Vibration and Operational Characteristics of a Composite-Steel (Hybrid) Gear

Robert F. Handschuh, Kelsen E. LaBerge, Samuel DeLuca and Ryan Pelagalli

Hybrid gears have been tested consisting of metallic gear teeth and shafting connected by composite web. Both free vibration and dynamic operation tests were completed at the NASA Glenn Spur Gear Fatigue Test Facility, comparing these hybrid gears to their steel counterparts. The free vibration tests indicated that the natural frequency of the hybrid gear was approximately 800 Hz lower than the steel test gear. The dynamic vibration tests were conducted at five different rotational speeds and three levels of torque in a four square test configuration. The hybrid gears were tested both as fabricated (machined, composite layup, then composite cure) and after regrinding the gear teeth to the required aerospace tolerance. The dynamic vibration tests indicated that the level of vibration for either type of gearing was sensitive to the level of load and rotational speed.

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Introduction

Drive systems of the future will be required to have increased power-to-weight ratios in addition to reduced maintenance, noise, and cost. These requirements have been under constant evolution, resulting in improvements over the last several decades. Materials, manufacturing, processing, lubricants, and advanced drive system analysis have led to drive system designers pushing the technology and steadily making improvements (Refs. 1–5). Typically these technology improvements are implemented when drive system upgrades occur and when new vehicles are designed. The implementation is timed in this manner to reduce costs, as qualification of a drive system for rotorcraft is an expensive process.

Power-to-weight ratio is one of the most important drive system attributes. Parametrically the resultant gearbox and lubrication system weight, w_t, for rotorcraft of all types, sizes, number of engines, and number of main rotors is shown (Fig. 1) (Ref. 6). Here, the parametric value, γ, is a function of the input and output speeds v_{in} and v_{out}, respectively (both in revolutions-per-minute), of the drive system and the transmitted power P (in horsepower). Once this calculation is made, the anticipated weight of the gearbox and lubrication system, w_t, can be found. Using advanced technologies results in drive systems that are lighter weight for a given gear ratio and power level such that they fall below the curve shown (Fig. 1). This improved capability offers the rotorcraft extended range and/or payload.

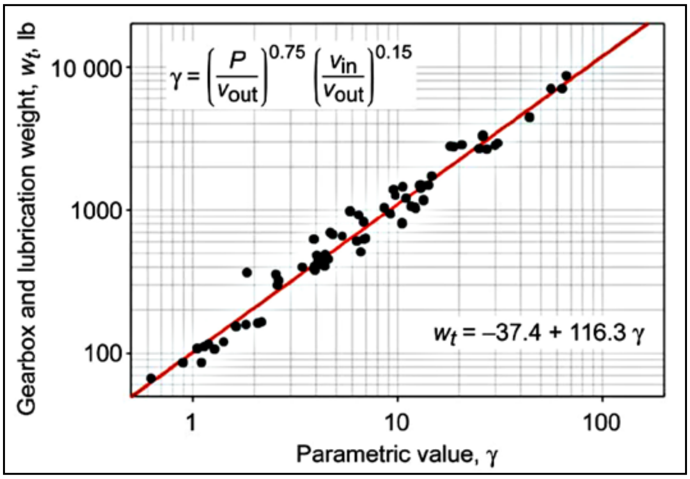

Figure 1 Parametric curve fit for rotary-wing aircraft gearbox and lubrication system weight (Ref. 6).

Materials play an important role in improving the power-to-weight ratio. Currently rotorcraft drive systems utilize lightweight structure materials (aluminum and magnesium) for the housing and minimize gear weight via careful analysis and machining. Generally, rotorcraft gears have only enough mass for load carrying ability. Minimizing the mass of a gear leads to a lack of heat storage capability. This attribute can cause problems during a primary lubrication system failure in which gears would operate under starved or dry conditions. Therefore all operation scenarios need to be carefully considered.

Composite materials have been considered for some rotorcraft drive system housings, such as the work contained in Reference 7.

Table 1 Composite and aerospace steel material properties.

Property	Composite material	AISI 9310 gear steel
Modulus of elasticity, Pa (psi)	Tension: 44×10^9 (6.4×10^6) Compression: 42×10^9 (6.1×10^6)	200×10^9 (29×10^6)
Poisson's ratio	0.3	0.29
Density, kg/m^3 (lb/ft^3)	1800 (112)	7861 (491)
Thermal conductivity, W/m•°C (Btu/h.ft.°F)	T700 fiber-axial: 9.4 (5.43)	55 (32)
Useful maximum temperature as gear material, °C (°F)	150 (302)	175 (347)
Coefficient of thermal expansion, 10^{-6} m/m-K^{-1} (10^{-6} in/in·°F^{-1})	In plane: 2 (1.1)	13 (7.3)
Failure strain, percent	Tension: 1.9 Compression: 0.94	
Elongation, percent		15

Many issues were worked out to attain project success. However, cost and some other technical issues yet to be resolved have been a roadblock to incorporating this technology into production.

In addition to structural components, there have also been recent efforts to incorporate lower-density composites in dynamic components, such as shafts and gears. This report focuses on the potential application of composite material in rotorcraft drive system gears. The web of the test gear was replaced with composite material. The material properties of the composite material used in this study are compared with those of a typical aerospace gear material in Table 1. One property that is of real importance is the density. The density of the composite material used in this study is approximately 25 percent of that of typical gear steel. Also, there was an anticipated benefit expected that the material change should help with mesh-generated vibration and noise that is transmitted from the gears to the shafts and bearings.

The objective of this study is to describe how composite webbed gears (referred to here as "hybrid" gears) were fabricated and the resultant effect on the gear natural frequency, transmitted vibration, and noise.

Hybrid Gear Manufacture

The hybrid gears manufactured in this study followed the process as described (Reference 8); a brief description follows.

The test gear design used for this study has the design shown (Table 2). Figure 2 provides a pictorial explanation of the hybrid gear assembly process. A hexagonal region was machined out of a steel gear leaving two steel gear components: a gear rim with the teeth and a hub region for attachment to the facility shafting. The braided pre-preg composite material (*fibrous material pre-impregnated with a particular synthetic resin used in making reinforced plastics*) was built up in a fixture around the steel hub and rim as shown (Fig. 2, steps 3–8). A total of 36 layers of composite material were built up and then cured in the fixture (step 9) at a final temperature of 177°C (350°F). The fixture for fabrication and curing used the inner diameter of the hub and the gear measurement over pins in an attempt to keep the gear teeth aligned with the axis of rotation. An example of the cured gear is shown (Fig. 3).

Table 2 Gear design for this study.

Number of teeth	42
Module, mm (Diametral pitch 1/in)	2.12 (12)
Circular pitch, mm (in.)	6.650 (0.2618)
Whole depth, mm (in.)	4.98 (0.196)
Addendum, mm (in.)	2.11 (0.083)
Chordal tooth thickness, mm (in.)	3.249 (0.1279)
Pressure angle, deg	25
Pitch diameter, mm (in.)	88.9 (3.5)
Outside diameter, mm (in.)	93.14 (3.667)
Measurement over pins, mm (in.)	93.87 (3.6956)
Pin diameter, mm (in.)	3.66 (0.144)
Backlash ref., mm (in.)	0.15 (0.006)
Tip relief, mm (in.)	0.013-0.018 (0.0005-0.0007)
All-steel gear weight, kg (lbf)	0.3799 (0.8375)
Hybrid gear weight, kg (lbf)	0.3242 (0.7147)

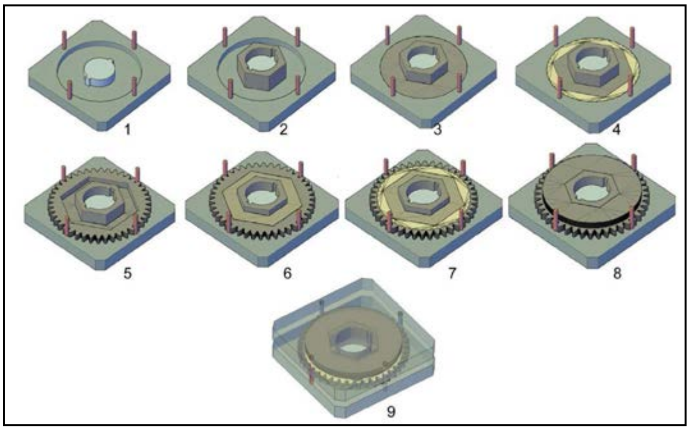

Figure 2 Hybrid gear assembly process.

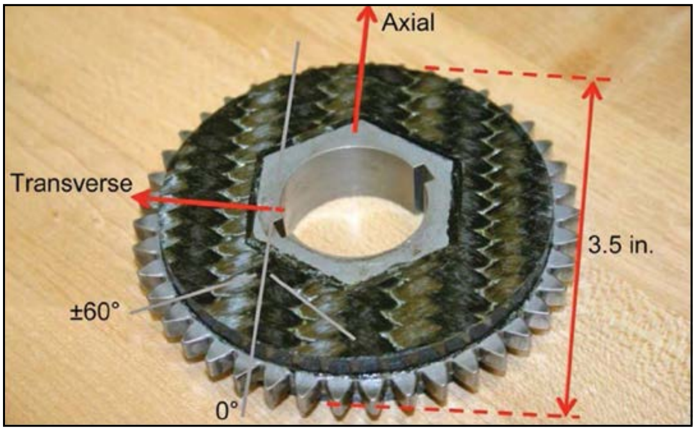

Figure 3 Assembled and cured hybrid gear.

A total of four gears (two pairs) were manufactured and used in this study. The first gear pair (A) was used for fatigue testing (to be described later), while the second pair (B) was used for both static and dynamic vibration tests. Because of material differences, i.e. — coefficient of thermal expansion — the initial gear geometry was not preserved after the curing process, thus leading to significant pitch variation and other anomalies compared with the original steel aerospace gear. Figure 4 provides example pitch variation measurements for two of the four hybrid gears after curing. These pitch variations were greater in gear set A than in set B. This resulted in a reduction in backlash, normally around 0.15 mm (0.006 in.), of 0 to 0.038 mm (0 to 0.0015 in.). Even with the reduced backlash, gear set A withstood fatigue testing without damage to the contact surfaces.

To reduce the pitch variations and return the tooth geometry to its original form, the teeth on gear set B were reground after performing initial vibration tests. The geometry of one of the re-ground gears is shown in Figure 5. Although the tooth geometry was restored, the re-grind process resulted in excess backlash of 0.58 to 0.64 mm (0.023 to 0.025 in.), which is nearly 4 times the normal steel gear backlash. Some

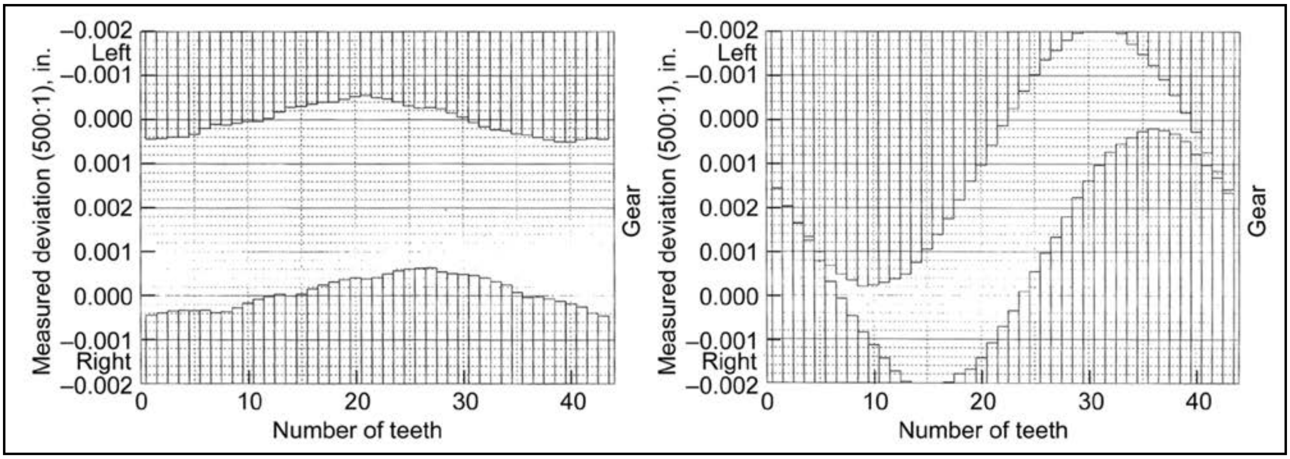

Figure 4 Measured pitch variation (index) of two assembled hybrid gears.

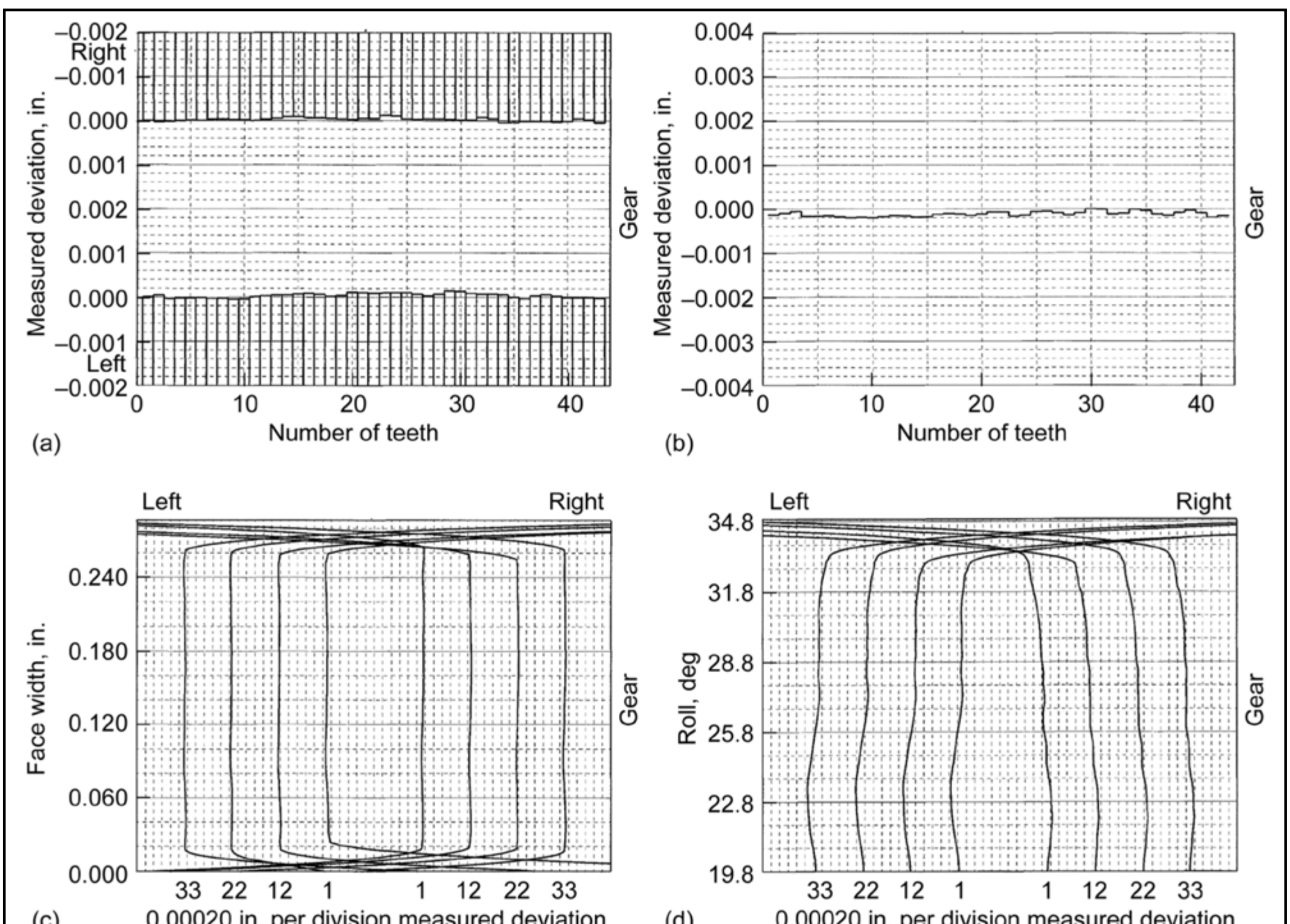

Figure 5 Hybrid gear geometry measurements after regrind; (a) pitch variation (index); b) pitch line runout; c) lead; d) involute.

modifications would be needed in adapting this process to production, such as waiting until after composite curing to perform the final tooth grind to minimize repeat effort.

Free Vibration Modes

In an effort to understand how adding a composite web to a spur gear affects the vibration and noise generated and/or transmitted, first free-free vibration tests were performed in the form of impact tests. Impact tests were performed on both the composite hybrid gear and its steel counterpart. The vibration response was collected with an accelerometer connected to the gear's inner bore via a small aluminum bracket affixed (glued) to the bore to provide a rigid mount that was easily removed. The gear was suspended by an elastic band. Using an instrumented modal hammer, the gear was impacted radially in line with the accelerometer. This is slightly different from the setup previously reported, that placed the accelerometer in the axial direction adjacent to the bore (Ref. 8). Figure 6 shows the test setup and accelerometer bracket. Data acquisition was triggered by the instrumented hammer and was collected at a sample rate of 50 kHz. Figure 7 shows representative vibration plots of both a hybrid and a steel gear. Although the majority of the vibration has diminished in the composite gear after 5 ms, the steel gear vibrates out past 10 ms.

The accelerance for the steel gear impact tests is shown (Fig. 8); the first natural frequency for the steel gear is around 7,000 Hz. This is consistent with a previous finite element analysis (FEA) that provided estimated natural frequencies at 7,187 and 7,270 Hz (Ref. 8). Similarly, the natural frequencies seen at 12.1 and 12.7 kHz are also close to FEA approximations. Those seen at 15.2, 16.5 and 17.2 kHz were not seen in previously published axial impact tests—likely due to the fact that the mode shapes around 15 and 16 kHz involve only radial motion, as was seen in the FEA-estimated mode shapes. Since damping for any particular mode is related to the width of the peak in the frequency spectrum, the sharp spikes at 12.1, 12.7, 16.5 and 17.2 kHz represent the modes with limited damping, whereas the modes at 7 and 15.2 kHz exhibit a higher level of damping.

Hybrid gear impact tests displayed (Fig. 9) show the first natural frequency around 6,400 Hz; this is consistent with

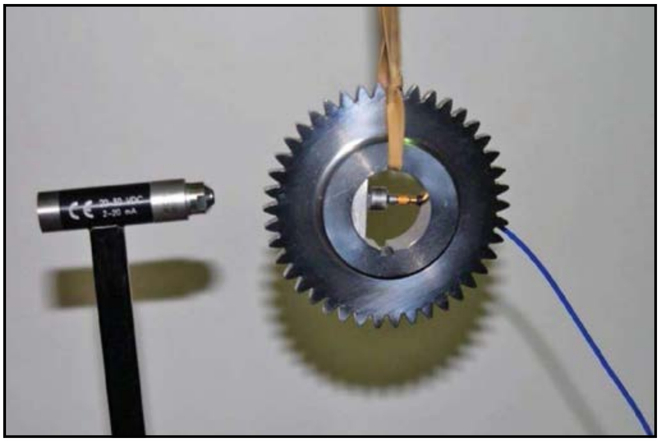

Figure 6 Impact test setup (with steel gear).

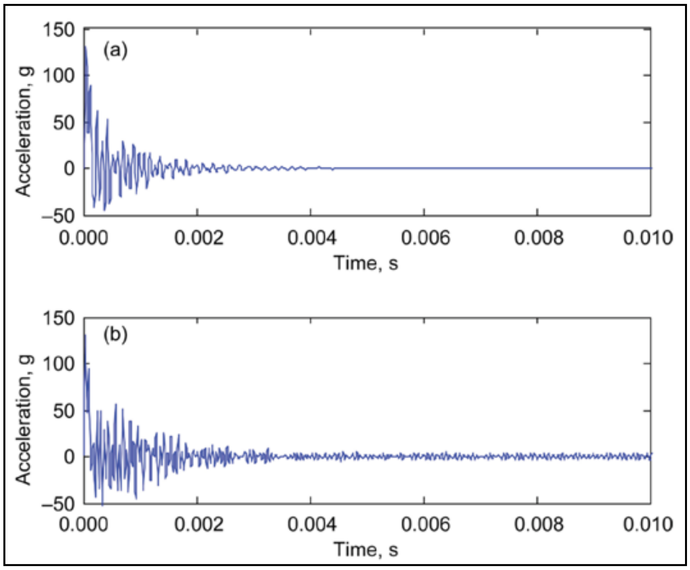

Figure 7 Vibration response of typical impact tests; a) hybrid gear; b) steel gear.

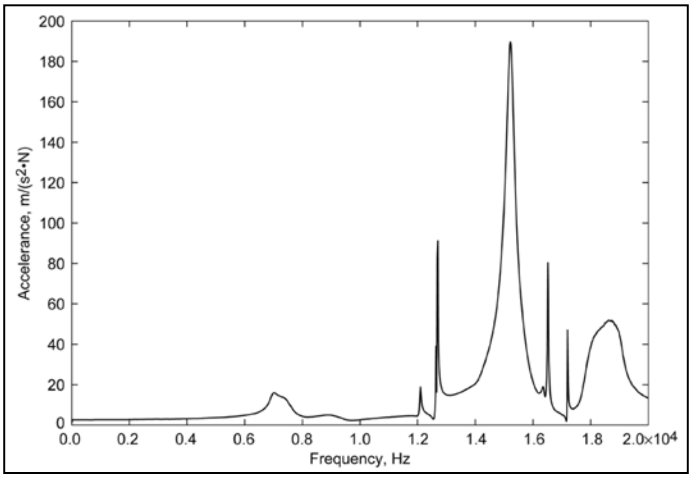

Figure 8 Steel gear accelerance magnitude (average of 10 impacts).

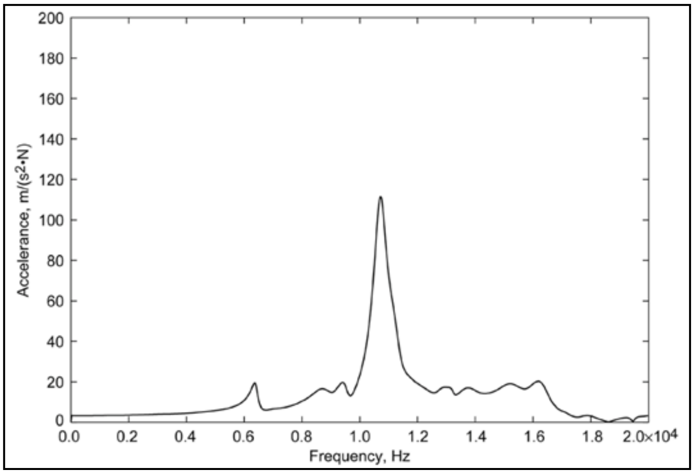

Figure 9 Hybrid gear accelerance magnitude (average of 10 impacts).

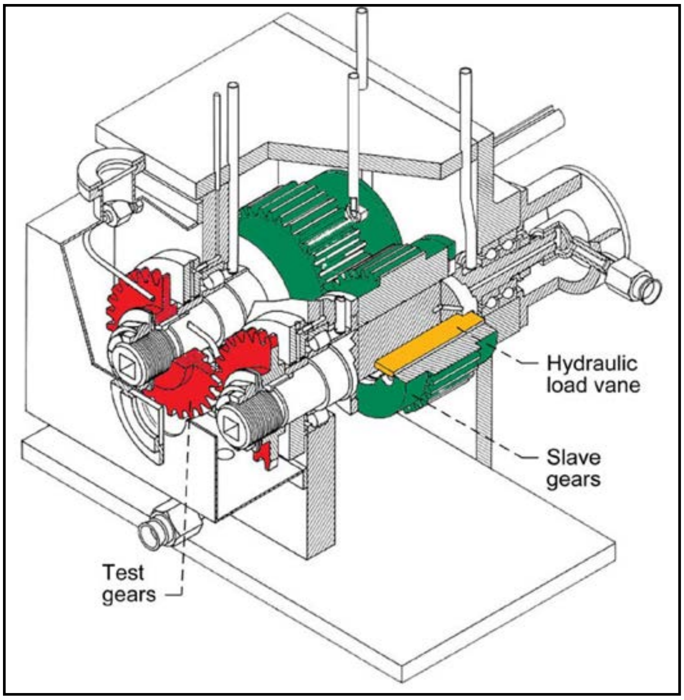

Figure 10 NASA Glenn spur gear contact fatigue facility test rig.

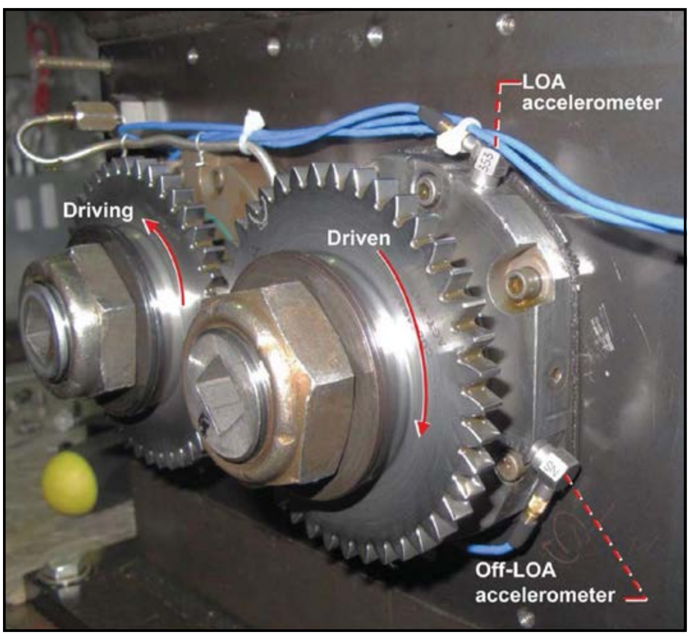

Figure 11 Location of accelerometers on driven shaft of test rig.

previous testing (Ref. 8). These results also show a strong natural frequency at 10.7 kHz. It is difficult to separate the other resonant frequencies, as they are highly damped and overlap. Neither of these natural frequencies is consistent with those identified by the FEA that predicted the first frequency to be higher than that of the steel gear—at 7,780 Hz (Ref. 8). Further modeling efforts are needed to obtain useful hybrid gear FEA predictions, with a focus on ply arrangement effects and a better model of the interface between the steel and composite material.

Test Facility

The dynamic tests reported herein were conducted on the NASA Glenn spur gear fatigue test rig (Fig. 10), a closed-loop test rig that operates at speeds up to 10,000 rpm and 57.9 N·m (513 in·lb) torque. Speed and load are independently adjustable; load is adjusted by varying the supply pressure to the rotating torque actuator located in one of the slave gears. From FEA the torque mentioned above induces a bending stress of 0.212 GPa (30.8 ksi) and a contact stress of 1.17 GPa (170 ksi). These stresses were calculated using the full three-dimensional FEA method described in Reference 9.

Vibration data were attained using accelerometers with a bandwidth of 10 kHz with ±5 percent accuracy (18 kHz at ±10 percent). The accelerometers were mounted as shown (Fig. 11), such that one was parallel to the line of action (LOA) of the gears and the other perpendicular (off-LOA). Data were captured via laboratory analog-to-digital converters and read into a computer at a rate of 50 kHz.

Experimental Results

The experimental results of the three types of tests performed are described in the following sections.

1. Vibration. Dynamic vibration data was taken with test gears installed in four different configurations: steel driving steel, steel driving hybrid, hybrid driving steel, and hybrid driving hybrid. It is important to note that the hybrid gears used for the dynamic tests were set B that were reground to aerospace tolerances, as discussed previously. The tests were performed at five different shaft speeds and at three levels of torque (load pressure). The rotational speeds were 2,500, 5,000, 7,500, 8,750 and 10,000 rpm and the torque levels were at 20.5, 39.2, and 57.9 N·m (182, 347, and 513 in·lb, respectively). From Figure 11, the driving gear is the one on the left and the driven gear is on the right. All vibration data were taken from the right-side seal housing that is in direct contact with the shaft support bearing.

Root-mean-square (RMS) vibration levels for each test are shown (Fig. 12) for both the LOA and off-LOA accelerometers. Results show that the highest vibration was experienced when the hybrid gear was driving the steel gear. There exists—when comparing the hybrid driving steel results with the steel driving hybrid results—a significant reduction in vibration. With the driving gear mounted on the left and the driven gear mounted on the right, adjacent to the accelerometers, we see that lower vibration levels are experienced when the hybrid gear is mounted next to the sensors. The hybrid gear pair shows some improvement over the steel pair at

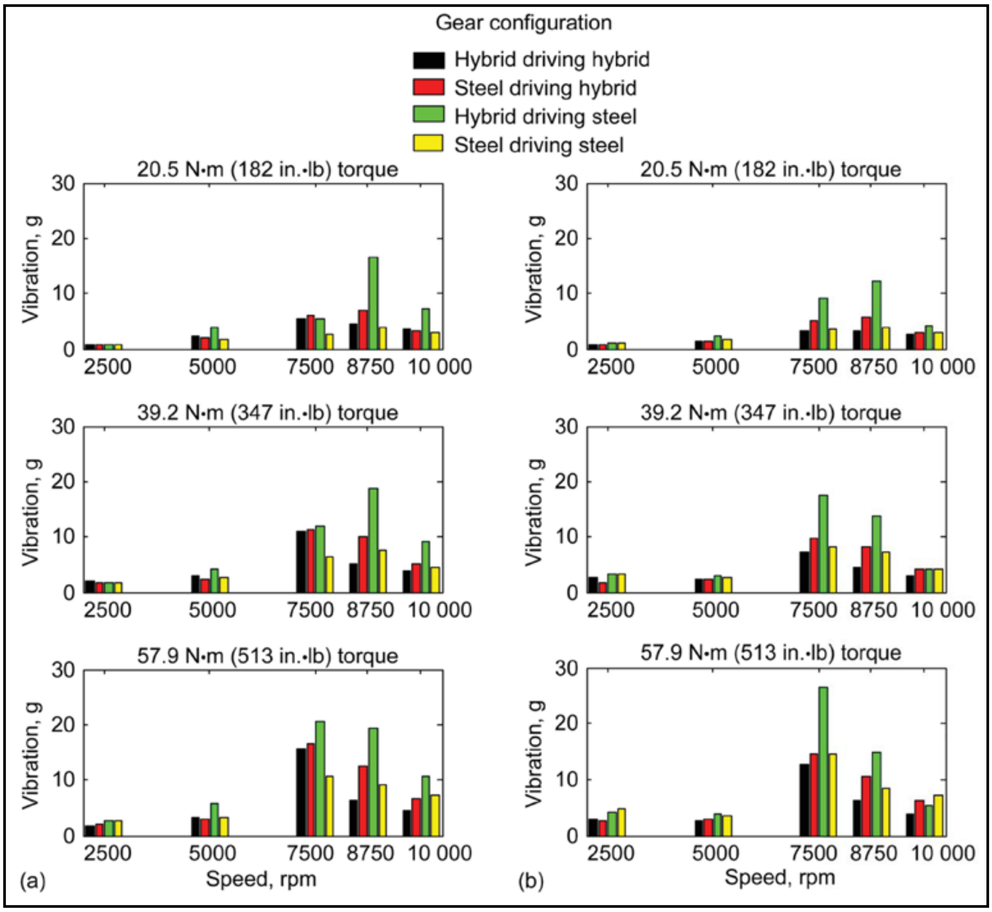

Figure 12 Comparison of root-mean-square (RMS) vibration signal of gears in different configurations; a) along the line of action (LOA); b) perpendicular to the line of action (off-LOA).

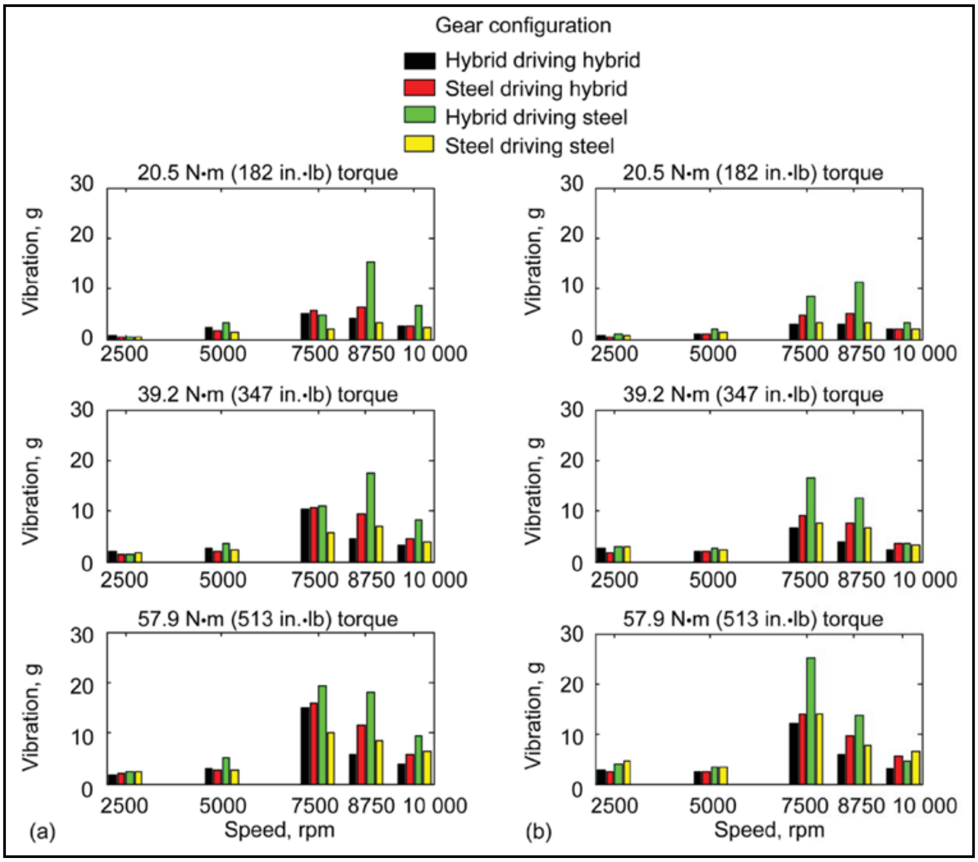

Figure 13 Comparison of RMS synchronous signal average of dynamic vibration of gears in different configurations; a) along the line of action (LOA); b) perpendicular to the line of action (off-LOA).

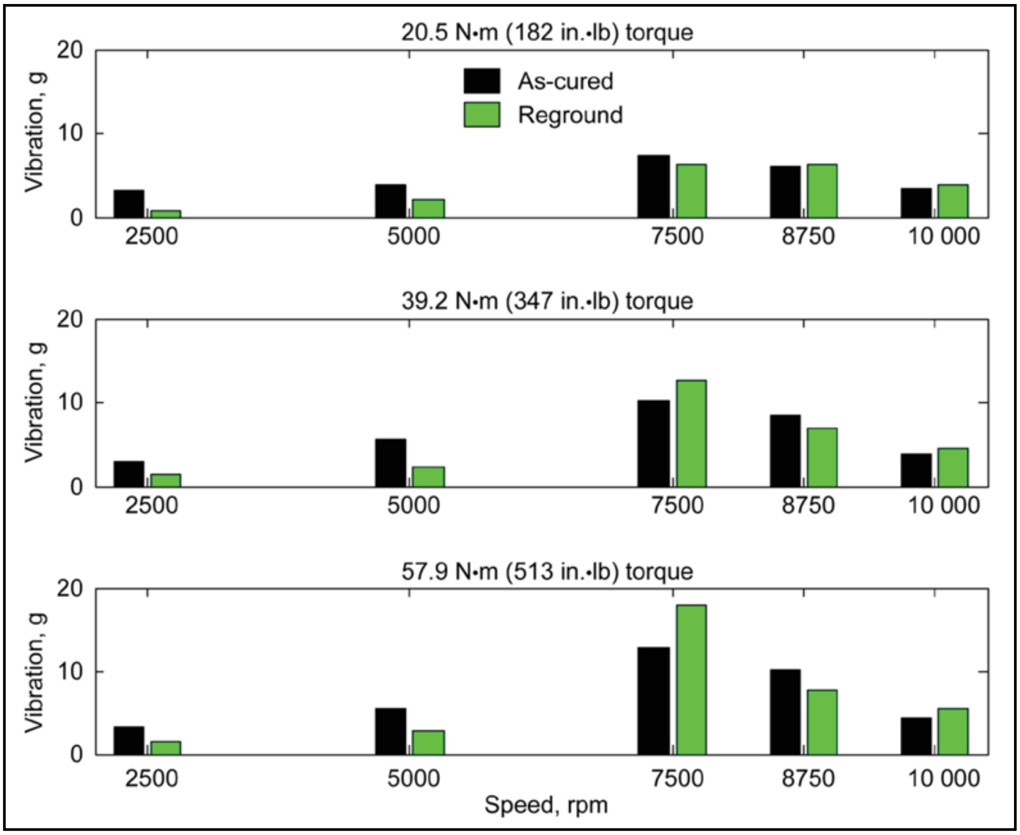

Figure 14 **Comparison of RMS vibration level along the line of action (LOA) between a hybrid gear pair before and after regrind.**

the higher rotational speeds and loads for the LOA accelerometer, and shows a small overall improvement when comparing to off-LOA results.

The raw data were also averaged over 20 shaft rotations to obtain the time-synchronous average. This process is performed to remove nonsynchronous noise from the data. The RMS of the average is shown (Fig. 13) and provides similar results to those shown in Figure 12.

Dynamic tests were performed both before and after gear set B was re-ground, providing a comparison between the two geometries; these vibration results are shown (Fig. 14). The re-grind process is shown to reduce the vibration by as much as half at lower speeds. The results presented could be improved by optimizing the composite curing process to minimize the distortion that occurs. The need for re-grinding after the composite cure process may not be eliminated, but the amount of material removal required to bring the geometry back to aerospace quality could be reduced.

2. Noise. While the vibration data was being taken, a sound pressure level measurement was made. The measurement was taken using a single, hand-held acoustic probe that uses the A-weighted scale. The probe was held at a distance of 0.4 m (~16 in.) at the height of the centerline of the meshing gears, pointed directly at the test gear cover. The RMS-averaged data are shown (Fig. 15). The noise data produced similar trends, as did the vibration data discussed previously.

3. Fatigue. Hybrid gear set A was used for endurance testing as a proof of concept. The gear set used in this test was put into the facility after composite curing and without any re-grinding of the gear tooth profiles. A single long-term test was conducted to examine the durability of the composite-steel bond. The test was run for one billion cycles at 10,000 rpm and 48.6 N·m (430 in·lb) at 49°C (120°F) oil inlet temperature. This corresponds to a calculated contact stress of approximately 1.35 GPa (195 ksi) and a bending stress of 0.20 GPa (29.2 ksi). The gears survived without any visual signs of de-bonding or degradation of the composite material.

Conclusions

Based on the results attained in this study, the following conclusions can be made:

- A hybrid gear mesh has shown promise for operation in a simulated aerospace environment with the possibility of reducing component weight.
- Hybrid gears need to be further processed to maintain aerospace gear quality.
- The hybrid gears were fatigue-tested to one billion stress cycles at representative loads and speeds, with no resulting degradation.
- The hybrid gears exhibited a lower natural frequency than the standard steel gears of the same size and dimension.
- Dynamic vibration results indicated the strong dependence of operational speed on the measured results. The hybrid driving hybrid configuration exhibited the lowest vibrations — but only at the higher speeds and loads.
- Vibration tests performed with the hybrid gear driving the steel gear typically exhibited the highest RMS vibration levels — particularly at the higher rotational speeds.
- The A-weighted, RMS sound-level measurements trended similarly to those of the vibration results. **PTE**

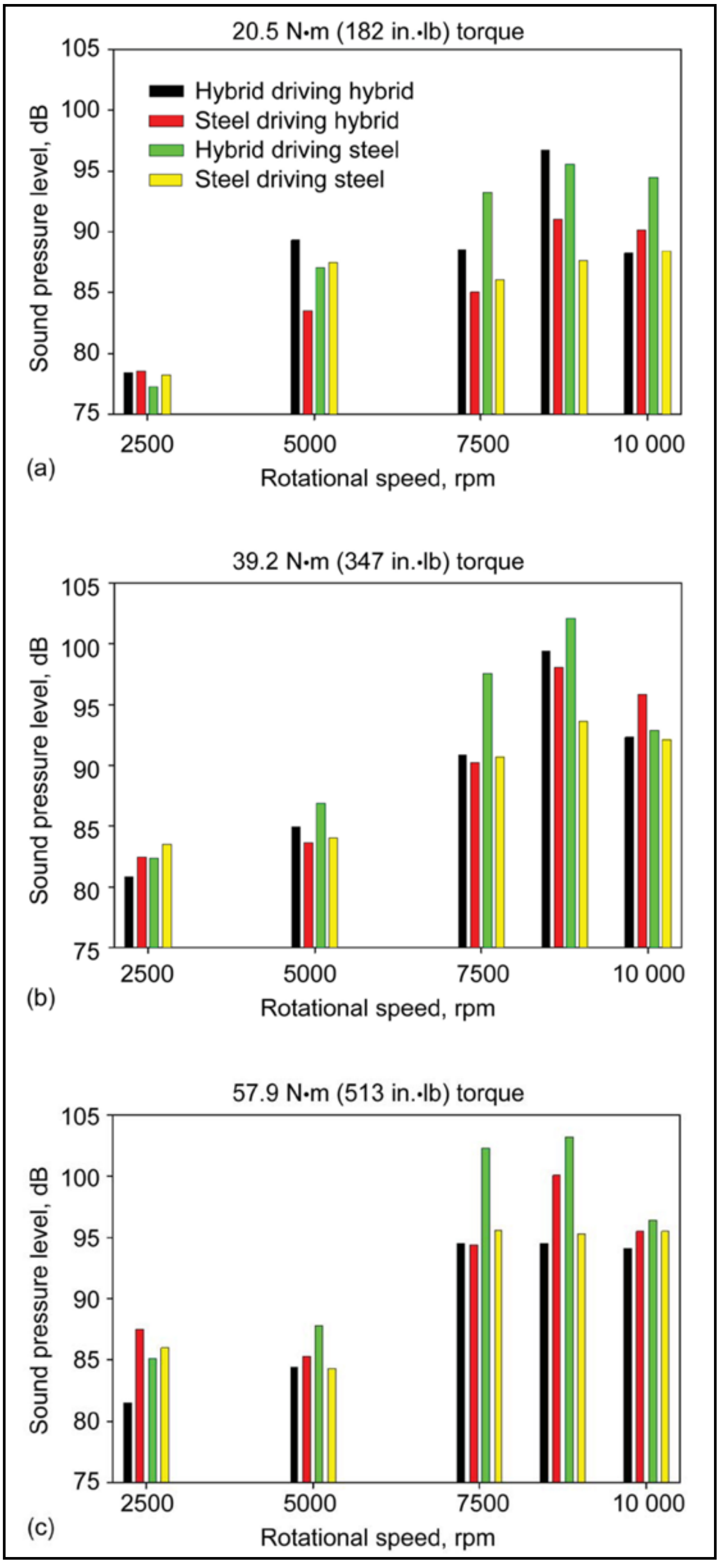

Figure 15 Effect of rotational load and speed on RMS A-weighted sound pressure level for different meshing conditions.

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