Application of Face-Gear Drives in Helicopter Transmissions

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ABSTRACT

The use of face gears in helicopter transmissions was explored. A light-weight, split torque transmission design utilizing face gears was described. Face-gear design and geometry were investigated. Topics included tooth generation, limiting inner and outer radii, tooth contact analysis, contact ratio, gear eccentricity, and structural stiffness. Design charts were developed to determine minimum and maximum face-gear inner and outer radii. Analytical study of transmission error showed face-gear drives were relatively insensitive to gear misalignment, but tooth contact was affected by misalignment. A method of localizing bearing contact to compensate for misalignment was explored. The proper choice of shaft support stiffness enabled good load sharing in the split torque transmission design. Face-gear experimental studies were also included and the feasibility of face gears in high-speed, high-load applications such as helicopter transmissions was demonstrated.

INTRODUCTION

The Advanced Rotorcraft Transmission (ART) program is an Army funded, joint Army/NASA program to develop and demonstrate lightweight, quiet, durable drivetrain systems for next generation rotorcraft (Bill, 1990). One contract team participant, McDonnell Douglas Helicopter Company (MDHC)/Lucas Western Incorporated, developed a novel split torque ART configuration using face gears (Bossier and Heath, 1990, 1991). The geometry and design of face gears and computerized simulation of their meshing have been developed by another member of the team, the University of Illinois at Chicago.

Manufacturing of face gears was proposed many years ago by the Fellows Corporation. Face gears have had widespread use in low power applications but have not had much development of design and manufacturing practice for high power use.

The theory of face-gear drives has not been developed sufficiently for the needs of the designers and manufacturers. Publications in this area in English by E. Buckingham (1949), and D.W. Dudley (1962) can be considered only as a brief description of face-gear drives. J. Davidov (1950), and F.L. Litvin and L.I. LiBurkin (1968) have published the results of their investiga-

SPLIT TORQUE DESIGN

The idea of the split of torque is illustrated with Fig. 1. Figure 1(a) shows the alternative version of the torque split by two spiral bevel pinions, a and b, designed as one rigid body. Figure 1(b) shows the second version of the split of torque when a single spur (or helical) pinion is in mesh with two face gears. The advantage of the second version is that the transmitting forces transmit a reduced load on the bearings in comparison with the version shown in Fig. 1(a). A second advantage is that the pinion is a conventional spur (or helical) gear compared to a complex spiral bevel design with two pinions.

The general configuration of the MDHC/Lucas ART design is illustrated conceptually in Fig. 2, although there have been some changes in details since this drawing was made. There are two
predicted payoff is greatly reduced weight and cost compared to conventional design. The pinion which serves the two face gears is a conventional spur gear with an even number of teeth. If the spur gears were rigidly located between the two face gears, precise torque splitting would be very unlikely. The spur gear has a free-floating mount which allows self-centering between the two face gears. It will be shown analytically (see next section) that precise torque splitting (with ±1.0%) will take place.

More importantly, torque splitting between two driven gears by a free-floating spur-gear pinion has been used for many years in truck transmissions. The first known truck application was the experimental Road Ranger transmission produced by the Fuller Transmission Division of the Eaton Manufacturing Company in 1961. Truck transmissions using this principle have been in production since 1963. In addition to accurate torque splitting, it was found that gear noise was reduced and gear life was increased. Thus the use of a free floating pinion as a torque-splitting device is well substantiated.

FINITE ELEMENT STRUCTURAL ANALYSIS FOR THE SPLIT TORQUE GEAR DRIVE

The success of a split-torque gear train design depends on the equal division of the torque to the two output shafts. Conceptually, the floating pinion design makes the system of the pinion shaft a two-force member. The transmitted forces on the two diametrically opposite meshing points on the pinion have to balance each other to achieve equal torque splitting.

The analytical effort to validate the split torque concept was conducted with the use of a finite element method. To analyze the deflection and the percentage of torque splitting in real condition, the elasticity of the gear structure and the shaft has to be considered first. The finite element model provides an accurate approach to include the stiffness and the deflection of the gear structure. The overall model of the split-torque gear train is shown in Fig. 3. The model has been used to analyze the torque splitting percentage for different support conditions as shown in Table I. The stiffnesses of the front and rear support of the pinion shaft were varied to determine their effect on torque split (cases 1 to 4, Table I). The contacts between the pinion and the two face gears were modeled using gap elements. The torque split was determined using gap element reaction forces as calculated using finite element analysis. Among the first four cases studied, the most even torque split was provided in case 4 when the stiffness of the shaft's front support was $1.1 \times 10^6$ N/m ($6.0 \times 10^4$ lb/in.). This is an order of magnitude less than a typical bearing support and is realistic.
Influence of Gear Eccentricity

The influence of gear eccentricity is important for determination of conditions of the split of torques when one pinion is in mesh with two face gears, and the pinion and the gears have eccentricity. Due to transmission errors the driven face gears will perform rotation with slightly different angular velocities, and this means that the split of torques will be accompanied with deflections of tooth surfaces.

Transmission error is defined as,

$$\Delta \phi_{T} = \phi_{T2} - \frac{N_1}{N_2} (\phi_{T1} - \phi_{T1})$$

(1)

where $N_1$ and $N_2$ are the number of teeth of the pinion and the gear, respectively; $\phi_{T1}$ and $\phi_{T2}$ are the angles of rotation of the pinion and the gear, respectively; $\phi_{T1}$ is the value of $\phi_{T1}$ that corresponds to $\phi_{T2} = 0$.

The results of investigation show that the transmission error curve due to eccentricity of the pinion or the gear is an approximate harmonic curve. The periods of these curves are the time periods for one revolution of the pinion and the gear, respectively. The transmission errors for the case with eccentricity of both the pinion and the gear is a periodic function as shown in Fig. 4. The period of this curve is determined by the lowest common multiple of the numbers of teeth of the pinion and the gear.

The great advantage of face-gear drive with an involute pinion is that the pinion teeth are equidistant and have a common normal. This means at the end of the meshing of a pair of teeth and the beginning of meshing a next pair of neighboring teeth both tooth pairs will have a common normal. Therefore, the change of tooth meshing at the transfer point will not cause a jump of angular velocity.

<table>
<thead>
<tr>
<th>Case number</th>
<th>Support of pinion shaft</th>
<th>Gear meshing clearance, mm (in.)</th>
<th>Split torque, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Front-end spring rate, N/M (lb/in.)</td>
<td>Rear-end spring rate, N/M (lb/in.)</td>
<td>Due to backlash</td>
</tr>
<tr>
<td>1</td>
<td>0 (Free float)</td>
<td>= (Restrained)</td>
<td>0.0 (0.0)</td>
</tr>
<tr>
<td>2</td>
<td>=</td>
<td>=</td>
<td>56.87</td>
</tr>
<tr>
<td>3</td>
<td>$1.1 \times 10^6 (6.0 \times 10^5)$</td>
<td>$1.1 \times 10^6 (6.0 \times 10^5)$</td>
<td>53.11</td>
</tr>
<tr>
<td>4</td>
<td>$1.1 \times 10^7 (6.0 \times 10^6)$</td>
<td>$1.1 \times 10^7 (6.0 \times 10^6)$</td>
<td>51.41</td>
</tr>
<tr>
<td>5</td>
<td>=</td>
<td>=</td>
<td>51.55</td>
</tr>
<tr>
<td>6</td>
<td>=</td>
<td>=</td>
<td>52.86</td>
</tr>
<tr>
<td>7</td>
<td>=</td>
<td>=</td>
<td>81.66</td>
</tr>
<tr>
<td>8</td>
<td>$1.1 \times 10^7 (6.0 \times 10^6)$</td>
<td>$1.1 \times 10^7 (6.0 \times 10^6)$</td>
<td>50.54</td>
</tr>
<tr>
<td>9</td>
<td>=</td>
<td>=</td>
<td>40.97</td>
</tr>
</tbody>
</table>

* $1.1 \times 10^6$ N/M (6.0 $\times 10^5$ lb/in.) is the translational spring rate for typical bearing and housing support in helicopter transmission.
This statement is correct as well for eccentric conventional involute gears. This implies that noise and vibration are relatively insensitive to gear misalignment.

BASIC TOPICS OF GEAR DESIGN AND MANUFACTURING

Generation of Face-Gear Drives with Localized Bearing Contact

The generation of the face gear by a shaper is shown in Fig. 5. The shaper and the gear rotate about intersecting axes with angular velocities $\omega(s)$ and $\omega(2)$ that are related as follows

$$\frac{\omega(s)}{\omega(2)} = \frac{N_2}{N_3}$$

(2)

The designations of $s$ and 2 indicate the shaper and the face gear, respectively.

If the face gear is generated by a shaper that is identical to the pinion, the process of generation simulates the meshing of the pinion with the face gear being in line contact at every instant. In reality, such type of contact cannot be implemented due to its sensitivity to misalignment. The errors (tolerances) of assembly and manufacturing can cause separation of the contacting surfaces and results in the undesirable contact at the edge. To avoid this, it is necessary to use a shaper with a larger number of teeth. The difference is denoted as $\Delta N = N_s - N_1$ ($N_1$ is the number of the pinion teeth; $\Delta N$ ranges from 1 to 3).

The geometric aspects of localization of bearing contact are illustrated with drawings of Fig. 6. We may imagine that three surfaces $\Sigma_1$, $\Sigma_1$, and $\Sigma_2$ are in mesh with each other simultaneously. Surfaces $\Sigma_s$ and $\Sigma_2$ are in line contact at every instant in the process for generation. Surfaces $\Sigma_s$ and $\Sigma_1$ are also in line contact being in an imaginary internal engagement as shown in Fig. 6. The imaginary meshing of the shaper and the pinion may be considered as a meshing with the following features: (1) the center distance $B$ depends on the difference $\Delta N$ of the number of teeth of the shaper and the pinion; (2) there is an instantaneous axis of rotation that intersects the extended center distance $O_sO_1$ at point $P$ and is parallel to the axes of the pinion and the shaper; (3) the instantaneous line of contact of $\Sigma_2$ and $\Sigma_1$ is a straight line $L_{s1}$ that is parallel to the axes of the shaper and the pinion; $M$ is the point of intersection of $L_{s1}$ with the plane of drawings; (4) surface $\Sigma_s$ and $\Sigma_2$ are in line contact (at $L_{s2}$) at every instant; and (5) surfaces $\Sigma_2$ and $\Sigma_1$ are in point contact and the instantaneous point of contact is the intersection of $L_{s1}$ and $L_{s2}$.

The location of point $P$ can be determined as the intersection of the common tangent to the base circles of the shaper and the pinion with the extended center distance $O_sO_1$ (Fig. 6). $PM$ is the common normal to the involute shapes of the shaper and the pinion.

The contact of the pinion and the face-gear surfaces under the load is a contact over an elliptical area; the center of such an ellipse is the theoretical contact point of $\Sigma_2$ and $\Sigma_1$.

The input design data for an example of a face-gear drive are given in Table II. These data are used for computations demonstrated in the following sections.

<table>
<thead>
<tr>
<th>Table II.</th>
<th>INPUT FACE-GEAR DRIVE DESIGN DATA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft angle, deg</td>
<td>80</td>
</tr>
<tr>
<td>Pinion number of teeth</td>
<td>28</td>
</tr>
<tr>
<td>Gear number of teeth</td>
<td>107</td>
</tr>
<tr>
<td>Diametral pitch</td>
<td>8</td>
</tr>
<tr>
<td>Pressure angle, deg</td>
<td>28</td>
</tr>
</tbody>
</table>

Meshing of the Shaper and the Face Gear

The shaper tooth surface $\Sigma_2$ and the face-gear tooth surface $\Sigma_2$ contact each other at every instant at a spatial line $L_{s2}$. Contact lines on $\Sigma_s$ and $\Sigma_2$ are shown in Fig. 7. The contact lines on $\Sigma_s$ and $\Sigma_2$ are derived from the following equations (Litvin, 1989):
1. Contact lines on the shaper surface (Fig. 7 (a)) are defined as
   \[ r_s(u_0,\theta_s) \cdot \psi^{(2)} = f(u_0,\theta_s,\phi_s) = 0 \quad (3) \]

2. Contact lines on the face-gear surface (Fig. 7 (b)) are determined as
   \[ r_2(u_0,\theta_2,\phi_2) = f(u_0,\theta_2,\phi_2) = 0 \quad (4) \]

Here, \((u_0,\theta_0)\) are the Gaussian coordinates of the involute shaper surface (see Appendix A in Litvin et al., 1992) and \(\phi_s\) is the generalized parameter of motion.

Tooth surface \(\Sigma_2\) of the face gear is represented by Eqs. (4) in three-parametric form with an implicit relation between parameters \((u_0,\theta_2,\phi_2)\). Fortunately, the equation of meshing
   \[ f(u_0,\theta_2,\phi_2) = 0 \quad (5) \]

is linear with respect to \(u_0\) and this enables us to eliminate \(u_0\) and represent \(\Sigma_2\) in two-parametric form as
   \[ r_2 = r_2(\theta_2,\phi_2) \quad (6) \]

Limitations of Face-Gear Tooth Surface

The length of the tooth surface of a face gear is limited, due to the possibility of undercutting by the shaper in the dedendum area and the pointing of the teeth in the addendum area (Fig. 9).

The investigation of conditions of nonundercutting of the face gear is based on the theorem that has been proposed by Litvin (1989). There is a limiting line \(L\) on the generating surface (shaper surface \(\Sigma_3\)) that generates singular points on face-gear surface \(\Sigma_2\). The limiting line on \(\Sigma_3\) can be determined with the equation
   \[ V_f(u_0) + V^{(2)} = 0 \quad (7) \]

Here, \(V_f(u_0)\) is the velocity of contact point in its motion over \(\Sigma_3\); \(V^{(2)}\) is the sliding velocity of the shaper with respect to the face gear. The reflection line of the conjugate meshing part and the fillet on the face-gear tooth surface is designated by \(L_s\) as shown in Fig. 7(b). More details are given in Appendix B in Litvin (1992).

The pointing of teeth (Fig. 7(b)) means that the tooth thickness on the top of the tooth becomes equal to zero. The location of the tooth pointing area may be determined by considering the intersection of the two opposite tooth surfaces at the top land of a tooth.

Computer programs for determination of limitations of the length of the face gears have been developed at the University of Illinois at Chicago. A quick review of results obtained are represented in the following charts.

Figure 9 shows the minimum and maximum radius factors for the face gear with various gear ratio \(m_{22}\) and the shaper tooth numbers. In this example, the shaft angle is 80° and the pressure angle is 20°. The program is sufficiently general that it has the ability to generate design charts over a wide range. Knowing the values of minimum and maximum radius factor we can obtain the values of \(L_1\) and \(L_2\) (Fig. 10) by multiplying the radius factors by \(N_2/2P\) where \(N_2\) is the tooth number of the face gear and \(P\) is the diametral pitch. Using this method, \(L_1\) and \(L_2\) are in units of inches. For design convenience, a unitless design parameter \(u_1 = 1P\) where \(1 = L_2 - L_1\) is usually considered. This parameter is similar to the parameter that express the ratio \(l/m\) where \(m = 25.4/P\) is the module of spur or helical gears. For power transmissions it is desirable to keep \(u_1 \geq 7\). Our investigation shows that this can be obtained with the gear ratio (number of face
Pressure angle = 20°
Standard tooth height
Shaft angle = 80°

\[ m_{s2} = 3 \]
\[ m_{s2} = 4 \]
\[ m_{s2} = 5 \]

(a) Minimum radius factor.

(b) Maximum radius factor.

Figure 9.—Face-gear minimum and maximum radius factors.

gear teeth divided by number of pinion teeth) \( \geq 3.8 \). Using the data in Table II, we have obtained \( L_1 = 160 \text{ mm (6.3 in.)} \), \( L_2 = 193 \text{ mm (7.6 in.)} \).

Computerized Simulation of Meshing and Contact of Pinion and Face Gear

The bearing contact of pinion and face-gear tooth surfaces \( \Sigma_1 \) and \( \Sigma_2 \) is localized using the technique described in the previous section. \( \Sigma_1 \) and \( \Sigma_2 \) are in point contact at every instant. The computerized simulation of meshing and contact of \( \Sigma_1 \) and \( \Sigma_2 \) (Tooth Contact Analysis; TCA) can provide information on transmission errors and the shift of bearing contact that is caused by pinion-face-gear misalignment.

The idea of TCA is based on equations of tangency of contacting surfaces (Fig. 11). Such equations express that the contacting surfaces \( \Sigma_1 \) and \( \Sigma_2 \) have, at any instant, a common position vector and collinear normals at their contact point \( M \). For more details see Litvin (1989) (and Appendix C in Litvin, 1992).

Our investigation shows that the gear misalignment (change of the shaft angle, crossing of axes instead of intersection, axial displacement of face gear) does not cause transmission errors. This is a great advantage of face-gear drives in comparison with spiral bevel gear drive.

However, gear misalignment does result in the shift of the contact path on the gear surfaces. The patterns of the bearing contact can be determined considering the motion of the instantaneous contact ellipse over the pinion-gear tooth surfaces in the process of meshing. The dimensions and orientation of the instantaneous contact ellipse can be found if the principal directions and
curvatures of the contacting surfaces are determined at the current point of surface contact (Litvin, 1992). The equations for computation of principal curvatures and directions are given in Appendix D in Litvin (1992). The elastic approach of the surfaces is considered as known.

It is possible to control the location of the bearing contact by changing of the machine angle $\gamma_m$ that is formed by the axes of the shaper and the face gear. However, the small magnitude of $\Delta \gamma_m$ can be only implemented with a very precise control of $\gamma_m$.

Figure 12 shows an example of the face-gear bearing contact prediction. The shift of bearing contact caused by gear misalignment and change in machine angle is given in Litvin (1992).

![Contact path](image)

**Figure 12.—Aligned face-gear drive: Localized bearing contact with $\Delta N = 3$.**

**Theoretical and Real Contact Ratio**

The contact ratio $m_c$ is determined with the equation

$$m_c = \frac{\phi(3) - \phi(1)}{360^\circ}$$

Here; $\phi(2)$ and $\phi(1)$ represent the angles of rotation of the pinion that correspond to the beginning and the end of meshing for one pair of teeth; $N_1$ is the number of pinion teeth. Angles $\phi(1)$ and $\phi(1)$ can be determined from drawings of Fig. 7(b) that show the instantaneous contact lines referred to angles of pinion rotation. Taking into account that for drawings of Fig. 7(b) the stepsize of $\phi_1$ is 3°, the number of contact lines that cover the surface of face gear is 10, and $N_1 = 28$ (see Table II), we obtain that the theoretical value of $m_c$ is 2.33.

The localization of bearing contact is accompanied with the reduction of contact ratio, since the number of potential contact ellipses is reduced. Using an approach that is similar to the one discussed above, we have determined that $(\phi(3) - \phi(1))$ is 20.8°, and the real contact ratio is 1.62.

**Generation of Face Gears by Rack-Cutter**

The installation of the rack-cutter and the gear is as shown in Fig. 13. While the gear performs rotational motion about its axis, the rack-cutter performs translational motion in $\sigma_p$ axis.

The undercutting and pointing of the gear tooth surface limit the length of the generated face gears. For those design data listed in Table II, the tooth length $P$ of the generated face gear is 0.3 in., which is too small for practical application (because $q_l = P/l = 2.4 < 7$).

![Diagram](image)

**Figure 13.—Installation of the rack-cutter.**

**EXPERIMENTAL TESTS**

Experimental tests on face gears were performed in the NASA Lewis spiral bevel gear rig (Handschuh et al., 1992). The face gears tested (Fig. 14) were basically a half-size version of the MDHC/Lucas ART design. The gears were 16 pitch with 28 teeth on the pinion and 107 on the face gear. The shaft angle was 90° to accommodate the rig. The gears were made of Maraging 300 steel per AMS 6514. The pinions were nitrided and ground with a case hardness of $R_C$ 58. The face gears were shaper cut and hardened to $R_C$ 52. For the tests, 100-percent design speed and torque were defined as 19 000 rpm pinion speed and 68 N/m (600 in./lb) pinion torque for a power of 135 kW (180 hp).

![Test gears](image)

**Figure 14.—Test gears.**

The NASA Lewis spiral bevel gear rig (Fig. 15) operates on a closed loop or torque-regenerative principle. Two sets of pinion/face gears are used in the loop with the two pinions connected by a cross shaft. The outputs of the two face gears are connected through a helical gear mesh. A hydraulic loading system is connected to the helical mesh which puts a thrust load on the mesh, and thus, the torque in the loop. A variable speed motor is connected by a belt to the loop and powers the test stand.

A limited amount of test gears were available for test (four pinions and four face gears). The objective of the tests were to demonstrate the feasibility of face gears and determine the failure modes for high power applications. Four sets of gears successfully completed 26-hr (30×10^6 pinion cycles) endurance runs at 100-percent speed and torque. The gears were run at 74 °C (165 °F) oil inlet temperature using an ample supply of DOD- L-85734 lubricant at about 0.8 gpm per mesh. The contact pattern on the teeth was good and developed on the full tooth of the face gear. The pinion teeth showed normal wear. The face-gear teeth, however, had some surface distress. The teeth from the test side
The teeth from the slave side (face gear driving the pinion) had small pit lines in some instances in the middle region of the teeth. The gears were subsequently run 26 hr at 200-percent torque and 100-percent speed. One test (two sets of gears) lasted the 26 hr with the pinions showing moderate wear and the face gears showing increasing surface distress. The second test (the additional two sets of gears) was suspended after about 10.5 hr due to a tooth breakage on one of the face gears (slave side). The breakage originated from the surface pit line from the previous test.

The results, although limited, demonstrated the feasibility of face gears in high-speed, high-load applications such as helicopter transmissions. Face gears which were basically a half-scale version of the MDHC/Lucas ART design were tested in the NASA Lewis spiral bevel rig. The pinions and the gears showed good contact patterns and ran at 100- and 200-percent design torque. However, the face gears did have some surface distress.

CONCLUSIONS

The use of face gears in helicopter transmissions was explored. A light-weight, split torque transmission design utilizing face gears was described. Face-gear design and geometry were investigated. Topics included tooth generation, minimum inner and maximum outer radii, tooth contact analysis, contact ratio, gear eccentricity, and structural stiffness. Face-gear experimental studies were also included. The following results were obtained:

1. The feasibility of face gears in high-speed, high-load applications such as helicopter transmissions was demonstrated through experimental testing. Face gears which were basically a half-scale version of the MDHC/Lucas ART design were tested in the NASA Lewis spiral bevel rig. The pinions and the gears showed good contact patterns and ran at 100- and 200-percent design torque. However, the face gears did have some surface distress.

2. Analytical transmission error studies showed face-gears were relatively insensitive to gear misalignment. Tooth contact, however, was affected by misalignment resulting in a shift of the contact on the tooth surfaces. A method of localizing contact by changing tool settings of the generating machine was explored.

3. The length of the face-gear tooth width was limited due to possible undercutting by the shaper in the dedendum area and pointing of the teeth in the addendum area. Design charts were developed to determine minimum inner and maximum outer radii.

4. A finite element analysis of the pinion and face-gear structure in a split torque design provided data on load sharing. Among the cases studied, an even torque split was provided when the stiffness of the pinion shaft front support (close to the face-gear mesh) was about an order of magnitude less than a typical bearing support.

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REFERENCES


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