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# Experimental Investigations About the Power Loss Transition Between Churning and Windage for Spur Gears

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*Oil sump lubrication is commonly used in gearboxes. When considering consistent speeds, oil immersion is usually set to low level in order to reduce associated power losses. This configuration is already used in some parts of helicopter mechanical transmissions, and it is under consideration as a lubrication solution for future electric powertrain where gearbox input speeds may be very high. The gear drag power losses are generally evaluated from either a churning power loss model for classic oil sump lubrication or a windage power loss model for oil jet lubrication. One may thus wonder how to estimate drag losses when considering a gear that only a small part is immersed. In this study, the authors investigate the transition between churning and windage phenomena for a spur gear. A series of torque measurements on a single spur gear rotating in an oil bath at numerous oil immersion levels have been carried out. Based on these results, a criterion to indicate which power loss model to use is proposed.*

**Keywords:** windage losses, churning losses, drag losses, power losses, transmissions, gears, gearbox efficiency, friction, gears, lubricants

## 1 Introduction

The design of mechanical transmissions relies on the capability to correctly estimate all of the most significant losses depending on the application. As far as gears are concerned, the load-dependent losses generated by friction are usually separated from the losses independent of load induced by pocketing, windage, and churning. Windage and churning deal with drag losses but are based on two different physical aspects: windage considers a homogenous medium around the moving gears (air or air/oil homogeneous mixture, etc.) while churning is associated with one disturbed free-surface between air and lubricant. When tangential speed is limited, oil sump lubrication method is usually employed inducing churning losses as it is the case for automotive transmission. When now considering high-speed gears, churning power losses would be so great that oil injection using nozzles is preferred, leading then to windage effects. Depending on the application, drag losses are thus calculated using either a windage model or a churning one. For example, Stavitsky et al. [1] and more recently Concli and Gorla [2] have listed both gear windage and churning losses models. Additional information on windage losses can be found in the gear windage review proposed by Eastwick and Johnson [3]. However, recently oil sump lubrication is under consideration for electric powertrain gearboxes. Although those gearbox input speeds may be very high, this type of lubrication is interesting because it meets the constraints of simplicity of the automotive industry. Neourouh et al. [4] showed that it is possible to use splash lubrication with high-speed gears using either gear flanges or low oil immersion. In this latter case, churning and windage phenomena can no longer be treated separately. The purpose of this article is to investigate these configurations that are neither windage nor churning phenomenon uniquely.

The article is organized as follows. Based on literature review, first the description of mechanisms involved in both windage and churning drag losses is given leading to an appropriate definition essentially based on fluid flow instead of fluid physical properties. Then, a campaign of torque measurements is carried out in order to better understand the contribution of each loss. A criterion that indicates if one loss prevails on the other one is finally proposed for spur gears.

## 2 Windage Losses

Windage corresponds to the resistance experienced by a body moving through a fluid of any kind, whether liquid or gaseous, and therefore, work has to be provided to maintain its movement. In the case of gearboxes, windage power losses correspond to the power necessary to entrain air surrounding the rotating parts. It has been seen that those losses can be major in the case of high-speed applications [5]. It is commonly accepted that those losses are significant when gear peripheral speed exceeds 50 m/s [6–8]. In addition to speed, gear geometry plays a major role.

The first substantial study on gear windage losses was carried out in 1984 by Dawson [9]. He measured the resistive torque of several large both spur and helical gears. Based on measured torque using disks and gears, Dawson highlighted the tooth contribution as the major one in the generation of total windage losses in comparison with the viscous power loss induced by the two gear faces (Fig. 1). The loss generated by teeth can represent in his experiments 90% of the total gear windage losses. Later, a series of tests were conducted by Winfree [10] on a spiral bevel gear spinning at very high angular velocity to design baffle that would produce the highest efficiency. A decrease of 78% on windage power losses was observed at best when both the inner and outer

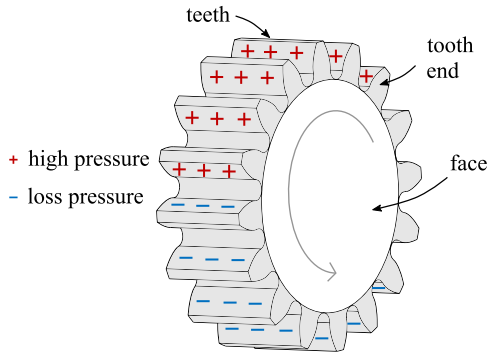


Fig. 1 Origin of the drag on a gear

ends of the gear teeth were taped. This confirms Dawson’s earlier observations on cylindrical gears.

As a consequence of these observations, the windage losses are traditionally broken down into a tooth contribution on the one hand and of both gear faces on the other hand [6,7,9,11]. The total resistive torque ( $T$ ) when written as a function of peripheral speed ( $V_T$ ) reads

$$T = A \cdot V_T^\gamma \quad (1)$$

with  $A$  a constant and  $\gamma$  the power on speed. It can be seen from the experiments that  $1.8 \leq \gamma \leq 2.0$  depending on each contribution [6,7,9,12,13]. Indeed, tooth contribution is generally proportional to speed to the power 2, while the face contribution is found to be proportional to speed to the power 1.8 [14]. This type of decomposition also makes it possible to separate the two physical phenomena of windage: the pressure (or shape) drag and the friction (or viscous) drag. The tooth contribution mainly comes from pressure drag, which is linked to the tooth shape and the pressure differences generated between front and rear tooth surfaces during rotation (Fig. 1). The faces contribution comes from the friction drag generated by the viscous shear which takes place in the fluid entrained by faces. These two drag sources have recently been calculated using computational fluid dynamics (CFD) numerical simulations. A numerical study on spur gears was carried out at first by Hill et al. [15]. Numerical results showed that the pressure drag prevails on the viscous drag in the case of common power transmission gears using sufficiently big teeth to transmit high mechanical power. Hill et al. [15] and Marchesse et al. [16] revealed also the three-dimensional complexity of the flow pattern independent of spur gear geometry and speed: the air from the gear side is drawn axially, circulation then takes place over the entire volume, and finally, the flow is expelled by centrifugal effects (Fig. 2). These results confirm Dawson hypotheses about the direction of the flow around spur gear teeth, and its importance in windage power losses generation [9]. As a result, based on a series of numerical results, Marchesse et al. [17] emphasized the link between tooth depth and windage power losses. Hill et al. [18] investigated several spur gear geometries, and it appears that the dominant physical mechanism contributing to windage losses is the pressure field associated with diversion and impingement of the high-speed relative flow on the leading tooth surface. This is in accordance with the previously mentioned Winfree’s experiments [10] since the pumping effect produced by the bevel gear rotation is mitigated by the stop of axial suction on both sides of the teeth.

Those windage flow mechanisms that are described earlier are valid to gears surrounded by any kind of homogenous medium. As an example, Dawson [19] used water in his experiments instead of air and noticed that the torque was also proportional to rotational speed to the power 2.0. Later, Gorla et al. [20] investigated experimentally and numerically the power losses generated by a rotating gear in a gearbox fully filled with oil and therefore without any free-surface inside. In this configuration, results

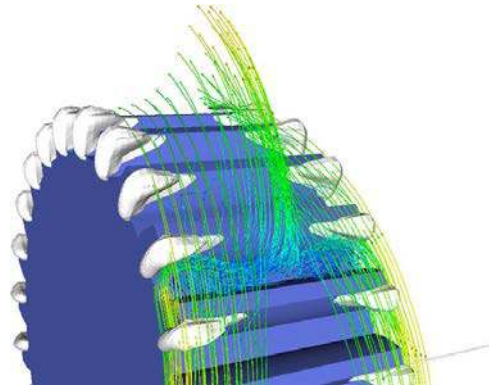


Fig. 2 Predicted relative streamlines and isosurface of high pressure region using the CFD method. (Reproduced with permission from Hill et al. [15]. © 2008 by American Helicopter Society.)

indicate that the torque is likely to be proportional to  $V_T^\gamma$  with  $\gamma \geq 1.9$ . It will be seen in Sec. 3 that  $\gamma$  value dramatically reduces when churning phenomenon occurs.

### 3 Churning Losses

For gearboxes using splash lubrication, an oil bath forms with surrounding air a free-surface that is highly disturbed by the regular passage of gear teeth of the partly immersed gear. Since the medium around the gear is no more homogeneous, windage point of view does not stand. Oil density is much greater than air density so that churning losses can be significant even at low speed, contrary to windage losses. The free-surface position is often expressed as an oil immersion level, the latter being defined as the dimensionless ratio of the immersion height ( $h$ ) of the gear in the bath and either the pitch or the outside radius of the gear. In this study, the outside radius ( $R_0$ ) is taken as the reference radius (Fig. 3). Papers in literature show churning losses increasing with oil level [21–27]. Polly et al. [27] measured the power loss of cylindrical gears rotating in oil bath for a wide range of oil levels. It appears that an almost linear relationship is measured between power loss and oil static level.

It appears from previous studies that geometrical parameters having a first-order influence on churning losses can be reduced to gear external dimensions such as diameter and width. In 1975, Terekhov [21] concluded, as a result of his experiments, that the tooth module had practically no effect on churning losses. Changenet et al. [26] measured resistive torque of eight cylindrical gears and a rotating disc. They observed that a cylindrical disc with the same external dimensions (i.e., diameter and width) as a cylindrical toothed wheel generates similar churning losses. The same conclusion has been drawn on spiral bevel gears by Laruelle et al. [28] as they measured resistive torque of four spiral bevel wheels and two equivalent smooth cones. In addition to power loss, Leprince et al. [29] showed that the oil flow expelled by a spur

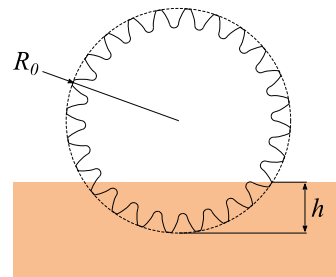
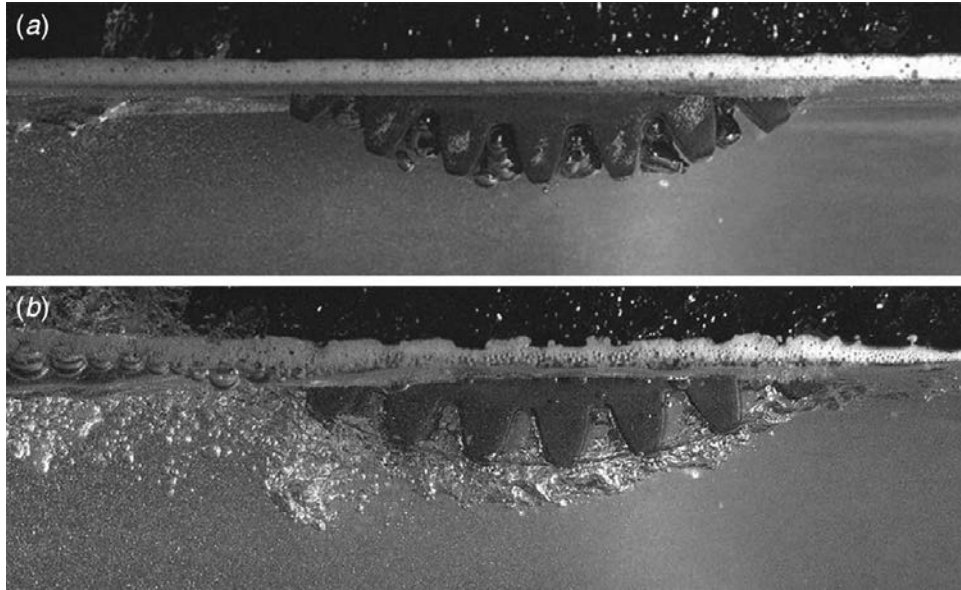


Fig. 3 Geometrical parameters of a gear immersed in an oil bath

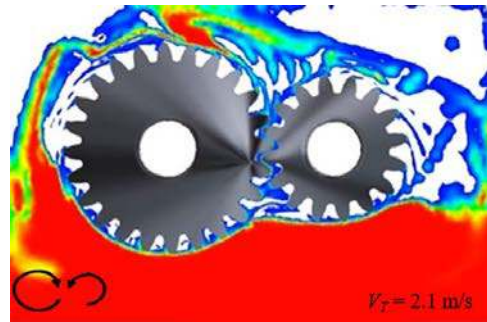


**Fig. 4 Influence of tangential velocity on the air arrangement in the bath: (a)  $V_T = 0.5$  m/s and (b)  $V_T = 2.5$  m/s—photography of the immersed part of a spur gear rotating in a water bath, clockwise rotation ( $h/R = 0.2$  and  $\nu = 1$  cSt). (Reproduced with permission from Kristóf and Várhegyi [30]. © 2016 by Periodica Polytechnica Mechanical Engineering.)**

gear is similar to the one of an equivalent cylinder disc. Recently, Várhegyi and Kristóf [30] setup a facility based on the use of distilled water to photograph the boundary layer of the submerged portion of a spur gear across the bath. It has been noticed that small air bubbles are at first trapped between the submerged teeth at low speed (Fig. 4(a)), and when a critical speed is reached, a boundary layer is created around the gear in addition to the inter-tooth gap completely filled with air (Fig. 4(b)). The free-surface then forms an almost perfect cylinder that is impermeable to oil. Chernoray and Jahanmiri [31] have also visualized the air trapped between teeth of a pinion using a similar device but this time using a bath of transparent mineral oil. Thanks to a particle image velocimetry (PIV) system measuring the speed field in oil bath, they were able to establish that the thickness of the boundary layer is the same for a cylindrical gear and for a smooth cylinder with the same external dimensions. These observations confirm that the churning phenomenon could induce the formation of an air boundary layer between the gear/disc and the oil bath for sufficient speeds as suggested by Luke and Olver [32]. This means that teeth have very little effect on churning losses whenever air bubbles are still trapped between teeth. In addition to the experimental approach, numerical methods have also been investigated to study their capabilities to correctly simulate free-surface flows. Among them volume of fluid (VoF) [33] and smoothed-particle hydrodynamic (SPH) [34] methods are the most employed. For example, Fig. 5 illustrates an instantaneous oil volume fraction distribution generated by a spur gear rotating in an oil bath [35].

Many investigations emphasized the presence of flow regimes in the oil bath depending on oil properties, immersion depth, and gear speed. It has also been observed that the churning torque, when expressed as Eq. (1), evolves with speed at a power so that  $0.47 \leq \gamma \leq 1.0$  [24,26,32,36,37]. This slower evolution in comparison with the one noticed for windage confirms that depending on the environment, i.e., homogeneous or not, physical aspects are totally different for windage and churning losses.

Recently, Neuroth et al. [4] showed that it is possible for electric vehicle transmission to use splash lubrication with high-speed gears using low oil immersion. One may now wonder what happens if sufficiently low immersion depth is considered so that only a small part of the gear is immersed. In this case, experiments exhibit a weak disturbance of the free-surface, and a large tooth part would therefore experience windage-like losses. This would

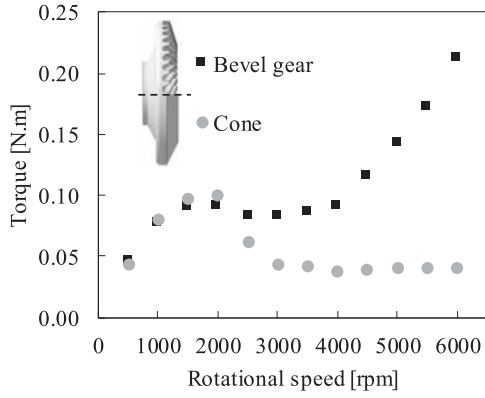


**Fig. 5 Oil fraction simulated by CFD in the symmetrical plane of the spur gear.  $V_T = 2.1$  m/s,  $(h/R)_1 = 0.1$  and  $(h/R)_2 = 0.4$ ,  $\nu = 480$  cSt. (Reproduced with permission from Liu et al. [35]. © 2018 by MDPI.)**

also be the case if a great portion of the oil under the immersed tooth were expelled so that a sufficient quantity of air would be present in this region. Section 4 deals with this discussion.

#### 4 Churning—Windage Transition

Quiban et al. [38] recently pointed out a transition between churning and windage phenomena in their investigations on spiral bevel gears. When studying torque evolution with angular velocity, a constantly increasing evolution is not exhibit as it would be the case in windage effect or churning effect alone. It has been observed that churning power losses at low speed transformed through a transition to windage power losses at high speed (Fig. 6). The authors reported that this transition was initiated by an oil bath free-surface displacement at moderate and high speeds. This left the gear running almost completely in a mono-phasic fluid (i.e., the surrounding air) as the free-surface was pushed away from the gear. The transition is well marked for two reasons: (i) the torque evolutions at low speed and at high speed are different and (ii) the tooth influence is insignificant at low speed but is crucial at high speed (comparison between cone and bevel gear in Fig. 6). The model for estimating drag losses is therefore questionable in a bath lubricated gear, and this problem would stand also for spur gears when



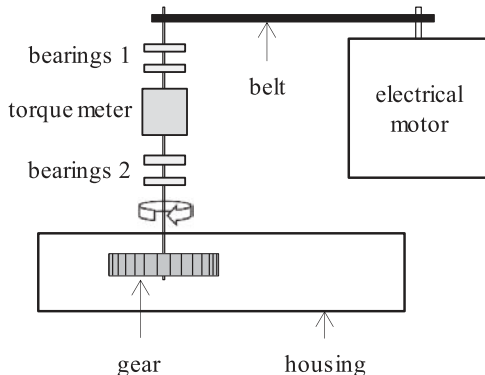
**Fig. 6** Measurements of resistive torque of gear/cone rotating in a boil bath. Highlight of the transition from churning to windage. Data from Quiban et al. [38].

low immersion depths are used. It is then interesting to investigate the influence of immersion depth on the type of losses, i.e., windage losses or churning losses.

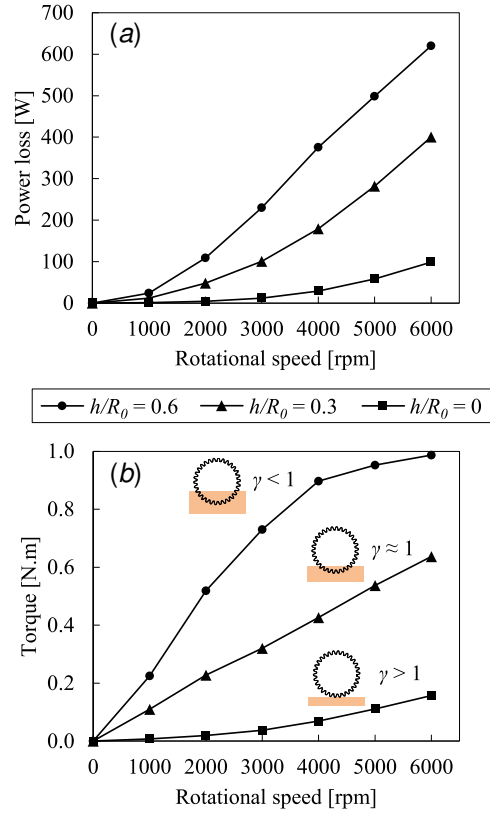
**4.1 Experimental Apparatus.** A detailed presentation of the experimental test rig employed here (Fig. 7) can be found in Ref. [38]. The housing has internal dimensions 394 mm × 266 mm × 100 mm. One face made of Plexiglas let the possibility to visualize oil flows and the immersion can be handled using volumetric pumps. Torque is measured with  $\pm 0.002$  N · m accuracy, and the torque meter is also used to measure speed. In order to eliminate bearing loss contributions, the bearing drag torque is experimentally determined as a function of speed (e.g., it has been found that a pair of bearings generates a drag torque of 0.02 Nm at 3000 rpm) and then subtracted from global torque measurements to isolate the churning net contribution. A thermocouple is used to measure the lubricant temperature and to ensure that thermal equilibrium is obtained.

Measurements were carried out with a single 30-tooth, 5-mm module spur gear of 24 mm face width immersed in MIL-PRF-23699 (NATO O-156) standard aeronautic oil at 40 °C. Corresponding oil kinematic viscosity and density equal 41 cSt and 838 kg/m<sup>3</sup>, respectively. Due to this viscosity value, the influence of temperature is not investigated here since viscous effects are negligible compared to inertial effects as suggested by previous measurements on that gear [26]. Gear rotational speed was varied up to 6000 rpm, and oil immersion was settled at different static levels starting from  $h/R_0 = 0.6$  and decreasing until no oil remains.

**4.2 Experimental Results.** A typical series of results are shown in Fig. 8(a) which gives the power loss as a function of



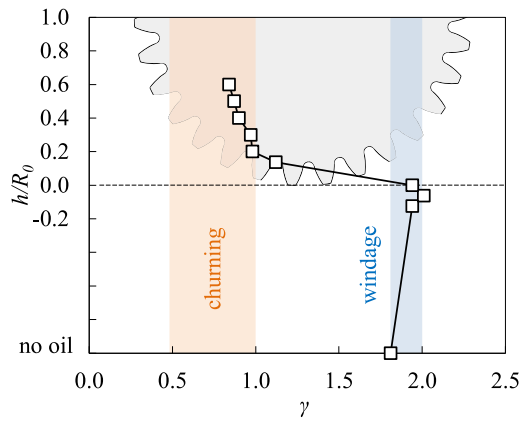
**Fig. 7** Top view of the test rig



**Fig. 8** Evolution of (a) power loss and (b) resistive torque with rotational speed for three different oil immersions

rotational speed for three different static immersion levels  $h/R_0 = \{0; 0.3; 0.6\}$ .  $h/R_0 = 0$  oil immersion corresponds to the free-surface reaching the gear tip edge at rest. Curve evolution with speed is more distinguished between the three oil immersion levels when torque is plotted instead of power (Fig. 8(b)). Next results are thus presented with torque instead of power measurements. The torque evolutions with speed are different depending on the free-surface level. When  $h/R_0 = 0.3 - 0.6$ , the torque is proportional to  $V_7^\gamma$  with  $\gamma \leq 1$  what is close to the one observed by Luke and Olver [32]. This is in accordance with churning phenomenon. They also measured a slight plateau of torque for higher speeds and suggested that it is probably due to windage effect displacing the free-surface such that the immersion depth on the gear was reduced. When  $h/R_0 = 0$ ,  $\gamma$  becomes greater than 1 which is in accordance with windage phenomenon. Oil immersions in the range 0–0.3 are commonly used in gear transmission systems, especially for pinion, which generally dips less than wheel in oil bath. It is therefore possible that choosing one value in this range would lead to a torque evolution in the windage-churning transition. It can be noticed that the torque increases constantly with speed as it was already the case on cylindrical gear churning losses but not observed for bevel gears.

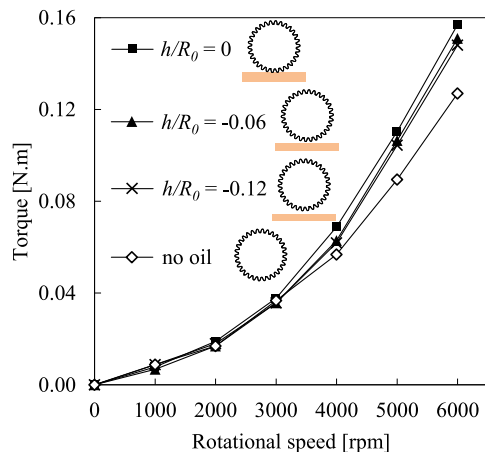
In order to study the transition between churning and windage phenomena, a series of torque measurements as the ones presented in the Fig. 8 have been carried out for numerous oil immersion levels:  $h/R_0 = [0.6, 0.5, 0.4, 0.3, 0.2, 0.14, 0, -0.06, -0.12, \text{no oil}]$ .  $h/R_0 < 0$  means that oil bath level is under the gear tip. For example, for  $h/R_0 = -0.06$ , the oil bath level is 5 mm below the gear tip, and for  $h/R_0 = -0.12$ , the oil bath level is 10 mm below gear tip.  $h/R_0 = 0.14$  is the configuration where one tooth is completely immersed in oil. It can be noted that torque loss values was found to evolve quasi linearly with oil static level in the range  $0.2 < h/R_0 < 0.6$  and at all speeds: for example, at 5000 rpm, the power loss increase equals 165 W when  $h/R_0$  increases from 0.2 to 0.4, which is similar to the increase between



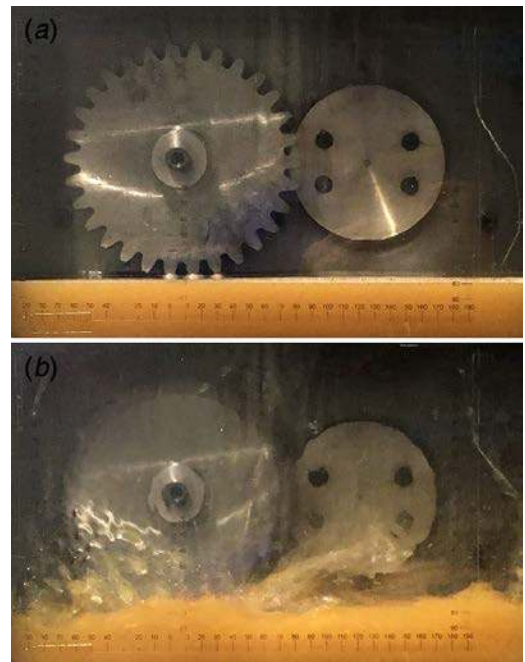
**Fig. 9 Highlight of the transition between churning and windage phenomena based on the values of the power  $n$  in correlation with oil immersion ratio**

$h/R_0=0.4$  and  $0.6$ . This observation is in accordance with Polly's measurements on cylindrical gears [27]. The value of  $\gamma$  in Eq. (1) is estimated with a power regression on torque results from 2000 rpm to 6000 rpm (Fig. 9). It can be noticed that the value of  $\gamma$  is within 0.84 and 2.04 with a distinguishable gap as the oil free-surface passes through teeth level. For oil immersions equal or greater than  $h/R_0=0.2$ , the values of  $\gamma$  are between 0.84 and 0.99 meaning that the torque evolution with speed is in accordance with churning phenomenon. For oil immersions equal or less than  $h/R_0=0$ , the values of  $\gamma$  are between 1.85 and 2.04, in accordance with windage phenomenon. When considering oil immersion between  $0 < h/R_0 < 0.2$ , drag losses cannot be estimated with a churning loss model alone and contribution of both churning and windage should be then employed. This latter has not been previously observed since only few published studies deal with low oil immersion, and authors mostly used relative oil immersions greater than 0.25.

Power  $\gamma$  is observed to vary from the configuration without oil and the one when oil level is below the outside radius (Fig. 9). It means that even if gear tip does not touch the bath at rest, there is still an oil influence on torque evolution. Figure 10 shows torque evolution with speed for oil free-surface levels that are at the edge of gear tip or lower. Although torque evolutions with speed are similar whatever the distance to free-surface from gear tip, an increase of torque values can be noticed as the oil free-surface approaches the gear tip. The torque is increased of a maximal value of 25% at 6000 rpm when oil bath level reaches gear tip.



**Fig. 10 Evolution of resistive torque with rotational speed for oil free-surface level under gear tip**



**Fig. 11 Photography of the gear (a) at rest and (b) rotating at 6000 rpm (50 m/s) with the oil free-surface level set at the edge of gear tip**

This is mainly due to oil projections entrained by air flows as can be seen in Fig. 11. The air in “no oil” configuration is then replaced by an air/oil mixture for the rotating gear increasing the surrounding fluid density. Marchesse et al. [17] demonstrated numerically that windage power losses are proportional to fluid density around rotating gears. As a consequence, when gears rotate near the free-surface of an oil bath, windage loss evaluation should then employ equivalent fluid density to be more realistic instead of using pure air properties.

## 5 Conclusion

The first series of torque was measured for one single spur gear rotating in an oil bath at three oil immersion levels. One observed when  $h/R_0=0.3-0.6$  that torque evolution is in accordance with churning physical aspect noticed in previous investigations. On the contrary, windage power losses were recovered when the free-surface reaches the gear tip edge at rest (i.e.,  $h/R_0=0$ ). The transition between these two types of losses was then investigated in a second series of experiments highlighting three immersion ranges. As it is expected when  $h/R_0 \geq 0.2$  churning losses were measured, whereas for  $h/R_0 \leq 0$  windage losses were identified. It appears now that when  $0 < h/R_0 < 0.2$  both churning and windage physical aspects should be considered to calculate power losses. Moreover, for oil immersion level below the gear tip, windage power losses can be 25% greater than the ones noticed when there is no oil due to oil sump entrainment.

## Conflict of Interest

There are no conflict of interest.

## Data Availability Statement

The data sets generated and supporting the findings of this article are obtainable from the corresponding author upon reasonable request. The authors attest that all data for this study are included in the paper.

## Nomenclature

- $h$  = gear static immersion depth, m  
 $A$  = constant  
 $R$  = gear pitch radius, m  
 $T$  = resistive torque, N · m  
 $R_0$  = gear outside radius, m  
 $V_T$  = peripheral speed, m/s

## Greek Symbols

- $\gamma$  = power on speed for torque evolution with speed (Eq. (1))  
 $\nu$  = kinematic viscosity, m<sup>2</sup>/s

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