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ADVANCED PROP-FAN ENGINE TECHNOLOGY (APET) SINGLE- AND COUNTER-ROTATION GEARBOX/PITCH CHANGE MECHANISM

FINAL REPORT
by

C. N. Reynolds

COMMERCIAL ENGINEERING
PRATT & WHITNEY
UNITED TECHNOLOGIES CORPORATION

PREPARED FOR:

NASA
National Aeronautics and
Space Administration

Lewis Research Center
Cleveland, Ohio
44135

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16. ABSTRACT

Volume I reports the preliminary design of advanced technology (1992) turboprop engines for single-rotation Prop-Fans, the conceptual design of the entire propulsion system, and an aircraft evaluation of the resultant designs.

Volume II reports the preliminary design of advanced technology gearboxes and conceptual designs of pitch change mechanisms for single- and counter-rotation Prop-Fan applications. The single-rotation gearbox is a split path, in-line configuration; the counter-rotation gearbox is an in-line, differential planetary design. The pitch change mechanisms for both the single- and counter-rotation arrangements are rotary/hydraulic.

The advanced technology single-rotation gearbox yields a 2.4 percent improvement in aircraft fuel burn and a one percent improvement in operating cost relative to a current technology gearbox. The 1992 counter-rotation gearbox is 15 percent lighter, 15 percent more reliable, 5 percent lower in cost, and 45 percent lower in maintenance cost than the 1992 single-rotation gearbox. The pitch controls are modular, accessible, and external. Research and technology plans will verify the necessary advanced technologies.

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SECTION 1.0
SUMMARY

SECTION 1.0
SUMMARY

The APET studies, Volume I of this report, have shown that an advanced, geared Prop-Fan propulsion system can lower transport aircraft operating costs substantially. Relative to an aircraft powered by an equal technology turbofan engine, a single-rotation Prop-Fan can lower fuel burn by 21 percent and direct operating costs by 10 percent. With counter-rotation, the benefits are even greater -- 31 percent and 14 percent, respectively. To attain the full potential of these benefits, an advanced gearbox is a critical technology.

In this phase of the APET program, the preliminary design of an advanced technology gearbox was completed for both single and counter-rotation applications. The design of each gearbox was accomplished as a separate contractual effort. The scope of work in each program included the preliminary mechanical design of the gearbox, the conceptual design of the pitch control mechanism, and the formulation of research and technology plans.

For a single-rotation Prop-Fan, the selected gearbox design is a split path, in-line configuration. It is a modular, compact arrangement using 17 gears and 15 bearings. The gearbox is designed for a wing-mounted tractor installation and has provisions for opposite hand rotation. Its design also accommodates an externally located pitch control. In the single-rotation effort, two designs of the split path gearbox were completed. One used current technology, and one used advanced technology available by 1988. Each design was analyzed in terms of structure, performance, and operating economics. Results of these analyses clearly showed that the advanced technology yields large fuel and economic gains. The net result is a gain of 2.4 percent in fuel burn and one percent in operating cost. Major features contributing to these benefits are stronger gear/bearing materials, advanced gear toothshape, a modulated lubrication system, an aerodynamic scavenge system, and an external pitch control. Hamilton Standard conceptually designed the pitch control which is a rotary hydraulic concept.

For a counter-rotation Prop-Fan, the selected gearbox design is a straddle-mounted, in-line differential planetary configuration. This design allows maximum installation flexibility, because it is adaptable to either pusher or tractor application with no modifications necessary. It is a modular design using 12 bearings and seven gears. The counter-rotation gearbox incorporates technology features similar to those in the single-rotation gearbox. Relative to single-rotation, the counter-rotation configuration is simpler and offers significant advantages. For equal horsepower, it is 15 percent lighter and 15 percent more reliable. Its acquisition cost is five percent lower, and its maintenance cost is 45 percent lower.

To achieve the full potential of these gearbox designs, verification of the following advanced technologies is necessary: gear and bearing materials, the lubrication supply and scavenge system, and the gear and bearing system. Critical technologies for the pitch control are the capacitor signal transfer, high pressure hydraulics, and rotating electronics. The plan to verify these technologies consists of a series of individual technology evaluations, followed by an integrated test program with the multipurpose gearbox rig. The plan will ensure technology verification by mid 1987.

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SECTION 2.0
INTRODUCTION

SECTION 2.0
INTRODUCTION

Pratt & Whitney's work under the NASA-sponsored Advanced Prop-Fan Engine Technology (APET) Definition Study, Volume I of this report, has shown that an advanced, geared Prop-Fan propulsion system offers significant savings in fuel burn and direct operating cost. Relative to an equivalent technology turbofan engine, the single-rotation Prop-Fan reduces fuel burn by as much as 21 percent and cuts direct operating costs by up to 10 percent. The counter-rotation Prop-Fan reduces fuel burn by as much as 31 percent and direct operating costs by up to 14 percent. For either system, an advanced technology gearbox is a critical technology. A gearbox enables optimizing the design of the two major components: the power turbine and the Prop-Fan. A reduction gearbox uniquely permits:

-- a high-speed power turbine with a smaller diameter and fewer stages for the best efficiency; and

-- a low-speed, lightly loaded Prop-Fan for maximum efficiency and lower noise levels.

This report summarizes the single-rotation and counter-rotation gearbox design efforts accomplished under NASA contract NAS3-23045.

The objectives of the gearbox/pitch control preliminary design program were to:

- o establish preliminary mechanical designs of single-rotation and counter-rotation reduction gearboxes;
- o establish pitch control conceptual designs that are compatible with the selected gearbox concepts; and
- o formulate a research and technology plan for critical gearbox/pitch control technologies.

The preliminary design of the single-rotation reduction gearbox/pitch control consisted of three technical tasks. They were as follows:

Task VII - The Preliminary Design of a Single-Rotation Gearbox

In Task VII, Pratt & Whitney developed a preliminary mechanical design of a single-rotation reduction gearbox. For comparison, there were two gearbox configurations. One used current technology, and the other used advanced technology. The results of performance, structural, and economic analyses determined the benefits of advanced technology.

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Task VIII - The Conceptual Design of a Single-Rotation Pitch Control

Under Task VIII, Hamilton Standard established a conceptual design of an advanced pitch control which is compatible with the advanced Prop-Fan blades and the advanced gearbox.

Task IX - The Research and Technology Plan

Under Task IX, the research and technology plan defines critical technologies for both gearbox and pitch control designs. The plan lays out a program that will verify technology readiness by 1987 to ensure engine certification by the mid 1990's.

The preliminary design of the counter-rotation reduction gearbox/pitch control consisted of three tasks. They were as follows:

Task XI - The Preliminary Design of a Counter-Rotation Gearbox

In Task XI, Pratt & Whitney developed a preliminary mechanical design of a counter-rotation reduction gearbox using advanced technology features.

Task XII - The Conceptual Design of a Counter-Rotation Pitch Control

In Task XII, Hamilton Standard established a conceptual design of a pitch control which is compatible with advanced Prop-Fan blades and the advanced counter-rotation gearbox.

Task XIII - The Research and Technology Plan

Under Task XIII, the research and technology plan defined critical technologies for both the counter-rotation gearbox and pitch control designs. This research and technology plan is essentially the same as the single-rotation system's; therefore, the plans are combined in this report.

Section 3.0 of this report summarizes the major results of the preliminary designs of the single-rotation and counter-rotation reduction gearbox/pitch controls. Section 4.0 is a discussion of results from the single-rotation and counter-rotation programs. Section 4.1 deals with the preliminary design of a single-rotation Prop-Fan gearbox and includes discussion of advanced and current technology gearbox designs. Section 4.2 presents a conceptual design of a single-rotation pitch control. Section 4.3 details the preliminary design of a counter-rotation Prop-Fan gearbox. Section 4.4 describes the conceptual design of a counter-rotation pitch control. Section 4.5 summarizes the research and technology plans for the gearbox and the pitch control programs. Section 5.0 presents the conclusions and recommendations.

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SECTION 3.0
SUMMARY OF RESULTS

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SECTION 3.0
SUMMARY OF RESULTS

3.1 Introduction

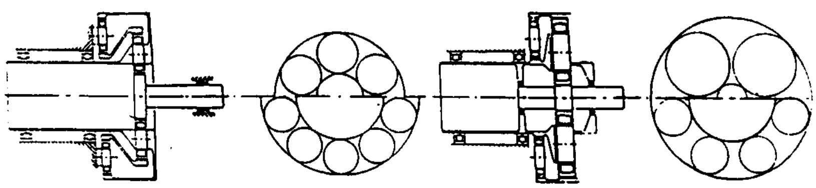
As part of the APET Definition Study, Pratt & Whitney completed preliminary designs of single-rotation and counter-rotation reduction gearbox/pitch controls. The APET single-rotation gearbox program consisted of a ten-month technical effort. The counter-rotation program also consisted of a ten-month technical effort. The scope of the work for each program covered three major areas. They were: the preliminary gearbox design, the conceptual pitch control design, and the definition of the research and technology plan.

3.2 Task VII -- The Preliminary Design of a Single-Rotation Gearbox

3.2.1 The Gearbox Refinement Studies

Results of earlier phases of the APET Definition Study, Volume I of this report, identified a split path, in-line gearbox as the most attractive for an advanced single-rotation Prop-Fan. However, the original gearbox configuration had a large number of gears and bearings. This penalized gearbox efficiency and reliability and increased maintenance costs. An additional effort was undertaken to optimize this gearbox.

On the basis of a refinement analysis, an alternate split path gearbox was configured. As shown in Figure 3.2-1, the refined configuration has five fewer gears and four fewer bearings than the original design. The reduction in the number of parts significantly simplifies the mechanical configuration.



Original concept | **Alternate concept (final design)**

Achievements

- 5 fewer gears/4 fewer bearings
- Improved efficiency 0.3%
- Achieved commonality for opposite rotation
- Improved durability 12% (MTBUR)
- Lowered maintenance cost 25%
- Incorporated external advanced technology pitch control

Figure 3.2-1 Selected Configuration - The reduced number of parts significantly simplifies the mechanical configuration, improves performance and durability, and lowers maintenance cost.

The results from the refinement analyses showed that the alternate configuration offers a moderate efficiency gain along with a substantial improvement in durability and lower maintenance costs. Another feature of the alternate split path gearbox is the remote location of the pitch control system. The pitch control's external position improves the gearbox's reliability and simplifies maintenance. The alternate split path gearbox is easily adaptable to opposite hand rotation. Almost all components are common to both rotations, with no need for a separate reversing stage.

3.2.2 The Preliminary Design of the Advanced Technology Gearbox

The single-rotation gearbox design, shown in Fig. 3.2-2, uses many advanced technologies including advanced materials in the bearings, gears, and housing. High-strength materials permit smaller, lightweight bearings to operate at higher loadings and to achieve a bearing set life of 18,000 hours. The gears and the gearbox housing are also designed with advanced, lightweight materials. The use of advanced materials reduces the gearbox's size and envelope.

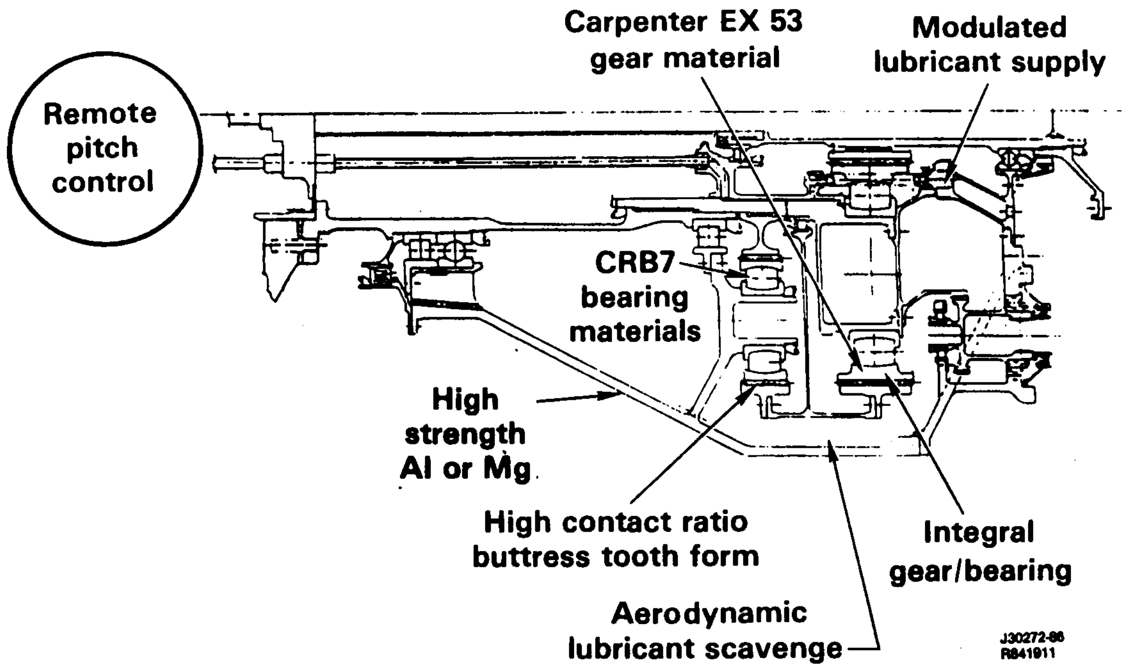


Figure 3.2.2 Advanced Technology Gearbox - Many advanced technologies are incorporated in the design.

A prominent feature of the advanced gearbox is Pratt & Whitney's modulated lubricant supply and aerodynamic scavenge system. These features reduce the oil supply to the gearbox at part power (e.g., cruise) while more effectively scavenging the oil. This improves the gearbox's overall efficiency to 99 percent at typical cruise operating conditions.

Component modularity is an integral part of this design. Accessibility to routine maintenance components has been greatly improved to maximize on-wing maintenance capabilities. A principal feature of component modularity is the external location of the pitch control.

3.2.3 The Preliminary Design of the Current Technology Gearbox

For comparative analysis, the alternate split path gearbox concept was designed with current technology. Analysis showed that current technology significantly compromises reliability, maintainability, and performance. State-of-the-art materials especially restrict bearing and gear design. Bearings must be substantially larger to accommodate the lower material/lubricant load carrying capacity and higher centrifugal loads. The gear teeth faces and bearing sizes must also become larger to compensate for higher loads, as is shown in Figure 3.2-3. The overall effects of the current technology's deficiencies are to make the gearbox larger in diameter, longer, heavier, and less efficient.

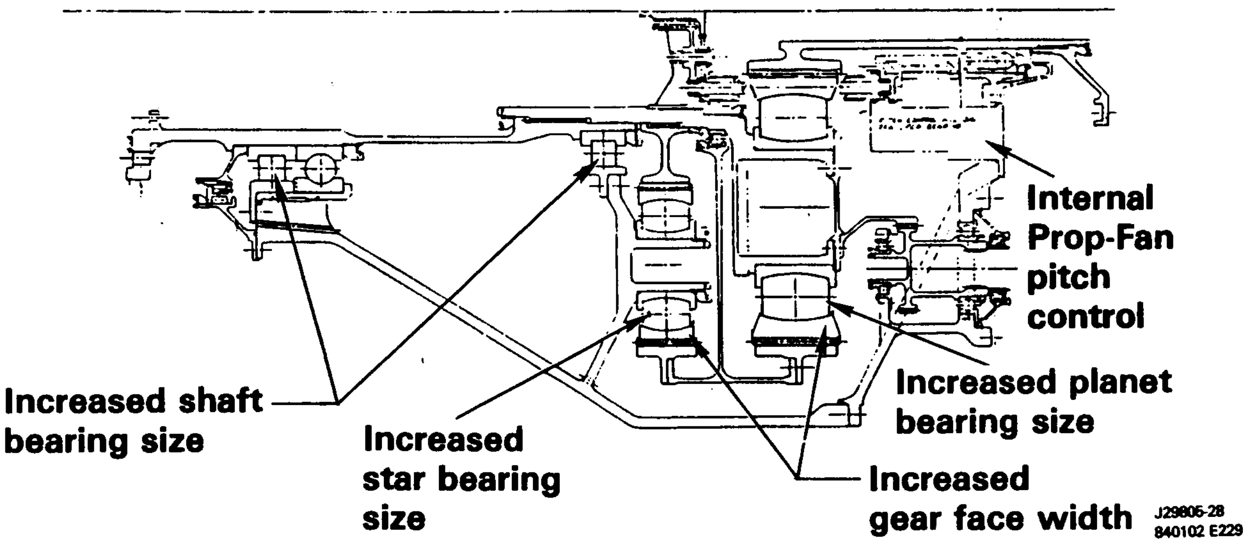


Figure 3.2-3 Current Technology Gearbox - The gearbox is larger in diameter, longer, heavier, and less efficient than the advanced technology gearbox.

With the current technology, the pitch control system must be contained within the gearbox. This adversely effects both reliability and maintenance costs.

3.3 Task VIII -- The Conceptual Design of a Single-Rotation Pitch Control

Hamilton Standard conducted a conceptual design study to provide an advanced flight-weight pitch control unit which is compatible with the in-line gearbox design. Prior to the conceptual design, Hamilton Standard performed a trade study to select a concept for further design effort under the APET contract. The selected concept, shown in Figure 3.3-1, incorporated a rotary hydraulic actuator with hydraulic and electrical power generated within the Prop-Fan assembly. The Prop-Fan also incorporates digital electronic control and a rotary capacitor signal transfer assembly. All pitch control components are of a modular design.

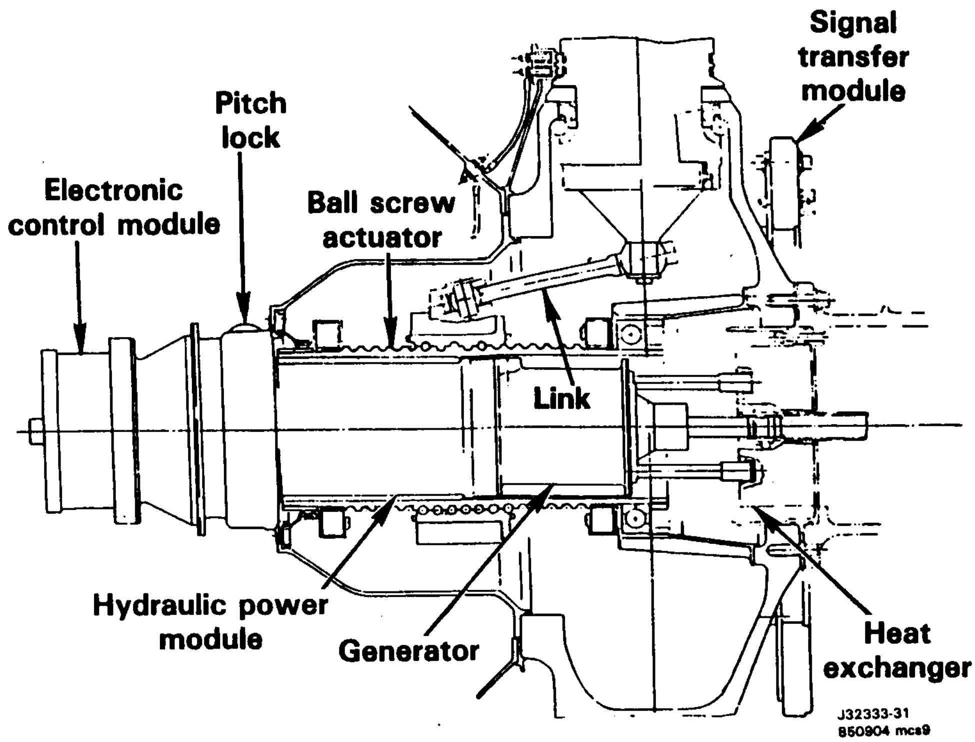


Figure 3.3-1 Pitch Control Drawing - This design incorporates a rotary hydraulic actuator with hydraulic and electrical power generated within the Prop-Fan assembly.

3.4 Task XI -- The Preliminary Design of a Counter-Rotation Gearbox

3.4.1 Optimization Studies

Results from an analysis conducted in Contract NAS3-23043 (Counter-Rotation Propeller Gearbox Study) concluded that the in-line differential planetary gearbox had the best overall performance rating out of ten concepts under consideration. Criteria for rating the concepts were reliability, efficiency, maintenance, acquisition cost, pitch control access, and weight. The in-line differential planetary concept proved to have the lightest weight, the fewest gears and bearings, the lowest acquisition and maintenance cost, and the highest efficiency.

The selection of the in-line differential planetary concept in the optimization study was essentially the starting point of the preliminary design effort. This gearbox concept appears in Figure 3.4-1.

To optimize the in-line differential planetary concept, we considered five different structural arrangements. The parameters from the conceptual studies were used in this evaluation. Results from the analysis indicated that the straddle-mounted arrangement was the best, because it reduced the installation length of the gearbox and provided better control of shaft/ring gear vibration.

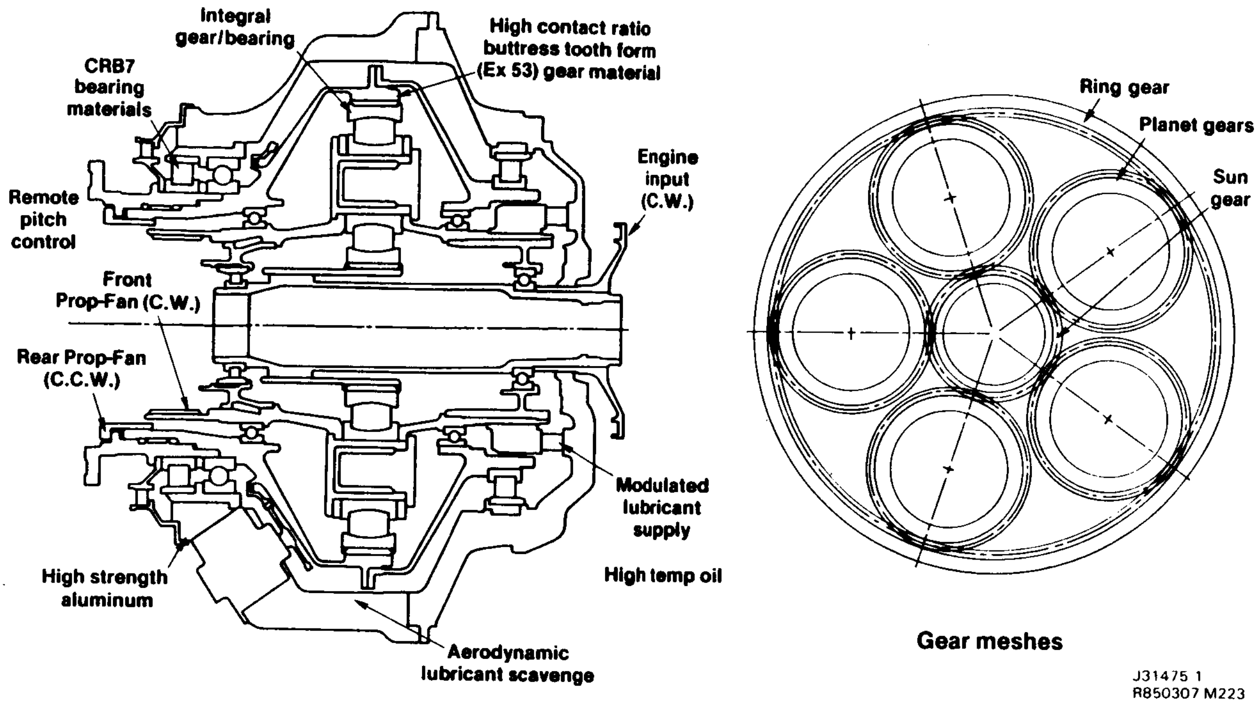


Figure 3.4-1 Counter-Rotating Differential Planetary Gearbox - The selection of this concept was the starting point of the preliminary design effort.

3.4.2 Mechanical Design

The selected in-line differential planetary configuration, shown in Figure 3.4-2, features advanced technologies in the design. Advanced materials increase reliability, lower weight, and permit using smaller gears and bearings. Consequently, advanced materials make for a lighter and smaller gearbox. To reduce weight further, advanced materials are also planned for the housing.

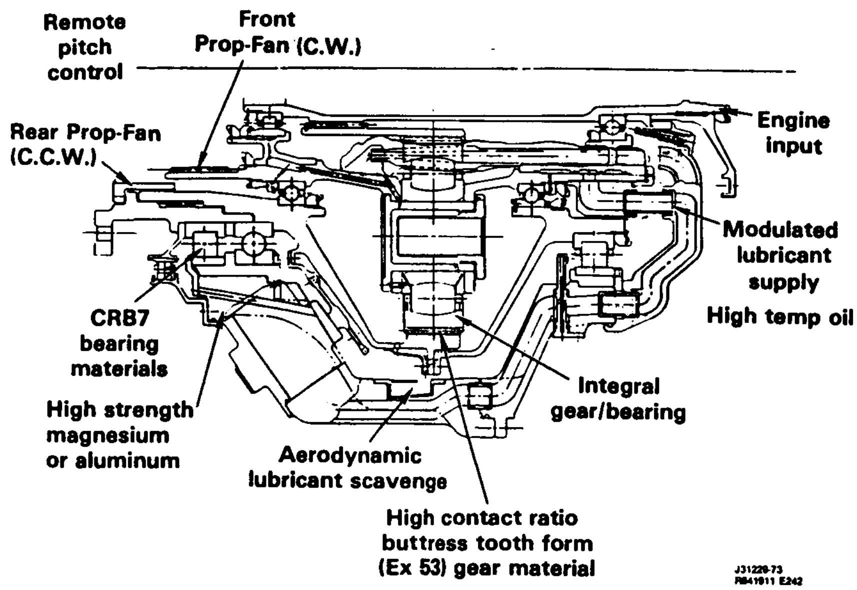


Figure 3.4-2 Advanced Technology Features - A remote pitch control, advanced materials, and an advanced lubrication system and lubricant provide greater reliability in a lighter gearbox.

As in the single-rotation configuration, a modulated lubrication supply and aerodynamic scavenge system is a design feature of the counter-rotation system. These features reduce the oil supply at part power and more effectively scavenge the oil. They contribute significantly to the predicted efficiency of 99 percent at cruise operation.

Component modularity is a very important part of this gearbox design. Greater accessibility to routine maintenance components such as the pitch control, for example, has drastically improved on-wing maintenance capabilities.

3.5 Task XII -- The Conceptual Design of a Counter-Rotation Pitch Control

Hamilton Standard designed an advanced, flight-weight pitch control unit which is compatible with the design of the counter-rotation gearbox. This design, which appears in Figure 3.5-1, is an adaptation of the previous pitch control developed in the trade studies and the single-rotation effort. Like the single-rotation design, the counter-rotation concept incorporates a rotary hydraulic actuator with hydraulic and electrical power generated within the Prop-Fan assembly. A digital electronic control and a rotary capacitor signal transfer assembly are also in the Prop-Fan. Like the single-rotation design, the counter-rotation pitch control incorporates a modular design.

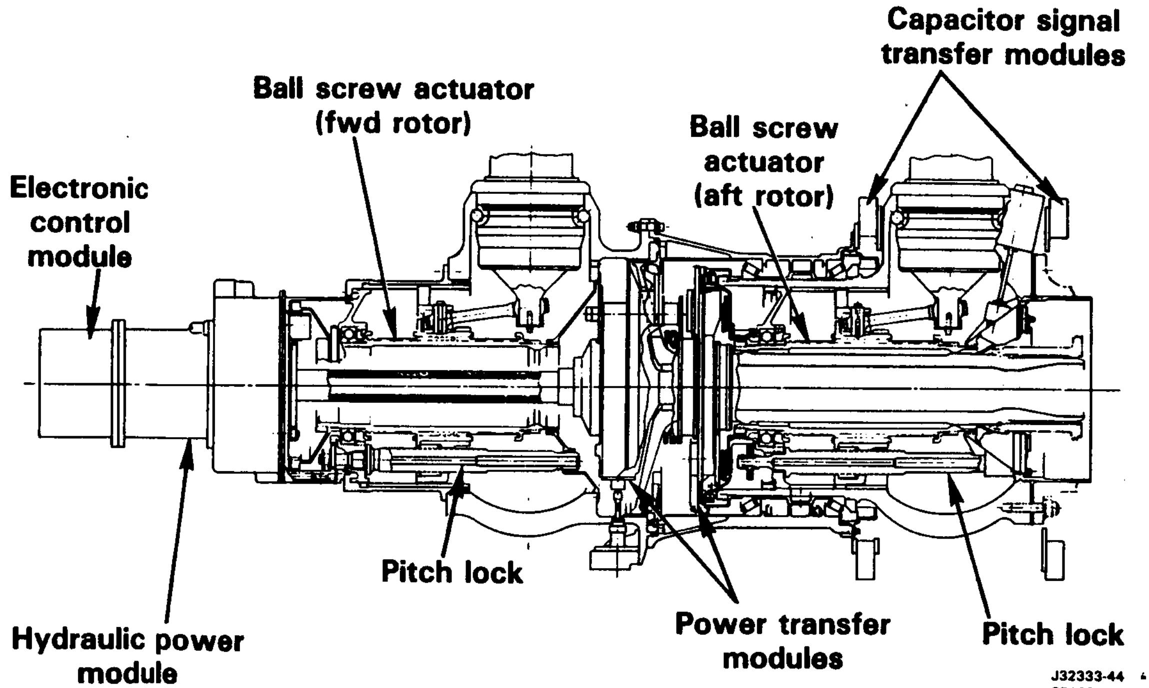


Figure 3.5-1 Rotary Hydraulic Pitch Control Concept - The design is an advanced, flight-weight pitch control unit which is compatible with the counter-rotation design developed in the trade studies and single-rotation effort.

3.6 Tasks IX and XIII -- Research and Technology Plan

To realize the full potential of an advanced, geared Prop-Fan propulsion system, test verification of several key technologies is necessary. For the gearbox, critical areas include materials, gear and bearing mechanical components, and lubrication system elements. For the pitch control, critical technologies are the capacitor signal transfer, high pressure hydraulics, and rotating electronics.

As shown in Figure 3.6-1, the plan for verifying these technologies consists of a multiyear effort ending in mid 1987. Individual gearbox technologies will first be evaluated and refined and then tested in the Pratt & Whitney Multipurpose Gearbox Rig Program at simulated Prop-Fan gearbox operating conditions. The four test builds of the multipurpose rig will ensure complete technology verification. It is necessary to progress with a program to verify the pitch control technologies shown in Figure 3.6-1.

The overall scope and timing of this plan are important steps toward bringing Prop-Fan propulsion and its large payoffs to the aviation industry by the early 1990's.

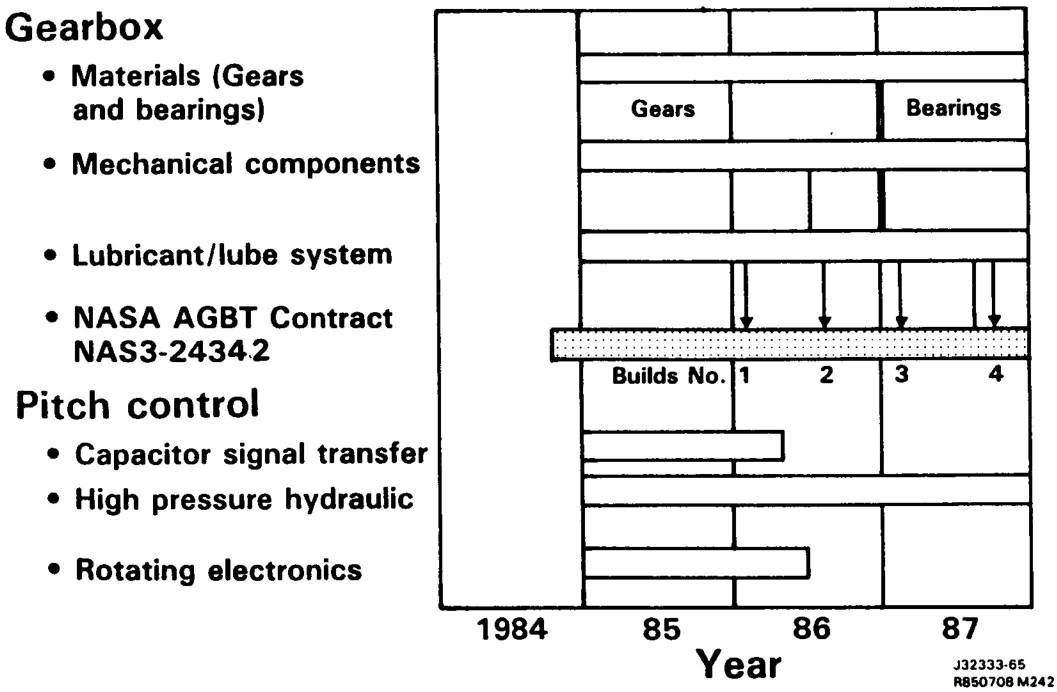


Figure 3.6-1 Gearbox/Pitch Control Overall Technology Plan - The four test builds of the multipurpose rig, supported by component rig programs, will ensure complete technology verification.

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SECTION 4.0
DISCUSSION OF RESULTS

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SECTION 4.1 -- DISCUSSION OF RESULTS

TASK VII -- THE PRELIMINARY DESIGN OF A SINGLE-ROTATION GEARBOX

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4.1 Task VII -- The Preliminary Design of a Single-Rotation Gearbox

4.1.1 Introduction

The objective of Task VII was to complete a preliminary mechanical design of a single-rotation reduction gearbox which would improve reliability and operating efficiency to meet the requirements of future Prop-Fan propulsion systems. The preliminary mechanical design and analyses consisted of three phases. They were: 1) the design and analysis of an advanced technology system; 2) the design and analysis of a current technology system; and 3) a comparison of the two designs to help assess and quantify the advantages of advanced technology.

The results of the previous Advanced Prop-Fan Engine Technology Definition Study identified the split path planetary system as the best design concept. Combining this concept with significant technological advances in materials, structures, and the lubrication system provided the framework for the present preliminary design of the gearbox.

The selected design configuration is a more refined version of the single-rotation, split path planetary system identified in the base APET Definition Study contract. This design emphasizes efficiency, long life, low maintenance cost, low initial cost, and high aircraft dispatch reliability.

4.1.2 Design Goals and Requirements

The design goals for an advanced technology gearbox reflect the requirements for improved reliability and efficiency. The overall design goals are to increase the mean time between unscheduled removal (MTBUR) to more than 15,000 hours and to increase cruise efficiency to 99 percent or more. The 15,000 hour MTBUR goal reflects the airline requirement that a Prop-Fan system must be as reliable as the fan section in present turbofan engines. The aim of the 99 percent efficiency goal is to minimize gearbox inefficiency due to internal power losses, thereby reducing the size and resulting drag of the air/oil cooler necessary for dissipating the heat attributed to this inefficiency. Past experience has shown that the MTBUR for turboprop gearboxes varies from 4000 to 8000 hrs., while cruise efficiency has been 98 percent. Making MTBUR three times greater and increasing cruise efficiency by a percentage point (i.e., cutting losses in half) will make advanced technology gearboxes considerably more cost effective than current turboprop gearboxes.

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In addition to requiring greater reliability and efficiency, the future Prop-Fan gearbox will operate in a propulsion system that requires two to three times the horsepower of present systems. Current turboprop gearbox drive systems cover a range of horsepower up to 5,000. Present projections indicate that future Prop-Fans will require gearbox drive systems in the range of 10,000 to 15,000 hp. To cover these larger powers, this preliminary design study focused on a 12,000 hp gearbox drive system.

Table 4.1.2-1 shows the advanced materials and lubricants that will improve durability and performance and contribute to a lighter system. Materials such as Cartech EX53 and Cartech CRB7 will allow higher gear design stresses and improved bearing life factors which will increase MTBUR. Cartech EX53 is 20 percent stronger than a conventional technology material, and its use will substantially reduce gear size. Advances in bearing materials are especially important, because they enable lightweight bearings to operate at higher speeds and greater loadings with much longer life. High strength aluminum or magnesium alloys will reduce housing weight, and advances in lubrication fluids will significantly improve load carrying ability, increase operating temperature, and reduce the size of the air/oil heat exchanger.

Table 4.1.2-1 Advanced Materials and Lubricants

	Current technology	Technology assumed available by 1988
Gears — materials	AMS 6265	Vasco X-2M or Cartech EX-53
Bending fatigue limit		
Unidirectional, psi	50,000(1)	60,000(1)
Reversed bending, psi	41,000(1)	49,000(1)
Hertz stress limit, psi	126,000(1)	151,000(1)
Pitch line velocity limit, ft/min	30,000	35,000
Bearings — materials	VIM VAR M50	Cartech CRB7
System design life requirement (L10), hr.	18,000	18,000
Material/lubrication life factor	6-12	20 — 30
Housings — materials	Aluminum, Magnesium	High strength Aluminum, Magnesium
Lubricant — fluids	Mil 23699 Type II	Synthesized Hydrocarbon Fluid (SHF)
Oil inlet temp, °F	150-210	210-270
Allowable temperature rise, °F	40-50	80-100
Load carrying ability, lb./in.	2000-3500	4000-4500
Scoring temperature index, °F	276	400

(1) Typical gear allowable stress — 3 sigma with a coefficient of variation = 0.1 for 10^{10} cycles.

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Several ground rules ensured that the gearbox design is acceptable to the aviation industry. Major design prerequisites governing the design include the following:

- Maximize reliability by transferring Prop-Fan loads directly to the aircraft -- By providing the most direct load path from the Prop-Fan to the aircraft, deflections or slopes that could cause gear tooth or bearing wear are minimal. Vibration isolators are used when mounting the gearbox to the aircraft to reduce the prop and gearbox vibration transmitted to the airplane.

- Minimize the gear misalignment produced by the deflection of the gearbox and drive shaft -- By using state-of-the-art finite element analysis, the design of the gearbox housing, shafts, and gears reduces slope and deflection at critical bearing and gear mesh locations. This significantly improves reliability, because most gearbox durability problems result from excessive misalignments.

- Minimize the number of bearings and gears -- Each bearing and gear mesh is a critical item in determining the gearbox reliability. The reliability of the gearbox improves directly with the elimination of a bearing or gear mesh.

- Provide an easily maintainable, external pitch control -- One of the major maintenance problems with current in-line gearboxes is the inaccessible, internal location of the Prop-Fan pitch control mechanism. Any maintenance of this pitch control requires pulling the gearbox from the aircraft and disassembling it to gain access to the pitch control. To improve accessibility, the pitch control is separate from the gearbox, so maintenance of the pitch control unit can proceed without removing the gearbox from the aircraft.

- Maximize component modularity -- Modular construction enables on-wing maintenance capability and maximum aircraft dispatch reliability. By designing systems composed of sub modules for routine maintenance activities, maintenance doesn't require removing the power plant. Examples of such systems are oil jets, carbon seals, the pitch control, etc.

Table 4.1.2-2 summarizes specific operating parameters and Prop-Fan drive and cooling requirements. The operating parameters include transferring 12,000 hp to the Prop-Fan blades at 1,145 rpm. This matches the drive requirement for a ten-bladed Prop-Fan with a diameter of 4.07 m (13.35 ft). The loads that the Prop-Fan imposes on the gearbox include the Prop-Fan weight of 635 kg (1,400 lbs), a thrust load of 88,520 N (19,900 lbs), and the 1P shear and moment loads. The Prop-Fan cooling requirements include: 1) providing 13 lbs of oil a minute to the Prop-Fan pitch control unit at a maximum oil supply temperature of 76.7°C (170°F) and a pressure of 0.483 MPa (70 lbs psi), and 2) accepting this oil back after it has cooled the Prop-Fan pitch control system.

Table 4.1.2-2 Prop-Fan Gearbox Design Characteristics

Prop-Fan drive requirements	
Max power, HP	12,000 -
Gear ratio	7-11
Prop diameter, M (ft)	3.5 (11.6)
Tip speed, m/sec (ft/sec)	228.6 (750)
Output shaft speed, rpm	1233
Max output torque, N-m (ft-lb)	38054 (28067)/ 31135 (22964)
Max total prop thrust, N (lb)	92656 (20,830)
Max '1P' moment, N-m (ft-lb)	8921 (6580)/9857 (7270)
Max '1P' shear, N (lb)	7317 (1645)/6361 (1430)
Max gyro moment at 0.2 rad/sec, N-m (ft-lb)	3186 (2350)/3186 (2350)

*Distance from CG to prop/gearbox shaft flange interface

Prop-Fan cooling requirements	
Oil flow kg/min (lb/min)	15.4 (34)
Max oil inlet temperature, °C (°F)	76.7 (170)
Max inlet oil pressure	Ambient
Max temperature rise, ΔT, °C (°F)	10.0 (50)

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In addition to the above specific operating parameters, the variations of operating conditions throughout the flight envelope also contributed to the design of the gearbox. Table 4.1.2-3 summarizes the flight mission profile chosen to represent a typical short range mission (741.3 km or 400 nmi). This typical mission assumes a flight profile in which most of the time is spent climbing and descending from a cruise altitude of 10,668 m (35,000 ft). While a cruise mach number of 0.8 was used in this study, previous work has indicated that whether the cruise speed is Mach 0.7 or 0.8, there is no significant effect on flight duration times.

Table 4.1.2-3 Flight Mission Profile For Gearbox Duty Cycle Analysis

Condition	Duration (minutes)	Altitude 304.8 M (1000 ft)	Flight speed (MN)	Power (% max)	Prop-Fan speed (% max)
Taxi (Ground idle)	5.0	0	0	2 – 5	20 – 70
Takeoff	1.5	0 – 1.5	0 – 0.39	100	95 – 100
Climb	2.4	1.5 – 10	0.39 – 0.5	88 – 81.3	100
	3.8	10 – 20	0.5 – 0.6	81.3 – 70	100
	8.9	20 – 30	0.6 – 0.74	70 – 58.7	100
	5.9	30 – 35	0.74 – 0.8	58.7 – 53.3	100
Cruise	20.0	35	0.8	43.3	100
Descent	20.0	Variable	Variable	2 – 5	30 – 70
Approach	3.0	Variable	Variable	20 – 25	75 – 100
Reverse	0.5	0	0.2 – 0	22 – 6	60 – 80
Taxi (ground idle)	5.0	0	0	2 – 5	20 – 70

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4.1.3 Reference Gearbox Design

NASA and Pratt & Whitney's APET definition study provided the starting point for the present preliminary design study. This previous effort surveyed all known gearbox drive concepts and identified five in-line and four offset concepts for study. These concepts are shown in Figure 4.1.3-1 and Figure 4.1.3-2.

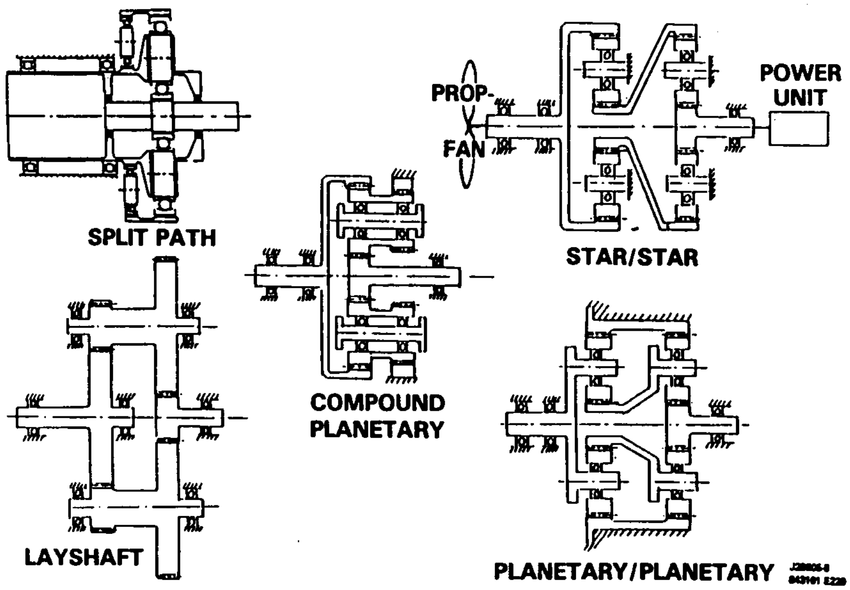


Figure 4.1.3-1 In-Line Gearbox Concepts From NASA APET Study - These selections result from NASA and Pratt & Whitney's APET definition study (Volume I).

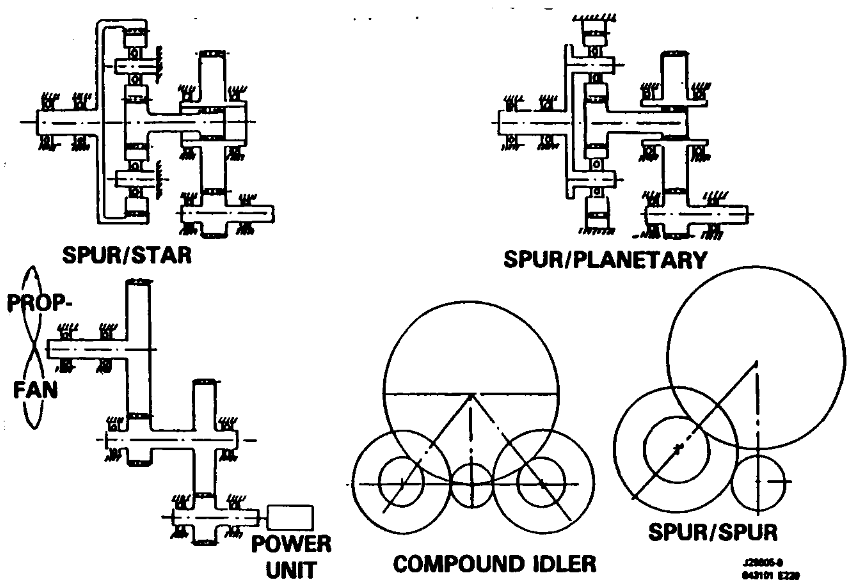


Figure 4.1.3-2 Offset Gearbox Concepts From NASA APET Study - The selections result from NASA and Pratt & Whitney's APET definition study (Volume I).

The five in-line concepts were the planetary/planetary system, the star/star system, the compound planetary, system, the lay shaft arrangement, and the split path concept.

Of the five in-line concepts, the split path was selected for further study, because it offers the lightest weight and smallest diameter. The planetary/planetary system was rejected, because the first stage planetary cage speed was so high that the resultant centrifugal force on the bearings significantly reduced their life. The star/star system was eliminated, because it was too large and heavy. The compound planetary had a competitive weight, but the centrifugal force from the pinion gears degraded the pinion bearing life. The simple lay shaft arrangement was rejected, because it was very large and heavy.

The four offset concepts were the spur/spur system, the spur/planetary system, the spur/star system, and the compound idler system.

Of the four offset concepts, the compound idler gear system was selected, because it is simple and highly efficient. The spur/spur system was rejected, because it was too large and heavy. The spur/planetary system offered a competitive weight, but it compromised efficiency and had a relatively large spur stage. The spur/star system was rejected, because it was relatively heavy due to the loss in reduction ratio associated with a star system.

Comparing the split path and compound idler gearbox concepts determined which arrangement had the best overall performance. The results, summarized in Figure 4.1.3-3, indicate that the split path is appreciably better than the compound idler, offering 1.4 percent lower fuel burn and 0.4 percent lower operating costs.

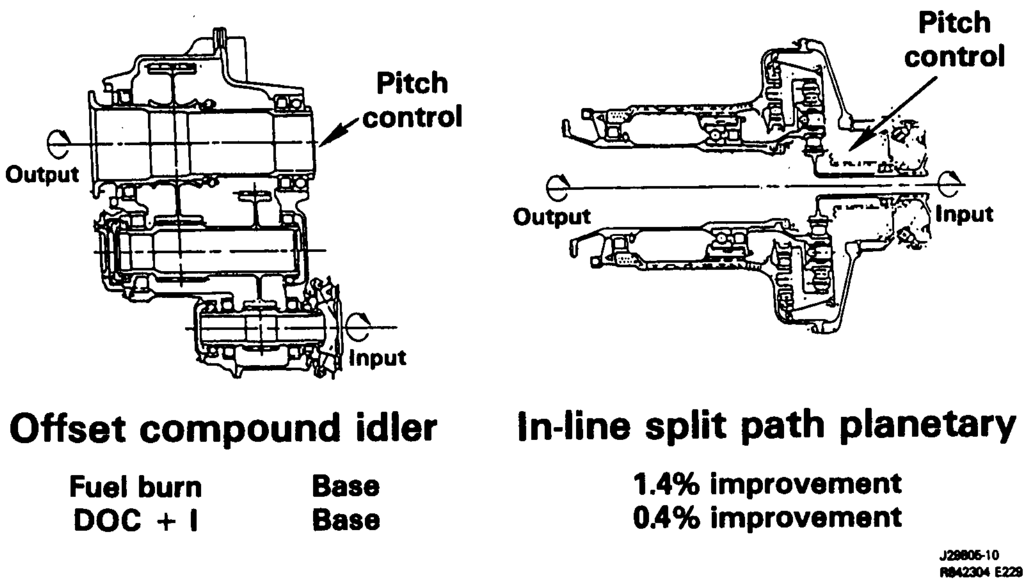


Figure 4.1.3-3 Best Gearbox Concepts From APET Definition Study (Volume I) - The split path concept is significantly better than the compound idler.

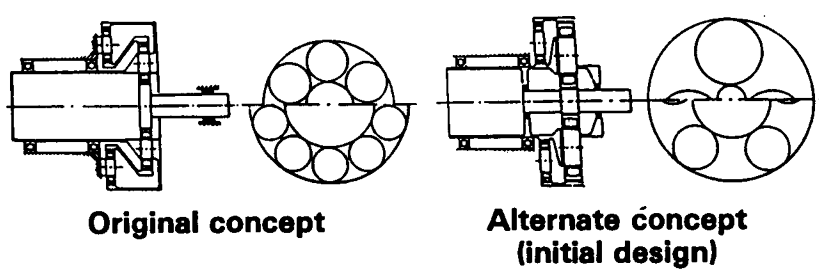
The benefits of the in-line split path are due to its weight reduction of 93 kg (205 lbs) and diameter reduction of 5.1 cm (two in) which also results in a smaller nacelle diameter. However, while the split path configuration is relatively compact, accessibility to the internally located pitch control was a major concern requiring further study. The relative complexity of the split path gearbox (the large number of gears and bearings) was also an area requiring further study.

In summary, this early study identified that the split path concept was preferable and that a preliminary design study should be conducted to:

- o Remove the pitch control from the gearbox,
- o Simplify the gearbox by reducing the number of gears and bearings, and
- o Provide capability for driving the Prop-Fan located on the other side of the fuselage in the opposite direction to reduce cabin noise.

4.1.4 Refinement Studies

Prior to the preliminary mechanical design effort, Pratt & Whitney conducted a series of analyses aimed at refining the design and performance of the reference split path gearbox concept. This original concept was relatively light and compact, but it contained 22 gears and 19 bearings which adversely impacted gearbox cost, maintenance cost, and durability. Figure 4.1.4-1 shows diagrams of the original concept and an alternate concept, as well as the goals for refining the original concept. Refining the original design provided an opportunity to simplify this design and to improve efficiency, durability, interface with the pitch control, and most important, commonality of components for opposite rotation.



Goals

- **Reduce number of gears and bearings**
- **Improve efficiency**
- **Opposite rotation with maximum hardware commonality**
- **Improve durability**
- **Improve maintenance**
- **Incorporate advanced technology remote pitch control**

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Figure 4.1.4-1 Split Path Concept Optimization - Refining the design improved efficiency, durability, interface with the pitch control, and component commonality with opposite rotation.

4.1.4.1 Design Commonality for Opposite Rotation

The split path gearbox concept allows both conventional and opposite output shaft rotations without the separate reversing stage which is necessary in other single-rotation gearboxes, the compound idler design for example. This is important, because the power loss associated with a separate reversing stage is substantial. The alternate split path concept is even more advantageous, because the only essential prerequisite for obtaining opposite rotation is changing the connections between planetary members. As a result, the switch from conventional to opposite hand rotation does not degrade the system's efficiency. In fact, there is a high degree of commonality.

Figure 4.1.4-2 illustrates the method of achieving opposite hand rotation. The shaded areas of each diagram identify gears, gear shafts, and bearings that are common to both conventional and opposite rotation. The unshaded connecting shafts are the only parts unique to each gearbox. The net result is that the switch from conventional to opposite rotation requires only a few new parts.

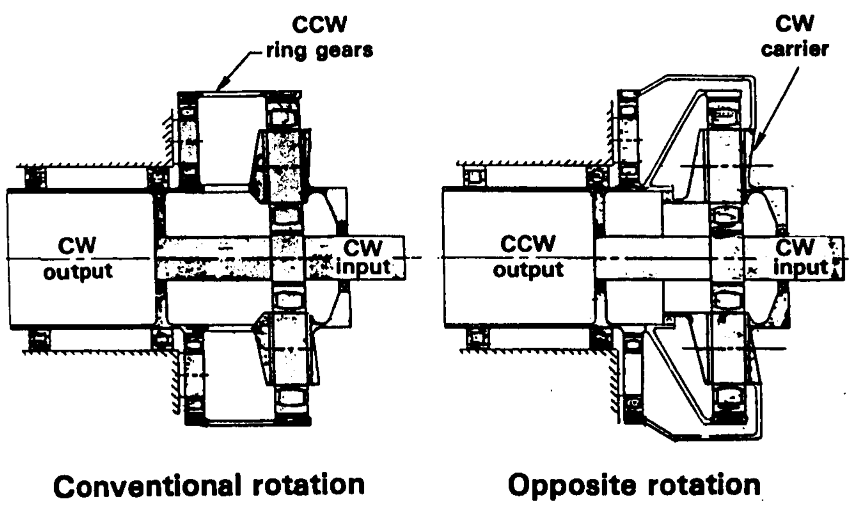


Figure 4.1.4-2 Method of Achieving Opposite Rotation - Changing from conventional to opposite rotation requires only a few new parts.

4.1.4.2 Gear Ratio Split Analysis

Optimizing the gear ratio split for the alternate split path gearbox centered on finding a compromise between the weight of gears and bearings and the number of gears and bearings.

Figure 4.1.4-3 shows constant reduction ratio lines plotted on the planet stage and star stage ratio coordinates (R_1 and R_2, respectively) which identify potential stage ratio combinations for a conventional rotation system;

where reduction ratio = $1 + R_1 + \frac{R_1}{R_2}$

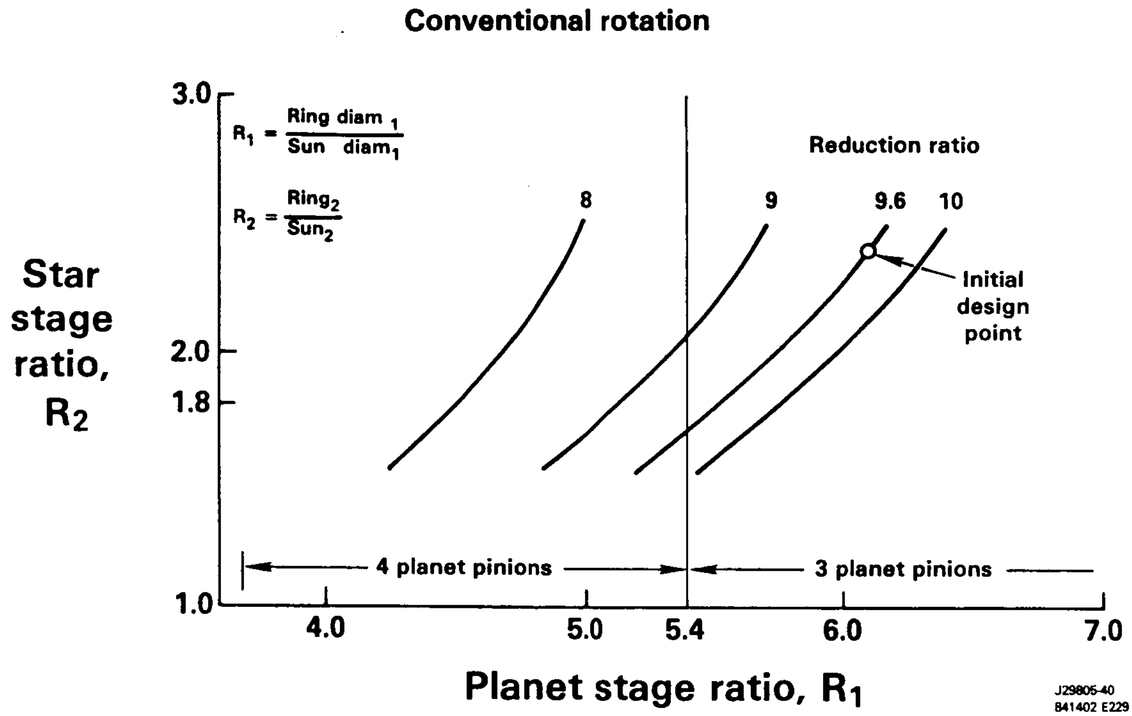


Figure 4.1.4-3 Gear Ratio Split Optimization - This optimization focused on finding a compromise between the weight of gears and bearings and the number of gears and bearings.

The initial alternate split path design assumed stage ratios of 6.1 and 2.4 for a 9.6 reduction ratio (indicated by the initial design point on the chart). Three planet pinions and five star pinions represent a minimum practical part count. However, reducing the planet stage ratio to less than 5.4 lowers weight. A stage ratio reduction of this size is enough to allow the use of four pinions. This makes the stage size significantly smaller. The star stage ratio should be held above 1.8 for adequate star pinion bearing size and life. Under these constraints, the reduction ratio becomes 9.2. Figure 4.1.4-4 illustrates the compromise outlined above.

Figure 4.1.4-5 shows a second set of constant reduction ratio lines on the stage ratio coordinates. These lines identify the potential planet stage and star ratio combinations for opposite rotation;

$$\text{where reduction ratio} = -\left(R_1 + \frac{1 + R_1}{R_2}\right)$$

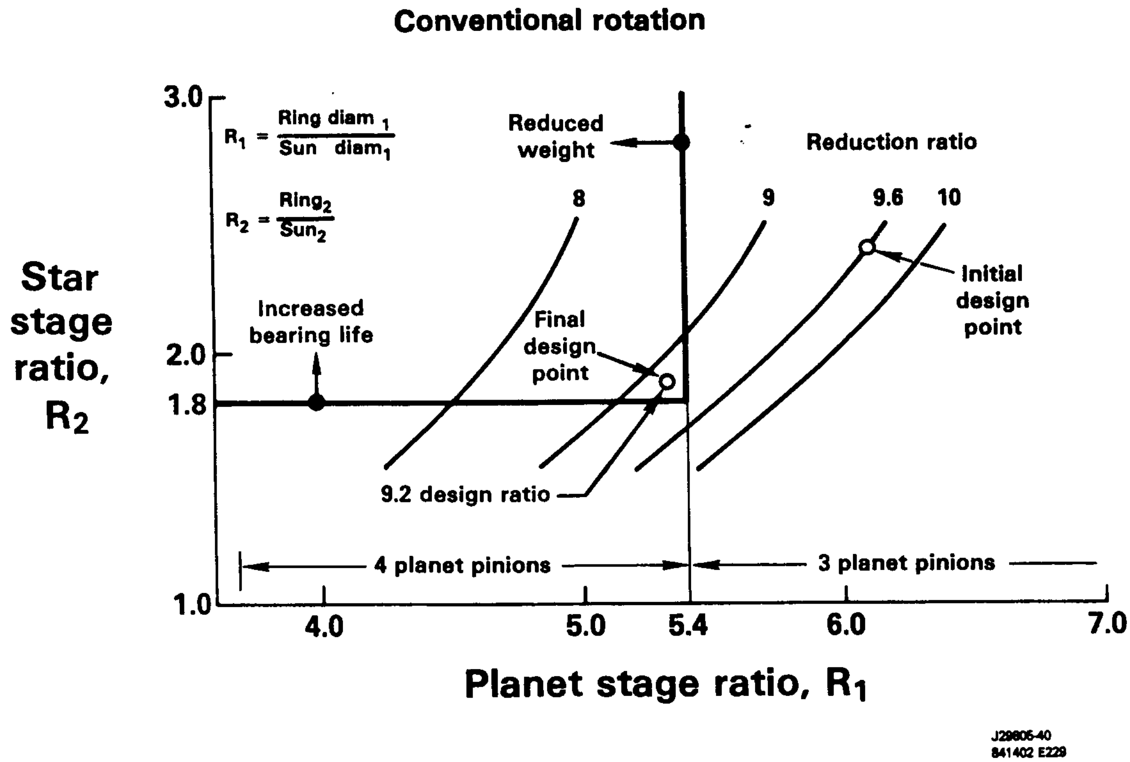


Figure 4.1.4-4 Gear Ratio Split Optimization - According to this compromise, the star stage ratio is held above 1.8 and reduction ratio becomes 9.2.

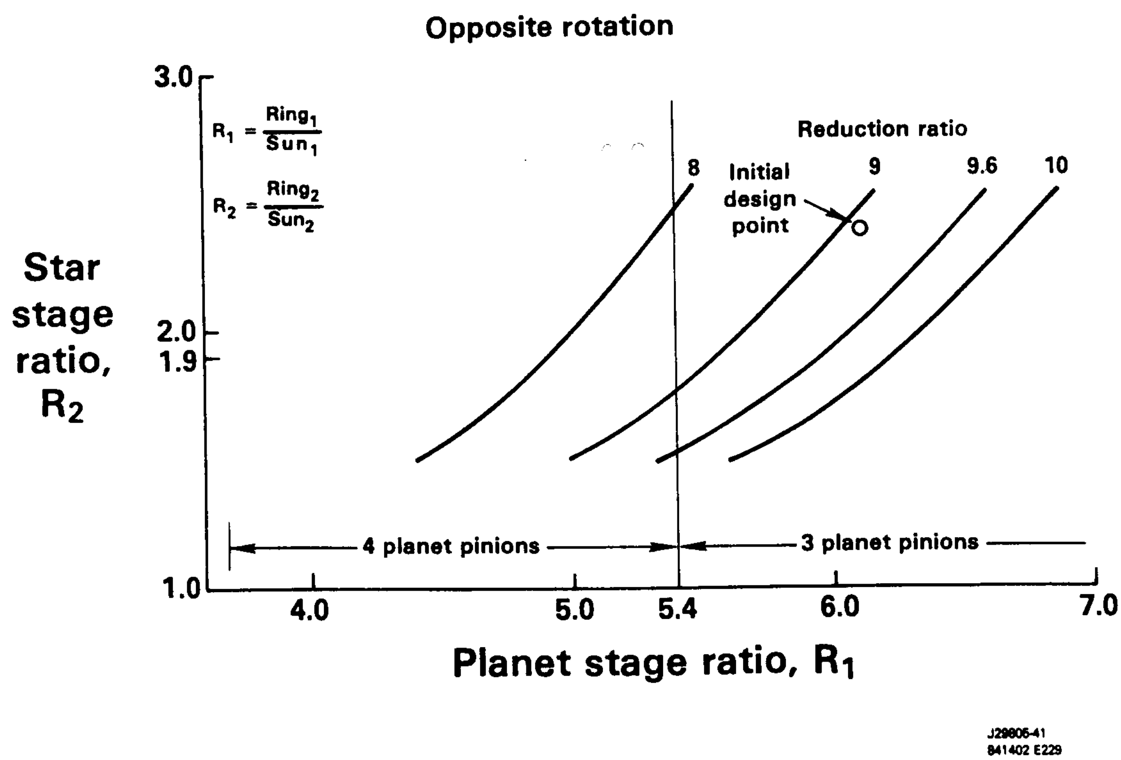


Figure 4.1.4-5 Gear Ratio Split Optimization - This chart shows the potential planet stage and star ratio combinations for opposite rotation.

At the 6.1 and 2.4 stage ratios used in the initial design, the opposite rotation reduction ratio is 9.1 (indicated by the initial design point in Figure 4.1.4-6). Applying the same logic as previously used for conventional rotation, the final design with four planet pinions results in a design point reduction ratio of 8.73. Therefore, by changing the way the gears connect to the drive system, opposite rotation can be accomplished using the same gears. The only compromise is a slight change in reduction ratio, since the conventional rotation reduction ratio is 9.2 and the opposite rotation is 8.73. This is equivalent to about a 500 rpm difference in power turbine speed (10,000 rpm input speed vs. 10,500 rpm). The power turbine design will be optimized at 10,150 rpm; the resultant difference of 250 rpm will have a negligible affect on performance.

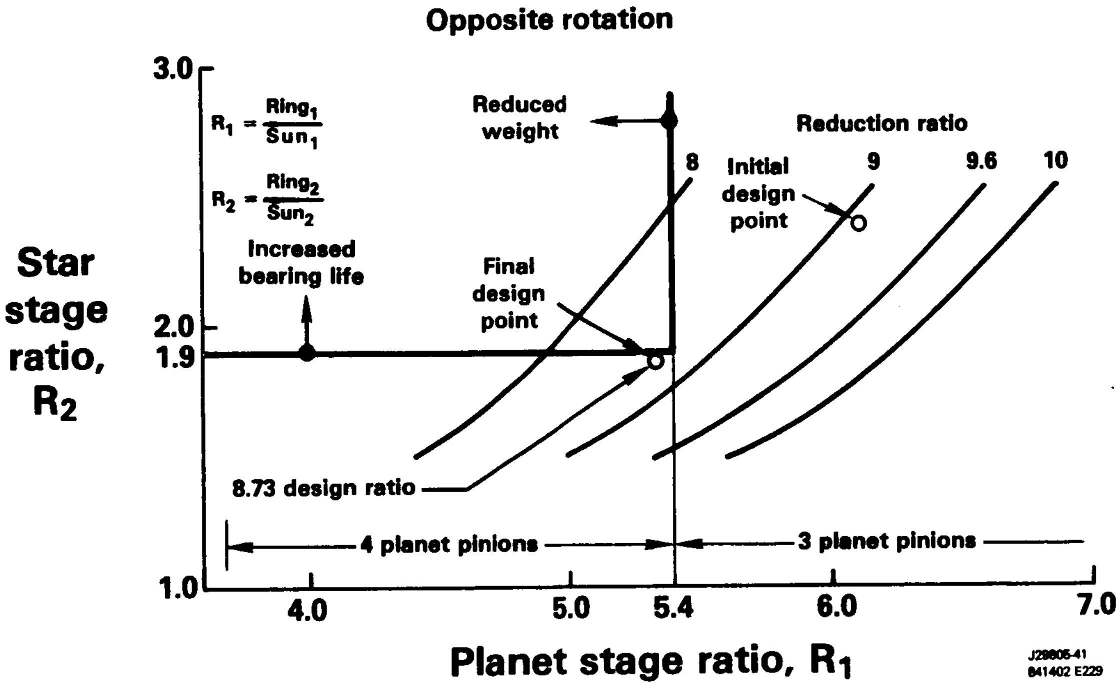


Figure 4.1.4-6 Gear Ratio Split Optimization - The final design point reduction ratio is 8.73.

Selection of reduced planet and star stage ratios and increased pinion count allows the ring gear diameter in both stages to be smaller. As Figure 4.1.4-7 shows, this reduces the gearbox envelope by about 5.1 cm (two in) and contributes significantly to ensuring a more compact and lighter gearbox arrangement.

On the basis of these refinement analyses, Pratt & Whitney has configured a final alternate split path gearbox. Figure 4.1.4-8 outlines the advantages of this configuration when compared to the original concept. The alternate split path gearbox contains five fewer gears and four fewer bearings for a significantly smaller number of parts. Moreover, the system's durability and maintenance costs improve substantially, while efficiency improves moderately. A major achievement mentioned in Figure 4.1.4-8 is the remote or external location of the pitch control module. This is particularly important in terms of maintainability and overall design simplicity.

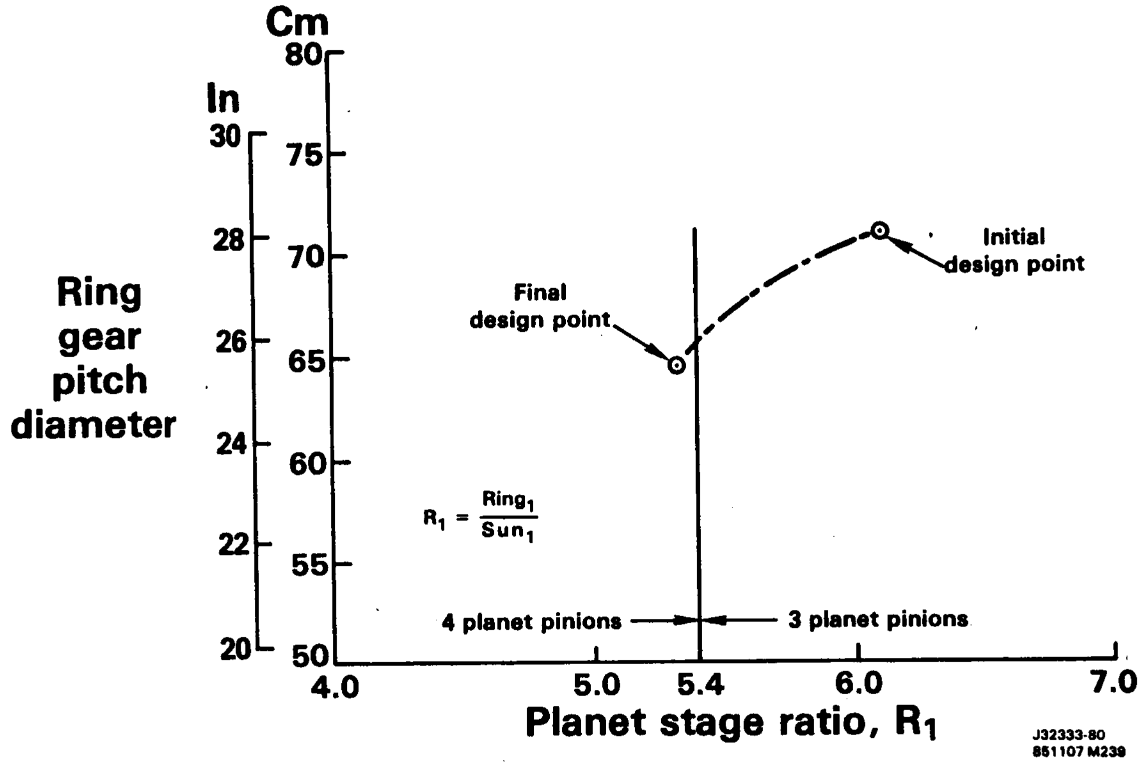
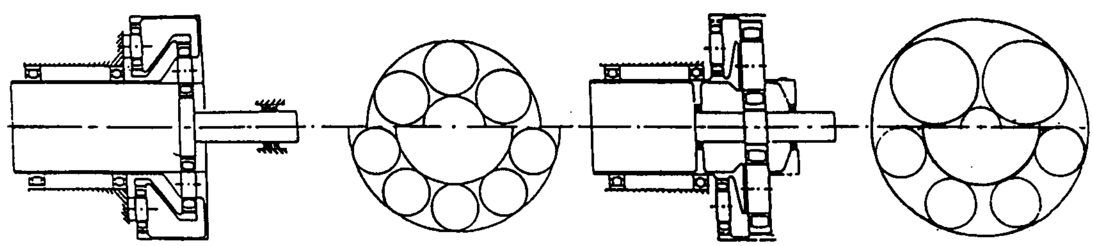


Figure 4.1.4-7 Gear Ratio Split Optimization Reduces Gear Size - The total reduction of the gearbox envelope is about two inches.



Original concept

Alternate concept (final design)

Achievements

- **5 fewer gears/4 fewer bearings**
- **Improved efficiency 0.3%**
- **Achieved commonality for opposite rotation**
- **Improved durability 12% (MTBUR)**
- **Lowered maintenance cost 25%**
- **Incorporated advanced technology remote pitch control**

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Figure 4.1.4-8 Advantages of the Selected Configuration - The alternate design is smaller and lighter. It is also more efficient, maintainable, and durable.

4.1.5 Advanced Technology Gearbox Preliminary Mechanical Design

The advanced technology gearbox preliminary mechanical design effort consisted of three program elements. The first was the design and analysis of an advanced technology system. The second was the design and analysis of a current technology system, and the third was a comparative assessment of the two designs to quantify the advantages of advanced technology.

For the advanced technology gearbox design (discussed in Section 4.1.5.1), several gear arrangements were evaluated to determine the optimal design. Weight costs, parts count, durability, and efficiency are some of the parameters which influenced the selection of the final split path configuration. This design covers both conventional and opposite hand rotation and incorporates a number of advanced technologies.

The preliminary mechanical design of a current technology gearbox (discussed in Section 4.1.5.2) provided a base to which the advanced technology design could be compared. The current technology gearbox incorporates state-of-the-art materials, bearings, and lubricants. This design covers conventional rotation only.

The comparative assessment of the two designs (discussed in Section 4.1.5.3) entailed structural analyses, performance and economic assessments, and integration analysis.

4.1.5.1 The Mechanical Design of the Advanced Technology Gearbox

This section will discuss the features and maintainability of the conventional and opposite hand rotation advanced technology gearboxes.

4.1.5.1.1 Design Description

The split path gearbox has two stages which are coupled through the ring gear of the first stage and the ring gear of the second stage. The second stage carrier is the only ground link in the transmission. The second stage sun gear is connected to the output shaft. This, combined with the first stage carrier providing torque reaction, transmits power directly to the output shaft. The proportion of the total power the first stage delivers is dependant on the individual stage's ratio and is 66.5 percent; the second stage delivers the balance of 33.5 percent to the output shaft. The arrangement has inherent advantages over the original split path gearbox candidate. A reduction of five gears and four bearings results in a substantial improvement in durability and maintenance cost and a moderate improvement in efficiency. Figure 4.1.5-1 illustrates the path of the transmission power through the gearbox.

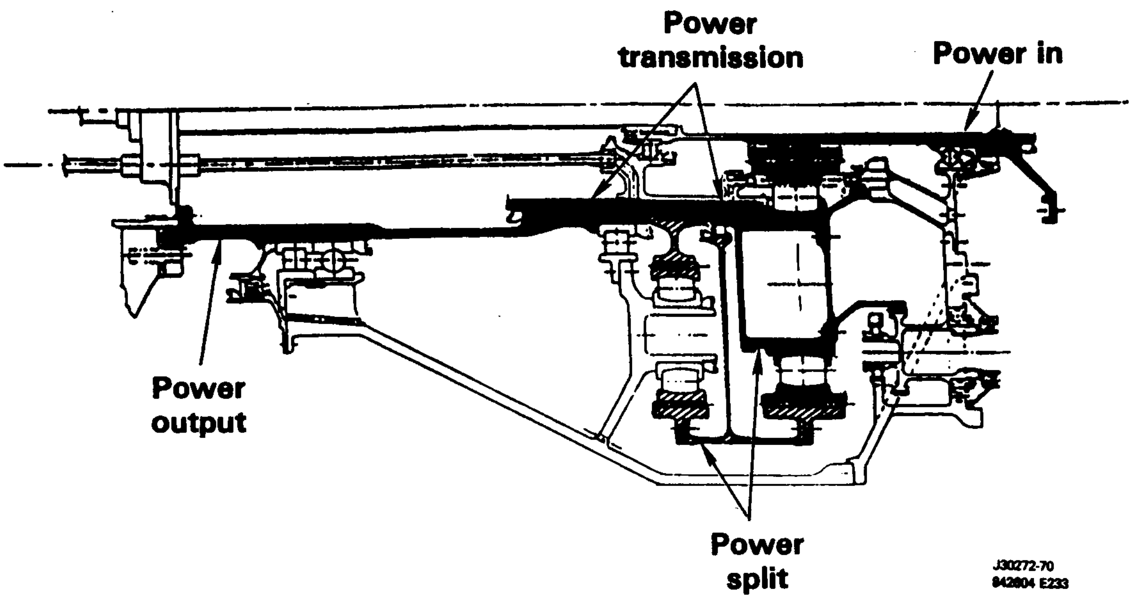

Figure 4.1.5-1 Split Path Two Stage Planetary Gearbox Power Split In Percent Is 66.5/33.5 - This figure shows the path of the transmission power through the gearbox.

For opposite hand rotation, all gears and bearings are the same as those necessary for conventional rotation. As shown in Figure 4.1.5-2, the six new parts necessary are: three connecting hubs, a modified carrier, a modified output shaft, and an intershaft bearing. Opposite hand rotation can be provided by freeing the planetary stage's carrier from the output shaft and connecting it to the ring gear of the star stage. In addition, the ring gear of the planetary stage connects to the output shaft. With opposite hand rotation, the reduction ratio changes from 9.2/1 to 8.73/1. This is equivalent to a change of about 500 rpm (an input speed of 10,000 vs. 10,500 rpm). The power split changes to 63 percent/37 percent, and the carrier speed decreases from 1144 rpm to 610 rpm. The lower carrier speed reduces the centrifugal load generated by the planet gear and increases bearing life. A detailed description of the bearing loads is in Section 4.1.5.3.

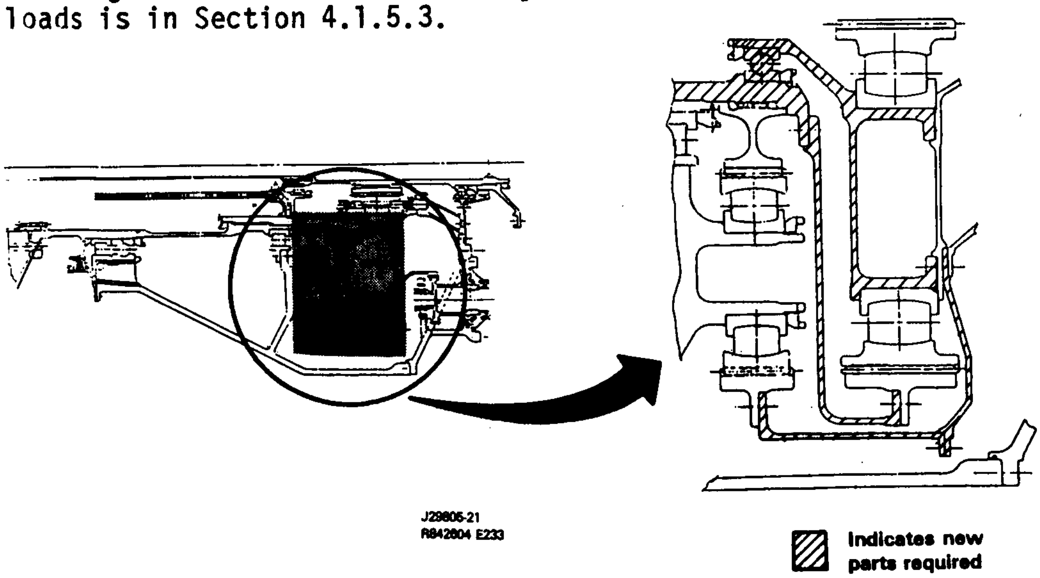

Figure 4.1.5-2 Minimum Number of New Parts Required for Opposite Hand Rotation - The six parts are three connecting hubs, a modified carrier, a modified output shaft, and an intershaft bearing.

Applying advanced technology to the alternate split path gearbox results in fewer parts, greater reliability, greater efficiency, easier and less frequent maintenance, and longer life. Some of the technological improvements are:

- o Advanced materials for both gears and bearings,
- o Component modularity,
- o A modulated lubrication system,
- o An aerodynamic oil scavenge system, and
- o An advanced gear tooth form.

Figure 4.1.5-3 shows where these features appear on the gearbox.

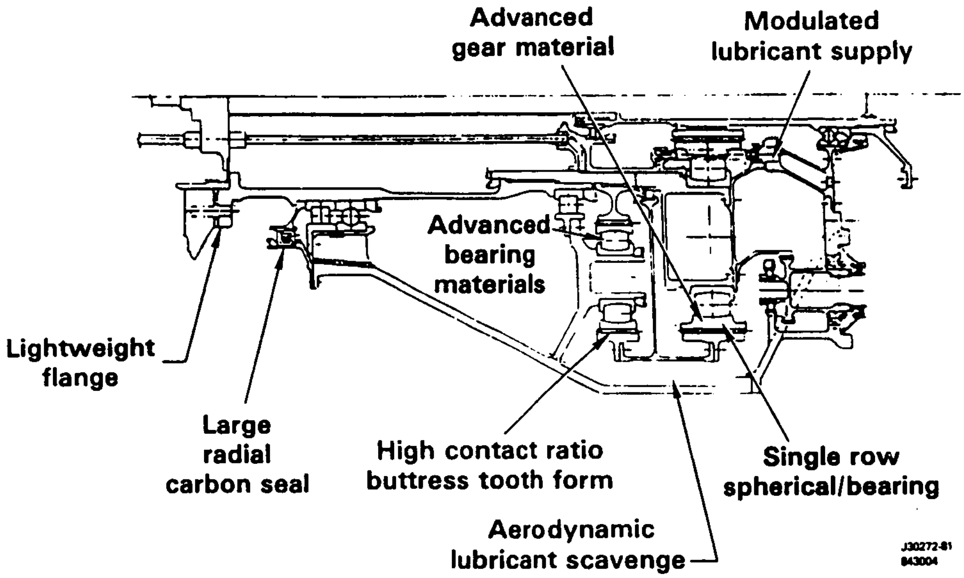

Figure 4.1.5-3 Advanced Gearbox Technology Features -- Using advanced technology results in fewer parts, greater reliability, greater efficiency, easier and less frequent maintenance, and longer life.

Extensively applying high strength materials in the gearbox design permits smaller and lighter bearings, gears, and housings. For example, bearings are 30 percent smaller, and the gear face width is 20 percent narrower. Smaller and lighter bearings and gears reduce the size of the gearbox envelope making it more compact and easier to integrate into the airframe.

Component modularity is important for lowering maintenance costs. The advanced technology gearbox design uses fewer components and simplifies maintenance by making normal maintenance parts readily accessible for on-wing replacement. A major design accomplishment in this area is the removal of the pitch control system to a location external to the gearbox.

As discussed in the lubrication section, the modulated lubrication system is a unique system that promotes greater efficiency and cuts cruise cooling requirements by 50 percent. Included in the lubrication system is an aerodynamic oil scavenging system which reduces power loss and monitors the in-line quality of oil.

The gearbox also provides cooling oil, generator drive power, and hydraulic drive power to the Prop-Fan unit. The cooling oil transfer tubes, mounted on the carrier/shaft, extend forward into the Prop-Fan unit and supply pressurized oil to the Prop-Fan system. This oil flows through the Prop-Fan unit and returns to the gearbox to be scavenged with the gearbox oil. A shaft which is splined to the sun gear drive shaft drives the Prop-Fan generator and hydraulic pump.

4.1.5.1.2 Maintainability

An important design issue is the maintainability of the gearbox system. The primary maintenance considerations guiding the design of the gearbox are:

- o Modularity,
- o Improved accessibility to major components,
- o On-wing maintenance, and
- o Condition monitoring.

The major subassemblies in the Prop-Fan system are the pitch control, the Prop-Fan, and the gearbox modules. For easy removal, each module has minimal interface requirements. As mentioned earlier, the prominent feature is the removal of the pitch control from the gearbox's internal structure. Moving the pitch control from the interior of the gearbox to a remote location substantially simplifies maintenance of both the pitch control and the gearbox.

To minimize the aircraft downtime, routine gearbox maintenance items such as the gearbox filters, carbon seals, oil pumps, and oil nozzle jets are accessible for on-wing maintenance. Figure 4.1.5-4 shows where these items appear on the gearbox. The prop side radial carbon seal and seal land are both replaceable without removing the gearbox. The carbon seal elements can be replaced without removing the Prop-Fan (segmented carbon elements), but the seal land replacement requires removing the Prop-Fan module. Oil pumps, filters, and approximately 70 percent of the oil nozzles are on-wing replaceable. The rear external carbon seal and the oil transfer carbon seals all require removing the engine drive shaft before replacement.

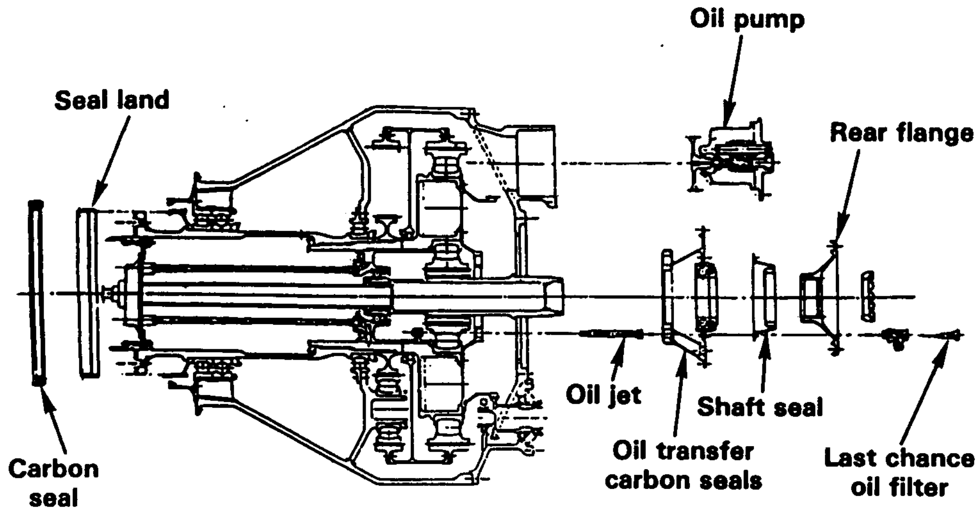

Figure 4.1.5-4 On-Wing Maintenance Capabilities - Easily maintained items such as gearbox filters, carbon seals, oil pumps, and oil nozzle jets minimize aircraft downtime.

A condition monitoring system improves both the maintainability and reliability of the gearbox. Oil in and out temperature and pressure measurements are closely monitored. In addition, a magnetic chip detector and a vibration/noise sensor monitor the oil to detect any debris that might restrict the flow and any change in vibration/noise characteristics that would indicate deterioration of the gears and bearings. A combined deaeratory debris monitor is part of the scavenge loop of the lubrication system. This unit is highly efficient in trapping and detecting debris, because it has full flow-monitoring characteristics. The monitor system, which will be upstream of the filters, offers a very high probability of a first-pass catch of a failure related particle. Ultra fine filtration does not affect this system.

The early detection of lubrication system problems by capturing and counting magnetic wear particles may save many hours of unscheduled maintenance and avoid the dangers of unexpected failures. The detection system interfaces with a computerized condition monitoring system and registers each particle capture as it occurs. The severity of wear, determined by the size of captured particles and the rate at which they are captured, can then be translated into permissible hours of safe operation before removal is necessary.

4.1.5.2 The Mechanical Design of the Current Technology Gearbox

Design features and maintainability of the current technology gearbox are discussed in this section.

4.1.5.2.1 Design Features

As stated earlier, the preliminary mechanical design of a current technology gearbox provided a basis for comparative evaluation of the advanced technology design. State-of-the-art materials, bearings, and lubricants were used. This design covered conventional rotation only. Although opposite hand rotation has slightly higher structural demands, changes in the overall results were judged to be minor.

The structural properties of currently available materials are 20 percent lower than those projected for the advanced materials available in the 1990's. This means all current technology gears require increased face widths to accommodate lower allowable stress levels. The results are larger gears and bearings and, consequently, a larger gearbox. This is apparent in the cross-sectional views of the advanced and current technology gearboxes presented in Figure 4.1.5-5.

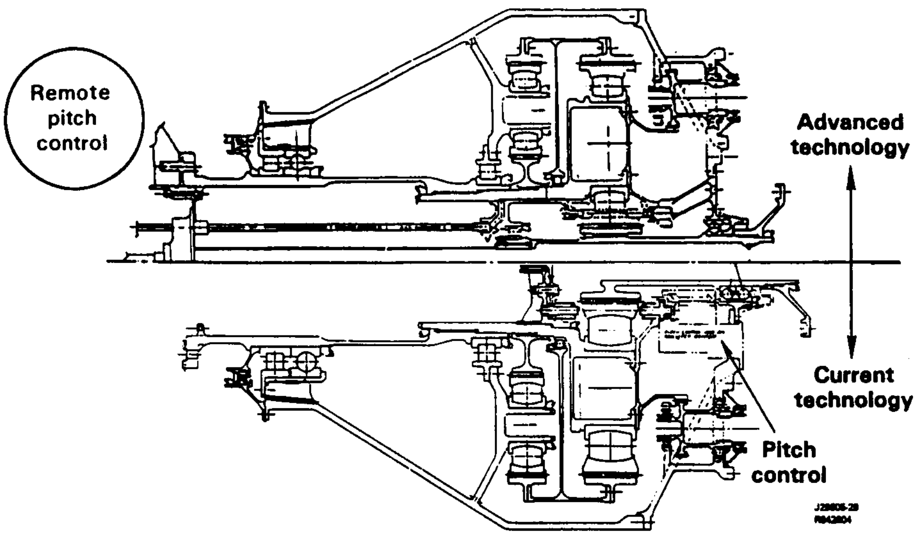

Figure 4.1.5-5 Comparison of the Advanced and Current Technologies - The current technology gearbox is 2 inches larger in diameter and 200 pounds heavier than the advanced technology gearbox.

The increases in the planet gear face width contribute directly to the larger load exerted on the planetary bearing. The additional planet gear weight generates a higher G-force thereby increasing the planet bearing load. This requires a larger bearing to accommodate the higher load and maintain bearing life. Excessive loads, as well as a non-modulated lubrication system, contribute to larger bearing friction losses and a lower overall gearbox efficiency.

The preliminary design study showed that the mechanical design based on current technology will be 5.1 cm (2.0 in) larger in diameter and 93 kg (205 lb) heavier to meet standard design criteria. When compared to the advanced gearbox design, current technology imposes many constraints on the design. Conventional materials and lubricants, including an integrated pitch control, all contribute to a more complex, larger gearbox.

4.1.5.2.2 Maintainability

One of the major limitations of the current technology design is the internal location of the pitch control unit. This has an adverse effect on component accessibility and on maintenance cost. With the pitch control located inside the gearbox, partial gearbox disassembly is necessary for removing the pitch control mechanism and making any repairs. This reduces on-wing maintenance capability and increases propulsion system down time. The maintenance cost ($/flight hours) for the current technology design in 20 percent more than that of the advanced technology design. The opposite hand rotation gearbox maintainability and maintenance cost comparisons are similar in nature.

4.1.5.3 Analyses

In this section, the advanced technology gearbox design is compared to the current technology gearbox design. The comparison is based on structural analysis, performance and economic assessment, and propulsion system integration evaluations.

4.1.5.3.1 Structural Analysis

Multiple analyses verified the structural design of both the advanced and current technology gearboxes. The analyses covered three major areas: shaft stress, gear stress, and bearing life. Opposite hand rotation dictated the allowable design, because it incurs slightly higher stresses than conventional rotation. Table 4.1.5-1 shows the allowable stress levels for current and advanced technology materials and lubricants.

Table 4.1.5-1 Advanced Materials and Lubricants

	Current technology	Technology assumed available by 1988
Gears — materials	AMS 6265	Vasco X-2M or Cartech EX-53
Bending fatigue limit	Base	+20%
Hertz stress limit, MPa (psi)	Base	+20%
Pitch line velocity limit, M/min (ft/min)	Base	+15%
Bearings — materials	VIM VAR M50	Cartech CRB7
Material/lubrication life factor	Base	2 to 3 times
Housings — materials	Aluminum, magnesium	High strength aluminum, magnesium
Lubricant — fluids	Mil 23699 Type II	Synthesized Hydro-carbon Fluid (SHF)
Oil inlet temperature, °C (°F)	Base	+15.6 (+60)
Allowable temperature rise, °C (°F)	Base	2 times
Load carrying ability, N/mm (lb/in)	Base	+35%
Scoring temperature index, °C (°F)	Base	+35%

[1]Typical gear allowable stress — 3 sigma with a coefficient of variation = 0.1 for 10^{10} cycles

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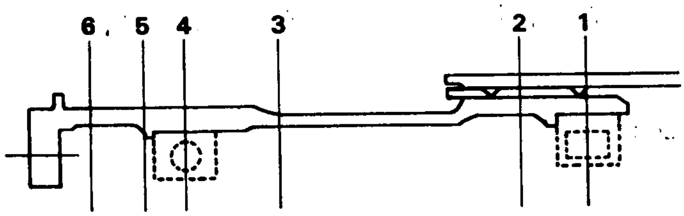
Shaft Analysis

Table 4.1.5-2 presents the propeller shaft stress summary. The shaft is designed as a rigid structure to minimize deflections and subsequent gear mesh misalignment under prop loads. The allowable shaft stress levels came from Pratt & Whitney's previous experience with similar shaft structures. These values apply to both current and advanced technology gearboxes, because stiffness requirements established the shaft design criteria. The prop shaft stress was calculated at various sections along the shaft, using a 1P moment of 23,184 Nm (17,100 ft-lb) and a gyroscopic load at 0.2 radians per second of 8,135 Nm (6000 ft-lb). Stress concentration due to bending stresses was added to the nominal stresses at various sections for evaluating the fatigue of the prop shaft under gyroscopic loads. The fatigue stress levels under a 1P moment load were calculated by including the effect of stress concentration on both bending and torsional stresses. Because shaft thickness was set by rigidity requirements to minimize slope and deflections, stresses are much lower than allowables and are within acceptable levels.

Table 4.1.5-2 Shaft Stress Summary



Shaft location	Allowable (combined)	Stresses, Mpa (psi) Bending	Axial	Torsional	Combined
1	103.4 (15,000)	19.4 (2,810)	6.0 (875)	67.2 (9,750)	70.5 (10,230)
2	172.4 (25,000)	41.6 (6,030)	14.8 (2,140)	150.1 (21,770)	157.7 (22,870)
3	172.4 (25,000)	121.5 (17,615)	14.8 (2,140)	150.1 (21,770)	157.7 (22,870)
4	103.4 (15,000)	84.8 (12,300)	4.3 (630)	43.9 (6,360)	46.2 (6,700)
5	172.4 (25,000)	65.1 (9,440)	6.9 (1,000)	66.6 (9,660)	70.4 (10,210)
6	172.4 (25,000)	77.9 (11,300)	8.1 (1,180)	68.4 (9,925)	72.9 (10,580)

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Gear Stress Summary

Table 4.1.5-3 describes the gear geometry, and Table 4.1.5-4 presents the gear stress summary for both the current and advanced technology gearboxes. The power split necessary for opposite hand rotation governed the gear design. As can be seen in Table 4.1.5-4, the sun pinion gear mesh in the planetary system is the most highly stressed in the opposite hand rotation gearbox. With the gears sized for this system, the stresses are lower for conventional rotation. However, the gears in the star stage are not stress limited because face widths must be larger to accommodate the larger bearings required for longer bearing set life.

Table 4.1.5-3 Gear Geometry Comparison For Single-Rotation

	Present technology		Advanced technology	
	1st stage	2nd stage	1st stage	2nd stage
Gear type	Spur	Spur	Spur	Spur
Diametral pitch	8.0	8.0	8.6	8.6
Pressure angle	22.5	22.5	22.5	22.5
Face width, cm (in.) (Sun gear)	9.40 (3.7)	5.72 (2.25)	8.89 (3.5)	4.95 (1.95)
Number of sun gear teeth	41	117	41	117
Sun gear pitch diameter, cm (in.)	13.018 (5.125)	37.148 (14.625)	12.109 (4.76744)	34.556 (13.60465)
Number of planet gear teeth	89	51	89	51
Planet pitch diameter, cm (in)	28.258 (11.125)	16.193 (6.375)	26.286 (10.34884)	15.063 (5.93023)
Number of ring gear teeth	219	219	219	219
Ring gear pitch diameter, cm (in.)	69.533 (27.375)	69.533 (27.375)	64.681 (25.46510)	64.681 (25.46510)

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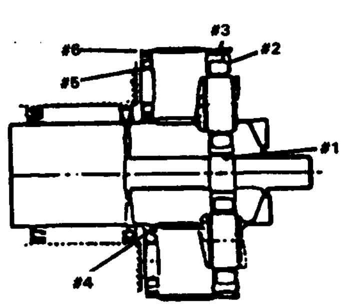
Table 4.1.5-4 Gear Operating Condition Comparison

Single rotation

ADVANCED TECHNOLOGY
OPPOSITE HAND ROTATION

	1st STAGE			2nd STAGE		
	1 SUN	2 PLANET	3 RING	4 SUN	5 PLANET	6 RING
COMPRESSIVE STRESS, MPa(psi)	1027.3(149,000)	1027.3/ 462.9 (149,000/67,138)	462.9(67,138)	1041.1(151,000)	1041.1/668.8 (151,000/97,000)	668.8(97,000)
BENDING STRESS, MPa(psi)	350.3(50,800)	353.0/331.6 (51,200/48,100)	319.9(46,400)	306.8(44,500)	308.9/301.3 (44,800/43,700)	242.7(35,200)
LOAD CARRYING ABILITY N/mm (LB/IN)	397.5(2270)	427.7(2442)	483.7 (2646)	436.4(2762)	436.1(2490)	409.8(2340)
CONTACT RATIO	1.652	1.652/1.775	1.775	1.678/7743	1.7443	1.7443
PITCH LINE VELOCITY M/MIN (FT/MIN)	3800.9(12,470)	3800.9(12,470)	3800.9(12,470)	1240.5(4070)	1240.5(4070)	1240.5(4070)
SLIDING VELOCITY (FT/MIN)						
FLASH TEMP. RISE (°F)						
SURFACE FINISH MICRO-METERS (MICRO-INCHES)	0.38(15)	0.38(15)	0.38(15)	0.38(15)	0.38(15)	0.38(15)



Single-Rotation

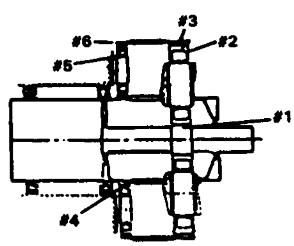
J32333-53
R851510 mcs

Table 4.1.5-4 (continued) Gear Operating Condition Comparison

Single rotation

ADVANCED TECHNOLOGY
CONVENTIONAL ROTATION

	1st STAGE			2nd STAGE		
	1 SUN	2 PLANET	3 RING	4 SUN	5 PLANET	6 RING
COMPRESSIVE STRESS, MPa(psi)	999.7/(145,000)	999.7/451.6 (145,000/65,500)	451.6(65,500)	870.8(126,300)	870.8/599.2 (126,300/86,900)	599.2(86,900)
BENDING STRESS MPa(psi)	330.9(48,000)	347.5/303.4 (50,400/44,000)	292.3(42,400)	245.9/(35,670)	259.6/242.0 (37,650/35,100)	195.1(28,300)
LOAD CARRYING ABILITY N/mm(LB/IN)	378.3(2160)	406.3(2320)	441.3(2520)	378.3(2160)	344.3(1966)	329.2(1880)
CONTACT RATIO	1.652*	1.652/1.775	1.775*	1.678*	1.678/1.744	1.744*
PITCH LINE VELOCITY M/MIN (FT/MIN)	3992.9(13,100)	3992.9(13,100)	3992.9(13,100)	1240.5(4070)	1240.5(4070)	1240.5(4070)
SLIDING VELOCITY M/MIN (FT/MIN)	301.8(990)		60.7(199)	67.1(220)		6.7(22)
FLASH TEMP. RISE °C (°F)						
SURFACE FINISH MICRO-METERS (MICRO-INCHES)	0.38(15)	0.38(15)	0.38(15)	0.38(15)	0.38(15)	0.38(15)



Single-Rotation

J32333-18
R851510 mcs

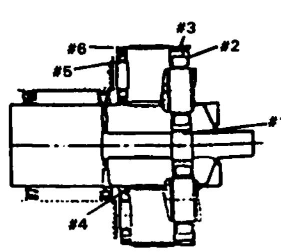
Table 4.1.5-4 (continued) Gear Operating Condition Comparison

Single rotation

PRESENT TECHNOLOGY
CONVENTIONAL ROTATION

	1st STAGE			2nd STAGE		
	1 SUN	2 PLANET	3 RING	4 SUN	5 PLANET	6 RING
COMPRESSIVE STRESS, MPa(psi)	866.0(125,600)	866.0/419.9 (125,600/60,900)	419.9(60,900)	782.6(113,500)	782.6/557.1 (113,500/80,800)	557.1(80,800)
BENDING STRESS MPa(psi)	277.9(40,300)	285.4/268.8 (41,400/38,980)	263.4(38,200)	200.6/(29,100)	215.1/209.6 (31,200/30,400)	165.5(24,000)
LOAD CARRYING ABILITY N/mm(LB/IN)	350.3(2000)	367.8(2100)	402.8(2300)	332.7(1900)	315.2(1800)	303.0(1730)
CONTACT RATIO	1.652	1.652/1.775	1.775	1.678	1.678/1.744	1.744
PITCH LINE VELOCITY M/MIN (FT/MIN)	4297.7(14,100)	4297.7(14,100)	4297.7(14,100)	1332.0(4370)	1332.0(4370)	1332.0(4370)
SLIDING VELOCITY M/MIN (FT/MIN)	362.7(1190)		54.9(180)	91.4(300)		50.3(165)
FLASH TEMP. RISE °C (°F)						
SURFACE FINISH MICRO-METERS (MICRO-INCHES)	0.51(20)	0.51(20)	0.51(20)	0.51(20)	0.51(20)	0.51(20)

*FACE WIDTH REDUCED BY 17% FOR HCR EFFECT



Single-Rotation

J32333-18A
R851510 mcs

The gear loading per inch of face width is relatively consistent for all three gearboxes. However, it should be noted that the second stage is slightly different from the first stage. This is because the load path in the second stage is the reverse of what takes place in the first stage. In the second stage, the ring gear drives the planet gear which in turn drives the sun gear. Therefore, the gear face widths are in reverse order, with the sun gear having the smallest face width.

Gear tooth stresses are acceptable for both current and advanced technology gearboxes.

Bearings

The collective life of the bearing system is the single most important factor controlling gearbox durability. Initial studies indicate that the objective of 15,000 hours MTBUR for the gearbox requires a bearing system that operates with a 50,000 hour mean time between failure. The equivalent 90 percent survival B_{10} life objective is 18,000 hours. This system objective is the governing factor in selecting bearing sizes for highly loaded applications in the gearbox.

Advanced technology materials and lubricants are necessary if life goals are to be met without excessive bearing size and weight. The fatigue life comparison of Table 4.1.5-5 shows that advanced technology bearings exceed the bearing set life goal, while current technology bearing set life falls short of the goal. The first stage planetary bearing capacity cannot be increased enough to carry the high centrifugal load of the larger planet gear.

Table 4.1.5-5 Bearing Life Summary

Average bearing set meets life goal

		Advanced technology		Current technology
	Number of bearings	Conventional rotation life, hrs.	Opposite rotation life, hrs.	Conventional rotation life, hrs.
Ring gear retainer	1	10^6	10^6	10^6
Input shaft (ball & roller)	2	10^6	10^6	10^6
Pinion (sphere) roller 1st stage	4	76,000	155,000	60,000
Pinion (sphere) roller 2nd stage	7	145,000	76,000	115,000
Output shaft roller aft	1	200,000	200,000	400,000
Output shaft roller forward	1	180,000	180,000	350,000
Output shaft ball	1	120,000	120,000	150,000
Bearing set life		18,250 Average		16,360

- **Average bearing set life goal — 18,000 hours (L_{10})**

Self-aligning, single-row, spherical roller bearings position the first and second stage planet and star pinion gears. These bearings carry gear mesh reaction loads and planet centrifugal loads. They also protect gear mesh alignment from pinion carrier thermal or mechanical distortion and promote uniform loading across the face of the gear. Single-row bearings provide the potential for improved high-speed operation and friction losses lower than those of the more widely used double-row bearing. Single-row bearings also offer an advantage of lower weight and a fewer number of parts.

Since the pinion gears are integral to the bearing outer rings, gear and bearing sizes and proportions are closely interrelated. To reduce gearbox weight and to maximize efficiency, gear and bearing diameters are as small as roller length limitations and bearing life objectives permit. A roller length to diameter ratio of 1.4 is adequate for controlling roller skewing and for limiting slippage and friction in roller contacts with the races.

The 18,000 hour bearing system life objective dictates that individual bearing lives in one stage exceed 75,000 hours, provided that other bearings in the system are close to 150,000 hours or higher. Table 4.1.5-5 also shows the bearing life distribution for the advanced technology split path planetary gearbox in both conventional and opposite rotation versions.

The first stage planetary bearings are critical in conventional rotation for determining gear stage diameter and bearing system life. The second stage star bearings are critical in opposite rotation. First stage bearing load is due to a combination of gear mesh reaction and gear rim centrifugal loads. The combined load is greater in conventional rotation, because the pinion carrier speed is substantially higher causing high centrifugal loads. Second stage bearing load is due solely to gear mesh reaction load. The load is higher in opposite rotation, because applied torque is from the first stage carrier and is substantially greater than the torque from the ring gear in conventional rotation.

Pinion gear rim thickness and bearing internal geometry were adjusted to match the particular application. The gear rim must carry bending moments from gear tooth loads and from distributed roller loads. The gear tooth separating load component tends to ovalize the gear rim and load the rollers directly under the gear mesh. The maximum roller load determined the relative race curvatures necessary for containing the roller contact without excessive edge stress at the contact extremities. Preliminary analyses determined gear rim thicknesses and bearing internal geometry selections. Duty cycle calculations determined individual bearing lives.

Table 4.1.5-5 shows that the output shaft ball and roller bearings are the only other locations with lives low enough to influence the bearing system life. The roller bearings carry all the moment and shear forces the Prop-Fan applies to the outer shaft. The aerodynamic content of these forces changes continuously through the flight cycle, while Prop-Fan weight and imbalance loading remain constant. The output shaft ball bearing carries only the Prop-Fan aerodynamic thrust loads which are applied in both directions, forward and reverse. As in the planet pinion design analysis, a detailed description of the aircraft flight cycle is used to calculate bearing loads and lives.

Output shaft roller bearing loads are transferred to a high expansion alloy gearbox housing through the support lines. The high thermal growth of housing diameters at operating temperatures implies loose bearing internal clearance, loose housing fits, or a combination of both. When loosely mounted bearings support a shaft which is highly moment loaded, angular displacement of the shaft causes significant misalignment of the bearings. Misalignment influences bearing internal geometry selection and bearing life. Adequate spacing between the bearings holds misalignment to acceptable levels.

Advanced technology gearbox bearing locations, selected bearing types and sizes, and speed factor (DN) speed levels are listed in Table 4.1.5-6. The corresponding data for current technology gearbox bearings are found in Table 4.1.5-7.

Table 4.1.5-6 Advanced Bearing Selection Summary

Advanced

Location Number	Location Name	Bearing Type*	Bearing size, mm Bore	O.D.	Width	Speed Factor
						XRPM
1	Planet roller, 1st stage	Spherical, 1 row	135	262.86**	60	580,000
2	Planet roller, 2nd stage	Spherical, 1 row	60	150.63**	45	160,000
3	Output shaft roller, front	Cylindrical, DFI	280	365	40	320,000
4	Output shaft ball, front	Split inner ring	280	365	40	320,000
5	Output shaft roller, rear	Cylindrical, DFI	250	335	40	290,000
6	Ring gear retainer roller	Cylindrical DFI&O	250	290	18	440,000
7	Input shaft roller, front	Cylindrical DFO	75	120	22	700,000
8	Input shaft ball, rear	Deep groove radial	85	135	24	890,000

*DFI = Double flanged inner
DFO = Double flanged outer
** Gear pitch diameter

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Table 4.1.5-7 Current Technology Bearing Selection Summary

Location Number	Location Name	Bearing Type*	Bearing size, mm Bore	O.D.	Width	Speed Factor
						mm XPRM
1	Planet roller, 1st stage	Spherical, 1 row	120	282.58**	84	440,000
2	Planet Roller, 2nd stage	Spherical, 1 row	55	161.93**	62	145,000
3	Output shaft roller, front	Cylindrical, DFI	280	380	48	320,000
4	Output shaft ball, front	Split inner ring	280	380	48	320,000
5	Output shaft roller, rear	Cylindrical, DFI	250	345	46	290,000
6	Ring gear retainer roller	Cylindrical, DFI&O	250	290	18	440,000
7	Input shaft ball, front	Counterbore outer	80	130	24	840,000
8	Input shaft ball, rear	Counterbore outer	80	130	24	840,000

*DFI = Double flanged inner
DFO= Double flanged outer
**Gear pitch diameter

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851610 mcs21

4.1.5.3.2 Lubrication System Analysis

Gearbox Oil System

There are several factors which contribute to efficiency loss in gearboxes. These include losses from windage and churning, gear friction, and bearing friction.

While heat rejection losses due to bearing and gear friction vary directly with power, much of the total power loss is due to other factors which are independent of power. These factors include windage drag losses on shaft and gear surfaces and oil churning losses in gearbox and bearing cavities. Figure 4.1.5-6 shows combined windage and churning losses in relation to gear and bearing friction.

At low power settings typical of cruise, windage and churning losses are dominant. The 99 percent efficiency level predicted at full power decreases to nearly 98 percent at typical cruise power levels. Since windage and churning losses are related to oil flow through the gearbox, less power will be lost with appropriate oil flow reduction at reduced power.

To minimize the efficiency degradation at cruise, the design employs a unique, modulated lubricant supply system. This system meets component cooling requirements at low-power and full-power flight conditions without reducing either supply pressure or oil jet velocity.

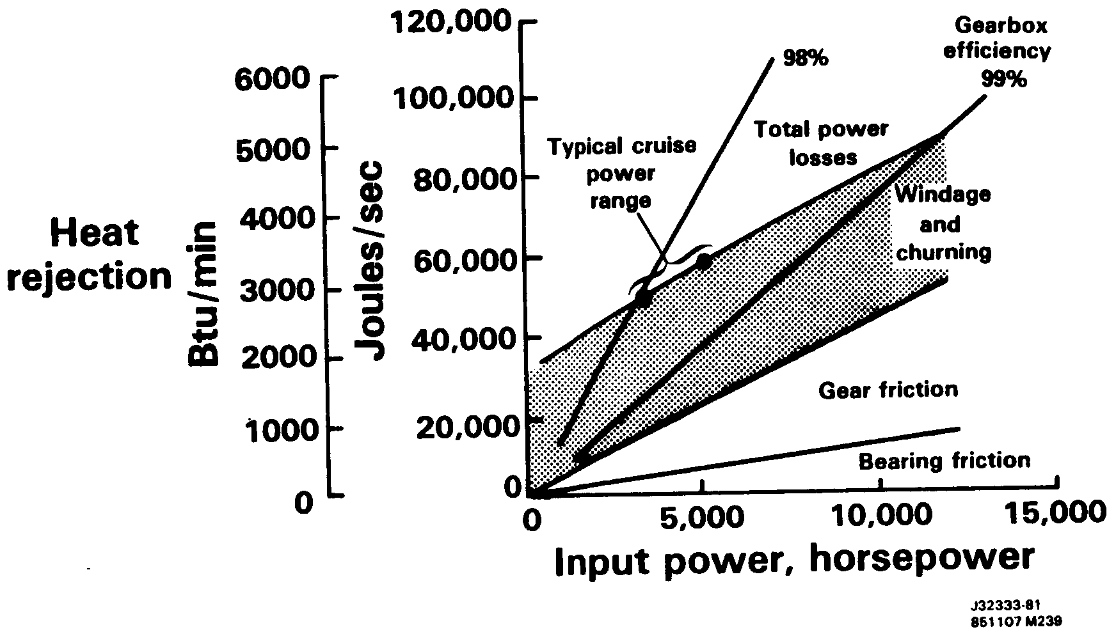

Figure 4.1.5-6 Lubrication Losses - Windage and churning dominate conventional lubrication system total losses.

The result is optimal oil system performance which cuts power loss at cruise to provide about 99 percent efficiency at both cruise and takeoff as indicated in Figure 4.1.5-7.

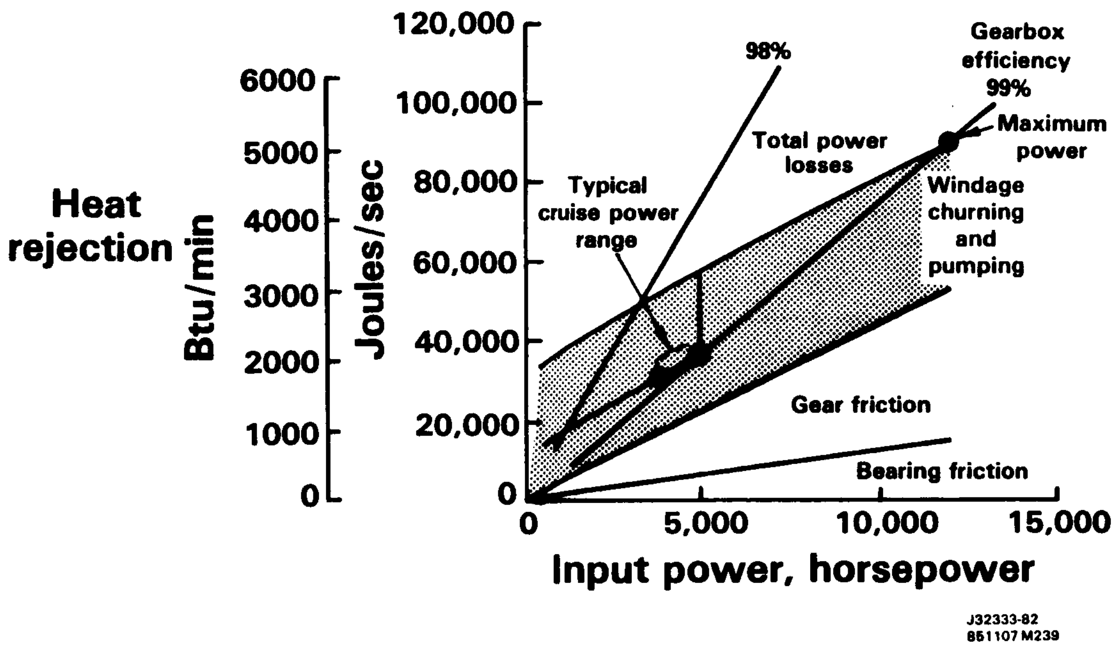

Figure 4.1.5-7 Lubrication Losses - Modulated lubricant system improves cruise efficiency.

The modulated lubrication system features a dual line supply which is composed of primary and secondary oil lines feeding primary and secondary oil jets. A two-position control valve turns off flow to secondary jets at low power. This system maintains full pressure to primary jets and ensures positive oil jet penetration and cooling effectiveness at all conditions.

Included in the lubricant system is an advanced aerodynamic oil scavenge system which reduces power loss and provides in-line condition monitoring. The scavenge system works as follows:

- An oversized oil scavenge pump draws air through the gearbox housing;
- Air flow suppresses oil splash and promotes flow into the scavenge lines, reducing windage and churning losses;
- Air returns to the gearbox and oil to the oil tank after separation in a vortex chamber; and
- The vortex airfoil separator also serves as a particle centrifuge, causing particles such as metal chips to concentrate in the oil system boundary layer; this ensures positive detection by the magnetic particle sensor which is at the inlet to the oil tank.

Engine Oil System Study

Studies show that separating the gearbox oil system from the engine oil system offers many distinct advantages. These include:

- Allowing the use of the lubricant best suited for each system;
- Containing debris within each system to improve reliability;
- Not allowing engine gaspath air leaks to contaminate gearbox oil;
- Allowing the use of 3 micron gearbox oil filters designed for optimal particle removal for the gearbox environment; and
- Reducing gear and bearing wear and corrosion, thus extending gearbox life.

Figure 4.1.5-8 is a diagram of separate oil systems for the engine and gearbox. A return-to-tank fuel system is shown, but the more conventional once-through system can be used.

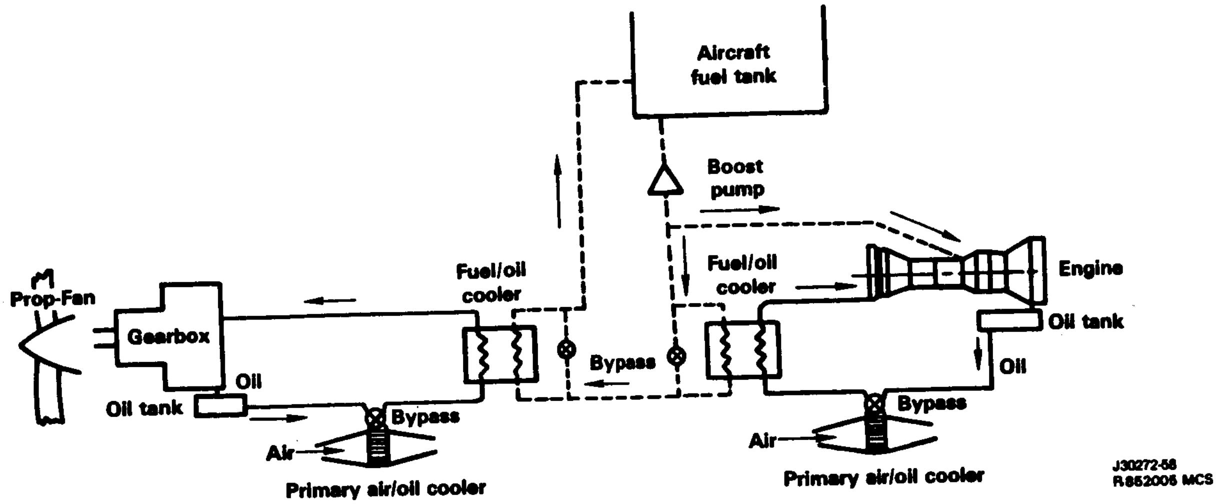

Figure 4.1.5-8 Candidate Oil Cooling System for Advanced Turboprop Engine - This arrangement has a return-to-tank fuel system.

Preliminary heat load calculations for both the gearbox and the engine oil systems are complete, and the sizes of both systems have been determined. While the gearbox's oil flow rates and heat rejection into the oil are somewhat greater than the engine values, the gearbox's oil tank volume can be smaller. This is because lower temperature, smaller oil consumption, and better air removal before the oil tank reduce the time the oil must stay in the oil tank. Table 4.1.5-8 presents gearbox and engine oil system requirements.

Table 4.1.5-8 Advanced Turboprop Engine Oil System Requirements

Dimensions: 1258 cm^2 (195 in^2 face area 5.7 cm (2.25 in) thickness
Core and header weight (WET) = 15 Kg (33 lb)

	SLTO	Cruise	Ground idle
Oil flow, lb/min	68.0 (150)	29.5 (65)	15.0 (33)
Oil in temp., °C (°F)	536 (280)	599 (315)	554 (290)
Oil out temp., °C (°F)	410 (210)	482 (250)	482 (250)
Air flow	205	33	17
Air in temp., °C (°F)	194. (90)	3 (−16)	194 (90)
Air out temp., °C (°F)	410 (210)	482 (250)	482 (250)
Air Mach No. at heat exchanger inlet	0.027	0.012	0.0025
Difference in air pressure, MPa (psi)	.0022 (0.32)	.0010 (0.14)	.0003 (0.04)
Q, J/sec (Btu/min)	89,658 (5100)	35,600 (2025)	11,427 (650)

Analysis has provided an estimate of the size and operating characteristics of the supplementary air/oil cooler for the gearbox. Table 4.1.5-9 illustrates the air/oil cooler operating characteristics. Heat rejection values, Q, reflect the use of the modulated oil supply which reduces loss at cruise power and below. The values for air flow and temperature rise reflect the use of appropriate air flow control devices.

Figure 4.1.5-9 illustrates a typical air/oil heat exchanger concept as it might appear relative to the engine air inlet. While a single engine inlet is evident, the arrangement easily accommodates a bifurcated inlet. The air/oil heat exchanger has dual inlets with variable bypass valves. Dual inlets for the cooler are downstream of the engine inlet to reduce and/or eliminate interference and interactions between the engine inlet and the cooler inlets. For flight conditions where there is insufficient pressure drop across the heat exchanger for effective heat dissipation, an ejector is used to ensure the proper airflow through the airflow cooler. The cooler inlets incorporate flaps which are adjusted at the proper times to minimize secondary losses.

The conservative requirement that the air/oil cooler carry the maximum heat load of the gearbox determined the size of the air and fuel oil coolers. The resulting cooler size is modest. It represents only 15 percent of the nacelle's frontal area. The analysis of propulsion system integration recognizes all of the gearbox heat removal penalties. The effects of air cooler weight and nacelle drag are less than one percent of fuel burn and 0.2 percent of the direct operating cost.

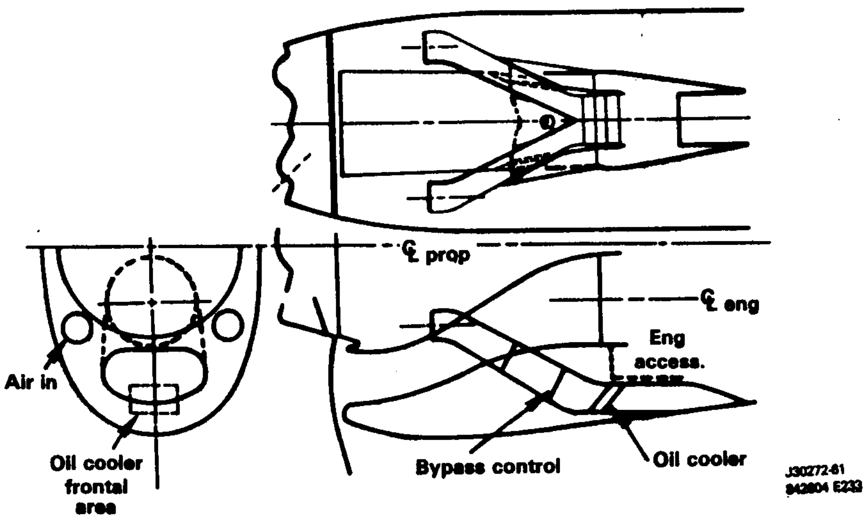

Figure 4.1.5-9 Air/Oil Heat Exchanger Concept - This arrangement accommodates a bifurcated inlet.

4.1.5.3.3 Gearbox Comparative Assessment

The current and advanced technology gearboxes were compared from an airlines' fuel burn and operating cost standpoint. As in the APET Contract NAS3-23045, we used a 120 passenger twin engine aircraft for the evaluation. The evaluation's ground rules were:

- o 3,335.7 km (1,800 nmi) aircraft design mission
- o 0.75 Mn cruise at 10,668 m (35,000 ft) attitude
- o 741.3 km (400 nmi) typical mission for the evaluation
- o Fuel price = $1.50/gal.

Fuel Burn

The two major engine related elements of fuel burn in a given mission are weight and component efficiency as it relates to specific fuel consumption. Table 4.1.5-9 compares both gearboxes in terms of these aspects.

The use of advanced technology materials along with advanced gear tooth forms permits the use of smaller and lighter gears and bearings for the advanced technology gearbox. This reduces the gearbox maximum diameter by 5.1 cm (two in) relative to that of the current technology, and the net result is a weight savings of 93 kg (205 lb) for the advanced technology gearbox.

Table 4.1.5-9 Alternate Split Path Gearbox Summary

	Current technology	Advanced technology
No. gears	15	15
No. bearings	17	17
Diameter, cm (in)	87.4 (34.4)	81.8 (32.2)
Cruise efficiency, (%)	98.2	99.2
Weight, kg (lb)	535.2 (1180)	442.3 (975)
Acquisition cost	Base	−10%
Reliability (MTBUR), hrs	15,600	23,300
Prop-Fan pitch control	Internal	External
Maintenance cost	Base	−20%
Rel. fuel burn	**Base**	**2.4% improve.**
Rel. DOC + I	**Base**	**1.0% improve.**

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The use of the modulated lubrication supply system along with the aerodynamic scavenge system in the advanced technology gearbox resulted in a one percent improvement in cruise efficiency relative to that of the current technology gearbox.

Improving gearbox efficiency has a doubling effect on fuel burn. Gearbox inefficiency (power losses in the gearbox) results in larger gearbox oil heat generation and, in turn, results in larger air/oil cooling requirements. For instance, if the gearbox efficiency was improved from 98 percent to 99 percent, then the resultant power losses in the gearbox would be halved from two percent to one percent. This results in one half the gearbox oil heat generation. This, in turn, halves the air/oil cooler size, which significantly reduces its installed drag and weight.

The net effect of the advanced gearbox's one percent efficiency improvement over the current gearbox is a 2.4 percent improvement in mission fuel burn.

Direct Operating Cost

A dominant factor in an airline's decision whether or not to purchase new equipment is the direct operating cost (DOC) of the new aircraft relative to existing systems. The three most important engine-related elements in DOC equations are mission fuel burn, engine acquisition cost, and engine maintenance cost. To represent a realistic direct operating cost for an airline, we have also included the cost of capital in our assessment of DOC. The cost of capital rate used in this evaluation is 15 percent per year.

Fuel Burn:

As discussed earlier, advanced technology materials and gear tooth forms permit using smaller and lighter gears and bearings in the advanced technology gearbox. The net reduction is 93 kg (205 lb). The smaller gears and bearings permit the gearbox diameter to be smaller. This, in turn, allows a smaller nacelle with less drag for tractor applications. The use of an advanced lubricant and a new modulated lubrication system results in a one percent improvement in gearbox efficiency and much smaller air/oil cooler requirements. All of these factors result in the advanced technology gearbox having a 2.4 percent fuel burn advantage over the current technology gearbox.

Acquisition Cost:

The advanced technology gearbox's smaller and lighter gears, bearings, and housings also result in this design having a 10 percent lower acquisition cost than that of the current technology design.

Maintenance Cost:

Because it has an externally located pitch control, the advanced technology gearbox is considerably more reliable than the current technology design. In fact, the current technology gearbox has only two-thirds the MTBUR hours of the advanced technology design. Current technology's lower reliability rating and greater acquisition costs result in a total maintenance cost 20 percent greater than that of the advanced technology.

In a total economic evaluation, the combined savings of 2.4 percent in fuel burn, ten percent in acquisition cost, and 20 percent in maintenance cost give the advanced technology gearbox a one percent advantage in direct operating cost plus interest over the current technology gearbox. This advantage represents a considerable contribution to improved operating economics.

4.1.5.3.4 Propulsion Systems Integration

This section presents information concerning propulsion systems integration. The section covers conceptual nacelles, propulsion system mounting, and component (accessories, heat exchanger) provisions.

Nacelle Configuration

The over-the-wing "tractor" installation was selected for Prop-Fans, because it provides adequate ground clearance for a typical low-wing commercial aircraft. This installation is compatible with both the in-line and offset Prop-Fan reduction gearboxes. The two configurations are shown in Figures 4.1.5-10 and 4.1.5-11, respectively. The in-line gearbox provides a slimmer nacelle than the offset gearbox, because the in-line gearbox is smaller in diameter and the resulting arrangement has a smaller diameter. Alternate mounting locations (e.g. tail mounted engines) were not considered for the single-rotation tractor.

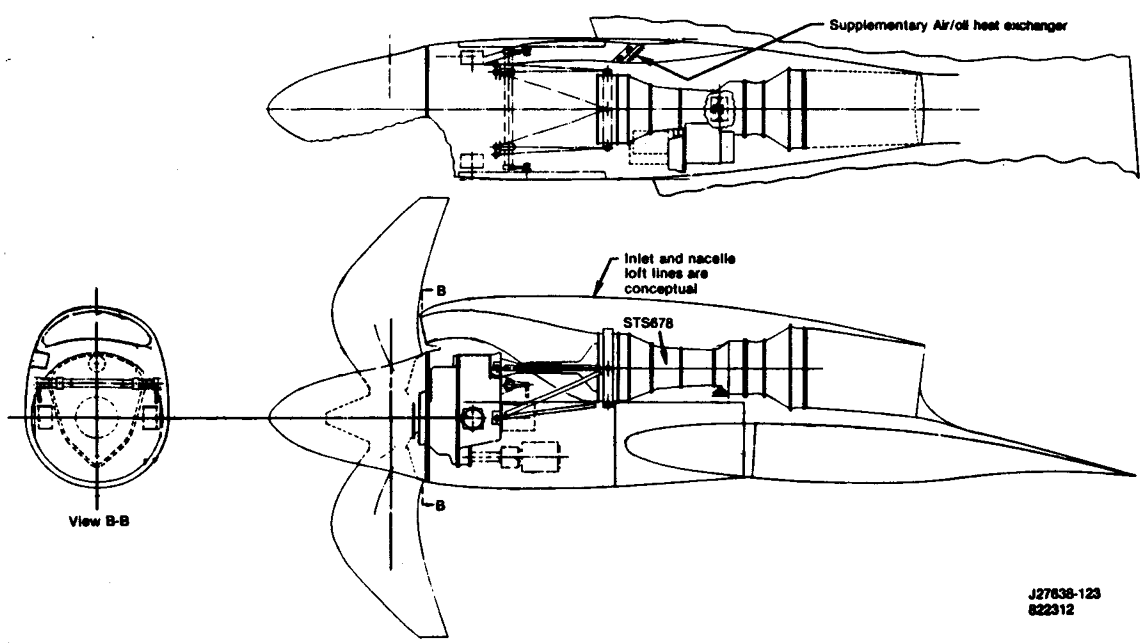


Figure 4.1.5-10 Conceptual Nacelle Design For An Offset Gearbox Installation - This design is compatible with both the in-line and offset Prop-Fan reduction gearbox.

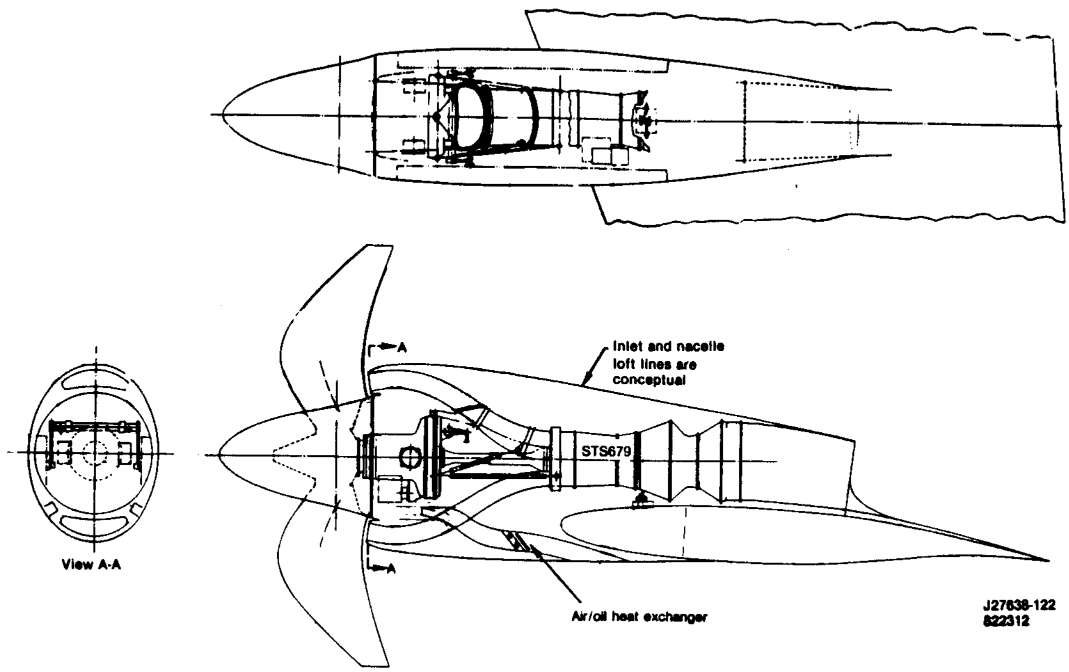


Figure 4.1.5-11 Conceptual Nacelle Design For An In-Line Gearbox Installation - The nacelle in this arrangement is slimmer, because the in-line gearbox is smaller in diameter.

The resulting nacelle for the offset compound idler gearbox has a maximum nacelle to Prop-Fan diameter ratio of 0.32 while the nacelle for the in-line gearbox results in a 0.28 diameter ratio due to the gearbox's smaller diameter.

The external aerodynamic lines for the nacelle are conceptual in nature and provide proper fairing for the Prop-Fan. The final aerodynamic nacelle lines would be tailored to the flow field of the specific aircraft application. NASA and the airframe manufacturers are conducting detailed studies to tailor the nacelle and aircraft wing to minimize aerodynamic interference losses. The nacelles identified in the study provided the basis for the mechanical design used in the engine/aircraft evaluation (Task IV).

Propulsion System Mounting

Based on input from the aircraft manufacturers, the "integrated" engine and reduction gear mount system is the primary engine mount system for the single-rotation tractor powerplant. Figure 4.1.5-12 is a schematic of this configuration. There are two mount planes. One is at the reduction gearbox, and the other is at approximately the engine's center of gravity. For this propulsion system mounting arrangement, a structurally stiff truss joins the Prop-Fan reduction gearbox to the gas generator (engine). This truss is capable of transferring moment and shear loads between the two components. The engine casing is stiffened to minimize compressor tip clearance increases. The aircraft nacelle provides the primary propulsion system support structure which consists of two axial beams. Both beams are cantilevered forward of the wing box structure on either side of the powerplant and joined together at the forward end by a bulkhead. The bulkhead provides pick-up points for the front mount plane while the structure attached to the wing box supports the rear mount plane. A Prop-Fan torque reaction system handles the large Prop-Fan torque while allowing the front mount to be sized for thrust, maneuver loads, and vibration isolation. The isolation of engine/Prop-Fan generated vibration is a major requirement for passenger comfort. The torque link system may not be necessary if vibration isolators are stiff enough to absorb Prop-Fan torque and permit the powerplant to translate freely in response to vibration while absorbing Prop-Fan torque.

Figure 4.1.5-13 shows the "integrated" engine and reduction gear mounting system for the in-line gearbox configuration. With this concept, a portion of the inlet duct is structurally tied to the engine and the gearbox to avoid structural links in the aerodynamic flowpath of the inlet, which would result in small performance and engine inlet distortion penalties.

While the "integrated" engine and reduction gear mounting system has been selected for the Prop-Fan propulsion system, there are several technical issues which require study beyond the scope of the current contract. These issues are summarized below.

- Powerplant/aircraft structural dynamics studies
  - Axial location of engine relative to gearbox and wing box
  - Shock isolation trade studies
  - Effect of these factors on wing flutter

- Integrated engine and gearbox structure
  - Structural links between engine and gearbox
  - Primary structure with inlet between engine and gearbox
  - Engine and gearbox attachments to the nacelle and wing

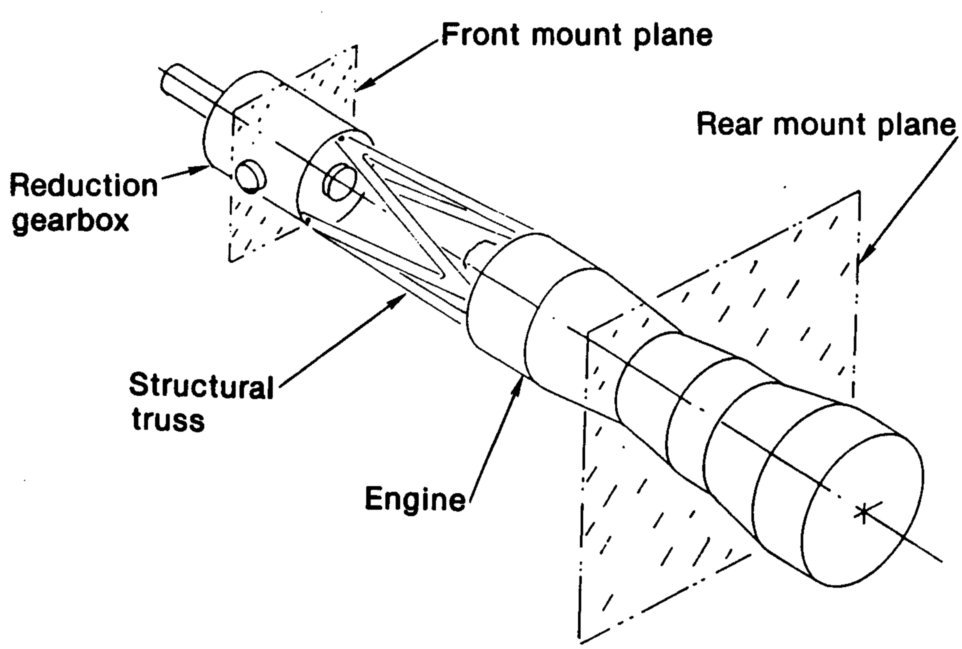


Figure 4.1.5-12 Schematic of "Integrated" Engine and Reduction Gearbox Mounting Scheme - In this arrangement there are two mounts: one is at the reduction gearbox, and the other is roughly at the engine's center of gravity.

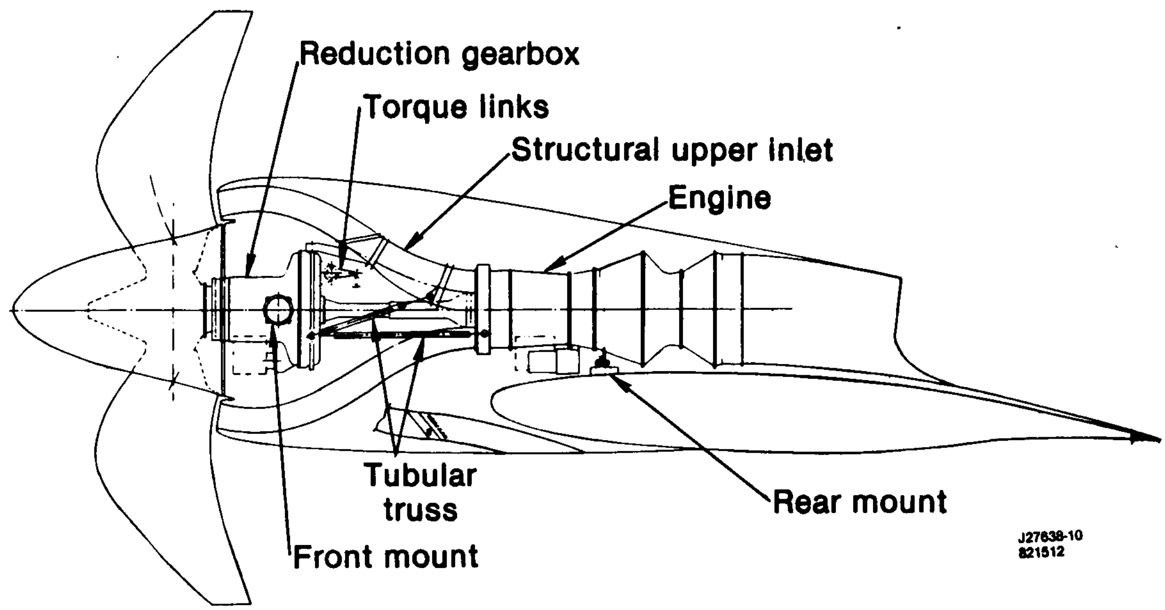


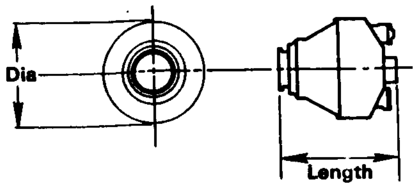
Figure 4.1.5-13 "Integrated" Engine and Reduction Gearbox Selected For In-Line Configuration - A portion of the inlet duct is structurally tied to the engine and the gearbox to avoid structural links in the aerodynamic flowpath of the inlet.

The high spool of the engine provides output power for engine accessories and input for the starter. Two power takeoff sources are available for aircraft accessories. The engine high spool can provide this power or the Prop-Fan reduction gearbox provides an optional power output for aircraft accessories.

The final choice of aircraft accessory location will require coordination with the airframe manufacturer and will involve both configurational and performance trades.

Scaleability

The advanced technology gearbox is designed for approximately 12,000 hp. The gearbox design can be scaled to accommodate the range between 8,000 hp and 16,000 hp without any major change in design. The changes that do take place are in weight, maximum diameter, and length. As Figure 4.1.5-14 shows, there is a nearly linear relationship between torque and weight. As the torque increases, the gearbox weight becomes proportionately heavier. However, maximum diameter and length do not share the same relationship. As the torque increases, maximum diameter and length increase moderately. The greater the torque becomes, the slower the increase in maximum diameter and length.

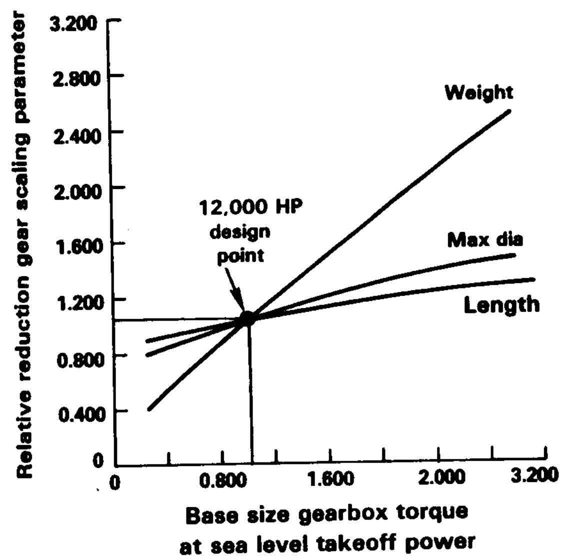


$$W_{GB} = W_{GB}\,(\text{base}) \times \left(\frac{GR}{GR_{base}}\right)^{.15}$$

Where: $GR_{base} = 8.23$
GR = desired gear ratio
W_{GB} (base) = 12,000 HP gearbox wt.

<u>Reference</u>

Weight, kg (lb) = 374.2 (825)
Length, cm (in) = 71.1 (28.0)
Diameter, cm (in) = 80.0 (31.5)
Total torque = 69,147 N-m (51,000 ft-lb)



$$\text{Scaling ratio} = \frac{\text{Torque}_{new}}{\text{Torque}_{base}}$$

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Figure 4.1.5-14 Turboprop Reduction Gear Scaling - As the figure shows, there is a nearly linear relationship between torque and weight.

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SECTION 4.2 -- DISCUSSION OF RESULTS

TASK VIII -- THE CONCEPTUAL DESIGN OF A SINGLE-ROTATION PITCH CONTROL

Table of Contents for Section 4.2

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4.2 Task VIII -- The Conceptual Design of a Pitch Control

4.2.1 Introduction

Task VIII provided an advanced, flight-weight pitch change control and mechanism conceptual design which is compatible with the in-line gearbox design of Task VII. Prior to the conceptual design, Hamilton Standard conducted a conceptual trade study to select a concept for further study under the APET contract. This section presents a discussion of Hamilton Standard's design of an advanced pitch control and mechanism. The first two parts of the section are concerned with a current technology overview and a discussion of pre-APET trade studies. The pitch control trade study segment has two parts. The first details the power system, and the second details the control system. The last part of the section describes the conceptual design of the selected pitch control concept.

4.2.2 Current Technology Overview

Blade pitch controls on new commuter turboprops generally incorporate a number of mechanisms and features which are representative of today's technology. A linear hydromechanical actuator with a metering valve and a mechanical pitch lock are basic components of the pitch change mechanism and are mounted in the rotating hardware. Mechanical, hydraulic, and electrical inputs must be transmitted from the fixed, nacelle-mounted components (i.e., the gearbox).

Rotary mechanical inputs position the metering valve and pitch lock and utilize either differential gearing or a bearing-mounted ball screw to transmit rotary motion across the rotating interface. High-pressure oil is transmitted to the metering valve and actuator through a low clearance oil transfer bearing and transfer tubes. Electrical power for ice protection is transmitted to the turboprop through contact brushes running on a rotating slip ring assembly.

The turboprop assembly drawing shown in Figure 4.2.2-1 defines a current pitch control concept adapted to an offset gearbox installation. The offset gearbox permits pitch control components to be mounted in accessible modules on the axis of rotation. This minimizes the impact on the gearbox design and greatly improves maintainability. Other features include the relatively small diameter oil transfer bearing and compact differential gearing in the regulator module and the small drum-type slip ring module. These features contribute to a more reliable system with less weight. Turboprops installed on the current generation of large commuter aircraft incorporate most of these features.

Figure 4.2.2-2 shows a current technology concept for transmitting rotary mechanical and hydraulic pitch control inputs to a turboprop installed on an in-line planetary gearbox. In this configuration, the drive shaft from the engine restricts access to the axis of rotation from the rear of the gearbox.

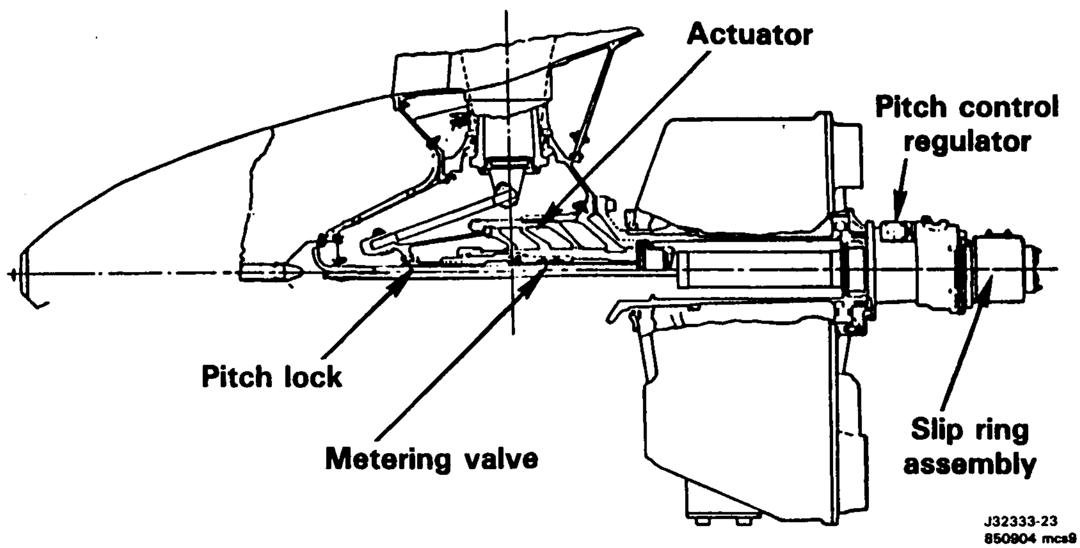

Figure 4.2.2-1 Turboprop Offset Gearbox Configuration - The offset installation permits using accessible pitch control components.

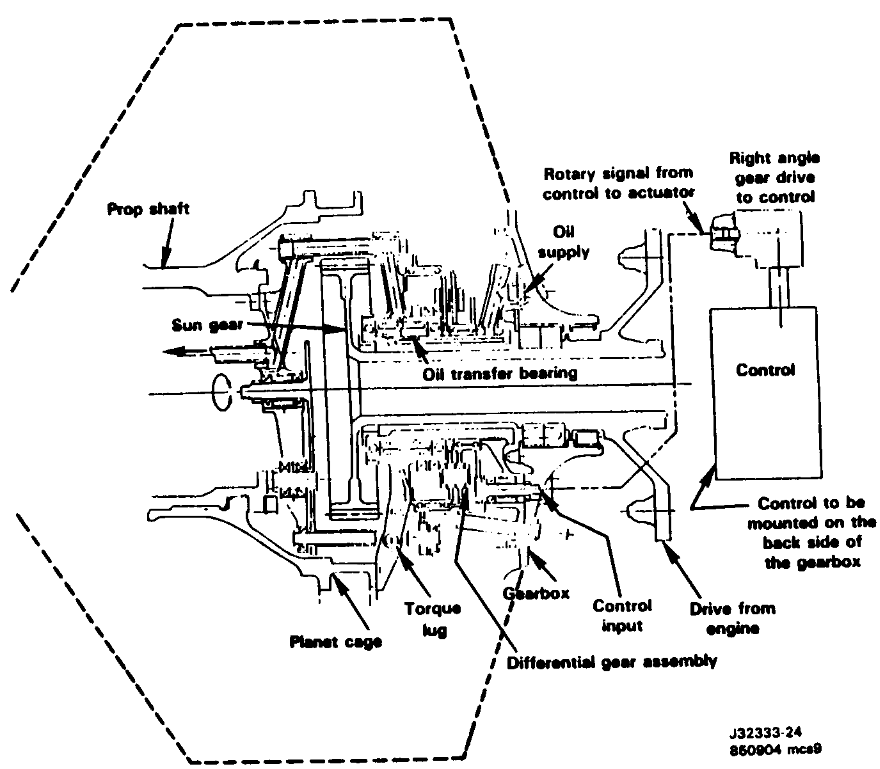

Figure 4.2.2-2 Turboprop In-Line Current Technology Gearbox Installation - This system uses non-modular pitch control inputs.

Therefore, the mechanical signal must be transmitted from the rear face of the gearbox housing to the turboprop, through differential gearing, around the sun gear shaft and lay shafts, and through the planet cage and additional gears to reach the axis of rotation. Similarly, high pressure pitch change oil must be transmitted through a large diameter (high leakage) transfer bearing, around the sun gear shaft and oil transfer tubes, and through the planet cage to the turboprop shaft.

Unlike the offset gearbox configuration, the integration of non-modular pitch control inputs within the in-line gearbox introduces several complexities. In addition to the complex gearing and large diameter transfer bearing, there is a significant impact on the gearbox design. The overall effects are a reduction in reliability and an increase in maintenance costs. This configuration emphasizes the need to develop advanced pitch control systems that are more reliable and more maintainable.

4.2.3 Trade Studies

Prior to the APET single-rotation Prop-Fan pitch control study, Hamilton Standard conducted company-funded pitch control trade studies to select the advanced technology concepts which were subsequently used for the APET single-rotation Prop-Fan and counter-rotation Prop-Fan studies. The primary criterion was that the pitch control system be adaptable to any gearbox configuration with minimal impact on the gearbox design. The pitch control was divided into two parts: a power system and a control system. A comprehensive matrix of the most viable concepts was prepared for each system, and each matrix was evaluated separately.

4.2.3.1 The Power System

The power system matrix in Figure 4.2.3-1 shows several concepts of pitch change mechanisms with prime movers and power supplies on either the stationary side (the gearbox) or the rotating side (the Prop-Fan) of the rotating interface. Several methods of power transfer across the rotating interface were considered. All components were evaluated and compared using the following parameters listed in order of importance, starting with the most critical:

- safety
- reliability
- maintainability
- weight
- performance (accuracy of blade angle control for Synchrophasing (R))

- o acquisition cost
- o impact on gearbox
- o technical risk
- o envelope
- o heat generation (efficiency)

These evaluation parameters were weighted and used in conjunction with a forced decision rating technique.

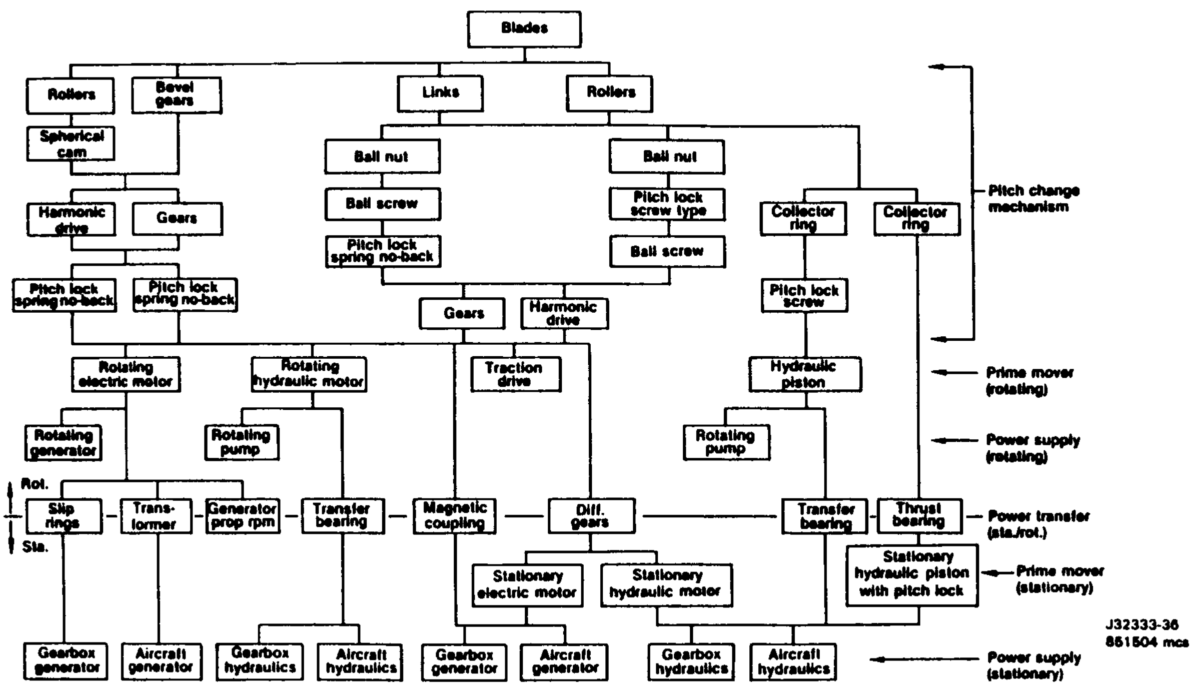

Figure 4.2.3-1 Prop-Fan Power System Matrix - Several methods of power transfer across the rotating interface were considered.

The first round of rating concepts of the same function eliminated several concepts. This resulted in a matrix of seven power systems represented by the shaded boxes in Figure 4.2.3-2. Pitch change mechanisms at the far left of the matrix were eliminated primarily because of weight penalties associated with the large bevel gears and cams necessary for actuating the Prop-Fan blades mounted in the large diameter hub. The ball screw and ball nut, coupled with a spring no-back pitch lock, in the left center of the matrix were eliminated because of unsatisfactory blade angle control. The backlash necessary to release and engage the pitch lock caused excessive hysteresis in the pitch control loop. Most of the power transfer components on the rotating interface were eliminated because of their poor reliability relative to systems incorporating dedicated Prop-Fan mounted power supplies. In addition, slip rings incur high maintenance costs; transformers and generators driven at Prop-Fan rpm are heavy; oil transfer bearings have poor reliability and maintainability for the large diameters necessary for in-line gearbox installations; and the thrust bearing that transmits pitch change and pitch lock loads across the rotating interface rates low on reliability, maintainability, and weight.

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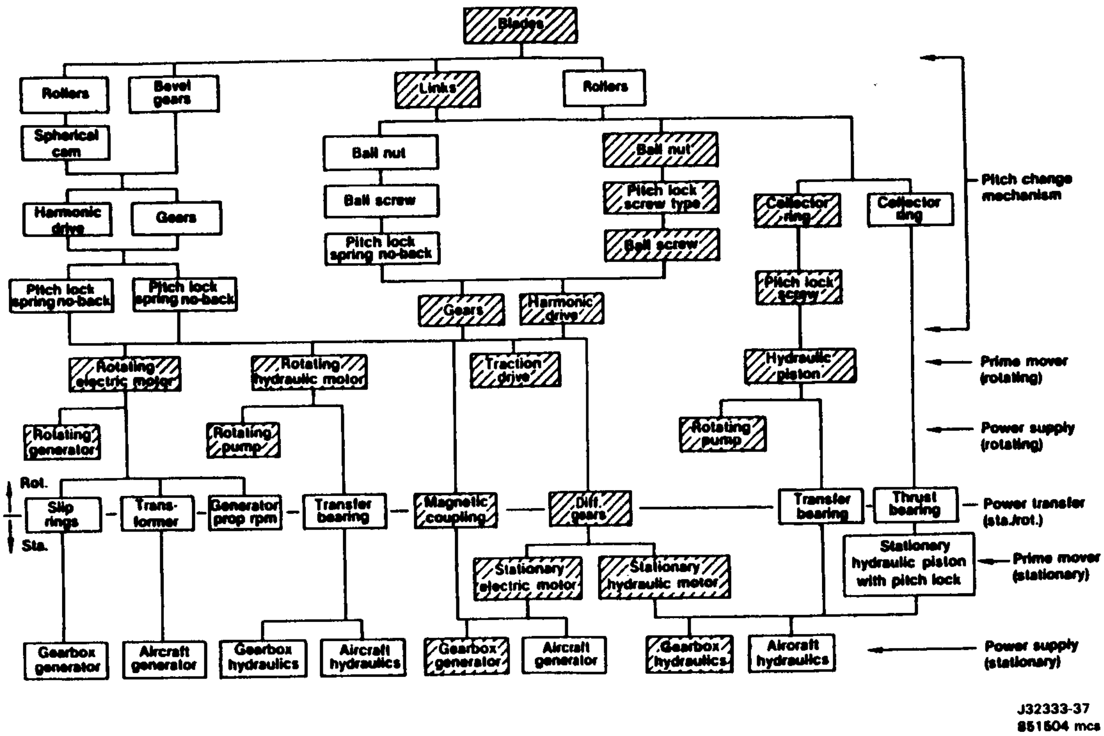

Figure 4.2.3-2 Prop-Fan Intermediate Power System Matrix - The shaded boxes represent the seven power systems chosen for further study.

One of the final seven power system candidates incorporates a linear hydraulic piston which acts directly on a collector ring, links, and blade trunnions (crank arms) to change pitch. The remaining six systems incorporate a ball screw, which when rotated, translates a ball nut and links to change blade pitch. Either a traction drive or motors (electric or hydraulic; rotating or stationary) can drive the ball screw. Hydraulic pumps and motors are considered to be gear types operating at a system pressure of 41.4 MPa (6,000 psi). Electric generators and motors are the samarium-cobalt permanent magnet brushless type with appropriate electronic controls. The required motor size for maximum pitch rate is approximately 25 hp.

The magnetic coupling is an electric motor mounted on the rotating interface with the stator fixed to the gearbox and the rotor driving the ball screw through appropriate gearing. During fixed pitch operations, the rotor reacts blade torque and rotates at a reference rpm dependent on Prop-Fan speed. Rotor speed is increased or decreased from the reference speed to raise or lower pitch. The traction drive is a toroidal variable ratio type with associated planetary gearing. This type of traction drive was selected over a constant ratio, multi-stage roller traction drive, because it offered a mechanical method of providing bi-directional, variable speed pitch control.

The seven power system concepts were evaluated and compared. They are as follows in order of ranking.

1. Hydraulic piston actuator
2. Ballscrew, hydraulic motor
3. Ballscrew, electric motor
4. Ballscrew, differential gears, hydraulic motor
5. Ballscrew, magnetic coupling
6. Ballscrew, differential gears, electric motor
7. Ballscrew, traction drive

The simplicity of the hydraulic piston concept prompted the highest ratings for reliability, performance, and cost. Consequently, the hydraulic piston concept received the highest total rating. Of the remaining ball screw concepts, electric motor drives rated second to hydraulic motor drives, because they are less reliable and heavier. Differential gear concepts rated lower on reliability, because they have a higher parts count. The toroidal traction drive was rated low on reliability, performance, and technical risk. This left the hydraulic piston actuator (linear hydraulic) and the hydraulic motor drive ballscrew (rotary hydraulic) as the two final candidates for further study.

The rotary hydraulic concept rated higher than the linear hydraulic concept in terms of weight and adaptability to counter-rotating Prop-Fans. The latter was a consideration secondary to the evaluation parameters listed at the beginning of this section. Both concepts have a minimum impact on the gearbox, but the rotary hydraulic power system was selected for the APET pitch control conceptual design study, because of its low weight and adaptability to counter-rotating Prop-Fans. This system is highlighted by the shaded boxes in the power matrix of Figure 4.2.3-3. Gearbox interface requirements for this self-contained hydraulic power system are minimal, consisting only of a high-speed pump drive shaft from the sun gear and a nominal amount of cooling oil flow.

4.2.3.2 The Control System

Figure 4.2.3-4 is a diagram representing a digital electronic aircraft propulsion control system in which a full-authority digital electronic engine control (EEC) coordinates and commands engine fuel flow, compressor vane positions, and Prop-Fan blade angle to control power and rpm. The engine and the Prop-Fan provide diagnostic feedback to the control. Report coverage of this system appeared in a NASA-sponsored study completed in 1978 (Report No. CR-135192). The system is still desirable for advanced Prop-Fans. The control system matrix, shown in Figure 4.2.3-5, identifies different methods of transmitting a blade pitch command signal to the Prop-Fan power matrix from the EEC. Several methods of transmitting the digital signal across the rotating interface to a Prop-Fan mounted electronic controller are shown with several types of blade angle (β) feedback sensors. A stationary nacelle mounted electronic controller was also considered in this analysis.

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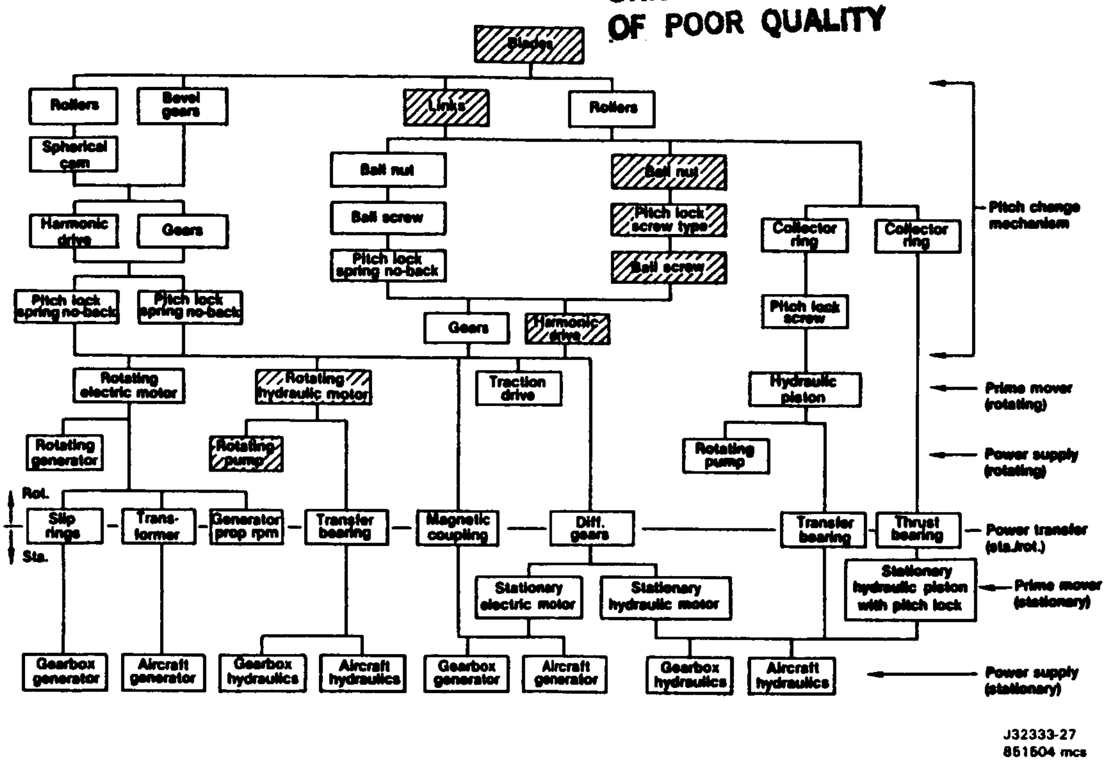

Figure 4.2.3-3 Selected Rotary Hydraulic Prop-Fan Power System From Trade Study - This system is lighter and more adaptable to counter-rotating Prop-Fans.

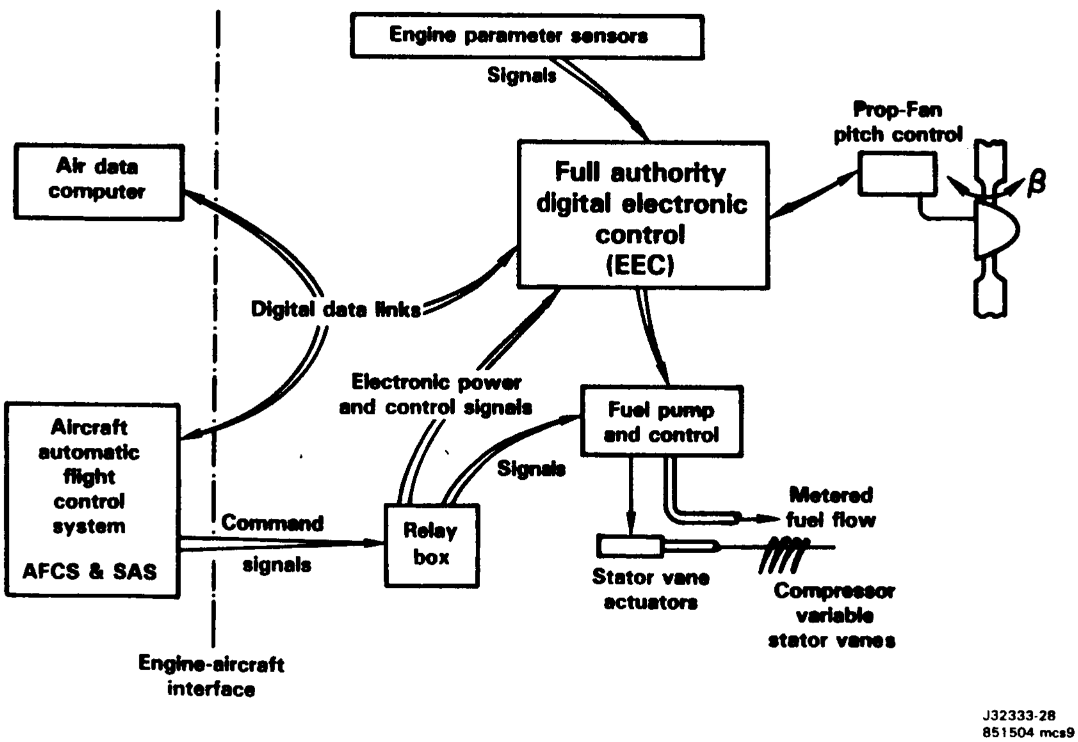

Figure 4.2.3-4 Prop-Fan Propulsion System Control Diagram - The engine and Prop-Fan provide diagnostic feedback to the control.

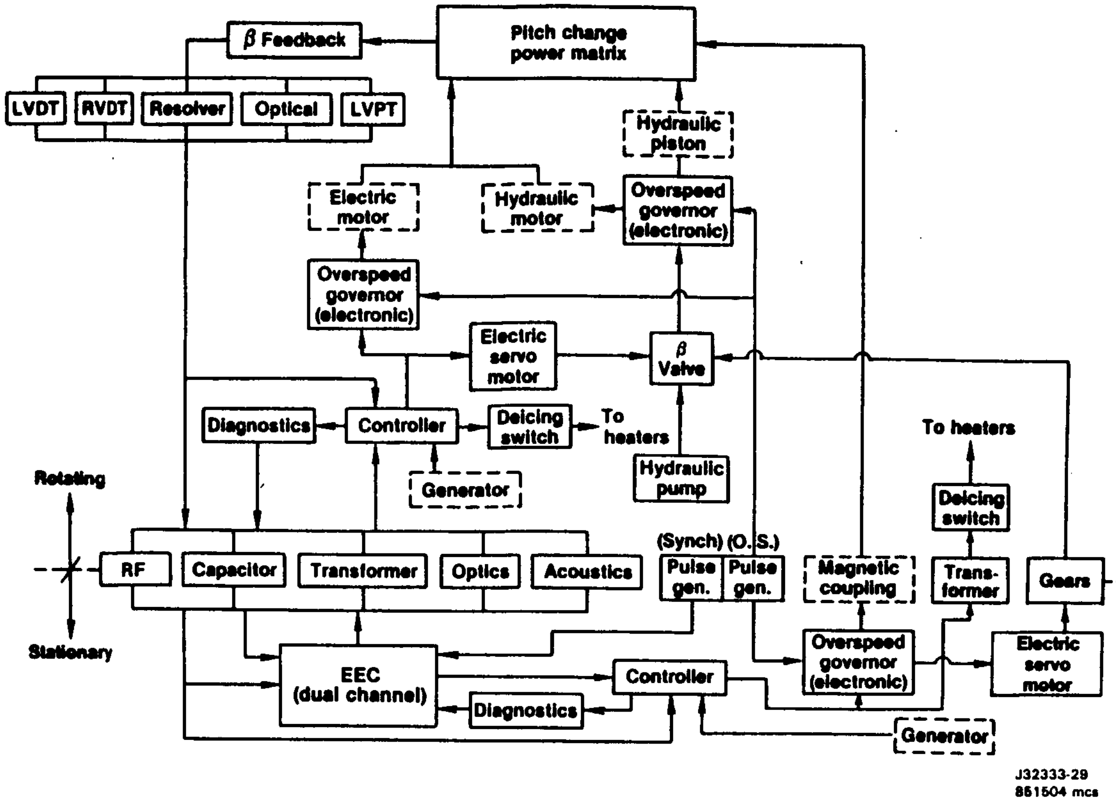

Figure 4.2.3-5 Prop-Fan Control System Matrix - This figure identifies different methods of transmitting a signal to the Prop-Fan power matrix from the EEC.

All control system components were evaluated and compared utilizing parameters and weighting factors similar to those employed in the power system study. These parameters are listed as follows in order of importance, starting with the most critical:

- safety
- reliability
- maintainability
- acquisition cost
- accuracy (Synchrophasing control)
- weight
- technical risk
- adaptability (to single-rotating and counter-rotating Prop-Fans, in-line and offset gearbox configurations)
- envelope

Five blade angle feedback displacement sensors were considered. They were: (1) linear variable differential transducer (LVDT), (2) rotary variable differential transducer (RVDT), (3) linear variable phase transducer (LVPT), (4) resolver, and (5) optical encoders. The LVDT and LVPT measure linear displacement. The RVDT and resolver measure rotary displacement, while optical encoders can measure either linear or rotary displacements.

A comparison of the sensors ranked them as follows:

1. LVPT

2. LVDT

3. RVDT

4. Resolver

5. Optical

All sensors provided sufficient accuracy, but each differed significantly in reliability, maintainability, and cost. The first three rated sufficiently higher than the last two to qualify as candidates for selection. The LVDT and RVDT measure displacement as a function of output voltage amplitude. Both are widely used today. In contrast, the LVPT represents a relatively new technology. It measures displacement as a function of phase difference of two output voltages, and unlike the LVDT, it does not require an analog/digital converter. Because of this latter feature, the LVPT rates slightly higher than the LVDT. However, the RVDT was chosen over the LVPT, because the RVDT is more adaptable to the rotary hydraulic power system previously selected for the APET study.

Five methods of transmitting digital control signals across the rotating interface were evaluated. These are: (1) radio (RF), (2) capacitor, (3) optics, (4) transformer, and (5) acoustics. Following the evaluation, they ranked as they appear below.

1. Capacitor

2. Transformer

3. Optics

4. Radio (RF)

5. Acoustics

Rating variations were based primarily on reliability, with particular emphasis on susceptibility to external interference. Optics rated lower than the capacitor and the transformer concept, because it is more difficult to protect optical components from contamination than to shield the capacitor and transformer from electromagnetic interference (EMI). Radio and acoustics were eliminated, because they are very difficult to protect from radio frequency (RF) and acoustic interference. The capacitor concept was selected, because it is simple and more reliable.

Three of the five control system concepts shown in Figure 4.2.3-5 utilize a fractional horsepower D.C. electric servo motor to position a metering valve to provide high pressure oil to either a linear piston or a gear motor prime mover. The servo motor and its electronic controller are mounted in the rotating Prop-Fan in two of these concepts and on the stationary gearbox in the third concept. The remaining two concepts incorporate an electronic controller to control a large D.C. electric motor (approximately 25 hp) prime mover. One of these concepts has the controller and motor mounted in the rotating Prop-Fan, and the other concept has the controller mounted on the gearbox to control the motor (magnetic coupling).

Comparative evaluation of the control systems resulted in the ranking shown below.

1. Electric servo motor, metering valve, hydraulic motor
2. Electric servo motor, metering valve, hydraulic piston
3. Electric servo motor, gears, metering valve
4. Electric motor (magnetic coupling)
5. Electric motor

The first two servo motor control systems are identical and share the same rating. They differ only in the prime movers being driven. Their rating is significantly higher than the ratings of the remaining three concepts. The third servo motor system was penalized on reliability and accuracy for transmitting the control input to the metering valve through differential gearing. The two large electric motor control concepts received low ratings, because the solid state components currently available for large motor and generator controls are less reliable and significantly larger than those for small motors and generators. Considerable research and development effort is underway to improve this technology for use in aerospace applications (e.g., the all electric aircraft). When electrical prime movers become competitive with hydraulic prime movers, the rotary pitch change mechanism can be easily adapted to either system.

The pitch control system components selected for the rotary hydraulic power systems are highlighted by the shaded control matrix boxes in Figure 4.2.3-6. Interface with the gearbox is minimal and consists of a support bracket for the stationary half of the capacitor signal transfer coupling and a high-speed generator drive shaft from the sun gear. This is the same shaft that drives the pumps in the power system.

In summary, the trade studies showed that:

a) The linear hydraulic actuator rates slightly higher than the rotary hydraulic actuators, but the latter is lighter and appears to be more adaptable to counter-rotating Prop-Fans; both are viable concepts for future study.

(b) Hydraulic systems are more reliable and have a higher power density capability than electrical systems.

(c) Using a power supply located on the rotating side of the interface is more reliable than transmitting power across the rotating interface from the stationary side.

(d) The capacitor control signal transfer across the rotating interface is simple and reliable.

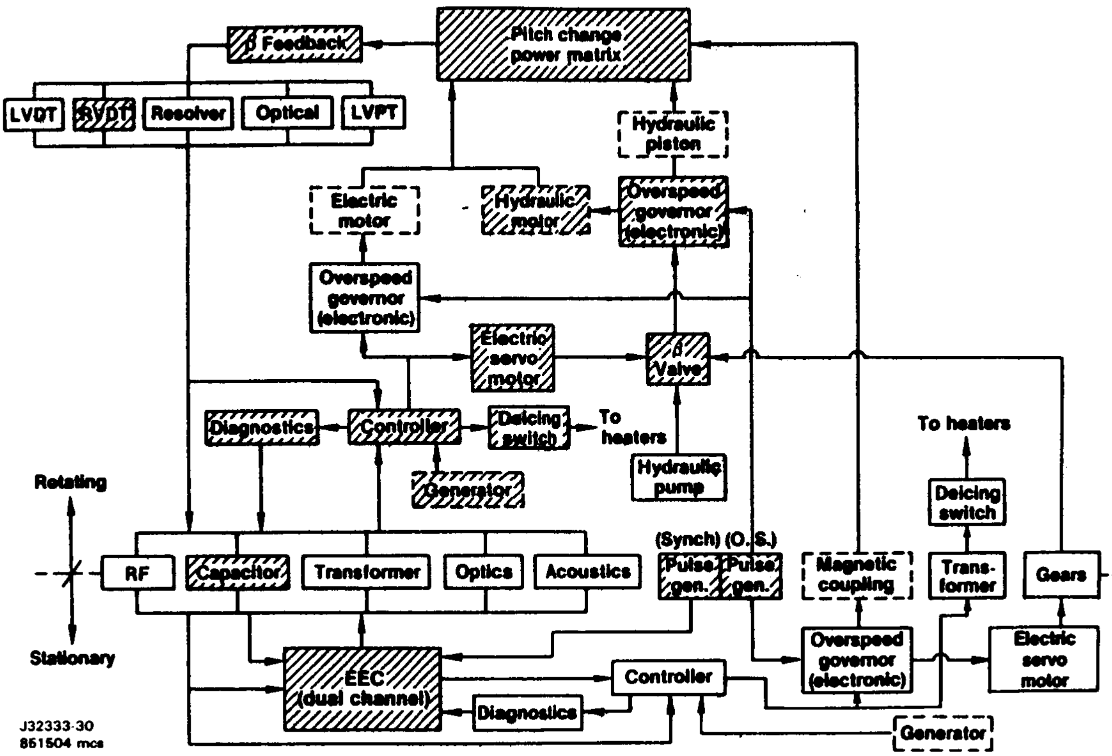

Figure 4.2.3-6 Selected Control System Rotary Hydraulic Concept - The shaded control matrix boxes highlight the pitch control components selected for the rotary hydraulic power systems.

4.2.4 Conceptual Design of Selected Concept

The two primary design objectives for the APET pitch control conceptual design were:

- o To minimize impact on the gearbox, and
- o To maximize accessibility and maintainability.

To meet these objectives, Hamilton Standard implemented a modular pitch control design which is in the rotating Prop-Fan assembly. This simplifies the interface with the gearbox, improves gearbox reliability, lowers maintenance cost, and reduces pitch change maintenance cost by providing accessible, easily maintainable modules. The conceptual design incorporates the rotary hydraulic pitch control system components selected in the trade study.

4.2.4.1 Description of the Pitch Control Concept

The Prop-Fan is flange-mounted to the gearbox output shaft through curvic face splines at the rear face of the hub. A single-row angular contact ball bearing retains each blade in the hub. An external blade clamp provides additional support for static blade pitch operation. A fixed amount of oil in the hub lubricates the blade retention bearings. A lip seal at the blade root seals the hub and prevents external leakage. A sectional assembly drawing of the Prop-Fan pitch control concept is shown in Figure 4.2.4-1. Blade trunnion arms, which are splined to the inboard end of the blades, rotate the blades about the pitch axes. Links with spherical rod-end bearings connect the trunnion arms to a ballscrew nut which translates to change blade pitch. The ballscrew is straddle-mounted on hub-mounted support bearings. Link forces impose a torque on the ball nut which is reacted by an integral lug sliding in a slot in the hub-mounted forward housing.

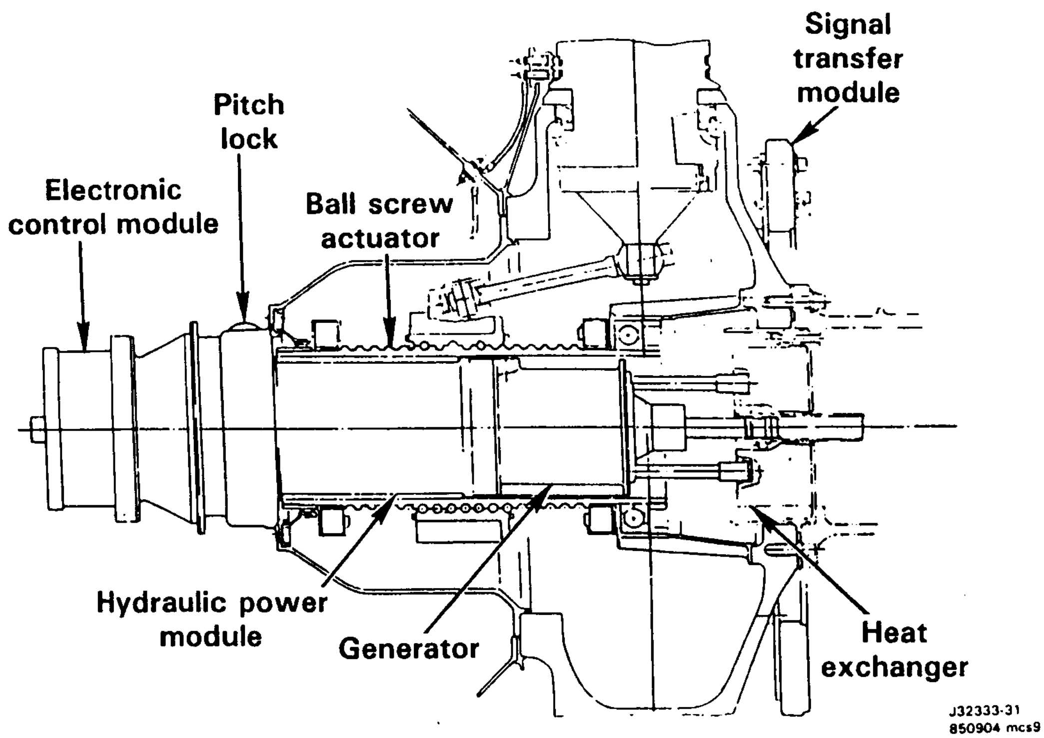

Figure 4.2.4-1 Pitch Control Drawing - This figure highlights pitch control features.

A hydraulic power module drives the ball screw. This module consists of a harmonic drive, a hydraulic gear motor, a four-way metering valve (beta control), a mechanical in-place pitch lock, pumps, oil sumps, pressure regulating and relief valves, and a generator. A bolted flange secures the power module on the Prop-Fan forward hub-mounted housing. The ball screw increases and decreases the pitch. Harmonic drive rotates the ball screw in response to pressurized oil applied to the high or low pitch side of the hydraulic drive motor. An irreversible worm gear mesh acts as a pitch lock. The worm gear, which is splined to the ball screw, rotates with the ball screw as a direct indication of blade angle position. A small axial gap is maintained between the end of the worm and the hub-mounted power module housing. The worm is free to translate. This prevents the blade pitch from decreasing toward low pitch more than one degree if hydraulic power inadvertently fails anywhere in the blade operating range.

A small bi-directional D.C. servo motor drives the pitch lock worm to control pitch upon command from the electronic control module. Each rotational position of the worm gear represents a discrete blade angle setting in the operating range. An RVDT measures this position. The RVDT is geared to the worm gear and fed back to both the electronic control module and the nacelle-mounted EEC.

4.2.4.2 Hydraulic System

Figure 4.2.4-2 is a diagram showing the functional relationship between the actuator, pitch lock, and the hydraulic components. The hydraulic system is designed to conserve power and reduce heat generation. The Prop-Fan pitch control operates over ninety-five percent of the time at power levels less than twenty percent of peak power. This is because peak pitch rate power is necessary only for large blade angle excursions such as reversing and feathering. For commercial aircraft, these operations comprise less than five percent of the total operating time.

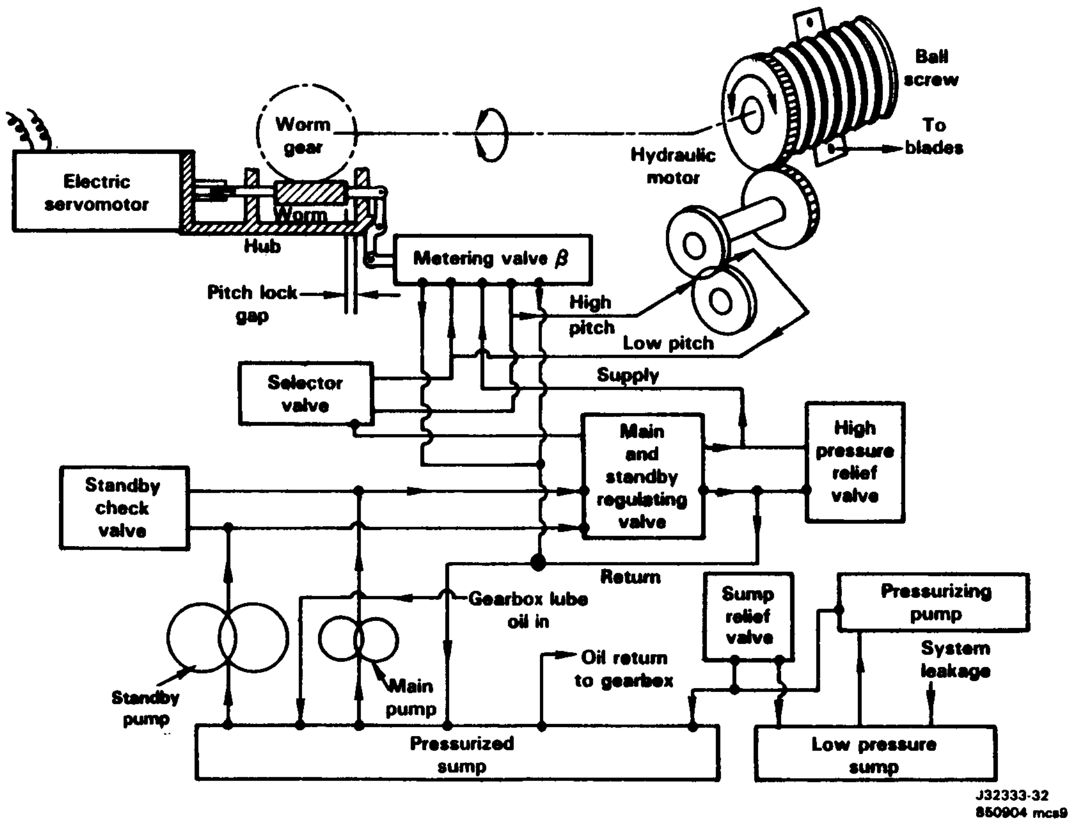

Figure 4.2.4-2 Hydraulic System Diagram - This diagram shows the functional relationships between the actuator, pitch lock, and the hydraulic components.

A small displacement main gear pump supplies high pressure oil to the hydraulic motor via the beta metering valve for all low-power pitch control requirements. Although the pump can provide the peak system pressure set by the high pressure relief valve, the pump supply (discharge) pressure is regulated to a few hundred psi above motor operating pressure requirements. The main and standby regulating valves accomplish this by regulating main pump supply pressure to the metering valve at a level slightly above the higher of high and low pitch pressures as indicated by the shuttle selector valve. This pressure regulation, coupled with the small pump size, reduces pitch control power generation to the low levels necessary for most of the flight spectrum. A standby gear pump with approximately four times the capacity of the main pump circulates oil back to the pressurized sump at low pressure (low power) most of the time. When the beta metering valve is positioned for high flow (pitch rate), the regulating valve and standby check valve combine both the standby pump flow and the main pump flow, at high pressure, to provide the required high power. This is a transient condition, and heat generation is minimal.

A pitch control system pressure versus weight trade study showed that 41.4 MPa (6,000 psi) is the optimal pressure for minimum weight. However, 32.8 MPa (4,750 psi) was selected because it results in higher reliability and lower cost for a weight penalty less than a percent of pitch control weight. A small scavenge pump charges the pressurized sump to 0.52 MPa (75 psi) minimum. This scavenge sump is on the atmospheric sump where system leakage collects. This pressure ensures that the main and standby high-speed pumps are adequately supplied with oil to prevent cavitation. Cooling oil from the gearbox lube system circulates through the hydraulic power module to mix directly with pitch control oil and return filtered to the gearbox cooler.

A high speed shaft from the gearbox drives the power module pumps and generator on the axis of rotation. The generator is a light-weight, samarium-cobalt, permanent magnet, externally-commutated A.C. type. The electronic control module rectifies the A.C. output to D.C. Dual generator windings provide separate voltage supplies for pitch control and blade deicing. An overrunning clutch at the generator drive shaft permits the generator to operate as a motor for static ground operation of the pitch control. Auxiliary ground cart power, supplied to the generator with the engine inoperative, drives the pumps to develop pressurized oil for pitch change.

4.2.4.3 Electronic Control System

The electronic control module incorporates the printed circuit boards and solid state components required to:

a) provide control of the D.C. servo motor under pitch control command from the nacelle-mounted, full-authority, digital Electronic Engine Control and from separate overspeed pitch control circuitry in the module,

b) transmit blade angle feedback and other diagnostic signals to the Electronic Engine Control, and

c) provide power switching for blade deicing.

A rotary capacitor signal transfer module, located at the rear of the hub, transmits serial digital pitch control signals bi-directionally between the Electronic Engine Control and the rotating electronic control module. The transfer module contains two electrical paths. Each path consists of two parallel annular metal disks, one on each side of the rotating interface, separated by an air gap.

Under normal operating conditions, the electronic control module provides only blade pitch control on command from the Electronic Engine Control. All intelligence for governing rpm, Synchrophasing, feathering, reversing, and ground handling is in the dual-channel Electronic Engine Control. This permits the more complex electronic control circuitry to be in the nacelle where it is more accessible for maintenance and for modification of control parameters. In the event of either an erroneous signal or loss of signal from the Electronic Engine Control, the electronic control module has a solid-state speed governor with separate power supply, circuitry, and speed sensor that will govern rpm at a set percentage of normal rpm. The flight may then continue with only the loss of Synchrophasing and reversing capability. Provision is made to conduct a pre-flight check of this back-up control circuit.

Blade pitch angle change originates with a requirement and a command signal from the EEC to change a discrete amount toward either high or low pitch. The signal is transmitted across the capacitor signal transfer module to the electronic control module. The electronic control module powers the D.C. servo motor to rotate the pitch lock worm and to translate the metering valve spool through a linkage. Pressurized oil, metered to the hydraulic motor, causes the ball screw and worm gear to rotate, translating the worm in the opposite direction, thereby nulling the valve. The ball screw and worm gear will continue to rotate as long as the motor is rotating. The pitch lock gap between the worm and ground toward low pitch is continuously maintained within one degree of blade angle (i.e., full metering valve authority is sustained within the pitch lock gap). The RVDT continuously measures the blade angle position which is fed back to the control. The control terminates the signal when the commanded angle is reached.

4.2.4.4 Maintainability Features

The modular component design of the pitch control concept satisfies the primary design objectives of minimum impact on the gearbox and maximum accessibility and maintainability for any gearbox configuration. After removal of the Prop-Fan spinner, the electronic control module can be easily removed by removing bolts from the mounting flange and by then pulling the module forward on guide pins to release the plug-in wiring connectors. Removal of the D.C. servo motor mounting bolts permits the motor and associated reduction gearing to be removed as a unit from the hydraulic power module. The hydraulic power module, including the generator, can be removed on guide dowels after taking out the mounting flange bolts. A hoist and lifting fixture are not necessary for removal of the module. Check valves are incorporated in the oil transfer tubes to seal against oil loss when disengaged from the gearbox cooling oil tubes.

Access is gained to the blade links and ball screw assembly for inspection or maintenance action by removing the conical support housing from the hub at the bolted flange. The hydraulic power module need not be removed for this operation. Individual blades can also be removed and replaced, if required, as follows:

(a) disconnect the blade link at the trunnion arm,

(b) disengage the deicing brush assembly from the blade slip rings,

(c) remove the external split clamp and lip seal from the hub,

(d) move the blade into the hub a small distance and remove the retention bearing balls, self-contained in a flexible plastic retainer, and

(e) remove the blade from the hub.

The capacitor signal transfer module is fabricated in segments that are easily removed for replacement or repair.

It is possible to remove and replace all Prop-Fan components without removing the hub or gearbox from the aircraft. The user can decide the extent to which this disassembly is necessary on the aircraft.

In Figure 4.2.4-3 there are three critical technologies that require additional development before their use in an advanced Prop-Fan propulsion system. The rotary capacitor signal transfer module requires an efficient shielding system to prevent electro-magnetic interference (EMI). The electronic control components must be mounted and packaged in the module to withstand the G-field environment of the rotating Prop-Fan (approximately 40 G's per inch of radius from the axis of rotation). The hydraulic gear pumps and gear motors must be developed for the high speed, high pressure application of the power module. A research and technology plan, defining the programs required for technology development, has been prepared and is included later in this report.

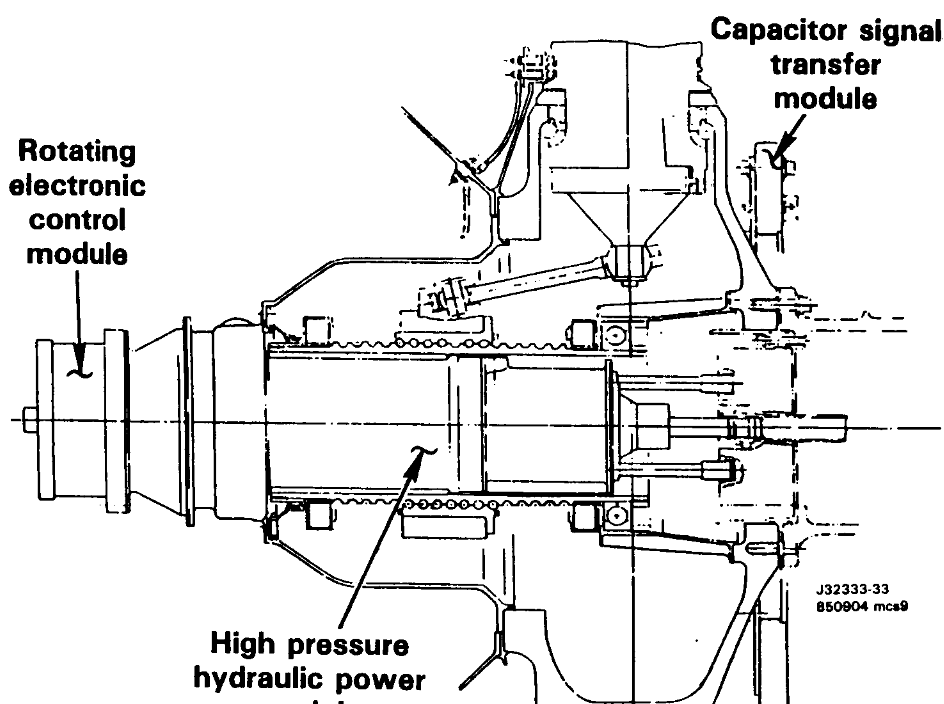

Figure 4.2.4-3 Advanced Technology Components Requiring Additional Development - A plan to test this technology is included in Section 4.5.

4.2.5 Pitch Control Parameters

The primary Prop-Fan design parameters used in the conceptual design of the advanced technology pitch control were blade pitch slew rates, blade angles, and blade twisting moments. The following sections discuss these parameters.

4.2.5.1 Slew Rates

Table 4.2.5-1 presents the blade pitch slew rate requirements for various Prop-Fan operating conditions. Normal slew rate requirements for most of the flight spectrum are low. Blade pitch angle is essentially constant at each flight condition with small excursions of less than ± 0.1 degree during Synchrophasing . Synchrophasing is a fine-tuning control of blade pitch through very small angles that do not require high slew rates.

Table 4.2.5-1. Slew Rates

Condition	Blade Pitch Rate (deg/sec)
Normal control	0-3
Synchrophasing	0-1
Feathering	15
Reversing	15
Ground Operation (engine inoperative)	0-3

The aircraft requirements normally set the maximum slew rate based on the time necessary to reach full reverse angle on landing. The rates shown are based on the capability to reverse fully from flight idle in three seconds. These rates are judged to be satisfactory for advanced turboprop propulsion systems. However, different rate requirements can be easily satisfied with minor changes to the pitch control.

4.2.5.2 Blade Pitch Angle Settings

The blade angle settings given in Table 4.2.5-2 are for various operating conditions. Angles are specified at the blade 3/4 radius. Pratt & Whitney indicates that the engine can start with the blades at any angle including feather. Therefore, the minimum Prop-Fan torque blade angle is somewhat academic for this propulsion system. The mechanical in-place pitch lock sets the emergency blade angles. The pitch lock follows approximately one degree below any commanded blade angle.

Table 4.2.5-2. Blade Angle Settings

Condition	$\beta_{3/4}$ (degrees)
Takeoff (0 Mn)	+32
Maximum climb (0.3 Mn)	+41
Cruise (0.75 Mn)	+55
Flight idle (0.3 Mn)	+38
Maximum reverse	- 7
Feather	+85
Minimum Prop-Fan Torque (static conditions)	0
Emergencies	<1 below setting when condition occurs

4.2.5.3 Blade Twisting Moments

The pitch control system must be capable of rotating the blades about the pitch axis, to counteract the total blade twisting moment. The total moment comprises the following individual twisting moments:

a) centrifugal, acting toward flat pitch,

b) aerodynamic, acting toward either high or low pitch depending on the flight condition, and

c) friction, acting to impede motion toward either high or low pitch.

Centrifugal twisting moment results from centrifugal forces on the blade mass as a function of distance from the pitch axis and makes up most of the total moment. Highly swept Prop-Fan blades have significantly higher twisting moments than more conventional blades with less sweep because of the increase in overhang from the pitch axis.

The maximum total blade twisting moment that the pitch control must overcome to move ten blades toward high pitch is 58,074 Nm (514,000 in-lb). The maximum total twisting moment required to hold the blades in position is slightly less than this value due to exclusion of the friction moment. It is this reduced moment that the pitch control or the pitch lock must react to hold the blades at a fixed blade angle setting.

4.2.6 Weight

Results from a weight analysis of the pitch control conceptual design were compared with the data base from the original APET contract study (Hamilton Standard report SP06A82, "Prop-Fan Data Package: Weights," May 1982). The net result is an 11 percent reduction in total Prop-Fan weight relative to an off-set gearbox data base (SP06A82, concept of which appears in Figure 4.2.2-1) and 12 percent reduction relative to an in-line gearbox data base (modified SP06A82, concept of which appears in Figure 4.2.2-2). These reductions are attributable to the advanced technology pitch control design concept, because the weight of the remainder of the Prop-Fan parts is the same as the data base references.

4.2.7 Reliability

A component failure rate and unscheduled removal rate analysis was performed for all the pitch control modules. These rates were then added to the respective rates of the remaining Prop-Fan hardware to arrive at the total Prop-Fan system rates.

Failure rate is defined as any event chargeable to the hardware. Removal rates include additional non-chargeable events such as maintenance damage, unsubstantiated removals (no failures), and accident and foreign object damage (FOD) where applicable, in addition to the chargeable removal rates. The mean time between unscheduled removals for all causes is the inverse of the total removal rate.

The MTBUR of 5300 hours for the advanced technology Prop-Fan system is derived in Table 4.2.7-1. The table is based on Prop-Fan assembly removals as well as removals of replaceable components such as the electronic control, the power module, and the spinner.

Table 4.2.7-1 Unscheduled Removals (All Causes)

Component	Removal Rate (events/1000 flight hours)
Spinner	0.0086
Disk and Aft Fairing	0.0029
Blades	0.0530
Forward Cover and Fairing	0.0055
Electronic Control Module	0.0490
Capacitor Coupling	0.0002
Power Module	0.0594
Non-Modular Components	0.0100
Total:	0.1886
MTBUR = (1/.1886) (1000) =	5,300 hours

The current technology, "internal" pitch control is applicable to an in-line gearbox configuration. It is described as "internal," because a portion of the pitch control is inside the gearbox. The current technology external pitch control is applicable to an offset gearbox configuration where all the pitch control hardware can be outside the gearbox.

The advanced technology pitch control is external to the gearbox and is applicable to both the in-line and the offset gearbox configurations. A Prop-Fan with the internal, current technology pitch control exhibits an eight percent loss in reliability (mean time between unscheduled removals) relative to a Prop-Fan with an external, current technology pitch control concept.

The Prop-Fan with external, advanced technology pitch control represents an improvement of 96 percent in MTBUR over the baseline external current technology pitch control Prop-Fan system defined in NASA report CR135192 "Study of Turboprop Systems Reliability and Maintenance Costs," June 1978, Table 4.4-I, page 231. Baseline removal rates were revised for ten blades instead of eight to compare with the advanced technology Prop-Fan system.

The "chargeable" mean time between unscheduled removals for a Prop-Fan with the external advanced technology pitch control concept is 9,000 hours This represents an improvement of 105 percent over the 4,400 hours for the baseline Prop-Fan system with an external current technology pitch control concept.

The predicted MTBUR (chargeable events) of 26,800 hours for a Prop-Fan with the external advanced technology pitch control concept is for occurrences where removal of the entire Prop-Fan assembly is required due to failure. This represents an improvement of 46 percent over the 18,400 hours for a Prop-Fan with the baseline external current technology pitch control concept.
Table 4.2.7-2 is a summary of the Prop-Fan reliability for both current and advanced technology pitch control concepts.

Table 4.2.7-2

PROP-FAN RELIABILITY SUMMARY

	CURRENT TECH. External	CURRENT TECH. Internal	ADVANCED TECH. External
o MTBUR Prop-Fan Assy. Chargeable* (hrs.)	18,400	18,400	26,800
o MTBUR Prop-Fan Assy. & Components			
Chargeable* (hrs.)	4,400	-8%	9,000
All Causes** (hrs.)	2,700	-8%	5,300

* Due to Hardware Failure

** Due to Hardware Failure and all other causes

4.2.8 Costs

4.2.8.1 Maintenance Cost

The Prop-fan module includes the electronic control, the hydraulic power module, ball screw actuators, and the capacitor signal transfer module. Maintenance costs for the Prop-Fan with the advanced technology pitch change system were estimated utilizing an on-condition philosophy established for the Prop-Fan. This philosophy is in line with present day turboprop field service experience and involves repair or replacement of only the faulty module, as determined by built-in health-monitoring diagnostics.

The maintenance cost was developed for the ten bladed, 4.1 m (13.35 ft) diameter Prop-Fan by considering all the elements of maintenance, namely:

1. Scheduled inspections
2. Unscheduled line repairs
3. Unscheduled removals

Scheduled inspections consist of four basic checks:

- A walk around check which is performed routinely and as a minimum check every 10 flight hours,
- A line check which is performed approximately every 35 hours,
- A base check which is performed approximately every 1,000 hours (can coincide with periodic check of the engine or aircraft) in which the spinner is removed, and
- A major check which is performed approximately every 18 months (about 4,500 operating hours) to coincide with a major shop aircraft check.

Unscheduled maintenance includes blade line repairs and unscheduled removals of major components such as the spinner, disc and aft fairing, pitch change modules, blades, and forward cover and fairing. A significant factor in the maintenance cost of Prop-Fan hardware is the design philosophy at Hamilton Standard. This philosophy includes designing both the Prop-Fan blade and hub for infinite life. Consequently, these items will only require replacement in the event of an accident or significant foreign object damage (FOD). Blades are repairable for all FOD except cases where spar damage is evident. Therefore, there will be no life limit on major parts, and accordingly, there will be low maintenance cost associated with scrap. Another design characteristic is the absence of major components that will be subject to replacement due to wear. Periodic replacement of the few secondary parts subject to wear is not a significant contributor to the maintenance cost.

Maintenance cost estimates for the unscheduled removals of the Prop-Fan system were obtained by adding removals of all the advanced technology pitch change modules to maintenance costs of the spinner, blades, disc, and fairing. Costs for unscheduled removals reflect both line manpower and shop costs to repair faulty components.

The maintenance cost projections for the advanced turboprop propulsion system were generated by multiplying line and shop labor cost estimates (converted to dollars using 1984 fully burdened labor rates) and material charges per maintenance action by the corresponding rate of maintenance action or repair. The line and shop labor cost estimates are based on industrial engineering evaluation of the design in conjunction with historical data for similar hardware.

Parts cost per event were developed using estimated acquisition costs and historical data relating per repair material costs to acquisition costs on a percentage basis. The Cost Engineering Group developed the Prop-Fan acquisition costs by analyzing the hardware as defined on the concept drawings and the developed parts list. This analysis uses standard techniques for estimating production hardware costs including comparisions with costs for similar parts currently in production. The maintenance manhours per 1,000 flight hours include both scheduled inspections and all unscheduled maintenance. The parts cost assumes 1984 economy and includes all unscheduled maintenance. Based on the maintenance philosophy Hamilton Standard established for the Prop-Fan system, all unscheduled actions have been accounted for. This includes maintenance actions where hardware is removed as well as actions where repair is accomplished on the aircraft.

The Prop-Fan with an internal current technology pitch control exhibits a one percent increase in maintenance cost relative to the base Prop-Fan with an external, current technology pitch control concept. The total maintenance cost for the Prop-Fan with an advanced technology pitch control represents a nine percent decrease from the baseline Prop-Fan configuration. (Reference Hamilton Standard Report SP04A82, "Prop-Fan Data Package: Maintenance Estimates," May 1982 as escalated for 1984 economy.) Table 4.2.8-1 is a summary of the Prop-Fan maintenance costs relative to the base for both current and advanced technology pitch control concepts. The lower maintenance cost of the advanced system results primarily from a reduction in the frequency of maintenance actions.

Table 4.2.8-1
PITCH CONTROL MODULE MAINTENANCE COST SUMMARY
(For In-Line Configuration)

	CURRENT TECH.		ADVANCED TECH.
	External	Internal	External
Maintenance Cost	Base	+ 1%	-9%

4.2.8.2 Acquisition Cost

The Prop-Fan with an internal current technology pitch control has an acquisition cost which is approximately two percent greater than a Prop-Fan with an external current technology concept. The acquisition cost for a Prop-Fan with the external advanced technology pitch control concept is approximately equal to the baseline external current technology concept. Acquisition costs estimates were developed as described in the section on maintenance costs.

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SECTION 4.3 -- DISCUSSION OF RESULTS

TASK XI - THE PRELIMINARY DESIGN OF A COUNTER-ROTATION GEARBOX

Table of Contents for Section 4.3

List of Illustrations for Section 4.3

List of Illustrations for Section 4.3 (continued)

List of Tables for Section 4.3

4.3 Preliminary Mechanical Design of a Counter-Rotation Reduction Gearbox -- Task XI

4.3.1 Introduction

The objective of Task XI was to complete a preliminary mechanical design of a counter-rotation reduction gearbox to meet the requirements of future Prop-Fan propulsion system.

The design of the APET counter-rotation gearbox is based largely on the results of the NASA-sponsored Counter-Rotating Propeller/Gearbox Study (NAS3-23043). The study identified a differential planetary gear system as offering the greatest potential for a counter-rotating Prop-Fan. This concept, along with a number of advanced technologies, provided the basis for the APET gearbox preliminary design effort.

4.3.2 Design Goals and Requirements and Reference Gearbox Design

Design Goals and Requirements

The design goals and requirements for an advanced technology counter-rotation gearbox are the same as those discussed in the single-rotation section with the addition of two requirements. They are:

- Separate the engine and airframe accessories from the Prop-Fan drive gearbox -- Early turboprop gearboxes had the accessories supported on and driven by the prop drive gearbox. Over 50 percent of the maintenance problems with these gearboxes was associated with the accessory drive parts. Remotely mounting the aircraft accessories locates them in an area similar to that of today's turbofan engine accessories. If they do develop problems, they can be replaced as sub modules without replacing the prop drive gearbox.

- Provide accurate and quantified condition monitoring information -- By integrating various condition monitoring parameters such as chip detection, temperature, vibration, and pressure measurements and by processing this information in a computer tied in with the engine control, a state-of-the-art condition monitor can flag maintenance action requirements accurately.

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Table 4.3.2-1 summarizes specific operating parameters and Prop-Fan drive and cooling requirements for a counter-rotating system. The operating parameters include transferring 12,000 hp to the Prop-Fan blades at 1,233 rpm. This matches the drive requirements for a 6 x 6 bladed, 3.5 m (11.6 ft) diameter Prop-Fan. The loads that the Prop-Fan impose on the gearbox include the prop weight of 816.5 kg (1,800 lb), a thrust load of 92,556 N (20,830 lb), and the 1P aerodynamic shear and moment loads. The Prop-Fan cooling requirements include providing a minimum of 15.4 kg (34 lb) of oil to the Prop-Fan pitch control unit at a maximum oil supply temperature of 338°C (170° F) at ambient pressure and then accepting this oil back with a 122°C (50° F) increase in temperature after it has cooled the Prop-Fan pitch change system.

Table 4.3.2-1 Prop-Fan Gearbox Design Characteristics

Prop-Fan drive requirements	
Max power, HP	12,000
Gear ratio	7-11
Prop diameter, M (ft)	3.5 (11.6)
Tip speed, m/sec (ft/sec)	2286 (750)
Output shaft speed, rpm	1233
Max output torque, N-m (ft-lb)	38054 (28067)/ 31135 (22964)
Max total prop thrust, M (lb)	92656 (20,830)
Max '1P' moment, N-m (ft-lb)	8921 (6580)/9857 (7270)
Max '1P' shear, N (lb)	7317 (1645)/6361 (1430)
Max gyro moment at 0.2 rad/sec, N-m (ft-lb)	3186 (2350)/3186 (2350)

*Distance from CG to prop/gearbox shaft flange interface

Prop-Fan cooling requirements	
Oil flow kg/min (lb/min)	15.4 (34)
Max oil inlet temperature, °C (°F)	76.7 (170)
Max inlet oil pressure	Ambient
Max temperature rise, ΔT, °C (°F)	10.0 (50)

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In addition to the above specific operating parameters, the variations of operating conditions throughout the flight also determine the design of the gearbox. Table 4.3.2-2 summarizes the flight mission profile representing a typical short range (741.3 km or 400 nmi) mission. This typical mission assumes a flight profile where most of the time is spent climbing and descending from a cruise altitude of 10,668 ms (35,000 ft). While Mach 0.8 was used in this study, previous work has indicated that the flight duration times are not significantly affected whether the cruise speed was Mach 0.7 or Mach 0.8.

Table 4.3.2-2 Flight Mission Profile for Gearbox Duty Cycle Analysis

Condition	Duration (minutes)	Altitude 304.8 M (1000 ft)	Flight speed (MN)	Power (% max)	Prop-Fan speed (% max)
Taxi (Ground idle)	5.0	0	0	2 – 5	20 – 70
Takeoff	1.5	0 – 1.5	0 – 0.39	100	95 – 100
Climb	2.4	1.5 – 10	0.39 – 0.5	88 – 81.3	100
	3.8	10 – 20	0.5 – 0.6	81.3 – 70	100
	8.9	20 – 30	0.6 – 0.74	70 – 58.7	100
	5.9	30 – 35	0.74 – 0.8	58.7 – 53.3	100
Cruise	20.0	35	0.8	43.3	100
Descent	20.0	Variable	Variable	2 – 5	30 – 70
Approach	3.0	Variable	Variable	20 – 25	75 – 100
Reverse	0.5	0	0.2 – 0	22 – 6	60 – 80
Taxi (ground idle)	5.0	0	0	2 – 5	20 – 70

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Reference Gearbox Design

The Counter-Rotating Propeller/Gearbox Study (NAS3-23043) Pratt & Whitney conducted in 1983 for NASA provided the starting point for the present preliminary design study. This previous effort surveyed all known gearbox drive concepts and identified five offset and five in-line concepts for further study. These concepts are shown in Figures 4.3.2-1 and 4.3.2-2. A forced decision selection methodology for screening the gearbox concepts was used. Rating parameters, summarized in Table 4.3.2-3, were ranked in terms of importance as determined by previous experience. As indicated, the range of parameters covers performance, economic, and installation related considerations.

The five offset gearbox candidates were:

- The dual compound idler
- The dual compound idler with reversing idler
- The dual compound bevel
- The spur with reversing idler
- The spur-differential planetary

The five in-line gearbox candidates were:

- The planetary with reversing bevel
- The compound planetary
- The multiple compound idler
- The split path planetary
- The differential planetary

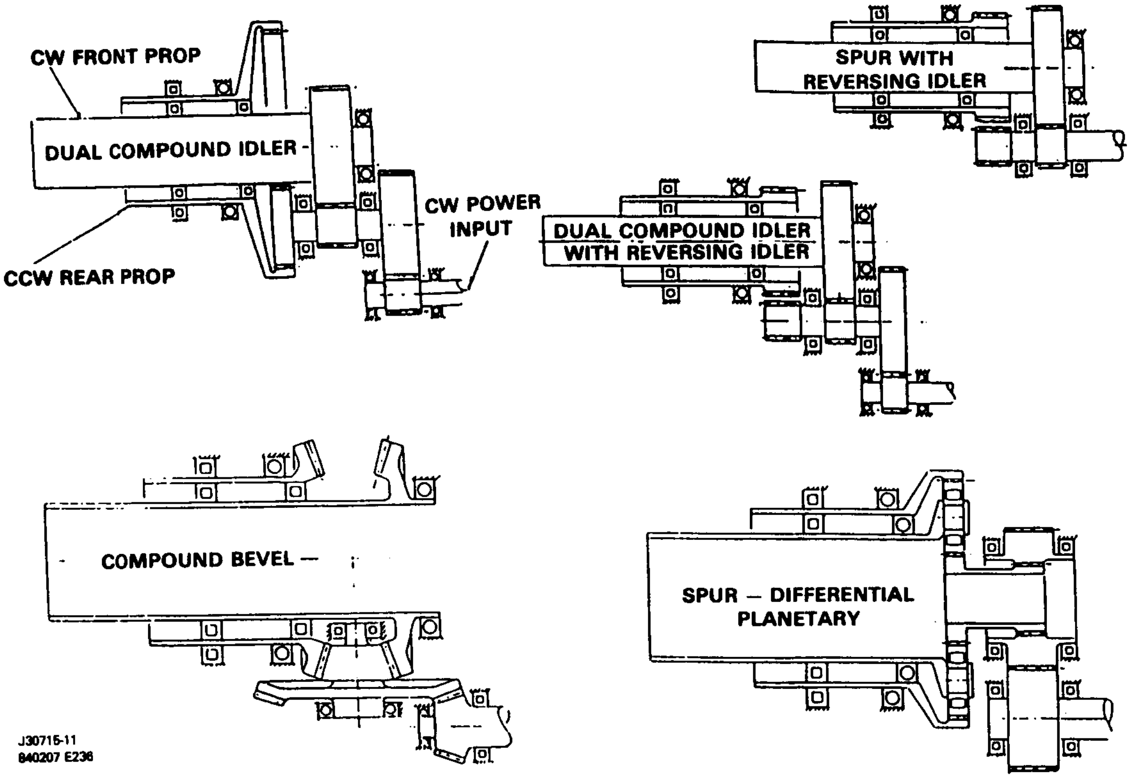

Figure 4.3.2-1 Offset Counter-Rotating Gearbox Candidates - Concepts were rated according to the forced decision analysis.

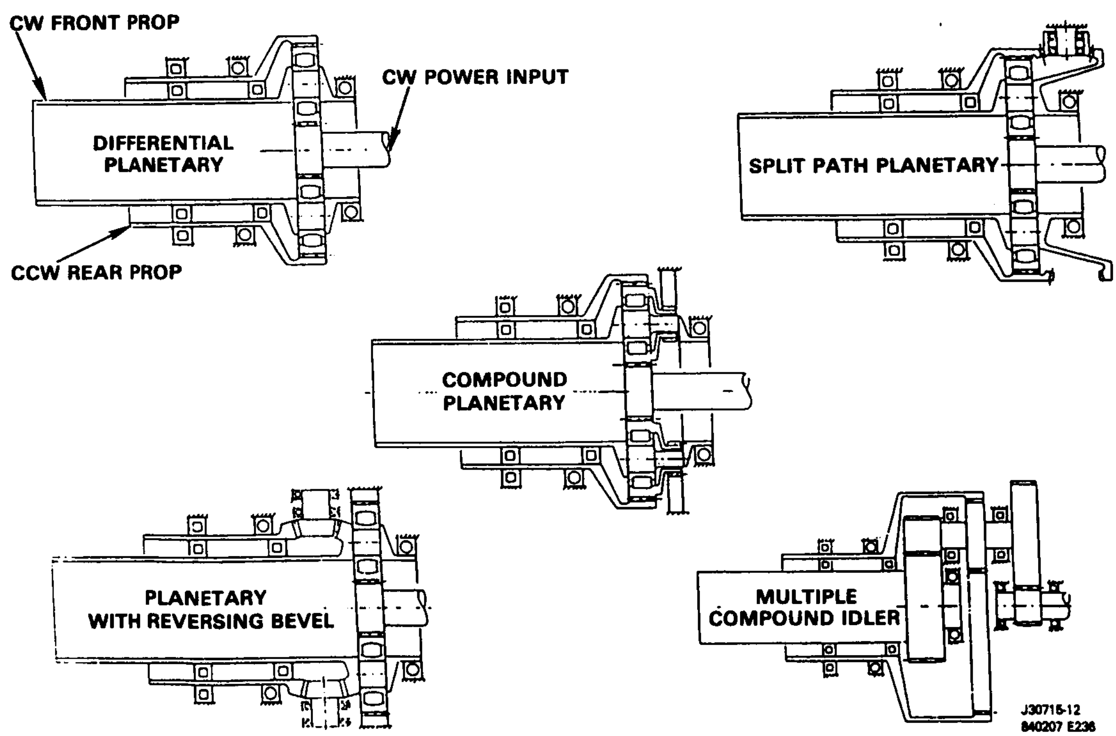

Figure 4.3.2-2 In-Line Counter-Rotating Gearbox Candidates - Concepts were rated according to forced decision methodology.

Table 4.3.2-3 Counter-Rotating Reduction Gear Forced Decision Evaluation Parameters

	Weighting factor (Property emphasis coefficient)
Reliability	0.18
Efficiency	0.17
Maintenance	0.13
Acquisition cost	0.12
Pitch control accessibility	0.12
Weight	0.11
Technical risk	0.08
Ease of scaling	0.04
Acoustic signature	0.03
Spatial envelope	0.02
	1.00

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Of the five offset concepts, the dual compound idler and the dual compound idler with reversing idler were rejected, because they are relatively heavy and complex, and they have large diameters. The compound bevel was rejected, because it is very heavy, and the spur with reversing gear was rejected, because it is too large. The spur-differential planetary concept was selected.

Of the five in-line candidates, the planetary with reversing bevel was rejected, because it is heavy and inefficient. The compound planetary was rejected, because it is relatively complex and less reliable. The multiple compound idler and the split path planetary were rejected, because they are heavy, complex, and difficult to maintain. The differential planetary was chosen, because it is simple, light, and efficient.

Figure 4.3.2-3 summarizes the results of the forced decision selection process. As shown, the differential planetary system has the highest figure of merit. It is superior in nearly all categories, especially in reliability, efficiency, maintenance, and acquisition cost. The only category where the in-line differential system does not rate high is accessibility to the pitch control system. At the time of this study, the pitch control in the in-line system was located in the gearbox, and any servicing of the pitch control required disassembly of the gearbox. This gave an advantage to the offset system, because the pitch control in the offset system could be serviced without disassembly of the gearbox.

In conclusion, this early study clearly identified the differential planetary gear system as being the best choice for counter-rotation Prop-Fans. In addition, this study identified that a remote pitch control system is necessary for the optimum in-line gearbox drive system.

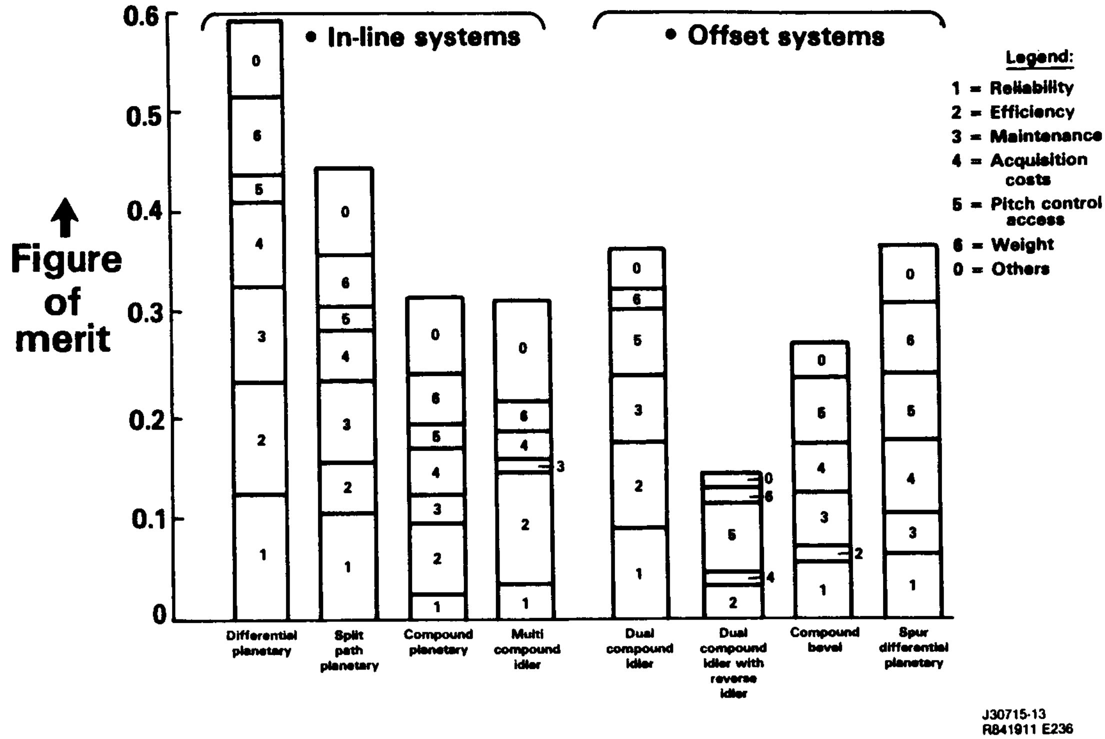

Figure 4.3.2-3 Differential Planetary Concept Has Greatest Overall Potential - This concept is superior in reliability, efficiency, maintenance, and acquisition cost.

4.3.3 Refinement and Mechanical Design

4.3.3.1 Refinement Analysis

The selection of the differential planetary in-line arrangement was, in essence, the starting point of the preliminary design effort. This effort consisted of design refinement studies which provided a basis for selecting the best configuration.

The differential planetary concept is the most simple epicyclic (planetary) counter-rotation system. In a tractor propeller configuration, this arrangement operates as follows. The front prop is driven through the planet pinion carrier, while the rear prop is driven through the ring gear. The carrier rotates in the same direction as the input shaft, and the ring gear rotates in the opposite direction. The blade pitch of each propeller controls the power split and relative propeller speed. Hamilton Standard determined that using equal Prop-Fan speeds was best for this arrangement, and this results in an unequal torque distribution to each prop.

Four different structural arrangements were configured using the differential gearing to optimize the differential planetary concept. Each system has a unique support structure, and each was evaluated using the critical parameters from the conceptual studies. The fifth gearbox arrangement in this study is a non-differential grounded system. The inclusion of the non-differential grounded system was prompted by concern over controlling rotor speeds in a failure mode, although Hamilton Standard is confident that this is not a problem in any failure mode.

4.3.3.1.1 Candidates

The five in-line planetary candidates evaluated in the optimization studies were:

- o The straddle-mounted
- o The cantilevered
- o The close coupled
- o The inter prop
- o The grounded planetary (split path)

Straddle-Mounted

The straddle-mounted designation relates to the prop shaft/ring gear support bearings which are fore and aft of the gear set. This design appears in Figure 4.3.3-1. This design provides a substantial wheelbase for the prop shaft and reduces the overall length of the gearbox. The carrier and sun gear shaft support bearings are carried within the prop shaft as intershaft bearings. This arrangement minimizes the motion at each gear mesh when the prop shaft reacts under prop loads.

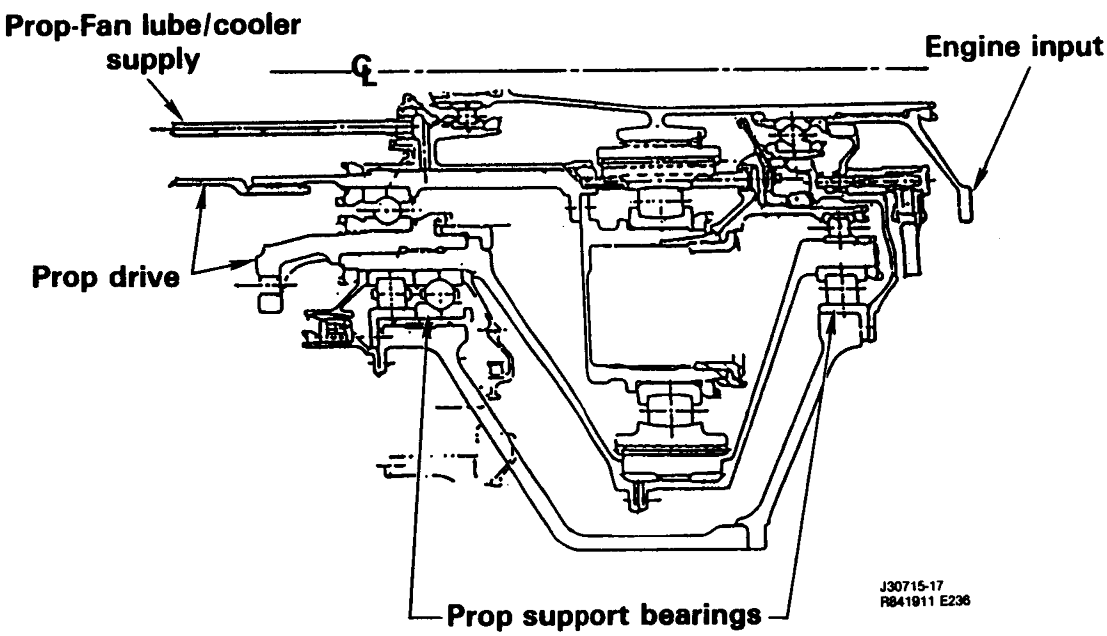

Figure 4.3.3-1 Straddle Mounted Differential Planetary - In this design, the prop shaft/ring gear support bearings are fore and aft of the gear set.

Cantilevered

In the cantilevered design, shown in Figure 4.3.3-2, the rear support bearings for the ring gear and carrier are forward of the gear set. In this design, the Prop-Fan loads are taken directly from the shaft to the support structure in the housing and do not go through the gear mesh.

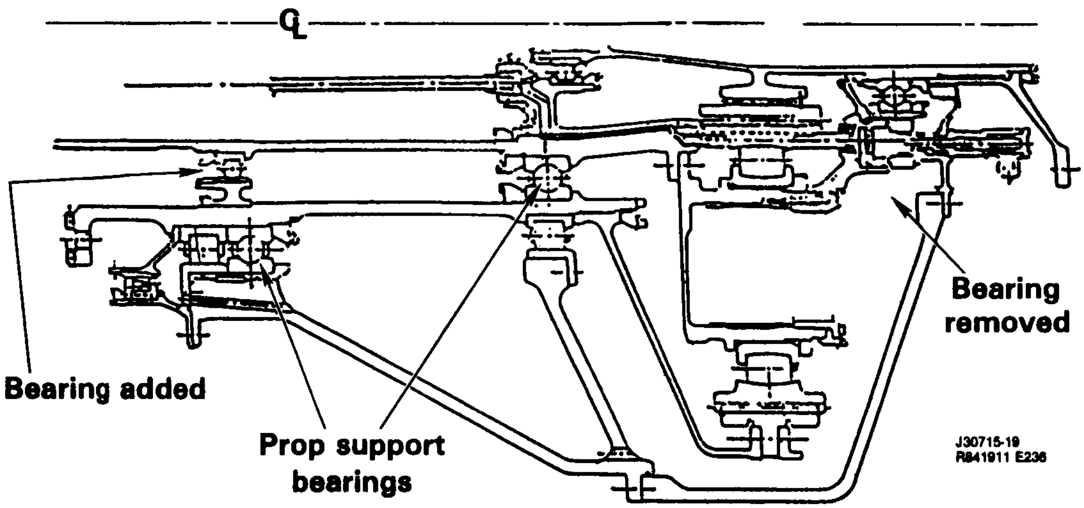

Figure 4.3.3-2 Cantilevered (Ring Gear/Carrier) Differential Planetary Gearbox Concept - In this design, the rear support bearings for the ring gear and carrier are forward of the gear set.

The penalty of this arrangement is the additional 15.24 cm (six in) necessary for providing an adequate wheel base to support the Prop-Fan shaft. As in the straddle-mounted design, the prop shaft supports the support bearings for the carrier and sun gear shaft as intershaft bearings.

Close Coupled

The overall propulsion system length is a key factor in minimizing installation problems. In the close coupled and inter prop configurations, a successful effort was made to reduce the length of the combined Prop-Fan gearbox package.

The close coupled concept, shown in Figure 4.3.3-3, attached the reduction gearing directly to the second stage Prop-Fan through the ring gear drive shaft. This shaft now provides both torque and housing for the gearbox. It extends into the engine and is supported on bearings grounded to the engine housing. Potential problems relating to this configuration were primarily in the lubrication system. A rotating oil supply pump could not be removed without completely disassembling the gearbox. Concern over the reliability of bypass valves in a G-field and a complex oil supply/scavenge transfer system were some of the problems related to this system.

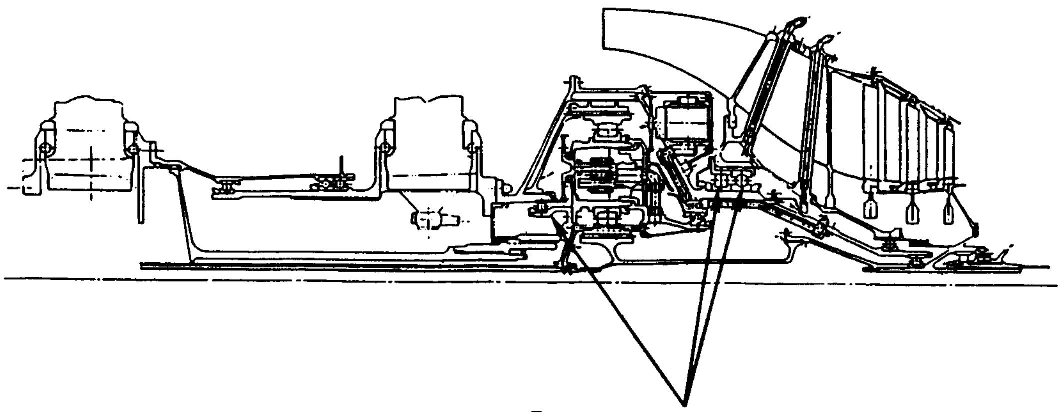

Figure 4.3.3-3 Close Coupled Differential Planetary Gearbox Concept - In this design, the reduction gearing is attached directly to the second stage Prop-Fan through the ring drive shaft.

Inter Prop

The inter prop gearbox, shown in Figure 4.3.3-4, provides the most compact installation. In this arrangement, the planetary gear set is integrated into the Prop-Fan by supporting the carrier from the forward prop and by coupling the ring gear to the aft prop hub. A segmented quill shaft, supported at mid-span, drives the sun gear from the engine. The prop pitch control is within the gear system, but it works independently of the reduction gear set.

The advantages of this system are shorter length and lighter weight, but the Prop-Fan pitch change mechanism and gearbox share the oil in all the systems being designed. This and loss of modularity offset these advantages.

Grounded Planetary (Split Path)

The split path planetary concept, shown in Figure 4.3.3-5, converts the differential planetary configuration to a grounded system with a fixed speed ratio for each propeller. As in the differential planetary, the planet pinion carrier drives the front prop. The forward prop and input shaft rotate in one direction, while the rear prop rotates in the other direction. The housing supports the multiple idler gears, grounding the differential system and imposing equal and opposite prop speeds at any propeller power split. Changes in propeller pitch in this system cannot influence the propeller power or speed split, and this could simplify the propeller pitch control for this system. However, this arrangement did not compare well in most evaluating categories. The additional number of gears and bearings had a negative impact on reliability, efficiency, maintenance cost, acquisition cost, and weight.

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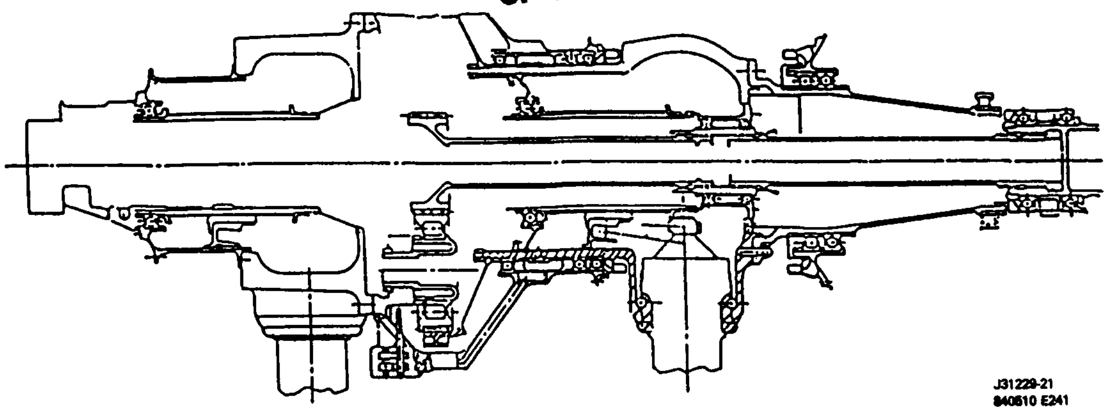


Figure 4.3.3-4 Inter prop Differential Planetary Gearbox Concept - Of the five candidates, this arrangement is the most compact.

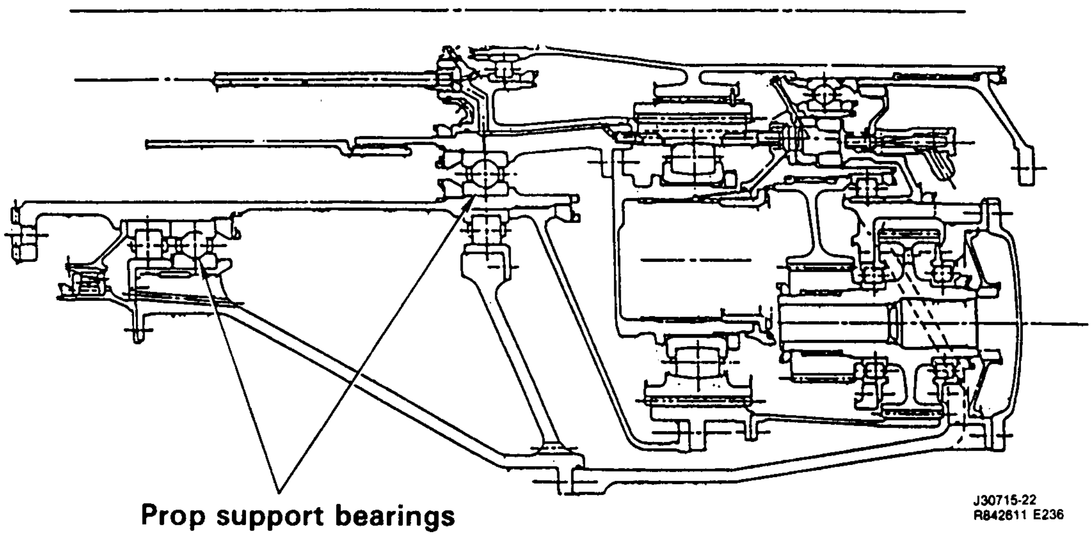


Figure 4.3.3-5 Grounded Planetary Gearbox - This concept converts the differential planetary configuration into a grounded system with a fixed speed ratio for each prop.

4.3.3.1.2 Evaluation

The five concepts were evaluated using the forced decision screening process in which seven parameters, weighted in terms of importance, determined the best concept. The parameters were: reliability, efficiency, maintenance, acquisition cost, pitch control risk, weight, and installation considerations. The analysis included sizing the gears and bearings and conceptually designing each configuration to identify the number of gears, bearings, and spacial envelope requirements. This information provided preliminary estimates to assess the reliability, technical risk, and installation considerations.

Figure 4.3.3-6 summarizes the results of this evaluation. Note that the split path grounded system did not compare well with the other arrangements. Weight, maintainability, reliability, and efficiency parameters penalized this arrangement. The straddle-mounted and cantilevered arrangements rated close, with a slight advantage given to the straddle-mounted, because it has a shorter installation length. The forced decision analysis identified both of these designs as being superior in most categories, so both were chosen for further study before a final selection was made.

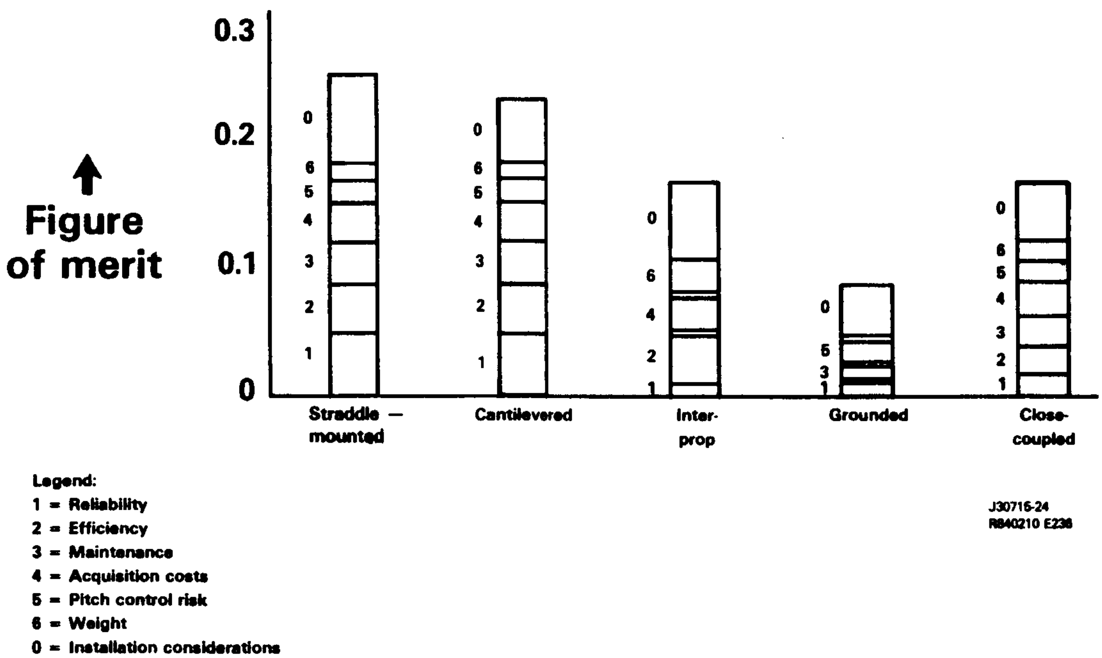


Figure 4.3.3-6 Counter-Rotating Gearbox, Forced Decision Analysis Comparison - The straddle mounted and cantilevered arrangements were chosen for further study.

The highlights of this evaluation are as follows:

Reliability

A common lubrication system necessary for both the inter prop and close coupled arrangement and an increased part count for the grounded system caused lower system reliability in all three of these gearbox arrangements.

Efficiency

The additional gears and bearings necessary in the grounded split path gearbox increased power losses in this system. All of the other gearbox arrangements were very close in efficiency levels.

Maintainability

A significant drawback of the inter prop gearbox is its reduced modularity. High maintenance costs will directly reflect this. The grounded system also received a low rating, because it has a high parts count. The close coupled arrangement received a low rating as well, because it suffers from the buried oil pumps located in the carrier posts.

Acquisition Costs

This comparison came out a draw for four of the five gearboxes. The only loser was the grounded system, because its high parts count raised the estimated acquisition cost.

Pitch Control Risk

Hamilton Standard's assessment determined that the only gearbox to have a problem with the pitch control was the inter prop. Combining the two systems allows debris from one to contaminate the other. This increases the risk of potential problems.

Weight

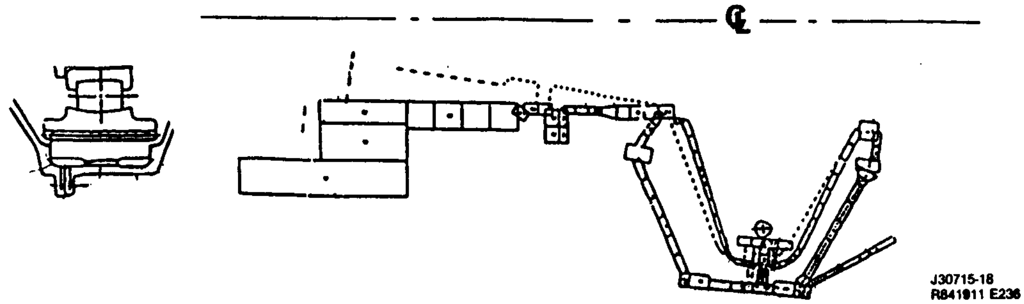
The weight evaluation determined that the grounded system is the worst arrangement in terms of weight, because of its high parts count. The interprop was the best by a slight margin.

Installation Compatability

Both pusher and tractor installations were considered for each gearbox evaluation. Accessory drive requirements, installation length, and torque measuring devices were all factors influencing the selection process. The results of the comparison gave the straddle-mounted arrangement a slight edge over the other gearboxes.

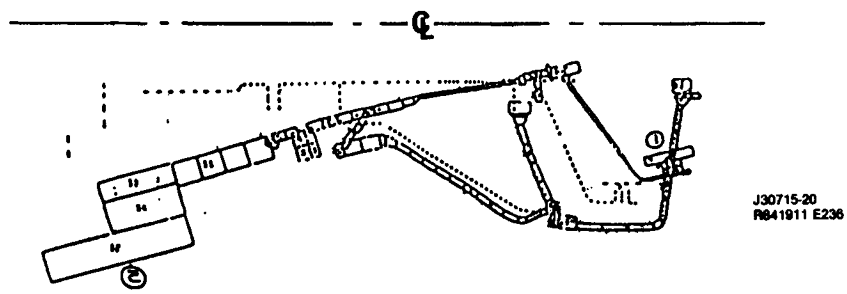
To compare fuel burn and direct operating costs, additional analysis of the straddle-mounted and cantilevered arrangements evaluated the impact of the prop shaft loading on the gear mesh and on the overall system. A shell analysis was conducted under a 1.5 and 1P shear load of 26,689 N (6,000 lb) to determine the impact of shaft and hub deflection on the gear mesh misalignment. A slope of 0.00101 cm/cm (0.0004 in/in) is acceptable in normal gear operation. The analysis showed that the resultant 0.00035 cm/cm (0.00014 in/in) misalignment is well within the allowable limit. Figure 4.3.3-7 shows the results of the straddle-mounted analysis, and Figure 4.3.3-8 represents the cantilevered analysis. The slope and deflection analysis highlights the major advantages and disadvantages of each arrangement. The cantilevered arrangement supports the gear package as a unit. Deflection of the prop shaft cannot generate slope differences between gears, because all gears move as a unit. However, the carbon seal located on the input shaft is grounded to the housing; therefore, there is a slope difference generated between the carbon seal land located on the rotor and the seal. The calculated slope is 0.00254 cm/cm (0.001 in/in) which is considered excessive for this type of seal arrangement.

Table 4.3.3-1 shows the gearbox technical evaluation comparison. The ratings of the two designs are essentially equal in terms of fuel burn and direct operating costs. A slight advantage in reliability, maintainability, and weight, as well as a 15.2 centimeter (six in) reduction in length, brought about the selection of the straddle-mounted arrangement.



Relative slope at ring/planet gear mesh — 0.00014

Figure 4.3.3-7 Straddle Mounted Concept Shell Analysis - The analysis shows calculated gear slope/misalignment from prop loads to be acceptable.



1. Relative slope at ring/planet gear mesh = 0

2. Slope at drive coupling end = 0.001

Figure 4.3.3-8 Cantilever System Shell Analysis - The analysis shows no significant misalignment from prop load.

Table 4.3.3-1 Gearbox Technical Comparison

System evaluation

	Straddle	Cantilever
Efficiency	Base	Base
Reliability (MTBUR), hr	Base	−5800
Maintainability $/EFH	Base	−0.25
Weight, Kg (lbs)	Base	+9.1 (+20)
Cost difference	Base	−4%
Technical risk	—	Ring shaft vibration
Gear mesh misalignment	Acceptable	0
Installation length, cm (in.)	Base	+15.2 (+6)
• Fuel burn	Base	+0.04%
• DOC	Base	−0.03%

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4.3.3.2 Mechanical Design

As shown in Figure 4.3.3-9, a preliminary design of the selected straddle-mounted arrangement uses advanced technology features. The advanced technologies are the same as those found in the single-rotation design. These features include a modulated lubrication system with an aerodynamic lubricant scavenge system, advanced bearing and gear materials, integral gear and bearings, and high contact ratio buttress gear tooth form.

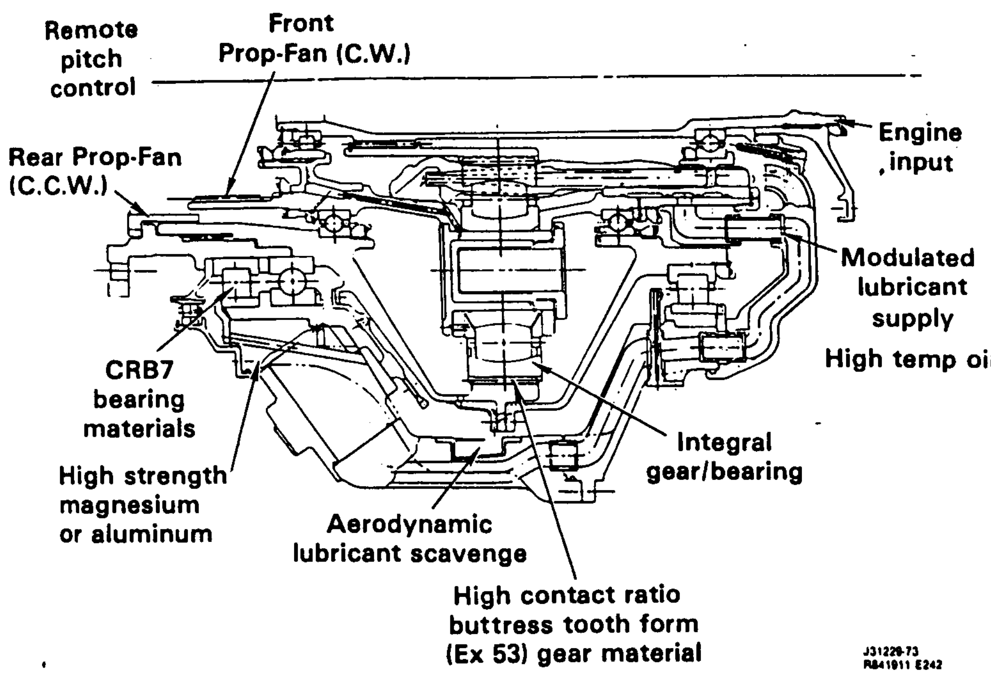

Figure 4.3.3-9 Advanced Technology Features - The advanced technology features for the Counter-Rotation gearbox are the same as those used in the Single-Rotation design.

Design of the planetary gear set included a planet pinion optimization study and a review of the arrangement's impact on the gas generator power turbine. Initially, a reduction ratio of 8.6 to 1 was established. This requirement came from a preliminary power turbine design study and Prop-Fan rpm requirements. This reduction ratio limited the planetary pinion count to a maximum of four. A small change in the reduction ratio (8.23 to 1) would allow the use of five pinions with a potential weight savings. Both four and five pinion designs were considered for the final design.

The selection process for sizing the planetary gearset is based on building a matrix of gearsets using two variables, the sun gear tooth count and pitch. All possible choices of planetaries were tried using a range of pitches from four to 16 and varying the sun gear tooth count from Nmin = Pitch dia x pitch to five percent greater than the minimum. This analysis identified three gearsets (one four pinion and two five pinion) that met all of the basic requirements. The concepts then underwent further refinements.

A trade study considering the power turbine speed, flowpath modification, efficiency, and weight was undertaken. As a result of the study, the smaller five pinion design was selected, because it has a slight advantage in weight, fuel burn, and direct operating costs. (See Table 4.3.3-2.)

Table 4.3.3-2 Summary for Four and Five Pinion Gear Design

	4 pinion	5 pinion
No. of bearings	6	7
No. of gears	11	12
Efficiency	Base	Base
MTBUR, hours	31,800	30,000
Acquisition cost, $	Base	−1000
Maintenance cost, $	Base	+0.16
Weight, Kg (lb)	Base	−11.3 (−25)
Fuel burn	Base	−0.03%
DOC + I	Base	−0.001%

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The high-power and high-speed levels required for the proposed Prop-Fan installations require larger gears and faster pitch line velocities than are currently used in turboprop transmissions. The proposed counter-rotating gearbox will have gear speeds nearly twice that of existing transmissions. The gear tooth dynamic loading is directly related to both speed and accountable tooth tolerances. As gear speeds increase, the dynamic increment on the nominal loading will increase. The dynamic increment is that part of the load induced by imperfections in gear manufacturing (tooth profile and relative position). No gear will ever be perfect; therefore, such imperfections are always be present. Another contributing factor to dynamic loading is elastic deformation of the teeth. As the teeth deflect, the load on a given tooth fluctuates which in turn causes cyclic tooth loading. Due to the high pitch line velocities inherent in the counter-rotating gearbox, the dynamic load will need very precise control. The greatest single contribution will be tooth form. Conventional spur gears operate with contact ratios of 1.4 to 1.7 and are noisy and prone to vibration at high speeds. There are two possible approaches to resolving the above problems: either using helical gearing or using high contact ratio (HCR) spur gears. Helical gears require a more complex bearing support system; therefore, the high contact ratio approach was used. High contact ratios are achieved by using teeth of a relatively low pressure angle, which brings about a tall, slender configuration. This form suffers a high root bending stress. To compensate for this, a buttress tooth form was used to lower stress levels in the root area.

The technologies applied to counter-rotation gearbox bearings are the same as those applied to single-rotation bearings. A two to threefold increase in material and lubricant life relative to current technology allows the use of smaller bearings and gears which reduces gearbox weight and power loss. Several changes will yield increased life factors. Improved material composition and processing will raise fatigue strength and retard surface originated fatigue (through improved corrosion and wear resistance). Better lubricants will increase lubricant film thickness. Reduced surface roughness will further retard surface originated fatigue, and fine lubricant filtration will help preserve the initial surface finish.

The planet bearing application in counter-rotation represents a greater technical challenge than single-rotation, because the higher ring gear speed increases planet gear speed and rolling element centrifugal loading.

While not reflected as increased DN level because the counter-rotation bearing cross section is heavier and the bore diameter (D) smaller, the speed effect on bearing friction and wear is important. The planet carrier speed presents a technical challenge for both single-rotation and counter-rotation applications and appropriate technology programs must assure roller pocket and cage land durability.

Other critical bearings in the counter-rotating gearbox are those supporting the output shaft which carries the Prop-Fan assembly. This is the ring gear shaft or the rear prop shaft in the counter-rotating tractor installation.

As in the single-rotation gearbox, high thermal growth of the high expansion light alloy housing leads to loose operating fits and clearances and, in the presence of Prop-Fan moment loads, prop shaft angular displacement causes bearing misalignment. While bearing geometry and capacity modifications can accommodate moderate amounts of misalignment (in the range of 0.0015 to 0.0020 radians), the relatively large Prop-Fan mass and moment arm of the counterrotation installation led to a study of alternative prop shaft bearings.

The first option, the combination of a ball bearing with cylindrical roller bearings spaced apart is the conventional or baseline configuration and is the base to which the other concepts are compared (see Figure 4.3.3-9). In this arrangement, the ball bearing carries only thrust loads fore and aft, while the cylindrical roller bearings support radial reactions to all shear and moment loads applied by the Prop-Fan. The span between roller bearings determines the reaction load magnitude and bearing misalignment.

As shown in Figure 4.3.3-10, an option with two tapered roller bearings spaced apart offers two advantages. The number of bearings supporting the prop shaft is reduced by one, improving reliability and maintainability. The tapered roller bearings increase the effective span between bearings, improving transfer of Prop-Fan loads to the gearbox housing. When shaft and axial housing displacements properly preload the tapered roller bearings, the bearings will operate without radial clearance, minimizing shaft radial displacement and bearing misalignment. The technical challenge is to cause the relative axial differential thermal growth between shaft and housing to reduce bearing clearance thus compensating for the effect of radial differential thermal growth which increases bearing clearance. The intended clearance values can be obtained providing that thermal and preload effects are consistent and predictable and the shaft and housing components have no plastic deformation.

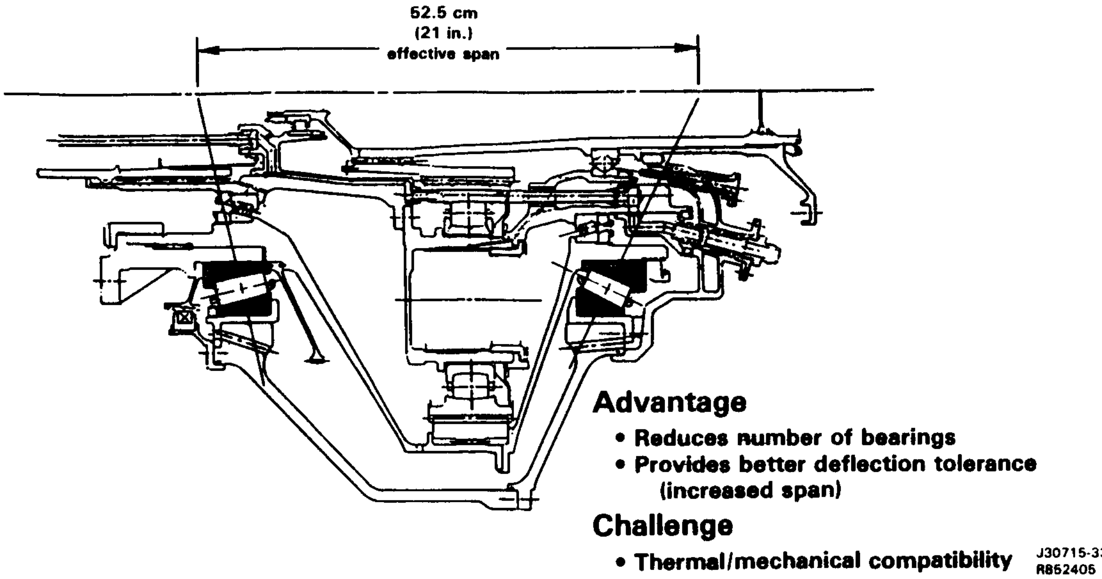

Figure 4.3.3-10 Tapered Roller Bearings (Spaced Apart) Provide Potential Benefits - The net advantages are improved reliability and maintainability.

As shown in Figure 4.3.3-11, another option is duplex tapered bearings with a cylindrical roller bearing. This is closer to the base, but with some potential advantages over that option. While the effective span is unchanged relative to the base, the duplex bearings provide a more direct path for moment transfer from the shaft to the housing. This effect could potentially reduce prop shaft bending and ring gear distortion, thereby improving load sharing between the planet gears. Shaft displacement and bearing misalignments will be reduced relative to the base but not as much as the first option. Moment loading of the duplex tapered bearings will require increased bearing size and weight relative to both options.

Direct moment transfer reduces deflections for increased tolerance to shock and vibration

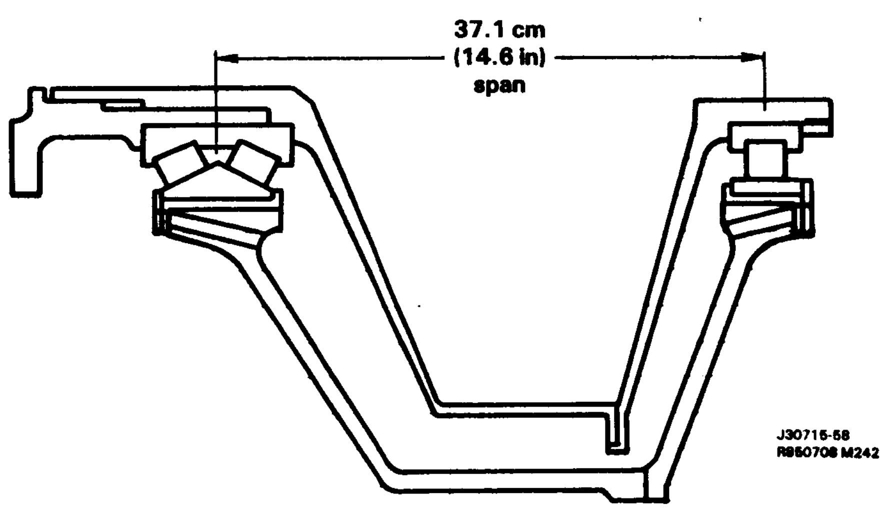

Figure 4.3.3-11 Duplex Tapered Bearings Transfer Shaft Bending Moments Directly to the Housing - This could potentially reduce prop shaft bending and ring gear distortion.

The use of tapered bearings in the second and third options increases bearing losses relative to the base. This effect is due to the axial load that must be applied to every loaded roller in proportion to the roller normal load and roller cone angle to keep the roller in equilibrium. This axial load is applied through sliding contact with the roller guide flange and causes bearing friction drag and power loss.

Table 4.3.3-3 shows an overall comparison of the three designs examined for the prop shaft support bearings. A qualitative assessment of the technical risks assigned to each appears in terms of a relative risk/benefit ratio. It was on the basis of this factor that the baseline ball and cylindrical roller bearing option was selected for the counter-rotation gearbox.

Table 4.3.3-3 Optimal Bearing System Risk/Benefit Analysis

Ball and roller bearings were selected

	Ball and cylindrical	Spaced tapered rollers	Duplex tapered rollers and cylindrical roller
Load support	Base	Improved	Improved
Damaged propeller operation	Base	Improved	Improved
Cost	Base	Reduced	Base (±)
Weight	Base	Reduced	Increased
Power loss	Base	Increased	Increased
Risk/benefit ratio	Base	Highest	Increased

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4.3.3.2.1 Maintainability

Maintainability of the differential planetary gearbox presents a substantial improvement over gearboxes of the past. The differential planetary gearbox is much simpler than previous gearboxes, having fewer parts contained in the single stage gear set. It has seven gears and twelve bearings, and it is a concept that lends itself to on-wing maintenance, as is shown in Figure 4.3.3-12. This has been achieved by providing easy access to normal maintenance items such as carbon seals, oil pumps, last chance filters, and oil nozzle jets.

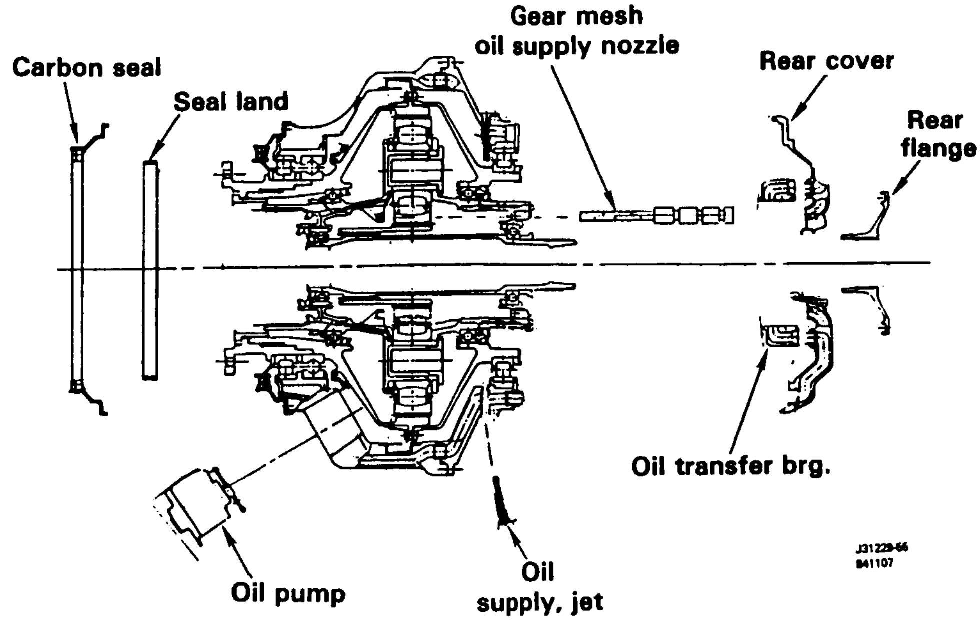
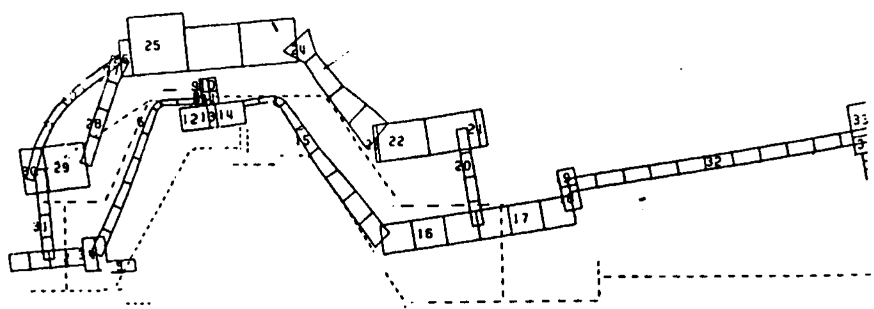


Figure 4.3.3-12 On-Wing Maintenance Capabilities - Easy access is provided to items such as carbon seals, oil pumps, last chance filters, and oil nozzle jets.

4.3.3.2.2 Structural Analysis

Shaft and Housing

A structural analysis was completed on both the housing and Prop-Fan shaft to ensure control of gear mesh misalignment, minimal slope at major bearing locations, and structural integrity of both shaft and housing at maximum operating load conditions. Results are shown in Figure 4.3.3-13. Loads considered were 1.5G and 1P shear prop load of 26,689 N (6,000 lb), 17,083 Nm (12,600 ft-lbs) gyro moment from a steady state basis, and the loss of a single-blade Prop-Fan shell and fill.



	ΔSlope — forward bearing (rad) (0.038 cm (0.015 in)			Ring gear		Max housing stress (bend-MPa (psi))		Max housing stress (bend-MPa (psi))	
Loading	(Defl)	(RC)	(Total)	Slope (rad)	Defl, cm (in)	Actual	Allowable	Actual	Allowable
1.5G + 1P 2721.6 Kg (6,000 lbs)	0.0009	0.0009	0.00181	0.0	0.00132 (0.00052)	8.27 (1,200)	N/A	65.3 (9,470)	N/A
Gyro moment 17083.3 N-m (12,600 ft-lbs)	N/A	N/A	N/A	N/A	N/A	29.7 (4,308)	N/A	206.8 (30,000)	206.8-275.8 (30-40,000)
Shell and fill Blade loss 9480.1 Kg (20,900 lbs)	N/A	N/A	N/A	N/A	N/A	42.1 (6,100)	68.9 (10,000)	332.7 (48,260)	1103 (160,000)

Figure 4.3.3-13 Deflection and Stress Analysis - The design meets structural requirements.

Several iterations were made in sizing both front and rear ring gear hubs in order to minimize prop shaft deflections. Hub stiffness was the key in controlling the slope and deflection at the ring gear, and by optimizing hub angle and wall thickness, an acceptable deflection was obtained. The prop shaft slope at the bearing locations required fine tuning of the gearbox housing as well as the shaft. Figure 4.3.3-13 also summarizes various load conditions and their impact on both gear mesh and bearing slopes. As indicated, the design meets the structural requirements.

A normal load of 1.5G + 1P generated a slope of 0.00228 cm/cm (0.0009 in/in) at the forward prop shaft bearing locations. The bearing internal radial clearance created an additional slope of 0.00228 cm/cm (0.0009 in/in). The combined total of 0.00459 cm/cm (0.00181 in/in) is still within the acceptable level of 0.01016 cm/cm (0.004 in/in) that bearing analyses and proven bearing experience recommend. The slope generated at the ring gear is essentially zero, but the deflection of 0.00132 cm (0.00052 in) is significant in that the ring gear flexibility should accommodate this motion to maintain load sharing among the planet gears.

Once per flight type loads (limit case) and major blade failure (ultimate case), in which the blade shell and filler are lost, do not apply to bearing and gear slope limitations. However, they do impact both housing and shaft design.

The gyro moment generates a cyclic load on the prop shaft, and predictions are that life limitations will be met. For the ultimate case, the loss of blade shell and fill, the goal is to prevent complete destruction of the support system. This type of failure imposes a cyclic load on the housing. An eight minute shut down period was used as a criterion to establish life limitations for the housing. This is equivalent to 10,000 cycles or a 68.9 MPa (10,000 psi) stress limitation. Actual housing stress is 42.1 MPa (6,100 psi) under blade shell and fill loss.

Gear Analysis

For preliminary design of the high contact ratio gears, a simplified gear analysis procedure was used. In this procedure, the gears were designed with conventional tooth form. The resultant gear face width was reduced by 17 percent to reflect the level of size reduction achievable by using high contact ratio gearing. This 17 percent reduction factor was chosen after consideration of a design study Sikorsky conducted on conventional versus high contact ratio gears for the Black Hawk transmission.

Table 4.3.3-4 shows a comparison of conventional gearing and high contact ratio spur gears. The notable difference between them is the gear tooth profile. The Sikorsky Black Hawk (UTTAS) planetary gearset uses a conventional low contact ratio (1.676, 1.771) gear tooth profile, and the advanced planetary gearset uses high contact ratio (2.087, 2.077) gear teeth. The significance of this comparison is that it shows the advantage of high contact ratio gear teeth in lowering stress leads. This translates directly into the ability to reduce gear face widths and, subsequently, to reduce weight. Stress level reductions with high contact ratio vary from 20 to 40 percent at equivalent face widths. A straight reduction in face width was not used, because there were other parametric considerations, including higher sliding velocities. Therefore, the face width reduction ratio factor of 17 percent, used on the APET gearbox, was chosen to account for potential scoring problems.

Table 4.3.3-4 Planetary Gear Technology Background

	Sun	Planet	Ring
UTTAS planetary gears			
Number of teeth	62	83	228
Diametral pitch, cm (in)	22.497 (8.857)	22.497 (8.857)	22.497 (8.857)
Pitch diameter, cm (in)	17.7800 (7.0000)	23.8025 (9.3711)	65.3854 (25.7423)
Pressure angle	22°30′	22°30′	22°30′
Face width, cm (in)	9.144 (3.600)	7.531 (2.965)	7.569 (2.980)
Contact ratio, min	1.676	1.771	
Bending stress, MPa (psi)	321.7 (46,654)	381.2/368.1 (55,289/53,382)	334.6 (48,527)
Contact stress, MPa (psi)	954.0 (138,367)	953.9/504.4 (138,347/73,154)	503.7 (73,050)
Advanced planetary gears			
Number of teeth	62	83	228
Diametral pitch, cm (in)	22.497 (8.857)	22.497 (8.857)	22.497 (8.857)
Pitch diameter, cm (in)	17.7800 (7.0000)	23.8025 (9.3711)	65.3854 (25.7423)
Pressure angle	20°	20/23°	23°
Face width, cm (in)	9.144 (3.600)	7.531 (2.965)	7.569 (2.980)
Contact ratio, min	2.0870	2.0771	
Bending stress, MPa (psi)	224.6 (32,573)	277.4/272.6 (40,227/39,532)	239.8 (34,787)
Contact stress, MPa (psi)	793.7 (115,114)	794.4/409.7 (115,211/59,421)	409.6 (59,408)

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Gear teeth structural design considerations included hertz stress, bending stress, and scoring resistance. The ideal design is one in which the margin of safety is equal for each criterion. Gear bending stress and scoring resistance are a function of diametral pitch, whereas hertz stress is not. Therefore, a gear tooth is not usually limited by bending since in the initial design, the bending stress level will be lowered by reducing the number of teeth in each member, while the hertz stress remains essentially the same. Table 4.3.3-5 is a summary of the gear tooth analysis. Note that the stress levels have not been refined to take full advantage of the allowable stress. This conservative approach was taken to account for potential scoring factors and excessive gear misalignment due to high prop loads and unequal load sharing between planets.

4.3.3.2.3 Bearings

As in the single-rotation gearbox, bearing system life strongly influences overall durability. The 18,000 hour bearing system B_{10} life objective was set to obtain a gearbox MTBUR period of greater than 15,000 hours. This system life objective, together with bearing quantity and technology level, determines how bearing sizes are selected for the critical applications in the gearbox.

Table 4.3.3-5 Gear Operating Condition

Sun/Pinion	Actual MPa	Actual (PSI)	Allowable MPa	Allowable (PSI)
Bending Fatigue				
Unidirectional (SUN)	307.3	(44,575)	413.7	(60,000)
Reversed Bending (PINION)	325.8	(47,260)	337.8	(49,000)
Hertz	894.9	(129,800)	1041.1	(151,000)
Pitchline Velocity m/min (ft/min)	5058	(16,595)	10,668	(35,000)
Sliding Velocity m/min (ft/min)	403	(1,323)	—	—
Surface Finish - Micrometers (Microinches)	0.38	(15)AA	—	—

Pinion/Ring	Actual MPa	Actual (PSI)	Allowable MPa	Allowable (PSI)
Bending fatigue				
Unidirectional (Ring)	270.7	(39,260)	413.7	(60,000)
Reversed bending (Pinion)	287.5	(41,700)	337.8	(49,000)
Hertz	497.2	(72,110)	1041.1	(151,000)
Pitchline velocity m/min (ft/min)	4164	(13,660)	10,668	(35,000)
Sliding velocity m/min (ft/ min)	82	(269)	—	—

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Since the number of planetary bearings has been reduced from eleven in single-rotation to five in counter-rotation and the number of bearings supporting propeller reaction loads is unchanged, the total number of critical bearing locations in the differential planetary gearbox is roughly half that of the split path planetary single-rotation gearbox. This allows individual bearing lives in the counter-rotation gearbox to be lower than bearing lives in the single-rotation gearbox for the same 18,000 hour system life. Table 4.3.3-6 identifies the counter-rotation gearbox bearing locations, selected bearing types and sizes, and "DN" speed levels.

Table 4.3.3-6 Bearing Selection Summary

Location number	Location name (tractor installation)	Bearing type*	Bearing size, mm Bore	O.D.	Width	Speed Factor
						MM XRPM
1	Planet roller	Spherical, 1 row	75	212.16**	72	510,000
2	Rear prop shaft roller,front	Cylindrical, DFI	340	430	42	420,000
3	Rear prop shaft ball, front	Split inner ring	340	430	42	420,000
4	Rear prop shaft roller, rear	Cylindrical, DFI	360	450	42	440,000
5	Front prop shaft ball, front	Split inner ring	254	304.8	25.4	630,000
6	Front prop shaft ball, rear	Split inner ring	254	304.8	25.4	630,000
7	Input shaft roller, front	Cylindrical, DFI	100	150	24	900,000
8	Input shaft ball, rear	Deep groove radial	100	160	28	900,000

*DFI = Double flange inner
DFO = Double flanged outer
**Gear pitch Diameter

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Single-row spherical roller bearings support the set of five planetary gears under operating conditions similar to the first stage planetary bearings in single-rotation. Spherical bearings promote uniform load distribution at each gear mesh, and the single-row bearing is potentially better for high-speed operation than the double-row version.

Planet gear size and loading dictate planet bearing size again, because the gear is integral with the bearing outer ring. Bearing and gear diameters are reduced, until the bearing cross section is matched to the gear and roller dimensions and are in suitable proportion to gear and roller pitch diameters. The selected bearing cross section contains 10 rollers, 3.6 cm (1.42 in) in diameter and 5.1 cm (2.0 in) long, with a 13.6 cm (5.37 in) pitch diameter. The B_{10} fatigue life of each planetary bearing is 65,000 hours. The fatigue lives of all bearings in the counter-rotating gearbox appear in Table 4.3.3-7. The calculated bearing life of 18,900 hours exceeds the 18,000 hour goal.

Table 4.3.3-7 Bearing Life Summary

Location number	Location name (tractor installation)	Bearing quantity	Bearing life (L_{10}) hours
1	Planet roller	5	65,000
2	Rear prop shaft roller, front	1	61,000
3	Rear prop shaft ball, front	1	250,000
4	Rear prop shaft roller, rear	1	390,000
5	Front prop shaft ball, front	1	>500,000
6	Front prop shaft ball, rear	1	>500,000
7	Input shaft ball, front	1	>500,000
8	Input shaft ball, rear	1	>500,000
	Bearing set life	(L_{10})	18,900 hours
	Set life goal	(L_{10})	18,000 hours

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The collective life of the bearing system is the single most important factor controlling gearbox durability. Initial studies indicate that the objective of 15,000 hour MTBUR for the gearbox requires a bearing system that operates with a 50,000 hour mean time between failure. The equivalent 90 percent survival B_{10} life objective is 18,000 hours. This system objective is the governing factor in selecting bearing sizes for highly loaded applications in the gearbox.

4.3.3.2.4 Lubrication System Study

The lubrication system study involved two system lubrication requirements. They were the lubrication of the gearbox and the lubrication of the engine.

Gearbox Lubrication

The counter-rotation gearbox lubrication system employs most of the features selected for the single-rotation gearbox except for supply system details which reflect differences in gear quantity and bearing arrangement. Special features common to both systems include Pratt & Whitney's two-stage modulated oil supply and the aerodynamic scavenge concepts described in Section 4.1.5.3.2. There are added lubrication system features which have been identified for use in the counter-rotation gearbox. Most of these are applicable to single-rotation. The lubrication system schematic, shown in Figure 4.3.3-14, identifies features described in this section.

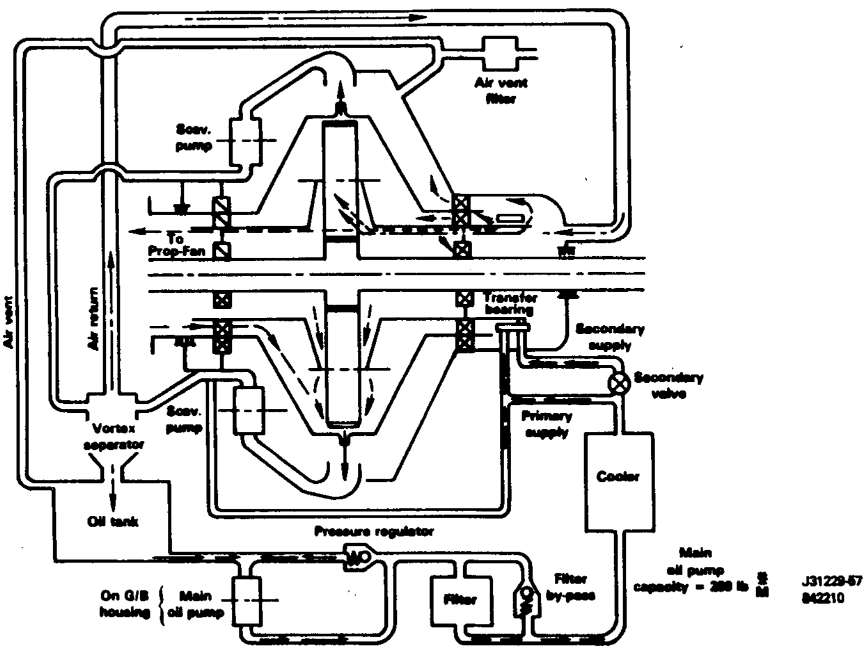

Figure 4.3.3-14 Lubrication System Schematic - This system has additional features which have been identified for use in the counter-rotation arrangement.

A high pressure positive displacement gear pump supplies oil to the gearbox in conjunction with a pressure regulating valve which holds a pre-set discharge pressure. The pressure is set high enough to assure adequate oil jet penetration to gear tooth flanks in the extended addendum high contact ratio gear mesh planned for this gearbox. The regulating valve bypasses some flow back to the pump inlet, compensating for metering jet and leakage flow area variables in the supply system.

High pressure oil is cooled and filtered before delivery to the gearbox. The cooler transfers oil heat to ambient air. The oil filter is an ultra-fine disposable element with a three micron rating. The filter is protected by a warning device which signals excessive pressure drop in advance of filter bypass.

Oil flow is divided into primary and secondary streams. The secondary stream is filtered with a two position shut-off valve. The valve will be designed to fail open and will be closed only when gearbox power transmission is at the cruise level or below as outlined in the Section 4.1.5.3.2 which describes the modulated oil supply concept.

The main oil flow paths to bearings and gears are fitted with "last-chance" screens which will intercept any foreign particles too large to pass through the smallest of the metering jets in the lubrication system. These screens are accessible for inspection and cleaning from outside of the fully assembled gearbox.

A major fraction of the primary oil supply and all of the secondary oil supply will be transferred from stationary flow lines in the housing to rotating passages in the planet gear carrier through an oil transfer bearing. The fixed member of the bearing receives oil through jumper tubes from the housing. It is fitted closely around the rotating member on the carrier shaft and is free to follow small displacement of that shaft. The close clearance between bearing stator and rotor limits oil leakage to small amounts (less than 10 percent of flow).

The rotating flow passages in the planet carrier deliver oil to the multiple planetary gear meshes, to the planet bearings, and to input and carrier shaft bearings. The gear mesh supply comes through an array of small jets which distribute oil across the face of each gear. Two sets of jets are necessary, one each for primary and secondary supply systems.

After passing through various gear and bearing lubrication sites, oil is thrown outward to the inside surface of the ring gear rotor. Oil is discharged from the ring gear rotor through rows of holes at its outer radius and is collected by a scroll to scavenge in the gearbox housing. The scavenge collector carries a mixture of air and oil from the interior of the gearbox housing to oversized scavenge pumps. The pumps are made large enough to carry a quantity of air sufficient to promote the scavenge process at critical locations in the gearbox as explained in the aerodynamic scavenge discussion of Section 4.1.5.3.2.

The air/oil mixture discharged from the scavenge pumps enters a vortex separator where air is discharged from one outlet and oil from another. Air is returned to the gearbox and oil to the tank. Air enters the gearbox at one end of the planet carrier shaft to help reduce oil churning around the sun gear.

For condition monitoring, the vortex separator contains a magnetic chip detector at the oil discharge line. The line is oriented so that dense solid particles are centrifuged toward the detector. This enhances the effectiveness of the detector.

The oil tank is located close to the gearbox and oriented so that pressure loss to the inlet of the main oil pump is minimal. The hot oil tank arrangement outlined above is commonly used in aircraft engine oil systems.

Engine Oil System

The engine oil system proposed for the single-rotation gearbox installation would be duplicated in a counter-rotation environment. The considerations which lead to separating the gearbox oil system from the engine oil system apply to counter-rotation as well. A plan for integrating the engine fuel/oil cooling system with the gearbox air/oil cooler was suggested for the single-rotation installation and is repeated here in Figure 4.3.3-15. This alternative return-to-tank system would offer some flexibility in configuring air/oil coolers and managing heat loads throughout the flight cycle.

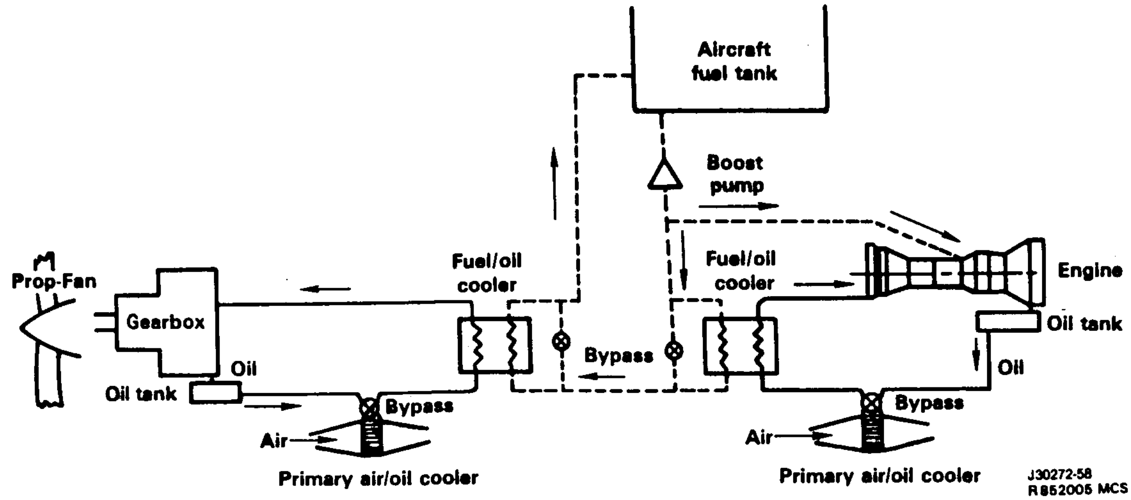

Figure 4.3.3-15 Candidate Oil Cooling System for Advanced Turboprop Engine - The alternative return-to-tank system offers flexibility in configuring air/oil coolers and managing heat loads throughout the flight cycle.

Predicted gearbox efficiency for counter-rotation closely matches single-rotation efficiency. The size of the gearbox oil system including the air/oil cooler closely follows the parameters identified for single-rotation in Section 4.1.5.3.2. Table 4.3.3-8 is a summary of oil system heat load, flow requirements, and tank sizes for both the engine and the gearbox.

Table 4.3.3-8 Advanced Turboprop Engine Oil System Requirements (12,000HP)

12,000 horsepower size engine

	Gearbox and Prop-Fan	Engine and power turbine
Heat to oil, joules/sec (Btu/min)	89,658-105,480* (5100-6000*)	73,836 (4200)
Oil flow, kg/min (lb/min)	68-91* (150-200*)	57 (125)
Oil volume, liters (gallons)	10.6-14.0* (2.8-3.7*)	12.9 (3.4)
Residence time, sec (full tank)	8	12
Tank size (oil + air), liters (gallons)	12.9-17.0* (3.4-4.6*)	14.4 (3.8)

*Higher value includes Prop-Fan

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4.3.3.2.5 Propulsion System Installation

The counter-rotation gearbox adapts easily to both tractor and pusher installations. This section will deal with considerations necessary for incorporating the counter-rotating Prop-Fan reduction gearbox into both tractor and pusher installations. Included are conceptual nacelles, propulsion system mounting, and component (accessories, heat exchanger) provisions. Aircraft manufacturers are currently considering aft fuselage mounted powerplants for 100 to 150 passenger aircraft. This will be the prime consideration of this section.

Tractor Requirements

Applying the counter-rotation Prop-Fan reduction gearbox to a wing-mounted tractor system requires no configurational changes to the mounting system relative to the in-line reduction gearbox. This is because the two gearboxes are very similar in shape and external configuration and have mount provisions incorporated into the gearbox housing. The consideration of aft fuselage mounted powerplants for 100 to 150 passenger aircraft prompted another evaluation of both the mounting arrangement and the inlet section of the tractor.

Locating the powerplant in the aft fuselage region of the aircraft results in a side-mounted arrangement. This is shown schematically in Figure 4.3.3-16 which shows typical three point pick-up mount reactions. The arrangement is similar to a single-rotation Prop-Fan tractor mount. The arrangement anticipates a shock mounted system and shows redundancy in the front mount plane. Considerations of tangential reactions at the front mount plane will result in a more determinate mount system.

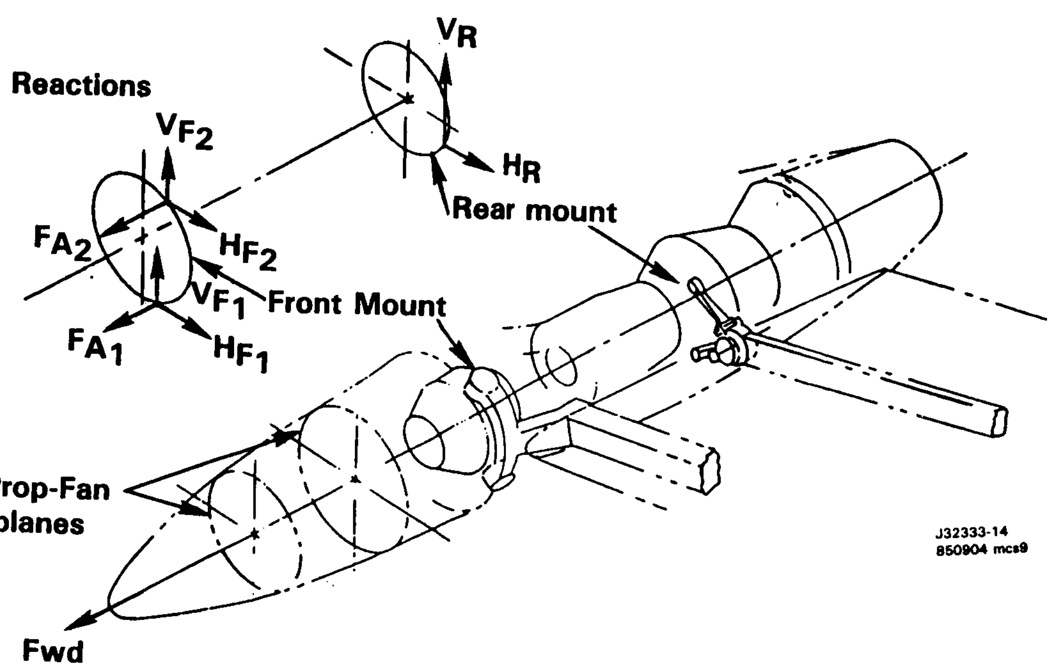

Figure 4.3.3-16 Aft Fuselage Mounted Tractor Installation Mount Schematic - This figure displays the typical three point pick-up mount reactions.

Based on discussions with airframe manufacturers, the inlet configuration was revised from the over/under bifurcated ducting of the single-rotation to one that has both inlets above the engine centerline. This appears in Figure 4.3.3-17. Initial indications are that any placement variation of the bifurcated inlets spaced 120 to 180 degrees apart can be accommodated.

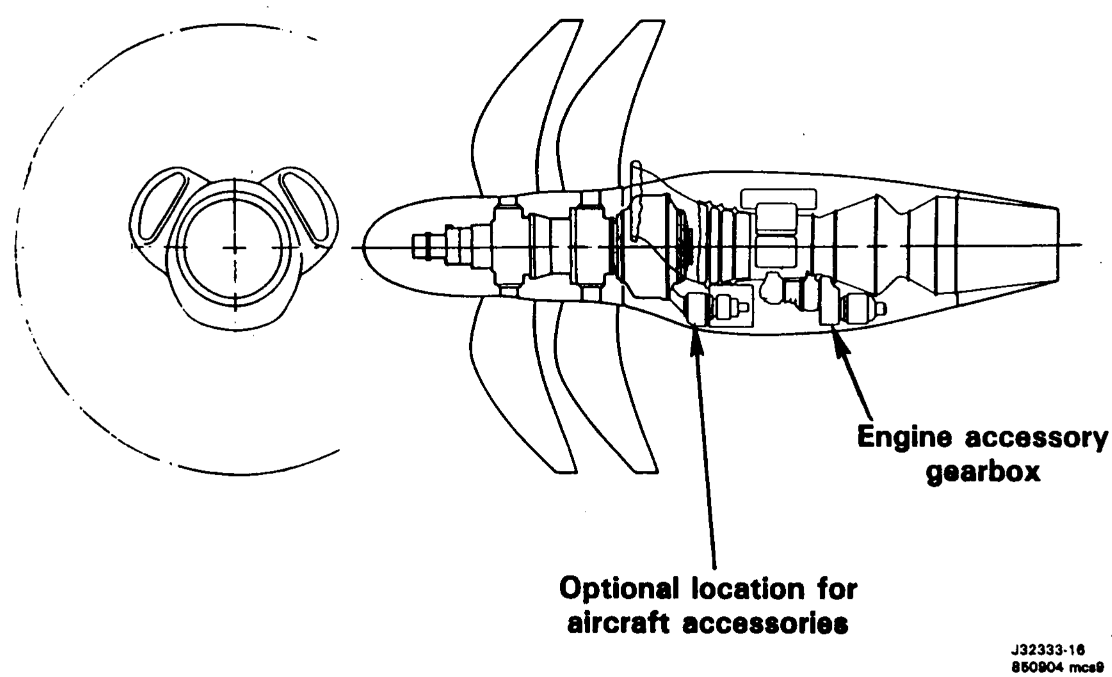

Figure 4.3.3-17 Tractor Installation - In this arrangement, both inlets are above the engine centerline.

Accessory and component locations for the tractor configuration are also shown in Figure 4.3.3-17. Featured is the option of providing power to the airframe accessories either through the high spool or the gearbox. The high spool drives the engine-only accessories and the starter.

Pusher Requirements

The counter-rotation Prop-Fan reduction gearbox is readily adaptable to a pusher installation. As with the tractor configuration, the pusher mount system evolves from a long beam/truss structure joining the gas generator (engine) to the Prop-Fan reduction gearbox to a fairly compact, short, coupled design in which a short conical section attaches the Prop-Fan reduction gearbox to the turbine exhaust case. Mount provisions are no longer incorporated into the gearbox housing. This simplifies the outer case significantly.

Figure 4.3.3-18 shows the initial configuration for the aft fuselage mounted pusher. This features a multiple beam/truss structure joining the Prop-Fan reduction gearbox to the turbine exhaust case of the engine. The beam/truss provided:

- o sufficient axial length to remove a section of the drive coupling,
- o pick-up region for the rear mount, and
- o space to allow exhaust gases to be ducted out to the free stream.

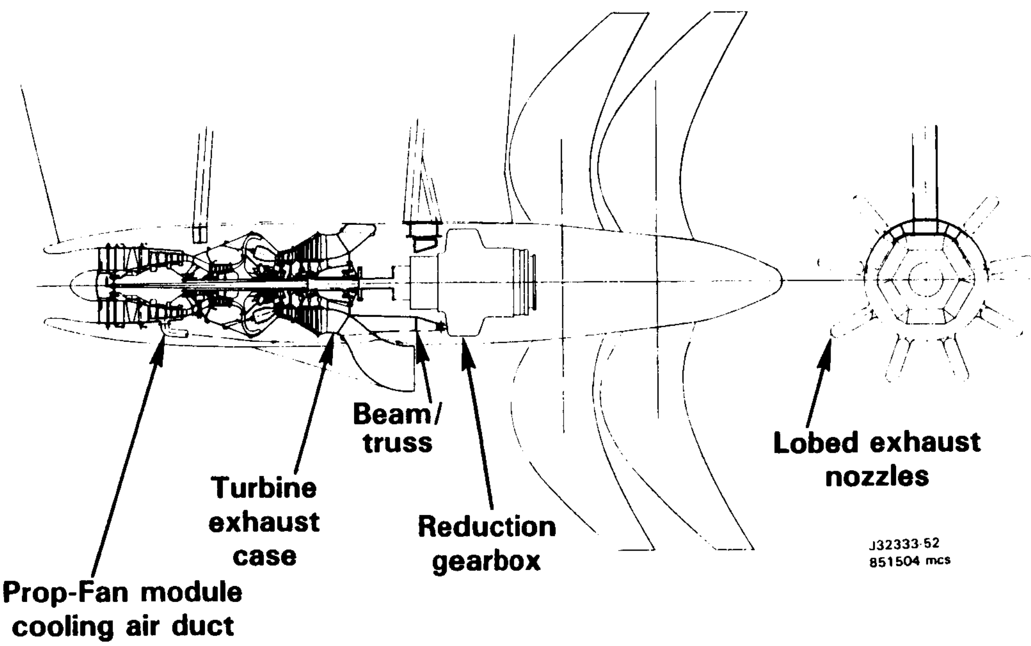

Figure 4.3.3-18 Aft Fuselage Mounted Pusher Installation - This features a multiple beam/truss structure joining the Prop-Fan gearbox to the turbine exhaust case of of the engine.

Final selection of an exhaust nozzle, ranging from narrow multilobe nozzles for low mean temperatures to a full annular nozzle resulting in direct exhaust impingement on the Prop-Fan blading, will be dependent on model testing. Most likely, a multilobed nozzle configuration will evolve, resulting in a mean gas temperature of about 932°C (500°F) at the blade root at hot day takeoff operation.

Discussions with the airframers revealed the desire for a short, compact package in which placement of the inlet of the engine should be as far aft as possible to allow sufficient room for aircraft service vehicles in the rear fuselage area. This consideration led to the short coupled turbine exhaust case to Prop-Fan reduction gearbox design shown in Figure 4.3.3-19.

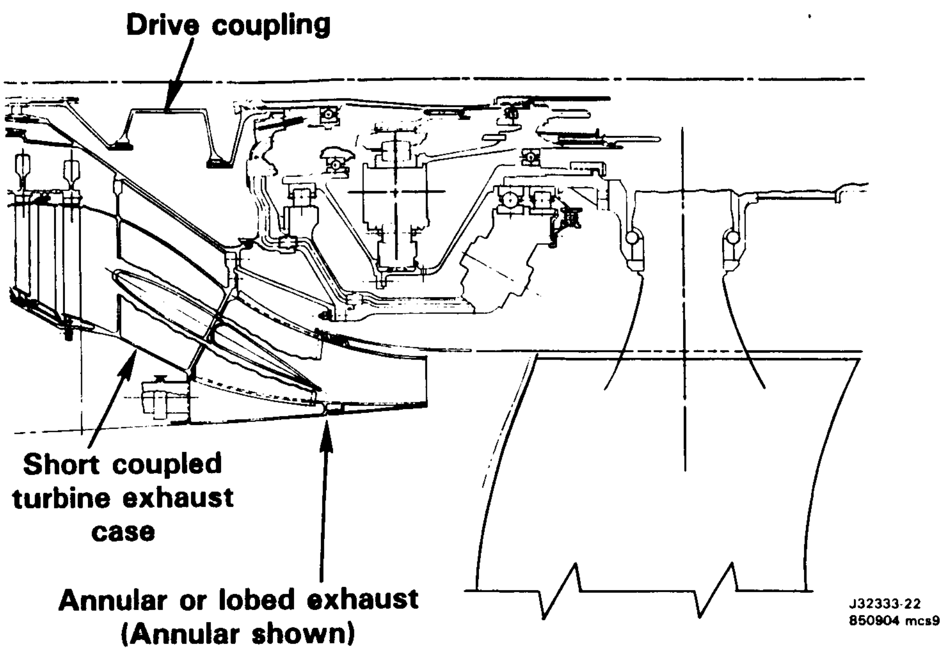

Figure 4.3.3-19 Aft Fuselage Mounted Pusher Installation - This installation is in keeping with the airframer's desire for a short, compact package in which placement of the inlet is as far aft as possible.

The short coupled turbine exhaust case design provides for:

- either full annular or lobed exhaust nozzles,
- ducting of cooling air to the Prop-Fan module and mixing with buffer air available from an enlarged first stage low compressor (see Figures 4.3.3-18 and 4.3.3-19), and
- rear mount plane located on the turbine exhaust case.

The mount system, as shown in Figure 4.3.3-20, takes vertical and side loads at the front mount on the engine inlet case, while the rear mount on the turbine exhaust case takes thrust and moment restraint as well as vertical and side loads. The rear mount plane was chosen to take most of the mount reactions because it is near the propulsion system center of gravity and would have to be strong enough to take most of the reaction loads. The resultant mount system is redundant, so the vibration isolators at the mount points will be designed with suitable stiffness to control the mount load reactions.

Accessory and component locations for the pusher are shown in Figure 4.3.3-21. Like the tractor, options for power sources for airframe accessories are available. This is accomplished by providing two possible power sources to the single accessory gearbox. The high spool power is driven through a conventional towershaft within the engine, while the Prop-Fan power turbine can provide power through a towershaft within the turbine exhaust case. This drives an angle gearbox and jackshaft connected to the accessory gearbox.

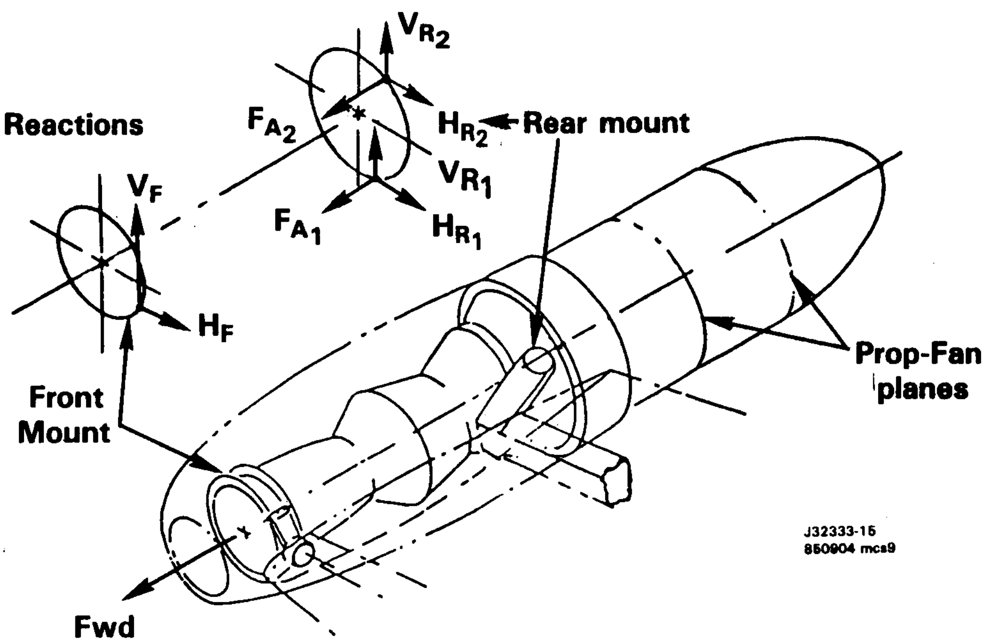

Figure 4.3.3-20 Aft Fuselage Mounted Pusher Installation Mount Schematic - The rear mount plane is located on the turbine exhaust case.

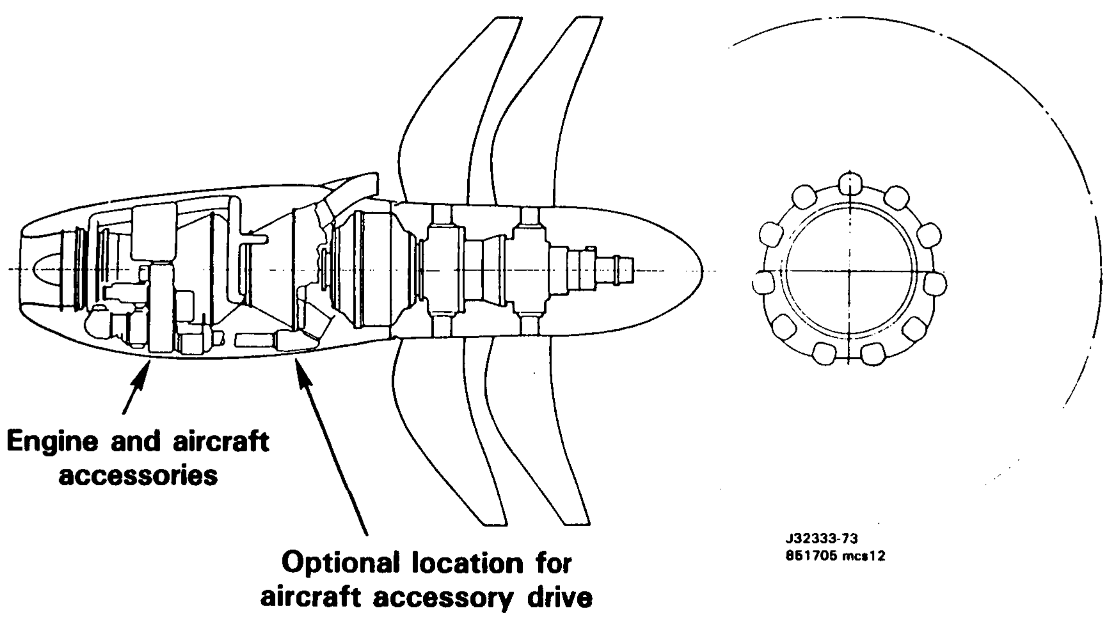

Figure 4.3.3-21 Typical Pusher Prop-Fan Propulsion System - Options for power sources for airframe accessories are available.

Gearbox Air/Oil Cooler Consideration

The propulsion system configuration places the air/oil cooler on the lower portion of the nacelle. This is shown for both tractor and pusher in Figure 4.3.3-22. The features are a fixed inlet and variable exhaust. An ejector provides for low-speed, high-power operation.

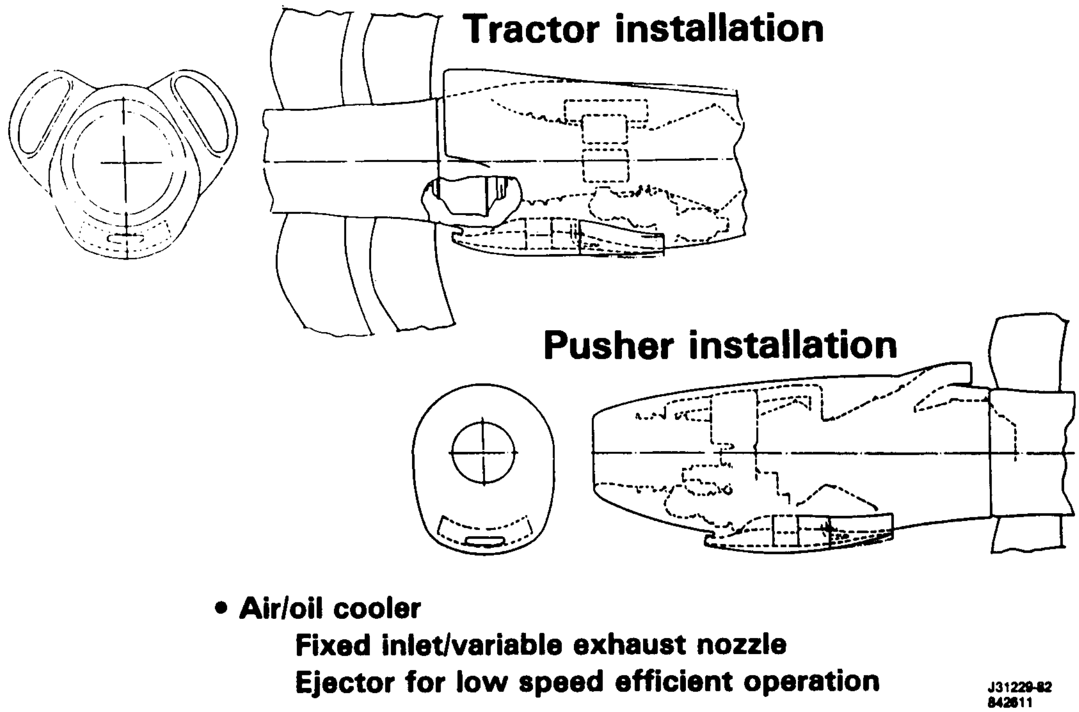

Figure 4.3.3-22 Nacelle Mounted Air/Oil Cooler - The air/oil cooler is on the lower portion of the nacelle.

For a propulsion system in the 12,000 hp category, the oil cooler would weigh approximately 45.4 kg (100 lb), have a frontal area of approximately 1,290 sq cm (200 sq in) and be approximately 12.7 cm (five in) in depth. It's a two-pass type radiator with a fixed inlet and Variable Exhaust nozzle area. (An injector is provided for static and low aircraft speed operation.) The cruise (0.8 mn) drag is estimated to be approximately 40 N (nine lb) or about a 0.2 percent fuel burn penalty.

4.3.3.2.6 Performance and Economic Assessment

Gearbox Efficiency

The lubrication system is the key item in governing the efficiency of large high-speed gearboxes. The meshing losses represent only a portion of the total potential power losses. How the oil flow through the gearbox is controlled is a very important factor in achieving greater efficiencies. If windage and churning are to be minimized, the oil flow within the gearbox must be strictly managed. Oil should be fed into the mesh and removed without the gears churning it. The goal of the modulated lubrication supply and the aerodynamic scavenge systems described in Section 4.3.3.2.4 is to eliminate this churning.

Table 4.3.3-9 is a summary of gear and bearing losses. It also includes estimated oil pump loss and projected windage and churning losses. Note that the predicted gear mesh losses and the bearing losses were analyzed at the maximum torque condition and at a reduced torque (cruise) condition. The major share of the power loss takes place at the sun/planet mesh. This is because the gear mesh sliding velocity at the sun gear is much higher than the gear mesh sliding velocity at the ring gear. The sliding velocity is greater at the sun gear, because the line of action is much longer in the sun/planet external mesh. The remaining losses are evenly distributed among the remaining bearings, pumps, and planet/ring gear mesh.

Table 4.3.3-9 Gearbox Power Losses Summary

	Max torque takeoff loss-HP	Cruise loss-HP
Gears		
Sun/pinion	50.3	15.2
Pinion/ring	5.6	1.7
Bearings		
Planetary (5 ea)	21.9	14.7
Prop shaft ball	2.6	1.0
Prop shaft roller (FWD)	1.6	3.4
Prop shaft roller (AFT)	1.3	2.2
Carrier (2 ea)	0.9	0.9
Sun shaft (2 ea)	2.1	2.1
Lube pumps (3 ea)	10.0	9.0
Windage and churning (ESTIM)	5.4	3.3
Total power loss -	101.7 HP	53.5 HP
Efficiency -	99.15%	98.97%

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Reliability

Reliability predictions for mechanical systems such as reduction gearboxes can only be determined for the mature system level. The mature reliability level reflects the basic capability of the design, and it is the level which will prevail throughout most of the useful life of the system under evaluation. This phenomenon does not imply the existence of an exponential failure distribution (constant failure rate) for each part, but rather the stabilizing effect of contributing factors such as:

- Replacement of parts prior to wear out,
- Mix of old and new parts, and
- Inherent randomness of failures operating in the tail areas of the distribution.

The prediction begins with a review of Pratt & Whitney's past gearbox and bearing designs. These include:

- o 25 years and hundreds of millions of hours of accessory gearbox experience in turbojet and turbofan commercial engines; and
- o The background data from the PT-6 turboprop reduction gearbox -- There have been 26,000 PT-6 engines delivered. They average over 10,000,000 flight hours per year, and their MTBUR, which approaches 100,000 hours per unscheduled removal, is the best in the industry.

The new system reliability prediction is based on actual service experience of predecessor designs whose operational environments are most similar to the design under consideration. This experience comes from accessory gearboxes and from Pratt & Whitney Canada's turboprop gearboxes. Once the basic component history (in this case, bearings, gears, seals, etc.) has been selected, adjustment factors based on our best engineering judgment are derived to reflect anticipated differences in parameters such as speed, load, maintenance philosophy, etc. These adjustment factors are then applied to the basic component failure rates to obtain the new failure rate. The parameter of primary concern, the MTBUR, is then calculated by taking the reciprocal of the sum total of the new failure rates.

A comparative type analysis was conducted for the major components using actual engine failure rates reflecting Pratt & Whitney's past experience with main shaft bearings, seals, and accessory drive gears. This was supplemented by system life analysis studies considering design criteria lives. Independent studies prompted further confidence in these estimates. Sikorsky and Pratt & Whitney Canada conducted the studies, which showed estimates predicting failure rates to be within ten percent of Pratt & Whitney's.

The MTBUR predictions for both the single and counter-rotation gearboxes with advanced technology are shown below. There are several key features in the design for achieving good MTBUR lives. These include:

- o simplifying the design to minimize the number of gears and bearings;
- o designing for long life bearings which past experience tells us is a weak link;
- o using higher strength advanced materials; and
- o moving the pitch control from the gearbox to the Prop-Fan rotor system.

Each of these factors has been considered in the single-rotation and counter-rotation gearbox designs.

The collective life of the bearing system is a most important factor in controlling gearbox durability. Studies indicate that the objective of 15,000 hours MTBUR for the gearbox requires a bearing system that operates with a 50,000 hours mean time between failure. The equivalent 90 percent survival B_{10} life objective is 18,000 hours. This system objective is the governing factor in selecting bearing sizes for highly loaded applications in the gearbox.

With bearing B_{10} lives set above 18,000 hours, the resultant main train MTBUR predictions are 26,200 hours for the single-rotation and 31,000 for the counter-rotation gearbox. The difference reflects the reduced numbers of gears and bearings for the counter-rotation design. When one considers the effects of an optional prop brake and the aircraft accessory pad, the final MTBUR values are 23,300 and 27,100 hours respectively.

Weight

The advanced technology design was based on a 12,000 hp gearbox with a scalable range from 5,000 hp to 22,000 hp. Scaling curves, shown in Figure 4.3.3-23, provide a method for estimating both weight and size of gearboxes in the power range. The abscissa represents the torque ratios of gearboxes above and below the base size of 12,000 hp. To determine the new gearbox weight or torque size, select the appropriate torque factor and extend a line vertically to interact the weight curve. Read the relative scaling parameter on the ordinate. Multiply this number by the base gearbox weight to arrive at the new gearbox weight. The same procedure will determine length and diameter of the new gearbox.

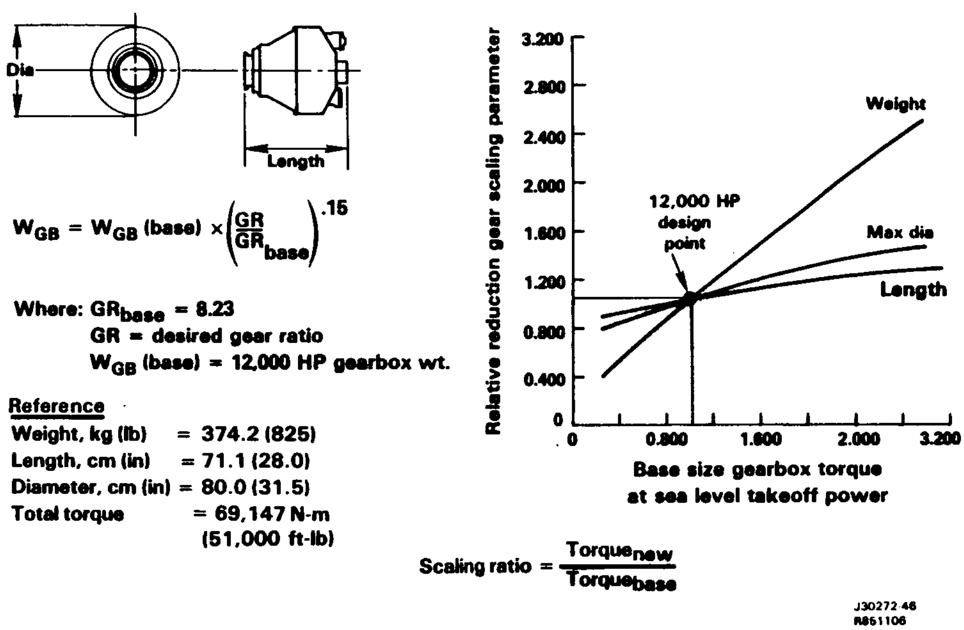

Figure 4.3.3-23 Turboprop Reduction Gear Scaling - The curves provide a method for estimating both weight and size of gearboxes in the power range.

4.3.3.2.7 Counter-Rotation Versus Single-Rotation Gearboxes

The following section reviews the results of the previous discussion and compares the two gearbox systems.

Both systems use in-line planetary designs. The major advantage of the planetary system is its compact size which allows small diameter nacelle contours. While the single-rotation gearbox has a relatively large number of gears (15) and bearings (17), the counter-rotation differential planetary system is very simple with only seven gears and 12 bearings. Since both systems have planetary gear arrangements, the identified advanced technologies are very similar.

Both systems use advanced gear materials with high contact ratio buttress gear tooth forms. These advanced technologies have a significant payoff in both systems by minimizing the size of the planet pinion gears. This in turn reduces the centrifugal load on the planet pinion support bearing and allows the size and weight of the system to be significantly smaller.

Both systems benefit from advanced technology lubrication and scavenging systems. The use of a modulated lubricant supply allows reducing oil flow at cruise which is predicted to improve the efficiency at cruise significantly. In addition, both systems have an aerodynamic scavenging system that uses the air/oil swirl associated with planetary systems to assist scavenging the oil out of the gearbox.

<u>Key Comparisons of Single-Rotation and Counter-Rotation Advanced Technologies</u>

- Both systems have integral gear-spherical roller planetary bearings. The counter-rotating bearing size is in between that of the single-rotation first and second stage planetary bearings. The DN level of 0.51 for the counter-rotating bearing is slightly lower than the 0.58 level for the first stage on the single-rotation. Both systems are designed for comparable lives.

- Both systems have relatively large diameter prop shaft support bearings. The single-rotation bearing bore diameter of 280 mm (11.02 in) is slightly smaller than the counter-rotating diameter of 340 mm (13.39 in). The life of 61,000 hours for the counter-rotation bearing is lower than the 100,000 hours for the single-rotation bearing, but the total system life is comparable, because the counter-rotating system has a smaller number of bearings.

- Both systems exceed the reliability goal by at least 15,000 hours before unscheduled removals. The predicted reliability of the single-rotating gearbox was 23,300 hours. This was achieved despite the relatively large number of gears and bearings, while the simpler counter-rotation gearbox has a predicted reliability of 30,000 hours. (See Table 4.3.3-10.)

  These high levels of reliability result from very high bearing lives and maintainability features such as the remote pitch change module, replaceable shaft oil seals, and easily maintainable oil jets and oil filters. There are also a minimal number of ancillary units driven by the gearbox, units such as oil pumps and aircraft accessories.

Table 4.3.3-10 Gearbox Reliability Predictions

	Single-rotation	Counter-rotation
	Split path planetary	Differential planetary
• Number of gears	15	7
• Number of bearings	17	12
• Bearing system B_{10} life	18,250 hours	18,900 hours
• Main train MTBUR* (gears, bearings, casings, etc)	26,200 hours	31,000 hours
• Main train MTBUR* (including prop brake and accessory pads)	23,300 hours	27,100 hours

MTBUR = Mean time between unscheduled removals

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- Both systems have similar gear designs. Focusing attention on the sun gear, the counter-rotation sun gear diameter is in between that of the first and second stages of the single-rotation system. This gear is designed to similar tooth bending stress levels. The pitch line velocity is higher for the counter-rotating gear. The gears for both the single-rotation and the counter-rotation APET preliminary designs were analyzed consistently to take credit for high contact ratio tooth forms without going through the complex procedure of detailed design of high contact ratio teeth.

Features common to both single-rotation and counter-rotation gearboxes include materials and lubrication systems. The method of transferring oil to the carrier has evolved to a more stable system that is not sensitive to rotor deflections. The single-rotation gearbox used carbon face seals as the transfer agent. Misalignment of the carbon rubbing surface has the potential for high leakage and instability. It is felt that the oil transfer sleeve used on the counter-rotating gearbox is much more stable. Another advantage of this arrangement is that it lends itself to accommodating the modulated lubrication system. The dual supply requirement is accomplished by two separate inlet ports and is channeled through the transfer bearing into separate cavities on the transfer interface with the rotating member.

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SECTION 4.4 -- DISCUSSION OF RESULTS

TASK XII -- THE CONCEPTUAL DESIGN OF A COUNTER-ROTATION PITCH CONTROL

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4.4 Task XII -- Conceptual Design of a Counter-Rotation Pitch Control

4.4.1 Introduction

Under Task XII, a conceptual design study provided an advanced, flight-weight pitch change control and mechanism design which is compatible with the in-line counter-rotation gearbox design of Task XI. Prior to the conceptual design, Hamilton Standard funded and conducted a conceptual trade study to select a concept for further study under the APET contract. The study was described in detail in Section 4.2.3, Trade Studies. The results of the trade studies are applicable to both single and counter-rotation.

This section presents a discussion of the conceptual design of a counter-rotation pitch control and mechanism. The first part of the section gives a current technology overview, and the second describes the APET design effort for the advanced counter-rotation pitch control system.

4.4.2 Current Technology Overview

The current technology available for counter-rotation is primarily the same as that available for single-rotation. The major difference is that the counter-rotation technology must accommodate two rotors. As with single-rotation, counter-rotation blade pitch controls generally incorporate a linear hydro-mechanical actuator with a metering valve and a mechanical pitch lock in the rotating hardware. Mechanical, hydraulic, and electrical inputs must come from the fixed, nacelle-mounted components (i.e., the gearbox).

Rotary mechanical inputs position the metering valve and pitch lock and utilize either differential gearing or a bearing-mounted ball screw to transmit rotary motion across the rotating interface. High pressure oil is transmitted to the metering valve and actuator through a low clearance oil transfer bearing and transfer tubes. Electrical power for ice protection goes to the turboprop through contact brushes running on a rotating slip ring assembly. Figure 4.4.2-1 shows the current pitch control concept.

The sectional drawing of Figure 4.4.2-2 shows current technology for transmitting rotary mechanical and hydraulic pitch control inputs to a counter-rotating turboprop installed on an in-line planetary gearbox. In this configuration, the drive shaft from the engine restricts access to the axis of rotation from the rear of the gearbox. Therefore, the mechanical signal must come from the rear face of the gearbox housing to the turboprop, through differential gearing, around the sun gear shaft and lay shafts, and through the planet cage and additional gears to reach the axis of rotation. Similarly, high pressure pitch change oil must go through a large diameter (high leakage) transfer bearing, around the sun gear shaft and oil transfer tubes, and through the planet cage to the turboprop shaft.

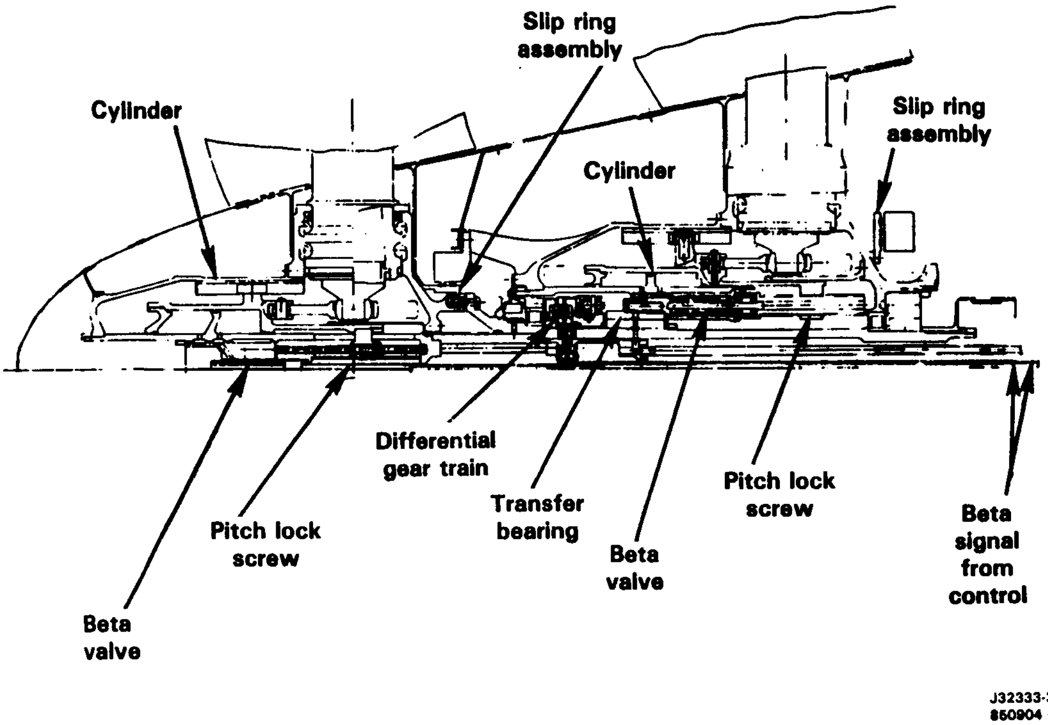

Figure 4.4.2-1 Current Technology Counter-Rotating Turboprop Assembly Pitch Change Mechanism - This technology is primarily the same as the single-rotation current technology.

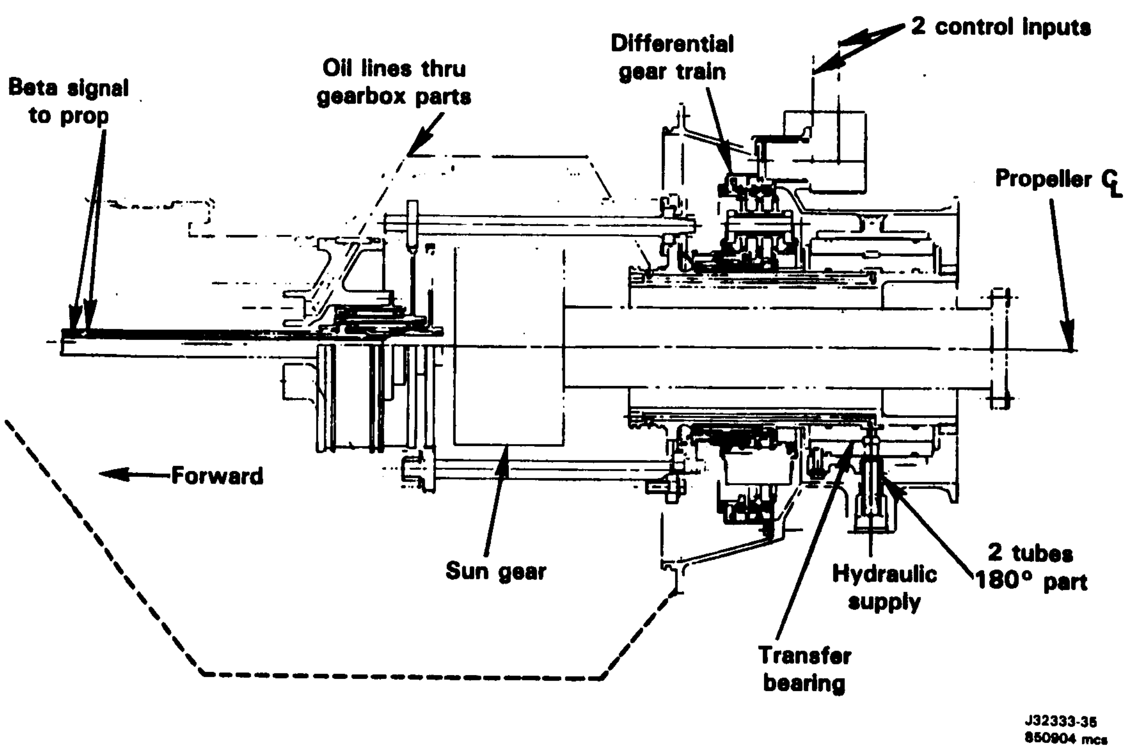

Figure 4.4.2-2 Current Technology Counter-Rotating Pitch Change Control Mechanism - Transmittal of rotary mechanical and hydraulic pitch control inputs are shown.

Integrating non-modular pitch control inputs within the in-line gearbox introduces several complexities. In addition to the complex gearing and large diameter transfer bearing, there is a significant impact on the gearbox design. The overall effects are a reduction in reliability and an increase in maintenance costs. This configuration emphasizes the need to develop advanced pitch control systems that are reliable and easily maintainable.

4.4.3 Conceptual Design of Selected Concepts

The two primary design objectives for the APET pitch control conceptual design were:

(1) to minimize impact on the gearbox, and

(2) to maximize accessibility and maintainability.

Implementing a modular pitch control design helped attain these objectives. In this concept, the pitch control is within the rotating Prop-Fan assembly. This simplifies the interface with the gearbox, improves gearbox reliability and maintenance cost, and reduces pitch change maintenance cost by providing accessible, easily maintainable modules.

This APET conceptual design study refined the rotary and linear pitch control concepts selected in the trade studies for single-rotation (Section 4.2.5). The study also evaluated the configurations to choose the best one for a counter-rotating Prop-Fan in a tractor installation. Evaluation of the rotary hydraulic concept continued, and a new "non-modular" linear hydraulic concept was generated for comparison. The "non-modular" concept incorporates a power module and electronic control mounted in each hub to reduce parts count at the possible expense of maintainability.

A variation of the rotary concept called rotary/linear was also studied. This concept incorporates a linear actuator in the forward hub of the rotor system and retains the rotary concept in the rear of the hub, thus reducing the total number of parts.

4.4.4 Description of Pitch Control Concepts

The counter-rotating Prop-Fan is flange mounted to the gearbox output shaft through curvic face splines at the rear face of the aft hub. This flange reacts all counter-rotation Prop-Fan mounting loads and drives the aft rotor. The planet carrier of the gearbox output shaft drives the forward rotor through a splined quill shaft.

Each blade is retained in the hub with a single-row angular contact ball bearing. An external blade clamp provides additional support for static blade pitch operation. Blade retention bearings are lubricated by a fixed amount of oil in the hub. A lip seal at the blade root prevents external leakage. Figure 4.4.4-1 shows a sectional assembly drawing of the counter-rotation Prop-Fan with rotary hydraulic pitch control (Concept 1). Blade trunnion arms, splined to the inboard end of the blades, rotate the blades about the pitch axes. Links with spherical rod-end bearings connect the trunnion arms to a ball screw nut assembly in each rotor which translates to change blade pitch. Each ball screw is straddle-mounted on hub mounted support bearings. Link forces impose torques on each ball nut which are reacted by an integral lug riding in a slot in the forward and aft hub-mounted housings.

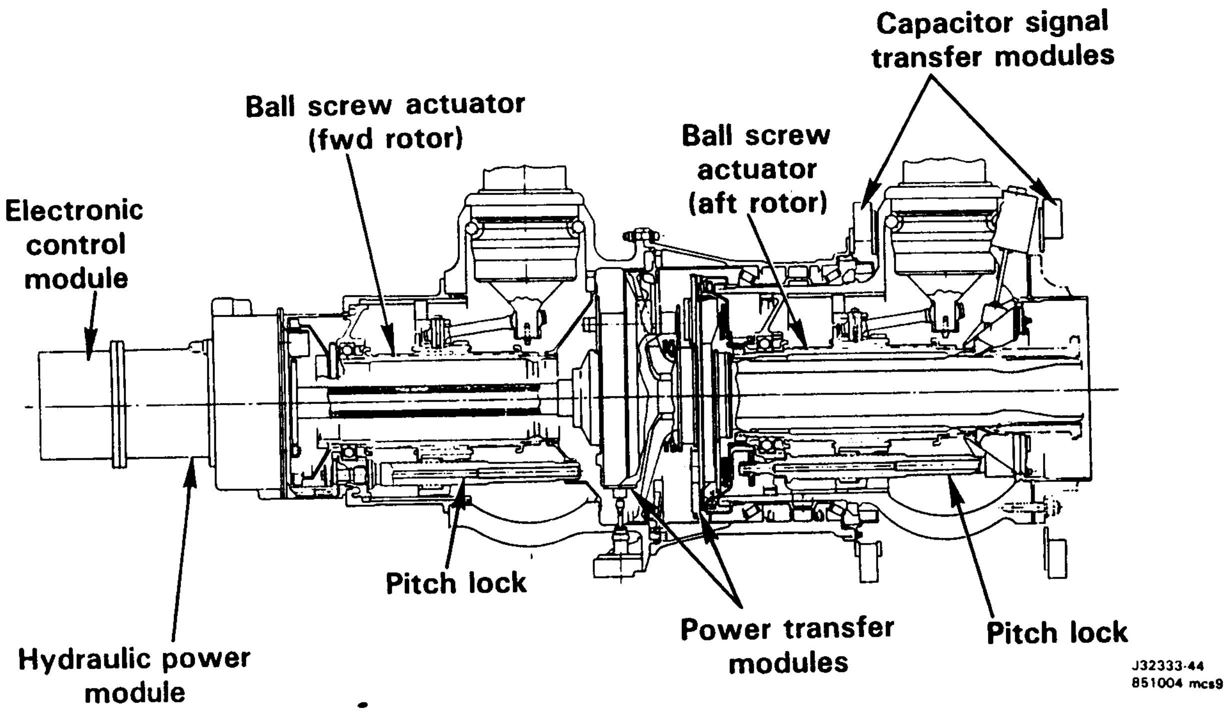

Figure 4.4.4-1 Rotary Hydraulic Pitch Control (Concept 1) - Major components of the system are shown.

A hydraulic power module drives the ball screws. This module consists of drive gearing, hydraulic motors, four-way metering valves (beta control), mechanical in-place pitchlocks, pumps, oil sumps, pressure regulating and relief valves, and a generator. A bolted flange is used to mount the power module on the counter-rotation Prop-Fan forward hub-mounted housing. In response to pressurized oil applied to the high or low pitch side of hydraulic drive motors, drive gearing rotates the ball screws which increase or decrease the blade pitch. An irreversible acme screw and nut acts as a pitchlock in each rotor. The pitchlock nuts are integral with the ball nuts. A small axial gap, between the end of the pitchlock screw and the hub-mounted actuator bulkhead during operation, prevents the blade pitch from decreasing by more than one degree toward low pitch if hydraulic power is inadvertently lost anywhere in the blade operating range.

A small bi-directional D.C. servo motor drives the pitch lock screw to control pitch upon command from the electronic control module. Each rotational position represents a discrete blade angle setting in the operating range. An RVDT, geared to the servo motor, measures these positions which are fed back to both the electronic control module and the nacelle-mounted EEC.

4.4.4.1 Hydraulic System

Figure 4.4.4-2 is a diagram showing the functional relationship between the actuator, pitch lock, and the hydraulic components for each rotor. The design of the hydraulic system conserves power and reduces heat generation. Over ninety-five percent of counter-rotation Prop-Fan pitch control operating time is at power levels less than twenty percent of peak power. This is because commercial aircraft require peak pitch rate power only for large blade angle excursions (i.e., reversing and feathering).

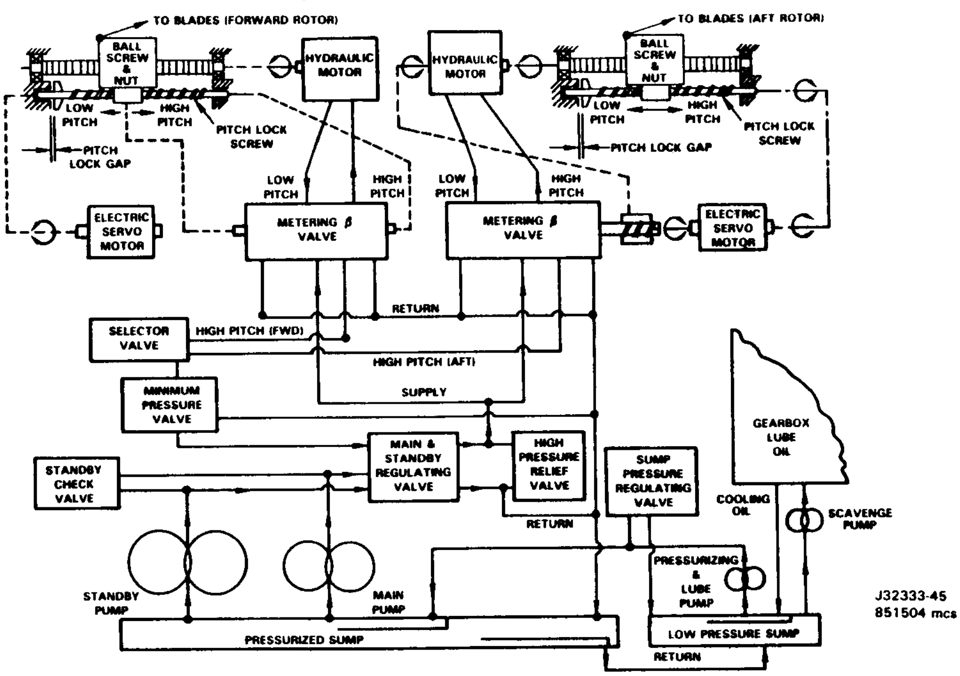

Figure 4.4.4-2 Rotary Hydraulic System Diagram (Concept 1) - This shows the functional relationship between the actuator pitch lock and the hydraulic components for each rotor.

A small displacement main gear pump supplies high pressure oil to each actuator motor via a beta metering valve for all low power pitch control requirements. Although the pump can provide the peak system pressure set by the high pressure relief valve, the pump supply (discharge) pressure is regulated to a few hundred psi above actuator operating pressure requirements. The main and standby regulating valve accomplishes this by regulating main pump supply pressure to the metering valve at a level slightly above the higher of the two high pitch pressures as indicated by the shuttle selector valve. This pressure regulation, coupled with the small pump size, reduces pitch control power generation to the low levels necessary for most of the flight spectrum with minimum heat generation. A standby gear pump with approximately four times the capacity of the main pump circulates oil back to the pressurized sump at low pressure (low power) most of the time. When the beta metering valves are positioned for high flow (pitch rate), the regulating valve and standby check valve combine both the standby pump flow and the main pump flow, at high pressure, to provide the required high power. This is a transient condition and heat generation is minimal.

A pitch control system pressure versus weight trade study showed that 41.4 MPa (6,000 psi) is the optimal pressure for minimum weight. However, 32.8 MPa (4,750 psi) was selected, because it results in higher reliability and lower cost for a weight penalty less than two percent of pitch control weight. The lubrication pump, located between the rotors, charges the pressurized sump to 0.52 MPa (75 psi) minimum. This pressure ensures that the main and standby high speed pumps are adequately supplied with oil to prevent cavitation. The lubrication pump also circulates cooling oil from the gearbox lubrication system through the pressurized sump to mix directly with pitch control oil and return filtered to the gearbox cooler. The differential rotation of the two rotors drives the power module pumps and forward generator through a differential planetary gear system and concentric, geared transfer shafts.

The generator is a light-weight, samarium-cobalt, permanent magnet, externally-commutated A.C. type. The electronic control module rectifies the A.C. output to D.C. Dual generator windings provide separate voltage supplies for pitch control and blade deicing. An overrunning clutch, at the generator drive shaft, permits the generator to be powered as a motor for static ground operation of the pitch control. Auxiliary ground cart power, supplied to the generator with the engine inoperative, drives the pumps to develop pressurized oil for pitch change. A separate generator for deicing of the aft rotor blades is mounted on the aft hub near the flange mounting face. The relative rotation of the two rotors drive this generator through a bevel gear set.

4.4.4.2 Electronic Control System

The electronic control module incorporates the printed circuit boards and solid state components necessary for:

(a) providing control of the D.C. servo motor for each rotor under pitch control command from the nacelle-mounted, full-authority digital EEC and from separate overspeed pitch control circuitry in the module,

(b) transmitting blade angle feedback and other diagnostic signals from each rotor to the EEC, and

(c) providing power switching for blade deicing.

Rotary capacitor signal transfer modules, located at the rear end of each hub, transmit serial digital pitch control signals bi-directionally between the EEC and the rotating electronic control module. Each transfer module contains two electrical paths. Each path consists of two parallel annular metal disks, one on each side of the rotating interface, separated by an air gap.

Under normal operating conditions, the electronic control module provides only blade pitch control on command from the EEC. All intelligence for governing rpm, Syncrophasing, feathering, reversing, and ground handling is in the dual-channel EEC. This permits the more complex electronic control circuitry to be in the stationary nacelle where it is more accessible for maintenance and for modification of control parameters. In the event of either an erroneous signal or loss of signal from the EEC, the electronic control module has a solid-state speed governor, with separate power supply circuitry and speed sensor, that will govern rpm at a set percentage of normal rpm. The flight may then continue with only the loss of Syncrophasing and reversing capability. Provision is made to conduct a pre-flight check of this back-up control circuit.

Blade pitch angle change originates with a requirement and a command signal from the EEC to change pitch a discrete amount toward either high or low pitch. The signal moves across the capacitor signal transfer modules to the electronic control module which powers two D.C. servo motors to rotate the pitch lock screws through gearing and to translate the metering valve spools. Rotating a valve mounted screw translates the aft rotor valve, while a linkage system, coupled to the forward pitch lock screw, translates the forward rotor valve. Concentric geared tubes and a differential provide the drive coupling to the aft rotor ball screw and pitch lock screw. Pressurized oil is then metered to the system hydraulic motors which drive the ball screw nut assemblies to the commanded blade angle position and null the metering valves. The in-place pitch lock gap between the screw and ground toward low pitch is continuously maintained within one degree of blade angle (i.e., full metering valve authority is sustained within the pitch lock gap). RVDT's continuously measure blade angle position in each rotor and feed the positions back to the control to terminate the signal when the commanded angles are reached.

4.4.4.3 Maintainability Features

The modular component design of the rotary hydraulic pitch control system satisfies the primary design objectives of minimum impact on the gearbox and maximum accessibility and maintainability for any gearbox configuration. After removal of the counter-rotation Prop-Fan spinner, the electronic control module can be easily taken out by removing bolts from the mounting flange and then pulling the module forward on guide pins to release the plug-in wiring connectors. Removal of the electro-mechanical module mounting bolts permits the D.C. servo motors, RVDT's, and associated reduction gearing to be removed as a unit. The hydraulic power module can be taken out by removing mounting flange bolts. The forward and aft generators, lubrication pump, and scavenge pump can each be removed and replaced without disturbing other components.

Access is gained to the blade links, ball screw, and pitch lock of the forward rotor for inspection, maintenance, or replacement by removing the cylindrical support housing bolts from the hub at the mounting flange and sliding the housing forward. Blades in the forward rotor can also be removed and replaced, if necessary, as follows:

(a) disconnect the blade link at the trunnion arm,

(b) disengage the deicing brush assembly from the blade slip rings,

(c) remove the external split clamp and lip seal from the hub,

(d) move the blade into the hub a small distance and remove the retention bearing balls which are self-contained in a flexible plastic retainer, and

(e) remove the blade from the hub.

The capacitor signal transfer modules are in segments that are easily removed for replacement or repair. Remaining counter-rotation Prop-Fan components incorporate modular design to facilitate shop maintenance. Replacement of the aft rotor blades requires removal of the aft alternator drive gearing to gain access to the blade links. This requires removal of the counter-rotation Prop-Fan assembly from the aircraft. Hamilton Standard is pursuing a simpler method of aft rotor blade replacement in separate design studies.

4.4.4.4 Alternate Concepts

Figure 4.4.4-3 shows a sectional assembly of the counter-rotation Prop-Fan concept with rotary/linear pitch control (Concept 2). Figure 4.4.4-4 is a diagram showing the functional relationship between the actuator, pitchlock, and hydraulic components for each rotor. This concept is essentially the same as Concept 1 except that a linear hydraulic actuator replaces the ball screw actuator in the forward rotor. The actuator piston is stationary and is straddle-mounted on support rings attached to the hub. The blade links are attached to the actuator cylinder which translates to change blade pitch. Torque restraint of the cylinder is accomplished by an integral cylindrical bushing which rides on a shaft fixed to the piston support rings. The pitchlock nut is integral with the actuator cylinder. The screw is supported on bearings in the piston support rings. Blade pitch angle control signals power the D.C. servo motor to drive the pitchlock screw through gearing. Linkage attached to the pitchlock screw causes translation of the metering valve spool. High or low pitch pressure is directed to the appropriate side of the piston. As the actuator cylinder translates, the pitchlock nut returns the metering valve to null position through the pitchlock screw and linkage.

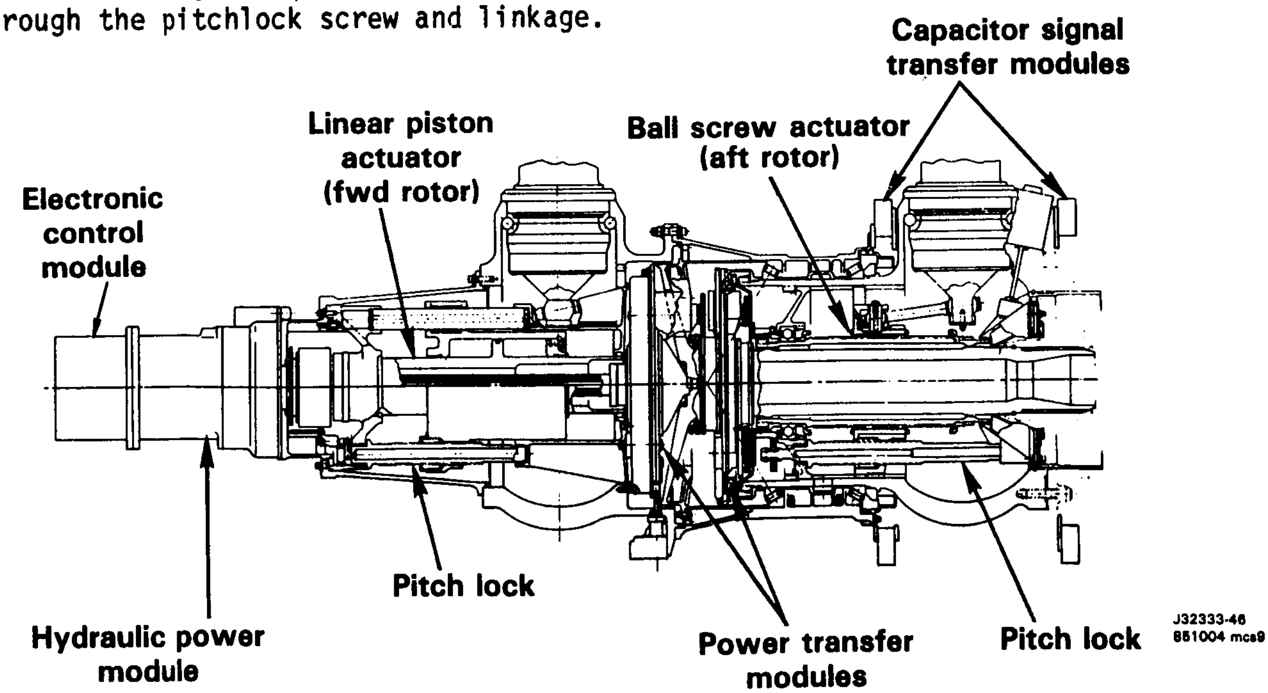

Figure 4.4.4-3 Rotary/Linear Hydraulic Pitch Control (Concept 1) - Major components of the system are shown.

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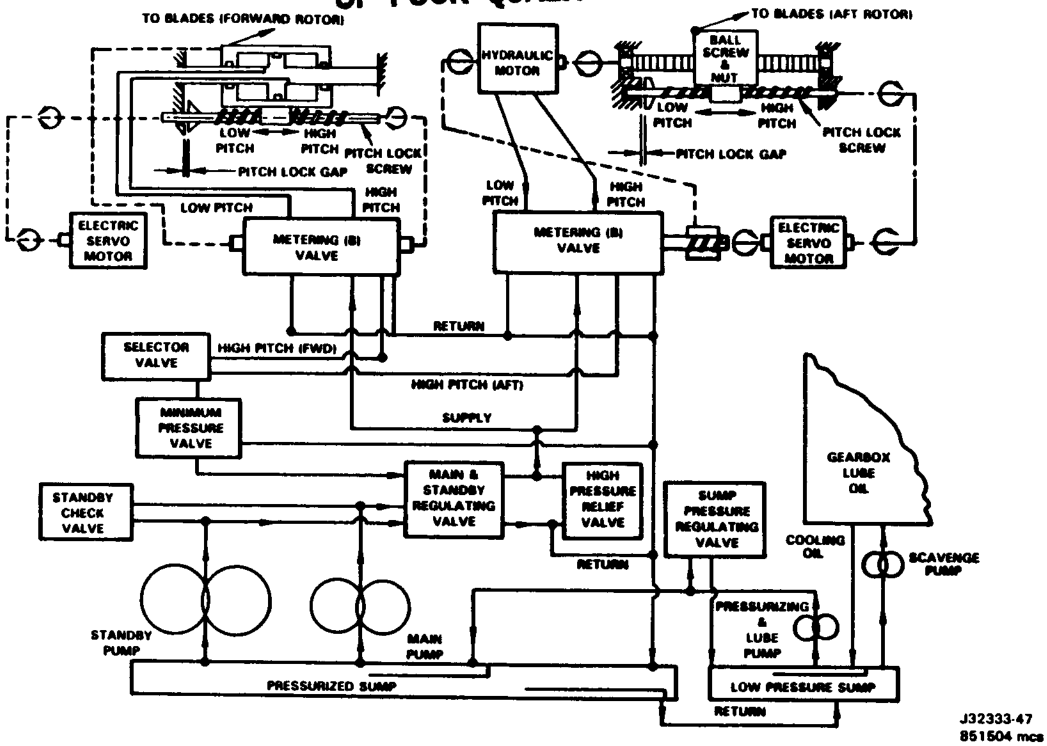

Figure 4.4.4-4 Rotary/Linear Hydraulic System Diagram (Concept 2) - This shows the functional relationship between the actuator, pitch lock, and hydraulic components for each rotor.

A sectional assembly of the counter-rotation Prop-Fan concept with linear pitch control (Concept 3) is shown in Figure 4.4.4-5. Figure 4.4.4-6 is a diagram showing the functional relationship between the actuator, pitchlock, and hydraulic components for each rotor. This concept incorporates two linear hydraulic actuators, as described for Concept 2, to change blade pitch of both rotors. Both pistons are stationary and are straddle-mounted on support rings attached to the hubs.

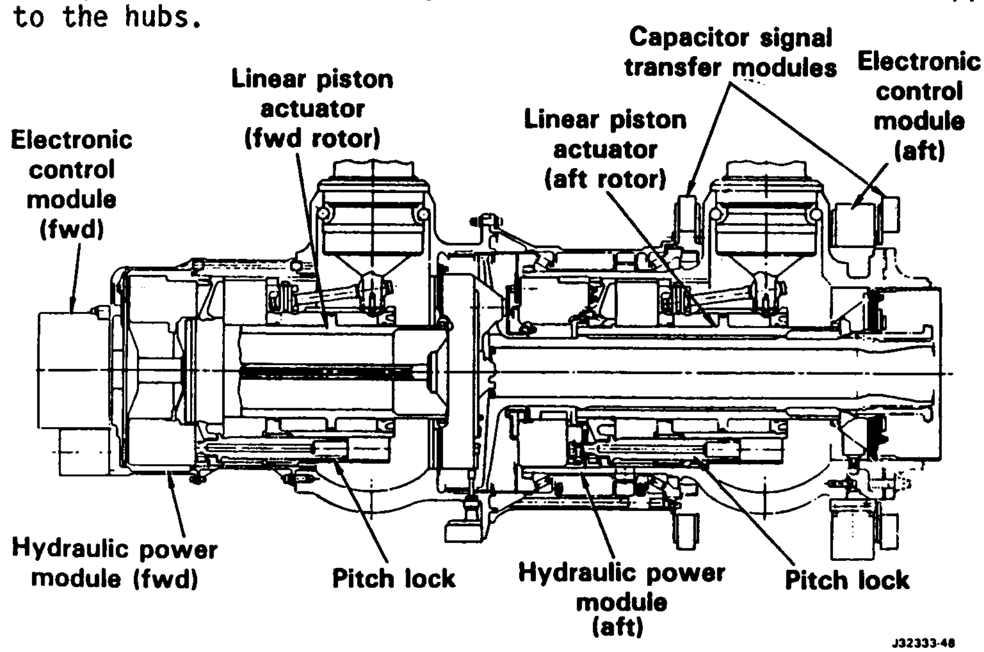

Figure 4.4.4-5 Linear Hydraulic Pitch Control (Concept 3) - Major components of the system are shown.

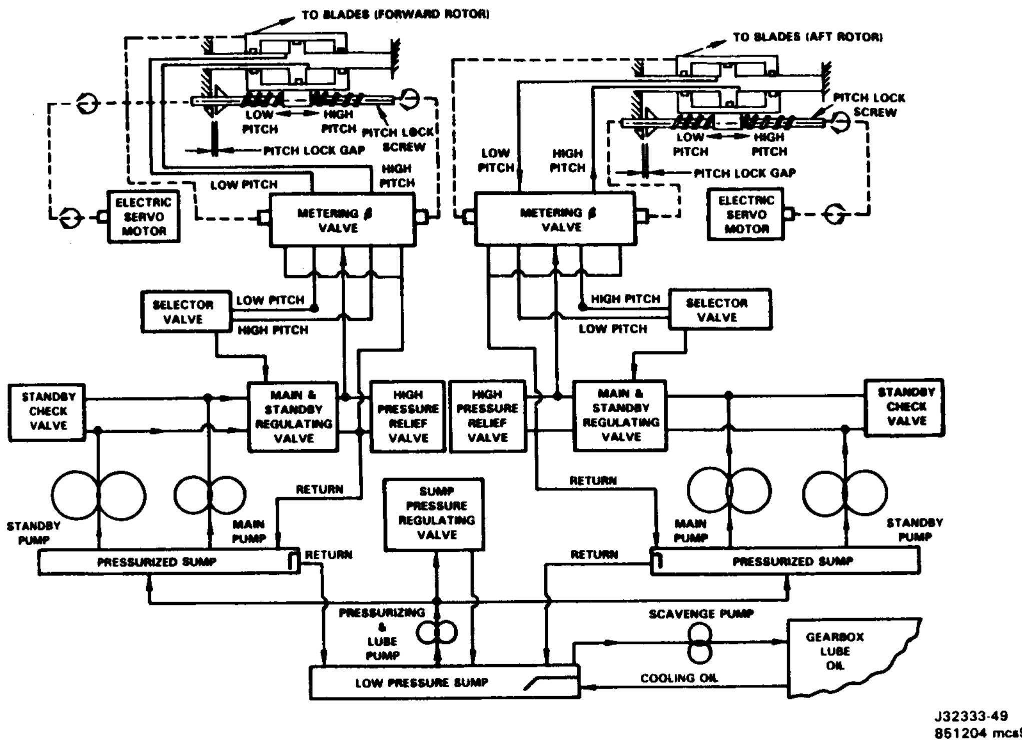

Figure 4.4.4-6 Linear Hydraulic System Diagram (Concept 3) - This shows the functional relationship between the actuator, pitch lock, and hydraulic components for each rotor.

Blade links in each rotor are attached to the actuator cylinder and change blade angle as the cylinder translates. Torque restraint of the actuator cylinders is accomplished by integral lugs riding in slots in the forward and aft housings. The pitchlock nuts for the pitchlock screws are integral with the actuator cylinders, and the screws are bearing-supported on the piston support rings. Blade pitch angle control signals power the D.C. servo motors through gearing to drive the pitchlock screws. Linkage attached to the pitchlock screws causes translation of the metering valve spool, sending high or low-pitch pressure to the appropriate side of the piston. As the actuator cylinder translates, the pitchlock nut returns the metering valve to null position through the pitchlock screw and linkage. Each rotor actuator is powered by an individual hydraulic power module incorporating a D.C. servo motor, drive gearing, a 4-way metering valve (beta control), main and standby pumps, a pressurized sump, pressure regulating and relief valves, and a generator. The power module for the forward rotor is bolted to the forward hub-mounted hous- ing, and the aft rotor power module is bolted to the aft hub-mounted housing. Individual electronic control modules provide pitch control for each rotor. The aft module is mounted in an annular segment near the counter-rotation Prop-Fan assembly mounting flange, and the forward module is mounted at the forward end on the axis of rotation. Servicing the forward rotor actuator and removing blades is essentially the same as described for Concept 1. Servicing the aft rotor actuator and the hydraulic power module and removing blades require the removal of the counter-rotation Prop-Fan assembly from the aircraft.

The three concepts were rated to the same evaluation parameters used in the trade studies and the results are below in order of rank.

1. Linear

2. Rotary

3. Rotary/Linear

Rating scores for the three concepts were close, but the linear concept rated slightly higher than the rotary concepts based on cost and technical risk. The rotary and rotary/linear concepts rated essentially the same, but the rotary is favored because of the slightly lower cost due to commonality of the fore and aft actuators.

The rotary concept was selected instead of the linear concept, because of the following additional considerations:

- Location of the electronic control and hydraulic power module on the forward end of the Prop-Fan facilitates providing power for static check-out and mounting instrumentation for diagnostics.

- The ball screw actuator is more adaptable to using Prop-Fan rotational energy to feather the blades mechanically if normal power is lost.

- Lower system pressure can be used with the rotary concept with minimal weight penalty, because it is significantly less weight sensitive to pressure level than the linear piston concept.

As shown in Figure 4.4.4-7, technology development for the three components is necessary prior to their inclusion in an advanced Prop-Fan application. An efficient shielding system against electro-magnetic interference (EMI) must be developed for the rotary capacitor signal transfer module. Electronic control components must be mounted and packaged in the module to withstand the G-field environment of the rotating Prop-Fan (approximately 40 G's per in radius from the axis of rotation). Hydraulic gear pumps and gear motors must be developed for the high speed, high pressure application of the power module.

A research and technology plan has been prepared and is included later in this report. The plan defines the programs necessary for technology development.

Figure 4.4.4-8 shows the modular pitch control design features which are expected to reduce line and shop maintenance costs substantially. All electrical, electronic, and hydraulic components can be replaced in modules on the aircraft. The remaining modular components facilitate shop maintenance actions.

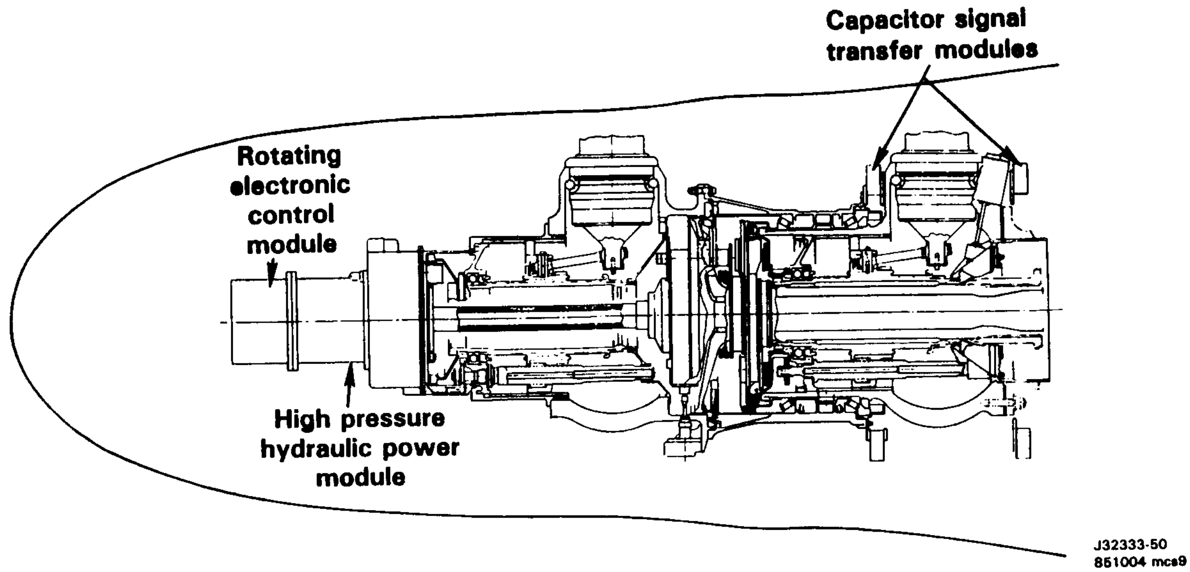

Figure 4.4.4-7 Advanced Technology Features Requiring Additional Development - The development of these features is necessary before their use in an advanced Prop-Fan.

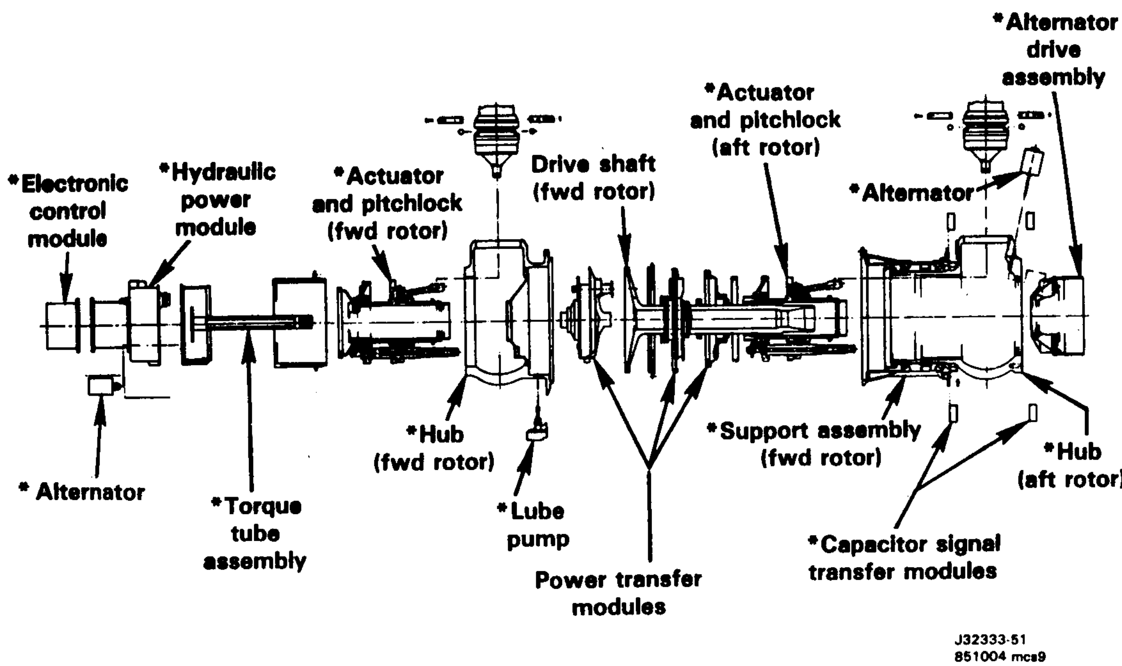

Figure 4.4.4-8 Modular Design Concept of Prop-Fan and Pitch Change Mechanism Assembly - These features will reduce line and shop maintenance costs.

4.4.4.5 Pitch Control Parameters

The primary Prop-Fan design parameters guiding the conceptual design of the advanced technology pitch control were blade pitch slew rates, blade angles, and blade twisting moments. A discussion of each follows.

4.4.4.5.1 Slew Rates

Table 4.4.4-1 shows blade pitch slew rate requirements for various Prop-Fan operating conditions. Normal slew rate requirements for most of the flight spectrum are low. Blade pitch angle is held essentially constant at each flight condition with small excursions of less than ±0.1 degree during Syncrophasing. Syncrophasing is a fine-tuning control of blade pitch through very small angles that do not require high slew rates.

Table 4.4.4-1 Slew Rates

Condition	Blade Pitch Rate (deg/sec)
Normal control	0-3
Syncrophasing	0-1
Feathering	15
Reversing	15
Ground Operation (engine inoperative)	0-3

The aircraft requirements normally set the maximum slew rate based on the time necessary to reach the full reverse angle on landing. The rates shown are based on the capability to reverse fully from flight idle in three seconds. These rates are judged to be satisfactory for advanced turboprop propulsion systems. However, different rate requirements can be easily satisfied with minor changes to the pitch control.

4.4.4.5.2 Blade Pitch Angle Settings

Table 4.4.4-2 gives the blade angle settings for each rotor for various operating conditions. Angles are specified at the blade 3/4 radius. Pratt & Whitney indicates that the engine can start with the blades at any angle including feather. Therefore the minimum Prop-Fan torque blade angle is somewhat academic for this propulsion system. Emergency blade angles are set by the mechanical in-place pitch lock which follows approximately one degree below any commanded blade angle.

Table 4.4.4-2 Blade Angle Settings

Condition	$\beta_{3/4}$ (degrees) Fwd Rotor	Aft Rotor
Feather	+85	+85
Flight Idle	+41	+39
Cruise (0.8 Mn)	+58	+55
Max. reverse	-15	-15
Min. Prop-Fan Torque (static conditions)	-1	-1
Emergencies	< 1 below beta setting	

4.4.4.5.3 Blade Twisting Moment

The pitch control system must be capable of rotating the blades about the pitch axis, counteracting the total blade twisting moment. The following individual twisting moments comprise the total moment:

(a) centrifugal, acting toward flat pitch,

(b) aerodynamic, acting toward either high or low pitch, depending on the flight condition, and

(c) friction, acting to impede motion toward either high or low pitch.

Centrifugal twisting moment results from centrifugal forces on the blade mass as a function of distance from the pitch axis and makes up most of the total moment. Highly swept Prop-Fan blades have significantly higher twisting moments than more conventional blades with less sweep because of the increase in overhang from the pitch axis.

The maximum total blade twisting moment that the pitch control must overcome to move twelve blades toward high pitch is 37,285 Nm (330,000 in-lbs). The maximum total twisting moment necessary to hold the blades in position is slightly less than this value due to exclusion of the friction moment. It is this reduced moment that the pitch control or the pitch lock must react to hold the blades at a fixed blade angle setting.

4.4.4.6 Weight

Results of a weight analysis conducted on the advanced pitch control conceptual design showed the weight to be the same as the weight of the baseline pitch control concept, shown in Figures 4.4.2-1 and 4.4.2-2, which originated in NASA report CR 168258, "Technology and Benefits of Aircraft Counter-Rotation Propellers," December 1982. Total Prop-Fan module weight equals the weight of the Prop-Fan and the weight of the pitch control and is the same as the baseline weight provided in Figure 79 of the reference report. This represents an improvement, because the advanced concept has the additional advantage of modularity for improved maintainability with no increase in weight.

4.4.4.7 Counter-Rotation Prop-Fan Pusher Installation

The advanced pitch control concept is easily adapted to either the pusher or tractor Prop-Fan installations. Position of the ball screw actuators changes from forward of the blades for a tractor to aft of the blades for a pusher. This changes the load direction in the blade links and requires the pitch lock gap to be relocated from one end of the lock screw to the other end. The pusher installation also requires engine-supplied cooling air inside the Prop-Fan spinner to maintain the thermal environment for the electronic control module within acceptable limits.

4.4.4.8 Reliability

A component failure rate and unscheduled removal rate analysis was performed in a manner similar to that for the single-rotation Prop-Fan discussed previously.

For the advanced technology Prop-Fan System, the MTBUR of 2,600 hours is derived in Table 4.4.4-3. It is based on Prop-Fan assembly removals, as well as removals of replaceable components such as the electronic control, hydraulic power module, electric motor module, and spinner.

This MTBUR represents an improvement of 73 percent over the 1,500 hour MTBUR for the baseline Prop-Fan system defined in NASA report CR 168258 (Figures 76, 77, and 81 on pages 265, 266, and 270, respectively).

The predicted MTBUR (chargeable events) of 13,800 hours for the advanced technology Prop-Fan system is based on only those failures that require removal of the entire Prop-Fan assembly. This represents a significant improvement over the 4,900 hours for a Prop-Fan utilizing the current technology system defined by the pitch control concept in NASA report CR 168258. The improvement in MTBUR is a result of the high reliability of the individual components in the advanced pitch control system and the modularity that allows many components to be replaced on line.

Table 4.4.4-3 Unscheduled Removals (All Causes)

Component	Removal Rate (Events/1000 Flight Hours)
Spinner	0.0086
Cover and fairings	0.0110
Blades	0.0688
Disks and fairings	0.0058
Aft power transfer module	0.0165
Forward power transfer module	0.0070
Aft actuator module	0.0118
Forward actuator module	0.0110
Hydraulic power module	0.0664
Electric drive module	0.0280
Electronic control module	0.0836
Aft deicing electronic control	0.0297
Lube pump module	0.0169
Alternator drive assy.	0.0037
Aft alternator	0.0034
Forward alternator	0.0033
Aft signal transfer assy.	0.0002
Forward signal transfer assy.	0.0002
Other	0.0054
Total:	0.3813

MTBUR = (1/0.3813)(1000) = 2,600 hours

Table 4.4.4-4 is a summary of the Prop-Fan reliability for both current and advanced technology pitch control concepts.

Table 4.4.4-4 Prop-Fan Reliability Summary

	Current Tech	Advanced Tech
o MTBUR, Prop-Fan Assy. (chargeable), hrs	4,900	13,800
o MTBUR, Prop-Fan Assy. & Components (all causes), hrs	1,500	2,600

4.4.4.9 Prop-Fan Failure Mode Considerations

The counter-rotation Prop-Fan pitch change mechanism and control logic has been evaluated to assure the design philosophy provides for safe operation. The purpose of this section is to summarize the evaluation with regard to both the Prop-Fan and its resulting impact on the selected gearbox.

The fundamental premise of the Prop-Fan safety philosophy is that fail fixed is fail safe and that uncontrollable decrease pitch is unacceptable. An in-place pitch lock for protection against mechanical and hydraulic failures prevents uncontrollable decrease in pitch. Redundant overspeed limiting provides protection against control failures which decrease pitch. Additional safety features include fuel limiting capability in the engine fuel control to prevent overspeed and fuel cutback as a function of measured torque to prevent transmitting excessive torque. Also, although fixed pitch operation on the pitchlock is considered flight safe, the pilot has the option to feather the blades with a separate analog emergency feather signal which bypasses the normal digital control.

Safety features are also incorporated in the full-authority digital electronic engine control which both commands and provides diagnostic feedback data to and from the engine and Prop-Fan. If loss of the EEC is experienced, the Prop-Fan reverts to 100 percent rpm speed control using the electronic control mounted on the Prop-Fan.

Both grounded and ungrounded gearboxes were considered in terms of the specified safety philosophy. For a grounded gearbox installation, the blade pitch angle controls the torque in each rotor drive path. For an ungrounded gearbox installation, the blade pitch angle controls the rpm of each rotor drive path. The results of trade studies presented in this report are applicable to both systems. However, the counter-rotation Prop-Fan pitch control concepts discussed in this report are based on an ungrounded differential planetary gearbox system.

A preliminary failure mode and effects analysis (FMEA) indicates that the Prop-Fan safety philosophy also protects the gearbox. To summarize the key results of the FMEA, three failure modes will be discussed. They are: 1) failure of the forward rotor blade angle toward feather with power on; 2) feathering of the forward rotor during in-flight shutdown with the aft rotor pitch locked, and 3) feathering of the aft rotor during in-flight shutdown with the forward rotor pitch locked. For condition 1), the forward rotor will decrease its speed, while the aft rotor will maintain its speed through the governing system. The engine speed will decrease with the torque limiter regulating its output. For conditions 2) and 3), the feathered rotor will rotate in the same direction as the aft rotor at a low speed, the pitch locked rotor will rotate at less than 100 percent speed, and the engine rotor will rotate at less than 50 percent speed.

Similar ungrounded gearboxes have been successfully used on two Russian applications: the Tupolev TU-95 "Bear" and the Antonov AN-22 "Cock." This experience provides evidence to confirm the results of analytical studies regarding the safety of the ungrounded gearbox.

4.4.4.10 Costs

4.4.4.10.1 Maintenance Cost

Maintenance costs for the Prop-Fan with the advanced technology pitch change system were estimated utilizing an on-condition philosophy established for the Prop-Fan as indicated in the single-rotation Prop-Fan (Section 4.2) discussed previously.

The maintenance cost was developed for the 6 by 6 bladed, 3.54 m (11.6 ft) diameter Prop-Fan.

The total maintenance cost for the Prop-Fan with an advanced technology pitch control represents a 19 percent decrease from the baseline Prop-Fan relative maintenance cost referenced in NASA report CR 168258, Figure 82. Baseline costs were escalated for the 1984 economy and adjusted to the 3.54 m (11.6 ft) diameter. The lower maintenance cost of the advanced system is primarily the result of reducing the frequency of maintenance actions and increasing modularity.

4.4.4.10.2 Acquisition Cost

The acquisition cost for a Prop-Fan with an advanced technology pitch control concept is approximately equal to the baseline current technology concept. Acquisition cost estimates were developed as described in the section on maintenance cost.

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SECTION 4.5 -- DISCUSSION OF RESULTS

TASKS IX AND XIII -- THE RESEARCH AND TECHNOLOGY PLAN

Table of Contents for Section 4.5

List of Illustrations for Section 4.5

List of Tables for Section 4.5

4.5 Research and Technology Plan - Tasks IX and XIII

4.5.1 Introduction

Results from the base APET Study Contract, the single-rotation gearbox preliminary design analysis, and the counter-rotation gearbox preliminary design analysis have established advanced gearbox and pitch control designs for both the single and counter-rotation configurations. Figures 4.5.1-1 and 4.5.1-2 show areas requiring further technology verification for the single-rotation and counter-rotation designs, respectively. A comparison of these two gearboxes shows the similarity of the required technologies. Verification of these technologies will assure meeting the gearbox and pitch control design goals for performance and durability for either singleor counter-rotation turboprop applications. In addition these technologies have wide applicability to the transmission systems of both military and commercial future helicopter and geared turbofan propulsion systems.

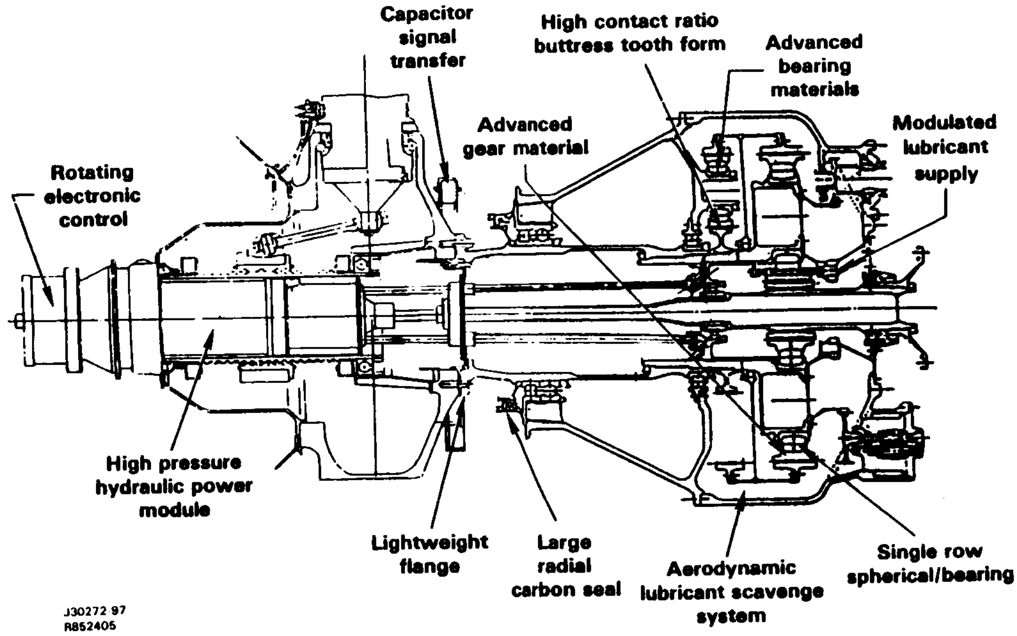

Figure 4.5.1-1 Single-Rotation Areas Requiring Technology Verification - Verifying these technologies will assure meeting performance and durability goals.

The focal point of the gearbox technology plan is the NASA Advanced Gearbox Technology (AGBT) Contract program. It is structured to evaluate each component technology and its interactive effects in a systems environment. Testing will be conducted in a large-scale gearbox configuration at operating conditions simulating a typical flight cycle. Testing of some supporting technology has already begun. This technology is in the areas of gear and bearing material development, the design of planetary bearings, and the development of advanced lubricants and lubrication systems. The verification of other supporting technologies should be conducted in a parallel effort.

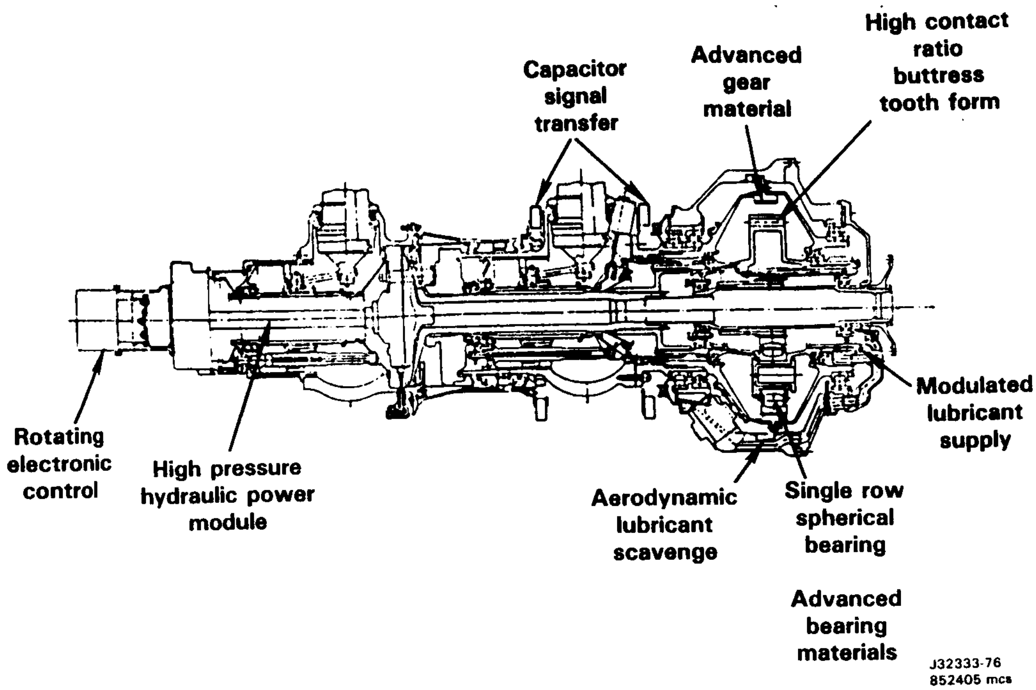

Figure 4.5.1-2 Counter-Rotation Areas Requiring Technology Verification - The technology required for counter-rotation is very similar to single-rotation's.

This multiyear technology verification program, which started in 1984, is a critical element in the overall NASA Advanced Turboprop Program which will introduce Prop-Fan propulsion to the aviation industry by the 1990's. For the program to be effective, comprehensive component rigs and materials research and technology programs must support the verification of a Prop-Fan gearbox in the AGBT program. Figure 4.5.1-3 presents a plan for integrating the various research and technology programs.

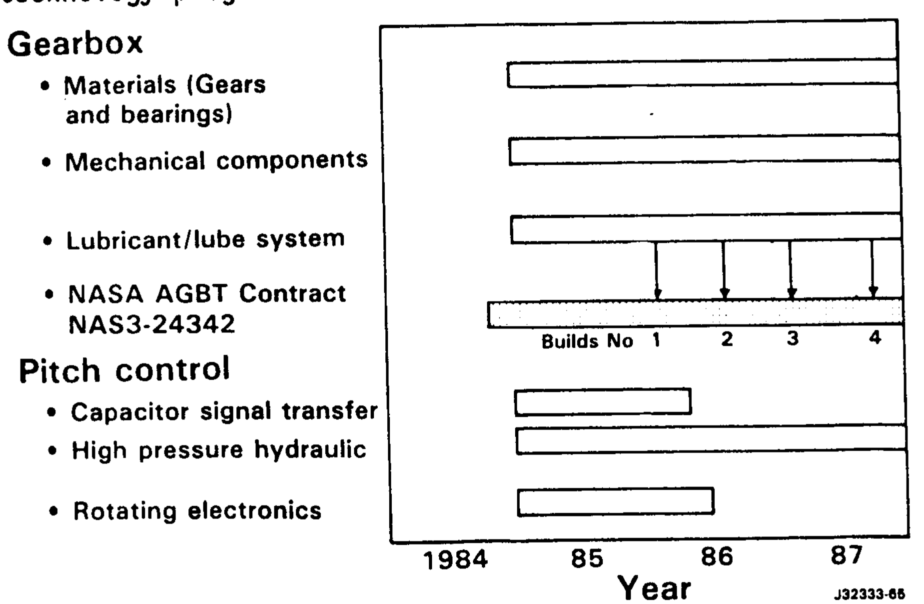

Figure 4.5.1-3 Gearbox/Pitch Control Overall Technology Plan - All the gearbox component technology programs can be tested in the AGBT program rig.

All of the gearbox component technology programs can be tested in the AGBT program rig. The pitch control technologies will be verified before flight testing.

The following sections present gearbox and pitch control research and technology plans in detail. Program schedules are aligned with a mid 1987 technology readiness objective. This is consistent with NASA and Pratt & Whitney's plans to ensure certification of a Prop-Fan powered aircraft in the early 1990's.

4.5.2 Gearbox Research and Technology Plan

Supporting gearbox technology plans cover a wide range of disciplines including materials and durability, mechanical components, and lubricant and lubrication systems. Technology requiring verification is the same for both the single and counter-rotation gearbox systems. The program plans take advantage of existing gearbox test rigs at Pratt & Whitney and at other divisions of United Technologies Corporation. The following plans are suitable for generic technology programs which would benefit a wide range of gearbox applications.

4.5.2.1 Materials Technology Plan

Application of advanced materials to the gears and bearings in the advanced technology gearbox is necessary for meeting weight and durability goals. These new materials must operate at higher temperatures, with improved fracture toughness and better resistance to bending fatigue, surface fatigue, and wear.

Plans for improving gear and bearing technology will help to meet these objectives. Some of these tasks are part of the AGBT Program. The plans are as follows:

- o evaluate advanced materials and establish design requirements,
- o conduct single element rig tests,
- o perform full-scale component testing, and
- o incorporate advanced technology gears and bearings in the AGBT rig to evaluate performance and durability.

Figure 4.5.2-1 presents a schedule of the overall plan for verifying materials technology.

<u>Gears</u>

Materials development for gears will consist of performing comparative property tests of candidate material specimens and obtaining design properties. Specimens will be machined and their mechanical properties will be evaluated under laboratory conditions. Results will be made available to the design department. After the specimen tests, prototypes will be fabricated and tested.

To meet the schedule shown in Figure 4.5.2-1, early selection of promising materials is essential. Examples of such materials are EX-53 or Vasco X-2 for replacing AISI 9310. Scoring and pitting resistance of these advanced materials will be established and processing variables will also be examined.

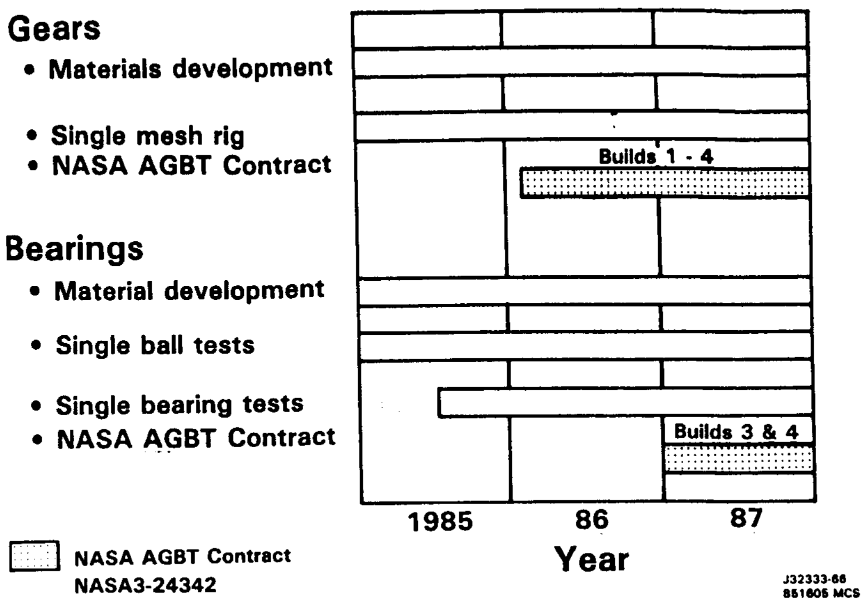

Figure 4.5.2-1 Gearbox Materials Research and Technology Plan - To meet this plan, early selection of materials is essential.

Figure 4.5.2-2 is a schematic of an existing single-mesh gear rig. This rig is used to test gear tooth form and to evaluate advanced materials and lubricants. The rig consists of two sets of test gears, a loading mechanism, and driver motor and train. By using helical loading gears, the axial load cylinder applies torque to the shafts, loading the test gears against each other. The power from the drive overcomes the frictional forces in the system. The frictional forces are a fraction of the loading forces.

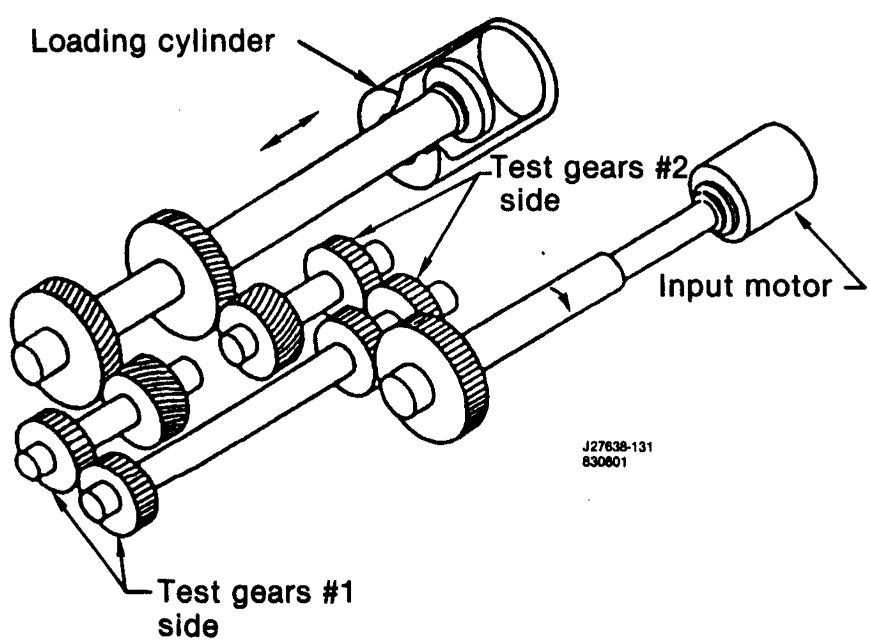

Figure 4.5.2-2 Single Mesh Gear Test Rig - This rig can be used to evaluate gear tooth forms and to evaluate materials and lubricants.

Bearings

The evaluation of advanced technology bearings is similar to that of the advanced technology gears. First, advanced material bearings are evaluated in specimen test rigs. Design properties are derived from the test results, and the results are made available to the Design Department. Second, promising materials are evaluated in a single ball test rig. Third, full-scale bearing component tests evaluate the advanced material and lubricating properties of advanced lubricants.

Bearing materials presently in use are VIM VAR M50. For future applications, we will need stronger materials such as Cartech CRB7 which is a technology projected for use in 1988.

Figure 4.5.2-3 shows a bearing test rig concept. This rig permits dynamic testing of full-scale bearings under controlled loading conditions. The rig can be used to obtain data on planet bearing friction and wear, as well as data on lubrication and cooling characteristics of advanced lubricants. It will also evaluate the rolling contact fatigue properties of candidate gear materials for a planetary gear bearing with an integral gear and bearing outer ring.

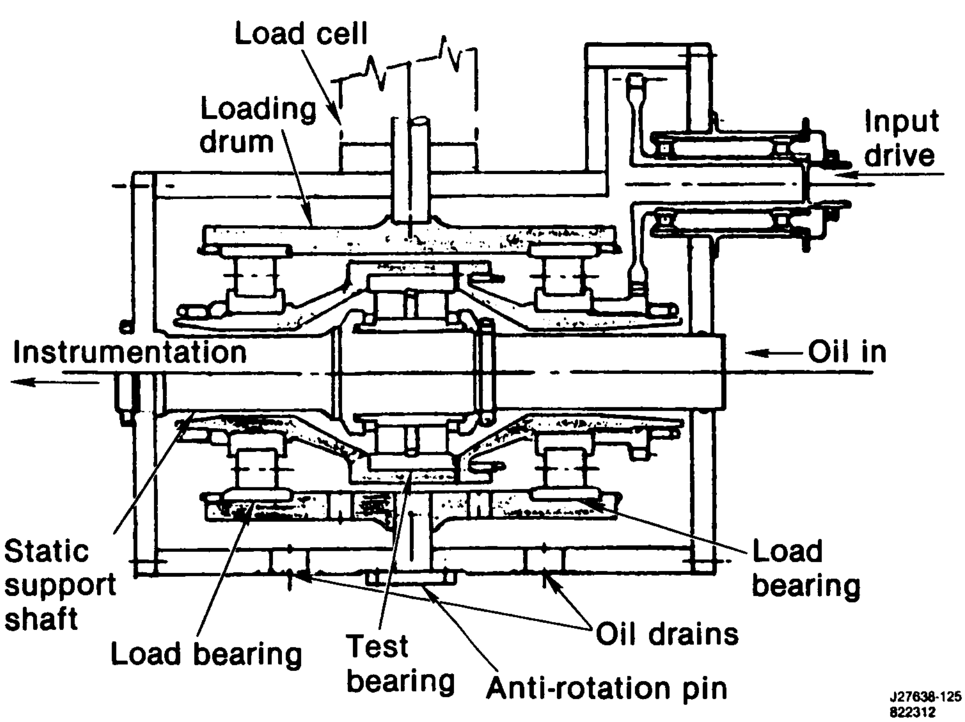

Figure 4.5.2-3 Single Bearing Test Rig - This rig will provide data on planet bearing friction and wear.

4.5.2.2 Mechanical Components Technology Plan

Several mechanical components of advanced gearboxes require separate technology demonstration tests. These components include planetary gear sets, planet pinion bearings, a large output shaft seal, and output shaft flange connections to the Prop-Fan. Figure 4.5.2-4 shows the plan for testing these components. The mechanical component tests shown in this plan will be conducted parallel to operation of the multipurpose gear technology rig.

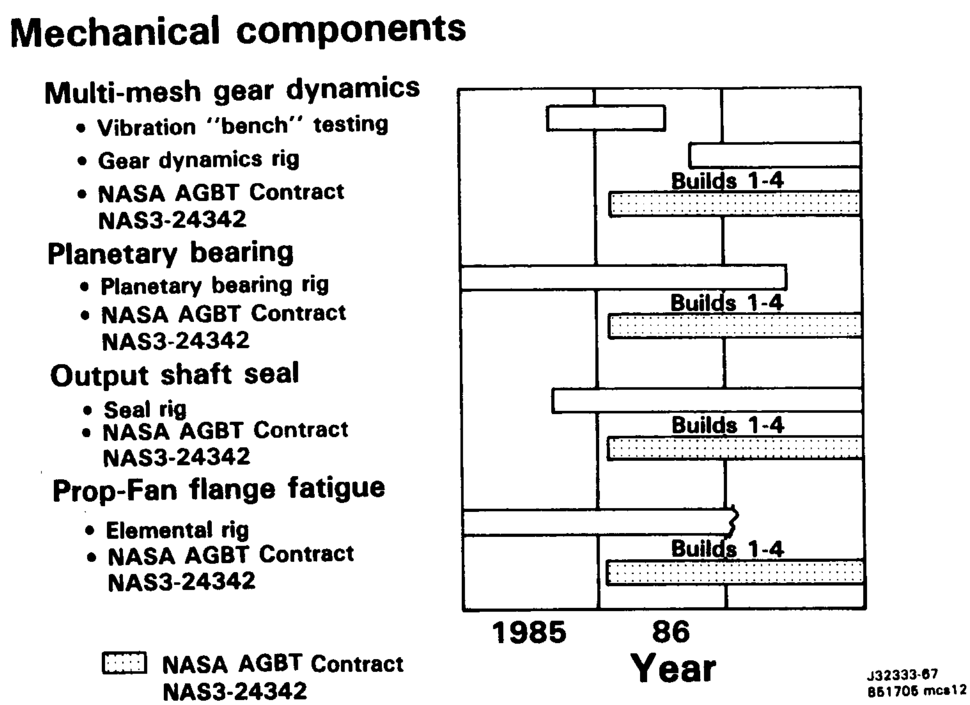

Figure 4.5.2-4 Gearbox Research and Technology Plan - The mechanical component tests can be conducted parallel to multipurpose rig operation.

MultiMesh Gear Dynamics

The multimesh gear dynamics program consists of vibration bench tests of critical components and specialized gear system dynamics tests.

The large-diameter, interconnected ring gear of the planetary gear set is an example of a specific component verification requirement. Natural frequencies and nodal patterns of the ring gear assembly must be identified and adjusted to provide resonance-free operation.

Tests of a complete planetary stage in a full-scale gear dynamics rig will follow bench testing to measure and modify natural frequencies. A single stage of the split path planetary gearbox will be tested under a light load (2 to 5 percent of rated power) to measure the gear stage's dynamic response over a wide range of speeds and under various controlled disturbances. These disturbances will include such factors as sun gear imbalance, planet pinion or star misalignment, and tooth spacing error in any gear in the system. Stress and displacement measurements at key locations will define system dynamic response.

These data will identify system damping factors and establish critical tolerances for improved durability at high operating speeds. The gear dynamics rig will also provide a means to evaluate corrective changes in the event that dynamics problems arise as the complete gearbox is tested in the multipurpose gear rig. Individual dynamics rig components are readily accessible for extensive instrumentation and rapid modification.

Planetary Bearing

A single-row spherical roller pinion bearing is used in both planet and star stages of the split path planetary gearbox. These applications represent a wide range of operating conditions, particularly with gearboxes for both conventional and opposite rotation. Conditions in the planetary stage are especially severe, because bearing loads are due to the combined effects of gear loads and centrifugal forces. While a complete gearbox is necessary for exposing a planetary bearing to the total gearbox environment, a relatively uncomplicated bearing rig which duplicates centrifugal force effects is vital to the component technology plan.

The pinion bearing rig consists of planet pinion bearings and gears mounted in a planet carrier, which is driven at propeller shaft speed, and a sun gear which engages the planet gears and is driven at input shaft speed. Centrifugal loads, speeds, temperatures, misalignments, and lubrication features of the complete gearbox are duplicated.

Bearing performance and durability characteristics will be examined and critical features will be modified to establish their effects on bearing operation. Of special interest is the effect of centrifugal loading on the wear of roller separator pockets and support lands and on stresses in the separator. Bearing misalignment and its effect on friction and wear of roller guide surfaces is also critical.

The results of pinion bearing technology testing will enhance gearbox performance and ensure adequate understanding of major factors which affect bearing durability.

Output Shaft Seal

The Prop-Fan output shaft seal is at a much larger diameter and higher surface speed than propeller shaft seals for established turboprop gearboxes. The use of a conventional elastomeric lip seal is not considered viable due to the high speed and the extended service life of the Prop-Fan. The requirement for negligible oil leakage and effective foreign particle exclusion dictates the use of a rubbing contact seal. Severe radial excursions, due to large propeller side loads which change direction through each flight cycle, favor the use of a face seal, while the large diameter and on-the-wing maintainability requirements indicate a radial seal design. Innovative sealing devices and material technologies are sought to enhance seal effectiveness and obtain service lives consistent with gearbox reliability objectives.

A suitable seal test rig will duplicate all essential sealing environmental factors including size, speed, seals, fluids, temperatures, pressures, displacements, and foreign particle contamination. Test measurements will include oil and air leakage rates, temperatures, heat generation, and sealing element wear rates. The test program will lead to the identification of seal candidates for full-scale gearbox testing and performance verification.

Prop-Fan Flange Fatigue

The Prop-Fan assembly is mounted on the gearbox output shaft by means of flanges on each part clamped together by a set of nuts and bolts. The bolted joint carries a complex combination of aerodynamic and mechanical loads. The flanges rotate through a pronounced fixed radial load and bending moment due to the combined effects of propeller aerodynamics and weight. The resulting cyclic stresses are superimposed on steady stresses due to the axial thrust load and the tangential power transmission load. The intensity of the cyclic stress can severely compromise flange life by causing flange surface damage due to fretting wear, followed by fatigue origins in the fretted surfaces.

The flange fatigue elemental rig will establish contact pressure and surface shear stress and displacement design guidelines for prevention of serious surface damage and exposure to fatigue cracking. The rig will also provide a comparison of fretting damage resistance in materials and coatings. Test data will be gathered for a range of contact pressure and contact slip amplitudes. Test results will ensure the design of minimum weight flanges without compromising gearbox reliability.

4.5.2.3 Lubricant and Lubrication System Technology Plan

Gearbox durability and performance are highly sensitive to lubricant and lubrication system characteristics. Lubricant characteristics must be well matched to the mechanical system and materials, so that surfaces are well protected and thermal behavior is controlled. Advanced lubrication system concepts, which are designed for improved lubrication effectiveness with low windage losses, must also be evaluated and perfected.

The plan shown in Figure 4.5.2-5 leads to improved protection of gearbox component surfaces at higher temperatures and at reduced oil flow rates. This results in improved durability and better efficiencies than those currently realized.

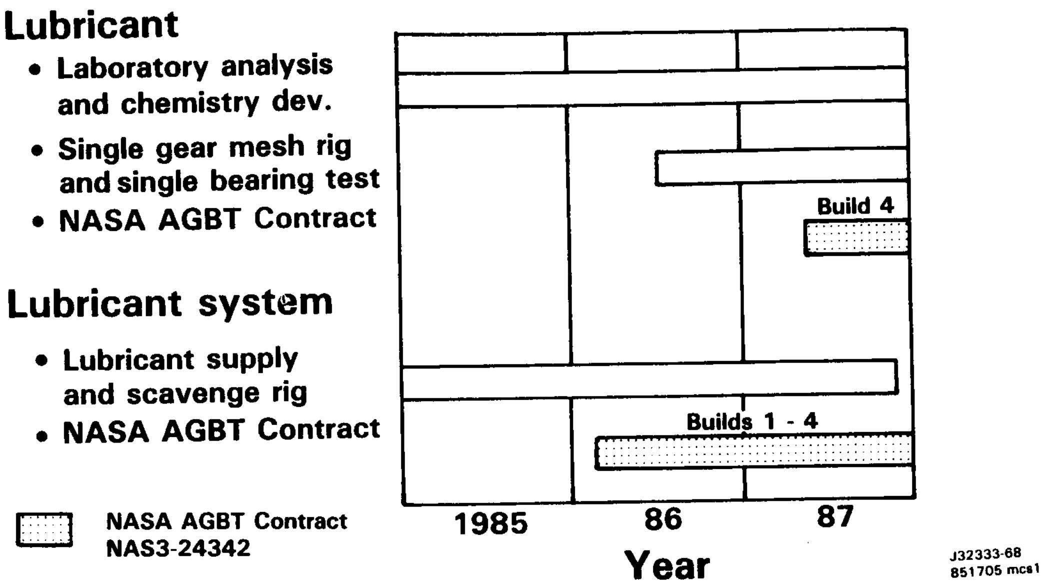

Figure 4.5.2-5 Gearbox Research and Technology Plan - Improved lubricants and lubrication system results in improved durability and higher efficiencies than those currently realized.

Lubricant

Table 4.5.2-1 summarizes the benefits of improved lubricant technology for the bearing, fuel burn, and dispatch reliability. The bearing life factors, which reflect improved material, lubricant, and surface finish, are projected to be in the range of 20 to 30 times "catalog" ratings for 1988 technology, whereas current technology features are in the six to 12 range. If current lubricants must be used in 1988, the life factor range is 15 to 28. The corresponding decrease in bearing system life is 20 percent. This translates into 3,000 hours, assuming bearing sizes are not adjusted. The net effect on fuel burn is significant, which should provide the incentive necessary to address advances in lubricant technology.

The acquisition of lubrication technology begins in the suppliers' laboratories. It then proceeds in conjunction with many of the same rig tests used for gear and bearing materials.

In the first phase of the lubricant technology acquisition, Pratt & Whitney will work closely with major oil companies to define the design requirements and desired oil properties. The oil companies will use this information to produce oil chemistry which will meet these requirements. The oil companies will also supply these oils to Pratt & Whitney for evaluation.

After the initial laboratory evaluations, the oils will be incorporated in gearbox component test rigs for further evaluation. Ultimately, promising oils will be evaluated in the AGBT multipurpose gear rig. The results will be compared with the current technology oils.

Table 4.5.2-1 Debits If Lubricant Technology Does Not Materialize

	Technology assumed available by 1988	Debit if lube technology is not available
• Bearing material/lubrication life factor	Base	10 to 25%
• Efficiency	Base	0.3%
• Bearing system life (L_{10}) hrs	Base	20%
• Gearbox reliability (MTBUR) hrs	Base	10%
• Fuel burn	Base	0.6%
• Dispatch reliability (measured in equivalent DOC)	Base	0.2%

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Lubrication System

Lubrication system verification requires a specialized rig which models critical sections of the gearbox and lubrication system. Unique design features such as the oil supply modulation and aerodynamic scavenge system will be evaluated and refined. These tests will provide design input and will support the AGBT multipurpose rig rests.

A dual-purpose test rig will be designed and fabricated. This rig, shown in Figures 4.5.2-6 and 4.5.2-7, will be used to:

a) develop an improved method of scavenging the oil from the gearbox housing; and

b) develop an improved method of supplying oil to the critical gear meshes such as the sun-pinion gear and a method to scavenge the oil from these regions.

Figure 4.5.2-6 shows a unique system for aerodynamic scavenging of oil from the gearbox housing. Scavenging is accomplished by introducing air and oil into a rotating drum that simulates the ring gear support rotor. Oil and air drain out of this rotating drum, simulating the swirling air/oil mixture in the gearbox housing. A replaceable outer housing sector is provided where the swirling air/oil leaves the rotating drum. This replaceable housing sector allows testing different aerodynamic scroll, louvered, and gravity collector scavenge schemes.

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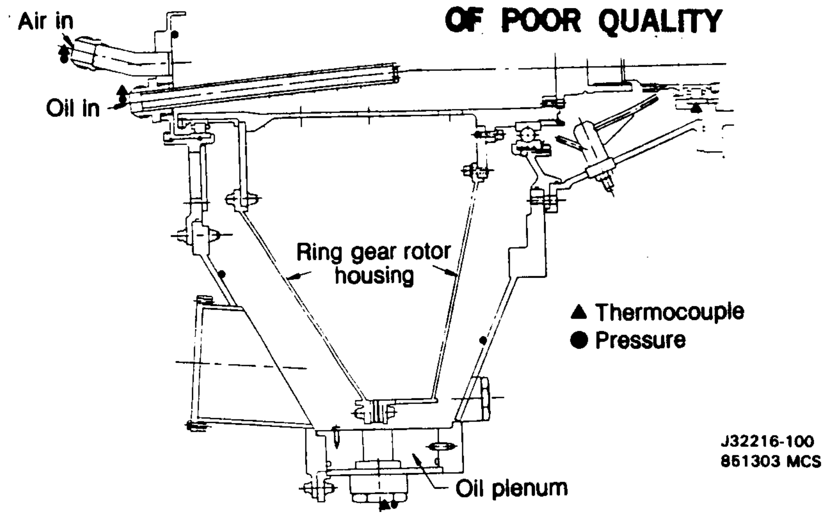

Figure 4.5.2-6 Scavenge and Lubricant Supply Rig - In this configuration the dual-purpose test rig will permit optimization of lubricant scavenge system.

In the second configuration, Figure 4.5.2-7, the rig will be adapted to evaluate the oil supply and scavenging for critical gear meshes. For instance, for the sun-pinion gear shown, the pinion carrier is mounted on the rig housing. The sun gear is driven by the rig counter-rotating gear system. Since the carrier is not rotating in this lubrication rig, the sun-pinion mesh region can be observed through the windows provided in the housing. This feature will enable easier evaluation of oil supply tube arrangements. Baffles for scavenging and air flow schemes to assist scavenging will also be evaluated. Since the carrier rotates in a counter-rotating gearbox, this rotation is expected to help scavenge the oil, further improving the efficiency of the system.

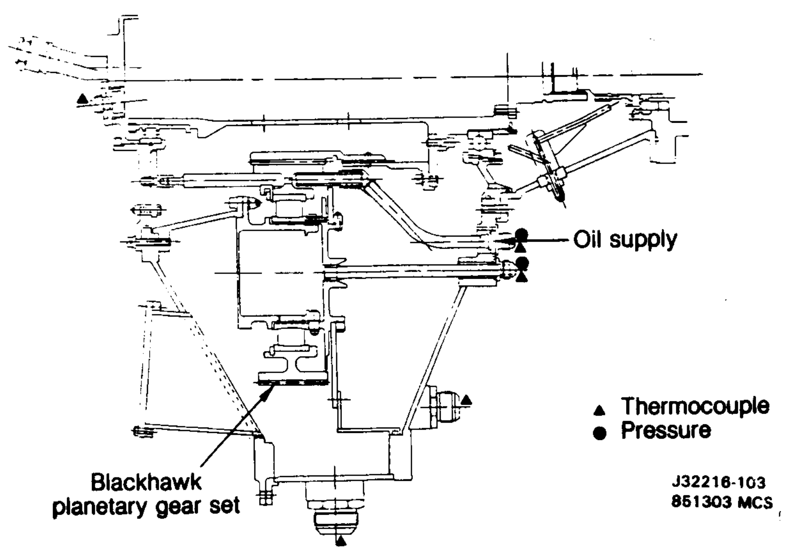

Figure 4.5.2-7 Scavenge and Lubricant Supply Rig - In this configuration the dual-purpose test rig will permit optimization of sun and planet gear lubrication system.

4.5.3 Pitch Control Research and Technology Plan

The conceptual design of a pitch change mechanism identified advanced component technologies that require technology programs for substantiation. These component technologies are the capacitor signal transfer, the high pressure rotating hydraulic power module, and the rotating electronic control module.

Figure 4.5.3-1 shows the pitch control technology plan.

The proposed program will address three technology features mentioned above:

- o Capacitor signal transfer,
- o High pressure rotating hydraulic power module (gear pump and motor), and
- o Rotating electronic control.

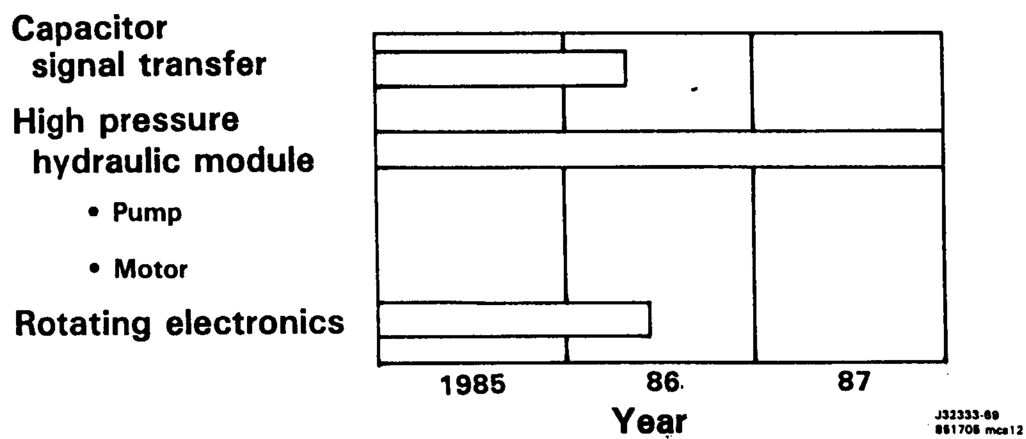

Figure 4.5.3-1 Pitch Control Technology Plan - The conceptual design identified technologies requiring verification.

4.5.3.1 Capacitor Signal Transfer

Current turboprops transmit an electrical signal across a rotating interface by using brushes and slip rings. This method has inherent problems, the most notable is carbon buildup due to brush wear and susceptibility to contamination from oil. These problems require frequent maintenance.

The use of a capacitor signal transfer eliminates these shortcomings. The major area of concern with regard to this concept is twofold. The first concern is the concept's susceptibility to electromagnetic interference (EMI) and vulnerability to lightning strike interference. The second concern relates to ensuring that the capacitor itself does not emit electromagnetic interference.

The program will include the design, fabrication, and testing of a shielded capacitor system. This system, which will be adaptable to an existing propeller barrel, will include a breadboard transmitter/receiver and will be subjected to an EMI survey test for susceptibility and emission. If necessary, additional shielding systems will be designed, fabricated, and tested.

Lightning transient tests will determine if the capacitor ring can withstand high voltage transients without damage. This program will result in a control signal transfer technique that is adaptable to both current and future turboprop systems and will eliminate the need for brushes and slip rings. Figure 4.5.3-2 shows the program schedule for the capacitor signal transfer.

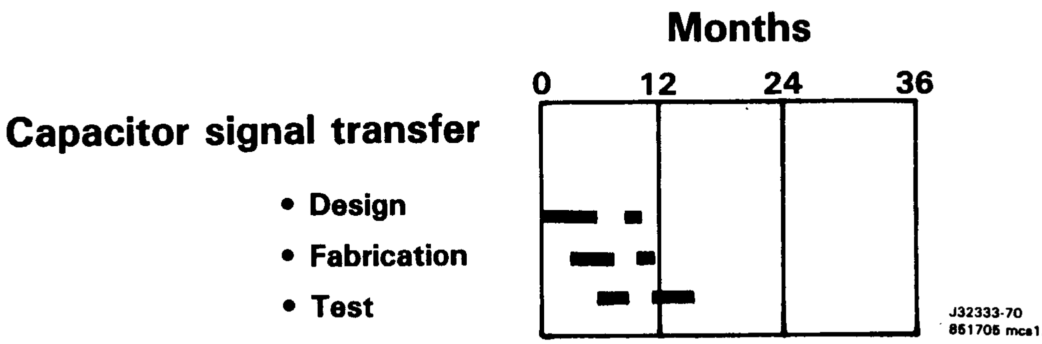

Figure 4.5.3-2 Program Schedule For The Capacitor Signal Transfer - This program will result in a pitch control signal transfer technique that will eliminate the need for brushes and slip rings.

4.5.3.2 High Pressure Hydraulic Power Module

Current turboprop systems use low pressure hydraulics, with the hydraulic power components mounted on the stationary side. The oil necessary for changing pitch is supplied to the rotating components through a transfer bearing. Experience has demonstrated that large diameter transfer bearings, which are common with many current systems, require frequent maintenance. Furthermore, Hamilton Standard conducted independent studies which indicate that using high pressure hydraulics can reduce the system weight.

A system has been devised for changing pitch on future turboprops with high pressure hydraulic supply components. For in-line gearbox systems, the components would be mounted on the rotating portion of the system. This eliminates the need for a transfer bearing and permits removal of hydraulic pitch change hardware from the gearbox. A concept has been developed whereby this can be achieved with a reliable, small diameter transfer bearing without impacting the gearbox. The use of high pressure hydraulics reduces size and weight of these components for optimized installation and maintenance. The objective of the pitch change technology program for single-rotation and counter-rotation in-line gearbox configurations is to establish an acceptable gear pump and gear motor that will operate at 32.75 MPa (4,750 psi) in a rotating environment, with an operating life design goal of 30,000 hours.

The approach is to design and build both a gear pump and gear motor sized for the requirements of a potential Prop-Fan system. Testing will determine torque characteristics, leakage, endurance, and susceptibility to cavitation. Gear motor testing will also include measurement of break out differential pressure and assessment of low speed characteristics due to the requirement for low friction.

This program will establish the feasibility of a 32.75 MPa (4,750 psi) gear pump and gear motor and will define hardware that is suitable for development on advanced pitch change systems. Figure 4.5.3-3 shows the program schedule for high pressure hydraulics.

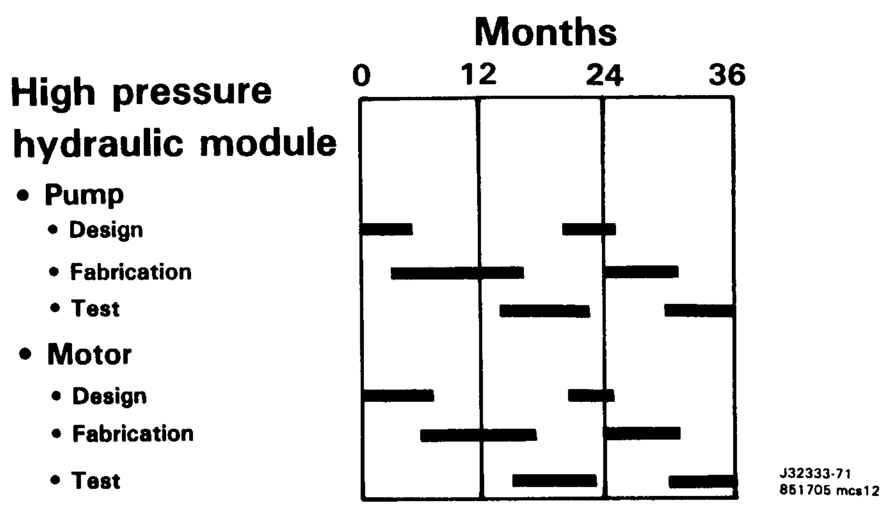

Figure 4.5.3-3 Program Schedule For The High Pressure Hydraulics - The use of high pressure hydraulics in advanced pitch control results in a reduced size and weight for optimized installation and maintenance.

4.5.3.3. Rotating Electronics

Current turboprop systems use hydromechanical pitch change controls that are mounted on the stationary side of the system. Mechanical signal transfer is necessary from the control to the pitch change actuator, across the rotating interface. This task is particularly difficult for in-line gearbox configurations and results in pitch change hardware being located within the gearbox. This has a significant impact on gearbox reliability and maintenance cost.

Replacing the hydromechanical system with rotating electronics will greatly simplify the signal transfer across the stationary-to-rotating interface. There is some concern regarding the ability of the electronic circuits to operate and survive in a high level "g" field. The objective of this technology program is to determine both the operational characteristics and the survivability of the electronic controller (the interfacing electronics package for signal conditioning, feedback signals, and control of the electro-hydraulic servo motor) when mounted and operating in a rotating field.

It is first necessary to establish the environmental requirements. Concepts for the structural packaging of the electronics for survival in this environment will be developed, and a breadboard differential input digital data transmitter/receiver circuit will be constructed for dynamic test evaluation. These tests will include both whirl tests and vibration tests over the total frequency spectrum anticipated for Prop-Fan mounted hardware. This program will establish the feasibility of a rotating electronic control and will define hardware which is suitable for development for advanced pitch change systems. Figure 4.5.3-4 shows the program schedule for rotating electronics.

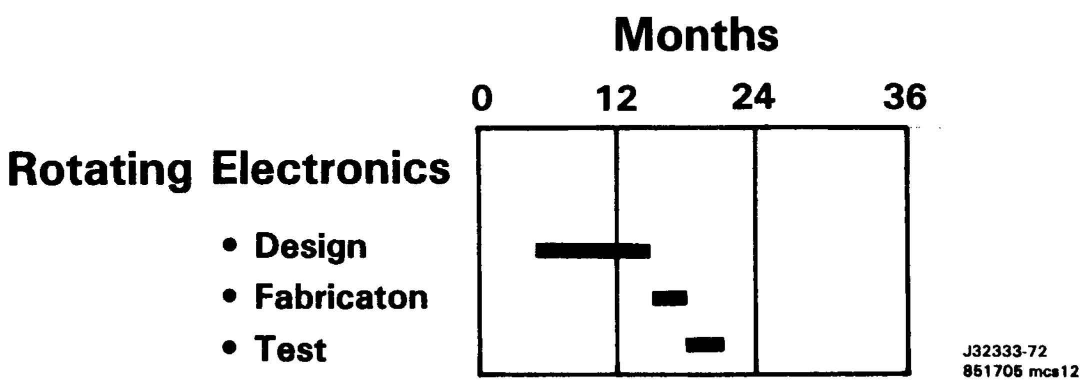

Figure 4.5.3-4 Program Schedule For The Rotating Electronics - The program will establish the feasibility of a rotating electronic control and will define hardware for development.

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SECTION 5.0
CONCLUDING REMARKS

SECTION 5.0
CONCLUDING REMARKS

APET studies have shown large potential payoffs -- 31 percent fuel burned and 14 percent Direct Operating Cost -- for an advanced technology geared counter-rotation Prop-Fan system relative to a turbofan propulsion system in future short range aircraft.

The work accomplished under these two phases of the APET (Advanced Prop-Fan Technology) program has clearly shown that advanced technology offers significant payoffs when used in the design of future Prop-Fan gearboxes. Whether for single or counter-rotation, using advancements in materials, lubrication systems, lubricants, and gear and bearing geometry provides far superior performance and operating economics than does current technology. Advanced technologies allow a simple, more compact gearbox, because fewer and smaller bearings and gears are necessary. They have also allowed a design that greatly improves component accessibility for on-wing maintenance. One noteworthy example of this is the removal of the pitch control mechanism from the gearbox.

Besides quantifying and qualifying the benefits of advanced technology, a gearbox technology plan will demonstrate technology verification by mid 1987. This timing is critical because it allows for full-scale Prop-Fan development in the late 1980's and certification in the early 1990's. Of major importance, the technologies requiring demonstration are identical for both single and counter-rotating gearbox systems. Moreover, they have wide application for use in both commercial and military advanced helicopter and geared turbofan engines.

Overall, the earlier APET Definition Study and the preliminary gearbox design efforts have provided an essential technology base. Continuation of this work with a gearbox technology verification program is a very important step in bringing the large benefits of a geared Prop-Fan propulsion system to commercial aviation.

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Abbreviations and Symbols

AGBT	Advanced Gearbox Technology
AL	Aluminum
APET	Advanced Prop-Fan Engine Technology
Btu	British thermal units
Btu/min	British thermal units per minute
°C	Degrees Centigrate
Cm	Centimeter
CW	Clockwise
CCW	Counter Clockwise
deg/sec	Degree per second
DN	Speed Factor
DOC + I	Direct Operating Cost Plus Interest
EEC	Electronic Engine Control
°F	Degrees Fahrenheit
Ft	Feet
Ft-lb	Foot-Pounds
Ft/min	Feet per minute
Ft/sec	Feet per second
HCR	high contact ratio
hp	horse power
hrs	hours
in	inch
in-lb	inch-pounds

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Kg	Kilogram
Kg/min	Kilogram per minute
Lb	pounds
Lb/min	pounds per minute
LVDT	Linear variable differential transducer
LVPT	Linear variable phase transducer
M	meter
MG	magnesium
min	minutes
mn	mach number
MPa	Mega-Pascals
MTBUR	Mean Time Between Unscheduled Removals
N	Newtons
NASA	National Aernautics and Space Administration
N-m	Newton-meters
N/Mm	Newtons per millimeters
NMi	Nautical Mile
No.	Number
psi	pounds per square inch
R_1	planet stage ratio
R_2	Star stage ratio
Rad/sec	radians per second
RVDT	Rotary variable differential transducer
SHF	Synthesized hydrocarbon fluid
β	blade angle
℄	center line

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Hamilton Standard Division, UTC
Windsor Locks, CT 06096
Attn: J.A. Baum, M.S. 1-2-11 (1)

Hamilton Standard Division, UTC
Windsor Locks, CT 06096
Attn: S. H. Cohen, M.S. 1-2-11 (1)

Hamilton Standard Division, UTC
Windsor Locks, CT 06096
Attn: B. S. Gatzen, M.S. 1-2-11 (2)

Hamilton Standard Division, UTC
Windsor Locks, CT 06096
Attn: M. G. Mayo, M.S. 1A-3 2 (2)

Hartzell Propeller Products
P. O. Box 1458
1800 Covington Avenue
Piqua, OH 45356
Attn: A. R. Disbrow (1)

Lockheed-California Company
P. O. Box 551
Burbank, CA 91503
Attn: A.R. Yackle,
Bldg. 90-1, Dept. 69-05 (2)

Lockheed Georgia Company
86 South Cobb Drive
Marietta, GA 30063
Attn: W.E. Arndt, M.S. D/72 17,
Zone 418 (4)

Lockheed-Georgia Company
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Marietta, GA 30063
Attn: D.M. Winkeljohn, M.S. D/72 79,
Zone 419 (1)

Pratt & Whitney
United Technology Corporation
Commercial Products Division
400 Main Street
East Hartford, CT 06108
Attn: J. Godston, M.S. 118-26 (1)

Pratt & Whitney
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Commercial Products Division
400 Main Street
East Hartford, CT 06108
Attn: A. H. McKibbin, M.S. 163-12 (1)

Pratt & Whitney
United Technology Corporation
Commercial Products Division
400 Main Street
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Attn: C. Reynolds, M.S. 118-26 (6)

Pratt & Whitney
United Technology Corporation
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Attn: C. V. Sundt, M.S. 118-27 (1)

Pratt & Whitney Aircraft
United Technologies Corporation
Military Products Division
P. O. Box 2691
West Palm Beach, FL 33402
Attn: L. L. Coons, M.S. 711-69 (1)

Pratt & Whitney Aircraft
United Technologies Corporation
Military Products Division
P. O. Box 2691
West Palm Beach, FL 33402
Attn: W. King, M.S. 702-05 (1)

Pratt & Whitney Aircraft
United Technologies Corporation
Military Products Division
P. O. Box 2691
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Attn: H. D. Snyder, M.S. 711-67 (1)

Pratt & Whitney Aircraft
United Technologies Corporation
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P. O. Box 2691
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Attn: S. Spoleer, M.S. 702-50 (1)

Pratt & Whitney Aircraft
United Technologies Corporation
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P. O. Box 2691
West Palm Beach, FL 33402
Attn: H.D. Stetson, M.S. 713-09 (1)

Sikorsky Aircraft
Transmission Engineering
North Main Street
Stratford, CT 06601
Attn: R. Stone, M.S. S-318A (3)

Williams International
2280 West Maple Road
P. O. Box 200
Walled Lake, MI 48088
Attn: Edward Lays, M.S. 4-9 (1)

Air Canada
Dorval Base H4Y 1CZ
Quebec, Canada
Attn: Goeff Haigh - Zip 14 (1)

Air Canada
Dorval Base H4Y-1CZ
Quebec, Canada
Attn: G.H. Jones - Zip 66 (1)

Air Transport Association
1709 New York Avenue, NW
Washington, DC 20006
Attn: D.J. Collier (1)

Delta Air Lines Inc.
Hartsfield Atlanta Intn'l Airport
Atlanta, GA 30320
Attn: J.T. Davis, Engr Dept. (2)

Federal Express
P. O. Box 727-4021
Memphis, TN 38194
Attn: B. M. Dotson, M.S. 4021 (1)

Ozark Air Lines Inc.
P. O. Box 10007
Lambert St. Louis Airport
St. Louis, MO 63145
Attn: Phil Rogers, Engrg Dept. (1)

Trans World Airlines Inc.
605 Third Avenue
New York, NY 10016
Attn: Engineering Dept. (2)

United Air Lines
San Francisco International Airport
San Francisco, CA 94128
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(2)