A procedure is presented for performing three-dimensional stress analysis of spiral bevel gears in mesh using the finite element method. The procedure involves generating a finite element model by solving equations that identify tooth surface coordinates. Coordinate transformations are used to orientate the gear and pinion for gear meshing. Contact boundary conditions are simulated with gap elements. A solution technique for correct orientation of the gap elements is given. Example models and results are presented.

Introduction

Spiral bevel gears are used to transmit power between intersecting shafts. One such application is in helicopter transmission systems. In this critical application, the gears operate at relatively high rotational speed and transmit substantial power (i.e., 1500 HP at 21,000 rpm).

Prior research has focused on spiral bevel gear geometry [1-6] to reduce vibration and kinematic error, improve manufacturability and improve inspection. Stress analysis is another important area of ongoing research. Accurate prediction of contact stresses and tooth root/fillet stresses are important to increase reliability and reduce weight. Tooth flexibility will shift the contact zone and path, alter and change the contact area and stresses. Finite element analysis of gears in mesh, including tooth flexibility, will model contact stresses better than a Hertzian calculation.

Much effort has focused on predicting stresses in gears with the finite element method. Most of this work has involved parallel axis gears with two dimensional models. Only a few researchers have investigated finite element analysis of spiral bevel gears [7, 8].

Finite element analysis has been done on a single spiral bevel tooth using an assumed contact stress distribution [9]. The research reported here will utilize the numerical solution for spiral bevel surface geometry to study gear meshing. Pinion tooth and gear tooth surfaces will be developed based on the gear manufacturing kinematics. The individual teeth are then rotated in space to create a multi-tooth model (4 gear and 3 pinion teeth). The tooth pair contact zones are modeled with gap elements. The model development procedure and finite element results are presented.

Equations for Tooth Surface Coordinates

The system of equations, briefly summarized here, required to define the coordinates of a face-milled spiral bevel gear surface were developed by Handschuh and Litvin [9].

A conical cutting head, attached to a rotating cradle, swings through the work piece. Parameters \( U \) and \( \theta \) locate a point on the cutting head in coordinate system \( S \), attached to the cutting head as shown in Fig. 1 and described by the following equations.

\[
\begin{align*}
    r_c &= \begin{pmatrix}
        r \cot \psi - U \cos \psi \\
        U \sin \psi \sin \theta \\
        U \sin \psi \cos \theta \\
        1
    \end{pmatrix} \\
    \end{align*}
\]  

The roll angle of the cradle \( \phi_c \) is used to locate the rotating cradle with respect to the fixed machine coordinate system \( S_m \). Parameters \( U \), \( \theta \) and \( \phi_c \), along with various machine tool settings can be used to completely define the location of a point on the cutting head in space.

Since the kinematic motion of cutting a gear is equivalent to the cutting head meshing with a simulated crown gear, an equation of meshing can be written in terms of a point on the cutting head (i.e., in terms of \( U \), \( \theta \) and \( \phi_c \)). The equation of meshing for straight-sided cutters with a constant ratio of roll
between the cutter and work piece is given in [1, 5, 6] as:

\[(U - r \cot \psi \cos \psi) \cos \gamma \sin \tau + S[(m - \sin \gamma) \times \cos \psi \sin \theta \mp \cos \gamma \sin \gamma \sin(q - \phi_c)]
\]

\[\pm E_m(\cos \gamma \sin \psi + \sin \gamma \cos \psi \cos \tau) - L_m \sin \gamma \cos \psi \sin \tau = 0 \quad (2)\]

The upper and lower signs are for left and right hand gears, respectively. The following machine tool settings are defined [5, 6, 9]:

- \(\psi\) = cutting blade angle
- \(\tau = (\theta \mp q \pm \phi_c)\)
- \(q\) = cradle angle
- \(\gamma\) = root angle of work piece
- \(E_m\) = machining offset
- \(L_m\) = vector sum of change of machine center to back and the sliding base
- \(m\) = the relationship between the cradle and work piece for a constant ratio of roll
- \(S\) = radial location of cutting head in coordinate system \(S_m\)
- \(r\) = radius of generating cone surface

This is equivalent to:

\[f_1(U, \theta, \phi_c) = 0 \quad (3)\]

Because there are 3 unknowns, \(U, \theta, \text{ and } \phi_c\), three equations must be developed to solve for the surface coordinates of a spiral bevel gear. The 3 parameters \(U, \theta\) and \(\phi_c\) are defined relative to the cutting head and cradle coordinate systems \((S_c\) and \(S_c\)) respectively. These parameters can be transformed through a series of coordinate transformations to a coordinate system attached to the work piece. Or \(U, \theta, \phi_c\) can be mapped into \(X_w, Y_w, Z_w\) in coordinate system \(S_w\) attached to the work piece. These transformations, used in conjunction with two other geometric requirements, give the two additional equations.

The correct \(U, \theta\) and \(\phi_c\) that solves the equation of meshing, must also, upon transformation to the work piece coordinate system \(S_w\), result in an axial coordinate \(Z_w\) that matches with a preselected axial position \(Z\). (See Fig. 2)

\[Z_w - Z = 0 \quad (4)\]

This equation along with the correct coordinate transformations [see Eq. (11)] result in a second equation of the form:

\[f_2(U, \theta, \phi_c) = 0 \quad (5)\]

A similar requirements for the radial location of a point on the work piece results in the following:

\[r - (X_w^2 + Y_w^2)^{1/2} = 0 \quad (6)\]

The appropriate coordinate transformations (see Eq. (11)) convert Eq. (6) into a function of \(U, \theta, \text{ and } \phi_c\).

\[f_3(U, \theta, \phi_c) = 0 \quad (7)\]

Equations (3), (5) and (7) form the system of nonlinear equations necessary to define a point on the tooth surface.

**Solution Technique**

An initial guess \(U^0, \theta^0, \phi_c^0\) is used to start iterative solution procedures. Newton's method is used to determine subsequent values of the updated vector \((U^k, \theta^k, \phi_c^k)\) [10].

Where the vector \(Y\) is the solution of:

The \(3 \times 3\) matrix in the preceding equation is the Jacobian matrix and must be inverted each iteration to solve for the vector. The equation of meshing, function \(f_1\) is numerically
differentiated directly to find the terms for the
\[
\begin{align*}
U^k & = U^{k-1} + \left( Y_{1}^{k-1} \right) \\
\theta^k & = \theta^{k-1} + \left( Y_{2}^{k-1} \right) \\
\phi^k & = \phi^{k-1} + \left( Y_{3}^{k-1} \right)
\end{align*}
\]
(8)

\[
\begin{bmatrix}
\frac{\partial f_1(U^{k-1})}{\partial U} & \frac{\partial f_1(\theta^{k-1})}{\partial \theta} & \frac{\partial f_1(\phi^{k-1})}{\partial \phi} \\
\frac{\partial f_2(U^{k-1})}{\partial U} & \frac{\partial f_2(\theta^{k-1})}{\partial \theta} & \frac{\partial f_2(\phi^{k-1})}{\partial \phi} \\
\frac{\partial f_3(U^{k-1})}{\partial U} & \frac{\partial f_3(\theta^{k-1})}{\partial \theta} & \frac{\partial f_3(\phi^{k-1})}{\partial \phi}
\end{bmatrix}
\]

\[
\begin{bmatrix}
Y_{1}^{k-1} \\
Y_{2}^{k-1} \\
Y_{3}^{k-1}
\end{bmatrix}
\]

\[
\begin{bmatrix}
f_1(U^{k-1}, \theta^{k-1}, \phi^{k-1}) \\
f_2(U^{k-1}, \theta^{k-1}, \phi^{k-1}) \\
f_3(U^{k-1}, \alpha^{k-1}, \phi^{k-1})
\end{bmatrix}
\]
(9)

Jacobian matrix. Function \( f_2 \) and \( f_3 \) cannot be directly differentiated with respect to \( U, \theta \) and \( \phi \). After each iteration \( U^{k-1}, \theta^{k-1}, \phi^{k-1} \) (in the cutting head coordinate system \( Sc \)) are transformed into the work piece coordinate system, \( S_w \), with the series of coordinate transformations as given in Eq. (11).

\[
\begin{bmatrix}
X_w \\
Y_w \\
Z_w
\end{bmatrix}
= [M_{wc}]
\begin{bmatrix}
r \cot \psi - U \cos \psi \\
U \sin \psi \sin \theta \\
U \sin \psi \cos \theta
\end{bmatrix}
\]
(10)

where:

\[
[M_{wc}] = [M_{waf}(\phi_c)][M_{ap}][M_{pm}][M_{msf}(\phi_c)][M_{mc}]
\]
(11)

Each matrix [\( M \)] above represents a transformation from one coordinate system to another. (See Appendix I for the specific matrices.)

Functions \( f_2 \) and \( f_3 \) are evaluated by starting with an initial \( U_k, \theta_k \) and \( \phi_k \), performing the transformation in Eq. (11) and evaluating Eqs. (4) and (6). The numerical differentiation of \( f_2 \) and \( f_3 \) is performed by transforming \( U^k + \text{inc}, \theta^k + \text{inc}, \phi^k + \text{inc} \) (where inc is a small increment appropriate for numerical differentiation into \( X_w + \text{inc}, Y_w + \text{inc}, Z_w + \text{inc} \). Equations (4) and (6) are then used to evaluate the numerical differentiation. Function \( f_1, f_2, f_3 \) required to the Jacobian matrix are updated each iteration. The iteration continues until the \( Y \) vector is less than a predetermined tolerance. This completes the solution techniques for a single point on the spiral bevel gear surface.

The four corners of the active profile are identified from the tooth geometry plane as shown in Fig. 2. Point 1 on the surface is chosen to be the lowest point of the active profile on the toe end. The initial guess to start the procedure has to be sufficiently close to the correct solution for convergence to occur. The solution for the first point proves to be an adequate initial guess for any subsequent points on the surface.

Subsequent interior surface points are found by incrementing \( r = (X^2 + Y^2)^{1/2} \) and \( Z \). By adjusting the increments used, a surface mesh of any density can be calculated. The process is repeated four times for each of four surfaces; gear convex, gear concave, pinion convex and pinion concave. Software was written to solve for the tooth surface coordinates. Additional software converted the surface coordinates into input commands for the 3D solid modeler PATRAN [11]. PATRAN was used to create the nodes and elements for the FE code NASTRAN [12].

Since all four surfaces are generated independently, additional matrix transformations are required to obtain the appropriate orientation for meshing. The proper convex and concave surface orientation (for both the gear and pinion) is found by fixing the concave surface and rotating the convex surface until the correct tooth thickness is obtained. The correct angle of rotation is obtained by matching the tooth top land thickness with the desired value.

**Gear and Pinion Orientations Required For Meshing**

After generating the pinion and gear surface as described above, the pinion cone and gear cone apex will meet at the same point as shown in Fig. 3. This point is the origin of the fixed coordinate system attached to the work piece being generated. To place the gear and pinion in mesh with each other rotations described in the following example are required:

1. The 19th tooth is selected for meshing. This corresponds to rotating the generated tooth 190 deg. CW about the \( Z \) axis. (For this example, \( N_t = 36 \) total teeth on the
For the general case the gear tooth is rotated \(360/N_x/2 + 1\) deg CW about its axis of rotation \((Z_w)\). This corresponds to selecting the \(i\)th tooth of the gear to be in mesh where \(i = N_x/2 + 1\).

(2) The pinion is rotated by 90 deg. CCW about the \(Y\) axis. Note: rotation (1) corresponds to selecting a different tooth on the gear to be in mesh; however, rotation (2) corresponds to physically rotating the entire pinion until it meshes with the gear.

(3) Because the four surfaces are derived independently, their orientation is random with respect to meshing. The physical location of the gear and pinion after rotations (1) and (2) correspond to the gear and pinion in mesh with severe interference. To correct the interference the pinion is rotated CW about its axis of rotation \((Z_w)\) until surface contact occurs. For this example the rotation was 3.56 deg. Figure 4 shows an example of a simulated gear pair meshing. The generated pinion tooth was copied and rotated 12 times and the generated gear tooth was copied and rotated 36 times.

**Contact Simulation**

The tooth pair mesh contact point can be located by a method described by Litvin [6] or by a search technique. Pairs of finite element node points (one on each tooth surface) are evaluated until the pair with smallest separation distance is obtained. (A finer finite element mesh would yield greater resolution.) Once the contact point is established, a vector normal to the surface at the contact point is calculated.

The intersection of the normal vector on the pinion at the contact point with the gear surface identifies the second point required to define the gap element. Additional gap elements are obtained by taking additional vectors from other pinion surface finite element nodal points (parallel to the contact point normal vector), and calculating where they intersect the mating gear surface. Finite element nodal points on the gear surface are located to the intersection points of the normal vectors and the gear tooth surface.

The vector normal to the cutting surface is given in [5] as:

\[
\mathbf{n}_m = \begin{bmatrix} \sin \psi_r \\ \cos \psi_r \sin \tau \\ \cos \psi_r \cos \tau \end{bmatrix}
\]

These equations lead to a system of 3 nonlinear algebraic equations. These three equations, along with the equation of meshing for the gear surface, provide a system of 4 equations and 4 unknowns \((u, \theta, \phi_r, b)\). These equations are solved with Newton’s method described earlier. The intersection of the normal from node 1 on the pinion with the gear surface is now obtained.

This procedure is used to locate the intersection of normals from all points on the pinion surface (in the contact zone) with the gear surface. The gear tooth surface is remeshed utilizing the intersection points as shown in Fig. 5. Gap elements are connected between corresponding nodal points on the pinion and the intersection points on the gear surface.

**Finite Element Model**

An example analysis was performed using gears from the NASA Lewis Spiral Bevel Gear Test Facility. In this case, the left hand pinion mates with the right hand gear. Counter clockwise rotation of the pinion results in the concave surface on the pinion mating with the convex gear surface.

The design data for the pinion and gear are given in Table 1. The design data are used with methods given in [6] to determine the machine tool settings for the straight sided cutter data given in Table 2.

The finite element gear pair model, shown in Fig. 6, contains 4 gear teeth and three pinion teeth. The model had 10,101 nodes (30,303 degrees of freedom) and 7596 eight noded 3D brick elements. The analysis was done on a general purpose FE code [12].
The three pinion teeth are fixed in space with zero displacements and rotations. This is done by setting \( x \), \( y \) and \( z \) displacements equal to zero on the four corner nodes of each rim section. The gear is constrained to rotation about its axis of rotation. The gear is loaded with a torque of 9450 in lbs on the gear by applying 4725 lb force located 2.0976 from the gear axis of rotation.

At the orientation chosen between the pinion and gear two pairs of teeth were in contact. One pair had contact near the middle region of the tooth and another pair had contact near the toe (i.e., about to go out of mesh). The solution iterated four times before reaching equilibrium with four gap element closed in each contact zone and one closed in the edge contact region. Stress contours for the pinion tooth with contact are shown in Fig. 7.

The average nodal minimum principal stresses in the main contact zone average \(-204,000\) psi with a maximum of \(-299,900\) psi and a minimum of \(-103,574\) psi. The corresponding elemental stresses average \(-103,500\) psi with a maximum and minimum of \(-123,900\) psi and \(-79,500\) psi, respectively. The nodal stresses are higher because of load concentration from the gap elements. These stress ranges bracket the estimated hertzian stresses for the gear set under the same load conditions. Contact with only 4 gap elements, along with large stress gradient among adjacent nodes indicate the need for a finer finite element mesh for improved stress prediction.

### Table 1 Pinion and gear design data

<table>
<thead>
<tr>
<th></th>
<th>PINION</th>
<th>GEAR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of teeth pinion</td>
<td>12</td>
<td>36</td>
</tr>
<tr>
<td>Dedendum angle, deg</td>
<td>1.5666</td>
<td>3.8833</td>
</tr>
<tr>
<td>Addendum angle, deg</td>
<td>3.8833</td>
<td>1.5666</td>
</tr>
<tr>
<td>Pitch angle, deg</td>
<td>18.4333</td>
<td>71.5666</td>
</tr>
<tr>
<td>Shaft angle, deg</td>
<td>90.0</td>
<td>90.0</td>
</tr>
<tr>
<td>Mean spiral angle, deg</td>
<td>35.0</td>
<td>35.0</td>
</tr>
<tr>
<td>Face width, mm (in)</td>
<td>25.4(1.0)</td>
<td>25.4(1.0)</td>
</tr>
<tr>
<td>Mean cone distance, mm (in)</td>
<td>81.05(3.191)</td>
<td>81.05(3.191)</td>
</tr>
<tr>
<td>Inside radius of gear blank, mm (in)</td>
<td>5.3 (0.6094)</td>
<td>3.0 (3.449)</td>
</tr>
<tr>
<td>Top land thickness, mm (in)</td>
<td>2.032 (0.080)</td>
<td>2.489 (.098)</td>
</tr>
<tr>
<td>Clearance, mm (in)</td>
<td>0.762 (0.030)</td>
<td>0.92964 (0.0366)</td>
</tr>
</tbody>
</table>

### Table 2 Generation machine settings

<table>
<thead>
<tr>
<th></th>
<th>PINION</th>
<th>GEAR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius of cutter, r, in</td>
<td>2.9656</td>
<td>3.0713</td>
</tr>
<tr>
<td>Blade angle, ( \psi ), deg</td>
<td>161.9543</td>
<td>24.3374</td>
</tr>
<tr>
<td>Vector sum, ( L_1 )</td>
<td>0.03849</td>
<td>-0.05181</td>
</tr>
<tr>
<td>Machine offset, ( E_m ), in</td>
<td>0.15457</td>
<td>-0.17426</td>
</tr>
<tr>
<td>Cradle to cutter distance, s, in</td>
<td>2.94780</td>
<td>2.80104</td>
</tr>
<tr>
<td>Cradle angle q, deg</td>
<td>63.94</td>
<td>53.926</td>
</tr>
<tr>
<td>Ratio of roll, ( M_r )</td>
<td>0.30838</td>
<td>0.32204</td>
</tr>
<tr>
<td>Initial cutter length, ( u ), in</td>
<td>9.59703</td>
<td>7.42534</td>
</tr>
<tr>
<td>Initial cutter orientation, ( \theta ), deg</td>
<td>126.83544</td>
<td>124.36389</td>
</tr>
<tr>
<td>Initial cradle orientation, ( \phi ), deg</td>
<td>-0.85813</td>
<td>-11.38663</td>
</tr>
</tbody>
</table>

### Conclusions

A multi tooth finite element model was used to perform three-dimensional contact analysis of spiral bevel gears in mesh. Four gear teeth and three pinion teeth are generated by solving the equations based on gear manufacturing kinematics, that identify tooth surface coordinates. The gear and pinion are orientated for meshing with coordinate transformations. Software was written to solve for the tooth surface coordinates, and create input commands for a 3D solid modeler. The solid modeler created the nodes and elements used as input for a general purpose finite element code.

Surface stresses are evaluated with gap elements. The gear surface is remeshed with nodal points relocated to identify...
the correct gap element orientation. Initial FEA stress results compare favorably with calculated hertzian contact stresses. However; large stress gradients between adjacent nodes in the contact zones indicate a need for greater finite element mesh refinement.

References
11 PATRAN Users Manual Ver. 2.4 PDA Engineering, Costa Mesa, CA.
12 MSC-NASTRAN Users Manual, Ver. 65, MacNeal-Schwendler Corp., Los Angeles, CA.

APPENDIX I

Coordinate transformations involving both rotation and translation require mixed matrix operations of multiplication and addition. Matrix representation of coordinate transformations will need only multiplication of matrices if position vectors are represented by homogeneous coordinates [1]. The following 4 x 4 matrices are required to transform the homogeneous coordinates of a point on the cutting head to a point on the work piece.

Matrix \( [M_{ce}] \) transforms coordinate system \( S_e \) into system \( S_m \) attached to the frame.

\[
[M_{ce}] = \begin{bmatrix}
1 & 0 & 0 & 0 \\
0 & \cos q & \mp \sin q & \mp \sin q \\
0 & \pm \sin q & \cos q & s \cos q \\
0 & 0 & 0 & 1
\end{bmatrix}
\]  \hspace{1cm} (16)

Matrix \( [M_{m_s}] \) transforms coordinate system \( S_s \) into system \( S_m \) attached to the frame.

\[
[M_{m_s}] = \begin{bmatrix}
1 & 0 & 0 & 0 \\
0 & \cos \phi_c & \mp \sin \phi_c & 0 \\
0 & \pm \sin \phi_c & \cos \phi_c & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]  \hspace{1cm} (17)

Matrix \( [M_{pm}] \) transforms coordinate system \( S_m \) into system \( S_p \) which orientates the apex of the gear being manufactured.

\[
[M_{pm}] = \begin{bmatrix}
\cos \delta & 0 & -\sin \delta & -L_m \sin \delta \\
0 & 1 & 0 & \pm E_m \\
\sin \delta & 0 & \cos \delta & L_m \cos \delta \\
0 & 0 & 0 & 1
\end{bmatrix}
\]  \hspace{1cm} (18)

Matrix \( [M_{ap}] \) transforms coordinate system \( S_a \) into system \( S_p \) and locates the apex of the gear being manufactured with respect to \( S_m \).

\[
[M_{ap}] = \begin{bmatrix}
\cos \mu & 0 & \sin \mu & 0 \\
\mp \sin \phi_w & \cos \phi_w & 0 & 0 \\
-\sin \mu & 0 & \cos \mu & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]  \hspace{1cm} (19)

Matrix \( [M_{wa}] \) transforms coordinate \( S_a \) into system \( S_w \) attached to the work piece.

\[
[M_{wa}] = \begin{bmatrix}
\cos \phi_w & \pm \sin \phi_w & 0 & 0 \\
\mp \sin \phi_w & \cos \phi_w & 0 & 0 \\
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]  \hspace{1cm} (20)