NUCLEAR ROCKET OPERATIONS
SACRAMENTO PLANT

REPORT NO. RN-DR-0131
TO

ENDURANCE TESTING OF TURBINE POWER CONTROL
VALVE ACTUATOR BENDIX MODEL
NO. NT-D1, SN 17

NERVA PROGRAM

CONTRACT SNP-1

NOVEMBER 1967

MASTER

NUCLEAR PRODUCTS AND SERVICES GROUP

AEROJET-GENERAL CORPORATION
SACRAMENTO, CALIFORNIA

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REPORT NO. RN-DR-0131
ENDURANCE TESTING OF TURBINE POWER CONTROL
VALVE ACTUATOR, BENDIX MODEL
NO. NT-D1, SN 17

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Bendix Failure Analysis of the Nutator Transmission of Turbine
Power Control Valve Actuator Model NT-D1, Serial No. 17

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I. INTRODUCTION

The data in this report were obtained from the Endurance Test portion of the qualification test program, which was performed on the Bendix Model NT-D1 Turbine Power Control Valve Actuator, Serial No. 17 and completed in the NRO Controls Laboratory on 15 June 1967.

The objectives of the actuator life test are to determine the service life of the actuator under functional and environmental conditions, with the exception of radiation, that simulate those expected during actual engine operating conditions and to determine the rate of deterioration of the actuator's performance in these environments (Reference RN-S-0425).

II. SUMMARY

Prior to this test, the test actuator had been subjected to 60.4 hours of usage under various conditions. The endurance test consisted of sinusoidal cycling of the actuator at 0.3 cps against a resistive torque whose peak amplitudes were 400 lb-in. in the clockwise (opening) direction, and 80 lb-in. in the counterclockwise direction (looking into the actuator output shaft end). During the torque cycling, the actuator mounting flange temperature was reduced during a 1-hr period to -230°F; increased to +600°F during a 2 to 2-1/2 hr period, then was cooled to 400°F, at which point the torque cycling was stopped. Later information revealed that the output shaft had inadvertently reached temperatures of approximately 1000°F (1460°R). Total time of endurance test cycle, including the time to return to ambient temperature and to complete the performance reference test was 8 hr. Total time of operation for this test actuator is 68.4 hr; NDC51A, the criterion to which this model TPCV actuator was procured, established the operating life at 20 hr.

Pretest and posttest data, as well as data recorded during the test, were not significantly different. In view of this fact and the extensive

prior usage, the actuator is considered to have exhibited sufficient endurance for application in the technology engine program.

Following the endurance test and during a test stand torque calibration test, it was discovered that the actuator exhibited limit cycling when stepped to a certain position. Further investigation revealed that the actuator was prone to limit cycling at intervals of 1.3° of actuator position, while operation between these positions appeared normal. The limit cycling was at a frequency of 25 cps and at a 0.3° peak-to-peak amplitude. Subsequent disassembly of the actuator revealed transmission gear wear and rough-running bearings. This bearing and gear wear may have resulted from temperatures at the input shaft that were higher than originally planned. Discoloration of the shaft and seal was evidence of the high temperature. However, inadequacies in the transmission gearing design appear to be major and possibly more significant contributing factors.

III. TECHNICAL DISCUSSION

A. TEST CONDITIONS

The test item was a Turbine Power Control Valve (TPCV) Actuator, Bendix Model No. NT-D1, Bendix Part No. 2775030, Serial No. 17, (referred to as the actuator in this report).

Prior to the endurance test described in this report, the assembly had been subjected to a total of 60.4 hr of actuation under various conditions and for various purposes. Table 1 presents a summary of the data.

Included in the preceding value of 60.4 hr is a 4-hr endurance test stand calibration period, during which the actuator was sinusoidally cycled at 0.1 cps against an opposing torque of approximately ± 80 lb-in. In addition, during the last 1-1/2 hr of cycling, the actuator temperature was reduced to -155°F before being returned to room temperature.

B. ENDURANCE TEST SETUP

Figure 1 shows the test fixture assembly. Schematics of the test console, actuator servoamplifier, and a block diagram of the test setup are shown in Figures 2, 3, and 4. The actuator output shaft is coupled to a shaft that passes through a dummy valve body to an Inland torque motor. The Inland torque motor is capable of providing a resistive torque that can be modified to simulate actual TPCV resistive torque while the actuator is being subjected to the simulated environments of temperature, pressure, and flow. Figure 5 shows a schematic of the Inland torque motor electrical system.

The whole Inland torque motor is enclosed in a pressure vessel and is sealed to the dummy TPCV body on the side opposite from the actuator mounting flange to preclude the need for a high-pressure rotating seal interface between the two.

TABLE 1

SUMMARY OF OPERATIONAL TIME ON ACTUATOR NO. 17
PRIOR TO THE ENDURANCE TESTS

<u>Activity</u>	<u>Hours</u>	<u>Comments</u>
Assembly, Run-In, and Performance Tests	1.5	Performed at Bendix
Acceptance Tests	2.5	*
Vibration Tests	13.3	
Hot TPCV Valve Testing	20.7	Performed in "H" area
Humidity Test	1.5	
Drop Test	0.3	
Computer Work (Analog Engine Simulation)	7.5	
Miscellaneous Pressure, Torque Tests	9.1	
Endurance Test Stand Calibration	<u>4.0</u>	
Total Testing Prior to Endurance Test	60.4	

*Unless otherwise noted all tests were performed in the
NRO Controls Laboratory at the Aerojet Nimbus facility

Figure 1 - Actuator Test Fixture Drawing and Photos
Sheet 1 of 3

Figure 1 - Actuator Test Fixture Drawing and Photos

Sheet 2 of 3

Figure 1 - Actuator Test Fixture Drawing and Photos

Sheet 3 of 3

Figure 2 - Schematic of Test Console

Figure 3 - Schematic of Actuator Servoamplifier

Figure 4 - Block Diagram of Endurance Test Setup

Figure 5 - Inland Torque Motor Command Signal Input

The dummy TPCV body is fitted with a collar in the vicinity of the actuator mounting flange, which is suitable for retaining liquid nitrogen for low temperature, and with a wrap-around flange heating element just below the LN₂ collar for high-temperature conditioning.

During the temperature torque cycling tests, continuous recordings were made of the actuator command signal and feedback voltages, of the Inland torque motor command signal voltage, and of the eight thermocouples that were positioned as follows: two were on the actuator base and one each was on the actuator gear motor, actuator potentiometer base, Inland torque motor base, Inland torque motor top, heat exchanger, and helium gas inlet fitting. An additional thermocouple was used as a high-temperature controller input signal. In addition, the actuator inlet pressure, the dummy TPCV body pressure, and the torque motor enclosure pressure were continuously observed.

The test system electrical setup is shown in the schematic diagram Figure 2.

C. SEQUENCE OF OPERATIONS

Following the installation of the actuator in the torque motor stand, testing was performed as follows:

1. Amplifier Gain Test
2. Performance Reference Test
 - a. Step Response Test
 - b. Frequency Response Test
 - c. Threshold Test
3. Temperature Torque Cycling Test
4. Performance Reference Test (Repeat (2) above)
5. Amplifier Gain Test

D. PROCEDURES AND RESULTS

1. Amplifier Gain Test

a. Procedure

With the actuator installed into the test stand but with the shaft spline coupling disconnected (i.e., no load), and no gaseous He supply to the gear motor, amplifier output current was measured for various known input voltages.

b. Results

A curve of amplifier gain in milliamps/millivolt before and after the temperature-torque cycling test is shown in Figure 6. The curves indicate that the gain is virtually linear in the range tested, and no change in gain occurred during the temperature torque cycling test.

2. Step Response Test

a. Procedure

With the unit installed as noted in Section III.D.1.a but with actuator helium supply pressure at 215 psia, the actuator was positioned to 45° nominal (5 volts on the recorder command trace). By means of the function generator, a 0.33 cps $\pm$ 1° square wave was superimposed on the nominal position signal.

b. Results

The following data show the response values before and after the temperature torque cycling test:

Figure 6 - Amplifier Gain Tests Before and After Temperature-Torque Cycling

	<u>Before</u>	<u>After</u>
Rise time, sec	0.025	0.015
Overshoot, %	16.0	19.0
Settling time, sec	0.088	0.19

3. Frequency Response Test

a. Procedure

With the actuator installed as indicated in Section III.D.2.a and by the use of a function generator, a $\pm 1^\circ$ sine wave was superimposed on the nominal position signal at frequencies of .01, 0.1, 0.5, 0.8, 1, 2, 4, 6, 8, 10, 12, 16, 20, 25, and 30 cps. A Weston, Boonshaft, and Fuchs Transfer Function Analyzer, Model No. DA-410A, S/N 6001, was used in the test to provide a digital readout of the phase angle and amplitude ratio, from which Bode plots were obtained.

b. Results

Figures 7 and 8 are plots of amplitude and phase angle-vs-frequency, respectively, resulting from the tests made before and after the temperature torque cycling test. Figure 9 is a plot of the amplitude ratios before and after, and Figure 10 shows both phase lags for comparative purposes.

The before and after figures show that the differences in gain and phase plots are not significant and can be attributed to normal system nonlinearities and accuracy of measurements; i.e., these data indicated no appreciable change in performance resulting from the test.

The crossover frequency (where the phase lag is 90°) is 20.5 and 21.8 cps for the before and after testing, respectively. The specification requires that the crossover frequency will be greater than 8 cps.

Figure 7 - Gain and Phase Lag vs Frequency Before TPCV
Temperature-Torque Cycling Test

Figure 8 - Gain and Phase Lag vs Frequency After TPCV
Temperature-Torque Cycling Test

Figure 9 - Amplitude Ratios Before and After Temperature-Torque Cycling Test

Figure 10 - Phase Angle Before and After Temperature-Torque Cycling Test

4. Threshold Test

a. Procedure

The initial test conditions were identical to those of Section III.D.3.a, and the sine-wave frequency was 0.1 cps. The amplitude of the sine wave was increased then decreased as required to indicate the presence of, then the lack of response to the command until the threshold of the response was determined.

b. Results

The threshold before and after the temperature torque-cycling test was $\pm 0.075^\circ$.

5. Temperature Torque Cycling Test

a. Procedure

The test unit (still mounted on the stand) was connected to the test stand torque motor by means of the splined coupling. The electrical portion of the test setup was in accordance with Figures 2, 3, 4, and 5.

With the test specimen electrically energized and the command potentiometers set at a nominal position of 45° (5 volts), a pressure of 215 psia was applied to the actuator inlet. The test stand helium pressure was then increased to 100 psi with a continuous slow bleed off through the hand vent valve (Figure 1).

The test stand Inland torque motor command potentiometer was energized and adjusted to null relative to the test stand feedback potentiometer. Then the torque motor was energized.

Again, by means of the function generator, a 0.3 cps sine wave was superimposed onto the 45° command position signal with sufficient amplitude to generate ± 240 lb-in. of torque at the peak, (by driving the test stand torque motor off its setpoint). The test specimen nominal command signal was then adjusted so that the peak torque in the TPCV opening direction was 400 lb-in. while the peak was 80 lb-in. in the closing direction. This mode of torque cycling was continued throughout the 5-hour duration of the test, while the actuator interface flange temperature was first cooled to -265°F, then heated to +600°F in accordance with the temperature-time profile shown in Figure 11. Also as shown in Figure 10 the output shaft dynamic seal pressure was set at 100 psia initially, raised to 200 psia during the warm up period, then to 300 psia during the heat soak period, and finally reduced to atmosphere, at which point heating was terminated.

c. Results

During the temperature torque cycling test, no changes in actuator performance were evident as indicated by the values of actuator current and feedback voltage, the Inland torque motor preamp voltage input, and feedback signals. Figures 12, 13, 14, and 15 show actual traces of the indicated parameters taken (1) at the beginning of the test, (2) during the low temperature phase, (3) during the high temperature phase, and (4) at the end of the test.

As already indicated, according to the results of the before and after performance reference tests, apparently no significant changes in actuator performance occurred as a result of the test. Bode plots of the frequency data for both the before and after performance reference tests are shown in Figures 7 through 10 for purposes of comparison as indicated in Section III.D.3.b.

Figure 11 - Time-Temperature Profile, TPCV Actuator
Temperature-Torque Cycling Test, 14 June 1967

Figure 12 - Beginning of Temperature Torque Cycling Test

Figure 13 - Low-Temperature Portion of Temperature-Torque Cycling Test

Figure 14 - High-Temperature Portion of Temperature-Torque Cycling Test

Figure 15 - End of Temperature-Torque Cycling

Following the endurance test and during a subsequent torque calibration test, a limit cycling of the actuator was encountered at one specific actuator position (in the vicinity of the 45° set point used during the test). This condition prompted a search to determine if the actuator could be made to limit cycle at other positions as well. The search revealed that the actuator was prone to limit cycling at approximately 1.3° intervals of actuator position throughout its entire 90° operating range, at a frequency and amplitude of approximately 25 cps and 0.3° peak-to-peak, respectively. (Note also that from the results of the amplifier gain test, the actuator servo amplifier gain had not changed during the test).

In view of the preceding circumstances, it was concluded that some change may have occurred within the actuator during (or after) the temperature torque cycling test, and an investigation of the irregular performance was made.

Examination of the actuator when it was removed from the endurance test stand revealed that discoloration was visible on the actuator output shaft and the actuator mounting flange (Figure 16). Also the output shaft coupling appeared to have been subjected to severe heating.

Subsequent disassembly of the actuator revealed the following:

1. Discoloration of the output shaft had progressed to the inside face of the shaft, leaving that area a gold color.
2. The actuator bearings were somewhat rough running, particularly those at the actuator output end. ✓

Figure 16 - Actuator Output Shaft After Temperature-Torque
Cycling Test

3. The actuator transmission gear teeth appeared to have been fretted or galled, a condition that was most evident at or near the ends of the teeth. In addition, some of the teeth appeared to have been shaved down the face, with the shaving (in each case) deposited near the root of the tooth (Figures 17, 18, and 19). This shaving condition appeared to be the most severe on the driven gear. The condition was more severe on the side resisting the 400 lb-in. torque load, which was applied during the temperature torque cycling test.

4. Manual rotation of the gear motor required an irregular application of torque (with respect to position). Subsequent measurement of torque provided the following data:

- a. Running torque; 18 to 25 in.-grams, clockwise;
20 to 30 in.-grams, counterclockwise.
- b. Breakaway torque: 29 in.-grams, clockwise;
32 in.-grams, counterclockwise (maximum).

5. On disassembly of the gear motor, the face of the gear teeth had some discontinuity of the molybdenum disulfide coating. However, this condition was only visible under high magnification, and probably had no adverse effect on gear motor operation.

6. The gear motor body had a small groove running around the inlet side of one of the driving gear cavities. Apparently a particle had been lodged between the tip of a tooth and the body during a portion of the gear's rotation (Figure 20).

7. A test of the two potentiometers was conducted to determine if they had been subject to adverse effects during the temperature torque cycling test.

Figure 17 - Transmission Gear Teeth After Temperature-Torque Test, High Torque (400-in.-lb) Side

Figure 18 - Transmission Gear Teeth After Temperature-Torque Test, Low Torque (80-in.-lb) Side

Figure 19 - Closeup of Gear Teeth After Temperature-Torque Test

Figure 20 - Gear Motor Body Showing Posttest Scratch

The test consisted of a standard linearity check described schematically in Figures 21 and 22.

The results of these tests are also shown in these figures. The data are typical for potentiometers of this type and are within specification limits.

6. Posttest Evaluation

a. Test Temperature

As already indicated, due to the degree of discoloration of the actuator input shaft, compared with that of the input flange (which was instrumented), it was suspected that the shaft reached higher temperatures during the endurance test than those recorded at the flange. To make an approximate determination of the shaft temperature, a series of Inconel samples were subjected to 2-hour helium atmosphere soaks, at temperatures ranging from 600°F to 1500°F in increments of 100°. The helium was taken directly from the supply system used for the endurance test. Discoloration of the samples resulted in a color spectrum (Figure 23) and various parts of the actuator were compared with this spectrum. The comparison indicated that the input shaft temperature reached approximately 900° to 1000°F (conservatively). Figure 23 shows areas on the output shaft together with the corresponding sample.

b. Transmission Gear Tooth Wear

Because of the uneven transmission gear tooth wear, and the loss of or change in gear tooth surface that was detected following the endurance test, the transmission design was reviewed as a possible causal factor.

Figure 21. Potentiometer Posttest Schematic and Results
(Terminals 1, 2, and 3)

Figure 22. Potentiometer Posttest Schematic and Results
(Terminals 16, 17, and 18)

POST-TEST VIEW SHOWING MATCHING PORTION
OF OUTPUT SHAFT FLANGE

700°F - NO COLOR CHANGE

800°F - SLIGHT GOLD
TINGE SHOWING

900°F - DEFINITE GOLD
COLOR WITH TRACES OF
PURPLE

1000°F - DEEP BLUE AND
PURPLE

1100°F - PREDOMINANTLY
BLUE

1200°F - BLUE TO GREEN

1400°F - LIGHT GREEN TO
BLACK

1500°F - DEEPER GREEN -
BLACK

COMPARISON OF COLOR CHANGES WITH
EXPOSURE OF INCONEL SPECIMENS IN
A HELIUM ATMOSPHERE

CLOSE UP OF 900°F COLOR SAMPLE AND
CORRESPONDING OUTPUT SHAFT COLOR

CLOSE UP OF 1000°F COLOR SAMPLE AND
CORRESPONDING OUTPUT SHAFT COLOR

Figure 23 - Posttest Views of Input Shaft Flange and
Corresponding Temperature Samples

Although it was apparent that the initial choice of base materials and heat treatment of the gear teeth left room for improvement, spectrographic analysis of the shavings found near the root of the teeth indicated the shavings did not contain iron, i.e., no 17-4 PH steel base material. Rather, the shavings were composed mainly of silver, with traces of molybdenum and silicon, which are constituents of the gear tooth coating only, (a silver plate with baked-on molydisulfide coating). The basic gear material, therefore, must be considered adequate, despite the merits of the initial choice.

c. Gear Tooth Configuration

Relative to gear tooth configuration, a preliminary 10x-size layout of the teeth in mesh indicated that due to a difference in the pressure angle of the output and input gears, the sharp corners of gear tooth tips were contacting the mating teeth and causing scuffing or shaving down of the faces.

A subsequent extensive investigation of this possibility was accomplished at Bendix, the results of which substantially verified preliminary findings by Aerojet. A complete report of this Bendix study is contained in Bendix Research Laboratories Report No. CA-23105, "Failure Analysis of the Nutator Transmission of Turbine Power Control Valve Actuator Model NT-D1, Serial No. 17," which is included in this report as Appendix A.

d. Seal

Of interest in the actuator post-endurance test evaluation was the condition of the seal.

Microscopic examination of the seal showed it to be in good condition. Some buildups were present, which appeared to be carbon or outgassed carbon binder material due to the high-temperature environment. The seal surface was a 4 microin. finish with a 6 microin. circumferential scratch, which probably did not affect the sealing function.

7. Conclusions

a. The results of the endurance testing described in this report, together with the extensive operating time the test actuator S/N 17 was subjected to before the start of the Endurance Test, verify that the units useful life is significantly longer than will be required for the XE testing application. In addition the high-temperature environmental conditions to which the actuator was subjected during the temperature torque cycling test were much more severe than had been originally intended.

b. Despite the preceding conclusion, the tests helped to reveal inadequacies in the actuator transmission gear design, which led to corresponding improvement of future hardware. Specifically the nutating gear has been redesigned with (1) a larger pressure angle, (2) a decreased addendum, and (3) radii at the tips and a decreased root radius, all of which are intended to eliminate the shaving action. Appendix A shows the details of these changes.

The changes will be effective on the Model NT-D2 actuators that are shortly to be delivered. Therefore, it is recommended that the endurance test described in this report be repeated using the new design. Also, the test equipment is to be modified so that the input shaft temperature will be used for control in place of the valve-actuator interface flange, in order to eliminate overheating.

APPENDIX

BENDIX FAILURE ANALYSIS OF THE
NUTATOR TRANSMISSION OF TURBINE
POWER CONTROL VALVE ACTUATOR MODEL
NT-D1, SERIAL NUMBER 17

ATTACHMENT

FAILURE ANALYSIS OF THE NUTATOR TRANSMISSION OF TURBINE
POWER CONTROL VALVE ACTUATOR
MODEL NT-D1, SERIAL NUMBER 17

INTRODUCTION

After completing one life cycle in the life test, TPCV actuator, Model NT-D1, Serial No. 17 was found to limit-cycle. The rotation of the output shaft was checked and found to be running rough. Aerojet-General personnel carefully disassembled the actuator and examined the transmission. Metal chips were discovered at the bottom of the gear teeth of the output gear, and the gear teeth faces appeared as though they were abnormally worn over about one-third of their area.

Bendix personnel visited Aerojet-General to examine the actuator parts, particularly the transmission gears, and to review the history of the actuator.

This attachment presents the detailed failure analysis of the transmission of actuator Serial No. 17. Included in the following sections are a description of the failed gears, a history of actuator No. 17, a history of the nutator transmission for TPCV actuators, an analysis of the gear teeth geometry, a review of gear materials, and conclusions and recommendations.

1 DESCRIPTION OF NUTATOR GEARS OF TRANSMISSION UNDER ANALYSIS

A sketch of the transmission (Part No. 2775033) used in the Model NT-D1 TPCV actuators is shown in Figure 1-1. In this sketch the output gear (Part No. 2775728) and the nutating gear (Part No. 2775815) under analysis are indicated. Noted in the sketch is vertex of the pitch cones of the two gears which is also the theoretical point about which the nutating gear nutates. The plus and minus direction of rotation are indicated.

1.1 Output Gear (Part No. 2775728)

Of the two gears, the output gear showed the most significant signs of wear. There was tooth contact throughout the entire length of the teeth, but the areas at the outer and inner portions of the tooth face showed signs of aggravated wear or scuffing. (See Figure 1-2.) On almost every tooth face that would be in contact when moving against the 400 in-lb load, there was an area at the outer portion of the tooth face that showed signs of scuffing with a metal chip at the bottom of tooth face. This area had the most significant signs of wear. It is not known at this time whether the chips at the bottom of the teeth are chips of the base metal of the gear or of the plating applied in the lubrication process. The metal chips can be seen in the photograph of the gear in Figure 1-3. There were also indications of scuffing on the inner portion of the tooth faces but they were not nearly so severe as those at the outer portion.

On the faces in contact when opposing the 80 in-lb load, there were signs of scuffing and metal chips on the inner portion of almost every tooth face. (See Figure 1-4.) Although there were metal chips in nearly every tooth, the wear was not so severe as on the outer portion of the opposite face. There were also signs of scuffing at the outer portion of the tooth face and metal chips in occasional teeth.

1.2 Nutating Gear (Part No. 2775815)

The nutating gear showed less signs of aggravated wear than the output gear. There was tooth contact along the entire length of tooth, but there were signs of scuffing at the inner and outer ends of the tooth face. (See Figure 1-5.) There were metal chips at the bottom of a few of the teeth, but these instances were relatively minor in comparison to the output gear. (See Figures 1-6 and 1-7.)

Figure 1-1 - Transmission Assembly (Part No. 2775033)

Figure 1-2 - Output Gear (Part No. 2775728)

Figure 1-3 - Chips at Bottom of 400 in-lb Face of Output Gear

Figure 1-4 - Chips at Bottom of 80 in-lb Face of Output Gear

Figure 1-5 - Nutating Gear (Part No. 2775815)

Figure 1-6 - Nutating Gear, Showing Wear

Figure 1-7 - Nutating Gear, Showing Wear

2 HISTORY OF ACTUATOR S/N 17

Prior to the life tests of actuator S/N 17, the actuator was subjected to and used in a number of tests. The operating time in these tests totaled 61 hours. The tests included vibration, frequency response, stall torque, and checkout tests of the actuator and the use of the actuator in engine simulation, test-stand checkout, and hot-valve tests. The life test was initiated without the refurbishment of the actuator.

In the life test, the actuator was mounted on a turbine power control valve that has been made part of the test fixture. The butterfly shaft was coupled to an Inland torquemotor to allow torque loads to be accurately programed and applied. To simulate temperature the turbine power control valve body was heated using a Calrod heater. Cryogenic temperatures were produced by cooling the valve body and actuator with liquid nitrogen. During the life tests the torque load was 400 in-lbs in a CCW and 80 in-lbs in CW direction. The input command to the actuator was a 0.3-cps sine wave with 6-degree peak-to-peak amplitude. The actuator position was 45 degrees. The time-temperature profile of the life cycle is shown in Figure 2-1. Also indicated in the figure is the output shaft dynamic seal pressure. The temperature indicated in the Figure was that measured by a thermocouple attached to the actuator at a dynamic-seal mounting screw. A life cycle was approximately five hours. Actuator performance was evaluated before and after the life cycle. After the cycle was completed, it was found that the actuator would limit-cycle and that the rotation of the output shaft without pneumatic power was rough. The actuator was carefully disassembled. The transmission gears of the actuator were found to be as described in Section 1. The total operating time of the actuator was approximately 70 hours.

Figure 2-1 - Temperature Profile in Life-Test Cycle

3 HISTORY OF TRANSMISSION DESIGN

The nutator transmission has been used in all Turbine Power Control Valve Actuators since the initiation of the B series of TPCV actuators. The transmission of the B-series actuators was designed to meet the requirements of BRL actuator specification DS-563. The actuator load as function of valve position was defined. The maximum valve load torque was 248 in-lbs; the friction load was 60 in-lbs; and the inertia load of the valve was considered negligible. The duty cycle of the actuator was not defined. The transmission designed had a ratio 82:1 and was capable of handling a 10 in-lb input torque. The gears of the transmission were made of 17-4PH. This material was selected for its high fatigue and yield strength combined with its excellent retention of ductility at cryogenic temperatures. The gear geometry, cone angle, pressure angle, and pitch selected for this initial transmission and the material selected have been used in all subsequent nutator transmissions built for TPCV actuator. Transmission Part No. 2151312 was used in the Models NT-B2 and NT-B3 actuators. Two transmissions of this design were life tested. The first transmission was run under maximum load at constant speed for four hours and three minutes. At the completion of the testing of this transmission, the gears showed significant signs of wear. A number of minor changes, including the increasing of the face width of the gears, were made in the second transmission. The second transmission was subjected to tests of temperature shock, constant-load endurance, and low-temperature cyclic load endurance in which a peak load of 383 in-lbs was applied. Tests of this transmission were concluded after 7.33 hours. While there were indications of wear at the conclusion of the tests of the second transmission, they were not considered excessive. The details of the testing of the transmission are presented in report number BPAD 863-4-15122R.

The NT-B4 actuator used transmission Part No. 2775005. This transmission was essentially the same transmission as Part No. 2151312 except that the single double-row angular-contact bearing was replaced with a pair of angular-contact bearings.

In the Model NT-C actuators, some refinements were made in transmission design. The face width of gears was increased but the gear geometry remained unchanged.

The Model NT-D1 actuators were designed to AGC specification NDC-51A. In this specification, a maximum output torque of 600 in-lbs and life of 20 hours were specified. No duty cycle was specified. The transmission was redesigned but gear geometry and material were retained.

The transmission of the Model NT-D2 actuator is essentially the same as that of Model NT-D1 actuator. In the specification for the Model NT-D2 actuator, AGC 90080, maximum torque is 600 in-lb; the actuator load as function of position is defined; a life of 100 hours is specified; no duty cycle is indicated.

4 ANALYSIS OF GEAR GEOMETRY

4.1 Description of Gear Mesh

A schematic diagram of the gear teeth is shown in Figure 4-1. The pitch lines of the gears are described by cones whose vertices are coincident when the gears are properly mated. The pitch-cone angle of the nutating gear is larger than the pitch cone angle of the rotating gear. When the transmission is assembled, the center lines of the gears are displayed by an angle equal to the difference in the pitch-cone angles. The gears, then, are meshed at the pitch line on one side of the gear but are not meshed on the diametrically opposite side of the gear. A sectional view taken perpendicular to the pitch line of the meshed gear describes the specified pressure angle. In operation, the center line of the nutating gear describes a cone with the vertex at the pitch-cone vertex. As the gear nutates, it becomes disengaged from the output gear at one point and becomes meshed at another point. There is a one-tooth difference between the nutating and output gears, and for each nutation cycle, the output gear is advanced $1/N$ of a revolution (where N is the number of teeth on the output gear).

4.2 Gear Geometry as Presented on Drawings 2775815 and 2775728

The nutating gear and the output gear are manufactured according to drawings 2775815 and 2775728, respectively. In order to obtain proper operation of the gear set, the pitch-cone and pressure angles of the gear teeth must be accurately machined and the proper tooth configuration must be generated. The gear teeth and spaces can be considered truncated pyramids as illustrated in Figures 4-2 and 4-3. The tooth spaces (and thus the teeth) may be generated using a form tool (mill cutter or grinding wheel). The vertex of the form-tool angle (Figure 4-4) moves along the line AC of Figure 4-3. It is to be noted that the line AC is at a different angle from the pitch line. The cutting-tool angle is therefore defined when viewing the tooth space along the line AC, and must be different from the pressure angle which is defined by viewing the space along the pitch line. On drawings 2775815 and 2775728, then, the tooth space angles are given as 19 degrees, 28 minutes, 46 seconds, and 20 degrees, 35 minutes, 15 seconds, and should be used for gear manufacturing only. If the gears are cut with these angles along the proper angle for the line AC (Figure 4-3), the proper pressure angle should be obtained when the gears are mated.

4.3 Analysis of Gear Geometry

An analysis of the present gear geometry was performed to determine if the dimensional information given on drawings 2775815 and 2775728 was correct. The equations to obtain the pressure angle from the given dimensions were derived using the sketches shown in Figures 4-5 and 4-6. Refer to Figure 4-5; the following equations were derived for the nutating gear (Part No. 2775815):

Figure 4-1 - Schematic Diagram of Transmission Gear Mesh

Figure 4-2 - Typical Gear Tooth

Figure 4-3 - Typical Tooth Space

Figure 4-4 - Sketch of Gear-Cutting Tool

Figure 4-5 - Schematic Diagram of Geometry for Nutating Gear (Part No. 2775815) (83 Teeth)

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$$\tan \theta = \frac{d}{l_1} \quad (1)$$

$$\tan \theta_2 = \frac{d}{l_4} \quad (2)$$

$$l_1 = r \tan (\phi - \phi_1) \quad (3)$$

$$l_4 = r \sin (\phi - \phi_1) \quad (4)$$

where

ϕ = pitch cone angle

ϕ_1 = angle of line AC (Figure 4-3)

θ = pressure angle

θ_2 = cutting tool angle

Other dimensions are as shown on the sketches.

Substituting equations (2), (3), and (4) into equation (1) results in

$$\tan \theta = \frac{r \sin (\phi - \phi_1) \tan \theta_2}{r \tan (\phi - \phi_1)} \quad (5)$$

which can be simplified to

$$\tan \theta = \cos (\phi - \phi_1) \tan \theta_2 \quad (6)$$

Equation (6), then, relates the pressure angle to the pitch-cone angle, the cutting-tool angle and the angle along which the form tool is moved while cutting the tooth space. Similar relations can be obtained for the output gear using Figure 4-6. The final equation for the pressure angle of the gear is

$$\tan \theta' = \cos (\phi_1' - \phi') \tan \theta_2' \quad (7)$$

where the angles are as defined above. (Primed quantities refer to output gear.)

The data given for manufacturing the two gears is as follows:

Nutating Gear

$$\phi = 30 \text{ deg}$$

$$\phi_1 = 27.3486 \text{ deg}$$

$$\theta_2 = 19.4794 \text{ deg}$$

Output Gear

$$\phi = 31.175 \text{ deg}$$

$$\phi_1' = 33.6806 \text{ deg}$$

$$\theta_2' = 20.5875 \text{ deg}$$

Calculated pressure angles for the nutating and output gears are 19.4588 degrees and 20.5655 degrees, respectively. It is seen, therefore, that the manufacturing dimensions given on the drawings do not result in a pressure angle of exactly 20 degrees as specified. The gears will not properly mesh at the pitch line when assembled in the transmission.

In order to obtain a clearer picture of how the gears mesh, the gear geometry was calculated at $r = 1.25$ inches and $r = 1.62$ inches (inner and outer radii of the gear teeth measured along the pitch line). Typical gear mesh for the two radii are shown in Figures 4-7 and 4-8, which are drawn several times size. It is seen from these figures that, due to the improper angles, the gears do not mesh along the pitch line. It is also seen how the rotating gear can be "shaved" by the nutating gear as the two come into mesh. Additionally, Figure 4-7 indicates that the teeth of the gears can bottom on the radius of the spaces at the smaller values of radius r . This correlates with the wear pattern of the faces of the teeth of both gears.

The transmission gears can be reworked to obtain the same pressure angle by reworking the nutator gear only. The teeth can be

Figure 4-7 - Typical Gear Mesh at $r = 1.25$ inches

Figure 4-8 - Typical Gear Mesh at $r = 1.62$ inches

machined at a slightly larger angle than the 19 degrees, 28 minutes, 46 seconds specified on drawing 2775815. The addendum of the teeth will also have to be decreased to prevent interference at the bottom of the rotating gear space. At the same time, the corners of the teeth should be broken with a slight radius to help alleviate any "shaving" action. It may also be desirable to deepen the nutating gear space by decreasing the root radius. With these modifications, the nutating gear will require shimming at assembly to assure coincidence of the pitch-cone vertices.

5 GEAR MATERIALS

The physical properties of a group of stainless steels that warrant consideration for this application, including the 17-4PH stainless steel presently being used, are presented in Table 5-1. All materials presented in this table have sufficient physical strength and ductility.

It has been indicated that the galling characteristics of the gear material may have an effect on the short life achieved by the failed parts. The 17-4PH stainless steel is considered to have relatively good wear characteristics; thus is recommended for the making of gears. The Armco Steel Corporation has conducted galling tests on various stainless steels in which loads causing galling were determined. The results of these tests are included in Table 5-1. The data indicates that the 410- or 416-series of stainless steels have greater resistance to galling than 17-4PH. The report also indicates that improved galling characteristics, approaching that of the 410 and 416 steels, could be achieved with 17-4PH by using mating parts with differing hardnesses.

In the TPCV actuator application, the mating parts are lubricated with a dry-film lubricant, and environmental and load conditions differ considerably from those of the galling tests. Therefore, it is difficult to evaluate with any degree of certainty the effect of the galling characteristics of the materials on the life they achieve. This is not to say that in future designs it would not be advisable to change the material from 17-4PH to either a series 410 or 416 stainless steel.

Table 5-1 - Material Review for Nutator Gears

Characteristic		Material									
		17-4 PH				410		416		431	
Tempering Temperature		900	925	1025*	1150	1000	300	1000	300	500	900
Tensile Strength	Room Temp. -320°F	190,000 263,000	170,000	155,000	135,000	145,000	195,000	137,000	176,000	193,000	200,000
Yield Strength	Room Temp. -320°F	170,000 243,000	155,000	145,000	105,000	115,000	150,000	110,000	138,000	148,000	156,000
% Elongation in Two Inches	Room Temp. -320°F	4-10 7	10 >7	12 >7	16 >7	20	15 7	15 10	9.5	17	18
IZOD Impact (ft-lb)	Room Temp. -320°F	**20	5	15-35	30-50	20	35	25	20	52	42
Hardness	BHN	388/488	375/438	331/401	277/352	300	390	285	363	388	415
	Rockwell C	40/47	38/45	35/42	28/32	32	41	30	38	41	44
Galling Load (psi)***			6000	6000	5000		20,000	30,000			

* Temperature of 17-4 PH used in present gears

** Minimum impact properties for 17-4 PH in position 900 are not acceptable.

*** "Stainless Steel Galling Characteristics," H. Tanczyn, Armco Steel Corporation, April 20, 1955 issue of "Steel" Magazine.

6 CONCLUSIONS AND RECOMMENDATIONS

It was concluded from the failure analysis that scuffing of the gear teeth and the resulting metal chip at the bottom of gear teeth was due to the difference in the pressure angle of the nutating and output gears and the sharp corners on the gear tooth tips. The chips found at the bottom of the teeth are probably chips of the plating material from Magnaplate process used to lubricate the gear teeth. The conclusion that the chips are from the plating process is drawn because it is not probable that the base material of gears would generate a chip, since both gears are of the same hardness. The plating material is soft in comparison to the base material and therefore a gear tooth with a sharp corner could shave the softer material.

In support of the foregoing conclusion, it is also concluded that it is unlikely that the galling characteristics of the base material contributed significantly to the failure. The gear teeth of the transmission are lubricated with a dry-film lubricant that actually prevents the contact of the base materials and lowers the coefficient of friction between the contacting faces of the gear teeth. Once the dry-film lubricant has deteriorated to the point where the base materials of the two gears come into contact, the transmission efficiency should decrease and the wear of the gear teeth increase. It is questionable to what extent that galling characteristics of the base material would limit the life of a transmission after the base materials come into contact. A review of materials indicated that the 416-series stainless steel did have better resistance to galling and could be substituted for the 17-4PH material presently being used.

It was noted in reviewing the history of the transmission design that the actuator duty cycle, loads, and environments as a function of time, have never been defined or specified. Therefore, it is not possible to establish the adequacy of the transmission for this application.

It is recommended that gear teeth of the nutating gear be changed to have the same pressure angle as the output gear and that the corners on the tips of the gear teeth be broken. It is not possible to predict to what extent the life of the transmission will be improved by correcting the pressure angles and breaking the sharp corners of the gear teeth. This can be established only through testing of the transmission. It is also recommended that the required duty cycle of the actuator be established and be made part of the actuator specification to allow adequacy of the transmission to be determined.