CFD Analysis of Gear Windage Losses: Validation and Parametric Aerodynamic Studies

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A CFD method has been applied to gear configurations with and without shrouding. The goals of the work have been to validate the numerical and modeling approaches used for these applications and to develop physical understanding of the aerodynamics of gear windage loss. Several spur gear geometries are considered for which experimental data are available. Various canonical shrouding configurations and free-spinning (no shroud) cases are studied. Comparisons are made with experimental data from the open literature, and data recently obtained in the NASA Glenn Research Center Gear Windage Test Facility. The results show good agreement with experiment. The parametric shroud configuration studies carried out in the Glenn experiments and the CFD analyses elucidate the physical mechanisms of windage losses as well as mitigation strategies due to shrouding and newly proposed tooth contour modifications.

Nomenclature

- $\forall$ volume
- $\mathbf{r}$ radial vector
- $\mathbf{V}$ velocity vector in the relative frame of reference
- $A_\theta$ tangential projection of the area of the grid face
- $C_p$ pressure coefficient
- $D$ wall damping coefficient
- $d_h$ measure of the normal distance to the nearest wall
- $k$ turbulence kinetic energy
- $L_{ref}$ reference length
- $p$ pressure
- $p_\infty$ freestream pressure
- $q$ $\sqrt{k}$
- $\mathbf{r}$ radial coordinate of grid face centroid
- $r_{\text{base}}$ base radius of gear
- $S$ area
- $T^*(x)$ torque per unit width
- $V^*$ local normalized projected relative velocity magnitude
- $V_x$ velocity x-component
- $V_y$ velocity y-component
- $V_{ref}$ reference velocity
- $C_{\mu}, C_1, C_2, \alpha$ turbulence model coefficients
- $Pr_q, Pr_\omega$ turbulent Prandtl numbers
- $\varepsilon$ turbulence dissipation rate
- $\mu$ molecular viscosity
- $\mu_t$ eddy viscosity
- $\omega$ $\varepsilon/k$
- $\rho$ density
- $\omega$ angular velocity
- $\tau$ shear stress tensor

Introduction

Gearbox windage losses refer to the power losses associated with rotational deceleration torques exerted on spinning gears by aerodynamic forces (pressure and viscous) within the air-oil atmosphere present within a gearbox. Windage losses are a source of significant heating and fuel consumption in rotorcraft and other VTOL systems. The weight and packaging constraints inherent in these systems require the gearing components to be both lightweight and heavily loaded. Attendant to this, the gears are required to operate at high rotational speeds where windage losses
become significant with respect to other gearbox losses (rolling, sliding, and lubrication losses).

A host of experimental studies of gearbox windage have appeared in the literature. These studies employ either closed loop systems or treat isolated gears where windage (and other) losses are determined by measuring spin-down velocities once the gear and shaft assembly is disconnected from the drive torque. Generally, these studies parametrize gear geometry elements, rotational speed, enclosure geometry, and lubrication system characteristics (flow rate, jet location, lubricant rheology), and use dimensional analysis to develop correlations for the power losses which have proven useful in the design process.

It has been shown experimentally by Winfree and Dawson that geometric modifications of the near-gear flow path, such as shrouding and baffle configurations, can significantly reduce both windage losses and lubricating oil consumption (80% and 40% reductions observed, respectively, in Ref. 1). Such modifications are therefore of significant interest in rotorcraft transmission design, but to date, only these empirical studies are available. Accordingly, there is a need for more systematic and prototypical experimental data and a need for improved understanding of the physical mechanisms involved in the schemes used to reduce these aerodynamic losses.

Recently, a gear windage test facility has been developed at the NASA Glenn Research Center. This facility is expected to produce the most comprehensive set of gearbox windage data to date, and is providing CFD validation data required in the present NASA sponsored research effort. The test facility is designed to parametrize the effects of spur gear geometry, shroud geometries, different lubrication system configurations, system pressures and temperatures, and gear meshing on windage loss. The first data sets from the Glenn rig are published here. This data includes a single spur gear spinning in air for a set of four axisymmetric axial/radial shroud configurations.

CFD practitioners have also recently begun to analyze gear windage aerodynamics towards the goal of developing a more general design approach. Recent 2-D studies were performed using the commercial flow solver Fluent, where a side correlation factor was used to account for 3-D effects (although these authors state that work is under way to extend their simulations to 3-D). Marchesse et al. used a similar approach as Al-Shibib et al., but used a structured grid with the flow solver ANSYS CFX. They also extended their 2-D model into three dimensions, using one tooth passage with periodic boundary conditions as their computational domain. Imai et al. investigated 3-D bevel gears in mesh in an air-oil atmosphere, modeling the gears as porous bodies. Hill et al. implemented a 3-D, unstructured, overset moving mesh method and applied it to isolated spur gears in air and validated the method against data due from Diab et al.

The validated capability developed in our earlier work has motivated, among other things, the focus of the present study: to use a CFD model to interrogate the physics associated with gear windage aerodynamics. The attendant goal of this is to develop design guidance principles for methods to minimize these power losses in rotorcraft, and potentially other high speed gear systems. This has been and will continue to be enabled by the availability of the Glenn Research Center data sets.

The paper is organized as follows: First, the governing equations and numerical procedures are reported. Next a series of numerical studies are performed that provide insight into several of the physical mechanisms of windage losses including the role of gear shrouding. Next, an overview of the NASA Glenn Gear Windage Test Facility is provided along with a selection of data obtained to date. CFD simulation of the Glenn windage loss tests are presented and discussion of these comparisons are provided. Lastly, we present results of a numerical experiment, guided by the findings described in the paper, that demonstrate a benefit in windage performance beyond that observed with shrouds alone.

**Theoretical Formulation**

**Governing Equations**

We focus in this paper on the physical mechanisms of gear windage loss, and mitigation schemes for reducing them. The available data shows that windage losses already become significant at low subsonic tip Mach numbers. Accordingly, in this work we restrict ourselves to incompressible analysis. Also, as will be shown, much can be learned from systems that are either shroud-free or have completely axisymmetric shrouds (i.e., fully enclosed radial shrouds) so we consider single-tooth domains that are periodic and steady in the frame of reference of the spinning gear. The conservation of mass and momentum can be written in integral conservation law form for a system rotating with constant angular velocity \( \omega \) as:

\[
\int_{\Sigma} \rho \mathbf{V} \cdot d\mathbf{S} = 0 \tag{1}
\]

\[
\int_{\Sigma} \rho \mathbf{V} \mathbf{V} \cdot d\mathbf{S} = - \int_{\Sigma} \rho d\mathbf{S} + \int_{\Sigma} \mathbf{F} \cdot d\mathbf{S} - \int_{\Sigma} \rho \omega \times (\omega \times \mathbf{r}) d\mathbf{S} - \int_{\Sigma} 2\rho (\omega \times \mathbf{r}) d\mathbf{S} \tag{2}
\]

where \( \mathbf{V} \) is the velocity vector in the relative frame of reference. It was shown in Hill et al. that a sublayer resolved two-equation turbulence model performed better than a high-Reynolds number form with wall-functions in predicting viscous losses on a spinning disk. Accordingly, in this work we adopt \( \overline{\tau} = (\mu + \mu_t)(\nabla \mathbf{V} + (\nabla \mathbf{V})^T) \) and a sublayer resolved \( q-\omega \) turbulence model due to Coakley is used. The dependent variables in this model are related to the turbulence kinetic energy, \( k \), and the turbulence dissipation rate, \( \varepsilon \), through \( q = \sqrt{k} \) and \( \omega = \varepsilon/k \). In this model, the eddy viscosity is obtained from:

\[
\mu_t = \rho C_{\mu} D q^2 / \omega \tag{3}
\]
where \( C_\mu = 0.09 \) and \( D \) is the near wall damping function:

\[
D = 1 - e^{-\alpha \rho \bar{d}_n / \mu}
\]

(4)

where \( \alpha = 0.02 \) and \( \bar{d}_n \) is a measure of the normal distance to the nearest wall. The modeled transport equations for \( q \) and \( \omega \) are:

\[
\int_S \rho q \mathbf{V} \cdot dS = \int_S \left( \mu + \frac{\mu_t}{Pr_q} \right) \nabla q \cdot dS + \int_S \frac{\rho q}{\nu} \left( C_\mu D S \omega - \omega \right) \ dS
\]

(5)

\[
\int_S \rho \omega \mathbf{V} \cdot dS = \int_S \left( \mu + \frac{\mu_t}{Pr_\omega} \right) \nabla \omega \cdot dS + \int_S \rho \left( C_1 C_\mu S - C_2 \omega^2 \right) \ dS
\]

(6)

CFD Code and Numerics

OVER-REL

OVER-REL is a parallel, cell-centered, incompressible, finite-volume code based on structured overset multiblock grids and the time-marching, pseudo-compressibility formulation of Chorin. Inviscid fluxes are formulated from the Roe-approximate Riemann solver and extended to third-order accuracy through the MUSCL scheme. Second-order accurate central differences are utilized for the viscous fluxes. Numerical derivatives are used to calculate the flux Jacobians. A symmetric Gauss-Seidel method is applied to solve the resulting linear system of equations. In the present application, the code’s turbomachinery analysis instrumentation is employed; all simulations are carried out for a single gear tooth, with periodic boundary conditions, in a non-inertial frame-of-reference rotating with the gear. OVER-REL is not the code used for our other gear-windage research, NPHASE-PSU, since it does not support compressible flow, multiphase flow, or unstructured meshes (necessary for complex gearbox geometries). However, it is better suited than NPHASE-PSU to the present physics exploration analyses due to its efficiency for cases where a single “blade-row”/gear tooth can be used (i.e., axisymmetric shroud).

Overset meshing: SUGGAR and DiRTlib

As illustrated below, the tight shroud clearances studied lead to mesh topology constraints that in turn lead to poor quality block-structured meshes. OVER-REL is instrumented with overset meshing capability, which enables high quality meshes for these configurations.

The overset assembly process is performed using the SUGGAR code, a general overset grid assembly code with the capability to create the domain connectivity information at node and/or element centers for general grid topologies. For static grid assemblies with no motion between component grids (as here) the grid assembly is a pre-processing step. The inter-grid interpolations produced by SUGGAR use an unweighted least square procedure.

OVER-REL is instrumented with DiRTlib, a solver neutral library that encapsulates the functionality required by the solver to utilize the overset domain connectivity information (DCI) produced by SUGGAR. DiRTlib is independent of the CFD solver’s data structures, and can therefore be used with any solver. The CFD code calls a few functions to initialize the library, load the DCI interpolation, transfer the data to appropriate processors in a parallel execution environment, and apply the interpolated data as boundary values at inter-grid boundary points.

Results

Loss Mechanisms and Budgets in Gear Windage

In their 2004 paper, Diab et al. studied four unshrouded, spur gears and a disk spinning freely in air (i.e., no shrouding). The gears varied in diameter, width and tooth count - as summarized in Figure 1. Figure 2 shows results published previously by the present authors. There it is observed that the NPHASE-PSU simulations exhibited very good agreement with experiment in modeling windage losses for these configurations.

In this work, free spinning simulations were first carried out for Diab Gear 1 and disk using OVER-REL. Non-overset multiblock structured meshes were employed for both as shown in Figure 3. Near-wall grids were constructed to return wall cell \( y^+ \) values < 1 and wall normal stretching ratios < 1.2 everywhere in order to adequately resolve the high Reynolds number boundary layers that arise. The grid topologies, near-wall grid spacing, and grid stretching ratios were maintained as closely as possible between the gear and disk meshes. Per the periodic boundary conditions employed, one tooth passage (2\( \pi / 72 \)) is modelled for both configurations. In order to stably time-march the OVER-REL solution, a very small inflow velocity and a pressure outflow boundary were included in adjacent to the maximum and minimum axial boundaries upstream and downstream of the rotating elements. This artificial through-flow velocity was successively reduced to where no perceptible changes in loss values were returned.

<table>
<thead>
<tr>
<th>Pitch diameter (mm)</th>
<th>Tooth width (mm)</th>
<th>Module (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear 1</td>
<td>288</td>
<td>30</td>
</tr>
<tr>
<td>Gear 2</td>
<td>144</td>
<td>30</td>
</tr>
<tr>
<td>Gear 3</td>
<td>144</td>
<td>60</td>
</tr>
<tr>
<td>Gear 4</td>
<td>144</td>
<td>60</td>
</tr>
<tr>
<td>Disk</td>
<td>300</td>
<td>30</td>
</tr>
</tbody>
</table>

Fig. 1 Diab Gears 1-4. (Only one side of symmetry plane is shown for each gear.) Adapted from Hill et al. 14

Figure 2 includes the OVER-REL results for Gear 1 and very good agreement with experiment is seen there as
Fig. 2 Comparison of results from experiment, NPHASE-PSU, and OVER-REL for Diab Gears 1-4.

Fig. 3 Comparison of Diab Gear 1 and disk grids.

well. Figure 4 shows that for Gear 1 the pressure torque associated with the integrated pressure difference between leading and trailing tooth surfaces, dominates the loss budget. As the rotation rate increases, viscous losses remain a nearly constant fraction of total loss (10%). Figure 5 shows a comparison of loss results between Gear 1 and the disk. The total losses are much smaller for the disk, which by symmetry are due to viscous shear alone. This large difference between disk and Gear 1 is particularly striking if we compare their similar geometries shown in Figure 6.

In Figure 7, the torque per unit span contributed by viscous shear is compared for the disk (up to its outer radius) and Gear 1 (up to its base radius). The geometry of these systems requires that the all pressure spin-down torque is due to the pressure differences between the leading and trailing tooth surfaces. Figure 7 illustrates that the viscous losses are very similar between these configurations (indeed the small differences in predicted viscous power values are due to very small grid differences) indicating that 3D effects (i.e. non-axisymmetric) associated with pressure forces are almost completely responsible for the significant increase in loss for the gear.

The physical mechanisms associated with the dominant spin-down pressure torque are studied by interrogating the CFD results. In Figures 8-11, several 3D visualizations are presented for the 850 rad/s case. In Figure 8, a number of relative frame-of-reference streamlines, colored by local static pressure, are plotted. These streamlines are seeded close to the gear face and teeth and integrated in both directions. Some of the high speed (in the relative frame) tangential flow near the gear face plane is diverted into the tooth passage, where strong secondary flows are evident. By symmetry this axial transport arises on both sides of the gear and therefore leads to impingement of oppo-
sitely directed flow and radial ejection of momentum near the gear centerline. Figure 9 shows an axial view of the same streamlines illustrating the complex secondary motions and an indication of the radial ejection angle. Also evident is the radially outward component of flow close to the face for \( r < r_{base} \). An axial projection of relative velocity vectors is shown in Figure 10 at a plane halfway between the gear face and gear centerline. A vector density of 0.5 (vector plotted for approximately every other grid point) is applied for clarity. Contours of local normalized projected relative velocity magnitude are included \( (V^* = \sqrt{V_y^2 + V_x^2/\omega r}) \). Two counter-rotating passage vortices are present. Peak normalized secondary velocity magnitudes near \( \frac{1}{2}\omega r \) are observed indicating the strength of these secondary motions. The flow at this axial location is reminiscent of a rearward facing step and/or cavity flow with attendant vortical recirculation regions. Figure 11 shows the same plot but near the gear centerline. Here one sees the very significant radial ejection quite clearly. The flow has a component directed upstream (against the relative flow) near the leading surface at the tip radius. The magnitude of the ejection flow induces significant blockage and we see values of \( V^* \) much less than 1 well beyond the tip radius.

In Figure 12 contours of static pressure coefficient, \( C_P = (p - p_\infty)/0.5\rho V_{tip}^2 \), are plotted for the leading gear tooth surface. As seen above, some of the high speed tangential flow near the gear face plane is diverted into the tooth passage. There is a stagnation where this flow
impinges on the leading surface, near the face. The leading surface pressure field can be compared with the much lower surface pressures on the trailing surface shown in Figure 13. The net spin-down torque due to pressure effects can be represented by the difference between the leading and trailing surface pressure coefficients, which is shown in Figure 14. Clearly the net torque is dominated by the impingement observed in Figures 8 and 12. In Figure 15 the torque per unit width is plotted vs. distance from the gear face. Here, torque is nondimensionalized as:

$$T^*(x) = \frac{\int_{r_{inner}}^{r_{tip}} \Delta p dA d\theta}{dx} \left( \frac{1}{2\rho \bar{V}^2_{ref}} \right)$$  \hspace{1cm} (7)$$

where $\Delta p$ is the pressure difference between the grid faces on the leading and trailing surfaces (which have identical x-r vertex coordinates), $dA_\theta$ is the tangential projection of the area of the grid face, and $r$ is the radial coordinate of the grid face centroid. We see that indeed it is the near face region that dominates the pressure windage loss torque.

**Gear Shrouding**

As mentioned in the introduction, several researchers have measured improvements in gear windage loss performance when shrouds of various configurations are employed.\(^1\)\(^-\)\(^3\) In this section we study the aerodynamics of shrouding in the context of geometrically simple configurations and draw several conclusions that may impact design considerations. The Diab Gear 1 configuration is studied. Four notional shrouding arrangements are considered, nominally Large-Axial-Large-Radial, Small-Axial-Large-Radial, Large-Axial-Small-Radial, and Small-Axial-Small-Radial. Table 1 and Figure 16 quantify the shroud dimensions. These four shroud configurations were chosen to be representative of the extrema of the full-shroud NASA Glenn tests as detailed further below. The smaller axial and radial shrouds clearances gave rise to mesh topology constraints that in turn lead to poor quality block-structured meshes unless overset meshes are used. Figure 17 shows a cross-sectional view of the overset mesh topology for a Small-Radial case.

<table>
<thead>
<tr>
<th>Axial Clearance ($R_{tip}$)</th>
<th>Radial Clearance ($R_{tip}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0970</td>
<td>0.1733</td>
</tr>
<tr>
<td>0.0044</td>
<td>0.0044</td>
</tr>
</tbody>
</table>

In Figure 18, the predicted windage loss vs. rotation rate is plotted for the free wheel case, validated and studied above, and the four shrouded configurations. All four of the shrouds give rise to very significant improvements...
Fig. 14  $\Delta C_P$ between gear leading and trailing tooth surfaces (1/2 of symmetrical gear shown). Diab Gear 1, 850 rad/s.

Fig. 15 Pressure torque per unit width vs. axial coordinate (1/2 of symmetrical gear shown). Diab Gear 1, 850 rad/s.

Fig. 16 Four notional shroud configurations for Diab Gear 1 geometry. Figure is to scale. Black lines define the gear face, tip radius and base radius, dark blue and red lines indicate the Large-Axial and Small-Axial shrouds, green and cyan lines indicate the Large-Radial and Small-Radial shrouds.

in windage losses. The Large-Axial-Large-Radial shroud provides a 68% decrease in loss at 850 rad/s; the Small-Axial-Small-Radial shroud approximately a 81% decrease at this speed. Indeed, distinguishing the performance gains between the shrouded cases is facilitated by not including the free spinning results as in Figure 19. Examining this plot we see that reducing the axial and radial clearances from large to small provide approximately the same level of additional benefit over the Large-Axial-Large-Radial case, with the reduction of the radial clearance providing somewhat more benefit in this particular case. Applying both clearance reductions together provides the maximum benefit.

Figures 20-22 show the same three pressure coefficient plots reported above, but here for the Large-Axial-Large-Radial shroud case, again at 850 rad/s. Several observations are forthcoming. Firstly, the load distribution on the leading surface shown in Figure 20 is qualitatively similar to the unshrouded case (Figure 12), including the stagnation region associated with near-face flow diverted into the tooth passage. However, the range of $C_p$ values here are much smaller than for the free wheel case. (Note that the absolute values of $C_p$ should not be compared between cases here as this level is determined in the incompressible flow solver using a domain “exit” value of $p_\infty = 0$).
The trailing surface (Figure 21) exhibits a much more axially uniform pressure distribution than the free wheel case (Figure 13), and again the range of $C_P$ values here are much smaller. Most importantly, in Figure 22 we observe a highly edge loaded $\Delta C_P$ between leading and trailing surfaces, as observed for the free wheel case, but with a very reduced range of $\Delta C_P$ which of course leads to reduced torque. Figure 23 shows contours of pressure and surface shear stress lines on the leading surface for the four shrouded cases at 850 rad/s. There is can be seen that the general features of impingement/stagnation near the face remain present for all of the configurations.

In Figure 24, the torque per unit width is plotted vs. distance from the gear face for all four shrouds at 850 rad/s. All four configurations exhibit edge loaded profiles, with their integral consistent with the net loss trend reported in Figure 19. Comparison with Figure 15 shows the dramatic reduction in pressure torque that has been achieved for these configurations, but with retention of the basic features of the pressure torque distribution.

A final exploration study of viscous losses was performed on the shrouded Diab gears. Figure 25 shows a comparison of viscous losses per unit span for the unshrouded case, illustrating the increase in shear torque with rotation rate and span. The viscous loss per unit span is plotted for each of the four shrouded cases and the unshrouded case at 850 rad/s in Figure 26. There is it ob-
served that face shear is smaller for all of the shroud cases due to the "couette-flow-like" rotational boundary layer that arises in the presence of an outer axial boundary. Also of interest in this figure is the increase in viscous loss for the Small-Axial shrouds compared to the Large-Axial shrouds. These losses are smaller than for the unshrouded case but suggest that viscous losses can increase a percentage of total loss for very small axial shroud clearances.

**NASA Glenn Gear Windage Test Facility**

Recently, a gear windage test facility has been developed at the NASA Glenn Research Center. This facility is expected to produce the most comprehensive set of gearbox windage data to date, and is providing the CFD validation data required in the present NASA sponsored research program. The test facility is designed to parametrize the effects of spur gear geometry, shroud geometries and sizes, different lubrication system configurations, system pressures and temperatures, and gear meshing on windage loss. It is instrumented to provide data on windage power loss (spin-down tests), oil fling-off temperatures, and pressure (at baffle walls). The test stand employs an open loop regenerative torque system to spin two gears in mesh. It can also be used to test individual isolated gears. Figure 27 is a diagram of the test facility. Figure 28 is a photograph of the gearbox assembly with the top cover removed, showing the test gears without shrouds. In this work we report data from the first series of experiments: air-only, single-gear, fully enclosed axial and radial shrouds. The single gear that is studied is a 13 inch pitch diameter spur gear, the topmost gear in Figure 28. The gear has 52 teeth and a tooth width of 1.12 inches. The gear has some modest geometric complexities compared to the idealized Diab gear studied above, including...
tooth chamfering, root rounding and narrowing of the gear width below the base radius to reduce gear weight/inertia. These features are shown in Figure 29 and were accommodated in the CFD model. The test fixture was designed to accommodate numerous shroud spacing configurations. Specifically, the baffle housing, pictured in Figure 30, has adjustable axial and radial wall clearances. The housing allows for up to 36 different fully-enclosed baffle configurations to be tested in the facility - report the data for and analyze the four extrema: Large-Axial-Large-Radial, Large-Axial-Small-Radial, Small-Axial-Large-Radial, and Small-Axial-Small-Radial. Table 2 lists these four axial and radial clearance dimensions.

### Table 2  Shroud Wall Clearances Studied

<table>
<thead>
<tr>
<th>Axial Clearance (in.)</th>
<th>Radial Clearance (in.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.030</td>
<td>0.030</td>
</tr>
<tr>
<td>1.170</td>
<td>0.655</td>
</tr>
</tbody>
</table>

Speed data is measured using inductive pickups that read a 60 tooth disc on the end of each of the shafts. The output from the sensor (pulse / sine wave) is sent to a frequency to voltage converter. The output from the convertor is then sent to a National Instruments card and read by Labview. Data was taken at 10 Hz.

Data from the NASA Windage Test Stand was measured in the following manner. In the tests reported here, there was only an input gear and the gear was operated in a totally enclosed shroud (no drain holes in the shroud). During the operation of the test facility discrete speeds were maintained and data (speed, torque, oil fling-off temperatures, and pressure at shroud walls) were taken. This was also
done at the same speeds for a shaft-only configuration (the test gear was removed). Driving torque data was taken at twelve discreet speeds with and without the gear present. The net effect of the gear was found by subtracting the shaft only torque from the total torque to operate the shaft and gear shroud arrangement together. Windage loss power is computed using the known rotational inertia of the gear (measured separately in a pendulum fixture).

**Experimental and Computational Results for the Glenn Facility Tests**

Grids were developed for the 13-inch pitch diameter spur gear configuration using similar topologies as the Diab cases. To date three shrouded configurations have been selected for study. Table 2 lists the four baffle configurations studied computationally; the “Small-Axial-Large-Radial” configuration has not yet been run in the NASA test rig.

Figure 31 shows comparison of the Glenn data and OVER-REL predictions for the Large-Axial-Large-Radial, Large-Axial-Small-Radial and Small-Axial-Small-Radial shroud configurations. Between 5 and 12 experimental data points were taken for each speed line. These data were curve fit using a cubic polynomial - resulting R values were all above 0.997 - and these curves are plotted in Figure 31. The data shows the same trends as the idealized Diab gear CFD studies reported above. Specifically, the Large-Axial-Large-Radial shrouding exhibits the highest loss levels, and the Small-Axial-Small-Radial shrouding exhibits the lowest loss levels. The benefit realized by reducing both clearances is somewhat more substantial than for the Diab case. The CFD results are seen here to provide fairly good agreement with measured values. An interesting observation in the CFD results is that the Large-Axial-Small-Radial and Small-Axial-Large-Radial results are nearly identical along the entire speed line.

The qualitative correspondence between the Glenn and idealized Diab cases presented earlier, suggests that the same physical loss mechanisms are acting. Figure 32 shows a view of predicted relative streamlines colored by pressure for the Large-Axial-Large-Radial shroud case at 700 rad/s. This image exhibits several of the features observed for the idealized case including diversion of the near face flow into the tooth passage and impingement upon the leading surface, strong axial secondary vorticity in the tooth passage, strong ejection of this flow near the tooth centerline and radial flow of the near-face streamlines below $r_{base}$.

**Design Alternatives**

It was demonstrated above that axial and radial shrouding can reduce windage losses. Some of the physics of these loss reduction schemes were studied there. Despite the experimentally and computationally observed differences in loss magnitudes between unshrouded and various
shrouded configurations, in all cases a significant component of the torques associated with spin down arose from impingement onto the leading surface of the high velocity relative-frame flow drawn into the tooth passage. Accordingly, in this section we return to the geometrically idealized Diab Gear 1 configuration, and experiment numerically with four proposed tooth geometry modifications aimed at mitigating this impingement and attendant spin down torque. Figure 33 shows an oblique view of the four alternative geometries considered: 1) leading surface tooth-edge rounding, 2) leading+trailing surface tooth-edge rounding, 3) double slots on the top of the teeth and, 4) trailing surface ramp.

Figure 34 shows a comparison of these four simulations with the baseline Large-Axial-Large-Radial case. The leading surface rounding and double slot geometries return nearly identical windage loss. The leading+trailing sur-
face rounding returns somewhat higher loss. However, the net loss obtained using the trailing surface ramp is approximately 30% lower than the baseline configuration. The torque per unit width for the five geometries are plotted in Figure 35. There it can be seen that the ramp configuration exhibits much smaller torques within the tooth channel and this clearly results in the reduced integrated loss for the entire gear.

To further elucidate the physics involved in these numerical studies, Figures 36-40 are presented. In each of these figures, predicted surface pressure coefficients contours are plotted along with selected surface skin friction lines. The baseline and two rounded geometries exhibit largely same qualitative flow features, with the rounded cases “smearing” the leading face impingement and trailing face detachment gradients. The tooth slots were conceived to “flush” the peak axial vorticity/low pressure regions with higher velocity incoming relative flow thereby reducing the axial pressure gradient and thereby the diverting the high relative velocity near-face flow into the passage. This appears here to not have achieved the desired result. The trailing surface ramp geometry did have a significant impact on the aerodynamics. Specifically, the relative flow near the face is turned away from the gear. This turning induces a local pressure rise on the ramp which contributes to spin-down torque. However, this flow has been diverted away from the tooth enough that subsequent diversion of this flow into the tooth passage has been virtually eliminated, resulting in almost no pressure rise on the leading surface. This gives rise to the much small torques shown in Figure 35. So the improved net performance of the ramp configuration observed in Figure 34 is clearly due to the reduced integrated tooth passage torque more than offsetting the increased torque associated with the ramp turning itself.

![Fig. 34 Windage loss predictions for the baseline tooth geometry (Diab Gear 1 with Large-Axial-Large-Radial shrouds) and four geometric alternatives.](image1)

![Fig. 35 Torque per unit width predictions for the baseline tooth geometry (Diab Gear 1 with Large-Axial-Large-Radial shrouds) and four geometric alternatives at 850 rad/s.](image2)

![Fig. 36 Predicted surface pressure coefficient and skin friction lines for baseline tooth geometry.](image3)

![Fig. 37 Predicted surface pressure coefficient and skin friction lines for tooth geometry alternative: leading surface rounding.](image4)
Fig. 38 Predicted surface pressure coefficient and skin friction lines for tooth geometry alternative: leading+trailing surface rounding.

Fig. 39 Predicted surface pressure coefficient and skin friction lines for tooth geometry alternative: tooth slot.

Fig. 40 Predicted surface pressure coefficient and skin friction lines for tooth geometry alternative: trailing surface ramp.

Conclusion

This paper has summarized a number of CFD studies focused on the aerodynamics of gear windage losses. The goals of the work have been to validate the numerical and modeling approaches used for these applications and to develop physical understanding of the aerodynamics of gear windage loss. Comparisons are made with experimental data from the open literature, and data recently obtained in the NASA Glenn Research Center Gear Windage Test Facility. Parametric shroud configuration studies carried out in the Glenn experiments and the CFD analyses elucidate important physical mechanisms of windage losses as well as mitigation strategies due to shrouding and newly proposed tooth contour modifications.

The following conclusions apply: 1) The Glenn data and CFD analyses show that axial and radial gear shrouding are effective in significantly reducing gear windage losses both independently and when employed together. 2) For all configurations studied, 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. 3) Shrouding mitigates the magnitude of this mechanism but not its dominance in the loss budget. 4) The CFD results show good agreement with the Glenn experiments. 5) The Glenn data shows quite similar loss trends to the idealized shrouded gear configurations studied computationally. 6) Viscous losses are reduced by shrouding as well, but are a small component of the total loss so this effect is not important. However, small axial clearances were observed computationally to increase viscous losses compared to larger shroud clearances, suggesting that this effect could become important for even smaller clearances and at higher speeds. 7) The CFD studies suggested a set of possible geometric tooth modifications to further reduce windage loss. One of these appears quite promising and the authors anticipate studying this in the Glenn facility.

Our continuing work in the gear windage aerodynamics area focuses on: 1) Further validation for shrouded gears as more Glenn data becomes available, 2) Multi-phase flows, 3) Overset gridding in the context of gear tooth contact, and 4) Further evolution of design guidance for minimizing windage losses.

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References