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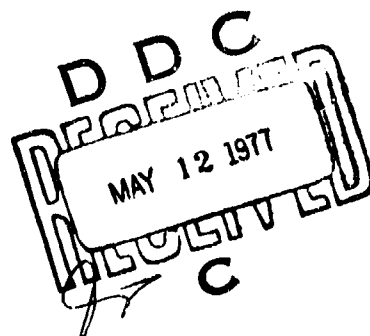


**3000-HP ROLLER GEAR TRANSMISSION DEVELOPMENT PROGRAM
Volume VI - Reliability and Maintainability Report**

Sikorsky Aircraft
Division of United Technologies
Stratford, Conn. 06602

January 1977

Final Report



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Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

This report is the final of six volumes of the final report under this contract. The objective of this program is to conduct research on the feasibility of a high-reduction-ratio, 3000-hp roller gear transmission.

This report covers the reliability and maintainability testing of the roller gear unit to determine the effect of extended time operation. A failure mode and effects analysis and a reliability analysis were conducted following a regenerative test of two roller gear drive units.

James Gomez, Jr., Propulsion Technical Area, Technology Applications Division, served as project engineer for this effort.

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20. ABSTRACT (Cont'd)

A reliability analysis shows that a fully developed roller gear transmission offers comparable reliability to a conventional two-stage planetary unit. However, relating the development testing of the roller gear transmission to project the reliability is premature at this stage.

The analysis does show that on-condition maintenance of the roller gear transmission is feasible.

A summary of the testing and development of the roller gear transmission is included. Detailed reports on the design, manufacture, and bench and aircraft testing have been published separately.

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PREFACE

This report is the final volume of six volumes dealing with the development of a 3000-horsepower roller gear transmission system. This program was conducted by Sikorsky Aircraft for the Eustis Directorate of the U. S. Army Air Mobility Research and Development Laboratory under Contract DAAJ02-69-C-0042 (Task 1G162207AA7201). The program was conducted under the auspices of Mr. James Gomez and Mr. Leonard Bartone of USAAMRDL. Mr. P. FitzGerald and Mr. L. Burroughs were the program managers at Sikorsky Aircraft.

Appreciation is extended to Mr. B. Trustee of the Reliability and Maintainability Group, Sikorsky Aircraft, for the reliability analysis of the roller gear transmission.

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INTRODUCTION

High-speed helicopter transmissions normally incorporate planetary arrangements to achieve speed reductions from the engine to the main rotor. While this type of reduction unit has operated satisfactorily, there has been continual research to develop alternate ways to obtain the reduction in speed and to increase the efficiency of the helicopter transmission. One that has emerged is the roller gear drive. As the name implies, the roller gear drive combines both gears and rollers; the rollers are used to position the gears in a planetary arrangement and to react gear teeth radial forces, the gears themselves transmit power.

A parametric study of the roller gear drive⁽¹⁾ was started in 1963 by TRW Incorporated under the direction of the inventor, Dr. A. L. Nasvytis. The roller gear concept was investigated over a range of 12,000 to 30,000 rpm, through input powers of 250 to 3000 hp and with reduction ratios of 20:1 to 100:1. During the study, no significant obstacles were found that would prevent the roller gear reduction drive from being developed, and it was concluded that the drive was a viable candidate for use in helicopter transmissions. TRW followed up the recommendations of the report with the design, fabrication, and testing of a 200-hp experimental roller gear drive unit with a 70:1 reduction ratio.⁽²⁾ Results of a 1000-hour endurance test confirmed that the roller gear drive was a high-efficiency unit. The report concluded with the recommendation that a full-sized unit be designed, fabricated, and tested to determine the technical feasibility of using the roller gear drive in a helicopter.

In 1968 and 1969, the Bell Helicopter Company of Fort Worth, Texas, conducted an engineering design study to determine the feasibility of employing the roller gear concept in a transmission for the UH-1 helicopter.⁽³⁾ This study compared the roller gear drive to the existing UH-1 transmission and a new three-stage planetary design. Five different roller gear drives of 42.8:1 reduction ratio were designed and analyzed. The drives were designed for 1140 hp with 13,890 rpm input and 324 rpm output. The primary study criteria were cost, weight, efficiency and reliability. Results showed that in the areas of efficiency and reliability, the roller gear was the potentially superior design. The roller gear drive ranked last only in fabricability/cost of the areas examined, while ranking second to the new three-stage planetary in weight.

TRW continued investigation of the concept with the design and fabrication of a 1100-hp, 11,300-rpm input, 34.8:1 reduction roller gear drive.⁽⁴⁾ Dynamic testing of the unit in a regenerative test facility showed an efficiency of 98.9 percent. Results of the test, which was halted after 76.5

hours due to test rig malfunctions, indicated that the basic design concept of the roller gear drive was sound and efficient in the transmission of high powers for high reduction ratio helicopter transmissions.

Sikorsky Aircraft first examined the roller gear drive in 1966 with a feasibility study to replace the two-stage planetary of the CH-54A helicopter main transmission with a roller gear drive reduction unit.⁽⁵⁾ It was found, however, that the required overall reduction ratio of 9.69:1 was too low to take full advantage of the benefits of a roller gear drive, and it was recommended that an evaluation of a roller gear drive be conducted at the inception of a new aircraft program where no hardware or configuration restraints existed. While the study concluded that the roller gear drive concept was not applicable to the CH-54A, this program and subsequent independent research and development efforts led to the present roller gear program.

In March 1969, Sikorsky Aircraft proposed a program to design, fabricate and test a transmission, incorporating a roller gear drive for the Sikorsky S-61 series helicopter. A contract was awarded in June 1969 for the program outlined in Figure 1. The design, manufacture, bench test and aircraft tiedown test phases of this program are reported in References 6, 7, 8 and 9, respectively.

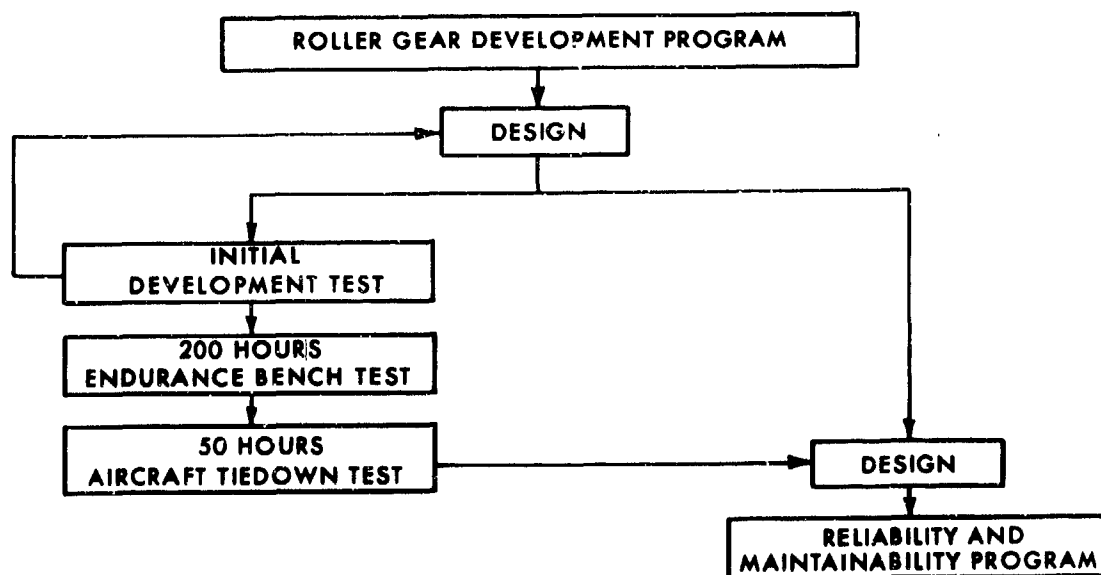


Figure 1. Outline, Roller Gear Development Program.

DISCUSSION

ROLLER GEAR DRIVE DESCRIPTION

The roller gear drive shown in Figure 2 reduces an input rotational speed of 4045 rpm at the sun gear to an output speed of 203 rpm at the ring gear through two rows of compound gears. It is configured in a star arrangement with seven first-row pinions and seven second-row pinions rotating about their own centers (Figure 3). Seven spherical roller bearings are used to locate each of the second-row pinions and react the transmitted torque. The sun gear is driven by an input quill driveshaft through a loose internal spline and is supported in the axial direction by the rollers of the seven first-row pinions. Each first-row pinion contains two outer spur gears which mate with the sun gear and an inner spur gear which mates with two second-row pinions. The first-row pinions are accurately positioned at one inside point by the rollers of the sun gear and at two outside points by the rollers of the second-row pinion. This three-point support is inherently stable and dispenses with the need for a bearing support. The inner rollers of the first-row pinions contain flanges which constrain these pinions in the axial direction.

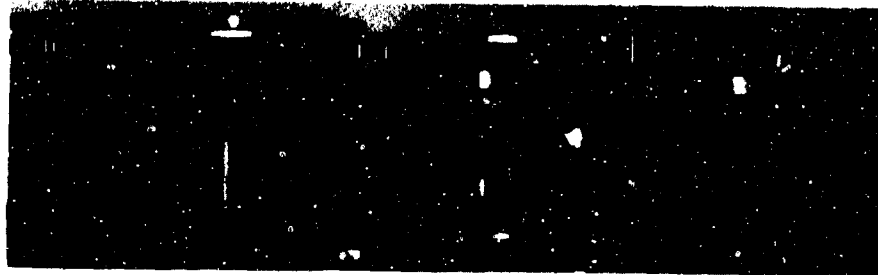
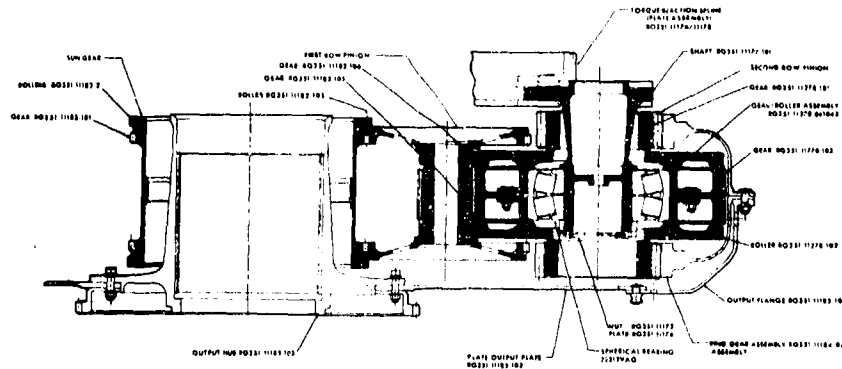


Figure 2. Roller Gear Drive.

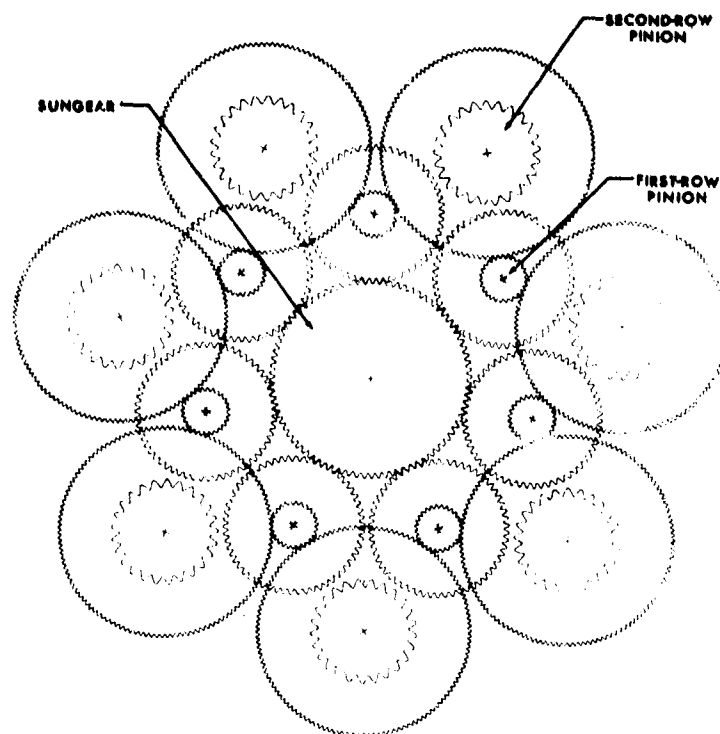


Figure 3. Star Arrangement, Roller Gear Unit.

The second-row pinions are dynamically positioned by the rollers of the first-row pinion and the gear mesh forces from the second-row pinion and ring gear. Spherical bearings provide axial positioning and react the transmitted torque. The internal clearance of these bearings is such that under the worst case of roller tolerances and deflections, the bearing does not react loads in the radial direction; only tangential loads resulting from the torque through the roller gear drive are reacted.

The basic data of the roller gear components is presented in Table 1.

TABLE 1. BASIC GEAR DATA: ROLLER GEAR COMPONENTS.				
Gear	Number of Teeth	Pitch Diameter (in.)	Diametral Pitch	Pressure Angle (deg)
Sun Gear	84	8.89077	9.448	22.5
First-Row Outer Gear	58	6.13887	9.448	22.5
First-Row Inner Gear	27	2.04282	13.217	25
Second-Row Inner Gear	126	9.53318	13.217	25
Second-Row Outer Gear	25	4.47788	5.583	30
Ring Gear	154	27.58374	5.583	30

The reduction ratio of the roller gear drive is given by the following equation:

$$RR = \frac{(58)(126)(154)}{(27)(25)(84)} = 19.848$$

The roller loads are a function of the gear loads and roller gear geometry. Whenever torque is transmitted to the roller gear drive, tangential and radial gear tooth loads are induced, Figure 4. The rollers react normal loads resulting from the gear pressure angle θ .

The roller gear drive is a self-preloading unit and has no roller loads when at rest. However, as soon as power is applied, all the roller gear members move radially inward from the second-row pinion/ring gear mesh forces, thereby generating roller loads. The roller loads at each contact point are presented in Figure 5. To ensure ideal gear operating conditions,

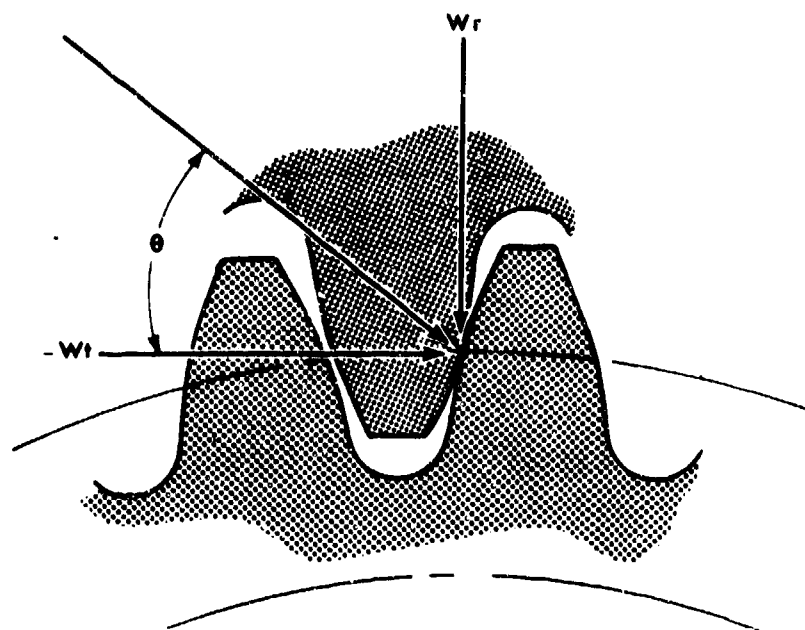


Figure 4. Gear Teeth Mesh Forces.

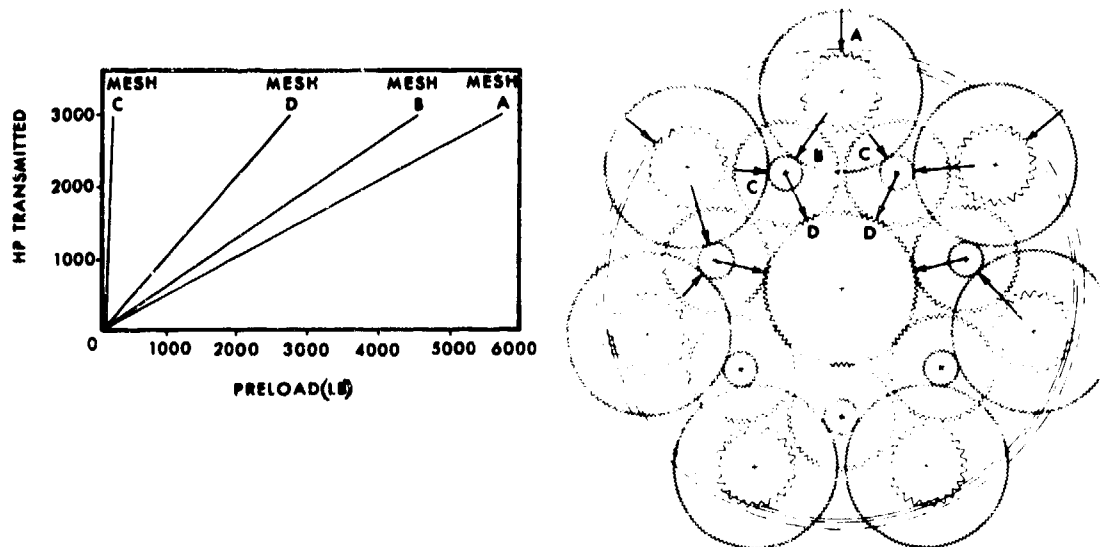


Figure 5. Roller Preload Forces.

the rollers, which control the gear operating center distance, are designed to equal the sun gear/first-row pinion pitch diameters when transmitting full power. The rollers are designed slightly oversize to compensate for the compressive deflections induced by the roller preload forces. The first- and second-row pinion preloads at B (4500 lb) and C (140 lb), however, differ so greatly that no deflection compensation is made. Table 2 gives the free-state roller diameters.

TABLE 2. ROLLER DIAMETERS.	
Location	Diameter (in.)
Sun Gear Roller	8.8917
	8.8913
First-Row Pinion Outer Roller	6.1395
	6.1391
First-Row Pinion Inner Roller*	2.0431
	2.0427
Second-Row Pinion Roller*	9.5335
	9.5331
*Roller size not corrected for deflection.	

The roller gear drive also features cantilevered mounted spherical bearing posts. Each of the seven posts reacts 10,940 lb when the drive is transmitting 3000 hp. The base of each post is sandwiched between a carrier plate which transmits the reaction torque through a splined connection to ground.

A more detailed description of the roller gear components is presented in Appendix A.

ROLLER GEAR MAIN TRANSMISSION

The roller gear unit was designed as the last stage in a three-stage reduction gearbox. A schematic of the entire transmission is shown in Figure 6.

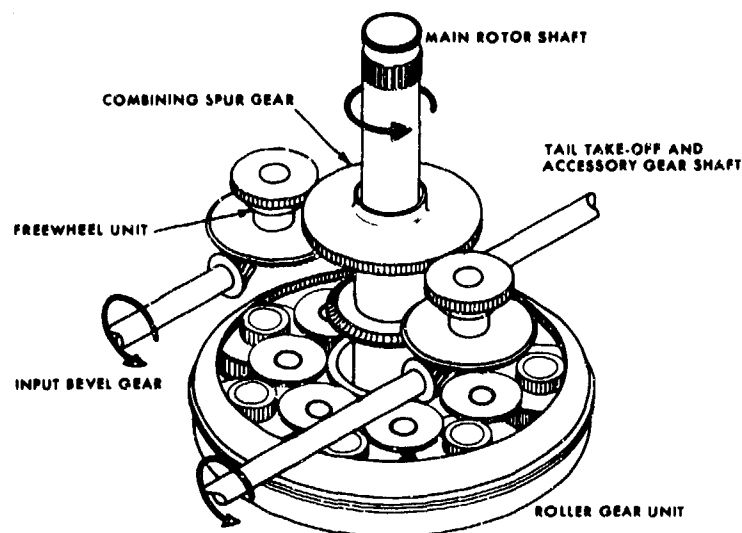


Figure 6. Roller Gear Transmission Schematic.

Power is fed from two turboshaft engines at 18,966 rpm to a pair of spiral bevel gear meshes. These first-stage spiral bevel meshes each have a reduction ratio of 3.05:1 which reduces the speed at the output bevel gear to 6223 rpm. Concentric with each output bevel gear shaft is a ramp-roller type overrunning clutch which permits single-engine operation and allows the rotor to overrun in the event of engine malfunction or engine shutdown. The output cam shaft of the ramp roller clutch drives a second-stage spur gear mesh where power from each engine is combined. At this spur gear mesh, the shaft speed is reduced from 6223 rpm to 4045 rpm.

The output shaft of the second-stage spur gear drives the roller gear unit and the accessory and tail drive section of the transmission.

The actual arrangement of these components is shown in the cross-sectional drawing of Figure 7.

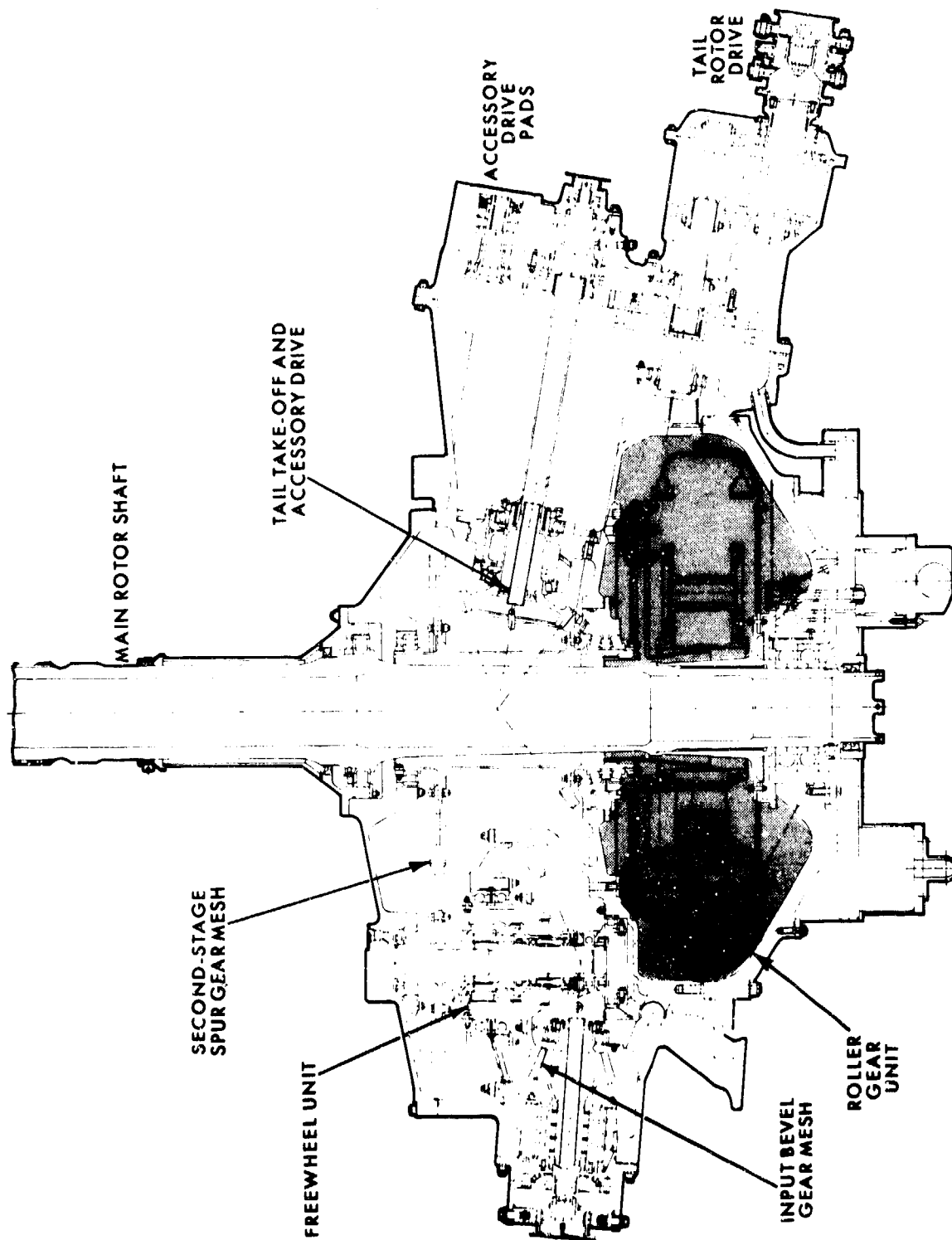


Figure 7. Cross-Section, Roller Gear Transmission.

BASELINE PLANETARY TRANSMISSION

In a separately funded program, a two-stage planetary transmission was designed for the same power as the roller gear transmission. This transmission, shown in Figure 8, was used to compare the relative merits of the roller gear unit with those of the conventional planetary unit.

The two-stage planetary, shown in Figure 9, has an overall reduction of 14.247:1. To remain compatible with the requirements of the roller gear transmission, the combining spur gear ratio was modified to obtain a 2.14:1 reduction; thus, with the input bevel arrangement remaining identical to that of the roller gear transmission, the overall ratio of 93.4:1 is maintained.

The two-stage planetary incorporates built-up carrier plates and cantilevered planet pinions. Within each pinion, double-row spherical roller bearings transmit torque to the rotating carrier plates.

The basic data of the two-stage planetary is presented in Table 3.

TABLE 3. TWO-STAGE PLANETARY, BASIC DATA.				
Gear	No. of Teeth	Pitch Diameter (in.)	Diametral Pitch	Pressure Angle (deg)
<u>First-Stage</u>				
Sun Gear	69	8.625		
Pinions (5)	86	10.75	8.0	22.5
Ring Gear	241	30.125		
<u>Second-Stage</u>				
Sun Gear	111	13.875		
Pinions (8)	65	8.125	8.0	22.5
Ring Gear	241	30.125		

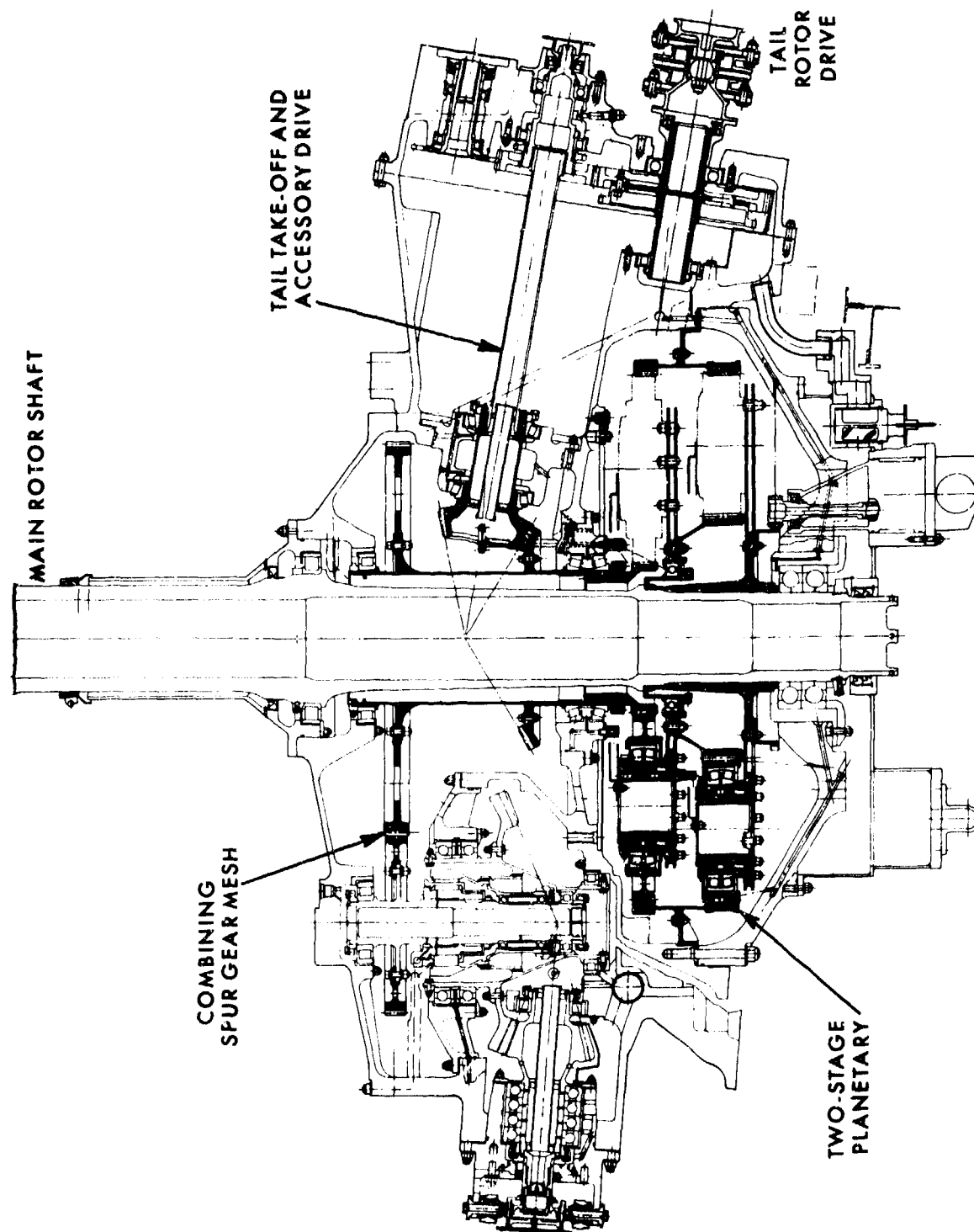
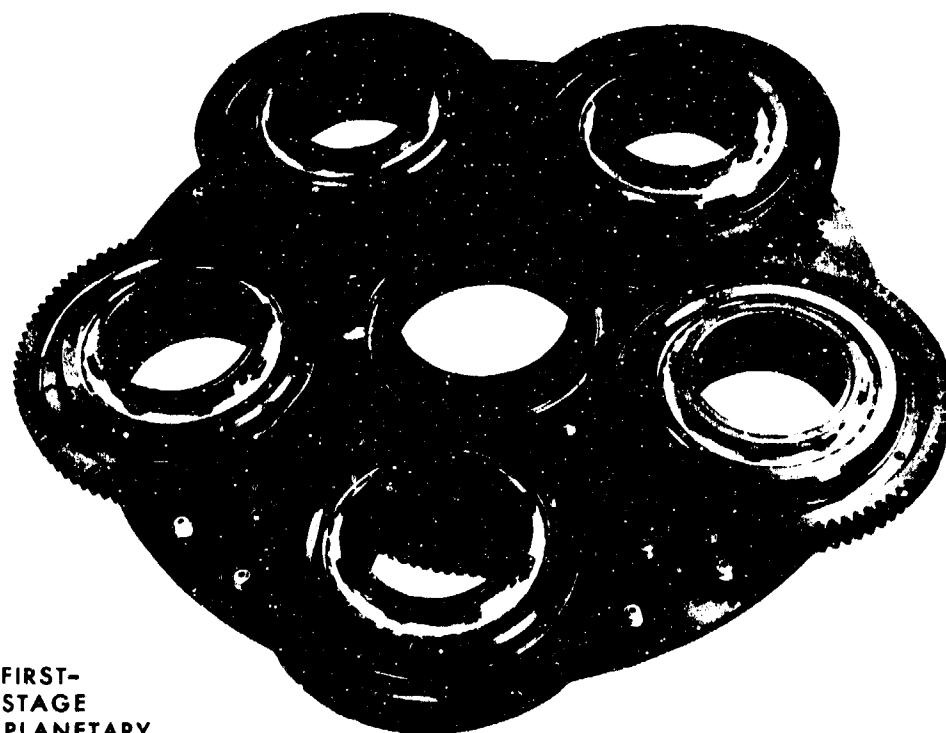
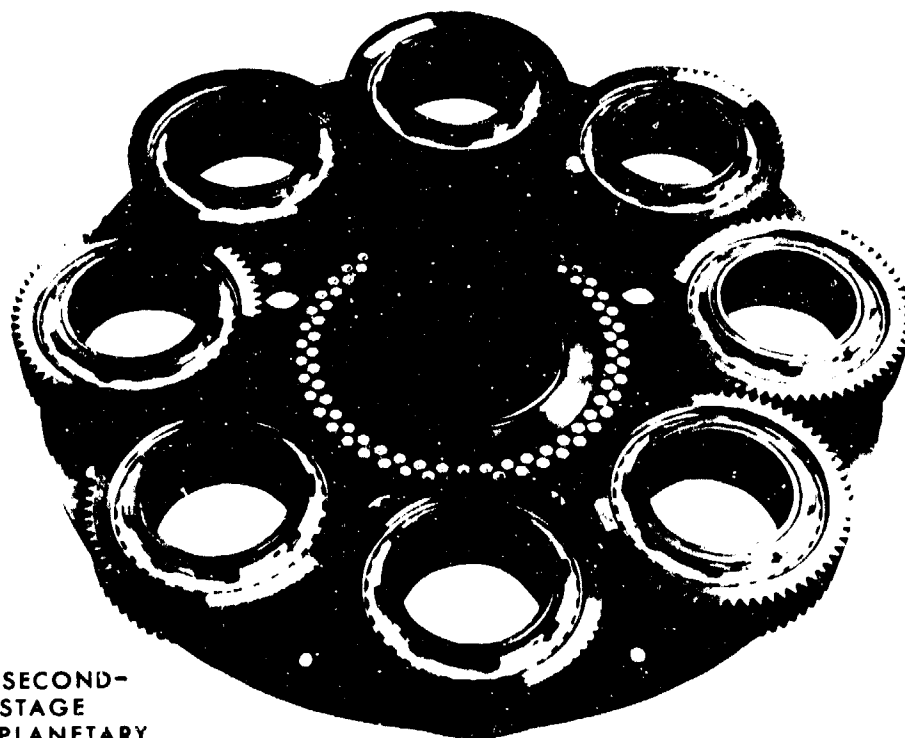


Figure 8. Baseline Transmission, Two-Stage Planetary.



FIRST-
STAGE
PLANETARY



SECOND-
STAGE
PLANETARY

Figure 9. Two-Stage Planetary Transmission.

DESCRIPTION OF SIKORSKY TRANSMISSIONS

The analysis used to establish the relative reliability of the roller gear and baseline planetary transmissions is based on historical data accumulated from Sikorsky S-61, S-64 and S-65 transmissions. This data has been obtained from over 300,000 flight hours of the three aircraft and was assumed to be appropriate to the roller gear transmission because of the basic transmission similarities. A brief description of each of the aforementioned transmissions follows:

S-61 Transmission

The Sikorsky S-61 helicopter transmission accepts power from two T-58 turboshaft engines having an output speed of 18,966 rpm and supplies power to the main rotor rotating at 203 rpm. The main transmission, shown schematically in Figure 10, incorporates an input spur gear mesh. Installed in each gear shaft is a freewheel unit, the output of which is integral with a helical pinion. This mates with a helical gear which transfers the power from both engines onto a single shaft. The combined power is transferred through spiral bevel mesh to a single-stage planetary reduction unit which supplies power to the main rotor shaft. The basic data for the S-61 planetary is given in Table 4.

TABLE 4. S-61 PLANETARY UNIT, BASIC DATA.				
Gear	No. of Teeth	Pitch Diameter (in.)	Diametral Pitch	Pressure Angle (deg)
Sun Gear	54	6.750)		
Pinion (5)	71	8.875)	8.0	22.5
Ring Gear	196	24.5)		

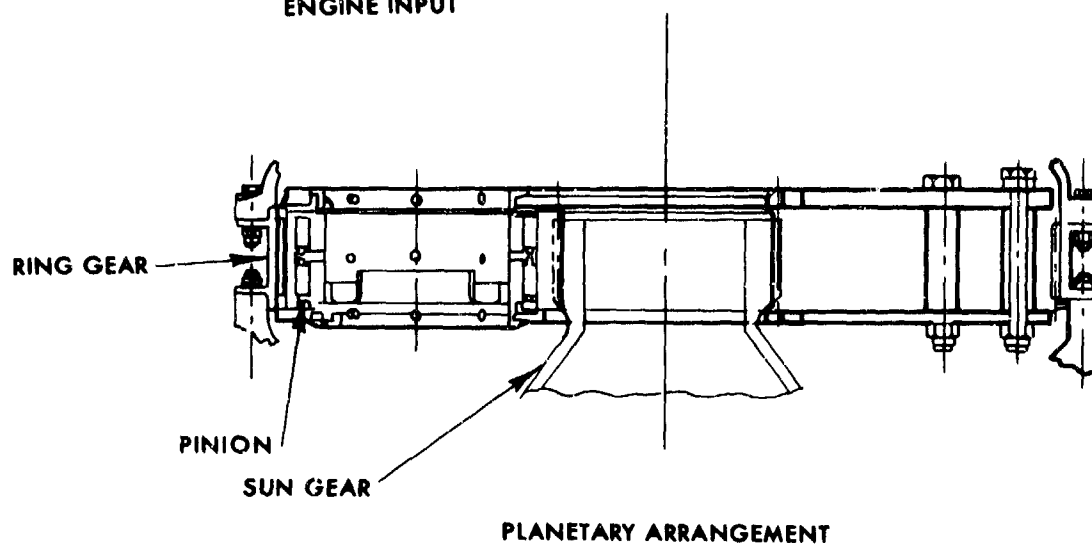
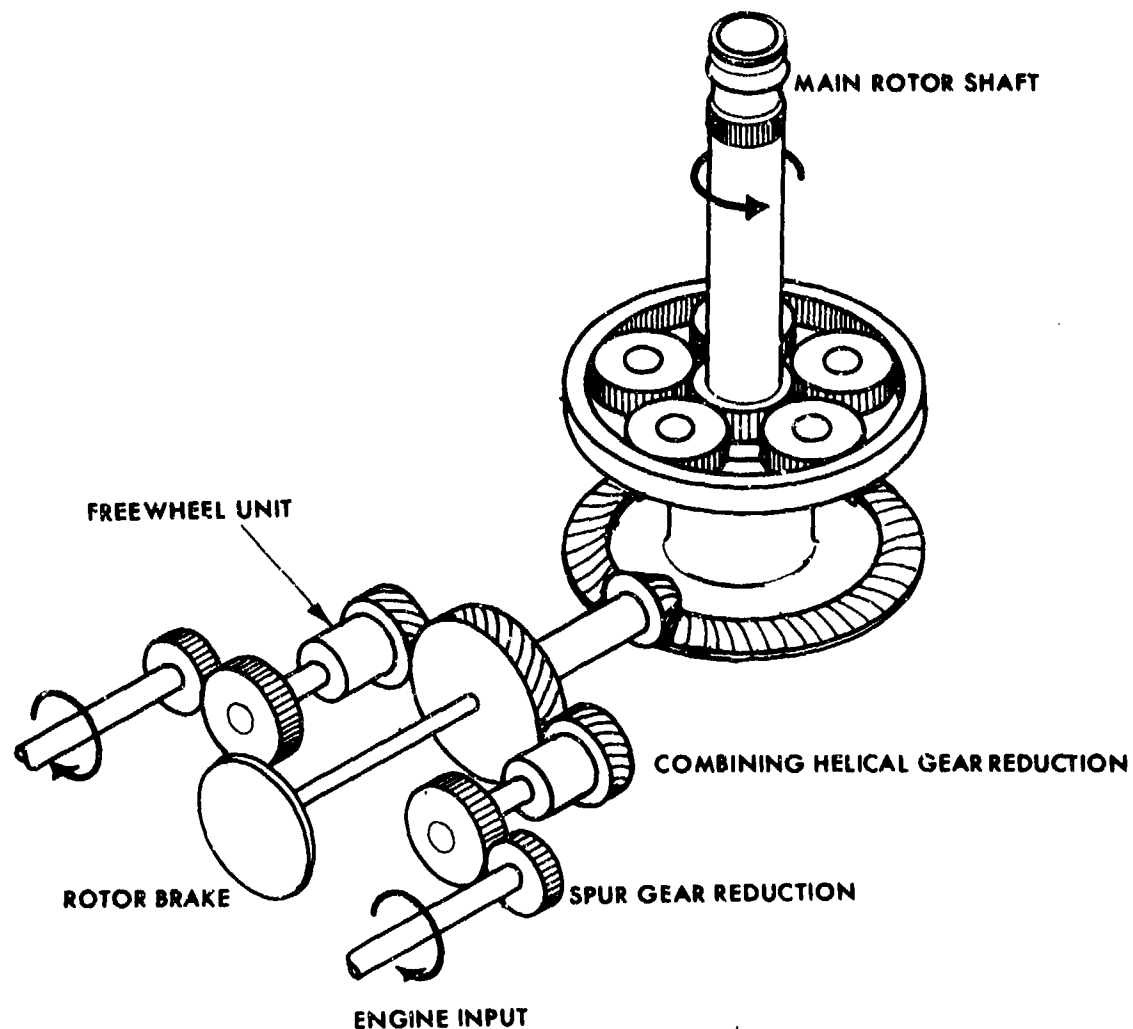


Figure 10. S-61 Transmission Schematic.

S-64 Transmission

A layout of the S-64 main transmission is shown in Figure 11. This transmission is designed to transmit 6600 hp to the main rotor shaft while reducing the speed from 9000 rpm at the engines to 185.4 rpm at the main rotor shaft. Power is transmitted from each engine through a first-stage bevel mesh and a ramp-roller type overrunning clutch to a second-stage bevel mesh where the power from the two engines is combined. A quill shaft attached to the output of the second-stage bevel transfers the combined power to a two-stage planetary with an overall reduction ratio of 8.14:1. The carrier output of the second-stage planetary then transfers power to the main rotor shaft through a splined connection. The gear data for each of the two stages of the S-64 planetary is presented in Table 5.

TABLE 5. S-64 PLANETARY UNIT, BASIC DATA.

TABLE 5. S-64 PLANETARY UNIT, BASIC DATA.				
Gear	No. of Teeth	Pitch Diameter (in.)	Diametral Pitch	Pressure Angle (deg)
<u>First-Stage</u>				
Sun	78	11.142)	7.0	22.5
Pinion (7)	55	7.8571)		
Ring	188	26.857)		
<u>Second-Stage</u>				
Sun	166	20.750)	8.0	22.5
Pinion (18)	32	4.00)		
Ring	230	28.750)		

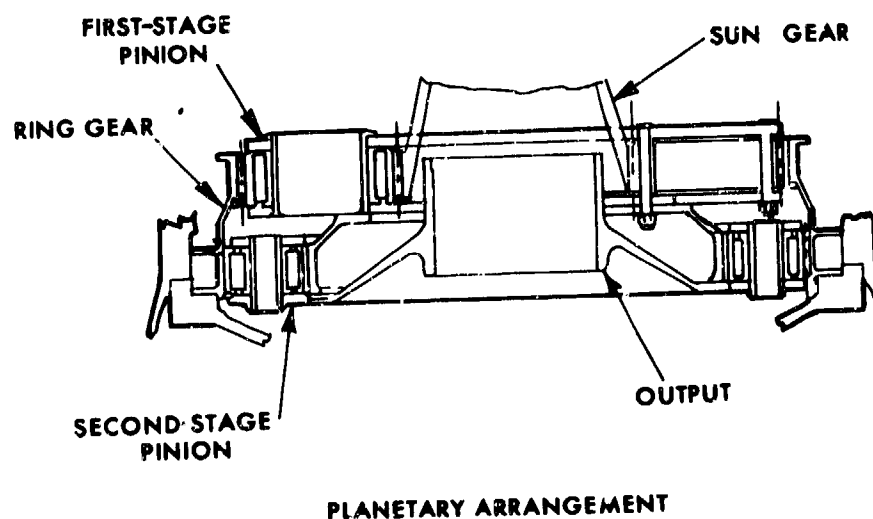
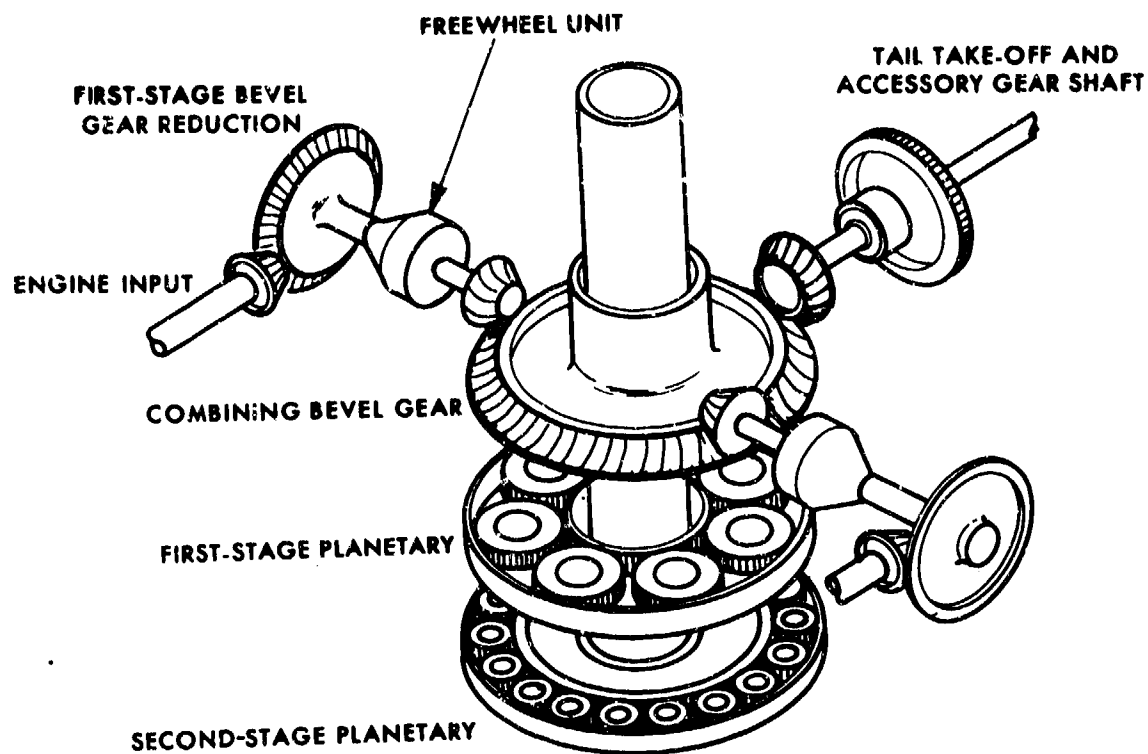


Figure 11. S-64 Transmission Schematic.

S-65 Transmission

The S-65 main transmission is shown in Figure 12. Power is transmitted from the engines to two nose gearboxes (not shown) where the speed is reduced from 13,600 rpm to 6023 rpm. It is then transmitted from the nose gearboxes through a spiral bevel mesh where power from the two engines is combined. The output gear of this mesh is connected to the sun gear of the first of two planetary stages by a quill shaft. The output of the second-stage planetary is splined directly to the main rotor shaft. Table 6 summarizes the gear data of the two planetary stages.

TABLE 6. S-65 PLANETARY UNIT, BASIC DATA.				
Gear	No. of Teeth	Pitch Diameter (in.)	Diametral Pitch	Pressure Angle (deg)
<u>First-Stage</u>				
Sun	69	11.50)	6.0	22.5
Pinion (7)	50	8.333)		
Ring	169	28.166)		
<u>Second-Stage</u>				
Sun	105	17.500)	6.0	22.5
Pinion (12)	33	5.500)		
Ring	169	28.166)		

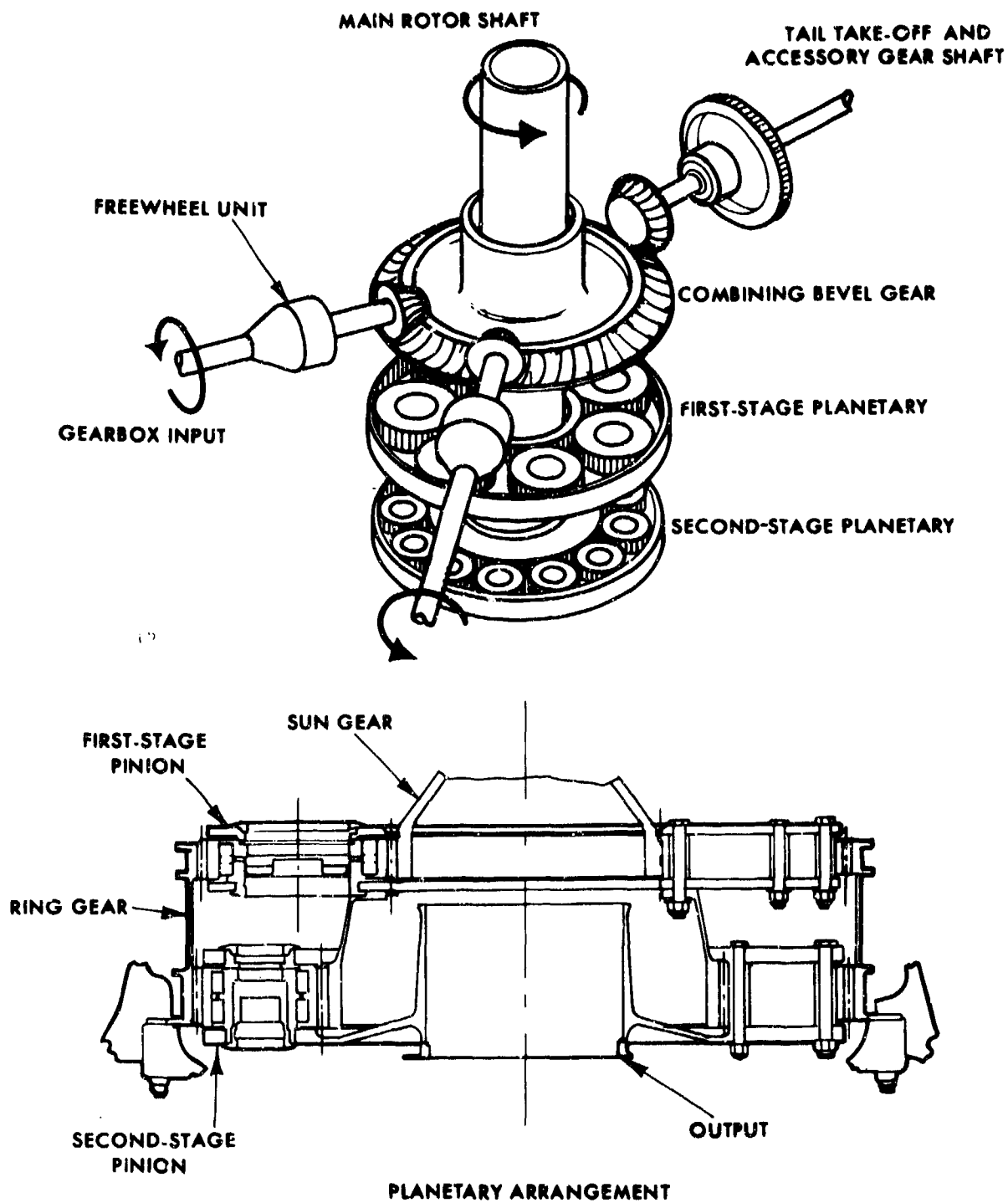


Figure 12. S-65 Transmission Schematic.

ROLLER GEAR TEST PHILOSOPHY

As with all new mechanical concepts, a test program has to be developed to substantiate the performance parameters predicted in the design. The roller gear drive, although not new in concept, is new in that it is the largest roller gear drive system developed thus far.

During development testing, the probability of a failure is relatively high. As testing continues, the probability decreases and should, if the initial test is representative of the actual environment, become substantially lower as testing continues. If the initial testing fully duplicates the service requirements, the probability of detecting a mode of failure should approach the hypothetical plot of failure versus elapsed time (Figure 13). If the level of the test spectrum is too low, then a low service life, i.e., poor reliability, often results. The testing then occurs in service where serious logistic and retrofit problems are experienced. If, on the other hand, too severe a test is conducted, either a too-low TBO results from "must-pass" test requirements, or, alternatively, the transmission becomes overdesigned with components which were subjected to test stresses that would never be experienced in service.

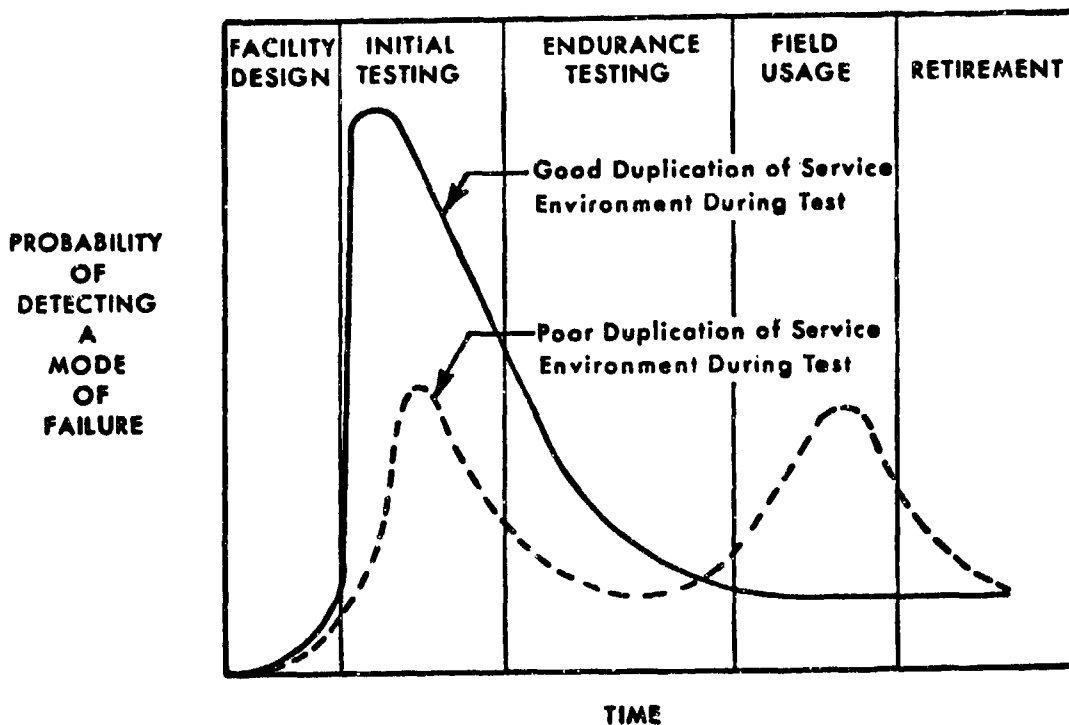


Figure 13. Effect of Environmental Duplication on the Probability of Detecting New Modes of Failure.

The test program developed for the roller gear drive had four primary objectives:

1. To demonstrate that the roller gear drive system met the design requirements.
2. To thoroughly develop and "debug" the roller gear drive components.
3. To determine the "modes of failure" of the roller gear drive components and the system.
4. To demonstrate that all catastrophic modes of failure are out of the planned operating range of the transmission, and that all failures are detectable.

The roller gear drive test program included a no-load lubrication test, regenerative bench endurance testing and aircraft tiedown testing of the complete roller gear drive transmission, and regenerative testing of roller gear drive units.

INITIAL AND ENDURANCE TESTING

The primary purpose of initial testing was to identify problems with the roller gear drive and make the necessary design changes prior to endurance testing. Included in this testing were the following:

1. No-load lubrication test; the primary objective was to optimize lubrication and eliminate restrictive oil passages.
2. Gear pattern tests to develop the contact pattern of the bevel gears.

The endurance test, which was of 200-hours duration, was conducted in accordance with the requirements specified in MIL-T-5955. This specification sets forth the minimum requirements of load, speed and duration that have to be accomplished on a "must pass" test. For this test, the design was held fixed to prove that the design met the preestablished goals. During this test phase, the efficiency of the transmission system was measured.

AIRCRAFT TIEDOWN TESTING

The tiedown test provided for the integration of the transmission with the complete aircraft. The aircraft tiedown test demonstrated compatibility of the roller gear drive with the deflections and torque fluctuations actually experienced by transmissions on a flight aircraft. Testing was conducted in accordance with MIL-T-8679, 50-hour Preliminary Flight Approval. In addition to evaluating compatibility of the subsystems, the test investigated performance and operating characteristics of all the test dynamic and flight control systems.

RELIABILITY AND MAINTAINABILITY TESTING

This test was designed to determine the effect of extended running time of the roller gear drive unit. Since all other components in the roller gear transmission were of conventional configuration and hence of proven durability, the R&M test was designed to evaluate wear and strength characteristics of the roller gear unit components. Again, power-time load spectrum requirements were equivalent to or greater than MIL-T-8679.

Brief discussions of the particulars of these test programs are presented in Appendixes B and C.

TEST RESULTS

REVIEW OF TEST RESULTS

The accumulative time attained by the roller gear units throughout the various phases of testing totaled 1347:31 hours. During this period, there were seven component fractures and test stoppages due primarily to electron-beam weld related problems. Only two test stoppages could be attributed to the roller gear and its modus operandi. These were the bending tooth fracture of the first-row pinion at 17:18 hours of the R&M test and the gear tooth spalling, again of the first-row pinion at 295:50 hours.

Throughout the test program, no significant fatigue wear of the roller elements occurred other than edge/corner interference. It has been demonstrated that the roller gear principle is a feasible means of attaining a high reduction for helicopter transmissions.

The weld-related fractures were development problems that can be associated with the application of a relatively new form of welding applied to advanced transmission system components. The welding constraints imposed by the design and complexity of the transmission components were unique. Rotational placement of the components was held to very tight limits to ensure adequate gear tooth case depth of the finished components. The application of excessively high energy welds was found to result in tempering of the roller surfaces. The initial weld parameters were changed to increase the weld energy to ensure 100 percent fusion of the second-row pinion gear/flange weld after separation of the gear from the flange occurred due to incomplete fusion.

In the initial design of the weld joints, the effect of weld porosity and incomplete fusion was not fully realized. Although poor contours on the face of an EB weld can be improved by using cosmetic weld passes, the inherently poor root-surface contours in a partial penetration weld cannot be improved except, to a limited extent, by defocusing the beam. In a partial penetration weld, spiking occurs due to irregularly shaped melting patterns at the weld root. It has the characteristics of solidification and shrinkage defects and results in voids shown in the cross-section view through the root of the first-row pinion initial weld configuration (Figure 14).

Design improvements included complete weld penetration and the cleanup of the weld face and root surfaces, the elimination of four of the six welds in the redesign of the second-row pinion, and the development of ultrasonic inspection for quality assurance of the welds.

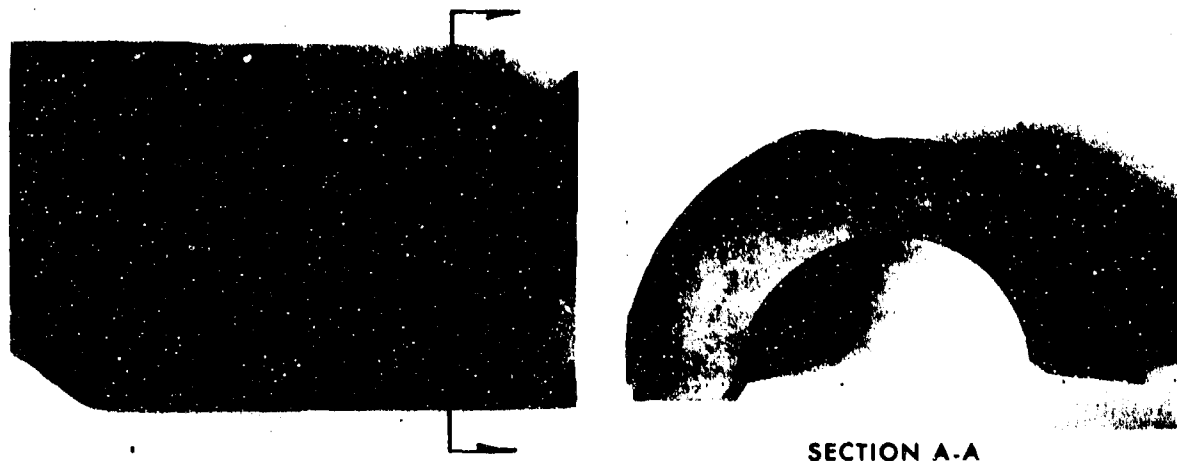


Figure 14. Electron-Beam Weld Root Voids.

The bending tooth fracture of the first-row pinion cannot be fully explained, for no material abnormalities were found other than a relatively low Rc56.5 hardness of the carburized case, the design requirements being Rc58-64. The carburized case depth at the root radius was .009 inch, as opposed to a required .010/.025 inch depth. Manganese phosphate process pitting was evident at the origin site. The heavily crystallized nature of manganese phosphate retains lubricating oil and prevents galling; however, etching of the gear surface occurs. It was from one such etched pit that the fracture originated. The calculated bending stress on these pinion teeth was 39,260 psi, well within the design allowable of 55,000 psi for 3σ reliability.

The spalling of the first-row pinion S/N 49 occurred after 499:35 hours of test operation, during which the teeth accumulated 3.5×10^7 cycles. The final 273 hours of test operation were at the 3000-hp maximum power condition. Spalling is recognized as gear tooth material fatigue resulting from pitting, whereby particles of material have broken out of the gear tooth surface after continued operation.

Initial pitting can, in some circumstances, terminate with no further destruction of the tooth surface. However, progressive pitting fatigue increases rapidly into spalling until the remaining tooth surface area not impaired is insufficient to transmit the load; progressively increasing damage can occur to destroy the gear tooth surface completely.

Various studies have been conducted on the mechanism of fatigue crack initiation. Bartel & Kass⁽¹⁰⁾ found from electron microscopic studies that plastic flow of the surface is followed by pits similar to that caused by electrostatic discharge across an oil film. General weakening of the surrounding surface determines whether the pit develops into progressive pitting.

Each time gear teeth go into mesh, they roll and slide on each other with the result that the surface and subsurface metal is subjected to tensile and compressive stresses. Metal is displaced, resulting in subsurface shear and tensile stresses. Frictional forces impose additional surface compressive and tensile stresses from which, under repeated stress cycles, surface failure can occur.

Contact stresses are higher in the dedendum of the driving gear at the point of tooth contact than those elsewhere due to the shorter radii of tooth curvature.

ROLLER GEAR MAINTAINABILITY

The roller gear unit is a very dense package with all gears interlocked. Consequently, the inspection and/or replacement of parts can be accomplished only when the unit has been completely disassembled.

The roller gear unit is a complete modular assembly comprising of a sun gear, seven first- and second-row pinions, a ring gear, and a torque reaction splined carrier plate assembly. Ease of maintainability is assured as the unit can be readily removed from its operating environment. Removal of seven hold-down bolts, located in the second-row pinion posts, allows the roller gear unit to separate at the torque reaction splines.

ASSEMBLY, ROLLER GEAR UNIT

The assembly and disassembly of the roller unit was accomplished on an assembly rig fixture. This tool centrally locates the sun gear and accurately positions the first- and second-row pinions with dowel pins located in jig-bored holes (Figure 15).

Assembly of the first- and second-row compound gears requires indexing of the gear teeth. Each first- and second-row gear assembly is identifiable by a master tooth, to which all seven gears of each row are machined identically. This tooth has to be angularly positioned with the gear teeth of the mating pinions for correct assembly.

The positions of the master gear teeth on the sun, first- and second-row pinions are identified on the side face of the adjacent rollers and assembled in accordance with the instructions (Appendix D). These instructions specify the procedures adopted to ensure correct assembly of the roller gear unit. An assembly gage checks the alignment of the first-/second-row pinion mesh.

The toggle angle generated by the first-row/second-row geometry prevents the placement of the last second-row pinion into the nest of gears in the normal manner. The last second-row pinion and two mating first-row pinions to be assembled are positioned as a set and then moved into position as shown in Figure 16. The second-row pinion posts are then installed followed by assembly of the output flange and hub assemblies. Assembly of the two halves of the ring gear complete the roller gear unit.



Figure 15. Assembly Fixture, Roller Gear Unit.

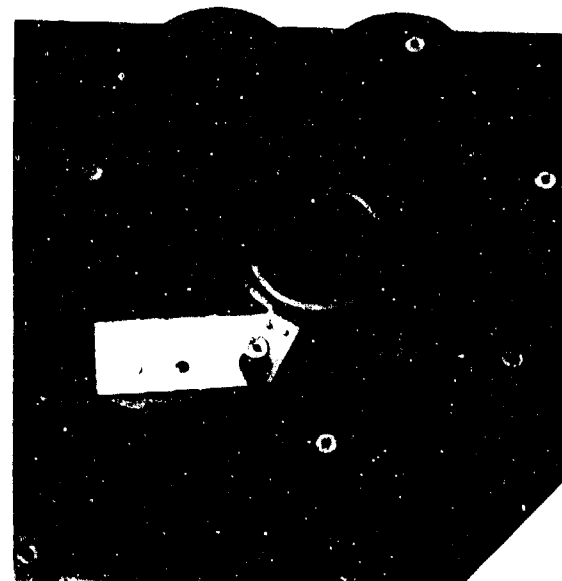
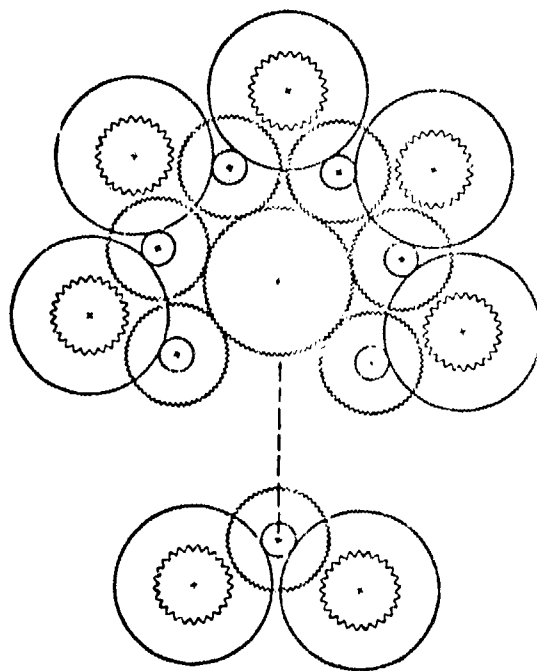


Figure 16. Gage, Gear Tooth Indexing.

MAINTAINABILITY

As with all systems, it was found that familiarity with the assembly of the roller gear drive increased productivity. By retaining the same personnel for the assembly and disassembly of the unit, the mean time to repair was reduced by approximately 50 percent from that originally scheduled.

Although special attention was required during assembly of the unit, the stringent timing requirements were easily checked when assembling the ring gear. If the timing of any one tooth mesh was off, the ring gear would not fit. Thus, an easy check was assumed before assembly of the complete unit had proceeded too far.

The dense packaging of the roller gear unit can, in the event of a primary fracture, result in a high degree of secondary debris damage. During initial testing of the roller gear transmission gearboxes in the regenerative test facility, the fracture of the first-row pinion gear teeth resulted in considerable secondary damage to the second-row gears. Similarly, while conducting the failure mode test in the R&M facility, the first-row pinions sustained debris damage from the fracture of a second-row pinion that resulted in the scrapping of all first- and second-row pinions. Only compartmentization of the two roller gear units utilized in the R&M test precluded debris particles from contaminating the lower unit. The high contamination rate from debris is inherent in the design of the roller gear unit with its multitude of gear meshes and rolling contact surfaces. In a two-stage planetary transmission, compartmentization of the stages can limit the extent of secondary damage to a single stage. The compactness and flat plane of the roller gear unit prevents such compartmentization.

A comparative count, Table 7, of the meshes in a two-stage planetary illustrates the inherent complexity of the roller gear unit.

TABLE 7. QUANTITATIVE COMPARISON OF TRANSMISSION MESHES.

Dynamic Element	Baseline		Roller Gear Unit
	Two-Stage First-Stage	Planetary Second-Stage	
Sun Gear			
Gear Mesh	5	8	14
Roller Mesh			14
Intermediate Pinions			
Gear Mesh			14
Roller Mesh			28
Ring Gear			
Gear Mesh	5	8	14
Bearings	5	8	7
Total Dynamic Contacting Elements	15	24	91

The analysis shows a reduction of approximately 60% in the number of dynamic contacting elements for the two-stage planetary unit. These elements can be further subdivided by compartmentizing the two stages.

Although the roller gear unit is more readily maintainable, i.e., it can be removed as a complete module as shown in Figure 17, a compartmentized two-stage planetary would require the removal of each stage. This modularization does not enhance the unit's survivability in the event of a fracture of a dynamic component.

As previously mentioned, the secondary debris damage sustained by only a few pinions has necessitated the replacement of the seven pinions that comprise a set. These pinions were manufactured in matched sets and no two sets were fabricated with master gear teeth ground to identical dimensions. Interchangeability can only be attained with the fabrication of all first- and second-row pinions to identical dimensions.

Support costs could be further reduced, and further usage of parts could be attained by reversing the pinions to drive on the back side of the gear teeth. The contact and bending stresses of the roller gear teeth are all one-way, i.e., torque is transmitted on one side of the gear teeth at all times. Thus, in the event of surface damage to a gear tooth, all pinions could be "flipped-over" to drive on the coast side. For this to be accomplished, the timing of the gear teeth on the coast side would have to be held to tolerances similar to those on the drive side.



Figure 17. Roller Gear Unit Module.

FAILURE MODE AND EFFECTS ANALYSIS

For a helicopter transmission, where a failure in the primary drive train can be catastrophic, it is important that potential failure modes are identified and evaluated. The failure mode and effects analysis (FMEA) determines transmission system failure modes and categorizes their effects on aircraft reliability and performance. Those failure modes which potentially might result in the loss of main rotor power, or tail rotor power, or loss of all hydraulic or electrical power are categorized as safety-of-flight failure modes. Failure modes which would result in gearbox removal are categorized as dynamic component removal failure modes.

The FMEA was conducted by first considering the failure modes of each gearbox component. Each failure mode was analyzed to determine its effect on assembly, gearbox, drive system and aircraft performance. When a failure mode could be placed in either category, depending upon its severity, the most severe was chosen. Maintenance inspections and fault warning systems were then analyzed to determine their applicability. Potential safety-of-flight failure modes that could be detected by maintenance inspections and which eliminate their occurrence in flight were classified as dynamic component removal failure modes. If an inspection did not completely eliminate the possibility of a failure mode, it was not recategorized.

When all the individual failure modes were analyzed and all the failure modes that comprise each category defined, they were summarized by generic component. The term generic component indicates a broad class of components. Bearings were divided into four generic components: ball bearings, spherical roller bearings, cylindrical roller bearings, and tapered roller bearings. Gears were divided into two generic components: spiral bevel gears and spur gears.

ROLLER GEAR TRANSMISSION

Table 8 lists the generic components which have safety-of-flight or dynamic component failure modes as their most severe effect. The table is arranged by gearbox section, as shown in Figure 18, and generic component.

As shown in Table 8, the roller gear unit has the highest number of safety-of-flight failure modes. This is to be expected in a transmission final reduction stage where a single torque path transmits power to the main rotor shaft. The input section, on the other hand, reflects the high reliability that is inherent in dual power paths.

A thorough detailed investigation of the roller gear drive assembly was conducted, which expands on the effect of the failure modes. This analysis, presented in Appendix F, lists the possible failure modes, their effects on the roller gear drive, the probable methods of failure detection, and the design parameters that minimize the occurrence of a catastrophic failure.

TABLE 8. ROLLER GEAR TRANSMISSION, FAILURE MODE SUMMARY.				
Section	Generic Component	Failure Mode	Number of Safety-of-Flight Failure Modes	Number of Dynamic Com- ponent Removal Failure Modes
Input bevel and Second-Stage Spur Gear Mesh	Spiral Bevel Gear	Excess Wear	-	5
		Spalling	-	5
		Scoring	-	5
		Tooth Fracture	-	5
		Web Shaft Crack	1	5
Spur Gear	Spur Gear	Excess Wear	-	3
		Spalling	-	3
		Scoring	-	3
		Tooth Fracture	-	3
		Web/Shaft Crack	1	3
Spline	Spline	Wear Fretting	-	10
Ball Bearing	Ball Bearing	Cage Fracture	-	18
		Spalling	-	18
		Smearing	-	18
		Excess Wear	-	18
Lock Ring	Lock Ring	Fracture	-	8
Bearing Retainer	Bearing Retainer	Crack, Shear	-	16
		Crack	-	8
Shaft	Shaft	Spline Wear	-	8
Roller Bearing	Roller Bearing	Cage Fracture	1	7
		Spalling	-	7
		Smearing	-	7
		Excess Wear	-	7

TABLE 8. (Continued)

Section	Generic Component	Failure Mode	Number of Safety-of-Flight Failure Modes	Number of Dynamic Com- ponent Removal Failure Modes
Input bevel and Second-Stage Spur Gear Mesh (Continued)	Tapered Roller	Cage Fracture	2	2
		Spalling	-	2
		Smearing	-	2
		Excess Wear	-	2
	FWU Housing	Crack	-	2
		Excess Wear	-	2
	FWU Cage	Cage Fracture	-	2
		Spalling	-	28
	FWU Roller	Excess Wear	-	28
		Brinelling	-	28
Main Shaft	Spring	Fracture	-	4
	Nut	Loose	-	270
	"O" Ring	Leakage	-	8
	Shaft Seal	Leakage	-	2
	Housing	Crack	-	2
	Bolts	Shear	-	272
	Spline	Wear, Fretting	1	1

TABLE 8. (Continued)

Section	Generic Component	Failure Mode	Number of Safety-of-Flight Failure Modes	Number of Dynamic Com- ponent Removal Failure Modes
Main Shaft (Continued)	Ball Bearing	Cage Fracture	2	2
		Spalling	-	2
		Smearing	-	2
		Excess Wear	-	2
	Lock Ring	Fracture	-	1
		Crack, Shear	1	2
	Shaft	Crack	1	1
		Spline Wear	-	1
	Roller Bearing	Cage Fracture	1	1
		Spalling	-	1
		Smearing	-	1
		Excess Wear	-	1
	Nut	Loose	1	17
	"O" Ring	Leakage	2	4
		Leakage	2	3
	Housing	Crack	2	2
	Bolts	Shear	-	21

TABLE 8. (Continued)

Section	Generic Component	Failure Mode	Number of Safety-of-Flight Failure Modes	Number of Dynamic Com- ponent Removal Failure Modes
Tail Take-Off and Accessories	Spiral Bevel Gear	Excess Wear	-	1
		Spalling	-	1
		Scoring	-	1
		Tooth Fracture Web/Shaft Crack	- 1	1 1
	Spur Gear	Excess Wear	-	13
		Spalling	-	13
		Scoring	-	13
		Tooth Fracture Web/Shaft Crack	- 2	13 13
	Spline	Wear, Fretting	2	12
	Ball Bearing	Cage Fracture	2	14
		Spalling	-	14
		Smearing	-	14
	Lock Ring	Fracture	-	18
	Bearing Retainer	Crack Shear	-	3
		Crack	1	5
	Shaft	Spline Wear	-	2
	Roller Bearing	Cage Fracture	2	8
		Spalling	-	8
		Smearing	-	8
		Excess Wear	-	8

TABLE 8. (Continued)

Section	Generic Component	Failure Mode	Number of Safety-of-Flight Failure Modes	Number of Dynamic Com- ponent Removal Failure Modes
Tail Take-Off and Accessories (Continued)	Tapered Roller Bearing	Cage Fracture	2	2
		Spalling	-	2
		Smearing	-	2
		Excess Wear	-	2
Single Specimens	Nut "O" Ring Housing Bolts	Loose	-	41
		Leakage	-	4
		Crack	-	3
		Shear	-	50
Roller Gear Unit	Spur Gear	Excess Wear	-	42
		Spalling	-	42
		Scoring	-	42
		Tooth Fracture	-	42
		Web/Shaft Fracture	42	42
		E/B Weld Deteriora- tion	28	28
		Excess Wear	-	44
		Spalling	-	44
Roller	Roller	Scoring	-	44
		E/B Weld Deteriora- tion	-	30
		Wear, Fretting	2	2
		Fracture	-	7
Spline Lock Ring	Spline Lock Ring	Fracture	-	7

TABLE 8. (Continued)

Section	Generic Component	Failure Mode	Number of Safety-of-Flight Failure Modes	Number of Dynamic Com- ponent Removal Failure Modes
Roller Gear Unit (Continued)	Spherical Roller Bearings	Cage Fracture	7	7
		Spalling	-	14
		Smearing	-	14
		Excess Wear	-	14
	Nut	Loose	-	404
	"O" Ring	Leakage	-	7
	Plate Assy	Crack	-	1
	Bolts	Shear	7	411
	Flange	Fracture	1	1

RELIABILITY ANALYSIS

In order to determine the relative reliability of the roller gear transmission, a hazard function analysis was performed on both the baseline planetary and roller gear transmissions. This analysis provided a means of comparing the baseline planetary transmission to a fully developed roller gear transmission on the basis of projected mean time between removals. An additional analysis on the roller gear was performed by computing the reliability of the roller gear transmission based on test and by projecting this figure along a trend curve to establish the reliability of a mature roller gear drive. A comparison of these two reliability figures for the roller gear transmission provides an indication of the present stage of development of the roller gear transmission.

HAZARD FUNCTION ANALYSIS

Hazard function analysis is a sophisticated technique for determining the reliability of a component or system of components based on actual field data. This technique is also useful for predicting the reliability of a system for which no field data exists but which is composed of generic components similar to those of existing systems for which field data does exist. For this program, the transmissions of the S-61, S-64, and S-65 helicopters provided the field data which was used to predict the reliability of both the baseline planetary transmission and the roller gear transmission.

The reliability analysis technique for estimating gearbox hazard functions is based upon the reliability distribution being defined by the Weibull reliability function. The Weibull distribution defines the cumulative reliability, $R(t)$, as follows:

$$R(t) = e^{-(t/\theta)^\beta}$$

where

- β = the shape parameter (dimensionless)
- θ = the size parameter (hours)

A hazard function expresses, for a particular failure mode of a component, its failure rate as a function of time. A hazard function of a particular failure mode of a particular generic component has the following form:

$$h(t) = \frac{\beta}{\theta} \left[\frac{t}{\theta} \right]^{\beta-1} \quad (1)$$

where

- $h(t)$ = the hazard function, i.e., the failure rate as a function of time.

Both the size and the shape parameters are derived from actual field data. The shape parameter describes how the failure rate behaves with time. It can readily be seen from equation (1) that a shape parameter of 1 implies a constant failure rate, a shape parameter of greater than 1 implies a failure rate which increases with time, and a shape parameter of less than 1 implies a failure rate which decreases with time. The size parameter is basically a measure of the magnitude of the failure rate. For example, if two components have equal shape parameters, at a given time, the component with the larger size parameter will have the lower failure rate. The size and shape parameters for the various failure modes of each generic component of the transmissions under consideration are presented in Appendix E.

The hazard function of a particular component is related to its reliability according to the Weibull relation:

$$R(t) = e^{-\int_0^t h(\tau) d\tau} \quad (2)$$

where

e = base of the natural logarithm
 τ = a dummy variable used to evaluate the integral
 $R(t)$ = the reliability of the component as a function of time.

For a system such as a helicopter transmission, it is assumed that the failure of a single component results in the failure of the whole system. This type of system is known as a series system. For a series system, the system reliability is related to component reliability by multiplication of all component reliabilities. This can be expressed mathematically by the following

$$R_s(t) = \prod_{i=1}^K R_i(t) \quad (3)$$

where

$R_s(t)$ = overall system reliability
 K = number of generic components
 R_i = reliability of i th component
 $\prod_{i=1}^K$ = a convention used to indicate that the R_i 's are multiplied together.

When comparing two transmission systems, it is generally preferable to use mean time between failures (MTBF) or mean time between removals (MTBR) when evaluating their relative reliability. The MTBF of a system is related to its reliability by the following

$$MTBF = \int_0^{\infty} R_s(t) dt \quad (4)$$

In general, the upper limit of the integral can be replaced by a finite time since the integral for most systems converges and thus changes very little after a certain time period. If it is assumed that any failure causes component removal, as it was here, MTBR is related to MTBF according to the following relationship

$$MTBR = MTBF \quad 1 - e^{-TBO/MTBF} \quad (5)$$

where

TBO = time between overhauls

ROLLER GEAR HAZARD FUNCTION ANALYSIS

In order to make the projected reliability figures for the baseline and roller gear transmissions more representative of advanced transmissions, it was necessary to adjust the size parameters used in the hazard function analysis. This adjustment was made by incorporating the trends shown in Figure 18. This figure shows the horsepower/reliability trend for both pre-1970 and post-1970 transmissions. The pre-1970 curve is based on actual field data. The post-1970 was shifted upwards on the basis of test data currently available for post-1970 transmissions.

The adjustment of the size parameters was accomplished as follows. First, the MTBR of a post-1970 transmission of 2250 prorated horsepower was found from Figure 18 to be 1850 hours. The size parameters based on field data were then adjusted so that a hazard function analysis of the baseline planetary transmission produced a MTBR of 1850 hours. These adjusted size parameters were then used in the hazard function analysis of the roller gear transmission.

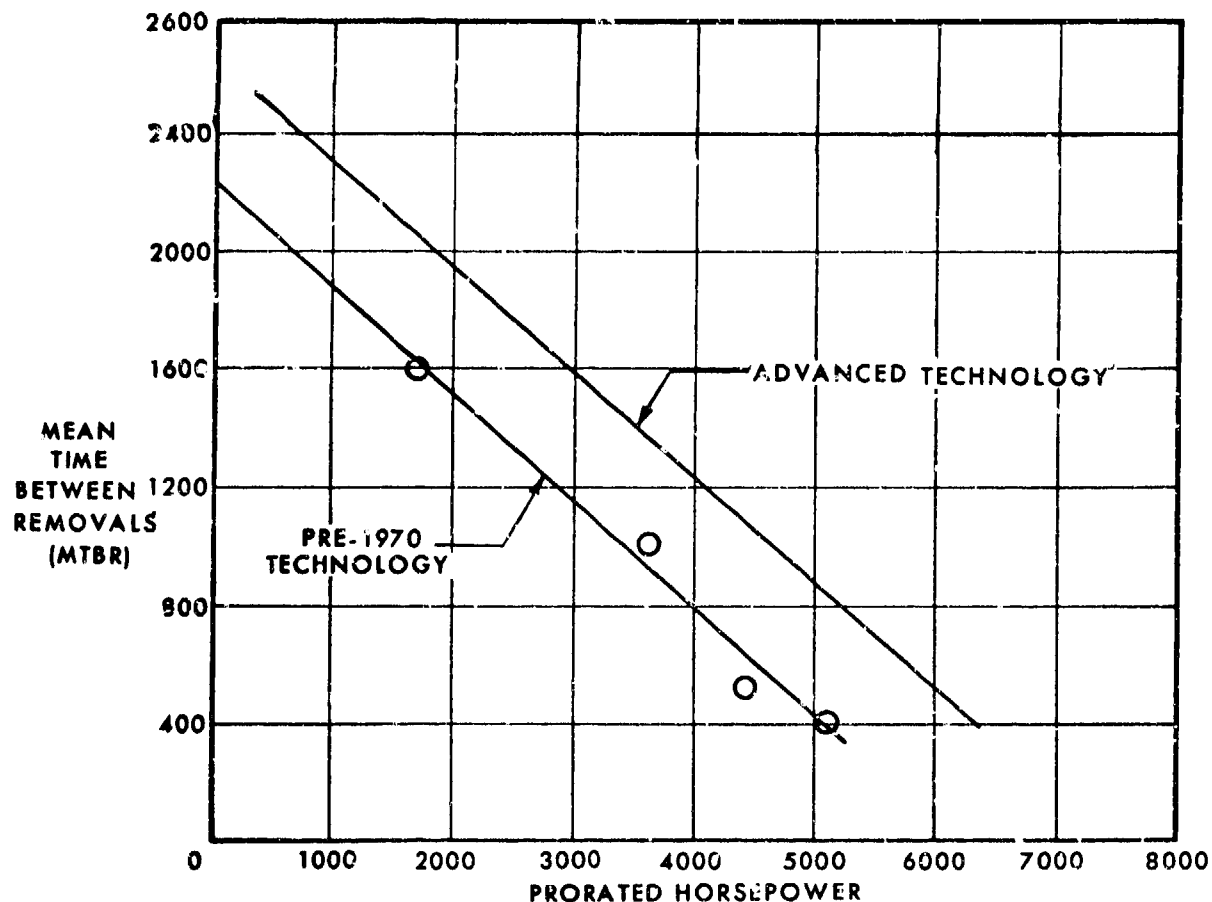


Figure 18. Transmission MTBR vs Prorated Power.

Two basic types of failure modes were considered in the analysis of the roller gear transmission.

- | | |
|------------------------------------|--|
| (1) Safety-of-Flight Failure Mode | A failure mode that causes either immediate forced landing, injury to the crew, or catastrophic loss of the vehicle. |
| (2) Dynamic Component Failure Mode | A failure mode that causes the removal of a component and replacement with a like item. |

The first hazard function analysis was performed for the roller gear transmission on the basis of safety-of-flight failure modes alone. The results of this analysis are presented in Figure 19. The size and shape parameters of the various safety-of-flight failure modes are tabularized in Table E-5 of Appendix E. As can be seen from Figure 19, the safety-of-flight failure rate is essentially constant with time. Any

flight system for which the safety-of-flight failure rate is constant with time qualifies for on-condition maintenance. On-condition maintenance means that there is no scheduled overhauls for the system since an overhaul would not appreciably affect safety-of-flight. This implies a TBO of infinity which reduces equation (5) to the following

$$MTBR = MTBF \quad (6)$$

Next, a hazard function analysis of the roller gear transmission was performed on the basis of the failure modes which result in dynamic component removal.

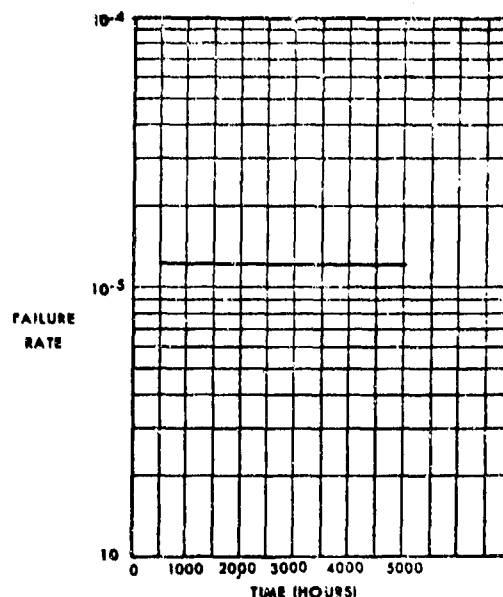


Figure 19. Safety-of-Flight Hazard Function.

The hazard function analysis based on failures which result in dynamic component removal showed a MTBR of 1750 hours for the roller gear transmission compared to the 1850 hour MTBR for the baseline planetary transmission. Since like generic components of each transmission had exactly the same size and shape parameters, the difference in calculated MTBR resulted essentially from a "higher" parts count for the roller gear than for the planetary transmission. Appendix E shows, in tabularized form, the number of generic components contained in each transmission, as well as the adjusted size and shape parameters for each failure mode of each generic component. The difference of 100 hours in MTBR between the roller gear and the baseline planetary transmission is not considered significant with this type of analysis. Therefore, hazard function analysis shows that a fully developed roller gear drive would have essentially the same reliability as that of a conventional two-stage planetary.

It is necessary to point out that this analysis presumed that the roller gear drive was a fully developed concept whose hazard function parameters followed those of fully developed planetary transmissions.

ROLLER GEAR RELIABILITY ANALYSIS BASED ON TEST

In order to assess the reliability of the present roller gear drive, MTBR's were computed on the basis of test alone. This was done by converting bench and aircraft tiedown test hours to "flight" hours by the ratio of prorated test power to the design prorate of 2250 horsepower. Table 9 summarizes this procedure. This analysis showed that for a confidence level of 50 percent, the roller gear transmission based on test has a MTBR of 150 hours, while for a 95-percent confidence level, its MTBR is only 90 hours. If the reliability growth of the roller gear transmission is assumed to follow that of the S-61, shown in Figure 19, mature reliabilities of 660 hours and 390 hours for 50 percent and 95 percent confidence levels, respectively, may be expected for the roller gear transmission. Several factors must be considered, however, when calculating these MTBR's. First, the testing on which the MTBR's are based was primarily development testing of a radically new concept. One would expect higher failure rates with this kind of test program than with the bench testing of a conventional transmission. Second, the MTBR trend curve of the S-61 on which the mature reliabilities are based is a relatively slow one. The S-61 transmission has a finite TBO. The reliability of a system with finite TBO's tends to improve more slowly than that of a system designed for on-condition maintenance. This occurs because it takes longer to uncover high time failure modes and thus, improvements are made more slowly. The MTBR trend curve of the roller gear transmission would be expected to show much quicker MTBR improvement since it is an on-condition system.

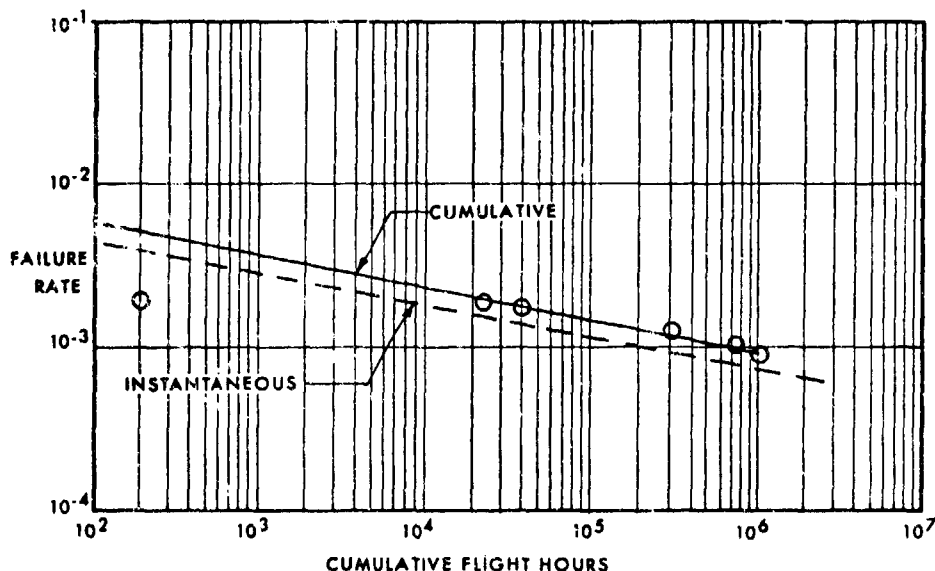


Figure 20. S-61 Transmission Trend Development.

TABLE 9. TEST HOUR - FLIGHT HOUR CONVERSION FACTORS.

Test	Prorated Horsepower	Test Hours	Equivalent Flight Hours
Bench/Initial	2472	53:00	53:23
Bench/Initial	2558	19:00	20:13
Bench/Initial	2348	60:48	62:01
Endurance	2100	400:00	388:24
Aircraft	2559	59:00	53:12
R&M	2348	42:00	42:50
Failure Mode	2771	115:30	128:40
R&M	2261	34:36	34:36
R&M	2900	591:40	678:39

DISCUSSION OF RESULTS

The roller gear test MTBR figures obviously fall far short of the projected MTBR of 1750 hours based on theoretical hazard function analysis. Whether or not the roller gear transmission could achieve its projected reliability depends on several unknowns. First, since nearly all of the failures associated with the roller gear drive were related to electron-beam welds, the quality control of the welds would have to be greatly improved for the roller gear transmission to meet its projected MTBR. Second, it is yet to be determined what effect, if any, the closer tolerance requirements of roller gear components have on roller gear reliability. Third, it is questionable whether the growth curve of the conventional S-61 transmission is applicable to the roller gear transmission. Until these questions are resolved, any assessment of the reliability of a roller gear transmission is premature and should be regarded as such.

CONCLUSIONS

1. The hazard function analysis shows the roller gear transmission to have essentially the same reliability as the baseline planetary transmission. However, this analysis assumed that the roller gear was a completely developed transmission whose components displayed the same failure rates as those of the baseline planetary.
2. A reliability analysis of the roller gear transmission was performed on the basis of testing conducted during this program. This type of assessment is not considered indicative of the potential reliability of the roller gear transmission since the testing included failures related to electron-beam weld development. Many of these development problems have since been resolved.
3. As the roller gear is a highly compact unit, it is susceptible to severe secondary damage in the event of the failure of a single component.
4. The single module design of the roller gear unit makes it readily replaceable in the field. However, a thorough inspection of the roller gear unit requires complete disassembly of the unit because of its compactness. Moreover, the matched set requirement of the pinions makes it expensive to repair. It may be possible to relax the tolerance requirements, thereby permitting replacement of individual pinions. However, this will require a study to determine the effect of relaxed tolerances on load sharing.

RECOMMENDATIONS

Further testing of the roller gear unit is required to determine if analytical predictions which show comparable reliabilities between a fully developed roller gear unit and a two-stage planetary are realistic. However, before further testing is conducted, it is recommended that the potential of the roller gear drive for helicopter transmissions be assessed. Although it offers a compact, high-efficiency final reduction stage, its geometric characteristics limit its application. Furthermore, it is expensive to fabricate when compared to a two-stage planetary and it is highly susceptible to secondary damage.

Therefore, before any further testing of the roller gear drive is conducted, it is recommended that a study be made to determine its role in future helicopter transmissions.

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APPENDIX A

COMPONENT DESCRIPTION

MATERIAL

The material used for the manufacture of the roller gear drive components was SAE 9310 steel processed to AMS 6265 specification. This carburizing steel, a premium quality consumable electrode vacuum melt (CEVM) material, was supplied as forgings for fabrication of the gears and rollers of the roller gear components. The chemical compositions of SAE 9310 to AMS 6260/6265 specification is given in Table A-1.

TABLE A-1. AMS 6260/6265 COMPOSITION.		
Element	Percent	
	Min	Max
Carbon	0.07	0.13
Chromium	1.00	1.40
Manganese	0.40	0.70
Molybdenum	0.08	0.15
Nickel	2.95	3.55
Silicon	0.20	0.35
Phosphorus	-	0.025
Sulfur	-	0.025
Iron	Balance	

The gear teeth and rollers were selectively case-hardened to produce a surface of R_c58-64 with a core hardness of R_c30-34 .

COMPONENT FABRICATION

The complexity of the roller gear components, with stepped pinion assemblies and roller diameters larger than the root diameter of the adjacent gear teeth, necessitated fabricated gear assemblies. Various fastening devices, some incorporating splines to time the gears, locknuts and shrink fits were considered, but were rejected on the basis of low reliability, complexity, weight or difficulty of holding assembly tolerances. Typical of the dimensional tolerances required are those shown for the first-row pinion (Figure A-1). In addition to these individual component tolerances, the seven first- and second-row pinions of the roller gear unit were required to be identical within the individual dual component tolerances.

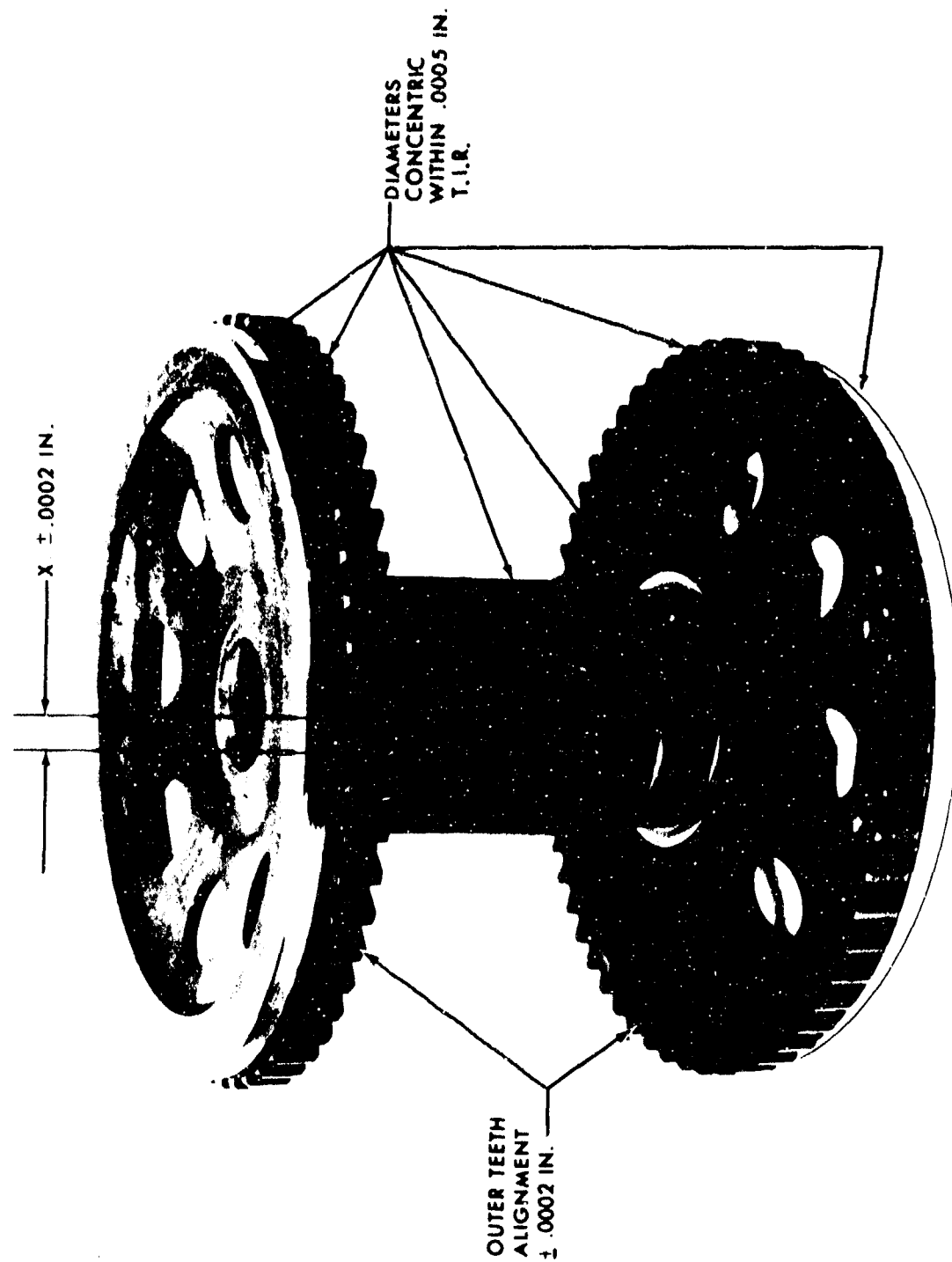


Figure A-1. Tolerance Requirements, First-Row Pinion.

To achieve these tolerances, and manufacture the gears in accordance with aircraft gearing requirements, the gears and rollers were fabricated as individual parts and electron-beam-welded to form an assembly.

Electron-beam-welding is one of the more recently developed fusion weld processes and produces extremely clean welds at very high depth-to-width ratios. The basic principle is illustrated in Figure A-2. Electrons are released from the surface of a low-voltage, high-current cathode by thermionic emission. The electrons are then accelerated toward the target by a high voltage anode and then focused into a narrow intense beam by passing through coils of successively higher potential, in much the same way a lens acts on a beam of light. Upon impingement on the work piece, the kinetic energy of the electrons is converted into thermal energy, causing localized melting and fusion of the part. The entire process is accomplished in a vacuum chamber evacuated to a pressure of 1×10^{-4} mm Hg or less. The molten metal is thereby exposed to less than 2 parts per million of external contaminant and eliminates the problem of interstitial gases which tend to weaken a weld.

Although there are many advantages to electron-beam-welding, including weld integrity and speed, important considerations are its minimal heat effect and its high depth-to-width ratio. The fusion process is so rapid that very little heating occurs in the surrounding metal with the result that there is very little distortion of the welded part, a necessary condition for the close tolerances required by the roller gear components.

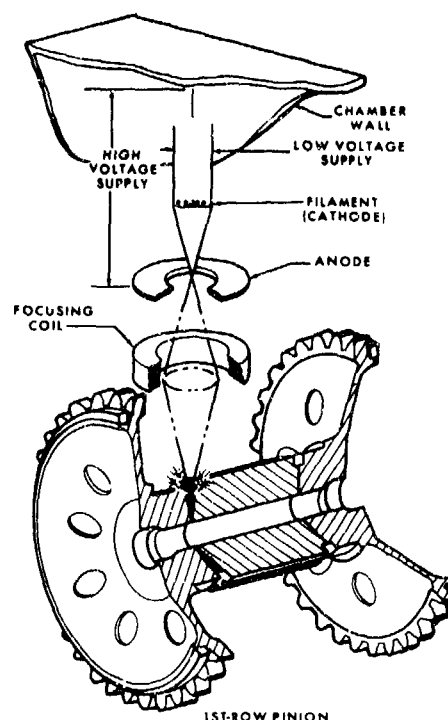


Figure A-2. Electron-Beam-Weld Schematic.

SUN GEAR

The sun gear consists of three elements, a central gear shaft and two identical end rollers (Figure A-3). Case-hardened rollers are electron-beam-welded to the gear shaft after the gear teeth have been case-hardened and finish machined.

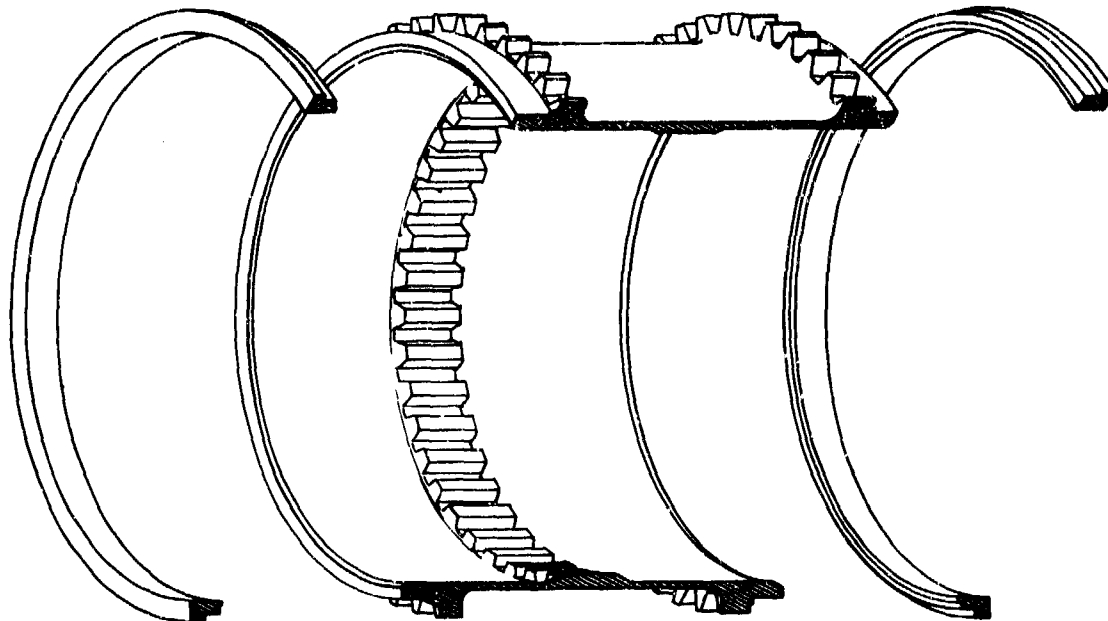


Figure A-3. Sun Gear, Exploded View.

An input spline is located equidistant between the gears to equalize torsional wind-up of the shaft at the gear teeth and assure proper load sharing between the two gears. The maximum permissible misalignment across the gear teeth is limited to .0004 inch. The electron-beam-welds are located at the centroid of the section between the roller diameter and inside diameter of the gear shaft. At this point, the tensile stresses due to bending of the roller rim are a minimum, the shear stresses are a maximum. Highest weld stresses are therefore in the direction of maximum weld strength.

The initial weld configuration incorporated an undercut between the roller and gear shaft to accommodate weld splatter. However, when the weld did not fully penetrate into the undercut, voids remained at the root of the weld, lending incomplete fusion of the joint. To eliminate the stress risers that this induced, the root area of the weld was removed as shown in the modified configuration (Figure A-4).

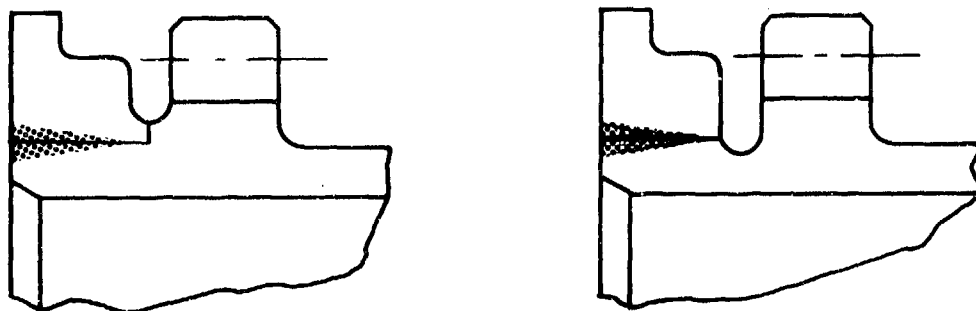


Figure A-4. Initial and Modified Weld Root Relief.

FIRST-ROW PINION

The first-row pinion consists of a central gear member, two identical outer gears, and two outer rollers (Figure A-5). The design configuration permits the inner gear to be carburized, hardened, and finish machined prior to assembly of the outer gears. A close tolerance fit on the locating diameter of the inner and outer gears ensures concentricity, while allowing the semifinished outer gear teeth to be rotated into alignment with the inner gear prior to electron-beam-welding. After welding, the locating diameters are machined off and the outer gears are finish machined to an index tooth on the inner gear to within $\pm .0002$ inch.

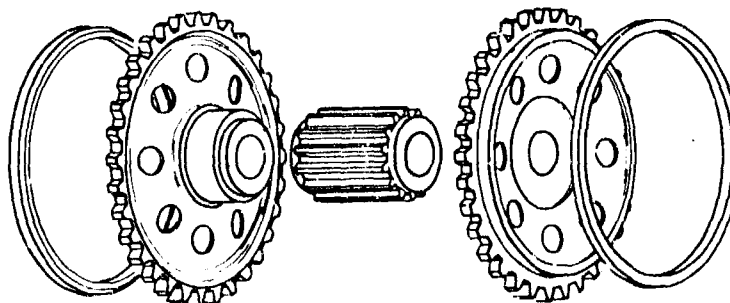


Figure A-5. First-Row Pinion, Exploded View.

The set of seven pinions has to have the identical "X" dimension, shown in Figure A-6, within $\pm .0002$ inch. The "X" dimension can be of any value provided all seven pinions are machined identical. By measuring to the drive side of the inner and outer gear teeth, tooth spacing and thickness tolerances are not included. After finish machining of the outer gears, the rollers are electron-beam-welded and finish ground concentric to the pitch diameter of the gears.

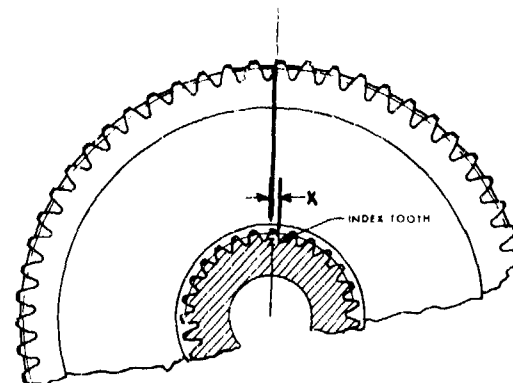


Figure A-6. Gear Teeth Timing, First-Row Pinion.

The initial design of the first-row pinion was configured with a longitudinal weld to join the outer gears to the inner gear (Figure A-7). However, initial development testing resulted in fracture of the small diameter roller, and it became apparent that the weld was too deep for the constraints imposed. To increase the depth of weld penetration would induce too much heat and temper the case-hardened surface of the roller, thereby reducing the required R_c 58-64 surface hardness. A butt-welded joint was designed which eliminated the problem.

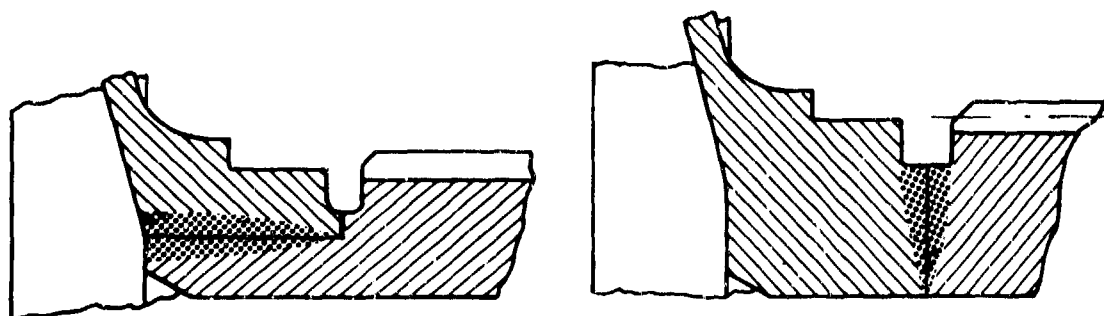


Figure A-7. Weld Configurations, First-Row Pinion.

The outer roller weld was modified to remove root weld porosity. This was accomplished by machining the root area of the weld similar to that accomplished on the sun gear.

SECOND-ROW PINION

The second-row pinions contain the spherical roller bearings which react the transmitted torque. The pinion incorporates a split bearing bore as shown in Figure A-8. The split design enables the center gear to be completely machined before being matched drilled and assembled to the outer gear and roller assemblies to within $\pm .0002$ inch. The outer gears are completely finished before being electron-beam-welded to the roller flange. The rollers are finish machined upon assembly of the completed gear. An interference fit of $.003$ to $.004$ inch between the bearing and gear locates the bearing outer race. Sixteen bolts transmit the torque from the large gear to the small gear. Threaded extraction holes are provided to permit jack-out of the bearing.

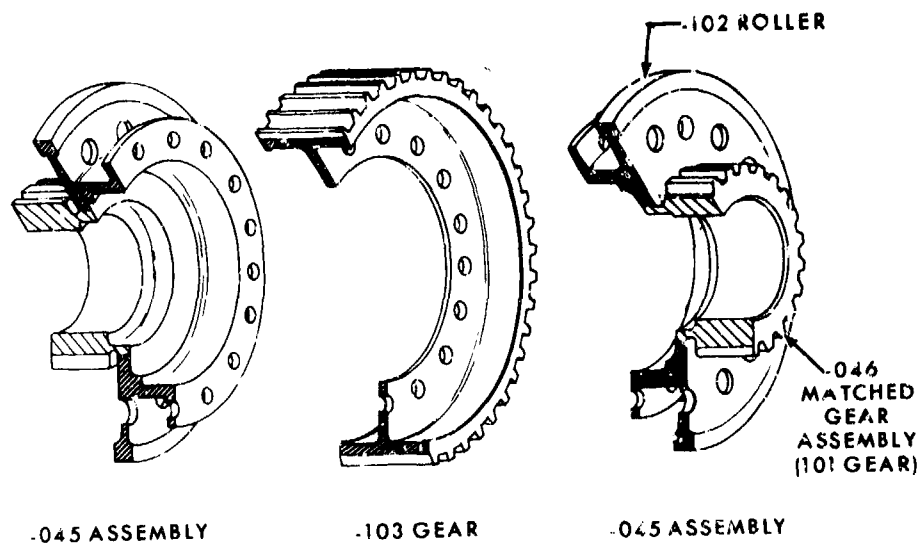


Figure A-8. Second-Row Pinion, Exploded View.

The initial pinion design, shown in Figure A-9, consisted of an assembly of seven fabricated parts: two identical ex.3 gears, two rollers, a central shaft, a center gear, and an end flange bolted to the center gear web by taper-shank bolts. Fracture at the root of the weld between the central shaft and gear flange during bench testing, (8) necessitated the redesign of this gear.

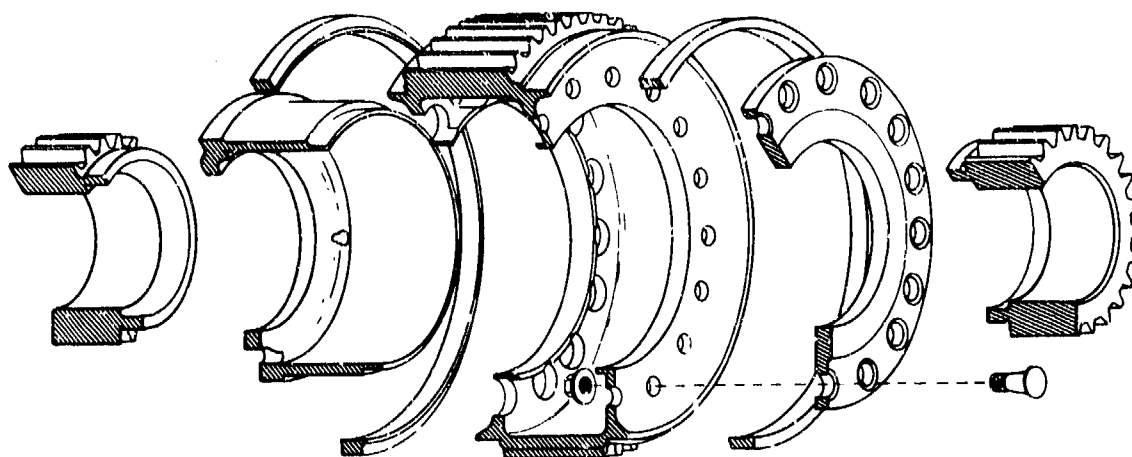


Figure A-9. Second-Row Pinion Initial Design, Exploded View.

RING GEAR

The ring gear consists of two nearly identical gears which are fastened together during assembly of the roller gear unit by countersunk head bolts (Figure A-10). These tapered bolts position the ring gears relative to each other within $+ .0002$ inch. The output flange assembly bolts transmit the output torque of the roller gear unit.

Gear teeth are final machined parallel to each other, within the $+ .0002$ inch tolerance. The curved web of the gear absorbs the high vibratory ring gear stresses induced by the 30° pressure angle of the gear teeth.

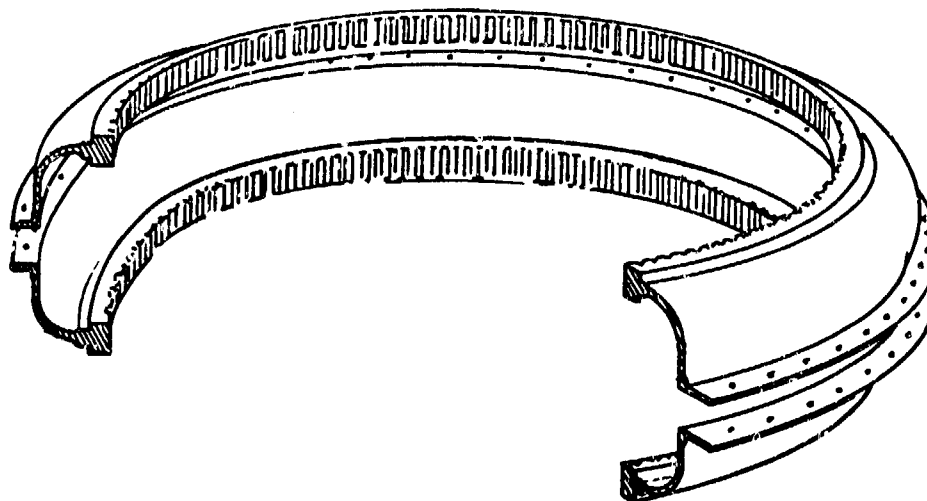


Figure A-10. Ring Gear, Exploded View.

REACTION POST ASSEMBLY

The torque induced in the roller gear unit is reacted by a spline assembly which permits different rates of thermal expansion between the ground reaction and the steel components of the roller gear drive.

Spherical bearings attached to cantilevered posts react the torque generated within the roller gear drive. These posts, designed for constant stress throughout the tapered wall section, react a 10,940-lb tangential load at 3000 hp and induce a slope of .0018 inch/inch in the spherical bearings. The cantilevered posts are sandwiched between a built-up plate assembly that is bolted and doweled to an external spline. Seven bolts, one through each of the posts, support the roller gear unit.

SPHERICAL ROLLER BEARING

The spherical roller bearings are of the two-row design. The rollers and inner and outer races are fabricated from vacuum melt steel; the one-piece cage is fabricated from silicon-iron bronze. The cage rides on the inner race and is silver-plated all over. The roller paths of the inner and outer races are machined to 6 AA (arithmetical average) surface finish.

APPENDIX B

DEVELOPMENT TEST PROGRAM

As in the development of radically new components and systems, there exists a development phase through which the parts progress. During this initial development phase, various minor malfunctions are uncovered and design modifications made. The relationship between design, development, manufacture, and testing is shown in Figure B-1.

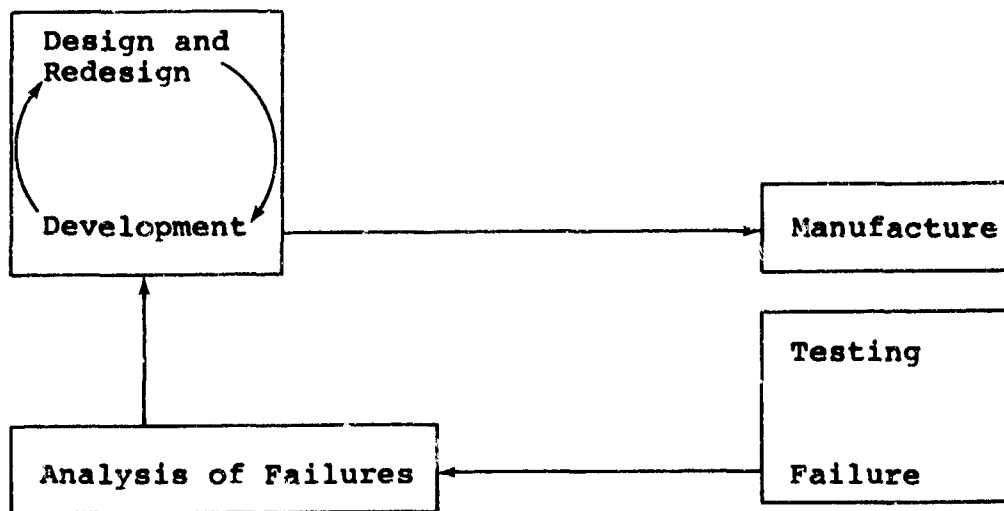


Figure B-1. Product Development Relationship.

In the development of the roller gear drive, initial testing preceeded a 200-hour development test. During this initial bench testing phase, the design of the first-row pinion was modified to incorporate an electron-beam butt welded joint, and the second-row pinion gear/flange EB weld parameters were revised. A 200-hour bench endurance test followed the initial development test. Then a 50-hour test was conducted on an aircraft that was physically tied to the ground. This test revealed a weakness in the structure of an electron-beam weld in the second-row pinions. Further testing, in the regenerative R&M test facility, caused fracture of this pinion though not at the defect for which the test was being conducted. This fracture necessitated the redesign of the second-row pinion and changes to the sun gear and first-row pinion electron-beam weld joints.

A review of the foregoing test and related roller gear fractures is presented in Table B-1. The accumulated test

TABLE B-1. ROLLER GEAR DEVELOPMENT.

Test Phase	Test Time Between Failures (hrs:mins)	Description of Failure	Accumulative Test Time (hrs:mins)
Bench/Initial Testing*	26:30	First-Row Pinion Roller Spall	53:00
Bench/Initial Testing*	9:30	(Second-Row Pinion Gear/Flange Fracture	
Bench/Initial Testing*	9:30	(First-Row Pinion Weld Cracks	72:00
Bench/Initial Testing*	20:54	First-Row Pinion Tooth Breakage	113:48
Endurance	200:00	Magnaglow Cracks Second-Row Pinion	513:48
A/C+	50:00	Second-Row Pinion Roller Cracks	563:48
R&M	21:00	Test Interrupted to Conduct Failure Mode Test	605:45
Failure-Mode+	57:45	Second-Row Pinion Fracture	721:75
R&M	17:18	First-Row Pinion Tooth Fracture	755:51
R&M	295:50	First-Row Pinion Spall	1347:31
*Reported in USAAMRDL TR 73-98C			
+Reported in USAAMRDL TR 73-98E			

hours constitute the total number of hours attained by both roller gear units undergoing testing in the bench aircraft, and R&M regenerative test facilities. Of these development problems, only the second-row pinion gear/flange fracture and the fracture of the second-row pinion were of a catastrophic nature. The other did not seriously affect the function of the roller gear units.

ROLLER SPALLING, FIRST-ROW

After 26:30 hours of bench testing, microscopic examination of a spalled first-row pinion small-diameter roller revealed fractures initiating from voids in the electron-beam weld. These fractures ran parallel with the surface of the roller at a depth of 1.5 inches (Figure B-2).

Subsurface stresses produced by the roller contact forces caused cracks to extend from the voids to the surface of the roller. The voids were the result of the depth of the weld and the weld parameters developed to avoid inducing too much heat into the pinion, which could temper the case-hardened surface of the roller.

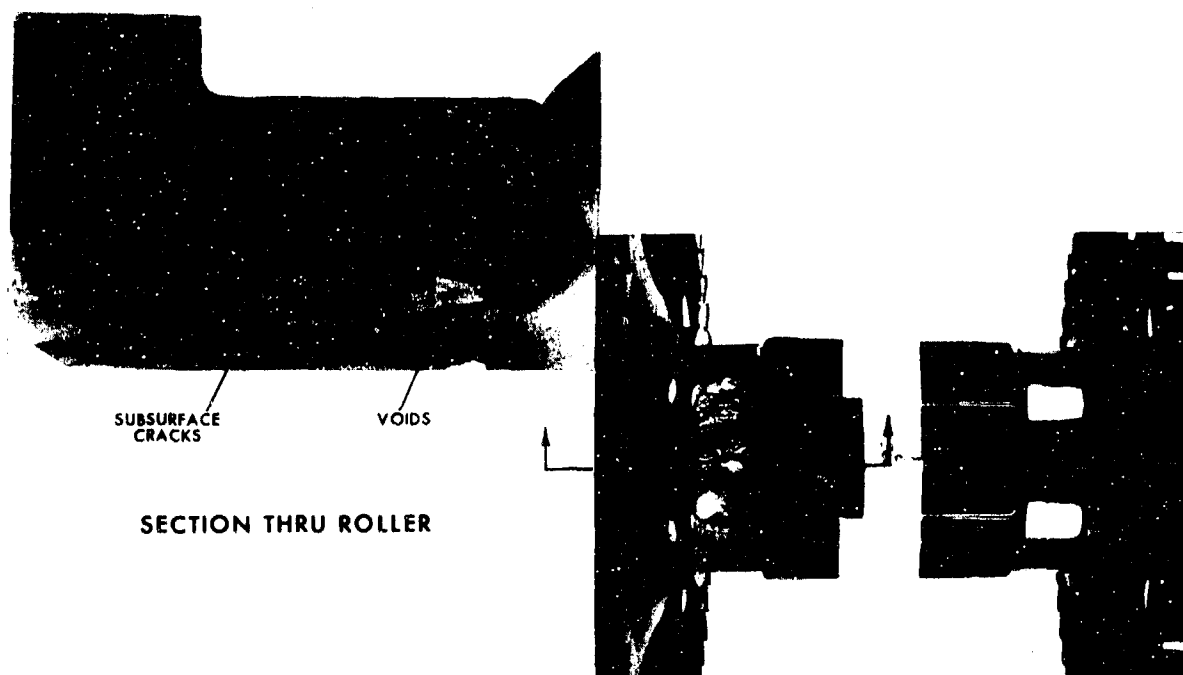


Figure B-2. First-Row Pinion, Roller Spalling.

PINION WELD CRACKS, FIRST-ROW

Testing with this same pinion design continued for another 9:30 hours, whereupon ultrasonic inspection revealed similar cracks in the weld area. The pulse echo ultrasonic inspection method was developed as a result of the previously-mentioned fracture. Prior to ultrasonic inspection, there was no reliable method of determining the integrity of the welds. Radiographic inspection proved to be unreliable for determining if fusion of the joint surfaces had taken place during EB welding. In addition, the small-diameter bore in which the film was placed necessitated many exposures to obtain X-rays perpendicular to the surface under examination. Magnetic particle inspection revealed flaws only on the entrance side of the weld. Even then, a missed seam was often masked by a cosmetic EB weld which was sometimes included to fill in an undercut formed during a deep weld.

GEAR/FLANGE FRACTURE, SECOND-ROW

The failure of magnetic particle inspection to reveal an incomplete weld also led to the fracture of a second-row pinion shown in Figure B-3. It was found that machining of the welded joint to remove traces of weld splatter had smeared metal over an incomplete joint which magnetic particle inspection had failed to detect. To prevent incomplete fusion of this weld from reoccurring, the undercut area between the gear and the flange was increased to allow a heavier EB weld blast shield to be included. This permitted the changing of weld parameters to produce a wider area of fusion.

Hereafter all welds were subjected to ultrasonic inspection.

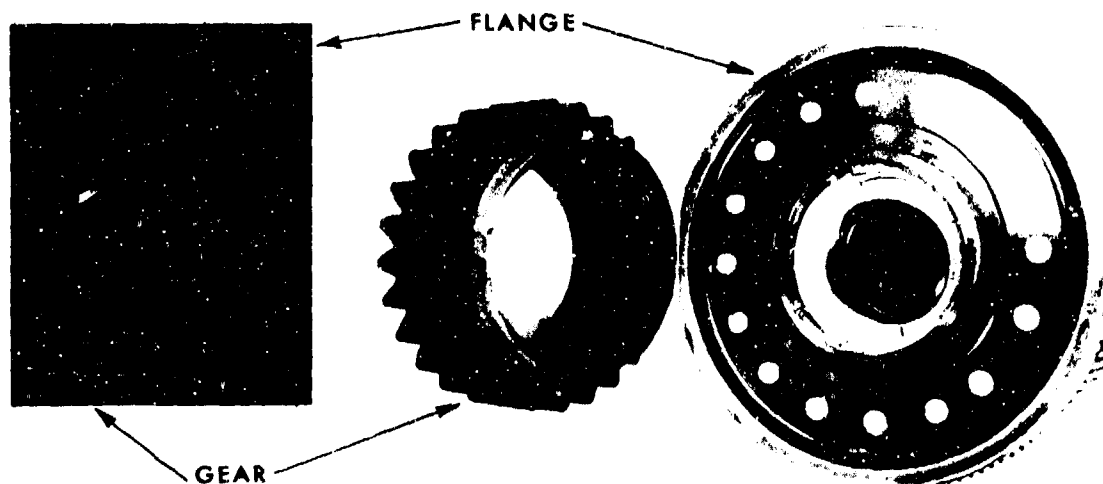


Figure B-3. Second-Row Gear/Flange Fracture.

GEAR TOOTH BREAKAGE, FIRST-ROW

With the incorporation of the butt weld design first-row pinion and modified second-row pinions, initial testing was continued. After approximately 21 hours of regenerative bench testing of the roller gear transmissions, the teeth of the first-row pinion, 27-tooth gear fractured (Figure B-4). Examination of the fractured gears revealed that the weld heat-affected zone extended into the carburized zone at the root of the gear teeth. This caused a transition interface which acted as a stress concentration between the compressive stress zone of the carburized teeth and the residual tensile stress zone of the weld. To prevent reoccurrence of this, the groove distance between the roller and gear was increased, thereby ensuring that the heat-affected area would not encroach into the carburized layers at the gear root.

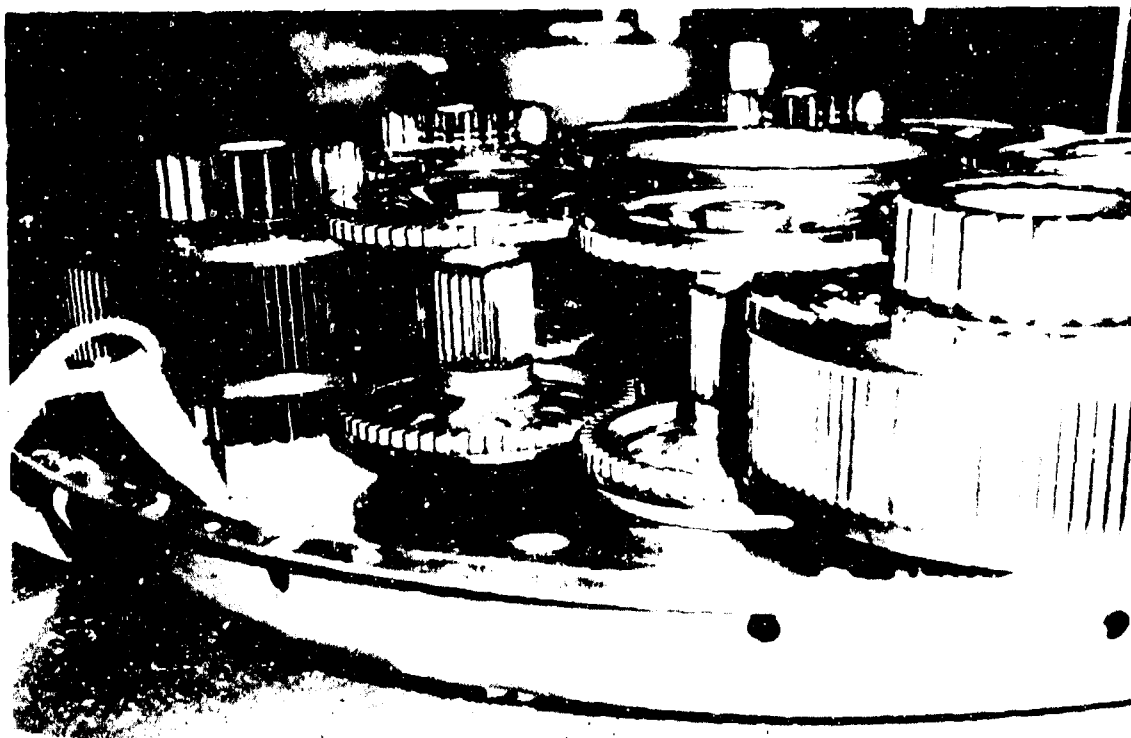


Figure B-4. First-Row Pinion Tooth Breakage.

BEARING BORE CRACKS, SECOND-ROW PINIONS

At the conclusion of the 200-hour endurance test, inspection revealed cracks in the second-row pinions in both the dummy (or slave) and the test roller gear transmissions. Cracks in three second-row pinions in the dummy gearbox extended from the gear/flange weld. These particular components were fabricated prior to the previously-mentioned second-row pinion fracture and used in the 26:30 hours and 20:54 hour tests. Prior to installation, the gear/flange assemblies were polished and inspected; however, on dissection of the cracked assemblies, voids were found on the exit side of the weld. None of the second-row pinions in the test gearbox, which incorporated the wider weld, exhibited any signs of cracks in this particular area, thus proving the wider weld to be successful. Four of the second-row pinions in the test gearbox did show crack indications in the bearing bore (Figure B-5). Metallographic examination revealed that fatigue cracks extended from voids at the weld root.

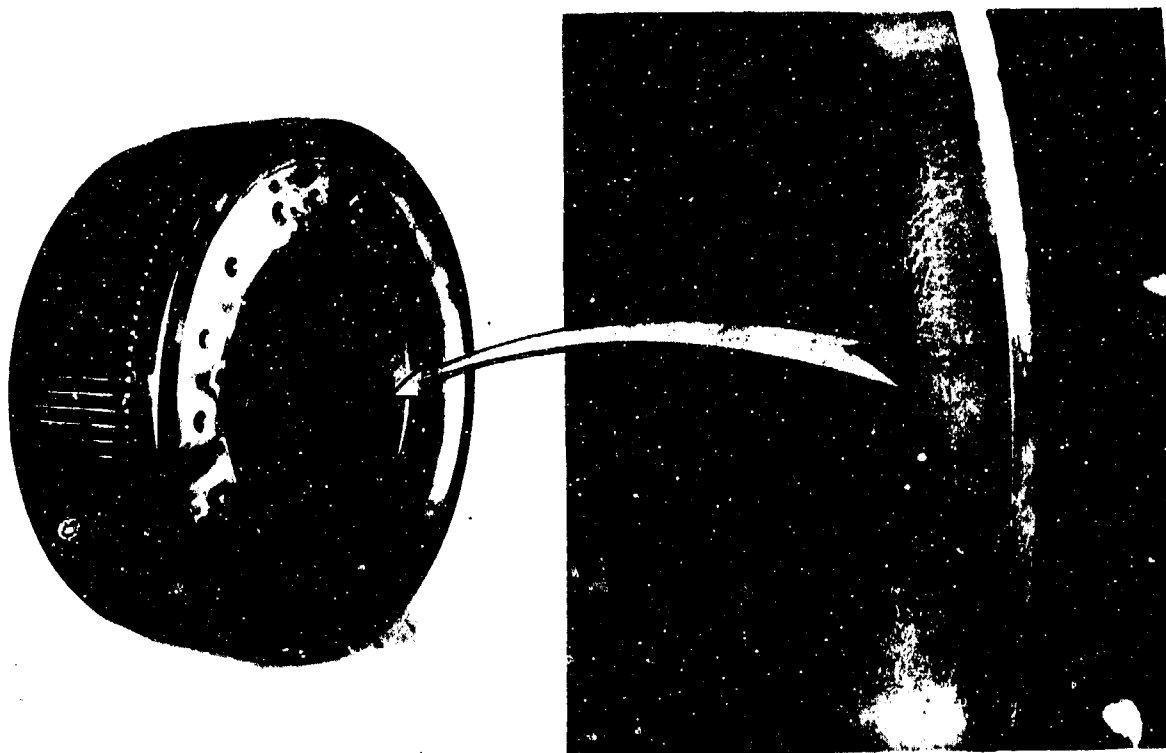


Figure B-5. Bearing Bore Cracks, Second-Row Pinion.

FRACTURE, SECOND-ROW PINION

Further testing of the roller gear unit continued with the transmission retrofitted on a tiedown Sikorsky S-61 aircraft (Figure B-6). The purpose of the test was to evaluate the dynamic characteristics of the aircraft engine, rotor and control systems when interfaced with the roller gear transmission.

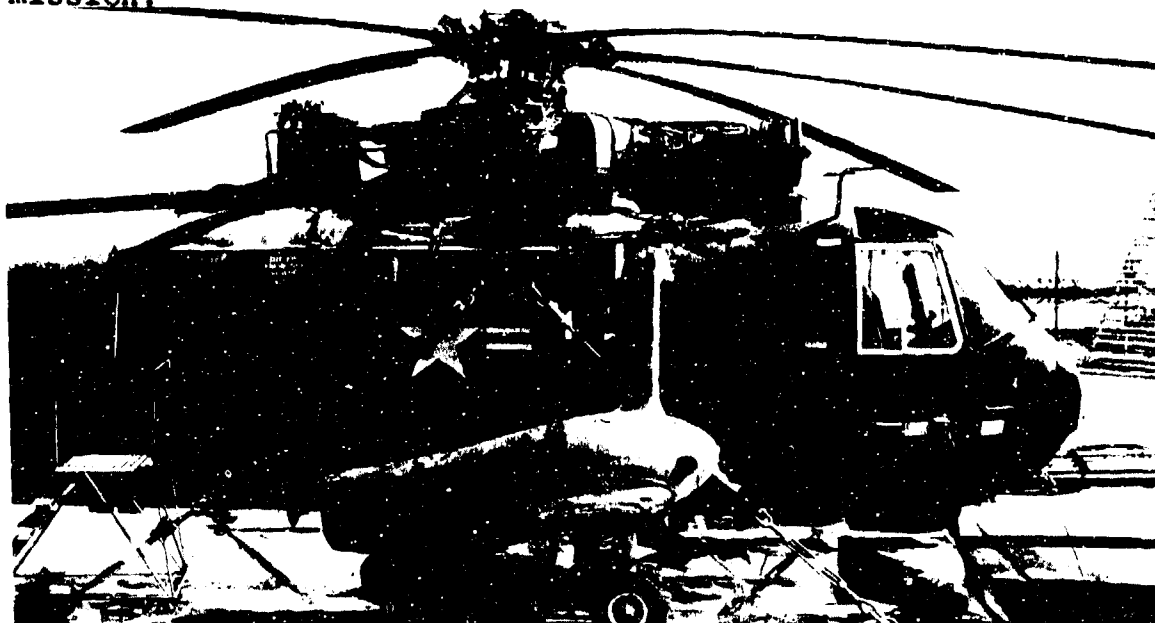


Figure B-6. Aircraft Tiedown Test.

Prior to assembly of the roller gear transmission, the electron-beam welds in the sun, first-row and second-row pinions were subjected to ultrasonic inspection, and facsimile recordings, c-scans, were obtained. At the completion of the 50-hour tests, the inspection was repeated and the recordings were compared. Degradation was visible in 8 of the 14 welds. The degraded area from pinion S/N 63 was removed, Figure B-7, to reveal a fractured zone 1.0 inch by .22 inch wide. Metallurgical inspection revealed that intergranular cracks at the weld pull-out zone developed between the coarse grained melt zone and the finer grained heat-affected zone.

A replacement for the dissected pinion was obtained, and the roller gear unit was subjected to further testing in the reliability and maintainability test facility to obtain crack propagation rates for the "cracked" roller welds. Ultrasonic inspections at 15 and 25 hours test time (at 3000 hp) showed no significant increase in flaw size. At 57:45 hours test time, fracture of a second-row pinion occurred (Figure B-8). Examination revealed that initial fracture occurred at the bearing bore weld, Figure B-9, and a secondary fracture occurred in the area of the outer roller weld due to the large

leverage load from the unsupported ring gear mesh forces. Ultrasonic inspection of the roller welds revealed no significant increase in the size of the flaws under investigation. This fracture of the second-row pinion resulted in a redesign of the second-row pinion. Testing of the roller gear units with the incorporation of this latest design pinion continued in the reliability and maintainability test facility.



Figure B-7. Ultrasonic Crack Detection.

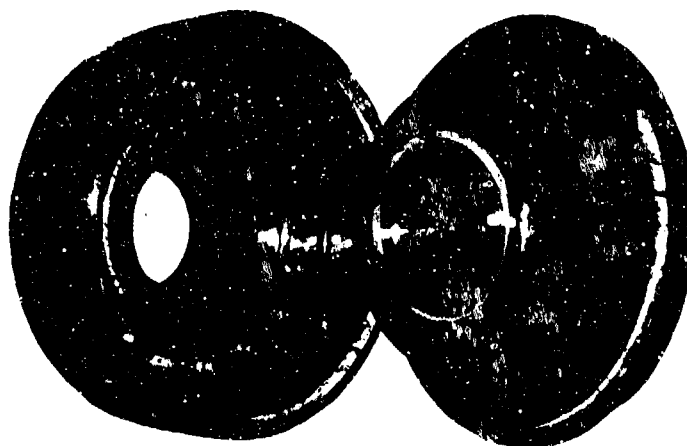


Figure B-8. Fracture, Second-Row Pinion.

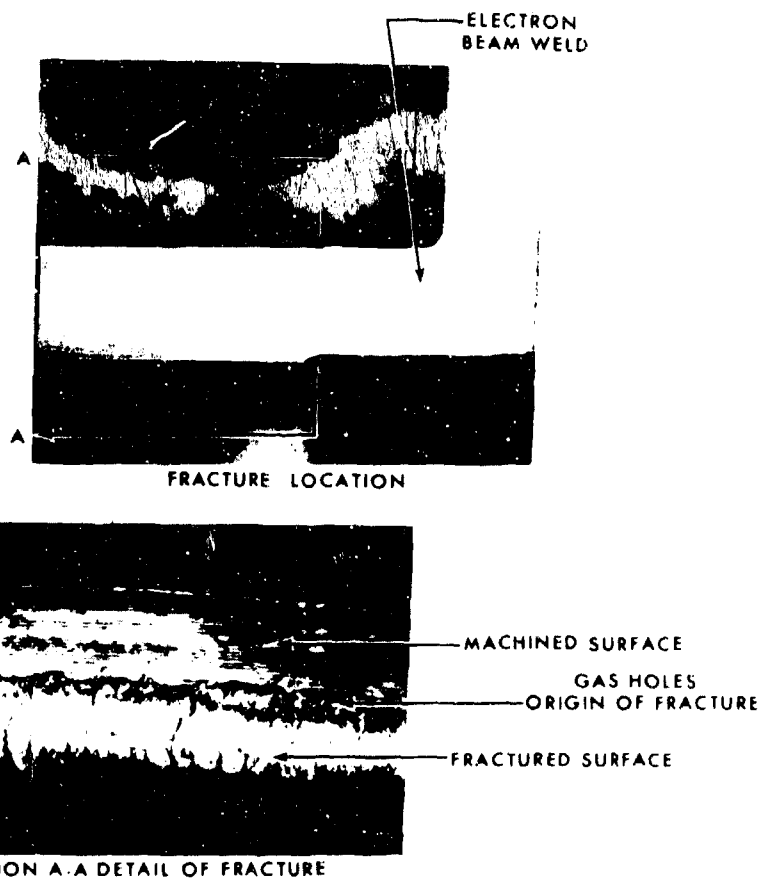


Figure B-9. Fracture Location.

APPENDIX C

RELIABILITY AND MAINTAINABILITY TEST

INTRODUCTION

The R&M testing of the roller gear unit was conducted in a regenerative test facility. This facility employed two identical roller gear units mounted back-to-back (Figure C-1). The high- and low-speed shafts of each unit are connected together to form a closed loop, whereby power can be recirculated throughout each roller gear unit. One unit is designated the test unit, while the other reacts the loads and speeds delivered by the test facility. The test load is applied by twisting one shaft relative to another to a prescribed torque, whereupon the shafts are locked in place. With the system brought up to speed, power is transmitted through the roller gear units, but is contained within the closed loop. Thus, the prime mover power requirements need only be equal to the power losses in the system, i.e., the power loss due to friction.

Requirements

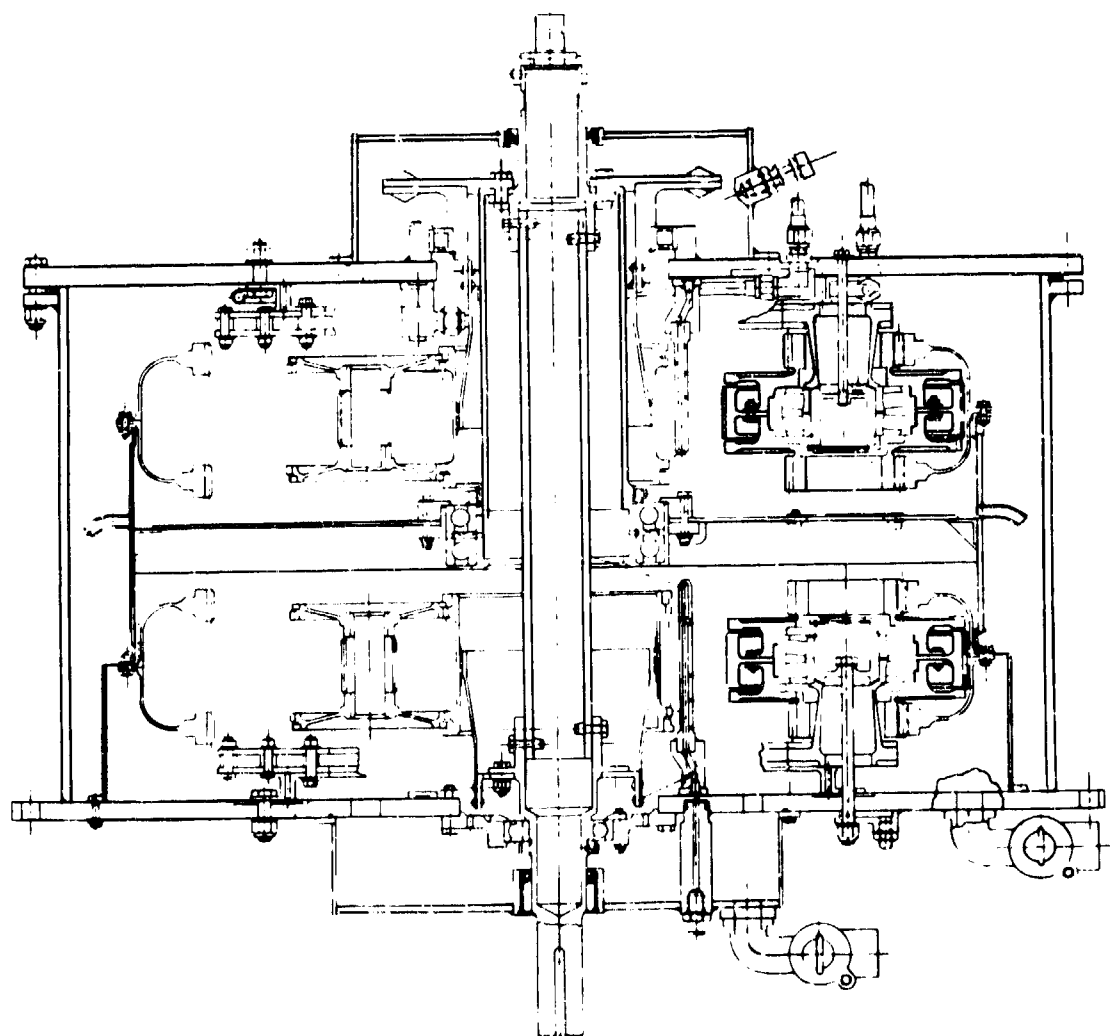
The specification for the test facility was to duplicate the installation of the roller gear unit in a helicopter transmission housing and apply maximum transmitted power. A brief outline of the specification is given in Table C-1.

TABLE C-1. TEST FACILITY SPECIFICATION.

Input Speed	4045 rpm
Torque	3895 ft-lb
Drive power (max)	150 hp
Oil flow (per unit)	10 gpm

FACILITY DESCRIPTION

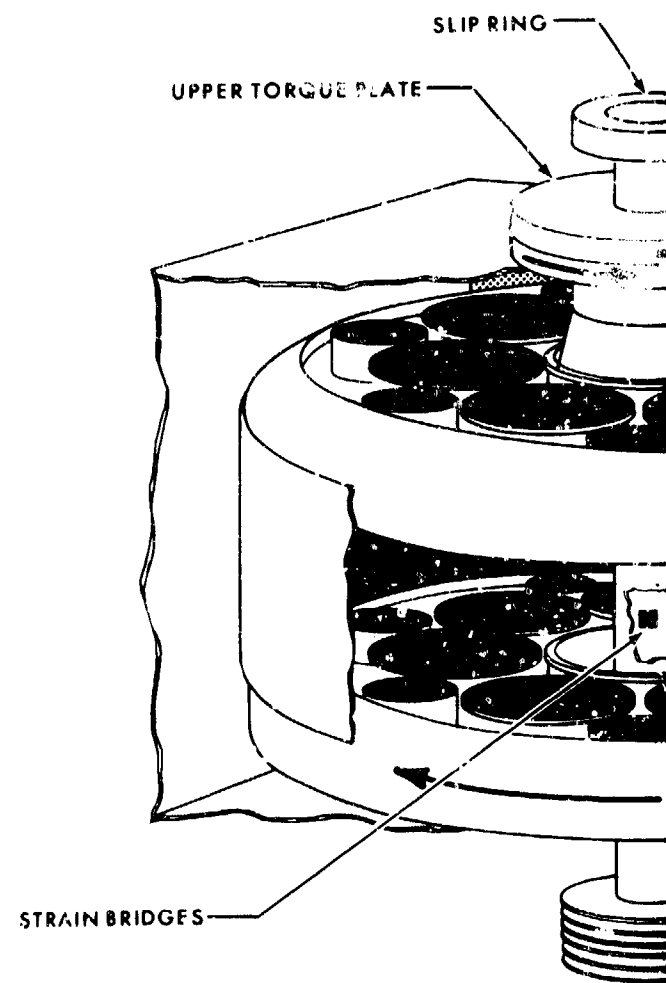
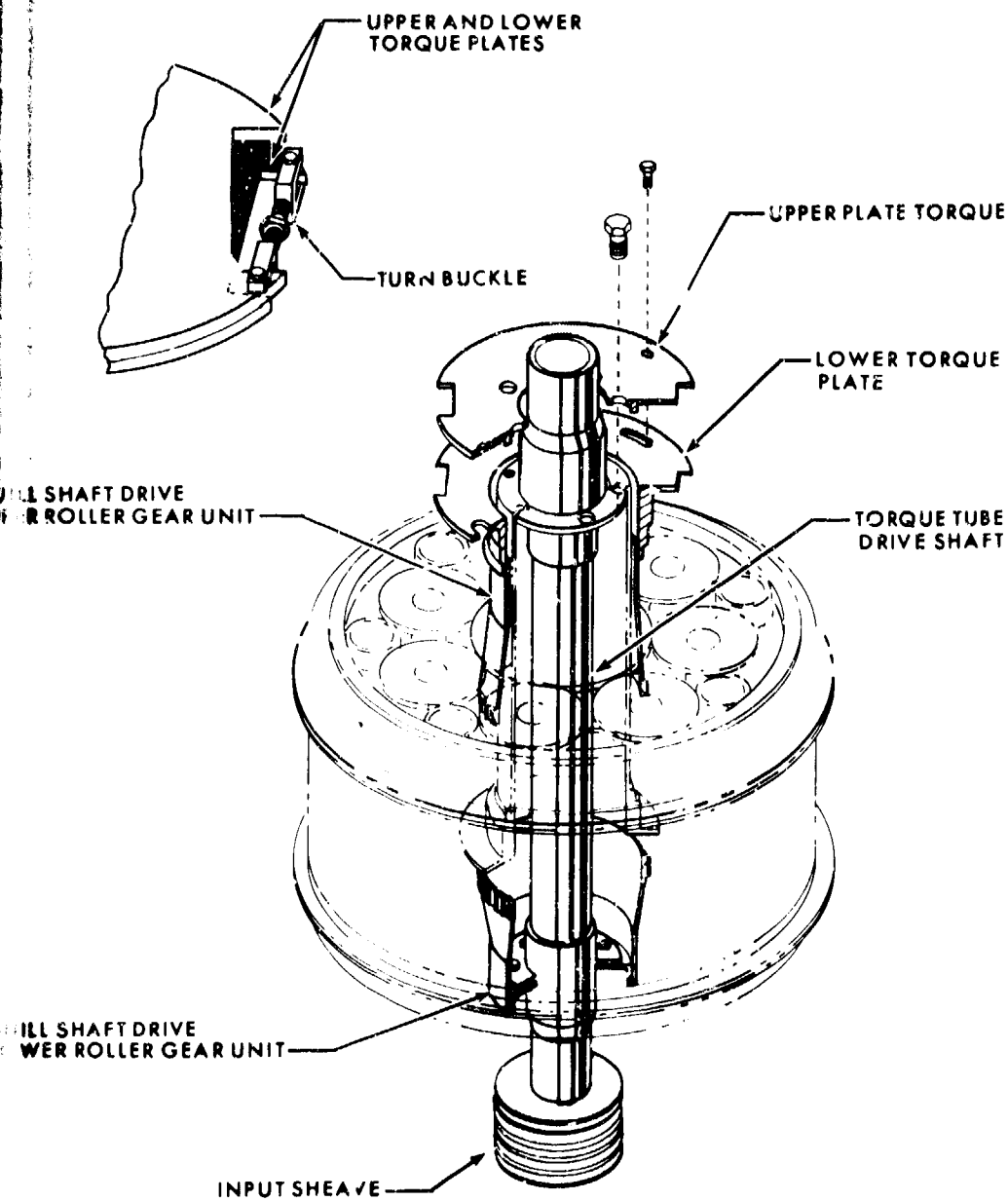
The facility consists essentially of an electrically driven gearbox within which are two roller gear units. The gearbox is mounted within a fabricated steelwork structure which locates the prime mover, the hydraulic reservoir, the oil/water cooler, and the electrically driven hydraulic pumps. The facility is housed in a brick-enclosed structure and is observed from an adjoining control room (Figures C-1 and C-2).



QUILL SHAFT DRIVE
UPPER ROLLER GEAR UP

QUILL SHAFT DRIVE
LOWER ROLLER GEAR

Figure C-1. Regenerative Roller Gear Test Facility.



UPPER PLATE TORQUE

UPPER TORQUE PLATE

SLIP RING

LOWER TORQUE PLATE

LOWER TORQUE PLATE

REACTION TORQUE /
UPPER ROLLER GEAR UNIT

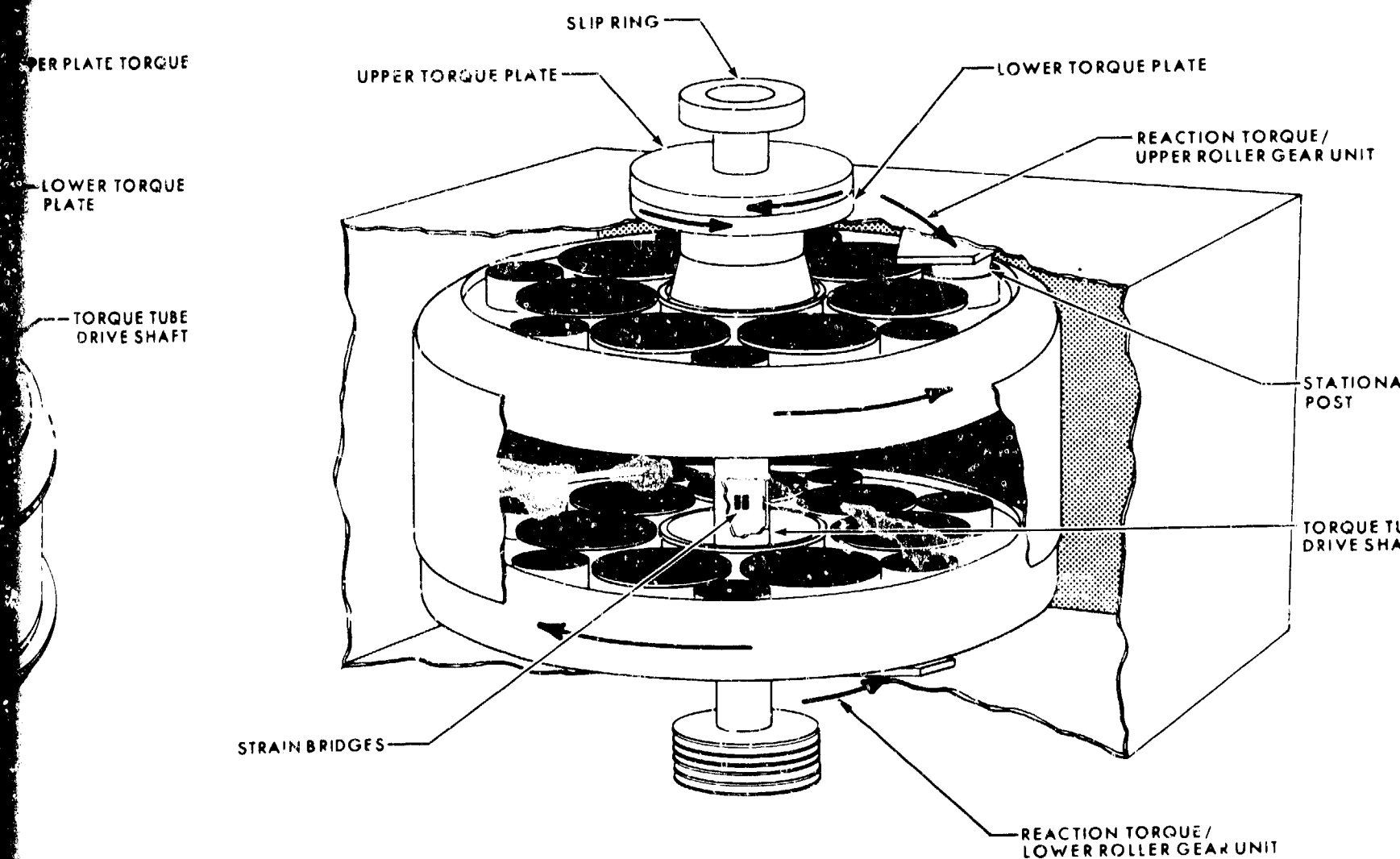
TORQUE TUBE
DRIVE SHAFT

STATIONARY
POST

TORQUE TUBE
DRIVE SHAFT

STRAIN BRIDGES

REACTION TORQUE /
LOWER ROLLER GEAR UNIT



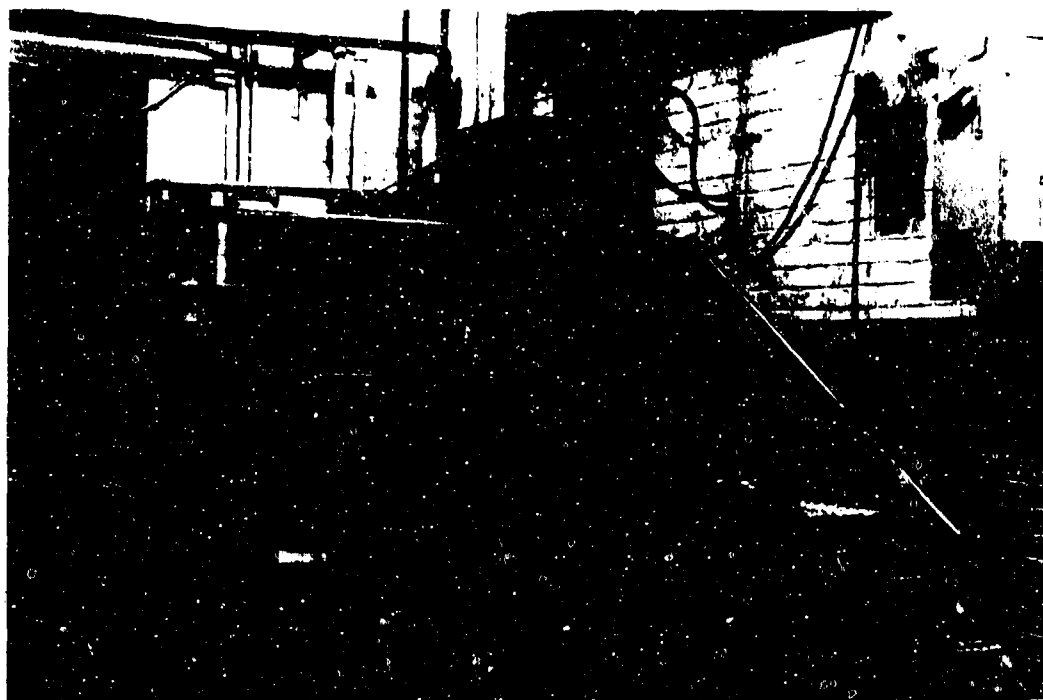


Figure C-2. Test Facility.

Drive

The power is supplied by a two-speed 150-hp A.C. motor. The drive from the 6000-rpm motor is through 16 vee-belts with differential sheaves providing the speed reduction from the gearbox. The gearbox sheave is straddle-mounted by two grease-lubricated, single-row, deep-grooved ball bearing pillow blocks. Between the gearbox input shaft and the sheave are two flexible couplings located on each side of a torque shaft. The couplings, comprised of stainless steel laminations separated by synthane spacers, absorb any misalignment between the sheave and gearbox. The power required by the motor is recorded on a dial indicator located on the control console.

Lubrication

Lubrication oil, to MIL-S-23699, is supplied to each roller gear unit. A central membrane between each roller gear unit prevents the upper roller gear unit drain-back oil from lubricating or contaminating the lower unit. Two separate chip detectors monitor debris from each roller gear unit and indicate, at the control console, an accumulation of magnetic particles.

Oil is delivered from a reservoir by a 25-gpm electrically driven pump at 75 psi through a 40-micron filter to an oil/water heater exchanger. The flow rate and pressure of the cooled oil is monitored prior to entry into each roller gear unit. Oil is scavenged from the gearbox through the chip detectors and strainers back into the reservoir, again by an electrically driven pump.

The lubrication of the upper roller gear unit duplicates the system that would be used in a helicopter gearbox. Seven probe jets extend down between the first-row gears and sun gear and direct oil to the gear meshes and roller contact points. A separate line feeds a circular manifold which delivers oil into the seven stationary posts that support the second-row pinions. Each bearing has a slotted upper face, which allows oil to drain through the bearing in the event that the primary mode of lubrication, through radial holes in the center of the inner race, should become blocked. Oil is centrifuged from the upper unit by the rotating membrane attached to the ring gear and is then scavenged through the combined chip detector and strainer.

The lubrication supply to the lower unit is essentially the same as that to the upper unit except for the supply to the spherical bearings. Here, oil is forced up through the retaining bolts and through radial holes in the head of the bolt. Oil is allowed to collect in the pocket of the inverted posts where it drains into the spherical bearings. Oil drains back into the sump through holes in the base plate, where it is scavenged through a combined chip detector and strainer.

The upper roller bearing of the facility is lubricated by a jet directed onto the bearing. Oil to the duplex bearing supporting the membrane is supplied from an orifice on each of the probe jets into a pocket on the inside of the bearing support shaft, where it is centrifuged through the bearing. The lower ball bearing is lubricated by the drain-back oil from the lower roller gear unit.

RELIABILITY AND MAINTAINABILITY TEST GEARBOX

The gearbox is designed to test two roller gear units in a back-to-back arrangement. The upper unit is suspended from the top cover, while the lower unit rests on the base. Each unit has splined connections to the gearbox housing, which reacts the torque transmitted through the spherical bearings.

Assembly

Figure C-3 shows the assembled unit both stationary and rotating prior to assembly into the R&M test gearbox. The R&M gearbox is assembled by positioning the roller gear unit so that the spline connection of the roller gear unit can mate with the internal spline bolted to the base of the R&M gearbox (Figure C-4). The lower quill shaft mates with the sun gear. The central membrane is then bolted to the lower ring gear and the upper roller gear unit, which is located by the upper ring gear and membrane flange. The top cover, complete with seven probe jets and internal spline to react torque, is placed in position and bolted to the housing. The R&M gearbox is then installed in the test facility.

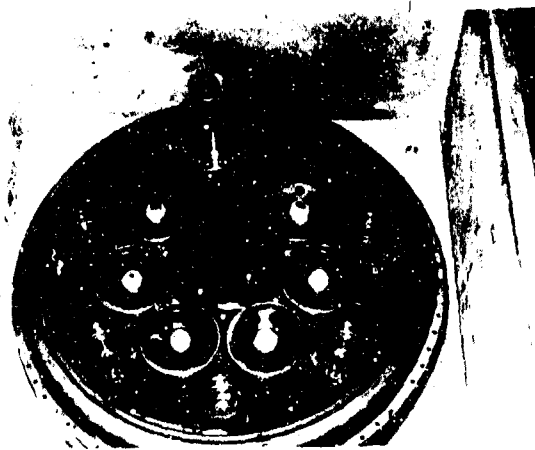


Figure C-3. Roller Gear Unit, R&M Test.

The input to the gearbox is a three-piece fabricated torque tube driveshaft. Attached to the upper end are two torque plates, one connected directly to the input driveshaft, the other to the sun gear of the upper roller gear unit. The lower end of the torque tube shaft is connected to the sun gear of the lower roller gear unit. Turnbuckles apply torque to the driveshaft by rotational displacement of the torque plates. Bolts clamp the torque plates together when the desired load level is obtained. The roller gear torque loop is completed by a fabricated support shaft which connects the ring gears of each unit. This shaft is welded to the central membrane and supported from the torque shaft by a very flexible slotted shaft. A duplex bearing set allows for the differential speed of the driveshaft and ring gear.



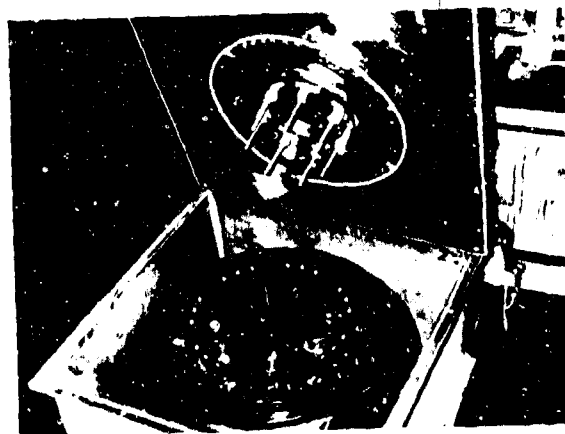
LOWER REACTION SPLINE



LOWER ROLLER GEAR UNIT



CENTRAL MEMBRANE



UPPER ROLLER GEAR UNIT

Figure C-4. Assembly, Roller Gear Units, R&M Test.

Problems were encountered with the torque plate drive to the upper roller gear unit. Two bearing retainer clips fastened to the shaft provided a stiff path for the drive torque and caused the fasteners to fatigue, with the resultant breakage of the quill shaft. A second shaft failed when fasteners sheared due to excessive clearance in the torque tube shaft holes when match drilling the quill shaft. A third shaft was modified and assembled with heavier fasteners; this fractured at 295:50 hours of R&M testing.

Instrumentation

Requirements for monitoring the performance of the roller gear unit during testing consisted of measuring the roller gear post deflections to determine load sharing characteristics and measurement of power losses within the system for efficiency calculations. In addition, oil flow, pressure and temperatures to each roller gear unit and the temperatures of the second-row spherical bearings and facility upper roller and lower ball bearings were recorded.

Load sharing of the second-row pinions was determined from monitoring the deflections of the stationary second-row pinion posts obtained from strain gauges bonded to the inside diameter of the posts. The power losses of the R&M gearbox were obtained by measuring the input torsional stress required to "turn-over" the roller gear units. The efficiency of the roller gear units was obtained by deducting the power absorbed by the facility bearings. The torsional strain was obtained from two strain gauge bridges bonded to the input torque shaft.

The spherical bearing temperature measurements were obtained by two wire, washer-type thermocouples attached to the spherical bearing posts.

TEST PLAN

The test requirement for the roller gear transmission specified a 300-hour endurance test. Within this test time, efficiency tests were to be conducted. The initial 22.5 hours consisted of spectrum running, whereupon continuous testing at the design power was conducted. The test was conducted as per Table C-2. The designed maximum continuous power for the 19.858:1 reduction ratio roller gear drive was 3000 hp at an output speed of 203 rpm.

TABLE C-2. R&M TEST POWER SPECTRUM	
Horsepower	Time (hrs:mins)
1440	6:00
2170	4:00
2400	10:30
2640	1:00
3000	274:20

R&M TEST COMPONENTS

Test Unit

The roller gear components of the test unit were all zero time parts except for the ring gear. The ring gear had been used in a 50-hour aircraft tiedown test and a 54-hour failure mode test, reported in Reference 9. Prior to assembly, the ring gear had passed magnetic particle inspection and visual and dimensional inspections.

The electron-beam welds in the sun gear, first-row pinions and second-row pinions were subjected to ultrasonic inspection by the pulse-echo method and C-scan recordings obtained. The spherical bearings assembled into the second-row pinions were also of zero time.

Dummy Unit

The first-row pinions and second-row pinions assembled in the dummy, i.e., lower roller gear unit, were zero time components. The ring gear had accumulated 75 hours and the sun gear 21 hours. Prior to assembly, all components were subjected to inspections and C-scan recordings obtained of the electron-beam welds.

TEST OPERATION

During the 300-hour test program, only one malfunction of the roller gear units occurred. This was the fracture of a first-row pinion small-diameter gear tooth at 17:15 hours. Apart from this, the roller gear operated flawlessly. Three fractures of the quill shaft that drives the upper roller gear unit occurred; however, their fracture had no bearing on the operation of the roller gear unit except for the downtime, which offered an opportunity to inspect the units.

Operation of the R&M test facility resulted in a high pitch noise. At the control console, the level was 93 dB and outside the test cell, 15-feet from the brick enclosure wall, the level was 89 dB. To isolate the cause, vibration recordings were taken at 10, 109, and 260 test hours.

The efficiency of the roller gear units was also determined during this test.

First-Row Pinion Tooth Fracture, 17:15 Test Hours

A total of 17 hours, 15 minutes of spectrum testing had been completed when, while transmitting 2400 hp, the test was stopped due to a change in pitch of overall noise. There was no visual indication from the temperature gauges or strain gauge instruments of a malfunction. Examination of the chip detector/strainers revealed a slight accumulation of magnetic particles from the area of the upper roller gear unit, the chips were insufficient to light the chip indicator on the control console. The detector for the lower unit was relatively clean.

Clean detectors were installed and the gearbox operated at no-load for less than one minute at 50% speed. The overall noise level differed from previous runs, and examination of the detectors revealed a further accumulation of magnetic particles. Examination of the upper roller gear unit revealed a tooth fracture of the small-diameter first-row pinion gear, serial number 67 (Figure C-5).

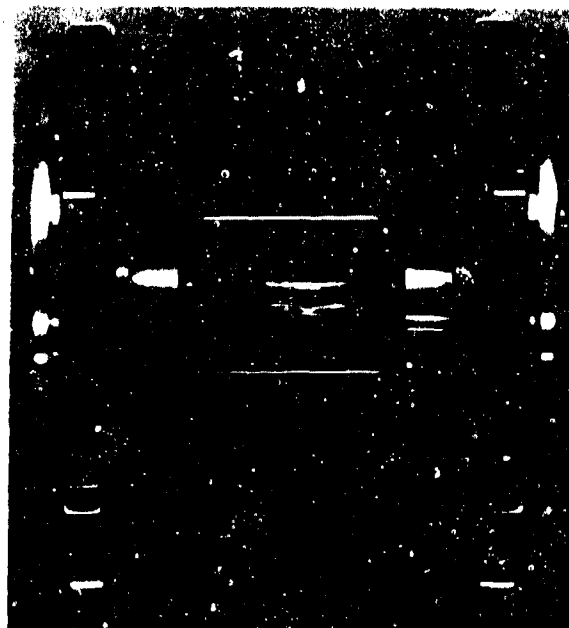


Figure C-5. Tooth Fracture, First-Row Pinion.

Preliminary examination of the fracture, which was located in the center of the gear face and which measured approximately 1.5 inches long, indicated a bending fatigue fracture. Magnetic particle inspection revealed a similar bending fatigue fracture originating on the drive side of a tooth, 5 teeth away. A hardness transverse across a sectioned tooth indicated a .017 inch case depth with a surface hardness of Rc56.5. Although this hardness is 1.5 points below that specified (Rc58-64), it is not considered completely contributory to the cause of fracture, though decreasing hardness does lower the allowable stress and/or the life of the gear. A dimensional check of gear tooth alignment, tooth timing, tooth-spacing and roller dimensions revealed them to be within drawing tolerances. Visually, the gear patterns were excellent and no indications of wear were visible on the rollers. This was true of all the gears in both roller gear units. Secondary damage sufficient to scrap the pinion was incurred by first-row pinion S/N 50 on two adjacent teeth. The sun gear also suffered a chipped tooth. Second-row pinions incurred slight debris damage.

All first- and second-row pinions (from both upper and lower roller gear units) showed interference between the corner radius on the first-row roller and shoulder and the radius on the second-row pinion roller. In some cases, especially on the top first-row pinion rollers of the lower unit, it resulted in pitting of the corner radius (Figure C-4), while in the majority of cases a polished line occurred. The outer-race of the spherical bearings showed evidence of movement in the second-row pinions which, although troublesome, was not considered detrimental to the operation of the roller gear unit.

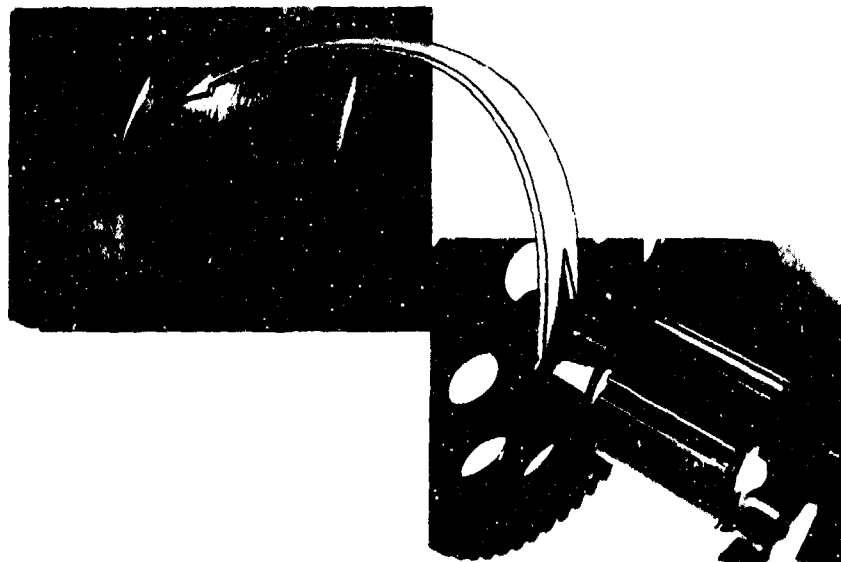


Figure C-6. Roller Interference, First-Row/Second-Row Rollers.

Metallurgical examination of the fractured pinion revealed a cracked tooth as well as the fractured tooth. The teeth failed in fatigue with origins in the root radius below the tooth wear pattern. Manganese phosphate process pitting was evident at the origin sites. This pitting, although normal with the process, is not desirable and could have been contributory to the fracture. However, the other pinions were subjected to the same process and no cracks were found. No direct cause for the failure could be found. The damage incurred required the replacement of the sun gear, first-row pinions and two second-row pinions.

Quill Shaft Fracture, 22:30 Test Hours

While operating at 2400 hp at 22:30 test hours, loss of torque occurred in the closed power loop of the roller gear units. Investigation revealed the fracture of the upper quill drive-shaft. Torsional fracture occurred along two diametrically opposed axes at a 45° angle to the spline. This caused the quill shaft spline diameter to collapse, causing disengagement from the mating sun gear spline and releasing the torque in the R&M test system.

A disassembly of the upper roller gear unit showed that it had not sustained any major debris damage. The rollers of the second-row pinions did sustain scratches and a first-row pinion, serial number 39, exhibited a distress band 1/8 inch wide on the top roller adjacent to the small diameter gear. Examination revealed a frosted zone with shallow peeling (Figure C-7). This type of distress was first noted on first-row pinion S/N 33 after the 200-hour bench test⁽⁸⁾ and on first-row pinions S/N 55 and 56 used in the 50-hour tiedown test.⁽⁹⁾

Pinion S/N 33 was subsequently used in a test of 21 hours duration and at the restart of testing after the previously-mentioned fracture at 17:15 hours. Examination of pinion S/N 33, after 26:15 hours, no longer showed the frosted zone or the degree of peeling which was evident after the 200-hour bench test (Figure C-8).

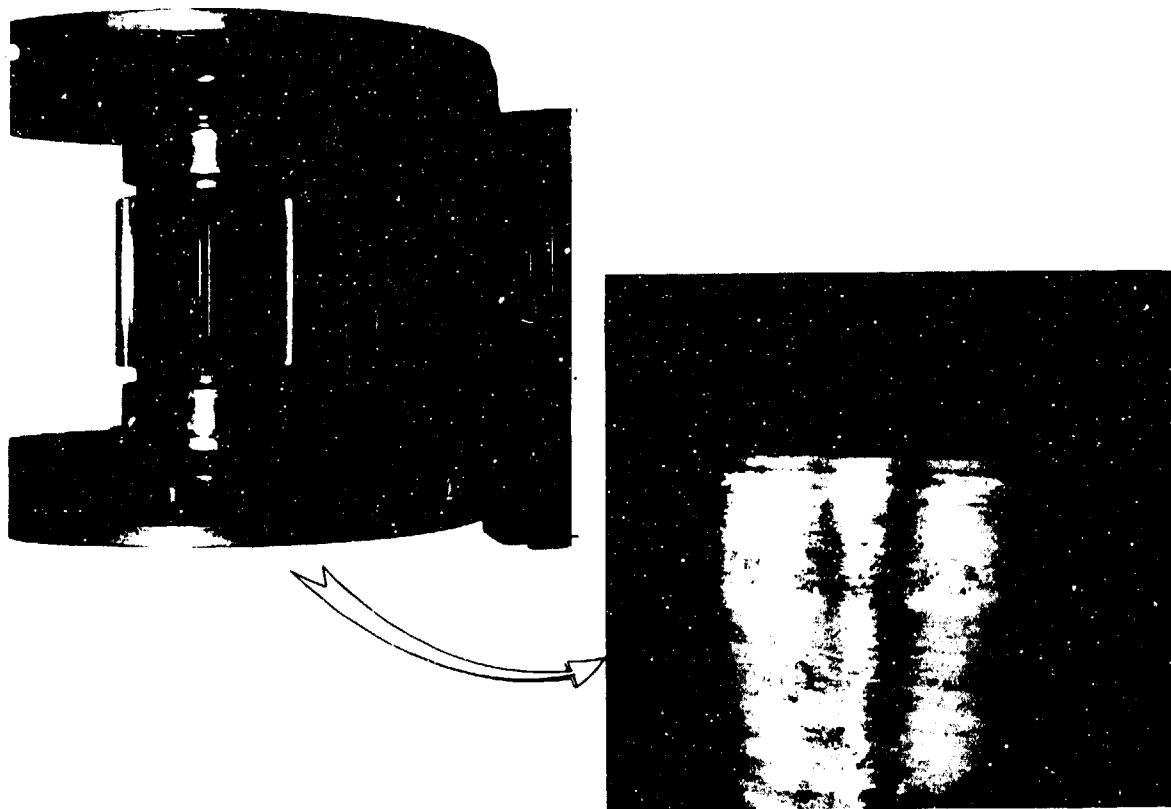


Figure C-7. Roller Distress, First-Row Pinion.

This type of self-healing phenomenon was noted by Franklin Institute Research Laboratories in Derivation of a Fatigue Life, Model for Gears, (11) wherein it was observed that shallow spalling of rolling contacting elements did not propagate deeper. Nonpropagation of surface spalling has also been noted by Onions & Archard, (12) who, in investigating the relationship between pitting of discs, suggested that the propagation of pre-pitting cracks that form pits can be inhibited by wear or plastic flow in the surface layers. It was concluded that, "a form of compliancy may be responsible for the fact that the cracks at the bottom of the shallow spall did not propagate under the Hertzian stresses or from lubricant-induced hydraulic pressure propagation." It is, therefore, concluded that the surface damage of the top roller surface of the first-row pinion, S/N 39, would not be detrimental.

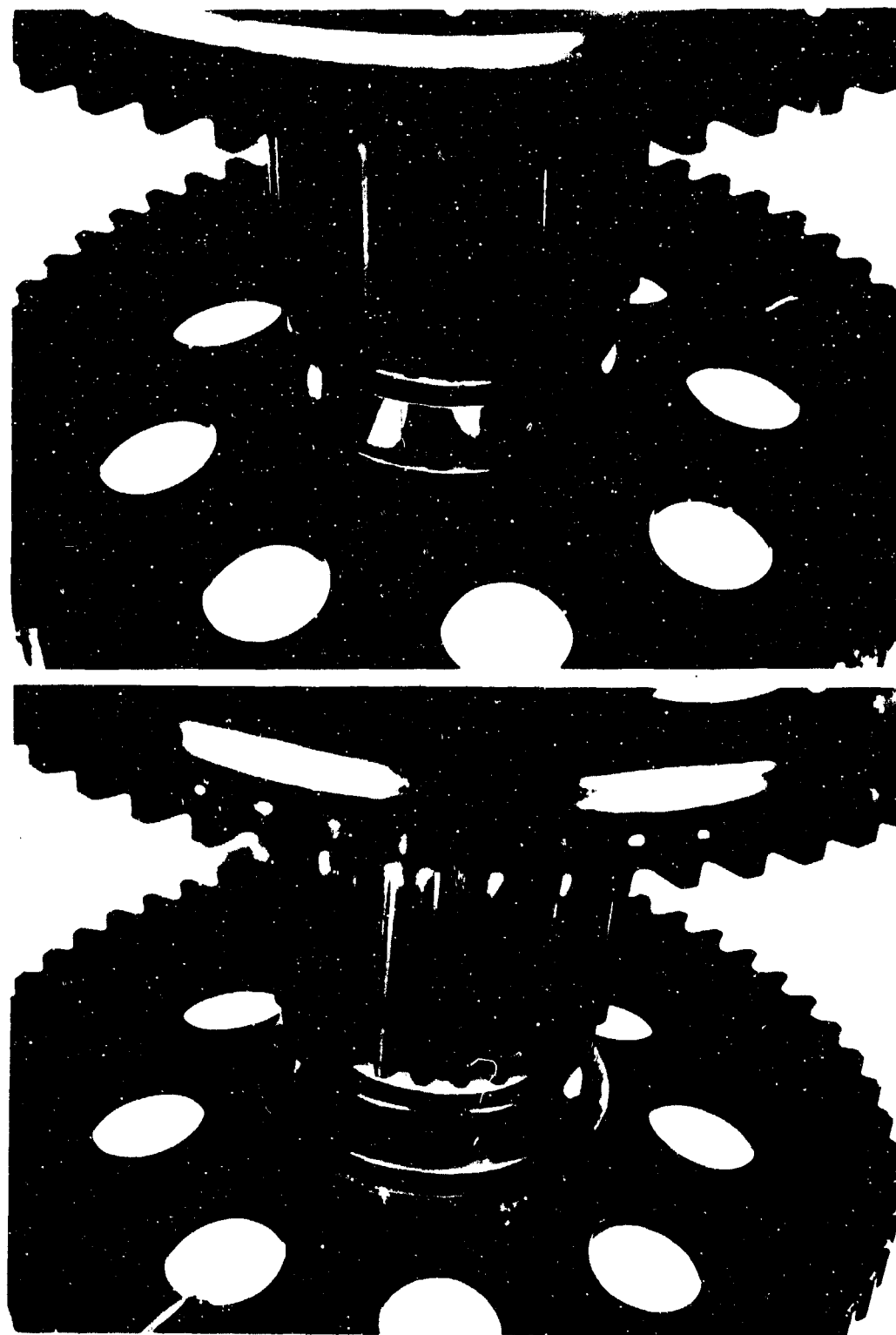


Figure C-8. Roller Healing, First-Row Pinion.

Efficiency Test

One advantage of the back-to-back R&M test gearbox was that the input power required was that which was necessary to overcome losses in the roller gear unit plus the losses in the support bearings. Thus, by measuring the input power to the gearbox and subtracting the losses in the ball, roller, and duplex support bearings, the power loss in the roller gear units was obtained.

The input torque shaft to the gearbox was strain gaged and calibrated. The roller gear units were operated with varying degrees of internal torque locked into the system and the power required to overcome the internal losses obtained. The results are tabulated in Table C-3.

TABLE C-3. FRICTION POWER LOSSES.	
Horsepower	Friction Torque (in.-lb)
750	380
1500	570
2450	780
3000	930

The bearing friction torque was calculated by the method presented by Harris.⁽¹³⁾ Total calculated bearing viscous torque was 20 in.-lb.

A plot of friction torque for the roller gear unit versus power is given in Figure C-9. This shows the efficiency of the roller gear unit to be 99% when transmitting 3000 hp. This figure compares favorably with the calculated efficiency of 98.9% given in the Design Report.⁽⁶⁾ Figure C-9 also depicts the actual efficiency of the roller gear transmission shown in Figure 7. These efficiency values were obtained by measuring the heat rejected by the transmission during the 200-hour endurance test of the Development Test Program.

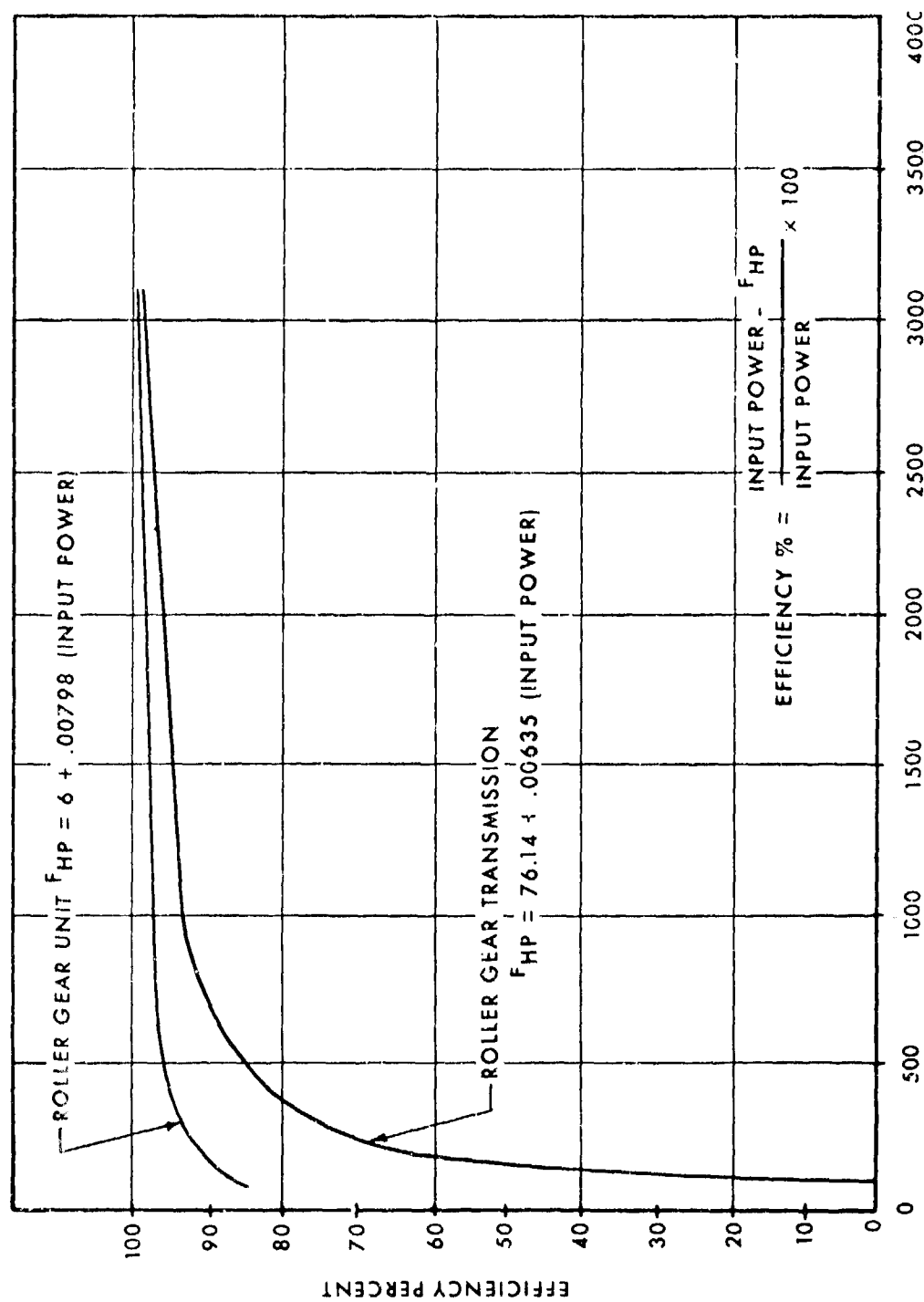


Figure C-9. Efficiency Chart, Roller Gear Unit and Transmission.

Second-Row Post Loads Survey

In planetary geared systems with more than three planets, the total load is not usually shared equally among the gears. In a three-planet system, it can be theoretically shown that if one member is allowed to float, the load will be divided equally among the three pinions. By ensuring that the load is shared equally, the reliability of the unit is considered to be improved, since the gears can be designed to known stress values.

The roller gear unit is comparable to the three-planet system in its capability to share the load among the planet pinions. It has been demonstrated during previous tests⁽⁸⁾ that within the accuracy of the instrumentation, the variation in load at the pinion posts is $\pm 5\%$ of transmitted power. This is achieved by a floating sun gear, a flexible ring gear and free-floating pinions in the first row of the two-row roller gear unit.

During operation of the reliability and maintainability test, the load sharing at the second-row pinions was again demonstrated. The second-row pinion posts in the upper roller gear unit were strain gauged and calibrated in the radial and tangential directions. The strains in the tangential and radial directions were measured and the loads obtained at varying powers.

The load distribution among the seven pinion posts showed a deviation from the mean of 8 percent in the tangential direction (Figure C-10). The loads in the radial direction were lost due to their very low magnitude and cross-torque interference in the strain bridges. Theoretically, there should be no radial load on the spherical bearing posts. The radial clearance in the spherical bearing prevents the bearing from reacting any loads in the radial direction.

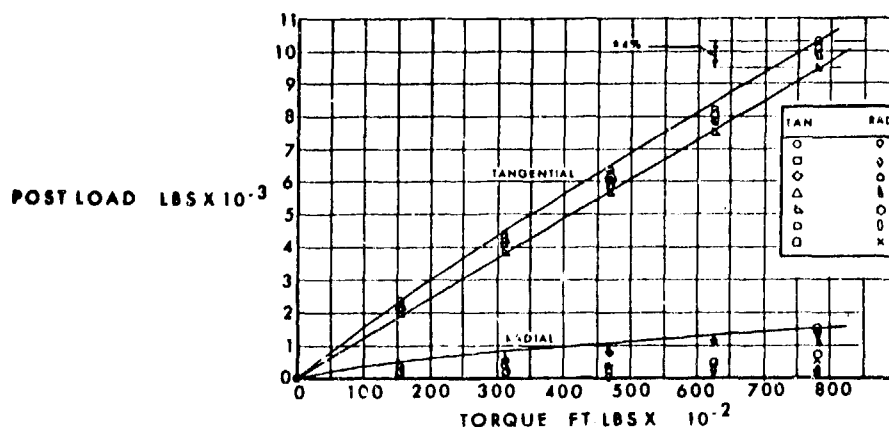


Figure C-10. Load vs Transmitted Power, Second-Row Pinion.

Temperature Survey

Throughout the endurance testing of the roller gear units, the temperature of the spherical roller bearings in the second-row pinions was continuously monitored as a means of detecting imminent failures. Similarly, the R&M gearbox bearing temperatures were monitored by thermocouples.

The flow of oil to each roller gear unit was also monitored, as was the inlet and outlet oil temperatures. The inlet oil temperature could be adjusted by varying the amount of cooling water allowed to pass through an oil/water heat exchanger.

Figure C-11 is the log sheet used to collect and record the temperatures. The values are typical of those witnessed throughout the test. The temperatures were initially displayed on a Brown recorder, which converted the thermocouple EMF onto a strip chart. Temperature data was collected on the log sheet every 15 minutes. A second log sheet, Figure C-12, was used to record oil flows and pressures from dial gauges.

A temperature differential of 12° to 18°F was noticeable between the spherical bearings of the lower and upper roller gear units. This was attributed to the method of lubricating the lower spherical bearings wherein incoming oil is transferred through bolts into the cavity of the second-row pinion post. Oil filled the cavity until it reached the feed hole close to the top of the post where it then bled into the bearing. Consequently, the temperature of the post (to which the thermocouple is bolted) is kept low by the constant volume of oil continuously being replenished by cool incoming oil.

The oil supply to the upper spherical bearings is metered from the orifice in the circular manifold which surrounds all seven spherical bearings. Oil then drains directly into the spherical bearings through oil passages in the post which do not allow a head of oil to build up.

The differing lubrication systems also account for the differing pressures recorded at the manifolds prior to entry into the separate lubrication systems.

Inspections of the roller gear components showed no signs of distress due to lack of lubrication except for the input splines in both the upper and lower sun gears. At the 17:15 and 22:30 hour stoppages and at the final inspection it was noticed that fretting was occurring between the quill shaft and the sun gear splines. No remedial action was taken, as the fretting was not considered detrimental to the operation of the roller gear units.

ROLLER GEAR RELIABILITY TEST DATA SHEET 1

TEMPERATURES

TIME OF DAY	TEST TIME	REGEN SHP	FRICTION POWER	ROLLER POST TEMPERATURES												TEMPERATURES							
				U-1	U-2	U-3	U-4	U-5	U-6	U-7	L-1	L-2	L-3	L-4	L-5	L-6	L-7	OIL IN	OIL OUT	TEMP. IN	TEMP. OUT	TEMP. IN	TEMP. OUT
0715	261:00	3000	67	58	62	61	60	59	60	60	46	45	45	45	45	45	38	65	57	78	62	37	
0730	:15	3000	67	58	63	61	60	60	60	60	46	45	46	46	45	45	39	66	58	78	62	39	
0745	:30	3000	67	58	63	61	60	60	60	60	46	45	46	45	45	45	39	66	58	79	62	39	
0800	:45	3000	67	59	62	61	60	60	60	60	46	45	46	45	45	45	40	65	58	78	62	40	
0815	262:00	3000	67	59	62	61	60	60	60	60	46	45	45	45	45	45	40	65	58	78	62	40	
0830	:15	3000	67	59	63	61	60	59	60	60	46	46	46	46	46	45	39	65	58	79	63	40	
0845	:30	3000	67	58	62	61	60	59	60	60	46	46	46	46	45	45	39	66	58	79	62	40	
0900	:45	3000	67	58	62	61	60	59	60	60	46	45	45	45	45	45	39	65	58	78	62	40	
0915	263:00	3000	67	59	62	61	60	60	60	60	46	46	46	46	45	45	39	65	58	79	63	40	
0930	:15	3000	67	59	63	61	60	60	60	60	46	46	46	46	45	45	39	66	58	79	62	40	
0945	:30	3000	67	59	63	61	60	60	60	60	46	46	46	46	45	45	39	66	59	79	62	40	
1000	:45	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	39	66	58	79	62	40	
1015	264:00	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	40	66	59	79	63	41	
1030	:15	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	39	66	58	79	62	41	
1045	:30	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	39	66	58	79	63	41	
1100	:45	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	40	66	59	79	63	41	
				SHUT DOWN 11:00												RESTART 11:35							
1150	265:00	3000	67	58	61	60	59	58	59	59	45	45	45	45	45	44	39	64	57	78	61	40	
1205	:15	3000	67	58	62	61	59	59	60	60	45	45	45	45	45	45	39	65	58	78	63	40	
1220	:30	3000	67	59	62	61	60	60	60	60	46	46	46	46	45	45	39	66	58	79	62	41	
1235	:45	3000	67	58	62	61	60	60	60	60	46	46	46	46	45	45	40	65	58	79	62	41	
1250	266:00	3000	67	59	62	61	60	60	60	60	46	46	46	46	45	45	40	65	58	79	62	41	
1305	:15	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	46	40	66	59	79	63	42	
1320	:30	3000	67	59	62	61	60	60	60	60	46	46	46	46	46	45	40	65	59	79	62	42	
1335	:45	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	40	66	59	79	62	42	
1350	267:00	3000	67	59	63	61	60	60	60	60	46	46	46	46	46	45	39	66	58	79	62	42	

Figure C-11. Temperature Log Sheet.

ROLLER GEAR RELIABILITY TEST DATA SHEET 2

PRESSURES, FLOWS, MISC.

TIME OF DAY	TEST TIME	PRESSURE - PSIG			FLOW - GPM			REMARKS
		PGM DIS	MANI 1	MANI 2	UPPER RGD	LOWER RGD	MISC.	
0715	261:00	116	34	20	7.2	7.2		
0730	:15	116	34	20	7.3	7.3		
0745	:30	116	34	20	7.3	7.3		
0800	:45	116	34	20	7.3	7.3		
0815	262:00	116	34	20	7.3	7.3		
0830	:15	116	34	20	7.3	7.3		
0845	:30	116	34	20	7.3	7.3		
0900	:45	116	34	20	7.3	7.3		
0915	263:00	116	34	20	7.3	7.3		
0930	:15	116	34	20	7.3	7.3		
0945	:30	116	34	20	7.3	7.3		
1000	:45	116	34	20	7.3	7.3		
1015	264:00	116	34	20	7.3	7.3		
1030	:15	116	34	20	7.3	7.3		
1045	:30	116	34	20	7.3	7.3		
1100	:45	116	34	20	7.3	7.3		
		SHUTDOWN 11:00						
1150	265:00	116	34	20	7.2	7.2		RESTART 11:35
1205	:15	116	34	20	7.3	7.3		
1220	:30	116	34	20	7.3	7.3		
1235	:45	116	34	20	7.3	7.3		
1250	266:00	116	34	20	7.3	7.3		
1305	:15	116	34	20	7.3	7.3		
1320	:30	116	34	20	7.3	7.3		
1335	:45	116	34	20	7.3	7.3		
1350	267:00	116	34	20	7.3	7.3		

Figure C-12. Oil Flow and Pressure Log Sheet.

As there were no heat-related or oil starvation problems associated with the operation of the R&M test, temperature recordings proved to be purely academic.

Vibration Survey

In any high reduction gearbox, the elements of the power train producing the dynamic forces generate noise. The noise can be troublesome and yet, to a trained operator, a means of detecting impending malfunction.

The noise is a product of the gear clash frequency and its harmonics, the rolling element bearings and the attenuation of such frequencies through the gearbox housing structure. Monitoring the gearbox vibrations is often used as a means of detecting impending malfunction.

Vibration sweeps of the frequencies generated by the two roller gear units in the R&M gearbox were taken at 10 and 260 test hours. The objectives of these measurements were: 1) to determine the source of an objectionably high-pitched whine, 2) to monitor any surface degeneration or crack propagation, and 3) to determine the effect of operating time and load on the levels of vibration.

An accelerometer output was fed into a frequency analyzer that swept the frequency range covering residual harmonics of the roller gears meshing frequencies. The output of the frequency analyzer was simultaneously played into a level recorder which, in turn, plotted the amplitude of the accelerometer signal as a function of frequency to give a spectral analysis of the gear's vibrations.

The primary rotational frequency associated with geared systems is gear clash frequency. This is a function of the number of teeth and the rotational speed and corresponds to the motion induced by the consecutive meshing of the teeth. Gear clash frequency is given by

$$F = \frac{n}{60} \cdot N, \text{ Hz}$$

where

n = shaft speed, rpm
N = number of teeth

TABLE C-4. ROLLER GEAR MESH FREQUENCIES.				
Component	RPM n	Shaft Speed (RPS)	No. Teeth N	Tooth Mesh Frequency (f)
Sun Gear	4045	67.4	84)	5663
			)	
First-Row Pinion	5858	97.6	(58)	2636
			(27)	
			)	
Second-Row Pinion	1255	21	(126)	
			(25)	523
			)	
Ring Gear	204	3.4	154)	

In addition to the meshing frequency, mesh frequency harmonics and sidebands occur. Predominant among planetary systems are sidebands at the planet-pass frequency occurring when the planets pass a stationary point. Also, sidebands are produced from dynamic tooth force modulations from an eccentric gear. By periodically forcing the teeth into mesh, a cyclic loading pattern occurs, with a minimum and maximum mesh force occurring once per shaft revolution. As the eccentricity increases, sideband amplitudes increase. Tooth stiffness variations can also alter mesh properties to produce sidebands at shaft rotational speeds as can the distressed surface of gear teeth.

An accelerometer was mounted on a second-row pinion retaining bolt, which protrudes through the R&M gearbox (Figure C-13). The horizontal vibration sweeps of Figures C-14 and C-15 were obtained at 10 and 260 test hours, while operating the R&M gearbox at 3000 hp, 100% speed.

Predominant among the vibration frequencies recorded is the 2636 Hz mesh frequency of the first-row/second-row gears and its harmonics. Superimposed on this frequency is the fifth harmonic of the ring gear mesh frequency, i.e., $5 \times 523 = 2615$ Hz.

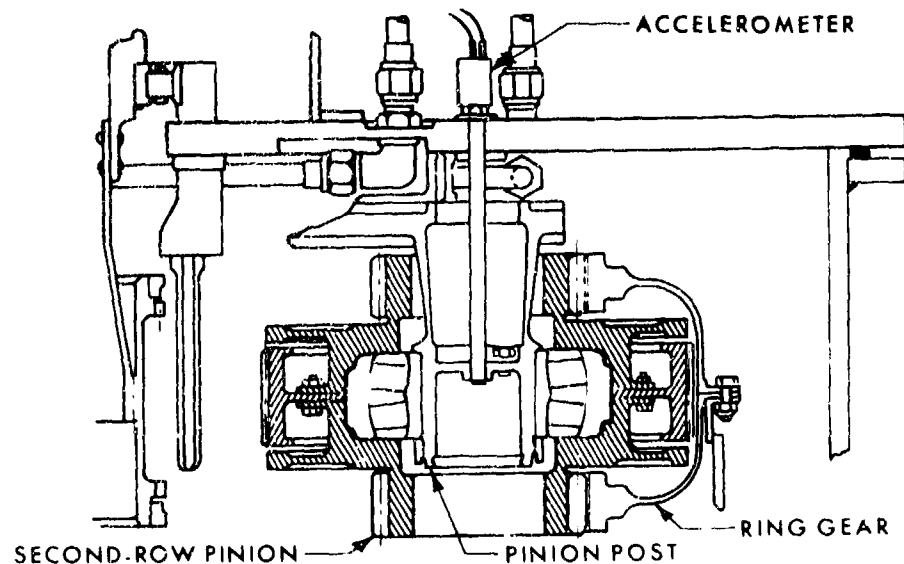


Figure C-13. Accelerometer Location, Second-Row Pinion Bolt.

Comparison of the two frequency sweeps shows a smoothing out of the ring gear harmonics with time. This is normal to expect as the minor discrepant errors incurred during manufacture are worn away and the unit becomes "run in."

Evident in the first recording are ± 130 Hz sidebands at the first, second, third and fourth harmonics of the first-/second-row mesh frequencies. On the 260-hour recording, the sidebands at the third harmonic correspond to the speed of the sun gear, 67 Hz. It would appear that run-out of the sun gear and/or driveshaft is imparting additional dynamic loads on the gear teeth and, consequently, the bearings. The importance of this type of force has yet to be assessed, but, during the running of the R&M test, three quill shafts driving the upper roller gear unit fractured. Run-out of the sun gear could have been a contributory cause.

The sideband amplitudes are of a relatively low magnitude when compared to the gear clash frequency values. Sidebands generally indicate a discontinuity in the normally smooth sequential operation of gear teeth meshing. Thus, when the recordings were obtained, the roller gear units were operating satisfactorily.

The 50 g's peak amplitude at 2636 Hz remained constant throughout the test program. This acceleration corresponds to a displacement of the pinion of ± 0.00003 inch occurring at a velocity of .42 inch/second. These very low values attest to the smoothness of the roller gear unit.

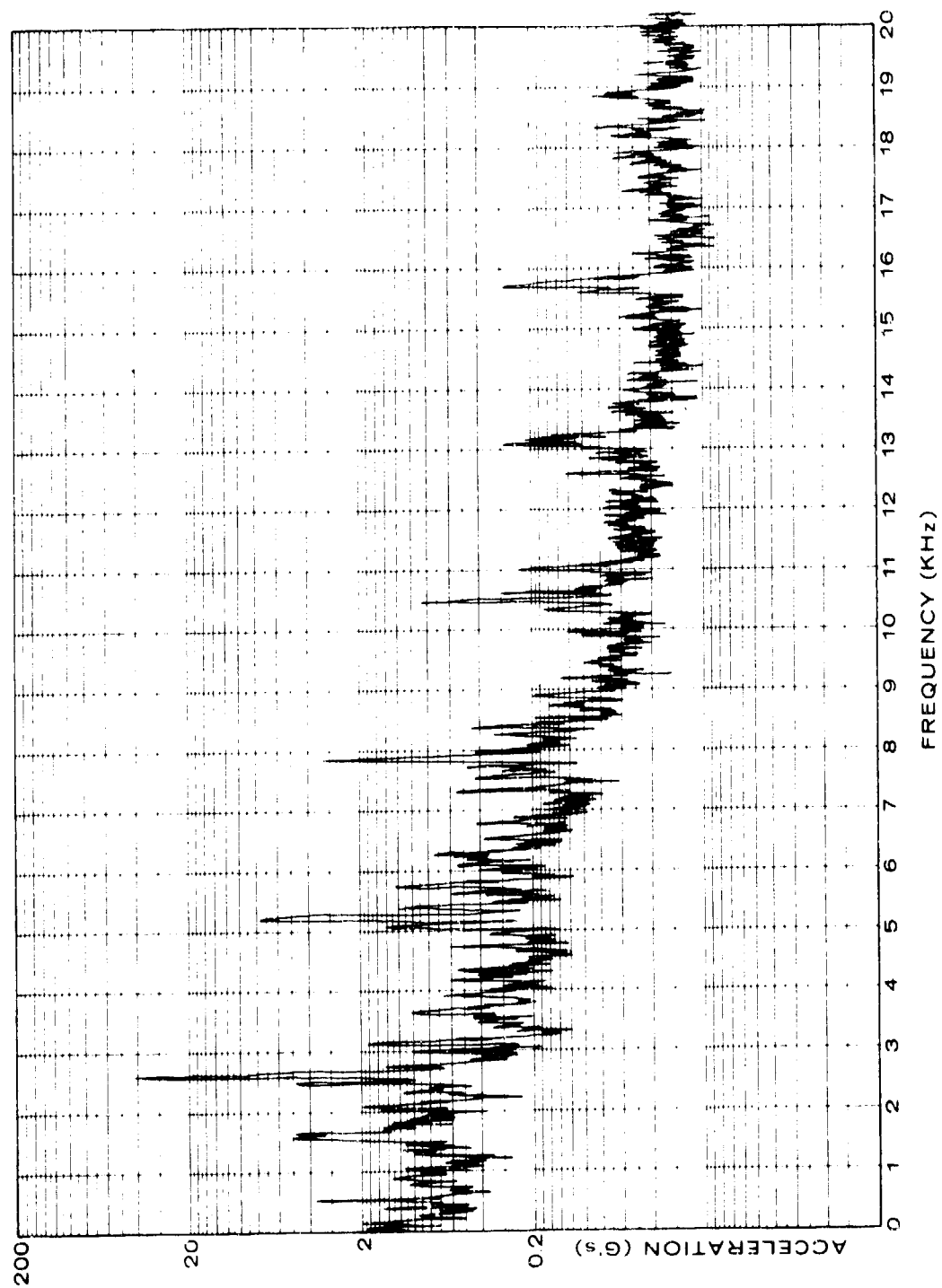


Figure C-14. Vibration Sweep, 10 Test Hours.

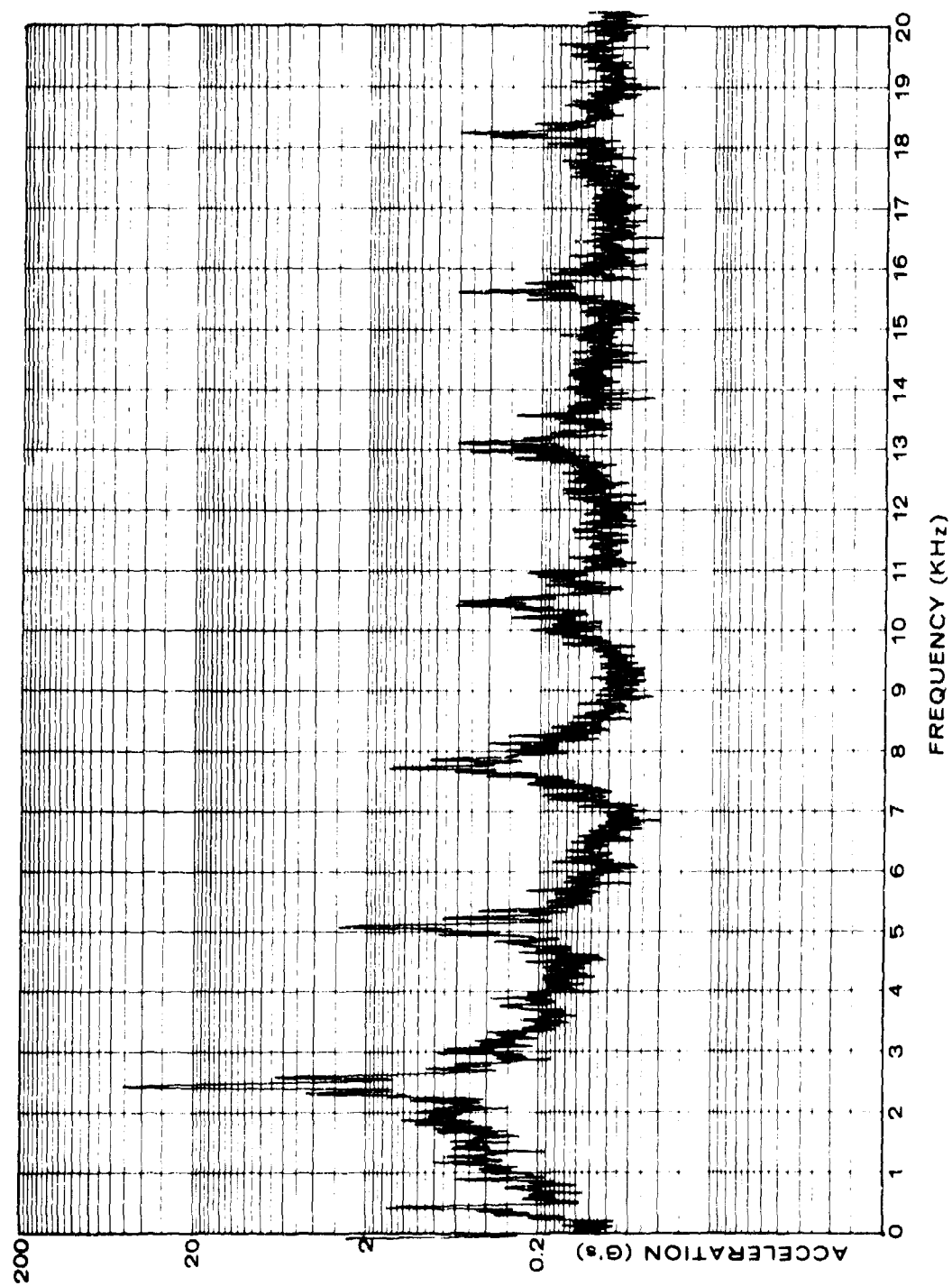


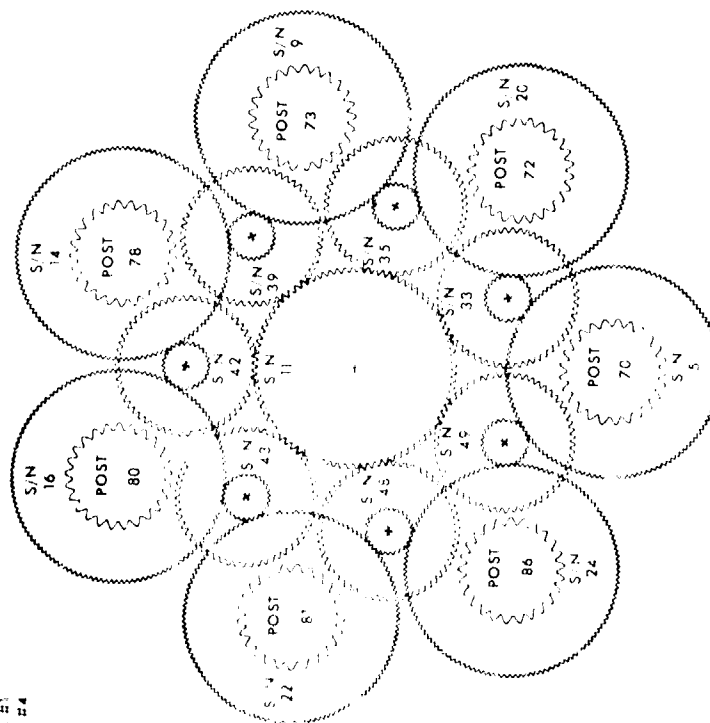
Figure C-15. Vibration Sweep, 260 Test Hours.

Post-Test Inspection

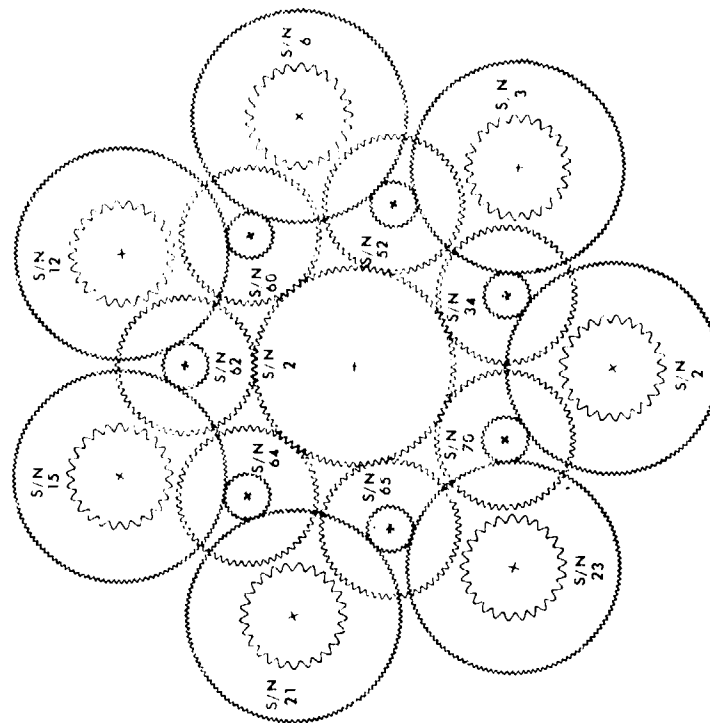
At the completion of 295:50 hours of testing of the roller gear units, an inspection of the individual components was conducted. A metallurgical examination of the upper quill shaft, which fractured at this time, was also conducted.

The components used and their arrangement in the upper and lower roller gear units is shown in Figures C-16 and C-17. The total test times and previous usage of the sun, first- and second-row pinions and ring gears are depicted in the charts of Figures C-18, C-19, C-20 and C-21, respectively. These show the complete program usage of the components, the particular dynamic tests they were subjected to and the primary cause of test stoppage and malfunction at the test hours accumulated. As can be seen from Figure C-19, the first-row pinions, set #6, utilized in the upper roller gear unit were previously used for the 21-hour R&M test and the 200-hour back-to-back regenerative bench test of the roller gear transmissions⁽⁸⁾. The total accumulated times on the components disassembled at the termination of testing are summarized in Table C-5.

43 #5
42 #7
39 #3
48 #6
35 #2
33 #1
42 #4



RING GEAR S/N 6
UPPER UNIT
R & M TEST 295.50 HOURS



LOWER UNIT
R & M TEST 295.50 HOURS

Figure C-16. Component Location, Test Unit.

Figure C-17. Component Location, Dummy Unit.

TABLE C-5. ACCUMULATED TEST TIME, R&M COMPONENTS.

COMPONENT	IDENT NO.	TEST	TEST TIME (hrs:min)
<u>Upper Unit</u>			
Sun Gear	S/N 14	Reliability & Maintainability	295:50
First-Row Pinions	Set #06	Bench Test (200 Hours) R&M Test #1 (21 Hours) Reliability & Maintainability (278:35 hours)	499:35
Second-Row Pinions	Set #2	Reliability & Maintainability (5 pieces) (S/N 20 & S/N 22)	295:50 278:35
Ring Gear	S/N 06	Aircraft Tiedown Test (50 Hours) Failure Mode Test (57:45 Hours) Reliability & Maintainability (295:50 Hours)	403:35
<u>Lower Unit</u>			
Sun Gear	S/N 02	R&M Test #1 (21 Hours) Reliability & Maintainability (295:50 Hours)	316:50
First-Row Pinions	Set #9	Reliability & Maintainability	295:50
Second-Row Pinions	Set #1	Reliability & Maintainability	295:50
Ring Gear	S/N 08	R&M Test #1 (21 Hours) Failure Mode Test (57:45 Hours) Reliability & Maintainability (295:50 Hours)	403:35

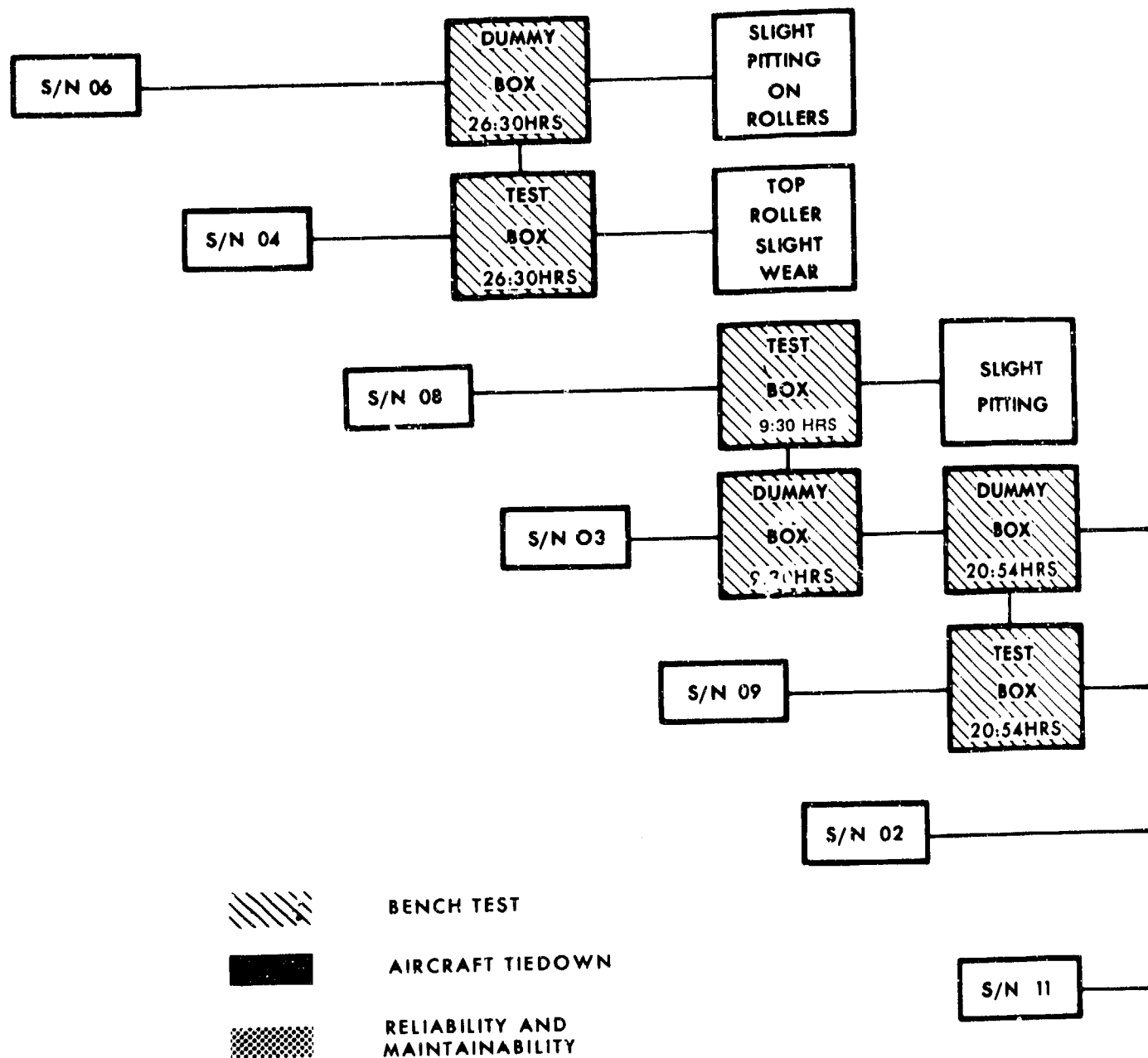
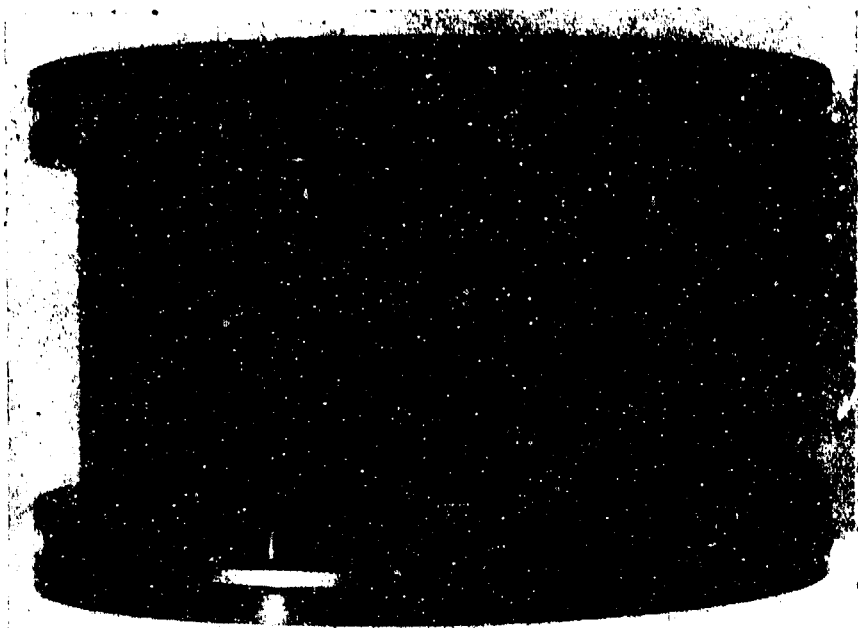
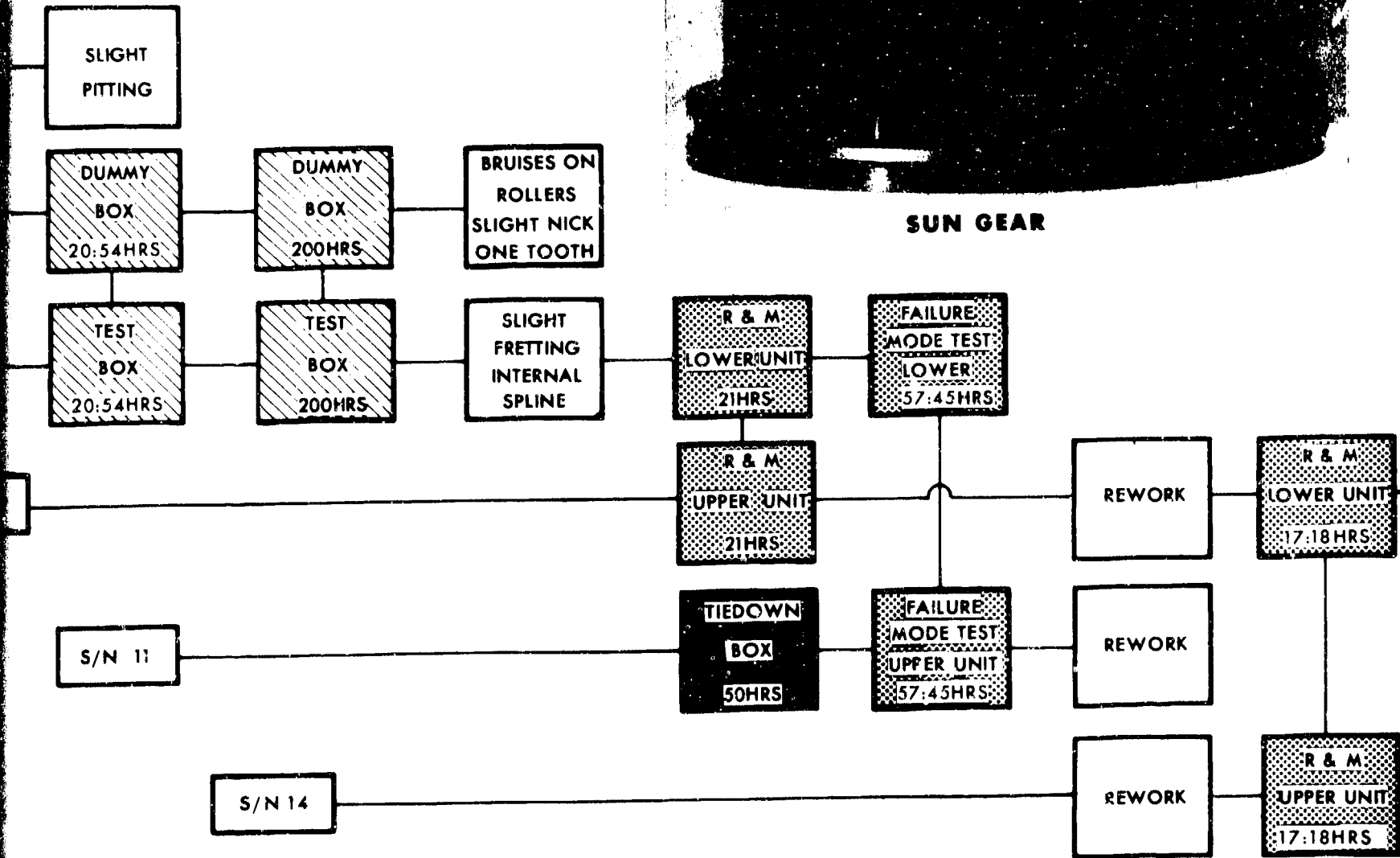


Figure C-18. Sun Gear Usage.

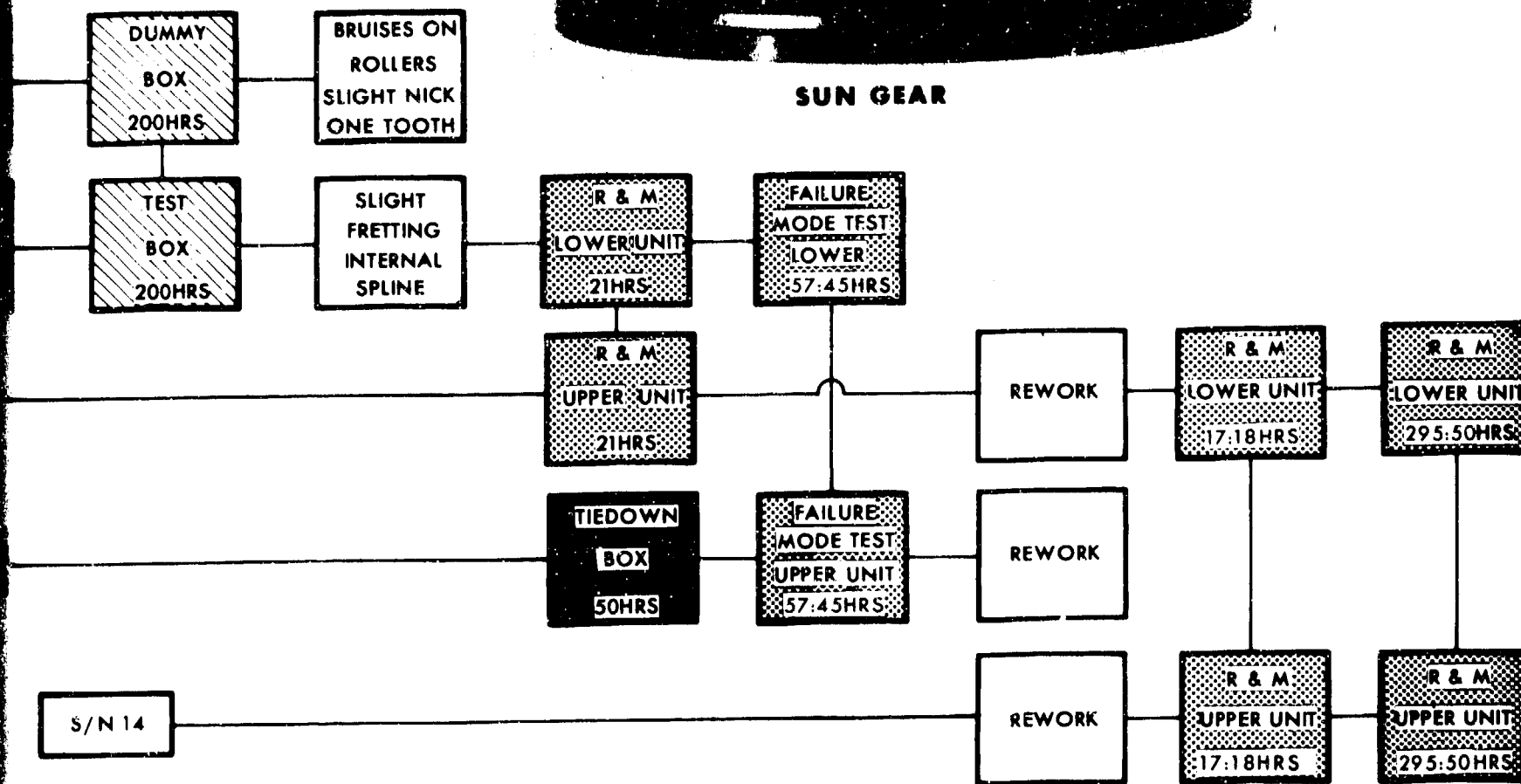


SUN GEAR





SUN GEAR



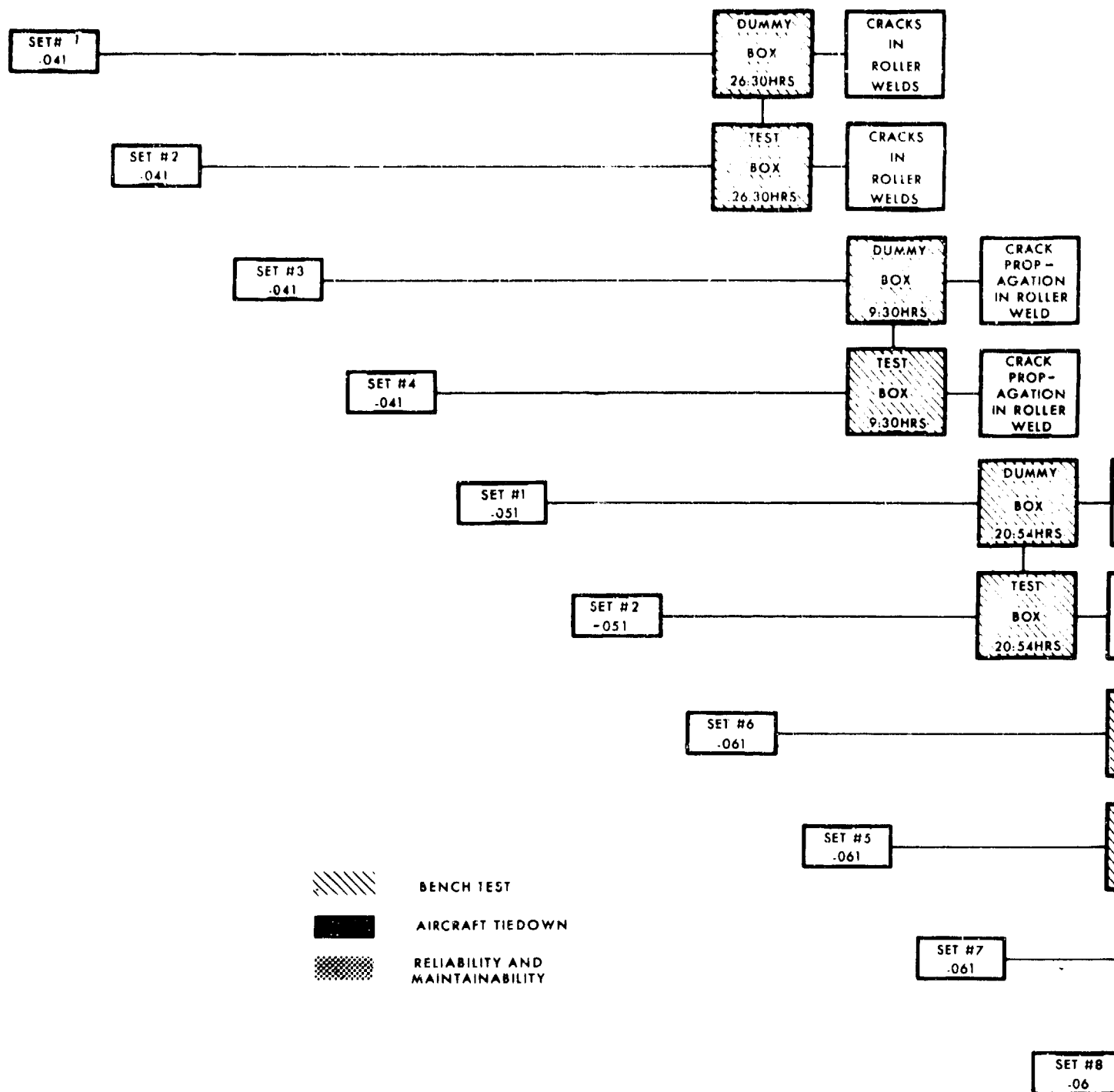


Figure C-19. First-Row Pinion Usage.

CRACKS
IN
ROLLER
WELDS

CRACKS
IN
ROLLER
WELDS

DUMMY
BOX
9.30HRS

CRACK
PROP-
AGATION
IN ROLLER
WELD

TEST
BOX
9.30HRS

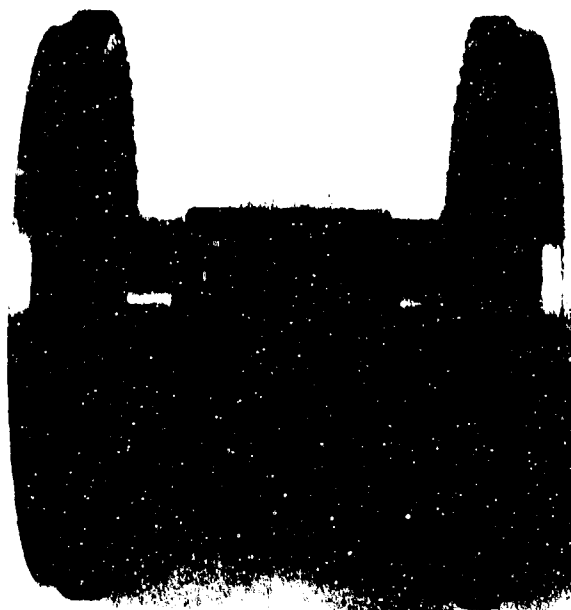
CRACK
PROP-
AGATION
IN ROLLER
WELD

DUMMY
BOX
20.54HRS

TEETH
FRACTURE
DUE
TO WELD

TEST
BOX
20.54HRS

TEETH
CRACKED
DUE
TO WELD



1st - ROW PINION

TEST
BOX
200HRS

R & M
UPPER UNIT
21HRS

REWORK

R & M
UPPER UNIT
295.50 HRS

SET #5
.061

DUMMY
BOX
200HRS

R & M
LOWER UNIT
21HRS

FAILURE
MODE TEST
LOWER UNIT
57.45HRS

SET #7
.061

TIEDOWN
BOX
50HRS

FAILURE
MODE TEST
UPPER UNIT
57.45HRS

DEBRIS
DAMAGE

SET #8
.06

REWORK

R & M
UPPER UNIT
17.18HRS

S/N72
TOOTH
FRACTURE

SET #9
.061

REWORK

R & M
LOWER UNIT
17.18HRS

R & M
LOWER UNIT
295.50 HRS

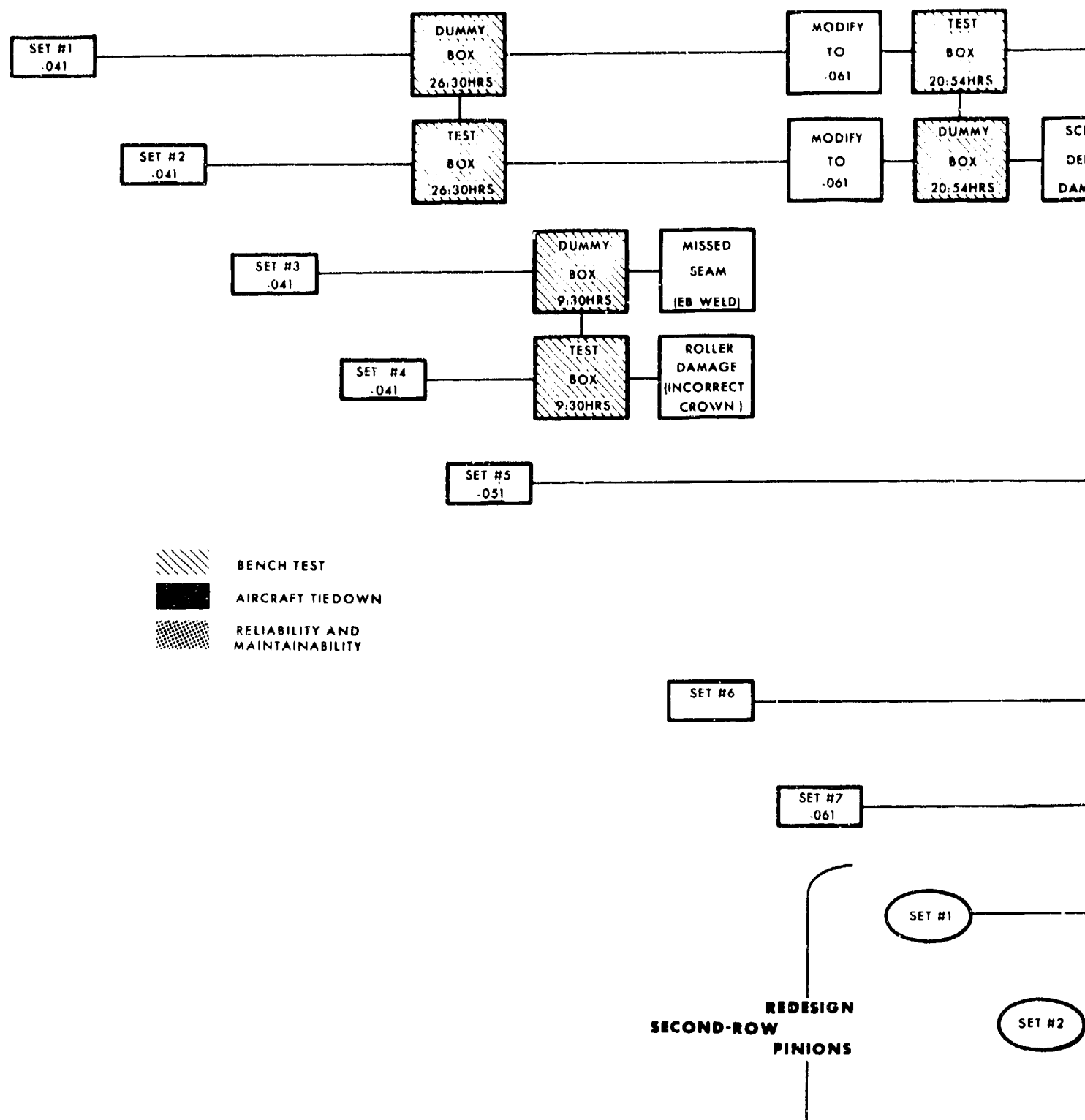
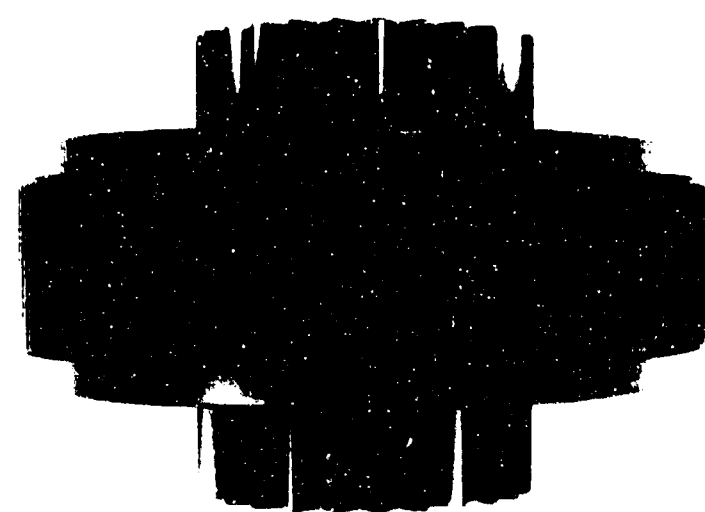
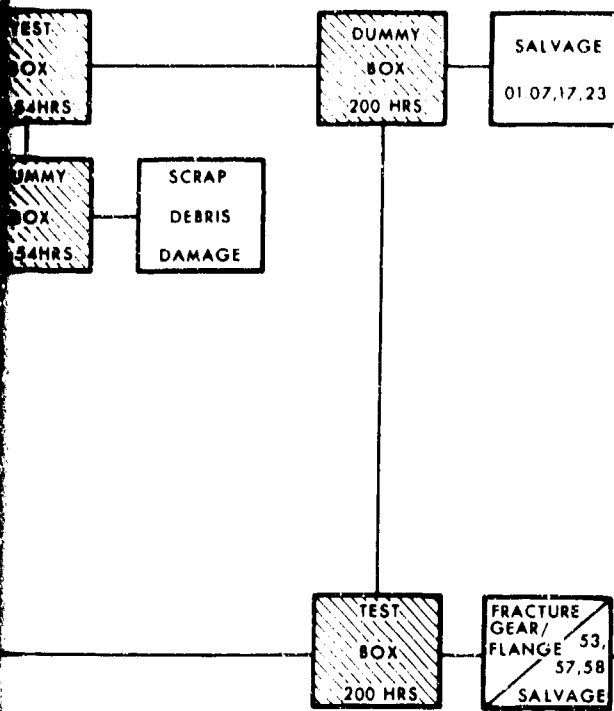
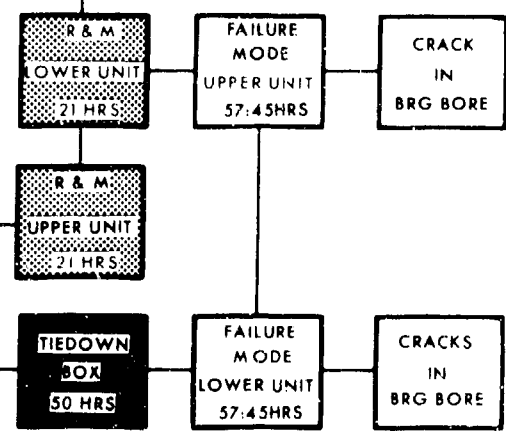


Figure C-20. Second-Row Pinion Usage.

CRAP
BRIS
AAGE

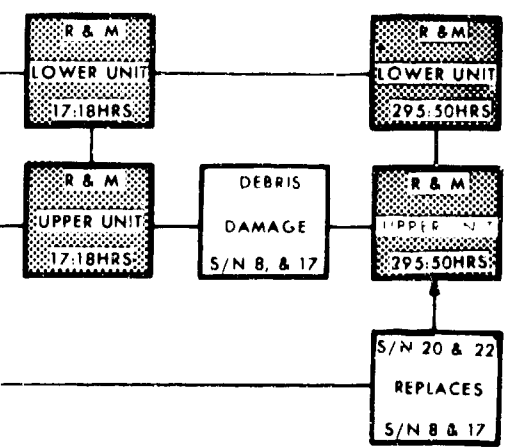


SECOND-ROW PINIONS



SET #2

SET #3



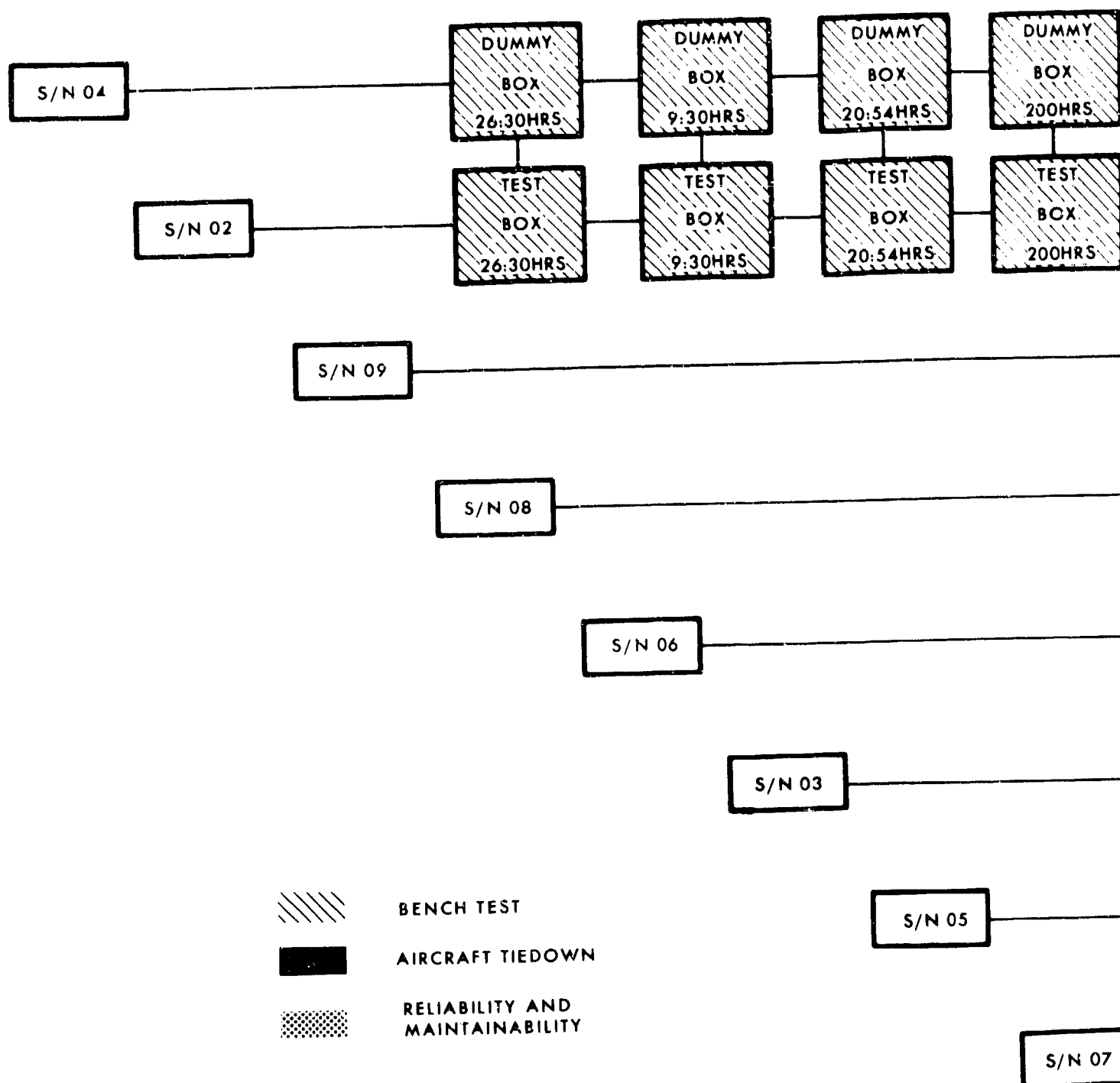
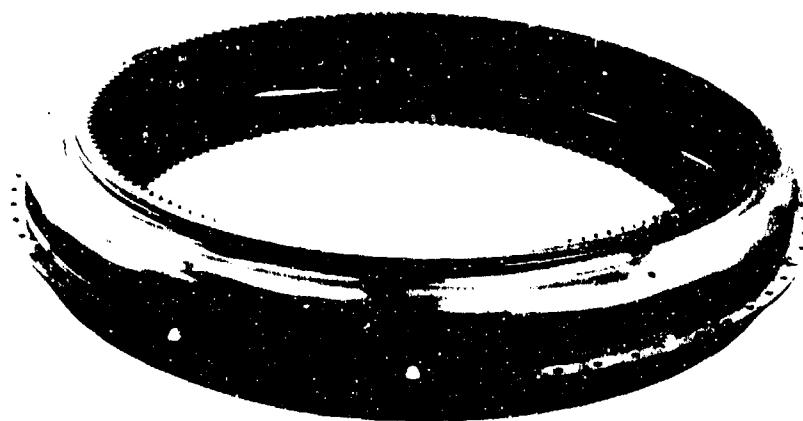
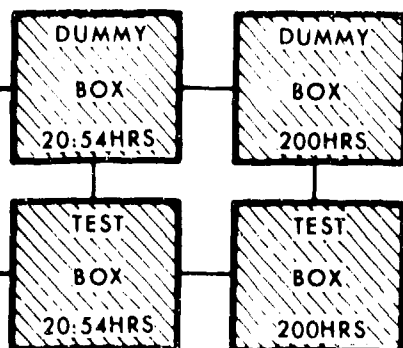
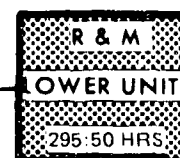
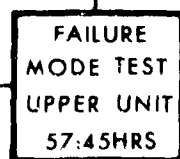
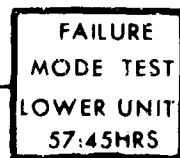


Figure C-21. Ring Gear Usage.



RING GEAR



03

S/N 05

S/N 07

NOT USED

TABLE C-6. UPPER ROLLER GEAR UNIT, FIRST-ROW PINIONS:
VISUAL INSPECTION 295:50 HOURS

S/N	TOP		MIDDLE		BOTTOM	
	ROLLER	GEAR	ROLLER	GEAR	ROLLER	GEAR
33	Excellent	Excellent	Intermittent spalling in corner. Journal excellent.	Frosted gear face, very slight pitting in dedendum.	Intermittent spalling in corner.	Excellent
35	Excellent	Excellent	Slight spalling in corner. Two very small spall areas.	Frosted gear face. Slight pitting in dedendum.	Few circular braille marks, otherwise excellent surface.	Excellent
39	Excellent	Excellent	Intermittent spalling in corner. Journal shows evidence of mating roller crown relief.	Frosted gear face. Very slight pitting in dedendum.	Intermittent spalling in corner. Slight frosting on edge. Journal excellent.	Isolated spalled area.
42	Excellent	Excellent	Spalled corner area. Random spalling in isolated areas.	Frosted gear face either side of pitch line	Intermittent spalling in corner.	Small isolated spalled area.
43	Excellent	Excellent	Uniform spalled corner. Isolated spalled areas on journal.	Frosted gear face. Slight pitting in dedendum.	Intermittent spalling in corner	Slight denting damage. Very slight spalled area.
48	Excellent	Excellent	Spalled corner. Journal excellent.	Severe spalled area in center of gear face. Pitting line along dedendum.	Very slight spalling in corner. Journal excellent.	Excellent
49	Intermittent spalled out areas. Minor scratches with spalling around edges.	One small pitted area, otherwise excellent.	Spalled corner. Spalled out areas on journal.	General frosted surface. Two large pitted areas on edge of gear.	Intermittent spalling in corner and on journal. Shoulder scuffed.	Random mechanical damage. aged areas.

TABLE C-7. UPPER ROLLER GEAR UNIT, SECOND-ROW PINIONS:
VISUAL INSPECTION 295.50 HOURS

S/N	GEAR			
	TOP	ROLLER	MIDDLE GEAR	LOWER GEAR
5	Slight pitting on high spots.	Journal excellent with slight roughness on edge.	Slight frosting on teeth.	Excellent Slight frosting in dedendum.
20	Debris damage on top of teeth; very slight pitting.	Excellent	Slightly frosted surface with pitting initiating in dedendum.	Excellent Slight frosting on teeth.
9	Debris damage to teeth. Slight frosting.	Excellent	Some frosting, slight pitting in dedendum.	Excellent Very slight pitting.
14	Incipient pitting on teeth.	Excellent	Very slight pitting in dedendum.	Journal excellent with slight roughness on edge. Slight frosting in dedendum, extending to pitch line.
16	Incipient pitting.	Nicks on roller edge. Journal excellent.	Very light pitting in dedendum.	Excellent Slight debris damage. Slight frosting on teeth.
22	Slight debris damage.	Excellent	Slight frosting in dedendum.	Excellent Very slight pitting at pitch line.
24	Very slight debris damage.	Excellent	Slight frosting in dedendum.	Excellent Slight frosting in dedendum.

TABLE C-8. LOWER ROLLER GEAR UNIT, FIRST-ROW PINIONS:
VISUAL INSPECTION 295:50 HOURS

S/N	ROLLER	TOP		MIDDLE		BOTTOM	
		GEAR	ROLLER	GEAR	ROLLER	GEAR	ROLLER
34	Excellent	Good wear pattern.	Intermittent pitting in radius area.	One large pit. Incipient pitting in dedendum below pitch line.	Slight pitting adjacent to roller edge.	Good	Excellent
52	Excellent	Excellent	Excellent	Full wear pattern, incipient pitting.	Excellent	Excellent	Excellent
60	Excellent	Excellent	Intermittent spalled areas on roller edge, rolling surface excellent.	Incipient pitting. Burnished face below dedendum.	Excellent	Excellent	Excellent
62	Excellent	Excellent	Excellent rolling surface. Slight spalling on roller edge.	Pitting in dedendum. Slight spalled areas on gear tooth face.	Spalling intermittently in radius area. Rolling surface excellent.	Excellent	Excellent
64	Excellent	Light corrosion on tooth face.	Excellent rolling surface.	Burnished surface. Two spall pits.	Slight corrosion on bearing surface, otherwise smooth.	Slight corrosion. Pitting on surface	Corrosion, pitting on bearing surface good.
65	Excellent	Excellent	Excellent rolling surface.	Burnished areas below pitch line.	Excellent rolling surface. Spalling damage adjacent to roller edge.	Some corrosion damage on teeth.	Excellent
70	Excellent	Burnishing, some corrosion.	Excellent	Burnishing and frosting below pitch line.	Excellent	Slight corrosion damage, light burnishing.	Excellent

**TABLE C-9. LOWER ROLLER GEAR UNIT, SECOND-ROW PINIONS:
VISUAL INSPECTION 295:50 HOURS**

S/N GEAR	TOP		MIDDLE GEAR		LOWER	
	ROLLER		ROLLER		GEAR	
2 Slight debris damage. Excellent wear pattern.	Exce'lent	Minor frosting at pitch line.	Two small pits on excellent rolling surface.	Minor dent on one tooth.		
15 Minor denting. Excellent wear pattern.	Good surface, slight pitted areas initiat- ing from scratches on surface (two places).	Minor frosting at pitch line.	Incipient pitting on roller edge. Slight mechanical damage.	Minor denting and pitting. Uniform wear pattern.		
12 Two small dents. Uniform wear pattern.	Excellent	Frosting at pitch line.	Excellent	Excellent wear pattern.		
6 Slight denting on two teeth.	Excellent	Excellent	Excellent	Uniform wear pattern.		
3 Slight debris damage. Skeved wear pattern.	Excellent	Slight frosting.	Excellent	Good wear pattern.		
21 Very slight debris damage. Good wear pattern.	Excellent	Slight frosting. Uniform wear pattern.	Excellent	Excellent		
23 Numerous small dents. Uniform wear pattern.	Excellent	Slight frosting at pitch line. Uniform wear.	Excellent	Uniform wear pattern.		

TABLE C-10. SUN GEARS: VISUAL INSPECTION 295:50 HOURS.				
S/N	TOP		BOTTOM	
	ROLLER	GEAR	GEAR	ROLLER
11	Slight debris damage. (UPPER UNIT)	Tool marks on gear teeth. Isolated frosted areas.	Slight pitting, emanating from tool marks on gear teeth.	Spalled band around lower edge of journal. Spalling initiating from debris damage areas.
2	Very light pitting. (LOWER UNIT)	Minor pitting at pitch line; frosted surface on teeth.	Incipient pitting.	Frosted surface on teeth.

TABLE C-11. RING GEARS: VISUAL INSPECTION 295:50 HOURS.		
S/N	UPPER GEAR	LOWER GEAR
08 (UPPER UNIT)	Excellent	Excellent
06 (LOWER UNIT)	Excellent	Excellent

Visual and Steroscopic Inspection Results

The primary cause of a stoppage of the reliability and maintainability test program was spalling of the first-row pinion small diameter gear S/N 48 (Figure C-22). The adjacent pinion, S/N 49, also showed spalling on the edge of the small diameter gear. Steroscopic examination of this gear tooth also revealed incipient spalling at the low point of single-tooth contact, i.e., along the dedendum of the gear tooth (Figure C-23). This type of pitting was evident in a number of first-row pinions. The second-row pinion mating gear teeth exhibited very slight pitting on the dedenda of the teeth and a slightly frosted surface (Figure C-24). However, it was on the driving gear tooth surfaces of the first-row pinions, with the short radii of tooth curvature, that the 'v' shaped pits were predominant.

The spall on the tooth of pinion S/N 48, Figure C-22, propagated from a series of small pits which had been joined by failure of metal between them. Ultimately, the relatively large piece of metal spalled from the surface. It has been shown that pitting from a surface origin is distinguished by initial crack propagation along a relatively shallow acute angle and usually has an exit angle more nearly perpendicular to the surface. In the early stages, the pitted area is shaped like an arrowhead, with the apex pointing in the direction of rotation (opposite to the direction of rolling) (Figure C-25). Continued operation of these gears (S/N 48 and 49) would eventually result in the destruction of the tooth flank, which would eventually cause fracture of the gear tooth.

It is not fully understood why the spall occurred on the end of the gear tooth of pinion S/N 49. No evidence of misalignment or end loading is evident in the illustration of Figure C-23. Some degree of chucking has occurred, as is evident from the intermittent spalling on the lower roller and second-row pinion roller outer edge interference (Figure C-26). The extent and degree of intermittent spalling is more pronounced on the pinions from the upper roller gear unit than on those from the lower unit. The former were utilized in the 200-hour bench test⁽⁸⁾ where they mated with "old design" second-row pinions. These pinions did not have the equal torsional stiffness that the redesigned pinions incorporate as a result of symmetry of design and may have resulted in some degree of torsional set in the first-row pinions. This could account for the intermittent spalling evident on the roller and the spall at the end of the gear teeth of pinion S/N 49.

Tables C-6 and C-7 show the visual inspection results of the first-row and second-row pinions from the upper roller gear unit. Tables C-8 and C-9 list similar inspection results from the lower unit, and Tables C-10 and C-11 list the sun and ring gear results, respectively.

Although slight debris damage had been incurred by the roller gear components, as far as could be determined, this did not result in the spalled areas. First-row pinion S/N 33 showed no reoccurrence of the distress which was evident on the roller after the 200-hour bench test.

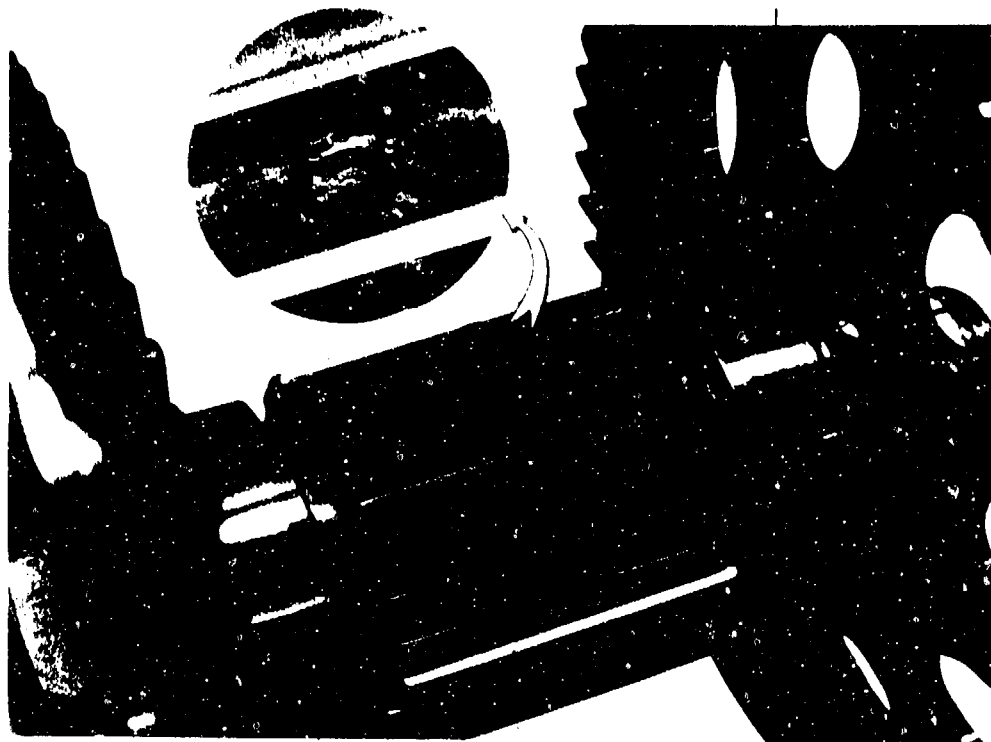


Figure C-22. Spalled Tooth, First-Row Pinion S/N 48.

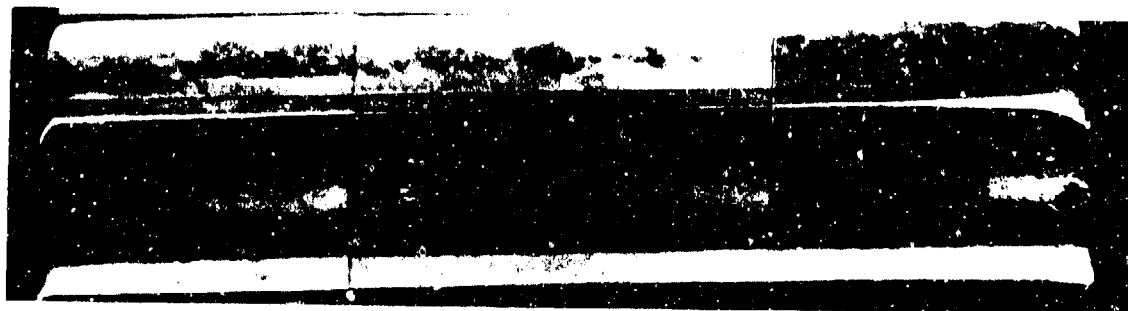


Figure C-23. Spalling and Incipient Spalling,
First-Row Pinion S/N 49.

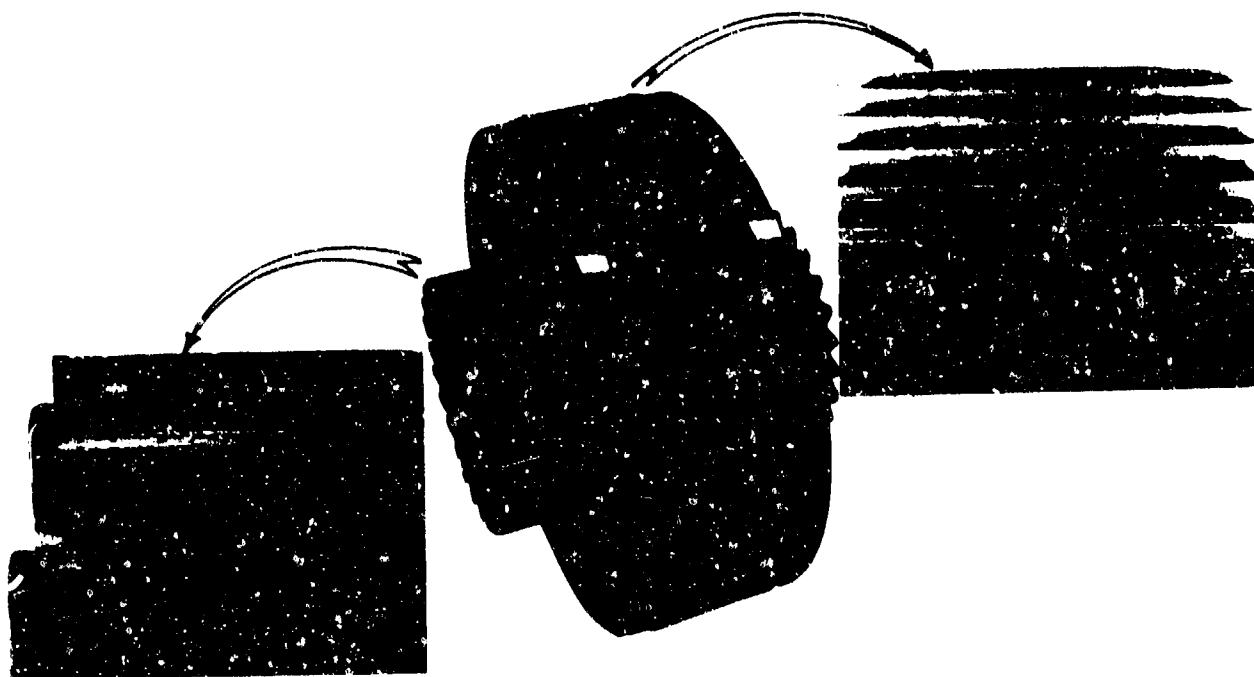


Figure C-24. Second-Row Pinion Teeth.

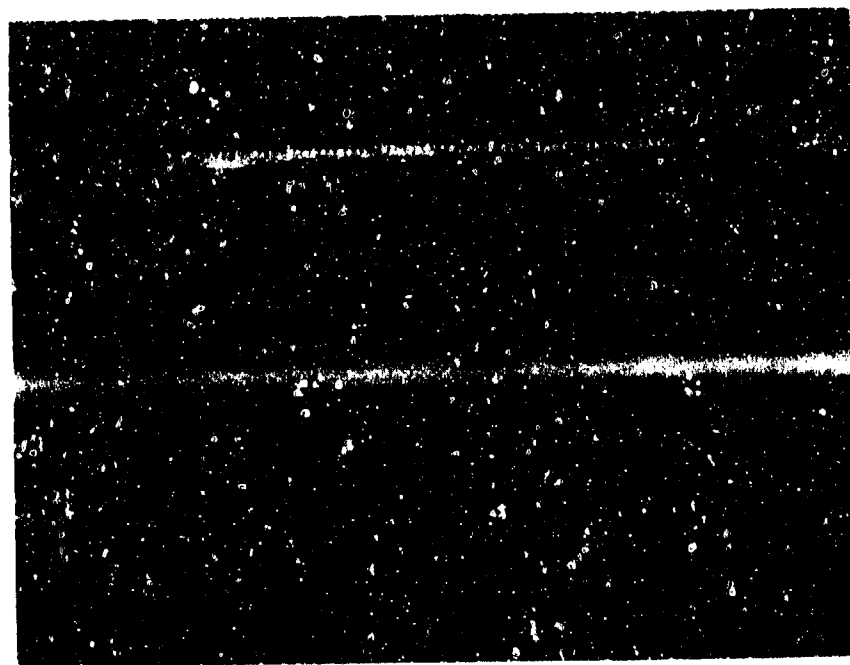


Figure C-25. Surface Pitting.

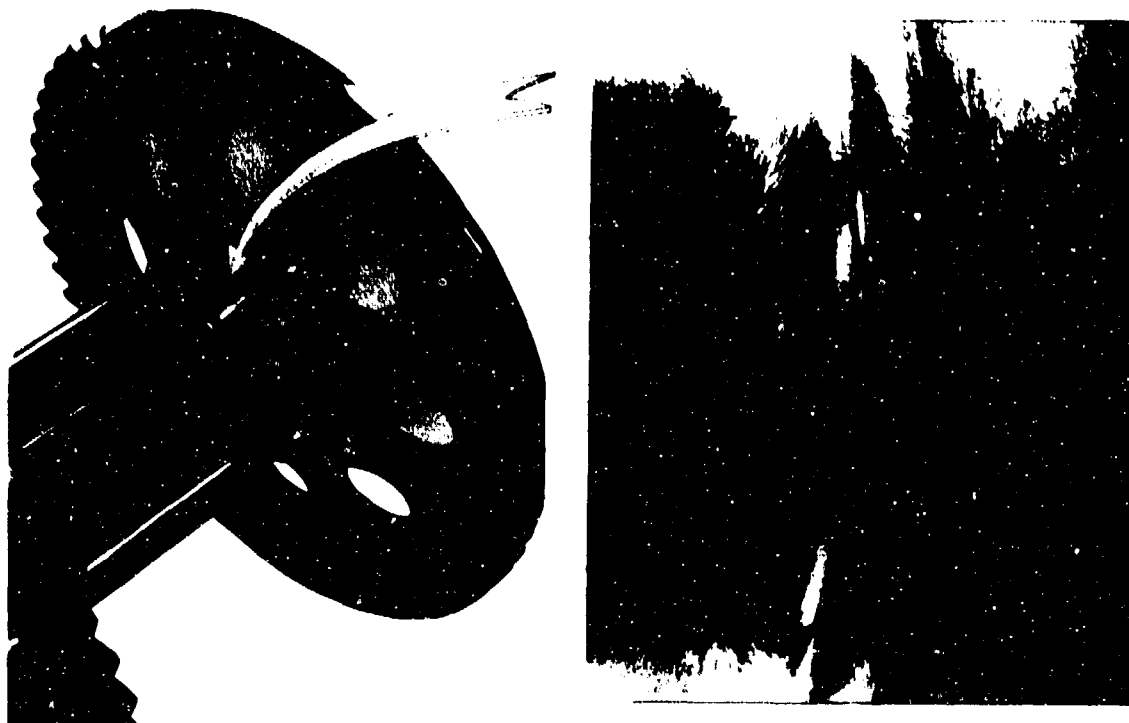


Figure C-26. Roller Interference.



Figure C-27. Roller Diameter, First-Row Pinion S/N 33.

Ultrasonic Inspection

Ultrasonic inspection of the electron-beam welds in the sun gear and first- and second-row pinions showed no degradation when the C-scans were compared to those obtained prior to the start of testing.

Ultrasonic inspection has shown to be a proven and reliable method of determining the integrity of the electron-beam welds. Facsimile recordings, C-scans, are compared to those obtained from a reference standard with known size defects and weld acceptance determined on a comparison basis. Other types of NDT were tried (magnetic particle and X-ray), but proved unsatisfactory.

Operating at frequencies above the normal range of audibility, ultrasonic energy, generated by electrically driven piezoelectric crystals, is introduced through a layer of coupling material into the component being tested. By molecular vibrations, the beam of ultrasound is transmitted through the workpiece. When a reflector, a flaw on the back side, is encountered, the beam is bounced back. Some of the reflected vibrations return along the path of entry to reenter the crystal where it is converted into an electrical signal for display on a cathode-ray tube for visual interpretation and/or displayed on a C-scan recording for permanent record (Figure C-28).

QUILL SHAFT FRACTURE, 295:50 TEST HOURS

The fractured quill shaft which stopped the R&M test at 295:50 hours is shown in Figure C-29. Two basic modes of fracture are evident, the transverse mode which acts along the line of maximum shear and the helical mode at 45° to the shaft axis along the plane of maximum tension. The multiple fractures precluded determination of the initial fracture or the sequence of fractures. However, vibration recordings of the roller gear units elucidated sidebands, indicating run-out of the sun gear. This would cause bending fatigue of the quill shaft, which could account for the transverse fracture origins.

Examination of the quill shaft revealed localized fretting damage around the fasteners from which the 45° fracture originated. Microstructural examination revealed no material abnormalities which could have attributed to the fracture, and it was concluded that the multiple fractures occurred from fatigue loadings.

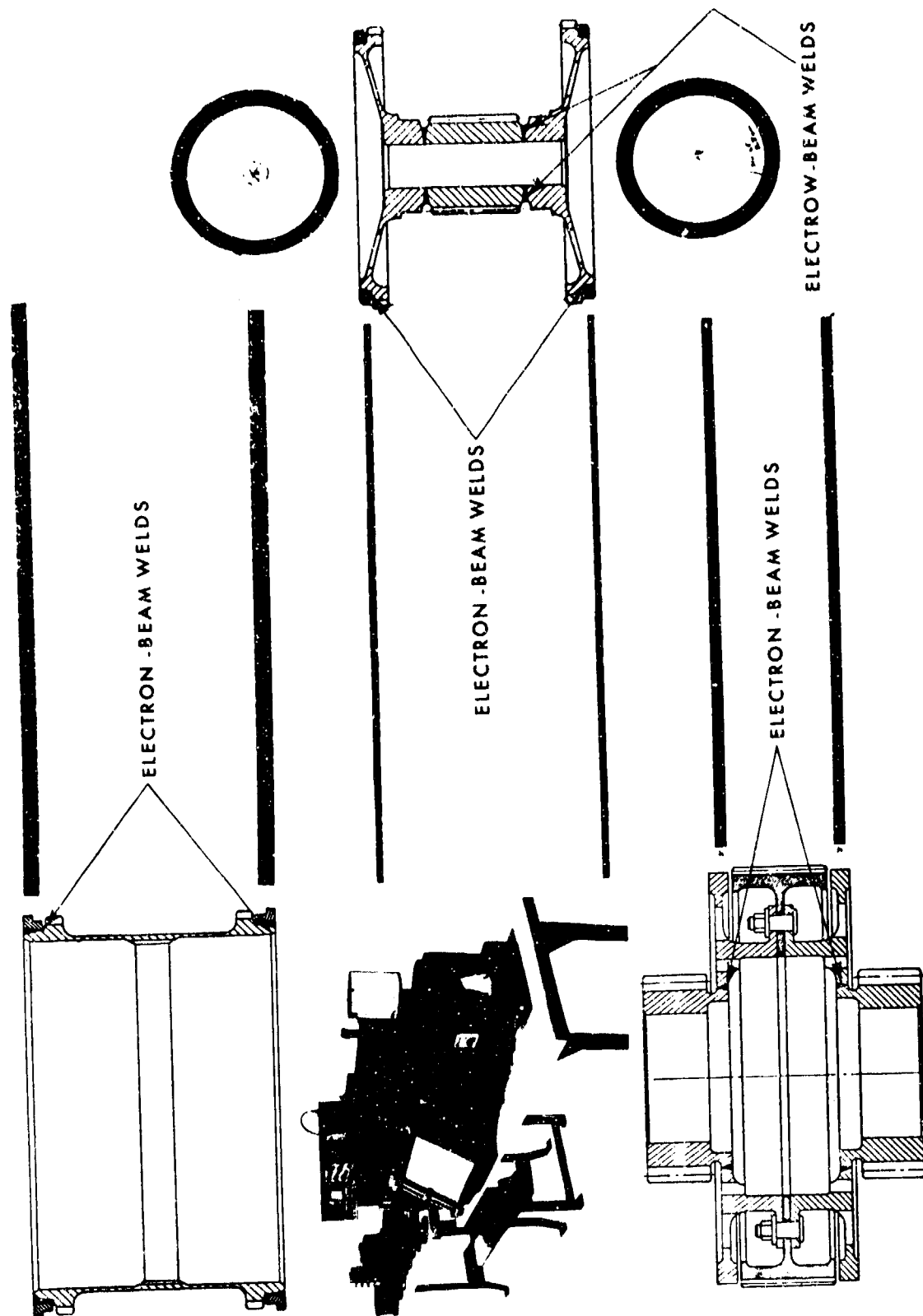


Figure C-28. "C" Scans, Electron-Beam-Welds.

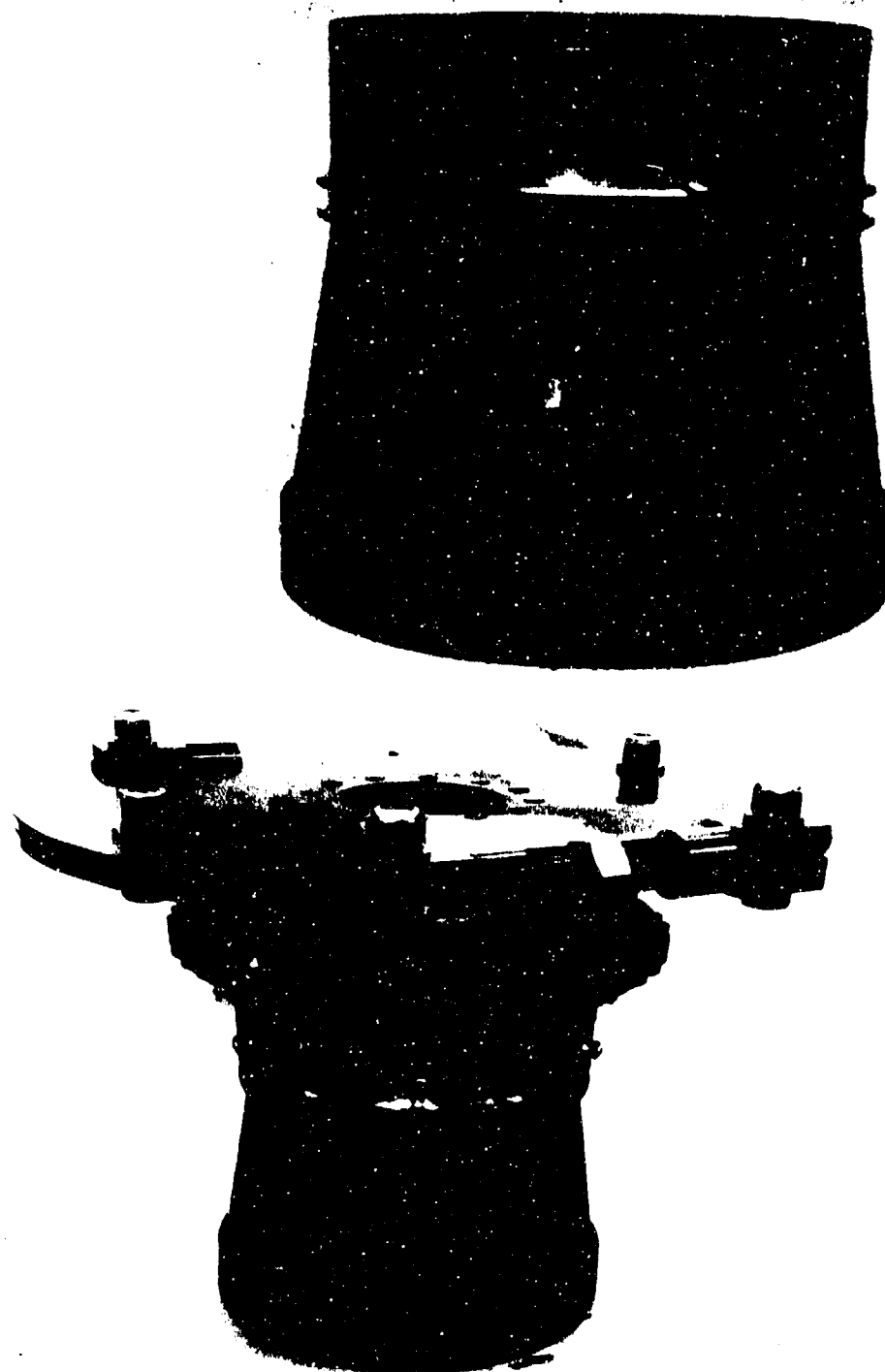


Figure C-29. Fractured Quill Shaft.

APPENDIX D

ASSEMBLY INSTRUCTIONS

Because of the unique geometry of the roller gear components, special attention had to be given to the assembly procedures of the roller gear unit. If assembly of the roller gear unit is attempted by first assembling all first-row pinions, then assembling all second-row pinions, it is found that the last second-row pinion cannot be placed in mesh with the first-row pinions. Assembly is prevented by the size of the angle between the meshes with the two first-row pinions, known as the toggle angle.

In order to overcome this difficulty, a special order of assembly was developed. The first step is to mate one of the first-row pinions with the sun gear. Next, a mating pair consisting of one first- and one second-row pinion is placed into mesh. This step is repeated until all but one first-row pinion and two second-row pinions are in place. The remaining three gears are then positioned as a set.

The second-row pinion bearing posts are then installed, followed by the placement of the output flange and hub assemblies. The two halves of the ring gear are then positioned to complete assembly of the roller gear unit. The completely assembled roller gear unit is shown in Figure D-1.

Visible in Figure D-1 are the timing marks which are critical to the assembly of the roller gear unit. Each first-row pinion is timed to the sun gear and the two second-row pinions with which it mates. Figure D-2 illustrates the tools that are utilized to ensure correct assembly of the roller gear unit.

The purpose of indexing instructions for the roller gear drive unit is to ensure assembly by the proper timing of the gear teeth. If the gear teeth are not timed properly, the roller gear drive teeth will not share the loads evenly.

ROLLER GEAR DRIVE ASSEMBLY

<u>Operation</u>	<u>Tools/Notes</u>
1. Install spherical bearing in each of the -042 assemblies of the second-row matched set assembly RG 351-11278-041 as shown in RG 351-11175.	

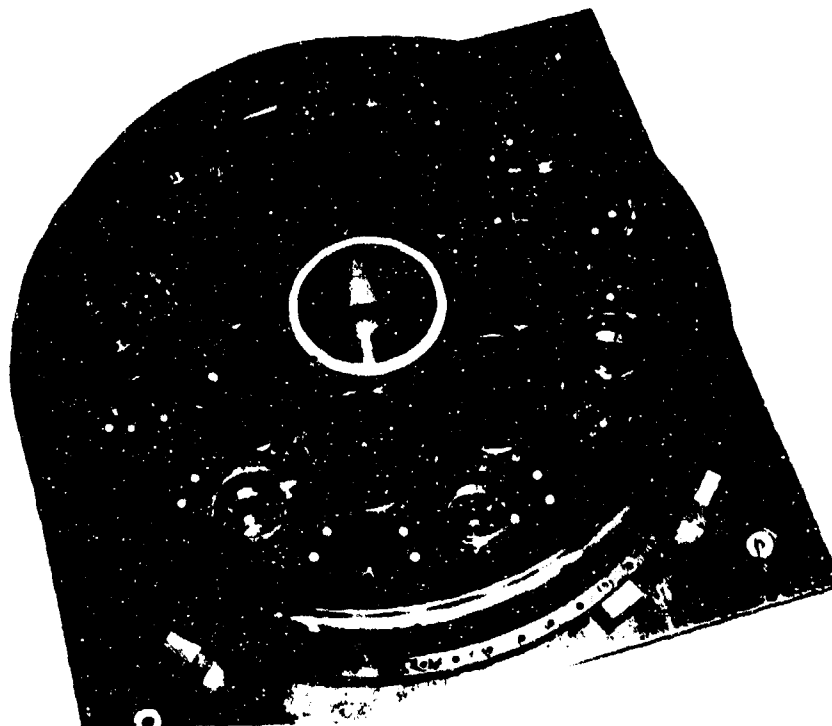


Figure D-1. Roller Gear Assembly.

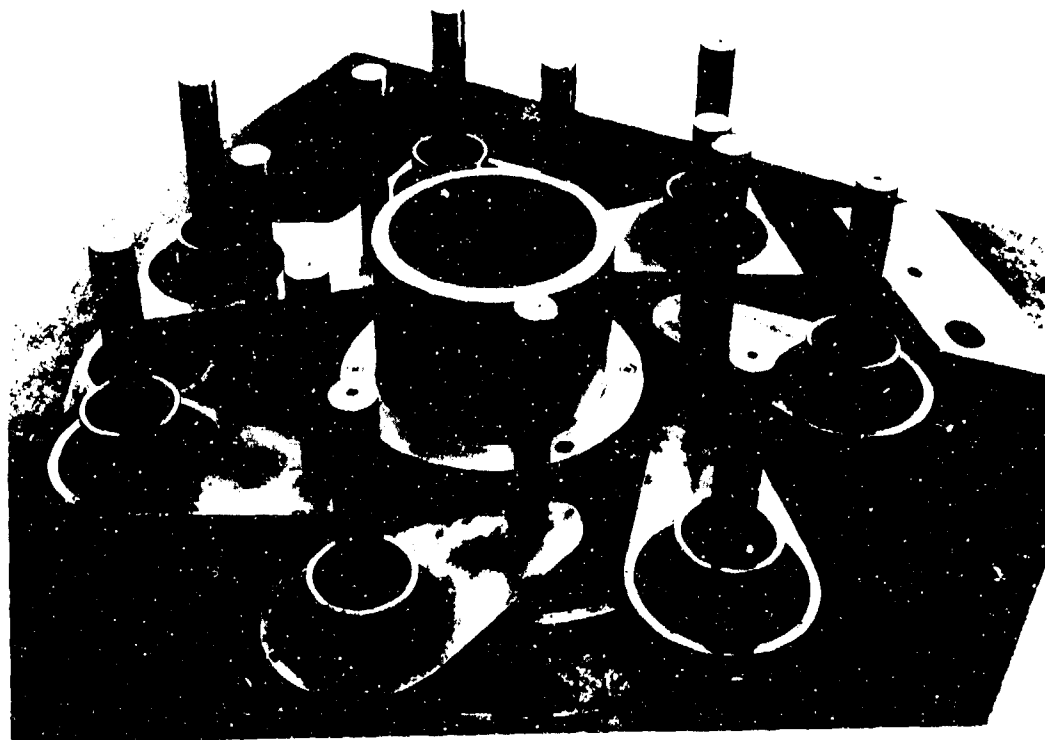


Figure D-2. Assembly Tools.

2. Mark the sun gear RG 351-11183- White lead or a suitable
-041 with white lead or some temporary marking material
other temporary marking
material as shown in Figure D-3.
Locate marks for the sun gear
on the upper surface of the
roller shoulder so that there
will be seven equally spaced
areas marked along the
periphery of the gear. This
is accomplished by starting
with any sun gear tooth space
and marking the roller
immediately above each adja-
cent tooth of space.

Count thirteen spaces,
including the previously
chosen space, and mark the
roller above the teeth adja-
cent to the thirteenth space.
As a check, there should be
seven equally spaced areas
around the periphery of the
sun gear.

3. Mark each of the -062 assemblies
of the first-row matched set gear
assembly RG 351-11182-061 on the
side marked "TOP" as illustrated
in Figure D-4. Mark the top sur-
face of the roller (58-tooth
gear only) in line with the gear
tooth marked with the letter "Z"
with white lead or some other
suitable temporary marking
material.
4. Mark each of the -042 assemblies White lead or a suitable
of the second-row matched set temporary marking material.
gear assembly RG 351-11278-041
on the side marked "TOP" as shown
in Figure D-5. Mark the top sur-
face of the 25-tooth gear in line
with the index tooth with white
lead or some other suitable temporary
marking material.

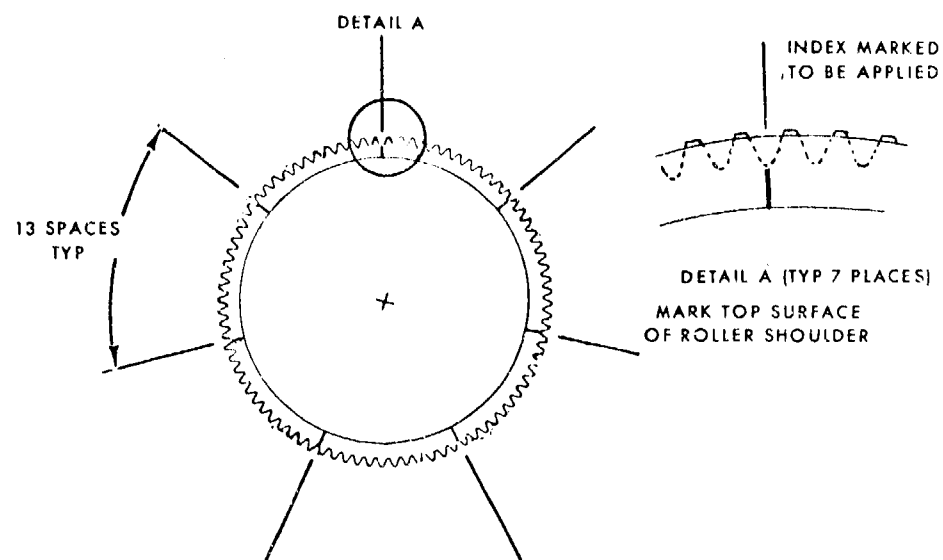


Figure D-3. Plan View, Sun Gear.

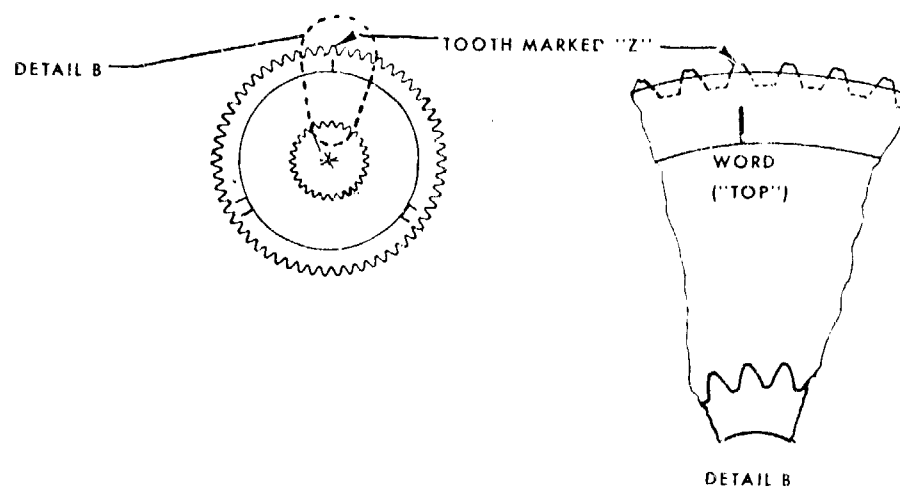


Figure D-4. Plan View, First-Row Gear.

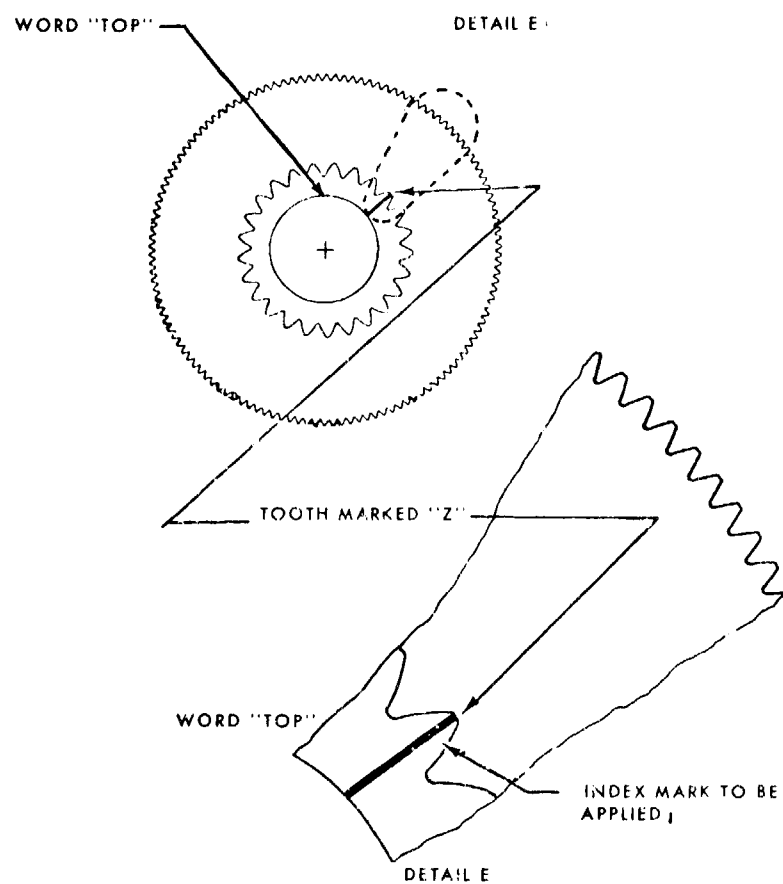


Figure D-5. Plan View, Second-Row Pinion.

Operation	Tools/Notes
5. Place the sun gear assembly RG 351-11183-041 onto center post of assembly fixture RG 370-35040-041, with index marks uppermost.	RG 370-35040-041
6. Install one first-row gear assembly RG 351-11182-062 on fixture RG 370-35040-041, with the side marked "TOP" up. Mesh marked tooth with marked tooth space of sun gear RG 351-11183-041 as shown in Figure D-6. Insert pin RG 370-35040-108.	RG 370-35040-108
7. Install disc RG 370-35040-109 into the bottom of RG 351-11278-042 second-row matched set gear assembly.	RG 370-35040-109
8. Install one second-row gear assembly of the matched set assembly RG 351-11278-041 with disc in place into fixture RG 370-35040-043 as illustrated in Figure D-7. Install second-row gear assembly with marked tooth opposite (180° to) set screw, with side marked "TOP" up.	RG 370-35040-043
9. Install one first-row gear assembly of the matched set assembly RG 351-11182-041 onto fixture RG 370-35040-043. Install gauge assembly RG 370-35040-045 into second-row gear assembly RG 351-11278-042. Insert pin RG 370-35040-108 through -045 gauge, first-row gear assembly RG 351-11182-062 and fixture RG 370-35040-043.	RG 370-35040-045 RG 370-35040-108

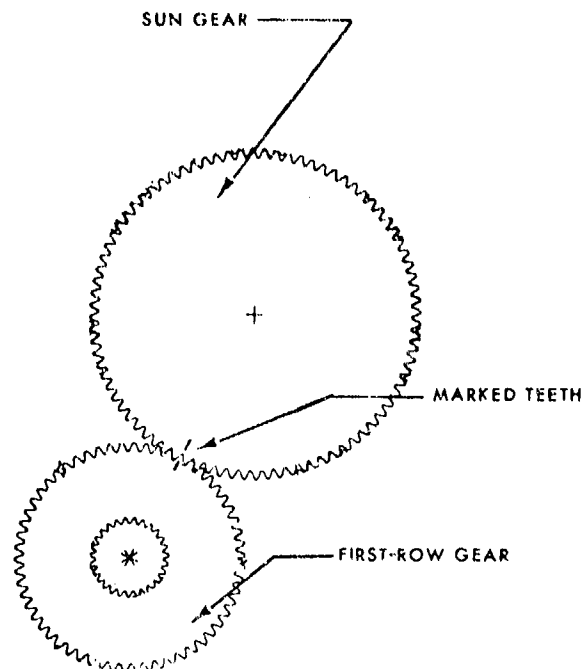


Figure D-6. Initial Assembly, Sun/First-Row Pinion.

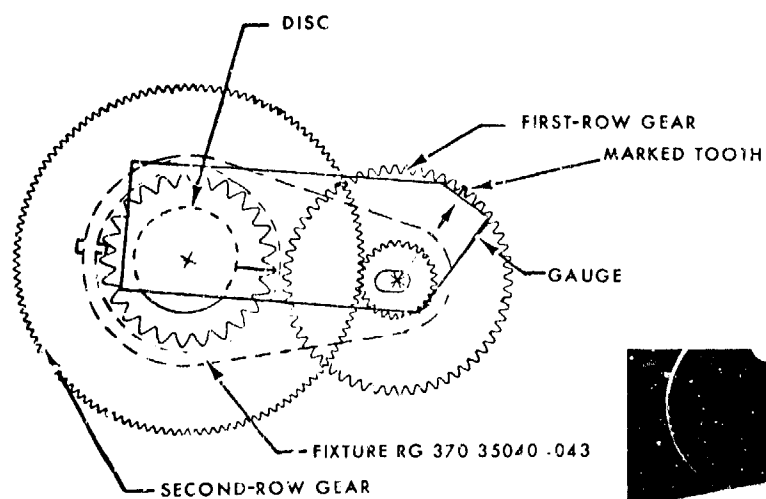


Figure D-7. Subassembly, First-/Second-Row Pinions.

Operation	Tools/Notes
The marked tooth of the second-row gear assembly RG 351-11278-042 must be visible through the hole in the gauge when arrow is pointing toward marked tooth on first-row gear assembly RG 351-11182-062.	
10. Install fixture RG 370-35040-043 with gears, gauge and pin onto assembly fixture RG 370-35040-041 and mesh marked tooth of first-row gear with marked tooth space of sun gear RG 351-11183-041 as shown in Figure D-8.	N.T.R.
Lift gears for engagement of roller shoulders.	
11. Insert pin RG 370-35040-108 located in first-row gear and fixture into assembly fixture RG 370-35040-041 hole.	N.T.R.
12. Remove gauge assembly RG 370-35040-045.	N.T.R.
13. Install an additional pin RG 370-35040-108 through second-row gear assembly RG 351-11278-042, disc RG 370-35040-109 and into hole in assembly fixture RG 370-35040-041.	RG 370-35040

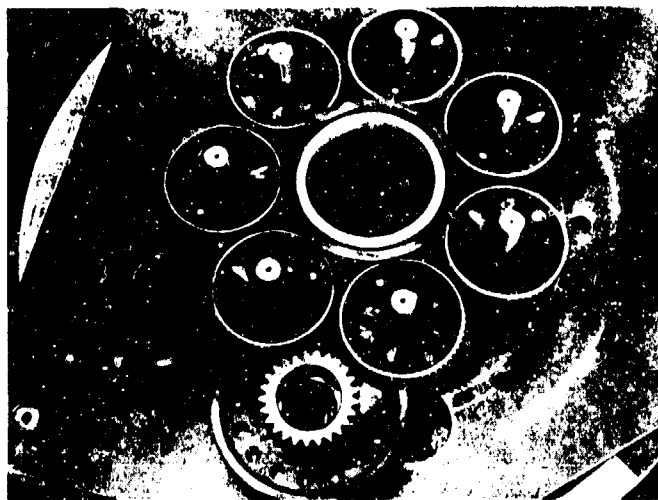
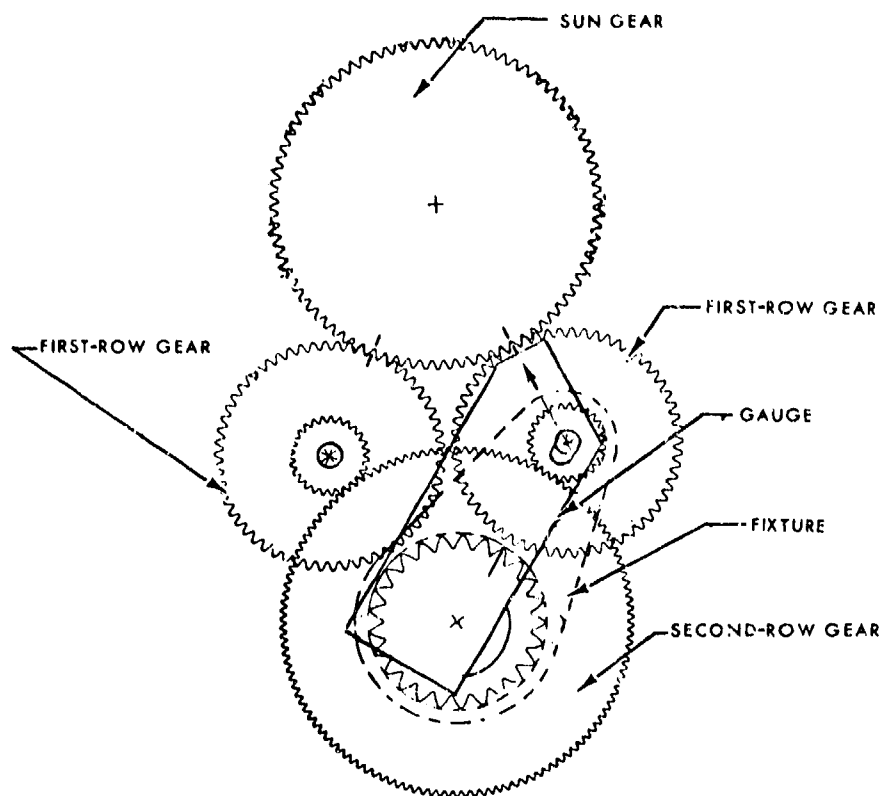


Figure D-8. Subassembly, Gear Cluster.

Operation	Tools/Notes
14. Repeat steps 6 through 12 for a total of five assemblies.	N.T.R.
The marked tooth of each of the five gears must be located in the same position relative to the gauge arrowhead.	
15. Install a disc RG 370-35040-109 into the bottom of RG 351-11278-042 second-row assembly.	RG 370-35040-110
16. Install the sixth second-row gear assembly RG 351-11278-042 with disc in place into fixture RG 370-35040-044 (3 gear fixture) as shown in Figure D-9. Install the second-row gear with the tooth marked "Z" on the 25-tooth gear 180° away from setscrew in fixture and with surface marked "TOP" uppermost.	RG 370-35040-044
17. Install the seventh first-row gear assembly RG 351-11182-062 onto fixture RG 370-35040-044. Install the gauge assembly RG 370-35040-045 on second-row gear assembly RG 351-11278-042. Insert pin RG 370-35040-108 through -045 gauge, first-row gear assembly RG 351-11182-062 into fixture RG 370-35040-044. The marked tooth on the second-row gear assembly RG 351-11278-042 must be visible through the hole in the gauge when the arrow is pointing toward the marked tooth of the first-row gear assembly RG 351-11182-042. Remove gauge.	RG 370-35040-108

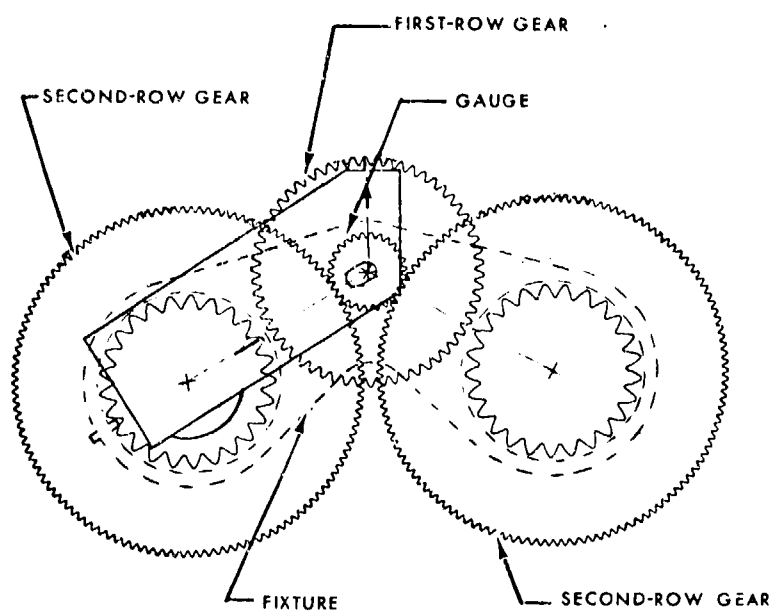


Figure D-9. Final Gear Cluster Assembly.

Operation	Tools/Notes
18. Install disc RG 370-35040-109 into bottom of remaining second-row gear assembly RG 351-11278-042.	RG 370-35040-110
19. Install the last second-row gear assembly RG 351-11278-042 with disc in place into fixture RG 370-35040-044. The second-row gear is not being indexed as yet and is indexed on final assembly.	
20. Slide fixture RG 370-35040-044 into place on assembly fixture RG 370-35040-041 as shown in Figure D-10. Lower the pin already installed per paragraph 17 into the hole in the fixture base. Install gauge assembly RG 370-35040-045 on last second-row gear assembly RG 351-11278-042. The marked tooth on the second-row pinion matched set assembly RG 351-11278-042 must be visible through the hole in the gauge when arrow is pointing toward the marked tooth of the first-row gear assembly RG 351-11182-062. Remove the gauge and install pins in the second-row gears.	
Repeat steps 18 and 19 if second-row gear assembly RG 351-11181 alignment is incorrect.	
21. By the use of bungee cord on the outer most gears, compress the entire roller gear drive unit inward radially toward the sun gear.	

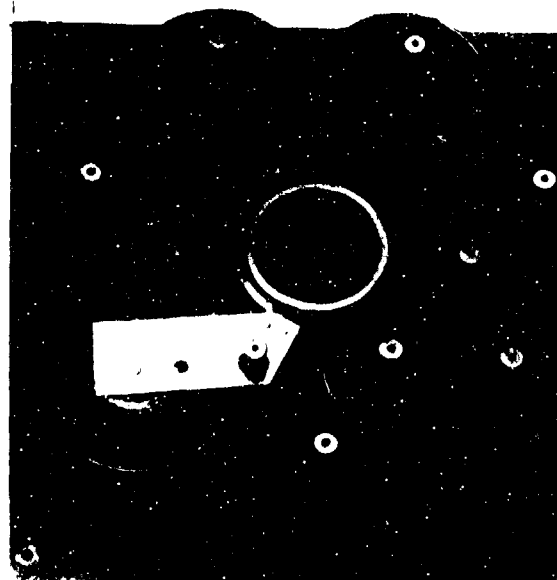
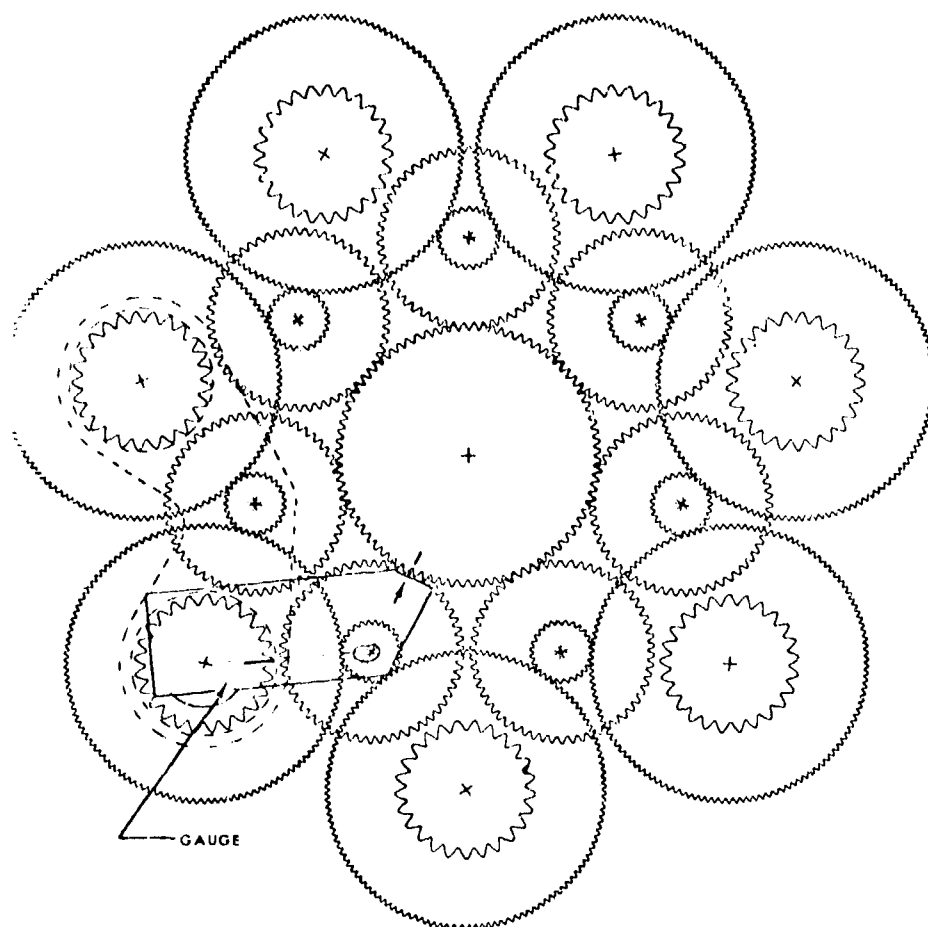


Figure D-10. Final Assembly.

Operation	Tools/Notes
22. Remove 7 pins, RG 370-35040 -108, from second-row pinions. Install the lower plate of matched plate assembly RG 351-11176 -041 with nut plates down. Install pinion shafts RG 351 -11177-101 (pack shafts in dry ice prior to assembly) into second-row gear assemblies RG 351-11278-042.	N.T.R.
23. Install spacers RG 351-11176-104, upper plate of plate assembly RG 351-11176-046, splined plate RG 351-11187-101, with all bolts per RG 351-11175.	
24. Rotate the entire roller gear drive assembly about the center post of fixture RG 370-35040-041 so that each pinion shaft in turn is directly located above the 4-inch-diameter clearance hole in the fixture base. Remove disc RG 370-35040-109. Install nut RG 351-11173-101 with wrench RG 370-35040-046. Seat the nut against the bearing but do not torque.	RG 370-35040-041
25. Repeat step 24 six times for a total of seven nuts.	
26. Assemble output flange assembly RG 351-11185-041, gear RG 351- 11169-101, and hub RG 351-11185- 103. Install hardware per RG 351-11175-041.	

Operation

Tools/Notes

27. Disassemble RG 351-11184-041 and install upper half of ring gear (half without counter sunk holes in flange) matched set RG 351-11184-041. Do not remove bungee cord. Wire this ring gear half temporarily to the second-row pinion assemblies. Attach the sling for lifting around plate assembly RG 351-11176-041. Install three AN 47-5 (or equivalent) eyebolts with sling wires attached into .437-20 threads in pinion shaft RG 351-11177-101. Remove seven remaining pins RG 370-35040-108.
28. Lift gear assembly off fixture and invert roller gear drive such that plate assembly is on bottom.
29. Torque nut RG 351-11173-101 as required per RG 351-11175-041 on each second-row pinion shaft. Install plate RG 351-11174-101 and ring RR-206.
30. Repeat step 29 six times for a total of seven nuts.
31. Install lower half of ring gear matched set RG 351-11184-041 such that all 16 taper lock holes are aligned. Remove bungee cord. Reinstall 16 taper locks per RG 351-11184.

APPENDIX E

HAZARD FUNCTION ANALYSIS

INTRODUCTION

Hazard function analysis is the statistical technique which was used to predict the projected reliability of both the baseline planetary and roller gear transmissions. The basic equations of this technique were presented in the discussion of reliability analysis. This section presents a brief discussion of the derivation of the size and shape parameters which form the basis of the hazard function. The discussion is presented only to give the reader some insight into how these parameters are derived from field service data.

For a more complete treatment, the reader is referred to USAAMRDL-TR-75-57, "Helicopter Drive System On-Condition Maintenance Capability."

This report also lists several references which can provide the reader with a more rigorous discussion of the mathematical derivation of the technique. The latter part of this appendix is comprised of tables which list the hazard function parameters used in this program.

DERIVATION OF SIZE AND SHAPE PARAMETERS

It will be useful here to repeat the basic hazard function relationship.

$$h(t) = \frac{\beta}{\theta} \left[\frac{t}{\theta} \right]^{\beta-1}$$

where

- $h(t)$ = instantaneous failure rate as a function of time
- β = shape parameter (dimensionless)
- θ = size parameter (hours)

In order to evaluate the size and shape parameters, it is first necessary to classify the various failure modes observed since each failure mode for a generic component has its own hazard function. It is then necessary to arrange the failures observed in ascending order of occurrence with time since overhaul. Once this is accomplished, a Weibull plot is made of percentage of items failed vs time since overhaul, as shown in Figure E-1. This plot shows the relationship of

$$\ln \ln \left[\frac{1}{1 - F(t_i)} \right] \text{ vs } \ln t_i$$

where

$F(t_i)$ is the probability of failure at time $t = t_i$.

A computerized procedure is then used to determine if a proper correlation exists between the

$$\ln \ln \left[\frac{1}{1 - F(t_i)} \right]$$

and the $\ln(t_i)$.

Briefly stated, however, it is necessary that a straight line relationship exist for the Weibull plot. The straight line is then assumed to have the following form

$$\ln \ln \left[\frac{1}{1 - F(t)} \right] = -\beta \ln \theta + \beta \ln t$$

The values of β and θ are then computed so that their values maximize the probability that the failures occurred the way they did. This is essentially a sophisticated best fit method. The β and θ so computed describe the hazard function for the particular failure mode of the generic component under discussion. This process is repeated for each failure mode of each generic component within the system for which the reliability is to be computed.

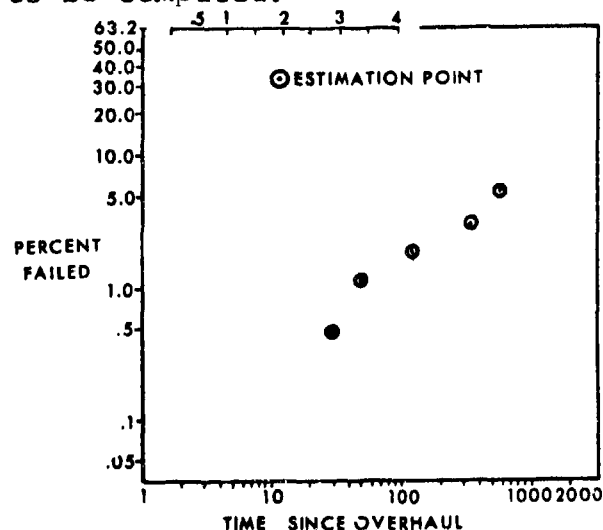


Figure E-1. Typical Weibull Plot.

The hazard function parameters computed for the hazard function analysis are listed in Tables E-1 through E-5. A description of their use in computing the reliability of the roller gear and baseline transmissions follows.

Table E-1 shows the differing parts count between the roller gear drive unit and the baseline planetary unit. These values were used to adjust the shape and size parameters, obtained from field experience of the S-64 and S-65 transmissions, to produce the hazard function parameters listed in Table E-2. From these values the projected MTBR of 1750 hours was computed for the roller gear drive.

Correlating the actual test results into a hazard function analysis resulted in the respective θ 's shown in Table E-3 for a maximum likelihood MTBR of 612 hours and, at 95 percent confidence, a MTBR of 1135 hours. These values reflect a 4.4 growth factor derived from the S-61 main transmission growth curve of Figure 20.

Table E-4 lists the main transmission generic components that are common to the roller gear and baseline transmissions and their associated hazard function parameters. Combining the parameters from Table E-3 with those of E-4 resulted in main transmission computed MTBR's of 394 and 657 hours, at 50 percent and 95 percent confidence levels respectively. The safety-of-flight hazard function parameters used to compute the failure rate depicted in Figure 19 are given in Table E-5. As a result of the constant failure rate shown in Figure 19, on-condition maintenance would appear feasible for a fully developed roller gear main transmission.

TABLE E-1. DYNAMIC COMPONENT FAILURE MODE QUANTITY COMPARISON.

Section	Generic Component	Failure Mode	Roller Gear Drive	Baseline Planetary
Roller Gear Drive Assy	Spur Gear	Excess Wear	47	18
		Spalling	47	18
		Scoring	47	18
		Tooth Fracture	47	18
		Web/Shaft Fracture	47	18
	Roller	E/B Weld Deterioration	42	-
		Excess Wear	44	-
		Spalling	44	-
		Scoring	44	-
		E/B Weld Deterioration	30	-
Spline Lock Ring Spherical Roller Bearings	Spline	Wear, Fretting	2	1
		Fracture	7	19
		Cage Fracture	7	13
		Spalling	14	26
		Smearing	14	26
	Nut "O" Ring Plate Assy Bolts Flange Ball Bearing	Excess Wear	14	26
		Loose	404	298
		Leakage	7	26
		Crack	1	2
		Shear	411	389
	Ball Bearing	Fracture	1	-
		Cage Fracture	-	1
		Spalling	-	1
		Smearing	-	1
		Excess Wear	-	1

TABLE E-2. FINAL REDUCTION DRIVE, HAZARD FUNCTION PARAMETER COMPARISON.

Section	Generic Component	Failure Mode	Roller Gear Drive		Baseline Planetary	
			β	θ (hrs)	β	θ (hrs)
Roller Gear Drive Assy	Spur Gear	Excess Wear	1.3	2.2x10 ⁴	1.3	1.7x10 ⁵
		Spalling	1.0	1.7x10 ⁶	1.0	9.0x10 ⁶
		Scoring	1.0	1.3x10 ⁵	1.0	9.0x10 ⁶
		Tooth Fracture	1.0	2.8x10 ⁶	1.0	9.0x10 ⁶
		Web/Shaft Fracture	1.0	2.3x10 ⁶	1.0	9.0x10 ⁶
		E/B Weld Deterioration	1.0	5.6x10 ⁶	-	-
Roller	Roller	Excess Wear	1.3	4.0x10 ⁵	-	-
		Spalling	1.0	8.2x10 ⁶	-	-
		Scoring	1.0	2.6x10 ⁸	-	-
		E/B Weld Deterioration	1.0	3.9x10 ⁷	-	-
		Wear, Fretting	1.33	4.4x10 ⁵	1.33	9.9x10 ⁶
		Fracture	1.0	3.9x10 ⁷	1.0	2.1x10 ⁷
Spline Lock Ring Spherical Roller Bearings	Spline Lock Ring Spherical Roller Bearings	Cage Fracture	1.0	9.1x10 ⁶	1.0	2.4x10 ⁷
		Spalling	1.3	3.8x10 ⁴	1.3	1.8x10 ⁵
		Smearing	1.0	4.0x10 ⁷	1.0	1.2x10 ⁶
		Excess Wear	1.1	4.6x10 ⁴	1.1	3.1x10 ⁶
		Loose	1.7	9.7x10 ⁵	1.7	4.8x10 ⁴
		Leakage	.79	1.5x10 ⁵	-	-
Nut "O" Ring Plate Assy Bolts Flange Ball Bearing	Nut "O" Ring Plate Assy Bolts Flange Ball Bearing	Crack	1.0	6.0x10 ⁵	1.0	2.0x10 ⁶
		Shear	1.0	6.5x10 ⁴	1.0	1.5x10 ⁴
		Fracture	1.0	1.2x10 ⁵	-	-
		Cage Fracture	-	-	1.0	2.4x10 ⁵
		Spalling	-	-	1.3	2.1x10 ⁸
		Smearing	-	-	1.0	4.4x10 ⁷
Excess Wear	Excess Wear	Excess Wear	-	-	1.1	6.1x10 ⁷

TABLE E-3. ROLLER GEAR DRIVE, TEST HAZARD FUNCTION PARAMETER PROJECTIONS.					
Section	Generic Component	Failure Mode	β	θ (hrs)	θ (hrs)
				50% Confidence	95% Confidence
Roller Gear Drive	Spur Gear	Excess Wear	1.3	1.3x10 ⁵	2.4x10 ⁵
		Spalling	1.0	3.2x10 ⁵	6.1x10 ⁵
		Scoring	1.0	2.7x10 ³	5.1x10 ³
Roller	Web/Shaft Fracture	Tooth Fracture	1.0	3.2x10 ³	6.1x10 ³
		Web/Shaft Fracture	1.0	2.0x10 ³	2.8x10 ³
		E/B Weld Deterioration	1.0	1.5x10 ⁵	2.8x10 ⁵
		Excess Wear	1.3	2.3x10 ³	4.4x10 ³
		Spalling	1.0	3.2x10 ³	6.1x10 ³
	E/B Weld Deterioration	Scoring	1.0	2.7x10 ⁷	5.1x10 ⁷
		E/B Weld Deterioration	1.0	3.8x10 ⁵	7.2x10 ⁵
		Wear, Fretting	1.33	2.4x10 ⁷	4.6x10 ⁷
		Fracture	1.0	3.8x10 ⁶	7.2x10 ⁷
		Cage Fracture	1.0	8.9x10 ⁶	1.6x10 ⁷
Spherical Roller Bearings	Spalling	1.3	3.7x10 ⁷	7.1x10 ⁷	
	Smearing	1.0	2.3x10 ⁵	4.4x10 ⁵	
	Excess Wear	1.1	3.7x10 ⁵	6.9x10 ⁵	
Nut	"O" Ring	Loose	1.7	3.7x10 ⁵	7.0x10 ⁵
Leakage		.79	2.8x10 ⁵	5.3x10 ⁵	
Crack		1.0	2.8x10 ⁵	1.1x10 ⁶	
Shear		1.0	6.4x10 ⁴	1.2x10 ⁵	
Fracture		1.0	1.2x10 ⁵	2.2x10 ⁵	

TABLE E-4. ROLLER GEAR/BASELINE PROJECTED HAZARD FUNCTION PARAMETERS.			
Section	Generic Component	Failure Mode	θ (hrs)
Input Bevel & Second-Stage Spur Gear	Spiral Bevel Gear	Excess Wear	1.4
		Spalling	5.2x10 ⁵
		Scoring	1.0
		Scoring	2.7x10 ⁶
		Tooth Fracture	1.0
		Web Shaft Crack	1.6x10 ⁶
		Excess Wear	1.0
		Spalling	1.6x10 ⁶
		Scoring	1.3
		Scoring	6.8x10 ⁵
Spur Gear	Spur Gear	Excess Wear	1.0
		Spalling	5.4x10 ⁶
		Scoring	1.0
		Scoring	5.4x10 ⁶
		Tooth Fracture	1.0
		Web/Shaft Crack	5.4x10 ⁶
		Wear Fretting	1.0
		Cage Fracture	2.6x10 ⁴
		Spalling	1.33
		Smearing	1.0
Spline Ball Bearing	Spline Ball Bearing	Excess Wear	1.3
		Fracture	2.3x10 ⁵
		Crack, Shear	1.0
		Crack	2.4x10 ⁷
		Spline Wear	1.1
		Cage Fracture	4.4x10 ⁶
		Spalling	5.1x10 ⁶
		Smearing	7.3x10 ⁴
		Excess Wear	1.0
		Brinelling	.98
Lock Ring Bearing Retainer Shaft	Lock Ring Bearing Retainer Shaft	Excess Wear	1.0
		Fracture	1.2
		Crack, Shear	1.0
		Crack	4.4x10 ⁷
		Spline Wear	1.3
		Cage Fracture	1.0
		Spalling	4.9x10 ⁵
		Smearing	4.4x10 ⁷
		Excess Wear	1.0
		Cage Fracture	1.1
Roller Bearing	Roller Bearing	Excess Wear	1.0
		Fracture	4.6x10 ⁵
		Crack, Shear	2.4
		Crack	3.8x10 ⁷
		Spline Wear	1.0
		Cage Fracture	4.0x10 ⁵
		Spalling	9.7x10 ⁵
		Smearing	2.8x10 ⁶
		Excess Wear	1.0
		Cage Fracture	1.1
Tapered Roller	Tapered Roller	Excess Wear	1.76
		Fracture	1.6
		Crack, Shear	1.1
		Crack	1.0
		Spline Wear	1.1
		Cage Fracture	2.2x10 ⁵
		Spalling	5.4x10 ⁶
		Smearing	1.1
		Excess Wear	1.1
		Cage Fracture	1.0
FWU Housing	FWU Housing	Excess Wear	1.0
		Fracture	1.2
		Crack, Shear	1.0
		Crack	4.4x10 ⁷
		Spline Wear	1.3
		Cage Fracture	1.0
		Spalling	4.9x10 ⁵
		Smearing	4.4x10 ⁷
		Excess Wear	1.0
		Cage Fracture	1.1
FWU Cage	FWU Cage	Excess Wear	1.0
		Fracture	4.6x10 ⁵
		Crack, Shear	2.4
		Crack	3.8x10 ⁷
		Spline Wear	1.0
		Cage Fracture	4.0x10 ⁵
		Spalling	9.7x10 ⁵
		Smearing	2.8x10 ⁶
		Excess Wear	1.0
		Cage Fracture	1.1
FWU Roller	FWU Roller	Excess Wear	1.76
		Fracture	1.6
		Crack, Shear	1.1
		Crack	1.0
		Spline Wear	1.1
		Cage Fracture	2.2x10 ⁵
		Spalling	5.4x10 ⁶
		Smearing	1.1
		Excess Wear	1.1
		Cage Fracture	1.0

TABLE E-4. (Continued)

Section	Generic Component	Failure Mode	B	θ (hrs)
Input Bevel & Second-Stage Spur Gear (Cont'd)	Spring	Fracture	1.0	2.1x10 ⁶
	Nut	Loose	1.7	5.1x10 ⁵
	"O" Ring	Leakage	.79	2.5x10 ⁵
	Shaft Seal	Leakage	1.95	8.0x10 ³
	Housing	Crack	2.38	4.5x10 ⁴
Main Shaft	Bolts	Shear	1.0	2.2x10 ⁴
	Spline	Wear, Fretting	1.33	1.4x10 ⁵
	Ball Bearing	Cage Fracture	1.0	1.2x10 ⁵
		Spalling	1.3	1.3x10 ⁶
		Smearing	1.0	2.2x10 ⁸
		Excess Wear	1.1	3.3x10 ⁷
	Lock Ring	Fracture	1.0	4.1x10 ⁷
	Bearing Retainer	Crack, Shear	.98	6.1x10 ⁵
	Shaft	Crack	1.0	4.1x10 ⁷
	Roller Bearing	Spline Wear	1.2	4.9x10 ⁸
		Cage Fracture	1.0	3.1x10 ⁸
		Spalling	1.3	2.2x10 ⁶
		Smearing	1.0	3.1x10 ⁸
		Excess Wear	1.1	6.0x10 ⁷
	Nut	Loose	1.7	2.6x10 ⁵
	"O" Ring	Leakage	.79	6.0x10 ⁵
	Shaft Seal	Leakage	1.95	6.5x10 ³
	Housing	Crack	2.38	7.7x10 ⁴
	Bolts	Shear	1.0	2.9x10 ⁵

TABLE E-4. (Continued)

TABLE E-4. (Continued)				
Section	Generic Component	Failure Mode	δ	θ (hrs)
Tail Take-Off & Accessories	Spiral Bevel Gear	Excess Wear	1.4	1.6x10 ⁶
		Spalling	1.0	1.4x10 ⁶
		Scoring	1.0	8.1x10 ⁶
		Tooth Fracture	1.0	8.1x10 ⁶
	Spline	Web/Shaft Crack	1.0	8.1x10 ⁶
		Wear, Fretting	1.33	2.2x10 ⁴
		Cage Fracture	1.0	1.7x10 ⁴
		Spalling	1.3	2.8x10 ⁵
	Ball Bearing	Smearing	1.0	3.1x10 ⁷
		Excess Wear	1.1	5.6x10 ⁶
		Fracture	1.0	2.3x10 ⁶
		Crack, Shear	.98	4.0x10 ⁵
	Lock Ring	Crack	1.0	8.2x10 ⁶
		Spline Wear	1.2	2.8x10 ⁸
Cage Fracture		1.0	3.9x10 ⁷	
Spalling		1.3	4.4x10 ⁵	
Roller Bearing	Smearing	1.0	3.9x10 ⁷	
	Excess Wear	1.1	9.1x10 ⁶	
	Cage Fracture	1.0	4.1x10 ⁷	
	Spalling	2.4	3.8x10 ⁴	
Tapered Roller Bearing	Smearing	1.0	4.6x10 ⁵	
	Excess Wear	1.3	9.7x10 ⁵	
	Loose	1.7	1.5x10 ⁵	
	Leakage	.79	6.0x10 ⁵	
Single Specimen	Housing	2.38	6.5x10 ⁵	
	Bolts	1.0	1.2x10 ⁵	
	Shear			

TABLE E-5. SAFETY-OF-FLIGHT HAZARD FUNCTION PARAMETER SUMMARY.

Section	Generic Component	Failure Mode	β	θ (hrs)
Input	Spiral Bevel Gear	Web/Shaft Crack	1.0	3.1×10^8
	Spur Gear	Web/Shaft Crack	1.0	1.6×10^{10}
	Roller Bearing	Cage Fracture	1.0	3.1×10^9
	Tapered Roller Brg	Cage Fracture	1.0	4.0×10^9
Main Shaft	Spline	Wear, Fretting	1.33	1.4×10^7
	Ball Bearing	Cage Fracture	1.0	1.2×10^8
	Bearing Retainer	Crack, Shear	.98	1.2×10^8
	Roller Bearing	Cage Fracture	1.0	3.1×10^{10}
	Nut	Loose	1.7	1.3×10^8
	Shaft	Crack	1.0	4.1×10^9
	Shaft Seal	Leakage	1.95	8.0×10^5
	Housing	Crack	2.38	7.7×10^6
Tail Take-Off & Accessories	Spiral Bevel Gear	Web/Shaft Crack	1.0	8.0×10^8
	Spur Gear	Web/Shaft Crack	1.0	8.1×10^6
	Spline	Wear, Fretting	1.33	8.6×10^6
	Ball Bearing	Cage Fracture	1.0	1.2×10^9
	Shaft	Crack	1.0	4.1×10^9
	Roller Bearing	Cage Fracture	1.0	1.6×10^{10}
	Tapered Roller Brg	Cage Fracture	1.0	4.0×10^9
	Spur Gear	Web/Shaft Crack	1.0	2.0×10^5
		E/B Weld Deterioration	1.0	1.5×10^7
Roller Gear Drive	Spline	Wear, Fretting	1.33	2.4×10^8
	Spherical Roller Brg	Cage Fracture	1.0	8.9×10^8
	Bolts	Shear	1.0	3.7×10^8
	Flange	Fracture	1.0	1.2×10^7

APPENDIX F

ROLLER GEAR DRIVE, FAILURE MODE AND EFFECTS ANALYSIS

A primary advantage of the roller gear drive is its relative compactness when compared to conventional planetaries. It offers a very high reduction ratio in a relatively small envelope. This very compactness, however, may lead to a higher degree of secondary damage in the event of the failure of a single component. In a helicopter transmission, where a failure in the primary drive system can be catastrophic, it is important that potential failure modes and the secondary effects associated with such failures be identified and evaluated. The technique used to evaluate the roller gear drive in this respect is called a failure modes and effects analysis (FMEA).

Failure mode analysis starts at the component level. The possible ways each component can fail are identified, and the effects of these failures on the next higher level of assembly are analyzed. This technique proceeds upward until the effects on the overall system are evaluated.

The functional block diagram, Figure F-1, shows the breakdown of the roller gear drive components used to conduct the FMEA. The drawing of Figure F-2 shows the various individual components which make up an assembly and their relationship to the roller gear drive assembly.

Table F-1 lists the potential failure modes and the effect each failure could have on the roller gear drive assembly. The probable symptom and means of detection are listed.

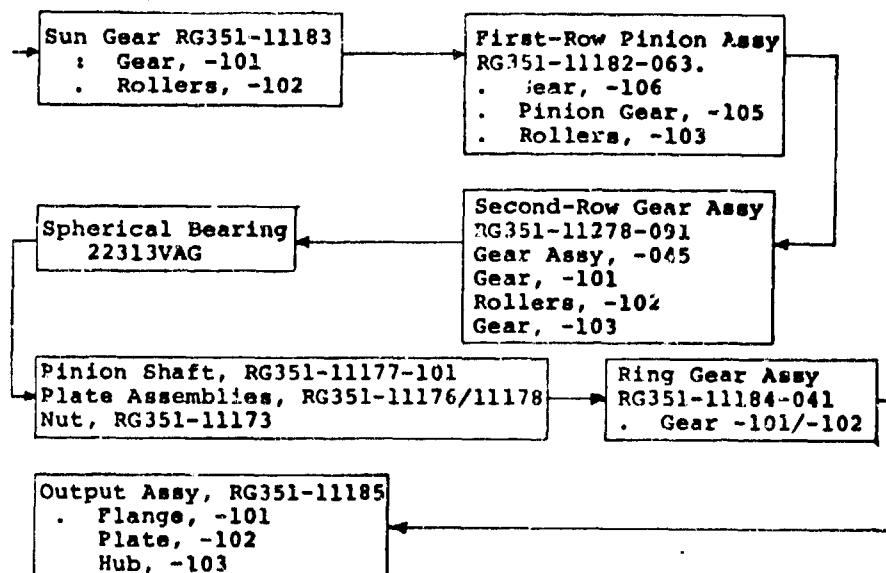
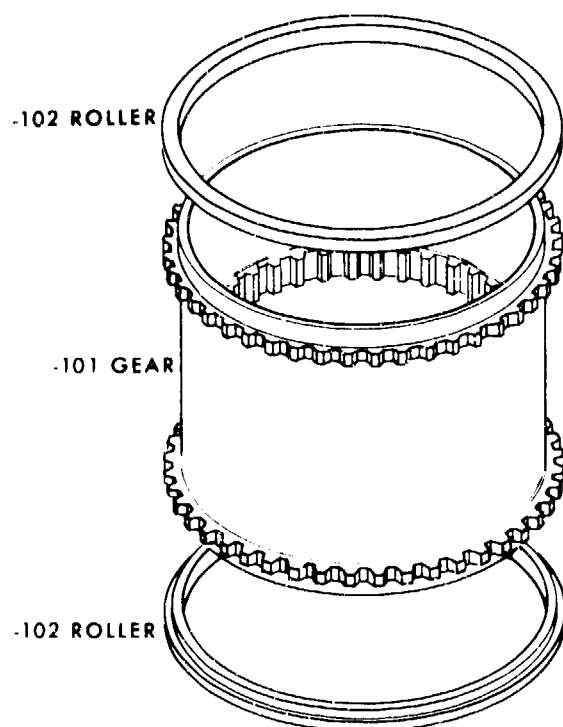
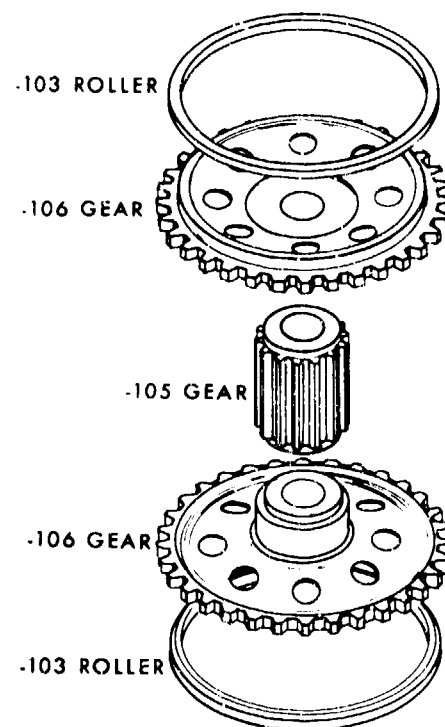


Figure F-1. Functional Block Diagram, Roller Gear Drive.

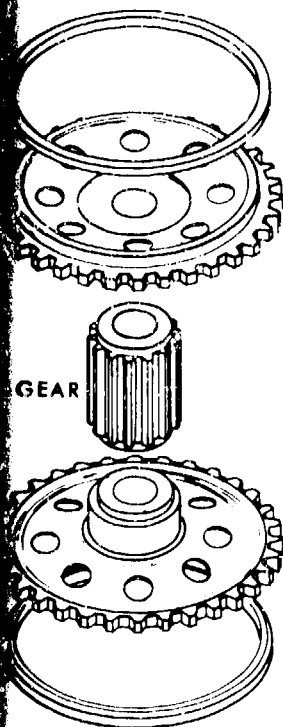


SUN GEAR
RG351-11183

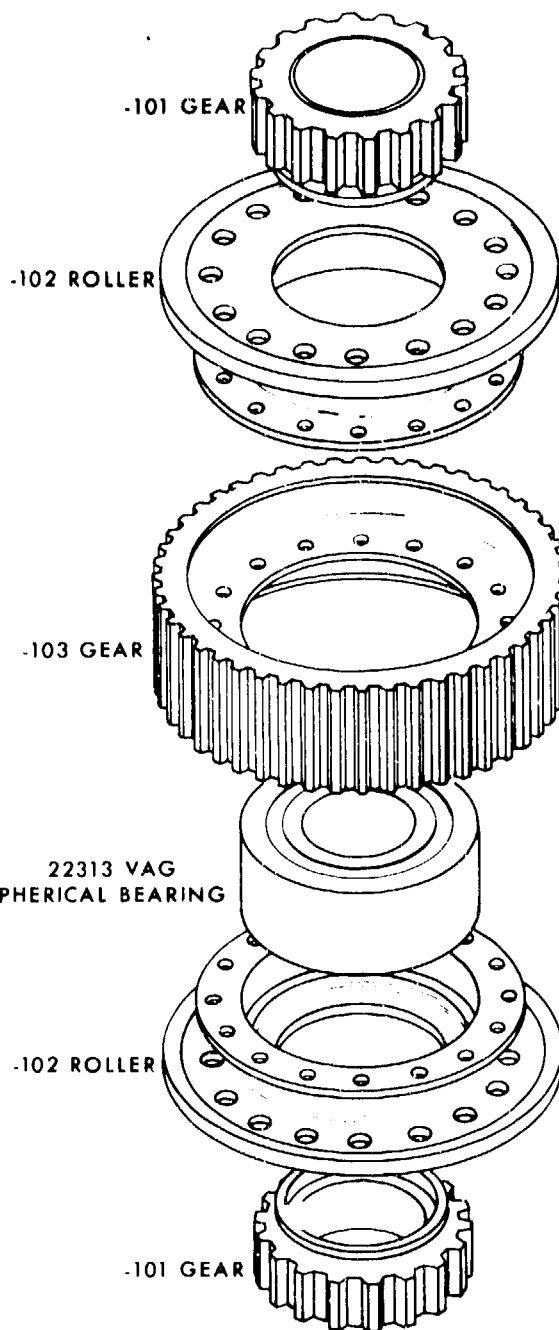


FIRST-ROW PINION
RE351-11182-063

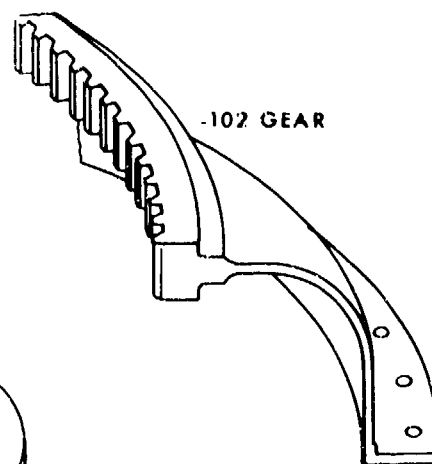
Figure F-2. Roller Gear Drive Components,
Exploded View.



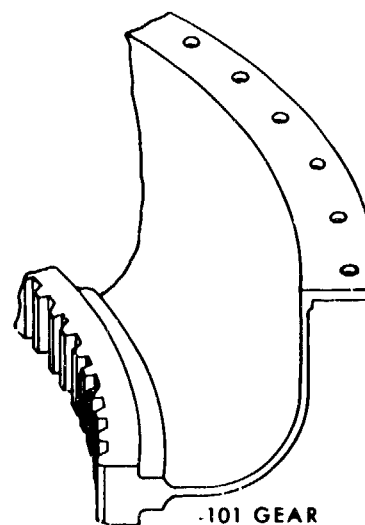
FIRST-ROW PINION
RE351-11182-063



SECOND-ROW GEAR ASSEMBLY
RG351-11278-041



-102 GEAR



-101 GEAR

RING GEAR
RG351-11184-041

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER C					
PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER C
RG351-11183-101	Sun Gear	1	Transmits torque from input shaft to first-row gear assembly.	Tooth fracture.	Secondary damage to other gear teeth and redistribution of stresses.
					Secondary damage to gear teeth and due to chip contamination.
					Complete loss would cause loss of function of the assembly.
				Scuffing of teeth.	Secondary damage to other gear teeth and rollers due to contamination.
				Crack in gear assembly (excluding gear teeth).	Propagation to could cause loss of function of RG
RG351-11183-102	Sun Gear Rollers	2	Reacts radial loads	Spline fretting.	None.
				Spalling/pitting	Secondary damage to other gear teeth and rollers due to contamination.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 1 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENTS	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
torque out shaft -row gear y. ro nta of ss e R ge eth me fr ss AD age eth m	Tooth fracture.	Secondary damage to other gear teeth due to redistribution of stresses. Secondary damage to other gear teeth and rollers due to chip contamination. Complete loss of teeth would cause loss of function of the RGD assembly.	Chip detector indicator and possible vibration/noise.	Negligible probability that complete loss of teeth could occur prior to detection of failure.
	Scuffing of teeth.	Secondary damage to other gear teeth and rollers due to metal contamination.	Teardown inspection at overhaul.	
	Crack in gear assembly (excluding gear teeth).	Propagation to fracture could cause loss of function of RGD assembly.	Teardown inspection at overhaul. Loss of MGB output.	
	Spline fretting.	None.	Teardown inspection at overhaul.	
	Spalling/pitting	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator.	
radial				

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER					
PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER
RG351-11183-102 (continued)	Sun Gear Rollers	2	Reacts radial loads.	Excessive wear of the roller surfaces. Crack in roller or weld.	Secondary damage to other gear teeth and rollers due to contamination. Secondary damage to other components due to chip contamination.
RG351-11182-063	First-Row Pinion Assem- bly (See also RG351-11182- 105 RG351-11182- 106)	7	Supports gears (-105 and -106) and transmits torque from gear -106 to gear -105	Crack in gear assembly (excluding gear teeth). Spalling/pitting of roller surface. Excessive wear of the roller surfaces.	Propagation to other components could cause loss of function of RG 351-11182-105 and -106. Secondary damage to gear teeth and due to metal chips. Secondary damage to gear teeth and due to metal chips.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 2 of 10)

FAILURE MODE	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
age eth o m .	radial	Excessive wear of the roller surfaces.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detect indicator or teardown inspection at overhaul.	
age nts ati		Crack in roller or weld.	Secondary damage to other components due to chip contamination.	Chip detector indicator or teardown inspection at overhaul.	Negligible probability that loss of roller would be catastrophic as gear outside diameter would react on root diameter of pin
o f oss GD	ts gears(-105 06) and its torque ear -106 to 105	Crack in gear assembly (excluding gear teeth).	Propagation to fracture could cause loss of function of RGD assembly.	Chip detector or teardown inspection at overhaul. Loss of MGB output.	Probability that chip detector would preclude catastrophic failure.
age d r con		Spalling/pitting of roller surface.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator.	
age d r con		Excessive wear of the roller surfaces.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator or teardown inspection at overhaul.	

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER C

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE ON ROLLER C
RG351-11182-106	First-Row Gear (See also: RG351-11182-063)	14	Transmits torque from sun gear to first-row gear assembly.	Tooth fracture.	Secondary damage gear teeth due to distribution of
				Scuffing of teeth.	Secondary damage gear teeth and due to chip con
RG351-11182-105	First-Row Pinion Gear (See also: RG351-11182-063)	7	Transmits torque from first-row gear assembly to second-row gear assembly.	Tooth fracture.	Complete loss of would cause loss function of the assembly.
					Secondary damage gear teeth and due to metal con
					Secondary damage gear teeth and due to chip con
					Complete loss of would cause loss function of the assembly.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 3 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
mits torque sun gear to -row gear bly.	Tooth fracture.	Secondary damage to other gear teeth due to re-distribution of stresses. Secondary damage to other gear teeth and rollers due to chip contamination. Complete loss of teeth would cause loss of function of the RGD assembly.	Chip detector indicator and possible vibration/noise.	Negligible probability that complete loss of teeth could occur prior to detection of failure.
	Scuffing of teeth.	Secondary damage to lower gear teeth and rollers due to metal contamination.	Teardown inspection at overhaul.	Conservative design minimizes probability of occurrence.
mits torque first-row assembly to d-row gear bly.	Tooth fracture.	Secondary damage to other gear teeth due to re-distribution of stresses. Secondary damage to other gear teeth and rollers due to chip contamination. Complete loss of teeth would cause loss of function of the RGD assembly.	Chip detector indicator and possible vibration/noise.	Negligible probability that complete loss of teeth could occur prior to detection of failure.

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE N UPON ROLLER GEAR
06351-11182-105 (continued)				Scuffing of teeth.	Secondary damage to gear teeth and roller due to metal contamination.
06351-11182-103	First-Row Pinion Rollers	14	Reacts sun/first-row gear mesh radial loads.	Spalling/pitting. Excessive wear of the roller surfaces. Crack in roller or weld.	Secondary damage to gear teeth and roller due to metal contamination. Secondary damage to gear teeth and roller due to metal contamination. Secondary damage to components due to chip contamination.
06351-11278-041	Second-Row Gear Assembly	7	Transmits torque from first-row pinion gear to ring gear	Crack in gear assembly (excluding gear teeth). Fretting between gear web and gear/flange assemblies. Dislodged pin (TL100-5-3) due to fracture or to backing off of unit (TLN-1001-5).	Propagation to fracture could cause loss of function of RGD assembly. None. Secondary damage to gearbox components chip contamination.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 4 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
sun/first-row gear mesh loads. mits torque first-row gear to gear	Scuffing of teeth.	Secondary damage to other gear teeth and rollers due to metal contamination.	Teardown inspection at overhaul.	Conservative design minimizes probability of occurrence.
	Spalling/pitting.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator.	
	Excessive wear of the roller surfaces.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator or teardown inspection at overhaul.	
	Crack in roller or weld.	Secondary damage to other components due to chip contamination.	Teardown inspection at overhaul.	Negligible probability, gear outside diameter would react load on root diameter of sun gear.
	Crack in gear assembly (excluding gear teeth).	Propagation to fracture could cause loss of function of RGD assembly.	Teardown inspection at overhaul. Loss of MGB output.	
	Fretting between gear web and gear/flange assemblies.	None.	Teardown inspection at overhaul.	
	Dislodged pin (TL100-5-3) due to fracture or to backing off of unit (TLN-1001-5).	Secondary damage to other gearbox components due to chip contamination.	Chip detector indicator.	Improper torquing of units could cause excessive load on pins.

TABLE - F 1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GE

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER GE
RG351-11278-041 (continued)					Loss of both will in loss of funct gear and possible quent damage to bution of stresses
RG351-11278-045	Second-Row Gear/Roller Assembly	14	Supports gears and roller, transmits torque to ring gear and reacts second- row gear/first-row pinion radial load.	Crack in gear assembly (excluding gear teeth).	Propagation to fr could cause loss tion of RGD assem
RG351-11278-101	Second-Row Pinion Gear	14	Transmits torque from second-row gear assembly to ring gear	Tooth fracture.	Secondary damage gear teeth due to distribution of s
					Secondary damage gear teeth and ro due to chip conta
					Complete loss of would cause loss tion of RGD assem
				Scuffing of teeth.	Secondary damage gear teeth and ro due to metal conta tion.
RG351-11278-102	Second-Row Gear Rollers	14	Reacts radial load.	Spalling/pitting.	Secondary damage gear teeth and ro due to metal conta tion.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 5 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
Transmits torque and transmits radial load to ring gear assembly. Second-row gear/first-row gear.	Crack in gear assembly (excluding gear teeth).	Loss of both will result in loss of function of gear and possible subsequent damage to redistribution of stresses.	Chip detector and/or teardown inspection at overhaul. Loss of MGB output.	Free-floating gear could result in catastrophic failure prior to chip detection.
	Tooth fracture.	Propagation to fracture could cause loss of function of RGD assembly.		
		Secondary damage to other gear teeth due to redistribution of stresses.	Chip detector indicator and possible vibration/noise.	
		Secondary damage to other gear teeth and rollers due to chip contamination.		
Transmits torque and transmits radial load to ring gear assembly. Second-row gear/first-row gear.		Complete loss of teeth would cause loss of function of RGD assembly.		Negligible probability that complete loss of teeth could occur prior to detection of failure.
	Scuffing of teeth.	Secondary damage to other gear teeth and rollers due to metal contamination.	Teardown inspection at overhaul.	Conservative design minimizes probability of occurrence.
	Spalling/pitting.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator.	

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER GEAR
RG351-11278-102 (continued)				Excessive wear of the roller surface.	Secondary damage to gear teeth and roll due to metal contamination.
RG351-11278-103	Second-Row Gear (See also: RG351-11181-044)	7	Transmits torque from first-row pinion to second-row gear assembly.	Tooth fracture. Scuffing of teeth.	Secondary damage to gear teeth due to distribution of stress. Secondary damage to gear teeth and roll due to chip contamination. Complete loss of teeth would cause loss of function of the RGD assembly. Secondary damage to gear teeth and roll due to metal contamination.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 6 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTONS	REMARKS
mits torque first-row n to second- gear assembly.	Excessive wear of the roller surface.	Secondary damage to other gear teeth and rollers due to metal contamination.	Chip detector indicator or teardown inspection at overhaul.	
	Tooth fracture.	Secondary damage to other gear teeth due to redistribution of stresses.	Chip detector indicator and possible vibration/noise.	
		Secondary damage to other gear teeth and rollers due to chip contamination.		
		Complete loss of teeth would cause loss of function of the RGD assembly.		Negligible probability that complete loss of teeth could occur prior to detection of failure.
	Scuffing of teeth.	Secondary damage to lower gear teeth and rollers due to metal contamination.	Teardown inspection at overhaul.	Conservative design minimizes probability of occurrence.

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER GEAR
22313VAG	Spherical Bearing	7	Reacts radial and axial loads; supports gear assemblies.	Spalling/pitting. Brinelling. Creep/spinning of inner race. Creep/spinning of outer race.	Secondary damage due to metal contamination. None. Secondary damage due to fretting of pinion bearing inner race wear of bearing/unfaced faces. Secondary damage due to fretting of second pinion bore/bearing diameter and/or wear of spacer/bearing end.
RG351-11177-101	Pinion Shaft	7	Supports second-row spherical bearing and gear assembly.	Crack in shaft.	Propagation to fracture could cause loss of function of RGD assembly due to loss of function of second-row gear assembly.
RG351-11176/11187	Plate Assemblies	3	Anchors pinion and gear assembly to gearbox housing.	Plate assemblies fail to anchor pinion shaft to housing due to sheared bolt, fractured plate or fractured spline. Spline fretting.	Loss of function of assembly due to loss of function of second-row gear assembly. None

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 7 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
radial and loads; s gear ies.	Spalling/pitting.	Secondary damage due to metal contamination.	Chip detector indicator and possible vibration/noise.	
	Brinelling.	None.	Teardown inspection at overhaul.	
	Creep/spinning of inner race.	Secondary damage due to fretting of pinion shaft/bearing inner race and/or wear of bearing/unit faces.	Teardown inspection at overhaul.	
	Creep/spinning of outer race.	Secondary damage due to fretting of second-row pinion bore/bearing outer diameter and/or wear of spacer/bearing end face.	Teardown inspection at overhaul.	
ts second-row cal bearing r assembly.	Crack in shaft.	Propagation to fracture could cause loss of function of RGD assembly due to loss of function of second-row gear assembly.	Teardown inspection at overhaul or loss of MGB output.	Multiple fasteners preclude loss of RGD output.
pinion and sembly to housing.	Plate assemblies fail to anchor pinion shaft to housing due to sheared bolt, fractured plate or fractured spline.	Loss of function of RGD assembly due to loss of function of second-row gear assembly.	Chip detector indicator and/or loss of power.	
	Spline fretting.	None	Teardown inspection at overhaul.	

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER GEAR
RG351-11173/ 11174	Second-Row Pinion Nut & Lock Plate	7	Retains bearing on pinion shaft.	Nut and lock plate fails to retain bearing on pinion shaft.	Second-row bearing gear assemblies be displaced vertical Secondary damage to gearbox components to displacement of bearing and gear assemblies.
RG351-11184-041	Ring Gear Assembly (See also: RG351-11184- 101 RG351-11184- 102)	1	Transmits torque from second-stage pinion to output flange assembly.	Crack in gear assembly (excluding gear teeth). Fretting between flange of gears -101 and -102. Dislodged bolt (TL100-53) due to fracture or to backing off of nut (MS21042-5).	Propagation to fail could cause loss of tion of the RGD as None. Secondary damage to gearbox components chip contamination Loss of all bolts result in separation gears -101 and -102 flange assembly (RG351-11185) caus loss of function of assembly.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 8 of 10)

FAILURE MODE	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
Improper torque of second-stage to output assembly.	bearing on pinion shaft.	Nut and lock plate fails to retain bearing on pinion shaft.	Second-row bearing and gear assemblies become displaced vertically. Secondary damage to other gearbox components due to displacement of bearing and gear assemblies.	Chip detector indicator and possible vibration/noise.	
		Crack in gear assembly (excluding gear teeth).	Propagation to failure could cause loss of function of the RGD assembly.	Teardown inspection at overhaul or loss of MGB output.	
		Fretting between flange of gears -101 and -102.	None.	Teardown inspection at overhaul.	
		Dislodged bolt (TL100-53) due to fracture or to backing off of nut (MS21042-5).	Secondary damage to other gearbox components due to chip contamination. Loss of all bolts will result in separation of gears -101 and -102 and flange assembly (RG351-11185) causing loss of function of RGD assembly.	Chip detector indicator or loss of MGB output.	Improper torquing of nuts could cause excessive load on bolts.

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GE

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER GE
RG351-11184-101/ -102	Ring Gear (See also: RG351-11184- 041)	2	Transmits torque from second-stage pinion to output flange assembly.	Tooth fracture.	Secondary damage gear teeth due to distribution of s
					Secondary damage gear teeth and rc due to chip conta
					Complete loss of would cause loss tion of the RGD a
				Scuffing of teeth.	Secondary damage gear teeth and rc due to metal condi tion.
RG351-11185-101	Output Flange	1	Transmits torque from ring gear to output plate.	Crack in flange.	Propagation to fi could cause loss tion of the RGD a
RG351-11185-102	Output Plate	1	Transmits torque from output flange to output hub.	Crack in plate.	Propagation to fi could cause loss tion of the RGD a
				Fretting between flange -101 and plate -102.	None.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 9 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
mits torque second-stage to output assembly.	Tooth fracture.	Secondary damage to other gear teeth due to re-distribution of stresses. Secondary damage to other gear teeth and rollers due to chip contamination. Complete loss of teeth would cause loss of function of the RGD assembly.	Chip detector indicator and possible vibration/noise.	Negligible probability that complete loss of teeth could occur prior to detection of failure.
	Scuffing of teeth.	Secondary damage to other gear teeth and rollers due to metal contamination.	Teardown inspection at overhaul.	Conservative design minimizes probability of occurrence.
	Crack in flange.	Propagation to fracture could cause loss of function of the RGD assembly.	Teardown inspection at overhaul. Loss of MGB output.	
	Crack in plate.	Propagation to fracture could cause loss of function of the RGD assembly.	Teardown inspection at overhaul. Loss of MGB output.	
mits torque ing gear to plate.				
mits torque output flange put hub.	Fretting between flange -101 and plate -102.	None.	Teardown inspection at overhaul.	

TABLE - F-1 DESIGN FAILURE MODE AND EFFECTS ANALYSIS: ROLLER C

PART NUMBER	COMPONENT	QTY	FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE UPON ROLLER C
RG351-11185-102 (continued)				Dislodged pin (HL20-8-4) due to fracture or to backing off of collar (HL86-8).	Secondary damage gearbox component chip contamination Loss of all pin result in separation plate -102 from -101, causing loss of function of RGD
RG351-11185-103	Output Hub	1	Transmits torque from output flange to rotor shaft.	Crack in hub. Dislodged bolt (NAS624-7) due to fracture or to backing off of nut (MS21042-4). Spline fretting. Fretting between plate -102 and hub -103.	Propagation to could cause loss of the RGD Secondary damage gearbox component chip contamination Loss of all bolt result in separation hub -103 from plate causing loss of RGD assembly None. None.

FAILURE MODE AND EFFECTS ANALYSIS: ROLLER GEAR DRIVE ASSEMBLY (SHEET 10 of 10)

FUNCTION	POSSIBLE FAILURE MODES IN ANTICIPATED ENVIRONMENT	EFFECTS OF FAILURE MODE UPON ROLLER GEAR DRIVE	METHOD OF DETECTION AND PROBABLE SYMPTOMS	REMARKS
Its torque output flange or shaft.	Dislodged pin (HL20-8-4) due to fracture or to backing off of collar (HL86-8).	Secondary damage to other gearbox components due to chip contamination. Loss of all pins will result in separation of plate -102 from flange -101, causing loss of function of RGD assembly.	Chip detector indicator or loss of MGB output.	Improper torquing of nuts could cause excessive load on bolts.
	Crack in hub.	Propagation to fracture could cause loss of function of the RGD assembly.	Teardown inspection at overhaul. Loss of MGB output.	
	Dislodged bolt (NAS624-7) due to fracture or to backing off of nut (MS21042-4).	Secondary damage to other gearbox components due to chip contamination. Loss of all bolts will result in separation of hub -103 from plate -102, causing loss of function of RGD assembly.	Chip detector indicator. Loss of MGB output.	
	Spline fretting.	None.	Teardown inspection at overhaul.	
	Fretting between plate -102 and hub -103.	None.	Teardown inspection at overhaul.	