



Experimental Study of the Influence of Speed and Load on Thermal Behavior of High-Speed Helical Gear Trains

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An experimental effort has been conducted on an aerospace-quality helical gear train to investigate the thermal behavior of the gear system as speed, load, and lubricant flow rate were varied. Temperature test data from a helical gear train at varying speeds and loads (to 5000 hp and 15000 rpm) was collected using thermocouple rakes and axial arrays. The instrumentation was able to capture the radial and axial expelled lubricant-air environment (fling-off lubricant) that is expelled during the gear meshing process. Effects of operational characteristics are presented.

Introduction and Background

High speed, heavily loaded, and lightweight gearing components are common-place in rotorcraft systems. These systems are expected to deliver high power from the gas turbine engines to the high-torque/low-speed rotor with reduction ratios in the range of 25:1 to 100:1 (refs. 1 to 14). Gearing systems in these extreme duty applications can also have thermal behavior issues due to the high pitch line velocities. While design considerations for bending and contact capacities are usually considered initially, high speed gearing design needs to carefully consider the consequences of pitch line velocities approaching 25000 feet per minute.

The thermal behavior characteristics of mechanical components is the least understood and has received the least amount of attention in the open literature (refs. 15 to 19). The thermal behavior of a system can cause a success from a load capacity (bending and contact stress) viewpoint into a failure from the resultant thermal induced failure (high operational temperatures, gear tooth scoring, and high drive system losses).

In rotorcraft drive systems, such as that of tiltrotors (fig. 1), a helical gear train is used to separate the parallel engine and rotor shafting on the aircraft. Therefore, the drive system is not only needed to provide the necessary reduction between the engine and rotor, but also has to make the system operate in emergency conditions (refs. 20 and 21) such as one engine inoperative (fig. 2) (ref. 22).

For the rotorcraft of interest in this study, the gearing between the engine and the rotor is composed of a series of helical gears (or a gear train is utilized). This

part of the drive system operates at very high rotational speed and carries the full power of the engine during normal operation and must operate at high load if the other engine fails.

In this type of arrangement the idler gears receive two thermal cycles per revolution. Since these gears have two thermal cycles per revolution and are extremely light-weight (low heat carrying capacity) the successful operation of the system in all possible normal and emergency conditions can be difficult.

The objective of this paper is to present the effects of speed, load, and lubricant jet pressure (flow) of the on the operating performance (power loss) and resultant fling-off temperatures. An enhanced measurement capability will be presented in this study. A high-speed helical drive train facility that utilizes full scale, aerospace quality components was used to generate the data presented in this study. The system can operate to 15000 rpm (to simulate the engine input rotational speed) and at power levels to 5000 hp.

Test Facility, Test Hardware, Data Acquisition, and Test Procedure

Test Facility

The test facility used for this study is shown in figure 3 (ref. 23). The facility is a closed-loop, torque-regenerative testing system. There is a test gearbox and slave gearbox that are basically mirror images of each

other. Each gearbox has an input gear, three idlers, and one bull gear. The gearboxes are joined together through the input gears and bull gears via shafting. The facility is powered by a 500 hp DC drive motor and its output speed is increased using a speed-increasing gearbox. The output of the speed-increasing gearbox then passes through a torque and speed sensor before connecting to the slave gearbox. The entire test stand configuration is shown in figure 4. Each gearbox has separate supply and scavenge pumps and reservoirs. Lubrication system flow rate is controlled using the supply pressure. Temperature is controlled via immersion heaters in the reservoir and heat exchangers that cool the lubricant returned from the gearboxes. Each lubrication system has a very fine 3-micron filtration. Nominal flow rate into the test or slave gearboxes at 80 psi is approximately 15 gpm.

The lubricant used in the tests to be described was a synthetic turbine engine lubricant (DoD-PRF-85734). This lubricant is used in gas turbine engines as well as the drive systems for rotorcraft.

Test Instrumentation

The test instrumentation used in this study beyond those of reference 24 included thermocouple rakes for locations across the face width and thermocouple arrays at the exit region from the helical gear axial pumping location. The test instrumentation measured the fling-off lubricant from the gears in the radial and axial directions. Locations of the two different measurements are shown in figure 5 (locations of rake and array probes). Shown in figures 6 and 7 are photographs of the instrumentation rakes and arrays respectively. The rake probes had 5 thermocouples across the face width and the array sensors had 9 thermocouples distributed as shown in figure 7. The thermocouple rakes were located at three positions as close as possible to the location of oil being flung radially out of mesh. The thermocouple arrays were centered at the axial point where the pitch diameter of the meshing gears meet.

Test Hardware

The test hardware used in the tests to be described is aerospace quality hardware. The basic gear design information is contained in table 1. The input and bull gear shafts have ball bearings to contain the resultant thrust loads whereas the idler gears only have roller bearings. The partially disassembled test gearbox is shown in figure 8. The bearing inner race is integral to the shafts on the idler gears and at other radially-loaded bearings on the input and bull gear shafts.

Shrouds for the gears were used to minimize the windage losses that high-speed gear systems possess but were partially or totally removed for some of the tests conducted. Figure 9 shows the shrouds installed in the test gearbox.

Data Acquisition

The test facility data system monitors three important facility parameters during operation. Speed, torque (supplied torque and loop torque), and temperature measurements were made during all the testing conducted. The test system loop torque is measured on the shaft connecting the bull gears from the test and slave gearboxes. A telemetry system was utilized in this location.

The data recording system used in this study has the capability of taking data from all parameters at a rate of one sample per second. The data is displayed to the test operator in real time. Data is stored in a spreadsheet format and each sensor can be viewed at any time during a test and when post processing the results.

Test Operation

The test procedure that was followed for collecting the data to be presented was the following. For a given set of conditions (speed, load, lubricant pressure and lubricant oil inlet temperature) the facility was operated for at least 5 minutes or until the temperatures of interest had stabilized.

Results and Comparisons

In high-speed gear trains power losses are due to many mechanisms. These mechanisms include the gear sliding and rolling losses, bearings, and seals. Also of great concern when pitch line velocities are high are the effects of the windage losses (due to the gearing members doing work on the ambient air-oil environment). In low speed systems, this last effect is not an issue but as pitch line velocity (size of the gearing and rotational speed) become high, this mechanism can be a substantial portion of the total losses. In some recent studies (refs. 24 to 26), these effects have been analyzed and experimentally quantified using direct effect tests that have been conducted. In figures 10 and 11, the effect of jet pressure and shaft speed is shown. Varying speed had the largest effect on the temperature measured. Varying the jet pressure had a much smaller effect on temperature change.

Fling-Off Data

As mentioned earlier, tests were run at a condition until steady state was achieved. A typical test is shown in figure 10. In this figure a series of tests at specific set points were conducted. The data for several key locations is shown during the test that lasted in excess of one hour. The test conditions for the data shown in figure 10 are contained in table 2. In this test, the speed and flow conditions were changed while keeping the load on the test gearbox nearly constant. The axial temperatures were from the center array thermocouple and the radial temperatures were from the mid-face width thermocouple position. A close up of part of the data is shown in figure 11. From this figure the typical relationship between the axial and radial fling-off temperatures from the mid face width and mid array locations are indicated. The axial fling-off temperatures were always higher for the idler-idler gear locations. The bull gear axial temperatures were always the lowest as would be expected given the size (heat capacity) of this component. The oil out temperature could be nearly 100 °F less than the highest fling-off temperature measured.

Shroud-Lubricant Jet Pressure Effects on Operational Performance

The effects of shrouding arrangement and lubricant jet pressure will now be described on the temperature increase in the lubricant and power loss on the gearing system. The test data to be presented were at two speeds for the test system (at 12500 and 15000 rpm) yet the load on the system was run at similar loading conditions.

For either measured parameter, temperature increase across the test gearbox or amount of power to drive the entire facility, are effected by the shrouding. In figure 12 the effect of shrouding condition and lubricant supply pressure (changes the flow rate of lubricant) are shown. At either speed condition the full shroud case was the best over all speed and load conditions except when the flow was reduce to approximately 60 psi. Using just the bottom half of the shrouds resulted in high temperature increase across the test gearbox. The no-shroud condition could not be conducted at 15000 rpm due to operational instability thought to be due to the lubricant-bull gear interaction.

A similar trend is shown in figure 13 for amount of power supplied to the entire test facility for shroud, bottom half shroud and no shroud conditions. Full shrouded gears require the least amount of drive motor power for the tests conducted.

Rake Data

An example of the data taken is shown in figure 14 for the 1st and 2nd idler location. The thermocouple numbering shown in figure 5(a) is used to describe where the temperature measurement was made. The effect of loop torque at two different speeds is shown. The highest temperatures across the face width position tended to be at the exit location. The maximum temperature found at this location for the conditions presented was on the order of 300 °F. A change in speed from 12500 to 15000 rpm constant torque caused an increase in temperature as high as 25 to 30 °F.

Array Data

An example of some of the data taken from the axial pumped lubricant-air environment for six conditions is shown in figure 15, and further described in table 3. The temperature measured for six different conditions for the thermocouple array at the 2nd and 3rd idler location is shown. The symbols shown in figure 15 are related to their location on the array sensor. Speed-load conditions I to III are for 12500 rpm and 3 load conditions (19000, 39000, and 58000 in•lb bull gear shaft torque). Conditions IV to VI are for 15000 rpm and the same three increasing load levels. Lubricant pressure was approximately 80 psi and oil inlet temperature of approximately 200 °F. The centered thermocouples in the radial direction indicated the highest temperatures at this array position. From the data shown the temperature could vary up to 40 °F over this small measurement region.

Summary and Conclusions

The thermal behavior of high-speed helical gear trains for various conditions has been presented. The facility utilizes aerospace - quality components and operates at pitch line velocities up to 25000 ft/min and at power levels to 3750 kW (5000 hp). Tests were conducted at high speed and loads with data taken using temperature rakes and arrays to measure the lubricant fling off temperatures. The following results were attained:

1. Speed and load affected lubricant fling off temperatures measured across the gear mesh face width and at the axial location due to the helical gear mesh axial pumping. Changing speed from 12500 to 15000 rpm had a more dramatic effect than increasing load from 30 to 100 percent.

2. Reducing the lubricant jet pressure from 80 to 60 psi reduced the power necessary to drive the facility, but the effect was rather small (approximately 5 hp) and caused the lubricant temperature difference between inlet and exit to increase up to 10 °F.
3. Shrouding for this particular facility produces the best results for the experiments conducted. Anything less than full shrouding of this gear train caused higher power loss and increase temperature difference between the inlet and exit lubricant temperature.

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TABLE 1.—BASIC GEAR DESIGN DATA

Number of teeth input and 2nd idler	50
1st and 3rd idler	51
Bull gear	139
Module, mm (diametral pitch (1/in.))	3.033 (8.375)
Face width, mm (in.)	67.2 (2.625)
Helix angle, degree	12
Gear material	Pyrowear EX-53

TABLE 2.—CONDITIONS FOR FIGURE 10 DATA. TESTS WERE RUN WITH THE OUTPUT TORQUE EQUAL TO 19000 IN•LB, 200 °F LUBRICANT INLET TEMPERATURE

Condition	Input shaft speed, rpm	Jet pressure, psi
A	START UP	81
B	12500	81
C	15000	81
D	12500	71
E	15000	71
F	12500	61
G	15000	61
H	15000	81

TABLE 3.—CONDITIONS FOR FIGURE 15 DATA. JET PRESSURE AND OIL INLET TEMPERATURE HELD CONSTANT AT 80 PSI AND APPROXIMATELY 200 °F, RESPECTIVELY

Condition	Input shaft speed, rpm	Loop power, hp	Drive motor power, hp
I	12500	1422	133
II	12500	2809	143
III	12500	4169	153
IV	15000	1672	195
V	15000	3420	208
VI	15000	5021	218



Figure 1.—Tiltrotor aircraft.

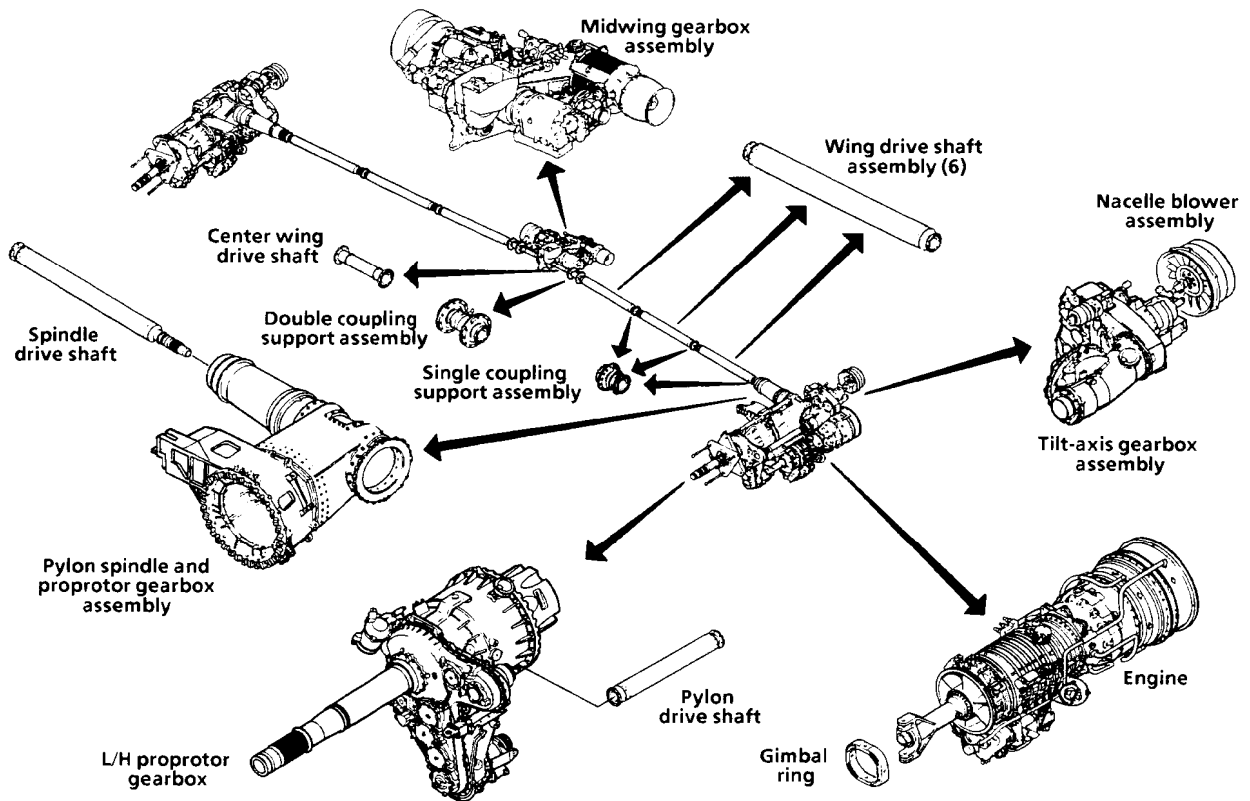


Figure 2.—Typical tiltrotor aircraft propulsion system arrangement.

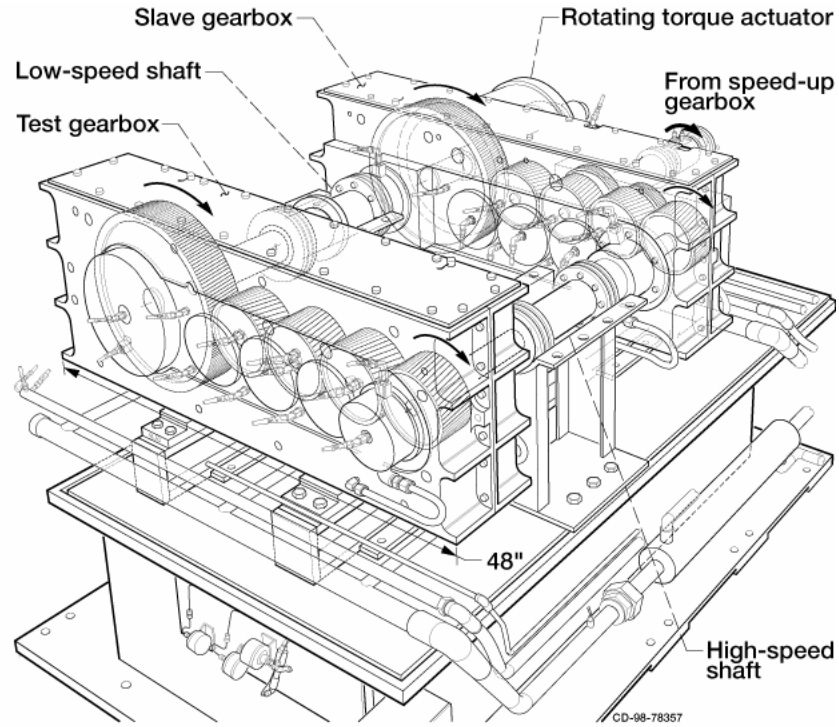


Figure 3.—NASA High-Speed Helical Gear Train Test Facility.

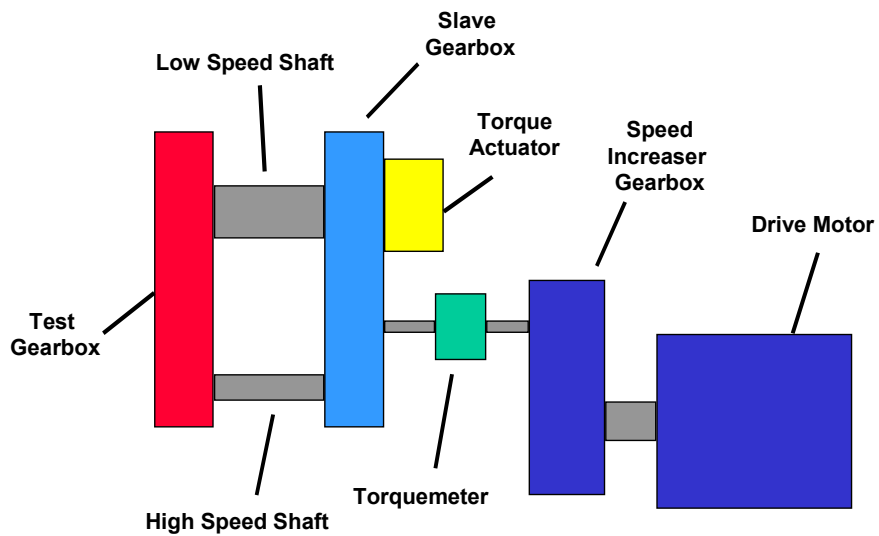


Figure 4.—Layout of NASA High-Speed Helical Gear Train Test Facility.

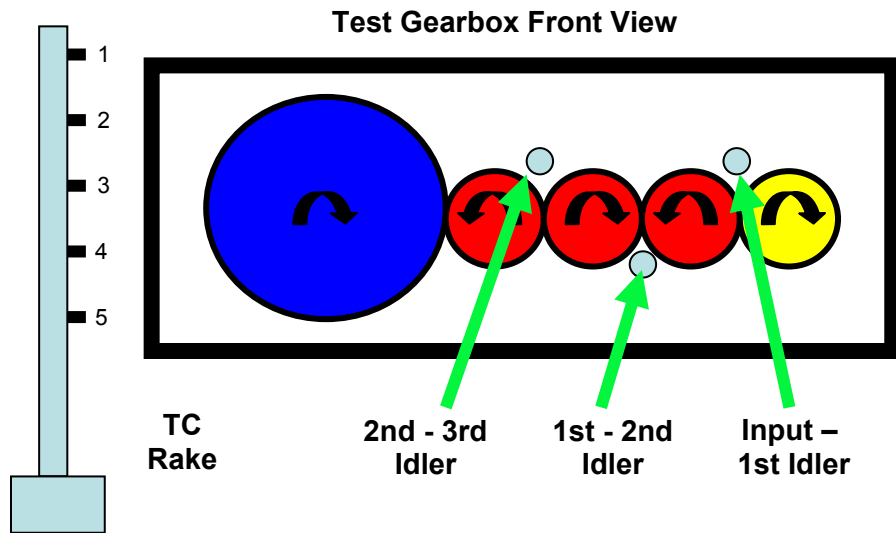


Figure 5(a).—Location and orientation of thermocouple rakes in the test gearbox.

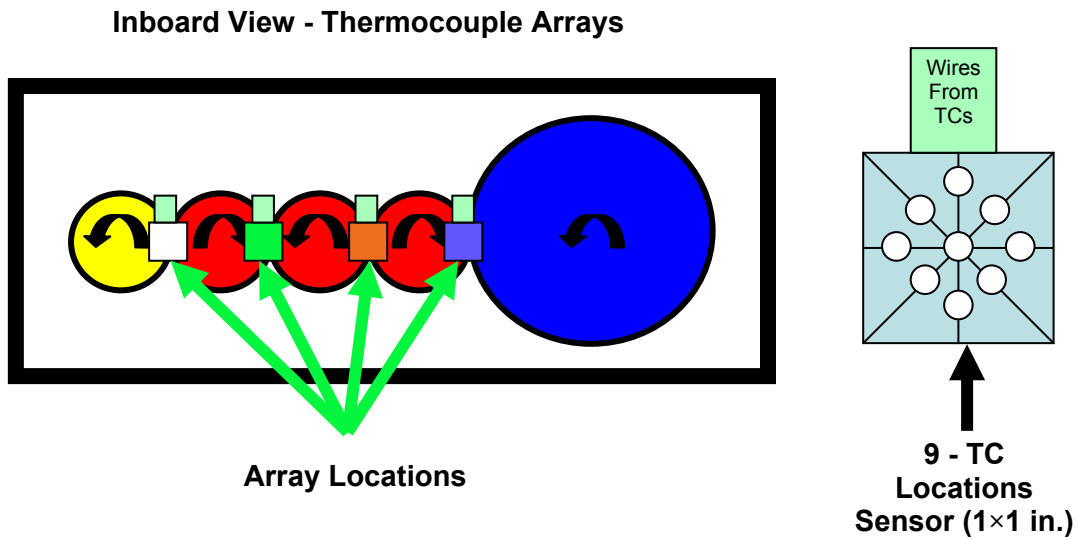


Figure 5(b).—Location and orientation of thermocouple arrays in the test gearbox.



Figure 6.—Photograph of thermocouple rake used in these tests.

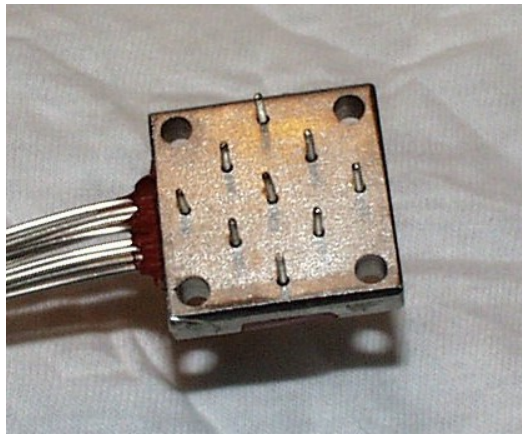


Figure 7.—Photograph of thermocouple array used in these tests.



Figure 8.—NASA High-Speed Helical Gear Train Test Facility components.

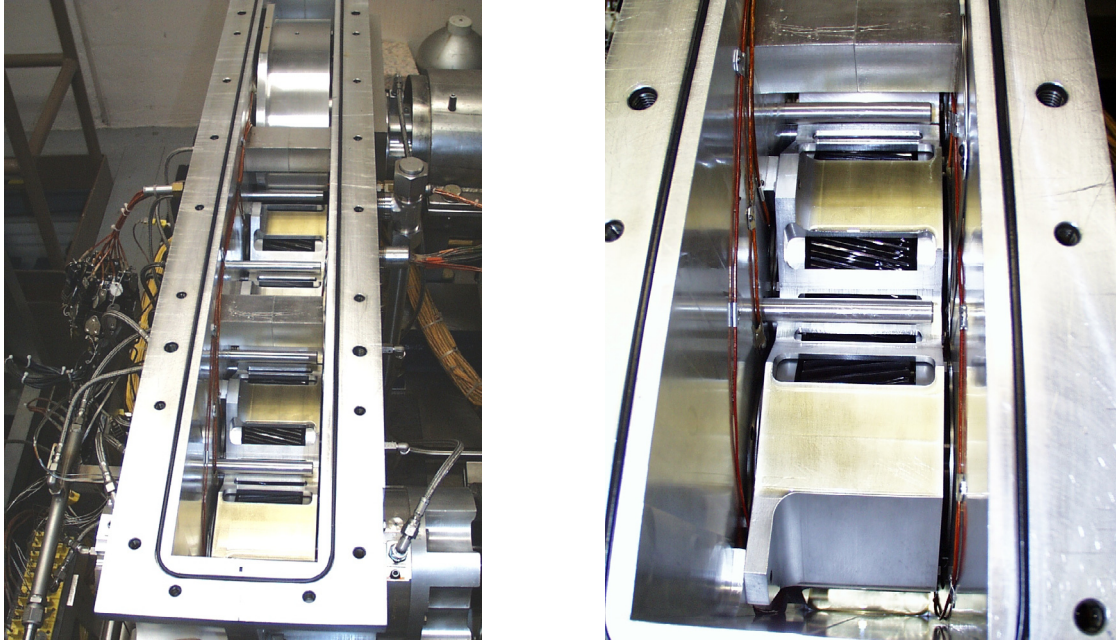


Figure 9.—Photographs of shrouding on the test gearbox (left entire gearbox and right photograph is a close up of the input-1st idler gear shrouds).

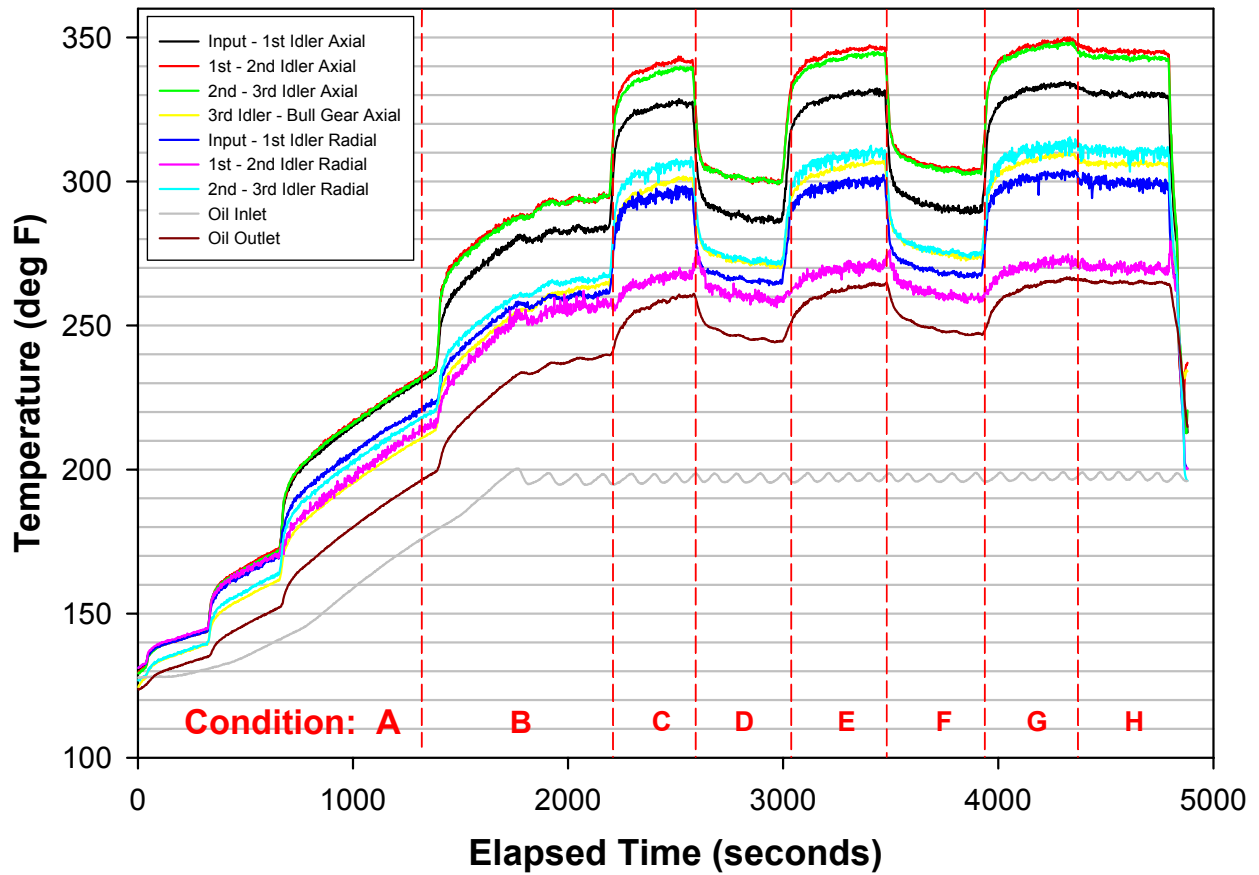


Figure 10.—Effect of operating condition on temperature.

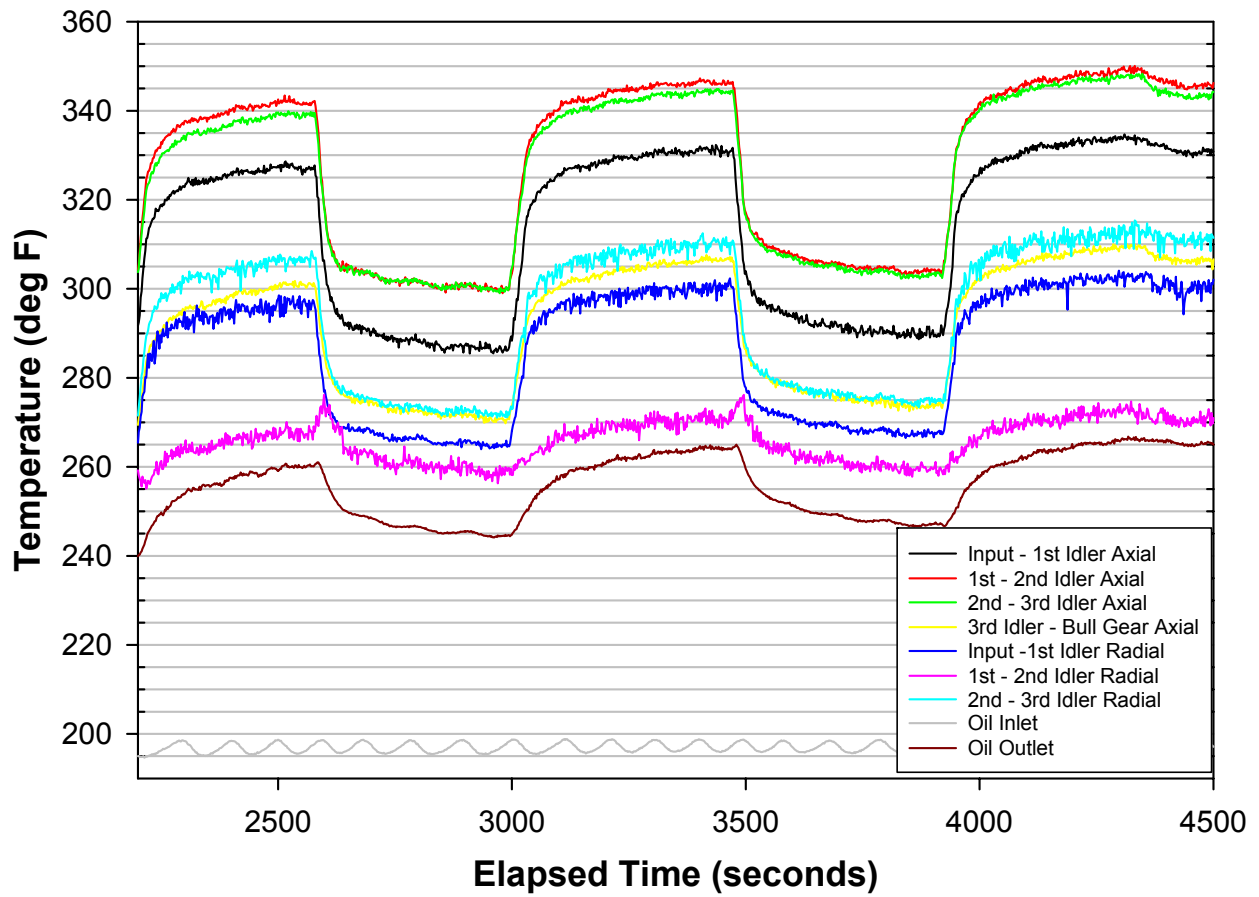


Figure 11.—Effect of operating condition on temperature (part of fig. 10).

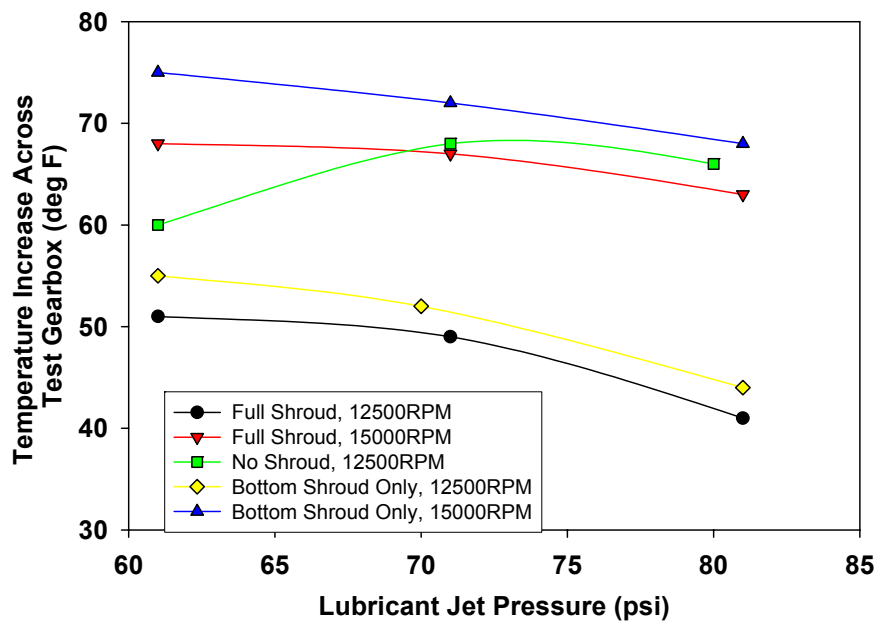


Figure 12.—Effect of lubricant jet pressure, input shaft speed, and shrouding on temperature increase across the test gearbox.

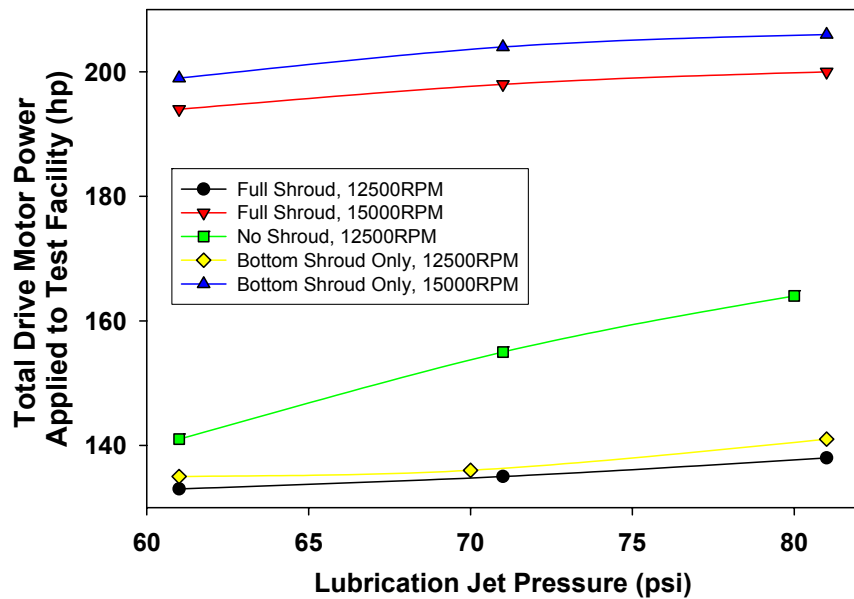


Figure 13.—Effect of lubricant jet pressure, input shaft speed, and shrouding on power to drive the test facility.

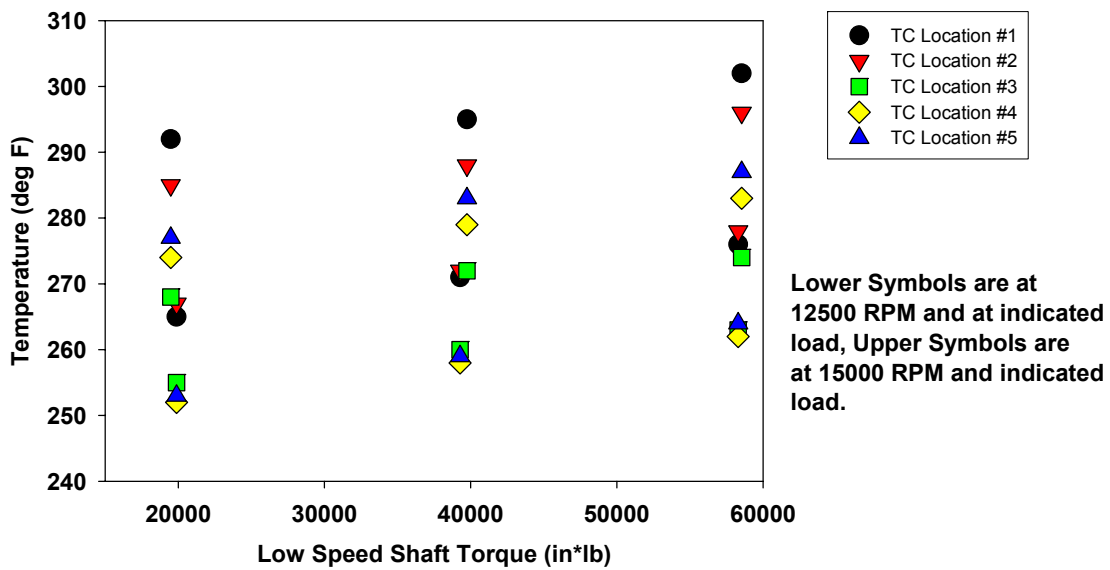


Figure 14.—1st and 2nd idler gears axial rake data, effect of speed on operating temperature, oil inlet temperature approximately 200 °F, 80 psi lubricant jet pressure.

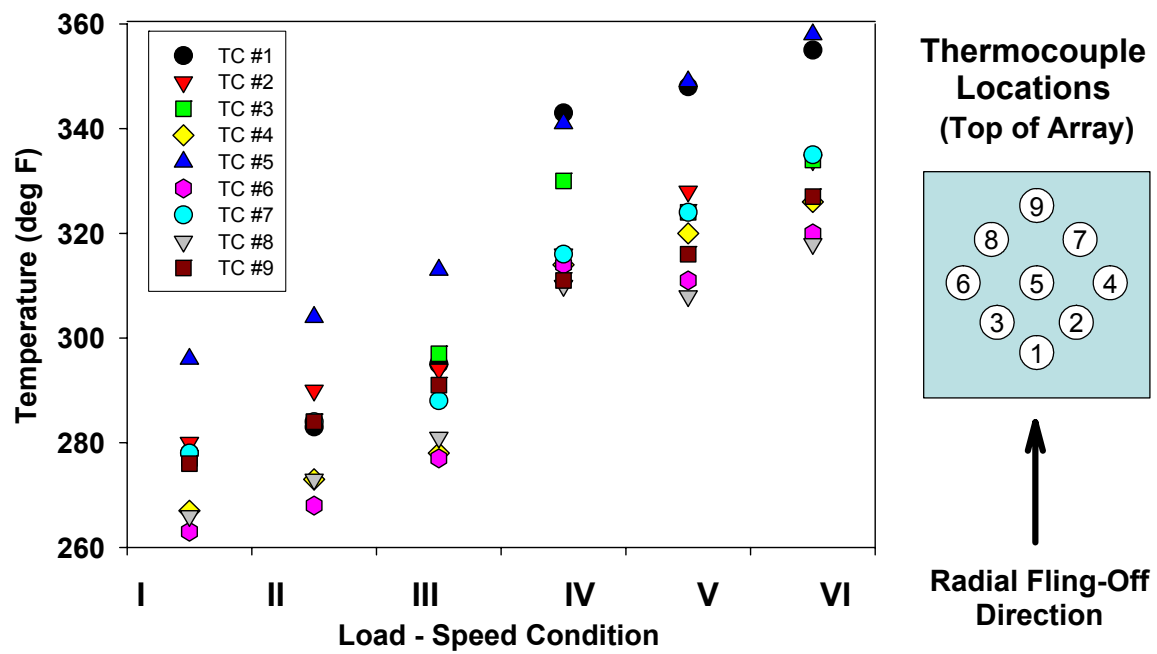


Figure 15.—2nd and 3rd idler array sensor data as a function of load condition.

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