

Aleksandar Dimić^{1*} , Nenad Kolarević¹ , Miloš Stanković² , Dimitrije Mihajlović⁴ 

WINDAGE LOSSES IN HIGH-SPEED CYLINDRICAL GEAR DRIVES

GUBICI VISOKOBRZINSKIH CILINDRIČNIH ZUPČASTIH PAROVA USLED STRUJANJA

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Adresa autora / Author's address:

¹⁾ University of Belgrade, Faculty of Mechanical Engineering, Belgrade Serbia *email: adimic@mas.bg.ac.rs,

A. Dimić <https://orcid.org/0000-0001-5495-0763> ;

N. Kolarević <https://orcid.org/0000-0003-2836-7405> ;

²⁾ Innovation Centre of the Faculty of Mechanical Engineering, Belgrade, Serbia

M. Stanković <https://orcid.org/0000-0002-4310-2184>

³⁾ EDePro - Engine Development & Production, Belgrade, Serbia

D. Mihajlović <https://orcid.org/0009-0009-5972-6926>

Keywords

- high speed gears
- windage losses
- cylindrical gears efficiency
- split path gearbox

Abstract

High-speed gears with a pitch line velocity exceeding 60 m/s are vital components in the transmission of power and motion in aircraft gearboxes. High demands are placed on them in terms of reliability and energy efficiency. At the same time, their operating conditions (load, speed, temperature) are very unfavourable. One of the main causes of power losses in such gears are windage losses. In this paper, an analysis of windage losses is carried out for a gearbox configuration applied in the aviation industry. Current analytical models are applied to calculate the windage losses. It has been shown that windage losses can be as much as 34 % of gear drive meshing losses and must be studied as such in detail in the design phase of high-speed gear drives.

INTRODUCTION

Gears are vital elements for the transmission and transformation of power and movement in all branches of industry. They are characterised by high efficiency and reliability in operation, even in extremely unfavourable operating conditions in terms of speed and load. On the other hand, they require very high production accuracy and are very technologically demanding. All these features are even more emphasised in high-speed gears used in the aviation industry gearbox. The main causes of power loss for these gear drives are: tooth mesh friction, bearing friction, churning, windage, oil pocketing and gears drag. The share of these losses in the total losses of the gear drive is presented in detail in /1/. According to /2/, the main sources of gear power loss at high gear speeds are found to be windage and churning. Windage power loss is defined as the power loss due to the fluid drag experienced by the gear when it is running in air or an air-oil mist /14/. In order to reduce windage losses in high-speed gears, solutions such as the application of shrouds and various gear design /3-5/, or even the correction of the face surfaces of the gears /6/ are applied. The effect of the shroud on the flow patterns in a tooth valley can be seen in Fig. 1.

Ključne reči

- visokobrzinski zupčanici
- gubici usled strujanja
- efikasnost cilindričnih zupčanika
- višegranski prenosnici snage

Izvod

Visokobrzinski zupčanici, kod kojih je na podeonoj kružnici brzina veća od 60 m/s, su vitalne komponente za prenos snage i kretanja kod vazduhoplovnih prenosnika snage. Njima se nameću visoki zahtevi u pogledu pouzdanosti i energetske efikasnosti. Istovremeno, njihovi radni uslovi (opterećenje, brzina, temperatura) su veoma nepovoljni. Jedan od glavnih uzroka za gubitak snage kod ovih zupčanika su gubici od strujanja. U radu je sprovedena analiza gubitaka od strujanja za jednu konfiguraciju prenosnika snage, primenjenu u vazduhoplovnoj industriji. Primenjeni su aktuelni analitički modeli u cilju određivanja gubitaka usled strujanja. Pokazano je da gubici mogu iznositi i do 34 % gubitaka usled sprežanja, koji se moraju uzeti u razmatranje pri konstruisanju visokobrzinskih zupčastih parova.

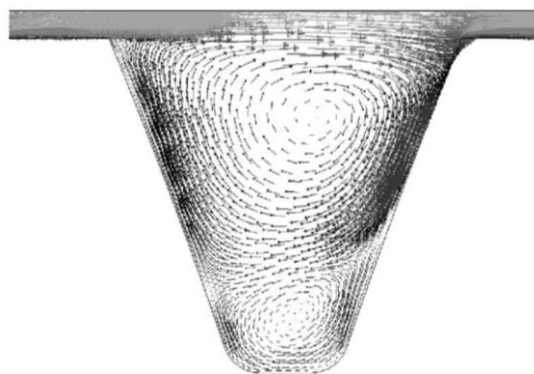


Figure 1. Velocity vector plot taken from /21/ showing secondary flows within a gear tooth valley for a spur gear rotating at 20000 rpm with a 1 mm peripheral shroud clearance.

A vector plot showing secondary flow patterns seen in a tooth valley is shown in Fig. 1: for a spur gear module of 1 mm, pressure angle of 20°, pitch circle diameter 200 mm, and face width 40 mm rotating at 20000 rpm. From Fig. 1, it can be seen that the flow is trapped within the tooth valley with recirculations formed in a similar manner to the flow in a cavity with a moving lid /21/. In /5/, the shrouding effect

had an approximately tenfold reduction in windage losses. In contrast, Fig. 2 shows the fluid flow around gear teeth for unshrouded conditions /8/. The appearance of turbulent flow affects the increase in movement resistance, i.e., power losses.

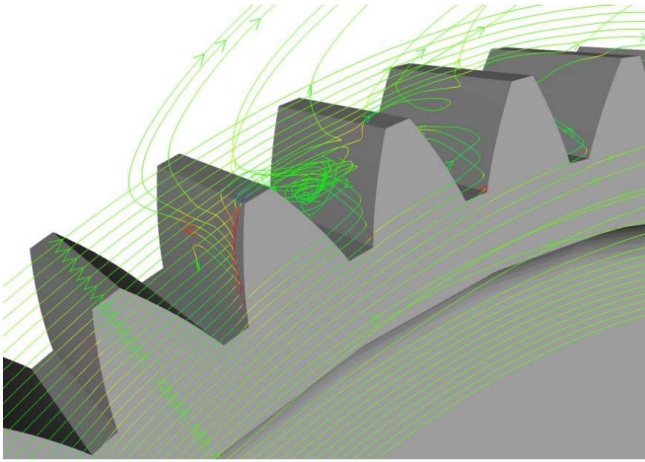


Figure 2. Three-dimensional relative frame streamlines for the unshrouded NASA Glenn 13-inch spur gear at 1000 rad/s, /8/.

Given that these losses in the gear drive occur as heat, special measures must be taken in lubricating and cooling such gears /7/. In addition to the numerical /8-11/, numerous experimental studies /9, 11-13/ are conducted to analyse the windage phenomenon. In these analyses, it is shown that windage losses can, in some cases, exceed meshing losses. In order to take into account windage losses in the gear drive design phase, various analytical models are formed. A review of these models can be found in /14/.

In this paper, a comparative presentation of the analytical models from Dawson /15, 17/, Anderson and Loewenthal /16/, and Diab et al. /18/, is given, and an analysis of windage losses for one configuration of a high-speed gearbox for the aerospace application is performed.

ANALYTICAL MODELS FOR WINDAGE LOSSES

• *Model 1:* Dawson /15/ presented a preliminary formula for windage power losses derived from his test results by treating the sides and periphery of a spur gear separately. Gear shrouding is accounted for by a scaling factor. The formula is obtained for spur gears within the variable range tested. According to this model, the expression for calculating windage losses is:

$$P_W = n^{2.9} \left(\underbrace{0.16 d_f^{3.9}}_{\text{sides}} + \underbrace{d_f^{2.9} b^{0.75} m^{1.15}}_{\text{periphery}} \right) 10^{-20} \Phi \lambda, \quad (1)$$

where: Φ is an unknown function of the effective density of air-oil atmosphere ($\Phi = 1$ indicates an oil free atmosphere); λ is the effect of the gear case. A λ of unity is for a gear operating in free space, 0.6-0.7 is for a gear in a large enclosure, and 0.5-0.6 is for tighter enclosures, i.e., a 'fitting' gear case. Other parameters are related to the gear: n - gear rotational speed (rpm); d_f - root diameter; b - gear face width; m - gear teeth module. This model can also be applied to helical gears, but with some care. Considering the positive influence of helix angle on windage losses, /13/, the results obtained on the basis of Eq.(1) should be reduced by a certain percentage.

• *Model 2:* The analytical formula provided by Anderson and Loewenthal /16/ is based on turbine rotor wheel analysis, where windage power losses are calculated with:

$$P_W = C_1 \left(1 + 2.3 \frac{b}{r} \right) \rho^{0.8} n^{2.8} r^{4.6} \mu^{0.2}, \quad (2)$$

where: C_1 is a constant (2.04×10^{-8} in S.I. units); b is gear face width; r is pitch radius; n is rotation rate in revolutions per minute; and ρ and μ are density (kg/m^3) and viscosity (centipoise) of the atmosphere, respectively. To account for the oil atmosphere within the gearbox, the density and viscosity of the atmosphere are corrected to reflect a 34.25 parts air to 1 part oil combination as reported in /19/. Their expression became:

$$P_W = C_2 \left(1 + 2.3 \frac{b}{r} \right) n^{2.8} r^{4.6} (0.028 \mu + C_3)^{0.2}, \quad (3)$$

where: C_2 and C_3 are 2.82×10^{-7} and 0.019, in respect. This equation, however, does not account for the characteristic properties of gear teeth, such as: diametral pitch, helix angle, etc.

• *Model 3:* To take into account the helix angle, in his extended paper, Dawson /17/ rearranged his equation based on further analysis and testing:

$$P_W = 1.12 \cdot 10^{-8} C' \rho n^{2.85} d_f^{4.7} \nu^{0.15} \lambda, \quad (4)$$

where: as before, λ is effect of gear enclosure; ν is kinematic viscosity (m^2/s); and C' represents a shape factor based on the face width to diameter ratio and the number of teeth the gear possesses. C' is obtained from Fig. 3 in /15/ for spur gears, although there is a large degree of experimental scatter within the results. For helical gears, an estimate is made based on Figs. 3-5 in Dawson's second paper, /17/.

• *Model 4:* The formula presented by Diab et al. /18/ calculates windage power loss as a function of fluid density ρ (kg/m^3), rotation rate ω (rad/s), pitch radius r , and a total windage loss coefficient C_i :

$$P_W = \frac{1}{2} C_i \rho \omega^3 r^5. \quad (5)$$

The power loss coefficient ($C_i = C_f + C_t$) is divided between the losses from the gear face (C_f) and from the gear teeth (C_t). The gear face contribution is:

$$C_f = \frac{2n_1\pi}{5-2m_1} \frac{1}{\text{Re}^{*m_1}} \left(\frac{R^*}{r} \right)^5 + \frac{2n_2\pi}{5-2m_2} \times \left[\frac{1}{\text{Re}^{*m_2}} - \frac{1}{\text{Re}^{*m_2}} \left(\frac{R^*}{r} \right)^5 \right], \quad (6)$$

where: n_1 and m_1 are coefficients for laminar flows and have values of 1.293 and 0.5, in respect. Coefficients n_2 and m_2 are for turbulent flows and have values of 0.072 and 0.2, respectively. Re^* is the critical Reynolds number ($\approx 3 \cdot 10^5$) between laminar and turbulent flow, and R^* is the critical radius separating the laminar and turbulent regions:

$$R^* = \sqrt{\frac{\mu \text{Re}^*}{\rho \omega}}. \quad (7)$$

The gear teeth contribution coefficient is:

$$C_t \equiv \xi \frac{z}{4} \left(\frac{b}{r} \right) \left[1 + \frac{2(1+x)}{z} \right]^4 (1 - \cos \phi)(1 + \cos \phi)^3, \quad (8)$$

where: z is number of teeth; x is profile shift coefficient; and φ is computed from: $\varphi = \frac{\pi}{z} - 2(\text{inv } \alpha - \text{inv } \alpha_A)$, where α and α_A are pressure angles at pitch point and tooth tip, in respect.

The tooth analysis assumes that the fluid is expelled from an active tooth area and that the pressure is uniform on the tooth. This has not been seen by the author [20], or others [14, 21].

ANALYSIS AND DISCUSSION

Estimates on the percentage effect of windage vary as the value is dependent on a number of different parameters. Townsend [22] identified the main influencing variables as:

- diameter of rotating elements,
- length of rotating elements,
- speed of rotation,
- web or gear-blank design,
- overall casing design,
- type of oil feed system,
- operating temperature and viscosity of oil,
- pressurisation of the casing.

All of the above parameters must be taken into account in the design process of high-speed gears. Figure 3 shows the configuration of the gearbox considered. It is a split path (two path) configuration with input rotational speed of 40000 rpm. After the combustion process, power is delivered to the gearbox via gear 1, on the same shaft as turbine T. The maximal power to be transmitted by the shown gearbox is 200 kW.

First, preliminary calculations are made based on which gear 2 is identified as the most vulnerable in terms of windage losses, due to the combination of a relatively large diameter and a high number of revolutions. Further analysis concerns the application of models described in the previous section to this specific gear. Geometric characteristics of gear 2 significant for calculation are given in Table 1.

On the basis of the diagramme in Fig. 4, the exponential growth of all models depending on number of revolutions can be concluded. All models have different gradients of change, and model 2 predicts the highest losses, while model 4 predicts the mildest. For the expected number of revolutions of gear 2, losses range from 0.05 to 0.36 kW depending on the applied model.

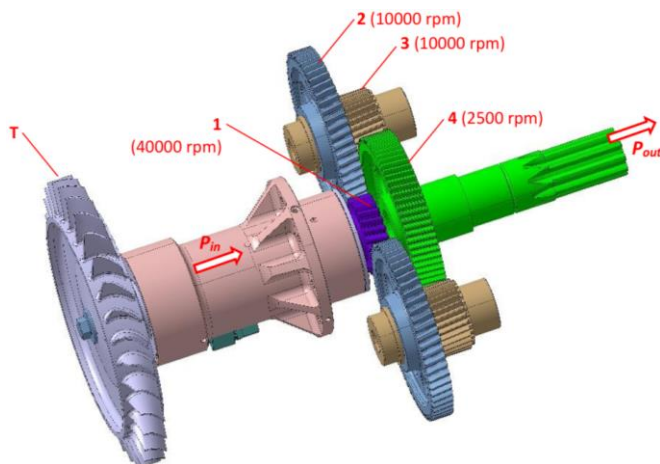


Figure 3. Split path gearbox configuration and position of the analysed gear.

Table 1. Gear 2 geometrical characteristics.

Symbol	Description	Value
z	number of teeth (mm)	72
m	module (mm)	2
x	profile shift coefficient	-0.36
α	Profile angle (°)	20
β	helix angle (°)	10
B	face width (mm)	20
d_f	root diameter (mm)	139.772
d	pitch diameter (mm)	146.221

For the given gear geometry and the nominal rotational speed of 10000 rpm, the pitch line speed of gear 2 goes up to 76.4 m/s. Gear lubrication conditions (closed case, forced lubricant injection) and lubricant type (density, viscosity, etc.) are taken into account by means of the corresponding factors in models described in the previous section.

The diagramme in Fig. 4 shows the applied windage loss calculation models for gear 2, in relation to rotational speed. Since it is a split path power system, all losses are shown for one cylindrical gear pair only.

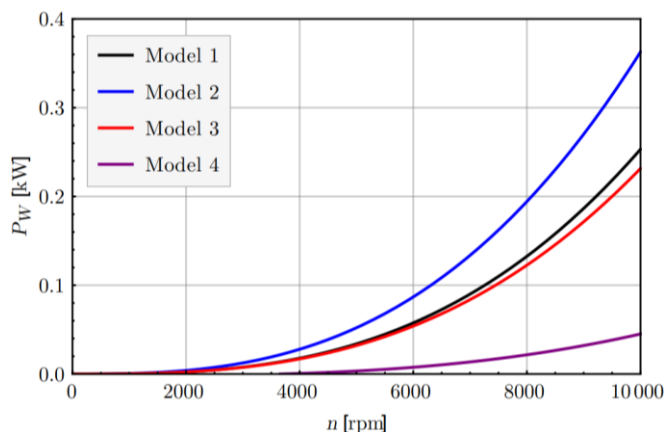


Figure 4. Windage losses for gear 2.

To perform a more precise evaluation, windage losses are compared with the main losses in the gear pair, i.e., the losses due to sliding during the gear teeth meshing. For this reason, the calculation of meshing power losses is carried out according to ISO TR 14179 standard, [23]. These losses are calculated for a constant torque value. This means that (as rpm is varied) the gear power is not constant. Meshing losses, denoted by P_M on the diagramme in Fig. 5, show an

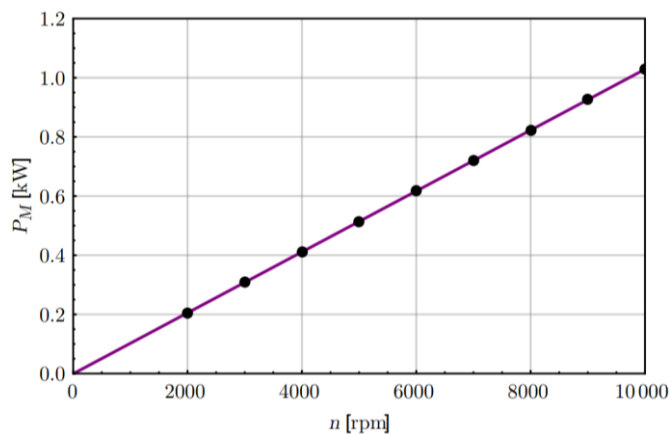


Figure 5. Meshing losses for gear 2.

approximately linear dependence on gear rotational speed. In addition, the intensity of these losses is several times higher than windage losses.

In the last step of the analysis, the factor ξ is introduced, which represents the ratio of windage losses and losses due to meshing:

$$\xi = \frac{P_M}{P_W} . \quad (9)$$

The diagramme of change of factor ξ as a function of the number of revolutions of gear 2, is shown in Fig. 6. On this diagramme $\xi_{\max}$ is windage loss ratio, according to the strictest model 2, while $\xi_{\min}$ is windage loss ratio according to the mildest model 4. The shaded area between boundary lines $\xi_{\max}$ and $\xi_{\min}$ represents the expected scatter for values of ξ . Based on this diagramme, it can be concluded that for the applied operating conditions and the geometry of gear 2, the expected losses due to windage are from 4 % to 34 % of the gear drive meshing losses. The upper limit value of factor ξ indicates a significant share in total losses, which contributes to increased movement resistance, lower efficiency and increased heating of the gear pair. The upper limit is more realistic because 3 out of 4 analysed models lean towards it. The lower limit, determined according to model 4, seems too optimistic, so it should be taken with care for the parameters used in this paper.

In order to give importance to all used analytical models, they are averaged, as shown by the black line ξ_{avg} on the diagramme in Fig. 6. This is done with the aim of obtaining an approximate, but general model for windage losses as a function of revolutions per minute only.

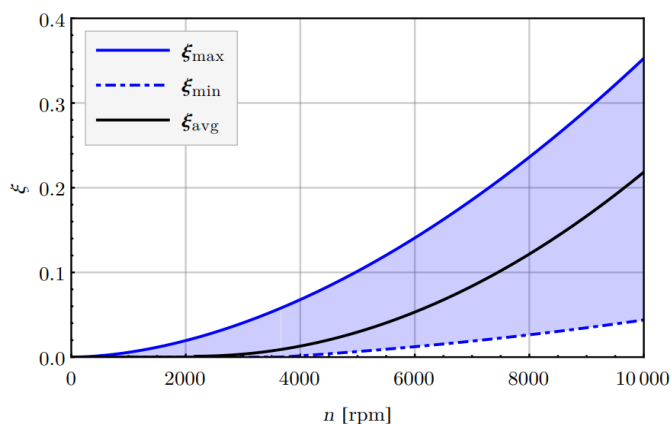


Figure 6. Boundary and average values for ξ .

Averaged values are approximated by a second-order polynomial, whose coefficients for the analysed case are:

$$\xi = 17.6 \cdot 10^{-3} - 15.33 \cdot 10^{-6} n + 3.54 \cdot 10^{-9} n^2 . \quad (9)$$

This curve, which includes 4 windage loss models, has a maximum loss value of 21.8 % at the gear 2 design speed of 10000 rpm. According to this model, the share of windage losses becomes pronounced after 6000 rpm, i.e., after pitch line speed of 45 m/s.

CONCLUSIONS

The paper deals with the analytical calculation of windage losses in high-speed gears. A comparative analysis of four representative models is performed by applying them to a

gear drive used in an aerospace application. The main conclusions for the analysed geometry of gear 2 and operating conditions are as follows:

- All analysed models show an exponential growth in relation to the number of revolutions per minute. Significant windage losses occur above 6000 rpm, i.e., 45 m/s pitch line speed for the given gear geometry.
- Anderson and Loewenthal model 2 is the most rigorous and gives the highest windage losses of 0.36 kW. In contrast, it is the Diab model 4 that gives the least windage losses of 0.05 kW.
- In relation to meshing losses, the proportion of windage losses is from 4 to 34 %.
- A general factor ξ is introduced that predicts the ratio of windage and meshing losses in relation to rpm. This factor takes into account the average values of all analysed models. Considering this factor, the expected windage losses at the design speed are about 22 %. Based on this value, certain measures will be taken to reduce windage losses of gear 2 (application of shroud).

The general conclusion is that all models are extremely sensitive to varying influential parameters such as: root diameter, gear face width, characteristics of oil and housing. That is why the formation of a general model is under question, i.e., each specific case of a gear drive must be analysed in detail for itself.

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