Effect of Advanced Component Technology on Helicopter Transmissions

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ABSTRACT

Experimental tests were performed on the NASA/Bell Helicopter Textron (BHT) 500-hp advanced technology transmission (ATT) at the NASA Lewis Research Center. The ATT was a retrofit of the OH-58C helicopter 236-kW (317-hp) main rotor transmission, upgraded to 373 kW (500 hp), with a design goal of retaining long life with a minimum increase in cost, weight, and size. Vibration, strain, efficiency, deflection, and temperature experiments were performed and the results were compared to previous experiments on the OH-58A, OH-58C, and UH-60A transmissions. The high-contact-ratio gears and the cantilever-mounted, flexible ring gear of the ATT reduced vibration compared to that of the OH-58C. The ATT flexible ring gear improved planetary load sharing compared to that of the rigid ring gear of the UH-60A transmission. The ATT mechanical efficiency was lower than that of the OH-58A transmission, probably due to the high-contact-ratio planetary gears.

INTRODUCTION

Over the past 30 years the helicopter has evolved into a valuable air vehicle for both military and commercial use. The design and performance requirements of helicopters are continuously becoming more demanding with specific goals of increasing life, reliability, and maintainability while decreasing weight and noise. Helicopter drive trains and in particular, transmissions, require advances in state-of-the-art technology in order to meet these requirements.

In most cases the technology to meet these challenges has been placed by military interests (1). Currently, the U.S. Army has a wide spectrum of helicopters in its inventory ranging from light observation to medium-lift cargo (such as the OH-58, UH-1, CH-47, UH-60, and AH-64). Much of the fleet became operational in the 1960’s and have continued in service through several versions (A, B, C, D series) with each version placing more demands on the drive train. Thus, advances in current drive train designs as well as innovative new concepts play an important role in meeting the next-generation rotorcraft needs.

In an effort to advance helicopter transmission technology NASA Lewis Research Center awarded a contract to Bell Helicopter Textron (BHT) for the design, development, and manufacture of a 373-kW (500-hp) demonstrator transmission. The design emphasis of the NASA/BHT advanced technology transmission (ATT) was to develop a 373-kW (500-hp) version of the OH-58C 236-kW (317-hp) main rotor transmission that would have long, quiet life at a minimum increase in cost, weight, and space. This was accomplished by implementing advanced technologies that have been developed during the last decade and making improvements dictated by field experience (2).

The objective of the present study is to report on the experimental testing performed on the NASA/BHT 500-hp ATT and compare the results to previous experiments on the OH-58A, OH-58C, and UH-60A transmissions. The tests were conducted in the NASA Lewis 500-hp helicopter transmission test stand, covering a range of torque and speed conditions. Vibration and efficiency experiments were performed on the ATT and compared to the OH-58A and OH-58C. Deflection and temperature experiments were performed on the ATT and compared to the OH-58A. Ring gear strain experiments were performed on the ATT and compared to the UH-60A.

APPARATUS

OH-58 Main Rotor Helicopter Transmissions

The OH-58 is a single-engine, land-based light observation helicopter. The helicopter serves both military (OH-58 Kiowa) and commercial (Bell Model 206 Jet Ranger) needs. The OH-58A main rotor transmission (Fig. 1) is rated for use at an engine output of 201-kW (270-hp) continuous power at 6180 rpm and 236 kW (317 hp) for 5 min at takeoff. The transmission is a two-stage reduction gearbox. For the first stage the input shaft drives a 19-tooth spiral-bevel pinion gear, which meshes with a 71-tooth gear. The bevel pinion shaft is mounted on triplex ball bearings and one roller bearing. The bevel gear shaft is mounted overhung on duplex ball bearings and one roller bearing.
A planetary mesh provides the second reduction stage. The bevel gear shaft is splined to a sun gear shaft. The 27-tooth sun gear drives three 35-tooth planet gears. The planet gears mesh with a 99-tooth fixed ring gear, which is splined to the top case. The planet gears are mounted on double-row spherical roller bearings and the bearings are attached to the planet carrier. Power is taken out through the planet carrier, which is splined to the last output shaft. The output shaft is supported by a ball bearing and a roller bearing. The overall reduction ratio of the main power train is 17.44:1.

The 71-tooth bevel gear also drives a 27-tooth accessory gear. The accessory gear runs an oil pump, and lubrication is supplied through jets located in the top and bottom cases.

The OH-58C main rotor transmission is identical to the OH-58A with the exception of the planetary arrangement. The OH-58C has four planets, compared to three in the OH-58A. The planet bearings in the OH-58C are double row cylindrical roller bearings, compared to spherical roller bearings in the OH-58A. The OH-58C also has a different planet carrier design, that being a two-piece carrier straddling the planet gears.

500-hp Advanced Technology Transmission

The design goal of the NASA/Bell Helicopter Textron (BHT) 500-hp advanced technology transmission (ATT) was to upgrade the OH-58C 236-kW (317-hp) version to 373 kW (500 hp) while retaining long life and low noise with a minimum increase in cost, weight, and size. This was accomplished by implementing advanced technology developed during the last decade and improvements dictated by field experience (2).

The ATT (Fig. 2) has the same basic configuration, the same number of teeth on the gears, and the same speed reduction ratio as the OH-58C transmission. The advanced technology components, concepts, and improvements incorporated in the ATT are:

1. High-contact-ratio planetary spur gears intended to enable higher torques to be transmitted and at the same time reduce noise and increase gear life.

2. A cantilever-mounted planetary ring gear intended to provide a flexible mount for a more uniform load distribution among planets, isolation of the meshing teeth from the housing for noise reduction, and reduced debris generated due to the elimination of a working ring gear spline.

3. An oil transfer mechanism designed in the planet carrier for improved lubrication to the planetary gear meshes and the sun gear spline.

4. A straddle-mounted bevel gear intended to enable higher torques to be transmitted without destroying the tooth contact patterns.

5. An improved sun gear spline design (crown hobbled and hardened) running in a bath of flowthrough oil (provided by the planet carrier) to prevent excessive wear.

6. Cleaner, stronger spiral-bevel gear steel (VIM-VAR 9310), vacuum carburized and shot peened, along with cleaner bearing steel (VIM-VAR M-50) intended to compensate for the increase in power transfer capability of the ATT with no decrease in gear or bearing lives.

The output of the test transmission input and output shafts are equipped with speed sensors, torque meters, and slip rings. Test transmission lubrication is supplied by an internal oil pump. The transmission oil can be cooled by a flight hardware air-oil cooler mounted on the transmission (with forced air provided to the cooler) or by an external oil-water heat exchanger. An external oil-pumping system located in the basement below the test stand is also available for the test transmission.

The 149-kW (200-hp) motor is equipped with a speed sensor and a torquemeter. The magnetic particle clutch is equipped with speed sensors on the input and output shafts along with thermocouples. A facility oil-pumping and cooling system lubricates the differential gearbox, the closing-end gearbox, and the bevel gearbox. The facility gearboxes are also equipped with accelerometers, thermocouples, and chip detectors for health and condition monitoring.

Instrumentation and Testing Procedure

First, the ATT was installed in the 500-hp test stand with a minimum amount of instrumentation. The transmission was run at a variety of speed and torque conditions as recommended by BHT as part of a break-in procedure. The speed and torque were gradually increased until full power was achieved. Upon completion of the break-in tests, the ATT was disassembled, inspected, instrumented, and reinstalled in the test stand. Vibration, strain, efficiency, deflection, and temperature experiments were then performed on the ATT. The experiments were all run independently, except for the efficiency and temperature tests, which were run at the same time.

During instrumentation, testing procedure, and data reduction of the ATT vibration experiments were patterned after the OH-58A vibration tests reported in Ref. 3. Seven piezoelectric accelerometers were mounted on the transmission housings (Fig. 4). The accelerometers were placed to cover a variety of locations and all three measurement directions: vertical,
longitudinal, and transverse. Each accelerometer pro-
duced a charge output proportional to acceleration.
The outputs were fed to charge amplifiers that produced
alternating-current (ac) voltages. The ac signals,
which represented actual acceleration as a function of
time, were recorded on magnetic tape for later
processing.

Ring gear strain experiments were performed on
the ATT to investigate load distribution in the plan-
etary system. Strain gages were mounted on the back
of the ATT cantilever-mounted (flexible) ring gear
(Fig. 5(a)). The gages measured strain in the tangen-
tial direction on the ring gear indicating hoop stress
levels. Strain gage conditioners provided the bridge
completion, excitation, amplification, and shunt cali-
bration. The conditioner output voltage represented
strain as a function of time and was recorded on mag-
netic tape for later processing. Strain experiments
were not available for the OH-58 spline-mounted (rigid)
ring gear to compare with the ATT. Results were
obtained, however, from OH-58A (Blackhawk) helicopter
transmission experiments. These ring gear strain tests
were performed in the NASA OH-58A transmission test
stand (4). Strain gages were installed on the back of the
OH-60A rigid-mounted ring gear in a similar fashion as those on the ATT
(Fig. 5(b)). A brief description of the OH-60A trans-
mision and the 3000-hp test stand is given in the
Appendix.

The ATT efficiency experiments were performed in
the same manner as the OH-58A tests of Ref. 5. The
test transmission and an oil-water heat exchanger
(used to cool the transmission oil and maintain it at
a constant temperature) were thermally insulated to
provide an adiabatic enclosure. Provisions were setup
to collect the water that flowed through the oil-water
heat exchanger. The weight of the accumulated water
and the difference in the heat exchanger water-outlet
and water-inlet temperatures were measured. From
these, the heat generation due to mechanical power
losses of the transmission was calculated. Transmis-
sion efficiency was determined by subtracting the power
losses from the measured input power, then dividing by
the input power. The lubricant used for the ATT effi-
ciency tests was that labeled "lubricant code K" in Ref. 5. Lubricant K was a synthetic gas turbine
engine oil with a viscosity of 5 centistokes at 99 °C
(210 °F).

Spiral-bevel gear deflection experiments were per-
formed on the ATT to investigate the movement of the
gear under load. A proximity probe was mounted close
to the spiral-bevel gear near the mesh point to mea-
sure the gear displacement in the vertical direction
under a variety of operating conditions (Fig. 6). The
probe output signal was a voltage proportional to the
clearance as a function of time, which represented the
clearance as a function of time, on the rotating planet
bearing (thermocouple 10) were

Vibration Tests

The ATT transient accelerometer outputs were
retrieved from tape and fed to a spectrum analyzer.
The spectrum analyzer converted the analog time traces
to digital format and routed the signals to a computer
for root-mean-square (rms) average calculations as
described in Ref. 3. The attenuation experiments were
compared to previous tests on the OH-58A (3) and
OH-58C (not previously reported). The rms average
accelerations for the accelerometers mounted on the
ATT, OH-58A, and OH-58C transmissions ranged from 2 to
11 g's rms for a variety of torque conditions (Fig. 7).
These results were from tests run at a transmission
input speed of 6060 rpm for the OH-58A and OH-58C,
and 6180 rpm for the ATT (6050 rpm is the maximum design
speed of the transmission while 6180 rpm is the opera-
tional limit speed). Currently, the Army operates the
OH-58 helicopters at 6180 rpm for the additional power
capability. For five out of the seven accelerometers,
the four-planet OH-58C had slightly less vibration
(approximately 1 g rms) compared to the ATT at the
same torque levels (accelerometers 1, 2, 4, 5, and 6;
Figs. 7(a), 7(b), 7(d), 7(e), and 7(f), respectively). The
OH-58A and the ATT had about the same levels of
vibration at the same torque conditions for the remain-
ing two accelerometers. The ATT had less vibration
(2 to 3 g's rms) than the three-planet OH-58A at the
same torque levels for five out of the seven accelerom-
eters (accelerometers 2, 3, 4, 5, and 7; Figs. 7(b),
7(c), 7(d), 7(f), and 7(g), respectively). Based on
the rms average calculations of the acceleration time
traces, the general conclusion is that the OH-58C trans-
mision had slightly less vibration than the ATT, and the
ATT had less vibration than the OH-58A.

The vibration contributions of the spiral-bevel meshes and planetary meshes were determined for the
ATT, OH-58A, and OH-58C. The contributions were calcu-
lated using power spectral density functions as
described in Ref. 3. The vibration test data is shown in
Fig. 8 for the OH-58A and OH-58C at 224 kW
(300 hp) and 6060 rpm and the ATT at 233 kW (313 hp)
and 6180 rpm. The planetary mesh vibration levels were
lowest for the ATT and highest for the OH-58A for
nearly all the accelerometers (Fig. 8(a)). Thus, the number of planets (four compared to three), the planet mounting (straddle compared to cantilever), and the bearing type (cylindrical roller compared to spherical) significantly reduced planetary mesh vibration when comparing the OH-58C to the OH-58A at about full-rated power conditions. Also, the high-contact-ratio gears and the cantilever-mounted ring gear slightly reduced planetary mesh vibration when comparing the ATT and OH-58C.

For nearly all the accelerometers the spiral-bevel mesh vibration levels of the ATT and OH-58C were pretty much the same (Fig. 8(b)). Thus, the bevel gear mounting (straddle or overhung) had little effect on vibration for this speed and torque condition. For five out of the seven accelerometers the spiral-bevel mesh vibration levels of the OH-58A compared to the OH-58C were quite different (accelerometers 2 to 5). It is not known why these levels were different since the bevel systems are the same in both transmissions.

**Planet Gear Load Sharing**

The ATT ring-gear strain gage signals were retrieved from tape and analyzed with an analog-to-digital converter and a computer. Typical traces of ring gear stresses as a function of planet-pass cycles are shown in Fig. 9(a) where the ATT operated at 373 kW (500 hp) and 6180 rpm transmission input speed. The ring gear strain gage signals from the OH-60A stress experiments (4) were retrieved from tape and analyzed in a similar manner as the ATT's. The results are shown in Fig. 9(b) for the OH-60A run at 2109 kW (2828 hp) and 258 rpm output rotor speed. These were the full-rated powers and speeds of both transmissions.

In Fig. 9, the peak stress values were connected by a dotted line for each transmission. Compare the trends of the values of the peak stresses for each of the two transmissions. The maximum peak stress in the meshing of a planet gear with the ring gear at the location of the strain gage. The data were analyzed using 50-peak traces even though only 20 were plotted in the figure. For the ATT the mean peak value was 190.2 MPa (27 590 psi). The standard deviation was 5.7 MPa (830 psi), which corresponded to a 3-percent coefficient of variation (the coefficient of variation equals the standard deviation value divided by the mean value, times 100). In comparison, the OH-60A had a mean peak value of 58.0 MPa (8410 psi) and an 8.0-MPa (1160-psi) standard deviation. This gave a 14-percent coefficient of variation. In addition, the ATT's maximum peak value was 71.0 MPa (10 300 psi), 22 percent above the mean. Thus, the ATT had improved planetary load sharing compared to that of the OH-60A.

Planetary load sharing is a function of many items such as sun gear, planet gear, or ring gear stiffness, number of planets, type of mounting, etc. The ATT and OH-60A transmission are quite different in design, size, and power capacity. A direct comparison of the two is an oversimplification. The flexible ring gear of the ATT, however, had a significant role in the improved planetary load sharing compared to the rigid ring gear of the OH-60A. The flexible ring gear also has a significant role in decreasing potential overloading caused by uneven load distribution among the planets.

Note the shape of the stress-cycle curves for the two transmissions. The ATT results indicated the ring gear was loaded in a rather smooth manner while the OH-60A results implied the ring gear had impact loading characteristics. Also note that the magnitudes of the ATT peak stresses were about three times higher than that of the OH-60A. This was expected due to the thin wall of the ATT ring gear, thus allowing greater deflections during loading. Follow-up analytical studies with the ring gears approximated as internally-pressurized, thin-walled cylinders were performed. These studies predicted that the ATT ring gear had a higher hoop stress value than the OH-60A. This was consistent with the results from the experimental strain tests.

**Efficiency Tests**

The ATT mechanical efficiencies ranged from 97.73 to 98.64 percent for a variety of torques and oil temperatures and 6180 rpm transmission input speed (Fig. 10). The efficiency was 98.64 percent at higher torque levels and decreased at lower levels. At a given torque the efficiency increased at higher oil temperatures due to the decrease in oil viscosity and density. The ATT mechanical efficiencies were 98.51 and 98.62 percent at full-rated power conditions (373 kW (500 hp) at 6180 rpm transmission input speed) for oil-inlet temperatures of 82 °C (180 °F) and 99 °C (210 °F), respectively.

The lubricant used for the tests was that labeled code K in Ref. 5. Shown on Fig. 10 are the OH-58A efficiency values taken from Ref. 5 using lubricant K and at the OH-58A's full-rated power and speed. The ATT was approximately 0.25 percent less efficient than the OH-58A at the same torque levels. OH-58C efficiency results were available (not previously reported) run with lubricant code D of Ref. 5 at an oil-inlet temperature of 82 °C (180 °F). The OH-58C mechanical efficiency was 98.84 percent at full-rated power and speed, compared to 98.60 percent for the OH-58A at the same conditions and same lubricant type. Thus, the ATT was less efficient than the three-planet OH-58C and the OH-58A.

The cause of the ATT lower efficiency is believed to be the high-contact-ratio planetary gears. From the ATT design study, BHT reported a 0.30-percent decrease in efficiency in their Model 222 planetary system when high-contact-ratio gears were used (2). This level of decrease in efficiency was consistent with the 0.25-percent decrease in efficiency measured during the ATT experiments. Also, analytical studies on gear mesh power losses indicated high-contact-ratio gears are generally less efficient than standard gears (6). The reason is the higher gear tooth sliding velocities of the high-contact-ratio tooth. The major goal of the ATT design, however, was to improve the transmission power-to-weight ratio. One way this was accomplished was employing high-contact-ratio planetary gear teeth. The high-contact-ratio mesh enabled an increase in the mesh power capability for the same size gears. There was a slight reduction in efficiency believed to be due to the high-contact-ratio gears, but the transmission as a whole maintained its relatively high mechanical efficiency.

**Spiral-Bevel Gear Deflection Tests**

The ATT dynamic proximity probe output of the spiral-bevel gear was retrieved from tape and analyzed with an analog-to-digital converter and a computer. The output represented the clearance between the gear and the probe. The initial gear-probe clearance at zero speed and load was recorded and the probe outputs at various speed and load conditions were subtracted from the initial clearance to determine the deflections of the spiral-bevel gear near the location of the mesh. The spiral-bevel gear deflections exhibited a linear relationship with respect to applied load for
torques in the range of 10 to 100 percent of full-rated. This linear relationship, however, had a no-load offset (i.e., the calculated deflection-load relationship predicted a deflection at zero torque). This indicated that there was excessive clearance in the spiral-bevel gear assembly. After a slight amount of load was applied on the gear, the clearance was removed and the gear exhibited the linear deflection-load relationship. Subtracting the no-load offset gave the linear relationship depicted in Fig. 11. Also shown in the figure are the spiral-bevel gear deflections of the OH-58A transmission from Ref. 7. The OH-58A deflection data also had their no-load offset subtracted out. Note that the ATT and OH-58A had about the same deflection-load relationship. This indicates that the spiral-bevel gear deflections in the vertical direction near the mesh point were not affected by the different bearing mounting conditions of the ATT (straddle mounted) and OH-58A (overhung mounted).

Further investigations of Fig. 11 revealed a deflection-torque slope of $0.54 \times 10^5$ P/m-m. Multiplying the deflection-torque slope by the bevel gear mean pitch radius of 3.05 cm and taking the reciprocal gave a stiffness value of $6.07 \times 10^7$ N/m (350000 lb/in.). Calculating the axial bearing stiffness value using the methods of Harris (8) gave a value of $4.24 \times 10^7$ N/m (2400000 lb/in.). Since the calculated axial bearing stiffness value was much higher than the measured gear assembly value, the gear deflection in the vertical direction near the mesh was not primarily caused by bearing deflections. The gear deflections were probably mostly due to gear body, gear shaft, or housing deflections. This offers an explanation why the deflections were not affected by the different bearing mounting of the ATT and OH-58A.

**Component Temperature Tests**

The ATT component temperatures were recorded during the efficiency experiments. Figure 12 depicts the temperature survey at 6180 rpm transmission input speed and three different torques. The transmission was enclosed with insulation and the results in the figure are for an oil-inlet temperature of 82 °C (180 °F). OH-58A component temperature data are also shown in the figure for a transmission input speed of 6060 rpm. The OH-58A data were taken from efficiency studies not previously published, with the transmission insulated and maintained at an 82 °C (180 °F) oil-inlet temperature.

The temperatures measured higher for the ATT compared to the OH-58A for nine of the eleven components instrumented. This was expected because the ATT was operating at higher torque values than those of the OH-58A. The measured oil flow rates were about 23 l/min (6 gpm) for the ATT and about 19 l/min (5 gpm) for the OH-58A. The measured temperatures ranged from 78 °C (172 °F) to 116 °C (241 °F). The spiral-bevel gear pinion components had the highest temperatures (thermocouples 1 to 6). Thermocouples 4 to 6 were mounted near the bevel gear out-of-mesh location to measure oil temperature as it was slung from the gear mesh. An interesting observation was that the ATT fling-off temperature distribution along the gear face was reversed compared to the OH-58A temperatures for thermocouples 4, 5, and 6, which were in ascending order for the ATT and descending order for the OH-58A). The planet bearings (thermocouple 10) measured low temperatures for both the ATT and OH-58A.

**SUMMARY OF RESULTS**

Experimental tests were performed on the NASA/Bell Helicopter Textron (BHT) 500-hp advanced technology transmission (ATT) at the NASA Lewis Research Center. The ATT was a retrofit of the OH-58C helicopter 236-kW (317-hp) main rotor transmission, upgraded to 373 kW (500 hp), with a design goal of retaining long life with a minimum increase in cost, weight, and size. The tests were conducted in the NASA Lewis 500-hp helicopter transmission test stand, covering a range of torque and speed conditions. Vibration, strain, efficiency, deflection, and temperature experiments were performed and the results were compared to previous experiments on the OH-58A, OH-58C, and UH-60A transmissions. The following results were obtained:

1. The ATT successfully operated at the full-rated design load of 373 kW (500 hp) at 6180 rpm transmission input speed.
2. Based on the overall rms average acceleration values, the OH-58C transmission had slightly less vibration than the ATT, and the ATT had less vibration than the OH-58A transmission.
3. The ATT planetary mesh vibration levels were slightly lower compared to those of the OH-58C transmission. Thus, the high-contact-ratio gears and the cantilever-mounted, flexible ring gear reduced vibration. The OH-58A planetary mesh vibration levels were higher compared to those of the OH-58C transmission. Thus, increasing the number of planets and straddle mounting the planets reduced vibration.
4. The ATT flexible ring gear improved planetary load sharing compared to that of the rigid ring gear of the OH-60A transmission.
5. The ATT mechanical efficiency was lower than that of the OH-58A transmission, probably due to the high-contact-ratio planetary gears.
6. The spiral-bevel gear deflections in the vertical direction near the mesh point were not affected by the different bearing mounting conditions of the ATT (straddle mounted) compared to the OH-58A (overhung mounted).

In general, the component temperatures measured higher for the ATT compared to the OH-58A at full-rated powers of both transmissions, probably due to the higher torque values of the ATT.

**REFERENCES**


APPENDIX

UH-60A Main Rotor Transmission

The UH-60A (Blackhawk) transmission has a twin engine power rating of 2109 kW (2828 hp) at an output rotor speed of 258 rpm. The transmission is a three-stage reduction gearbox consisting of two input modules and one main module. For the first reduction stage the two input modules each contain a spiral-bevel gear mesh. The second and third reduction stage is contained in the main module and accomplished by a spiral-bevel mesh and five-planet planetary system. The overall reduction ratio is 81.042:1. A more complete description of the transmission is given in Ref. 4.

NASA 3000-hp Helicopter Transmission Test Stand

The UH-60A (Blackhawk) transmission was tested in the NASA Lewis 3000-hp helicopter transmission test stand. The test stand consists of three regenerative power-flow loops (two inputs to the UH-60A transmission simulating the twin engine inputs, and a tail rotor output), all recirculating through the main rotor output. Torque through each loop is controlled independently as well as test stand speed. The test stand also has the capability of applying forces to the transmission output shaft simulating flight lift, moment, and drag rotor loads. A more detailed description of the test stand is given in Ref. 4.

FIGURE 1. OH-58A HELICOPTER MAIN ROTOR TRANSMISSION.
I. OUTPUT

- Shot-peened VIM-VAR 9310 Steel Bevel Gears
- VIM-VAR M-50 Steel Bearings

Figure 2. - NASA/Bell Helicopter Textron 500-HP Advanced Technology Transmission.
FIGURE 4. - ACCELEROMETER LOCATIONS ON OH-58A, OH-58C, AND ADVANCED TECHNOLOGY TRANSMISSION.
(a) FLEXIBLE RING GEAR, ADVANCED TECHNOLOGY TRANSMISSION.

(b) RIGID RING GEAR, UH-60A TRANSMISSION.

FIGURE 5. - STRAIN GAGE LOCATIONS ON ADVANCED TECHNOLOGY TRANSMISSION AND UH-60A TRANSMISSION RING GEARS.
FIGURE 6. - PROXIMITY PROBE AND THERMOCOUPLE LOCATIONS IN ADVANCED TECHNOLOGY TRANSMISSION (LOCATIONS ALSO APPLY TO OH-58A).
FIGURE 7. - TRANSMISSION VIBRATION LEVELS OF OH-58A, OH-58C, AND ADVANCED TECHNOLOGY TRANSMISSION AT FULL-RATED SPEED.
Figures 8 and 9 illustrate vibration levels and ring gear stress for OH-58A, OH-58C, and an advanced technology transmission. The planary mesh vibration levels and spiral-bevel mesh vibration levels are shown in Figures 8(a) and 8(b), respectively. Figure 9(a) depicts flexible ring gear stress for the advanced technology transmission at full-rated power, while Figure 9(b) shows rigid ring gear stress for the OH-60A transmission. These figures indicate load sharing among planets, demonstrating the improved performance of the advanced technology transmission.
OIL-INLET TEMPERATURE, \(0^\circ C\)

OPEN SYMBOLS DENOTE OH-58A
SOLID SYMBOLS DENOTE ADVANCED TECHNOLOGY TRANSMISSION

FIGURE 10. - MECHANICAL EFFICIENCIES OF OH-58A AND ADVANCED TECHNOLOGY TRANSMISSION AT FULL-RATED SPEED. LUBRICANT, TYPE K OF REFERENCE 5.
FIGURE 11. MESH POINT BEVEL GEAR DEFLECTIONS IN VERTICAL DIRECTION FOR OH-58A AND ADVANCED TECHNOLOGY TRANSMISSION AT FULL-RATED SPEED.
FULL-RATED TORQUE, %

△ 100
○ 75
□ 50

OPEN SYMBOLS DENOTE OH-58A
SOLID SYMBOLS DENOTE ADVANCED TECHNOLOGY TRANSMISSION

FIGURE 12. - COMPONENT TEMPERATURES OF OH-58A AND ADVANCED TECHNOLOGY TRANSMISSION AT FULL-RATED SPEED. TRANSMISSION OIL-INLET TEMPERATURE, 82 °C (180 °F). TRANSMISSION INSULATED.
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