

Article

Assessing the Meshing for Windage Power Loss Simulations of an Orthogonal Face Gear

Tiberiu-Daniel Pau ¹, Zoltan-Iosif Korka ^{1,2,*} , Dorian Nedelcu ¹ and Corneliu Hrimiuc ¹
¹ Doctoral School of Engineering, University Center Resita, Babes-Bolyai University, 320085 Resita, Romania; tiberiu.pau@ubbcluj.ro (T.-D.P.); dorian.nedelcu@ubbcluj.ro (D.N.); corneliu.hrimiuc@ubbcluj.ro (C.H.)

² Department of Engineering Sciences, University Center Resita, Babes-Bolyai University, 320085 Resita, Romania

* Correspondence: zoltan.korka@ubbcluj.ro; Tel.: +40-745-911-887

Abstract: In the current energy landscape, efficiency is a critical topic. Therefore, even in the case of geared transmissions, it is essential to predict and calculate power losses as accurately as possible from the design phase. There are mainly three categories of losses in a gear unit: friction—the power losses due to the contact between teeth in rotation on the one hand and the seals with the spindles on the other hand; churning—the power losses generated by the air–lubricant mixture compression around teeth roots during rotation; and windage—the power losses due to the teeth aerodynamic trail in the air–lubricant mixture. While the first two categories of losses are intensively studied in the literature, the papers focusing on windage power losses are less representative. An estimation of windage power losses can be performed by numerical simulation, and the accuracy of the results depends on the mesh density and the available computing power. The present study discusses the influence of meshing on the windage torque of an orthogonal face gear immersed in air and compares numerical results generated by SolidWorks 2025 Flow Simulation software with experimental data measured on a test rig.

Keywords: convergence; face gear; flow simulation; torque; windage



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1. Introduction

The windage power losses (WPLs) in gears, especially in high-speed applications, are a significant concern because they affect transmission efficiency. These losses occur due to the interaction between the rotating surfaces of the gears and the surrounding air or lubricant, resulting in energy dissipation [1–4]. Various studies [5–9] have explored methods to quantify and mitigate these losses. For instance, Ruzek et al. [10] revealed that the optimal modification of tooth geometry can decrease WPL, but there is an optimal modification beyond which losses increase.

Li and Wang [11] further developed a calculation method for windage power losses in spiral bevel gears, emphasizing the influence of gear speed and geometry on these losses and highlighting the role of lubricating oil in increasing windage losses. Their subsequent study [12] using oil injection lubrication confirmed the significant impact of lubrication parameters on windage power losses, providing a comprehensive analysis of the fluid dynamics involved.

In earlier research, Ruzek et al. [13] presented an original test device that allows the measurement of windage losses for a pinion–gear pair, as opposed to classical devices limited to a single rotating element. Their study confirmed that the WPL result is close to the result obtained by summing the losses for each element considered individually.

However, it appears that the air flow generated by one element may act on the other element to some extent, thus leading to overall WPLs slightly lower than those obtained by adding individual losses. The amount of power loss due to windage at the highest wind speeds can reach several kW, indicating that windage is essential in some applications.

In their study, Al-Shibl et al. [14] explored non-dimensional shroud spacings (ratio of gap to gear pitch circle diameter) between 0.005 and 0.05 at shaft speeds ranging from 5000 rpm to 20,000 rpm. While it was observed that the Computational Fluid Dynamics (CFD) data closely matched the experimental results, an optimum tolerance could not be identified. Additionally, it was found that modifying the tooth tip by adding a small chamfer on the leading edge reduced the WPL by about 6%. A small bevel increased total WPL by a similar amount, suggesting that the WPL may rise as the gear wears down.

Recently, Dai et al. [15] investigated the windage and churning behaviors in different gear types, finding that spatial intersecting cross-axis gears like bevel and face gears exhibit higher no-load power losses compared to parallel-axis gears.

The use of shrouds has been proposed as a strategy to reduce windage losses by enclosing the gears, as demonstrated by Dai et al. They involved the Lattice Boltzmann method to simulate and validate the reduction in windage power losses in shrouded spur gears [16]. Furthermore, they introduced a torque containment factor that accounts for air compressibility at high Mach numbers, leading to improved theoretical formulae for predicting windage power losses, which enhances the applicability of these predictions during the preliminary design stage of shrouded gears.

Handschuh et al. [17] revealed through experimental results that even a loose shroud on the gears, operating at a remarkable pitch line velocity, can significantly minimize windage power losses, showcasing the power of innovation and resilience in engineering.

Marchesse et al. [18] conducted experiments to better understand how single-side and double-side baffles affect the drag losses of helical and spur gears. Their findings revealed something important: for spur gears, using symmetrical flanges consistently helps to lower windage losses, no matter the size of the clearance. It is fascinating to see how even a single axial flange can make a difference in reducing these losses. On the other hand, when it comes to helical gears, they found that a flange positioned close to the discharge side does not have as significant an impact on reducing windage losses as one placed on the intake side.

As can be observed, these studies collectively offer valuable insights into the mechanisms of windage power losses and potential strategies for their reduction, crucial for enhancing the efficiency of gear transmissions in high-speed applications. While the rotational speed of gears, the geometrical characteristics of gear design, and the density of the surrounding fluid are undeniably significant factors, the specific impact of these elements on power loss is not yet fully elucidated. Furthermore, although various solutions have been proposed to reduce power loss, their overall effectiveness in diverse operational scenarios remains uncertain and requires further scholarly examination. In addition, recent research increasingly relies on the results obtained from numerical simulations using CFD software tools. It is well-known that the accuracy of these results largely depends on the refinement of the model and the available computing power.

Starting from this fact, in this study, we aimed to assess the influence of the used mesh density, with respect to the cell numbers, on the simulation results. For this purpose, we created a virtual model identical to the one used by Zhu and Dai [19], who experimentally measured the windage torque of a 3D-printed orthogonal face gear made of 9400 resin, which was placed inside a parallelepiped gearbox housing made of Plexiglas, without oil. The growing interest in researching orthogonal face gears stems from their application as a new type of transmission, particularly in aeronautical systems. These gears offer several

advantages, including a compact structure, a large contact ratio, and an effective power-splitting capability. In these applications, the gears operate at high rotational speeds, which can lead to windage losses caused by air resistance and turbulence within the gearbox. The simulation results from this study closely aligned with the experimental findings reported by Zhu and Dai [19], especially as the refinement grade increased. Similar outcomes were observed when a specific number of 689,605 cells was used to mesh the face gear.

The structure of this paper is as follows. Section 2 discusses the theoretical foundations of the study, including the computational methods employed for a generalized analytical model that evaluates windage drag torque. Section 3 outlines the materials and methods used in the research, while Section 4 presents the results obtained. Finally, the conclusions are provided in Section 5.

2. Theoretical Background

According to [20], because of the coexistence of windage and churning behaviors inside the gear housing, the windage drag torque T_{wind} can be obtained as a product of the windage contributions factor λ and the gear windage torque without the oil T_{wind0} :

$$T_{wind} = \lambda T_{wind0}, \quad (1)$$

The windage contributions factor is crucial for estimating both windage power losses and total power losses. Since windage losses are proportional to the total surface area exposed to the airflow, this factor is calculated [20] by taking the ratio of the air-wetted surface area to the total area of the gear (Figure 1):

$$\lambda = 1 - \frac{S_w}{S} = 1 - \frac{\varnothing_w}{\pi} = 1 - \frac{1}{\pi} \arccos\left(1 - \frac{h}{R_p}\right), \quad (2)$$

where

S_w —air-wetted surface of the gear;

S —total area of the gear;

$\varnothing_w$ —oil immersion angle of the gear;

h —oil immersion depth of the gear;

R_p —the pitch radius of the gear.

Expressing the pitch radius of the gear as

$$R_p = \frac{z m_n}{2 \cos \beta}, \quad (3)$$

where

z —number of teeth;

m_n —normal module;

β —gear helix angle,

Equation (1) becomes

$$T_{wind} = \left[1 - \frac{1}{\pi} \arccos\left(1 - \frac{h}{R_p}\right)\right] T_{wind0}. \quad (4)$$

Furthermore, the oil-wetted surface area, which includes the contact between the gear and the lubricating oil, can generally be expressed as the lateral surfaces of the front/rear faces and the tooth surfaces [21]:

$$S_w = R_p^2 (2\varnothing_w - \sin 2\varnothing_w) + 2R_p b \varnothing_w + \frac{4.5 z \varnothing_w b m_n}{\pi \cos \alpha_p \cos \beta}, \quad (5)$$

where

b —gear width;

α_p —pressure angle at pitch point.

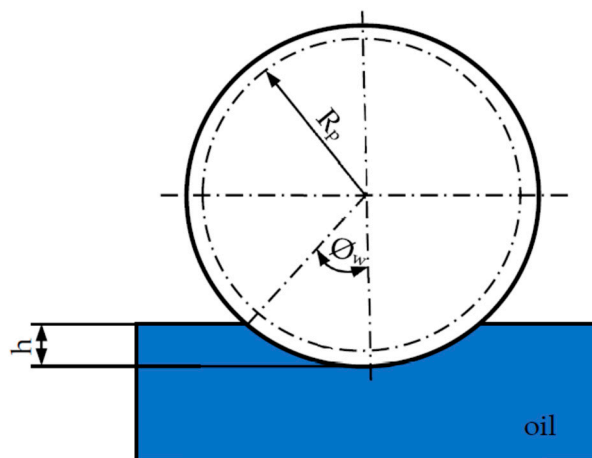


Figure 1. Parameters of a gear immersed in oil.

The windage drag torque and the related energy losses occurring without oil can be clearly divided into two components: the torque exerted on the flanks (T_{flank}) and the torque exerted on the teeth (T_{teeth}) [19]:

$$T_{wind0} = 2T_{flank} + T_{teeth} \quad (6)$$

Depending on the flow regime, based on boundary layer theory, Seetharaman et al. [22,23] proposed the following relations for the windage drag torque exerted on the flanks T_{flank} :

(a) for the laminar regime ($Re < 10^5$):

$$T_{flank}^{(L)} = 0.41 \pi \rho_{air} \nu_{air}^{0.5} R_p^4 \omega^{1.5} = 0.41 \pi \rho_{air} \nu_{air}^{0.5} R_p^{2.5} V^{1.5} \quad (7)$$

where

ρ_{air} —air density;

ν_{air} —kinematic viscosity of air;

V —tangential velocity at the pitch point;

(b) for the turbulent regime ($Re > 10^6$):

$$T_{flank}^{(L)} = 0.025 \pi \rho_{air} \nu_{air}^{0.14} R_p^{4.72} \omega^{1.86} = 0.025 \pi \rho_{air} \nu_{air}^{0.14} R_p^{2.86} V^{1.86} \quad (8)$$

For both upper mentioned cases, the Reynolds number is computed as [19]

$$Re = \frac{2\rho_{air} \omega R_p^2}{\mu_{air}} \quad (9)$$

where μ_{air} is the air viscosity.

Based on the concept of the active surface of teeth proposed by Diab et al. [24], the formula for the drag torque acting on the teeth surface, T_{teeth} , is expressed as

$$T_{teeth} = \frac{z}{8} \rho_{air} \xi b \left[1 + \frac{2(1+x_a)}{z} \right]^4 (1 - \cos\phi) (1 + \cos\phi)^3 (1 - \sin^2\beta) R_p^4 \omega^2, \quad (10)$$

in which

$$\phi = \frac{\pi}{z} - 2(\text{inv}\alpha_p - \text{inv}\alpha_A), \quad (11)$$

where

ξ —reduction factor caused by the inhibiting effect of the gearbox casing on the windage losses generated in the teeth space;

x_a —tooth profile shift coefficient;

α_A —pressure angle at tooth tip.

The windage torque is finally obtained as

(a) for the laminar regime:

$$T_{wind}^{(L)} = \left[1 - \frac{1}{\pi} \arccos \left(1 - \frac{h}{R_p} \right) \right] \left\{ 0.82 \pi \rho_{air} v_{air}^{0.5} R_p^4 \omega^{1.5} + \frac{z}{8} \rho_{air} \xi b \left[1 + \frac{2(1+x_a)}{z} \right]^4 (1 - \cos\phi) (1 + \cos\phi)^3 (1 - \sin^2\beta) R_p^4 \omega^2 \right\}, \quad (12)$$

or:

$$T_{wind}^{(L)} = \left[1 - \frac{1}{\pi} \arccos \left(1 - \frac{h}{R_p} \right) \right] \left\{ 0.82 \pi \rho_{air} v_{air}^{0.5} R_p^{2.5} V^{1.5} + \frac{z}{8} \rho_{air} \xi b \left[1 + \frac{2(1+x_a)}{z} \right]^4 (1 - \cos\phi) (1 + \cos\phi)^3 (1 - \sin^2\beta) R_p^2 V^2 \right\}. \quad (13)$$

(b) for the turbulent regime:

$$T_{wind}^{(T)} = \left[1 - \frac{1}{\pi} \arccos \left(1 - \frac{h}{R_p} \right) \right] \left\{ 0.025 \pi \rho_{air} v_{air}^{0.14} R_p^{4.72} \omega^{1.86} + \frac{z}{8} \rho_{air} \xi b \left[1 + \frac{2(1+x_a)}{z} \right]^4 (1 - \cos\phi) (1 + \cos\phi)^3 (1 - \sin^2\beta) R_p^4 \omega^2 \right\}, \quad (14)$$

or:

$$T_{wind}^{(T)} = \left[1 - \frac{1}{\pi} \arccos \left(1 - \frac{h}{R_p} \right) \right] \left\{ 0.025 \pi \rho_{air} v_{air}^{0.14} R_p^{2.86} V^{1.86} + \frac{z}{8} \rho_{air} \xi b \left[1 + \frac{2(1+x_a)}{z} \right]^4 (1 - \cos\phi) (1 + \cos\phi)^3 (1 - \sin^2\beta) R_p^2 \omega V^2 \right\}. \quad (15)$$

3. Materials and Methods

3.1. Test Stand and Experimental Results

As previously mentioned, Zhu and Dai [19] published experimental data measured on a 3D-printed face gear made of 9400 resin. Their test stand (Figure 2) included the following main components:

- a high-speed electric motor;
- a dynamic torque transducer;
- a bearing support
- 2 elastic couplings;
- transparent housing;
- a face gear inside the housing.

The relevant technical characteristics of the test stand, according to the information available in the cited reference, are as follows.

The high-speed electric motor is controlled by a frequency transformer to drive the shaft along with the face gear, up to a maximum rotating speed of 12,000 rpm. The windage losses are measured by involving a dynamic torque transducer with an accuracy of ± 0.001 Nm. The housing having the dimensions of $394 \times 266 \times 100$ [mm] is made of Plexiglas.

The geometrical parameters of the face gear are presented in Table 1.

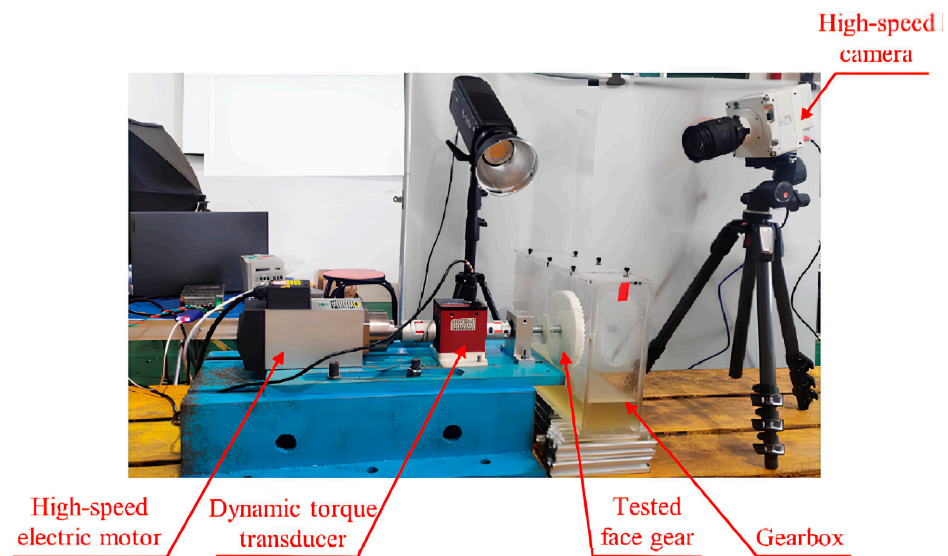


Figure 2. Photo of the test stand [19].

Table 1. Geometrical parameters of the face gear.

Parameter	Measurement Units	Value
Number of teeth	-	51
Normal module	mm	2.5
Pressure angle	°	25
Inner diameter	mm	128
Outer diameter	mm	164

The tests were performed at rotating speeds of 1000, 2000, and 3000 rpm; the evolution of the measured windage torque is illustrated in Figure 3.

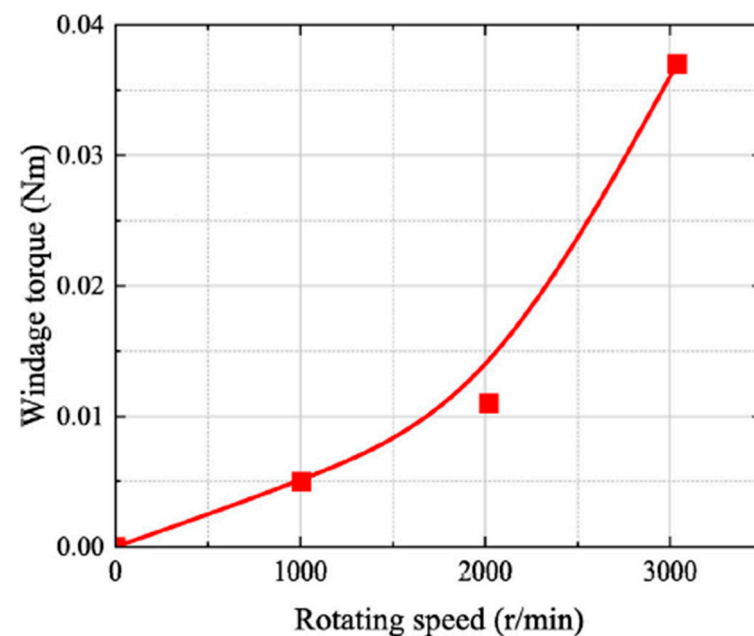


Figure 3. Variation of the windage torque with rotating speed [19].

3.2. 3D Model of the Gearbox

Starting from the above-mentioned information regarding the test stand provided in [19], the generation of the gear 3D geometry was initiated by using KISSsoft R2024

software [25]. The model was exported as an stp-file and was integrated into the assembly generated in SolidWorks 2025 software. The gear geometry and the 3D model of the gearbox assembly are shown in Figure 4.

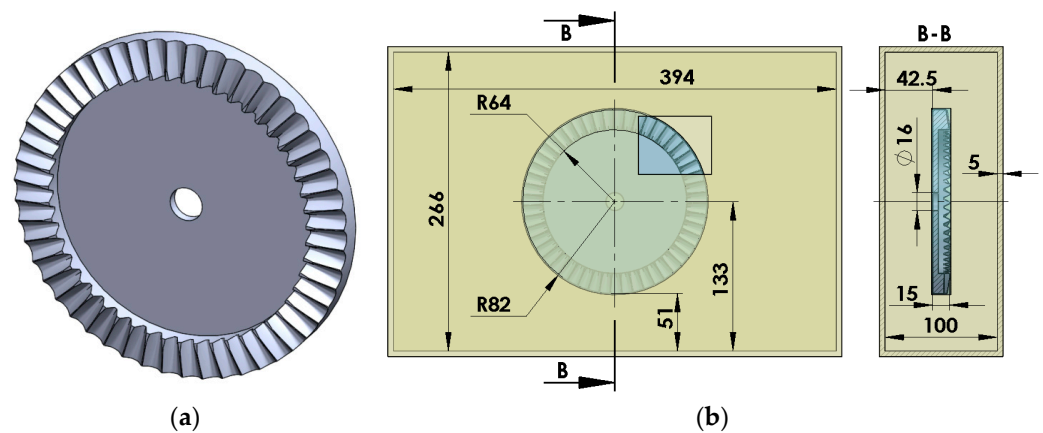


Figure 4. The 3D modelling of the test gear: (a) Gear geometry; (b) Assembly.

3.3. Flow Simulation Settings

To perform the simulation in the Flow Simulation module of SolidWorks, the following steps have been performed:

1. Creation of Flow Simulation project;
2. Definition of analysis type and control volume;
3. Specifying the boundary conditions;
4. Specification of convergence criteria;
5. Calculation of flow studies/projects;
6. Visualization of the results.

Table 2 presents the general settings of the simulation project. Additionally, the Rotating Region was configured to specify a local rotating frame of reference, allowing for the analysis of fluid flow through the model's rotating components.

Table 2. General settings of the simulation project.

Parameter	Setting
Fluid flow	activated
Time dependence	activated, max. time 5 s
Gravity	activated, 9.81 m/s ² along Y direction
Rotation	activated, Local region(s) (Sliding)
Analysis type	Internal
Geometry recognition	CAD Boolean
Exclude cavities without conditions	deactivated
Project fluid	Air (Gases)
Flow type	Turbulent

In this simulation, for defining the Rotating Region, we created a disc with a diameter of 166 mm and a width of 17 mm, which included the gear geometry. Additionally, the angular velocity of this component had to be specified (1000, 2000, or 3000 rpm).

The SolidWorks Flow Simulation module analyses the assembly geometry and automatically generates the computational domain that includes the analyzed geometry. The computational domain represents the region where flow calculations are conducted, and domain boundaries are parallel to the global coordinate system planes.

Meshing affects both the computation time and the accuracy of the results. To assess the refinement grade on the solution convergence, four mesh densities were imposed, shown in Table 3. The table also contains information regarding the hardware configuration of the computers that were used in the study. Figure 5 illustrates, for comparison, the four types of gear discretization.

Table 3. Mesh settings and computer configuration.

Project No.	Cells No.	Mesh Setting	Computer Configuration	Figure 5
1	35,964	Global mesh: Automatic, level 5 Local mesh: Channels, level 1 Advanced refinement Small Solid Feature Refinement Level: level 1	Processor 12th Gen Intel (R) Core (TM) i5-1240P, 1.70 GHz, 16 cores, installed RAM 16.0 GB, 64-bit operating system, x64-based processor	(a)
2	100,530	Global mesh: Automatic, level 6 Local mesh: Channels, level 1 Advanced refinement Small Solid Feature Refinement Level: level 1		(b)
3	316,775	Global mesh: Automatic, level 7 Local mesh: Channels, level 1 Advanced refinement Small Solid Feature Refinement Level: level 2		(c)
4	689,605	Curvature Level: level 3 Global mesh: Automatic, level 7 Local mesh: Channels, level 1 Advanced refinement Small Solid Feature Refinement Level: level 4 Curvature Level: level 5	Processor 11th Gen Intel (R) Core (TM) i9-11900K, 3.50 GHz, 16 cores, installed RAM 64.0 GB, 64-bit operating system, x64-based processor	(d)

SolidWorks Flow Simulation can create different mesh configurations: Manual or Automatic. In Automatic mode, the user imposes specific settings, based on which SolidWorks Flow Simulation generates the number of cells. Each setting is controlled by a slider control with values from 1 to 7 or 1 to 9. The higher the specified value, the more cells that are automatically generated. In our analysis, the following settings were used to control the mesh density:

- Global mesh—used to construct the initial computational mesh;
- Local mesh—Channels—used to specify an additional mesh refinement in the model's flow passages to obtain a more accurate solution;
- Advanced refinement—Small Solid Feature Refinement—allows one to capture relatively small features at the boundary between substances (fluid/solid, fluid/porous, porous/solid interfaces, or boundary between different solids) with a denser mesh by specifying the Refinement Level (denotes the smallest size to which the cells can be split);
- Curvature level—used to restrict the smallest size of the cells.

The aim of the analysis was not to impose a certain number of cells, but to successively generate a number of cells by increasing the configuration settings, leading to simulation values as close as possible to the experimental values. In this way, it is possible to obtain a configuration of settings usable for analyses of similar geometries.

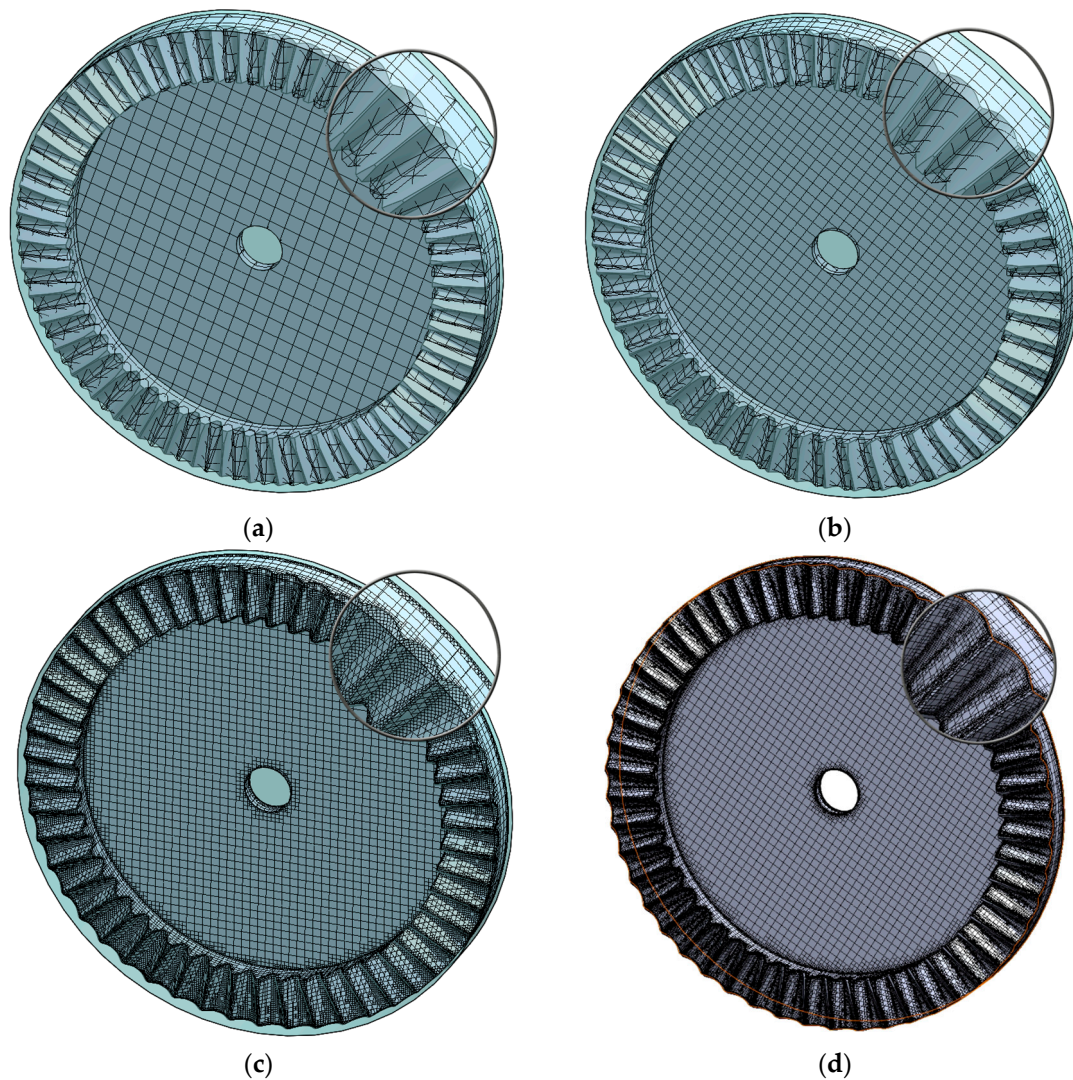


Figure 5. Gear meshing: (a) 35,964 cells; (b) 100,530 cells; (c) 316,775 cells; (d) 689,605 cells.

4. Results and Discussion

In parallel with the running of the four simulations, starting from the variation form illustrated in Figure 3, the curve was digitized, using the PyDigitizer module [26]. Digitizer is a module for digitizing 2D curves from images, created with the help of free and open-source resources, using Python 3.13 as a programming language. The installation kit is available as a dataset from Mendeley [27] with free installation and use. The operating procedure is explained in a video tutorial available on their YouTube channel [28]. The result of the digitization is shown in Figure 6, together with the variation equation.

After performing the simulation for the four defined mesh densities, the results that were obtained were as shown in Table 4. The table presents, in addition to the computed windage torque (WT) values at the three rotating speeds (1000, 2000, and 3000 rpm), the number of iterations, the central processing unit (CPU) time, and the hard disk drive (HDD) size in each study.

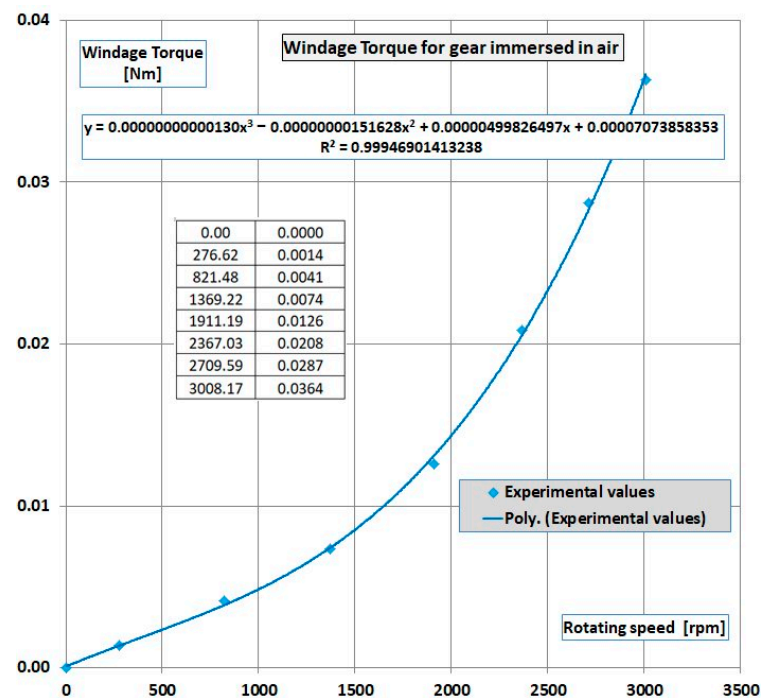


Figure 6. Result of digitizing the experimentally established variation curve of windage torque.

Table 4. Simulation results.

Project No.	Cells No.	Study No.	Rotating Speed [rpm]	Simulation WT [Nm]	No. of Iterations	CPU Time [min]	HDD Size [Mb]
1	35,964	1.1	1000	0.0012	163	1.77	167
		1.2	2000	0.0042	163	1.83	156
		1.3	3000	0.0094	163	1.90	156
2	100,530	2.1	1000	0.0018	230	4.90	438
		2.2	2000	0.0069	230	5.05	390
		2.3	3000	0.0184	230	4.53	394
3	316,775	3.1	1000	0.0034	344	22.75	1490
		3.2	2000	0.0134	360	23.60	1540
		3.3	3000	0.0300	344	22.53	1480
4	689,605	4.1	1000	0.0037	647	87.85	6610
		4.2	2000	0.0149	541	74.07	5650
		4.3	3000	0.0332	604	82.33	6220

To obtain a better understanding of the differences between the experimental results and the simulation results, Table 5 provides the percentage differences (Diff.) between these values. As [19] does not provide concrete data on WT, but only their variation curve (Figure 3), as a function of speed, the comparison was made with the values resulting from the digitization of this curve.

Table 5. Differences between experimental and simulation results.

Rotating Speed [rpm]	Digitized WT [Nm]	Simulation Results WT [Nm]/Diff. [%]							
		35,964 Cells		100,530 Cells		316,775 Cells		689,605 Cells	
		WT	Diff.	WT	Diff.	WT	Diff.	WT	Diff.
1000	0.0049	0.0012	75.5	0.0018	63.3	0.0034	30.6	0.0037	24.5
2000	0.0144	0.0042	70.8	0.0069	52.1	0.0134	6.9	0.0149	−3.5
3000	0.0365	0.0094	74.2	0.0184	49.6	0.0300	17.8	0.0332	9.0
Average Diff. [%]		73.5		55.0		18.4		10.0	

The analysis of the percentage differences between the experimental results and the simulations reveals a notable reduction as the number of mesh cells increases. This statement is also supported by the graph represented in Figure 7.

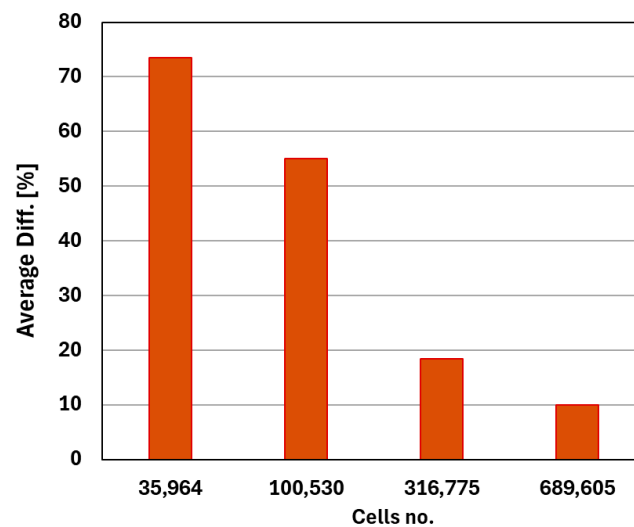


Figure 7. Percentage differences between experimental and simulation results.

As can be observed, an average percentage difference of 73.5% was obtained in the case of simulations performed with 35,964 mesh cells, indicating a substantial deviation between the experimental and simulated data. A significant deviation is also observed in the case of the usability of 100,530 cells. In contrast, this discrepancy diminishes significantly to 18.4% in the simulation with 316,775 cells and to 10% when the number of mesh cells is increased to 689,605.

For a clearer picture of the influence of mesh refinement on the simulation results, Figure 8 shows the variation of WT results as a function of speed and number of cells. This trend emphasizes the critical role of mesh refinement in enhancing the accuracy of simulation outcomes.

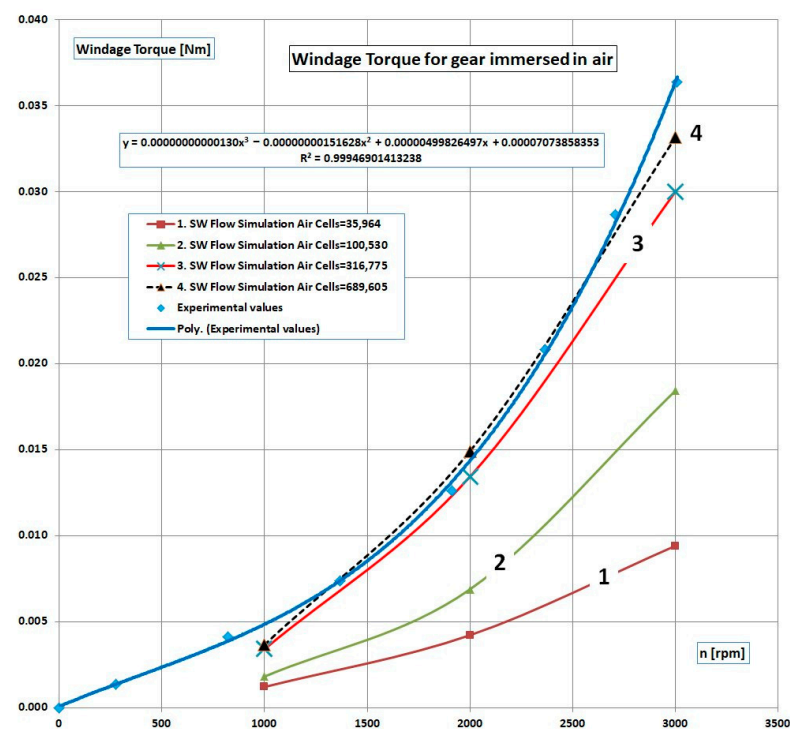


Figure 8. Comparison of experimental and simulation results.

The above conclusion is further reinforced by the convergence chart shown in Figure 9. It is evident that, as the number of cells increases, the windage torque values obtained from the simulation become increasingly similar to the experimentally measured values, and the convergence curve begins to flatten.

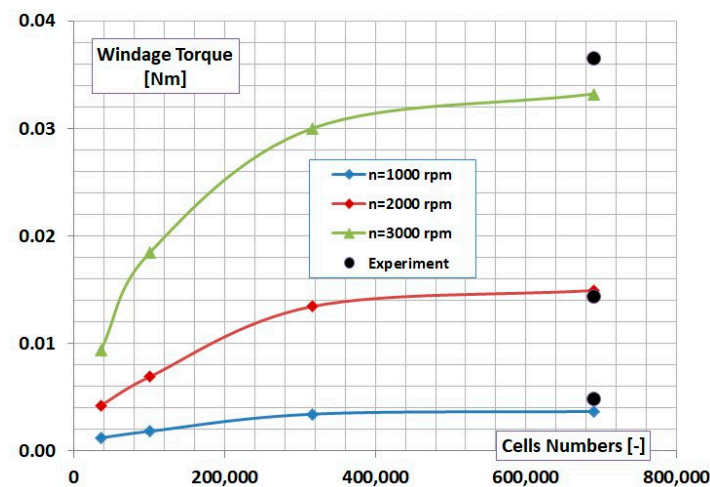


Figure 9. Convergence chart.

This finding is also confirmed by several research works [29–31], which also reveal that the fineness of discretization in numerical simulations significantly affects the accuracy of the results, as it determines the granularity of the computational model. Finer discretization generally leads to more accurate results but involves increased computational resources and time. In this sense, Figure 10 provides an image of the variation of computational resources (average CPU time and HDD size) consumed by the numerical simulation, depending on the number of cells.

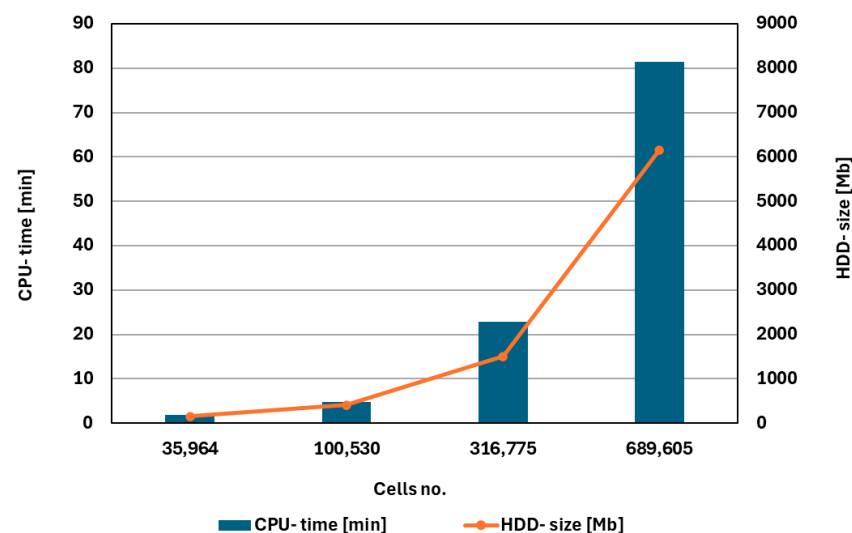


Figure 10. Influence of mesh refinement on computational resources.

As one can observe, discretization impacts both the computational efficiency and the precision of the outcomes. While finer grids improve accuracy, they also increase computational costs. Thus, in the present study, increasing the number of cells by 19.17 times requires 44.25 times longer computation time and 38.25 times higher HDD capacity. Therefore, it is essential to find an optimal balance between accuracy and computational efficiency, as overly coarse discretization can lead to significant inaccuracies. Consequently, while

finer discretization enhances accuracy, it is crucial to consider the associated computational costs and the feasibility of achieving such precision in practical applications.

In addition to the above-mentioned results, Flow Simulation in SolidWorks provides several graphical images that help to visualize fluid flow, pressure distribution, velocity fields, temperature gradients, and other key parameters in a design. Thus, Figures 11 and 12 exemplify the velocity distribution in the gear front plane and the flow trajectories inside the gear housing, respectively. These images offer several key benefits for engineers and designers, such as a better visualization of fluid behavior, quick identification of design issues, and enhancing product optimization.

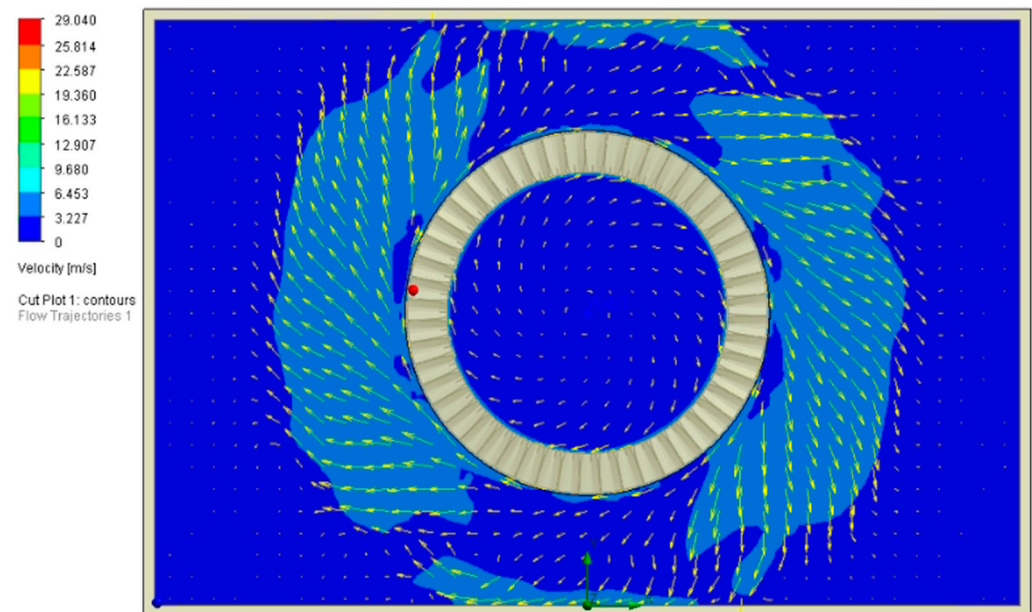


Figure 11. Velocity distribution in gear front plane.

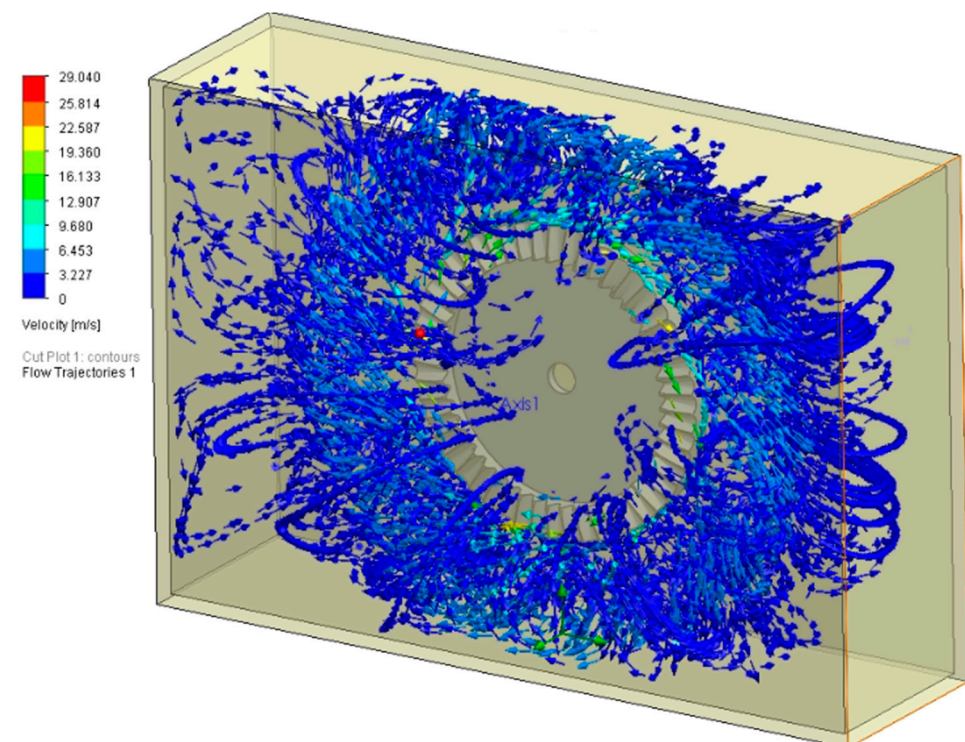


Figure 12. Flow trajectories inside the gear housing.

5. Conclusions

The simulation of gear windage power losses is a critical area of research, particularly in high-speed applications where efficiency is paramount. As orthogonal face gears are increasingly used, especially in aerospace applications, due to the advantages that they offer, the present study was designed to determine the effect of the meshing on the windage torque due to the interaction of the gear with the air. To achieve this, a virtual model that closely resembles the one used in previous research was developed.

Based on the analysis conveyed, it was observed that the windage losses obtained by computational fluid dynamics are very close to the experimental values, especially when the number of cells used in the simulation exceeded 6,000,000.

The analysis indicated an average percentage difference of 73.5% in simulations conducted with 35,964 mesh cells, highlighting a significant deviation between the experimental and simulated results. A considerable discrepancy was also noted when utilizing 100,530 mesh cells. In contrast, this discrepancy substantially diminished to 18.4% with the implementation of 316,775 cells and further decreased to 10% when the number of mesh cells was increased to 689,605.

It is essential to conclude that discretization influences both computational efficiency and the precision of the results obtained. While the implementation of finer grids enhances accuracy, it concurrently increases computational costs. In this study, an increase in the number of cells by a factor of 19.17 necessitated a computation time that was 44.25 times longer and required 38.25 times greater hard disk drive (HDD) capacity for the simulation in question. Therefore, it is imperative to achieve an optimal balance between accuracy and computational efficiency. We must be mindful that overly coarse discretization can result in notable inaccuracies and addressing this is essential for achieving reliable outcomes.

While these simulations provide valuable insights into windage losses, they also highlight the complexity of fluid interactions in gear systems. Future research could investigate alternative methods or materials to further reduce these losses, which may result in more efficient gear designs.

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Abbreviations

The following abbreviations are used in this manuscript:

B1, B2	Bearings
CFD	Computational Fluid Dynamics
CPU	Central processing unit
Diff.	Differences
DTT	Dynamic torque transducer
H	Housing
HDD	Hard disk drive
HSEM	High-speed electric motor

WPL	Windage power losses
WT	Windage torque

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