TORQUE SPLITTING BY A CONCENTRIC FACE GEAR TRANSMISSION

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ABSTRACT

Tests of a 167 Kilowatt (224 Horsepower) split torque face gearbox were performed by the Boeing Company in Mesa, Arizona, while working under a Defense Advanced Research Projects Agency (DARPA) Technology Reinvestment Program (TRP). This paper provides a summary of these cooperative tests, which were jointly funded by Boeing and DARPA. Design, manufacture and testing of the scaled-power TRP proof-of-concept (POC) split torque gearbox followed preliminary evaluations of the concept performed early in the program. The split torque tests were run using 200 N-m (1767 in-lbs) torque input to each side of the transmission. During tests, two input pinions were slow rolled while in mesh with the two face gears. Two idler gears were also used in the configuration to recombine torque near the output. Resistance was applied at the output face gear to create the required loading conditions in the gear teeth. A system of weights, pulleys and cables were used in the test rig to provide both the input and output loading. Strain gages applied in the tooth root fillets provided strain indication used to determine torque splitting conditions at the input pinions. The final two pinion-two idler tests indicated 52% to 48% average torque split capabilities for the two pinions. During the same tests, a 57% to 43% average distribution of the torque being recombined to the upper face gear from the lower face gear was measured between the two idlers. The POC split torque tests demonstrated that face gears can be applied effectively in split torque rotorcraft transmissions, yielding good potential for significant weight, cost and reliability improvements over existing equipment using spiral bevel gearing.

INTRODUCTION

Drive system engineers continuously strive to develop improvements in gear, shaft and bearing configurations, as well as investigating new materials, processes, modular design methods and improved technology components. The TRP program and the 2828 Horsepower (HP) Demonstrator Transmission program, both jointly-funded by DARPA and Boeing, were initiated to develop and refine transmission technologies which will provide increased power density (more power per unit weight and volume), increased reliability and reduced costs. This development will allow aircraft integration of future transmissions requiring these improvements.

The contracting agency for the TRP program was the NASA Glenn Research Center, working in conjunction with the U.S. Army Research Laboratory in Cleveland, Ohio. These two agencies also provided engineering and facilities support in the face gear durability tests [1-4] during the TRP and earlier Advanced Rotorcraft Transmission (ART) I Programs.

A split torque proof-of-concept test gearbox has been designed and built as part of the TRP program. This is a reduced-size, scaled-power 167 kW (224 HP) split torque face gear design. The design is configured with two face gears located face-to-face one over the other, resulting in a compact size and cylindrical shape for the gearbox. Two input pinions and two idler gears are also used in this torque splitting arrangement. Part sizes are decreased due to torque being divided in half at the input pinion meshes, which results in significant weight reduction and allows use of a larger reduction ratio in the space available. For these reasons, a high ratio concentric split torque face gear stage can replace two stages of a conventional transmission. Following gear tooth pattern development, TRP slow-roll tests were conducted which involved both single and dual pinion loading. The tests utilized a test rig built at Boeing in Mesa.

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Face gear grinding development work was performed by Boeing, the University of Illinois at Chicago [5-6] and Derlan Aerospace Canada (Milton, Ontario), a major producer of aerospace-quality gears. Derlan built the face gear grinding machine from the base up, as found required to create the machine configuration and operating capabilities needed to precision grind face gears. To date, the developed method has been used to produce the POC test gears, as well as two 2828 HP test gear sets. This DARPA-related work is based on mathematical principles of face gear geometry, tooth contact, grinding wheel motions, grinding wheel dressing and coordinate measurement.

DARPA TECHNOLOGY REINVESTMENT PROGRAM SPLIT TORQUE TESTS

Proof-of-Concept Gearbox Configuration

The DARPA TRP proof-of-concept (POC) test gearbox was designed to allow determination of torque splitting efficiencies of the concentric face gear split torque concept at representative tooth stress levels. Shown in Figure 1, this is a reduced-size, scaled-power 167 kW (224 HP) split torque concentric face gear design. The final design of the POC test transmission, which was completed in 1998, is configured with two face gears located face-to-face one over the other in the gearbox. Two input pinions and two idler gears are also used in the configuration. From each side of the gearbox, one of the input pinions drives in between the two face gears, creating an upper and lower mesh. Facilitated by a cantilevered bearing mount arrangement that allows the pinions to float, input torque from each pinion is divided evenly to the two face gears. The two idler gears recombine the torque fed to the lower face gear from the pinions back to the upper (output) face gear. Part sizes are decreased substantially by dividing torque in half before recombining it at an output gear. The geometry of face gears also provides an inherent capability to handle larger reduction ratios. The concentric split torque face gear stage eliminates one of two input stages of a conventional transmission and results in substantial savings of weight, cost and volume. For a face gear transmission similar to the POC design, weight reduction is currently estimated at 22%. Ongoing development programs at Boeing Mesa, through modifications to the face gear transmission configuration, are projecting a power density improvement of 35%.

Proof-of-Concept Gearbox Design Criteria

For the POC gearbox design, the 100% input shaft torque for tests is 200 N-m (1767 in-lbs). The equivalent 100% power level at each shaft is 84 kW (112 HP) for a pinion speed of 4000 rpm. The 100% design torque level for each upper and lower pinion mesh with the face gears is 100 N-m (883.5 in-lbs). This later torque value assumed a 50%-50% torque split from each pinion to the face gears for design sizing purposes. The 100% design torque level for each face gear at each of its two pinion and two idler meshes is 403 N-m (3571 in-lbs). The lower face gear transfers torque (divided out to it from the lower pinion meshes) to the upper face gear through the idler gears. The upper face gear combines the torque from its four meshes, to provide 1614 N-m (14,283 in-lbs). at the output shaft. A complete production transmission using the face gear configuration would include an output planetary being driven from the upper face gear, an arrangement similar to the intended aircraft configuration.

The gear material for the POC face gears, pinions and idlers is 9310 steel per AMS 6265. The test gears for slow roll tests were as-heat treated, quenched and tempered, with a hardness of R_c 34-38. Since a viable production design would be surface-hardened, it was found appropriate to use allowables similar to those for hardened gears when making actual bending stress comparisons based on slow roll test strain data. The POC gears are 2.03 module (12.5 diametral pitch), with the pinions and idlers having 24 teeth and the face gears each having 97 teeth. The gears are of American Gear Manufacturers Association (AGMA) Class 12 quality. The POC transmission includes a non-90° input shaft angle with the face gear axis. The pressure angle for the gear teeth is 25°. Tooth backlash is within a design range of 0.152 to 0.254 mm (0.006 to 0.010 inch) for the set. Surface roughness of active tooth profiles was measured at an average of 0.41 micrometer (16 micro-inches).

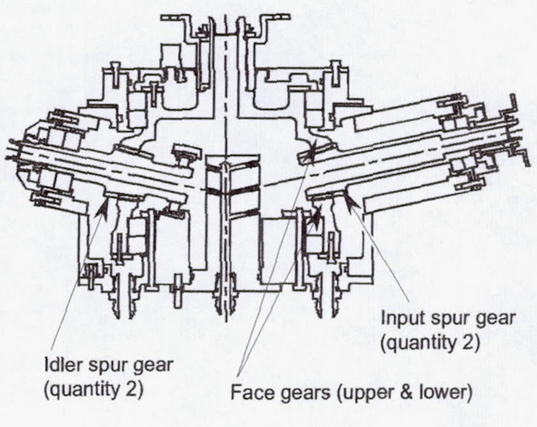

Figure 1. TRP Proof-of-Concept Gearbox

Drive hubs with keyway connections are used on the input and output shafts. All bearings and standard parts hardware for the test gearbox were purchased off-the-shelf per applicable quality standards. The gearbox housing weldment assembly was made from 6061 aluminum, T42 condition. The bottom cover was made from 6061 aluminum, T6 condition. The input, output and idler covers and sleeves were machined from 4340 steel.

Split Torque Test Objectives

The first objective of the split torque slow roll tests was to determine the quasi-static torque splitting efficiency of the input pinions as they mesh with the upper and lower face gears. The second objective was to determine relative percentages of load recombined from the lower face gear to the upper face gear through each of the two idlers used during dual pinion tests. The third objective was to investigate tooth bending stresses and stress distributions obtained from the strain measurements taken during tests.

Test Fixture and Instrumentation

The slow roll test fixture for TRP proof-of-concept gearbox tests was designed and built at Boeing Mesa in 2000. Shown in Figure 2, the fixture used weights pulleys and cables to provide specified input loading at the two pinions and output resistance at the upper face gear. The test gearbox was installed with the output shaft horizontal and the input shafts nearly horizontal off to each side, to provide good orientation of the pulleys which were attached to the shaft ends.

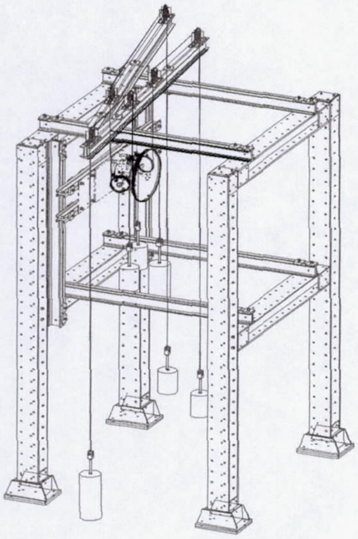

Figure 2. POC Gearbox Test Rig

Provisions for aligning the pulleys and cables with the test gearbox were included. For the test fixture design, weight values and pulley radii were selected to yield the required input torque values at the pinions and output resistance torque at the upper face gear, as specified in the POC gearbox design criteria. The pulleys were counterbalanced, with equal weight suspended off of either side of them. This assured that the input and output shafts were loaded in pure torsion, which was important to obtain representative testing of the design concept. An optical encoder was installed on the end of one pinion shaft to record angular position of the gears during tests. This allowed strain vs. position data to be plotted for all gears after each slow roll test run. Strain measurement equipment provided readout of gear tooth strains for the slow roll tests. Related instrumentation used in the test set-up included load cells to measure input/output torque, signal conditioning amplifiers, power supplies, multi-channel recording equipment, multiplex cards and analog/digital converter boards.

Tooth Contact Pattern and Backlash Adjustment

The POC face gears were first installed into the test gearbox using the mounting distances identified on the gear shafts by Derlan during manufacturing contact pattern development. Following pinion and idler installation, the initial tooth backlash and contact patterns (light load used) required some minor adjustments. Figure 3 shows the set-up used for measuring tooth backlash. The gear tooth pattern trends observed during the POC gear installation were similar to installation trends observed for the face gears used in the durability tests at NASA Glenn [1].

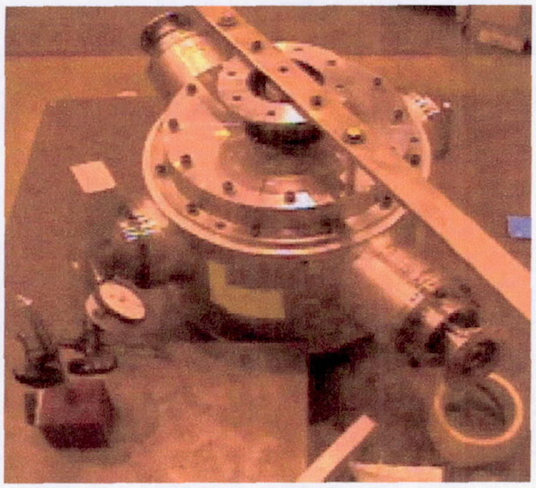

Figure 3. POC Gear Tooth Backlash Measurement

Full load contact pattern checks were conducted next. During full load pattern checks, the pinions and idlers were rotated 150 degrees back and forth a number of times, to set patterns at each of the eight face gear to pinion and idler meshes. Figure 4 shows contact patterns on a section of the upper face gear teeth. Inspections of the 100% load patterns suggested that no further changes were required prior to tests.

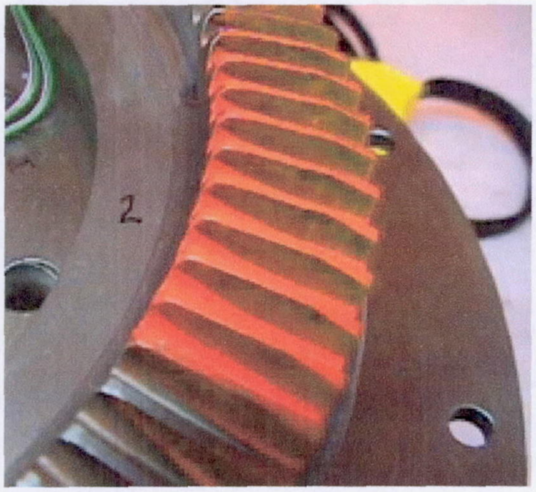

Figure 4. Full Load Tooth Contact Patterns Upper Face Gear Teeth

Split Torque Test Procedure

TRP tests of the proof-of-concept (POC) gearbox included slow-roll split torque tests performed for both single drive and twin drive input conditions. During single drive tests, one pinion and one idler were installed in mesh with the face gears. During twin drive tests, two pinions and two idlers were installed. All tests were conducted at the 100% test load level of 200 N-m (1767 in-lbs), applied at each input pinion. The pinions and idlers were slowly rolled a total of four rotations during each test run. This rolled each face gear about one full turn, so that all teeth of the meshing gears contacted each other. During each run, strain vs. position data was recorded for all strain gages on the instrumented teeth in mesh. Measurements were taken at about each .17° of face gear roll angle, and about each .70° of pinion and idler roll angle. Three identical runs were performed at each test condition, and strain vs. position curves were plotted after each set above to verify repeatability of data. The curves were also used to determine backlash changes needed to help equalize strain levels. All backlash changes were recorded, along with loading conditions and corresponding output data. Relative torque distribution percentages for each pinion mesh and between the idlers were not obtained until after a follow-on torque calibration was performed. The torque splitting efficiencies determined from these tests provided a positive indication of the relative advantages of the concentric face gear design concept.

Split Torque Test Results

Initial Testing

Prior to the formal start of the test, the complete POC gearbox assembly with Pinions 1 and 2 and Idlers 3 and 4 was installed and slow roll tested. This was intended as an initial look to verify correct operation of the entire system and to check out the instrumentation and test procedure.

Initial results showed unusually high upper face gear tooth bending strain at the Idler 4 mesh. When Idlers 3 and 4 were swapped, it was Idler 3 that demonstrated high strain. A misalignment of the housing bore at the Idler 4 location was suspected. The housing was carefully inspected and misalignment was noted at both idler bore locations. Material plugs were inserted and the housing idler bores were re-machined.

The re-worked housing was returned to the Structures Test lab and the POC gearbox reassembled. Though still showing higher strain for the mesh with Idler 4, the upper face gear strain results were altered due to the housing modifications resulting in more even strain at the two idler meshes.

In an attempt to equalize the upper face gear strain at the two idler meshes, the backlash for Idler 4 was increased reducing the amount of load recombined through Idler 4 while increasing load transmitted through Idler 3. The conclusion from this exercise was that backlash modification could be used to affect the distribution of load between idlers.

Formal Testing

As described in the previous section, the two pinion-two idler configuration was tested after the housing modifications. A final gearbox setup configuration was arrived at and three slow roll tests – POC Test Runs 140-142 - conducted for that configuration. Table 1 is a summary of all the formal POC slow roll testing.

The test runs were generally conducted in groups of three to insure that consistent runs were being achieved. Comparisons of repeat data show that the data recorded for a given gearbox configuration was very consistent with peak strains varying by less than 3%.

A total of 80 strain gages were applied to the six gear components – two pinions, two idlers and two face gears. The active length of the strain gages was 0.38 mm (0.015 inch). All gages were applied in the root fillet of the gear tooth aligned so as to measure strain due to tooth bending. The same strain gage configuration was used for both the pinions and idlers. For each pinion or idler, two diametrically-opposed teeth had strain gages applied at three locations along the length of the tooth face and on both sides of the tooth. For each face gear, four teeth approximately 90 degrees apart were instrumented with two strain gages along the face on both sides of the tooth. All face gear gages were applied toward the toe (inboard) end of the tooth – the changing profile of the face gear tooth yields a very tight root radius toward the heel (outboard) end of the tooth making gage installation in that region difficult. Figure 5, showing a cross-section through the pinion in mesh with both face gears, is useful for understanding the relative positions of the component strain gages and interpreting the strain plots to follow. Figure 6 is a schematic top view of the lower face gear showing strain gage locations. This figure is also useful in interpreting the face gear strain plots.

Table 1. POC Gearbox Test Identification Table.

Test Description	Test I.D.	Gear Components Included in Test[a]					
		Idler 4	Idler 3	Pin 2	Pin 1	UFG	LFG
2 Pinion, 2 Idler Tests	140						
	141	Yes	Yes	Yes	Yes	Yes	Yes
	142						
1 Pinion, 1 Idler Tests	210						
	211	Yes	No	No	Yes	Yes	Yes
	212						
	300						
	301	No	Yes	Yes	No	Yes	Yes
	302						
Single Idler Torque Calibration (idler operates as a pinion)	400						
	401	Yes	No	No	No	Yes	No
	402						
	500						
	501	No	Yes	No	No	Yes	No
	502						

[a] Pin=pinion, UFG=upper face gear, LFG=lower face gear.

Typical test results are presented below for each of the four types of POC transmission gear components – pinion, idler, upper face gear and lower face gear - using plots of strain versus sample number. As described previously, each data sample corresponds to approximately 0.70 degrees of pinion roll angle. All strain plots are from Run No. 141 of the full-up two pinion-two idler POC test configuration.

Figure 7 shows a typical pinion strain plot for the Pinion 1 tooth drive side gages. The strain peaks correspond to pinion tooth meshes with the upper and lower face gear. The strain gage numbers (SG1, SG2 and SG3) correspond to the numbers in Figure 5 (PIN1, PIN2 and PIN3) and indicate the strain gage location along the face – note that numbers increase going outboard. It is apparent that the pinion loading is biased toward the heel for the upper face gear mesh, as strain at the middle and outboard gages is predominant.

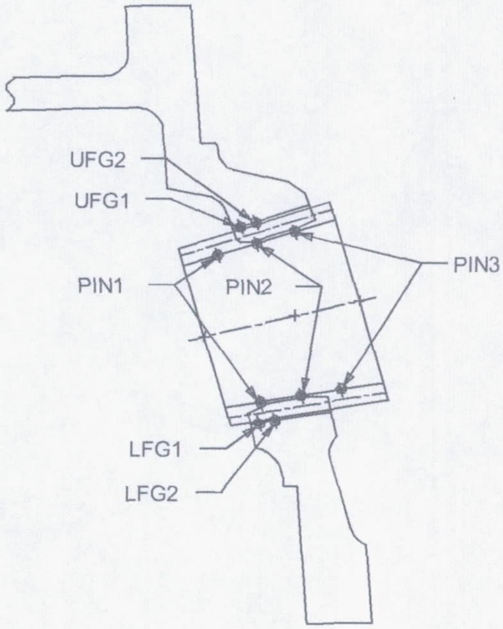

Figure 5. Strain Gage Locations.

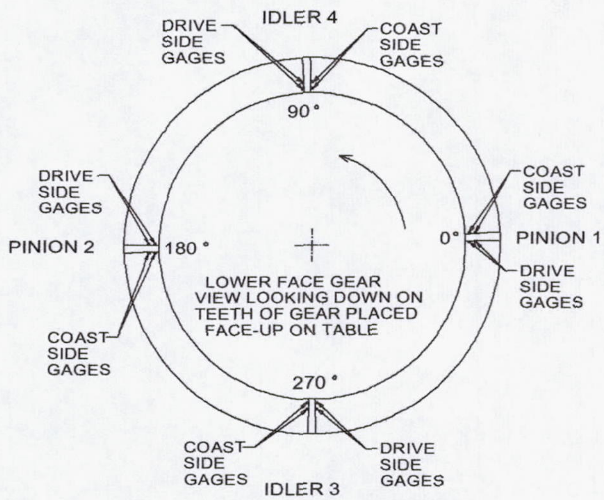

Figure 6. Designation of Pinions and Idlers.

This is further illustrated in Figure 8 which shows strain distributions for both the pinion and upper face gear (UFG) for their common mesh. This plot represents the simultaneous readings from gages along the length of the teeth. Data points are connected by dashed lines as the distribution between gages is not known. For each component, data shown represents the highest overall strain level for the mesh. For the lower face gear mesh, pinion loading is biased toward the toe as strain at the inboard and middle gages predominates - see Figures 7 and 9. This is as expected based on the offset of the POC face gears – see Figure 5.

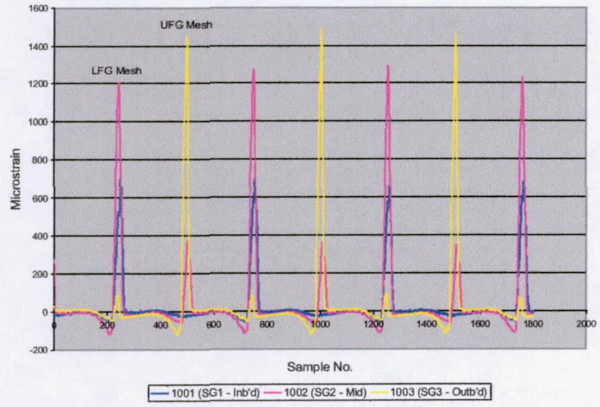

Figure 7. Typical Pinion Strain Output.

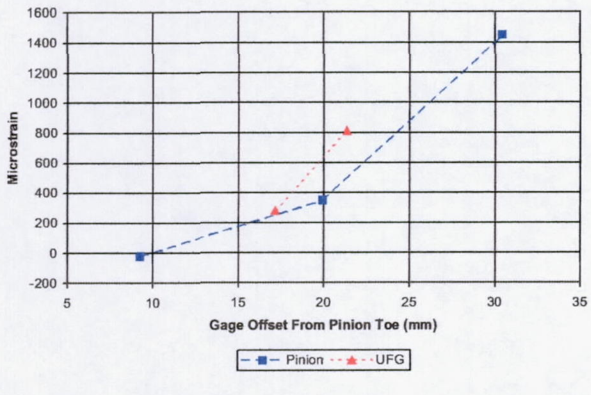

Figure 8. Strain Distribution for the Pinion/UFG Mesh.

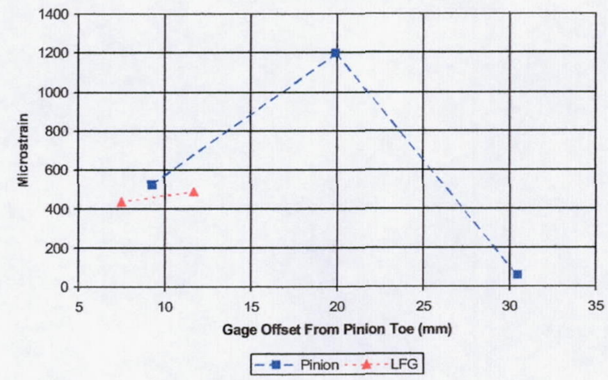

Figure 9. Strain Distribution for the Pinion/LFG Mesh.

A typical idler strain plot is shown in Figure 10 for Idler 3. Note the significant difference as compared to the pinion strain plot in Figure 7. The pinion is the driving member at both face gear meshes splitting the input torque between the upper and lower face gears. The purpose of the idlers is to transmit torque from the lower face gear to the upper face gear, therefore, the idler is the driven member at the lower face gear (LFG) mesh, and the driving member at the upper face gear (UFG) mesh. The same idler gages that are in tension while driving the upper face gear are in compression when the tooth they are on is being driven by the lower face gear.

Also notable in Figure 10 is the fact that while predominantly loading the central portion of the idler tooth, the lower face gear mesh produces negligible strain at the toe gage. Figure 11, a strain distribution plot for the idler/LFG mesh, also shows the idler loaded primarily at the middle gage location with very little strain near the toe on either the idler or lower face gear. (In Figure 11, strain gages are on the non-contacting sides of the teeth resulting in negative bending strains.) This contrasts with the pinion result in Figure 7. Based on the position of the lower face gear relative to the idler (Figure 5), and results shown in Figures 10 and 11, it appears that load remains biased toward the heel of the lower face gear throughout its mesh with the idler.

A strain plot for the upper face gear is shown in Figure 12 for tooth drive side gages. The inboard and mid strain gages appear to be loaded fairly evenly at the idler meshes. For the pinion meshes, the mid gage is loaded significantly more than the inboard gage indicating tooth loading is farther outboard at the pinion meshes relative to the idler meshes.

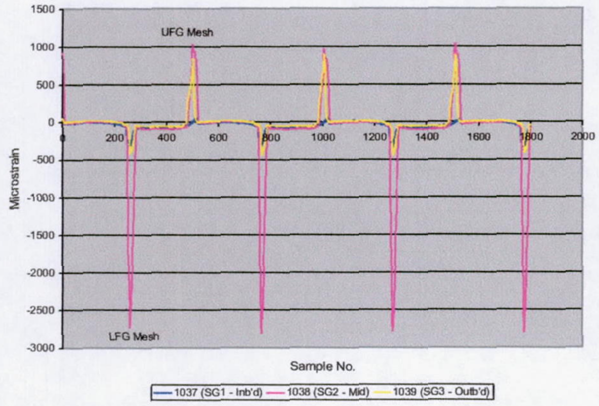

Figure 10. Typical Idler Strain Output.

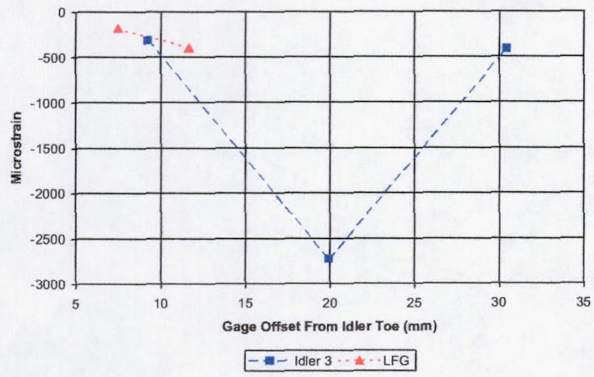

Figure 11. Strain Distribution for the Idler/LFG Mesh.

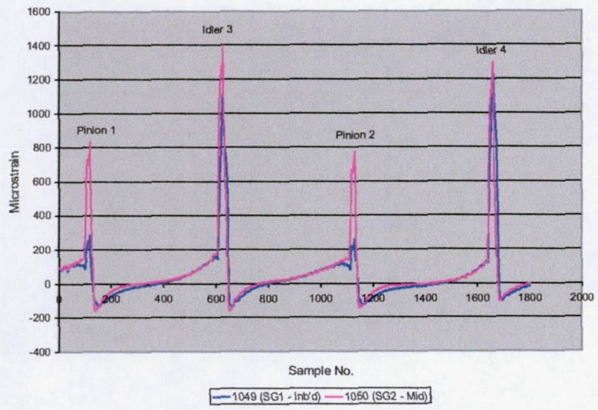

Figure 12. Typical UFG Strain Output.

There is an obvious difference in strain magnitude between the pinion and idler meshes. Strain magnitude is a function of tooth load position relative to the gage location as well as a function of load magnitude. It is believed that the differences in the UFG strains at the different meshes (pinions versus idlers) are more a function of tooth load proximity to the UFG gages. This conclusion is supported by Figure 13, a strain distribution plot for the idler/UFG mesh. Figure 13 shows the idler loaded primarily at its middle gage location which corresponds to the UFG strain gage locations in the toe region (Figure 5). Based on the results shown in Figures 12 and 13, the idlers appear to load the UFG teeth near the toe end; pinions appear to load the UFG farther outboard.

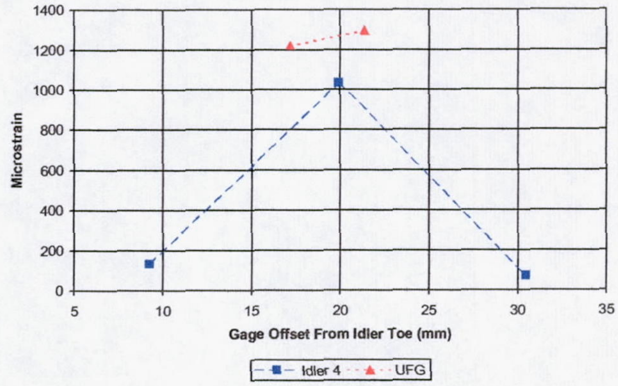

Figure 13. Strain Distribution for the Idler/UFG Mesh.

The lower face gear acts as a type of idler in that it transmits load from pinion to idler but does not transmit torque. Similar to the idlers, the lower face gear teeth are both driven and driving. At the pinion mesh, the lower face gear is driven; at the idler mesh, the LFG drives the idler, which in turn drives the upper face gear. The plot shown in Figure 14 shows output for typical LFG gages.

The LFG strain at the idler meshes is significantly less than at the pinion meshes. The load on the lower face gear tooth for the idler mesh is biased toward the heel, more so than for the pinion mesh, and so produces lesser strain output for the LFG gages which are in the toe region. This conclusion is supported by the fact that the middle gage strain is much greater than the inboard gage strain when the LFG tooth is in mesh with the idler; see Figure 14. These findings are consistent with those for the idlers.

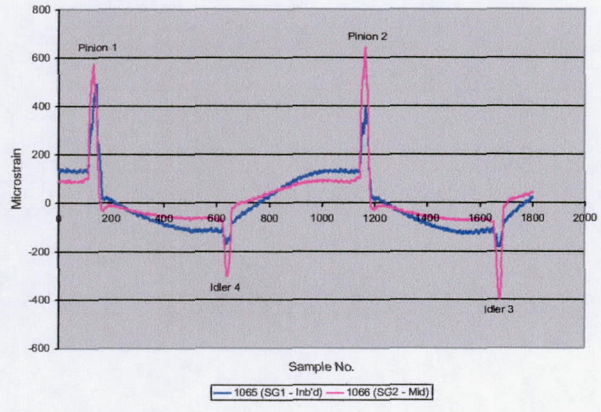

Figure 14. Typical LFG Strain Output.

Torque Split Determination

Background

One of the primary objectives of this POC test program was to determine and optimize, if necessary, the torque split between the upper and lower face gears. The means intended for this purpose were the gear tooth root fillet strain gages. It was assumed that measured bending strain would be proportional to load transmitted at the mesh.

The tapered configuration and offset between the upper and lower face gear (Figure 5) made use of the pinion gages undesirable for the purpose of torque split determination. It was hoped that the upper face gear would yield the torque split information as the geometry of the contact between the upper face gear and both the pinions and idlers was the same. Load transmitted by the idlers to the upper face gear (UFG) is the portion of input torque transmitted from the pinion to the lower face gear and then to the idlers. A cursory review of the results for UFG drive side gages shown in Figure 12 would seem to indicate that much more load was split to the lower face gear than the upper.

Closer scrutiny showed that the distribution of load across the face gear tooth differs between the pinion and idler meshes. For the pinion meshes, strain registered at the middle gage is much greater than that measured at the inboard gage. For the idler meshes, inboard and middle gages measure roughly the same strain. Furthermore, it is difficult to make an accurate load comparison based on two gages near one end of the tooth. As testing progressed, it became apparent that directly measured tooth bending strains would not be sufficient to determine torque split. Torque loading transmitted through a given mesh cannot be directly related to output from a gage or even group of gages as configured for this test.

Torque Calibration

To ascertain the torque split between the pinion upper and lower face gear meshes, it was decided to conduct a "torque calibration." The torque calibration is used to develop a relationship between pinion torque and gear tooth strain for the purpose of torque split determination. There are many factors that make this difficult to do analytically – e.g. high contact ratio, asymmetric face gear configuration and offset contact between the upper and lower face gear relative to the pinion. As such, these relationships were developed experimentally using the POC gearbox test stand.

For the torque calibration, only the upper face gear and a single modified idler were used. The lower face gear was in its normal position, but with no standard idler installed, it was "free-wheeling" offering no torsional resistance. The idler was temporarily modified by the attachment of a hub and pulley such that torque could be applied to the idler as if it were a pinion. A modified idler was used in lieu of a pinion as the idler design includes bearing support both inboard and outboard of the face gear meshes which was necessary for this calibration step for which the lower face gear was not loaded.

Torque was applied to the modified idler and reacted through the upper face gear to the output coupling. While under torque, the assembly was slowly rolled through nearly one revolution of the upper face gear (approximately four revolutions of the modified idler). This procedure was conducted at the modified idler torque level of 100 N-m (883.5 in-lbs) – the 50% torque level. The modified idler was rotated counterclockwise; output for all modified idler and upper face gear strain gages was recorded for this test. This procedure was conducted separately for each of the two idlers. For all the torque calibration tests, the idlers were installed in their own bores with the same setup used for the formal POC test program.

Torque Split Method

The method for torque split determination assumes that the sensitivities between transmitted tangential load and strain developed for the modified idler gages during the initial torque calibration step, are the same as the sensitivities for the idler gages when the idler is operating in its standard fashion. It is known that assumption is not true. Comparing strain output from the Idler 3 torque calibration (Figure 15) and the one pinion-one idler test (Figure 16) shows that the strain distribution changes significantly. Nevertheless, this revised method is used out of necessity to estimate the torque split results.

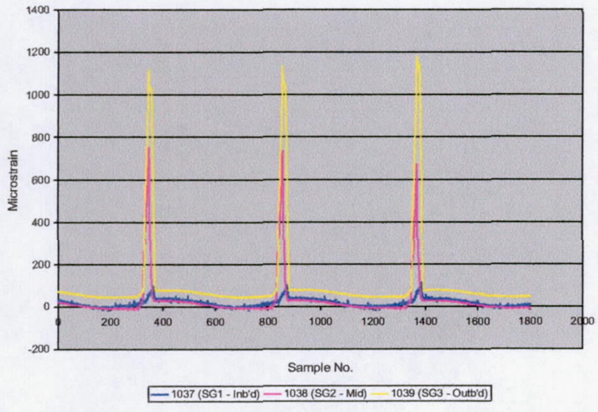

Figure 15. Strain Output for Idler 3, Run No. 501.

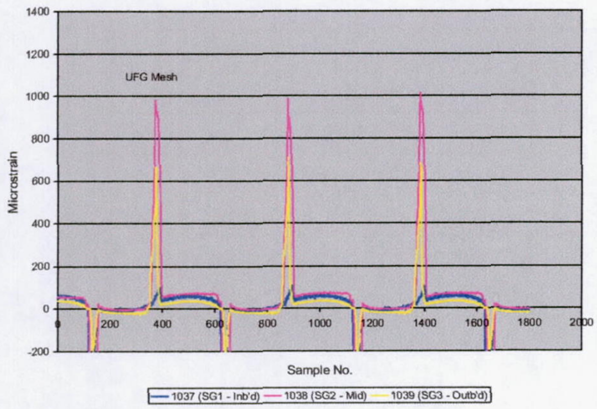

Figure 16. Strain Output for Idler 3, Run No. 301.

Derivation of Revised Torque Split Method

For the one pinion-one idler configuration:

$$T_P = W_{TPU}*R_U + W_{TPL}*R_L \quad (1)$$

(Torque split at pinion between upper and lower meshes)

where

R_U = pinion/idler pitch radius at center of UFG tooth
R_L = pinion/idler pitch radius at center of LFG tooth
W_{TPU} = tangential load at pinion/UFG mesh
W_{TPL} = tangential load at pinion/LFG mesh
T_P = pinion input torque

$$W_{TIU}*R_U = W_{TIL}*R_L \quad (2)$$

(Idlers transmit no torque.)

where

W_{TIU} = tangential load at idler/UFG mesh
W_{TIL} = tangential load at idler/LFG mesh

$$W_{TPL} = W_{TIL} \quad (3)$$

(The lower face gear transmits no torque.)

$$W_{TIU}*R_U = W_{TPL}*R_L \quad (4)$$

$$T_P = W_{TPU}*R_U + W_{TIU}*R_U \quad (5)$$

$$W_{TPU} = (T_P - W_{TIU}*R_U)/R_U = T_P/R_U - W_{TIU} \quad (6)$$

W_{TIU} is determined from the torque calibration. Torque is applied to the idler which has been modified by addition of a hub. In the torque cal, W_{TIU} is known and assumed equal to $W_{TIU} = T/R_U$ where T is the torque applied to the idler.

During this calibration, the bending gage strains were recorded. Ratios of $K_{IU} = W_{TIU}$/strain were calculated using the maximum or minimum strain, as appropriate, recorded during the complete roll. During the one pinion-one idler (1 and 1) test, from measured idler strain and using K_{IU}, W_{TIU} was determined. W_{TPU} was then calculated.

From the equations above, the fraction of torque transmitted at the upper pinion mesh is

$$\text{Upper Torque Split} = W_{TPU}/(W_{TPU} + W_{TIU}) \quad (7)$$

$$\text{Lower Torque Split} = 1 - \text{Upper Torque Split} \quad (8)$$

Ratios of $K_{PU} = W_{TPU}$/strain were calculated from the results of the 1 and 1 test.

This process was carried out for the combinations of the Modified Idler 4 torque calibration (Run No.401) and Pinion 1/Idler 4 1 and 1 test (Run No.211), as well as, the Modified Idler 3 torque calibration (Run No.501) and Pinion 2/Idler 3 1 and 1 test (Run No.301). K values were then available for both pinions and both idlers. These K values were used with the two pinion-two idler (2 and 2) test results in terms of strain to arrive at the overall upper/lower torque split at the pinions and the idler load-sharing split.

Because of the known inaccuracy in this method for determining torque split, K values for each gage of a given component were calculated and used to determine several values of W (tangential load) which were then averaged. W values based on K values for gages with known low output for a given mesh were not used to arrive at the average for W.

In order to provide an even broader sample from which to calculate average values for W, K values were developed for upper face gear gages as well. The only difference in the process described above for the pinion and idler gages was that the maximum or minimum strain value (as appropriate) was not calculated for the entire slow roll run, but only from the data samples encompassing the UFG mesh of interest. For example, to determine the tangential load for an idler/UFG mesh for the 1 and 1 test using a UFG drive side gage, only the maximum value for the portion of the UFG strain trace that encompassed the spike representing the idler/UFG mesh would be obtained. This maximum would then be multiplied by the appropriate K value to arrive at the tangential load, W.

Torque Split Results

Using the process described above, the results of the torque split determination are shown in Table 2. The overall trend is that the torque splits fairly evenly. The idler load-sharing results indicate that Idler 3 transmitted significantly more LFG load than Idler 4. This is likely due to the fact that the Idler 4 backlash was intentionally increased as previously described.

Table 2. Estimated Torque Split Results from 2 and 2 Test.

Description	Pinion/Idler Average	UFG Average	Combined Average
Upper torque split	49.2%	46.6%	48.0%
Lower torque split	50.8%	53.4%	52.0%
Idler 3 split	58.8%	55.1%	57.0%
Idler 4 split	41.2%	44.9%	43.0%

Strength Summary

Maximum tensile stresses based on the discrete gages along the face width, calculated from strains measured during POC Run No. 141 - the full-up two pinion-two idler test - are shown, by gear, in the row labeled "maximum measured" in Table 3. The input pinion torques for this test run were 200 N-m (1767 in-lbs), equivalent to a 100% maximum continuous torque level.

The row labeled "calculated spur pinion" presents stresses calculated using the AGMA tooth root bending stress formula [7] for a spur gear. Although the actual POC pinion is a Boeing-Proprietary tapered design, not straight, a standard section exists for which spur gear geometry data is defined on the drawing. It is data for this standard section that is used in the formula for tooth bending stress for a standard spur gear. For all calculated stresses, an even torque split is assumed.

Table 3. Tooth Bending Fatigue Strength (POC Run 141, 2 on 2).

Description	Gear Tooth Bending Stress in MPa (KSI)					
	Pin 1	Idler 4	Pin 2	Idler 3	UFG	LFG
Maximum Measured (100% MCP)	299 (43.3)	332 (48.1)	241 (34.9)	476 (69.0)	305 (44.3)	176 (25.5)
Calculated Spur Pinion (100% MCP)[a]	224 (32.5)	224 (32.5)	224 (32.5)	224 (32.5)	N/A	N/A
Allowable Bending Stress (Carburized and Hardened)[b]	517 (75.0)	362 (52.5)	517 (75.0)	362 (52.5)	517 (75.0)	362 (52.5)

[a] An even torque split is assumed.

[b] AGMA spur and helical gear allowable tooth bending stress for 10^7 cycles life at 99% reliability

The allowable bending stresses shown in Table 3 for carburized and hardened steel gears, the production heat treat condition, are taken from [7]. A reduction factor of 0.70 is applicable for the idlers and lower face gear that see reversed bending [7]. All measured stresses are within the allowable limit except for the Idler 3 stress.

The calculated spur pinion stresses are below the actual maximum measured stresses for all the pinions and idlers. The measured stresses for the pinions are fairly close to what was predicted by the spur gear calculation. The lower face gear stress is lower than either the pinion or idler stresses. The upper face gear stress, however, is higher than the pinion stresses but lower than the idler stresses. The face gear teeth are similar to those of a rack with a relatively wide base at the root. Because of this geometry, it was anticipated that the pinion/idler tooth bending stresses would be more critical than those of the face gears. This was not completely borne out by the test results. The face gear gages are near the toe of the face gear teeth, which is the region where the tooth root is thinnest and higher stresses might be expected (notwithstanding the fact that the root fillet radius increases in going from the heel to the toe). Additionally, strain data indicated that loading at the Idler 4/UFG mesh was biased toward the toe of the UFG tooth which leads to more localized, higher stresses at the UFG gage locations.

An unexpected result in Table 3 is the high stresses recorded for the idlers, particularly Idler 3. Of all the gears, and specifically the idlers, Idler 3 experiences the highest tensile bending strain. [Note that even though high compressive strains are shown in Figure 10 for the LFG mesh, gages on the opposite side of the idler tooth measured tensile strains of over 2300 $\mu\varepsilon$. These maximum positive strains are of lower magnitude than the minimum compressive strain of approximately -2800 $\mu\varepsilon$ (Figure 10) due to the radial component of applied tooth load which acts inward adding to compressive bending strain and subtracting from tensile bending strain.] It was anticipated that the idler stresses would be comparable with those of the pinion. As shown above, the maximum measured idler stress is almost 60% greater than the maximum measured pinion stress. The highest idler tooth bending stress was measured by an Idler 3 middle strain gage for the I3/LFG mesh. As stated previously, for the idler/lower face gear mesh, the load appears to be concentrated near the middle of the idler tooth and correspondingly near the heel end of the lower face gear tooth. The load is apparently concentrated to an excessive degree producing high loading on a very local region of the idler tooth leading to root bending stress much higher than was expected. The uneven torque split also contributes to the difference in peak strains between Idlers 3 and 4.

For a production face gear main transmission design, three idlers would be required instead of the two used for the full-up configuration of the POC gearbox. The intent is to reduce the bending stresses by using more idlers to transmit the load. Even if they were the same as the pinion stresses, it would be necessary to reduce the idler teeth bending stresses to a lower level than that for the pinions because the idlers experience reversed bending and are subject to a lower fatigue strength allowable.

Finite Element Analysis

Analysis Approach

To further investigate the high idler tooth bending stresses, a finite element analysis was conducted. Although the pinion was modeled, it was used to investigate the end loading that is apparent at the idler-lower face gear mesh. The pinion and idler have the same geometry and this model does not take into account the mounting stiffness of the pinion, therefore, it is more accurately used to represent the more rigidly-mounted idler.

The pinion finite element model was created from a solid Unigraphics (UG) model of the POC pinion, portions of both the upper and lower face gears that mesh with the pinion, and contact solids at the pinion/LFG and pinion/UFG meshes. The contact solids were created by slightly oversizing the pinion and face gears. The gears were then "rolled" through the mesh in increments within Unigraphics. At each increment, interferences between the pinion and face gear teeth surfaces were used to create "contact solids." [Deflection in the gear teeth is not accounted for in this method. Additionally, use of oversized gears, required to create the contact solids, may give an inaccurate indication of the contact ratio. The effect of these inaccuracies requires further investigation.] Unique sets of upper and lower contact solids were created for sequential increments of roll through the mesh. Note that both upper and lower meshes of the oversize gears include at least three teeth in contact simultaneously as demonstrated by the contact solids formed. Figure 17 shows an outline of the POC pinion and the contact solids formed by its mesh with the LFG at a certain roll position. As stated previously, POC test results indicated the highest strains for the LFG mesh (as opposed to the UFG mesh).

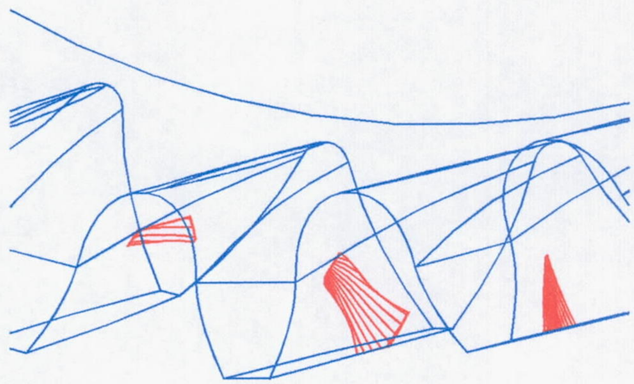

Figure 17. Input Pinion Showing Contact Solids on Three Adjacent Teeth for the LFG Mesh (View Looking Outboard).

Various sets of contact solids were reviewed to determine a set (pinion roll position) that yielded the worst case contact solid configuration. The configuration shown in Figure 17 was chosen as it included only three teeth in contact (rather than four teeth in contact that existed for some roll positions), middle tooth loading appeared maximized in comparison to the preceding and succeeding teeth, and the middle tooth load was high up on the tooth which produces higher bending stress than a low contact pattern.

A finite element model (FEM) of a portion of the pinion was created using the UG solid model. For the finite element model, load was applied to only a single tooth – the middle of the three loaded teeth shown in Figure 17. To facilitate this loading, the intersection of the middle tooth contact solid and the pinion was used to define a spline on the face of the pinion tooth. This spline was used to split the face of the tooth. The UG model was imported into the MSC Patran to create the finite element model. The geometry of the middle tooth contact surface was imported along with the model. The model was meshed using tetrahedral elements and refined using surface curvature to yield a finer mesh at the root fillet. Initially, half of one tooth was meshed; this was then mirrored and the tooth mesh duplicated twice to yield a FEM of three of the pinion teeth (Figure 18).

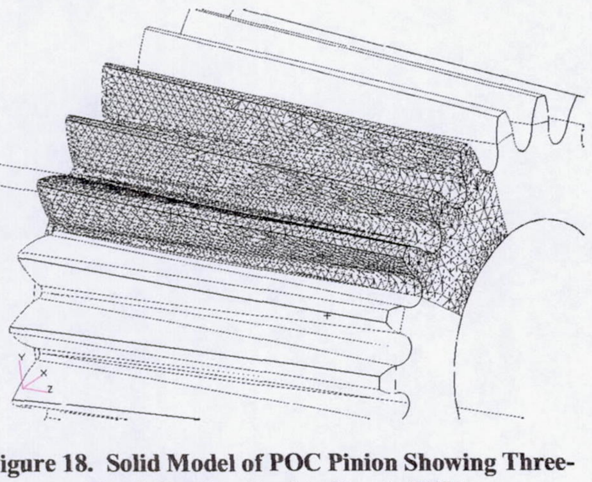

Figure 18. Solid Model of POC Pinion Showing Three-Tooth Portion Used to Create FEM.

The middle tooth was loaded using pressure applied to the contact surface, only. Three loading cases were run: 2a) moderate end loading that varied linearly from a maximum at the outboard end of the contact surface to zero at the inboard end, 3) uniform loading over the entire contact surface, and 4) loading that varied from a maximum at the outboard end of the contact surface to zero at the spanwise center of the contact surface. Cases 2a and 4 were intended to simulate varying degrees of end loading. Pressure loads for Cases 2a and 4 were created using "Fields" within Patran. The variation of pressure across the 21.4 mm (0.843 inch) length contact surface is shown in Figure 19.

The model was restrained in translation at all nodes in the plane at either circumferential end of the model. The magnitude of the pressure load applied in each case was chosen such that the total load applied to the contact surface would be roughly equal (see Figure 19). The FEM built in Patran was submitted to MSC Nastran for processing. A statement was added to the Nastran bulk data file prior to processing. This statement was used such that torque about the pinion axis could be captured. This information was essential to insuring the load cases were comparable and of the same magnitude as test loads.

Finite Element Analysis Results

In addition to making an adjustment for applied torque load, it was necessary to adjust the FEA stress results to account for the fact that the middle tooth loading yields only a fraction of the total torque transmitted at the pinion/LFG mesh (three teeth in contact - Figure 17). Factors affecting the torque contribution of each tooth contact include area of the contact, magnitude of the pressure load, radial distance of the contact area from the pinion axis, and angle between a vector normal to the contact surface and a vector in the tangential direction. As a simplifying assumption, it was assumed that the pressure on each contact surface was uniform and of the same magnitude. The properties of each of the contact areas are shown in Table 4. The geometric properties were determined using UG.

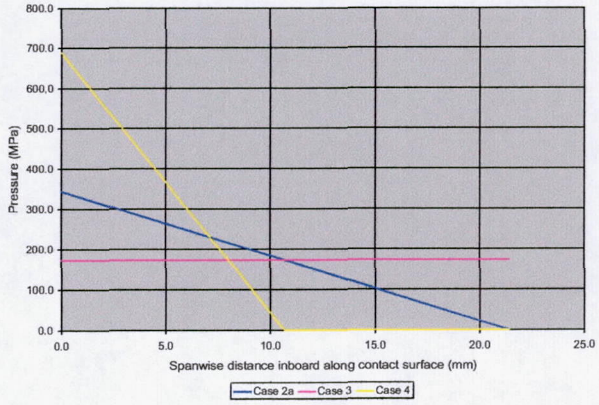

Figure 19. Pressure Distribution for the Various Loading Cases.

Table 4. Tooth Contact Area Properties.

Tooth Number	Area A mm^2 (in^2)	Radius R mm (in)	Angle α (degs)	$A*\cos(\alpha)*R$ mm^3 (in^3)
1	6.0 (0.0093)	24.5 (0.9654)	17.19	140.4 (0.00858)
2	26.5 (0.0410)	24.6 (0.9696)	25.59	588.0 (0.03585)
3	8.0 (0.0124)	23.0 (0.9055)	24.54	167.4 (0.01021)
Total	40.5 (0.0627)			895.8 (0.05465)

The 100% torque level for the POC test was 200 N-m (1767 in-lbs). Assuming a 50-50 torque split at the upper and lower face gear meshes, the torque transmitted at the LFG mesh was 100 N-m (883.5 in-lbs). Based on this torque, Table 4, and the simplifying assumption, the following equations can be written:

$$\Sigma(p*A*\cos\alpha*R) = T \tag{9}$$

$$p*\Sigma(A*\cos\alpha*R) = T \quad (10)$$

$$p = T/\Sigma(A*\cos\alpha*R) \quad (11)$$

where

p = uniform pressure at all tooth contact surfaces
T = torque transmitted at the mesh
= 100 N-m (883.5 in-lbs)
$\Sigma(A*\cos\alpha*R) = 895.8\ mm^3\ (0.05465\ in^3)$ from Table 4

$$p = 1000*100/895.8 = 111.6 \text{ MPa } (16{,}170 \text{ PSI}) \quad (12)$$

For each individual tooth, the following equation can be written:

$$T = p*A*\cos\alpha*R \quad (13)$$

For the middle tooth, $A \cos\alpha\ R = 588.0\ mm^3\ (0.03585\ in^3)$, therefore

$$T = 111.6*588.0/1000 = 65.5 \text{ N-m } (579.6 \text{ in-lbs}) \quad (14)$$

As stated previously, the torque applied for each Patran load case was obtained from the Nastran results. Using the torque actually applied and the torque that should have been applied to match the POC test, scale factors are developed for each load case in Table 5.

Table 5. FEM Load Case Scale Factors.

Load Case	Applied Torque N-m (in-lbs)	Required Torque N-m (in-lbs)	Scale Factor
2a	108.4 (959)	65.5 (579.6)	0.604
3	99.0 (876)	65.5 (579.6)	0.662
4	108.8 (963)	65.5 (579.6)	0.602

Within Patran, the scale factor was applied to the FEM results. Figures 20-22 show the scaled results for all three load cases. All three plots use the same color scale so that they can be compared directly. For Figures 20-22, the view is looking inward at the three modeled teeth (refer to Figure 18) with the toe of the pinion to the right and the heel to the left. The negative (compressive) stress region corresponds roughly to the contact surface where the pressure load is applied. Note that this contact surface is not parallel to the tooth root; it extends from the top of the pinion tooth face at the outboard end of the pinion-lower face gear mesh to the pitch line region at the inboard end of the pinion-LFG mesh. The band of high tensile stress is at the root of the loaded (middle) tooth and is a result of tooth bending. In going from uniform loading to moderate end loading to extreme end loading (Cases 3 to 2a to 4), the stress distribution gradually shifts toward the outboard end of the pinion-LFG mesh and the stress gradient increases as the applied load becomes more tightly focused. As expected, the peak stress also rises.

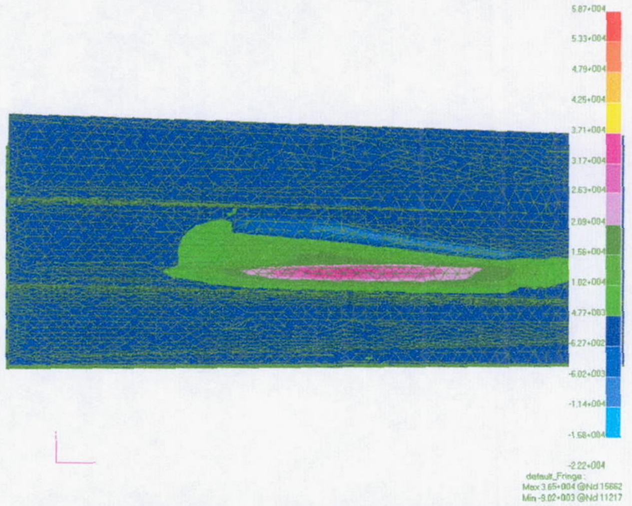

Figure 20. Maximum Principal Stress (PSI) Plot for Load Case 3 – Uniform LFG Loading.

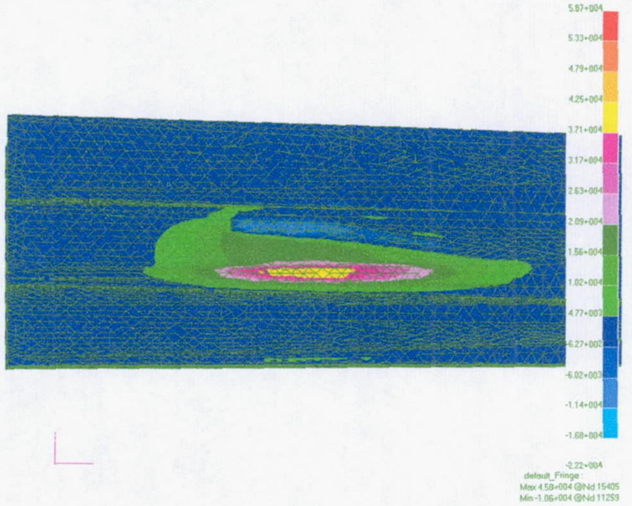

Figure 21. Maximum Principal Stress (PSI) Plot for Load Case 2a – Moderate LFG Outboard End Loading.

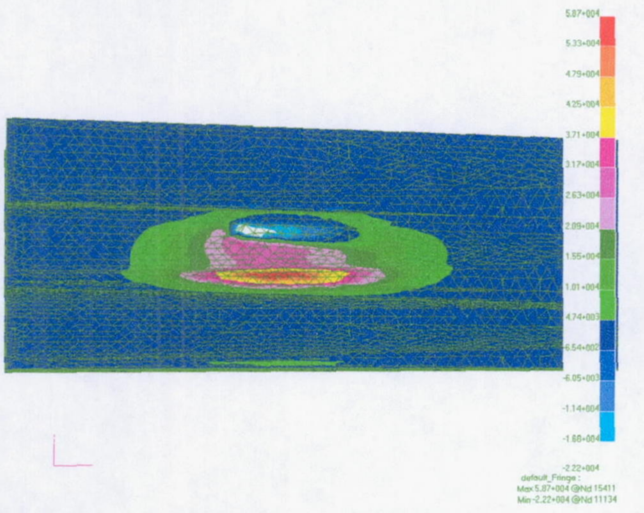

Figure 22. Maximum Principal Stress (PSI) Plot for Load Case 4 – Extreme LFG Outboard End Loading.

POC Test Results

Tooth root bending strains were measured directly during the POC slow roll test, however, this data also required adjustment before being used to compare to the FEM results. As stated above, the finite element analysis assumed a torque split of 50-50. In Table 2, the actual torque split is estimated to be 48% to the upper face gear and 52% to the lower face gear. Because the stresses measured for all pinion/idler meshes with the lower face gear theoretically correspond to 52% of the pinion input torque, all measured stresses for the lower face gear mesh of all pinions and idler were reduced by the factor 50/52 = 0.96. Additionally, data from Table 2 also indicates that load transferred from the lower face gear back up to the upper face gear was not evenly split between the idlers. Idler 3 transmitted 57% of the lower face gear load, while Idler 4 transferred 43%. Idler 3 strains were multiplied by the factor 50/57 = 0.88, and Idler 4 strains were multiplied by the factor 50/43 = 1.16.

Only pinion or idler gage sets that experience tension due to bending at the lower face gear mesh are of interest in this study. For each component, that yields two sets of three gages that are of interest. For each component, maximum measured stresses were compared for the two sets of gages. The set with the highest peak stress was chosen for comparison to the finite element analysis (FEA) results.

For all pinion/idler components, the central gage – Gage 2 – sees the highest strain of the three gages along the face width. Data for all three gages corresponding to the precise roll angle at which Gage 2 reaches a maximum for the entire test run is used for the FEA comparison.

Comparison of FEM Results, POC Test Results and AGMA-Predicted Results

The scaled FEM results are compared to adjusted results from the POC test in Figure 23. There are four meshes that match the geometry of the mesh analyzed – pinion/lower face gear. The four meshes are the Pinion 1, Pinion 2, Idler 3 and Idler 4 meshes with the LFG. Data from Run No. 141 of the POC test, the full-up two pinion-two idler configuration (Table 1), is used for this comparison. The calculated bending stress level, based on the AGMA formula [7] for spur gears (Table 3) is also included in Figure 23.

The solid lines in Figure 23 represent the FEM results and correspond to the three different pressure distributions applied to the pinion tooth contact area (Figure 19).

The discrete points correspond to data measured in the POC test and are defined in the chart key. The offset distances are based on the gage locations adjusted to account for differences in component shim thicknesses. Note that the pinion stresses fall within the range of the uniform (3) and moderate end loading (2a) cases. The idler stresses correlate fairly well with the extreme end loading case (4). The results of the finite element analysis support the conclusion that non-uniform end loading exists at the idler-lower face gear mesh.

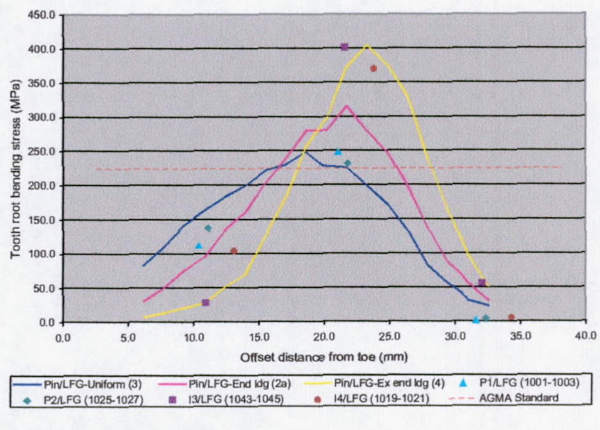

Figure 23. Comparison of POC Run No. 141 Measured Stresses to FEM Predicted Stresses.

Comparing the finite element results to the measured POC data and AGMA-calculated stresses shows very good agreement between measured and FEA data, however, the stress distribution along the length of the tooth is not uniform, as is assumed by the AGMA standard stress formula. Within the AGMA approach, an uneven distribution can be accounted for using a load distribution factor, K_m, however, standard Apache practice has been to set K_m=1.0 (assumes roughly uniform). The non-uniform stress distribution along the face gear pinion or idler tooth is due to at least three factors. (1) The face gear for this POC set does not mesh with the full length of the pinions/idlers. The face width of the lower face gear teeth is 21.3 mm (0.838 inch) compared to 41.5 mm (1.635 inches) for the pinion/idler teeth. (2) Lines of contact are not parallel to the gear axis which produces a variation in stress at the root. (3) As stated above, there appears to be significant end loading between the face gears and idlers.

Based on Figure 23, use of the AGMA standard spur gear formula without an adjustment factor appears to be a reasonable estimate of maximum bending stress in the pinion as predicted by FEA and measured in the POC test. Assuming the end loading problem with the idlers can be addressed and eliminated, AGMA spur gear stress levels should be achievable.

DARPA 2828 HP DEMONSTRATOR TRANSMISSION TEST PROGRAM

The demonstrator transmission designed as part of the DARPA 2828 HP transmission program is a full size, full speed split torque main transmission utilizing the same basic concentric face gear concept as the TRP gearbox, plus a planetary output stage. The 2109 kW (2828 HP) transmission was designed to fit the AH-64A Apache transmission test stand. This second program involved fabrication of the transmission components and full load split torque concept demonstration, which verified face gear operational capability at full scale and load. Similar to the POC gearbox, the cantilever arrangement allowed the pinion to float in mesh between the two face gears, and find the position in equilibrium where the forces were equal.

ROTORCRAFT DRIVE SYSTEM 21 PROGRAM

Boeing was recently awarded a contract to participate in the Rotorcraft Drive System 21 Program (RDS-21), which is a new program administered by the U.S. Army Applied Aviation Technology Directorate (AATD). The RDS-21 Program was initiated to develop and demonstrate critical drive system technologies to overcome weight, reliability and cost barriers faced by current technology as they apply to current DoD rotorcraft fleet, future unmanned air vehicles and rotorcraft platforms. The Boeing Rotorcraft operations in Mesa, Arizona was awarded four advanced technology demonstration projects under the RDS-21 program. Two of the projects are structured to advance the Boeing face gear technology providing operational, analytical and manufacturing validation. The remaining two projects integrate smart materials and non-destructive sensors in composite shafting and structural housings. These composite projects will be performed in the Boeing Rotorcraft Philadelphia facility. The face gear technology projects include design, analysis tool development, and testing of face gears for surface fatigue, single mesh bending fatigue, single tooth bending fatigue, dynamic bending fatigue and system endurance testing of a face gear split torque design. The face gear projects provide validated data for future implementation of the technology in manned and unmanned aircraft applications. One of these applications involves the Boeing Mesa Affordable Apache Drive System Program, which will include development of a higher horsepower face gear transmission as part of future AH-64D drive system upgrades. Figure 24 shows a cross section comparison of a conventional transmission and a face gear split torque transmission rated at the same horsepower.

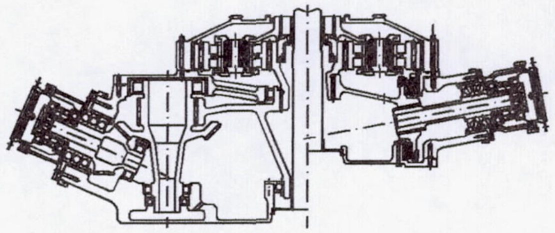

Figure 24. Left Side: Conventional 3 Stage; Right Side: 2 Stage Split Torque Face Gear

CONCLUSIONS AND RECOMMENDATIONS

1. The Boeing Company and the Defense Advanced Research Projects Agency (DARPA) have pursued jointly-funded face gear technology development in the Technology Reinvestment Program (TRP) and the 2828 Horsepower Demonstrator Transmission Program. Engineering and test facilities support was provided early in the TRP during successful face gear durability tests run at the NASA John H. Glenn Research Center and the U.S. Army Research Laboratory.

A scaled-power TRP split torque gearbox using face gears was designed and manufactured for use in split torque concept tests. Test investigations included face gear tooth contact pattern development followed by successful slow-roll tests performed for both single drive and twin drive input conditions.

2. The primary objective of this test program was to determine the torque split for a tapered, off-90-degree face gear transmission. All indications point to a nearly even torque split at the input pinions - 48% to the upper face gear mesh and 52% to the lower face gear mesh. The load sharing between idlers was not equal - 57% for Idler 3 and 43% for Idler 4 - but the testing demonstrated that the load

sharing between idlers could be adjusted by changing the relative amounts of backlash. It was only after all formal testing was completed that the torque calibration was conducted and pinion torque splits and idler load sharing could be determined. It is believed that the idler backlash can be adjusted to arrive at approximately equal load sharing.

3. Strain was not a reliable indicator of load transmitted at a given mesh as the distribution of tooth bending strain varied between similar components (pinions and idlers) and different meshes on the same component, i.e., the upper face gear. A torque calibration was performed in an effort to develop a relationship between load transmitted at a given mesh and measured strain. It was discovered that even these load/strain relationships changed depending on the assembly configuration and the magnitude of load. The torque calibration results were averaged over several strain gages to try and offset error, however, there may be some inaccuracy associated with these test results.

4. Maximum measured bending strain levels were higher than expected for all components. This was particularly true for the idlers. It is believed that some of the highest strains are contributed to by end loading which is apparent based on the tooth strain patterns. This type of loading was particularly evident at the heel end of the lower face gear in its mesh with the idlers. The situation of apparent end loading emphasizes the sensitivity of the face gear assembly to proper alignment. Initial testing indicated possible misalignment in the Idler 4 bore resulting in inspection and re-machining of both idler bores prior to conducting the formal test. The re-boring operation significantly affected the tooth strain distributions.

5. A reliable approach for the determination of torque and idler load split for future face gear assembly testing must be defined for development purposes. Ideally, this would be a simple, robust method that would yield rapid feedback in response to configuration changes.

6. If possible, gages should be added along the entire length of the instrumented gear teeth for the pinions, idlers and face gears. This will give a better picture of the strain distribution and is more likely to capture the maximum strain. High strains were measured during this test, and it is likely that the highest strains were not captured. Also, to aid in tooth contact ratio determination, one sector of four teeth in a row should be instrumented on each gear.

7. A finite element analysis of the pinion/idler tooth was conducted. Results correlated well with data measured during the POC test and verified the end loading between the idlers and lower face gear. Additional finite element analysis (FEA) tools should be developed to better understand tooth loading and stress distribution. Contact FEA should be attempted to obviate the need for many of the assumptions used in the FEA herein.

8. Means of eliminating the apparent end loading must be pursued. Changes in tooth geometry, particularly crowning/end relief, should be considered. Care should be given to the manufacture and inspection of the transmission housing to insure accurate bores. More uniform tooth load distribution will result in a lower maximum bending strains and surface contact stresses.

9. The Full Scale 2828 Horsepower Demonstrator Transmission Program involved operation of the as-designed 2109 kW (2828 HP) face gear transmission to demonstrate torque splitting and applicability of the design at high loads.

10. New methods have been developed for face gear grinding, grinding wheel dressing and coordinate measurement. Under this DARPA-related work, a face gear grinding machine was custom-built and operated by Derlan Aerospace Canada and has demonstrated the capabilities of finishing face gears to required case hardness, profile accuracy and surface finish for aerospace applications.

11. Face gear technology offers great promise for application in helicopter transmissions. The ability of face gears to provide high ratios of gear reduction and achieve self adjusting-torque splitting allows the use of transmissions requiring fewer reduction stages. This yields a better power to weight ratio, reduction in parts relative to multiple-stage designs and reduction in volume. The split torque face gear design offers improved reliability and reduction in operation and support (O & S) costs over existing conventional gearing designs used in large horsepower applications.

ACKNOWLEDGMENTS

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