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

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Management Summary

Minimizing gear losses caused by churning, windage and mesh friction is important if plant operating costs and environmental impact are to be minimized. This paper concentrates on mesh friction losses and associated scuffing risk. It describes the preliminary results from using a validated, 3-D 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 and generic procedures for minimizing losses are presented. Future development and further validation work is discussed.

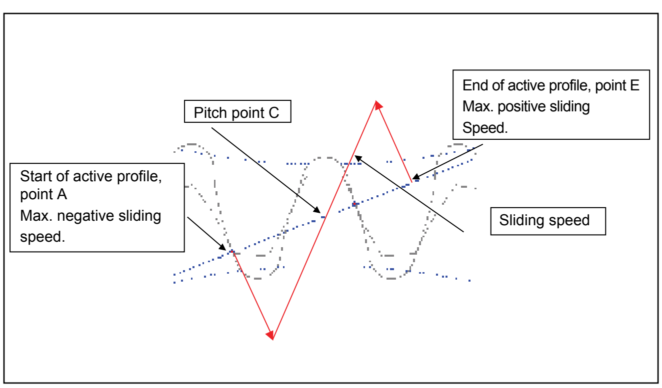

Figure 1—Variation in drive gear sliding speed with mesh phase (position).

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 microgeometry to account for elastic deflection and manufacturing errors; and have geometry that is mathematically straightforward 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—i.e., 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, which are 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, 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 that are costly and environmentally harmful. Scuffing risk is difficult to assess, and although there are two ISO Technical reports (Refs. 5–6) and the ANSI/AGMA standard 6011-I03/Annex B (Ref. 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 modeling of mesh friction in gears—including microgeometry 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 (Fig. 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 extreme pressure (EP) additives, generally sulfur-based, which 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 2—Scuffed gear sample from Design Unit's 160-mm center test rig.

Gear scuffing is difficult to predict, but several standards (DIN3991, ISO/TR13989-1 and ISO/TR 13989-2 (Refs. 5–7) provide procedures to estimate a safety factor for scuffing. These procedures use the gear macrogeometry, 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 microgeometry has on the local tooth surface loads and thus the localized scuffing risk. The microgeometry of a gear is the intentional departure from a standard gear form that can optimize its performance 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 modeling process.

Minimizing friction losses in gears is straightforward in principle, i.e.:

- 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 cor rections for calculated elastic deflections and improv ing the gear accuracy and alignment within the gear case.

continued

- 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 balancing the requirements of low-friction-loss gears with key requirements of maintaining power density, reliability, low noise and low cost, and to 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 (ν_g) shown in Equation 1:

$$P_L = \mu N v_g \tag{1}$$

Each of these quantities varies 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 manufac turing 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 fric tion at the pitch point will be different.
- The sliding speed (ν_g) varies linearly with distance along the path of contact from A to E, with $\nu_g = 0$ 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:

$$P_L = \frac{1}{b\,P_{et}} \int_{o}^{z=b} \int_{A}^{E} \left[\mu(x)\, F(x)\, v_g(x)\right] d_x\, d_z \quad [\mathrm{W}] \tag{2}$$

where:

P_{et}	base pitch (transverse), mm
$\mu(x)$	coefficient of friction
$F(x)$	local mesh load, N
$\nu_g(x)$	sliding speed, mm/s
b	face width, 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 designer's specified helix and profile correction), and dynamic loads are not accurately modeled 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 (German Engineering Federation) in Germany, completed a systematic review of gear losses with the ambitious target of producing a lubricant-free powertrain (Ref. 1). The effect of geometry parameters—i.e., module (tooth size), addendum modification, ratio, helix angle, pressure angle and face width on total gear losses was investigated. 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 also 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 also 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 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 understood 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 TE is minimized to reduce dynamic loads and gear noise.

Implementation of the Loss Calculation

GATES. In the early 1990s, the Design Unit (at Newcastle University) identified a need to improve the modeling of cylindrical gears and developed a 3-D FEA and TCA program to optimize the gear macrogeometry (module, helix angle, pressure angle, etc.), and gear microgeometry (flank relief/profile, tip/root relief and helix correction and crowning). It is used to estimate mesh forces, bending stresses, contact stresses and loaded 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 recirculating test rig (back-to-back configuration), that can be run at 6,000 rev/min and 8 MW (Refs. 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 improvement in usability, visualization of results and extension of its analysis range with the added capability to estimate mesh friction losses.

The model works in two stages:

- A 3-D FEA to establish the stiffness matrix of the gear flank. This requires the definition of the gear macrogeometry, bore or shaft size and torque direction and rotation directions. Provided the geometry is unchanged, it requires running only once and takes typically 5 to 15 minutes to run. Post-processing the FEA compliance data into a series of curves for compliance and stress is performed, thus defining the compliance of any point on the tooth surface. It allows up to 120 points per contact line to be used in the subsequent TCA analysis.
- A TCA that includes the arrangement, load conditions, gear geometry errors, mounting errors and detailed microgeometry. This takes typically one minute to run and is used to investigate relief strategy, sensitivity to alignment errors and gear manufacturing errors. It calculates contact loads, bending stress, contact stress (by analytical methods), mesh friction power loss, peak power loss, loaded TE, mesh stiffness variation, axial load shuttle for a defined load condition and speed. Typically, 32 phases of mesh are analyzed.

Table 1—160 mm center test rig gear specification

Parameter	160 mm center test gears	
	Z 1	Z 2
Teeth (z)	33	34
Module (M_n)	4.5	4.5
Pressure angle (α_n)	20°	20°
Helix angle (β)	18.3535°	18.3535°
Additional mod. coefficient (x)	0.0	0.0
Outside diameter (d_a)	166.61	171.39
Root diameter (d_f)	145.01	149.79
Tooth height (h/M_n)	2.4	2.4
Root fillet radius (ρ_{ao})	0.590	0.587
Facewidth (b)	44.0	44.0
Accuracy (ISO 1328)	5	5
Profile crown (C_α)	–	–
Helix crown (C_β)	–	–
Tip relief	–	–
Operating speed (rev/min)	3000	2911.8
Torque (Nm)	4000	4132

An important parameter that has not been discussed is the value selected for the coefficient of mesh friction. This value is speed- and lubricant-dependent and will vary with mesh position through the contact region. It is known that mesh friction will be higher at the start of mesh engagement, and that at the pitch diameter, the rolling or static friction coefficient will also be higher, but for the purposes of this initial analysis, it has been assumed to be constant through the mesh region. A coefficient of 0.05 has been used for all the initial analysis work, although this is higher than is measured in some studies (Ref. 2).

Scope of geometry modifications to minimize friction losses. Many gear designs that have the specific objective of reducing losses result in a geometry that is significantly different from conventional involute gear designs, and may not be suitable for existing stress analysis methods. It is important to realize that a low-loss design should deliver minimum noise and maximum power density as primary objectives, and thus low-loss is considered a secondary design objective. To adhere to appropriate design practice and commonly used standards such as ISO 6336, the scope of the geometry modi-

continued

fications in this project have been limited by:

- transverse contact ratios of $\varepsilon_\alpha > 1.0$
- non-stubbed tooth forms
- maintaining pressure angles to reduce excessively high mesh forces, shaft deflections and bearing loads

Example. The 160-mm power recirculation test rig will be used as part of the validation process with the helical gear geometry specified in Table 1. The gears are highly loaded in the rig with powers in excess of 1.3 MW.

To begin, a calculation package was used to determine the characteristics of the nominal design according to ISO 6336 (Authors' note: The version of ISO 6336:2006 used in the analysis includes Technical Corrigendum 1, issued June 1, 2008). Examination of the ISO 6336 stress analysis in Table 2 shows that the gears are highly stressed and that bending fatigue failure is probable (see bold-type data in Table 2). Increasing the module and reducing tooth numbers is the obvious solution to increase bending strength, but Höhn (Ref. 1) shows that increasing module will increase friction losses. Also, to maintain the integer overlap ratio, a higher helix angle is required—higher than commonly used by industry. Work at the Design Unit has shown that increasing bending fatigue strength by 40% is practical by shot peening, and thus, the gear bending strength should be acceptable (Ref. 4). This is significantly more than the 10% allowed with ISO 6336:2006.

A 1.03:1 ratio was selected to avoid uneven flank damage. With a nominal ratio close to 1:1, mesh sliding speeds

Table 2—ISO 6336 Stress Analysis

```
****************************************   pinion material Eh
*          DONTYNE SYSTEMS             *   wheel material  Eh
*       www.dontynesystems.com         *   hardness        HV       700      700
****************************************   material quality
* (University of Newcastle Upon Tyne   *   roughns flnk/µm Rz       6.0      6.0
*  Design Unit Report Style Format)    *   roughns root/µm Rz       15.0     15.0
****************************************   viscosity @ 40C nu          160
* Dontyne Systems ISO 6336 Rating      *   pitting permitted?       no       no
* V 4.10 Bld 10 14/04/2009 18:17:42    *   reversing duty?          no       no
* 33-34                                *
****************************************   applicatn factr KA           1.000
number of Teeth z       33      34         required life/h              1200
normal module   mn          4.500          load cycles     NL       2.2e+08 2.1e+08
transvrs module mt          4.776          mesh power/kW   P            1256.64
gear ratio      u           1.030          torque/Nm       T        4000.0  4121.2
centres         a           160.00         tr tang force/N Ft           50757.6
facewidth       b       44.00   44.00      speed/RPM       n        3000.0  2911.8
reference diam  d       157.61  162.39     pitch line speed/m/s         24.8
base diameter   db      147.02  151.48     tip relief/µm   Ca           30.0
tip diameter    da      166.61  171.39
root diameter   df      145.01  149.79     helix modifictn none
tooth depth     h       10.800  10.800     fav contact pattn posn verifctn?  no
internal diameter       0.00    0.00       wheel web thickness
norm pres angle alphan      20.0000        pinion offset   s            0.000
transv pres ang alphat      21.1217
wkng tr pr ang  alphawt     21.1217
ref helix angle beta        19.5778
base helix ang  betab       18.3535
prof shift coef x       -0.000  -0.000     hx dev elast/µm fsh          0.0
sum of "" coefs Sigx        -0.000         quality grade   q      5        5
bsc rack dedend hfP/mn 1.400    1.400      hx dev manuf/µm fma          5.0
bsc rk root rd rofP/mn 0.390    0.390      init'l misal/µm Fbetax       6.1
residual protub Spr/mn 0.000    0.000      run-in misal/µm Fbetay       5.2
root chord lgth sFn/mn 2.163    2.169      stiff(N/mm/µm)  cgamma       16.6
bending mom arm hFa/mn 1.093    1.095
root radius     roF/mn 0.583    0.581
tr contct ratio epsalpha    1.545          overlap ratio   epsbeta      1.043
-------------------------------------FACTORS-------------------------------------
resonance ratio N           0.529          dynamic fct     KvB          1.055
------------TOOTH FLANK----------------     ---------------TOOTH ROOT--------------
face load factr KHbeta      1.030          face load factr KFbeta 1.023     1.023
transv load fct KHalpha     1.000          transv load fct KFalpha      1.000
zone factor     ZH          2.376          form factor     YF       1.410    1.405
elasticity fctr ZE          190.272        stress conc fct YS       1.890    1.897
single pair fct ZB/ZD  1.000    1.000      notch parameter qs       1.854    1.867
cont ratio fctr Zepsilon    0.805          cont ratio fctr Yepsilon     0.687
helix ang factr Zbeta       1.030          helix ang factr Ybeta        0.837
life factor     ZN     0.956    0.957      life factor     YN       0.918    0.918
lub inf fct     ZLZVZR 0.978    0.978      notch sensy fct YdrelT   0.994    0.994
work-hardng fct ZW          1.000          surface factor  YRrelT   0.976    0.976
size factor     ZX          1.000          size factor     YX       1.000    1.000
-------CONTACT STRESS (N/mm²)----------     -------BENDING STRESS (N/mm²)----------
allw stress num sgHlm  1500.0   1500.0     allw stress num sigFE    922.0    922.0
permiss stress  sigHP  1402.2   1403.4     permiss stress  sigFP    820.6    821.2
"" (reference)  " ref  1466.5   1466.5     " (reference)   " ref    894.0    894.2
contact stress  sigH   1484.0   1484.0     root stress     sigF     617.5    617.0

safety factor   SH     0.94     0.95       safety factor   SF       1.33     1.33

min safety fctr SHmin       1.00           min safety fctr SFmin        1.40

Angles are in ° and distances in mm unless otherwise stated.
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1. The version of ISO 6336:2006 used in the analysis includes technical corrigendum 1 issued 1st June 2008

are balanced. With other ratios, sliding speed may be biased more at the entry side or exit side of the mesh. This effect can be minimized by changing the addendum modification factors (*x*) of the gears, but care should be taken to ensure that good design practice is maintained, the tooth crest width is acceptable (say > 0.30 m_n) and tooth bending strength and contact strength are maintained. Given these considerations, the macrogeometry was determined as suitable for high-performance gear testing using the following criteria:

1. The geometry is compatible with the scope of ISO 6336.
2. The geometry has an integer overlap ratio (ea) to minimize potential transmission errors and thus reduce dynamic loads.
3. The helix angle of 18.35º is within the range used by many industrial and automotive gears, so these are representative of 'real' gears.
4. Controlled shot peening is used to increase bending strength without increasing the module.
5. The sliding speeds are reasonably well balanced at entry and exit.
6. The test rig is very rigid and the gears well aligned, so alignment errors should be minimal under test load conditions (< 10 µm).
7. A hunting ratio was selected to ensure contact fatigue damage is evenly distributed around the gear.

The results from the initial *GATES* TCA are for gears under the subject load conditions without flank corrections. Figure 3 shows the predicted contact load (N/mm) over a single tooth with a plot of the length of roll of the driving gear with face width. The length of roll is from the start of active profile (SAP) of the driving gear at 12.62 mm at the root, to the end of active profile at +12.62 mm at the tooth tip. The two peaks are both close to 963.8 N/mm and occur at the entry and exit of the tooth into the mesh region (this is a helical gear). There are two dotted lines on the chart representing the theoretical start and end of active profile (contact region) of the gear pair. Contact is obviously occurring beyond this region and is due to elastic deflection of the teeth under load. This extended contact region (non-conjugate contact) occurs in the mesh region where the highest sliding speeds occur and thus the greatest potential for friction losses and scuffing failure. Modeling this region accurately is obviously of paramount importance for realistic friction losses and is the reason that the 3-D FEA and TCA method was selected to evaluate gear losses, instead of an analytical method based around an ISO 6336 procedure.

GATES estimates contact stress using classical analytical methods from the tooth load data and instantaneous radius of curvature, thus avoiding the need for a fine FE mesh to predict contact stress. Contact stress is illustrated in Figure 4 showing a maximum contact stress of 1,864.7 N/mm^2.

The combination of the tooth load data (Fig. 3) and sliding speed data (Fig. 5) are used to estimate the mesh friction

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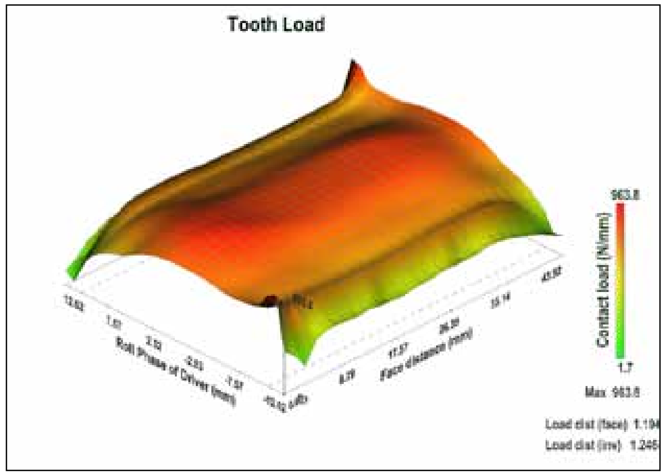

Figure 3—Tooth load: no flank relief.

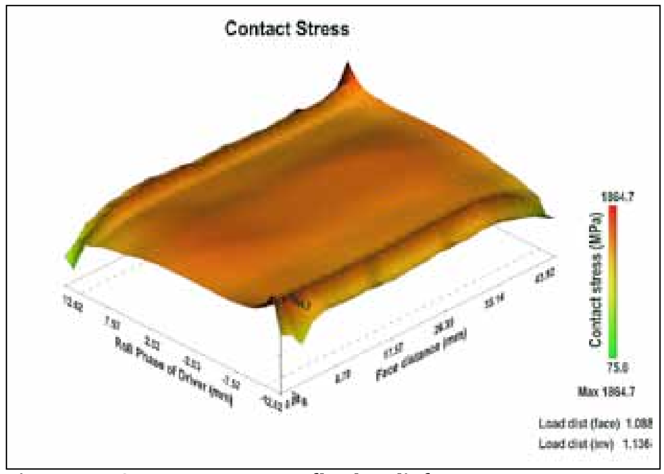

Figure 4—Contact stress: no flank relief.

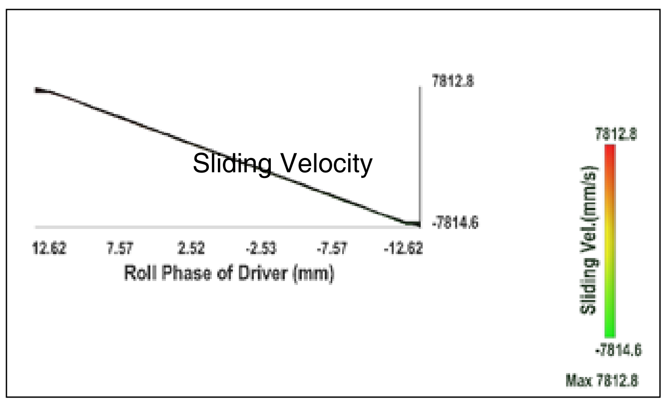

Figure 5—Mesh sliding speed, mm/sec.

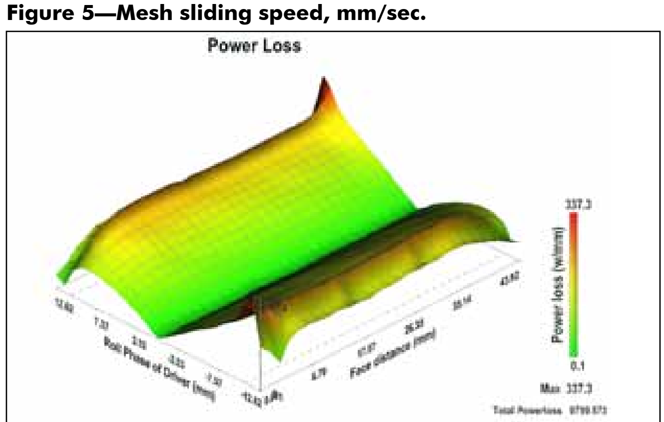

Figure 6—Mesh friction losses: no flank relief.

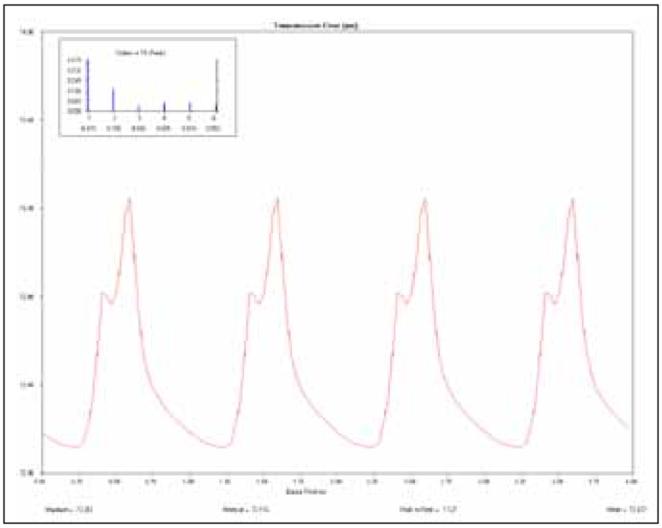

Figure 7—TE 1.27-mm pk-pk: no flank relief.

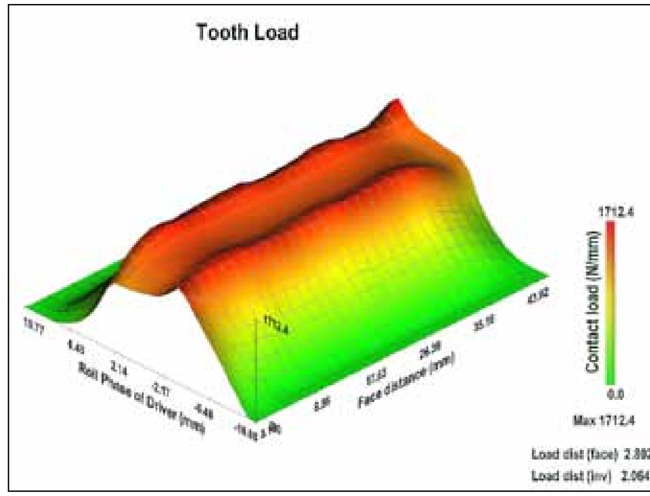

Figure 8—Contact load: 82-mm linear relief starting at the HPSTC contact stress.

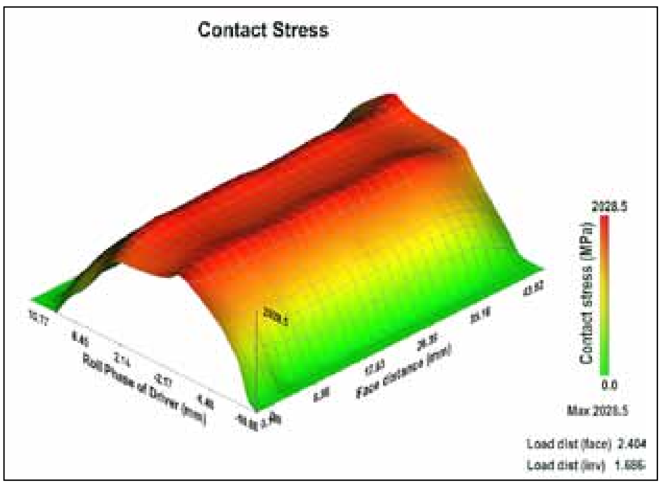

Figure 9—Contact stress: 82-mm linear relief starting at the HPSTC power loss.

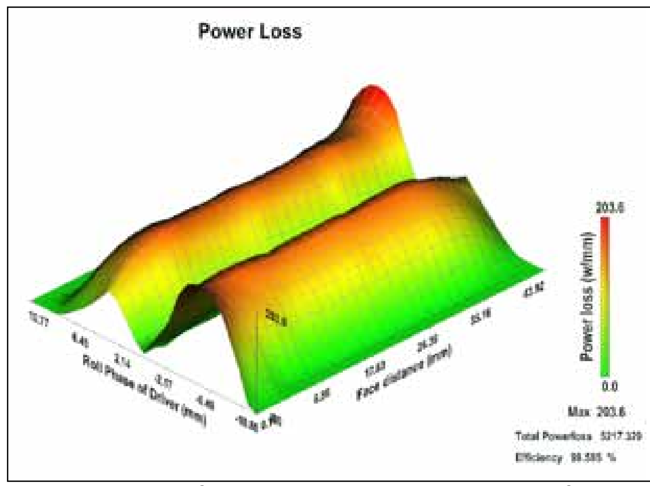

Figure 10—Mesh friction losses: 82-mm linear relief starting at the HPSTC.

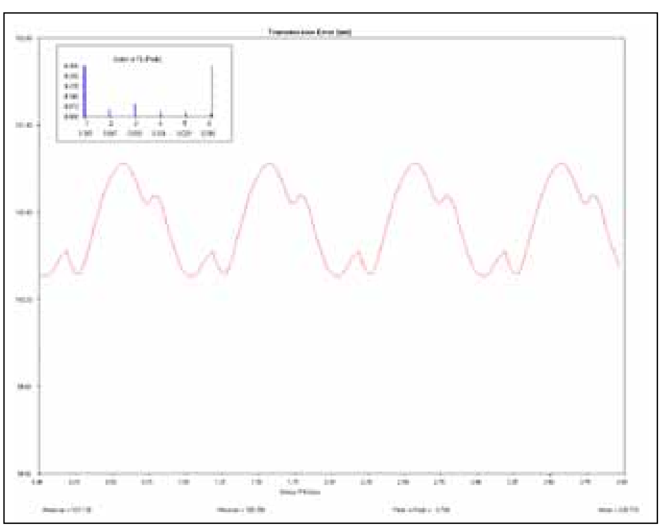

Figure 11—TE 0.784-mm pk-pk: 82-mm linear relief starting at the HPSTC.

losses (Fig. 6). It is important to note that the power loss calculation (Eq. 2) is modified to account for the extended area of contact (Fig. 3). The points A and E in Figure 1 and Equation 2 are extended due to this effective increase in the theoretical contact region.

The power loss of 9,799 W is the reference value for the proposed tests, equivalent to 99.220% mesh efficiency. A maximum predicted power loss of 337.3 W/mm indicates the maximum power loss region and thus where the most likely scuffing initiation point is likely to be.

Figure 7 shows the predicted loaded TE for the gears of 1.27 μm peak-to-peak value. This is low for gears without flank relief and is a result of the integer overlap ratio selected for the gear macrogeometry. Low TE will minimize the unknown effect of dynamic loads on the gears in the test program.

For this example, two tip relief strategies were tested with *GATES* that are applied in industry:

- Linear tip relief, starting from the HPSTC (highest point of single tooth contact)
- Parabolic tip relief starting from the pitch circle diameter

Table 3—Summary of 160mm Test Gears *GATES* Analysis Results					
Parameter	No relief	82 µm linear tip relief	82 µm parabolic tip relief	90 µm parabolic tip relief	100 µm parabolic tip relief
Maximum contact load, N	964	1712.4	1491.6	1532	1580.9
Maximum contact stress, N/mm^2	1864.7 (100%)	2028.5 (108.7%)	1885.7 (101.1%)	1911.1 (102.5%)	1941.4 (104.1%)
Power loss, W, (μ = 0.05)	9799 (100%)	5217 (53%)	5556 (56.7%)	5363 (54.7%)	5187 (52.9%)
Efficiency (μ = 0.05)	99.220%	99.585%	99.558%	99.573%	99.587%
Peak power loss, [W/mm]	337.3 (100%)	203.6 (60.3%)	201.3 (59.7%)	186.8 (55.3%)	191.3 (56.7%)
Transmission error, µm, (pk-pk)	1.27 (100%)	0.78 (69.2%)	0.95 (74.8%)	0.80 (62.9%)	0.83 (65.4%)

The amount of relief was estimated from the mean mesh deflection calculated from Equation 3:

$$F_{ro} = \frac{F_t}{b\, C_\gamma} \qquad (3)$$

where:

F_t tangential force, N

b face width, mm

C_γ combined mesh stiffness, N/mm/µm—use 14–18 as a guide value—ISO overestimates the stiffness, in our experience.

For the subject gears and load condition—and assuming that for a highly loaded gear (b = 44 mm and F_t = 50757.6 N), a mesh stiffness of 14 N/mm/mm is appropriate. Equation 3 predicts a mean mesh deflection of 82 µm. The mean mesh deflection value was used to define the tip relief amount and is always a good estimate for the sum of tip relief and crowning height combined. The resulting *GATES* analysis results for the linear relief strategy are illustrated in Figures 8–11and summarized in Table 3.

The results for 82 µm parabolic tip relief starting at the reference diameter are illustrated in Figures 12–15. Comparing the contact stress shape for the linear tip relief, Figure 9—with the parabolic relief in Figure 13—shows that peak stresses at the intersection of the linear tip relief with true involute form have vanished. This intersection point is often where micropitting failures are observed with its potential to initiate macropitting and flank-initiated bending fatigue failures. In general, the TE and losses are slightly higher with a parabolic relief strategy, although the overriding benefit of reduced contact stresses by 7% would dictate that a parabolic relief strategy is recommended.

The effect of increasing the amount of parabolic tip relief is shown in Table 3. Although the changes are small, increasing the tip relief to 90 µm clearly has some benefits and manufacturing variability will have little effect on gear performance. Additional work (Ref. 9) shows that peak contact stress with parabolic tip relief is less sensitive to profile slope and helix slope manufacturing errors.

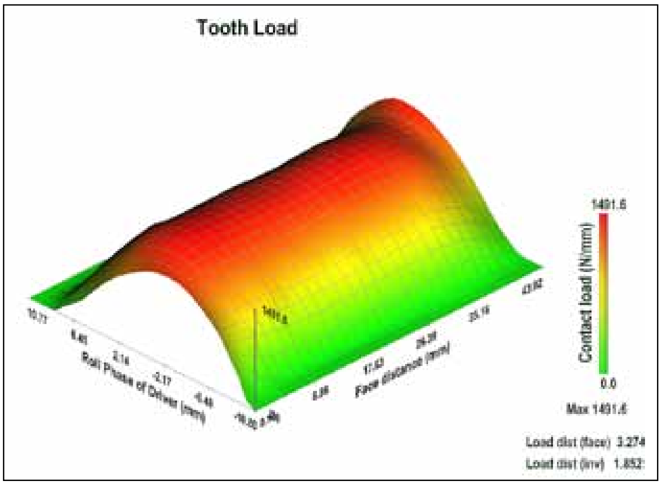

Figure 12—Tooth load: 82-µm parabolic relief starting at the reference diameter.

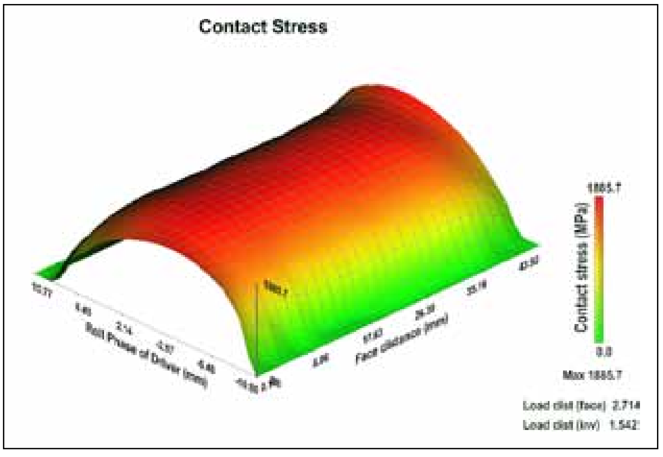

Figure 13—Contact stress: 82-µm parabolic relief starting at the reference diameter.

continued

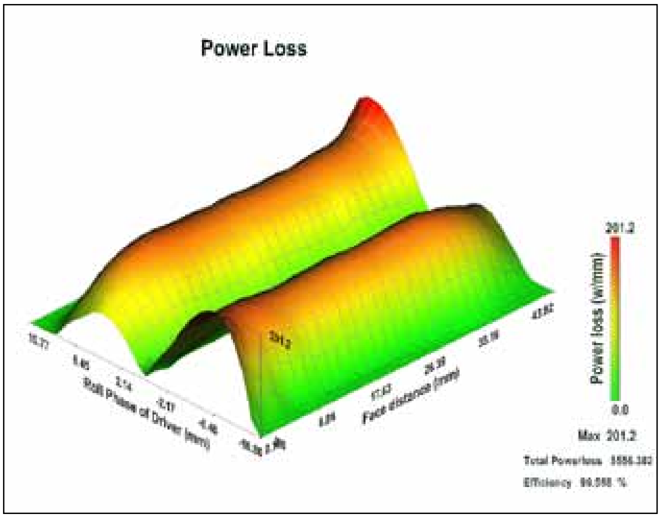

Figure 14—Mesh friction losses: 82-μm parabolic relief starting at the reference diameter.

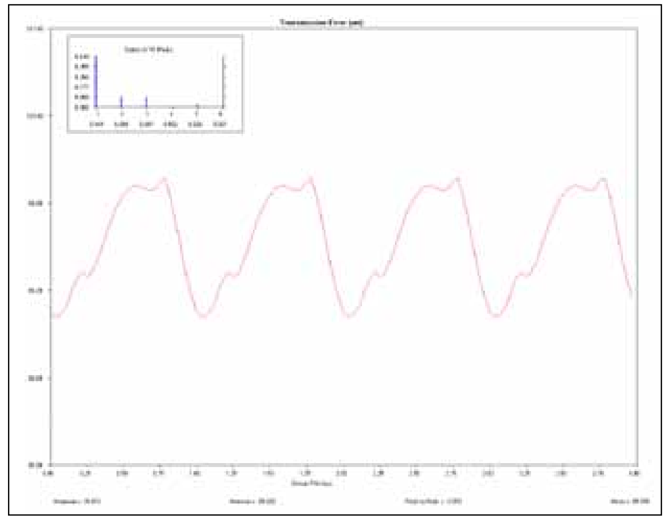

Figure 15—TE 0.95-mm pk-pk: 82-mm parabolic relief starting at the reference diameter.

A final benefit that is not illustrated in this example is that in most practical gears, helix crowning is required to minimize the increase in peak contact and bending stress due to gear alignment errors. The *GATES* analysis shows that minimizing the crown height applied to gears will minimize the increase in TE and contact stress (Ref. 10). The *GATES* TCA model includes alignment errors, profile slope errors and shaft and bearing compliance data to accurately assess these effects.

Applications

A number of "real gear applications" were investigated as part of the X-GEAR project. A few are summarized in an Appendix to this article, which can be found online at *http://www.geartechnology.com/issues/0810*.

Discussion

The work completed to date predicts that low-friction-loss gears can be designed without compromising performance while retaining standard gear geometry. Applying good design practice will result in gears with low friction loss and scuffing risk. Significant points to consider during the design are:

- Adequate contact stress and bending stress safety factor assessed in accordance with ISO 6336/AGMA 2101.
- Maximize permissible stresses using high-strength material and heat treatment.
- Maximize bending strength by increasing the root fillet radius.
- Minimize sliding speeds at the extremes of contact (start of active profile and end of active profile) to minimize scuffing risk and minimize losses by reducing module size.
- Shot peening should be considered to increase bending strength if supported by a cost/benefit analysis.
- Balance sliding speeds at the extremes of contact (start of active profile and end of active profile) to minimize losses with appropriate addendum modification factors (x).
- Maximize the length of contact line using helical gears with integer overlap ratio and thus minimize the change in length of contact line through different contact phases.
- Minimize noise and dynamic loads by minimizing TE using optimized macrogeometry and microgeometry.
- Optimize the tip relief strategy using the approximate Equation 3 and minimize discontinuities in the flank profile using parabolic tip relief. Apply appropriate helix correction to compensate for gear, shaft and gear case deflections and apply the minimum amount of crowning to correct for random manufacturing and alignment errors.
- Use current stress analysis software to the latest standards (ISO 6336/AGMA 2101), and where possible, use 3-D FEA optimization software to investigate the effect of random manufacturing errors on gear performance.
- It is recommended that gears be optimized using load data gathered from the measurement of in-service loads with appropriate strain gaging instrumentation. The load data should include transient loads and duty cycle from a representative operational duty. ISO 6336-6 should be used to evaluate the gear loads using a Miners sum cumulative damage analysis.

Conclusion

Using existing design tools such as ISO 6336 procedures in conjunction with good design practice discussed herewith should result in gears with lower mesh friction losses, lower scuffing risk and lower contact stress. Mesh friction losses can be minimized without compromising the performance of gear load carrying capacity and noise with standard tool geometry.

The paper has shown how a CAE tool has been employed to optimize performance characteristics beyond the scope of the international standards. Even marginal improvements of 0.2–0.5% represent a significant economic benefit in large-scale engineering. Using an experimentally validated design tool such as *GATES* allows individual gear sets to be optimized with confidence with particular reference to investigating the robustness of the design subject to random manufacturing and different operating conditions before manufacture and testing.

Future plans include the further validation of the model by testing at the Design Unit with back-to-back test rigs as part of the X-GEAR project. The refinement of the *GATES* model at the critical mesh regions at the start and end of active profile is also planned.

Acknowledgment

The authors gratefully acknowledge the support from the European Union, which funded the work reported in this paper as part of the X-GEAR project COLL-CT-2006-030433.

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