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RDS–21 Face-Gear Surface Durability Tests

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Abstract

Experimental fatigue tests were performed to determine the surface durability life of a face gear in mesh with a tapered spur involute pinion. Twenty-four sets of gears were tested at three load levels: 7200, 8185, and 9075 lb-in face gear torque, and 2190 to 3280 rpm face gear speed. The gears were carburized and ground, shot-peened and vibro-honed, and made from VIM-VAR Pyrowear 53 steel per AMS 6308. The tests produced 17 gear tooth spalling failures and 7 suspensions. For all the failed sets, spalling occurred on at least one tooth of all the pinions. In some cases, the spalling initiated a crack in the pinion teeth which progressed to tooth fracture. Also, spalling occurred on some face gear teeth. The AGMA endurance allowable stress for a tapered spur involute pinion in mesh with a face gear was determined to be 275 ksi for the material tested. For the application of a tapered spur involute pinion in mesh with a face gear, proper face gear shim controlled the desired gear tooth contact pattern while proper pinion shim was an effective way of adjusting backlash without severely affecting the contact pattern.

Introduction

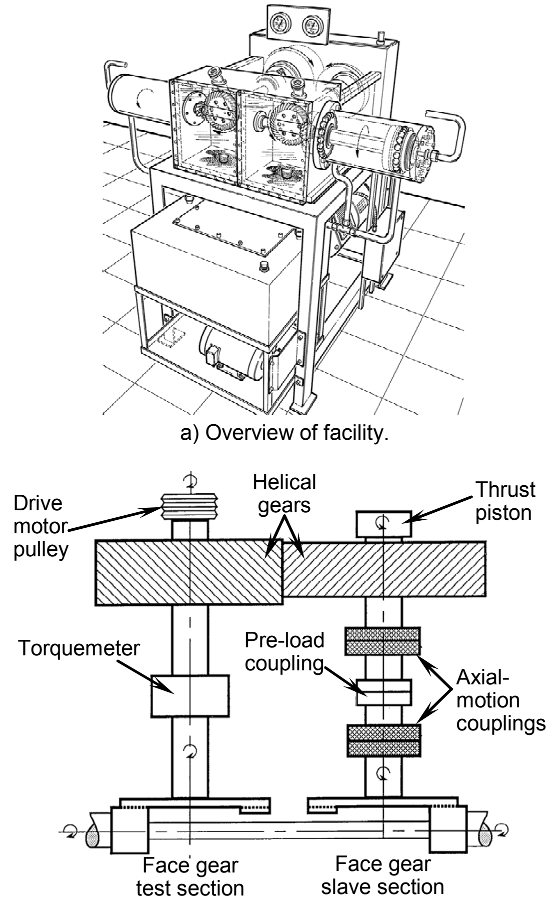
Military requirements and commercial economic survivability continuously drive the needs for reduced weight and increased power density for aircraft drive systems. Modern drive system components operate at high speeds, high loads, and high temperatures, and require state-of-the-art in design and materials. A few years ago, the U.S. Army funded a study named the Advanced Rotorcraft Transmission Program (ART I) to investigate improved concepts to reduce helicopter drive system weight and noise, and increase life (ref. 1). From that, the use of face gears in transmissions emerged as a way to achieve these goals.

Results from the original studies showed that a split-torque, face-gear transmission gave a 40% decrease in weight compared to a conventional design for an advanced attack helicopter application (ref. 2). Face gears, however, had no previous application in high-power systems. Much work had to be done to establish design guidelines. Early analytical work developed face-gear geometry, methods to control the tooth contact, and simulation of meshing and transmission error prediction (ref. 3) as well as tooth generation and methods to define limiting inner and outer radii (ref. 4). Initial experiments demonstrated the feasibility for the use of face gears in helicopter applications but depicted the need for high-accuracy, high-strength, carburized and ground gears (refs. 5 to 7). A DARPA-funded Technology Reinvestment Program (TRP) was established to develop face gear grinding techniques and demonstrate face gears in an actual helicopter gearbox. From this, design guidelines were refined and a grinding methodology using a worm wheel generator was developed (ref. 8). A face gear grinding machine was fabricated, a novel split-torque, face-gear transmission for the U.S. Army AH-64 Apache helicopter application was explored, and load sharing experiments were performed (refs. 9 and 10). The proof-of-concept tests demonstrated effective torque sharing and that face gears yielded good potential for significant weight, cost, and reliability improvements over existing equipment using spiral-bevel gearing.

Further advancements were made in face gear technology in support of the U.S. Army Rotorcraft Drive Systems for the 21st Century (RDS–21) Program performed by Boeing under agreement with the Aviation Applied Technology Directorate of the U.S Army Aviation and Missile Command. The geometry for tapered pinions and idlers for use in a split-torque, face-gear transmission were analyzed (refs. 11 and 12). Detailed face gear tooth contact and bending stress predictions based on finite element and contact solvers were developed (ref. 13). Further analytical studies investigating torque sharing among multiple pinions and idlers, as well as the effect of tail-rotor power taken from an idler, were performed (ref. 14). In addition to studies for the AH-64, face gear applications for the U.S. Army UH-60 Blackhawk helicopter were investigated (ref. 15).

In addition to the U.S., the interest in face gears grew in popularity abroad. A European program called FACET ("The development of face gears for use in aerospace transmission") was established amongst the U.K., France, Germany, and Italy (ref. 16). From this program, significant advancements have been achieved in face gear analysis in the areas of contact analysis of helical face gears, loaded meshing of face gears, and experiment validation of loads (refs. 17 to 19). Work in Japan investigated the use of face gears for rotorcraft application and consisted of prototype testing (ref. 20). Other work considered the stress analysis of face gears using a global-local finite element method (ref. 21). Additional work in Germany considered face gear manufacturing simulation (ref. 22). Work in Italy considered the effect of misalignment and profile modifications for face gears using the finite element method (ref. 23). Lastly, work in Spain looked at an enhanced approach for longitudinal plunging in the manufacturing of a double crowned pinion of a face gear mesh (ref. 24). As can be seen from these references, a considerable amount of effort has been invested in the analysis of face gears. Little work, however, has been made in basic development of design guidelines with respect to the fatigue life of face gears. With that said, endurance tests were performed in the current study in support of the RDS–21 program.

The objective of this study is to determine the surface durability life of a face gear in mesh with a tapered spur involute pinion. Experimental fatigue tests were performed at the Glenn Research Center. The effect of shimming on backlash and contact pattern was investigated. Preliminary tests were performed to evaluate a few pinion and face gear design parameters. Last, endurance tests were performed on twenty-four sets of gears at three loads levels.



a) Overview of facility.

b) Schematic view.

Fig 1. NASA Glenn spiral-bevel-gear, face-gear test facility.

Apparatus

Test Facility

The experiments reported in this report were tested in the NASA Glenn spiral-bevel-gear/face-gear test facility. An overview sketch of the facility is shown in figure 1a and a schematic of the power loop is shown in figure 1b. The facility operates in a closed-loop arrangement. A spur pinion drives a face gear in the test (left) section. The face gear drives a set of helical gears, which in turn, drive a face gear and spur pinion in the slave (right) section. The pinions of the slave and test sections are connected by a cross shaft, thereby closing the loop. Torque is supplied in the loop by physically twisting and locking a torque in the pre-load coupling on the slave section shaft. Additional torque is applied through a thrust piston (supplied with high pressure nitrogen gas), which exerts an axial force on one of the helical gears. The total desired level of torque is achieved by adjusting the nitrogen supply pressure to the piston. A 100-hp DC drive motor, connected to the loop by V-belts and pulleys, controls the speed as well as provides power to overcome friction. The facility has the capability to operate at 750 hp and 20,000 rpm pinion speed. A torquemeter in the loop on the test side measures torque and speed. The facility is also equipped with thermocouples, oil flow meters, pressure transducers, accelerometers, counters, and shutdown instrumentation to allow 24-hour unattended operation.

The test gears and facility bearings and gears were lubricated and cooled by a pressurized oil system. The lubricating fluid used was a synthetic base helicopter transmission oil conforming to the DOD-L-85734 specification. The test pinions and face gears were lubricated by jets which radially directed oil into the roots of the teeth on both the into-mesh and out-of-mesh sides. The nominal oil supply pressure was 80 psi and the nominal flow rate was 1.0 gpm for both the test section and slave section. Oil inlet temperature was set at 85 °F. An external vacuum pump connected to the oil tank worked as a scavenge system to remove the oil from the test gearboxes and bearing cavities and direct it to the sump. Also, the oil system was equipped with an oil-debris monitor as well as a 3-μm filter.

Test Gears

The design parameters for the pinions and face gears used in the tests are given in table I. A photograph of the test specimens is shown in figure 2. The set was primarily designed to fail in surface pitting fatigue mode. The set had a reduction ratio of 3.842:1. The set also had a diametral pitch of 10.6 teeth/in, roughly similar to the previous TRP design and current AH-64 replacement design. The face width of the face gears was 0.6 in.

The face width of the spur pinions was 0.8 in, significantly greater than the face gear to allow for backlash adjustment and optimization of tooth contact. The shaft angle was 90° to accommodate the facility. The pinions were slightly tapered, similar to the TRP design and current AH-64 replacement design, which allows the independent setting of backlash for the multiple pinions and idlers in the split-torque transmission application (ref. 9).

TABLE I.—TEST GEAR DESIGN DATA.

AGMA quality	12
Number of teeth; pinion, gear	19, 73
Diametral pitch	10.6
Pressure angle (deg)	27.5
Shaft angle (deg)	90
Face width (in); pinion, gear	0.8, 0.6
Hardness (Rc); case, core	62, 38
RMS surface finish	16
Material	X53 steel

The pinions and face gears were made from carburized and ground vacuum induction melting-vacuum arc remelting (VIM-VAR) Pyrowear 53 steel per AMS 6308 using standard aerospace practices. At 6000 lb-in face gear torque, the calculated AGMA contact stress index was 250 ksi and the calculated AGMA bending stress index was 72 ksi using approximate spur gear calculations per AGMA (ref. 25).

Test Gear Installation Procedures

Previous studies showed that proper pinion and face gear installation is a criteria for successful operation (ref. 7). Both pinion and face gear adjustments were made (fig. 3).

Figure 4 shows the effect of face gear adjustments on contact pattern and backlash while keeping a constant pinion position. It is clear that the face gear adjustment had a significant effect on both pattern and backlash. Moving the face gear out of mesh increased backlash and moved the contact pattern from heel to toe. On the other hand, moving the face gear into mesh decreased backlash and moved the contact pattern from toe to heel. Figure 5 shows the effect of pinion adjustments on contact pattern and backlash while keeping a constant face gear position. Moving the pinion into mesh decreased backlash, but had a relatively small effect on the face gear tooth contact pattern for the range of adjustments used. Moving the pinion out of mesh increased backlash, and also had a relatively small effect on the face gear tooth contact pattern. As expected, the contact pattern on the pinion tooth moved from heel to toe as the pinion was moved out of mesh (backlash increased), and from toe to heel as the pinion was moved into mesh (backlash decreased). However, since the pinion tooth width was wider than the face gear tooth width, the patterns still remained on the tooth. Thus, adjusting the pinion position was an effective way of adjusting backlash without severely affecting the contact pattern.

The installation procedure for the gears tested was then defined as follows. First, the test-side pinion and face gear were installed in the facility (with no cross shaft connected to the

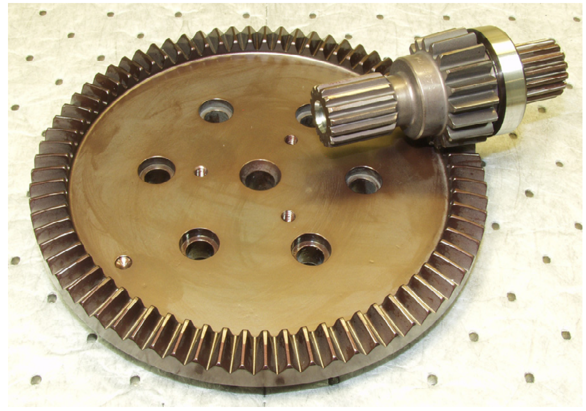


Fig. 2. Test gears.

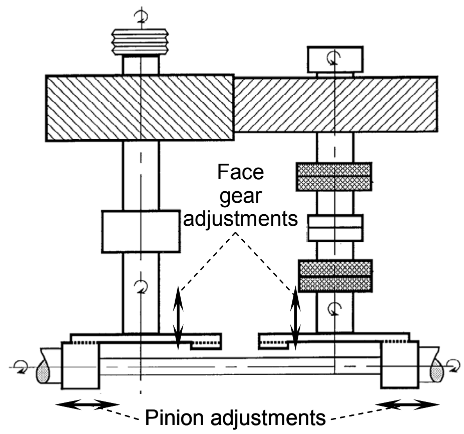


Fig 3. Pinion and face gear shim adjustments.

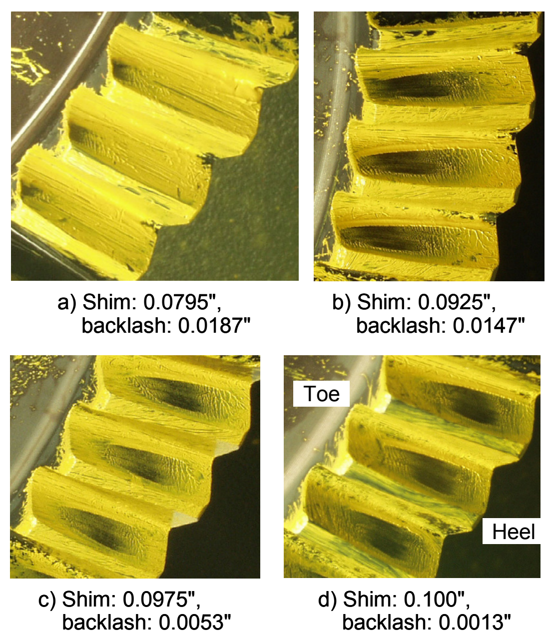


Fig 4. Effect of face gear shim on contact pattern and backlash (all photos shown are face gear patterns).

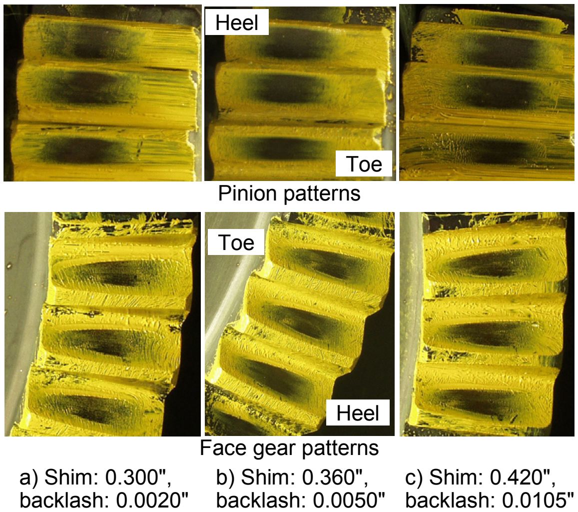


Fig 5. Effect of pinion shim on contact pattern and backlash.

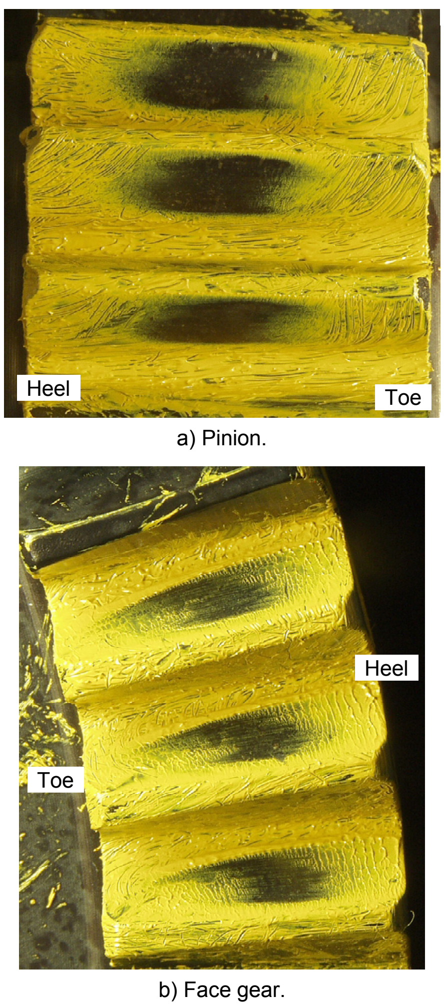


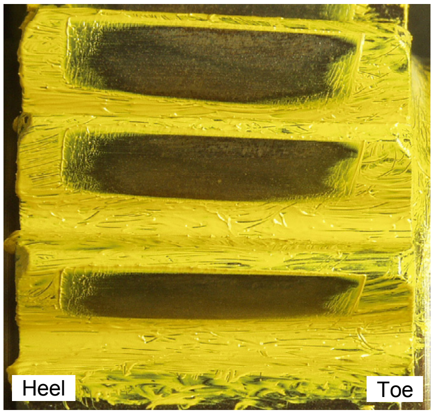
Fig 6. Typical contact pattern check, hand resistance.

pinion). Backlash measurements and no-load contact pattern checks were taken for the mesh as described above. A contact pattern biased slightly toward the heel on the face gear and a backlash of 0.006 to 0.010-in was required for this trial. The slight bias of pattern was required since the pattern shifted slightly toward the toe when full load was applied. If necessary, the face gear shim was first adjusted to achieve the proper contact pattern, and then the pinion shim was adjusted to achieve the proper backlash. This process was then repeated for the slave-side pinion/face-gear mesh. Figure 6 shows an acceptable no-load contact pattern. After proper shimming was achieved, the cross shaft was installed. Marking compound was then re-applied to all the pinions and gears and a loaded static roll test was performed. This was done by applying a moderate torque in the loop (through the load piston), manually rotating the complete assembly, and photographing the resulting contact patterns. Figure 7 shows a typical example of a tooth contact pattern check for a loaded static roll test. The objective of this procedure was to ensure that proper backlash and proper shimming was used, edge loading was prevented, and the contact pattern on the face-gear tooth was evenly spread under load.

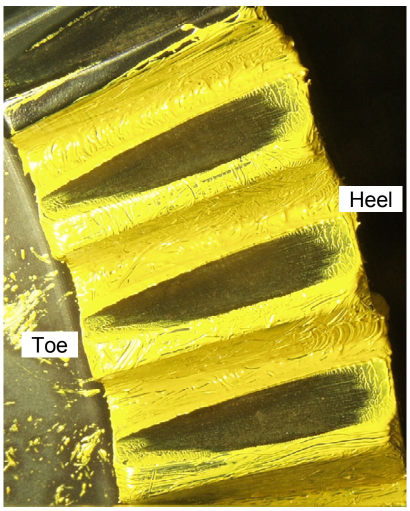
Test Procedure

The test procedure to evaluate the fatigue life of face gears was as follows. First, the selected test gears were installed with the proper shims as described above. Backlash measurements as well as un-loaded and loaded contact patterns were documented. After acceptable patterns and backlash, the gears were then run through a break-in procedure. This was a short 70-min run consisting of a gradual increase in speed and torque. The applied torque was obtained using only the load piston. After completion of the break-in run, the gears were inspected. The pre-load coupling was then adjusted to produce a face-gear torque somewhere between 3000 to 5000 lb-in. The gears were then run at required speed and torque for the specific test (torque adjusted using load piston). Facility parameters as well as high-frequency vibration monitoring with gear fault detection software and oil-debris monitoring were collected. During the tests, the gears were inspected at routine intervals (5 to 10 million face gear cycles). The gears were run until a surface durability failure occurred or a suspension was defined. A surface durability failure was defined as macro-pitting or spalling of at least 0.1-in continuous length along the contact area on any tooth of a tested pinion or face gear. Once completed, the gears were removed from the facility, cleaned, and photographed for documentation purposes.

An initial series of tests was done to evaluate various pinion and face gear tooth profiles. The concern was to avoid hard lines on the tooth surfaces. Hard lines are concentrated wear lines that are caused either at the beginning, end, or edge of contact and occur at high loads. Hard lines could cause premature pitting at the region of high contact stress and bias test results. High loads were required for the tests in order to obtain fatigue lives in a reasonable amount of time. Proper adjustments were made



a) Pinion.



b) Face gear.

Fig 7. Typical loaded contact pattern check, ~4400 lb-in gear torque.

to minimize edge contact. In addition, the face gears had slight crowning in the longitudinal direction of the tooth based on methods by Litvin (ref. 8). This also minimized edge contact. Thus, the concern of tip and root contact was studied. Various pinion tip and root relief profiles were tested (table II). In addition, face gear tooth tip designs with increased edge break were investigated (table III). These gears were finished using a rolled-brushing process. Six tests were performed at 6000 lb-in face gear torque and face gear speeds from 3280 to 4700 rpm, depending on the vibration levels of the test.

TABLE II.—PINION DESIGN VARIATION PARAMETERS.

Pinion design	Root relief (in)	Tip relief (in)	Number tested
Mod1	0.0004	0.0002	2
Mod2	0.0007	0.0005	2
Mod3	0.0010	0.0002	4
Mod4	0.0014	0.0002	5
Mod5	0.0012	0.0002	1

TABLE III.—FACE GEAR DESIGN VARIATION PARAMETERS.

Face Gear Design	Tip radius (in)	Number tested
Original	0.010	6
Rolled-tips	0.010	6
Increased-rolled-tips	0.020	2

After completion of the tooth profile evaluations, the Mod3 pinion design and Rolled-tips face gear design (tables II and III) were chosen for the endurance tests. In addition, the face gears were shot-peened and vibro-honed to match conditions proposed for the AH-64 replacement design. Twenty-four sets of this design were fabricated. Tests were performed at three load levels: 7200 lb-in face gear torque (275 ksi calculated AGMA contact stress), 8185 lb-in face gear torque (292 ksi contact stress), and 9075 lb-in face gear torque (307 ksi contact stress). Test speeds were 2190 to 3280 rpm face gear speed, depending on the vibration levels of the test. Note that for these test conditions, it was estimated that the test sets operated in the mixed elasto-hydrodynamic/boundary lubrication regime based on calculate oil film thicknesses.

Results and Discussion

Twenty-four sets of gears were tested as part of the endurance tests using the Mod3 pinion design and Rolled-tips face gear design. Twelve sets were tested on the left (test) side and twelve were tested on the right (slave) side. Tests were performed at three load levels: 7200 lb-in face gear torque (275 ksi calculated AGMA contact stress), 8185 lb-in face gear torque (292 ksi contact stress), and 9075 lb-in face gear torque (307 ksi contact stress). Test speeds were 2190 to 3280 rpm face gear speed, depending on the vibration levels of the test. Of the twenty-four sets tested, 17 resulted in spalling failures and 7 were suspended with no spalling. The number of cycles tested per set ranged from 32.7 to 590.9 million pinion cycles. For all the failed sets, spalling occurred on at least one tooth for all the pinions. In some cases, the spalling initiated a crack in the pinion teeth which progressed to tooth fracture. In some cases, spalling occurred on face gear teeth, although this was not the norm.

Figure 8 gives the results of all the twenty-four sets on a S/N (stress/cycle) plot. The vertical axis is calculated AGMA contact stress, S_c. The horizontal axis is pinion cycles. Again, all failures plotted are from spalling. Also plotted in figure 8 is a regression curve fit of the test data using a two-parameter power function (dotted line). The curve fit neglects suspended items. The result of the regression fit relating contact stress, S_c, to number of cycles, N, is

$$S_c = 422.5\, N^{-0.021} \qquad (1)$$

Lives using AGMA guidelines (ref. 25) for spur and helical gears are also plotted in figure 8. These lives were determined from the AGMA stress cycle factor, Z_n, as a function of stress cycles, N, where

$$Z_n = \begin{Bmatrix} 2.466\ N^{-0.056} & \text{for } N < 10^7 \text{ cycles} \\ 1.4488\ N^{-0.023} & \text{for } N > 10^7 \text{ cycles} \end{Bmatrix} \tag{2}$$

and

$$S_c = Z_n S_{ac} \tag{3}$$

where S_{ac} is the AGMA allowable contact stress. Here, the stress cycle factor, Z_n, is equal to one at N=10,000,000 cycles. The allowable contact stress was set to equal to S_{ac}=275 ksi, corresponding to a Grade 3 material in the AGMA guidelines (ref. 25) since the test gears were VIM-VAR steel. Using eqs. (2) and (3), relating contact stress, S_c, to number of cycles, N, gives

$$S_C = \begin{Bmatrix} 678.2\ N^{-0.056} & \text{for } N < 10^7 \text{ cycles} \\ 398.4\ N^{-0.023} & \text{for } N > 10^7 \text{ cycles} \end{Bmatrix} \tag{4}$$

which is plotted in figure 8. As can be seen from eqs. (1) and (4), the slope of the regression fit on the test data matches very closely to that of the AGMA for N>10^7 cycles.

Figure 9 is a Weibull plot of the endurance data points at the 275-ksi contact stress level. The data for the plot was created using methods of Johnson (ref. 26). The procedure plots the medium rank of the data point as a function of number of cycles. The medium rank is also adjusted for suspensions. The data is plotted on special Weibull logarithmic axes. Also included in the plot are 90% confidence bands. Specific lives from the analysis are tabulated in table IV. Shown are the L_1 life (1% probability of failure), L_{10} life (10% probability of failure), L_{50} life (50% probability of failure), and mean life as well as upper and lower 90% confidence band limits. For tests at 275 ksi contact stress, the Weibull slope was 1.67 and the failure index was 7 out of 12 (7 failures out of 12 total sets). Figure 10 is a Weibull plot of the endurance data points at the 292-ksi contact stress level. Here, the Weibull slope was 1.54 and the failure index was 4 out of 6. Figure 11 is a Weibull plot of the endurance data points at the 307-ksi contact stress level. Here, the Weibull slope was 2.23 and the failure index was 6 out of 6. Specific lives for these test conditions are also tabulated in table IV.

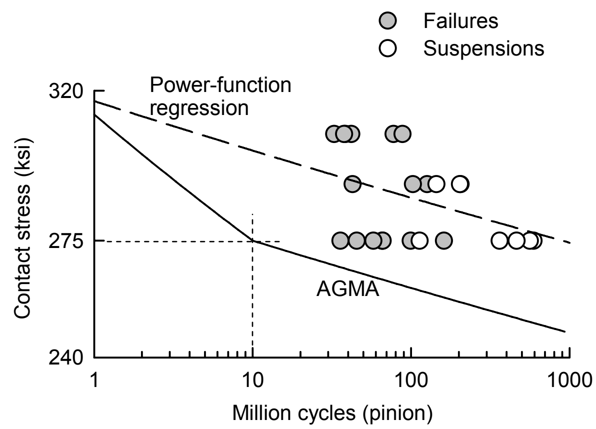


Fig 8. S/N (stress/cycle) plot results of face gear endurance tests.

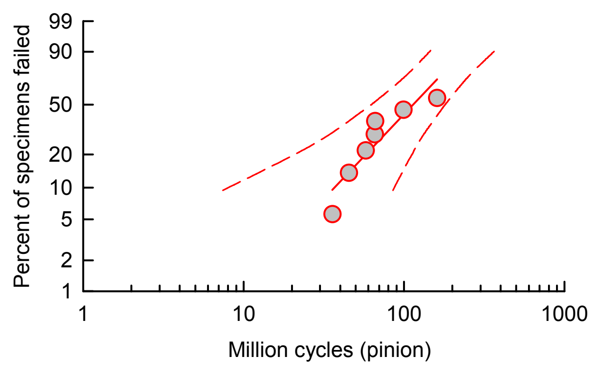


Fig 9. Weibull plot with 90% confidence bands, endurance runs data at 275 ksi contact stress.

TABLE IV.—SUMMARY OF FATIGUE LIVES FROM WEIBULL ANALYSIS, ENDURANCE TEST RESULTS

			Contact stress (ksi)		
			275	292	307
Gear set life, M Cycles (pinion)	L_1	Lower 90% Confidence Limit	---	---	---
		Experimental	9.0	9.4	7.8
		Upper 90% Confidence Limit	47.5	72.1	27.6
	L_{10}	Lower 90% Confidence Limit	8.1	4.3	6.4
		Experimental	36.8	43.2	22.2
		Upper 90% Confidence Limit	86.6	132.1	43.5
	L_{50}	Lower 90% Confidence Limit	62.4	61.2	31.8
		Experimental	114.0	147.2	51.8
		Upper 90% Confidence Limit	181.0	272.6	75.1
	L_{mean}	Lower 90% Confidence Limit	71.9	74.2	33.7
		Experimental	126.9	168.3	54.1
		Upper 90% Confidence Limit	198.0	302.8	77.7
Weibull slope			1.67	1.54	2.23
Failure index			7 out of 12	4 out of 6	6 out of 6

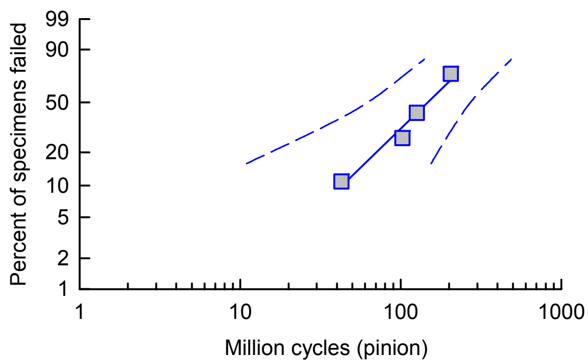


Fig 10. Weibull plot with 90% confidence bands, endurance runs data at 292 ksi contact stress.

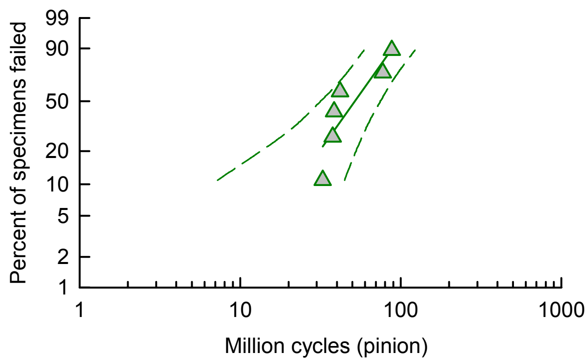


Fig 11. Weibull plot with 90% confidence bands, endurance runs data at 307 ksi contact stress.

Figure 12 combines all the data from the 275, 292, and 307-ksi contact stress levels (figs. 9 to 11) into a single plot. As seen from this figure and with table IV as well, the resulting lives at 275 ksi were less than that at 292 ksi. This inconsistency was due to two factors. The first was caused by a couple of relatively low-life outcasts at the 275-ksi stress level. Second was the number of failure points. Even though 24 sets were tested and 17 failures were produced, more data was needed to produce statistically significant results.

This is also evident in table V where the significance of the resulting lives at the 292-ksi and 307-ksi stress levels are compared to that at 275 ksi. This table was created using the methods of Johnson (ref. 26) in which the confidence that the lives of a tested population are greater (or less) than those of another tested population is explored. This confidence is based on the sample sizes (number of failures for both populations) and life ratios of the populations, either at the L_{10} life or mean life level. From table V, the confidence of the L_{10} life at 307 ksi stress relative to 275 ksi is 70%. This means that 70 times out of 100, the L_{10} population life at 307 ksi is less than that at 275 ksi. The confidence of the L_{10} life at 292 ksi stress relative to 275 ksi is only 54%. Confidence numbers in the 70 to 80% range are rather marginal in significance. Numbers in the 50 to 60% range indicate no significant difference. The only conclusion that can be drawn with high confidence is that the mean life at 307 ksi is statistically less than that at 275 ksi (confidence >99%).

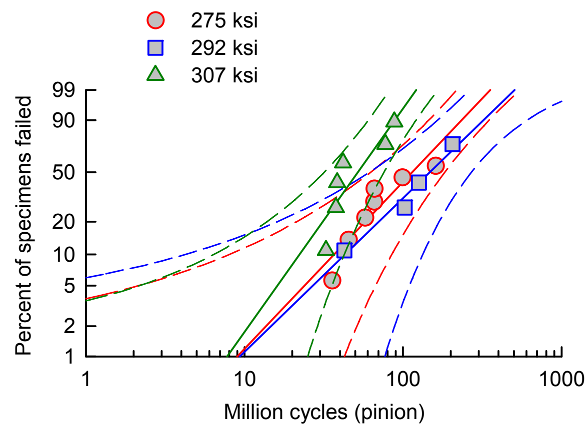


Fig 12. Weibull plots, endurance runs data, all stress conditions.

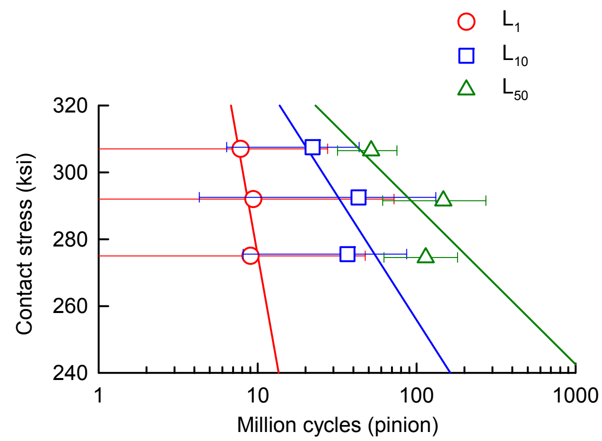


Fig 13. S/N (stress/cycle) plot results of face gear endurance tests at the 99% (L_1), 90% (L_{10}), and 50% (L_{50}) reliability levels.

TABLE V.—SIGNIFICANCE OF FATIGUE LIVES RELATIVE TO THOSE AT 275 ksi CONTACT STRESS

	Contact stress (ksi)	
	292	307
L_{10}	54%	70%
L_{mean}	79%	>99%

The L_1, L_{10}, and L_{50} lives from the Weibull analysis for the various stress levels are plotted in an S/N curve in figure 13 along with linear curve fits. The purpose was to define the AGMA allowable endurance limit from the test data. AGMA uses the L_1 life in its calculation. That is, the endurance allowable is defined as the load that results in 10 million cycles life for a reliability of fewer than one in 100 failures. From figure 13, the curve fit at the L_1 life level is:

$$S_c = 2146.5 - 267.32 \log(N) \tag{5}$$

From eq. (5), for N=10,000,000 cycles, S_{ac}=S_c=275 ksi. Thus, the AGMA endurance limit from the test data was the same as that for a Grade 3 material as recommended in the AGMA guidelines (ref. 25). The test results support use of the AGMA spur-helical gear method for estimating the pitting resistance of face gear sets.

Conclusions

Experimental fatigue tests were performed to determine the surface durability life of a face gear in mesh with a tapered spur involute pinion. The tests were performed at the Glenn Research Center in the Glenn spiral-bevel-gear/face-gear test facility. The effect of shimming on backlash and contact pattern was investigated. Preliminary tests were performed to evaluate a few pinion and face gear design parameters. Lastly, endurance tests were performed on twenty-four sets of gears at three loads levels. The following results were obtained:

(1) The AGMA endurance allowable stress for a tapered spur involute pinion in mesh with a face gear was determined to be 275 ksi for carburized and ground, shot-peened and vibro-honed, vacuum induction melting-vacuum arc remelting (VIM-VAR) Pyrowear 53 steel per AMS 6308. The tests produced 17 gear tooth spalling failures and 7 suspensions. For all the failed sets, spalling occurred on at least one tooth of all the pinions. In some cases, the spalling initiated a crack in the pinion teeth which progressed to tooth fracture. Also, spalling occurred on some face gear teeth, although this was not the norm.

(2) Even with 17 failure points, the statistical significance of the endurance tests results was marginal when tested at three load levels.

(3) For the application of a tapered spur involute pinion in mesh with a face gear, proper face gear shim controlled the desired gear tooth contact pattern while proper pinion shim was an effective way of adjusting backlash without severely affecting the contact pattern.

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13. SUPPLEMENTARY NOTES

14. ABSTRACT
Experimental fatigue tests were performed to determine the surface durability life of a face gear in mesh with a tapered spur involute pinion. Twenty-four sets of gears were tested at three load levels: 7200, 8185, and 9075 lb-in face gear torque, and 2190 to 3280 rpm face gear speed. The gears were carburized and ground, shot-peened and vibro-honed, and made from VIM-VAR Pyrowear 53 steel per AMS 6308. The tests produced 17 gear tooth spalling failures and 7 suspensions. For all the failed sets, spalling occurred on at least one tooth of all the pinions. In some cases, the spalling initiated a crack in the pinion teeth which progressed to tooth fracture. Also, spalling occurred on some face gear teeth. The AGMA endurance allowable stress for a tapered spur involute pinion in mesh with a face gear was determined to be 275 ksi for the material tested. For the application of a tapered spur involute pinion in mesh with a face gear, proper face gear shim controlled the desired gear tooth contact pattern while proper pinion shim was an effective way of adjusting backlash without severely affecting the contact pattern.

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