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# Investigations on CFD simulation for predicting windage power losses generated by helical gears

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## 1 Introduction

When considering high-speed gear transmission, the power loss due to the fluid drag torque generated by the gear rotation in air/oil mist may dominate the other losses [1]. This windage power loss (WPL) must be therefore studied in order to mitigate its impact. Numerous investigations have been then made in the past experimentally [2,3,4] or numerically [5,6,7]. Some of these studies proposed WPL model in addition to the results obtained. However the influence of helix angle is not clearly understood since most of the previous studies considered spur gears. Experiments have still shown that WPL value is modified when helix angle value becomes not null. For example Dawson [2] noticed a decreasing of WPL when the helix angle of large width gears is increased whereas opposite observation was made in Diab *et al.* investigation [8] when using highly different gear characteristics. More investigations are therefore needed to better understand the contribution of helix angle in WPL if appropriate model is wanted. The purpose of this paper is at first to adjust to helical gears a previous numerical model [9] using CFD method which was capable to estimate WPL generated by spur gears. This approach is validated comparing numerical predictions with experimental data [8] using identical configuration (Table 1). Further tests are then conducted all considering angular velocity equal to 700 rad/s in which geometrical characteristics (*i.e.* helix angle and width values) vary. Both WPL and volumetric rate expelled by the gears involved in the tooth region are observed here. It has been shown [10] indeed that WPL is highly linked to this volumetric flow rate and this latter could be then employed in an analytical model based on turbomachinery analogy.

Table 1 – Experimental gear characteristics for numerical approach validation.

Pitch diameter (mm)	width (mm)	module (mm)	helix angle (°)
288	30	4	15

## 2 Numerical approach

The previous simplified numerical approach which has been validated for spur gear by Pallas *et al.* [9] is reemployed here. In this approach the use of the entire gear is avoided and the contribution of the flank is ignored in the numerical computation but analytically estimated for the evaluation of total loss. The computational domain involving helix angle comprises one complete single tooth, the associated fraction of flank, and the surrounding air. The fluid is supposed to be pure air, the physical properties of which are evaluated at 25 °C (*e.g.*  $\rho = 1.185 \text{ kg/m}^3$  and  $\mu = 1.831 \times 10^{-5} \text{ Pa.s}$ ). The state equations are solved on structured fine grid near to the wall regions ( $y^+ < 5$ ) while unstructured grid is built for the other regions. Simulations are based on Reynolds-Averaged Navier-Stokes theory using the Shear Stress Transport  $k-\omega$  turbulence model. For all the computations, a nil velocity is imposed at all the nodes located on the gear tooth surfaces. Periodic conditions are applied to the two lateral surfaces where the fluxes of all variables are supposed to be identical. A velocity profile proposed in ref. [9] is imposed at the perpendicular surface near the tooth and an opening condition is imposed to the other external surfaces. Convergence is reached when both the equation residuals are low and the drag torque caused by windage keeps nearly constant amplitude.

## 3 Numerical results

The comparison between numerical predictions and experimental data is satisfying (Figure 1.a). Relative error deviation reaches 5 percent at maximum; this validates the numerical approach. One observes that due to the helix angle the airstream enters axially by one (suction) flank, a circulation then takes place in the intertooth, and this flow is finally mainly ejected by the other (expelling) flank (Figure 1.b). A small part of the ejection is noticed in the same region but above the tooth. This flow pattern is totally different from the one observed when

considering spur gear [6,7]. Indeed for this latter case the fluid ejection only comes from a region which is located in the middle of the gear width. A high static pressure region is observed here near the suction flank which highly contributes to the drag torque effect. When modifying tooth geometrical parameters one notices that both helix angle and width values have an influence on windage power loss (see WPL in Figure 2). One mainly notices that maximum values of WPL are observed when increasing these parameters (Figure 2.a). However when helix angle is maximum (*i.e.*  $\beta = 30^\circ$ ) this phenomenon is not totally recovered since the maximum values of WPL are observed both for minimum and maximum width tooth (*i.e.*  $WPL = 1607 \text{ W}$  when  $b = 20 \text{ mm}$ , and  $WPL = 1696 \text{ W}$  when  $b = 60 \text{ mm}$  where  $b$  is the tooth width). Furthermore, when it is not null, the helix angle value has a linear influence on WPL (Figure 2.b) which is all the more important as the width is low. The streamlines are similar as previously whatever both the helix angle and the width except for the case for which maximum helix angle value and minimum width value are used together (respectively  $\beta = 30^\circ$  and  $b = 20 \text{ mm}$ ). For this latter case the fluid is ejected only by the expelling flank region and no ejection takes place above the tooth.

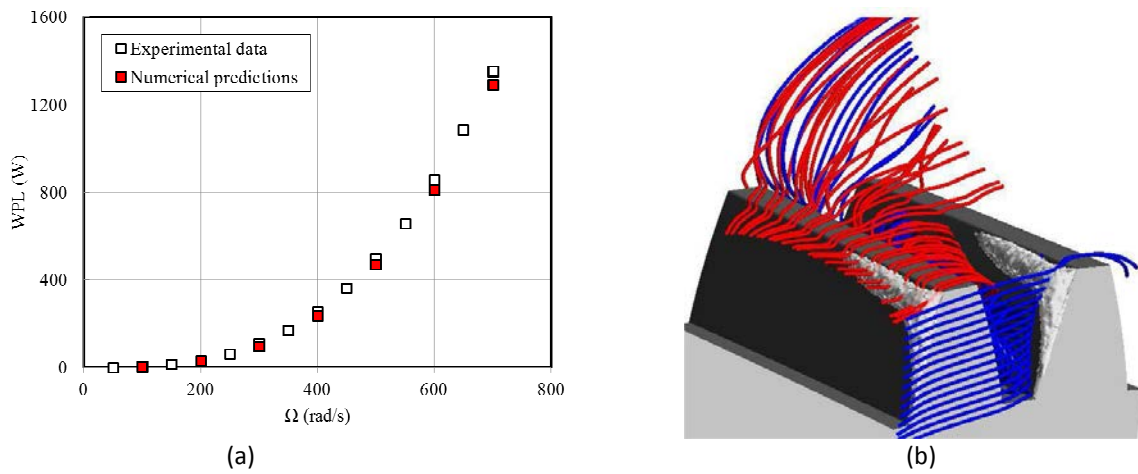


Figure 1: a, windage power loss (experimental data [8] and numerical predictions); b, streamlines in the tooth region and isosurface of a high pressure region.

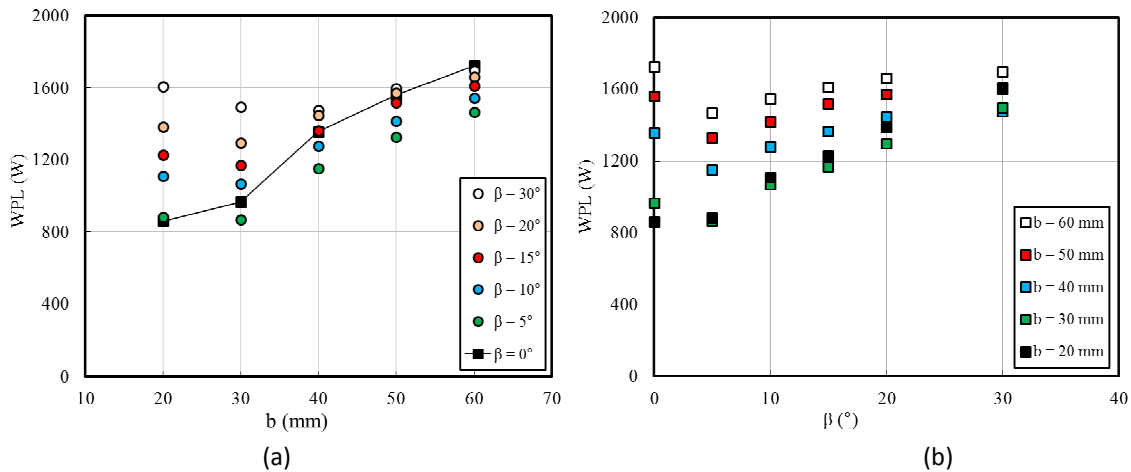
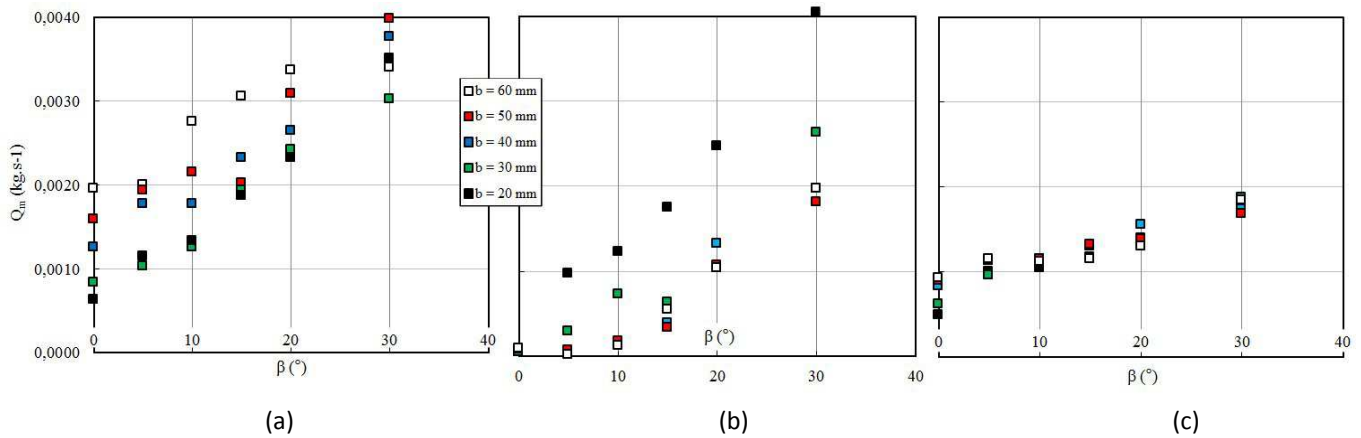


Figure 2: windage power loss evolution with width (a) and helix angle (b) when angular velocity equals 700 rad/s.

An analytical model which is based on classical approach used in turbomachinery has been proposed in previous study [10]. Hence volumetric flow rate which appears in the tooth region seems to play an important role in WPL. One notices here that as the helix angle value increases most of the volumetric flow rates involved in one intertooth region become more important (Figure 3). On the other hand the outlet volumetric flow rate passing through the tip radius surface remains nearly unchanged with a modification of the helix angle value.

The volumetric flow rate getting into the suction flank surface does not seem to depend on width value what is not observed when considering the other surfaces.



**Figure 3: inlet volumetric flow rate passing through the intertooth surface at tip radius (a), outlet volumetric flow rate passing through the expelling flank (b), and inlet volumetric flow rate passing through suction flank (c) – angular velocity equals 700 rad/s (these values do not represent the entire gear contribution but one single tooth).**

## CONCLUSIONS

This paper shows that the previous numerical method (*i.e.* CFD approach) which has been used for spur gear is still satisfying when considering helical gears. The influence of both width and helical angle values is noticed. As it was the case for the spur gears their contribution is complex. When increasing these two previous parameter values one mainly observes an increasing of both WPL and volumetric flow rate involved in the region near the tooth. The further investigation aims then at building a model for the volumetric flow rate in order to complete the previous WPL analytical model which is based on turbomachinery approach.

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