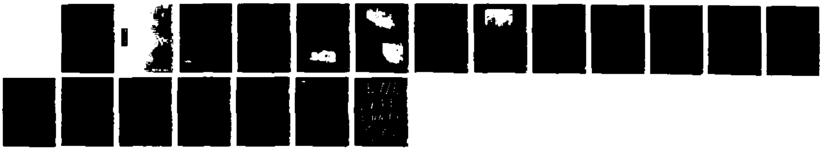
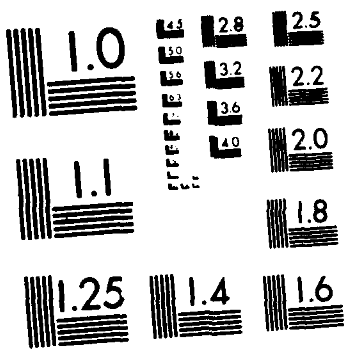
AD-A191 004 EFFICIENCY TESTING OF A HELICOPTER TRANSMISSION 1/1
PLANETARY REDUCTION STAGE..(U) NATIONAL AERONAUTICS AND
SPACE ADMINISTRATION CLEVELAND OH LE..
UNCLASSIFIED R F HANDSCHUH ET AL. FEB 88 NASA-E-3770 F/G 13/9 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963 A

NASA Technical Paper 2795

AVSCOM Technical Report 87-C-28

1988

Efficiency Testing of a Helicopter Transmission Planetary Reduction Stage

Robert F. Handschuh
Propulsion Directorate
USAARTA-AVSCOM
Lewis Research Center
Cleveland, Ohio

Douglas A. Rohn
Lewis Research Center
Cleveland, Ohio

NASA
National Aeronautics and Space Administration

Scientific and Technical Information Division

Summary

A parametric study was performed on a helicopter transmission planetary stage with a four-planet configuration. Isolating the planetary stage components allowed their effect on overall system efficiency to be assessed. Two planetary stages were used in a back-to-back, test-and-slave arrangement with separate lubrication systems. A total of 130 different conditions were tested. Test parameters included speeds to 1622 rpm and torque to 1840 N m (16 300 in. lbf). Two gear lubricants were used, with total flow rates per planetary stage from 100 to 190 cm^3/sec (1.6 to 3.0 gal/min) and oil-inlet temperatures from 60 to 99 °C (140 to 210 °F). Experimentally measured efficiency over all the test variables ranged from 99.44 to 99.75 percent.

Analytical performance predictions were made and compared with measured results. The analytical results were attained by modeling the test hardware and conditions. Computer programs that predict power loss for external gears, internal gears, and planet bearings were used. The analytical results predicted higher efficiencies than were measured experimentally.

Comparisons were made with the results of other investigators. The differences in results could be attributed to differences in the test conditions, the number of components, and the type of gearing.

Introduction

The effects of operating conditions and design features on the performance of gears, bearings, and complete transmissions have been studied by many investigators (refs. 1 to 12). In references 1 to 5 performance characteristics are experimentally and analytically determined for complete helicopter and turboprop gearboxes. References 6 to 8 present analytical procedures for determining the effects of load, nonstandard spur gears, high contact ratio, and lubricant churning on performance. The study described in reference 9 demonstrates how a special lubricant, for transmission use only, can benefit transmission performance and reliability. The type of gearing can also affect the performance that can be attained (refs. 10 and 11). The proper measurement system, as described in reference 12, is essential to attaining accurate test results. These analytical and experimental studies have focused on the operating conditions, the lubricants used, and even the effects of gear tooth geometry.

Helicopter transmissions are extremely efficient. Efficiency is typically above 95 percent for the complete helicopter transmission—220 to 2200 kW (300 to 3000 hp) depending on the operating parameters (speed, load, and operating temperature). However, slight changes can affect the complete transmission system. Changes in transmission operating parameters (e.g., the amount of lubricant needed or the operating temperature) can affect oil cooler size and thus transmission weight. Improving the efficiency will lower fuel usage and thus increase payload or aircraft range.

When testing a complete flight-ready helicopter transmission the conditions that can be imposed on the planetary gear train are limited. The design and operating conditions that can be reached by the entire transmission system dictate the range of parameters that the planetary system experiences. A test rig constructed for the test of a flight-ready transmission does not allow studies of the individual assemblies.

The objective of the present research was to parametrically study the effects of speed, load, lubricant type, flow rate, and oil-inlet temperature on the performance of a planetary section from a helicopter transmission. Isolating the planetary section allowed direct control over imposed operating conditions and more accurate efficiency measurements without interference from other components. To accomplish this objective, the planetary section from a U.S. Army OH-58 helicopter was used. These planetary components were tested to a maximum of 310 kW (420 hp). The rig was operated in a back-to-back, power-regenerative, closed-loop configuration. A total of 130 different conditions were tested during this parametric study. The results showed that the experimentally measured efficiency could range from 99.44 to 99.75 percent depending on the test parameters. These experimental results were compared with other experimental and computational results.

Apparatus and Procedure

Test Rig, Instrumentation, and Data Acquisition System

The test rig contained two planetary sections that were driven back to back; each section had four planets. From the description of the test hardware in table I the nonstandard design of the ring gear is evident. This ring gear was modified by the drop tooth method (ref. 13). The components used in

TABLE I.—PLANETARY GEAR TRAIN HARDWARE USED IN TEST AND SLAVE SECTIONS

[Planetary reduction ratio, 4.667. Planet bearing data (double row cylindrical): inside diameter, 46.5 mm (1.83 in.); outside diameter, 68.5 mm (2.697 in.); roller diameter; 11.0 mm (0.433 in.).]

Gear	Number of teeth	Module, mm	Diametral pitch, in.$^{-1}$	Pressure angle, deg	Pitch diameter	
					mm	in.
Sun	27	2.868	8.857	24.6	77.4	3.048
Planet	35	2.868	8.857	24.6	100.4	3.952
Ring[a]	99	2.778	9.143	20.2	275.0	10.828

[a]Drop tooth design.

these tests were from the U.S. Army's OH-58 helicopter transmission (fig. 1). This transmission has a reduction ratio of 17.44 overall, with the planetary stage contributing a ratio of 4.67. The planet gear carrier and sun gear used in the test program are shown in (fig. 2). The carrier contained the planet gears, the planet bearings, the planet bearing posts, and the mechanical connection to the output shaft.

The assembled rig (fig. 3) contained lubrication systems, manifolds, and individual oil jet lines. During the testing phase the facility and associated lubrication systems were insulated. This allowed test temperatures to be attained in a reasonable amount of time and allowed the rig to operate in a nearly adiabatic environment. The rig was surrounded by an insulated enclosure (fig. 4) for all tests conducted.

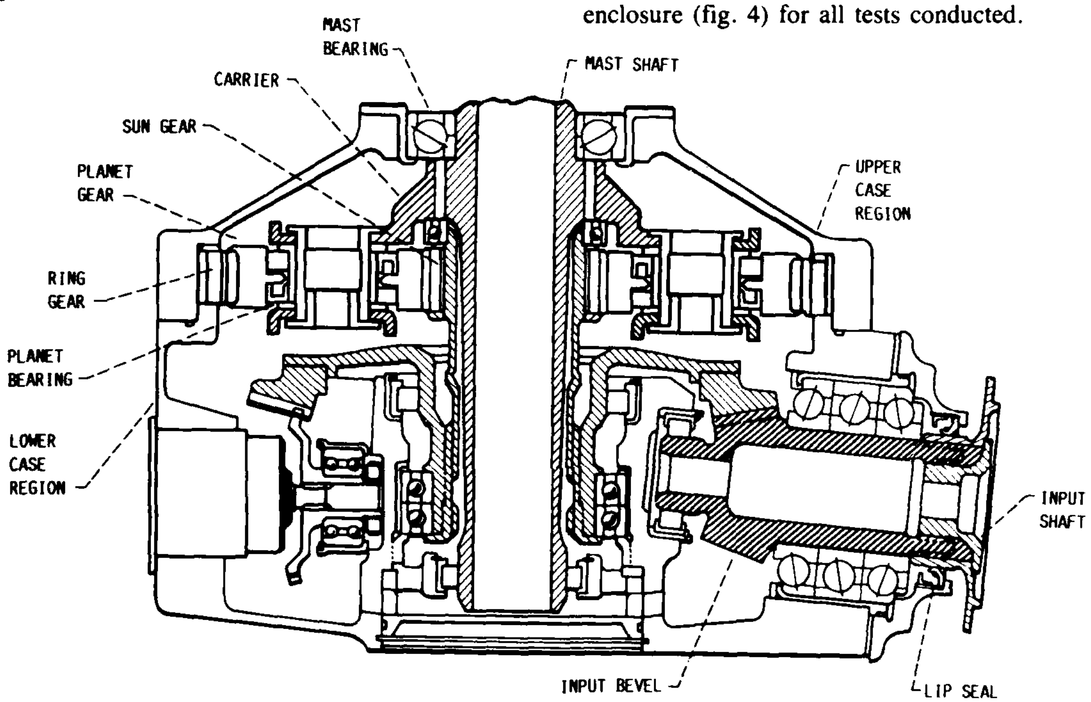


Figure 1.—Cross-sectional view showing four-planet arrangement in the OH-58 helicopter transmission.

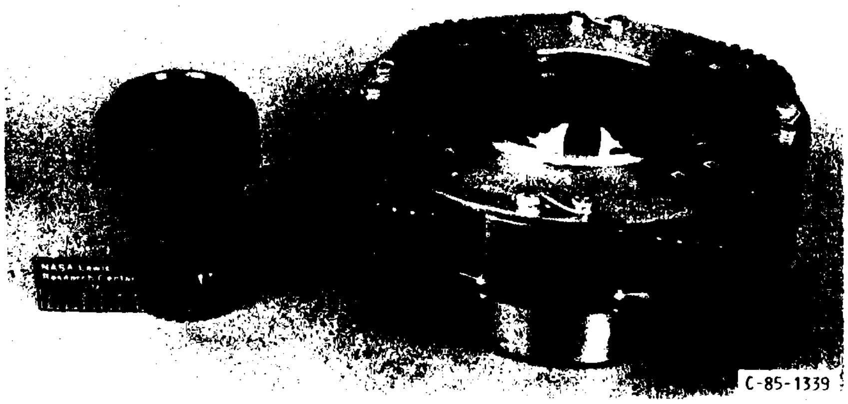


Figure 2.—Sun gear, planet gears, and planet gear carrier used in test facility.

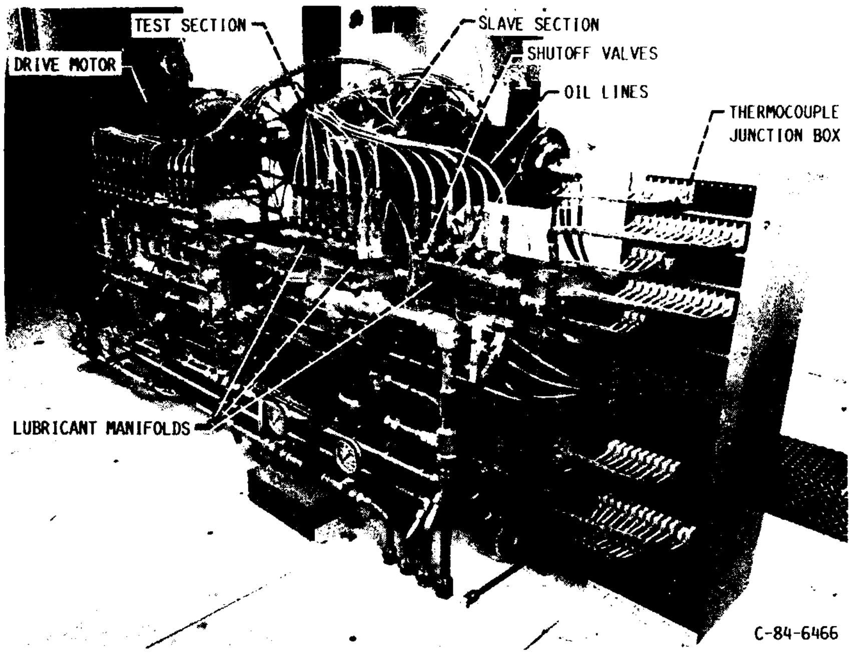


Figure 3.—Test facility showing lubrication distribution system.

The test rig (fig. 5) used a regenerative-torque, back-to-back configuration where a test and a slave planetary section were loaded against each other by using a rotating torque actuator. Hydraulic pressure was applied to the actuator to produce the loop torque. The slave-section sun gear and high-speed shaft were connected through the actuator to the test-section sun gear. The slave-section sun gear was twisted by the torque actuator relative to the high-speed shaft. The actuator allowed the torque to be adjusted while the shaft was rotating, The drive motor was required to supply enough power to overcome rig and test hardware losses.

The test rig was instrumented with flowmeters, pressure transducers, thermocouples, and strain gages. Some of these instruments were on the rotating components. Their signals were connected to rotating amplifiers that allowed transmission across sliprings. Strain gage bridges were used to monitor the shaft torque. Temperature signals from thermocouples on each of the planet bearings were also transmitted across sliprings. These thermocouples were used to monitor the interface between the bearing post and the inside diameter of the inner planet-bearing race.

The rest of the instrumentation was for measuring lubricant flow, temperature, and pressure. Each planetary stage was lubricated by a separate system consisting of a main supply that was then divided into three subsystems. Each subsystem lubricated a particular part of the planetary section. These lubrication areas were the sun–planet mesh (four nozzles), the planet–ring mesh (four nozzles), and the planet bearing area (six nozzles). Each subsystem had a manifold through which the flow was dispersed to the individual nozzles. Flow rate was controlled through shutoff valves that were in series with the manifold nozzles. Figure 6 shows the typical orientation of the lubrication jets in relation to the test hardware.

A torquemeter (full bridge type) located between the system drive motor and the test facility was used for measuring the facility tare, test-section, and slave-section losses. The output of the torquemeter was low-pass filtered to remove frequencies above 0.2 Hz. This yielded a time-averaged torque instead of the instantaneous torque that this instrument could measure.

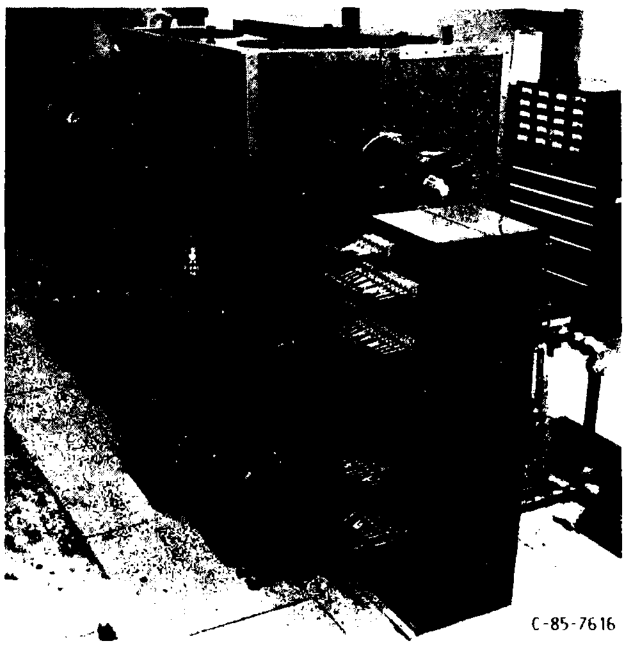
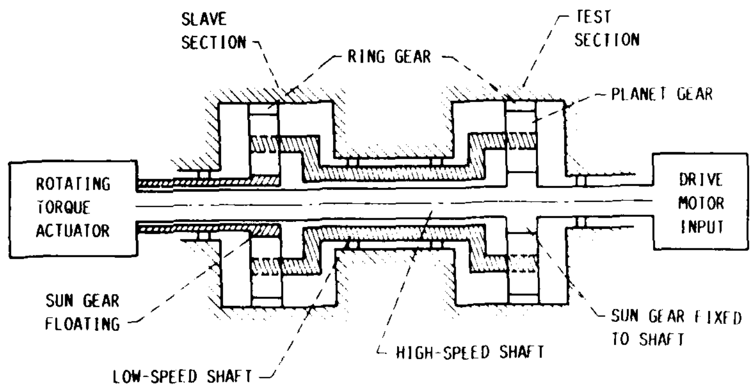


Figure 4.—Test facility after installation of insulation material on lubrication systems and Plexiglass enclosure filled with granulated cork insulation.



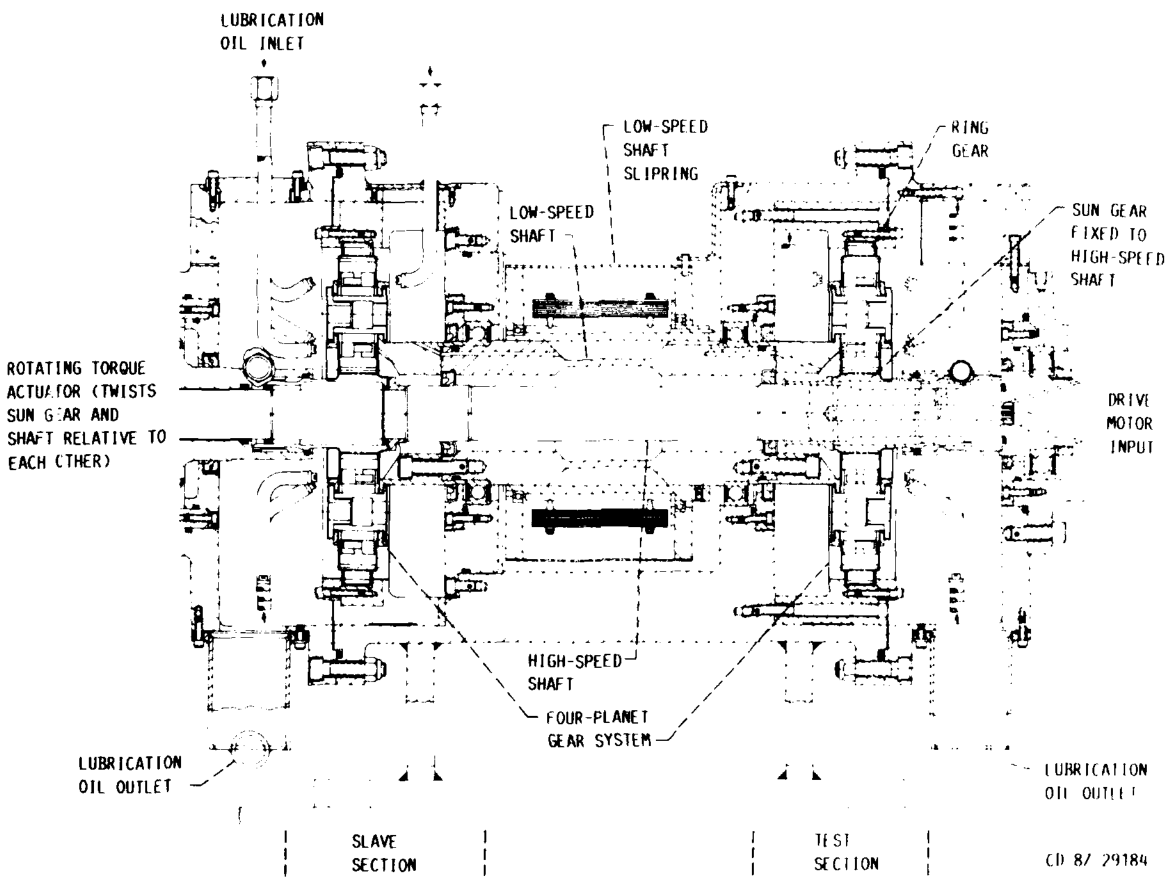


Figure 5.—Cross-sectional view of test gearbox showing key components.

Data were collected and stored by a remote mainframe computer to provide a chronological history of the test. This enabled post-test processing of key data. All data channels were updated every 2 sec while tests were being run.

Instrument Calibration

Several instruments required special calibration. These were the instruments for measuring input torque, high- and low-speed shaft torque, and lubricant flow rate. These data were used for determining the performance characteristics associated with the test program to be conducted. The input torque was measured by a commercially available torquemeter consisting of a full strain gage bridge on a rotating shaft. The torquemeter was bench calibrated over its full rated capacity by hanging weights at a fixed lever arm distance.

The two test rig shafts that carried the sun gear and output carrier torques were calibrated, with their strain gage torque

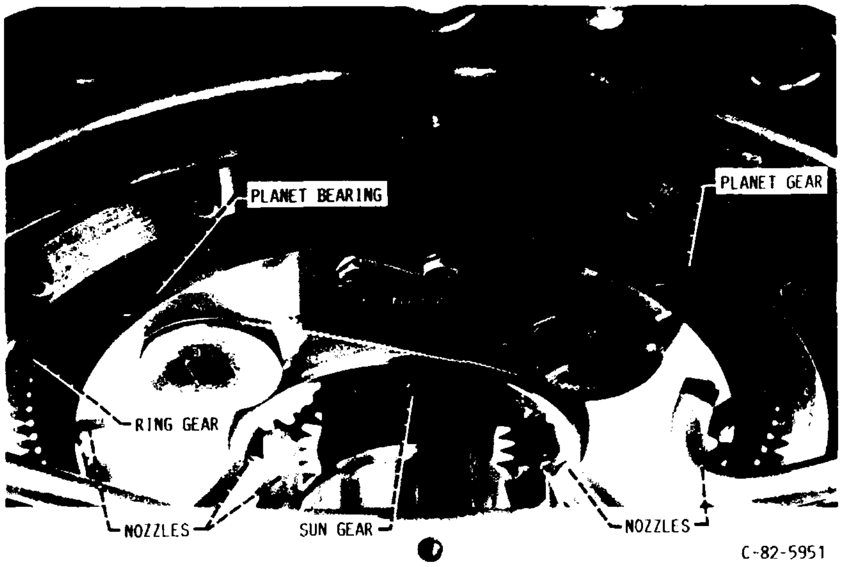


Figure 6.—Location of lubrication nozzles with respect to test rig gears and bearings.

bridges and rotating amplifiers, in a tensile test machine. The fixture used (fig. 7) provided the ability to load the shafts in pure torsion. Each shaft had two complete torque bridge circuits that were calibrated by axially loading the fixture. Each shaft was calibrated by varying the tensile load, which then was related to the applied torque in the shaft.

The volume rate of lubricant flow was measured with turbine flowmeters. Each lubrication system contained one main flowmeter plus a separate one for each of the three manifolds. Each manifold divided the flow equally to a number of nozzles, as discussed earlier. A universal viscosity flowmeter (calibrated with several fluids) was used in series with the individual test flowmeters, prior to testing, to develop a relationship between the output frequency of the universally calibrated flowmeter and that of each test flowmeter. During testing the oil temperature was measured and the oil viscosity calculated from known fluid properties. Next the output frequencies of the individual flowmeters were used to determine what the calibration flowmeter would have measured. Finally the volume flow rates were calculated on the basis of the universal viscosity flowmeter calibration.

Test Rig Tare Losses

The test rig tare losses are those due to components other than the gear meshes and planet bearings. The rig as shown in figure 5 requires a number of bearings and seals that are not part of the helicopter transmission's planetary stage to support shafts and to allow operation in the regenerative loop manner. Also, there are power losses attributed to the two sliprings used. Since the losses not associated with the planetary components are a part of the facility operation, these losses needed to be known. The tare loss associated with the bearings, seals, and sliprings were experimentally measured at the three oil-inlet temperatures used in the parametric tests.

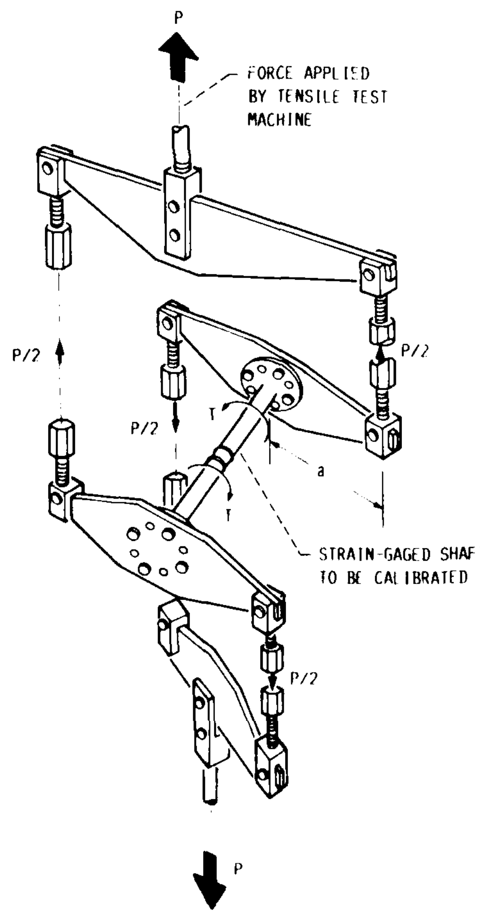


Figure 7.—Fixture used for shaft strain gage torque bridge calibration. Applied torque, $T = Pa$.

Since the test gears drive the shafts, and vice versa (at different operating speeds), a technique was required to drive the rig shafts and bearings by themselves. To accomplish this, the tare tests were conducted in a three-step fashion. The first test consisted of operating the high- and low-speed shafts locked together, in order to rotate the low-speed shaft and measure the losses associated with it. This was done by mechanically locking the sun gear to the carrier and required removing the two ring gears. The rig was then assembled and the insulating enclosure installed. The results of this test at the various speeds and temperatures are shown in figure 8(a). Next, the carriers were removed from the test and slave sections and the facility was again assembled to rotate and measure losses from only the high-speed shaft and the drive motor input shaft. Tests were conducted first at the lower operating speed of the carrier shaft (150 to 350 rpm) and then at the sun gear speed range that was used for the parametric tests (600 to 1620 rpm). These results are shown in figure 8(b). Now the contribution of the high-speed shaft could be subtracted from the locked-shaft condition and the low-speed-shaft losses determined. By using a linear fit of these data and the rotational speed ratio between the high- and low-speed shafts, a total system tare loss as a function of temperature and high-speed-shaft rotational speed was determined.

The last investigation determined the losses due to the shaft between the torquemeter and the input to the test section. This test was conducted by rotating only this shaft at two oil-inlet temperatures (60 and 93 °C; 140 and 200 °F). The torque required to rotate this component was nearly constant, approximately 0.9 N m (8 in. lbf), over the speed range 600 to 1620 rpm at both temperatures.

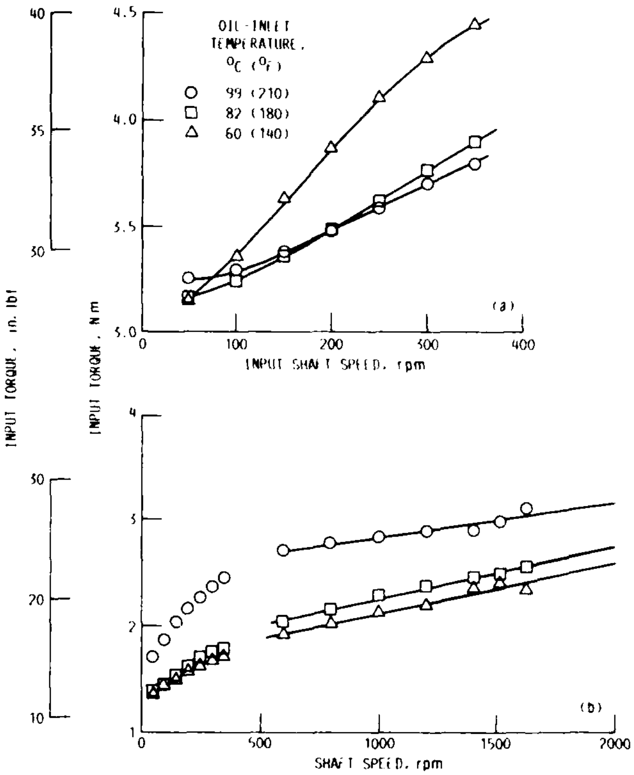
Test Procedure

For each test the first variable to be stabilized was the oil-inlet temperature. Once the required oil-inlet temperature was reached, the oil flow rate, the shaft rotational speed, and the load were set. The rig was then run at these conditions for typically 30 min in order to reach steady state before any data were taken. For each test 10 sets of data were taken at 1-min intervals. At each 1-min interval five successive 2-sec scans were averaged to produce one set. Thus 50 scans were averaged for each test. This was done so that fluctuations in the test measurements would be filtered (averaged) over the test. Then the conditions were changed and cycle repeated.

Results of Parametric Studies

A total of 130 different tests were conducted. The variables altered during the test program were rotational speed, load, lubricant flow rate, and oil-inlet temperature, as well as lubricant type. The maximum test conditions of 100 percent sun gear speed and 100 percent sun gear total torque were 1622 rpm and 1840 N m (16 300 in. lbf), respectively.

To measure the losses based on a given set of conditions, the torque required to drive the complete system was monitored. The tare losses at a given speed and oil-inlet temperature were subtracted from the total input torque. An example of the experimental data (fig. 9) shows the effect of varying oil-inlet temperature, speed, and oil flow rate. The net torque to rotate the test planetary section, because of its losses, was assumed to be half of the remaining torque after the rig tare losses were subtracted from the total torque. An equal split of the losses was assumed because, except for direction of rotation, the test and slave planetary sections were identical in their gear and bearing configuration, lubrication flow rate, lubricant type, and oil-inlet temperature for all tests conducted. An attempt was made to separately measure the power used by the two planetary sections on the basis of the ratio of heat rejected to their respective lubrication systems. However, the temperature differentials were too small to be accurately used at the lower power conditions.



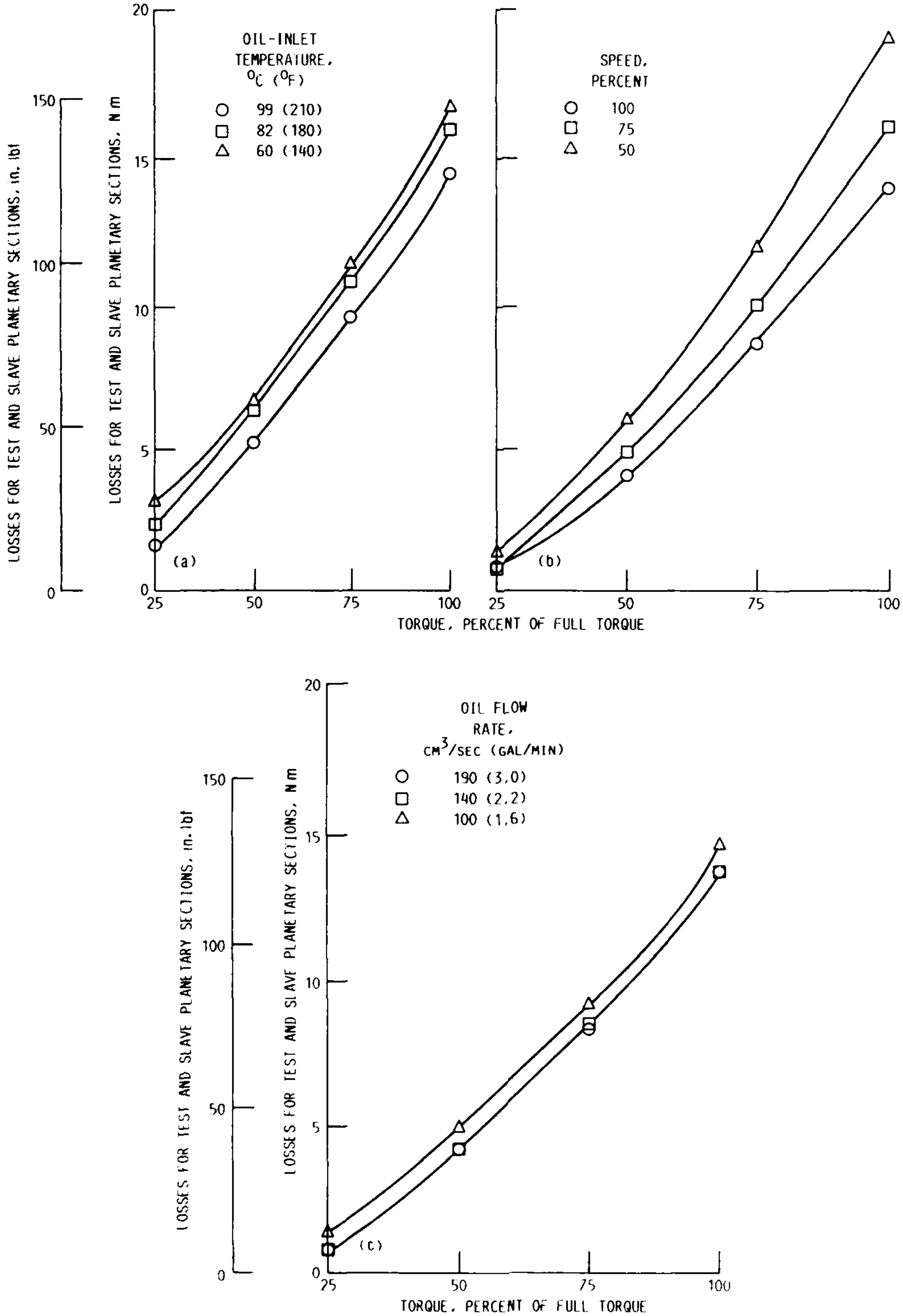
(a) High- and low-speed shafts locked together.
(b) High-speed shaft only.

Figure 8. —Experimental data showing how facility frictional losses (bearing, seal, and slipring losses) changed as function of speed and temperature.

Operating efficiency η was computed from the following equation:

$$\eta = \frac{\left| P_{T_{in}} - 0.5 \left(P_{DM} - P_{tare} \right) \right| \times 100}{P_{T_{in}}}$$

where $P_{T_{in}}$ is the total input power to the test planetary section. This equals the loop power on the high-speed shaft plus the drive motor power minus the input stub shaft losses. The net



(a) Lubricant K; oil-inlet temperature, variable; oil flow rate, 190 cm²/sec (3.0 gal/min); speed, 1622 rpm.
(b) Lubricant E; oil-inlet temperature, 99 °C (210 °F); oil flow rate, 190 cm²/sec (3.0 gal/min) ; speed, variable.
(c) Lubricant E; oil-inlet temperature, 99 °C (210 °F); oil flow rate, variable; speed, 1622 rpm.

Figure 9.—Experimental data showing total rig torque minus tare losses at variety of conditions for two lubricants tested.

TABLE II.—RESULTS FROM UNCERTAINTY ANALYSIS (REF. 14)

Test measurements and calculated facility losses	Test condition or calibration value, N m	Test condition or calibration value, in. lbf	Assumed accuracy, percent	Uncertainty in efficiency calculation, percent
Case 1: 100 percent torque; high accuracy				
High-speed shaft torque	1825.0	16 150.0	1	0.009
Total drive motor input torque	18.8	166.5	1	
Tare losses	4.5	40.0	5	
Input shaft losses	.9	8.0	5	
Case 2: 25 percent torque; low accuracy				
High-speed shaft torque	470.0	4 157.2	5	0.055
Total drive motor input torque	6.4	56.7	5	
Tare losses	3.9	34.8	10	
Input shaft losses	.9	8.0	10	

loss of the test and slave planetary sections equals the drive motor power minus the facility tare losses ($P_{DM} - P_{tare}$).

An uncertainty analysis (ref. 14) was performed on this equation. The accuracy of the result is a function of the test conditions and the uncertainties in the measured quantities. For this equation the worst situation is when the test rig torque is low (25 percent of full torque) and the inaccuracy of the measuring instruments is high. Two example cases (table II) show that the assumed acuracy along with the test conditions can affect the uncertainty in the measured efficiency.

The results, based on the experimental measurements, found from the preceding equation are shown in table III.

Results and Discussion

The two lubricants used in the parametric tests were the oils "E" and "K" from reference 2 (see table IV). Lubricant E is a formulated gear lubricant (dibasic acid ester), and lubricant K is a turbine engine oil (mixture of 99 percent PE (pentaerythritol ester) and 1 percent DPE (dipentaerythritol ester). In figure 10 the test-section efficiency is plotted as a function of percent of full sun gear speed (1622 rpm) for the two lubricants tested. The conditions of 99 °C (210 °F) oil-inlet temperature and 190-cm^3/sec (3.0-gal/min) oil flow rate were held constant. For both lubricants the efficiency increased with speed, except at 25 percent torque (the lowest torque tested). For all speeds the efficiency decreased with increasing percentage of full torque. The highest (99.75 percent) and lowest (99.44 percent) efficiencies measured for all the parametric tests are also shown in figure 10. These were found in the data for lubricant E.

TABLE III.—CALCULATED EFFICIENCY BASED ON EXPERIMENTAL RESULTS FROM PLANETARY TEST RIG

[100 Percent speed, 1622 rpm; 100 percent torque, 1845 N m (16 300 in. lbf).]

Oil inlet temperature, °C	Volume flow rate, cm³/sec	Speed, percent of total	Load, percent of total	Efficiency based on input torque, percent
Lubricant K				
60	190	100	25	99.49
			50	99.57
			75	99.54
			100	99.51
		75	25	99.53
			50	99.54
			75	99.53
		85	100	99.49
		50	25	99.63
			50	99.59
			75	99.53
			100	99.48
	140	100	25	99.57
			50	99.58
			75	99.56
			100	99.53
		75	25	99.59
			50	99.58
			75	99.54
		85	100	99.53
		50	25	99.59
			50	99.57
			75	99.51
			100	99.47
	100	100	25	99.57
			50	99.58
			75	99.56
			100	99.54
	76	100	25	99.54
			50	99.57
			75	99.56
50	190	100	50	99.54
			75	99.52
			100	99.49
82	190	100	25	99.57
			50	99.58
			75	99.56
			100	99.52
		75	25	99.56
			50	99.56
			75	99.52
			100	99.48
		50	25	99.54
			50	99.52
			75	99.48
			100	99.44
	140	100	25	99.65
			50	99.64
			75	99.62
			100	99.58
		75	25	99.68
			50	99.63
			75	99.59
			100	99.60
		50	25	99.66
			50	99.60
			75	99.55
			100	99.50
99	190	100	25	99.65
			50	99.64
			75	99.60
			100	[illegible]
		75	25	99.60
			50	99.62
			75	[illegible]
			100	[illegible]
		50	25	[illegible]

Oil inlet temperature, °C	Volume flow rate, cm³/sec	Speed, percent of total	Load, percent of total	Efficiency based on input torque, percent
Lubricant K (concluded)				
99	190	50	[illegible]	[illegible]
			[illegible]	[illegible]
			100	[illegible]
	140	100	25	[illegible]
			50	[illegible]
			[illegible]	[illegible]
			100	[illegible]
		75	25	[illegible]
			50	[illegible]
			75	[illegible]
			100	99.55
		50	25	99.62
			50	99.59
			75	99.54
			100	99.50
	100	100	25	99.63
			50	99.62
			75	99.62
			100	99.59
Lubricant E				
60	140	100	25	99.69
			50	99.63
			75	99.59
			100	99.49
	100	100	25	99.62
			50	99.62
			75	99.58
			100	99.54
82	190	100	25	99.75
			50	99.69
			75	[illegible]
			100	99.58
		75	25	[illegible]
			50	[illegible]
			75	[illegible]
			[illegible]	[illegible]
		50	25	[illegible]
			50	[illegible]
			75	[illegible]
			[illegible]	[illegible]
	140	100	[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
		75	[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
		[illegible]	[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
99	[illegible]	[illegible]	[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]
			[illegible]	[illegible]

[a] [illegible]
[b] [illegible]

The effect of oil-inlet temperature at 100 percent speed (1622 rpm) and at constant lubricant flow rate is presented in figure 11. It is evident that as the temperature of the oil was raised the efficiency rose for all the conditions shown. This type of performance response is what would be expected. The properties of the lubricant changed over this temperature range (table IV), and this affected the planetary power losses.

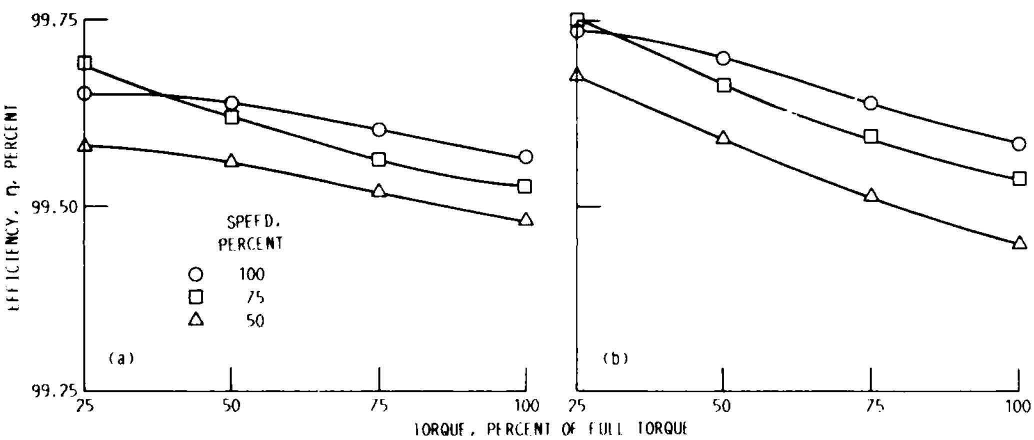
The last parametric effect to be discussed is the effect of oil flow rate (fig. 12). The oil flow rate was changed by closing the oil flow to certain nozzles. Nozzles were closed equally for both the test and slave planetary sections. As shown in figure 13, the lubricant was supplied to each planetary section from nozzle assemblies at several positions on each side. Nozzles to each area of the planetary sections (sun-planet mesh, planet-ring mesh, and planet bearings) were fed through a seperate manifold with control valves for each nozzle. Thus specific total flows were obtained by opening or closing a nozzle valve as indicated in figure 13. Closing the nozzles resulted in a slightly higher flow rate per nozzle (approx. 5 percent increase in flow rate per nozzle when changing from 190 to 100 cm^3/sec (3.0 to 1.6 gal/min) because of an increase in manifold pressure). Flow to each planetary area was maintained in all cases.

From figure 12 it is evident that for this distribution system there was an optimal oil flow rate. The tests were conducted at 190, 140, and 100 cm^3/sec (3.0, 2.2, and 1.6 gal/min). The 140-cm^3/sec (2.2-gal/min) oil flow rate proved to be best for lubricant K (fig. 12(a)). For lubricant E (fig. 12(b)) the 190- and 140-cm^3/sec (3.0- and 2.2-gal/min) flow rates gave similar results, but lowering to 100 cm^3/sec (1.6 gal/min) caused the performance to decrease. A different distribution of the oil, but with the same total flow rate, may further affect performance.

TABLE IV.—PROPERTIES USED FOR ANALYSIS AND CALCULATIONS

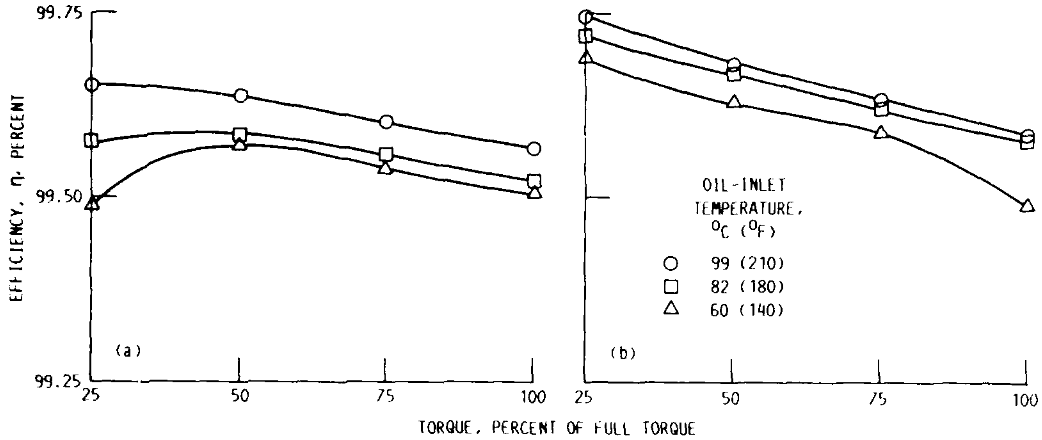
Lubricant[a]	Temperature		Thermal conductivity		Pressure viscosity coefficient		Temperature-viscosity coefficient		Kinematic viscosity		Specific gravity	Specific heat
	°C	°F	$\frac{W}{m\ °C}$	$\frac{Btu}{hr\ ft\ °F}$	$\frac{1}{GPa}$	$\frac{1}{psi}$	$\frac{1}{°C}$	$\frac{1}{°F}$	cSt	$\frac{ft^2}{sec}$		$\frac{Btu}{lbm\ °F}$ or $\frac{mcal}{mg\ °C}$
K (turbine engine oil)	40	104.0	0.115	0.066	11.40	7.86×10^{-5}	0.0223	0.0124	26.39	28.39×10^{-5}	0.9829	0.464
	82	179.6	.115	.066	----	-------	.0223	.0124	7.61	8.19	.9721	.495
	100	212.0	.115	.066	9.50	6.55	.0223	.0124	5.09	5.48	.9725	.507
E (formulated gear lubricant)	40	104.0	0.121	0.070	15.53	10.70×10^{-5}	0.0232	0.0129	33.91	36.50×10^{-5}	0.9322	0.680
	82	179.6	.121	.070	----	-------	.0232	.0129	8.91	9.59	.9211	.730
	100	212.0	.121	.070	11.51	7.94	.0232	.0129	5.87	5.48	.9201	.767

[a]From ref. 2



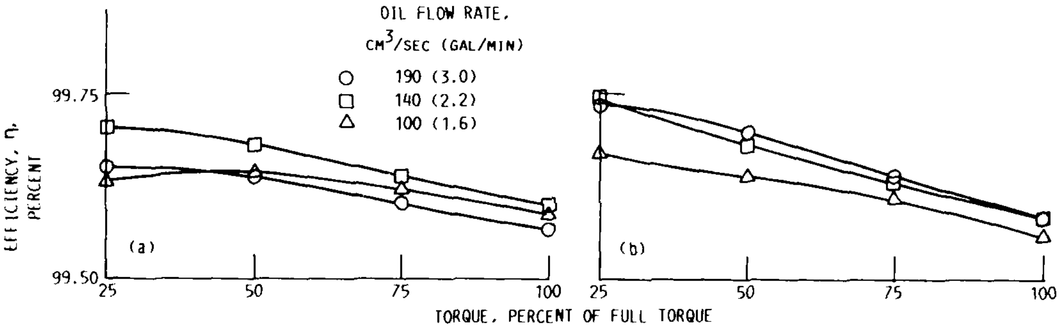
(a) Lubricant K.
(b) Lubricant E.

Figure 10.—Efficiency of test planetary section as function of torque at different rotational speeds for two lubricants. 100 Percent torque, 1840 N m (16 300 in. lbf); oil flow rate 190 cm^3/sec (3.0 gal/min); oil-inlet temperature, 99 °C (210 °F).



(a) Lubricant K; oil flow rate, 190 cm^3/sec (3.0 gal/min).
(b) Lubricant E; oil flow rate, 140 cm^3/sec (2.2 gal/min).

Figure 11.—Efficiency as function of torque at three oil-inlet temperatures. 100 percent speed, 1622 rpm.



(a) Lubricant K.
(b) Lubricant E.

Figure 12.—Efficiency as function of torque for three oil flow rates. 100 Percent speed, 1622 rpm; oil-inlet temperature, 99 °C (210 °F).

Comparison of Results Between Lubricants

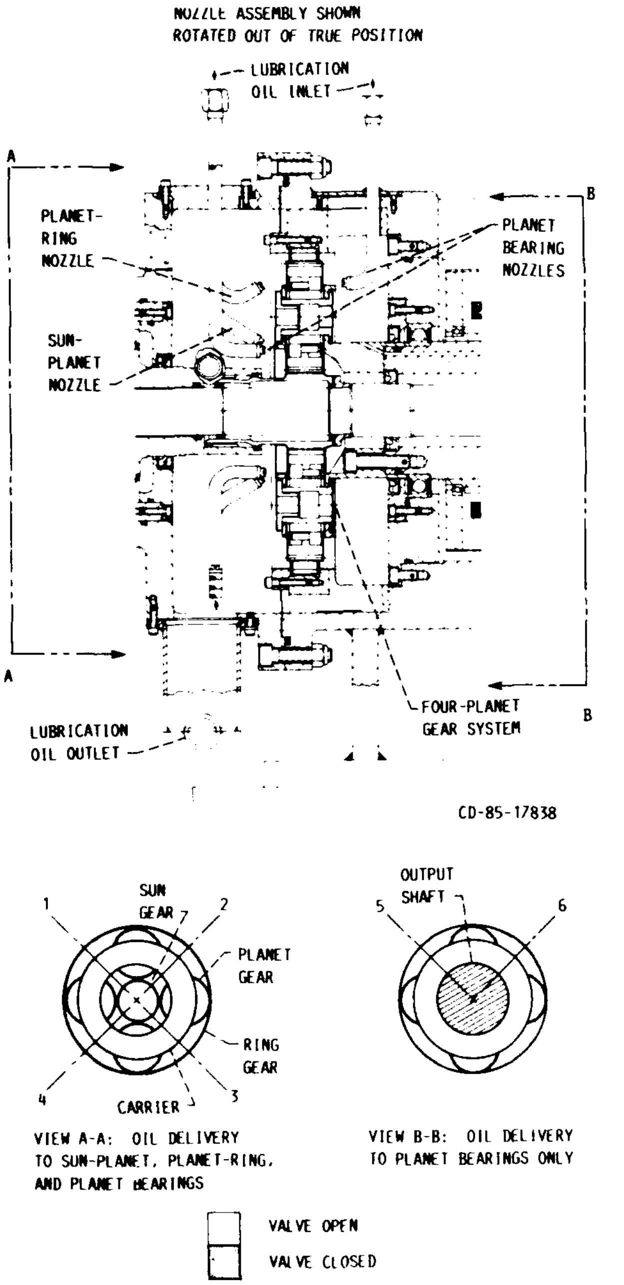
In the present parametric study lubricants K and E (ref. 2) were compared. These lubricants were chosen because they had performed quite differently from each other in that earlier investigation of a complete OH-58 transmission. The physical properties of the lubricants used in the present study, as well as in reference 2, were characterized by the study conducted in reference 15.

The results attained during the present comparison of the lubricants' performance were different from those found in reference 2. In reference 2 the lubricant K produced high efficiencies (98.7 to 98.8 percent), and lubricant E performed poorly in comparison (98.3 percent). In the results found in the present study lubricant E equaled or exceeded lubricant K at most conditions tested. A comparison of the results at 99 °C (210 °F) oil-inlet temperature, 190-cm^3/sec (3.0-gal/min) oil flow rate, and various percentages of torque and speed is shown in figure 14.

As indicated in reference 2 there was no reasonable correlation between efficiency and lubricant properties. Only a fair correlation was attained when the properties were corrected for pressure and temperature in the contact (ref. 4). This same trend was evident from the data gathered in the present study. Numerous attempts were made to use a multiple linear regression model to predict the efficiency based on test facility conditions and expected lubricant properties. The correlation between the regression equation and the experimental results was poor.

Planetary Gear Train Efficiency Analysis

To model the efficiency of a planetary gear train, many sources of power loss must be considered. As indicated in references 5 to 7 the losses found from meshing gear teeth include sliding, rolling, and windage losses. Also for the planetary system the losses due to the planet bearings must be considered. The sum of these loss components was used to predict the planetary stage efficiency.



VOLUME OIL FLOW RATE PER PLANETARY STAGE		NOZZLE LOCATIONS													
		SUN-PLANET GEAR MESH MANIFOLD				PLANET-RING GEAR MESH MANIFOLD				PLANET BEARING LUBRICATION MANIFOLD					
CM^3/SEC	GAL/MIN	1	2	3	4	1	2	3	4	1	2	3	4	5	6
190	3.0														
140	2.2														
100	1.6														
76	1.2														

Figure 13.—Oil nozzle orientation for planetary test- and slave-section lubrication and its effect on flow rate.

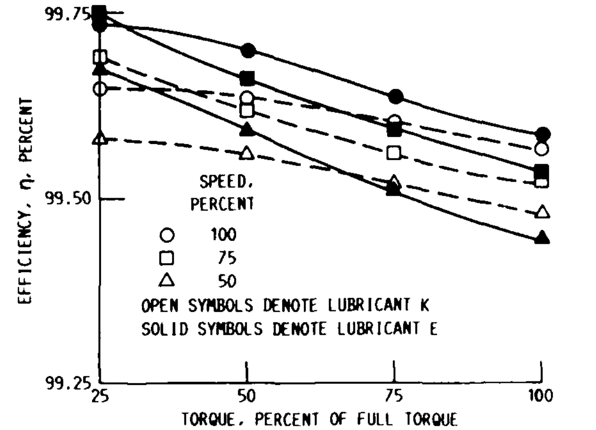


Figure 14.—Efficiency as function of speed for two lubricants tested at different values of full load (1840 N m; 16 300 in. lbf). Oil-inlet temperature, 99 °C (210 °F); oil flow rate, 190 cm^3/sec (3.0 gal/min); 100 percent speed, 1622 rpm.

To model the sliding losses, the coefficient of friction must be determined. Early methods for predicting the coefficient of friction in meshing gears are described in reference 16. The gear contact was simulated with rollers, and the effects of sliding velocity, oil temperature, and load were measured. An empirical model was formulated for the mineral oil tested in reference 16. The experimental results in reference 16 for coefficient of friction agreed within the expected range ($0.001 < \mu < 0.1$) for gears as described in reference 17. In a more recent study (ref. 18) seven synthetic lubricants and traction fluids were experimentally tested similar to reference 16. In the study of reference 18 two of the lubricants used were lubricants E and K of reference 2. Experimental data were gathered to develop a constitutive model for predicting the coefficient of friction for gear surfaces.

The rolling losses were predicted by using the technique of reference 6. This technique is similar to what would be used for a ball bearing. Since the gear tooth surfaces roll over each other (rolling velocity is a function of position along the line of action) an elastohydrodynamic (EHD) film forms. The EHD film developed on the surfaces resists the motion of the gears. To overcome the EHD film requires expenditure of power.

The windage loss model, as described in reference 6, was developed from investigations of turbine wheel windage. The model was developed to incorporate the windage losses of the gears on the basis of gear size, rotational speed, and assumed oil-air environment.

A computer program, PLANETSYS (ref. 19), was used to predict the planet bearing losses. Lubricant properties, bearing design data, planetary arrangement, and operating conditions were the input variables. The program predicts the steady or transient bearing performance, fatigue life, and many other important bearing parameters.

Comparison of Experimental and Analytical Results

The planetary gear train that was experimentally tested was modeled by the previously described analytical techniques. The gear parameters used for analysis are shown in table I and the lubricant properties in table IV. The drop tooth design used on the ring gear required making a choice as to which pressure angle would be used in the analysis. The parameters associated with the ring gear were chosen (as shown later, the analysis indicates a rather minor contribution from this gear mesh). The planet bearing program PLANETSYS was used with the assumption that the bearings operated in a fully flooded lubrication condition (the cavity region of the bearing was assumed to be full of lubricant). This condition was assumed because of the low rotational speed of the carrier and the amount of fluid supplied to this region at the 190-cm^3/sec (3.0-gal/min) condition used for comparison. Analytical losses were calculated by adding power losses from the external gears, the internal gears, and the planet bearings.

Predicted and measured efficiency results at 100 percent speed, 60 and 99 °C (140 and 210 °F) oil-inlet temperature, and different percentages of full torque are shown in figure 15 for both lubricants. The best correlation was found from the results at 60 °C (140 °F) for lubricant E. The results of the other tests, although not agreeing in actual value, did follow the same trends as the experimental data.

The differences between experimental and predicted results are probably due in part to oil pumping losses by the gears, which were not considered in the analytical model. This is not to be confused with the rolling or "pumping" loss associated with the formation of an EHD film that was previously discussed. Reference 8 shows that a significant amount of power is expended in pumping oil from the changing volume in between the gear teeth as the gear rotates through mesh. Increasing the gear pitch line velocity mitigates the problem because some of the oil between the teeth is flung off by gear rotation and there is less time for the jet to penetrate between the teeth (refs. 8 and 20). It is conceivable that with the introduction of oil pumping losses the existing model (ref. 6) would approximate the experimental results more closely.

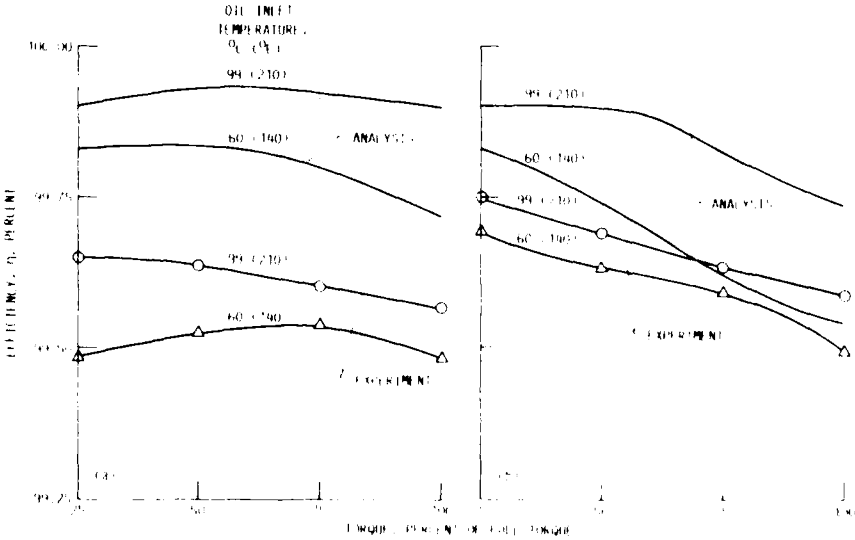
Another aspect that is not included in the model of reference 6 is the effect of lubricant volume flow rate. The model of reference 6 assumes that only a thin film of lubricant is present. Introducing the loss due to momentum exchange between the lubricating jets and the components under investigation would lead to an improved numerical tool.

Comparison With Results of Other Studies

The results found experimentally and analytically in this report are compared here with those found by other investigators. The main difficulty in the comparison is that the conditions, speeds, loads, and geometries of the geared systems differed.

Comparison with reference 2.—The test results given in reference 2 were the closest to the results of this study. A complete OH-58 helicopter transmission was used. Although the sun and ring gears were identical to those tested here, the number of planets and the type of planet bearings differed. The transmission configuration used in reference 2 had three planets that were supported by a double row of spherical bearings. The complete transmission also included a number of bearings and a spiral-bevel gear mesh not used in the planetary test rig.

The reference 2 data showed an increase in efficiency when the oil-inlet temperature was increased except for 2 of the 11 oils tested, including lubricant E. This result conflicts with what was found in the present study, where both lubricants K and E exhibited an increase in efficiency with temperature. It is conceivable that the contribution of the other components in the complete transmission affected the experimental outcome.



(a) Lubricant K.
(b) Lubricant E.

Figure 15.—Experimental and analytical efficiency as function of torque for full speed (1622 rpm) at two oil inlet temperatures.

***Comparison with reference* 3.**—Reference 3 decribes the test of a 2240-kW (3000-hp) helicopter transmission where efficiency increased with power and decreased with speed. Neither of these conclusions agrees with the results from the planetary test done here. Once again the number of spiral-bevel meshes and support bearings in the reference 3 study influenced the overall transmission performance.

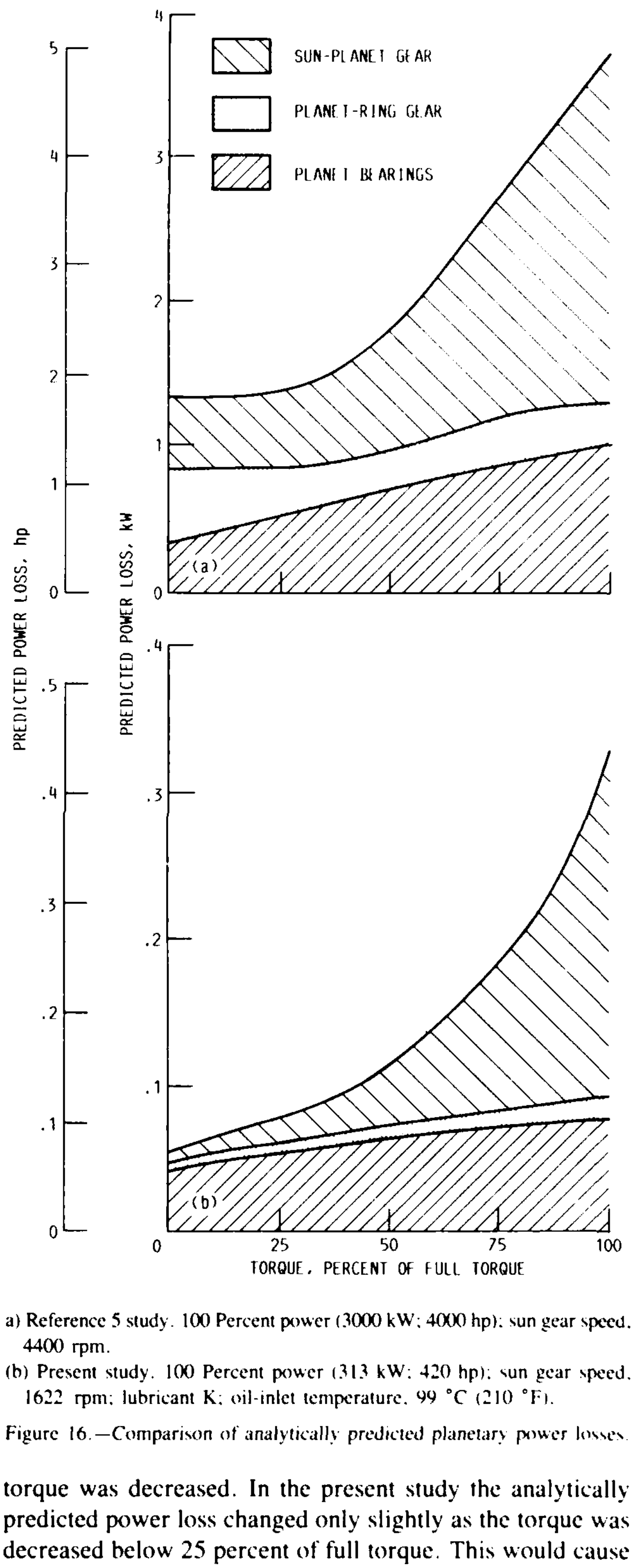
***Comparison with reference* 5.**—The study completed in reference 5 compared experimental and analytical results for a turboprop gearbox. Although the gearbox contained a planetary stage without spiral-bevel gears, there were accessory gears and bearings, pumps, and shaft support bearings. For the total transmission, at full speed, the analytical and experimental results showed that as the power increased so did the total transmission efficiency. The losses due to all of the transmission bearings found analytically in reference 5 increased only slightly over the full range of torque. Since the change in bearing power loss (ref. 5) with torque was low, this would contribute to an overall efficiency decay at low levels of input power.

The numerical analysis of reference 5 predicted nearly constant efficiency for the planetary components (i.e., sun-planet mesh, planet-ring mesh, and planet bearings) at full speed (approx. 99.85 percent at 1000 to 3000 kW (1340 to 4000 hp) of input power). In reference 5 (fig. 16(a)) the power loss due to the sun-planet gear mesh remained nearly constant (approx. 40 percent of the total power loss for the planetary stage) from very low power levels until approximately 30 percent of full power. As the power level was raised to 100 percent of full power (3000 kW; 4000 hp) the sun-planet losses increased to approximately 65 percent of the total power loss for the planetary stage.

In the present study (fig. 16(b)) the power loss, found by the analysis procedure, due to the sun-planet gear mesh increased linearly from very low power levels until it reached 25 percent of the total loss at approximately 30 percent of full power. It then increased parabolically to 75 percent of the total power loss at 100 percent of full power level.

Since the planetary power loss from reference 5 increased linearly with percentage of full power, this caused the efficiency predicted for the system (ref. 5) to remain constant over this range of power ($\eta \approx 99.85$ percent for 1000 to 3000 kW (1340 to 4000 hp) of input power). In the present study the power loss due to the sun-planet mesh, as well as the entire planetary stage, increased parabolically from 25 to 100 percent of full power (80 to 310 kW; 105 to 420 hp). This parabolic increase in power loss with load caused the efficiency to decrease with load. This effect was seen experimentally (figs. 10 to 12) and analytically (fig. 15).

One last comparison to note between reference 5 and the present study is the analytical results at low percentages of full torque (approx. 0 to 30 percent of full torque). In reference 5 below 30 percent of full torque the power loss due to the planetary stage remained constant (fig. 16(a)). This would cause the predicted efficiency to decay rapidly as the applied torque was decreased. In the present study the analytically predicted power loss changed only slightly as the torque was decreased below 25 percent of full torque. This would cause the efficiency to decay as torque was decreased below the 25 percent level.



(a) Reference 5 study. 100 Percent power (3000 kW; 4000 hp); sun gear speed, 4400 rpm.

(b) Present study. 100 Percent power (313 kW; 420 hp); sun gear speed, 1622 rpm; lubricant K; oil-inlet temperature, 99 °C (210 °F).

Figure 16.—Comparison of analytically predicted planetary power losses.

This effect for low power levels (less than 25 percent of full torque) was not shown in figure 15 because experimental data were not taken below this level. However, increasing torque from 0 to 25 percent of full torque at constant speed would increase efficiency with load over this range of torque. The same effect of increasing efficiency with torque from 0 to 25 percent of full torque would have been expected from the experimental facility had tests been conducted at these levels of applied load.

***Comparison with reference* 6.**—As mentioned in the Planetary Gear Train Efficiency Analysis section of this report, the sliding losses were predicted by using the constitutive model developed in reference 18 along with the analyses of references 5 to 7. That model will now be compared with one used in an earlier study (ref. 6). Reference 6 contains predicted results for a single-mesh gearbox as well as the basic method for predicting gear mesh losses used in reference 5 and in this report. For the gearbox presented in reference 6 the experimental as well as analytical results indicated that as speed increases efficiency decreases and as load increases efficiency increases. The analytical results of reference 6 are based on a lubrication friction model for MIL–L–7808 lubricant. The gear data from reference 6 were then used with the same analytical procedure used in the present study to determine the gear losses for lubricant K at 60 °C (140 °F) oil-inlet temperature. The results are shown in figure 17 for torques from 68 to 270 N m (600 to 2400 in. lbf), the values used in reference 6. In this torque range the predicted efficiency of the gear mesh increased with applied torque. The maximum loading conditions presented in reference 6, 270 N m (2400 in. lbf) of torque at 2000 rpm, produced a maximum bending stress of 31 KPa (4500 psi). This is a very lightly loaded system. This same gear configuration was then loaded from 340 to 1000 N m (3000 to 9000 in. lbf) of torque. The efficiency then remained nearly constant over this range, as is also shown in figure 17.

The gear mesh efficiencies (fig. 17) for the two lubricant models had similar trends over the full range of applied torque. The analytical model (using the friction coefficient of ref. 18) predicted, as did the MIL–L–7808 lubricant model (ref. 5), a decrease in efficiency if the load on the system was very light. Both models also indicated that at moderate to high loads the efficiency remained nearly constant.

***Comparison with reference* 10.**—Reference 10 compares Novikov and helical involute gears. The system was splash lubricated, so churning losses were included in the results. The results showed that efficiency increased with speed and decreased with load. These results were similar in trends to what was found in the present study.

Summary of Results

A parametric study was performed on a helicopter transmission planetary stage with a four-planet configuration.

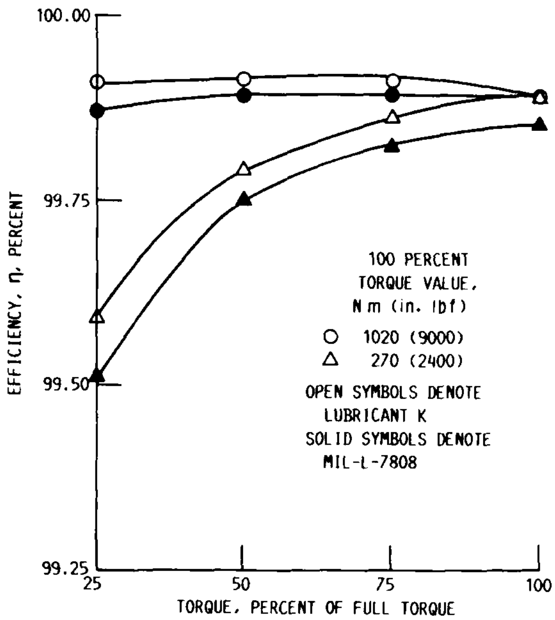


Figure 17.—Comparison of results of two different friction models for spur gears described in reference 6 as a function of load.

Two planetary stages were used in a back-to-back, test-and-slave arrangement with separate lubrication systems. Test parameters included sun gear speeds to 1622 rpm, and torque to 1840 N m (16 300 in. lbf). Two different gear lubricants were used, with total flow rates per planetary stage from 100 to 190 cm^3/sec (1.6 to 3.0 gal/min) and oil-inlet temperatures from 60 to 99 °C (140 to 210 °F). Experimentally measured efficiency over all the test variables ranged from 99.44 to 99.75 percent. Analytical performance predictions were made and compared with measured results. Comparisons were also made with the results of other investigators.

The conclusions that can be drawn from this study are that all the parameters varied during the tests affected the planetary section efficiency. The following specific results were obtained:

1. Efficiency generally increased with speed for constant torque except at 25 percent torque, where 75 percent of full speed was more efficient than 100 percent for several tests.

2. Efficiency decreased with torque at constant speed for all tests and analytical predictions (50 to 100 percent of full speed).

3. Efficiency increased with oil-inlet temperature for each speed and loading condition.

4. Efficiency was generally higher for lubricant E (formulated gear lubricant) than for lubricant K (turbine engine oil).

5. Efficiency was also affected by the lubricant flow rate. For lubricant K, lowering the flow from 190 to 140 cm^3/sec (3.0 to 2.2 gal/min) increased the efficiency, but further reduction to 100 cm^3/sec (1.6 gal/min) decreased efficiency. Lubricant E produced no change in efficiency at 190 and

140 cm^3/sec (3.0 and 2.2 gal/min), but lowering the flow rate to 100 cm^3/sec (1.6 gal/min) resulted in a performance decrease.

6. Experimental efficiency results were lower than those predicted, probably in part because the analytical model lacked the gear pumping losses. However, the efficiency as a function of load for constant speed and oil-inlet temperature followed similar trends in both the experimental and analytical results.

7. Comparing results between different investigations can be quite difficult. The loading conditions, the number of components, and the geometry of the components can greatly affect both measured and analytically predicted performance.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, September 21, 1987

References

1. Coy, J.J.; Townsend, D.P.; and Zaretsky, E.V.: Gearing. NASA RP-1152, 1985.
2. Mitchell, A.M.; and Coy, J.J.: Lubricant Effects on Efficiency of a Helicopter Transmission. Problems in Bearings and Lubrication, AGARD CP-323, AGARD, Paris, France, 1982, pp. 20-1 to 20-16. (NASA TM-82857.)
3. Mitchell, A.M.; Oswald, F.B.; and Schuller, F.T.: Testing of YUH-61A Helicopter Transmission in NASA Lewis 2240-kW (3000-hp) Facility. NASA TP-2538, 1986.
4. Coy, J.J.; Mitchell, A.M.; and Hamrock, B.J.: Transmission Efficiency Measurements and Correlations With Physical Characteristics of the Lubricant. Gears and Power Transmission Systems for Helicopters and Turboprops, AGARD CP-369, AGARD, Paris, France, 1984, pp. 20-1 to 20-15. (NASA TM-83740.)
5. Anderson, N.E.; Lowenthal, S.H.; and Black, J.D.: An Analytical Method To Predict Efficiency of Aircraft Gearboxes. AIAA Paper 84-1500, June 1984. (NASA TM-83716.)
6. Anderson, N. E.; and Lowenthal, S.H.: Spur-Gear Efficiency at Part and Full Load. NASA TP-1622, 1980.
7. Anderson, N.E.; and Lowenthal, S.H.: Efficiency of Nonstandard and High Contact Ratio Involute Spur Gears. J. Mechanisms Transmission Automation Design, vol. 108, no. 1, Mar. 1986, pp. 119-126. (NASA TM-83725.)
8. Ariura, Y.; et al.: The Lubricant Churning Loss in Spur Gear Systems. Bull. JSME, vol. 16, no. 95, May 1973, pp. 881-892.
9. Drago, R.J.; Mangione, P.J.; and Wendrzycki, R.B.: A Study of the Potential Benefits Associated With the Development of a Dedicated Helicopter Transmission Lubricant. Aircraft and Bearing Tribological Systems, AGARD CP-394, AGARD, Paris, France, 1985, pp. 6-1 to 6-11.
10. Ishisbashi, A.; and Yoshino, H.: Power Transmission Efficiencies and Friction Coefficients at Teeth of Novikov-Wildhaber and Involute Gears. ASME Paper 84-DET-58, Oct. 1984.
11. Martin, K.: The Efficiency of Involute Spur Gears. J. of Mech. Des., vol. 103, no. 1, Jan. 1981, pp. 160-169.
12. Tourret, R.; and Wright, E.P., eds.: Performance and Testing of Gear Oils and Transmission Fluids. Heyden/Institute of Petroleum, 1980.
13. Dudley, D.W.: Handbook of Practical Gear Design. McGraw-Hill, 1984.
14. Holman, J.P.; and Gajda, W.J.: Experimental Methods For Engineers, McGraw-Hill, 1978, pp. 44-51.
15. Present, D.L.; et al.: Advanced Chemical Characterization and Physical Properties of Eleven Lubricants. NASA CR-168187, 1983. (Avail. NTIS, AD-A131945.)
16. Benedict, G.H.; and Kelley, W.W.: Instantaneous Coefficients of Gear Tooth Friction. ASLE Trans., vol. 4, no. 1, Apr. 1961, pp. 59-70.
17. Jantzen, E.; and Buck, V.: Tribology in Aircraft Systems—Basic Principles and Applications. Aircraft and Bearing Tribological Systems, AGARD CP-394, AGARD, Paris, France, 1985, pp. 4-1 to 4-13.
18. Tevaarwerk, J.L.: Constitutive Modelling of Lubricants in Concentrated Contacts at High Slide to Roll Ratios. (IR-4, Transmission Research Inc.; NASA Contract DEN3-35.) NASA CR-175029, 1985.
19. Hadden, G.B.; et al.: A Computer Program for the Steady State and Transient Thermal Analysis of a Planetary Power Transmission System: User's Manual. (SKF-AT81D044, SKF Technology Services; NASA Contract NAS3-22690.) NASA CR-165366, 1981.
20. Akin, L.S.; Mross, J.J.; and Townsend D.P.: Study of Lubricant Jet Flow Phenomena in Spur Gears. J. Lubr. Technol., vol. 97, no. 2, Apr. 1975, pp. 283-288.

Report Documentation Page

National Aeronautics and Space Administration

1. Report No. NASA TP-2795 AVSCOM TR-87-C-28

2. Government Accession No.

3. Recipient's Catalog No.

4. Title and Subtitle

Efficiency Testing of a Helicopter Transmission Planetary Reduction Stage

5. Report Date

February 1988

6. Performing Organization Code

7. Author(s)

Robert F. Handschuh and Douglas A. Rohn

8. Performing Organization Report No.

E-3770

10. Work Unit No.

1L161102AH45
505-63-51

9. Performing Organization Name and Address

NASA Lewis Research Center and Propulsion Directorate,
U.S. Army Aviation Research and Technology Activity—AVSCOM,
Cleveland, Ohio 44135

11. Contract or Grant No.

13. Type of Report and Period Covered

Technical Paper

12. Sponsoring Agency Name and Address

National Aeronautics and Space Administration
Washington, D.C. 20546 and U.S. Army Aviation
Systems Command, St. Louis, Mo. 63120

14. Sponsoring Agency Code

15. Supplementary Notes

Robert F. Handschuh, Propulsion Directorate, USAARTA-AVSCOM.

16. Abstract

A parametric study of the efficiency of a 310-kW (420-hp) helicopter transmission planetary test section (four-planets) was performed. The purpose was to determine the planetary contribution to the overall transmission power loss. Test parameters varied were oil flow rate, oil inlet temperature, lubricant type, shaft speed, and applied torque. The measured efficiency over all the test variables ranged from 99.44 to 99.75 percent. These experimental results were compared with other experimental and computational results.

17. Key Words (Suggested by Author(s))

Gears
Transmission efficiency
Power loss

18. Distribution Statement

Unclassified – Unlimited
Subject Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No of pages	22. Price*
Unclassified	Unclassified	17	A02

*For sale by the National Technical Information Service, Springfield, Virginia 22161

END
DATE
FILMED
5-88
DTIC