Gear Design Effects on the Performance of High Speed Helical Gear Trains As Used In Aerospace Drive Systems

R. Handschuh
robert.f.handschuh@nasa.gov
Research Mechanical Engineer
NASA Glenn Research Center
Cleveland, Ohio, U.S.

C. Kilmain
ckilmain@bh.com
Director of Mechanical Systems
Bell Helicopter Textron Inc.
Fort Worth, Texas, U.S.

R. Ehinger
rehinger@bh.com
Manager IPT Engineering
Bell Helicopter Textron Inc.
Fort Worth, Texas, U.S.

E. Sinusas
esinusas@bh.com
Supervisor IPT Engineering
Bell Helicopter Textron Inc.
Fort Worth, Texas, U.S.

ABSTRACT

The performance of high-speed helical gear trains is of particular importance for tiltrotor aircraft drive systems. These drive systems are used to provide speed reduction / torque multiplication from the gas turbine output shaft and provide the necessary offset between these parallel shafts in the aircraft. Four different design configurations have been tested in the NASA Glenn Research Center, High Speed Helical Gear Train Test Facility. The design configurations included the current aircraft design, current design with isotropic superfinished gear surfaces, double helical design (inward and outward pumping), increased pitch (finer teeth), and an increased helix angle. All designs were tested at multiple input shaft speeds (up to 15,000 rpm) and applied power (up to 5,000 hp). Also two lubrication, system-related, variables were tested: oil inlet temperature (160 to 250 ° F) and lubricating jet pressure (60 to 80 psig). Experimental data recorded from these tests included power loss of the helical system under study, the temperature increase of the lubricant from inlet to outlet of the drive system and fling off temperatures (radially and axially). Also, all gear systems were tested with and without shrouds around the gears. The empirical data resulting from this study will be useful to the design of future helical gear train systems anticipated for next generation rotorcraft drive systems.

INTRODUCTION

Rotorcraft drive systems are critical to the high efficiency and lightweight needed from the propulsion system. Tiltrotor aircraft, as currently designed, utilize the drive system as a means to fly even when one engine is inoperative through the use of shafting and other gearboxes to connect the two rotors together. A sketch of the entire propulsion system is shown in Figure 1 and a close up of the wing tipped nacelle propulsion system is shown in further detail in Figure 2 (Ref. 1).

Also in a tiltrotor aircraft, the entire propulsion system is required to tilt from the vertical position (helicopter mode) to that of the horizontal position (forward flight—airplane mode). The unique capabilities allow this aircraft to fly at a high rate of speed in the airplane mode and land vertically, greatly enhancing the aircraft's usefulness in fulfilling a number of military missions.

The drive system contained within the prop-rotor gearbox connects the parallel shafts of the gas turbine engine to that of the propeller via a gear train of helical gears. The gears in this gear train operate at high rotational speeds that result in high pitch line velocity of the gears that can affect the overall drive system performance through an increase in windage losses. It is of the utmost importance for drive system efficiency to make the transition from the gas turbine engine to the propeller with the minimum amount of power loss. High power loss is absorbed by the lubricant or expelled through the gearbox housing in the form of heat. Therefore improved performance of the gearbox can result in more power available to the rotor, increased load capacity, or extended range.

The objective of this study is to experimentally determine how operating conditions, gear design, and gear shrouding can influence the performance of high-speed helical gear trains as used in a tiltrotor aircraft.

TEST FACILITY, TEST HARDWARE, AND TEST METHOD

Test Facility

The test facility used in this study is the High Speed Helical Gear Train Test Facility located at NASA Glenn Research Center (Ref. 2). The test facility arrangement is shown in Figure 3 and a sketch of the key test system components is shown in Figure 4.
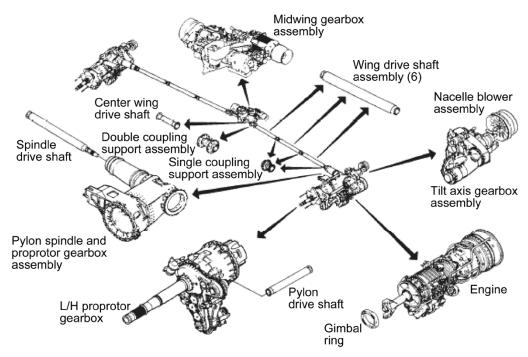
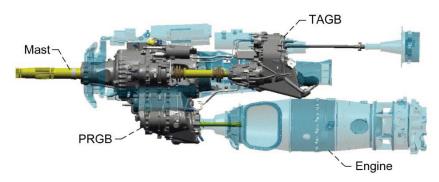**Figure 1: Tilt rotor propulsion system components (Ref. 1).**

Figure 2: Propulsion system components that reside within the aircraft nacelle. Tilt axis gearbox (TAGB), and prop-rotor gearbox (PRGB).
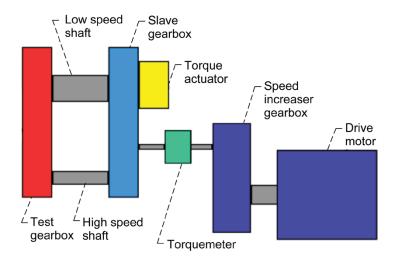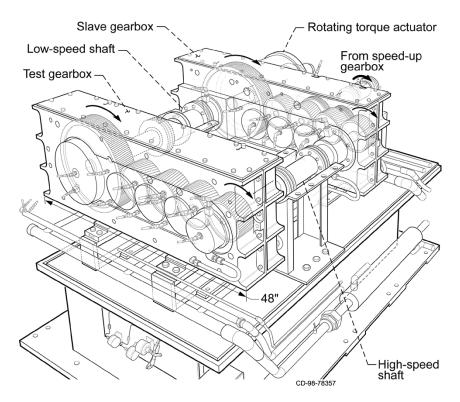

Figure 3: Test facility arrangement.

Figure 4: Sketch of test facility primary components.

Referring to Figure 3 for the discussion, the facility operates as a closed-loop test facility. Power is circulated from the test gearbox to the slave gearbox and then returns to complete the torque loop. A rotating torque actuator in the slave gearbox provides an adjustable loop torque while the drive motor must provide for all of the gear, bearing, and windage losses. With the current components, up to 5,000 hp can be circulated around the test to the slave gearbox loop. The high-speed shaft, simulating the power turbine shaft, can be rotated to 15,000 rpm. Most of the test conditions that will be reported in this paper have to do with the hover and forward flight speed conditions. The drive system input shaft rotates at 15,000 and 12,500 rpm, respectively at these conditions. Both gearboxes have separate lubrication systems that include supply and scavenge pumps, filters (3 μm), heaters, heat exchangers, etc. The test and slave gearboxes operate in a dry sump mode where the lubricant that is jet fed to the gears and bearings is removed immediately after lubricating and cooling via the scavenge pumps. For all the data presented in this report, the slave gearbox operating conditions were constant, at ~160 °F lubricant inlet temperature, and ~80 psi jet pressure.

Test Hardware

The gearing components used in the test program included four different gear designs. There is the baseline (currently used aircraft design), baseline with isotropic superfinishing (ISF), double helical, fine pitch, and increased helix angle designs. The design information for all four cases are provided in Table 1. The same gear material,

Pyrowear 53, was used in all gearing components and all were finished with the same surface finish and gear quality. A photograph of the input gear designs is shown in Figure 5. The gearing components are shown during their installation into the gearbox housing as shown in Figure 6 (baseline design).

Table 1: Basic Gear Design Information.

	Baseline Design	Double Helical Design	Fine Pitch Design	Increased Helix Angle Design
Number of teeth, input and 2nd idler/1st and 3rd idler/ bull gear	50/51/139	50/51/139	70/73/196	50/51/139
Normal module, mm, (diametral pitch, (1/in.))	3.033 (8.375)	2.540 (10.000)	2.142 (11.858)	2.9136 (8.7177)
Face Width, mm (in.)	66.68 (2.625)	78.23 (3.08)	66.68 (2.625)	66.68 (2.625)
Normal pressure angle, deg.	20	20	20	20
Transverse helix angle at pitch diameter, deg.	12	35	12	20

Test Method

The test facility was operated at all conditions long enough to establish steady state conditions. This typically took ~ 5 minutes to attain once the first test condition was reached. An example of this will be presented later in this paper. Data taken was stored remotely for playback if needed. The rate of data acquisition for all tests was 0.5 or 1.0 Hz.

Figure 5: Photograph of input gear designs. Baseline, baseline + ISF, double helical, fine pitch, and increased helix angle.
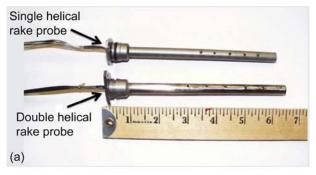

Figure 6: Baseline test hardware during installation.

TEST GEARBOX INSTRUMENTATION

The test facility provided for five operational condition measurements: drive motor power, drive motor speed, test system loop power, lubricant pressure and all the facility temperatures. Drive motor power to the facility is monitored via a commercially available torquemeter. Loop power is measured using a torque-bridge-telemetry system attached to the Bull Gear connect shaft between the test and slave gearboxes. A plethora of thermocouples monitor the lubricant, gearbox housing, bearings, and fling-off temperatures. Fling-off temperatures are found via two different probe types.

Rake and array thermocouple probes were designed and fabricated to indicate the lubricant temperature radially flung off and axially pumped respectively (Ref. 3). The two different probe types are shown in Figure 7. The rake probes had five thermocouples across the face width of the gear (six for double helical design) and the array probe had nine thermocouples in a 25.4- by 25.4-mm (1- by 1-in.) substrate.
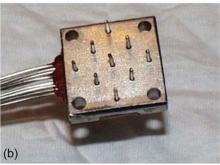

Figure 7: Rake probes (a) and array probe (b).

Both probes were located very close to the mesh position of the gears. The rake and array probes were located within the test gearbox as shown in Figure 8.

TEST DATA

Test Operation

Test operation was conducted in the following manner. First the rotational speed and applied load were established and then the temperatures of the facility were allowed to come to steady state once the oil inlet and outlet temperatures stabilized. An example of a typical time history of a test is shown in Figure 9 for the conditions given in Table 2 (Ref. 4). In the data to be presented, values from all important variables will be presented at a steady state operating point.

Table 2: Conditions for Figure 9. Gears were Baseline design, superfinished, and 160 °F oil inlet temperature.

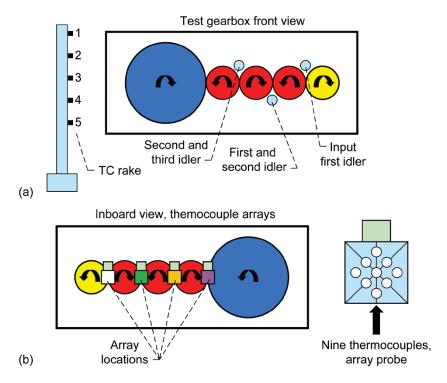| Condition	Input shaft speed (krpm)	Lower power (hp)	Temperature increase across gearbox (°F)	Drive motor power (hp)
A	Warm up			
B	12.5	1,379	50.6	138.0
C	12.5	2,801	55.1	149.2
D	12.5	4,170	59.7	160.1
E	15.0	1,657	73.8	201.9
F	15.0	3,366	79.1	213.2
G	15.0	4,986	83.0	225.1

Figure 8: Locations of the rake (a) and array (b) probes in the test gearbox.
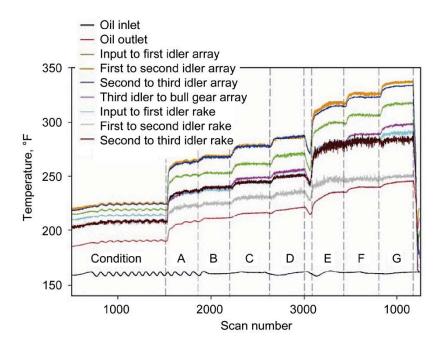

Figure 9: Temperature data at mid-face of rake and at array probe center for all sensor locations (conditions shown in Table 2, one scan = 2 sec, 160 °F oil inlet temperature) super-finished baseline design test results.

Spin Loss Data

In order to understand the drive system losses, experimental tests were conducted to full speed (except in some unshrouded cases) at approximately 10 percent of the full torque of the facility. The data generated was for the various gear designs, with and without shrouding. The test gearbox (top cover removed) with the shrouding installed is shown in Figure 10(a) and shrouding removed in (b).

The results from the shrouded and unshrouded tests are shown in Figure 11(a) and (b), respectively. The shrouded gear tests were run at ~ 160 °F lubricant inlet temperature. The unshrouded gear tests were run for most of the gear designs at two lubricant inlet temperatures (~160 and 200 °F).

The non-linear increase in drive motor power requirement for these tests resembles the windage power loss curves generated in Reference 5. The data plotted in all the curves here, and those to come later with respect to "drive motor power", refer to the entire test system (test and slave gearboxes). As shown in Figure 11(b), for the high helix angle gear design, the facility could not stably run at the 12,500 and 15,000 rpm conditions in the unshrouded case. This is an indication of the windage from the gears interrupting the scavenging of lubricant from the test gearbox.

A comparison of the double helical—outward pumping design, is shown in Figure 12. The data indicates the effect of having the shrouds installed. The drive system power requirement difference is a direct measurement of the shroud effectiveness at a given speed and torque combination. Note that rotational speed change is far more important than the level of applied load, meaning that the windage part of the losses is dominating the drive motor power requirements.
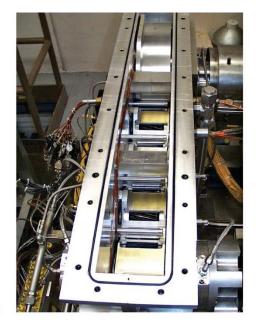
Gear Design Effects

An indication of how the different gear designs affect the performance for the same operating conditions for all four designs will be addressed in this section. In Figure 13 the design effect is shown at 160 °F lubricant inlet temperature and in Figure 14 at 250 °F lubricant inlet temperature as a function of applied bull gear torque at two different rotational speeds. Both figures were for shrouded gears.

It is apparent from these two figures that the rotational speed had the largest effect on the results for a given design. Higher lubricant inlet temperature reduced the power requirement, and the baseline or high helix angle designs produced the largest power requirements. The fine pitch design produced the lowest power requirements for all conditions shown. This result must be tempered with the fact that the test and slave gearboxes had this type of gearing, therefore the power savings of an individual gearbox would

be similar to that of the double helical gear design that were operated in the outward pumping arrangement.

As an example of how the lubricant inlet temperature affects the power loss of a given configuration is shown in Figure 15. In this figure the drive motor power is plotted versus bull gear shaft torque for two input shaft speeds for the baseline design. Higher lubricant inlet temperature reduces the power loss of the gear train at all speed and load conditions. For the baseline design this resulted in a ~10 hp reduction in power loss by increasing the lubricant inlet temperature from 160 to 250 °F.
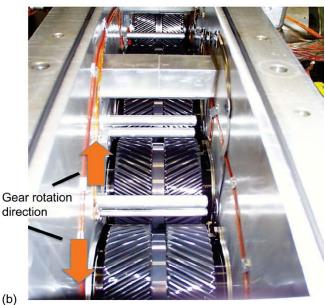

(a)

(b)

Gear rotation direction

Figure 10: Test gearbox with the shrouding installed and removed. (a) Shrouding installed (baseline). (b) Shrouding removed (double helical inward pumping).

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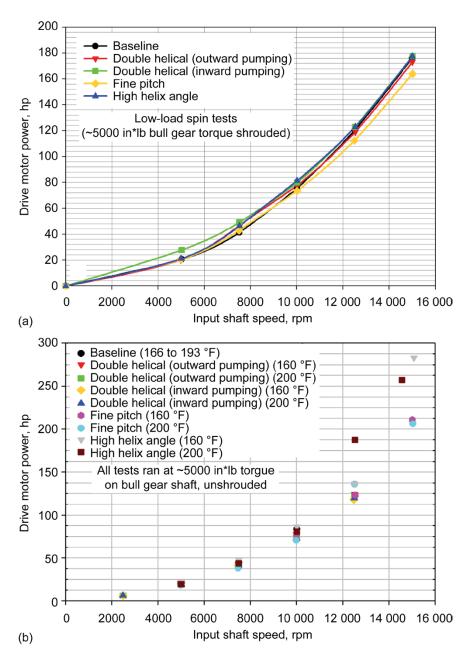
(a)

(b)

Figure 11: Spin loss data for (a) shrouded and (b) unshrouded conditions.
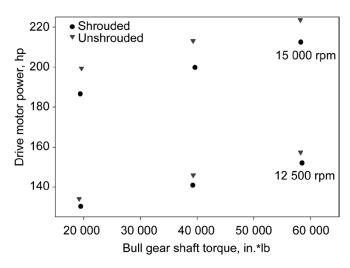
Figure 12: Effect of shrouding on drive motor power. Double helical gears, outward pumping 200 °F oil inlet temperature.
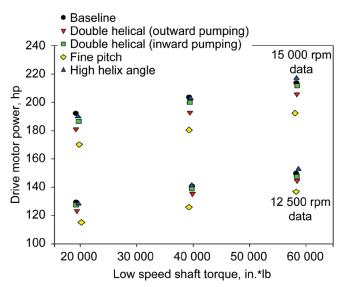

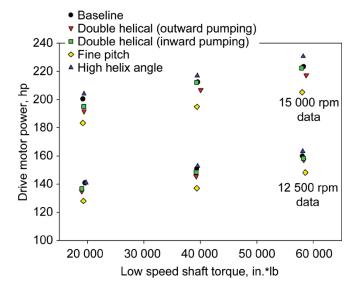**Figure 14: Effect of gear design (shrouded) on drive motor power, 250 °F lubricant inlet temperature.**

Figure 13: Effect of gear design shrouded on drive motor power, 160 °F lubricant inlet temperature.
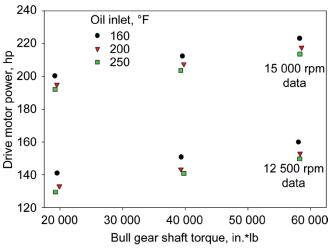

Figure 15: Lubricant inlet temperature effects on drive motor power required (gears shrouded).

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Lubrication Jet Pressure Tests

Three lubricant jet pressure (flow rate) conditions are shown for the two speed conditions and one level of applied load (~33 percent of full torque). The drive motor power required is shown in Figure 16 and the lubricant temperature increase across the gearbox (exit temperature minus inlet temperature) is shown in Figure 17. In either figure the lower symbol for a given design is the 12,500 rpm data and upper symbol is the 15,000 rpm data. As would be expected, higher flow rate of lubricant reduced the temperature change, but higher jet pressure (flow) increase the power loss for all conditions. The fine pitch design had the lowest drive motor required (note fine pitch gears were installed in both the test and slave gearbox) and minimum temperature increase for any rotational speed or loop torque requirement.

Internal Gearbox Instrumentation

The final comparison to be made in this paper is how the instrumentation inside the gearbox, fling off temperatures, were affected by gear designs and operation conditions. As described earlier, rake probes (radial) and array probes (axially) will be used to generate the data discussed here. The data presented was the maximum from any of the three rake probes or the four array thermocouple sensors. Generally speaking, the highest temperature locations were those at the idler gear positions.

An example of the rake probe data is shown in Figure 18. The lubricant inlet temperature for this data was 200 °F. Six different test configurations are shown all with the gears shrouded. As with all the other data presented in this study, the fine pitch gear design performed the best and rotational speed was a larger factor than applied load on the results. The fling off temperature from the rake probes could be in excess of 125 °F higher than the lubricant inlet temperature.

An example of the array probe data is shown in Figure 19. This data was also taken at the same inlet lubricant temperatures as the data from Figure 18. This data requires a little more explanation than the rake data. The array probe data is influenced by the axially pumped air-lubricant mixture due to the helical gear meshing action. For the single helical gear designs, the air-lubricant mixture expended from the ends of the teeth impinge directly on the array sensor. Therefore the single helical gear design data is clustered at a higher temperature than either of the double helical results. The outward pumping helical gears have approximately one-half of the face width before the air lubricant mixture impinges on the array sensor. The distance that the air-lubricant mixture is pumped in a single helical gear is the complete face width. Therefore single helical results would be expected to have a higher axially pumped measured temperature.
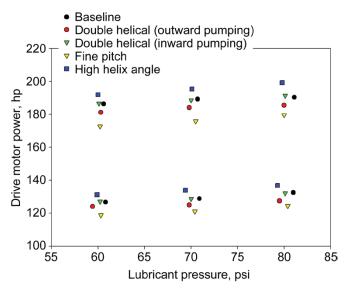

Figure 16: Lubricant pressure effects on drive motor power requirements. Shrouded gears, ~19,000 in.*lb torque, 200 °F oil inlet temperature.
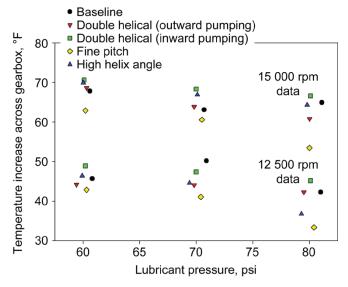

Figure 17: Lubricant pressure effects on temperature increase across the test gearbox, 200 °F oil inlet temperature.
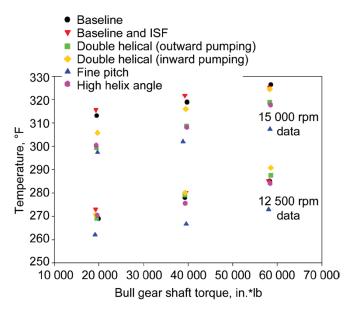
Figure 18: Rake probe lubricant fling-off data, maximum for a given speed and load condition, 200 °F oil inlet temperature.
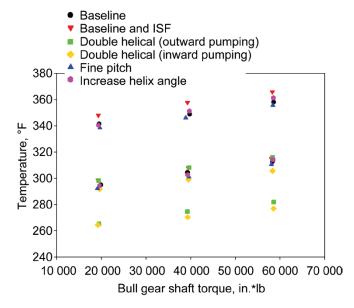

Figure 19: Array probe fling-off data, maximum values for a given speed and load, 200 °F oil inlet temperature. (Note: Data at higher symbol is at 15,000 rpm and lower symbol 12,500 rpm.)

CONCLUSIONS

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

1. High-speed gearing benefits from the use of shrouding when the pitch line velocity exceeds ~ 15,000 ft/min. At conditions above this pitch line speed, the windage losses can dominate those from other sources (gear meshing and bearing losses).
2. Gear design characteristics can also impact the drive system power losses. For the tests conducted in this study, the fine pitch gear design had the least power loss and lowest temperature increase of the lubricant across the gearbox.
3. Lubricant inlet temperature changes indicated that higher inlet temperature required less drive motor power for identical conditions for all designs.
4. Lubricant jet pressure (flow) affects the power loss and temperature change from the inlet to exit of the gearbox. Lower flow resulted in less power required, but resulted in an increase in temperature across the gearbox.
5. Special rake and array probes indicated that the temperature of the lubricant that is flung off radially and pumped axially far exceeds the bulk flow temperature exiting the gearbox. The temperature rise can exceed 125 °F radially (rake probe) and 165 °F axially (array probe) depending on the speed, load, and other conditions applied.

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