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POWER LOSSES OF GEAR SYSTEMS

This paper provides a review of experimental investigations and available models of gear load-independent power losses (windage losses, churning losses and air-oil pocketing power losses) for spur, helical, and bevel gears. The aim of the review is to provide a comprehensive compilation of published information on gear load-independent power losses to assist gearbox designers in identifying relevant experimental and modeling information. While it is clear from the review of published work that the rotational speed, gear geometrical parameters, degree of confinement, and density of the fluid surrounding the gear are important, the degree of effect and general solutions for reducing power loss are less clear. The motivation for this is that for some applications, this power loss can be a significant component, particularly lightly loaded high-speed applications.

Keywords: *transmissions, gears, windage loss, churning loss, air-oil pocketing loss, gearbox efficiency, empirical models*

Introduction. Environmental awareness is pushing mechanical engineers to develop mechanical systems, and in particular gear units, that have lower environmental impact. Such objective can be reached through different perspectives: reduce gear power loss and improve efficiency, reduce gear operating temperature, reduce friction between gear teeth, reduce gear load-independent power losses. The losses associated with meshing gears are important in the design of many industrial, marine, and gas turbine situations. Gearbox efficiency varies from 98% to 99% for the best designed high power applications. The highest rated gearboxes now exceed 100 MW [1], so for such a gearbox a 1 per cent power loss equates to 1 MW and this is not insignificant.

Power losses of a gearbox containing several gear pairs that are supported by shafts and rolling element bearings can be classified into two groups. The first group is comprised of load-dependent (friction-induced) power losses caused primarily due to contacting surfaces of gears and the bearings. The losses in the second group are independent of load and are often referred to as spin power losses. There are many sources of such losses, the primary ones being oil churning and windage that are present as a result of oil/air drag on the periphery and faces of the gears, pocketing/squeezing of lubricant from the cavities of the gear mesh, and viscous dissipation of bearings. While losses from these two groups are often comparable under high-load, low-speed conditions, the spin losses were shown to dominate over the load-dependent power losses at higher operating speed conditions. Of the total losses, for a typical gearbox, 40 per cent come from meshing, 50 per cent from bearings, and 10 per cent from windage and churning [2]. Windage power loss (WPL) 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. Churning power loss (CHPL) is defined as the power loss when a gear is running in an oil bath or is dipping into oil "slugs." During the meshing of high-speed spur or helical gears, the mixture of air and lubricant is successively compressed and expanded in the intertooth spaces giving rise to significant heating and power loss named as air-oil pocketing power losses.

Windage Power Losses. Estimates on the percentage effect of windage vary as the value is dependent on a number of different parameters. One of the critical parameters is the pitch line velocity; obviously, high velocities ([3] suggests 51 m/ s, and [4] suggests tangential speeds greater than 90–120 m/ s) produce greater stirred motion, so large gears rotated at high rotational rates are particularly vulnerable. Additionally, the lubrication flow rate and scavenge design are critical as these directly affect the properties of the fluid surrounding the gear [3], [5]. So, in a case where you have a high level of lubricant suspended around a gear with high pitch velocity, as is the case of an aero-engine, windage becomes a significant contributor to the power loss. It may only account for a few percent, but this can be critical. Traditionally, there have been two approaches to reducing WPL; the first is to use a shroud or baffle to enclose gears and the second is to positively pump the oil and air from the gearbox casing. Pumping from the gearbox casing is known as evacuating the gearbox [1], and allows a reduction of fluid density within the casing. This can result in up to 1% improvement in efficiency but can only be used in a limited number of situations.

Published gear windage power losses experiments, which have been few and far, can be grouped based on their primary focus. One group of studies focused on the measurement of air windage losses [6] by measuring the deceleration of a single gear or disk rotating in air, and then applying the kinetic energy theorem to obtain power loss from air drag. [7] used the tool of dimensional analysis to define an empirical windage moment coefficient in terms of speed, oil properties, Reynolds number, gear size, tooth parameters, and the geometry of nearby fluid flow obstructions such as close-fitting gear case walls. While applicable to a single disk or gear rotating in air, these models did not consider the effects of a meshing gear or impinging oil jet, and so cannot be validated using geared transmissions.

As an alternative, [8] and [9], [10] developed empirical models for meshed spur gears based on pitch radius, face width, rotational speed, and viscosity of the ambient fluid. Likewise, [11], based on measurements of high-speed, long addendum spur gears, reported that windage power loss was proportional to the 2.8th power of the rotational speed and also that the inertial losses resulting from the impinging oil jet were linearly proportional to the rotational speed. Here, the inertial losses from the impinging oil jet were shown to increase with oil jet pressure, and composed a significant portion of load-independent power loss.

Table 1 is divided into experimental and modeling studies into WPL for spur, helical, and bevel gears.

Oil churning power losses. Most of experimental studies considered a single gear, disk, or bladed rotor immersed in oil [12], [13], [14]. These oil churning studies were also devoted to developing empirical equations to obtain a dimensionless churning moment coefficient. [12] proposed four different flow regimes around a rotating disk fully submerged in fluid and correlated these flow regimes to Reynolds number and enclosure effects based on experimental results. Mann and Marston [13] studied friction drag of bladed and unbladed disks and related experimental results to a moment coefficient based on Reynolds number and axial clearance with the chamber, etc.

However, in the case of gears, there are fewer empirical models and, because of experimental difficulties, measurements of thermal performance and power losses have been limited. The first in situ temperature measurements date back to the classic works by [15], [16], while the specific studies on churning losses comprise those of [17], [18] and, more recently, [19] and [20].

Table 1

Experimental studies of windage power loss

		Authors	Object	Design parameters test spur gears	Conclusions	Empirical models: Determination of WPL
1	2	3	4	5	6	7
Spur Gears	windage due to single spur gears in isolation	[Dawson, P.H., 1984, 1988]	WPL vs speed, size, gear geometry, and shrouding for single spur gears in air	37 gears of root diameter 300–1160 mm with face widths of 32–187 mm and tooth modules of 2–24 mm	• Main contributors to WPL were the teeth. • Teeth may be acting as a centrifugal fan drawing in air axially at the ends of the teeth and ejecting it radially toward the middle of the face.	$N^{2.9}(0.16d^{3.9}+d^{295}b^{0.75}m^{1.15})*10^{-20}\Phi\,\lambda$ Φ=1-oil free atmosphere, λ=0.6-0.7-gear in free space, λ=0.6-0.7-gear in a large enclosure, λ=0.5–0.6-"fitting" gear case. $1.12*10^{-8}C^{*}\rho N^{2.85}d^{4.7}\nu^{0.15}\lambda$ C^{*} is obtained from [Dawson, P.H., 1984].
		[Lord, A. A., 1998]	Axial and peripheral shrouding single and meshed gears at speeds ranging from 0 to 20, 000 rpm.	Pitch diameters 200 mm, face width of 40 mm, and modules 1 and 5 Axial clearance from 1 mm to 10 mm	• Maximum WPL was experienced for a spur gear of Module 4 and that the minimum was seen for that of Module 1. • Shrouding a single gear in air reduces WPL to 25% of their unshrouded values. • With a1 mm peripheral clearance, WPL at high speeds was reduced by 75%. • Relationship between WPL and peripheral clearance was approximately linear.	Module 1 $2.9\rho\omega^{3}r^{3.51}m^{1.06}F^{0.42}$ Module 1.25-4 $2.9\rho\omega^{3}r^{3.51}m^{1.06}F^{0.42}$ Module 5 $2.9\rho\omega^{3}r^{3.42}m^{1.16}F^{0.42}$ The predictions from these expressions were found to lie within $\pm15\%$ of the experimental data.
		[Diab, Y., Ville, F., Changenet, C., and Velex, P., 2004]	Dimensional analysis the gear geometry at speeds ranging from 0 to 12, 000 rpm.	Pitch diameters from 144 mm to 300 mm, face width of 30–60 mm, and modules 4 and 6	• Speed appears as a key factor, since windage losses become very significant at high speeds. • Influence of the teeth is found to be significant with WPL ratio of about 5:1 when comparing a gear and a disk. • Introducing obstacles such as flanges can therefore reduce the air aspiration and ejection by the rotating teeth and modify the corresponding power loss.	$\frac{1}{2}C_t\rho\omega^{3}r^{5}$, with:$C_t=C_f+C_l$, C_f - dimensionless moment at the front and rear faces of the gear, C_l - dimensionless moment at the teeth of the gear = $\xi\frac{z}{4}\left(\frac{b}{r}\right)\left[1+\frac{2(1+x)}{z}\right]^{4}(1-\cos\phi)(1+\cos\phi)^{3}$ $C_f=60Re^{-0.25}\left(\frac{b}{r}\right)^{0.8}z^{-0.4}\left\{\left(\frac{h_1}{r}\right)^{0.56}+\left(\frac{h_2}{r}\right)^{0.56}\right\}$ h_1,h_2 – characteristic the presence of a deflector, a flange

Table 1 (continued)

1	2	3	4	5	6	7
Helical Gears	Closed-loop systems	[Anderson, N. E., and Loewenthal, S. H., 1983]	WPL vs speed, size, gear geometry,		• developed empirical models for meshed spur gears based on pitch radius, face width, rotational speed, and viscosity of the ambient fluid	$2.82 * 10^{-7} N^{2.8}\left(1 + 2.3\frac{b}{r}\right)(0.028\nu + 0.019)^{0.2}$
		[Petry-Johnson, T., Kahraman, A., Anderson, N. E., and Chase, D. R., 2008]	WPL at speeds from 0 to 10, 000 rpm.	Number of teeth 23 and 40, face widths of 14,7; 19.5 and 26.7 mm and tooth modules of 2.32 and 3.95 mm	•The 23-teeth gear pair, with m = 3.95 mm, is predicted to have more windage power loss than the 40-teeth gear pair having m = 2.32 mm. • Variation of the total WPL with face width is primarily due to the windage pocketing power loss	$0.025\pi\rho\omega^{2.86} r_a^{4.72}\mu^{0.14}$
	windage due to single spur gears in isolation	[Dawson, P.H., 1984, 1988]	Effect of vary-ing helix angle for a single gear	Helix angle from 0 deg to 50 deg Diameter of 0.514 m and a face/diameter ratio of 0.364	• power loss reduces with helix angle up to the 50 deg	$1.12 * 10^{-8} C^{*}\rho N^{2.85} d^{4.7} \nu^{0.15} \lambda$ C^{*} - shape factor, related to the number of teeth, face width to diameter ratio of the gear
		[Handschuh, R. F., and Kilmain, C. J., 2003]	efficiency of high-speed helical gear trains	Speeds were varied up to 15,000 rpm and loads up to 3.7 kW Number of teeth 50, 51 and 139, face widths of 67,2 mm and tooth mod-ule of 3.033 mm, helix angle 12 deg	• High gearing system rotational speed has a drastic effect on the efficiency of high-speed gear trains. • Windage losses will dominate the performance when light loads and high speed is applied to the gear meshing system. • When their gear system was operated in the range of 33–100% of full load and 83–100% full speed, the losses and, thus, efficiency were significantly influenced by windage, with WPL being nearly equal or exceeding those due to meshing.	$32C_2N^3r^5 + 16C_3N^3r^4b\left(\frac{R_f}{\sqrt{\tan\beta}}\right)$ C_2, C_3 – constants; R_f - is the rough surface adjustment factor Dawson's [Dawson, P.H., 1984, 1988] and Townsend's [Townsend, D. P., 1992] empirical models with correlations for windage, but made no comment on the accuracy of the correlations as their experiments measured total power loss rather than isolating windage.

Table 1 (continued)

1	2	3	4	5	6	7
		[Lord, A. A., 1998]	windage and meshing losses	Pitch diameters 200 mm, face width of 40 mm, and modules 1 and 5 Axial clearance from 1 mm to 10 mm	• WPL for a helical gear was lower than for a comparable spur gear. • reduction in WPL was due to the air motion around the gear teeth, with the flow into and out of spur teeth requiring a change in direction of 90 deg while the helical gear required a shallower change of direction.	Module 1 $2.9\rho\omega^3 r^{3.51} m^{1{,}06} F^{0.42}$ Module 1.25-4 $2.9\rho\omega^3 r^{3.51} m^{1{,}06} F^{0.42}$ Module 5 $2.9\rho\omega^3 r^{3.42} m^{1{,}16} F^{0.42}$
		[Diab, Y., Ville, F., Velex, PM., 2005]	influence of the proximity of a flange	Pitch diameters 288 mm, face width of 30 mm, and modules 4 mm, helix angle 15 deg	• The influence of rotation direction was noted, with a reduction of 38% noted for anticlockwise rotation for the best performing flange gap, while a 15% reduction was seen in the clockwise direction.	$\xi \frac{z}{4}\left(\frac{b}{r}\right)\left[1+\frac{2(1+x)}{z}\right]^4 (1-\cos\phi)(1+\cos\phi)^3(1-\sin^2\beta)$
Bevel Gears		[Winfree, D. D., 2000]	shrouding (baffling or containment) of a single spiral bevel gear	15 in. gear constructed from a rigid urethane compound	• Gears traveling over 51 m s must be shrouded. • Spiral bevel gear rotating direction has no effect on baffle design and that bevel gears pump from the inside to the outside across the tooth • Closing the gear inlet end reduces windage, churning, and power usage.	$\frac{v \times i}{745.7}$ with i - current (amps), v - voltage (volts). $C_f = \frac{P_W \times 10^{17}}{N^3 d^5 b^{0.7}}$ - windage correction factor

[17] developed empirical relations for a dimensionless moment coefficient from numerous experiments on gears rotating partially submerged in a fluid and identified separate power loss equations for meshed gears rotating upward or downward in an oil bath. [19] conducted friction torque tests with a simple bench setup using smooth disks of various diameters and face widths, which were partially submerged in high-viscosity oil, and compared these results to experimental observations with a gear (See table 2).

More recent efforts using similar methods include that by [21], [22], and [20].

[22] performed a number of experiments to determine churning loss in single and meshed spur gear pairs. They compared their experimental observations on spin power losses with the empirical formulations of [19] and [17] and found that contrary to what Bones had predicted, the spin power losses were not strongly affected by the viscosity of the lubricant. Furthermore, their observations called into question the attempt used to characterize spin power loss based on a Reynolds number dependent on lubricant viscosity.

[8] measured losses from jet-lubricated spur gear systems experimentally. They proposed an analysis of the power required to pump the oil trapped between mating gears.

[5], [23] analyzed the effect of rotationally induced windage on the lubricating oil distribution in the space between adjacent gear teeth in spur gears. The purpose of their study was to provide formulations to study lubricant fling-off cooling. They proposed that impingement depth of the oil into the space between adjacent gear teeth and the point of initial contact was an important aspect in determining cooling effectiveness.

[24] analyzed fluid flow in the meshing zone between spur gear pairs to assess the magnitude of the fluid velocity, temperature, and pressures that result from meshing gear teeth.

A more recent study by [20] investigated the influence of meshing gear on oil churning power losses by performing a number of gear oil churning experiments to come up with empirical formula for power losses. Parameters included were gear module, diameter and face width, speed, and lubricant viscosity. Their empirical formula (See table 2) suggested that the influence of viscosity on oil churning loses is insignificant with regard to viscosity at high speeds of rotation for single gears, corroborating similar findings from the experimental observations of [22].

Another relevant work by [21] also stresses this apparent lack of dependence of oil type on load-independent losses. In their experiments, measured gear and bearing power losses, and forged a balance between generated heat in the gearbox due to gears and bearings and the dissipated heat in the form of free and forced convection and through radiation as well, from housing and rotating parts, to calculate mean lubricant temperature.

Conclusions. This review describes a number of studies that have investigated gear windage and churning power loss. While it is clear from all of these investigations that the rotational speed, gear geometrical parameters, degree of confinement, and density of the fluid surrounding the gear are important, the degree of effect and general solutions for reducing power loss are less clear. The majority of the modeling methodologies are experimental correlations derived from specific experiments that have unique elements, making a general conclusion regarding the best methodology difficult. The methodologies do allow a general assessment of the expected levels of gear windage and churning present in a specific design and possible routes to reducing gear windage and churning power loss. It is clear from this review that a modeling methodology capable of being used for all gear types and configurations is required, which allows analysis of the fluid dynamics phenomena.

Table 2

Experimental studies of churning power loss

Authors	Object	Design parameters test spur gears	Conclusions	Empirical models: Determination of CHPL
[Terekhov, A. S., 1975]	Dimensional analysis the gear geometry.	High viscosity lubricants (from 200 to 2000 Cst), Low rotational speeds, and tested gears with modules ranging from 2 to 8 mm.	• Churning power loss is independent of gear tooth geometry. • Churning power loss depends on the flow regime. • Churning power loss were not strongly affected by the viscosity of the lubricant.	$P_{ch} = \rho\omega^3 br^4 C_m$ For laminar flows (10< Re <2250), if $Re^{-0.6}Fr^{-0.25} > 8.7 \times 10^{-3}$ $C_m = 4.57Re^{-0.6}Fr^{-0.25}\left(\frac{h}{r}\right)^{1.5}\left(\frac{b}{r}\right)^{-0.4}\left(\frac{V}{V_0}\right)^{-0.5}$ otherwise $C_m = 2.63Re^{-0.6}Fr^{-0.25}\left(\frac{h}{r}\right)^{1.5}\left(\frac{b}{r}\right)^{-0.17}\left(\frac{V}{V_0}\right)^{-0.73}$ For turbulent flows (2250< Re <36,000) $C_m = 0.373Re^{-0.3}Fr^{-0.25}\left(\frac{h}{r}\right)^{1.5}\left(\frac{b}{r}\right)^{-0.124}\left(\frac{V}{V_0}\right)^{-0.576}$
[Lauster, E., and Boos, M., 1983]	CHPL vs speed, immersion depth, gear geometry	Specific case of truck transmissions	• Churning power loss is independent of gear tooth geometry and flow regime.	$P_{ch} = \rho\omega^3 br^4 C_m$ $C_m = 2.95Re^{-0.15}Fr^{-0.7}\left(\frac{h}{r}\right)^{1.5}\left(\frac{b}{r}\right)^{-0.4}\left(\frac{V}{V_0}\right)^{-0.5}$
[Boness, R. J., 1989]	drag torque generated by discs and gears rotating in water, or in oil	Disks of various diameters and face widths	• Drag torque increases with an increasing Reynolds number. • Low viscosity lubricants generate higher losses.	$P_{ch} = \frac{1}{2}\rho\omega^3 S_m r^3 C_m$ For laminar flows (Re <2000) $C_m = \frac{20}{Re}$ For laminar flows (2000< Re <100,000) $C_m = 8.6 \times 10^{-4} Re^{1/3}$ For turbulent flows (100,000< Re) $C_m = \frac{5 \times 10^8}{Re^2}$
[Changenet, C., and Velex, P., 2007]	100 experiments influence of temperature on lubricant viscosity	Number of teeth 20 - 102, face widths of 14 and 24 mm and tooth modules of 1.5 - 5 mm	• To minimize energy losses, one possible way to improve churning losses consists in changing the shape of the casing. • Total loss is not equal to the sum of the individual losses associated with the pinion and the gear when considered apart	$P_{ch} = \frac{1}{2}\rho\omega^3 S_m r^3 C_m$ If $\omega rb/\nu = Re < 6000$ $C_m = 1.36Re^{-0.21}Fr^{-0.6}\left(\frac{h}{d}\right)^{0.45}\left(\frac{V_0}{d^3}\right)^{0.1}$ If $\omega rb/\nu = Re > 9000$ $C_m = 3.644Fr^{-0.88}\left(\frac{h}{d}\right)^{0.1}\left(\frac{b}{d}\right)^{0.85}\left(\frac{V_0}{d^3}\right)^{-0.35}$

Nomenclature

d = pitch diameter (m)
b = face width (m)
m = tooth module
N = rotational speed (rpm)
Re = Reynolds number
h = immersion depth of a pinion (m)
ρ = density
i = current (amps)
μ = dynamic fluid viscosity
β = helix angle (deg)
r = pitch radius (m)
z = number of teeth
ω = speed (rad/s)
Fr = Froude number
S_m = immersed surface area of the pinion (m^2)
V_0 = oil volume (m^3)
v = voltage (volts)
υ = kinematic viscosity (m^2/s)

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ВТРАТИ ПОТУЖНОСТІ В СИСТЕМАХ ЗУБЧАСТИХ КОЛІС

У статті наведено огляд експериментальних досліджень та наявних моделей втрат потужності (вентиляційні втрати, втрати на вихороутворення, втрати потужності при стисканні повітряно-мастильної суміші) для циліндричних, косозубих та конічних передач. Метою огляду є надання всебічної компіляції опублікованої інформації про втрати потужності незалежно від навантаження, щоб допомогти розробникам редукторів визначити відповідну експериментальну та модельну інформацію. Хоча з огляду опублікованої роботи зрозуміло, що важливі швидкість обертання, геометричні параметри передач, ступінь ущільнення та щільність рідини, що оточує передачу, ступінь ефекту та загальні рішення для зменшення втрат потужності менш зрозумілі. Сенс полягає в тому, що для деяких варіантів використання ця втрата потужності може бути важливою складовою, особливо для легко завантажених високошвидкісних механізмів.

Ключові слова: передачі, зубчасті колеса, вентиляційні втрати, втрати на вихороутворення, втрати потужності при стисканні повітряно-мастильної суміші, коефіцієнт корисної дії редуктора, емпіричні моделі

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