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Optimizing Gear Geometry for Minimum Transmission Error, Mesh Friction Losses and Scuffing Risk Through Computer Aided Engineering

By R.C. Frazer and B.A. Shaw,
Newcastle University, and
D. Palmer and M. Fish, Dontyne
Systems Ltd

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Minimizing gear losses caused by churning, windage and mesh friction is important if plant operating costs and environmental impact are to be minimized. The paper concentrates on mesh friction losses and associated scuffing risk. It describes the preliminary results from using a validated 3D Finite Element Analysis (FEA) and Tooth Contact Analysis (TCA) program to optimize cylindrical gears for low friction losses without compromising Transmission Error (TE), noise and power density. Some case studies are presented and generic procedures for minimizing losses are proposed. Future development and further validation work is discussed.

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Robert C. Frazer and B.A. Shaw, Newcastle University
and David Palmer and Michael Fish, Dontyne Systems Ltd

Introduction

Cylindrical involute gears have many advantages over other gears. They are relatively easy to manufacture with standard tools, insensitive to center distance change, can accommodate modifications in micro-geometry to account for elastic deflection and manufacturing errors, have geometry that is mathematically straight forward and relatively easy to measure. Standards covering the rating and analysis of cylindrical gears such as the ISO 6336 suite of standards and ANSI/AGMA 2101-D04 are well developed and applied worldwide.

Users of cylindrical gears demand continuous improvement such as increased power density, lower weight, reduced manufacturing costs, reduced noise, increased reliability and reduced operating costs. In recent years it has become more important to reduce environmental impact from plant operation. Cylindrical involute gears are inherently very efficient, typically 98–99.5% per mesh. However, small improvements in efficiency will minimize overall system losses and reduce lubricant and cooling system requirements.

Gearbox losses occur from a number of sources:

- Churning losses – due to lubricant agitation are geometry and speed (pitch line velocity) dependent. These can be minimized by using spray lubrication, dry sumps and internal gearbox baffles to minimize gear immersion as well as using smaller module and higher helix angle gears.
- Windage losses – are geometry and speed dependent and can be minimized by modification to geometry (smaller module and higher helix angle) running in partial vacuums or light gases.

- Mesh friction losses – which are affected by speed, load, coefficient of friction and gear geometry.
- Bearing losses – which are affected by speed, load and gear geometry.
- Seal losses – seal type and speed dependent.

The work described in this paper concentrates on mesh friction losses. In a 2 MW installation, losses of between 10 to 40 kW occur with efficiencies of 98–99.5%. This power is lost as heat which requires external cooling systems and control systems adding to the cost of the plant. Lower gear friction improves operational efficiency and if applied carefully it will reduce plant manufacturing costs and lubrication requirements.

Minimizing the mesh friction loss is of particular importance because it will also reduce the risk of scuffing failure and help to eradicate the need for lubricant additive packages which are costly and environmentally harmful. Scuffing risk is difficult to assess and although there are two ISO Technical Reports [5,6] and the ANSI/AGMA standard 6011-I03, annex B [7] published on the subject, the safety factors that result from the analysis procedures often conflict, reducing confidence in the results and evaluation procedure. The accurate modelling of mesh friction in gears including micro-geometry correction, manufacturing and alignment errors and accounting for elastic deflection under load is therefore important to minimize the mesh friction losses in cylindrical gear design and improve gear reliability.

The preliminary results from the development of a friction loss model are described in this paper. The work to model and minimize mesh friction losses was undertaken as part of a wide ranging project funded by the European Union named X-GEAR. This targeted specifically Wind Turbine and Automotive gear applications. However, the results from

this work are generic and applicable to all cylindrical gear transmissions.

Background

Gear mesh friction

Figure 1 illustrates that as the tooth pair moves along the line of action, the combination of sliding and rolling changes throughout the mesh cycle. Pure rolling occurs at the pitch point (point C in Figure 1), but as contact moves away from the pitch point sliding increases. Meshing gear pairs require a lubricating film to separate the gear surfaces but if

this film breaks down, a failure mode called scuffing occurs as illustrated in Figure 2. When the lubrication film breaks down the gear tooth surfaces instantaneously weld together and are then pulled apart due to the combination of rolling and sliding that occurs during the mesh cycle. Gear lubricants have been developed over many years to prevent this with EP additives, generally sulphur based, that bond to the gear surfaces and thus prevent metal to metal contact. However the additive packages make gear oil unpleasant to handle and significantly increase the environmental impact of the lubricant when it is disposed.

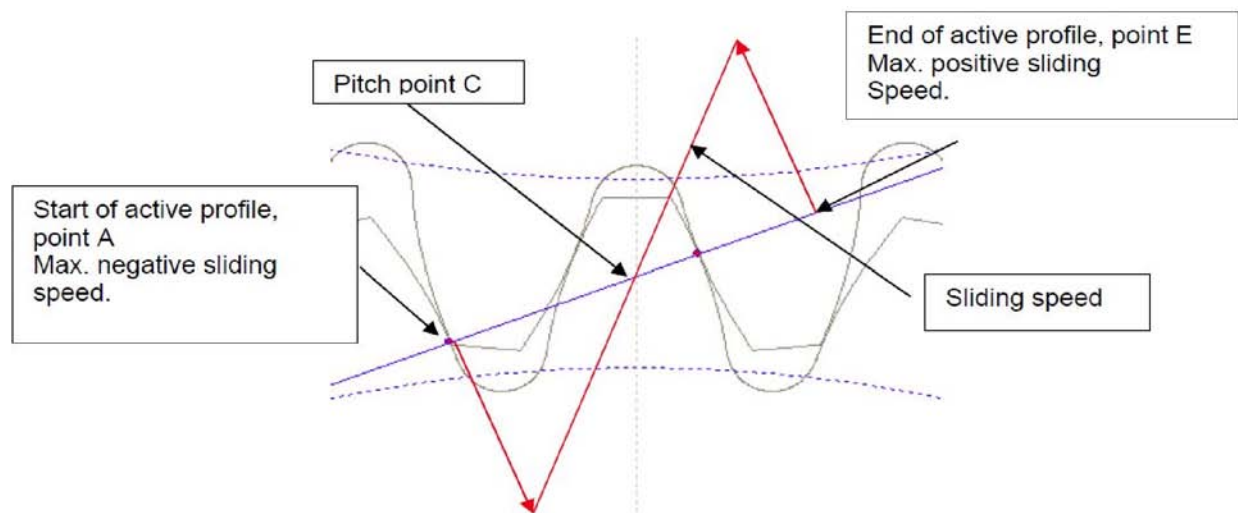


Figure 1. Variation in drive gear sliding speed with mesh phase (position)

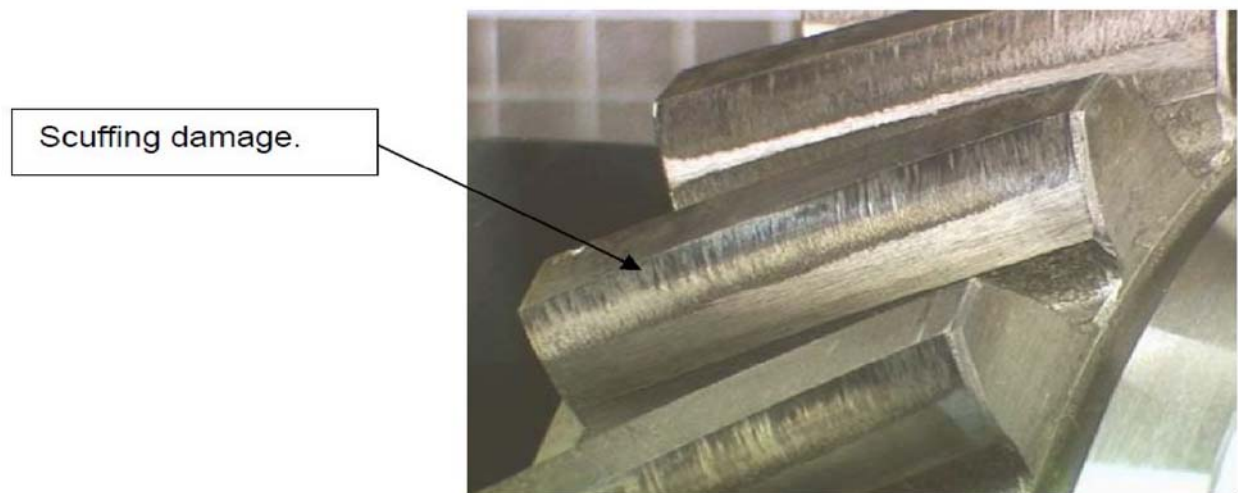


Figure 2. Scuffed gear sample from Design Unit's 160mm center test rig

Gear scuffing is difficult to predict but several standards (DIN 3991, ISO/TR 13989-1 and ISO/TR 13989-2 [5,6,7]) provide procedures to estimate a safety factor for scuffing. These procedures use the gear macro-geometry, the calculated load distribution factors from the gear accuracy, and the estimated shaft deflections to estimate scuffing risk. The standards provide good general guidance but fail to consider the important effect gear micro-geometry has on the local tooth surface loads and thus the localized scuffing risk. The micro-geometry of a gear is the intentional departure from a standard gear form that can optimize the performance of a gear by improving load distribution and minimizing transmission error and noise by compensating for deformation and misalignments present in all loaded systems. It is important that this is considered as part of the modelling process.

Minimizing friction losses in gears is straight forward in principle:

- Minimize sliding speed [m/s] by reducing the height of the gear teeth, either by using a smaller module or simply reducing the addendum and dedendum of the gear, sometimes known as a stub tooth gear form.
- Reduce the peak tooth loads by applying flank corrections for calculated elastic deflections and improving the gear accuracy and alignment within the gear case.
- Reduce self induced dynamic loads by minimizing transmission error.
- Reduce the mesh friction coefficient by improving lubricant additives, surface finish and applying low friction coatings.

In practice, consideration has to be given to balance the requirements of low friction loss gears with key requirements of maintaining power density, reliability, low noise, low cost and minimize sensitivity to manufacturing and alignment errors.

Friction power loss

Friction power losses (P_L) in gears are dependent on the normal force (F_N), coefficient of friction (μ) and the sliding speed of the surfaces (v_g) shown in equation 1.

$$P_L = \mu N v_g \quad (1)$$

Each of these quantities vary through the mesh cycle, depending on the instantaneous position of the mesh point between A (start of active profile) and E (end of active profile) shown in Figure 1, but they also vary across the face width (b) as discussed below:

- The normal force (F_N) depends on the number of teeth in mesh (affected by transverse contact ratio and overlap ratio) and load distribution due to manufacturing errors and elastic deflection of the gear teeth, the gear shafts and housing. The actual load is required at each phase of the gear mesh along the path of contact (x) from (A-E) in Figure 1 to accurately estimate gear losses.
- The coefficient of friction (μ) will vary with sliding speed. It is likely that the coefficient of friction will be higher at the entry to the mesh and the static friction at the pitch point will be different.
- The sliding speed (v_g) varies linearly with distance along the path of contact from A to E, with $v_g = 0$ at the pitch point 'C' in Figure 2. This can be calculated directly.

Thus the equation derived for power loss (W) in real gears, with geometry errors, flank relief and elastic deflection effects is given by equation 2.

$$P_L = \frac{1}{b P_{et}} \int_0^b \int_A^E [\mu(x) F(x) v_g(x)] dx dz \quad [W] \quad (2)$$

where

P_{et} base pitch (transverse), mm
 $\mu(x)$ coefficient of friction
 $F(x)$ local mesh load, N
 $v_g(x)$ sliding speed, mm/s
 b facewidth, mm

The accurate assessment of equation 2 requires accurate knowledge of the instantaneous mesh load $F(x)$. This is beyond the scope of standard gear stress analysis procedures defined in ISO 6336 (gear stress analysis standard) as detailed mesh deflection values, gear geometry (including predicted errors and the gear designers specified helix and profile correction), and dynamic loads are not accurately modelled in the standard.

Existing work

There has been much work in recent years to minimize mesh friction. The EU funded research project

'Oil Free Powertrain' (IPS-2001-CT-98006), a project coordinated by the VDMA in Germany, completed a systematic review of gear losses with the ambitious target of producing a lubricant free powertrain [1]. The effect of geometry parameters, namely module (tooth size), addendum modification, ratio, helix angle, pressure angle and face width on total gear losses was investigated and the results showed:

Module (M_n): reducing the module reduces sliding speed but increases root bending stress. This effect can be minimized by controlled shot peening or increasing the helix angle.

Pressure angle (α_n): increasing the pressure angle reduces the contact stress, but reduces the transverse contact ratio which can increase the mean tooth load and bearing loads.

Addendum (h_a): decreasing the addendum reduces the sliding speed but reduces the contact ratio and increases tooth stiffness (resulting in an increased sensitivity to geometry errors).

Helix angle (β): Increasing the helix angle increases the overlap ratio, increases the transverse pressure angle and reduces transverse contact ratio but increases axial bearing loads.

Addendum modification factor (x): increasing the addendum modification factor will increase the operating pressure angle and increase sliding speed at the tip unless the mating gear is also adjusted.

Topping factor (k): changing the outside diameter without changing the cutting tool is called topping the gear. It allows gear geometry to be changed without changing the manufacturing tool. Positive (+) topping reduces the outside diameter (d_a) of the gear.

Reviewing the data published by the 'Oil Free Powertrain project' shows that the strongest correlation between reducing power loss and geometry modifications is by either minimizing module or reducing the transverse contact ratio by using a stub tooth gear geometry. Both have the same effect of reducing sliding speed but the stub tooth geometry does not suffer from the reduced bending strength that affects the smaller module size.

These changes reduce the load contact line length, increasing contact loads and therefore also increasing contact stress. Two methods to compensate for this are to either to increase the face width, with a resulting increase in gear manufacturing costs (due to larger bearing spans, gear blanks, gear case and increased weight) or to increase the pressure angle of the gears (which increases the relative radius of flank curvature and thus reduces the Hertzian contact stress). It should be realized that for a given material and manufacturing route, the load carrying capacity of a gear is proportional to its volume. Thus a change in a one geometry parameter requires a proportional change in a second geometry parameter.

Many of these changes in geometry can have potentially conflicting effects on gear performance. Throughout this work to minimize losses it is imperative that good gear design practice is followed and that Transmission Error is minimized to reduce dynamic loads and gear noise.

Implementation of the loss calculation

GATES

In the early 1990s, The Design Unit identified a need to improve the modelling of cylindrical gears and developed a 3D FEA and TCA program to optimize the gear macro-geometry (module, helix angle, pressure angle etc), and gear micro-geometry (flank relief- profile tip/root relief and helix correction and crowning). It is used to estimate mesh forces, bending stresses, contact stresses and loaded transmission error (TE). The model, known as DU-GATES (Gear Analysis for Transmission Error and Stress) was initially validated by a series of tests using an instrumented power re-circulating test rig (back-to-back configuration), which can be run at 6000 rev/min and 8MW [3,8,9]. Transmission Error was verified by measuring dynamic bearing loads, and mesh stiffness by measuring load distribution across the face width with strain gauges. It has since been successfully used for optimizing gear designs in conjunction with ISO 6336 analysis methods on a wide range of applications including marine gear, automotive, aerospace and industrial transmissions over a 15 year period.

In 2007 the software was transferred to Dontyne Systems Ltd and renamed GATES. The development of the model continued with significant