations by Lord [10]. According to Table 10 and

Figure 15, when the gear pair is under oil-jet lubricated, the density has a stronger influence in reducing the windage power loss than kinematic viscosity.

6. Conclusions

- (1) Based on the VOF multiphase flow model, a CFD numerical calculation model considering the oil-gas two-phase flow is established to calculate the windage power loss of the gear pair. Through the calculation, the differential pressure force, viscous force, windage torque, turbulent kinetic energy, and turbulent dissipation rate of gear pair can be obtained
- (2) The numerical simulation analysis test of calculating the windage loss is designed by orthogonal test. The calculation formula of windage loss is obtained by fitting the simulation results. This formula can calculate the efficiency of gear system efficiently and accurately
- (3) Through the simulation analysis of the internal flow field of the gearbox, according to the velocity vector diagram, the variation characteristics of the flow field with time are obtained. The formation mechanism of the windage loss during oil injection lubrication is intuitively understood. At the same time, specific measures to reduce the windage loss are proposed to provide reference for improving efficiency and reducing temperature rise of the gears
- (4) The mechanical and energy characteristics of the windage power loss are proposed. It is concluded that the essence of the windage power loss in two-phase flow is that the differential pressure torque and friction torque of the fluid acting on each surface and meshing area of the gear. This is finally transformed into the kinetic energy and heat energy of the fluid around the gear, so that the gear pair consumes power
- (5) Through the analysis of the windage loss characteristics under different oil-jet parameters, it is shown that the oil-jet velocity has the greatest and proportional influence on the windage power loss. It mainly affects the volume fraction of lubricating oil in the meshing area. Secondly, the influence of the oil-jet temperature and oil-jet pressure on windage loss is inversely proportional, which mainly affect the turbulent kinetic energy and turbulent dissipation rate

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

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Conflicts of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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