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3000-HP ROLLER GEAR TRANSMISSION DEVELOPMENT PROGRAM
Volume I - Summary Report

Sikorsky Aircraft
Div of United Technologies Corp.
Stratford, Conn. 06602

January 1977

Final Report

Approved for public release;
distribution unlimited.

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 first of six volumes of the final report under this contract, and presents a summary of the work conducted in the total program. The objective of this program is to conduct research on the feasibility of a high-reduction-ratio 3000-hp roller gear transmission, for installation in an S-61 type helicopter.

Mr. James Gomez, Jr., Propulsion Technical Area, Technology Applications Division, served as project engineer for this effort, which was initially executed by Mr. Leonard M. Bartone, Military Operations Technology Division.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report presents a summary of the 3000-HP Roller Gear Transmission Development Program. During this program a helicopter transmission employing a 19.85:1 reduction ratio roller gear unit as the final reduction stage was designed, manufactured, and subjected to extensive testing. On the basis of the results of this program, it has been

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20. ABSTRACT (Cont'd)

determined that the roller gear is a viable concept for use in helicopter transmissions. Among the advantages it offers are compactness and high efficiency. The roller gear unit has the potential to offer weight savings over conventional type planetaries for reduction ratios greater than 12:1. The roller gear unit is, however, considerably more costly than conventional planetaries. Quality control of the electron-beam welds used in the fabrication of the roller gear components remains a problem. Detailed reports on the design, manufacture, bench testing, aircraft tiedown testing, and reliability and maintainability of the roller gear transmission have been published separately.

PREFACE

This report summarizes 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 M. Bartone. Mr. P. FitzGerald and Mr. L. Burroughs were the program managers at Sikorsky Aircraft.

Appreciation is extended to the entire transmission design group of Sikorsky Aircraft for their contributions to this program.

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INTRODUCTION

The designers of helicopter transmissions are continually being required to increase transmission power-to-weight ratio and efficiency, while reducing the size and cost of the power train. At the same time, reliability and maintainability standards have to be improved. In order to meet these demands, existing transmissions are continually being modified and uprated, and research is continually being conducted to develop new concepts for the transmission of power. This impetus resulted in the design of the roller gear transmission.

Sikorsky Aircraft began work on the roller gear concept in 1966 with a study of the feasibility of utilizing the roller gear drive in lieu of a two-stage planetary as the major reduction stage for the CH-54A helicopter. In this study,[(1)] it was found that a reduction stage of 9.69 to 1 was too low to fully realize the advantages of the roller gear drive. Subsequent independent research and development efforts led to the submittal of a proposal to USAAMRDL in March 1969. In June 1969, Sikorsky Aircraft was awarded a contract to build the roller gear drive for the power requirements of the Sikorsky S-61 series helicopter.

The roller gear drive evolved from a roller friction drive designed by TRW in 1961. A parametric study conducted by Dr. A. L. Nasvytis of TRW in 1965[(2)] concluded that a multi-row roller gear drive potentially offered the following substantial benefits over a conventional simple planetary:

- a) elimination of rolling element bearings in all pinions but the last row
- b) very efficient support of the pinions with the gears straddled by rollers, and
- c) ideal operation of the gears at the pitch line.

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(1) L. F. Burroughs, N. L. Chivaroli, CH-54A HIGH SPEED ROLLER GEAR TRANSMISSION FEASIBILITY STUDY, Sikorsky Engineering Report SER-64202, January 1970.

(2) Dr. A. L. Nasvytis and J. E. Bauer, PARAMETRIC STUDY ON THE ROLLER GEAR REDUCTION DRIVE, Thompson Ramo Wooldridge Inc., USAAVLABS Technical Report 64-29, U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, June 1976, AD 619294.

A 200-horsepower, 70:1 reduction ratio roller gear unit was designed, fabricated and tested for 1000 hours in a back-to-back regenerative arrangement by TRW of Cleveland, Ohio. This endurance test,[3] conducted in 1964 and 1965, confirmed that the roller gear drive was a high-efficiency unit suitable for future helicopter transmissions.

The Bell Helicopter Company of Fort Worth, Texas, conducted an engineering design investigation in 1968/1969 to determine the feasibility of employing the roller gear concept in a transmission for the UH-1 helicopter.[4] This study showed that in the areas of efficiency and reliability, the roller gear was potentially superior to the existing UH-1 transmission and a new three-stage planetary 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 subsequently fabricated an 1100 horsepower, 34.8:1 reduction ratio roller gear drive for the UH-1 helicopter transmission.[5] Efficiency of the unit, as determined by test, was 98.9 percent, which is high for a unit of such large reduction ratio. After 76.5 hours, testing was halted due to a test rig malfunction and minor design deficiencies. It was determined that future designs would require attention to the method of joining the rollers to the gears and also to the width of the roller end flanges. It was recommended that a full-sized unit be designed, fabricated and tested to determine the technical feasibility of using the roller gear drive in a helicopter.

This report summarizes the total program effort conducted by Sikorsky Aircraft to develop and incorporate a roller gear reduction drive unit into a helicopter transmission. This effort encompassed the design, fabrication and development tests of a two-row, 19.848:1 reduction ratio roller gear unit. This unit was incorporated as the final reduction stage in a 93.4:1 ratio helicopter transmission that was adapted to a modified Sikorsky Aircraft S-61 helicopter.

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(3) ENDURANCE TEST OF AN-1 ROLLER GEAR DRIVE, Thompson Ramo Woodridge Inc., USAAVLABS Technical Report 65-31, U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, August 1965, AD 471437.

(4) C. W. Bowen, C. E. Braddock, and R. D. Walker, INSTALLATION OF A HIGH-REDUCTION-RATIO TRANSMISSION IN THE UH-1 HELICOPTER, Bell Helicopter Co., USAAVLABS Technical Report 68-57, U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, May 1969, AD 855747.

(5) A. L. Nasvytis, and J. H. Hemlein, 1100-HP ROLLER GEAR DRIVE, TRW Mechanical Products Div., USAAVLABS Technical Report 70-3, Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, January 1970, AD 867795.

ROLLER GEAR DESIGN AND MANUFACTURE

ROLLER GEAR DRIVE CONCEPT

The roller gear drive is a combination of the pure roller transmission, which transmits power through friction in a planetary arrangement of preloaded rollers, and a conventional geared planetary or epicyclic gear train in a compound arrangement. The rollers, which are integral with and located on either side of the gear member, have outside diameters coincident with the gear pitch diameter. In addition to providing support (in place of bearings) for the gear members, they also contribute to the driving power as in the pure roller drive. A model illustrating the roller gear concept is shown in Figure 1. In this model, the relationship between the rollers (which form the support for the gears) and the geared portion of the roller gear drive system is shown. As a component of a helicopter main transmission, the roller gear drive is used in an epicyclic gear reduction in either a planetary or star system arrangement.

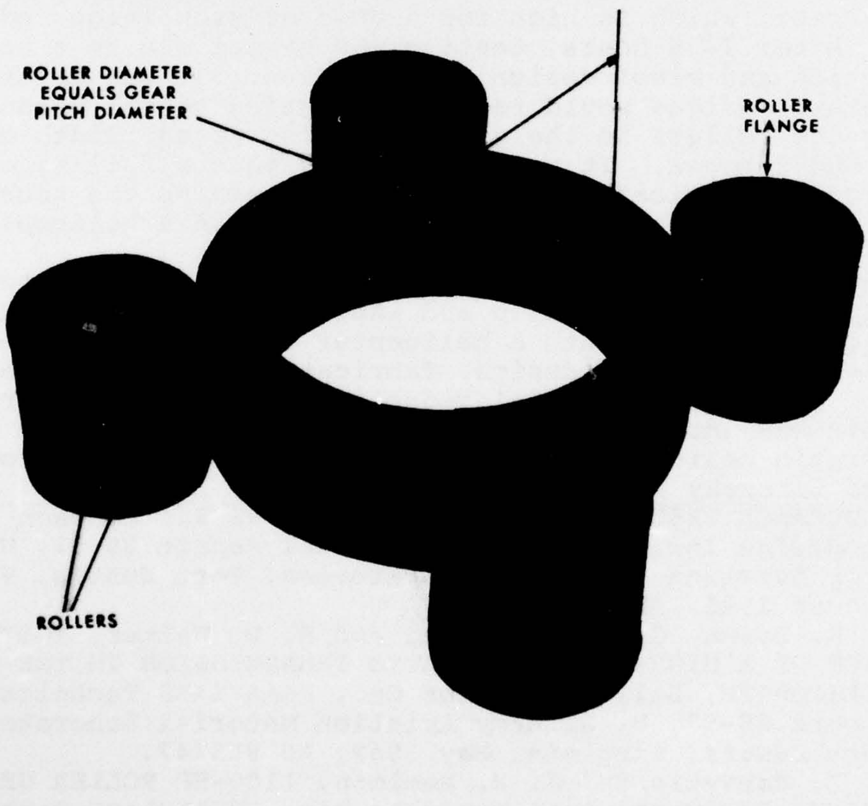


Figure 1. Roller Gear Component Concept.

The roller gear drive employs the same basic principle as that of a pure friction drive. A two-row compound planetary roller drive is shown in Figure 2. This friction drive illustrates the principle of a roller gear drive wherein the first-row rollers are supported at one inner point by the sun and at two outer points by the second-row rollers. The rollers serve the dual function of driving and supporting. The roller gear drive is merely a friction drive with gear teeth in place of the central portion of the roller. Whereas in the pure roller drive, torque is transmitted by friction; in the roller gear drive, torque is transmitted by involute gears, and rollers are used to position the components and provide kinematic stability.

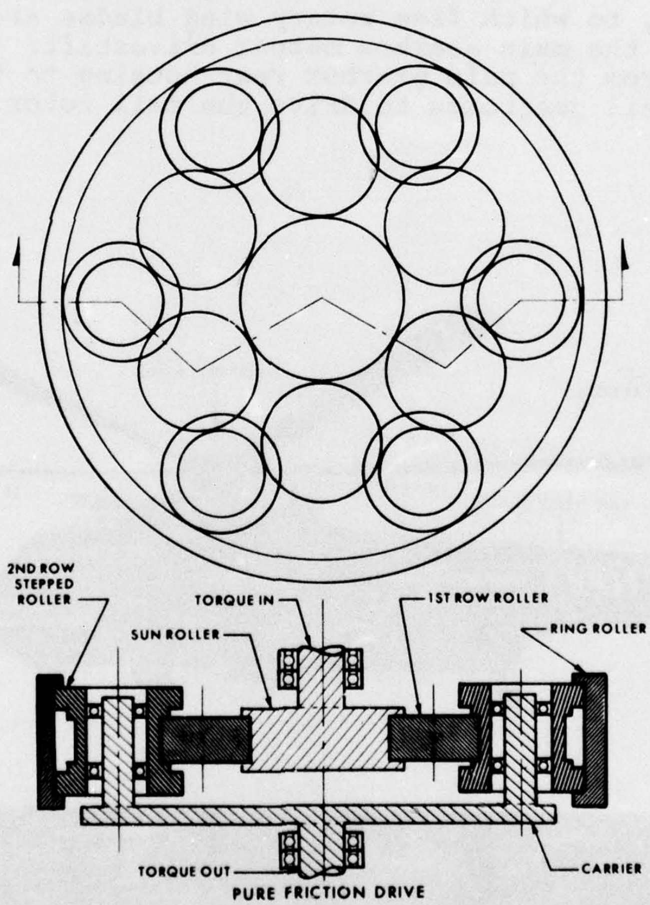


Figure 2. Pure Roller Friction Compound Planetary.

BASELINE S-61 HELICOPTER

In order to provide a realistic evaluation of the roller gear drive in a helicopter environment, a roller gear transmission was designed for a growth version of the S-61 series helicopter.

The design of the roller gear drive transmission was therefore dictated by the physical geometry and characteristics of the S-61 helicopter. This helicopter is a twin-turbine powered aircraft with a torque compensating tail rotor. The basic S-61 helicopter, Figure 3, has a gross weight of 21,000 pounds and is powered by two General Electric T58 engines, each developing 1250 horsepower at the 30-minute rating. The turbine engines are mounted side-by-side. Engine driveshafts transmit power directly into the aft positioned main gearbox. The rotary wing assembly, to which five rotary wing blades are attached, is splined to the main gearbox output driveshaft. Shafting extends aft from the main gearbox rear housing to the intermediate and tail gearboxes to drive the tail rotor.

Figure 3. Sikorsky S-61 Helicopter.

The main gearbox of the S-61 helicopter has four reduction stages, Figure 4. The first stage is a spur gear reduction which drives through a freewheel unit to a second-stage combining helical gear mesh. The third stage is a spiral bevel mesh with the driven gear concentric with the main rotor shaft. The driven gear of the spiral bevel mesh drives the tail takeoff as well as the fourth reduction stage, a single-stage planetary. The planetary is of the single-row type with sun gear input, carrier output, and ring gear fixed. The carrier drives the main rotor shaft.

In general, the lightest helicopter transmission results when the highest possible reduction ratio is located in the final reduction stage. With present helicopter transmission designs, this last stage usually consists of a one- or two-stage planetary reduction unit. Conventional planetaries, as used in helicopter transmissions, however, are presently limited to a reduction ratio of about 3:1 per stage. For the roller gear drive to be potentially superior to the planetary arrangement, it should have a reduction ratio greater than 12:1. Only above this ratio is there a potential weight saving and efficiency improvement with the roller gear drive.

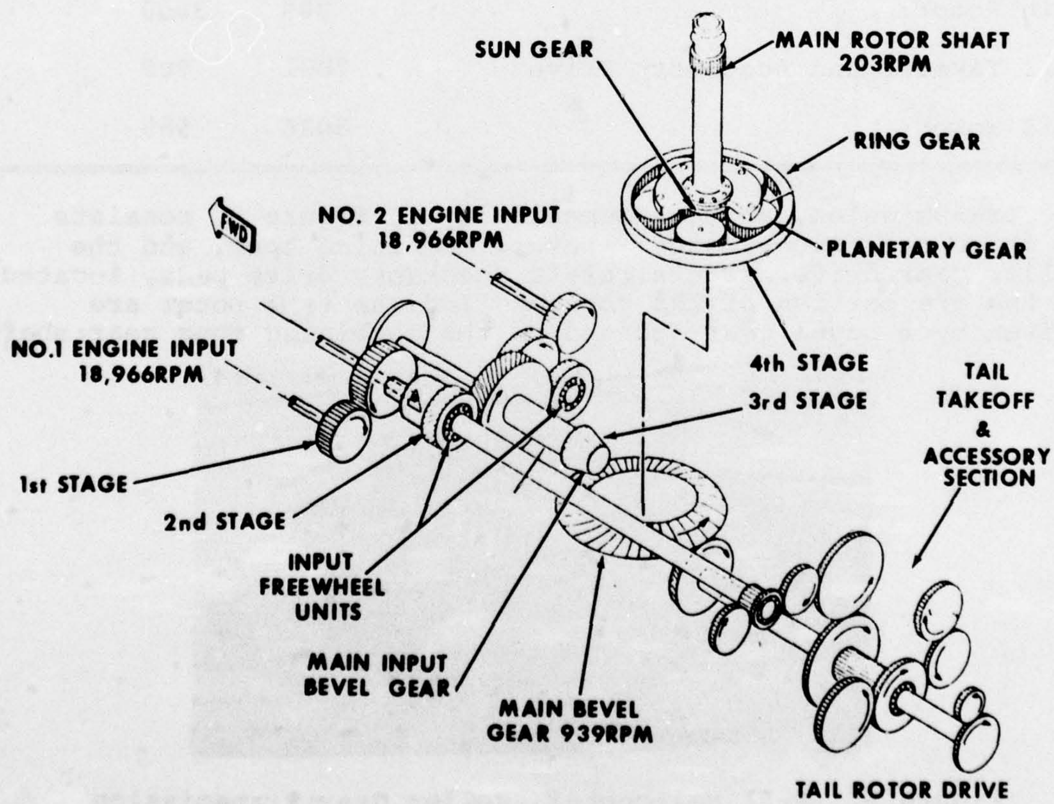


Figure 4. S-61 Main Transmission, Schematic.

ROLLER GEAR TRANSMISSION

The roller gear transmission was designed to replace the present S-61 transmission. The design criterion for the roller gear transmission was for a 27,000-lb gross weight S-61 helicopter powered by two YT58-GE-16 engines, each capable of delivering 1870 horsepower (3740 horsepower dual engine power) at 18,966 rpm, Figure 5. This required a roller gear drive transmission to transmit 3000 horsepower to the main rotary wing assembly. The design requirements for this transmission are given in Table 1.

TABLE 1. S-61 ROLLER GEAR TRANSMISSION DESIGN REQUIREMENTS.

Location	Speed (rpm)	Power (hp max)
Input Drives:		
Dual Engine	18,966	3700
Single Engine	18,966	1870
Main Rotor:	203	3000
Tail Takeoff and Accessory Drive	7031	700
Tail Rotor:	3026	565

The transmission, shown schematically in Figure 6, consists of three reduction stages: bevel, combining spur, and the roller gear drive. The aircraft accessory drive pads, located in the aft portion of the gearbox, and the tail rotor are driven by a bevel gear located on the combining spur gear shaft.

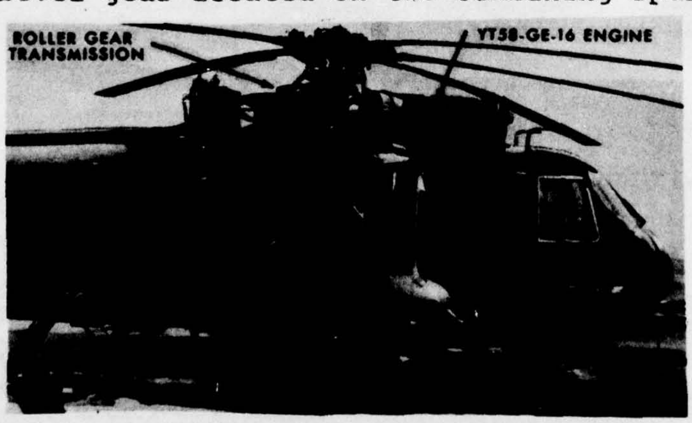


Figure 5. S-61 Helicopter, Roller Gear Transmission Arrangement.

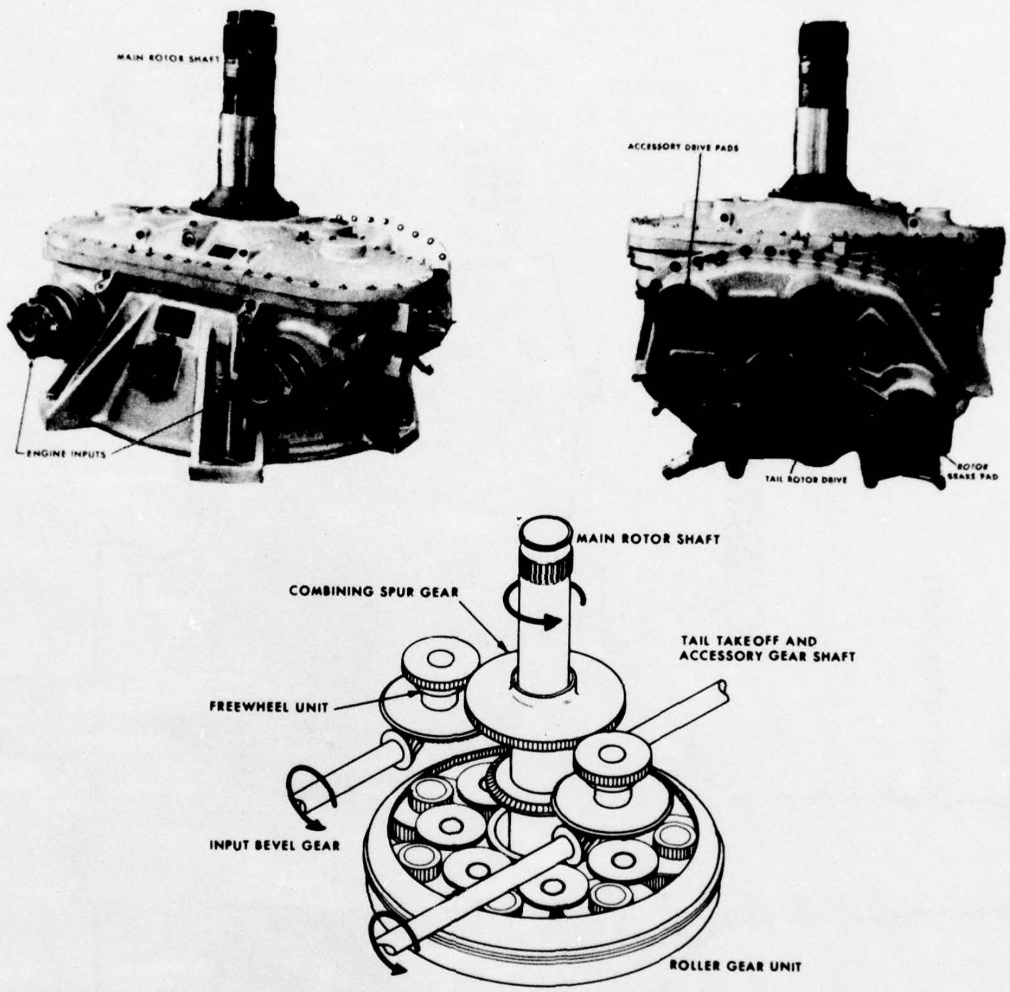


Figure 6. S-61 Roller Gear Transmission, Schematic.

Each engine transmits power at 18,966 rpm to a spiral bevel gear mesh. This first-stage spiral bevel mesh has a reduction ratio of 3.05:1, which reduces the speed at the output bevel to 6223 rpm. Concentric within each output bevel gear shaft is a ramp-roller type overrunning clutch, which permits single engine operation and also allows the rotor to overrun in the event of engine malfunction or engine shutdown. The output camshaft of the ramp roller clutch drives a second-stage combining spur gear mesh where the power from each engine is combined to a single torque path operating at 4045 rpm. The output of the second stage drives the roller gear unit and the accessory and tail drive system. The actual arrangement of these components is shown in the cross-sectional drawing, Figure 7.

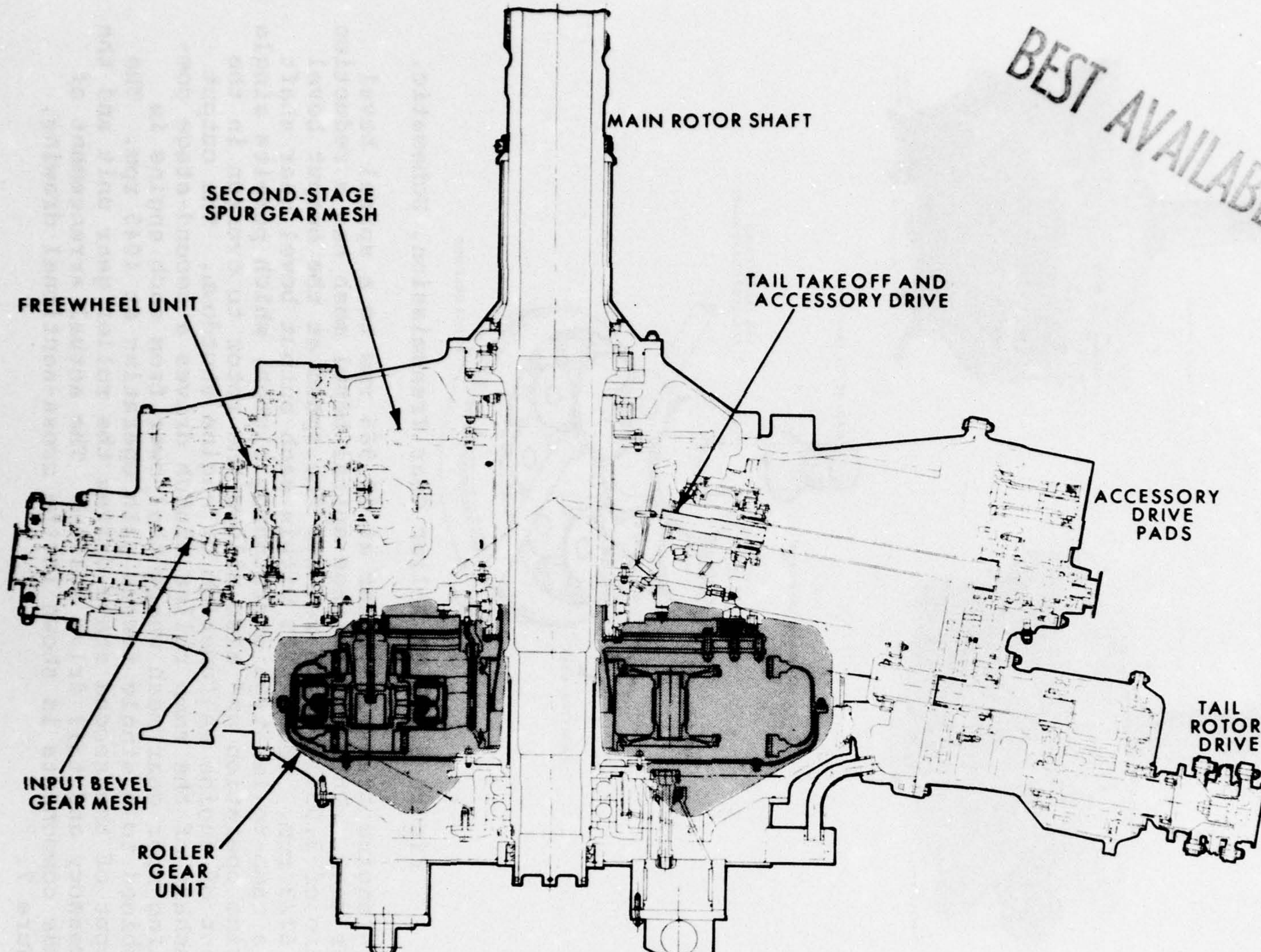


Figure 7. Roller Gear Transmission Drawing.

ROLLER GEAR DRIVE DESIGN

The final reduction stage of the roller gear transmission is the roller gear drive unit with a reduction ratio of 19.848:1. The unit consists of two rows of seven stepped pinions, wherein the second-row pinions react the transmitted torque, ring gear output, and sun gear input. A split power path at the sun gear and ring gear induces symmetrical loading for each mesh in the roller gear unit. Two rollers per mesh, whose diameter equals the gears' pitch diameters, straddle the gears of the sun, first-row pinion, and second-row pinion. The first-row pinion is positioned by the sun and second-row pinion rollers and is thus accurately located by three contact points. The second-row pinion is similarly positioned by the rollers of the first-row pinion and the separating component of the ring gear. This force ensures contact of the rollers at all times. The symmetry of the design of the rollers and the forces induced on them allows the planets to be held parallel. Flanges on the rollers provide axial location of the first- and second-row pinions. Torque is reacted on the roller gear drive unit through spherical bearings located in the second-row pinion. Power is thus transmitted by the gear teeth while kinematic stability is provided by the rollers.

This design eliminates planet bearings except in the last row, where they are necessary to transmit the reaction torque, and ensures parallel alignment of all elements within manufacturing tolerance. The roller gear drive unit has inherently more stable load-sharing characteristics than conventional planetaries due to the accurate positioning of the pinions by the rollers. Figure 8 shows the arrangement of the roller and gear members and the assembled unit. The basic gear data of the roller gear components is summarized in Table 2.

TABLE 2. 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

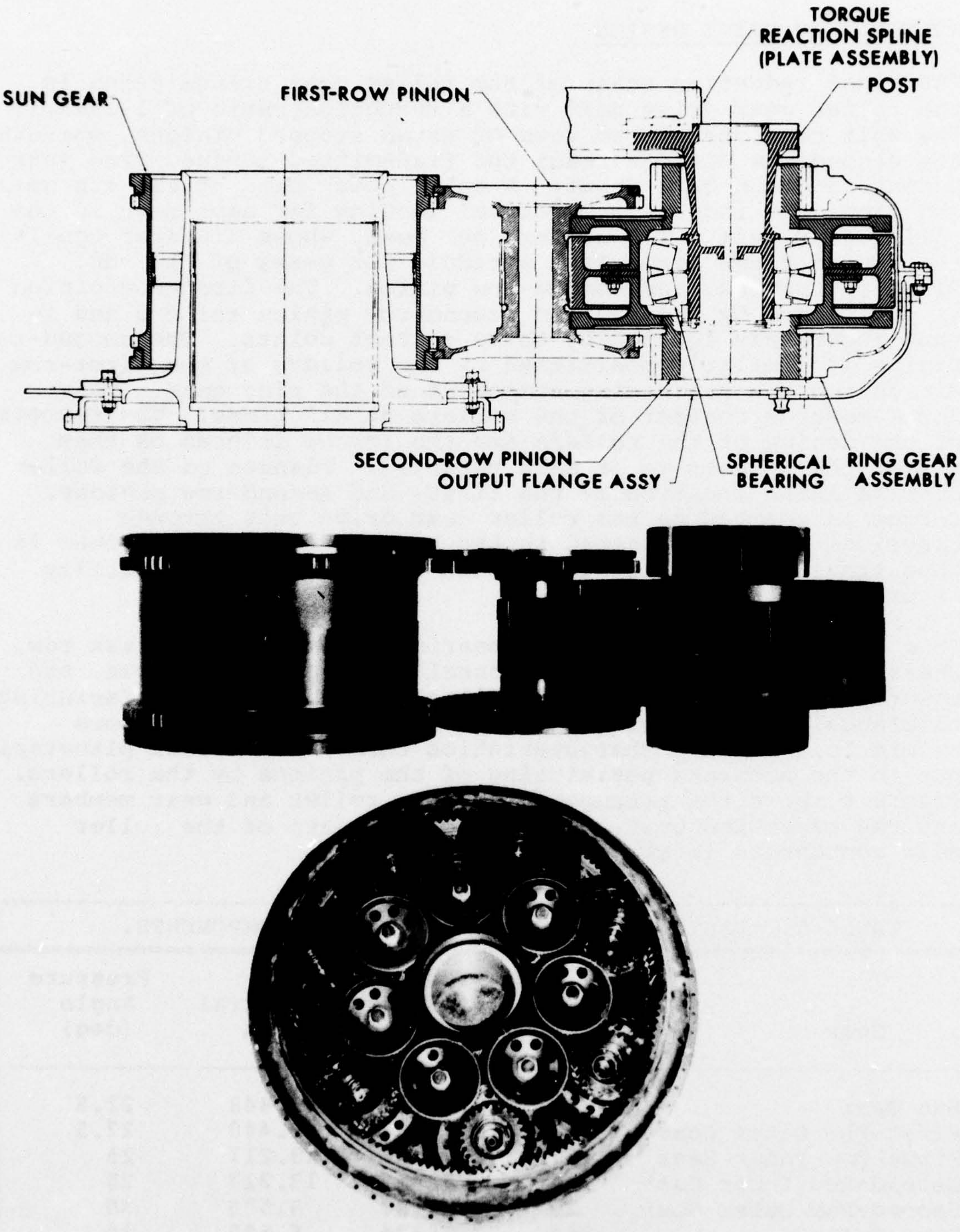


Figure 8. Roller Gear Drive: Final Reduction Stage - Roller Gear Transmission.

To ensure contact and proper location of the "free" pinions, earlier roller gear units used loading mechanisms that pre-loaded the second- and first-row pinions, holding them against one another and the sun gear. In these designs, the initial preload had to be sufficient to overcome the separating component of the normal gear force at the maximum power to be transmitted. The S-61 roller gear is self-preloading, having higher pressure angles at each successive gear mesh. When torque is applied to the unit, each component is forced radially inward due to the larger radial force on the outside of the component. This keeps all rollers loaded continuously during operation. Table 3 gives the net preload force induced by the gear tooth mesh separating component when transmitting 100% torque.

TABLE 3. GEAR PRESSURE ANGLE VS. PRELOAD FORCE.

Gear Mesh	Gear Pressure Angle (deg)	Preload Force (lb)
Sun Gear - First Row	22 1/2	2750
First-Row - Second-Row	25	4500
Second-Row - Ring Gear	30	5580

Figure 9 shows the radial roller forces as a function of transmitted torque.

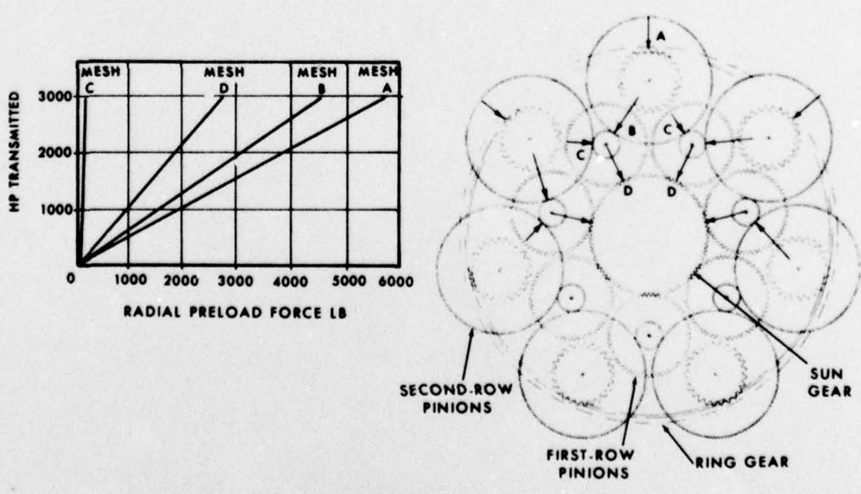


Figure 9. Gear Mesh Roller Preload Forces.

To assure equilibrium (i.e., preload) of the roller gear components, the spherical bearings within the second-row pinions were designed not to react any radial loads. This was done by providing sufficient internal clearance within the bearings that at maximum loads, no radial forces could be transmitted to the bearing posts.

The spherical bearing arrangement for the second-row pinions allows a flexible reaction post to be utilized, thereby achieving a relatively "soft mounted" roller gear unit. The free-floating sun gear, the floating first-row pinions and the ring gear provide a degree of flexibility that enhances equal load sharing among the pinions. The spherical bearing arrangement ensures parallellism of the second-row pinions when the parts are deflected by the torque reaction load.

ROLLER GEAR COMPONENT DESIGN

Proper operation of the roller gear unit depended heavily on close timing between the various elements of the compound gears. Both first- and second-row pinions of the roller gear unit consisted of matched sets wherein the drive side of the individual gear teeth were timed to within ±.0002 inch. The outer gear teeth were aligned to the same tolerance. Earlier roller gear units used press fits in the fabrication of the compound gears. This method, besides leading to bulky gear assemblies, proved to be very unreliable in the operation of the roller gear units. For increased reliability and weight savings, electron-beam welding was utilized in the fabrication of the roller gear assemblies.

Electron-beam welding, as depicted in Figure 10, is accomplished by focusing a narrow intense high-energy stream of electrons onto the joint to be welded. The kinetic energy of the electrons is converted to heat on impingement, thereby melting the material and fusing the pieces. This welding process is performed in a vacuum and produces a very clean weld. Although there are many advantages to electron-beam welding, including weld integrity and speed, an important consideration in choosing electron-beam welding for fabrication of the roller gear components is its minimal heat effect and its high depth-to-width ratio. The fusion process occurs so rapidly that very little heating occurs in the surrounding metal. This means that there is very little distortion of the welded part, a necessary condition for the close tolerances required by the roller gear components. In addition, the size of the heat-affected zone where the structure of the metal might be altered is minimized.

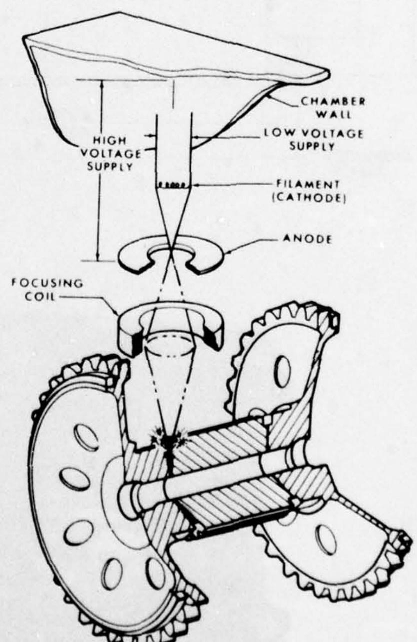


Figure 10. Electron-Beam Welding Schematic Representation.

However, the electron-beam welds proved to be the most troublesome aspect of the design of the roller gear components. These welds accounted for the majority of the problems encountered during testing and necessitated the redesign of the first- and second-row pinions. The following section discusses the design of each component and traces the evolution of the components through redesign to the final configuration.

Sun Gear

The sun gear, shown in Figure 11, consists of three elements: the shaft containing the two geared surfaces and two rollers which are electron-beam welded to the shaft after finish grinding of the gear teeth. The rollers are ground to size after welding. The sun gear changed the least of the welded components through the development of the roller gear drive. The only significant change from the original design of the sun gear was a deepening of the relief between the rollers and the gears so that the end of the weld was completely cleaned up.

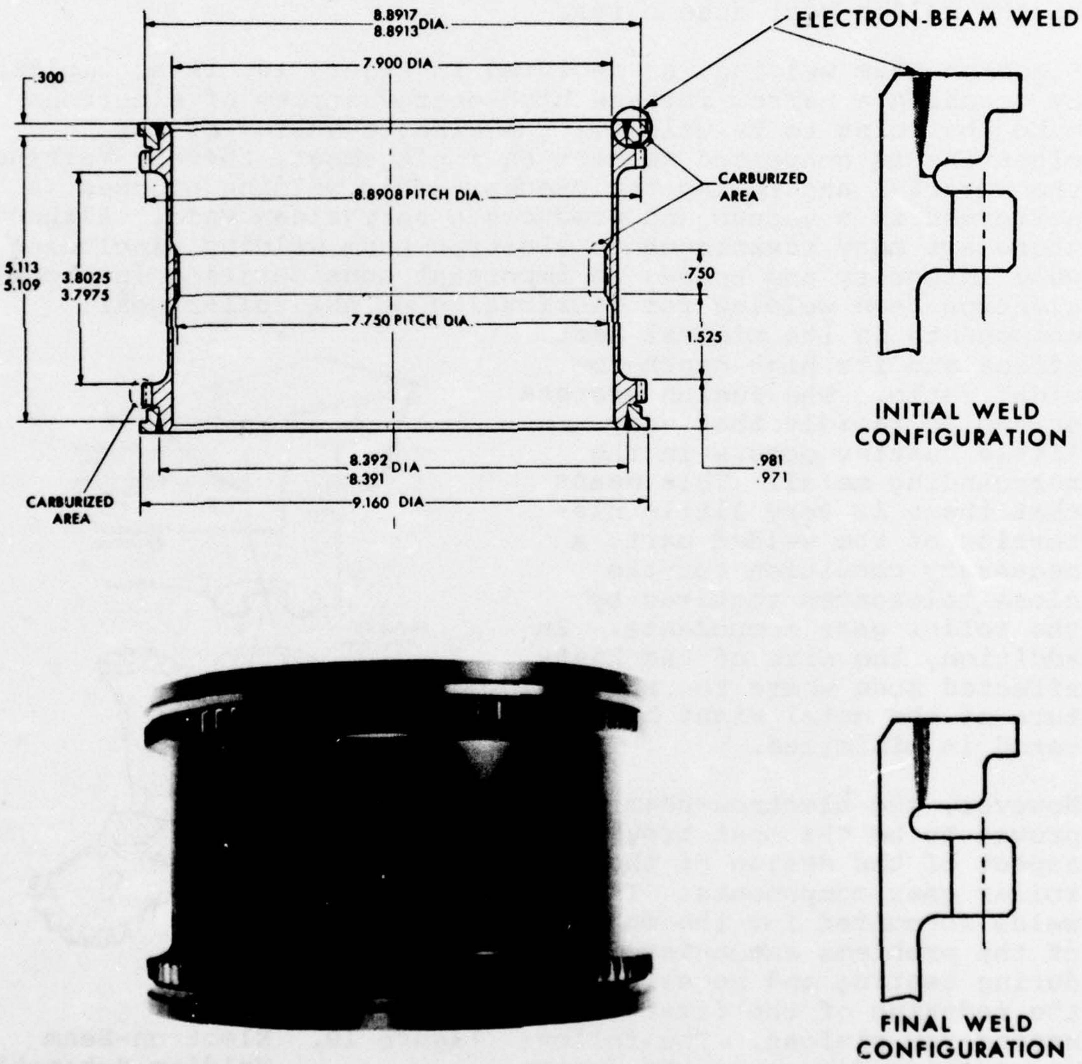


Figure 11. Sun Gear.

First-Row Pinion

These pinions were fabricated in matched sets of seven pinions per set. The outer gears were machined to a master indexed inner gear tooth to within ±.0002 inch. All seven pinions that comprised a set were required to be the same within .0004 inch. The pinion consisted of five elements. A central gear that is heat-treated is finish machined prior to electron-beam welding of the two outer gears. The outer gears, which are case-hardened prior to assembly onto the center gear, are then finish machined to the inner gear. Finally, two case-hardened rollers are electron-beam welded onto the outer gears and ground concentric to the gear pitch diameters.

The initial design of these pinions incorporated a longitudinal electron-beam weld under the small diameter roller, Figure 12. Voids at the root of the weld produced stress risers from which fatigue cracks propagated. The pinion was subsequently redesigned to incorporate the butt weld joint shown in Figure 13.

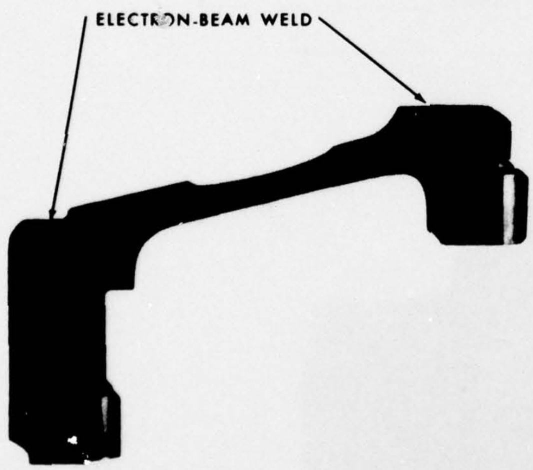


Figure 12. First-Row Pinion, Longitudinal Weld Configuration.

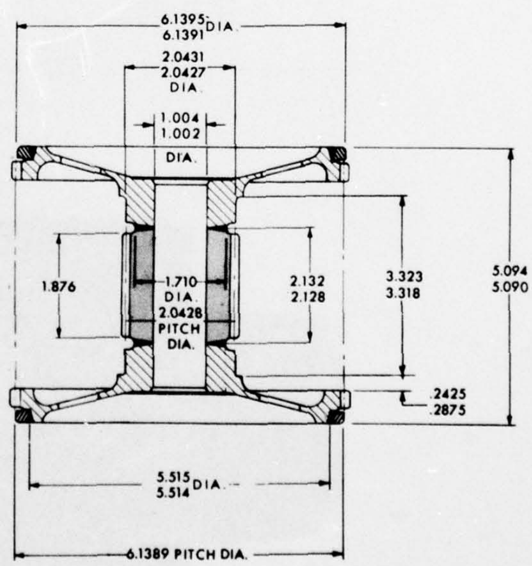


Figure 13. First-Row Pinion, Final Configuration.

Second-Row Pinion

The redesigned second-row pinion is shown in Figure 14. The spherical roller bearing, which reacts the transmitted torque, is sandwiched between the two end gear and roller assemblies. These end gears are electron-beam welded to the roller flange and fastened to the center gear by tapered bolts which align the end gears to a master index tooth on the center gear. These pinions are held to tolerances similar to those of the first-row pinions. A matched set of seven pinions is required for assembly into the roller gear unit.

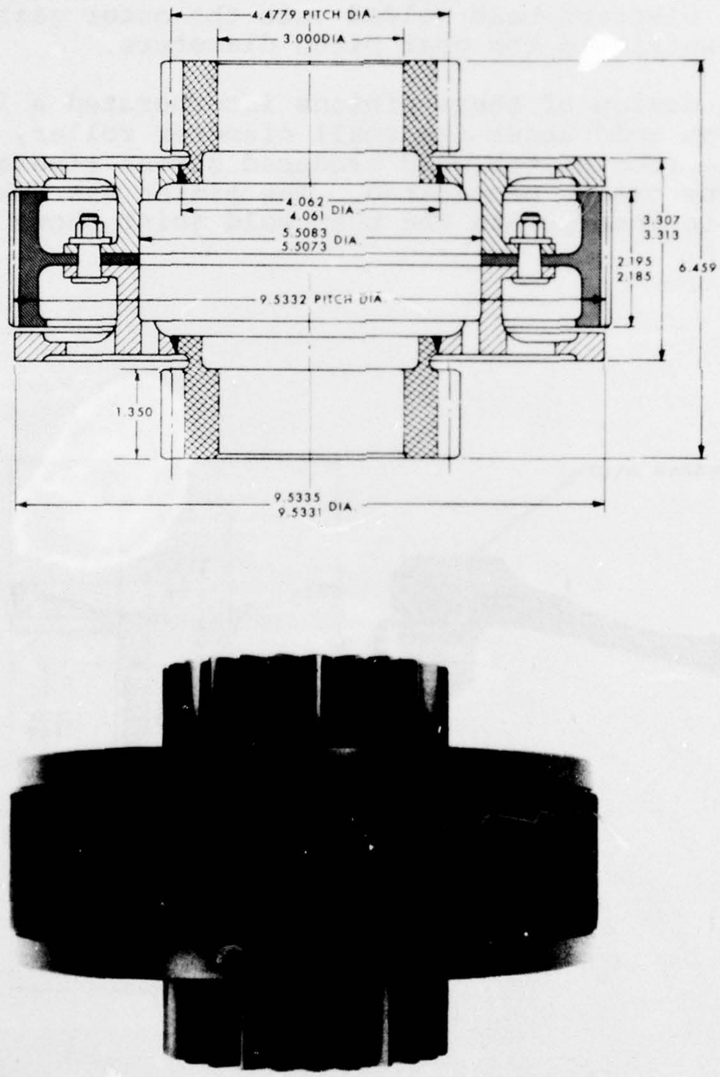


Figure 14. Second-Row Pinion, Final Configuration.

The original pinion design incorporated seven individual parts which required six electron-beam welded joints, Figure 15. Timing of the gear teeth was achieved by grinding the large diameter gear teeth relative to the small diameter gear teeth at the subassembly. The two case-hardened rollers were then welded and ground concentric to the large gear pitch diameter, whereupon the end gear and flange assembly was positioned (drilled and reamed) to line up the end gear teeth within ±.0002 inch.

A fracture of the end gear and flange assembly, Figure 16, during initial testing precipitated a minor design change to increase the size of the relief between the gear and flange. A thicker blast shield was positioned between the exit of the electron-beam weld and the gear face, thereby enabling a wider and stronger electron-beam to be utilized to ensure fusion of the joint. A second fracture involving this gear design resulted from fatigue fracture propagating from voids at the root of the weld to the inside bore, Figure 17. Secondary fracture then originated from the stress concentration notch at the roller weld due to the large leverage load from the unsupported ring gear mesh forces. This fracture resulted in the redesign of the pinion, whereby all but two electron-beam welds were eliminated.

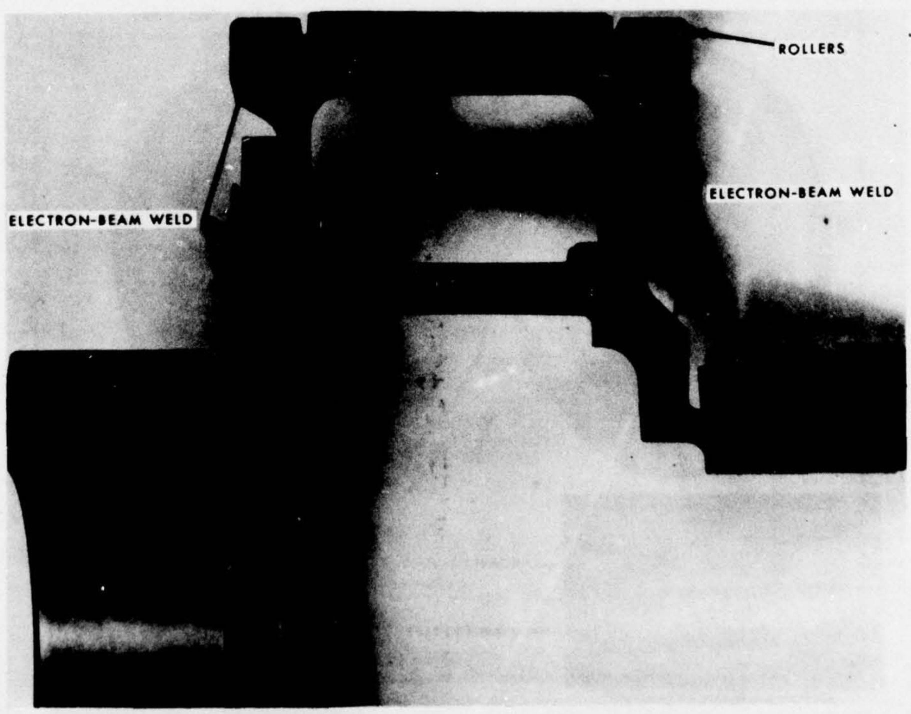


Figure 15. Second-Row Pinion, Original Configuration.

Figure 16. Gear/Flange Fracture.

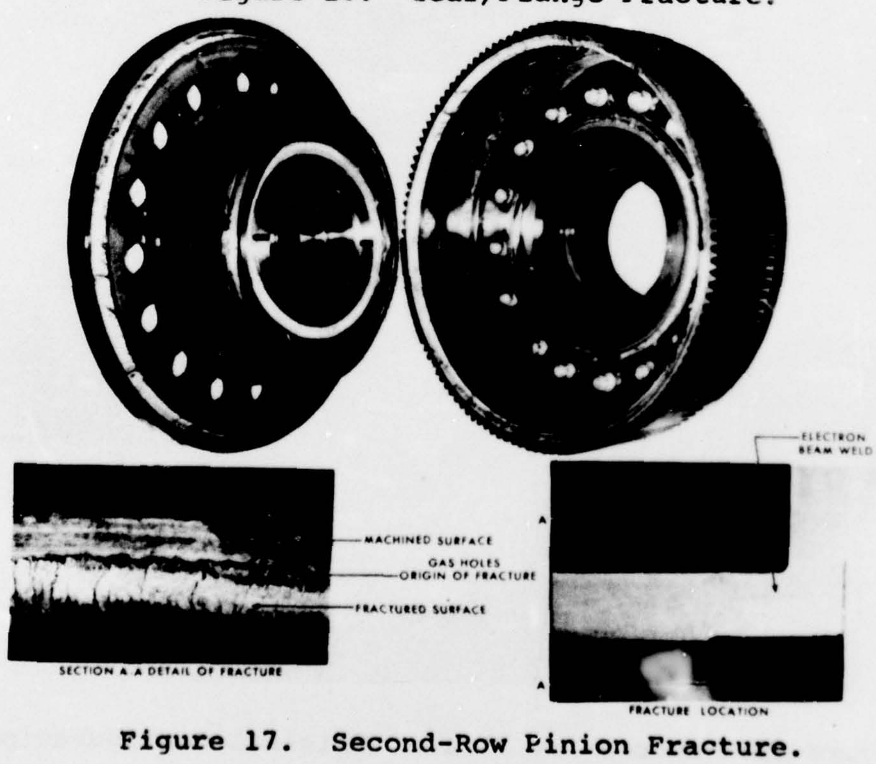


Figure 17. Second-Row Pinion Fracture.

Ring Gear

A prime consideration in the design of the ring gear, Figure 18, was sufficient flexibility to assure proper load sharing among the components of the roller gear drive. It was important that deflections of the main rotor shaft, to which the ring gear is splined, not be translated into misalignment of the ring gear itself. To this end, the webs connecting the main rotor shaft attachment to the ring gear were made as thin as possible. The gear flanges were designed for a predetermined deformation under load to enhance the load sharing among the pinions. The gear teeth were ground with the two halves of the ring gear assembled together. Alignment of the teeth of the two halves was held to within .0004 inch.

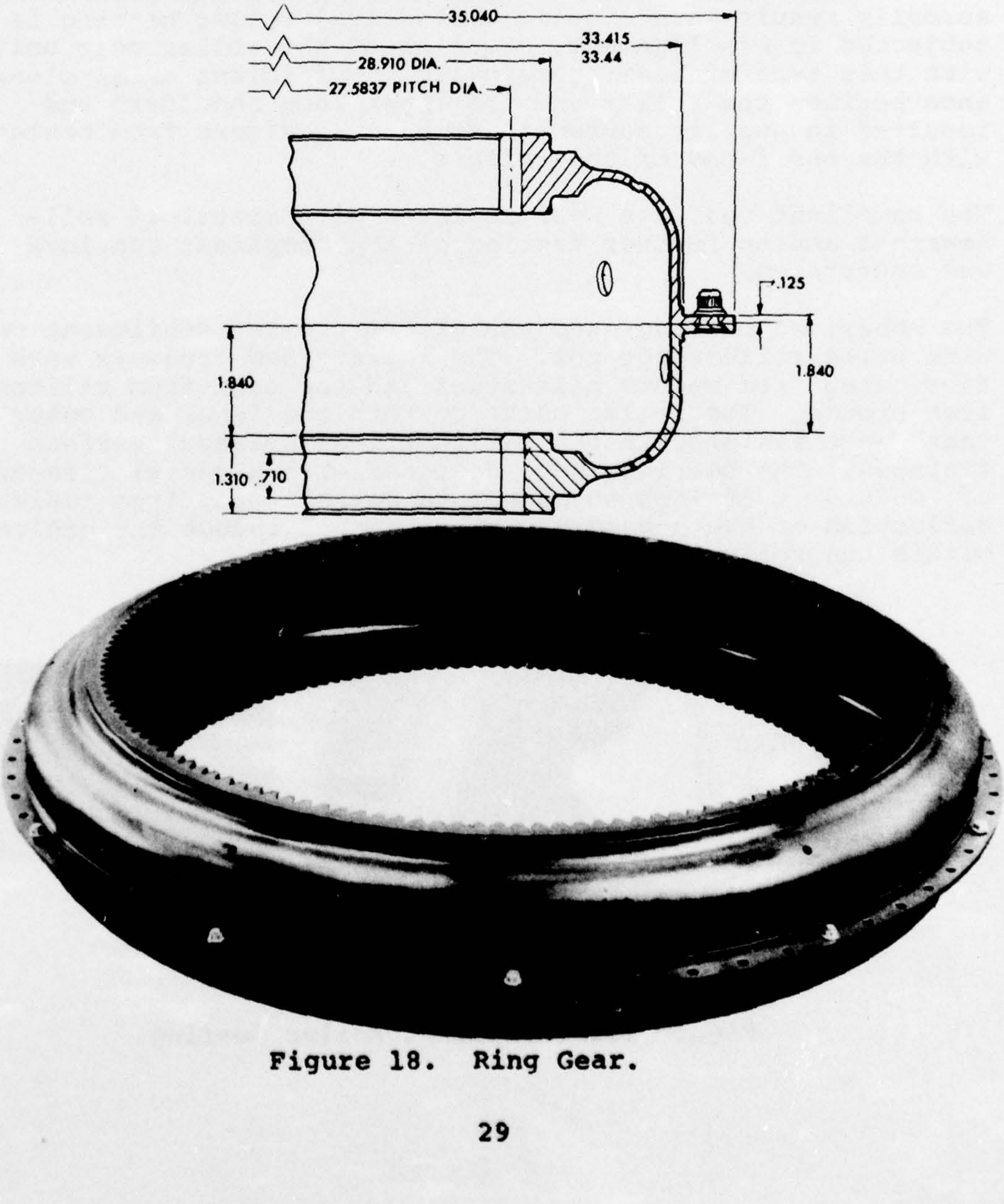


Figure 18. Ring Gear.

Second-Row Pinion Bearings

Two types of bearings were assessed during initial development testing of the roller gear units. Both types had identical envelopes and were interchangeable in the second-row pinions. The compliant type bearing, Figure 19, is an advanced state-of-the-art cylindrical type roller bearing wherein both ends of the rollers are recessed. It was designed to accommodate the calculated slope due to the deflection of the bearing posts under maximum power. The hollow ended rollers were developed with the objective of increasing the range of misalignment (i.e., slope) which a cylindrical roller bearing can accommodate. By allowing the roller ends to deform, the load is redistributed to relieve the high stresses which normally result when a solid cylindrical roller bearing is subjected to misalignment. Testing of the roller gear unit with this type of bearing revealed insufficient axial clearance between the roller and the inner race shoulders and resulted in heavily scored inner race shoulders from contact with the end faces of the rollers.

The compliant bearings were replaced with spherical roller bearings and no further testing of the compliant bearings was undertaken.

The spherical bearing used was of the two-row configuration with seven rollers per row. The rollers and raceways were fabricated from vacuum melt steel and the cage from silicon iron bronze. The roller paths on both the inner and outer races were finished to 6AA (arithmetical average) surface finishes. The bearings were designed with a radial clearance of .0076 - .0086 inch so as not to detract from free radial deflection of the second-row pinions that induce the preload within the roller gear unit.

Figure 19. Compliant Roller Bearing.

ROLLER GEAR COMPONENT MANUFACTURE

The manufacture of the roller gear drive components followed the normal high standard practices used in the aerospace industry. However, greater control was exercised in some areas to ensure equal load sharing among the roller gear components. These were the following:

- Parallelism of the gear teeth of split gears
- Indexing of the compound pinions
- Concentricity between rollers and gears
- Allowable tooth spacing errors

Typical of the tolerances to which the components were fabricated are the following:

- Basic roller diameter, ±.0001 inch
- Roller diameter deviation .0001 inch T.I.R. (total indicated reading)
- Concentricity
  - Pitch diameter of gears to each other .0005 T.I.R.
  - Roller diameters to each other, .0002 T.I.R.
  - Roller diameter to its gear, .0002 T.I.R.
- Tooth-to-tooth spacing error, .0002 inch
- Accumulated spacing error, .0008 inch
- Lead error not to exceed .0002 inch
- Involute error not to exceed ±.0003 inch.

The manufacture of the components to these tolerances required stringent shop control measures; however, it was the electron-beam welding that proved to be the most challenging. In the process of fabricating the roller gear components wherein the sun gear required two electron-beam welds, the first-row pinion four electron-beam welds and the initial design of the second-row pinions six electron-beam welds, a total of 72 welds per roller gear unit, welding problems other than the establishment of welding parameters were experienced.

Electron-beam welds are susceptible to porosity and spiking at the root of the weld. Figure 20 is an example of porosity which occurred in the initial design of the first-row pinion. Also visible is the incomplete fusion resulting from mis-alignment between the electron-beam and the seam. These weld defects caused cracks to propagate to the surface of the first-row pinion roller and resulted in extensive spalling of the roller surface. It is believed that the porosity results from the release of gases dissolved in the metal or trapped in the joint to be welded. Although welding is accomplished in a vacuum, the pilot diameter press fit that is used to position the two parts can trap gases which are released during the welding process.

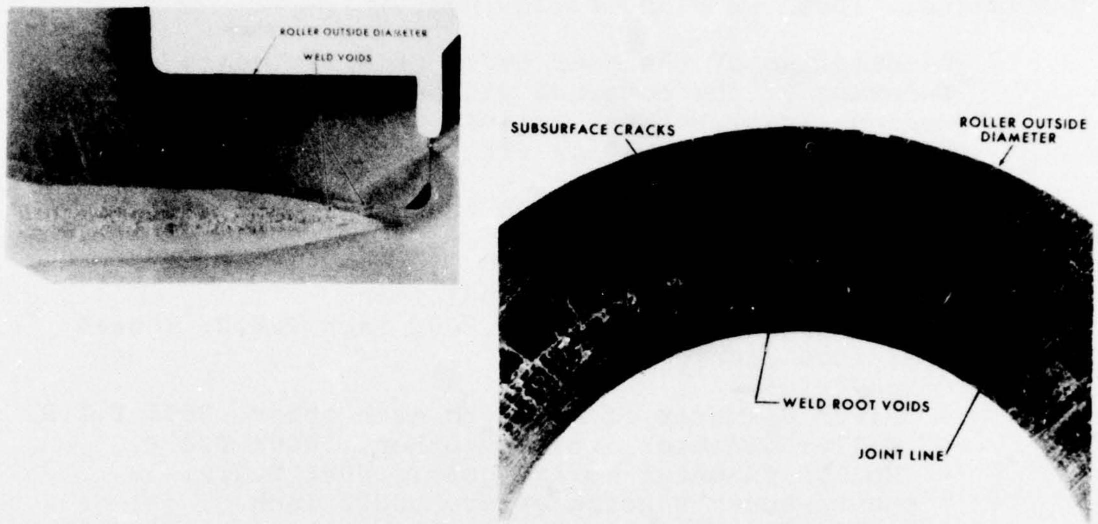


Figure 20. Weld Porosity, First-Row Pinion.

Although missed seam defects can be corrected by rewelding the area with increased power, this approach could not be utilized on this weld joint. Since the weld itself lay beneath a carburized case-hardened roller, the initial welding parameters were developed to minimize the amount of heat generated in the component to preclude tempering the hardness of the roller. An increase in the power intensity of the beam would have resulted in a "softer" roller and an undesirable change in the grain structure of the carburized case.

Incomplete fusion of the weld seam was further exemplified when a second-row pinion small diameter gear separated from the flange at the electron-beam weld joint, Figure 21. Examination revealed that although complete weld beam penetration had been achieved, fusion had not occurred where the center of the weld missed the mating surfaces. To ensure complete fusion, a more powerful welding beam was used. This required a thicker blast shield to be placed between the flange and gear to prevent metal spatter from pitting and damaging the end face of the gear.

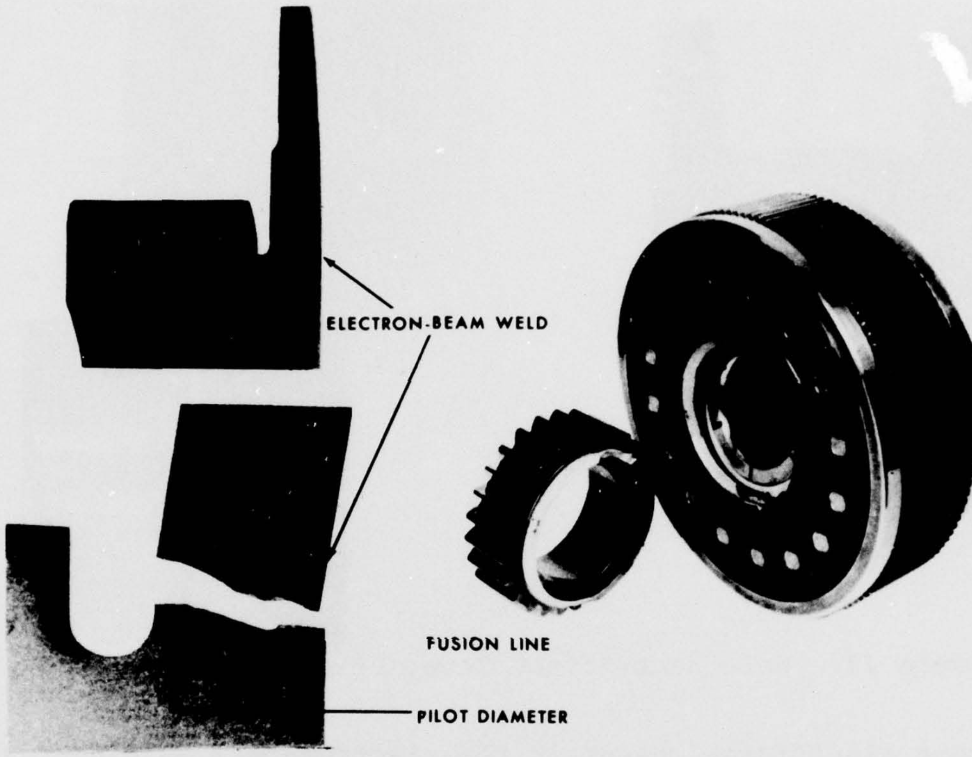


Figure 21. Incomplete Weld Fusion, Second-Row Pinion.

Another weld-related problem appeared with the fracture of gear teeth on the small diameter gear of the first-row pinion. Inspection of the fractured teeth revealed that cracking had originated near the roots of the teeth where the heat-affected zone adjacent to the weld had extended into the gear root (Figure 22). Metallurgical analysis revealed a transition interface where an area of compressive stress (carburized layer) bordered an area of tensile stress. This transition in the material from a state of tension to a state of compression led to a stress concentration at the edge of the heat-affected zone which, in turn, led to the failure of the gear teeth. This structural change occurred only where the beam overlapped the previously welded area. Existing parts were modified by increasing the groove width. In addition, the groove was shot peened to put the surface in compression.

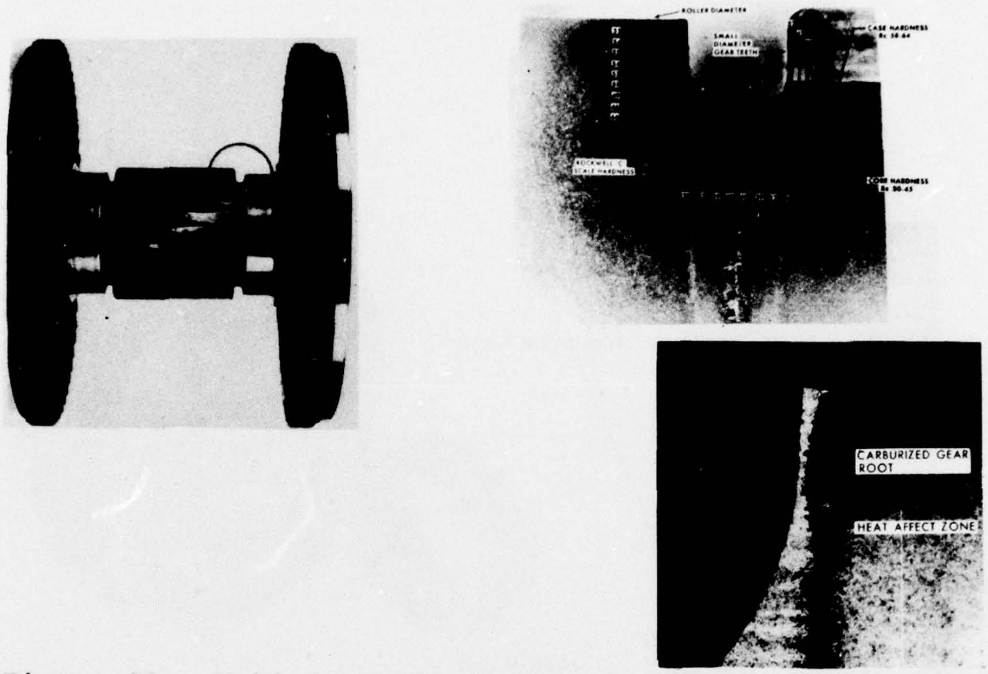


Figure 22. Weld Heat-Affect Zone, First-Row Pinions.

The most significant change in the electron-beam welding procedures resulted from cracks that appeared in the electron-beam welds of the rollers of the second-row pinions after aircraft testing of the roller gear transmission. Ultrasonic inspection of the welds showed indications of cracks that were not detected prior to the commencement of testing. Cracks were found in six of the seven second-row lower roller welds. Examination of the ultrasonic C-scans shows that degradation occurred along a line associated with weld pullout.

Investigation of the welding procedures revealed the use of identical welding schedules for both top and lower roller welds. The only difference in procedure was a 24-hour time lag which occurred between the welding and stress relieving of the lower roller.

It was concluded that the fracture that initiated from the exit side of the weld at the interface of the melt and heat-affected zones probably resulted from weakened grain boundaries and excessive residual stresses due to welding. As a result, the stress-relieving cycle was changed from 2 hours at 275°F to 5 hours at 325°F, and was to be accomplished immediately after welding.

ULTRASONIC INSPECTION

At the onset of the program, it was thought that conventional techniques could be utilized to inspect the electron-beam welds. However, magnetic particle and X-ray inspection proved to be unreliable in detecting weld defects. Magnetic particle inspection is suitable for surface or near-surface cracks, although this method failed to show the incomplete fusion of the second-row pinion gear/flange weld. This could be attributable to the difficulty in viewing the joint line below the gear root diameter, which is smeared over by subsequent machining and further hindered by the heat-affected zone marks. X-ray inspection proved ineffective in the detection of voids and missed seams due mainly to the geometry of the part which prevented proper location of the film.

The fracture of the first-row pinion small-diameter roller from voids in the weld spurred the development of ultrasonic inspection of the welds. In cooperation with Automation Industries of Danbury, Connecticut, a technique was developed in which facsimile recordings were obtained of all the welds. This method is illustrated in Figure 23. A piezoelectric search unit and the test piece are immersed in a liquid whereupon short bursts of high-frequency ultrasonic waves, generated by the search unit, are transmitted into the test piece. A discontinuity in the test piece causes an acoustic impedance mismatch which reflects some of the ultrasonic waves back to the search unit. The search unit converts this "echo" into an electrical signal, which is displayed on a cathode ray tube. Permanent displays of this process on recorder strips are known as C-scans. This technique proved to be extremely valuable in the detection of welding flaws. Perhaps the most striking example confirming the reliability of ultrasonic inspection was the recording of cracks in the second-row pinion rollers after testing of the roller gear transmission on the aircraft. Comparison of the C-scans taken before and after 50 hours of transmission testing revealed cracks in the lower roller welds. Removal of the degraded area (Figure 24) confirmed the ultrasonic inspection results.

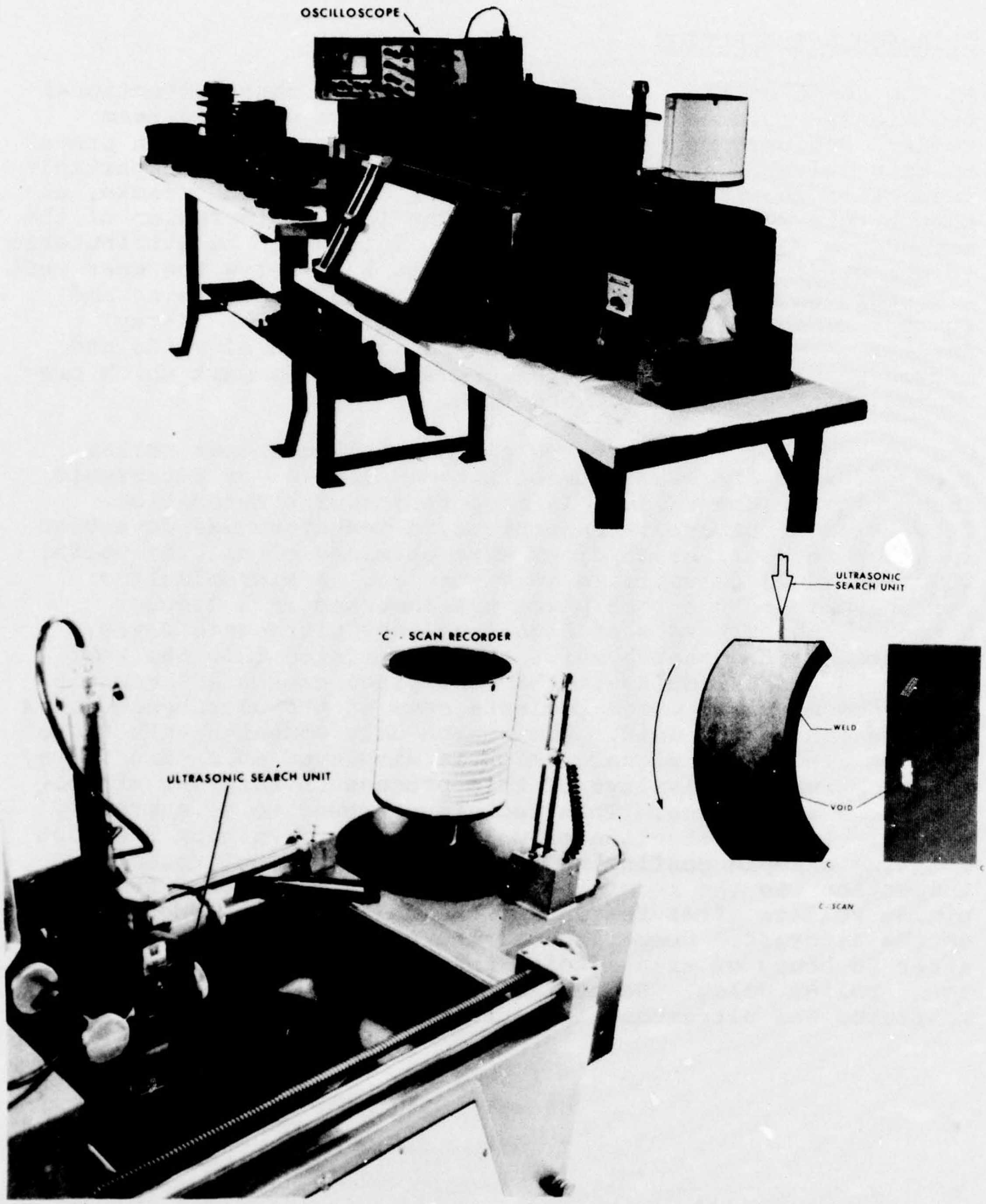


Figure 23. Ultrasonic Inspection of Electron-Beam Welds.

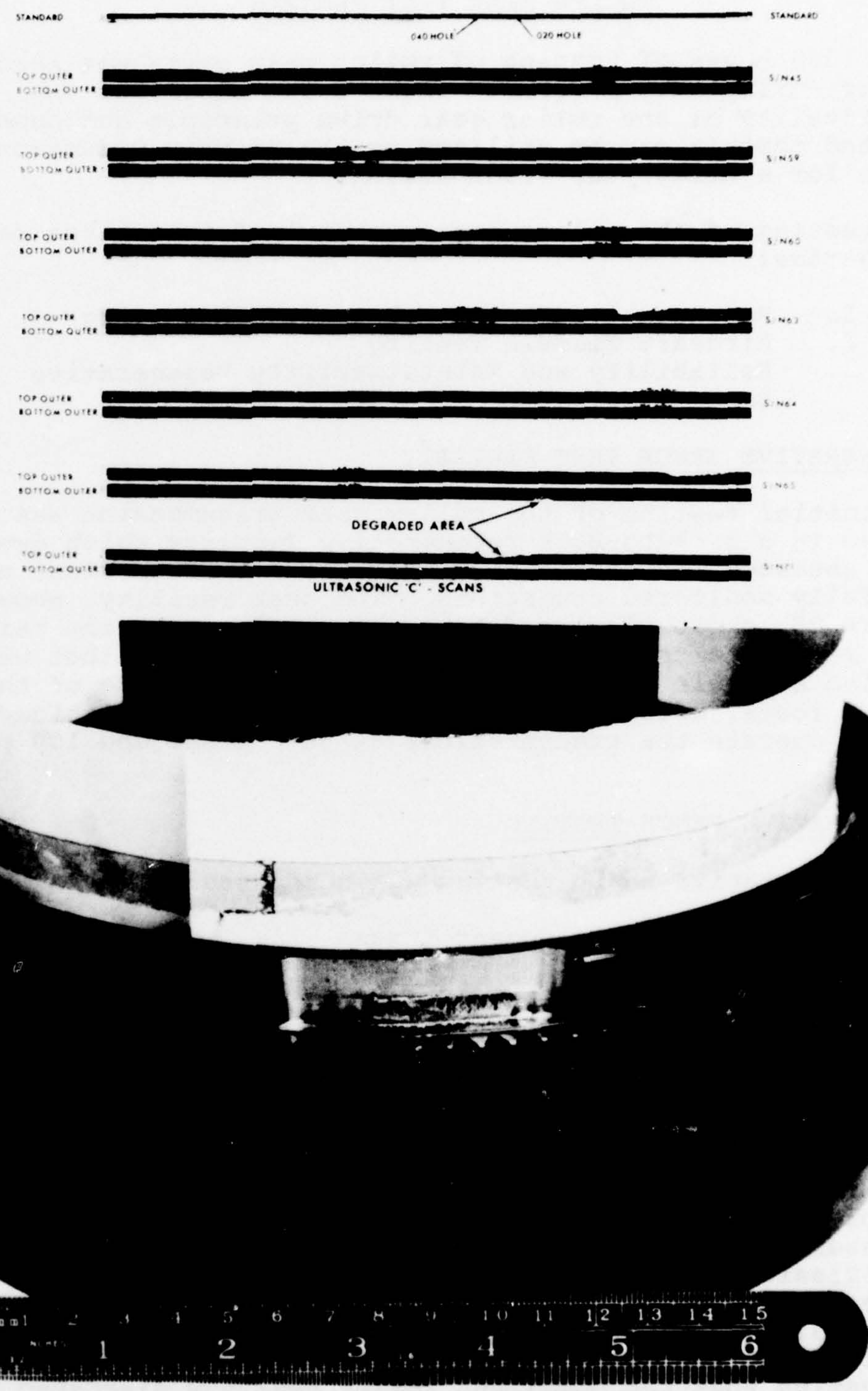


Figure 24. Cracked Lower Roller Weld, Second-Row Pinion.

ROLLER GEAR TEST PROGRAM

Over 1300 hours of testing of roller gear units was accomplished during this program. These tests confirmed the practicality of the roller gear drive principle and demonstrated that it can be utilized as the primary reduction drive for a helicopter transmission.

The testing of the roller gear was divided into three major categories:

1. Regenerative (back-to-back) Bench Testing
2. Aircraft Tiedown Testing
3. Reliability and Maintainability Regenerative Bench Testing.

REGENERATIVE BENCH TEST FACILITY

The initial testing of the roller gear transmission was conducted in a back-to-back regenerative facility which dynamically assessed the characteristics of the transmissions under carefully monitored conditions. The test facility, shown in Figure 25, employed closed torque loops for both the tail and main rotor shafts of two identical transmissions that were coupled at their inputs. Electric motors, outside of the main torque loops, were used to overcome the system friction torque and to operate the transmissions at 50 percent and 100 percent speed.

REGENERATIVE BENCH TESTING

The regenerative bench testing was conducted in three phases:

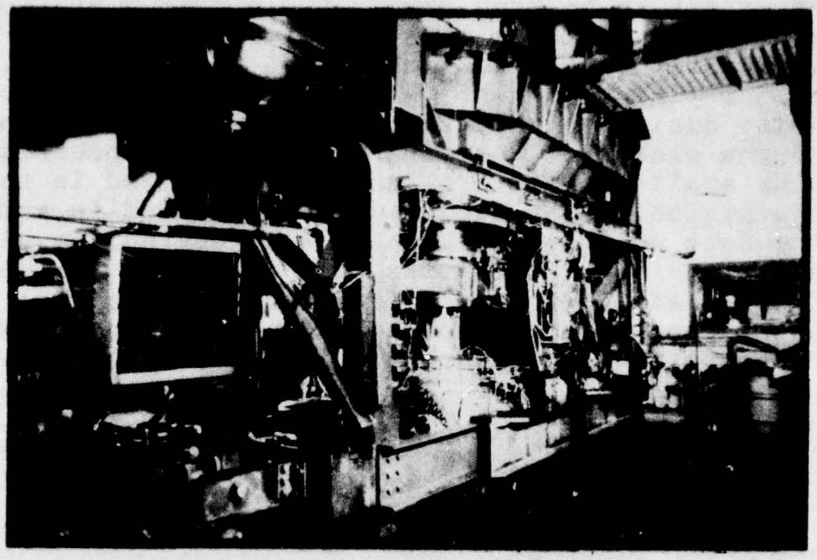
Gear pattern development tests

Initial development tests

200-Hour endurance test.

For these tests the primary instrumentation involved measurement of the main rotor and tail driveshaft torques, bearing temperatures, and reaction loads on the roller gear posts.

The primary purpose of the gear pattern development tests was to assure proper contact of all gears in the roller gear transmission. These tests revealed full-face contact on all meshes of the roller gear drive. Chamfering of the corner edge radius of the input bevel pinion teeth was the only modification that was necessary. It was during this phase of testing that the compliant roller bearings discussed earlier were evaluated.



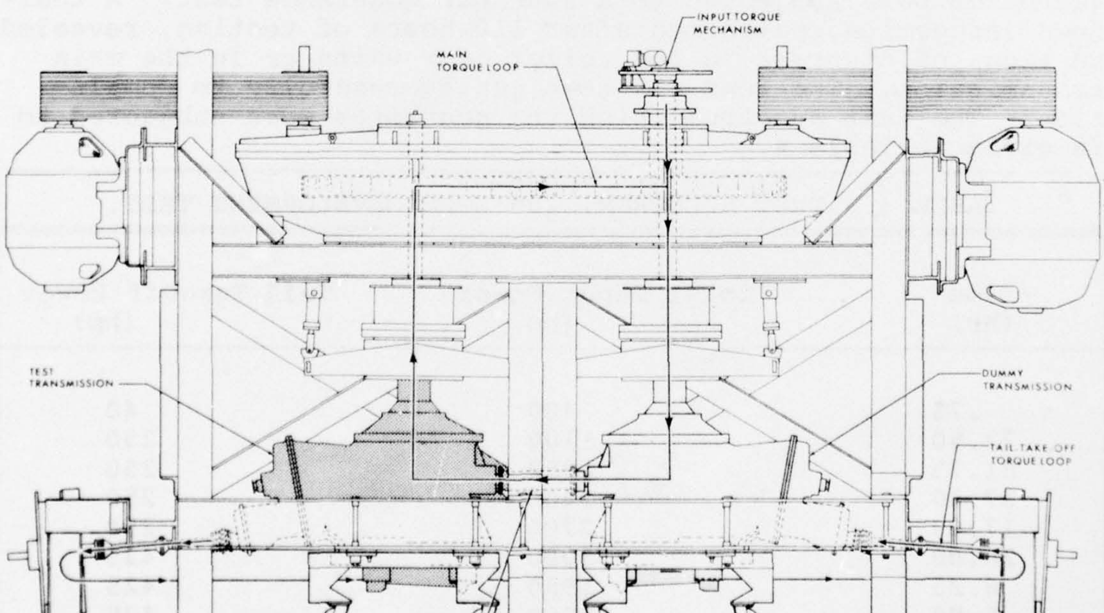


Figure 25. Roller Gear Transmission Bench Test Facility.

INITIAL DEVELOPMENT TESTS

As this was the first time that electron-beam welding was used to such an extent on so many components within a helicopter transmission, this test was considered to be important in providing data for the evaluation of the designs of the welded components. During this test, which was conducted at varying power levels, design modifications were made to both the first- and second-row pinions. The first-row pinions incurred fractures of the small-diameter roller which resulted in the butt weld design pinion. Subsequent testing resulted in a modification to the butt weld design because heat during the welding process had transformed carburized material near the root of the small gear leading to tooth fracture. The initial design second-row pinion was modified to allow more powerful welding parameters to be used to ensure complete fusion of the small-diameter gear to the flange. Scuffing of the shoulders of the first- and second-row pinion rollers due to inadequate lubrication was corrected by lubricating all dynamic contact points within the roller gear drive.

200-HOUR DEVELOPMENT TEST

To further evaluate the roller gear transmission, the two test gearboxes were subjected to a 200-hour endurance test. A tear-down inspection, performed after 110 hours of testing, revealed no signs of distress in the roller gear units or in the main transmission. The test was then run successfully to completion. The test spectrum which the gearboxes were subjected to is given in Table 4.

TABLE 4. TEST SPECTRUM, 200-HOUR DEVELOPMENT TEST.

Time (hr)	Total Input Power (hp)	Tail Takeoff Power (hp)
.75	400	40
22.50	1100	250
81.75	1950	250
37.50	2400	250
17.00	2700	250
26.00	3000	425
6.25	1950	425
6.00	3560	425
2.25	3700	425

Visual inspection of the rollers showed them to be generally in excellent condition. The second-row pinion rollers still retained their original phosphate finish. Two first-row pinions did exhibit a slight degree of flaking (Figure 26). However, subsequent testing showed that this was self-healing and, therefore, of no concern. It is evident that the surface distress, which is apparently caused by asperity interaction between the mating roller surfaces, does not necessarily cause stress concentrations sufficient to make themselves self propagating. In fact, a smoothing over and plastic spreading of the higher, nonpeeled surface occurs. This type of self-healing phenomenon was noted by Franklin Institute Research Laboratories,[6] wherein it was observed that shallow spalling of rolling contacting elements did not propagate deeper. 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 crack propagation."

Magnetic particle inspection revealed cracks in the bearing bore of the second-row pinions (Figure 27) originating from the stress riser resulting from the configuration of the weld. Fatigue cracks were also found in the gear/flange weld of the second-row pinions originating from a series of voids along the weld line.

A survey of the post loads, which were monitored continuously, showed equal load distribution among the seven pinions within ± 4 percent. An efficiency test, ran coincident with the 200-hour test, showed an efficiency of 98.4 percent for an input power of 3500 horsepower, which compares very favorably with those of conventional transmissions.

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(6) J. H. Rumbarger, L. Leonard, DERIVATION OF A FATIGUE LIFE MODEL FOR GEARS, Franklin Institute Research Laboratories, USAAMRDL Technical Report 72-14, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, May 1972, AD 744504.

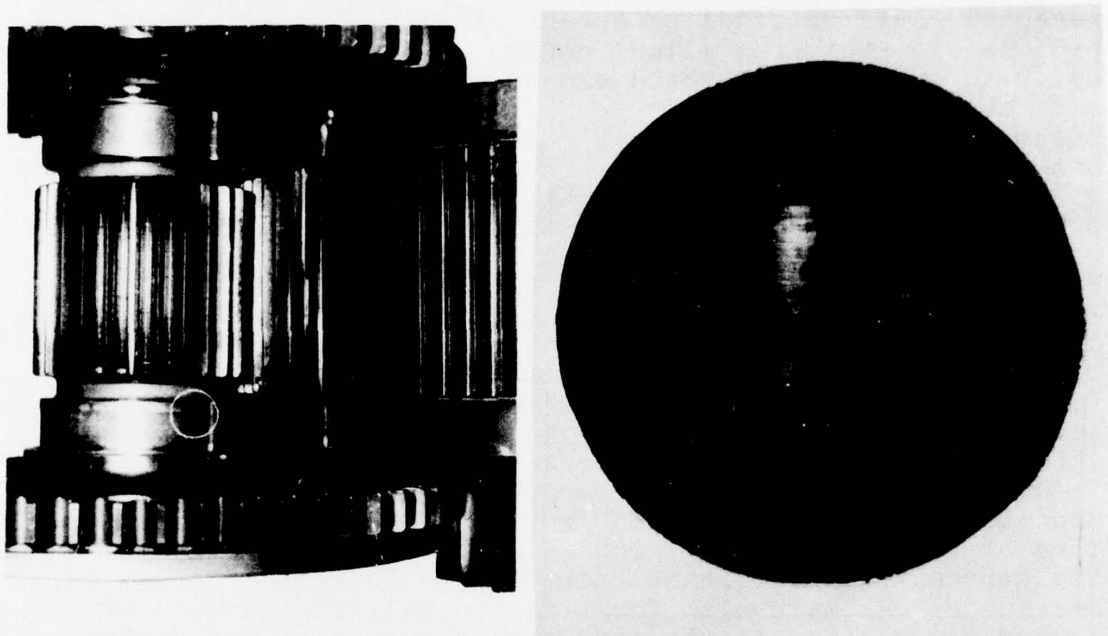

Figure 26. Surface Flaking, First-Row Pinion Roller.

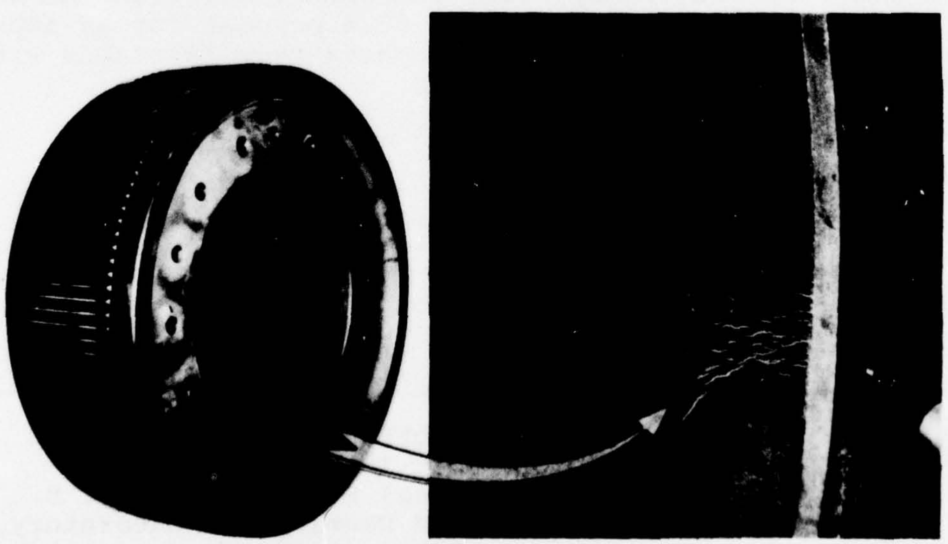

Figure 27. Second-Row Pinion Bearing Bore Cracks.

AIRCRAFT TIEDOWN TEST

Further testing of the roller gear transmission was conducted on a modified S-61 type helicopter tied to the ground (Figure 28) to simulate transmission interaction with the engines, rotors, and airframe in actual flight conditions. The aircraft tiedown test is considered to be more severe than flight, for the ground effect, induced by the close proximity of the rotors to the ground, produces vibrations much higher than those experienced in flight.

The aircraft itself was extensively modified to accommodate the two experimental YT58-GE-16 engines. The general arrangement of the engine installation is shown in Figure 29. The engine is supported at the front on support struts and at the rear on a gearbox gimbal. An engine driveshaft transmits the power from the engine free turbine to the input bevel pinions. Flexible drive couplings on either side of the driveshaft allow for misalignment between the engine and gearbox. Power to the gearbox is monitored by a torquemeter.

Throughout the 50-hour test, the transmission operated within allowable temperatures, maintained good oil pressure and experienced only minor operational problems. The actual test conditions are shown in Table 5. The total operational time of 57.6 hours exceeded the 50-hour test requirement; however, the actual power levels were less than planned due to engine performance degradation, primarily attributable to "hot-day" conditions. A post-test inspection of the roller gear unit showed a slight deterioration of two first-row pinion rollers, similar to that which occurred during the 200-hour bench test. The second-row pinions still retained the original black oxide coating. The gear teeth on all the components showed excellent contact patterns with only negligible signs of wear. Comparison of post-test ultrasonic inspection C-scans with those taken prior to the commencement of testing revealed cracks in the electron-beam welds of the second-row pinion rollers.

As a result of these cracks, a separate test was performed in the R&M test rig, which tests two roller gear units alone in a regenerative arrangement. During this test, the cracked second-row pinions were operated under load, and crack propagation was checked periodically. The test was terminated when the second-row pinion fractured (Figure 17). The initial fracture occurred at the bearing bore weld, a result of weld voids and stress intensity caused by the joint design. Examination of the remaining pinions showed fatigue cracks in the bearing bores. Although this test ended in a catastrophic mode of failure, there was no significant increase in the size of cracks in the roller welds than observed after the 50-hour tests.

TABLE 5. ROLLER GEAR AIRCRAFT TIEDOWN TEST POWER LEVELS.

Test Condition	Total Main Gearbox Input (hp)	Main Rotor Torque* (%)	Main Rotor Power (hp)	Tail Rotor Torque* (%)	Tail Rotor Power (hp)	Frictional Losses and Accessory Power (hp)	Drive Train Speed*** (%)	Time (hr)
Maximum	3390-3440	98-100	2940-3000	55-60	311-339	135	100	.5
3000 hp	3110-3480	89-100	2670-3000	55-62	311-350	133-135	100	1.6
2000 hp	2130-2410	60- 67	1800-2010	36-48	203-271	128-129	100	1.3
1000 hp	1080-1370	33- 38	990-1140	17-18	96-102	122-123	100	1.7
Flat Pitch	490	11	330	7	40	118	100	4.8
Takeoff High	3080-3600	89-105	2670-3150	45-65	254-367	133-136	100	2.5
Takeoff (Idle)	100	5	60	5	10	30	40	2.5
Military High	3290-3740	91-109	2730-3270	51-60	288-339	133-138	100	2.1
Military Low	2120-2350	60- 67	1800-2010	35-41	198-232	128-129	100	2.7
100% NRP	2976-3590	85-105	2550-3150	47-60	266-339	132-136	100	14.9
90% NRP	3120-3280	88- 94	2640-2820	49-58	277-328	133-134	100	3.9
80% NRP	2810-2940	80- 83	2400-2490	47-62	266-350	131-132	100	3.5
60% NRP	1960-2250	60- 70	1630-1970	35-41	180-218	127-129	91- 94	4.0
Overspeed	2490-3020	62- 80	2010-2640	38-55	232-342	129-132	108-110	4.0
Single Engine No. 1	1640-1750	45- 48	1350-1440	28-32	159-181	125	100	2.6
Single Engine No. 2	1630-1730	44- 48	1320-1440	27-31	153-175	125	100	2.7
Ground Idle	100	5	60	5	10	30	40	2.3

* 100% Main Rotor Torque = 3000 hp (77,312 ft-lb at 203.8 rpm).
** 100% Tail Rotor Torque = 565 hp (979 ft-lb at 3030 rpm).
*** 100% Drive Train Speed = 18,966 rpm Engine Driveshaft, 203.8 rpm Main Rotor, 1243 rpm Tail Rotor.

Figure 28. Roller Gear Tiedown Test Aircraft.

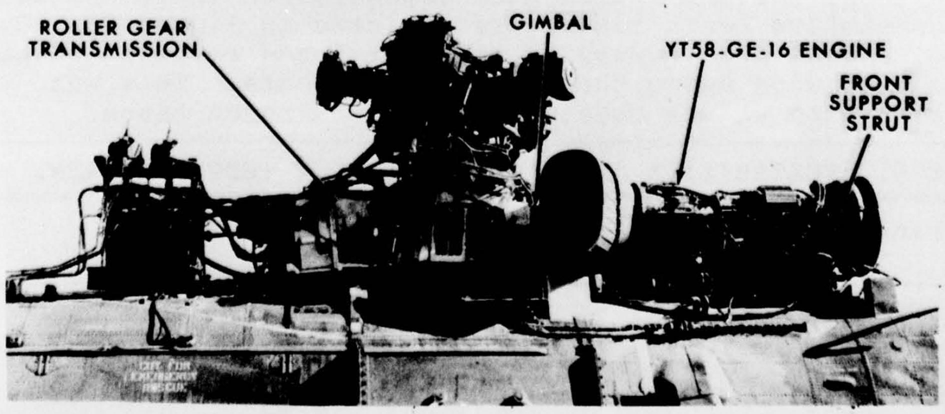

Figure 29. Engine/Transmission Installation on Tiedown Test Aircraft.

RELIABILITY AND MAINTAINABILITY TEST

This test, conducted in the facility shown in Figure 30, employs two roller gear units mounted back-to-back. The inputs of each unit are coupled together through a torque mechanism, and the ring gears are connected to form a closed torque loop whereby power can be regenerated in each roller gear unit. The test load is applied statically by twisting the input shafts relative to each other. With the system at operational speed, power is recirculated and contained within the roller gear units; thus, the prime mover power requirements need only to be equal to the power losses in the system, i.e., the power loss due to friction.

During the 300-hour test, the first-row pinions incurred a tooth fracture and spalling of the gear teeth. The fracture of a first-row pinion small-diameter gear tooth (Figure 31) was typical of that associated with gear tooth bending fatigue. No evidence of excessive or unequal loading was found. A contributing cause of the fracture was the presence of manganese phosphate pits which were found at the origin sites. The second malfunction occurred again on the small-diameter gear teeth of the replacement first-row pinions. After continued operation for 250 hours at the 3000-horsepower design power, spalling of the gear teeth (Figure 32) occurred on two adjacent pinions. Again, no maldistribution of loading was evident. Examination did reveal a low case-depth of the hardened surface which was inadequate for the loads. Table 6 shows the endurance test spectrum that was conducted.

Concurrent with the endurance test, an efficiency test was conducted by measuring the input power required to overcome the friction torque of the two roller gear units. It was found that an efficiency of .99005 percent was achieved by the roller gear units. Representative efficiencies for the roller gear unit and for the roller gear transmission (obtained during the regenerative bench tests) are depicted in Figure 33. During testing, a continual survey of the post loads revealed ± 4 percent load showing among the seven pinion posts. This was consistent with values obtained during the bench tests.

TABLE 6. RELIABILITY AND MAINTAINABILITY TEST SPECTRUM.

Transmitted Power (hp)	Accumulated Time (hrs)
1440	6.0
2170	4.0
2400	5.25
2640	6.25
3000	274.5

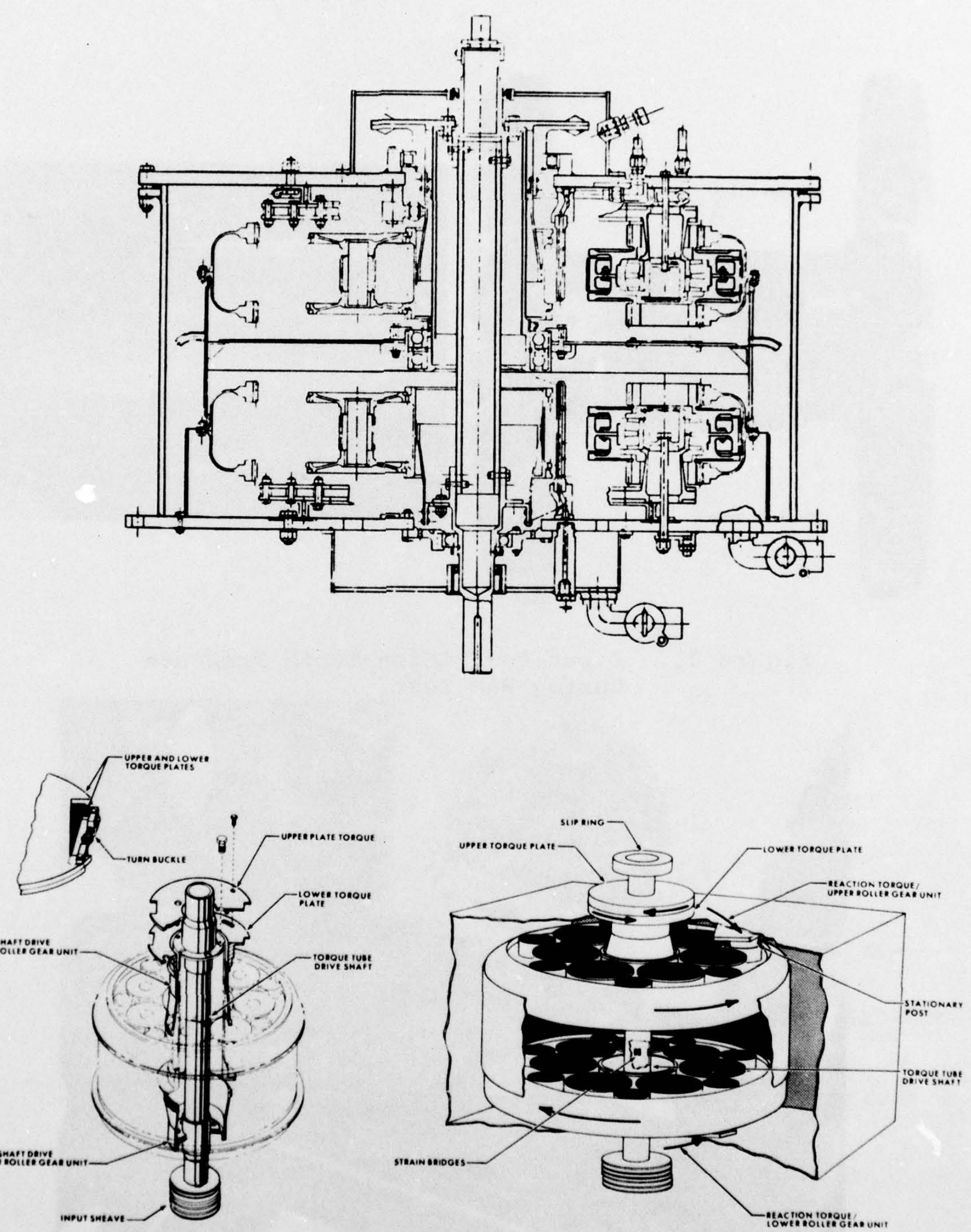

Figure 30. Reliability and Maintainability Regenerative Test Facility.

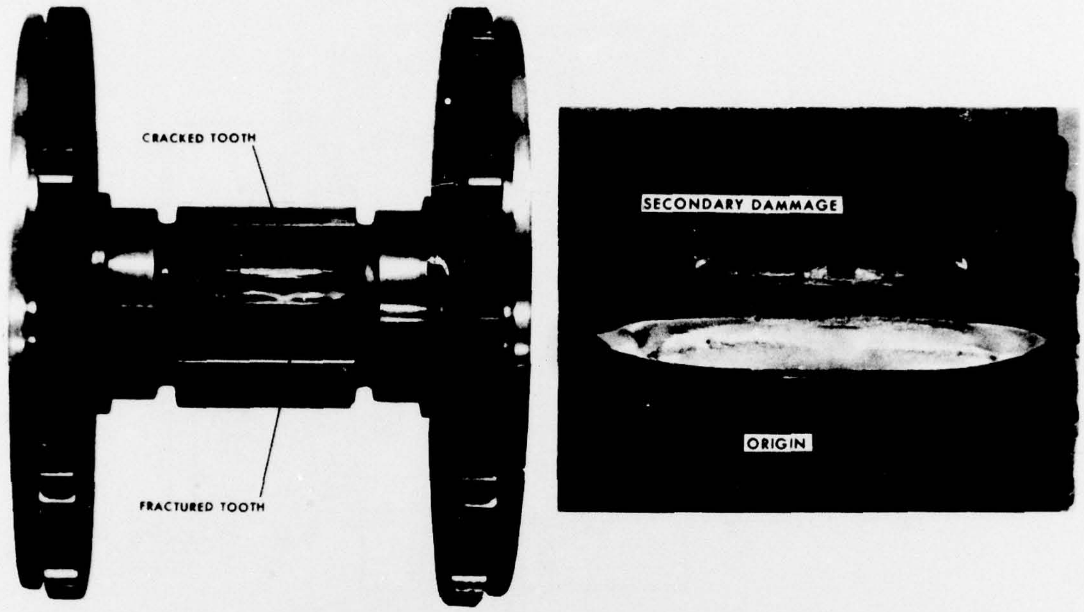

Figure 31. First-Row Pinion Tooth Fracture During R&M Test.

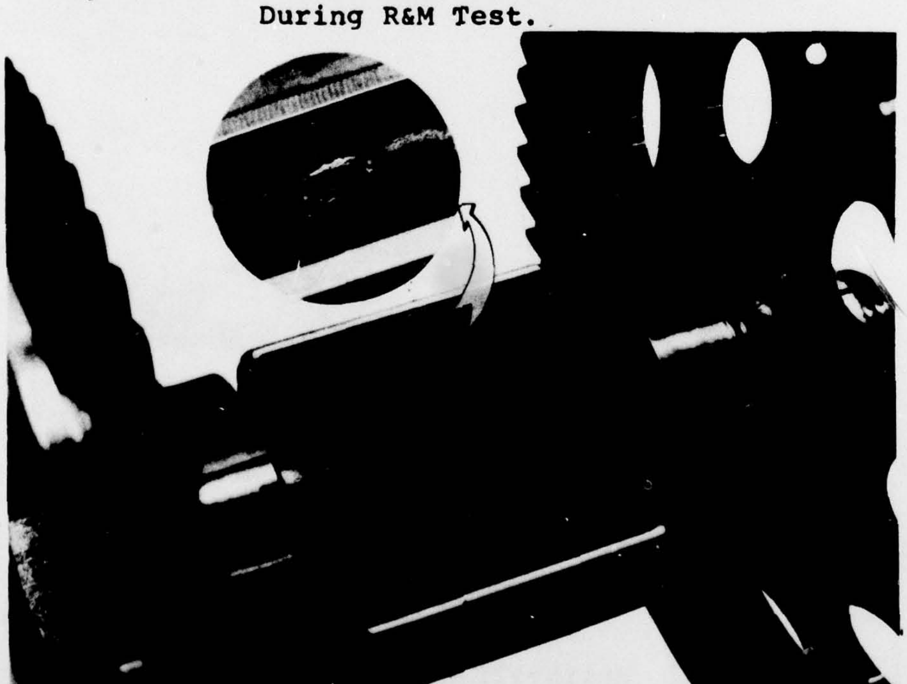

Figure 32. First-Row Pinion Tooth Spalling.

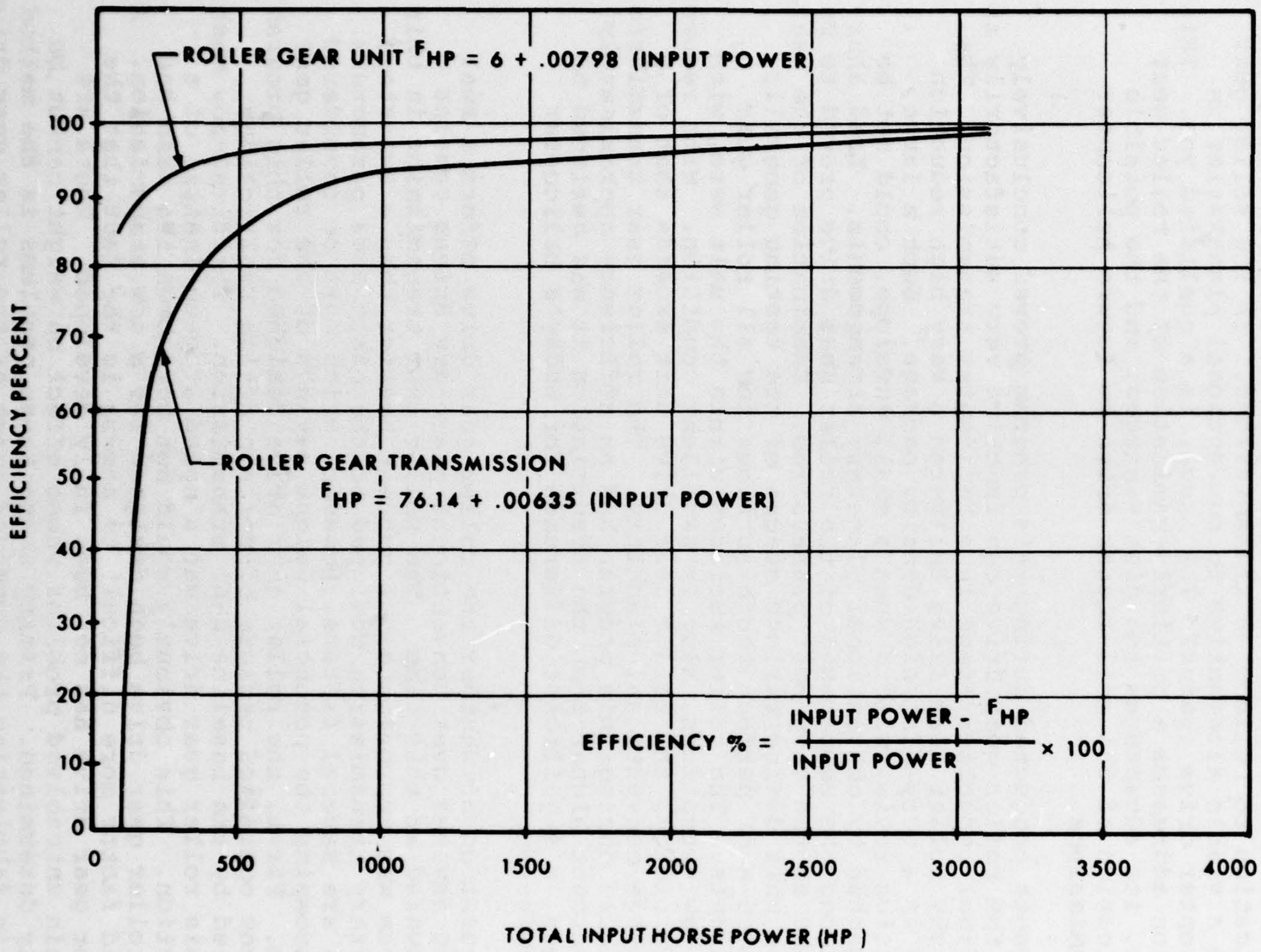

Figure 33. Roller Gear Unit and Transmission Efficiencies.

ROLLER GEAR EVALUATION

With the completion of the various test stages of this program, it remains to evaluate the roller gear with respect to its application to helicopter transmissions. Is the roller gear drive a viable alternative to conventional planetaries in helicopter drive trains? The answer is a qualified yes. This section addresses a critical examination of the roller gear drive, its advantages and disadvantages, and the possible application of the roller gear drive to future helicopter transmissions.

The tests conducted during this program proved conclusively that the roller gear drive can function very satisfactorily as the final reduction stage in a helicopter transmission. The Sikorsky roller gear drive delivered a very high reduction ratio in a very flat/high-density package. Such a large reduction ratio, within such a small envelope, could not be approached by conventional planetary arrangements. Load sharing among the components of the roller gear drive proved to be as good as the very best planetaries. Examination of the gears of the unit during various stages of the testing generally revealed near perfect tooth surfaces for all roller gear components. The roller surfaces within the unit were, with very few exceptions, also in excellent condition. With respect to efficiency, the roller gear drive unit exceeds that of a two-stage conventional planetary. The roller gear transmission developed during this program had an efficiency approximately 0.5 percent higher than the transmission it was designed to replace, a significant difference for today's helicopter transmission.

The question of whether the roller gear drive offers a real weight savings over conventional planetary drives remains unanswered at this time. The roller gear transmission in this program was approximately 1 percent heavier than a two-stage planetary transmission designed to the exact same parameters. There are several factors, however, which must be considered in assessing the potential weight savings of the roller gear drive. First, the roller gear drive designed for this program was not optimized because of certain design restrictions imposed by the baseline S-61 transmission. The first-row mesh of this roller gear drive was a speed step-up instead of a reduction. This obviously would not have been the case had the roller gear drive been designed for a new transmission. A second factor more difficult to assess is the fact that the roller gear drive has not been fully developed. There are certain unresolved problems whose effect on weight cannot be easily determined. Primary among these problems is the method used in fabricating the compound gears of the roller gear drive. Substantial progress was made during the course of this program in the development of electron-beam welding for fabrication of the compound gear components. The final test of these components in the Reliability and Maintainability test facility proved that electron-beam welding is indeed well suited to this purpose. During this test, two complete roller gear units were tested at 3000 horsepower for over 200 hours. At the completion of this test, not a single weld showed any sign of degradation. Quality control of the welds, however, remains a problem. At least one fracture experienced during this program resulted from discontinuities that were not detected during ultrasonic inspection. Unless the method of detecting minute welding flaws is improved, it would be necessary to design production roller gear components with heavier welds which would lead, of course, to heavier roller gear components.

The reliability of the roller gear drive is another area which, like weight, remains open to question at this time. It is very difficult to assess the reliability with any degree of confidence of any system that is not yet fully developed. Several aspects of the roller gear drive may be pointed out, however, which do tend to affect its reliability. On the basis of quantity of parts alone, the roller gear drive has a distinct advantage over conventional planetaries; in general, the fewer the parts, the more reliable the system. The fact that the roller gear drive employs fewer bearings than conventional reduction systems also tends to favor the roller gear with regard to reliability. Bearing failure is an important factor in the unscheduled removal frequency of transmissions. There are some practical considerations to be weighed, however, that tend to offset these potential advantages. The electron-beam welds within the roller gear components have not been consistent, and the long-term reliability of these components is still open to question because of this. Another aspect of the roller gear drive that could adversely affect its reliability is its inherent tolerance sensitivity. Although the roller gear components are made to an accuracy exceeding that of conventional aerospace gearing, manufacturing tolerances can adversely affect roller gear operation. The maintainability characteristics of the roller gear drive is far from ideal. Because of the compact nature of the unit, it is necessary to completely disassemble it to conduct a proper inspection. The fact that the roller gear drive is a complete modular assembly and is easily removed from the gearbox somewhat offsets this disadvantage.

As can be deduced from the above discussion, the manufacture of the compound gear components remains the most difficult problem with the roller gear drive. Because of their complexity, the roller gear components require special fabrication techniques and tolerance controls that are unnecessary for conventional gears. Much progress was made during this program toward perfecting the manufacturing process, but some problems remain. The use of electron-beam welding in the fabrication process offered a clear advantage over the earlier efforts using press fits and bolts. At this time, however, the desired degree of quality control of the welds has not been attained. During this program an ultrasonic technique was developed for inspection of the welds. While this method has proven to be extremely valuable, it did not detect weld defects smaller than .010 inch. This shortcoming was apparent after aircraft tiedown testing. Second-row pinion roller welds, which had exhibited no imperfections on initial inspection, had seriously deteriorated during load testing apparently because of undetected discontinuities. The close timing and tolerance requirements of the compound gear components leave the roller gear drive at a distinct disadvantage in fabricability when compared with conventional reduction systems.

Closely related to manufacture is cost. The price of any new system must be considered in a complete evaluation. At this time the roller gear drive is not cost competitive. Estimates place a roller gear production transmission at roughly double the cost of a conventional transmission designed to the same requirements. Further development of the manufacturing processes could somewhat reduce this cost differential, but barring unforeseen developments, it appears that the initial cost of a roller gear drive will be substantially more than that of a comparable conventional system.

When assessing the potential application of any new system to an aircraft, a trade-off study must be conducted to weigh the potential gain against the level of risk incurred. While the result of any study like this depends on many factors such as size of aircraft, type of mission, etc., several conclusions may be drawn on the basis of this program. First, none of the technical problems still associated with a roller gear drive appear unresolvable with present technology. A roller gear transmission should be able to be developed with a low technical risk. Second, it is probable that in the proper application, a roller gear drive can offer a substantial weight savings over a conventional planetary arrangement. It appears that a minimum last-stage reduction of 12:1 is necessary for any gearbox weight savings to be realized. The roller gear transmission can also be utilized to advantage where the space envelope allotted the transmission is very flat. Here the compactness and single plane reduction features of the roller gear can be very desirable. Of course, all of these advantages must be traded-off against the higher cost of the roller gear, the one aspect that appears unresolvable at present. However, It is very conceivable that in the proper application, the roller gear drive can lead to a more cost-effective aircraft despite its higher initial cost.

CONCLUSIONS

1. The roller gear drive is a high-efficiency, 99% unit that provides very high reduction ratios in a relatively compact package. The concept is fundamentally sound and can function successfully as the final reduction stage of a helicopter transmission.

2. Quality control during fabrication of the compound gear components remains an obstacle to the successful execution of the roller gear concept. Electron-beam welding, while producing relatively light components, has yet to produce these parts with consistently high enough quality to warrant their use in a production transmission.

3. The roller gear drive has the potential to offer significant weight savings, but realization of this weight savings depends on an efficient design and solution of the above-mentioned quality control problem.

4. The relative reliability of the roller gear drive cannot be assessed with certainty at this time. Since the roller gear drive is not yet fully developed, it cannot easily be compared with units which have been. The roller gear drive, like any system not completely developed, is still susceptible to problems which should eventually be eliminated through design improvements.

5. The roller gear drive is considerably more costly than conventional planetaries. Rigid tolerance and timing requirements, which are not required with conventional systems, add substantially to the cost of the unit.

RECOMMENDATIONS

On the basis of the results of this program, further investigation of the roller gear drive is warranted to assess its role in the future of helicopter transmission development. It is recommended that the roller gear drive be pursued with a two-phase program. The first phase of this program should address itself to the fabrication problem associated with the compound gear components. The objective of this phase should be to determine what procedures, both design and manufacture, should be instituted to produce consistently high-quality electron-beam welded components. Attention should also be given to further refining the weld inspection process. In addition, this phase should include an analytical effort to develop ways to reduce the cost of a production roller gear transmission. The second phase of the recommended program would be a comparative design investigation of the roller gear with conventional planetaries or another advanced gear reduction concept. This effort should include determination of the most efficient application of the roller gear concept and an accurate evaluation of the roller gear in this application with respect to both conventional and advanced concepts.

APPENDIX

ROLLER GEAR PROGRAM REPORT SUMMARIES

The following pages present brief summaries of the six volumes covering the 3000-HP Roller Gear Transmission Development Program.

Volume I - Summary Report (USAAMRDL-TR-73-98A)

This report presents a summary of the 3000-hp Roller Gear Transmission Development program conducted between June 1969 and June 1976. Included in this report are brief summaries of the roller gear drive concept, design and manufacture. Also outlined are the results of each phase of testing to which the roller gear drive was subjected. In addition, this report presents a critical evaluation of the roller gear drive which includes a discussion of the various advantages and disadvantages that a roller gear drive transmission offers when compared to a conventional planetary. Specific areas of the roller gear drive that require further development are also noted.

Volume II - Design Report (USAAMRDL-TR-73-98B)

This report covers the initial design of the 3000-hp roller gear transmission. A detailed discussion of the general design of a roller gear unit is presented including examples of various roller gear configurations and their associated envelope, rotation and ratio restrictions. The basic roller gear geometry, teeth indexing, and methods of preload are also discussed. The design of the Sikorsky roller gear unit is described in detail including the rationale behind the selection of the final configuration. Each component of the roller gear unit is also completely covered with respect to its design. Other topics dealt with in this report include the design of the transmission's primary drive components, the structural analyses and the efficiency and lubrication analyses of the roller gear unit and transmission.

Volume III - Roller Gear Manufacture (USAAMRDL-TR-73-98C)

This report presents a survey of the manufacturing methods used in the fabrication of the roller gear transmission components. The bulk of the material contained in this report deals with the roller gear unit components because of the unique manufacturing problems they presented. No attempt is made to describe in detail the more conventional manufacturing processes employed in this program. The most significant aspect of the manufacture of the roller gear components was the extensive use of electron-beam welding. This process is discussed in depth. Another topic which is dealt with extensively is the inspection methods used in attempting to inspect the electron-beam welds. Magnetic particle, x-ray, and ultrasonic inspection techniques are evaluated with respect to their success in detecting weld flaws. A discussion of the successful use of the magnesium casting alloy ZE-41A for the main housing is also included. Other topics covered are the manufacture of the main rotor shaft and the freewheel unit, and the assembly of the roller gear unit. Manufacturing procedures used in the fabrication of the roller gear unit components are included as Appendices.

Volume IV - Laboratory Bench Test (USAAMRDL-TR-73-98D)

This report presents the results of dynamic load tests performed on the roller gear transmission. The report contains the test procedures and results of the no-load lubrication test, the gear pattern development test, the initial development tests, and the 200-hour endurance test. It also reports the results of an efficiency test of the roller gear transmission made in conjunction with the 200-hour endurance test. Included with the discussion of the initial development tests are descriptions of design changes to the roller gear components which were made. In addition, this report contains detailed descriptions of the test facilities and instrumentation which were employed during testing. Logs of the various tests are included in an Appendix.

Volume V - Aircraft Tiedown Test (USAAMRDL-TR-73-98E)

This report deals with testing of the roller gear transmission in a tied down S-61 type aircraft. This test was performed to evaluate the performance of the roller gear transmission in an actual aircraft installation. A complete description of the tiedown aircraft is presented including the engine, transmission, fuselage and flight control systems. Static testing of these systems prior to start-up is described as are the test instrumentation and data acquisition systems. The results of the 50-hour endurance test performed on the tiedown aircraft are dealt with in detail. Also reported in this document are the results of testing of grease-lubricated tail and intermediate gearboxes with which the aircraft was equipped during the 50-hour test. As a result of this endurance test, a fatigue crack propagation test was performed with the same roller gear components used during the tiedown testing. This test, performed on the R&M test facility is discussed in this report.

Volume VI - Reliability and Maintainability Report
(USAAMRDL-TR-73-98F)

This report presents the results of a study to determine the reliability of the roller gear transmission relative to that of a conventional planetary design. Included in this report are the results of the R&M testing of the roller gear unit. In this test two roller gear units were tested simultaneously in a regenerative test facility. A reliability analysis, which compares projected roller gear reliability with two-stage planetary reliability, is presented in this report. Also included is a discussion of the maintainability aspects of the roller gear drive and a failure mode and effects analysis of the roller gear transmission.