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INTEGRATED ACCESSORY SYSTEMS FOR SMALL GAS TURBINE ENGINES

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Prepared for

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

EUSTIS DIRECTORATE POSITION STATEMENT

This report contains detailed information on the design and performance trade-offs associated with advanced control and accessory systems for small gas turbine engines. The information contained in the report and the findings and conclusions cited are considered to be accurate and appropriate to the work conducted. The results of this work will be considered in future planning of any work related to gas turbine engine controls or accessories.

Mr. Roger G. Furgurson of the Propulsion Technical Area, Technology Applications Division seved as project engineer for this effort.

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The intent of the Integrated Accessory Systems for the Small Gas Turbine Engines Program was to optimize the controls and accessory systems for the next generation of small turboshaft gas turbine engines for Army helicopter applications. A 1977 development time frame was considered. A primary objective of the program was to reduce vulnerability of the controls and accessory components without severely compromising other important design criteria.

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During the Phase I conceptual design phase, analyses and preliminary designs of Control and Accessory (C&A) configurations for both front- and rear-drive engines were accomplished.

Phase II included preliminary design of two candidate control and accessory systems and selection of one system for detailed design. A tower shaft control and accessory drive with an air turbine starter was selected, and a detailed system design of all C&A components was accomplished. The critical items of the C&A system were identified and test programs were recommended for Phase III.

Phase III included testing of the following critical items: (1) fuel pump inducer (inlet suction tests), (2) high-speed oil pump (cavitation tests), (3) electronics cooling techniques (performance tests), (4) power turbine overspeed sensor (performance tests), and (5) starter overrunning clutch (endurance tests).

SUMMARY

The intent of this program was to optimize the controls and accessory systems for the next generation of small turboshaft gas turbine engines for Army helicopter applications. A 1977 development time frame was considered for the engine. A primary objective of the program was to reduce vulnerability of the controls and accessory components without severely compromising other important design criteria. The effort was accomplished by closely coordinating the study, selection, and design of the controls and accessory (C&A) components with consideration for their integration within the engine.

The program was divided into three phases. During the Phase I conceptual design phase, analyses and preliminary designs of C&A configurations for both front- and rear-drive engines were accomplished. The candiate C&A systems were evaluated using weighted selection criteria, and two candidate rear-drive C&A systems (Tower Shaft/Air Turbine Starter, Cluster Gearbox/Air Turbine Starter) were recommended for further detailed design study.

With Army approval of the two recommended candidate C&A systems, the Phase II detailed design effort was initiated. The early part of the Phase II effort was directed toward analyses and selection of one system for detailed design. A tower shaft C&A drive with an air turbine starter was selected, and a detailed system design of all C&A components was accomplished. The critical items of the C&A system were identified and test programs were recommended for Phase III.

The Phase III critical item test programs were selected by the Army, and development testing of the following critical items was accomplished: (1) fuel pump inducer (inlet suction tests), (2) high-speed oil pump (cavitation tests), (3) electronics cooling techniques (performance tests), (4) power turbine overspeed sensor (performance tests), and (5) starter overrunning clutch (endurance tests).

Recommendations for future C&A system development programs were made.

PREFACE

This effort is reported according to the terms of Contract DAAJ02-73-C-0003, which is under the technical direction of Mr. R. G. Furgurson, Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia.

The program was conducted by the Advanced Controls Group, Florida Research and Development Center, Pratt & Whitney Aircraft, West Palm Beach, Florida. Mr. B. C. Miller was the Program Manager. Technical coordination at P&WA was provided by Mr. J. D. Estes; Mr. A. White coordinated the activities at the Chandler Evans Corporation.

This report carries the internal designation of FR-6983.

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SECTION I

INTRODUCTION

A. PROBLEM STATEMENT AND APPROACH

Engine technology for military helicopters is advancing toward more reliable, higher performing, lighter weight engines with low vulnerability and simplified maintenance. Much of the advance in this technology has been accomplished by increasing the rotational speed of the major engine parts, which has reduced engine size. To avoid large, heavy, and extremely complex control and accessory (C&A) systems, it has been necessary to keep the state of the art of the C&A systems abreast of engine development.

Previously funded Army programs have contributed to the reduction of C&A component cost, size, and improvement of packaging. The Integrated Accessory Systems for the Small Gas Turbine Engines program was directed toward the continued advancement and optimization of small gas turbine engine C&A systems through study and evaluation techniques to (1) combine the component functions into integrated modules and (2) integrate component functions into the basic engine design. The program closely coordinated the design of the C&A components and the accessory drive system with the design of the engine interfaces before the engine configuration was firmly established.

The major problem in meeting the goal of this program was to establish a C&A system that incorporates, at its proper level, each of the established performance and design criteria. Reliability is of major importance, but, in a military application, a reliable component that is also highly vulnerable to combat threats is a poor compromise. Integrating the components within the engine to reduce the exposed area, providing high-speed or dual-function components, or locating the components so that the critical items are shielded by the engine structure were means of achieving such reduced vulnerability. The development risk and cost of these integration techniques were considered to reflect the growing concern for cost reduction. The impact on overall engine performance and weight was reflected in the system design. Since an Army helicopter must be field maintainable, the C&A implementation also considered component repair or replacement. The problem was therefore not just one of component integration, but component integration in a manner which did not severely compromise other important operational criteria.

B. DESCRIPTION OF APPROACH

The program was divided into three phases: (1) Conceptual Design, (2) Detailed Design, and (3) Critical Item Fabrication and Test. These phases are described in the following paragraphs.

1. Conceptual Design Phase

a. Conceptual Design Phase - Front Drive Engine

(A program outline for this phase is shown in Figure 1.)

ESTABLISH PRELIMINARY ENGINE CONFIGURATION → **ESTABLISH CONTROL AND ACCESSORY REQUIREMENTS** → **ESTABLISH COMPONENT TECHNOLOGY LEVEL** → **ESTABLISH CANDIDATE STARTER AND DRIVE TECHNIQUES** → **ESTABLISH CANDIDA C&A SYSTEMS**

ESTABLISH PRELIMINARY ENGINE CONFIGURATION

Requirements:

- 3 lb/sec
- 600 shp
- Front End Drive
- N_{gg} = 65,000 rpm
- NPT = 26,000 rpm
- TIT = 2350°F
- PR = 10.0

Approach:

- Scale Configuration From Current and Previous Small Engine Programs
- Maintain Flexible Configuration

ESTABLISH CONTROL AND ACCESSORY REQUIREMENTS

Components:

- Fuel Control
- Fuel Pump
- Lubrication System
- Electrical Power Generator
- 15-hp, 20,000-rpm Accessory Pad
- Variable Geometry Actuators
- Compressor Bleed Valve
- Anti-Icing Valve
- Ignition System
- Engine Starter

Approach:

- Define Mission
- Establish Engine Performance at Extreme Flight Points
- Determine C&A Requirements

ESTABLISH COMPONENT TECHNOLOGY LEVEL

Approach:

- Review Previous C&A Contract Efforts
- Review Current and Proposed C&A Technology Programs
- Coordinate Technology Review With C&A Vendors,

ESTABLISH CANDIDATE STARTER AND DRIVE TECHNIQUES

Approach:

- Review Air Force Program "Gas Turbine Advanced Accessory Drive System Study"
- Define Candidate Drive Techniques
- Describe All Candidate C&A Systems in Terms of Accessory Drive Technique and Engine Interface

ESTABLISH CANDIDA C&A SYSTEMS

Figure 1. Program Outline for Phase I - Conceptual Design - Front Drive Engine

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ESTABLISH CANDIDATE C&A SYSTEMS

EVALUATE 10 CANDIDATE C&A SYSTEMS

Requirement:

Evaluate the 10 Most Promising C&A Systems and Perform the Necessary Trades to Select the Best Five Systems

Approach:

- Perform Preliminary Analysis To Size the Candidate C&A Components
- Prepare Engineering Sketches of Possible Interface Techniques
- Analyze Each To Determine the Advantages and Disadvantages and Relative Merits
- Select Five C&A Systems for Continued Study

ANALYZE FIVE CANDIDATE C&A SYSTEMS

Requirement:

Continue Analysis and Mechanical Design Studies of Five C&A Systems To Provide Trades To Reduce Candidates to Two Systems

Approach:

- Perform Mechanical Integration Studies To Define Interfaces
- Modify Engine Design To Improve Integration Where Possible
- Evaluate the C&A Systems To Determine Trades
  Criteria Shall Include:
  1. Reliability
  2. Vulnerability Resistance
  3. Development Risk
  4. Cost
  5. Weight and Volume
  6. Performance
  7. Maintainability
  8. Installation Flexibility

CONDUCT TRADE-OFF STUDY SUMMARIES AND SELECT TWO FRONT DRIVE C&A SYSTEMS

Requirement:

Select Two C&A Systems for Preliminary Design Based on Analysis and Mechanical Design Integration Efforts

Approach:

- Determine Trade Study Rating Factor
- Evaluate Candidate C&A Systems
- Prepare Summary Charts of Ratings
- Select Two Systems
- Review with Army Technical Representative

e Engine

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1. A preliminary front drive engine component sizing and flowpath arrangement was established. The basic engine components were guided by the specified engine description (Appendix A).

2. The required control and accessory components and their performance requirements were defined. These components included the following:

   Fuel Control
   Fuel Pump
   Engine Lubrication System
   Starter
   Electrical Power Generation for Engine Use
   Variable-Geometry Actuator
   Aircraft Accessory Pad (15 hp at 20,000 rpm)
   Compressor Bleed Valve
   Anti-Ice Valve
   Ignition System.

3. A review of previous USAAMRDL and other Government-sponsored programs was made, and component vendors were surveyed to establish component technology levels compatible with the engine development time frame of 1977.

4. The candidate control and accessory drive techniques were defined. This effort utilized the results of the Turbine Engine Advanced Accessory Drive System Study (AFAPL Contract F33615-72-C-1170) where applicable.

5. Candidate integrated control and accessory systems applicable to a front drive engine were identified, and 10 systems were selected on a qualitative basis for preliminary design evaluation.

6. A preliminary analytical and mechanical design analysis was conducted to evaluate the 10 candidate schemes and to recommend 5 candidates for further analysis.

7. Analytical and mechanical integration studies of the five selected systems were performed, and two integrated systems were recommended for further analysis.

b. Conceptual Design Phase - Rear Drive Engine

(A program outline of this phase is shown in Figure 2.)

1. A preliminary rear drive engine configuration was defined. The control and accessory components were the same as defined for the front drive arrangement.

2. Candidate control and accessory drive techniques were defined, and six systems were selected on a qualitative basis for preliminary evaluation.

3. Analytical and mechanical design analyses were conducted to evaluate the six systems and recommend two systems for further analysis.

4. An assessment of both the front and rear C&A drive/engine starter system candidates was made, and a recommendation was made to the Army for approval to proceed with the Detailed Design Phase.

2. Detailed Design Phase

(A program outline for this phase is shown in Figure 3.)

1. Preliminary design layouts were provided that define the packaging, interfaces, volumes, environment and mechanical implementation of the two candidate systems selected in the Conceptual Design Phase.

2. Design analysis support was provided for the layout effort to establish the component sizes and weights and the effects on overall system performance. Preliminary reliability, maintainability and vulnerability assessments of the two systems were made. Evaluations of relative cost, development risk and installation flexibility were also made.

3. Trade-off studies of the two systems on the basis of selection criteria and rating factors were conducted, and one integrated system was recommended for design.

4. Preliminary control and accessory system component specifications were established that identified the interface and performance requirements.

5. A design of the selected system control and accessory components, through a combined P&WA and vendor effort, and with sufficient detail to identify any high-risk areas, was made.

6. A design of the selected C&A/engine interfaces, in sufficient detail to show mechanical implementation and to identify any high-risk areas, was made.

7. The high-risk components or subsystems were identified and ranked in order of priority from a standpoint of further required development.

8. A test plan and a cost estimate for a test program to evaluate each identified high-risk component or subsystem were prepared.

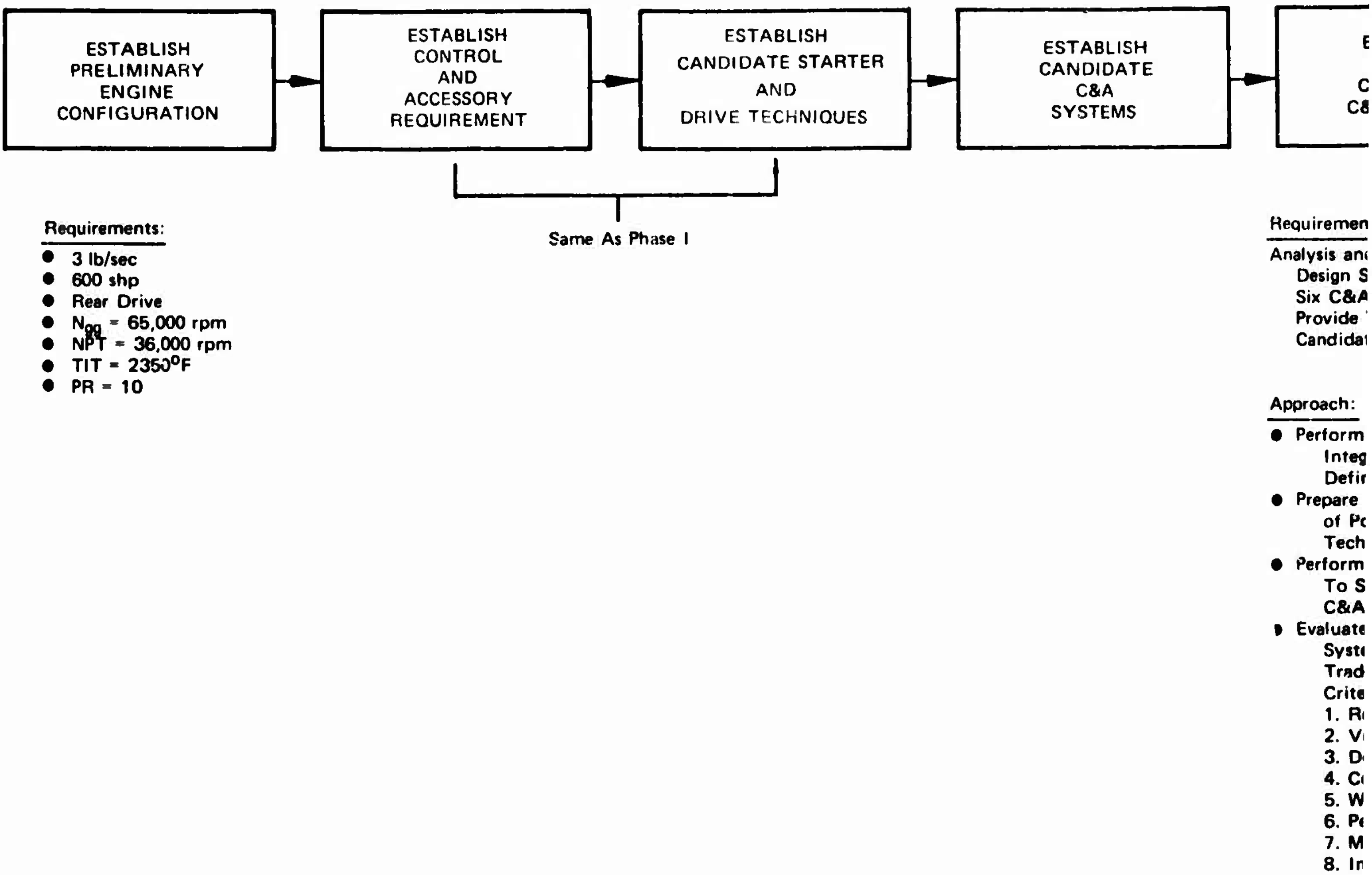


Figure 2. Program Outline for Phase I - Conceptual Design - Rear Drive Engine

EVALUATE SIX CANDIDATE C&A SYSTEMS → CONDUCT TRADE-OFF STUDY SUMMARIES AND SELECT TWO REAR DRIVE C&A SYSTEMS → EVALUATE REAR vs FRONT DRIVE SYSTEMS AND MAKE RECOMMENDATION FOR DETAIL DESIGN → OBTAIN APPROVAL OF CONTRACTING OFFICER → PHASE II

Requirement:

Analysis and Mechanical Design Studies of Six C&A Systems To Provide Trades To Reduce Candidates to Two Systems

Approach:

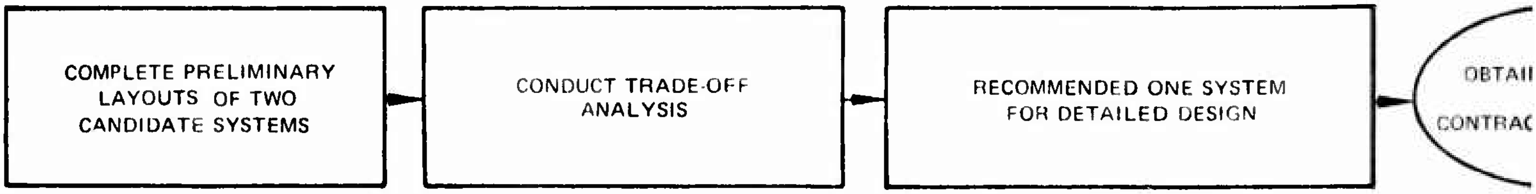
- Perform Mechanical Integration Studies To Define Interfaces
- Prepare Engineering Sketches of Possible Interface Techniques
- Perform Preliminary Analysis To Size the Candidate C&A Components
- Evaluate the C&A Systems To Determine Trades
  Criteria Shall Include:
  1. Reliability
  2. Vulnerability Resistance
  3. Development Risk
  4. Cost
  5. Weight and Volume
  6. Performance
  7. Maintainability
  8. Installation Flexibility

Requirement:

Select Two C&A Systems for Preliminary Design Based on Analysis and Mechanical Design Integration Efforts

Approach:

- Determine Trade Study Rating Factor
- Evaluate Candidate C&A Systems
- Prepare Summary Charts of Ratings
- Select Two Systems
- Review with Army Technical Representative



Requirements:

Provide Preliminary Design Layouts That Define Packaging Interfaces, Volumes, Environment, and Mechanical Implementation of the Two Candidate C&A Systems

Approach:

- Obtain the Design Characteristics of Each Control and Accessory
- Define Engine Interfaces for Selected C&A Systems
- Define Required Engine Configuration Changes
- Prepare Two Preliminary Layouts

Requirements:

Perform Design Analysis and Trade Studies Necessary To Evaluate Relative Merits of Each System

Approach:

- Evaluate the Performance Characteristics
- Review Layouts To Determine C&A Integration Impact on Engine Design
- Evaluate the C&A System Based on the Following General Performance and Design Criteria:
  1. Reliability
  2. Vulnerability Resistance
  3. Development Risk
  4. Cost
  5. Weight and Volume
  6. Performance
  7. Maintainability
  8. Installation Flexibility

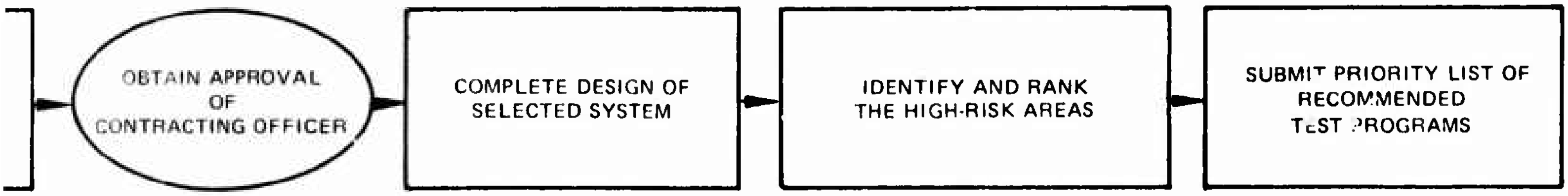
Requirements:

Select One C&A System Based on the Trade Study Results Review Selection With Designated Technical Representative

Approach.

- Apply Rating Factors to Trade-off Study Results
- Prepare Trade Study Results and Selection Information for Technical Review

Figure 3. Preliminary Program Outline for Phase II - Detail Design Phase



Requirements:

Continue Layout Design To Further Define the Final Integrated C&A System

Approach:

- Perform Detailed Analysis of Engine Interface
- Complete Design Layout of Critical Interfaces
- Complete Component Design Layout

Requirements:

Identify High Risk Components or Subsystems and Rank in Order of Priority

Approach:

- Critique Final C&A System to Identify High Risk Components and Subsystems
- Review High Risk Items With C&A Vendors to Determine Any Related Technology Programs

Requirements:

Prepare and Submit Test Program Plans in Priority Listing

Approach:

- Determine Tests Required for Experimental Evaluation of High Risk Components and Subsystems
- Define Required Rigs for Tests
- Provide Cost Estimates and Recommended Test Programs

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3. Fabrication and Test Phase

1. Test rigs to allow the selected tests to be conducted were fabricated. Test rigs and C&A hardware in existence from previous contracts were reviewed and utilized to the fullest extent possible.

2. The component or subsystem functional and endurance tests were conducted in accordance with the approved test plan to fully evaluate the high-risk aspects of the component or subsystem. Critical interface conditions for the components which were identified in the Detailed Design Phase were simulated.

3. The initial test results were evaluated, and the components were modified and retested as required for full evaluation.

4. After completion of all tests, the test results were evaluated for relative success or failure to meet the previously established performance and endurance goals.

SECTION II

REQUIREMENTS DEFINITION - FRONT-DRIVE POWER TURBINE

A baseline engine configuration was established as a starting point for the C&A optimization studies. The detailed engine requirements are outlined in the engine description, included in Appendix A. The engine requirements are summarized below:

- 3 lb/sec Airflow
- 600-hp Power Turbine
- 1977 Development Time Frame
- Dual Engine Installation
- Front Drive Power Turbine
- 15 hp, 20,000 rpm, PTO
- Emergency Oil System - 6-min Capacity

A baseline was derived from previous and current small engine design activities at FRDC. The engine technology was selected to be consistent with the development time frame.

A cross section of the basic engine without controls and accessories is shown in Figure 4. Since it was the intent of this study to optimize the overall C&A system, this layout was used only as a starting point for the study to establish the basic engine component sizes, flowpath, and bearing configurations. The engine design characteristics are outlined below:

- 65,000 rpm - Gas Generator
- 26,000 rpm - Power Turbine
- 2350°F - Turbine Inlet Temperature
- 10:1 Pressure Ratio - Compressor
- Semireverse Flow, Ejector Type - Inlet Particle Separator (IPS)

The engine configuration consists of a single-stage centrifugal compressor, an annular combustor, and a single-stage, cooled, axial turbine. The power turbine is a two-stage, uncooled, axial turbine with the shaft concentric with the gas generator rotor.

The centrifugal compressor has a separate inducer, pipe diffusers, and a variable inlet guide vane assembly. The centrifugal compressor offers the advantage of being less vulnerable than an axial or axial/centrifugal design.

The combustor is a full-annular, radial-inflow design using advanced cooling techniques and air-atomizing fuel nozzles. Combustor liner cooling is accomplished through the use of a finned, double-wall-construction technique, FINWALL®.

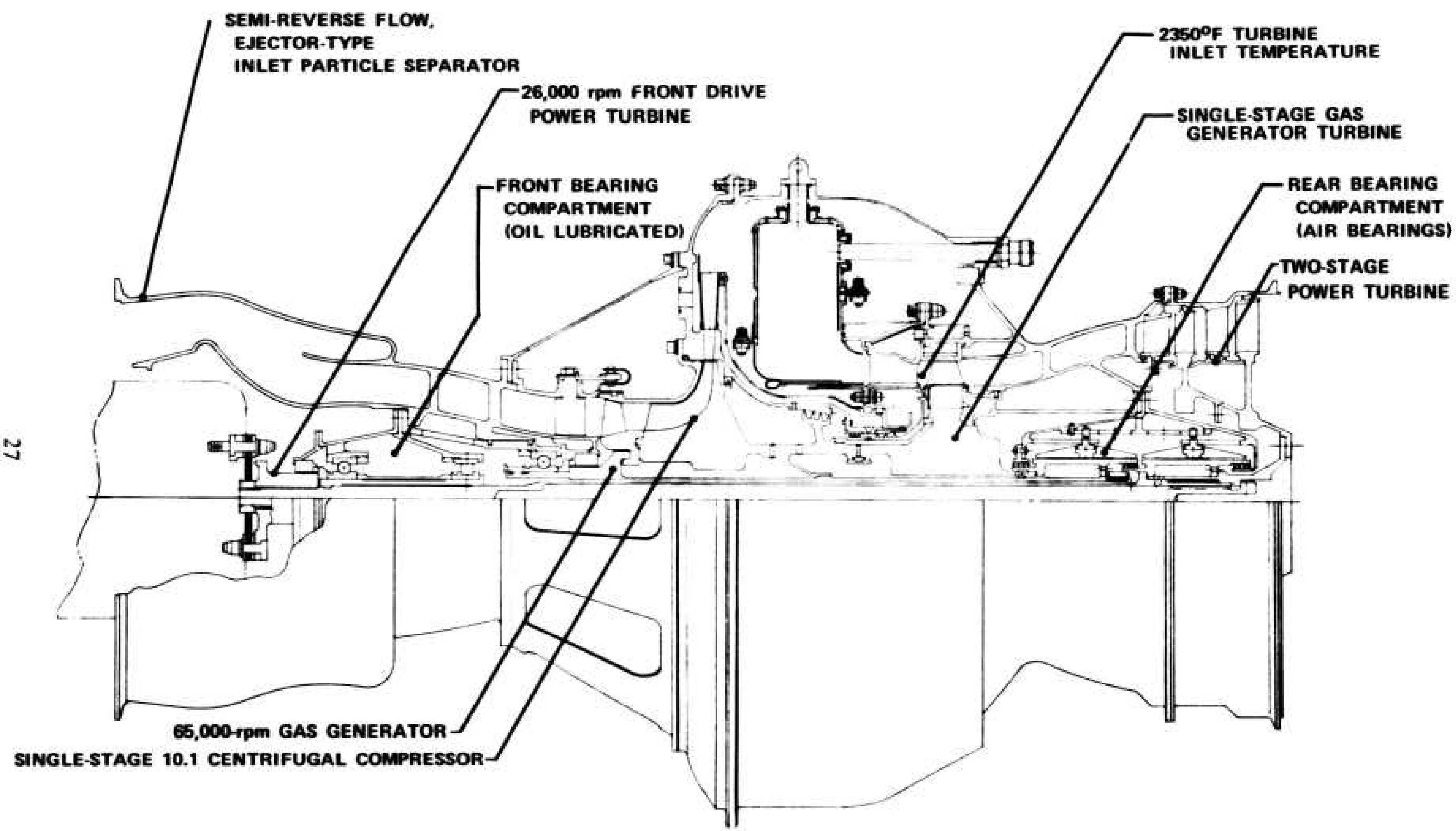

Figure 4. Baseline Engine Configuration

The gas generator turbine consists of a single high-work stage, with convectively cooled vanes and blades, and includes a thermal-response rotor tip shroud to control tip leakage. The control ring structure that surrounds the turbine blade row also adds vulnerability protection.

A two-stage, axial power turbine is necessary for acceptable engine performance with a front drive engine of this size. The rotor is limited in allowable operating speed due to critical speed of the power turbine shaft; therefore, two stages are necessary to obtain the desired turbine efficiency.

1. Bearings and Seals

An engine bearing and seal configuration, consistent with the 1977 development time frame, was established. The system was configured for low heat generation to minimize the size of vulnerable heat exchanger components. A backup oil system, with a 6-min emergency operational capability, was required. The recommended baseline engine bearing and seal configuration is described as follows:

1. Front Bearing Compartment

   Oil-Lubricated, Antifriction Bearings (3)
   Hydrodynamic Lift-off Seals (2)
   Carbon Intershaft Seal
   Elastomeric Damper

2. Rear Bearing Compartment

   Tilting Pad Air Bearings (2)

3. Heat Exchanger

   Air/Oil Heat Exchanger - Integral with front bearing compartment
   Oil Temperature - 200 to 250°F

4. Backup Oil System

   Oil Mist - 6-min capability

The hybrid air bearing/oil-lubricated bearing configuration was selected, based on the requirement that some type of oil-lubricated gearing to the gas generator shaft in the front compartment would most likely be required. Additionally, the oil-lubricated bearings have a greater capacity for the radial load that can be imposed by misalignment of the power turbine shaft with the airframe gearbox.

Air bearings are planned for the rear compartment to reduce engine vulnerability by eliminating oil system components and to reduce the size of the oil heat exchangers. A typical air bearing configuration, using tilted pad air bearings, is shown in Figure 5. Mechanical Technology, Incorporated, has successfully tested tilted pad air bearings for stop-start and high-temperature material evaluation.(1) Other manufacturers have successfully evaluated foil-type bearings. The final air bearing configuration will not be established as a part of this program, but the present state of the art indicates that, with a continuing research and development effort, air bearings are a reasonable risk for an engine starting development in 1977.

Air bearings in the rear compartment can operate at higher temperatures than the oil-lubricated system and, therefore, simplify the overall cooling and the rear compartment insulation requirements. The airflow requirements for the bearings are reduced by the higher operating temperatures due to the viscosity effects. Bleed air requirements are estimated at 0.05 to 0.10 lb/sec.

Hydrodynamic lift-off seals will be used where the available envelope allows. This will significantly reduce the oil cooling heat load due to elimination of the rotational friction. The rotating intershaft seal is proposed as a conventional carbon face seal because of the limited envelope available.

Hydrodynamic lift-off seals (shrouded Rayleigh step lift pads operated with a gas film separating the sealing faces) have been designed, fabricated, and tested at P&WA.(2) Tests have demonstrated the feasibility of operation at gas temperatures to 1200°F, pressure differentials to 300 psi, and sliding speeds to 500 ft/sec. Conventional contact seals are used in place of labyrinth seals when air and oil leakage is a problem. Their disadvantage is high wear.

Elastomeric dampers are recommended to replace oil film dampers to improve engine vulnerability. In-house IR&D testing of elastomeric dampers indicates adequate life, better tip clearance control, more design freedom with stiffness and damping coefficients, and better operation with rotor unbalance that might be caused by combat damage.

Engine heat generation has been reviewed in sufficient detail to establish the need for an air/oil cooler. A fuel/oil cooler will not have sufficient capacity at flight idle. Two air/oil cooler configurations were considered: (1) a heat exchanger located on the inlet OD that uses channeled pressurized oil, and (2) a heat exchanger located on the inlet ID that uses oil mist on the front compartment wall. The first approach uses a separate air/oil heat exchanger and transfers the heat to the air that is bypassed in the IPS. The recommended system uses the front bearing compartment as an integral heat exchanger and transfers the heat into the main engine airstream.

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(1) Swenson, K. R., N. M. Hughes, and D. F. Hever, EVALUATION OF GAS LUBRICATED HYDRODYNAMIC BEARINGS IN A GAS TURBINE ENVIRONMENT, Report No. AFAPL-TR-72-41, June 1972.

(2) Povinelli, V. P., and H. H. McKibbin, DEVELOPMENT OF MAINSHAFT SEALS FOR ADVANCED AIR BREATHING PROPULSION SYSTEMS, Phase I, Report No. NASA CR-72737, PWA-3933, and Phase II, NASA CR-72987, PWA-4263.

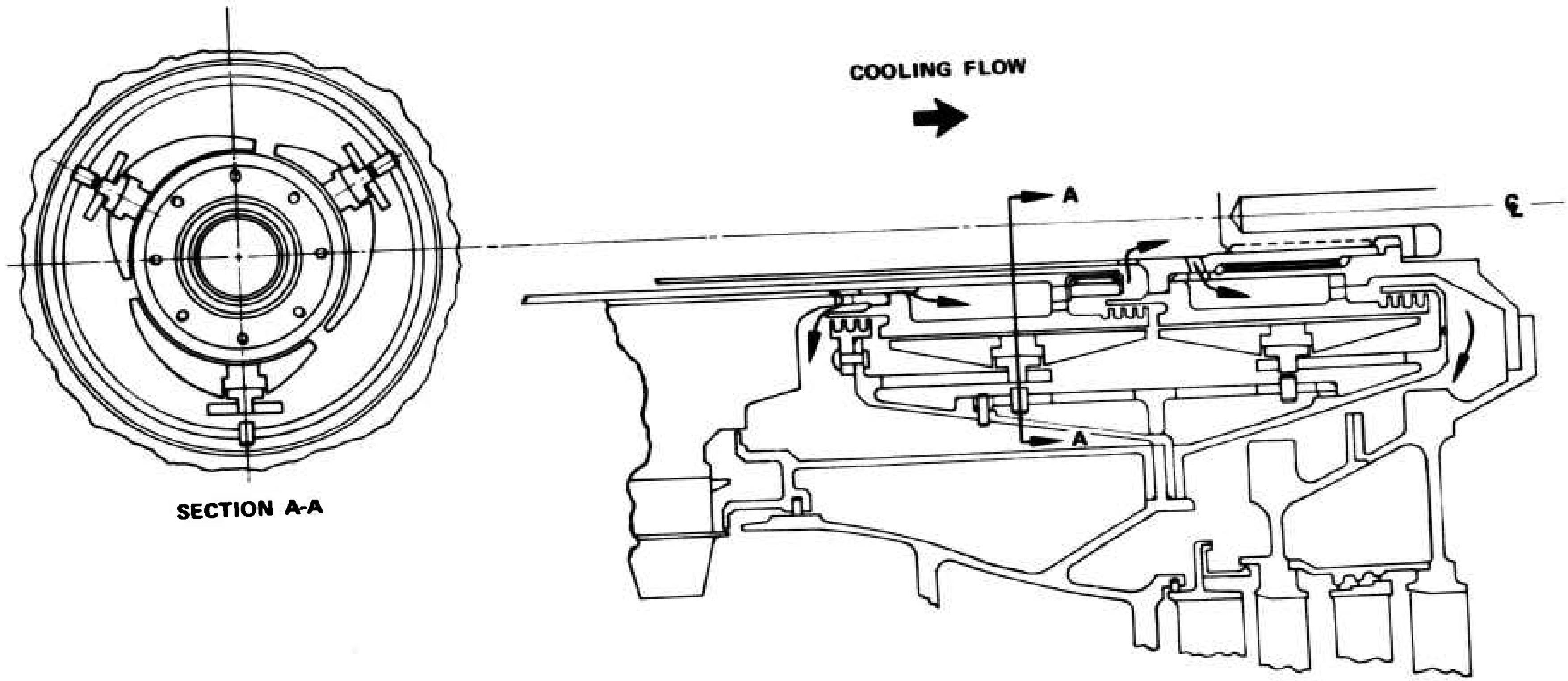

Figure 5. Tilting Pad Air Journal Bearing Rear Compartment Conceptual Design

Use of the finned inlet ID concept results in a performance penalty. A 3.2°F rise in average inlet temperature was calculated at flight idle. A 1°F rise in T_{t2} will increase SFC 0.13% and decrease SHP 0.4%. These values are based on a bulk temperature rise.

An emergency air/oil mist system is recommended to further reduce vulnerability. Emergency air/oil mist systems have been tested and proved feasible for 6-min operation.

The problems with air/oil mist are:

1. Determining the proper oil and air mixture for each bearing.

2. Cooling the inner races more effectively than the outer to control thermal distortion.

3. Scavenging air from the compartment.

4. Determining proper oil mist line velocity to avoid problems of condensation.

A simplified oil system that is used only in the front compartment is more amenable to an oil mist backup system than an engine with multiple oil-lubricated bearing compartments.

A detailed analysis of an air/oil mist lubrication system is presented in Section VII.

2. Inlet Particle Separator

The IPS is a semireverse flow separator, which is integrated into the compressor inlet case and uses scavenge airflow from a hot air ejector incorporated in the engine exhaust nozzle. A schematic of this system is shown in Figure 6. The fixed-geometry tailpipe ejector requires no valves or blower. The use of preswirl vanes is also not required. The design of the ejector must be carefully coordinated with the power turbine and IR suppressor designs to assure proper ejector performance over the operating range.

Current development work on erosion-resistant coatings for engine components indicates potential application in advanced engines.[3] The optimum engine dust protection system may incorporate both a simple IPS and basic erosion-resistant protection to the engine components.

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[3] McAnally, W. J., III, EROSION RESISTANT COATINGS FOR TITANIUM, Pratt & Whitney Aircraft, AMMRC CTR73-6, January 1973.

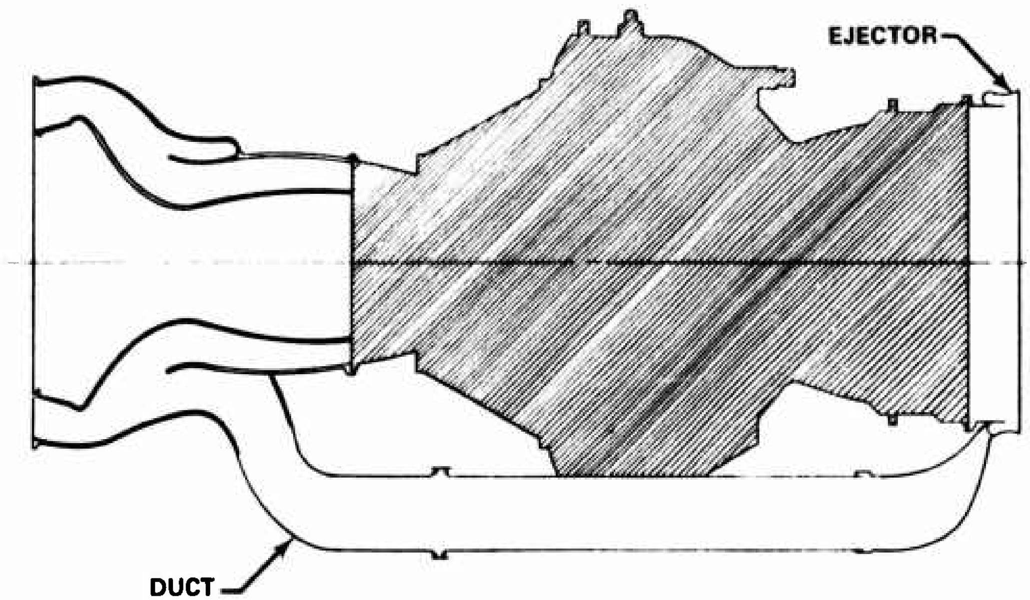


Figure 6. Semireverse Flow Inlet Particle Separator

A detailed design of the IPS was not established during this program. The IPS design was considered flexible to accommodate integration of the C&A components. Partial inlet or radial inlet IPS configurations, as shown in Figure 7, were considered.

3. Starter

The engine starting horsepower requirements were analyzed in detail, since they have a significant impact on the C&A system design. The engine rotor parasitic losses were based on applicable rig data and are shown in Figure 8. The minimum engine starting speed was based on a prediction of where stable compressor operation would be expected after ignition and is illustrated on the representative compressor map shown in Figure 9. The motoring torque for the engine for standard and cold day is shown in Figure 10. The starter was sized for a 2 ft-lb torque margin above the cold day requirements. The relationship of starter horsepower to light-off speed is also shown on Figure 10. For a sea-level cold-day start, the starter size was established at 16 hp at the light-off speed of 16,000 rpm. The required torque-speed relationship is shown in Figure 11. The starter was sized to accomplish a 30-sec, sea-level, standard-day start time.

4. Controls and Accessories

The requirements for the C&A components were established, and a preliminary definition of the component configuration was made to aid in the overall system selection. The Chandler Evans Corporation (CECO) was subcontracted to support definition and design of the control system components. The CECO effort under Contract DAAJ02-70-C-0002, Advanced Engine Control Program for 2 to 5 lb/sec Airflow Engines, was used to guide the control component designs, where applicable. The following C&A components are required:

Fuel Control and Sensors
Fuel Pump
Inlet Guide Vane Actuator
Compressor Bleed Valve
Anti-Ice Valve
Alternator
Ignition System
Starter
Lubrication System

A detail description of the component requirements and the component implementation is presented in Section VII.

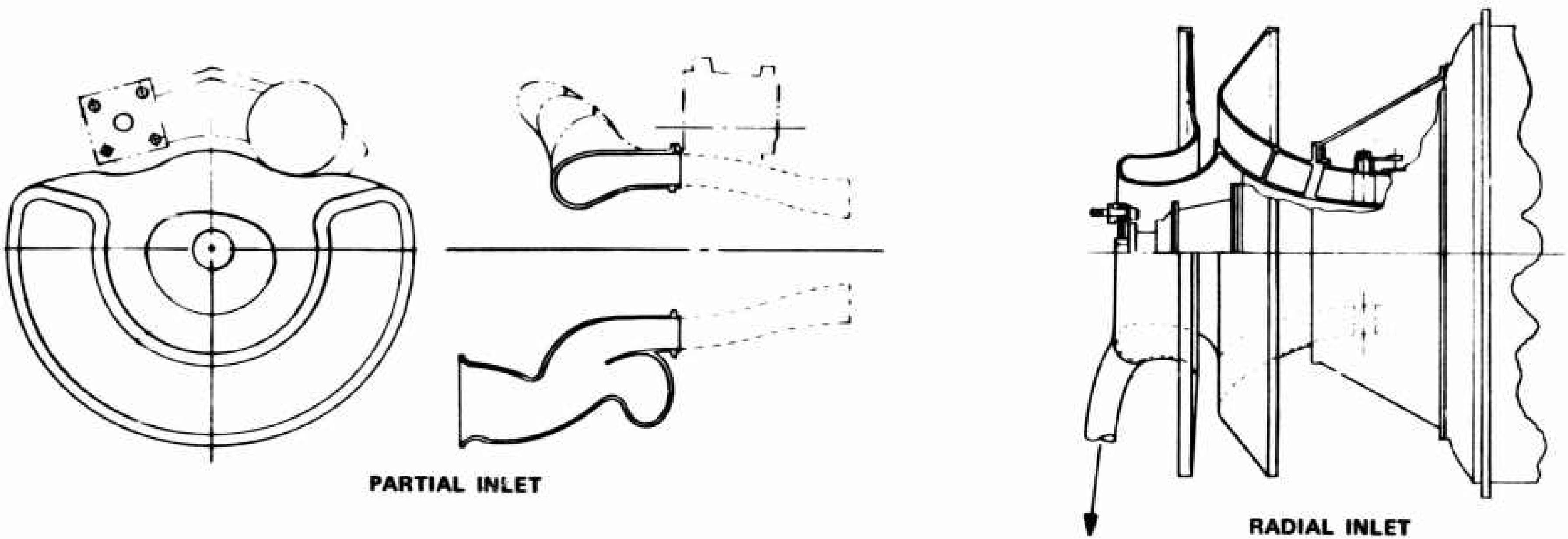


Figure 7. Inlet Particle Separator Alternate Inlet Configurations

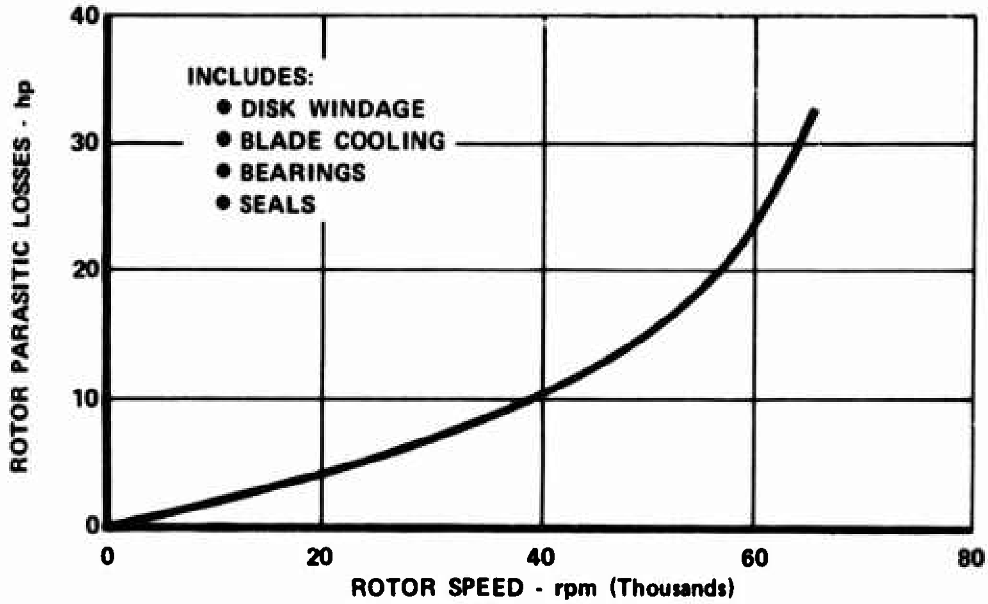


Figure 8. Engine Rotor Parasitic Losses

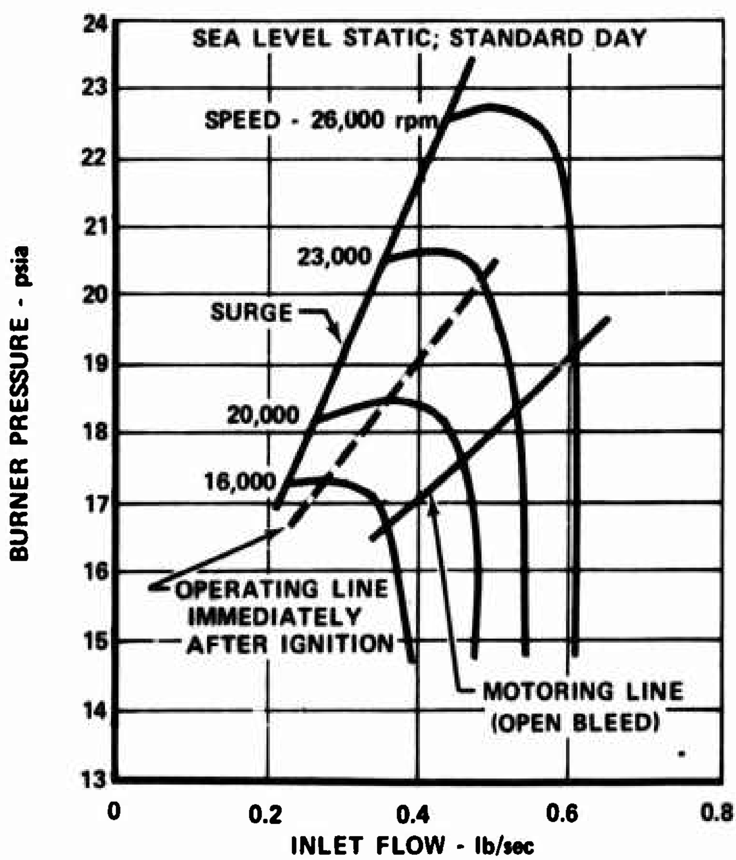


Figure 9. Compressor Map During Starting

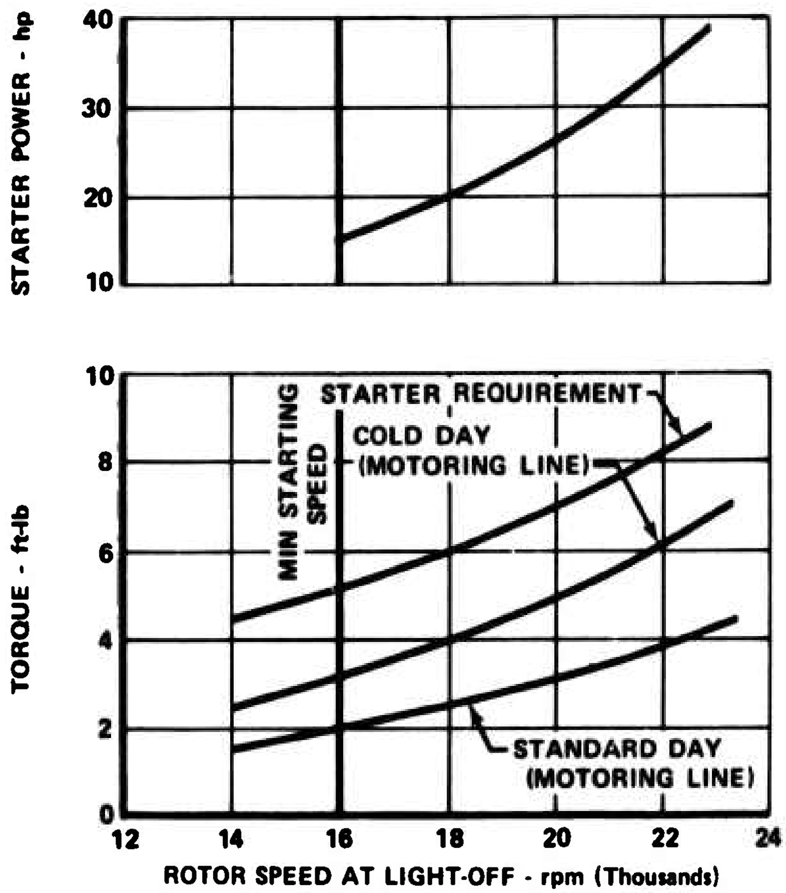


Figure 10. Torque and Horsepower vs Ignition Speed

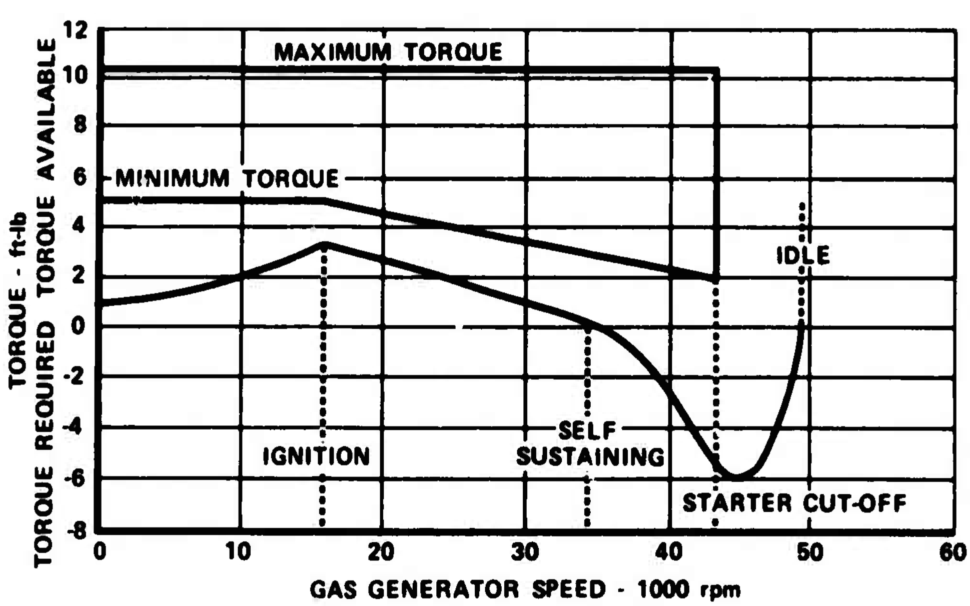


Figure 11. Starting Torque and Speed Requirement
(Sea Level/Cold Day)

SECTION III

SELECTION CRITERIA FOR CANDIDATE SYSTEMS

The following general performance and design criteria were utilized during the concept selection and design phases of this program as they applied to the overall engine C&A system under consideration and are presented in order of their estimated relative importance.

Reliability - The reliability (safety, mission and unscheduled maintenance) of the system considering the actual operational environment of U. S. Army helicopters.

Vulnerability Resistance - Of primary consideration is the vulnerability to small-arms fire. In addition to actual direct damage, the effects of secondary damage, fire and overall survivability were considered.

Development Risk - This is a judgment that was based on the present state of the art of the component being considered and its projected rate of development (assuming a moderate effort is specifically applied) over the total time frame involved, i.e., formal development initiation in 1977.

Cost - Projected production cost assuming a rate of 500 units per year over a total period of 10 years.

Weight and Volume - Installed weight and volume of the engine C&A system.

Performance - The ability of an individual component to perform its required function(s) and its effect on overall system performance.

Maintainability - The relative ability of an individual component or complete system to be maintained in an operational condition in the least time, at the least cost and with a minimum expenditure of support resources.

Installation Flexibility - The characteristics of the C&A system which provide the least overall design complexity relative to the engine's installation into a flight vehicle.

These criteria were established as a combined effort of the Army, P&WA, and CECO and then assigned weighted values on the basis of recommendations of the personnel associated with the program. The final weighted values for the criteria are outlined below.

Criteria	Weight - Percentage
Reliability	23
Vulnerability	19
Development Risk	14
Cost	12
Weight and Volume	10
Performance	9
Maintainability	7
Installation Flexibility	6
	100

The safety-oriented factors, reliability and vulnerability, received weighting which accounted for 42% of the total. Reliability in a manned aircraft is, of course, of prime importance and received a strong consideration in this analysis. Vulnerability in a manned aircraft that is used for a military application that has a high exposure to ground fire is another prime consideration, and in this case was rated approximately 80% of the value assigned to reliability.

The cost-oriented factors, development risk and production cost, received a total of 26% of the total weighting, with cost rated slightly below development risk. The development cost was reflected, to some degree, in the development risk evaluation. While these considerations did not consider all of the factors relating to life-cycle cost, it is not anticipated that a complete life-cycle cost analysis, which was beyond the scope of the program, would have influenced the overall ratings.

The mission effectiveness oriented parameters, weight and volume, and performance, received 19% of the total and were rated approximately equal. The operational oriented factors, maintainability and installation flexibility, received 13% of the total, with flexibility rated slightly below maintainability. Even though weight and volume, performance, maintainability, and installation flexibility received a total weight of only 32%, it is important to recognize that, while these criteria have a small weight in the basic system selection, their importance was not overlooked in the detailed design.

These general categories are summarized below:

	Rating Percentage
Safety	42
Cost	26
Mission Effectiveness	19
Operational Effectiveness	13

The primary goals of this program were safety and cost oriented, but the performance and operational considerations received more attention as the final system design was formulated.

To be considered a viable candidate, each system that was considered was required to exhibit a minimum level of performance in each criteria. For example, a system that caused an unrealistic engine performance (TSFC) penalty was not considered even though it was acceptable in the other areas. This step was necessary to preclude the possibility of selecting a system, based on established rating criteria, which would not be a realistic candidate.

SECTION IV

CONCEPTUAL DESIGN - FRONT DRIVE ENGINE

A. SELECTION OF TEN CANDIDATE SYSTEMS

The C&A drive systems considered for this program were divided into 17 schemes. The starter drive systems were broken into two basic divisions: mechanical drive input and integral. A matrix of the candidate systems is shown in Table 1. The matrix consists of the 17 C&A drive schemes combined with the two basic starter drive systems, including eight types of starter drive schemes. The total combination of C&A drive and starters creates 129 possible systems for study.

Each system was reviewed and several were logically eliminated. The remaining systems required analysis to allow a decision to be made as to whether the concept should have further consideration. The justification used for screening each system is specified on the matrix and summarized in Table 2. The 10 candidate systems selected for further evaluation are shown in Table 3.

1. Candidate C&A Drive/Starter Systems

During the conceptual design phase, the method of driving the C&A components and the method of starting the engine were emphasized. The C&A drive systems and the starter systems originally considered in the study are described below:

1. Candidate C&A Drives

   a. Mechanical

      (1) Tower Shaft Drive - Tower shaft drive through a gearbox with multiple gearing for the various required accessory drives

      (2) Cluster Gearbox - Cluster gearbox mounted about centerline of engine with multiple gearing for the various required accessory drives

      (3) Single-Speed Module - Tower shaft drive through a gearbox with a single drive shaft. All required accessory drives run a the same speed from a common shaft.

   b. Electrical

      (1) Integral Starter/Generator - Starter and generator use same winding, housings, etc., and are integrated into the gas generator rotor. The electric generator can drive one or multiple motors for the required accessory drives

      (2) Integral Generator - Electric generator is integrated into the gas generator rotor and provides power for the required accessory drives. Other starting means must be employed.

TABLE 1. COMPONE[illegible] ACCESSORY DRIVE SYSTEMS MATRIX

Starter Drive Systems		I Mechanic a Tower Shaft Gearbox	I Mechanic b Cluster Gearbox	I Mechanic c Tower Shaft Single Speed Module	II Electrical a Integral Starter/Generator	II Electrical b Integral Generator	III Pneumatic a Cold Gas Bleed	III Pneumatic b Bleed and Burn	III Pneumatic c Interturbine Bleed	III Pneumatic d Mixed Bleed	IV Hydraulic a Integral Hydraulic Pump	V Hybrid a I, III, or IV + Integral Generator	V Hybrid b IA + Electric Generator/Motor	V Hybrid c IA + Pneumatic Compressor/Motor	V Hybrid d IA + Hydraulic Pump/Motor	V Hybrid e Auxiliary Power Unit	V Hybrid f IA + Electr. Generator for Fuel Pump	VI Integral a Integral Fuel/Oil Pump + Alternator
Mechanical	1. A. Electric 1. APU/Generator 2. Battery	1	1	2-D	N/A	2-D	5-D	5-D	5-D	5-D	5-D	8-D	1	6-S	6-S	1	1	2-D
Mechanical	B. Hydraulic 1. APU/Hydraulic Pump 2. Accumulator Blowdown	1	1	2-D	N/A	2-D	5-D	5-D	5-D	5-D	5-D	8-D	6-S	6-S	1	7-D	1	2-D
Mechanical	C. Pneumatic 1. APU Bleed	1	1	2-D	N/A	2-D	5-D	5-D	5-D	5-D	5-D	8-D	6-S	1	6-S	7-D	1	2-D
Mechanical	D. Self Contained 1. Gas Turbine, 2. Piston Engine, 3. Wankel	1	9-S	2-D	N/A	2-D	5-D	5-D	5-D	5-D	5-D	8-D	6-S	6-S	6-S	7-D	1	2-D
Integral	2. A. Integral Electric	2-S	2-S	2-D 2-S	2-D 2-S	2-D 2-S	2-S	7-D 2-S	10-D 2-S	3-D 2-S	2-D 2-S	8-D 2-S	2-S 6-S	2-S 6-S	2-S 6-S	7-D 2-S	2-S	2-D 2-S
Integral	B. Integral Hydraulic	2-S	2-S	2-D 2-S	N/A	2-D 2-S	2-S	7-D 2-S	10-D 2-S	3-D 2-S	2-D 2-S	8-D 2-S	2-S 6-S	2-S 6-S	2-S 6-S	7-D 2-S	2-S	2-D 2-S
Integral	C. Integral Pneumatic 1. Cold Gas Impingement 2. Engine Ram/Closed Inlet, 3. Hot gas Impingement	1	1	2-D	N/A	2-D	1	7-D	10-D	3-D	2-D	8-D	6-S	6-S	6-S	7-D	1	2-D
Integral	D. Cartridge	4-S	4-S	2-D 4-S	N/A	2-D 4-S	4-S	7-D 4-S	10-D 4-S	3-D 4-S	2-D 4-S	8-D 4-S	4-S 6-S	4-S 6-S	4-S 6-5	7-D 4-S	4-S	2-D 4-5

TABLE 2. IDENTIFICATION NOTES FOR THE STARTER - C&A DRIVE SYSTEM MATRIX

1.	Considered a candidate C&A/starter system
2.	Not technically feasible or considered beyond 1977 state of development
3.	Excessive TSFC penalty
4.	Logistic requirements not compatible with mission
5.	If a mechanical drive is required for start, no advantage to another drive technique for the accessories
6.	For hybrid systems, the starter and accessory drive modes should be common
7.	Excessive reliability penalty
8.	An integral generator for control power only offers no advantage if a separate accessory drive device is available
9.	Excessive packaging penalty for the available space
10.	Excessive vulnerability penalty
	S - Starter Drive D - C&A Drive

c. Pneumatic

(1) Cold Gas Bleed - Engine airflow gas bleed upstream of the burner is used for providing power for the required accessory drives.

(2) Bleed and Burn - Engine airflow gas bleed upstream of the main burner is fed through a separate combustor, mixed with a fuel supply, and burned. The combustion products are then used to provide power for the required accessory drives.

(3) Interturbine Bleed - Engine airflow gas bleed downstream of the gas generator turbine is bled and used to provide power for the required accessory drives.

(4) Mixed Bleed - Engine airflow gas bleed is bled from the engine, both upstream of the burner and downstream of the gas generator turbine. The hot and cold gases are mixed to provide power for the required accessory drives.

d. Hydraulic

Integral Hydraulic Pump - A hydraulic pump is integrated with the gas generator rotor, and the hydraulic power from the pump is used to provide power for the required accessory drives.

e. Hybrid

(1) Any C&A Drive/Integral Generator - All accessories are driven by mechanical, pneumatic, or hydraulic power with an electrical generator integrated with the gas generator rotor to supply electrical power for the control and ignition system only.

(2) Mechanical/Electrical Interface - A tower shaft is used for driving an electric generator. The generator is used for powering the required accessory drives.

(3) Mechanical/Pneumatic Interface - A tower shaft is used for driving a separate air compressor. The compressor is used for powering the required accessory drives.

(4) Mechanical/Hydraulic Interface - A tower shaft is used to drive a separate hydraulic pump. The hydraulic pump is used to power the required accessory drives.

(5) Auxiliary Power Unit - A self-contained auxiliary power unit supplies power for the required accessory drives.

(6) Mechanical Drive/Electrical Fuel Pump Interface - An electric generator, integrated with the gas generator rotor, will provide electric power for driving a variable-speed pump. All other accessories are driven by tower shaft.

f. Integral

(1) Integral Fuel Pump, Oil Pump and Alternator - All the accessories are integrated into the gas generator rotor and operate at gas generator speed.

TABLE 3. TEN CANDI

		Con			
		Mechanical		Pneumatic	
		A. Tower Shaft	B. Cluster Gearbox	A. Cold Gas Bleed	A.
Starters — Mechanical Starter Drives	A. Electric Motor 1. APU/Generator 2. Battery	1 ⓧ	3 ⓧ		
	B. Hydraulic Motor 1. APU Supplied Hydraulics 2. Pressurized Blowdown	ⓧ	ⓧ		
	C. Pneumatic Motor 1. APU Bleed 2. APU Bleed + Fuel/Combustor/Turbine Starter	ⓧ	ⓧ		
	D. Auxiliary Power Unit 1. Gas Turbine 2. Piston Engine 3. Wankel Engine	ⓧ			
Starters — Integral Starter Drives	A. External Pneumatic Supply (Cold Gas Impingement)	2 ⓧ	4 ⓧ	5 ⓧ	
	B. External Pneumatic/Fuel Supply/Auxiliary Combustor (Hot Gas Impingement)	ⓧ	ⓧ	ⓧ	
	C. External Pneumatic/Fuel Supply for Engine Ram/Closed IGV's	ⓧ	ⓧ	ⓧ	

Legend:

▭ - Basic

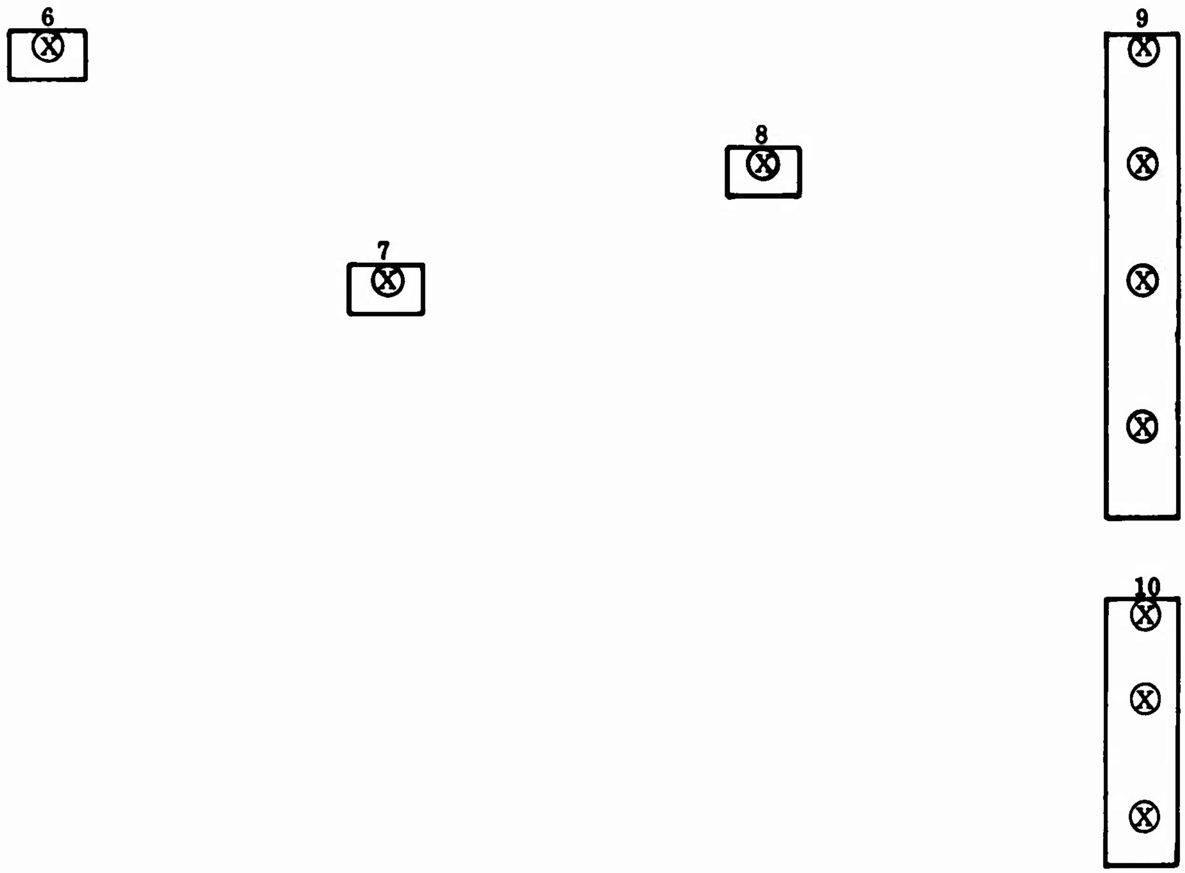
ⓧ - Deriv

:ANDIDATE SYSTEMS

Controls and Accessory Drives

Hybrid

A. Mechanical Gearbox/ Electrical Generator	B. Mechanical Gearbox/ Compressor	C. Mechanical Gearbox/ Hydraulic Pump	D. Mechanical Gearbox Driving Plus An Electrical Fuel Pump Drive



:

- Basic C&A Starter System
- Derivative Starter Configuration

2

2. Candidate Starter Systems

a. Mechanical

(1) Mechanical/Electric - An electric-powered starter driving a gearbox connected to the gas generator rotor. Starting power provided by APU/generator or battery system.

(2) Mechanical/Hydraulic - A hydraulic-powered starter driving a gearbox connected to the gas generator rotor. Starting power provided by APU/hydraulic pump or accumulator blowdown.

(3) Mechanical/Pneumatic - A pneumatic-powered starter driving a gearbox connected to the gas generator rotor. Starting power provided by APU bleed.

(4) Self-Contained Starter - A self-contained gas turbine, piston, or Wankel engine mechanically connected to the gas generator rotor.

b. Integral

(1) Integral Electric - An electric starter integrated with the gas generator rotor.

(2) Integral Hydraulic - A hydraulic starter integrated with the gas generator rotor.

(3) Integral Pneumatic - An external APU supplying bleed air, which is used in either of the three following methods: cold gas impingement, heat addition in the gas generator burner with closed engine inlet, and hot gas impingement using heat addition in an external burner.

(4) Cartridge - Solid grain hot gas generating device used for driving the gas generator rotor directly or through an intermediate system.

2. Preliminary Screening Analyses

The logic and analyses used to preliminarily screen the candidate C&A drive/starter system are summarized in the following section. These analyses reduced the candidate C&A drive/starter system to 10 basic categories. These 10 categories included 26 possible C&A drive/starter combinations. Each of the candidate C&A drive and starter systems considered is described below.

a. C&A Drives

(1) Mechanical

The candidate mechanical C&A drives considered were:

1. Tower Shaft - Considered a candidate and is shown in Figure 12.

2. Cluster Gearbox - Considered a candidate and is shown in Figure 13.

3. Tower Shaft/Single-Speed Module - The use of a 20,000-rpm, single-speed C&A module was discounted on the basis of incompatibility with the oil pump system. A single-stage 20,000-rpm oil pump element is not a valid mechanical configuration due to the disproportionate pumping element L/D (7:1). A two-stage oil pump, using a centrifugal inducer, was considered to be unduly complex when compared to a lower rpm single-stage positive-displacement pump. The centrifugal oil pump inducer is also subject to high blade loading at -65°F conditions.

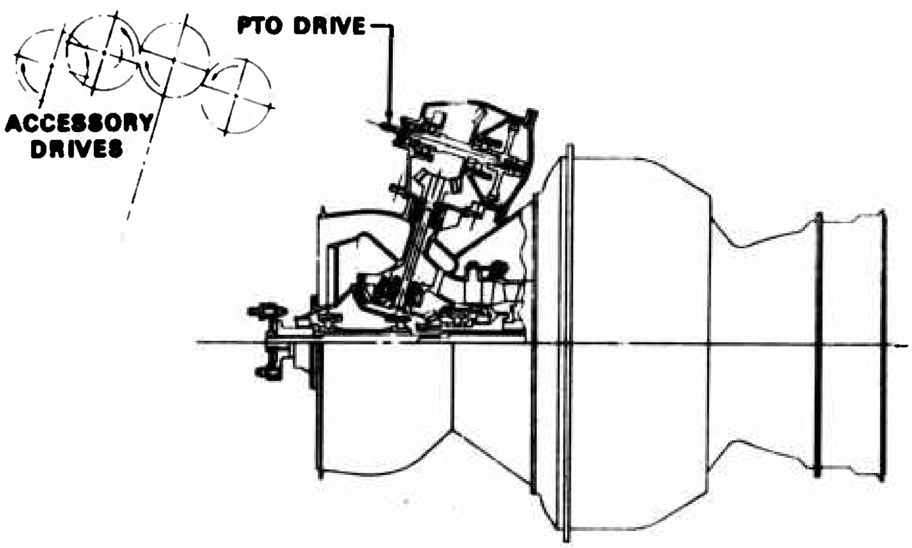


Figure 12. Candidate C&A Drive (Tower Shaft Drive)

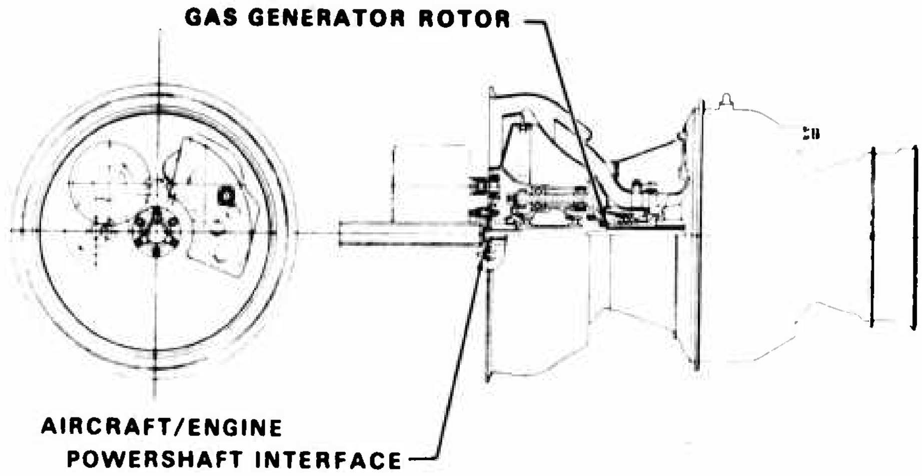


Figure 13. Candidate C&A Drive (Cluster Gearbox Drive)

(2) Electrical

The candidate electrical C&A drives considered were:

1. Integral Starter/Generator - The integral starter/generator illustrated in Figure 14 was discounted for the following reasons:

   a. Weight - The combined weight of the starter/generator, or generator, accessory drive motor, power processor, and engine cables was estimated at 54 to 58 lb, without cooling provisions, which is considered to be excessive for this engine size.

   b. Vulnerability - The physical volume of the starter/generator, motor, and power processor was estimated at approximately 500 in.^{3}, which would present a large exposed critical area in a vulnerability analysis.

   c. Tip Velocity - The tip velocity (60,000 ft/min) requirements exceed the current speeds used in aircraft electrical starter/generators and would require an advanced development effort for confirmation. Engineering development in this area is proceeding in the field of portable power packs using smaller diameter generator rotors operating at tip velocities of 63,500 ft/min.

      The development activity for this engine integration application is considered marginal with respect to the 1977 development time frame goal.

   d. Envelope - The envelope requirement dictates that the starter/generator or generator would have to be located upstream of the compressor, which would severely impact the inlet and IPS configurations due to the required diameter of 7.5 in. and the close axial location relative to the compressor inlet.

e. Maintenance - An integral starter unit would negate or severely complicate field removal and replacement of the starter unit.

f. Engine Critical Speed - The incorporation of the generator rotor would require moving the forward bearing support for the power turbine approximately 6 in. This would impose severe critical speed problems on the power turbine. Similarly, the incorporation of a 6-lb overhung mass (generator rotor) on the gas generator imposes critical speed problems on this system. Integration of a generator in this horsepower class is not considered to be technically feasible.

2. Integral Generator - The system was discounted for the same reasons outlined for the integral starter/generator, since the electrical component sizes are similar.

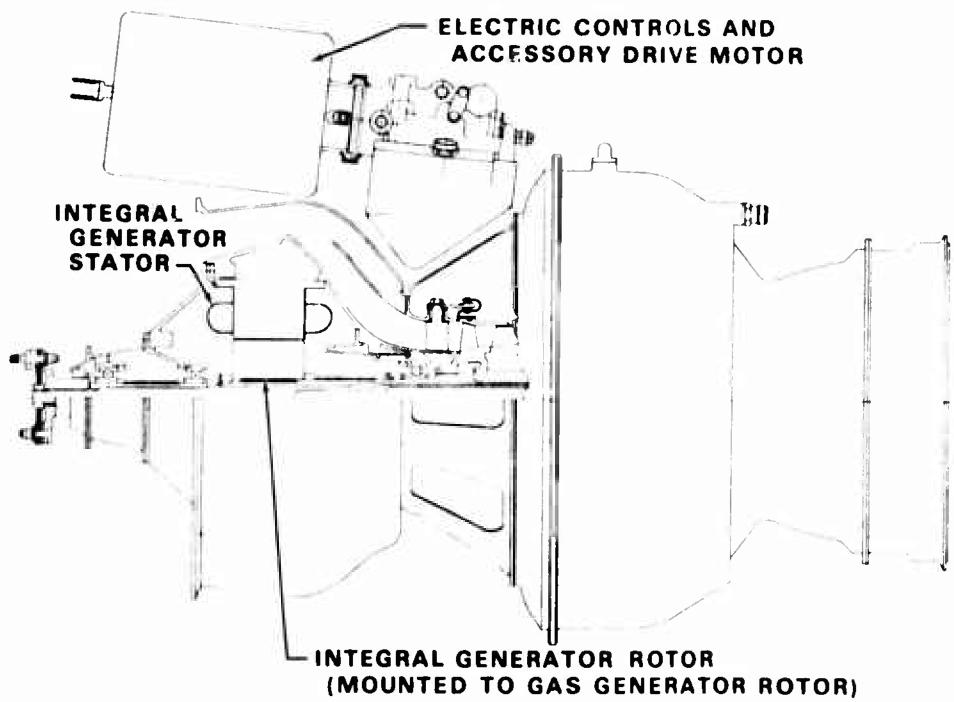


Figure 14. Integral Electric Starter/Generator

(3) Pneumatic

Pneumatic drive schemes were considered in combination with the integral starter systems only. The pneumatic drives in combination with a mechanical starter system were discounted on the premise that a mechanical drive system sized for the required starting power would be capable of a C&A drive, with no compromise in the drive system design. The study of the individual drive techniques is summarized below:

1. Cold Gas Bleed - Considered a candidate and is illustrated in Figure 15
2. Bleed and Burn - Discounted on the basis of complexity
3. Interturbine Bleed - Discounted on the basis of vulnerability
4. Mixed Bleed - Discounted on the basis of performance loss.

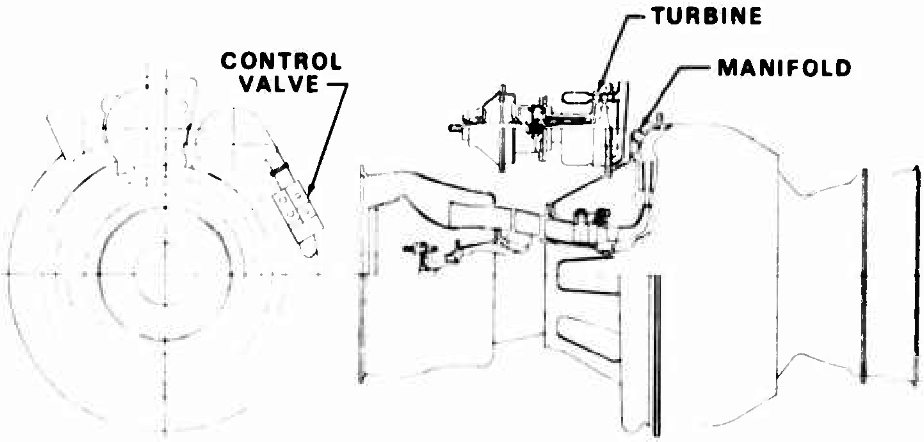


Figure 15. Candidate C&A Drive (Pneumatic Drive)

A discussion of the analysis follows.

The bleed systems, shown schematically in Figure 16, were analyzed at minimum and maximum engine power levels as follows:

1. Sea level ram/cold day (100% power) - (Maximum engine pressures, airflow, fuel flow, rotor speed, and turbine powers)
2. 20,000-ft ram/hot day (flight idle) - (Minimum engine pressures, airflow, fuel flow, rotor speed, and turbine powers).

The engine conditions at these points are summarized below:

Engine Power Condition	Engine Airflow, lb/sec	Compressor Pressure, psia	Compressor Temperature, °R	Interturbine Pressure, psia	Interturbine Temperature, °R	Ambient Pressure, psia
SLR/Cold (100%)	4.3	208	1110	63	2200	14.7
20,000-ft/Hot (Flight Idle)	1.25	21	925	10	1150	7.0

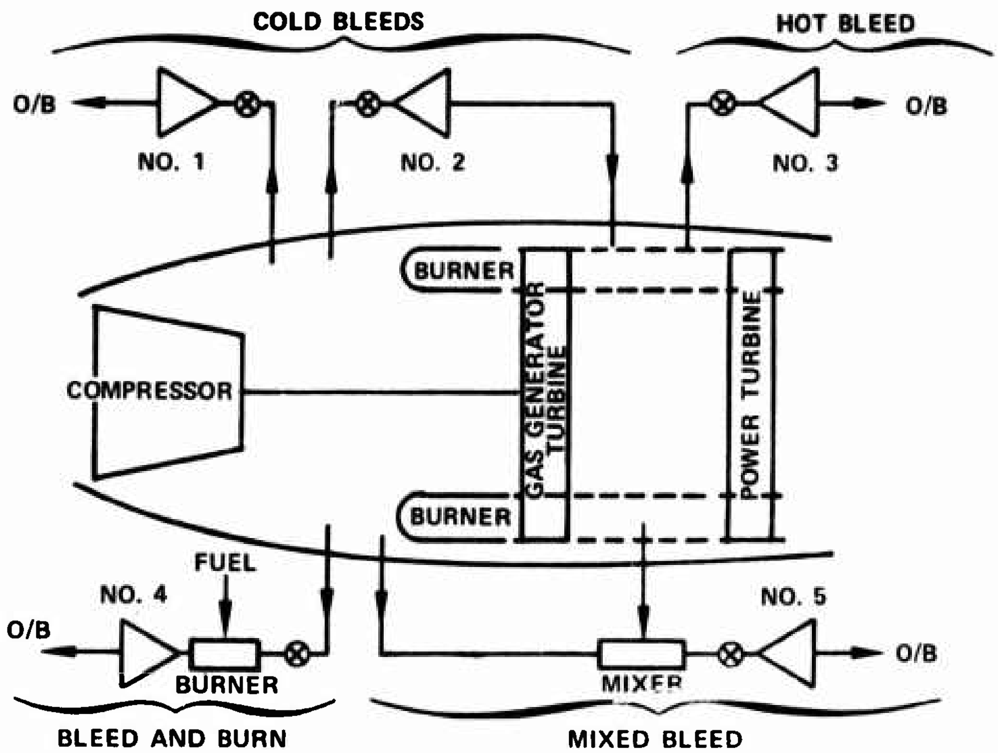


Figure 16. Candidate Pneumatic Drive Systems

The assumed accessory drive turbine requirements were: 25 hp at maximum engine conditions and 16 hp at minimum engine conditions. The program requirement of 15 hp for power takeoff at all flight conditions was the dominant factor in these requirements. Other ground rules for the accessory drive turbines are shown below:

Turbine Speed = 20,000 rpm at both flight conditions
Minimum $\Delta P/P$ for control valve and lines = 10%
Single-stage turbine
Maximum useful turbine $\Delta P/P$ = 4
Maximum turbine mean diameter = 5 in.
Minimum blade height = 0.30 in.

A turbine for driving the controls/accessories has a small power turndown (25 to 16 hp) between the two flight points investigated. This is in contrast to the gas generator engine turbine of 915- to 97-hp turndown or the power turbine of 860 to zero delivered horsepower. (Flight idle is full speed, but no delivered horsepower for the power turbine.)

Because of this trend, shown in Figure 17, the accessory turbines are sized at the low engine power point. Therefore, at the maximum power condition, the accessory turbine will deliver much more power than the required 25 hp. Throttling the accessory turbine system (with a control valve or variable admission turbine) to produce only the required 25 hp provides an inefficient turbine bleed system.

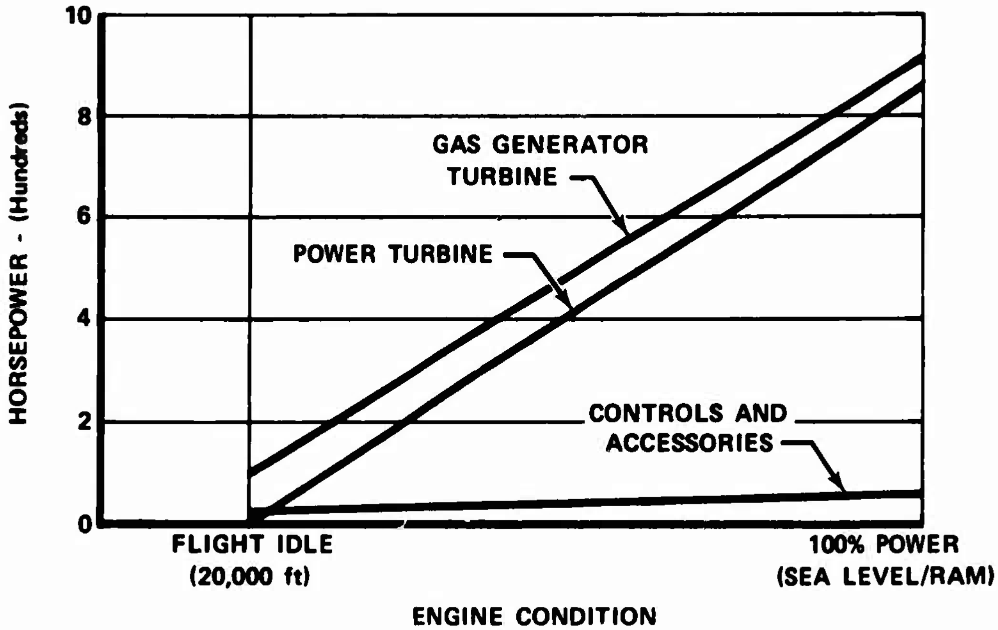


Figure 17. Turbine Horsepower Turndown Ratio

The required bleed flows and engine ΔSFC are given relative to a conventional tower shaft/gearbox for a constant power turbine delivered power and are shown in Table 4.

TABLE 4. TOWER SHAFT GEARBOX TURBINE BLEED PERFORMANCE

Accessory Drive	System Components	Turbine Flow, lb/sec	Engine ΔSFC, %
1. Cold (Exhaust Overboard)	Valve and Fixed Turbine	0.644	+ 8.3
2. Cold (Exhaust to Inter-turbine)	Variable Admission Turbine	0.875	+ 5.9
3. Interturbine	Variable Admission Turbine	0.465	+ 8.0
4. Bleed and Burn	Valve Fixed Turbine, Burner, and Fuel System	0.277 (air) +18.8 lb/hr (fuel)	+ 6.7
5. Mixed	Valve, Fixed Turbine, Mixer	0.40 (cold) 0.70 (hot)	+19.0

The two most promising systems of those investigated were (1) the cold bleed (exit overboard with a control valve) and (2) hot bleed (exit overboard with a variable admission turbine). The reasoning behind this selection was:

1. Cold Bleed With a Valve - It has a low performance penalty (+8.3% SFC) and small engine component changes, except for a 23% larger compressor. It is a conventional system, which has been used on other engines. With the flow exit overboard, no return line to the engine is required.

2. Cold Bleed (Exit to Interturbine) With a Variable Admission Turbine - Although this system has small engine effects, it is complicated with a variable admission turbine and has a vulnerable exit duct back to the engine interturbine.

3. Hot Bleed With a Variable Admission Turbine - It had good performance and small engine changes. It requires a variable admission turbine and has a vulnerable supply duct from the interturbine.

4. Bleed and Burn - This system had good performance and small engine changes. However, it is a complicated system with a separate burner, fuel system, and ignition system.

5. Mixed Bleed - This system's performance was bad and required two ducts.

The use of cold gas for a C&A turbine drive was considered to be more reliable and less prone to introducing secondary damage in the event of a hit, as compared to a hot gas system. In the event of failure or damage to the hot gas line, the 2200°R interturbine gas products would be discharged into the rear engine compartment and would provide a source of ignition to anything combustible in the area. The development risk and production cost of the hot gas components would be higher. The interturbine bleed system would have a slight performance advantage and might offer some weight advantage. Maintainability and installation flexibility would be similar for the two systems.

Therefore, the interturbine bleed was discounted in favor of a cold gas bleed based on the estimated superiority in reliability, vulnerability, development risk, and cost of the cold gas system.

(4) Hydraulic

Integral Hydraulic Pump - An integral hydraulic pump, shown in Figure 18, was not judged to be technically feasible on the basis of the minimum inside diameter imposed by the gas generator shaft. This physical limitation would require dynamic seal velocities of approximately 500 ft/sec and an excessive pump inlet pressure to suppress cavitation.

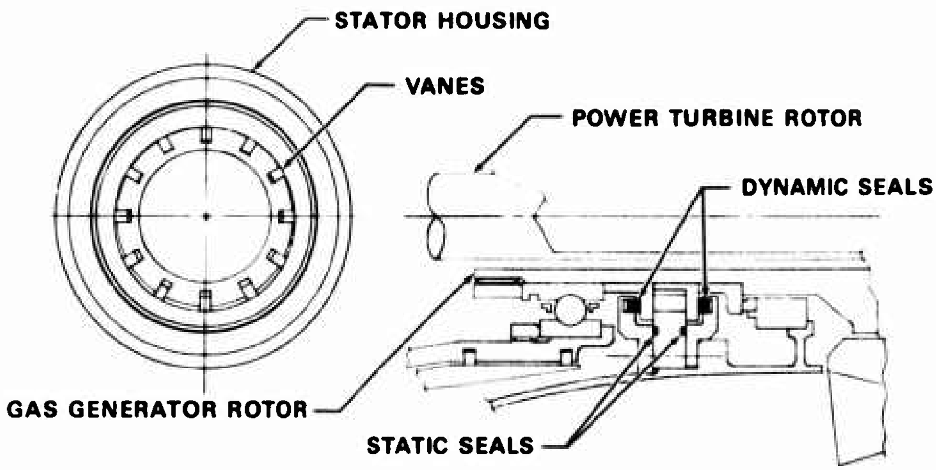


Figure 18. Integral Pumping Element

(5) Hybrid

A discussion of the hybrid systems considered is provided below:

1. Any C&A Drive - Integral Alternator - An integral alternator to provide 60w power for the engine ignition and electronic control was discounted. An integral generator was considered to be practical from an engine integration standpoint, but offered no vulnerability, weight, or reliability advantages over a generator that could be incorporated as part of a lower speed accessory or component. Separation of the generator from the electronic control unit was also not considered to be advantageous.

2. Mechanical/Electrical Interface - Considered a candidate with an external electrical starter.

3. Mechanical/Pneumatic Interface - Considered a candidate with an external pneumatic starter.

4. Mechanical/Hydraulic Interface - Considered a candidate with a hydraulic starter.

5. Auxiliary Power Unit - An auxiliary power unit for the C&A drives was discounted. This unit would consist of a JP-fueled, internal combustion engine that powered a C&A module. This system was eliminated on the basis of reliability, where the successful operation of the engine would depend on the functioning of two separate powerplants.

6. Mechanical Drive - Electrical Fuel Pump Interface - A variation of the mechanical drive systems was considered as a candidate; it used an electric generator/motor coupling for only the main fuel pump. This would simplify the flow control fuel bypass and subsequent heat rejection problems, by allowing an infinite variation of fuel pump speed to meet the engine requirements.

(6) Integral

Fuel and oil system pumping elements integral with the gas generator rotor were discounted because of the excessive inlet pressure requirements (up to 1000 psi) to suppress cavitation. The high inlet pressure requirement is a result of the 65,000-rpm operating speed and the tip speeds, which are established by the gas generator shaft diameter.

Oil pumps operating in a slinger configuration as an integral part of the engine rotors were briefly considered and discounted for the following reasons: (1) charging the center of the shaft(s) would require additional high velocity dynamic seals, which would add to the heat rejection and system complexity; (2) additional heat rejection to the oil would result from friction losses due to the relative shaft velocities; (3) there are technological unknowns with this pumping scheme, and they were considered beyond the development time frame for this contract.

b. Starters

(1) Mechanical

This category includes those starter systems that have a mechanical interface with the engine. These candidates, illustrated in Figure 19, are:

1. Electrical - Considered a candidate; APU/generator and battery supplies considered

2. Hydraulic - Considered a candidate; APU/hydraulic pump and accumulator blowdown supplies considered

3. Pneumatic - Considered a candidate; APU bleed and APU/external combustor supplies considered

4. Self-Contained Starter - Considered a candidate; will not be considered with the cluster gearbox C&A drive because of packaging. Gas turbine, piston and Wankel engines considered as candidates.

(2) Integral

This category includes those starter systems that do not require a mechanical interface with the engine. Three integral pneumatic systems were considered for further evaluation and are illustrated in Figure 20. A discussion of all the integral start system candidates is provided below:

1. Integral Electric - An electrical starter integral with the gas generator rotor was sized for this application and was approximately the same size as the integral starter/generator previously discussed. This system was discounted for the same reasons outlined for the integral electric C&A drive techniques.

2. Integral Hydraulic - An integral hydraulic starter was discounted because of the requirement for high-pressure dynamic seals on

the gas generator rotor. These seals would be required to direct the high-pressure, externally supplied fluid through a start turbine and would operate in the nonstarting mode at rubbing velocities of approximately 500 ft/sec due to the gas generator shaft diameter. The seals would require a lift-off feature, or would generate heat during engine operation above start and would require cooling. This overall system was not considered to be compatible with the development time frame.

3. Integral Pneumatic - Considered a candidate. Three integral pneumatic systems were established: cold gas impingement, cold gas supply using heat addition in the gas generator burner with a closed engine inlet, and hot gas impingement using heat addition in an external burner.

4. Cartridge - Cartridge starters for the primary starter mode were discounted because of the logistics involved in supplying starter grain assemblies. In addition, manual replacement of starter grains between starts or the development of a multistart cartridge grain feed system would be required. The potential hazard of carrying grain assemblies onboard a military aircraft was also considered as a disadvantage.

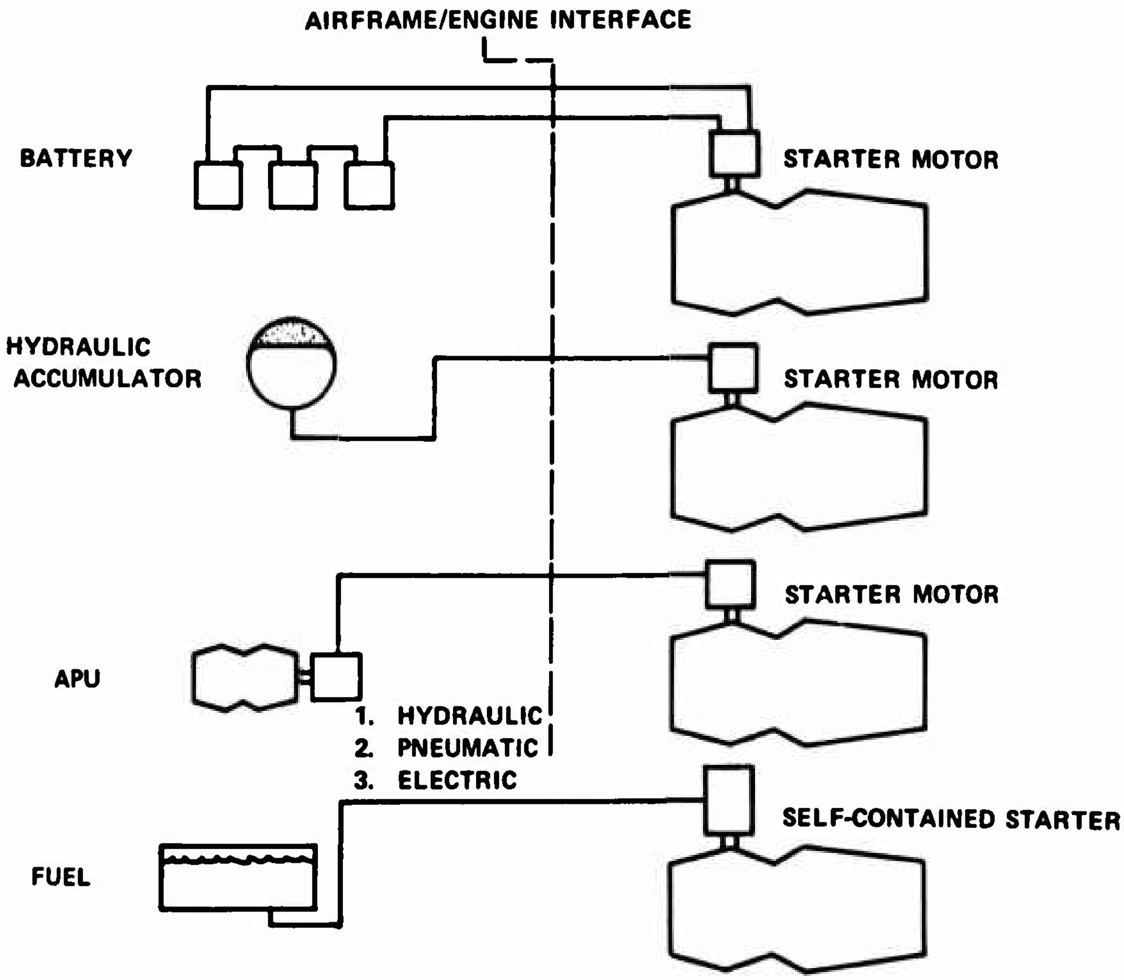


Figure 19. External Starter Systems

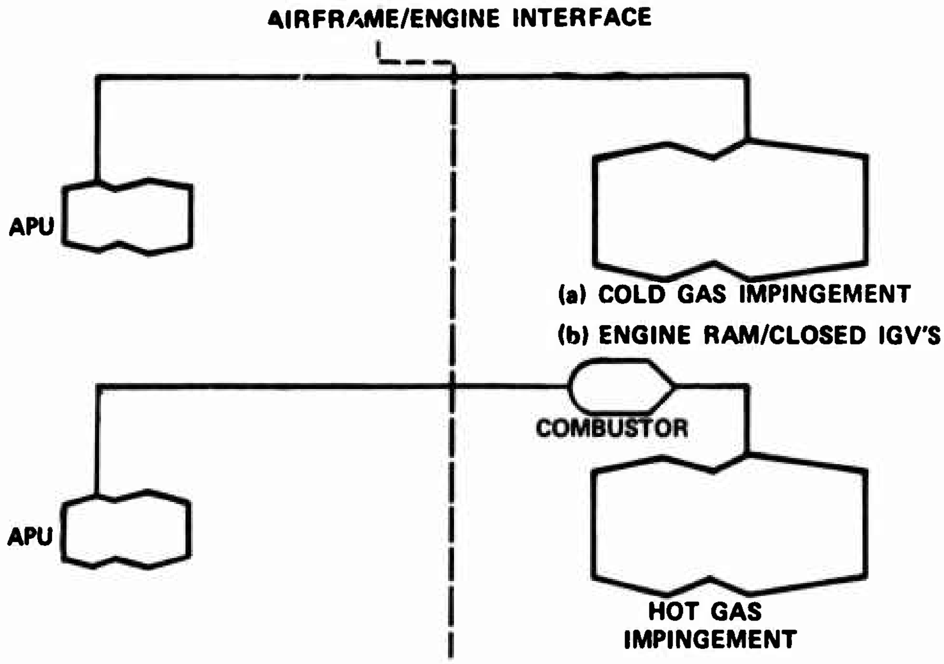


Figure 20. Integral Starter Systems

3. Selection of Five Candidate Systems

Of the 10 basic systems selected for further evaluation, there were 26 individual C&A/starter concepts. It was required to reduce the candidate configurations to five basic systems, using the established performance and rating criteria. The logic network used to screen the candidates is illustrated in Figure 21 and is summarized below:

1. Starters

   a. External

      - Select best airframe-powered starter (electric, hydraulic, or pneumatic)
      - Select best self-contained starter (gas turbine, piston or Wankel)
      - Compare and select best airframe-powered starter vs best self-contained starter.

   b. Integral

      - Select best integral start mode (cold gas impingement, hot gas impingement, engine ram/closed IGV's).

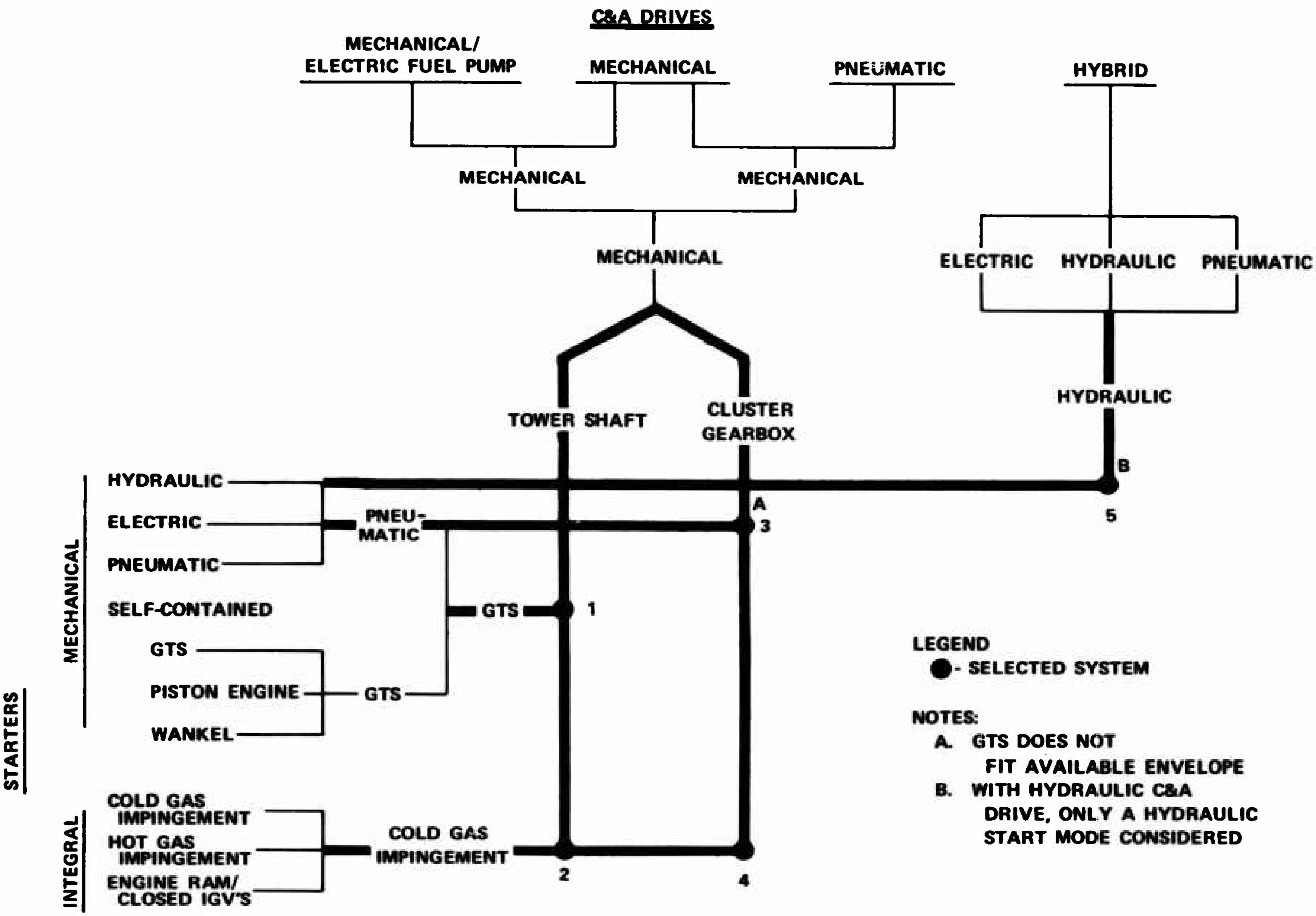


Figure 21. Logic Matrix for Selection of Five Systems

2. C&A Drives

- Select best of mechanical drive with electric fuel pump drive vs an all-mechanical drive.
- Select best of mechanical drive vs pneumatic drive.
- Select best hybrid drive (electrical, hydraulic, or pneumatic).

Based on the analysis, the five candidate systems outlined below were selected for further study:

- Tower Shaft Drive/Gas Turbine Starter
- Tower Shaft Drive/Cold Gas Impingement Starter
- Cluster Gearbox Drive/Air Turbine Starter
- Cluster Gearbox Drive/Cold Gas Impingement Starter
- Hybrid Hydraulic Drive/Hydraulic Starter.

The analysis is summarized in the following paragraphs.

a. External Starters

(1) Airframe-Supplied Power

A study was conducted to evaluate six different engine starter concepts for a twin-engine helicopter application based on the weighted assessment criteria previously established. The results of the study are summarized in Table 5 and show that the APU-pneumatic had the highest rating.

As indicated in Table 5, three of the starter concepts are powered by an auxiliary powerplant and use hydraulic, pneumatic, and electric starter motors, respectively. The fourth system uses a battery-powered electric starter and the fifth, an accumulator blowdown hydraulic system. The last system is a combustion/turbine starter, which requires a pressurized supply of air and fuel for the combustor, which has an integral turbine starter.

Table 6 summarizes the data obtained from the references and used in this evaluation. An explanation of how the assessment weightings were determined follows:

1. Reliability - MTBF's were computed for each starter concept by averaging the MTBF's given in the references for each component when more than one data point was available.

   The weightings were computed on the basis of giving the lowest MTBF system 23% (maximum weighting for reliability) and the other systems a lesser percentage proportional to their MTBF's.

TABLE 5. STARTER CONCEPT EVALUATION

Criteria	Weight Factor	Starter Rating: APU-Hydraulic	APU-Pneumatic	APU-Electric	Battery Electric	Acc. Hydraulic	Combustion Turbine
Reliability	23.0	11.3	23.0	15.0	18.3	15.0	18.3
Vulnerability	19.0	18.5	19.0	10.5	8.1	10.5	5.9
Development Risk	14.0	10.0	10.0	10.0	14.0	14.0	10.0
Cost	12.0	4.0	4.0	4.0	9.0	12.0	6.0
Weight and Volume	10.0	9.5	10.0	5.8	5.2	4.7	7.3
Performance	9.0	9.0	9.0	9.0	3.0	7.0	5.0
Maintainability	7.0	3.5	7.0	4.6	5.6	4.6	5.6
Installation Flexibility	6.0	3.0	3.0	3.0	6.0	4.0	2.0
Total	100.0	68.8	85.0	61.9	69.2	71.8	60.1

TABLE 6. STARTER CONCEPTS EVALUATION DATA

	Reliability Fail/1000 FH	Vulnerability	Development Risk	Cost $13,000 Normalized	Weight and Volume lb	Weight and Volume ft^3	Performance	Maintainability MMH/1000 FH	Installation Flexibility
APU-HYDRAULIC									
Auxiliary Powerplant	3.30	For this study vulnerability will be based on physical volume.	1975 technology	1.0	50	0.27	No known limitations.	33	Fuel and inter-connections limit flexibility.
(2) Starter and Controls	4.70				33	0.13			
(1) Pump	0.85				10	0.05			
Gearbox and Clutch	0.30				10	0.05			
	9.25				103	0.50			
APU-PNEUMATIC									
Auxiliary Powerplant	3.3		1975 technology	1.0	50	0.27	No known limitations.	16	Fuel and inter-connections limit flexibility.
(2) ATS and Controls	1.2				34	0.16			
	4.5				84	0.43			
APU-ELECTRIC									
Auxiliary Powerplant	3.30		1975 technology	1.0	50	0.27	No known limitations.	25	Large volume.
(1) Generator and Control	0.38				65	0.47			
(2) Starter and Control	3.00				60	0.30			
Gearbox and Clutch	0.30				10	0.05			
	6.98				185	1.09			
BATTERY-ELECTRIC									
Battery	2.4		State-of-the-art	0.44	154	1.40	Low temperature battery limitation.	20	Good flexibility.
(2) Starter and Control	3.0				60	0.30			
Gearbox and Clutch	0.3				10	0.05			
	5.7				224	1.75			
ACCUMULATOR-HYDRAULIC									
(1) Accumulator	0.4		State-of-the-art	0.33	195	1.04	-65°F pressure topping accumulator required.	25	Good flexibility
(2) Starter and Controls	4.7				33	0.19			
(2) Speed Ratio Package	0.3				4	0.02			
	5.4				232	1.25			
COMBUSTION-TURBINE									
(2) Combustion-Starters	4.2	Volume/1.5 because of redundancy.	1975 technology	0.67	64	1.60	-65°F pressure topping accumulator required for starter.	20	Large volume limitation.
Fuel and Air Supply	0.8				19	0.60			
Fuel and Air Hand Pumps	-				10	0.10			
Controls and Lines	0.4				7	0.06			
Gearbox and Clutch	0.3				10	0.05			
	5.7				110	2.41			

2. Vulnerability - Vulnerability was computed to be proportional to the volume of the starter. This was done because the probability of small-arms fire hitting any section of a helicopter is proportional to its exposed area. Nothing was done in this evaluation in consideration of locating components on the less vulnerable areas of the helicopter or secondary damage (fire) that could result from a hit. Those concepts that require fuel for their operation will be more vulnerable when secondary damage is considered.

3. Development Risk - The weights given each system for development risk were based on engineering judgment. All of the concepts being considered are state of the art, and the degree of development risk will be related to the weight and size reduction and performance design goals of each system. The starter concepts that use an APU were judged to be a little higher development risk primarily because the APU size and weight and performance are based on 1975 technology. However, none of the concepts are considered a high development risk.

4. Cost - Some cost data were available in the referenced reports; however, the weights given in Table 6 represent estimations of the relative costs. The cost of the other concepts was estimated in proportion to their relative complexity as compared with the APU systems.

5. Weight and Volume - The assigned weightings were determined by separately assessing weight and volume for each system based on the lowest weight, with the lowest volume getting 10 points and higher weights and larger volumes a proportionally lower number of points. The system with the highest combined point total for weight and volume was weighted 10, which is the maximum for this design parameter.

6. Performance - The ability of the APU-powered starter concepts to perform the starting function was judged to be equal. The battery electric system was evaluated the lowest because of its cold-day limitations. The accumulator-hydraulic system and the combustion starter system require pressurized oil, pressurized fuel, and air supplies, respectively, which will also have cold-day limitations. These limitations can be taken care of by cold weather pressure topping using hand pumps, but the performance was judged to be lower for this reason.

7. Maintainability - The maintainability of the starter concepts was evaluated based on multiplying the reliability by the average maintenance hours per maintenance action as determined from the references.

8. Installation Flexibility - Battery-electric starter was considered best since the starter has good flexibility in mounting to the engine and requires only a single electrical connection to

the airframe. All other systems required additional and more complex connections for airframe-to-engine interface and were downgraded accordingly.

(2) Self-Contained Starters

A survey was made to evaluate the potential of using an engine-mounted, self-contained, JP-fueled starter. Three basic engine types, gas turbine, piston engine, and the Wankel, were considered.

A starter specification was generated and several manufacturers were surveyed. The specification required the use of JP fuel and a provision for a manual secondary starting system.

Assuming that cost and development interest were not overwhelming disadvantages, the three candidates were compared as shown in Table 7 and rated as shown in Table 8.

The study showed that a gas turbine starter (GTS) would provide the lightest, least vulnerable, and most reliable overall starter package, but would require an expensive development effort. Packaging of a GTS may also be a limitation in certain installations.

TABLE 7. SELF-CONTAINED STARTER EVALUATION RATING

Criteria	Weight Factor	Starter		
		Gas Turbine	Piston Engine	Wankel
Reliability	23	23	12.7	17.0
Vulnerability	19	19	9.8	4.5
Development Risk	14	14	10.0	12.0
Cost	12	4	12.0	8.0
Weight and Volume	10	8	10.0	9.0
Performance	9	9	6.0	6.0
Maintainability	7	7	3.0	5.0
Installation Flexibility	6	6	5.0	5.0
Total	100	90	68.5	66.5

TABLE 8. ACCESSORY INTEGRATION PROGRAM STARTER CONCEPT EVALUATION DATA

	Reliability Failure/1000 FH	Vulnerability	Development Risk	Cost Normalized $13,000	Weight and Volume, lb	Weight and Volume, ft^3	Performance	Maintainability MMH/1000 FH	Installation Flexibility
GTS									
(2) Gas Turbine Starters	4.1	Volume	1975 technology	1.00	114	0.65	No known limitation	15	Engine mounting limitation
Two-Cycle Engine									
(2) Engines	7.1	Volume	1975 technology	0.33	95	0.78	No limitation when fuel injected	35	Engine mounting limitation
(2) Gearbox and Clutch	0.3				10	0.05			
	7.4				105	0.83			
Rotary-Wankel									
(2) Engines	5.2	Volume	1975 technology	0.50	100	1.74	No limitation when fuel injected	21	Large volume limitation
(2) Gearbox and Clutch	0.3				10	0.05			
	5.5				110	1.79			

The results of the survey and discussions with vendors are summarized in the following paragraphs.

a. Gas Turbine Starters

Gas turbine starters (GTS) in the 16-hp class are not being produced, but could be developed for a specific application. Development, qualification, and production costs would be relatively high, considering the requirement for an altitude relight. For most applications, an APU is used for engine starting and also provides other functions such as power generation or environmental conditioning. The development and production cost is, therefore, spread over several functional requirements.

The GTS configuration considered would incorporate a planetary reduction gear drive and an overrunning clutch. A hydraulic accumulator blowdown start system would be used. This configuration also allows the added flexibility of cold cranking the main engines to check fuel and electrical systems.

b. Piston Engine

Several commercial piston engine manufacturers were contacted but showed no interest in a starter development effort. The primary reason given was limited production quantities. No major technical difficulties in using a piston-engined starter were uncovered.

Starting at -65°F with JP fuel would require fuel injection or the use of a fuel preheater. A dual-clutch arrangement would be used. A centrifugal clutch would allow a smooth acceleration of the engine, and an overrunning clutch would be used to decouple the starter.

c. Wankel Engine

Curtiss-Wright Corporation was visited to review the use of a Wankel engine for a starter application. No technical difficulties were uncovered, during the discussions, which would limit the application of a Wankel. Little enthusiasm was shown for participation in a development effort for this application. Again, the reasoning used was the limited production quantities. Weight and volume estimates were provided.

An air-cooled Wankel was considered that dictated the use of large cooling fins and influenced the size and weight estimate. Starting at -65°F would dictate the use of a fuel injection system. The operating speed considered was 12,000 to 15,000 rpm and would require development of a 3000-psi fuel injection system. The reliability estimates for the Wankel were based on the relative complexity as compared to a conventional piston engine.

(3) Airframe-Supplied Power vs Self-Contained Starters

Based on the previous study information, the best engine-mounted, self-contained starter was then compared with the best engine-mounted starter that is supplied with power from an external source. In this study, the GTS was compared with the APU/pneumatic. The relative comparison is summarized in Table 9 and shows the GTS with a slightly higher rating.

TABLE 9. STARTER CONCEPT EVALUATION

Criteria	Weight Factor	Starter Concept - Rating: APU Pneumatic	Starter Concept - Rating: Gas Turbine
Reliability	23	21.0	23.0
Vulnerability	19	19.0	19.0
Development Risk	14	14.0	14.0
Cost	12	12.0	12.0
Weight and Volume	10	10.0	7.4
Performance	9	7.0	9.0
Maintainability	7	6.6	7.0
Installation Flexibility	6	6.0	5.0
Total	100	95.6	96.4

These systems were rated closely, primarily because of the similarity of the components. Both concepts use a gas generator and a turbine/gearbox for transfer of energy to the engine. The GTS configuration uses a dedicated system for each engine, while the APU/pneumatic uses a common gas generator and separate air turbine starters. It should be noted that the cost of the APU and GTS systems were rated similarly, assuming a specialized development effort for each system. This will not be a valid assumption if the APU is used for other aircraft functions and the cost is shared.

For this program, the GTS was used where the envelope allowed. For envelope-limited applications, or in installations where an APU is required for other airframe functions, the APU/pneumatic system is recommended.

b. Integral Starters

Three candidate systems were evaluated that provided means of starting the main engine without requiring a mechanical connection to the engine rotor. These systems use bleed air supplied by a separate APU and are illustrated in Figure 22.

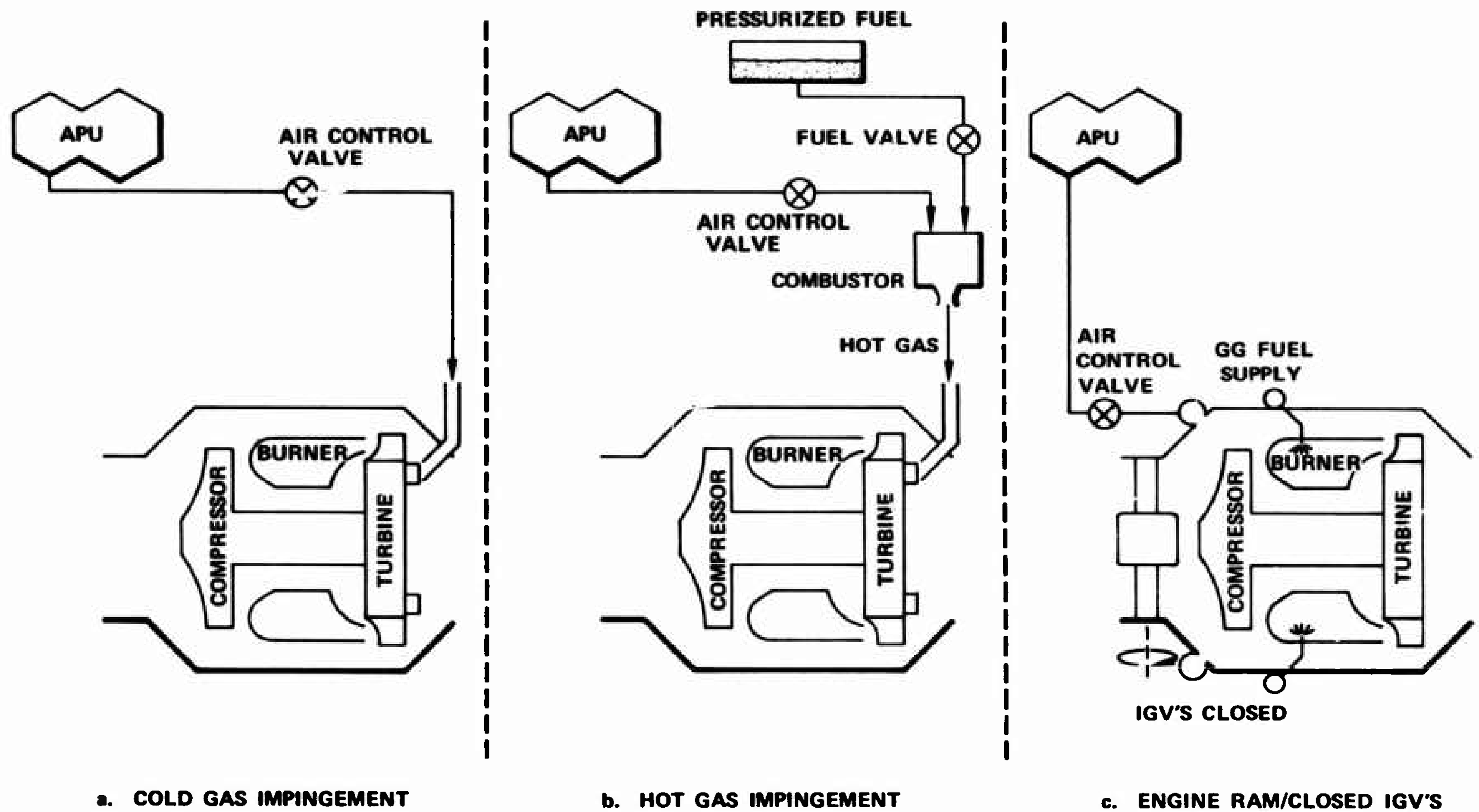


Figure 22. Integral Engine Starter (APU Bleed Air Supplied)

Air supplied by a pressurized bottle system was briefly considered but was discounted on the basis of size, weight, and the potential hazard involved with high-pressure pneumatic bottles subject to a military environment. A comparison was made of a pressurized bottle system and an APU. The following APU was chosen for comparison:

Bleed Flow	=	0.63 lb/sec
P/P	=	6.8
Volume	=	0.47 ft^3
Weight	=	42 lb
Torque at Ignition	=	5.2 ft-lb

Since the total gas mass (flowrate x time of starter operation) must be carried in the bottle, the starter torque was varied to reduce total mass of air, as shown in Figure 23. Even though flowrate increases with starter torque, the time to starter cutoff and total mass flow decreases.

Increasing tank pressure reduces the required tank volume, as shown in Figure 24. Very high pressures are necessary to obtain a volume competitive with an APU.

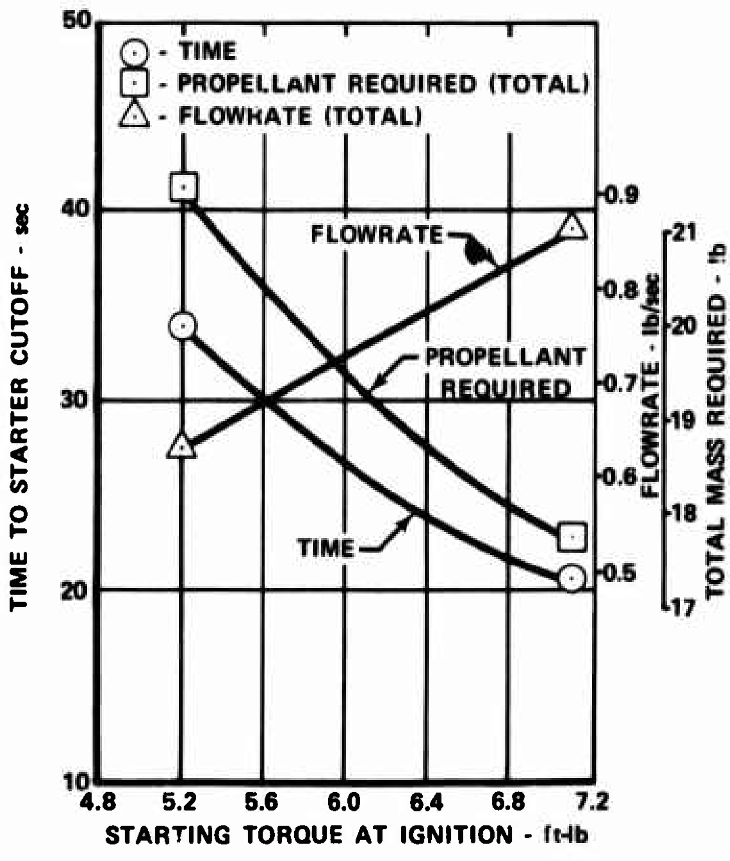


Figure 23. Impingement Starting Characteristics

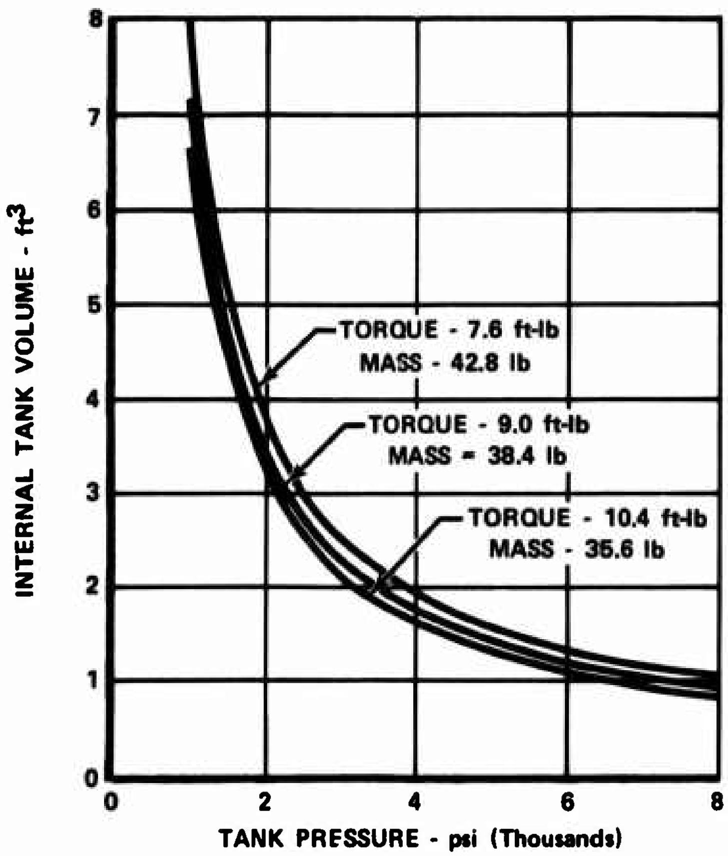


Figure 24. Tank Volume Requirements for Impingement Starting Gas Required for Two Starts

(1) Cold Gas Impingement

This system provides input energy to the gas generator rotor by impinging on the rotor surface with a high-velocity gas supplied by an external APU. For this application, several locations for the impingement area were considered. A location at the rear of the gas generator turbine rotor was selected and is shown in Figure 25. The addition of air at this location has a minimal effect on normal engine operation, as opposed to an injection point on the compressor, where the externally supplied air might interfere with normal compressor operation.

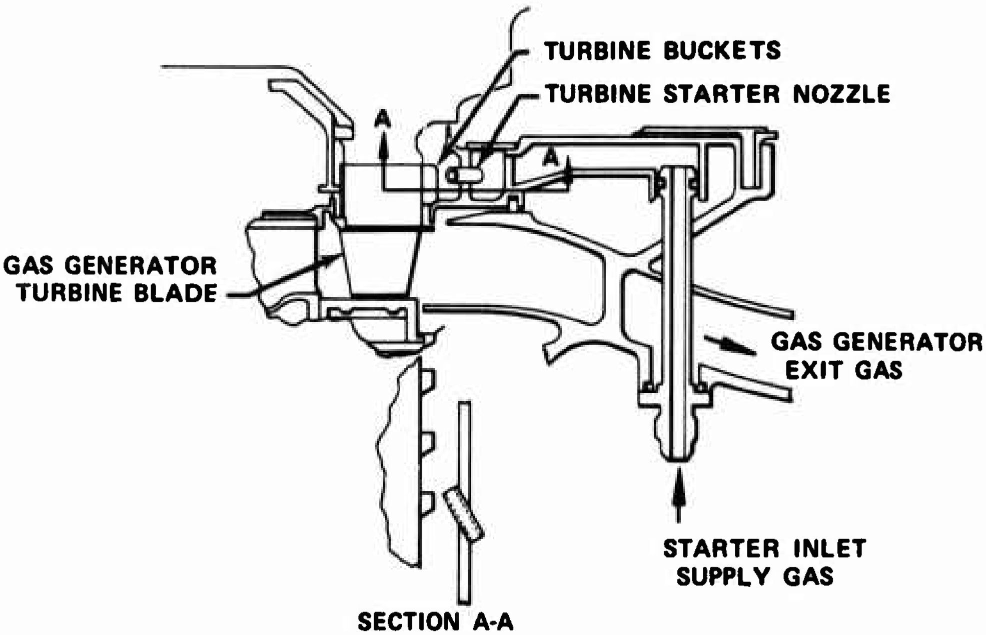


Figure 25. Impingement Starter

The characteristics of the selected cold gas impingement system are outlined below:

Nozzle	Gas T = 600°R Impingement Surface Diameter = 5.4 in.
Mach Number	1.0
Pressure Ratio	1.893
Flowrate	1.226 lb/sec
Total Orifice Area	1.928 $in.^2$

(2) Hot Gas Impingement

This system inputs energy to the gas generator rotor by impinging on the rotor surface with a hot gas, which is supplied by an auxiliary combustor. The combustor is engine-mounted close to the injection point and is provided air by an external APU. A schematic of the system is shown in Figure 22. The system requires the additional complexity of a separate ignition and fuel control system.

The characteristics of the hot gas impingement system are outlined below:

Mach Number	1.0
Gas Temperature	2460°R
Pressure Ratio	1.893
Flowrate	0.58 lb/sec
Total Orifice Area	1.348 $in.^2$
Mean Diameter of Turbine Buckets	5.4 in.

(3) Engine Ram/Closed IGV's

This starting method uses bleed air from an APU for supplying air to the engine burner during start. Air introduced into the engine burner is forced through the engine turbine by closing the compressor inlet area. The compressor IGV's are used to close the compressor inlet, as shown in Figure 26. With the compressor inlet closed, work required to pump inlet air is eliminated and, therefore, reduces the horsepower required to rotate the engine rotor to any specific speed. The air introduced into the engine combustor is capable of rotating the engine up to minimum fuel pumping speed, at which time fuel from the engine fuel system is also introduced into the combustor and ignited. Additional energy available from the combustion gases will now accelerate the rotor above a self-sustaining (34,000 rpm) rotational speed. The relationship of torque available vs torque required for the rotor parasitic losses is shown in Figure 27. The auxiliary air supply is now removed from the engine combustor, the IGV's are scheduled to their normal run condition, and the engine is accelerated to ground idle. This transient is illustrated on the compressor map in Figure 28. Since the transfer of air supply from the APU to the engine compressor requires some transient time, the initial speed at which the transfer is initiated must be high enough to allow for "coast-down" during the transient and retain an above-self-sustained speed at the completion of the transfer. The expected engine coast-down characteristics are shown in Figure 29. An APU flowrate of 0.42 pps at a pressure ratio of 1.9 will be required.

(4) Rating of Integral Starters

The numerical ratings for the three systems are shown in Table 10. The cold air impingement system received the highest rating (82.3), primarily because of the simplicity and proven capability. The engine ram with closed IGV's received the next highest rating (81.6) and was downgraded primarily because of development risk. The hot air impingement system received the lowest overall rating (72.3).

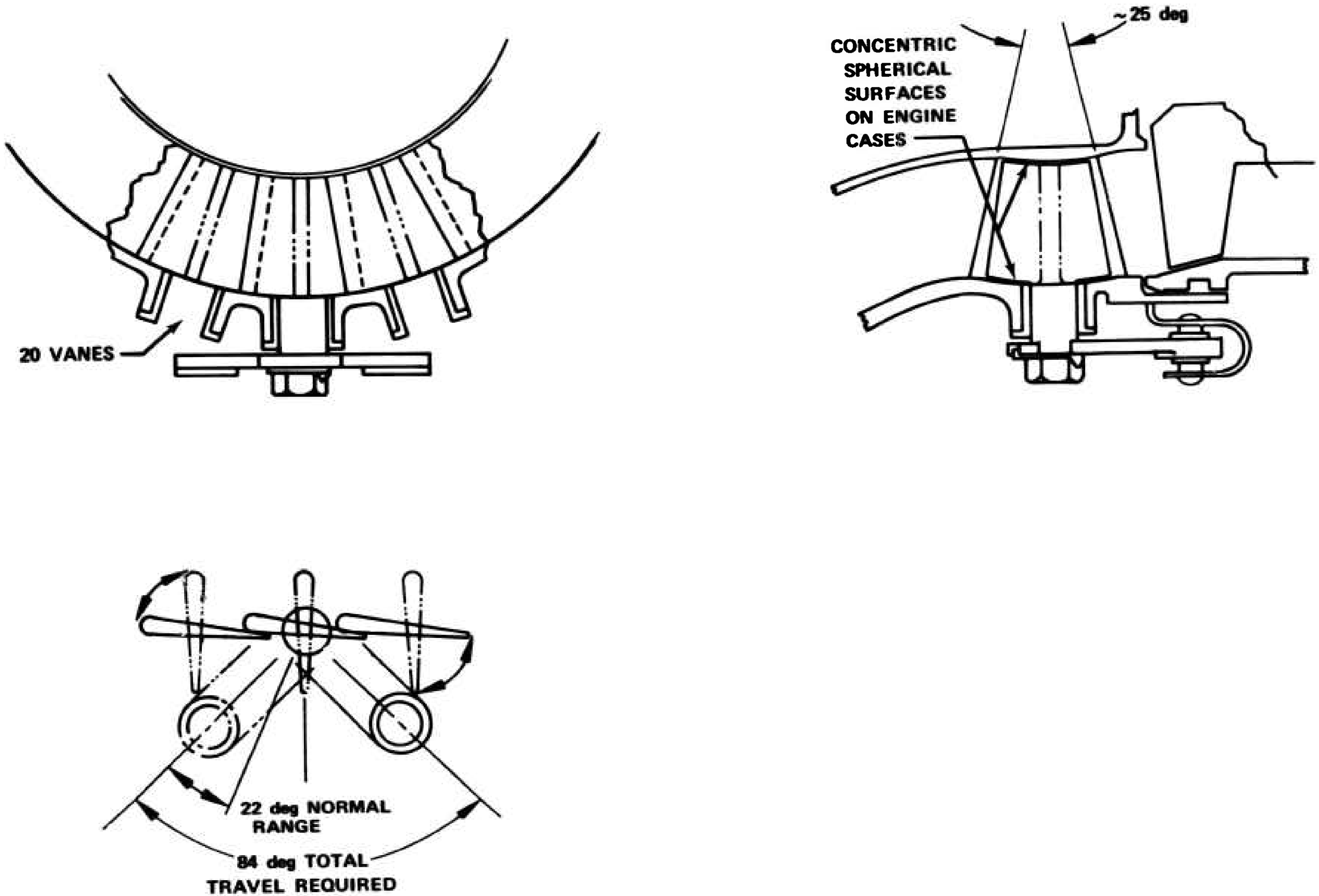


Figure 26. Variable Inlet Guide Vanes Used To Close Compressor Inlet

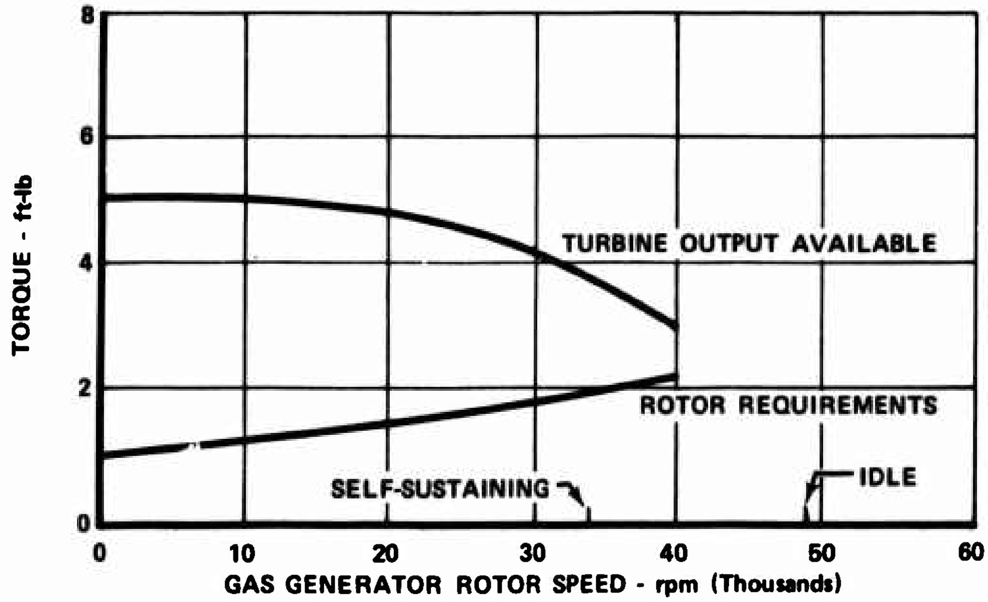


Figure 27. Torque Characteristics of Ram Start/ Closed IGV's

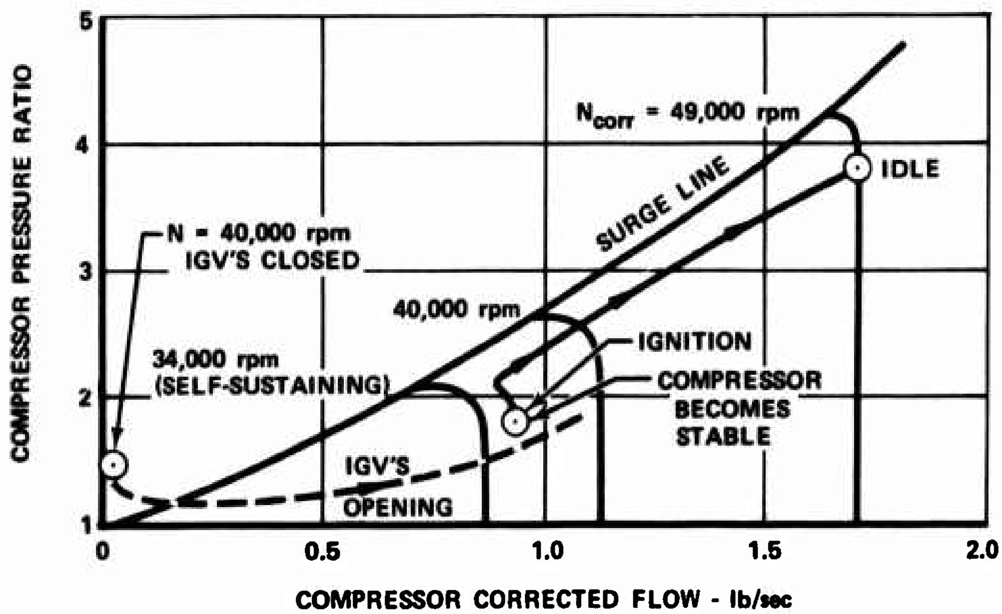


Figure 28. Compressor Map of Ram Start/Closed IGV's

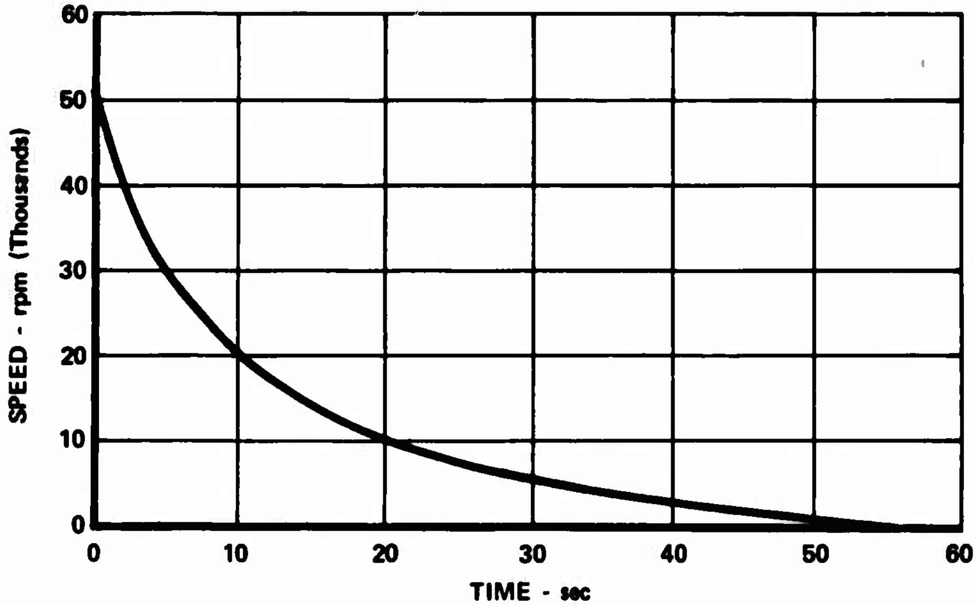


Figure 29. Engine Coast-Down Characteristic of IGV's Open

It is worthwhile to note that if the engine ram with closed IGV's was experimentally proven, the overall rating would be better than the cold air impingement system. The system would require slightly more engine complexity, but would reduce the airframe APU size by 66%, as compared to the cold gas impingement system. Experimental work in this area is justifiable. A discussion of the ratings is provided as follows:

1. Reliability - The cold air impingement system was considered to be the simplest and most reliable. The hot air impingement system was downgraded because it requires a separate igniter and fuel system for the small burner. The third system (with closed IGV's) requires sequencing of the IGV's and APU shut-off valve.

2. Vulnerability - The criteria used were volume and number of components (Table 11). The engine ram with closed IGV's was considered as the baseline (volume = 0.25 ft^3). The cold air impingement system had a large volume (1.08 ft^3) but a fewer number of components. The hot air impingement system required a burner plus a fuel and ignition system for both the engine and the APU and had a volume of 0.58 ft^3.

3. Development Risk - The cold impingement was considered the least risk. This system will start the engine; the only problem is the interface with the engine/rotor for an optimum engine system. The hot air impingement will also start the engine, but, in addition to its interface with the engine/rotor, the burner must be developed. The engine ram system with closed IGV's will require experimental evaluation to define the compressor recovery characteristics for the transient when the IGV's are opened. The compressor must recover from an unstable or stalled condition with the inlet closed and establish a normal compressor-engine operating line during the coast-down period.

TABLE 10. BLEED FROM APU FOR ENGINE STARTING (INTEGRAL STARTER EVALUATION RATING)

		Starting Method					
		Jet Impingement on Engine Rotor				Engine Ram	
		Cold Gas Impingement on Engine Turbine		Hot Gas Impingement on Engine Turbine		Cold Air for Engine Burner-Closed IGV's	
Selection Criteria	Weight Factor	Rating Factor	Rating	Rating Factor	Rating	Rating Factor	Rating
Reliability	23	1.0	23.0	0.8	18.4	0.8	18.4
Vulnerability	19	0.5	9.5	0.5	9.5	1.0	19.0
Development Risk	14	1.0	14.0	0.9	12.6	0.3	4.2
Production Cost	12	1.0	12.0	0.5	6.0	0.8	9.6
Weight and Volume	10	0.3	3.0	0.7	7.0	1.0	10.0
Performance	9	1.0	9.0	1.0	9.0	0.9	8.1
Maintainability	7	1.0	7.0	0.8	5.6	0.9	6.3
Installation Flexibility	6	0.8	4.8	0.7	4.2	1.0	6.0
Rating Total	100	-	82.3	-	72.3	-	81.6

TABLE 11. INTEGRAL STARTER EVALUATION DATA

System		Starting Method		
		Jet Impingement on Engine Rotor		Engine Ram
		Cold Gas Impingement on Engine Turbine	Hot Gas Impingement on Engine Turbine	Cold Air for Engine Burner-Closed IGV's
APU (1)	Bleed Flow, lb/sec/P/P_i	1.23/2.36	0.40/2.83	0.42/1.93
	Weight, lb	54	22	15
	Volume, ft^3	1.08	0.38	0.25
	Fuel, lb/hr	57.5	31	24
Burner and Fuel System	Weight, lb	N/A	3.0 (Each Engine)	Engine
	Volume, ft^3	N/A	0.0975 (Each Engine)	Engine
	Fuel, lb/hr	N/A	47.0	45
Totals(2)	Weight, lb	54	28	15
	Volume, ft^3	1.08	0.575	0.25
	Fuel, lb/hr	57.5	78	69

(1) APU and accessories (starter not included)

(2) Line from APU to engine not included.

4. Production Cost - The cold impingement has a larger APU, but everything else is standard. The hot impingement has a smaller APU, but requires a separate burner, ignition system, and a fuel pump. The engine ram requires the smallest APU, but more expensive IGV's and actuation system.

5. Weight and Volume - The sizes include the APU's accessories, except for their starter systems. For the hot air impingement system, the burner was considered as being mounted on the engine; therefore, two would be required for a twin-engine installation.

6. Performance - The engine ram system with closed IGV's was downrated because of the potential of a slight performance penalty due to the IGV design.

7. Maintainability - The hot air impingement system was downgraded because of the large number of components (burner, fuel nozzle, fuel pump, motor, and ignition system). The engine ram system with closed IGV's was downgraded slightly due to the special IGV's.

8. Installation Flexibility - Volume and number of components were used for the criteria. The engine ram had the smallest volume and thus was the baseline. The cold air impingement was downgraded because of its larger APU volume. The hot air impingement system was graded lowest because of volume and the requirement that the burner (10 in. length and 5 in. diameter) be mounted on the engine to prevent long, hot lines.

c. Pneumatic vs Mechanical C&A Drive

The cold gas bleed pneumatic turbine accessory drive system previously selected as the best pneumatic drive was compared with a tower shaft accessory drive system using the selected weighting criteria previously established.

The tower shaft system received a rating of 92.1% vs a rating of 69.1% for the cold gas bleed system, as shown in Table 12. Even though the cold gas bleed system was compared only to a tower shaft system in this study, a comparison with any other direct mechanical drive system such as a cluster gearbox arrangement would yield similar results because mechanical drive systems, in general, have similar advantages and disadvantages. Based on this analysis, the cold gas bleed system was eliminated as a candidate accessory drive system.

TABLE 12. CONTROLS AND ACCESSORY DRIVE SYSTEM EVALUATION RATING

Pneumatic vs Mechanical C&A Drive

		Tower Shaft		Pneumatic	
Criteria	Weight Factor	Rating Factor	Rating	Rating Factor	Rating
Reliability	23	1.0	23.0	0.60	13.8
Vulnerability	19	1.0	19.0	0.80	15.2
Development Risk	14	1.0	14.0	0.75	10.5
Cost	12	1.0	12.0	0.40	4.8
Weight and Volume	10	1.0	10.0	1.00	10.0
Performance	9	1.0	9.0	0.20	1.8
Maintainability	7	0.3	2.1	1.00	7.0
Installation Flexibility	6	0.5	3.0	1.00	6.0
Total Rating	100		92.1		69.1

The better of the two systems in each of the selection criteria is given a factor of 1.0, and the other system is given a factor that indicates its ranking in that category relative to the other. Figure 30 illustrates tower shaft and pneumatic turbine drive systems used for this study. The tower shaft system showed advantages in all areas except maintainability and installation flexibility. The reliability of the pneumatic system was downgraded due to the requirement for a pneumatic control valve and speed control loop.

d. Electrical vs Mechanical Fuel Pump Drive

Consideration was given to an electrically driven variable-speed pump as opposed to a conventional mechanically driven pump. The potential advantage of the electrically driven pump would be in simplification of the control system, in that the pump speed would be infinitely varied to set the desired main engine flow instead of an in-line throttle valve or a bypass valve. The electrical drive system will reduce the fuel pumping inefficiency and the heat rejection to the fuel. The drive systems were evaluated as shown in Table 13.

The mechanical drive was considered superior in all areas except performance, maintainability, and installation flexibility. The electric motor drive has the disadvantage of size and weight, which also impact the vulnerability assessment. The mechanical drive was selected for further consideration.

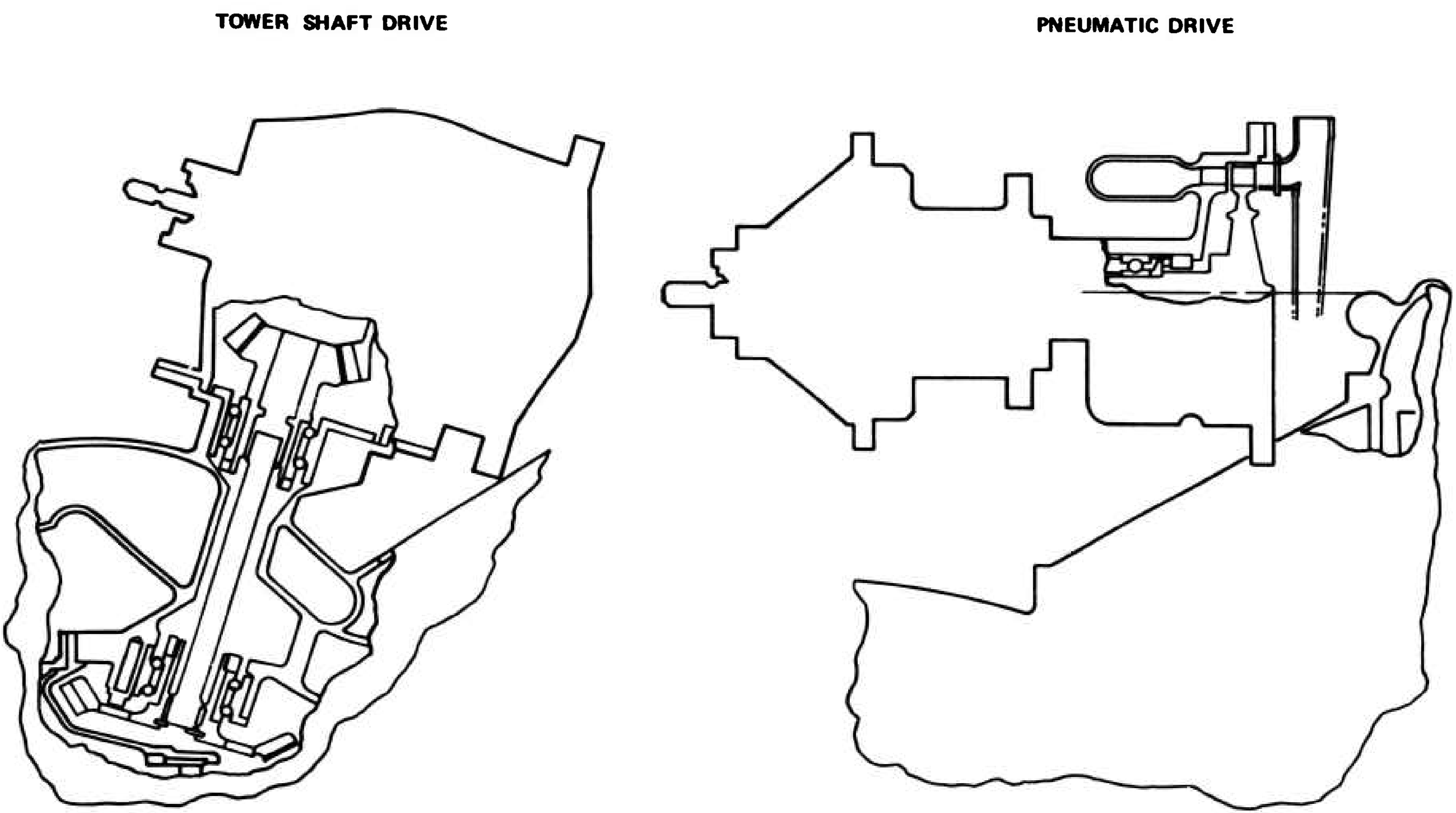


Figure 30. Tower Shaft and Pneumatic Turbine Drive Systems

TABLE 13. FUEL PUMP EVALUATION RATING

Mechanical vs Electrical Driven Pumps

Selection Criteria	Mechanical Drive Weight Factor	Mechanical Drive Rating Factor	Mechanical Drive Rating	Electrical Drive Rating Factor	Electrical Drive Rating
Reliability	23	1.0	23.0	0.9	20.7
Vulnerability	19	1.0	19.0	0.8	15.2
Development Risk	14	1.0	14.0	0.9	12.6
Cost	12	1.0	12.0	0.6	7.2
Weight and Volume	10	1.0	10.0	0.5	5.0
Performance	9	0.5	4.5	1.0	9.0
Maintainability	7	0.8	5.6	1.0	7.0
Installation Flexibility	6	0.5	3.0	1.0	3.0
Total	100		91.1		79.7

e. Hybrid C&A Pump Drive Systems

Three hybrid drive systems, shown in Figure 31, which used either a hydraulic, pneumatic or electrical interface between a tower shaft drive and the accessory package, were considered.

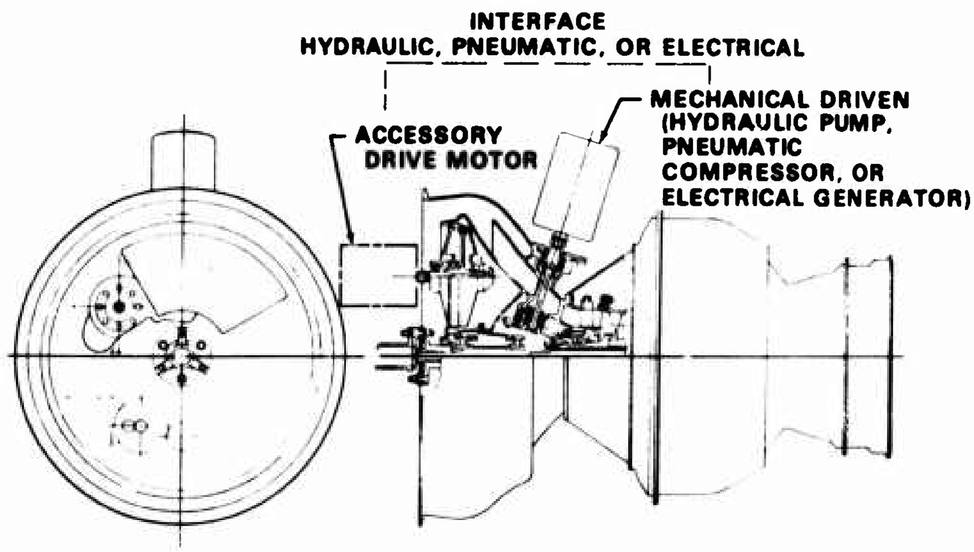


Figure 31. Candidate C&A Drive

The hybrids offered flexibility in the location of the C&A components in that only the drive motor must be located in the vicinity of the tower shaft. In all three cases, the drive motor also was used as a starter for the main engine.

The electrical system was discounted on the basis of the starter system studies, which showed that an electrical system had disadvantages in overall size, performance, and weight. In addition, the size and weight of the engine-mounted starter/generator and drive motor were not competitive with those of other systems in this horsepower class.

The hydraulic and pneumatic hybrid systems were studied in greater detail, since the hydraulic and pneumatic start systems were rated closely. The hydraulic starter/pump and drive motor and the pneumatic starter/compressor and drive turbine systems are shown schematically in Figures 32 and 33, respectively.

The mechanical/hydraulic hybrid system was selected for further evaluation. A relative ranking of the two systems is shown in Table 14.

A discussion of the relative rankings of the two systems is shown below:

1. Reliability - The relative complexities of the two systems are considered to be similar. The pneumatic system is rated higher due to the insensitivity to contamination and the demonstrated improved reliability of pneumatic drive components.

2. Vulnerability - The pneumatic system was downrated because of the large-diameter (approximately 7 in.) starter/compressor and drive turbine required.

3. Development Risk - The pneumatic system was downrated due to the development risk associated with the required convertible starter/compressor.

4. Cost - The relative costs of the two systems were judged to be similar.

5. Weight and Volume - The hydraulic system has the largest weight penalty, but the pneumatic system requires the largest volume. The hydraulic system was rated slightly higher on an overall weight and volume basis.

6. Performance - The hydraulic drive system (pump and motor combined) had the best overall efficiency, 73%, as compared to the pneumatic, 50%.

7. Maintainability - The pneumatic system was rated easier to maintain.

8. Installation Flexibility - The hydraulic system has the best installation flexibility based on smaller components.

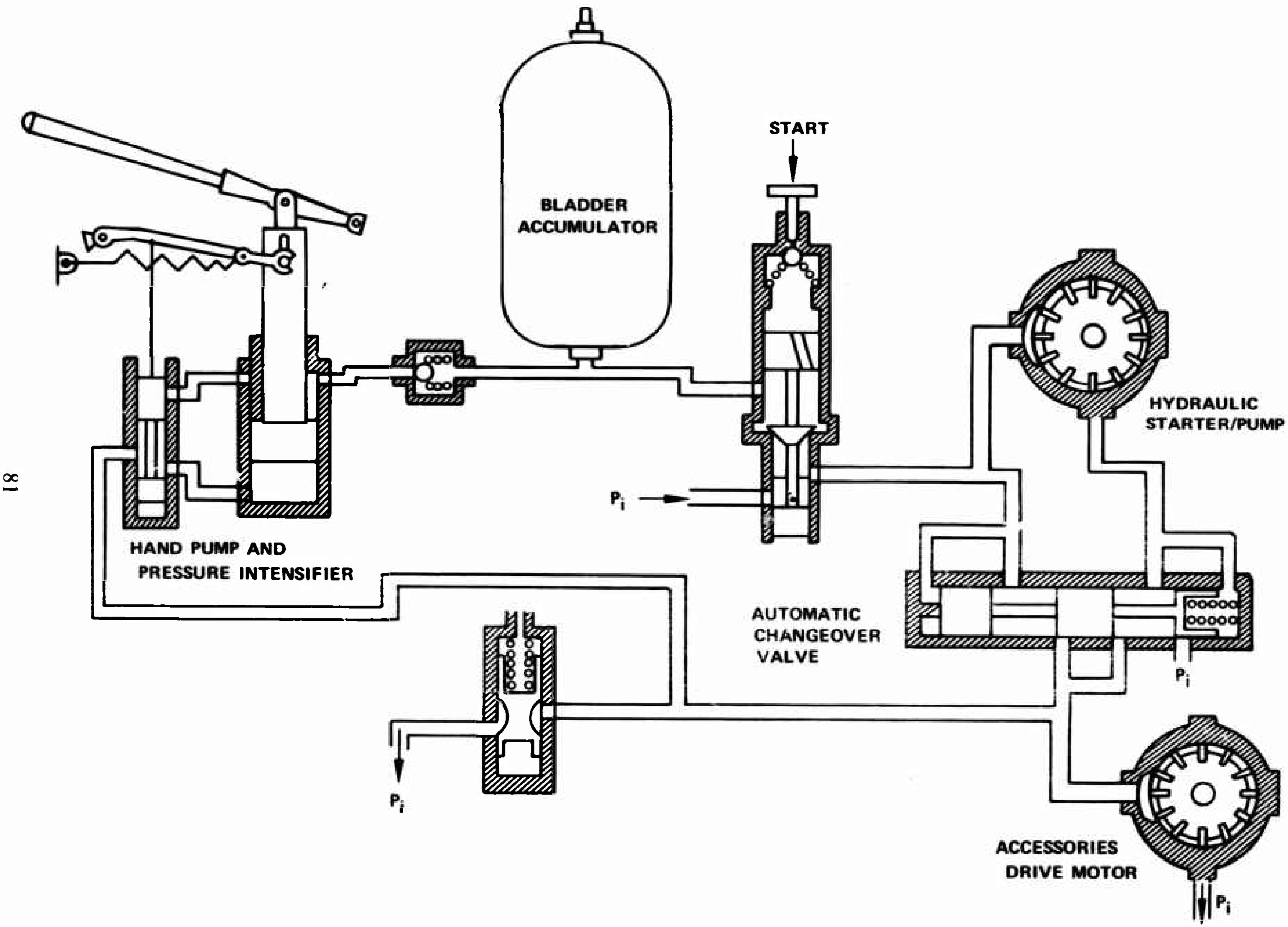


Figure 32. Hydraulic Starter and Accessories Drive

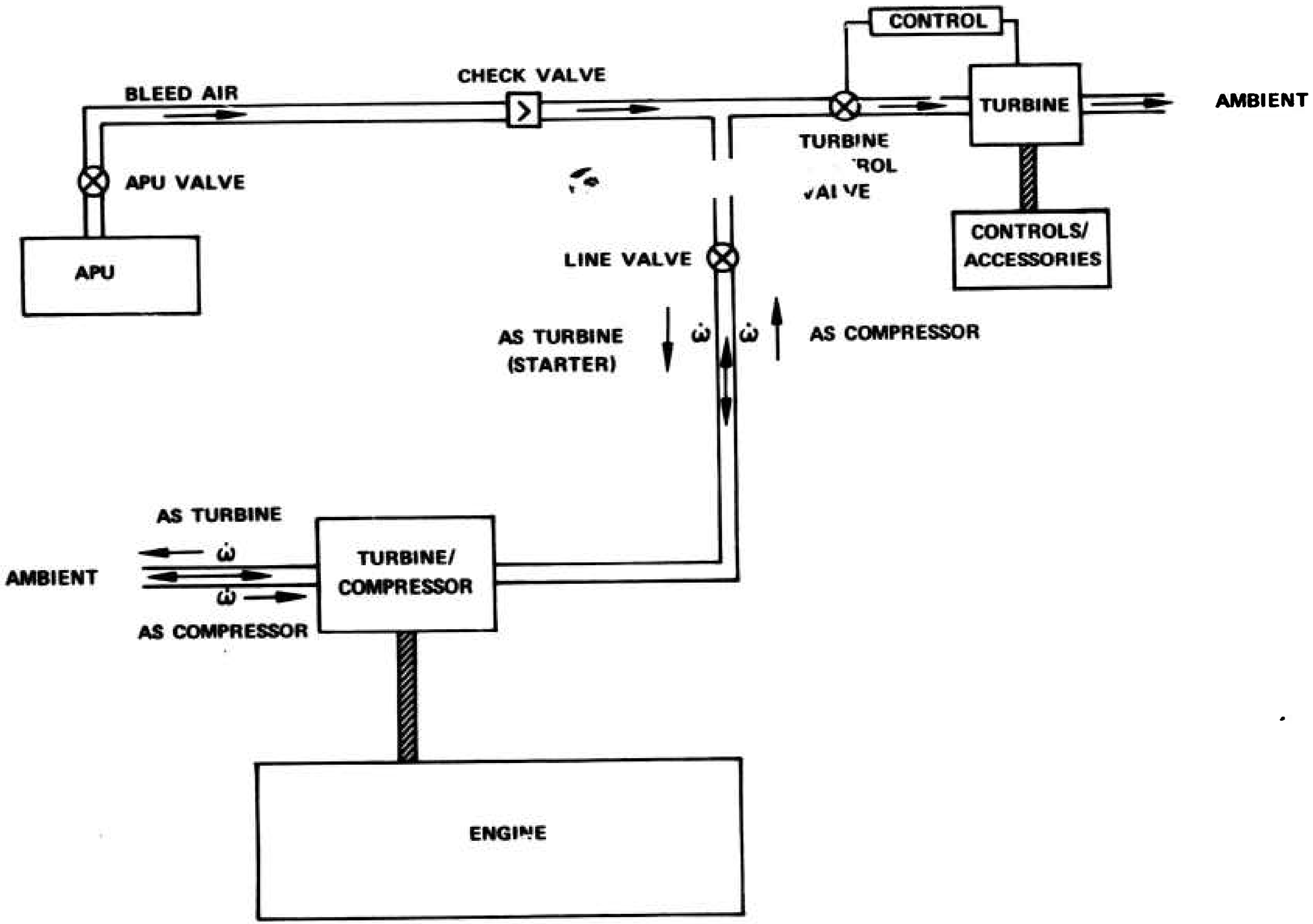


Figure 33. Pneumatic Starter and Accessories Drive

TABLE 14. HYBRID CONTROL AND ACCESSORY DRIVE SYSTEM

Hydraulic vs Pneumatic Evaluation Rating

		Hydraulic		Pneumatic	
Criteria	Weight Factor	Rating Factor	Rating	Rating Factor	Rating
Reliability	23	0.8	18.4	1.00	23.0
Vulnerability	19	1.0	19.0	0.60	11.4
Development Risk	14	1.0	14.0	0.80	11.2
Cost	12	1.0	12.0	1.00	12.0
Weight and Volume	10	1.0	10.0	0.90	9.0
Performance	9	1.0	9.0	0.80	7.2
Maintainability	7	0.8	5.6	1.00	7.0
Installation Flexibility	6	1.0	6.0	0.18	4.8
Total	100		94.0		85.6

4. Selection of Two Candidate Systems

Five control and accessory drive/starter systems were evaluated during this phase of the program. Preliminary design layouts of the five systems were made, and the systems were evaluated using the performance and rating criteria established in the Requirements Definition Phase. The control, fuel pump, PTO, and oil system components were considered to be common for this study. The five candidate control and accessory drive/starter configurations evaluated are outlined below.

a. C&A Component Description

(1) Tower Shaft C&A Drive/Gas Turbine Starter (TS/GTS)

A tower shaft-driven gearbox is used to drive a 65,000-rpm combination fuel pump and alternator, a 20,000-rpm PTO, and a 10,000-rpm oil pump. A self-contained gas turbine starter is mounted on a 20,000-rpm engine gearbox pad for engine starting. The starter consists of a gas generator, a hot gas power turbine, and a gear reduction system for the power turbine. An overrunning clutch is used to decouple the starter. This configuration is shown in Figure 34.

(2) Tower Shaft C&A Drive/Impingement Starter (TS/IS)

A tower shaft drive, as described in system No. 1, is used for the C&A components. Starting is accomplished by a cold gas impingement starter integral with the gas generator turbine. The starter is powered by an airframe-mounted APU, with a compressor discharge (cold gas) bleed. This configuration is shown in Figure 35.

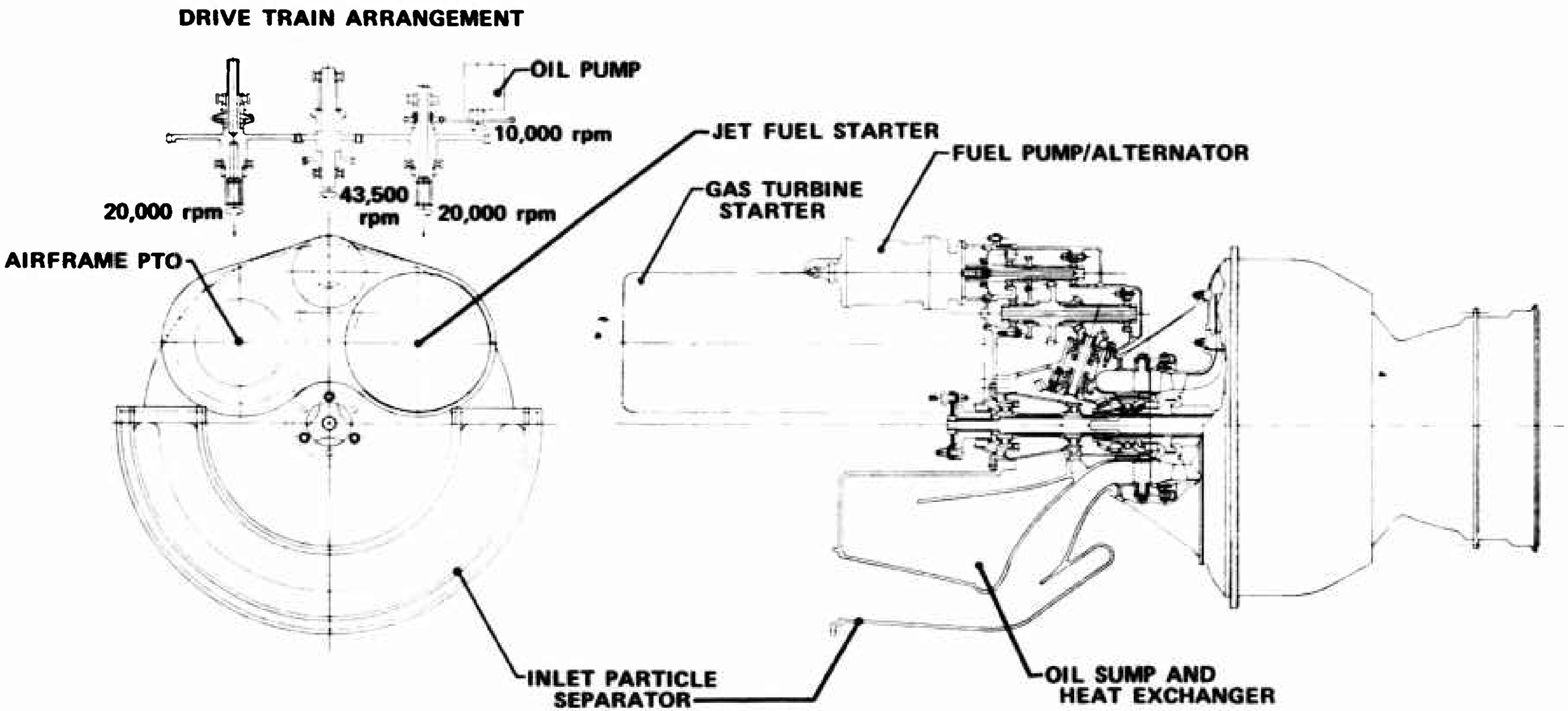


Figure 34. Tower Shaft/Gas Turbine Starter

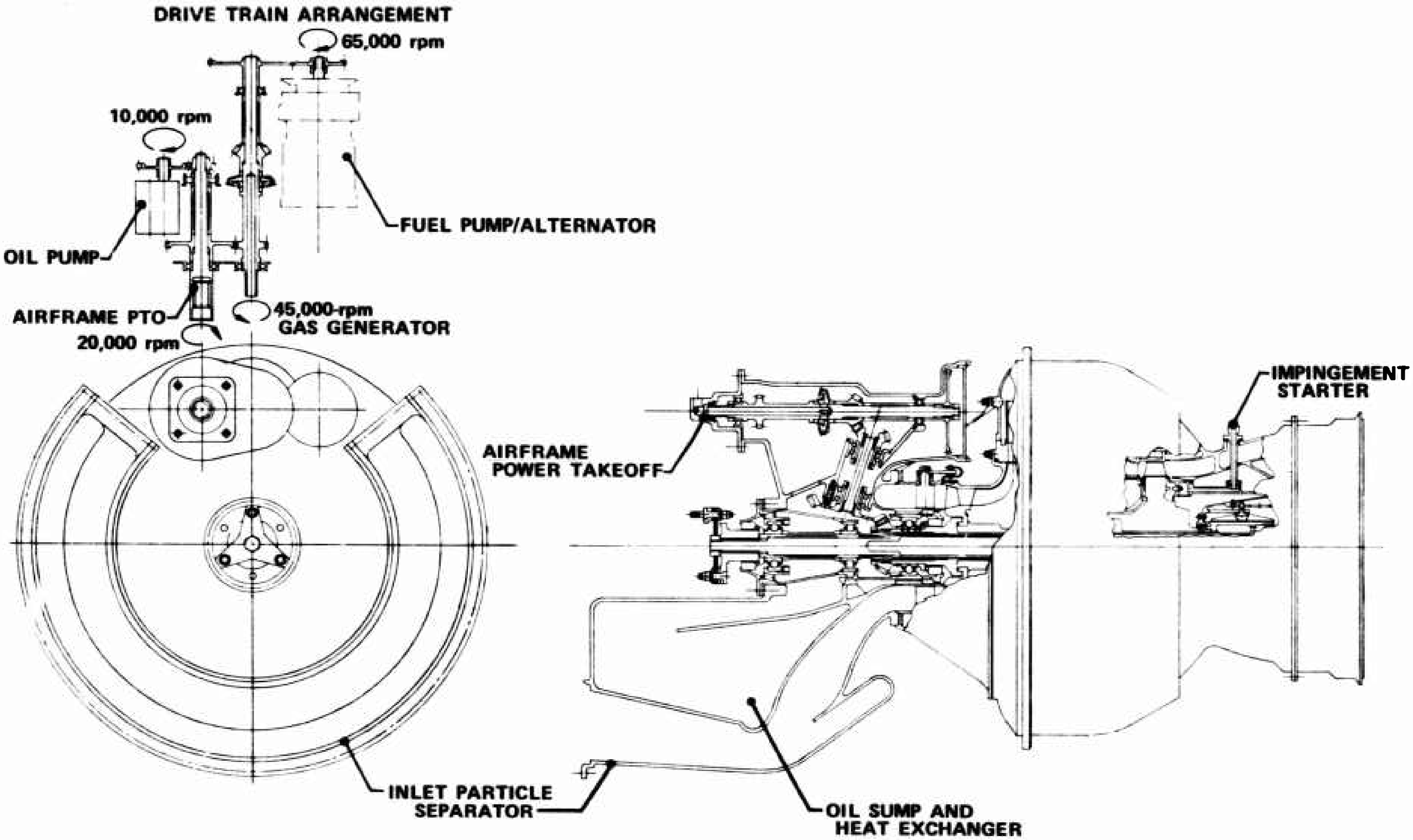


Figure 35. Tower Shaft Gearbox/Impingement Starter

(3) Cluster Gearbox C&A Drive/Air Turbine Starter (CGB/ATS)

The drives for the C&A components are provided by gears that are clustered around the gas generator drive shaft. The C&A component drive speeds are the same as described in system No. 1. An air turbine starter is used and is mechanically connected to the gas generator rotor through the cluster gearing. The air turbine starter consists of a high-speed turbine and a gear reduction system to couple with a 20,000-rpm pad. An overrunning clutch is used to decouple the starter system. This configuration is shown in Figure 36.

(4) Cluster Gearbox C&A Drive/Impingement Starter (CGB/IS)

The C&A drive is the same as described in system No. 3, and the starter system is the same as described in system No. 2. This configuration is shown in Figure 37.

(5) Hybrid Mechanical, Hydraulic C&A Drive/Hydraulic Starter (HD/HS)

The hybrid C&A drive used consists of a hydraulic starter/pump that is connected to the gas generator shaft by a tower shaft, and a hydraulic motor, which is used to power the C&A components. An accumulator blowdown system is used during the start transient to drive the hydraulic starter/pump and for powering the C&A drive motor. After the engine is self-sustaining, the hydraulic starter is switched to a pumping mode and is used to drive the C&A components. The configuration is shown in Figure 38.

b. Analysis

The five systems were assessed for each of the established rating criteria. The results are summarized in Table 15 and show the CGB/ATS starter and the TS/GTS as the two systems selected for detailed design. The ratings are discussed below:

1. Reliability - A reliability assessment was made for both the starter system and the C&A drive. The starter system reliability was given a 20% weighting and the C&A drive an 80% weighting. The results of the analysis are summarized in Table 16. The cluster gearbox drives were rated as being more reliable than the tower shaft drive because of fewer components in the drive train. The CGB/IS and CGB/ATS systems were rated closely together and ranked first and second, respectively. The TS/IS and TS/GTS systems were also rated closely together and ranked third and fourth, respectively. The HD/HS had the lowest reliability because of the hydraulic pump and drive motor in the drive train.

2. Vulnerability - The vulnerability of the total system was determined by an assessment of both the starter system and C&A drive system components. Since the protection afforded by the airframe is not known, the total volume of the airframe-mounted starter system components was used as the vulnerability criterion. The starter vulnerability was given a 25% weighting. The exposed vulnerable area of the engine-mounted C&A drive components was assessed and used as the vulnerability criterion. The C&A drive vulnerability was given a 75% weighting.

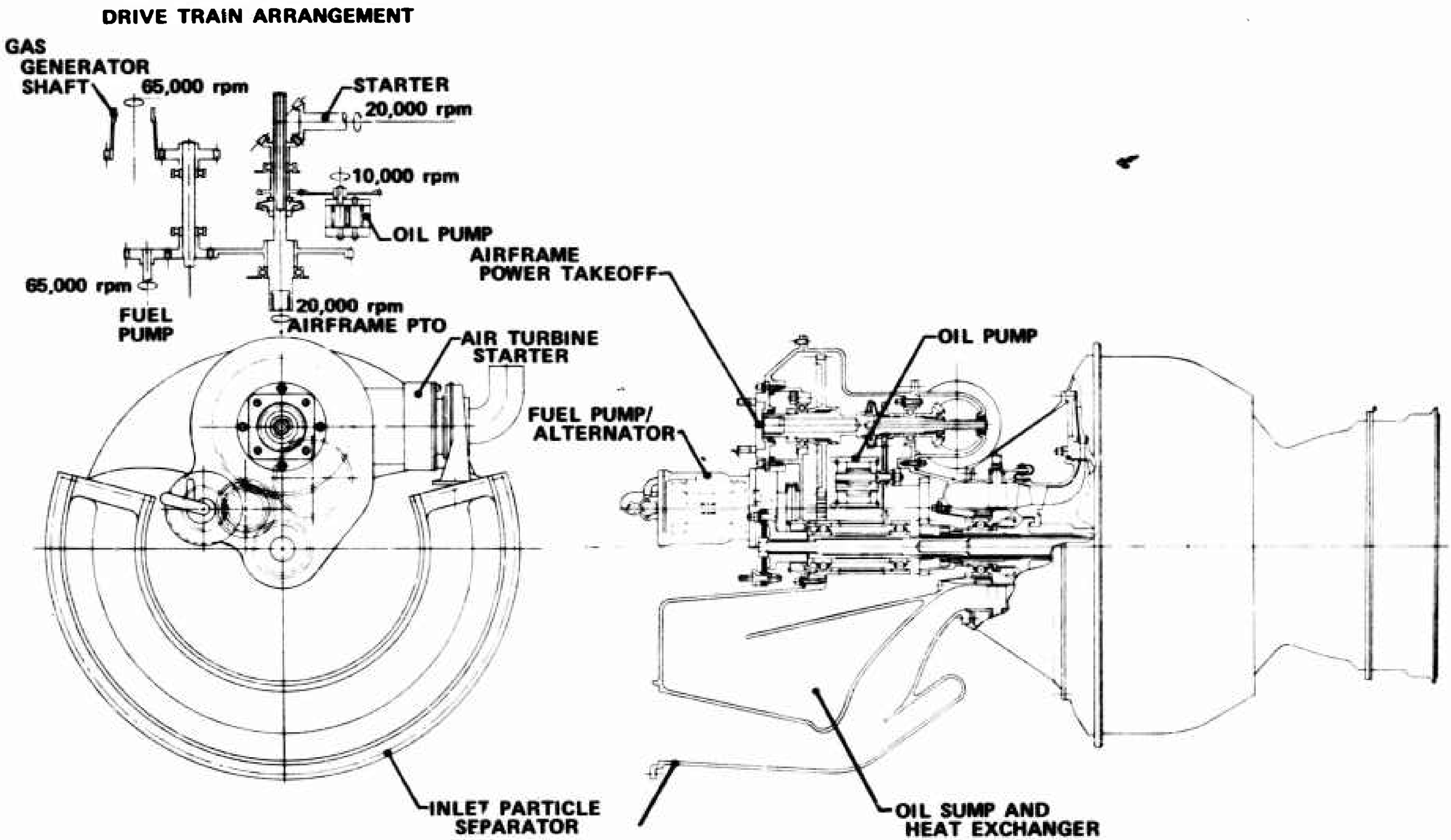


Figure 36. Cluster Gearbox/Air Turbine Starter

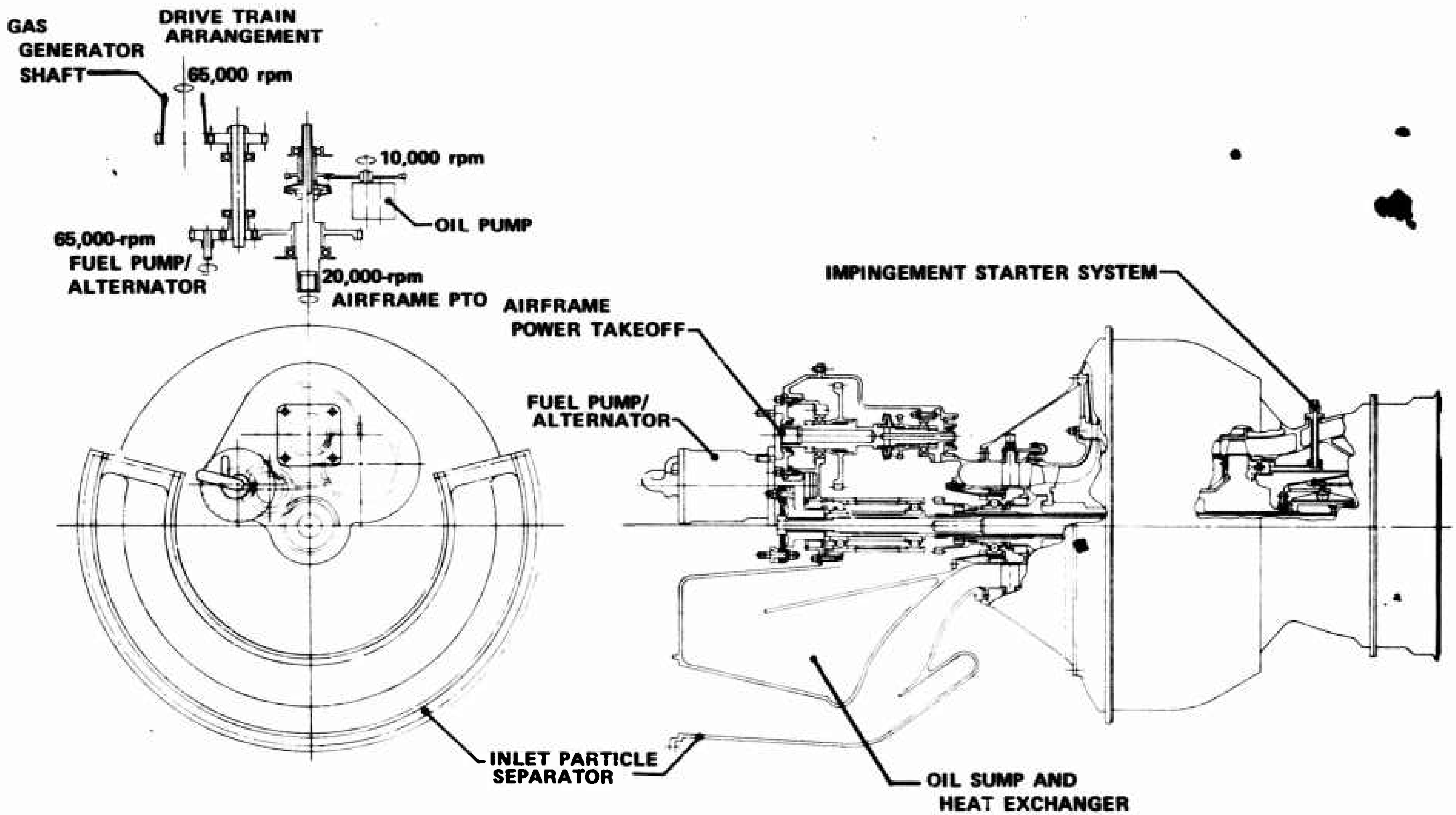


Figure 37. Cluster Gearbox/Impingement Starter

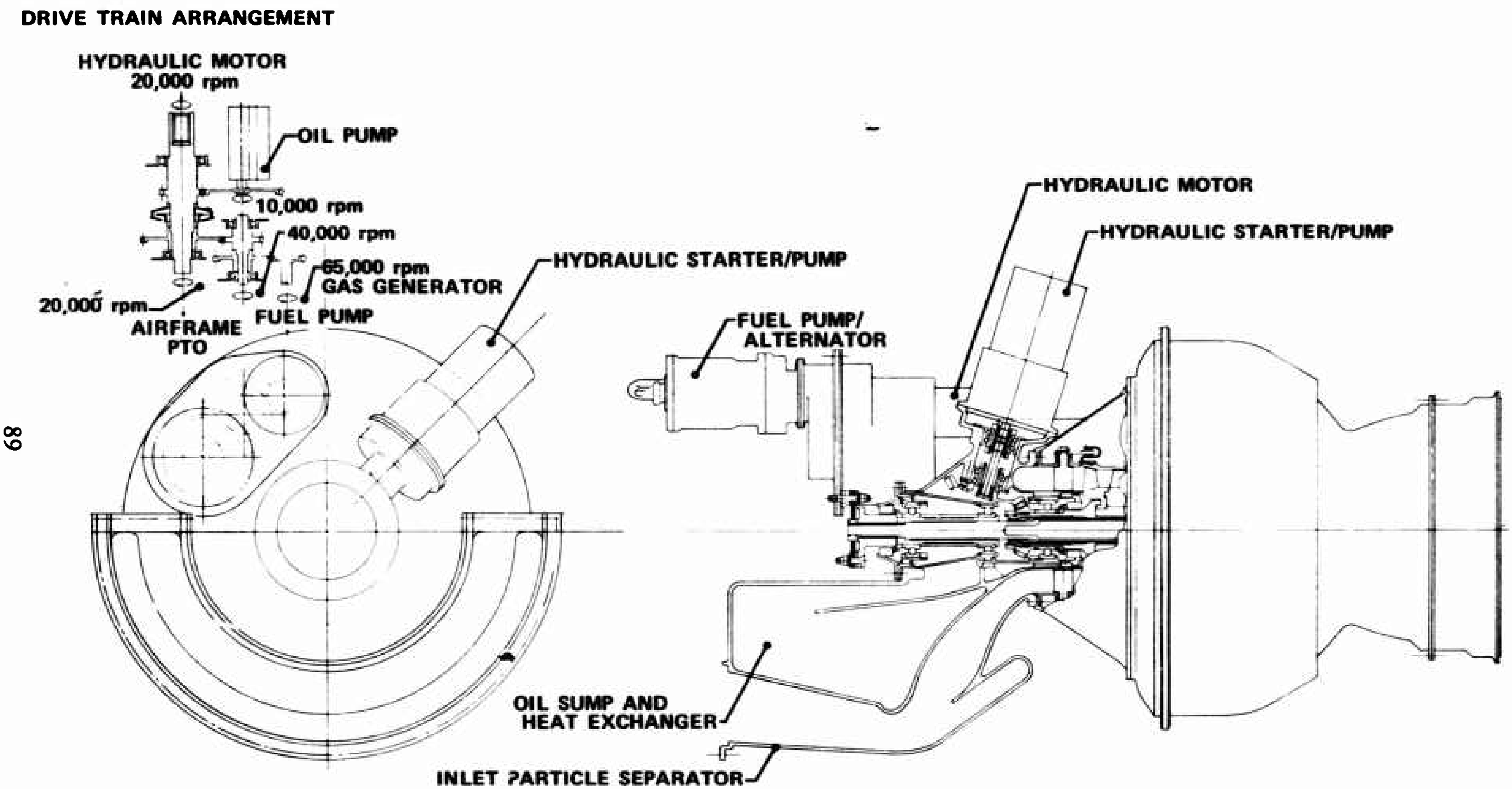


Figure 38. Hydraulic Drive/Hydraulic Starter

TABLE 15. RATING SUMMARY

C&A System Starter Configuration	Reliability, 23%	Vulnerability, 19%	Development Risk, 14%	Cost, 12%	Weight Volume, 10%	Performance, 9%	Maintainability, 7%	Installation Flexibility, 6%	Total Rating, %	Ranking
1. TS/GTS	16.2	19.0	14	4.8	7.5	9.0	5.3	4.7	80.5	2
2. TS/IS	16.7	12.9	14	4.4	5.2	8.6	4.8	3.7	70.3	4
3. CGB/ATS	22.7	16.3	14	3.8	10.0	8.8	7.0	6.0	88.6	1
4. CGB/IS	23.0	13.1	14	4.4	5.5	8.6	5.9	3.8	78.3	3
5. HD/HS	7.2	8.0	12	12.0	6.0	7.4	3.7	4.8	61.1	5

TABLE 16. RELIABILITY (RATING = 23%)

C&A System Starter Configuration	Start			Flight			Total Weight	Relative Weight	Rating, %	Ranking
	Component Failure Rate per 1000 hr	Relative Rating	Relative Weight, 20%	Component Failure Rate per 1000 hr	Relative Rating, %	Relative Weight, 80%				
1. TS/GTS	4.714	0.77	0.154	0.250	0.675	0.540	0.694	0.704	16.2	4
2. TS/IS	4.147	0.87	0.174	0.250	0.675	0.540	0.714	0.726	16.7	3
3. CGB/ATS	4.289	0.84	0.168	0.169	1.000	0.800	0.968	0.986	22.7	2
4. CGB/IS	3.985	0.91	0.182	0.169	1.000	0.800	0.982	1.000	23.0	1
5. HD/HS	3.604	1.00	0.200	1.286	0.132	0.106	0.306	0.312	7.2	5

The vulnerability assessments are summarized in Table 17. The TS/GTS system received the highest rating, primarily because the relatively large GTS afforded protection to the C&A drive components. The CGB/ATS was rated second and reflects a smaller engine-mounted starter. The TS/IS and CGB/IS system ratings reflect no protection from an engine-mounted starter and a larger airframe-mounted starter APU. The HD/HS system received the lowest vulnerability rating because of the large exposed vulnerable area of the hydraulic pump and drive motor.

3. Development Risk - The C&A drive system/starter configurations for all systems except the HD/HS were essentially state of the art and were judged to have a similar development risk. The HD/HS system requires a convertible starter motor/pump and a C&A drive motor that must be developed to be extremely reliable. The hybrid system was therefore the only system downrated for development risk.

4. Cost - The cost of the systems was based on estimated production pricing. The development cost of the components was not reflected in the cost evaluation, with the assumption that all systems would require the same relative amount of development effort. The HD/HS system had the lowest cost because of the lack of an APU or GTS in the start system.

   The cost of the other candidate systems was estimated to be essentially the same. It should be noted, however, that if the APU used in systems Nos. 2, 3, and 4 was used for other airframe services in addition to starting, then the cost could be shared and a higher rating assessed.

5. Weight and Volume - The weight and volume of the total starter/C&A drive system was assessed on the basis of preliminary component designs. The weight rating was given 50% of the total, and the volume rating was given the remaining 50%. The weight and volume tabulations are shown in Table 18. The tabulation shows that the CGB/ATS system has the lowest overall weight and volume and received the highest rating. The TS/GTS system was second, and the rating reflected a volume penalty for the two GTS systems. The HD/HS system was rated third and had the highest weight but the second lowest volume. The CGB/IS and TS/IS systems were rated fourth and fifth, respectively, primarily because of the large APU required for starting.

6. Performance - The performance of the systems was evaluated for both starting and during normal operation. The starter efficiency was evaluated in terms of the required starter fuel flow and was given a 10% weighting in the evaluation. The C&A drive performance was weighted at 90% and evaluated on the basis of drive efficiency. A tabulation of the performance factors is shown in Table 19. The mechanical drives were all rated at the same relative efficiency. The HD/HS efficiency was lower and reflected the product of efficiencies of the tower shaft drive, hydraulic pump, and hydraulic motor.

TABLE 17. VULNERABILITY (RATING = 19%)

C&A System/ Starter Configuration		Start: Start System Volume, in.^{3}	Start: Relative Rating	Start: Relative Weight, 25%	Flight: C&A Exposed Area, in.^{2}	Flight: Relative Rating	Flight: Relative Weight, 75%	Total Weight	Relative Weight	Rating, %	Ranking
1	TS/GTS	1335	0.89	0.22	31.4	1.00	0.75	0.97	1.00	19.0	1
2	TS/IS	3206	0.37	0.09	41.2	0.76	0.57	0.66	0.68	12.9	4
3	CGB/ATS	1187	1.00	0.25	40.6	0.77	0.58	0.83	0.86	16.3	2
4	CGB/IS	3288	0.36	0.09	40.6	0.77	0.58	0.67	0.69	13.1	3
5	HD/HS	2320	0.51	0.13	85.5	0.37	0.28	0.41	0.42	8.0	5

TABLE 18. WEIGHT AND VOLUME (RATING = 10%)

C&A System/ Starter Configuration		Weight: Weight, lb	Weight: Relative Rating	Weight: Relative Weight, 50%	Volume: Volume, ft^3	Volume: Relative Rating	Volume: Relative Weight, 50%	Total Weight	Relative Weight	Rating, %	Ranking
1.	TS/GTS	138	0.79	0.395	1.91	0.70	0.350	0.745	0.75	7.5	2
2.	TS/IS	201	0.54	0.270	2.72	0.49	0.245	0.515	0.52	5.2	5
3.	CGB/ATS	109	1.00	0.500	1.33	1.00	0.500	1.000	1.00	10.0	1
4.	CGB/IS	189	0.58	0.290	2.52	0.53	0.265	0.545	0.55	5.5	4
5.	HD/HS	276	0.39	0.195	1.64	0.81	0.405	0.600	0.60	6.0	3

TABLE 19. PERFORMANCE (RATING = 9%)

C&A System/ Starter Configuration	Start			Flight						
	Fuel Flow, lb/hr	Relative Rating	Relative Weight, 10%	Drive Efficiency, %	Relative Rating	Relative Weight, 90%	Total Weight	Relative Weight	Rating, %	Ranking
1. TS/GTS	35.0	0.39	0.04	98	1.00	0.90	0.94	1.00	9.0	1
2. TS/IS	57.5	0.00	0.00	98	1.00	0.90	0.90	0.96	8.6	3
3. CGB/ATS	45.0	0.21	0.02	98	1.00	0.90	0.92	0.98	8.8	2
4. CGB/IS	57.5	0.00	0.00	98	1.00	0.90	0.90	0.96	8.6	4
5. HD/HS	0.0	1.00	0.10	72	0.74	0.67	0.77	0.82	7.4	5

7. Maintainability - The systems were evaluated on a comparative basis for maintainability. The results of the evaluation are summarized in Table 20 and show the CGB/ATS system as the easiest to maintain. Comments on the five systems that guided this rating are summarized in Table 21.

8. Installation Flexibility - Installation flexibility of the systems was evaluated, including both system volume and installation complexity. The volume was weighted at 50%, and the installation complexity was weighted the remaining 50%. The results are summarized in Table 22 and show the CGB/ATS with the highest rating. The GTS/TS system was downgraded because of the potential interference with the airframe gearbox, and the TS/IS and CGB/IS systems were downgraded because of the large APU required for start.

TABLE 20. MAINTAINABILITY (RATING = 7%)

C&A Drive/Starter Configuration		Relative Rating	Rating, %	Ranking
1.	TS/GTS	0.75	5.25	2
2.	TS/IS	0.69	4.83	4
3.	CGB/ATS	1.00	7.00	1
4.	CGB/IS	0.72	5.94	3
5.	HD/HS	0.53	3.70	5

TABLE 21. MAINTAINABILITY COMMENTS

1. TS/GTS

   a. Two GTS units are required.

   b. Tower shaft is a more complicated design, with more bearings and gears.

   c. There is less flexibility in mounting than remote APU.

   d. There are fewer components than APU-driven ATS.

   e. Fuel filters and GTS controls add maintenance.

   f. GTS mounting has good accessibility.

TABLE 21. MAINTAINABILITY COMMENTS (Continued)

2. TS/IS

   a. Requirement for ATS is eliminated.

   b. Remote APU results in good mounting flexibility.

   c. Air lines, nozzles, and filters add maintenance problems.

   d. APU and APU starter are very large and heavy for a pneumatic system.

   e. Pneumatic supply lines are not a fire hazard and subject to leakage monitoring.

   f. Air nozzles may have an accessibility problem.

3. CGB/ATS

   a. Pneumatic supply lines are not a fire hazard and subject to leakage monitoring.

   b. Remote APU results in mounting flexibility.

   c. Configuration is smallest and lightest considered.

   d. Separate APU and ATS add an additional component.

   e. Integral gearbox is less complicated than tower shaft configuration, but must be disassembled to gain access to oil pump.

   f. Pneumatic system has fewer contamination problems than hydraulic system.

4. CGB/IS

   a. Same considerations as TS/IS, except for gearbox.

   b. Integral gearbox is less complicated than tower shaft configuration, but must be disassembled to gain access to oil pump.

TABLE 21. MAINTAINABILITY COMMENTS (Continued)

5. HD/HS

   a. Hydraulic system is same as initial starter; therefore, it is relatively compact.

   b. Accessories are driven hydraulically, can be remotely mounted, and are regulated independently of engine speed.

   c. Hydraulic pumps are run full time.

   d. Hydraulic systems are subject to frequent leakage problems, and they are also a fire hazard.

   e. Hydraulic systems are sensitive to contamination

   f. System is the heaviest of all considered.

B. FRONT-DRIVE STUDY CONCLUSIONS

The analysis conducted for a front drive engine configuration with a 15-hp PTO indicates that the controls and accessory drive are best accomplished by a mechanical drive. Either a cluster gearbox or a tower shaft arrangement can be considered.

Starter system studies have indicated that the best overall engine/airframe starter choice is the gas turbine starter or the air turbine starter. In installations where the development cost of a gas turbine starter or an APU for an air turbine starter is prohibitive, then a battery/electric or accumulator/hydraulic motor start system can be considered.

Certain observations from the initial design studies were considered in the recommendations for later program phases and are summarized below.

- The requirement for provision of a PTO is a major factor in configuring the gearbox. This relatively large, low-speed drive contributes significantly to gearbox size.

- Control and accessory packaging is compromised by a front-drive power turbine. Integral or direct-drive components are not possible.

- The basic engine design is not optimized for a front-drive power turbine, as compared to a rear drive, because of the impact of the power turbine shaft on the gas generator bore diameter.

TABLE 22. INSTALLATION FLEXIBILITY (RATING = 6%)

C&A System Starter Configuration	Volume: Volume, in^3	Volume: Relative Rating	Volume: Relative Weight, 50%	Location and Connection: Assigned Weight	Location and Connection: Relative Rating	Location and Connection: Relative Weight, 50%	Total Weight	Relative Weight	Rating, %	Ranking
1. TS/GTS	1335	0.89	0.445	30.6	0.61	0.305	0.745	0.789	4.7	3
2. TS/IS	3206	0.37	0.185	40.0	0.80	0.400	0.585	0.619	3.7	5
3. CGB/ATS	1187	1.00	0.500	44.5	0.89	0.445	0.945	1.00	6.0	1
4. CGB/IS	3288	0.36	0.180	41.6	0.83	0.415	0.595	0.630	3.8	4
5. HD/HS	2320	0.51	0.255	50.0	1.00	0.500	0.755	0.798	4.8	2

SECTION V

CONCEPTUAL DESIGN - REAR-DRIVE ENGINE

The study was continued to define the optimum C&A system for a rear-drive engine. The requirements of the baseline engine and C&A components were established. Engine definition studies were conducted and led to definition of the baseline engine configuration. The candidate C&A drives and starter techniques were defined, and a matrix of possible C&A drive/starter systems was established. The candidate systems were analyzed, and two systems were recommended for additional study.

A. ENGINE DEFINITION STUDIES

Prior to establishing the baseline rear-drive engine configuration, engine definition studies were conducted in several areas to evaluate feasibility of selected candidate configurations.

1. Air Bearing Application Study

The front-drive engine studies showed that the engine oil system represented a major portion of the engine's vulnerable area. To evaluate complete elimination of the oil system, further analyses were conducted to determine if an all-air-bearing engine was within the 1977 development time frame.

A preliminary engine layout, shown in Figure 39, was established, employing all air bearings. From this drawing, shaft critical speeds and air bearing loads were evaluated. The gas generator configuration has a rotor bending mode at 47,800 rpm, which is less than the expected idle speed. For the engine to pass through this mode without intolerable bearing loads and/or deflections, multi-plane balancing (which is not currently state of the art) would be required. The power turbine was not analyzed, but its critical speed was estimated to be acceptable.

This analysis showed that the thrust bearing size and location (more distance between bearings) created critical speed problems. Improvements in the gas generator critical speed could be obtained by reducing the diameter of the thrust bearing piston and the distance between bearings.

Bearing thrust and radial loads were established as shown below:

1. Engine Unbalanced Loads

   Gas Generator Thrust = 370 lb
   Gas Generator Radial = 77 lb
   Power Turbine Thrust = 1135 lb

2. Engine Maneuver Loads

   Gas Generator Thrust = 28 lb x 10g = 280 lb
   Gas Generator Radial = 14 lb x 10g = 140 lb
   Power Turbine Thrust = 15 lb x 10g = 150 lb

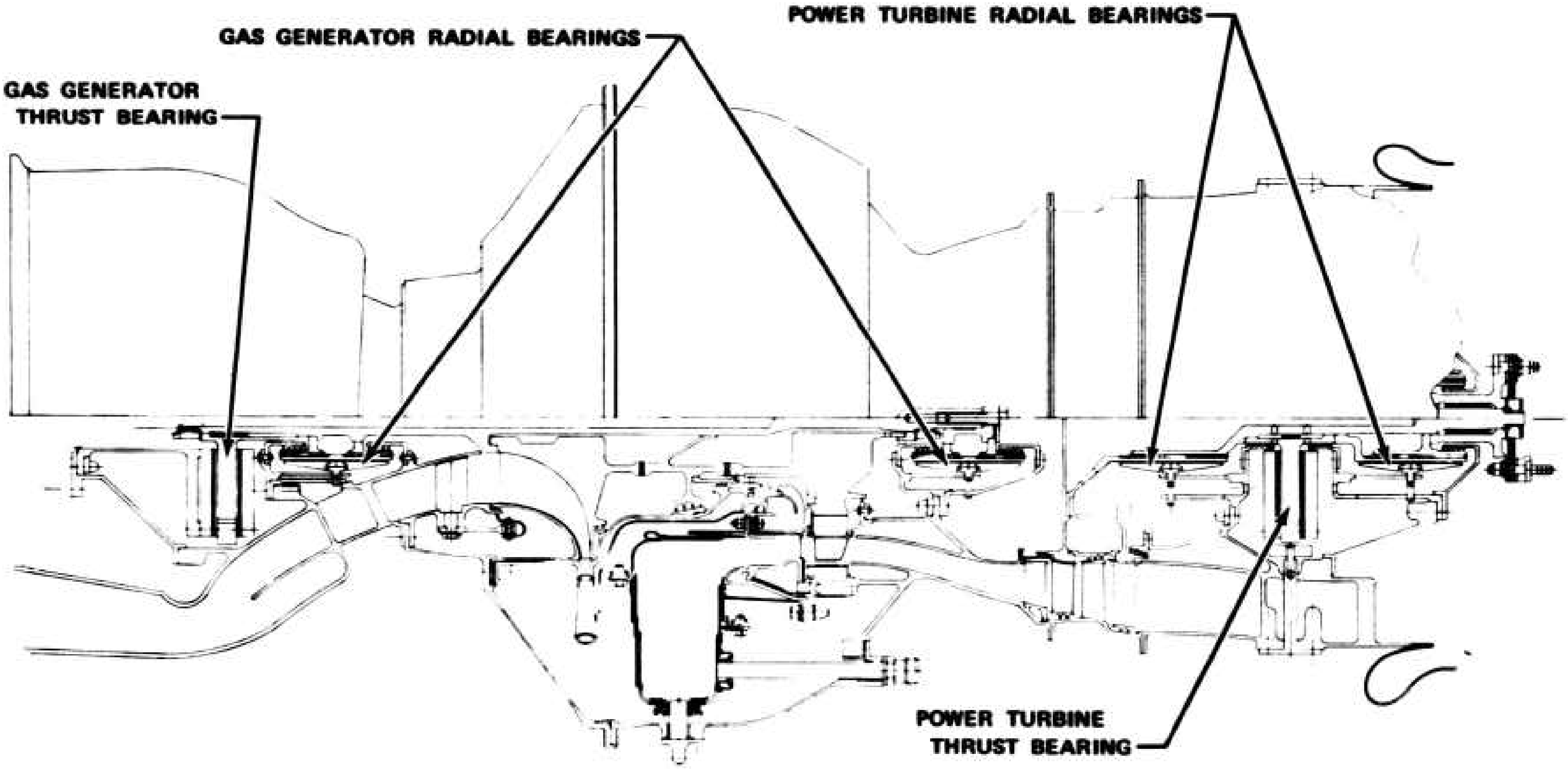


Figure 39. Air Bearing Engine Configuration

Based on the loads established, consideration was given to the feasibility of an all-air-bearing engine which would be consistent with a 1977 development time in frame. This was accomplished by reviewing experimental data and through conversations with Mechanical Technology, Incorporated (MTI). The literature survey indicated that the necessary analytical tools needed to determine operating characteristics of air bearings exist today, but the test data necessary to substantiate these theoretical predictions for air thrust bearings do not. Based on discussions with MTI, the bearing load/area cannot exceed 18 to 20 $lb/in.^2$ for thrust bearings and 40 to 50 $lb/in.^2$ for radial bearings. From thrust and radial loads calculated, radial bearings for the gas generator offer no problem at 12.3 $lb/in.^2$. The power turbine radial load/area was not calculated, but it is expected to be in the same order of magnitude. Thrust bearing load/area was much larger than allowable, with the gas generator at 32.5 $lb/in.^2$ and the power turbine at 64 $lb/in.^2$.

Based on present technology, thermal and dynamic instabilities are problems associated with the thrust balance pistons of the size used in the conceptual layout. Increasing the piston diameter, as would be required to lower the load per unit area, would increase instability problems. The effect or severity of bearing instability could be evaluated only by appropriate testing.

The adverse impact on engine critical speed of a larger thrust balance piston also is not desirable. Thrust balancing techniques are not expected to change by 1977. The load area ratio for the gas generator could be reduced by baselining an engine using a shrouded centrifugal compressor in combination with a radial inflow turbine. The thrust balance piston on the power turbine could be eliminated by allowing the airframe gearbox to support the power turbine thrust load. A conceptual airframe-gearbox-supported power turbine is shown in Figure 40.

In summary, for the selected baseline engine configuration, radial air bearings appear to be within the 1977 development time frame, while thrust bearings do not. The power turbine load distribution requirements greatly exceed the demonstrated thrust bearing capabilities. The gas generator thrust load distribution requirements are much closer to the demonstrated levels, but are sufficiently in excess of these levels to also be considered beyond the 1977 development time frame.

An engine without an oil system is still a most desirable configuration and should be further pursued. For this application, complete elimination of the oil system would require the following efforts: (1) further experimental development of air thrust bearings for the gas generator to increase their load-carrying capability by a factor of 2 to 3, and (2) use of an engine/airframe interface, where the power turbine is supported by the airframe gearbox.

2. Bearing Configuration/Arrangement Study

The bearing configuration and arrangement for the baseline engine were established from an optimization trade-off study. The study indicated that at least one oil-lubricated bearing would be required on each rotor to support the unbalanced thrust loads.

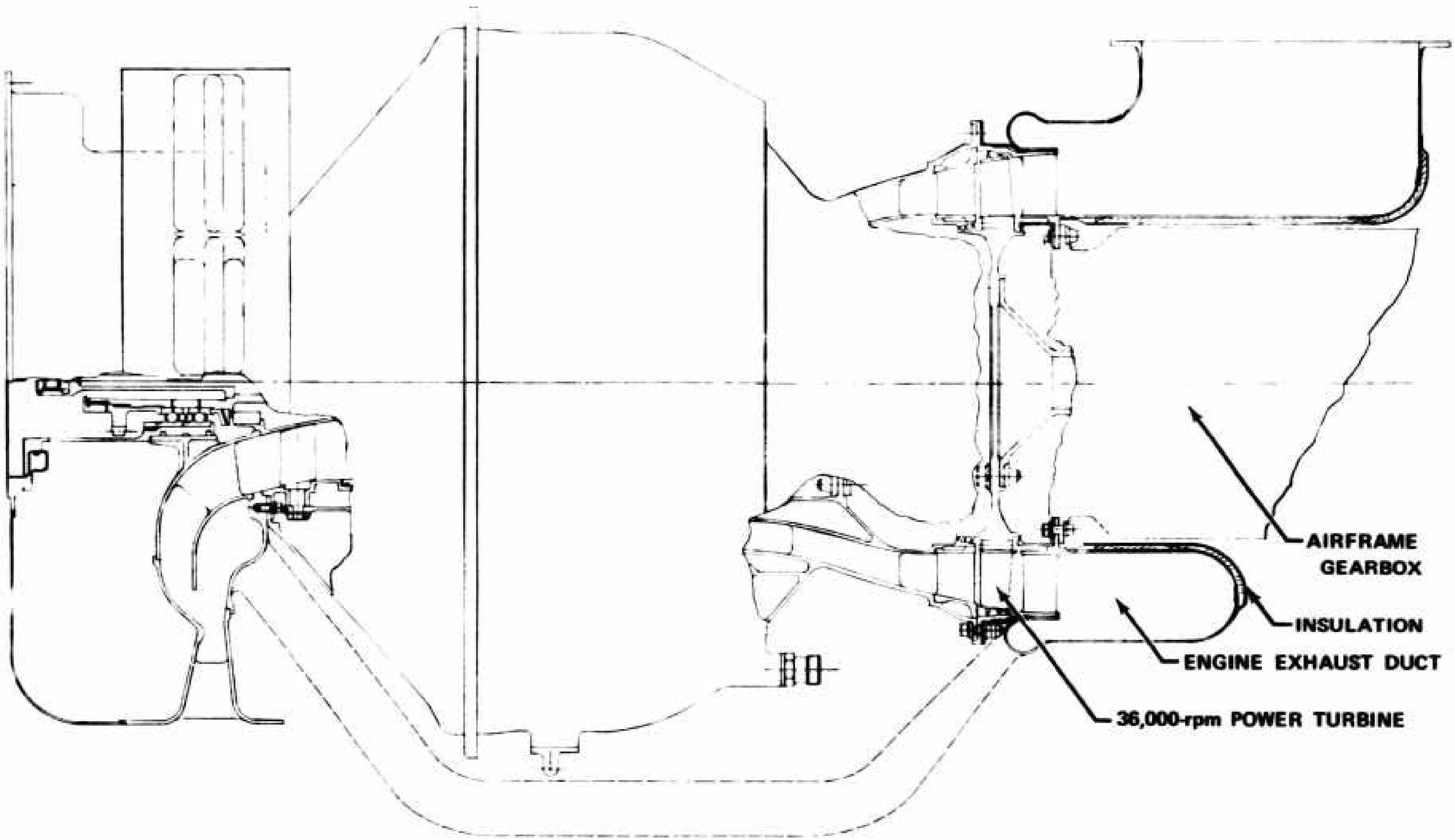


Figure 40. Airframe Gearbox-Supported Power Turbine

For the gas generator rotor, the forward bearing location was chosen for the thrust bearing. The forward location allowed close control of the centrifugal compressor clearances, which would not be possible with a thrust bearing location in the rear, since dimensional tolerances and thermal growth differences of the rotor assembly, relative to the compressor case, would require excessive compressor clearances. A radial air bearing was selected for the rear support of the gas generator, based on the adequate load-carrying capability of air bearings and the benefit of reducing the overall oil system volume and weight.

The heat generation summary for the selected configuration is shown in Table 23.

TABLE 23. BASELINE ENGINE HEAT GENERATION SUMMARY

	Heat Generation ~ Btu/min	
Compartment Location	SLS	Ground Idle
Front Compartment Ambient	-47.8	-41.1
Middle Compartment Ambient	Air Cooled	Air Cooled
Rear Compartment Ambient	+17.7	+3.1
Compartment Seal Friction	All Self-Acting ≈ 0	All Self-Acting ≈ 0
Seal Leakage	Negligible	Negligible
Gas Generator Thrust Bearing (Tandem)	66	42.5
Power Turbine Thrust Bearing	24.1	24.1
Gas Generator Journal	Air Cooled	Air Cooled
Power Turbine Journal	Air Cooled	Air Cooled
Gearbox	$\approx$58.0	< 58.0
Total	+118	<86.6

The bearing configuration for the power turbine was selected by comparison and rating of several candidate configurations. Study layouts of the configurations were made and relative comparison of the candidates accomplished. The candidate comparisons are outlined in Table 24.

The No. 4 candidate (rear overhung turbine with a forward air bearing, and a rear oil-lubricated bearing) was selected based on the shortest coupling length and superior vulnerability resistance. This configuration also had equal or superior ratings in all other areas when compared to the other candidates.

TABLE 24. BEARING CONFIGURATION CANDIDATES

	Baseline	2	3	4
Turbine Location	Forward	Forward	Center	Rear
Front Bearing Type	Oil Lubricated	Air	Air	Air
Rear Bearing Type	Oil Lubricated	Oil Lubricated	Oil Lubricated	Oil Lubricated
Air Seals	Two Shaft	Three Shaft	Three Shaft	Two Shaft
Oil Seals	Two Shaft	Two Shaft	Two Shaft	Two Shaft
Coupling Length	Baseline	Same as Baseline	Same as Baseline	Slightly Less Than Baseline
Overall Length	Baseline	Same as Baseline	Same as Baseline	Same as Baseline
Oil System Wetted Area	Baseline	Less Than Baseline	Much Less Than Baseline	Much Less Than Baseline
Oil System Heat Load	Baseline	Less Than Baseline	Less Than Baseline	Less Than Baseline
Bearing Compartment Services Required	One Air Supply One Oil Supply One Oil Scavenge	Two Air Supply One Oil Supply One Oil Scavenge	One Air Supply One Oil Supply One Oil Scavenge	One Air Supply One Oil Supply One Oil Scavenge
Critical Speed	OK	OK	OK	OK
Vulnerability Resistance	Baseline	Same as Baseline	Same as Baseline	Better Than Baseline

3. Integral Start Turbine

The integral start technique, previously selected and described in Section IV, used directional jets that impinged on buckets machined on the gas generator turbine. This approach, while simple in concept, required a relatively large APU (1.26 lb/sec at a pressure ratio of 2.36). The C&A systems using this integral start technique were, therefore, penalized due to the weight, volume, and vulnerability considerations of the APU.

In an attempt to make the integral start system a more viable candidate, consideration was given to means of improving system efficiency. A design evaluation was made of a separate, dedicated starter turbine integral with the gas generator rotor. The configuration evaluated is shown in Figure 41. This system used a 5.02-in. diameter turbine and a 30% admission nozzle, and required an APU flowrate of 0.37 lb/sec at a pressure ratio of 4.0.

The disadvantages of this design prevented its application. The most significant was the impact on engine critical speed. The starter turbine mass and increased engine rotor length reduced the critical speed margin below the desired 30%, considered minimum for the engine design. The second most significant impact was on overall engine performance. A 10-hp parasite drag was predicted for the starter turbine at 100% engine operating speed, and 1% of engine airflow leakage was predicted due to additional dynamic seal requirements. Pressure losses in the turbine exhaust case because of the large struts required for the start turbine air inlet and exhaust passages were anticipated, but were not evaluated. The turbine added 2.1 in. to the engine length and 8.1 lb to the overall weight. Therefore, this integral starter turbine was not recommended, and the impingement starter system was retained as the prime integral starter candidate.

B. BASELINE ENGINE DEFINITION

The engine requirements for the rear drive configuration were identical to those described in Section II, except that this engine (1) has a single-stage rear drive power turbine with a design speed of 36,000 rpm and (2) does not have a 15-hp, 20,000-rpm PTO.

The axial power turbine was sized to operate at 36,000 rpm to obtain the desired operating efficiency with a single stage. Since the rear-drive engine facilitates use of a short power turbine shaft, critical speed problems are not encountered at this higher rotor speed, as compared to the long shaft required for the front-drive engine configuration. A lower power turbine speed could be provided at the expense of a second turbine stage.

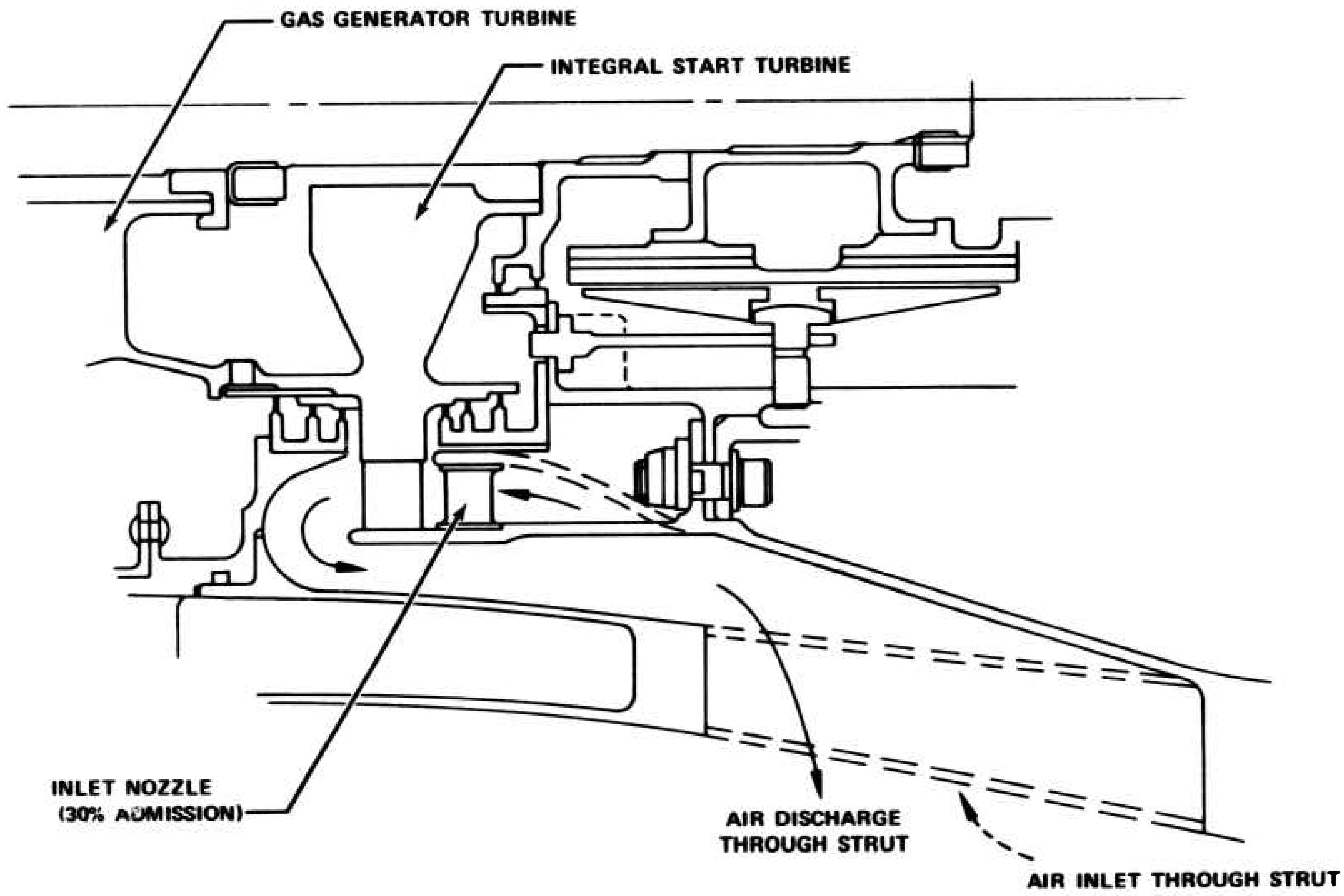


Figure 41. Integral Start Turbine

The baseline engine bearing and seal configuration is described as follows:

1. Front Bearing Compartment

   Oil-Lubricated, Antifriction Bearings
   Hydrodynamic Lift-off Seals
   Elastomeric Damper

2. Middle Bearing Compartment

   Tilting Pad Radial Air Bearings

3. Rear Bearing Compartment

   Oil-Lubricated, Antifriction Bearings
   Hydrodynamic Lift-off Seals

4. Heat Exchanger

   Air/Oil Heat Exchanger - Integral With Front Bearing Compartment
   Oil Temperature - 200 to 250°F

5. Backup Oil System

   Oil Mist - 6-min Capability

A cross-sectional view of the baseline engine configuration is shown in Figure 42.

C. CANDIDATE C&A DRIVE/STARTER DESCRIPTIONS

During this program phase, the method of driving the C&A components and the method of starting the baseline engine were considered. Systems that were evaluated as not technically applicable or beyond the state of the art for a 1977 development time frame during the study of a front-drive engine were not considered if the design requirements for application to a rear-drive engine were essentially the same. The C&A drive systems and the starter systems initially considered in the study are outlined below.

1. Candidate C&A Drives

   a. Mechanical

      (1) Tower Shaft Drive - Tower shaft drive through a gearbox with multiple gearing for the various required accessory drives

      (2) Cluster Gearbox - Cluster gearbox mounted about centerline of engine with multiple gearing for the various required accessory drives

      (3) Single-Speed Drive - Tower shaft or cluster gear drives a single-speed drive shaft. All required accessory drives would run in a tandem arrangement at the same speed from this shaft.

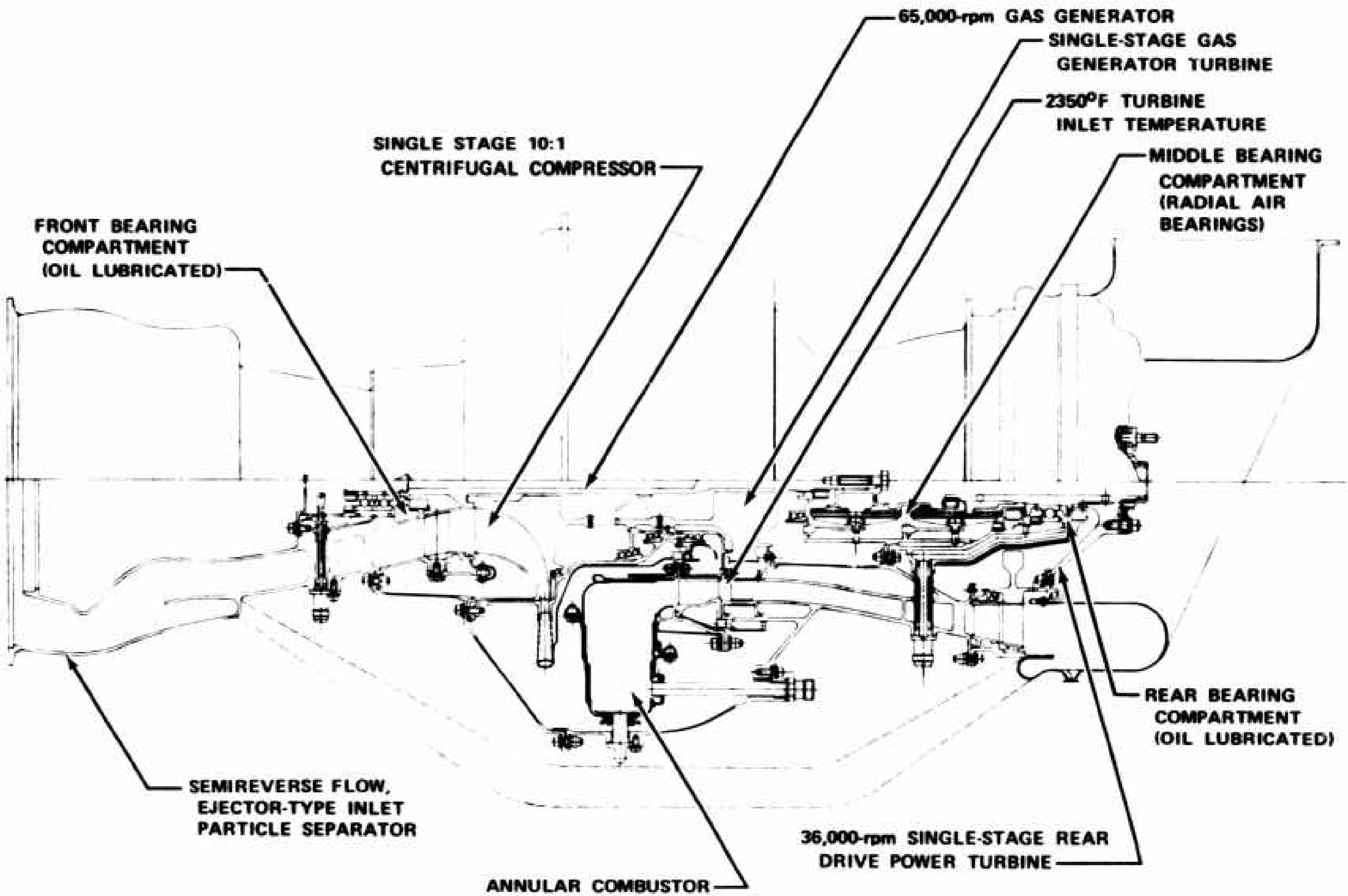


Figure 42. Baseline Engine Configuration

b. Pneumatic

(1) Cold Gas Bleed - Engine airflow gas bleed upstream of the burner is used for providing power for the required accessory drives.

(2) Bleed and Burn - Engine airflow gas bleed upstream of the main burner is fed through a separate combustor, mixed with a fuel supply, and burned. The combustion products are then used to provide power for the required accessory drives.

(3) Interturbine Bleed - Engine airflow gas bleed downstream of the gas generator turbine is bled and used to provide power for the required accessory drives.

(4) Mixed Bleed - Engine airflow gas bleed is bled from the engine, both upstream of the burner and downstream of the gas generator turbine. The hot and cold gases are mixed to provide power for the required accessory drives.

c. Hybrid

(1) Any C&A Drive/Integral Generator - All accessories are driven by mechanical or pneumatic power with an electrical generator integrated with the gas generator rotor to supply electrical power for the control and ignition system.

(2) Mechanical/Pneumatic Interface - A tower shaft is used to drive a separate compressor. The compressor is used to power the required accessory drives.

(3) Mechanical/Hydraulic Interface - A tower shaft is used to drive a separate hydraulic pump. The hydraulic pump is used to power the required accessory drives.

(4) Mechanical Drive/Electrical Fuel Pump Interface - An electric generator, integrated with the gas generator rotor, will provide electric power for driving a variable-speed fuel pump. All other accessories are driven mechanically.

(5) Mechanical Drive/Electrical or Pneumatic Oil Pump Interface - An electric generator, integrated with the gas generator rotor, or a pneumatic bleed from the engine compressor will provide power for a variable-speed oil pump. All other accessories are driven mechanically.

2. Candidate Starter Systems

a. Mechanical

(1) Mechanical/Electric - An electrically powered starter driving through a gearbox connected to the gas generator rotor. Starting power is provided by APU/generator or battery system.

(2) Mechanical/Hydraulic - A hydraulic-powered starter driving through a gearbox connected to the gas generator rotor. Starting power is provided by APU/hydraulic pump or accumulator blowdown.

(3) Mechanical/Pneumatic - A pneumatic-powered starter driving through a gearbox connected to the gas generator rotor. Starting power is provided by APU bleed.

(4) Self-Contained Starter - A self-contained gas turbine, piston, or Wankel engine mechanically connected to the gas generator rotor.

b. Integral

(1) Integral Pneumatic - An external APU supplying bleed air, which is used in either of the three following methods: (1) cold gas, (2) heat addition through combustion in the gas generator burner with closed engine inlet, and (3) hot gas using heat addition through combustion in an external burner.

1. Selection of Six Candidate Systems

The C&A drive systems considered for this phase were divided into 12 schemes. The starter systems were broken into two basic divisions: mechanical drive input and integral starters. A matrix of the candidate systems is shown in Table 25. The matrix consists of the 12 C&A drive schemes, combined with the 2 basic starter drive systems, including 7 types of starter drive schemes. The total combinations of C&A drives and starters created 84 possible systems for study.

Each candidate system was reviewed in light of the trade-off studies and analyses conducted during the analysis of a front-drive engine. If the analysis was directly applicable, the same conclusions were used. If the requirements had changed for the rear-drive engine, the previous results were reviewed in light of the new requirements, and additional analyses were conducted where required.

TABLE 25. CANDIDATE COMPONENT AND ACCESSORY DRIVE SYSTEMS

			Accessory Drives											
			Mechanical			Pneumatic				Hybrid				
			Tower Shaft	Cluster Gearbox	Single-Speed Module	Cold Gas Bleed	Bleed and Burn	Interturbine Bleed	Mixed Bleed	Any C&A Drive/ Integral Generator	Mechanical/ Pneumatic Inter-face	Mechanical/ Hydraulic Inter-face	Mechanical Drive Electrical Fuel Pump	Mechanical Drive/ Electrical Or Pneumatic Oil Pump
Starters	Mechanical	Mechanical/Electric	5-S	5-S	5-S	5-S 2-D	5-S 2-D	5-S 2-D	5-S 2-D	5-S	3-S 5-S 5-D	3-S 5-S 5-D	5-S 5-D	5-S 5-D
		Mechanical/Hydraulic	5-S	5-S	5-S	5-S 2-D	5-S 2-D	5-S 2-D	5-S 2-D	5-S	3-S 5-S 5-D	5-S 5-D	3-S 5-S 5-D	5-S 5-D
		Mechanical/Pnuematic	1	1	6-S	2-D	2-D	2-D	2-D	6-S	5-D	3-S 5-D	3-S 5-D	5-D
		Self-Contained	4-S	4-S	4-S	4-S 2-D	4-S 2-D	4-S 2-D	4-S 2-D	3-S	3-S 5-D	3-S 5-D	3-S 5-D	3-S 5-D
	Integral	Pneumatic - Cold Gas Bleed	1	1	1	5-D	5-D	5-D	5-D	1	5-D	5-D	5-D	5-D
		Pneumatic - Engine Ram, Closed IGV's	5-S	5-S	5-S	5-S 5-D	5-S 5-D	5-S 5-D	5-S 5-D	5-S	5-S 5-D	5-S 5-D	5-S 5-D	5-S 5-D
		Pneumatic - Hot Gas Impingement	5-S	5-S	5-S	5-S 5-D	5-S 5-D	5-S 5-D	5-S 5-D	5-S	5-S 5-D	5-S 5-D	5-S 5-D	5-S 5-D

S - Starter Consideration
D - Drive Consideration
1 Considered a candidate C&A drive/starter system
2 If a mechanical drive is required for start, there is no advantage to another drive technique for the accessories.
3 For hybrid systems, the starter and accessory drive modes should be common.
4 Excessive packaging penalty for available space
5 Discounted on the basis of Phase I trade-off studies
6 Starter horsepower requirements were not compatible with drive.

Six candidate systems were selected for evaluation and are outlined below:

1. Tower Shaft Drive/Air Turbine Starter
2. Tower Shaft Drive/Impingement Starter
3. Cluster Gearbox Drive/Air Turbine Starter
4. Cluster Gearbox Drive/Impingement Starter
5. Single-Speed Drive/Impingement Starter
6. Single-Speed Drive/Impingement Starter/Integral Alternator

An outline/discussion of the rationale and analyses required to obtain the selected systems follows:

1. Candidate C&A Drives

   a. Mechanical

      (1) Tower Shaft - Considered a candidate

      (2) Cluster Gearbox - Considered a candidate

      (3) Single-Speed Drive - Considered a candidate with the module operating at 15,000 rpm.

   b. Pneumatic

      The pneumatic drive system selected during the front-drive engine study was the cold gas bleed. This selection was also considered applicable to the rear-drive engine.

      During the front-drive analysis, the best pneumatic drive was compared with a mechanical drive, and the mechanical drive was selected, based on advantages in all areas except maintainability and installation flexibility with overall ratings of 92.1 (mechanical) vs 69.1 (pneumatic). The reduced performance penalty for a pneumatic drive for this application (less bleed air required due to deletion of PTO) would not significantly alter this evaluation. Therefore, pneumatic drive systems were discounted in favor of mechanical drives.

   c. Hybrid

      (1) Any C&A Drive/Integral Generator

      An integral generator was discussed and discounted for the front-drive engine, based primarily on the fact that several component drive pads were available that offered more favorable opportunities for integration of the generator as compared to the basic engine. For the rear-drive engine application, the integral generator was considered a candidate only for the single-speed module drive operating at 15,000 rpm.

(2) Mechanical/Pneumatic Interface

The front-drive analysis showed that the mechanical/hydraulic interface was preferred over the mechanical/pneumatic system, based on superior or equal ratings in all areas except reliability and maintainability. The overall ratings were 94.0 (hydraulic) vs 85.6 (pneumatic). This system selection was also considered applicable to a rear-drive engine without PTO, since the drive system components are sized by the starter requirements.

(3) Mechanical/Hydraulic Interface

Front-drive analyses of a mechanical/hydraulic drive showed that the mechanical/hydraulic drive had lower overall ratings in the areas of reliability, vulnerability, development risk, performance, and maintainability as compared to any mechanical drives. This analysis was also considered valid for a rear-drive engine without PTO since the drive system components were sized by the starter requirements.

(4) Mechanical Drive/Electrical Fuel Pump Interface

During the front-drive engine study, consideration was given to electrically driving the fuel pump from an integral generator. This system was discounted in favor of a mechanical drive based on higher ratings of the mechanical drive in all areas except performance, maintainability, and installation flexibility with an overall rating of 91.1 (mechanical) vs 79.6 (mechanical/electric). The same conclusions are applicable to a rear-drive engine without PTO.

(5) Mechanical Drive/Electrical or Pneumatic Oil Pump Interface

During this phase, the study was expanded to cover the possibility of an electrically or pneumatically driven oil pump. Since the oil system flow requirements are essentially a direct function of engine speed, no engine performance improvements were anticipated when compared to a mechanical drive. The advantage of an electrically or pneumatically driven oil pump would be eliminating or simplifying the gearbox. Since the baseline engine definition studies defined an engine with an oil-lubricated, gas generator forward bearing, no significant simplification of the overall oil system would be realized for this application. Based on the previous study of an electrically driven fuel pump, the only advantage to an electrically or pneumatically driven oil pump as compared to a mechanical drive would be in maintainability and installation flexibility with disadvantages in all other areas. Therefore, the electrical or pneumatic oil pump drive was discounted.

2. Candidate Starter System

The starter requirements for a rear-drive engine are the same as for a front-drive engine. The starter system studies conducted during the previous analysis were considered applicable. The candidate starter systems are outlined below.

a. Mechanical

(1) Mechanical/Electric - This system was discounted on the basis of the front-drive studies, which rated an APU/pneumatic starter superior to a battery/ electric starter. The overall ratings were 69.2 (electric) vs 85.0 (pneumatic).

(2) Mechanical/Hydraulic - This system was discounted on the basis of the previous studies, which rated an APU/pneumatic starter superior to an accumulator/ hydraulic starter. The overall ratings were 71.8 (hydraulic) vs 85.0 (pneumatic).

(3) Mechanical/Pneumatic - An air turbine starter system was considered a candidate.

(4) Self-Contained - A self-contained gas turbine starter (GTS) rated overall in the front-drive study as slightly better than an air turbine starter (96.4 GTS vs 95.6 ATS). The GTS would be used only where the envelope would allow. If an APU is required for any other airframe purpose, then an ATS would be preferred.

For this application, the starter packaging requirements are more critical because of the reduced gearbox size without PTO. For this study, no advantage in using a GTS as compared to an ATS was indicated.

b. Integral-Pneumatic

(1) Cold Gas Bleed - This system was considered a candidate.

(2) Engine Ram/Closed IGV's - This system was discounted on the basis of the front-drive studies, which rated a cold gas bleed system over an engine ram closed IGV's system based primarily on reduced development risk with an overall rating of 82.3 (Cold Gas Bleed) vs 81.6 (Engine Ram/Closed IGV's).

(3) Hot Gas Impingement - This system was also discounted on the basis of the previous studies, which rated a cold gas bleed system superior or equal to a hot gas impingement system in all areas except weight and volume with an overall rating of 82.3 (Cold Gas Bleed) vs 72.3 (Hot Gas Impingement).

2. Selection of Two Candidate Systems

The six C&A drive/starter systems were evaluated during this task. Preliminary design layouts of these systems were made, and each system was evaluated based on the selection criteria previously defined. From these candidates, the two C&A systems with the highest overall rating were selected for the detail design phase of the program. These systems were:

1. Tower Shaft Drive/Air Turbine Starter
2. Cluster Gearbox Drive/Air Turbine Starter

A discussion relating to the selection of the two candidates is presented in the following paragraphs.

a. Candidate C&A Drive/Starter Description

(1) Tower Shaft Drive/Air Turbine Starter (TS/ATS)

A tower shaft drive is used to drive a 65,000-rpm fuel pump/alternator package and a 15,000-rpm oil pump. An air turbine starter is coupled to the gas generator shaft through a 1:1 ratio bevel gear train. The air turbine starter consists of a high-speed air turbine with a cluster gear reduction. An overrunning clutch decouples the start system when the engine is self-sustaining. The TS/ATS configuration is shown in Figure 43.

(2) Tower Shaft Drive/Impingement Starter (TS/IS)

A tower shaft drive, as described for the TS/ATS configuration, is used for the C&A components. Starting is accomplished by cold gas impingement on buckets machined on the back side of the gas generator turbine. The bleed air is provided by an airframe-mounted APU. The TS/IS configuration is shown in Figure 44.

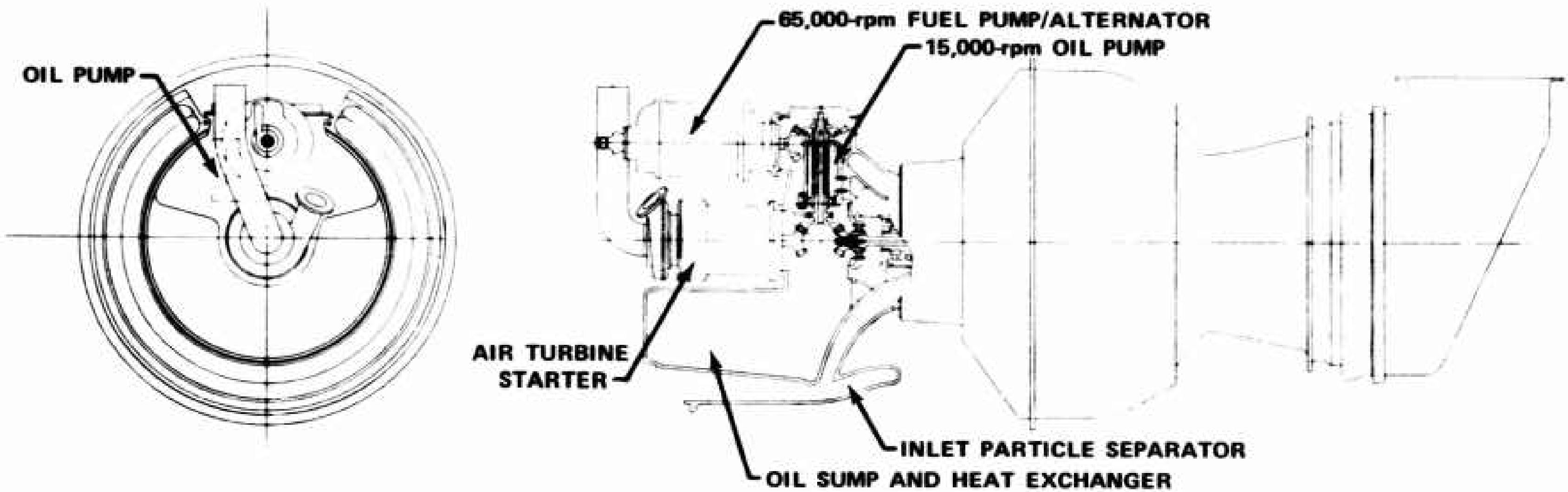


Figure 43. Tower Shaft Drive/Air Turbine Starter

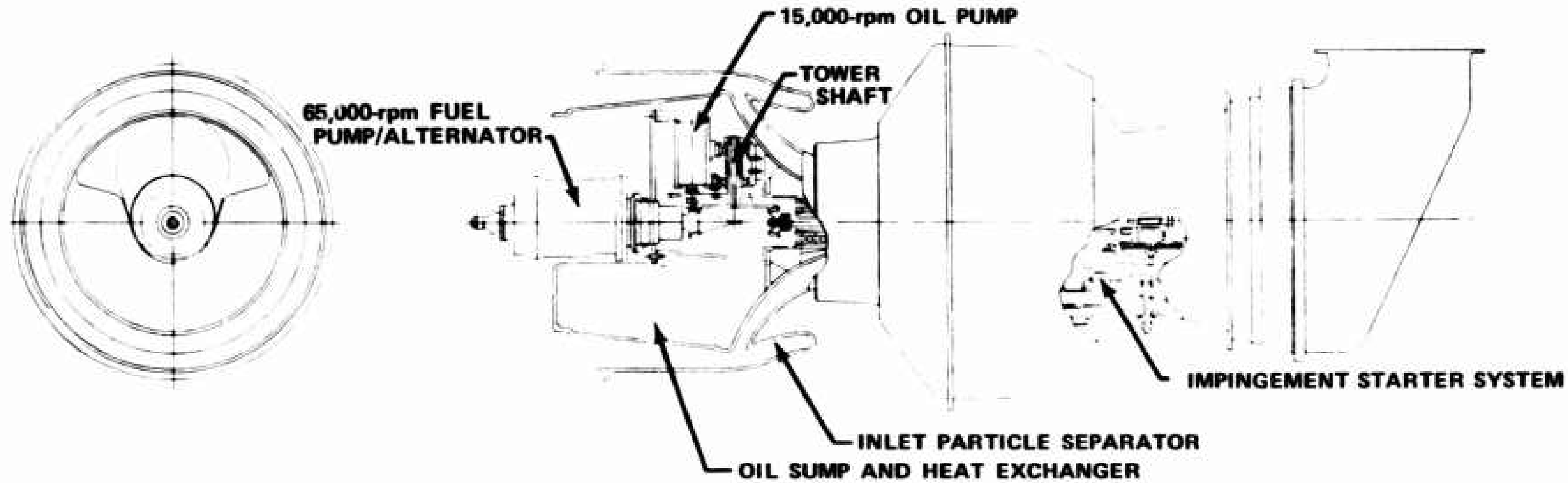


Figure 44. Tower Shaft Drive/Impingement Starter

(3) Cluster Gearbox Drive/Air Turbine Starter (CGB/ATS)

The drives for the C&A components are provided by gears clustered around the gas generator rotor. The C&A components and starter system are identical to those described for the TS/ATS. The CGB/ATS configuration is shown in Figure 45.

(4) Cluster Gearbox Drive/Impingement Starter (CGB/IS)

The C&A drive train is the same basic design as the CGB/ATS configuration described above. Starting is accomplished by the impingement start system described for TS/IS system. The CGB/IS configuration is shown in Figure 46.

(5) Single-Speed Drive/Impingement Starter (SSD/IS)

A single-speed shaft (15,000 rpm) is used to drive all C&A components. A 15,000-rpm integral fuel pump/alternator and a 15,000-rpm oil pump are coaxial with the drive shaft. The drive shaft is coupled to the gas generator rotor through a reduction ring-gear/pinion arrangement. Starting is accomplished by the impingement start system. The SSD/IS configuration is shown in Figure 47.

(6) Single-Speed Drive/Impingement Starter/Integral Alternator (SSD/IS/IA)

This drive configuration is similar to the SSD/IS, but has the alternator coupled directly to the gas generator rotor. The 65,000-rpm alternator design is used, and other components operate at 15,000 rpm. Starting is accomplished by the impingement start system. The SSD/IS/IA configuration is shown in Figure 48.

b. Analysis

Assessment of the six systems was made considering each of the rating criteria defined previously. The results are summarized in Table 26 and show the TS/ATS and the CGB/ATS as the two systems with the highest ratings. Discussion of the ratings is outlined below.

(1) Reliability

A reliability assessment was made for both the starter system and the C&A drive. The start system reliability was given a 20% weighting and the C&A drive an 80% weighting. The results of the analysis are summarized in Table 27. The air turbine starter system was rated as less reliable than the impingement starter due to the addition of the ATS to the pneumatic scheme. Gearbox reliability was based on total number of bearings, number of floating splines, and number of gear engagements contained in each gearbox. The SSD was downrated, primarily due to the number of bearings and gear engagements involved in the ring-gear/pinion coupling at the gas generator shaft.

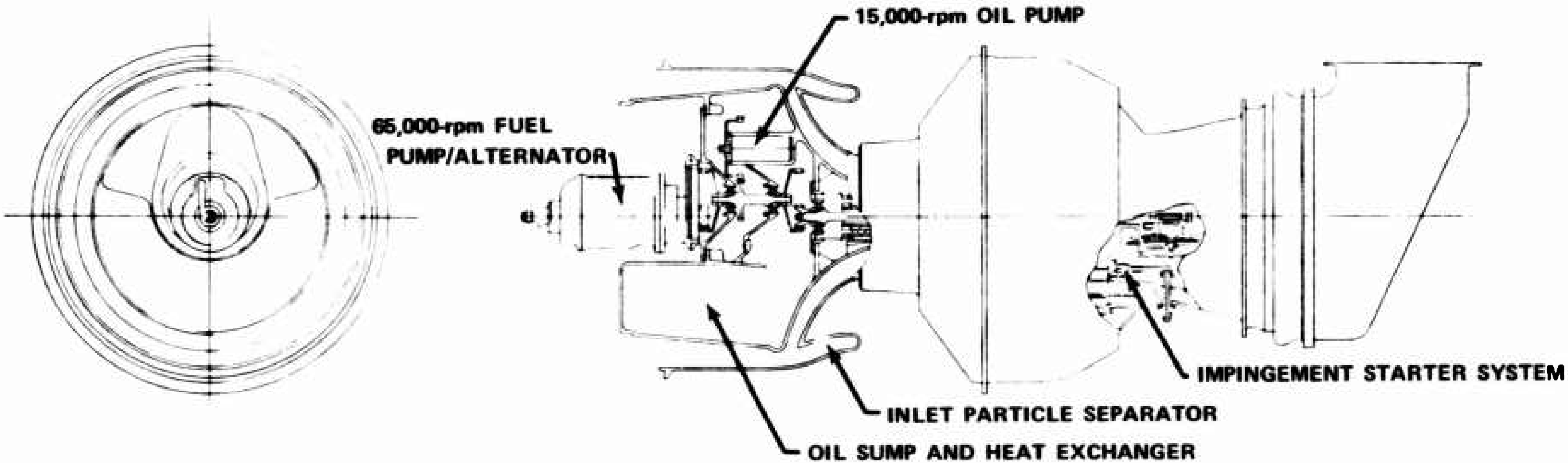


Figure 45. Cluster Gearbox Drive/Air Turbine Starter

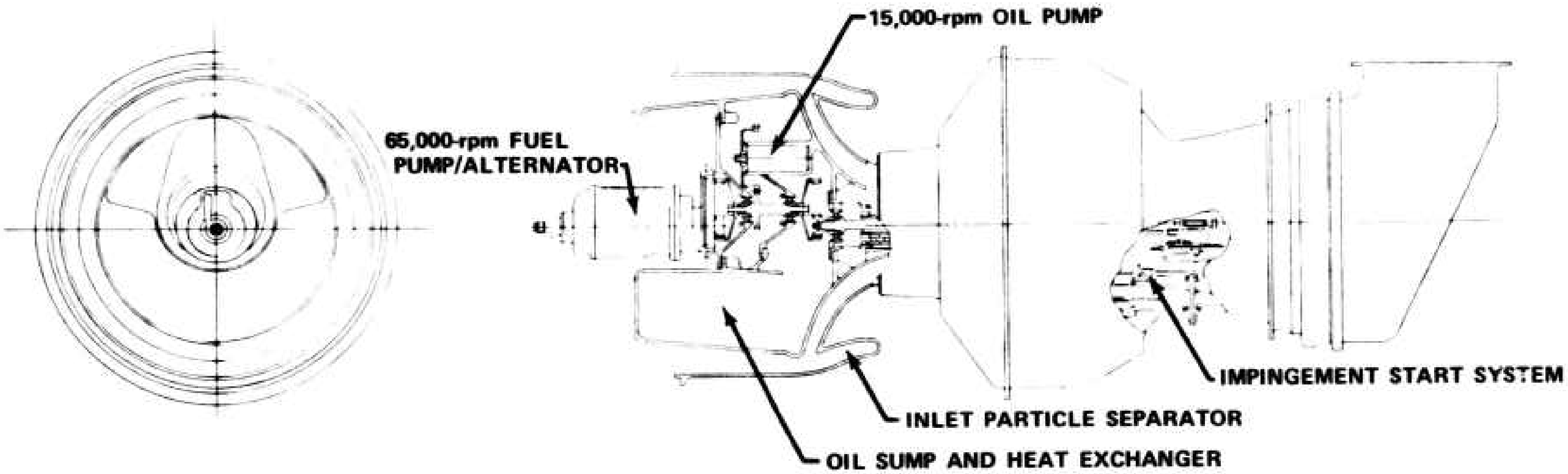


Figure 46. Cluster Gearbox Drive/Impingement Starter

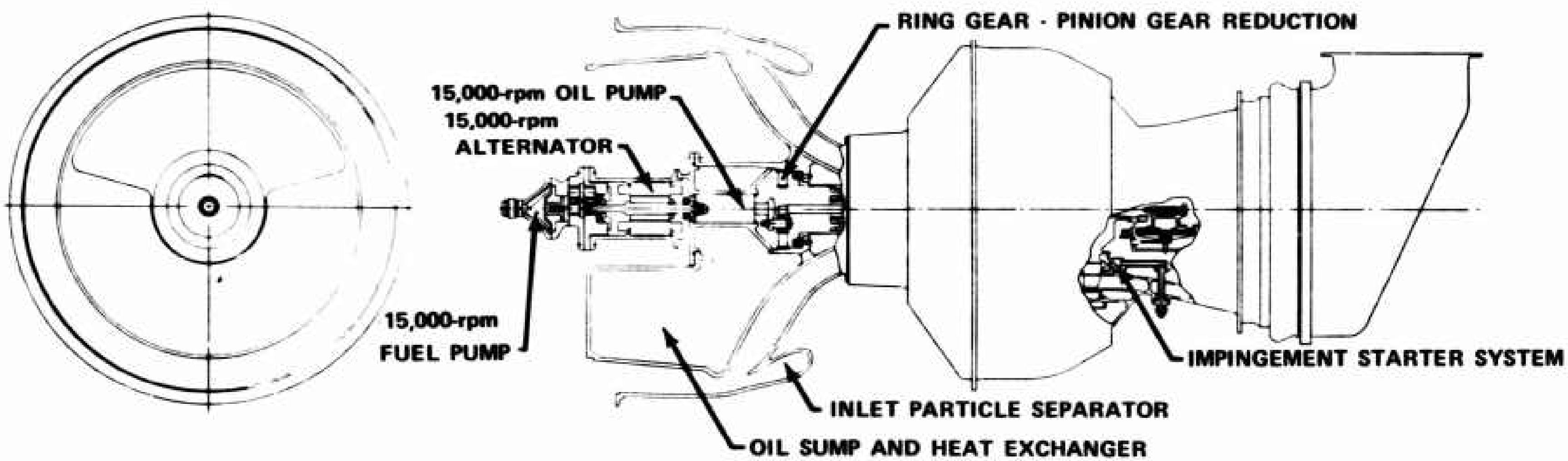


Figure 47. Single-Speed Drive/Impingement Starter

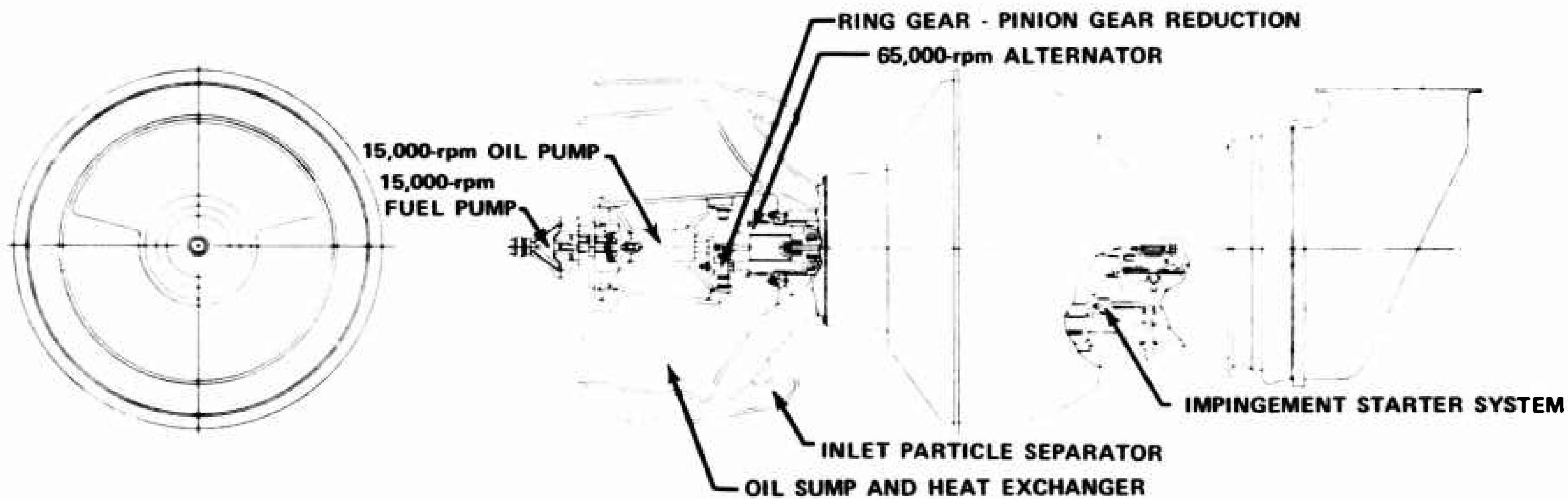


Figure 48. Single-Speed Drive/Impingement Starter/Integral Alternator

TABLE 26. RATING SUMMARY

C&A Starter Configuration	Reliability, 23%	Vulnerability, 19%	Development Risk, 14%	Cost, 12%	Weight and Volume, 10%	Performance, 9%	Maintainability, 7%	Installation Flexibility, 6%	Total Rating, %	Ranking
TS/ATS	21.0	19.0	12.6	11.4	10.0	9.0	6.8	6.0	95.8	1
TS/IS	23.0	11.1	12.6	11.3	4.5	8.7	6.2	4.6	82.0	3
CGB/ATS	21.0	16.4	12.6	11.4	9.6	9.0	7.0	5.9	92.9	2
CGB/IS	21.5	9.6	12.6	11.3	4.3	8.6	6.4	4.5	78.8	4
SSD/IS	16.9	10.5	14.0	12.0	4.3	8.6	6.2	4.5	77.0	5
SSD/IS/IA	16.2	10.5	12.6	11.7	4.1	8.6	5.6	4.4	73.7	6

TABLE 27. RELIABILITY (RATING = 23%)

	Starter*			Drive**			Total			
C&A/Starter Configuration	Component Failure Rate Per 1000 Starts	Relative Rating	Relative Weight, 20%	Component Failure Rate Per 1000 hr	Relative Rating	Relative Weight, 80%	Total Weight	Relative Weight	Total Rating	Ranking
TS/ATS	6.877	0.906	0.181	0.249	0.92	0.736	0.917	0.917	21.0	3
TS/IS	6.217	1.000	0.200	0.230	1.00	0.800	1.000	1.000	23.0	1
CGB/ATS	6.877	0.906	0.181	0.249	0.92	0.736	0.917	0.917	21.0	4
CGB/IS	6.217	1.000	0.200	0.249	0.92	0.736	0.936	0.936	21.5	2
SSD/IS	6.217	1.000	0.200	0.344	0.67	0.536	0.736	0.736	16.9	5
SSD/IS/IA	6.217	1.000	0.200	0.344	0.63	0.505	0.705	0.705	16.2	6

*Data Relate to One Installation (Two Engines)
**Data Relate to One Engine

(2) Vulnerability

The vulnerability of the total system was determined by an assessment of both the starter system and C&A drive system components. Since the protection afforded by the airframe is not known, the total volume of the airframe-mounted starter system components was used as the vulnerability criterion. The starter vulnerability was given a 25% weighting. The exposed vulnerable area of the engine-mounted C&A drive components was assessed and used as the vulnerability criterion. The C&A drive vulnerability was given a 75% weighting. The vulnerability assessments are summarized in Table 28. All of the candidate systems that use the impingement starter were penalized due to the requirement for a larger APU to provide the increased bleed air as compared to the ATS. The TS/ATS and the CGB/ATS received the highest ratings because of fuel pump shielding (in the bottom view) afforded by the ATS.

(3) Development Risk

The C&A drive system/starter configurations for all systems were essentially state of the art and were judged to have a similar development risk. Shaft seals will be required for the 65,000-rpm alternator and were considered an added development risk for those C&A drive configurations using this component, and the systems were penalized accordingly.

(4) Cost

The cost of the systems was based on estimated production pricing. The development cost of the components was not reflected in the cost evaluation, with the assumption that all systems would require the same relative amount of development effort.

Therefore, the cost of all six candidate systems was based on the number of components in the particular C&A system and was estimated to be essentially the same.

(5) Weight and Volume

The weight and volume of the total C&A drive/starter system were assessed on the basis of preliminary component designs. System weight was 50% of the total rating, and volume was 50% of the total rating. The weight and volume assessments are shown in Table 29. The tabulation shows that the TS/ATS system has the lowest overall weight and volume. The CGB/ATS system was second, and the rating reflected a slight volume penalty due to the larger gearbox. All systems using the impingement starter reflected a high volume due to the larger APU requirement and were downrated accordingly.

TABLE 28. VULNERABILITY (RATING = 19%)

C&A/Starter Configuration	Start Volume*			Exposed Area**			Total Weight	Relative Weight	Rating	Ranking
	Start System Volume, $in.^3$	Relative Rating	Relative Weight, 25%	C&A Exposed Area, $in.^2$	Relative Rating	Relative Weight, 75%				
TS/ATS	1020	1.00	0.250	41.06	1.00	0.750	1.000	1.000	19.0	1
TS/IS	3210	0.31	0.078	61.40	0.67	0.504	0.582	0.582	11.1	3
CGB/ATS	1020	1.00	0.250	50.34	0.82	0.613	0.863	0.863	16.4	2
CGB/IS	3210	0.31	0.078	71.83	0.57	0.428	0.506	0.506	9.6	6
SSD/IS	3210	0.31	0.078	64.71	0.63	0.476	0.554	0.554	10.5	4
SSD/IS/IA	3210	0.31	0.078	64.71	0.63	0.476	0.554	0.554	10.5	5

*Data Relate to One Installation (Two Engines)
**Data Relate to One Engine

TABLE 29. WEIGHT AND VOLUME (RATING = 10%)

	Weight*			Volume*						
C&A/Starter Configuration	Weight, lb	Relative Rating	Relative Weight, 50%	Volume, $in.^3$	Relative Rating	Relative Weight, 50%	Total Weight	Relative Weight	Rating	Ranking
TS/ATS	96.33	1.0	0.50	1212	1.00	0.50	1.00	1.0	10.0	1
TS/IS	178.99	0.54	0.27	3392	0.36	0.18	0.45	0.45	4.5	3
CGB/ATS	103.76	0.93	0.47	1239	0.98	0.49	0.96	0.96	9.6	2
CGB/IS	185.94	0.52	0.26	3439	0.35	0.17	0.43	0.43	4.3	4
SSD/IS	187.45	0.51	0.26	3484	0.34	0.17	0.43	0.43	4.3	5
SSD/IS/IA	189.81	0.50	0.25	3501	0.33	0.16	0.41	0.41	4.1	6

*Data Relate to One Installation (Two Engines)

(6) Performance

The performance of the systems was evaluated for both starting and normal engine operation. The overall starter performance was evaluated in terms of the required APU fuel flow and given a 10% weighting in the evaluation. The C&A drive performance was weighted at 90% and evaluated on the basis of drive efficiency. The gearbox efficiency was set at 98% and then adjusted lower, based on the total number of floating splines, gear engagements, and bearings for each C&A/Drive candidate. A tabulation of the performance ratings is shown in Table 30. The performance of all six candidates was rated essentially equal, primarily due to the common use of a mechanical accessory drive.

(7) Maintainability

The systems were evaluated on a comparative basis for maintainability. The results of the evaluation are summarized in Table 31 and show the CGB/ATS system as the easiest to maintain. Comments on the six systems that guided this rating are summarized below:

1. TS/ATS

   - Alignment of bevel gearing is critical.
   - Oil pump is externally accessible without gearbox disassembly.
   - Maintainability is considered a function of gearbox components: bearings, gears, and shafts (total of eleven).
   - The ATS is an additional gearbox-mounted component.

2. TS/IS

   - Alignment of bevel gearing is critical.
   - Removal of fuel pump/alternator assembly is required prior to oil pump removal.
   - Maintainability is considered a function of gearbox components: bearings, gears, and shafts (total of eight).
   - Maintenance or detailed inspection of engine starter cannot be performed without separation of engine case.

TABLE 30. PERFORMANCE (RATING = 9%)

C&A/Starter Configuration	APU Fuel Flow			Drive Efficiency			Total Weight	Relative Weight	Rating	Ranking
	Fuel Flow, lb/hr	Relative Rating	Relative Weight, 10%	Drive Efficiency, %	Relative Rating	Relative Weight, 90%				
TS/ATS	35.0	1.00	0.10	97.4	0.992	0.893	0.992	0.998	9.0	1
TS/IS	57.5	0.61	0.06	98.0	1.000	0.900	0.960	0.967	8.7	3
CGB/ATS	35.0	1.00	0.10	97.4	0.992	0.893	0.993	1.000	9.0	2
CGB/IS	57.5	0.61	0.06	97.4	0.992	0.893	0.953	0.960	8.6	4
SSD/IS	57.5	0.61	0.06	97.0	0.990	0.891	0.951	0.958	8.6	5
SSD/IS/IA	57.5	0.61	0.06	96.8	0.987	0.888	0.948	0.955	8.6	6

TABLE 31. MAINTAINABILITY (RATING = 7%)

	Relative Rating	Rating	Ranking
TS/ATS	0.97	6.8	2
TS/IS	0.89	6.2	4
CGB/ATS	1.00	7.0	1
CGB/IS	0.91	6.4	3
SSD/IS	0.89	6.2	5
SSD/IS/IA	0.80	5.6	6

3. CGB/ATS

- Gear alignment is less complex than TS arrangement.
- Accessibility to oil pump requires partial disassembly of gearbox.
- Maintainability is considered a function of gearbox components: bearings, gears, shafts (total of eight).
- The ATS is an additional gearbox-mounted component.

4. CGB/IS

- Gear alignment is less critical than TS arrangement.
- Accessibility to oil pump requires partial disassembly of gearbox.
- Maintainability is considered a function of gearbox components: bearings, gears, shafts (total of eight).
- Maintenance or detailed inspection of engine starter cannot be performed without separation of engine case.

5. SSD/IS

- Alignment of components is complex since all components are coaxial and, therefore, dependent.
- Accessibility to oil pump requires removal of fuel pump/alternator assembly.
- Maintainability is considered a function of gearbox components: bearings, gears, shafts (total of eight).

- Maintenance or detailed inspection of engine starter cannot be performed without separation of engine case.

- All gears lie in same plane, making access and disassembly less complex.

6. SSD/IS/IA

- Alignment of components is complex since all components are coaxial and, therefore, dependent.

- Accessibility to oil pump requires removal of fuel pump assembly.

- Accessibility to alternator requires complete disassembly of gearbox.

- Maintainability is considered a function of gearbox components: bearings, gears, shafts (total of eight).

- Maintenance or detailed inspection of engine starter cannot be performed without separation of engine case.

- All gears lie in same plane, making access and disassembly less complex.

(8) Installation Flexibility

Installation flexibility of the candidates was evaluated, including both the system volume and installation complexity. The volume was weighted at 50%, and the installation complexity was weighted the remaining 50%. The results are summarized in Table 32. The systems with the impingement starter were downrated due to the larger APU requirement. Installation complexity for all candidates was considered equal except that candidates with the ATS were slightly penalized because of this added component.

TABLE 32. INSTALLATION FLEXIBILITY (RATING = 6%)

	Relative Rating	Rating	Ranking
TS/ATS	1.00	6.0	1
TS/IS	0.76	4.6	3
CGB/ATS	0.98	5.9	2
CGB/IS	0.75	4.5	4
SSD/IS	0.75	4.5	5
SSD/IS/IA	0.74	4.4	6

D. REAR-DRIVE STUDY CONCLUSIONS

The analysis of the starter/C&A system for an engine with a rear-drive power turbine and no PTO requirement has shown that the C&A drive is best accomplished mechanically, using a tower shaft or cluster gearbox configuration. The gearbox provides a 65,000-rpm fuel pump/alternator drive, a 15,000-rpm oil pump drive, and an interface for the starter.

The recommended starter is an air turbine unit. For those installations where an APU is not desirable, the air turbine starter can be replaced with an electrical or hydraulic unit.

The starter/C&A study for a rear-drive engine has confirmed the observations outlined below, which were originally made as a result of the front-drive study.

- Elimination of the PTO greatly simplifies the engine gearbox design.
- C&A packaging is improved with a rear-drive engine.

Several basic advantages for a rear-drive power turbine engine configuration as compared to a front drive were also identified.

- Bearing DN and rotating seal velocities are lower, which should contribute to a more reliable engine.
- The gas generator turbine disk will be lighter due to a smaller bore.
- The power turbine critical speed will be higher and will allow the use of a single-stage power turbine.
- The hub/tip ratio of the gas generator compressor will be improved.
- Maintainability of the power turbine module is improved.

Based on these observations, it was recommended that further detailed design of a starter/C&A drive system be based on a rear-drive engine configuration.

A rear-drive engine configuration may not fit all installations, but it does represent an optimized powerplant package. For those airframe installations requiring a front-drive engine, it was not anticipated that the basic starter/C&A system components would change. The primary impact on the starter/C&A drive system would be nonoptimum packaging. The controls and accessory packaging for a front-drive power turbine engine configuration with an electric starter pad is shown in Figure 49.

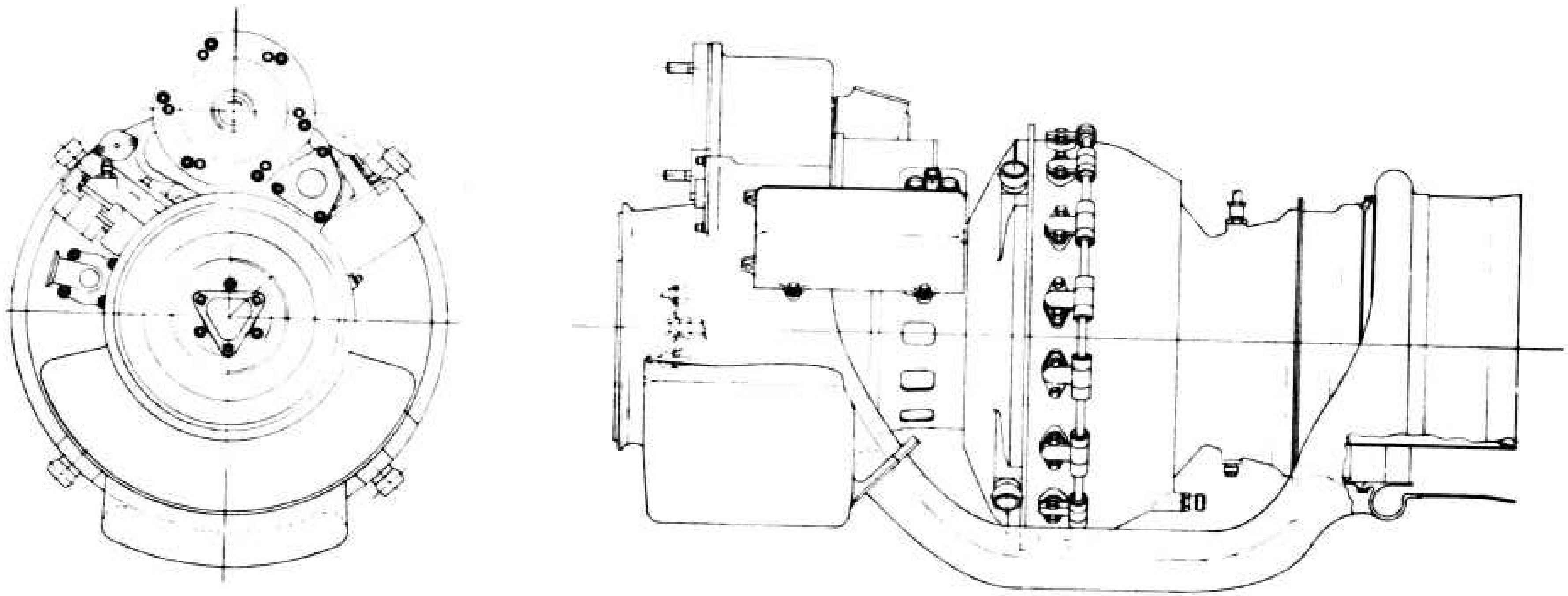


Figure 49. Front Drive Engine With Pad for Electrical Starter

SECTION VI

SELECTION OF SYSTEM FOR DETAILED DESIGN

A. CANDIDATE SYSTEMS

The conceptual design studies for a rear-drive power turbine described in Section V concluded that the C&A drive is best accomplished by a tower shaft gearbox or cluster gearbox arrangement. Both configurations use a gearbox-mounted air turbine starter. A description of the TS/ATS and CGB/ATS systems is given in the following paragraphs.

1. Tower Shaft Drive/Air Turbine Starter

Figure 50 details the TS/ATS configuration and shows the locations of the control and accessory components and the critical control system sensor interfaces. The tower shaft is coupled to the gas generator rotor through a 0.462:1 ratio spiral bevel gear set and is driven at 30,000 rpm. The alternator is incorporated on the tower shaft assembly. The alternator is a cartridge-type unit that runs in an oil-mist environment in a double-piloted cylindrical cavity above the engine centerline.

At the upper end of the tower shaft is a combination spiral bevel and spur gear. The spiral bevel gear drives a centrifugal fuel pump at 65,000 rpm through a 2.167 gear ratio. The fuel pump drive shaft also contains the air-oil separator and is mounted in an oval-shaped cover, which pilots on the upper end of the alternator and bolts to the inlet case. The cover, which also protects the oil pump gear, is sealed by a rubber O-ring.

A spur gear drives the oil pump at 15,000 rpm through a 0.5:1 gear ratio. The oil pump is mounted parallel to the alternator in another cylindrical cavity with cored passages for oil pump supply, scavenge, and discharge. The oil pump configuration is a high-displacement vane pump using a through-vane concept and includes a tandem scavenge-pump element.

The air turbine starter drives the engine through a quill shaft, supported radially and retained axially to facilitate starter installation. The quill shaft is retained by a cone-shaped cover when the starter is removed. An overrunning clutch within the starter disengages the starter turbine after the engine becomes self-sustained. The starter will be required to provide 5.2 ft-lb output at 16,000-rpm engine rotor speed, where ignition will occur and will operate up to 43,000-rpm engine self-sustaining speed.

The centrifugal fuel pump and inlet guide vane (IGV) actuator are integrated with the fluid controller located on top of the engine above the IGV unison ring. The electronic computer is located adjacent to the fluid controller. A finned heat exchanger section protrudes through the inlet duct to provide a heat transfer path to the inlet air to cool the electronic components.

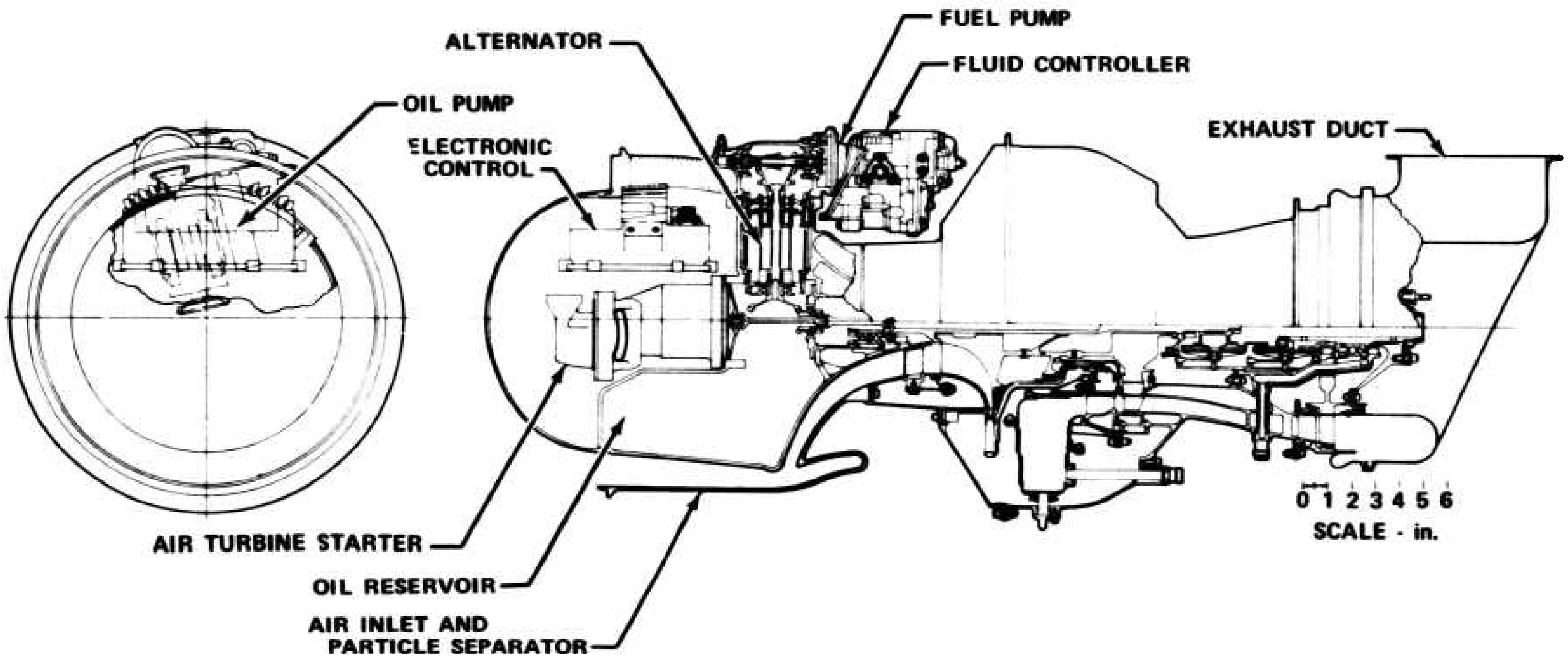


Figure 50. Tower Shaft/Air Turbine Starter Candidate

The torque sensor is a shaft twist-type indicator, designed into the power turbine shaft by means of three gear-toothed wheels adjacent to each other. One gear is attached to the output shaft and the other two are on the end of a torque tube that is concentric and splined to the output shaft. The three-gear system is a modification of the conventional phase displacement torque measuring system, where the third (position) gear compensates for shaft misalignment not accounted for by the torque and reference gears.

The power turbine overspeed sensor is a ring of cantilevers, splined to the forward end of the power turbine shaft. Centrifugal deflection of the cantilevers is used to generate a pneumatic overspeed pressure signal to the fluid controller.

The optical pyrometer is incorporated on top of the engine case with the aperture assembly sighting down and tangential to the gas generator blades. Installation in the engine requires a straight aperture assembly, with the detector assembly located within the electronics package.

The 65,000-rpm fuel pump (Figure 51) is contained within the hydromechanical fluid controller and results in the packaging arrangement shown in Figure 52. Weights and volumes of the alternator, pump, and fluid controller are listed in Table 33. The ATS and electronics package, common to both the TS/ATS and CGB/ATS systems, is also included in the table.

2. Cluster Gearbox/Air Turbine Starter

Figure 53 details the CGB/ATS configuration and shows the location of the control and accessory components. The control system sensor interfaces are the same as in the TS/ATS design. The CGB/ATS features a primary shaft driven by the gas generator rotor at 20,600 rpm through a 0.317:1 spur gear ratio. An internal gear is used to obtain a large speed reduction while keeping the two shaft centerlines as close together as possible.

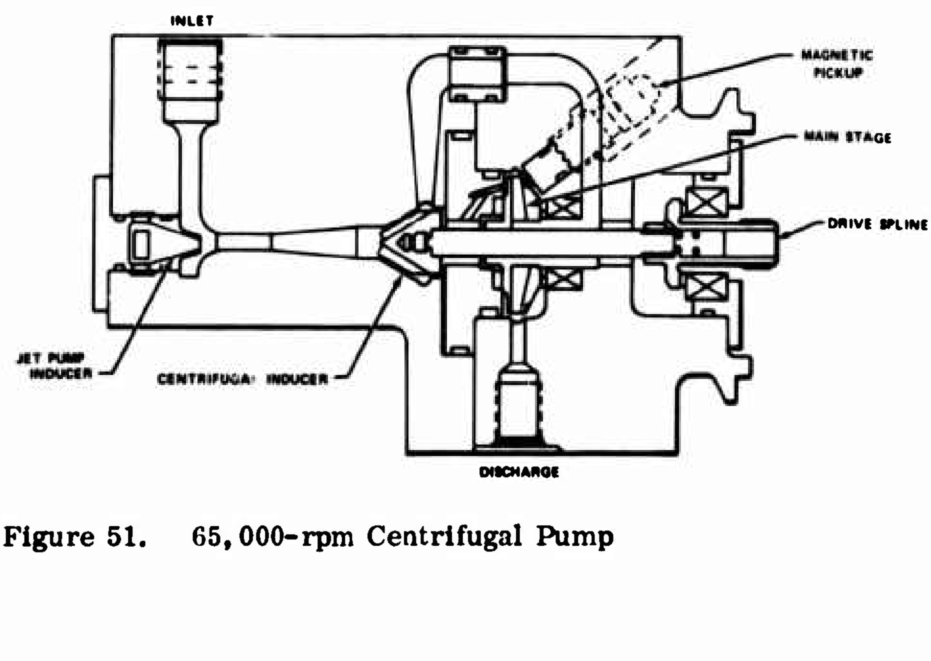


Figure 51. 65,000-rpm Centrifugal Pump

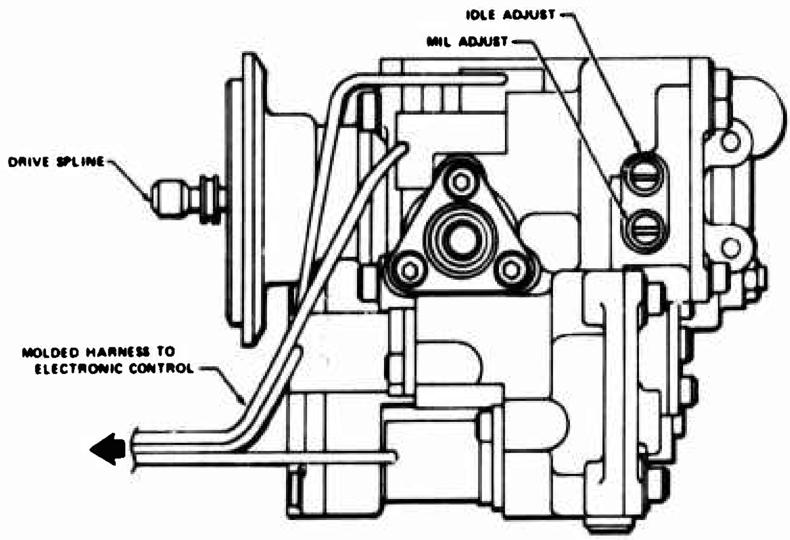


Figure 52. Tower Shaft Drive Fluid Controller and Fuel Pump

TABLE 33. COMPONENT WEIGHTS AND VOLUMES TS/ATS SYSTEM

Component	Weight, lb	Volume, $in.^3$
Alternator	2.6	9.9
Fuel Pump	1.8	16
Fluid Controller	7.4	74
Electronic Computer	3.8	76
Air Turbine Starter	9	140

The air turbine starter, common to both candidate configurations, meshes with a spur gear on the forward end of the primary shaft. An overrunning clutch within the starter disengages the starter turbine after the engine becomes self-sustaining.

The oil pump gear, located in the middle of the primary shaft, drives the oil pump at 15,000 rpm through a 1.728:1 gear ratio. The oil pump, also common to the TS/ATS design, is mounted in a cylindrical cavity in the inlet case and is connected to cored passages for oil pump supply, scavenge, and discharge.

The gear driving the oil pump is also used as an idler gear to drive the fuel pump drive shaft at 65,000 rpm through a 4.34:1 gear ratio. An air-oil separator is located on the fuel pump drive shaft.

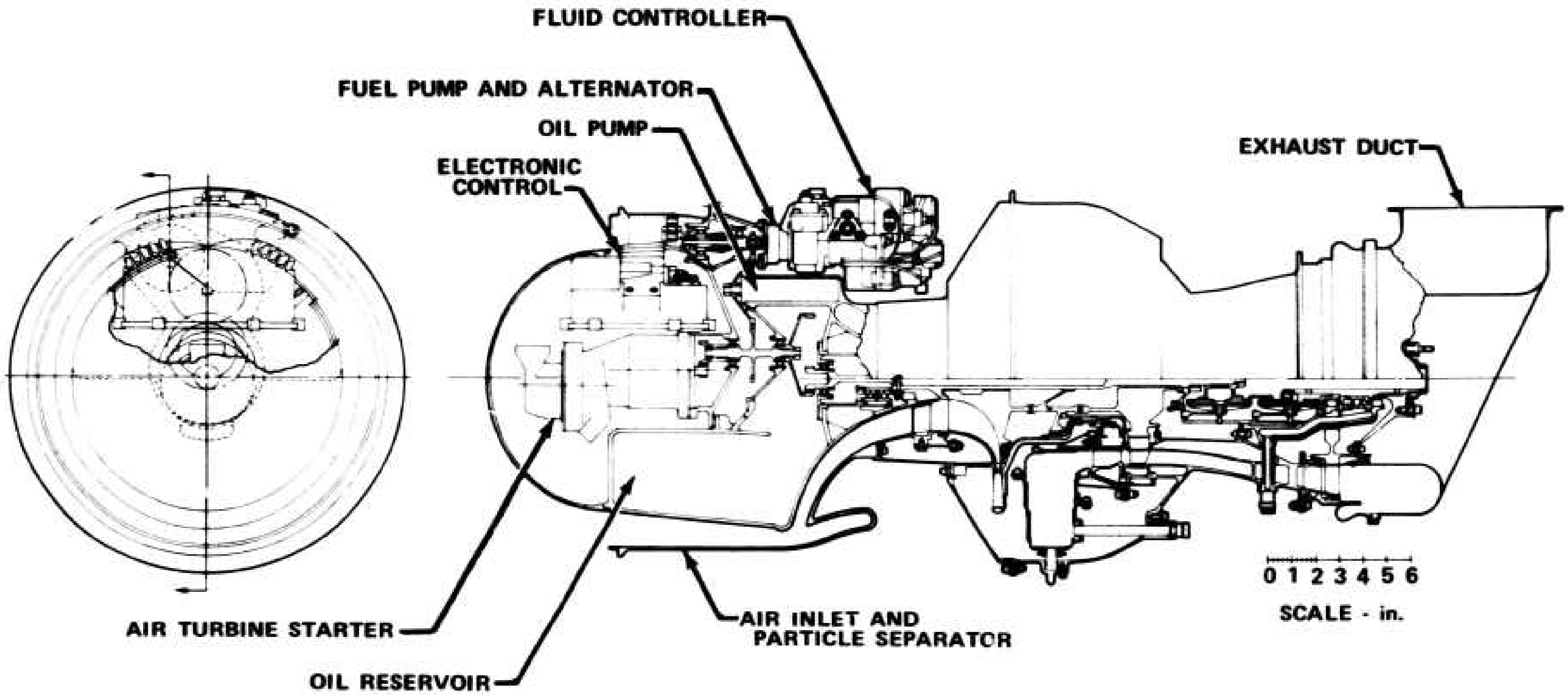


Figure 53. Cluster Gearbox/Air Turbine Starter Candidate

The alternator in the CGB system is incorporated into the fuel pump package as shown in Figure 54 and will run dry at a maximum speed of 65,000 rpm. The hydromechanical fuel-metering package and the electronic computer are mounted the same as in the tower shaft configuration.

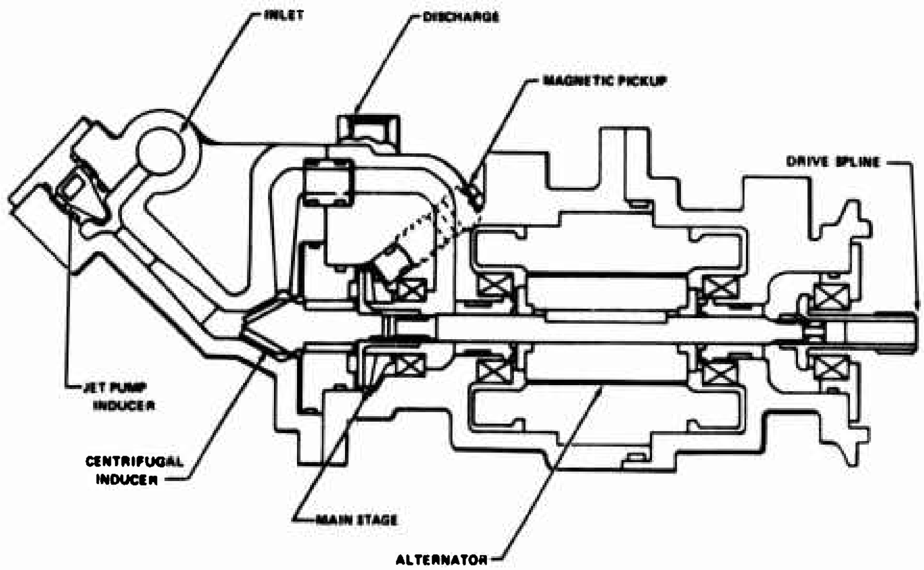


Figure 54. 65,000-rpm Centrifugal Pump/ Alternator

The fluid controller size and weight are detailed in Table 34, and the packaging arrangement is shown in Figure 55.

B. EVALUATION, RATING, AND SELECTION OF ONE SYSTEM

1. Summary

The tower shaft/air turbine starter (TS/ATS) control and accessory drive system was recommended for the detailed design phase, based on a detailed comparison with the cluster gearbox/air turbine starter (CGB/ATS) system. Design layouts of the two candidate systems were completed and evaluated using the weighted selection criteria previously established for the program. A summary of the evaluation is included in Table 35.

The two candidate systems used a common gas generator and power turbine. The common controls and accessories components included the starter, electronic computer, sensors, ignition system, compressor bleed valve, fuel pump, IGV actuator, and hydromechanical metering system. Components that were not common included the control and accessory drive (gearbox), starter drive, alternator, and hydromechanical system packaging. These analyses concentrated on evaluating the differences in the basic components, and the differences brought about by the variations in the overall component arrangement.

TABLE 34. COMPONENT WEIGHTS AND VOLUMES CGB/ATS SYSTEM

Component	Weight, lb	Volume, $in.^3$
Fuel Pump/Alternator	4.8	40
Fluid Controller	10.8	94

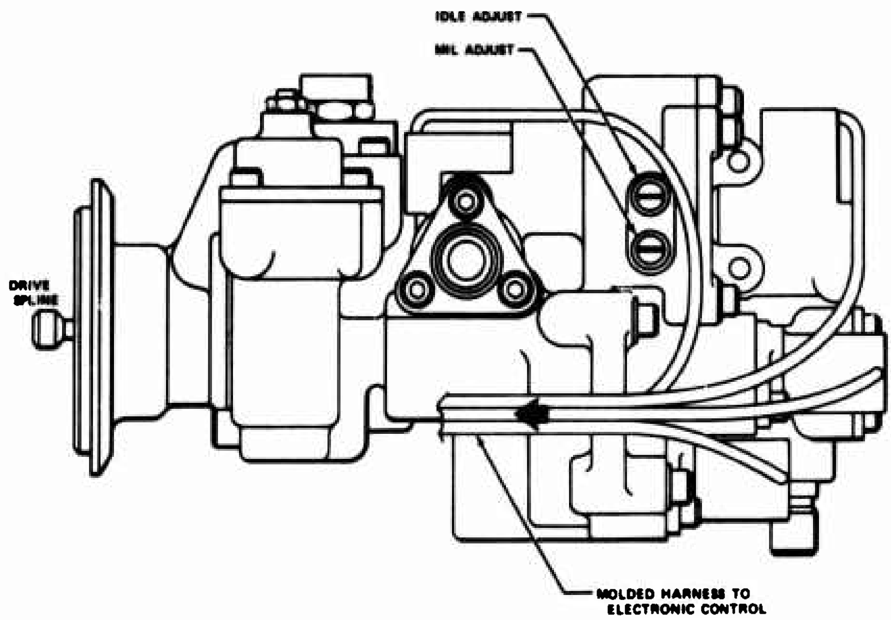


Figure 55. Cluster Gearbox Drive Fluid Controller and Fuel Pump/ Alternator

TABLE 35. OVERALL SYSTEM EVALUATION

Criteria	Criteria Weight, %	Systems Relative Rating TS/ATS	Systems Relative Rating CGB/ATS
Reliability	23	23	20.8
Vulnerability	19	19	14.8
Development Risk	14	14	13.5
Cost	12	12	12
Weight and Volume	10	10	7.9
Performance	9	9	7.9
Maintainability	7	7	6.5
Installation Flexibility	6	6	5.7
	100	100	89.1

The two systems were closely rated in all areas except vulnerability, where the TS/ATS system showed a distinct advantage. This was primarily due to the integration of the alternator with the tower shaft drive and the compactness of the tower shaft gearbox as compared to the cluster gearbox.

The TS/ATS system showed advantages in all other categories except cost, where the two systems were rated equally, with the resultant overall rating of 100 for the TS/ATS and 89.1 for the CGB/ATS system. These analyses reflected a clear choice for the TS/ATS system.

2. Analyses

The completed preliminary design layouts were reviewed and graded using the selection criteria described herein. The results of these analyses are summarized in the following paragraphs.

a. Reliability

The component failure rate estimates reflected inherent design features affecting reliability. These included susceptibility to dry starts, shock loading, potential of misalignment, and the design operating levels relative to the state of the art. The approach used "base" failure rates, i.e., similar component failure history from other P&WA turboshaft engines. Each component base rate was adjusted for differences in design and duty requirement, e.g., bearing DN's, environmental conditions, startup frequency, etc. The adjusted rates were then compiled to determine an overall failure rate for each scheme. The analyses showed a preference for the TS/ATS system as compared to the CGB/ATS system.

The CGB/ATS failure rate was greater due primarily to the larger number of bearings and seals required for the alternator. A point in favor of the TS/ATS scheme was that the alternator did not require additional bearings (eliminated one bushing), did not require shaft seals (eliminated two seals), and eliminated one spline coupling. Alternator bearing failures are among the more dominant alternator failure modes in field engines. Operating the alternator in the gearbox environment as compared to an ambient environment was not seen as a penalty to the TS/ATS system.

The relative reliability evaluation of the two candidate systems is summarized in Table 36.

Based on a total weight of 23 for reliability, a rating of 23 was given to the TS/ATS and a rating of 20.8 was given to the CGB/ATS system.

b. Vulnerability

The vulnerable areas were tabulated for both the start and in-flight modes. The start mode considered the engine-mounted starter and fuel system components, while the in-flight mode considered only the fuel system components. The results of the vulnerability analyses are shown in Table 37.

The TS/ATS had a vulnerability advantage due to the greater degree of protection provided for the alternator and the compactness of the tower shaft gearbox. Both schemes had good frontal protection. The greater tooth contact area of the spiral bevel gears enhanced survivability in the event of tooth damage to the TS/ATS system.

Vulnerable area estimates were made for four views of each scheme (front, bottom, left, and right). The vulnerable area was taken as the product of a component's projected area and the probability of a kill (given a hit). Estimates of kill probability were governed by:

1. Degree of protection afforded by neighboring components
2. Degree of protection provided by the engine structure
3. Survivability of the damaged component as a function of hit location, size, and velocity of missile.

Components analyzed included the alternator gears (web and teeth), bearings (races and rolling elements), shafts, and splines. The oil pump was included in the final numbers; however, it is felt that a direct hit would not cause an immediate kill.

The projectile used was a 30-caliber (0.303 in. diameter) bullet impacting with a 2500-ft/sec strike velocity at 0-deg obliquity.

TABLE 36. RELIABILITY SUMMARY

λ_b = Basic Failure Rate/10^6 Flight Hours
Shock = Shock Loading Factor
Dry Start = Dry Start Factor
λ_t = Total Failure Rate/10^6 Flight Hours

1. Tower Shaft/Air Turbine Starter

Component	λ_b	Shock	Dry Start	λ_t	Remarks
Gears					
Spur	1	2	1	2	1 spur/2 gear meshes
Spiral Bevel	4	1.5	1	6	2 spiral bevel meshes
Splines	4	1.5	1	6	4 spline couplings
Shafts	6	2	1	12	1 at 30,000 rpm 1 at 15,000 rpm 4 at 65,000 rpm
Bearings					
Rolling Element	5	1	2.5	12.5	3 at 65,000 rpm 2 at 30,000 rpm
Bushings	5	1	1	5	3 at 15,000 rpm 2 at 65,000 rpm (DN's within state of the art)

TABLE 36. RELIABILITY SUMMARY (CONTINUED)

1. Tower Shaft/Air Turbine Starter

Component	λb	Shock	Dry Start	λT	Remarks
Seals	5	1	5	25	3 at 65,000 rpm (oil) 2 at 65,000 rpm (fuel)
Oil Pump Element	1	1	1	1	Dual vane element (3 bearing supports)
Alternator	20	1	1	20	30,000 rpm (oil mist environment)
		Total		89.5	

2. Cluster Gearbox/Air Turbine Starter

Component	λb	Shock	Dry Start	λT	Remarks
Gears					
Spur	4	2	1	8	4 spur/gear meshes 3 spline couplings
Splines	3	1.5	1	4.5	
Shafts	6	2	1	12	4 at 65,000 rpm 1 at 20,300 rpm 1 at 15,000 rpm
Bearings					
Rolling Element	5	1	2.5	12.5	2 at 20,300 rpm 3 at 65,000 rpm
Bushings	6	1	1	6	3 at 15,000 rpm 3 at 65,000 rpm
Seals	7	1	5	35	1 at 41,000 rpm 2 at 65,000 rpm (oil) 4 at 65,000 rpm (fuel)
Oil Pump Element	1	1	1	1	Dual vane element (3 bearing supports)
Alternator	20	1	1	20	65,000 rpm (ambient environment)
		Total		99.0	

TABLE 37. VULNERABILITY ANALYSES

	Vulnerable Area, in.^{2}		Relative Rating		Absolute Rating		Total Absolute Rating
	Start	Inflight	Start	Inflight	Start 25%	Inflight 75%	
TS/ATS	20.2	18.7	1	1	0.25	0.75	1.0
CGB/ATS	25.2	24.2	0.8	0.77	0.20	0.58	0.78

Based on a total value of 19 points for vulnerability, the TS/ATS system was rated at 19, and the CGB/ATS was rated at 14.8.

c. Development Risk

The development risk of the two candidate systems was rated essentially the same, with a small advantage evident with the TS/ATS system. The gear trains and the alternator configurations were primarily considered in this evaluation, since the other C/A components were common.

The gearbox bearing DN's and gear pitch-line velocities were evaluated for both schemes and are tabulated in Figure 56 for the TS/ATS and in Figure 57 for the CGB/ATS. These levels of DN and pitch-line velocity were within the state of the art and were not considered as high-risk items. The development risk of using spiral bevel gears (TS/ATS) as compared to spur gears (CGB/ATS) was considered to be essentially the same.

The development risk of the overall CGB/ATS drive system was considered to be slightly higher due to the additional bushing and dynamic seals required in the alternator drive.

The alternator configurations for each system are summarized in Table 38.

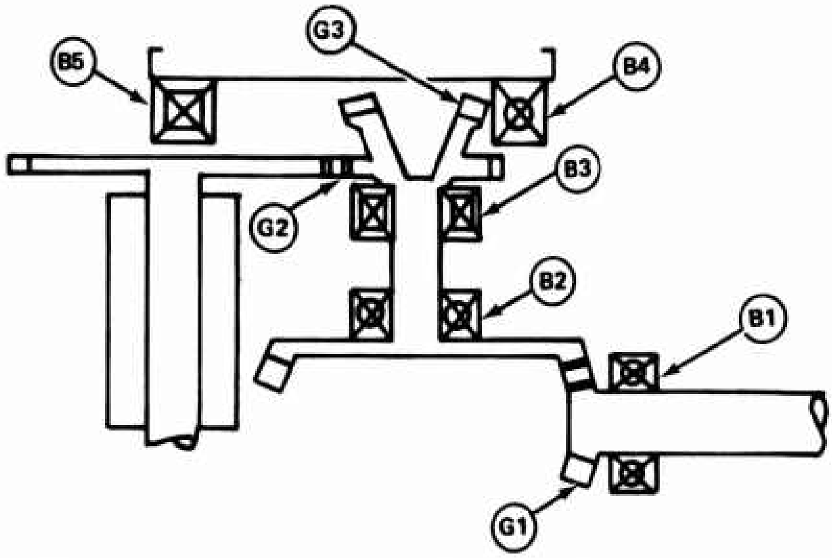
The development risk of the alternator electrical components was rated the same, based on no large operating temperature differential. The penalty for developing the additional bearings, seals, and splines for the CGB/ATS alternator is reflected in the gearbox development risk rating.

The development risk ratings of the drive system and components are summarized in Table 39.

Based on a total value of 14 points for development risk, a rating of 14 was given to the TS/ATS and a rating of 13.5 was given to the CGB/ATS.

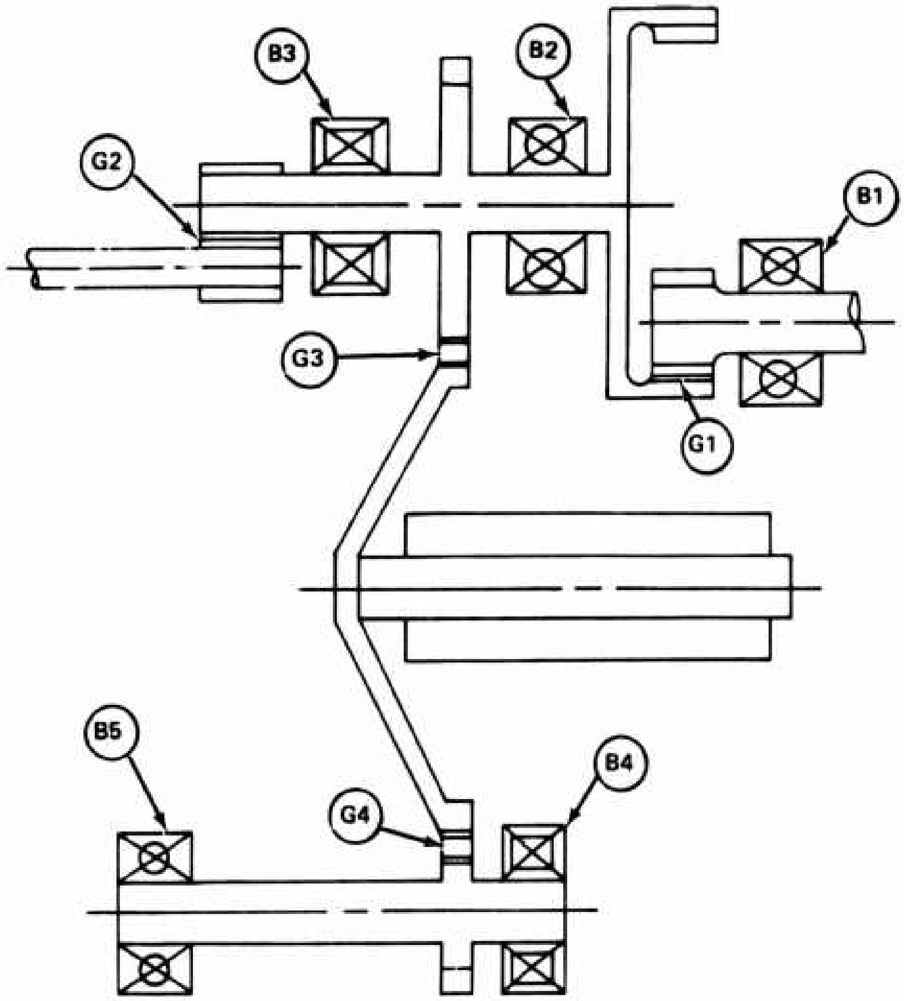
d. Cost

Estimated production pricing for the candidate C&A drive systems and the C&A components was determined. The gearbox component costs for both drive schemes were assessed to be so close as to be comparable. The estimated absolute cost of either drive system was $6,000. The control and accessory components for each drive system are compared in Table 40.



LOCATION	DIAMETER, mm	rpm	BEARING DN OR GEAR PITCH-LINE VELOCITY
B1	19	65,000	1.235×10^6
B2	12	30,000	0.36×10^6
B3	13	30,000	0.39×10^6
B4	12	65,000	0.78×10^6
B5	12	65,000	0.78×10^6
G1	25.4 55.1	65,000 30,000	17,017 ft/min
G2	36.2 73.3	30,000 15,000	11,250 ft/min
G3	42.7 19.7	30,000 65,000	13,191 ft/min

Figure 56. Tower Shaft Drive System Bearing DN's and Gear Pitch-Line Velocity



LOCATION	DIAMETER, mm	rpm	BEARING DN OR GEAR PITCH-LINE VELOCITY
B1	19	65,000	1.235×10^6
B2	16	20,287	0.324×10^6
B3	16	20,287	0.324×10^6
B4	10	65,000	0.650×10^6
B5	10	65,000	0.650×10^6
G1	26.2 83.8	65,000 20,287	17,527 ft/min
G2	26.2 13	20,287 40,973	5,470 ft/min
G3	61.2 82	20,287 15,137	12,800 ft/min
G4	82 19.1	15,137 65,000	12,800 ft/min

Figure 57. Cluster Gearbox Drive System Bearing DN and Gear Pitch-Line Velocity

TABLE 38. ALTERNATOR DESIGN REQUIREMENTS

	TS/ATS	CGB/ATS
Speed, rpm	30,000	65,000
Ambient Temperature, °C	121	80
Ambient Environment	Oil Mist	Atmospheric
Additional Drive Spline Required	-	1
Additional Seals Required	-	2
Additional Bushings Required	-	1

TABLE 39. DRIVE SYSTEM AND COMPONENT DEVELOPMENT RISK RATINGS

Component	Relative Weight, %	Relative Ratings TS/ATS	Relative Ratings CGB/ATS	Absolute Rating, % TS/ATS	Absolute Rating, % CGB/ATS
Oil Pump	20	1.0	1.0	20	20
Fuel Pump	20	1.0	1.0	20	20
Alternator (Electrical Components)	20	1.0	1.0	20	20
Gear Train	40	1.0	0.9	40	36
	100			100	96

TABLE 40. COMPARISON OF C&A DRIVE SYSTEM COMPONENTS

	TS/ATS	CGB/ATS
Hydromechanical Package/Pump/Alternator		$2,550
Hydromechanical Package/Pump	$2,280	
Alternator	225	
Electronic Computer and Sensors	3,400	3,400
Torque Sensor (Speed Pickups)	45	45
Power Turbine Overspeed Sensor	165	165
Compressor Bleed Valve	275	275
Total	$6,390	$6,435

A tabulation of the cost ratings is shown in Table 41. Based on a total weight of 12 points and no significant differences in the total cost, both systems were given a rating of 12.

TABLE 41. COST

Rating = 12		
C&A/Starter Configuration	Costs, $	Absolute Rating
TS/ATS	12,390	1.0
CGB/ATS	12,435	0.996

e. Weight and Volume

The weight and volume of the two candidate systems were assessed on the basis of preliminary engine system designs and the control and accessory system vendor analysis of the C&A components. Since the air turbine starter is common to both candidates, the tower shaft and cluster gearbox drives were evaluated on the basis of gears, gear shafts, bearings, and supporting members to determine the weight of each system. The volumes were determined directly from the preliminary design layouts.

The weight of the CGB drive is 3.6 lb. The weight of the TS drive is 3.8 lb. The number of bearings in each drive system is the same. The alternator/oil pump gear shaft and the cylindrical cavities for the oil pump and alternator account for a major portion of the TS drive system weight penalty.

The volumes of the CGB and TS gear trains were determined to be 110.5 and 86.6 $in.^3$, respectively. The major impact on the CGB volume is the axial oil pump cavity and length of the primary gear shaft, which displaces the air turbine starter axially forward 1 in.

The C&A components for each candidate system are common, with the exception of the alternator and hydromechanical package. The CGB arrangement, which incorporates the fuel pump, alternator, and fuel metering package into one unit, results in a combined package weight and volume of 15.6 lb and 134 $in.^3$, respectively. The tower shaft arrangement, which has a combined fuel metering package and fuel pump and a separate alternator, weighs 11.8 lb and has a total volume of 99.9 $in.^3$.

The combined weights and volumes for each candidate configuration are shown in Table 42.

TABLE 42. COMBINED WEIGHTS AND VOLUMES FOR CANDIDATE CONFIGURATIONS

	TS/ATS		CGB/ATS	
	Weight, lb	Volume, $in.^3$	Weight, lb	Volume, $in.^3$
C&A Drive	3.8	86.6	3.6	110.5
C&A Components	11.8	99.9	15.6	134.0
Total	15.6	186.5	19.2	244.5

The overall weight and volume rating of each candidate C&A drive/starter system is shown in Table 43. System weight was 50% of the total rating, and volume was 50% of the total rating. Based on a total rating of 10 points for weight and volume, the TS/ATS system was given a rating of 10 and the CGB/ATS system was given a rating of 7.9.

TABLE 43. WEIGHT AND VOLUME

C&A Starter Configuration	Weight (50%) Weight, lb	Relative Rating	Absolute Rating	Volume (50%) Volume, in^3	Relative Rating	Absolute Rating	Total Absolute Rating
TS/ATS	15.6	1.0	0.5	186.5	1.0	0.5	1.0
CGB/ATS	19.2	0.81	0.41	244.5	0.76	0.38	0.79

f. Performance

Each C&A drive/starter configuration was evaluated on the basis of engine inlet blockage due to the gearbox configuration and gearbox drive efficiency to determine the effect on system performance. Inlet blockage was 60% of the total rating, and drive efficiency was 40% of the total rating.

The tower shaft configuration was determined to have a 38-deg inlet blockage (Figure 58), while the cluster gearbox configuration has a 48-deg inlet blockage.

A method of rapidly assessing the sensitivity of a compressor to circumferential distortion has been developed by P&WA, East Hartford. This technique indicates that a 20% degradation in distortion tolerance of the engine would result with the CGB arrangement.

The C&A drive efficiency for each configuration was assessed to be equal because of the similarity in the number of bearings, shafts, and gears and the similarity in the shaft speeds. A tabulation of the performance ratings is shown in Table 44. Based on a total weight of 9.0 for performance, the TS/ATS was given a rating of 9.0 and the CGB/ATS was given a rating of 7.9.

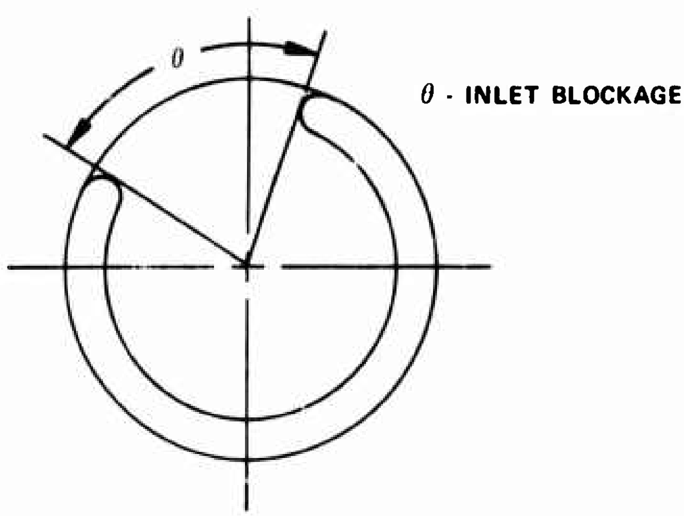


Figure 58. Tower Shaft Configuration With a 38-deg Inlet Blockage

TABLE 44. PERFORMANCE

Configuration	Drive Efficiency (40%) Relative Drive η	Relative Rating	Absolute Rating	Inlet Blockage (60%) Inlet Blockage, deg	Relative Rating	Absolute Rating	Total Absolute Rating
TS/ATS	1	1	0.4	38	1.0	0.6	1.0
CGB/ATS	1	1	0.4	48	0.79	0.48	0.88

g. Maintainability

The maintainability of the two candidate systems was evaluated, with the result that a slight preference for the TS/ATS system was seen as compared to the CGB/ATS system. The control and accessory component accessibility and gearbox complexity were also considered. This preference was primarily due to the better accessibility of the oil pump and alternator for the TS/ATS system. The maintainability evaluation is summarized in Table 45.

Based on a total weight of 7 for maintainability, the TS/ATS was given a rating of 7 and the CGB/ATS was given a rating of 6.5.

h. Installation Flexibility

The installation flexibility of the two candidate systems was evaluated considering the applicability of the configurations for alternate engine installations. The engine/airframe interfaces and basic component accessibility were also considered.

The engine/airframe interfaces for the starter, fuel supply, and electrical connections are the same for both systems. Accessibility to the C&A components is good in both configurations, provided that there is free access to the top of the engine. The major C&A components (starter, electronic computer, hydromechanical metering system, and fuel pump) are similarly located for both systems. The oil pump is considered to be less accessible in the CGB/ATS system and would require partial gearbox assembly. The evaluation is summarized in Table 46.

Based on a total weight of 6.0 for installation flexibility, a rating of 6.0 was given to the TS/ATS and a rating of 5.7 was given to the CGB/ATS.

TABLE 45. MAINTAINABILITY EVALUATION COMMENTS

	TS/ATS	CGB/ATS	Weight	Relative Rating TS/ATS	Relative Rating CGB/ATS	Absolute Rating TS/ATS	Absolute Rating CGB/ATS
I. Component Accessibility							
(a) Starter	(a) Remove nose cone to expose starter.	(a) Same as TS/ATS	10	1	1	10	10
(b) Electronic Computer	(b) Pull nose cone to expose the control. There is limited access to the connections at the rear of the control; releasing and securing the connections could be a problem.	(b) Same as TS/ATS	15	1	1	15	15
(c) Fuel Pump-Hydromechanical Control	(c) Secured with "V" band clamp. No apparent problem.	(c) Same as TS/ATS	15	1	1	15	15
(d) Alternator	(d) Remove hydromechanical control and upper part of gearbox to expose alternator.	(d) The alternator is part of the hydromechanical control. Replacement of the alternator will require that the control unit be replaced. This will require a complete engine trim run.	10	1	0.6	10	6
(e) Oil Pump	(e) The oil pump and alternator must be withdrawn and installed together with the gears meshed.	(e) The electronic computer, gearbox front cover, and the starter must be removed to replace the oil pump.	10	1	0.7	10	7
II. Gearbox Accessibility							
(a) Number of Gearbox Parts							
Gears and Gearshafts Bearings	5 3 Balls, 2 Rollers	5 3 Balls, 2 Rollers	20	1	1	20	20
(b) Gearbox Complexity	Compatible with current gearbox assemblies. Alignment of the spiral bevel gears may be a problem.	Compatible with current gearbox assemblies. Somewhat more complex internally than the TS/ATS scheme.	20	1	1	20	20
			100			100	93

TABLE 46. ENGINE/AIRFRAME INTERFACE AND COMPONENT ACCESSIBILITY

	Weight	Relative Rating		Absolute Rating	
		TS/ATS	CGB/ATS	TS/ATS	CGB/ATS
Engine/Airframe Interface Accessibility	50	1	1	50	50
Component Accessibility	50	1	0.9	50	45
				100	95

SECTION VII

DETAILED DESIGN OF SELECTED SYSTEM

Approval of the recommended tower shaft/air turbine starter control and accessory drive system was received from the Army, and further optimization of the C&A systems was initiated. To decrease vulnerability of the electronic computer, the control was moved to the top of the engine, adjacent to the fluid controller. Air cooling of the electronics housing would be provided by compressor inlet airflow.

In an effort to reduce engine length, starter configuration studies were accomplished to guide the definition of the available starter envelope.

Control and accessory component preliminary requirement specifications and final envelope drawings were prepared and submitted to the component vendors. Control system sensor interfaces and component interconnects and interfaces were defined. Consideration was also given to environmental conditions, cooling, routing, and accessibility.

Meetings were held with control and accessory system vendors to review the component specification requirements and to discuss the potential interfaces.

Garrett/AiResearch was contacted to provide input for the starter specification. The need for engine-supplied lubrication of the starter bearings and gears was established due to the high operating speed of the overrunning clutch. Implementation of an emergency clutch disengagement feature was also discussed and included as part of the starter requirements.

The oil pump mounting configuration was reviewed with Sundstrand Corporation to confirm the approach of using a cartridge-type pump that would pilot into a cylindrical cavity within the accessory gearbox. For installation flexibility, the spur gear drive will be an engine-supplied part. It was determined that an O-ring seal should be provided between the pump inlet and discharge cored passages and that Sundstrand would provide a shear section or equivalent mechanical feature as part of the pump drive section. Design analysis determined acceptable piloted cavity clearances on the basis of cavity material thermal growth and required gear meshes.

The Chandler Evans Corporation began final design of the control and fuel system components applicable to the tower shaft drive engine configuration. The following paragraphs detail the final C&A system components and the study of an emergency lubrication system.

Control and accessory component layouts were coordinated, and the final layout drawings were received from the component vendors and incorporated in the engine layout drawing. The C&A system components are described in the following paragraphs.

1. Electronic Control and Sensors

a. Electronic Computer

The electronic fuel control technology selected for this study is being developed by Chandler Evans for the U. S. Army Air Mobility Research and Development Laboratory, Eustis Directorate. This selection was made because the control was originally selected for development, based on design trade-off studies that are consistent with the goals of the subject program. The control is being developed for small (2- to 5-lb/sec airflow), advanced turboshaft engines for which formal development would begin in 1975.

The electronic computer is a hard-wired, special-purpose, hybrid computer, which provides all of the computation, control, scheduling, and logic function requirements for controlling the engine. It incorporates a 256 eight-bit "read only" memory for start and acceleration scheduling and provides IGV closed-loop scheduling and compressor bleed valve control. Both are controlled as a function of corrected gas generator speed.

The computer module houses the radiation pyrometer silicon chip for the turbine blade temperature limiter and the variable capacitance compressor discharge pressure sensor. These sensors are located within the computer housing to take advantage of the cooled environment. Cooling is required to enhance the reliability of the electronics and to isolate them from the engine temperature environment. The electric power dissipated in the computer is only about 6w, so the conventional self-heating problems are obviated; therefore, air or fuel cooling can be used.

The computer controls the IGV actuator and the position of the main metering valve through stepper motors. IGV and metering valve positions are fed back to the computer through resolvers.

The control/accuracy and sensing requirements are listed in Tables 47 and 48. The fly-by-wire approach has been incorporated into the electronic control design. Electrical PLA and collective pitch input from the airframe have been provided. The main fuel shutoff valve is solenoid operated. Provision for a redundant shutoff solenoid, using airframe power, can be added for independent pilot control. External adjustments of maximum and idle gas generator turbine speed are located on the package exterior. The computer can also accept cockpit trim of power turbine set speed. The starter cut-out signal is also provided to operate the starter inlet air solenoid valve.

The electronic computer layout is shown in Figure 59. Heat dissipation is accomplished by means of the front mounting flange that channels internal heat into the engine inlet case. All the C&A components are designed to endure a 50 g vibration environment radially from the engine centerline. The electronics integrity is maintained by means of vertically mounting the printed circuit boards, vibration absorbing grommets at the board mounting points, and viscoelastic material sandwiched into the printed circuit board multilayer construction.

TABLE 47. SENSOR REQUIREMENTS

Sensor	Control	Engine Condition Monitoring
Gas Generator Rotor Speed	X	X
Power Turbine Rotor Speed	X	X
Output Shaft Torque	X	X
Gas Generator Blade Temperature	X	X
Compressor Inlet Temperature	X	X
Burner Pressure	X	X
Engine Oil Temperature		X
Engine Oil Pressure		X
Magnetic Particle Detector(s)		X
Oil Level		X
Fuel Filter Bypass		X
Oil Filter Bypass		X
Engine Start (No.) and Time Counted		X

TABLE 48. FUEL CONTROL REQUIREMENTS

Requirement		Accuracy
Semiautomatic Start Sequencing		Pilot Initiates Fuel Flow
W_f/P_b Start, Acceleration, and Deceleration		± 2 lb/hr, ± 4%, ± 6%
Proportional Gas Generator Governor		± 1%
Isochronous Power Turbine Governor		± 3% Transient, ± 0.25% Steady-State
Inlet Guide Vane Scheduling		± 1% Stroke
Compressor Bleed Scheduling		Two-Position
Torque:	Limiting	115 ± 5%
	Display	See AV-E-8593
	Loading Sharing	± 5%
Turbine Blade Temperature:	Limiting	± 16°C
	Display	± 5°C
Power Turbine Speed Trim		85 to 115%
Power Turbine - Backup Overspeed Limiter		50-ms Response

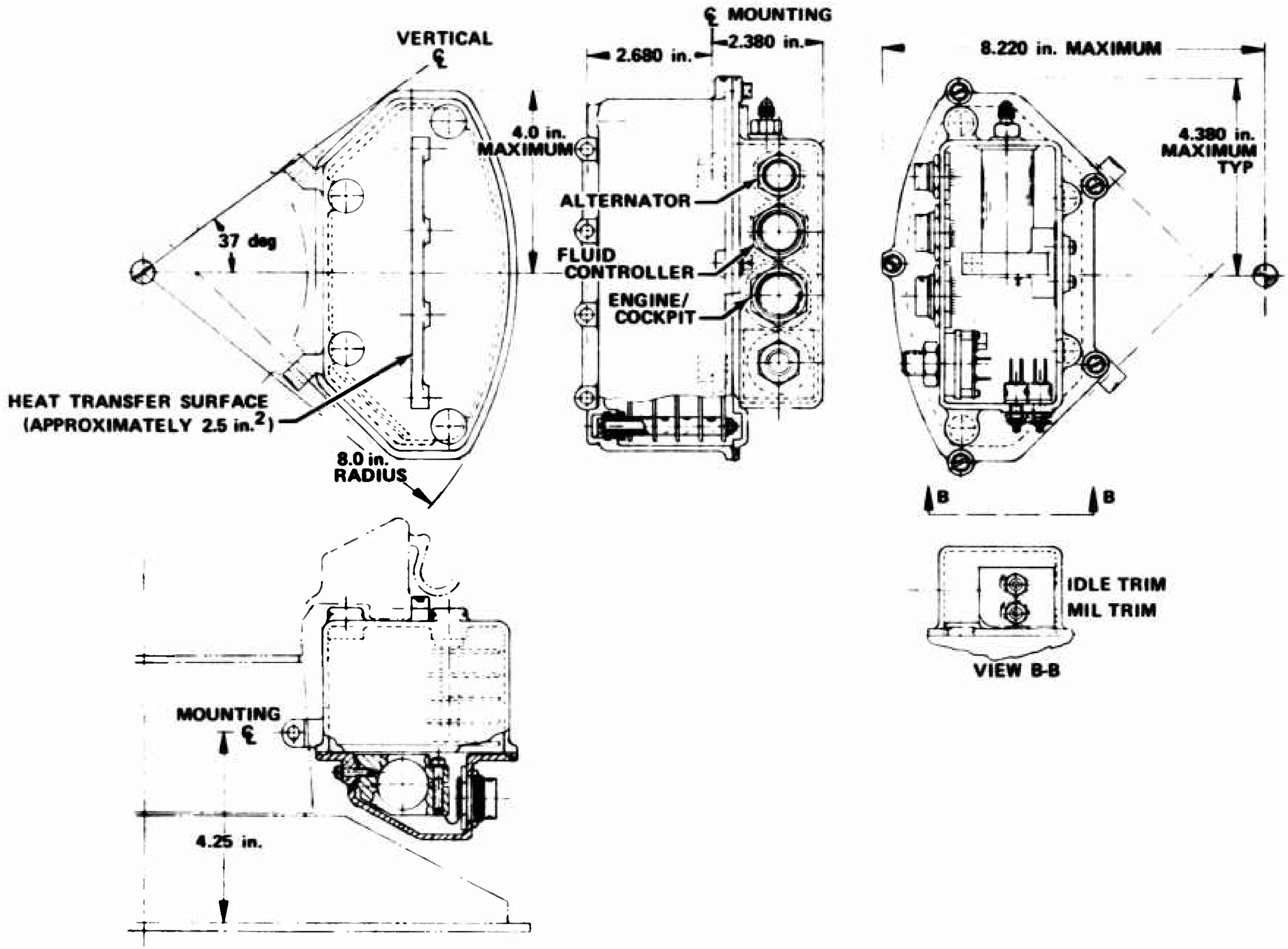


Figure 59. Engine Electronic Control

b. Torque Sensor

The torque sensor measures the twist of the power turbine shaft by means of three adjacent gear-toothed wheels, as indicated in Figure 60. One gear is attached to the output shaft, and the other two are attached to the end of a torque tube that is concentric and splined to the output shaft. The three-gear system is a modification of the conventional phase displacement torque system, in that the third (position) gear allows for compensation of shaft misalignment not accounted for by the torque and reference gears. Three magnetic pickups monitor the toothed wheels and provide an approximate sinusoidal signal to the electronic computer. When torque is applied to the shaft, the resultant twist causes circumferential displacement of the toothed wheels, which, in turn, causes a phase shift to occur between the signals generated by the magnetic pickups. This phase shift is demodulated and is presented as a measure of shaft torque. A thermistor is included with the pickups to enable the electronic unit to modify the phase shift signal to compensate for shaft temperature change. A more detailed study could indicate that the measurement of turbine blade temperature may be used to predict torque tube operating temperature, thus eliminating the requirement for a thermistor. The electronic module that transduces phase shift and modulates the output as a function of shaft temperature is located in the electronic control.

Installation of the torque sensor assembly is shown in Figure 61. Full-scale angular shaft twist has been calculated to be 5.6 deg and, coupled with the system accuracy requirements, has required the relative movement between the pickups to be held to a minimum. The three sensors and the temperature probe are, therefore, formed into one module. The module itself is supported radially, as indicated in Figure 60, to further minimize movement due to vibration and thermal growth. The electrical cable and magnetic pickups are air-cooled to maintain a maximum operational temperature of 450°F.

c. Radiation Pyrometer

Gas generator turbine blade temperature is measured by means of a lens-type radiation pyrometer. Installation of the probe in the turbine case allows a straight sighting tube to be used, thus allowing consideration of either a lens or aperture-type system. The aperture assembly requires the use of a glass rod to transmit the light back to the fiber-optic cable and is limited by the maximum usage temperature (1000°F) of that rod. Metal temperatures of the turbine shroud can reach 1900°F, and temperatures at the probe tip, 1200° to 1400°F. It is doubtful that the 1000°F glass rod temperature limit could be maintained unless cooled purge air is used.

The aperture system could also suffer from an erroneous maximum temperature signal due to a buildup in light intensity as the blade passes by the probe tip. The lens system has a more precise target area and would not experience this problem. Based on the above criteria, the lens approach was chosen for the C&A application and is shown in Figure 62.

The probe assembly is mounted in the engine combustor case and is surrounded by compressor discharge air at 750° to 800°F. Compressor air passes through small purge holes drilled into the sighting tube upstream of the lens to keep the sighting area purged. The maximum operating temperature (1400°F) of the lens will not be exceeded using the uncooled compressor air for purge.

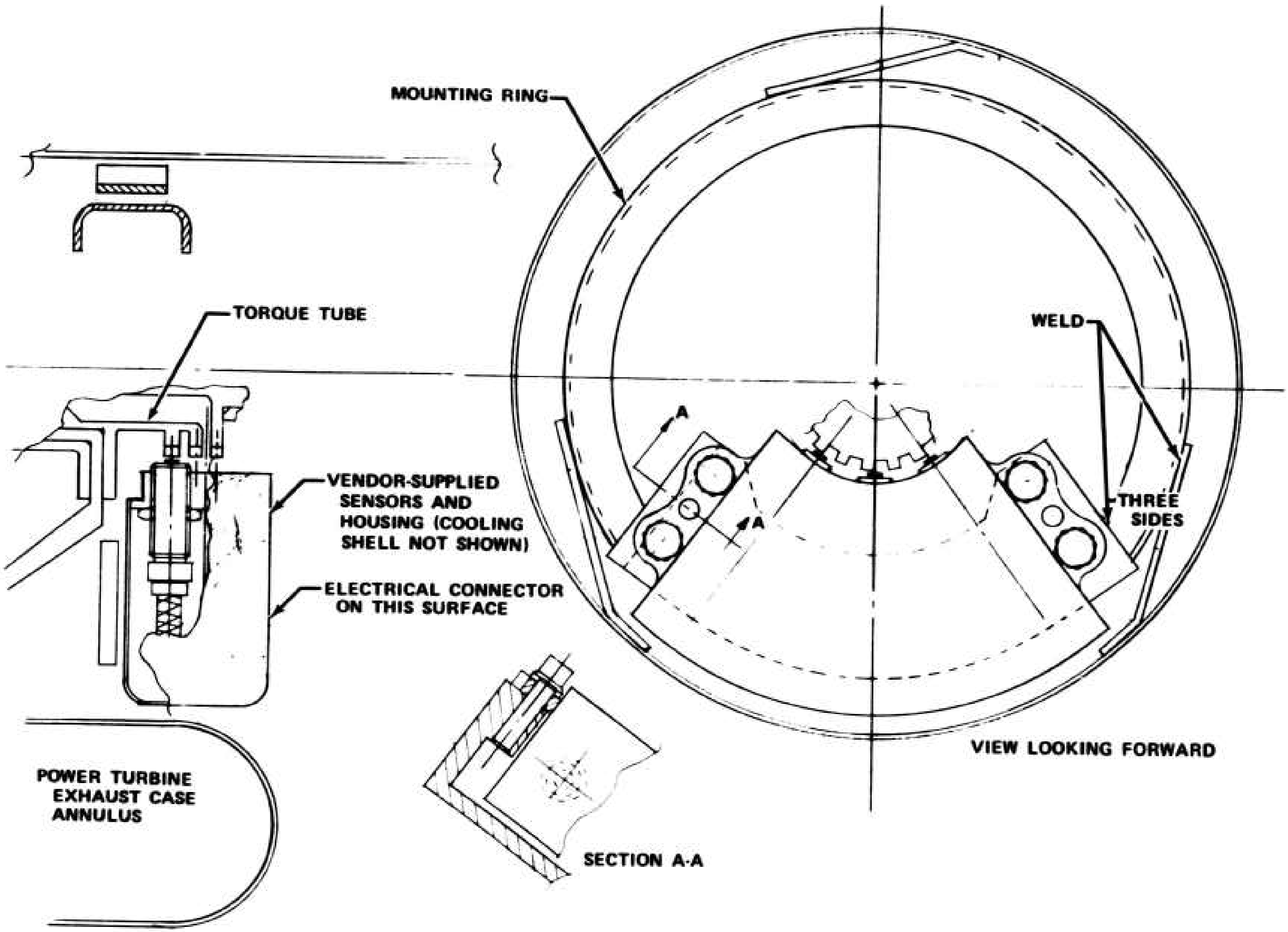


Figure 60. Torque Sensor Mounting Configuration

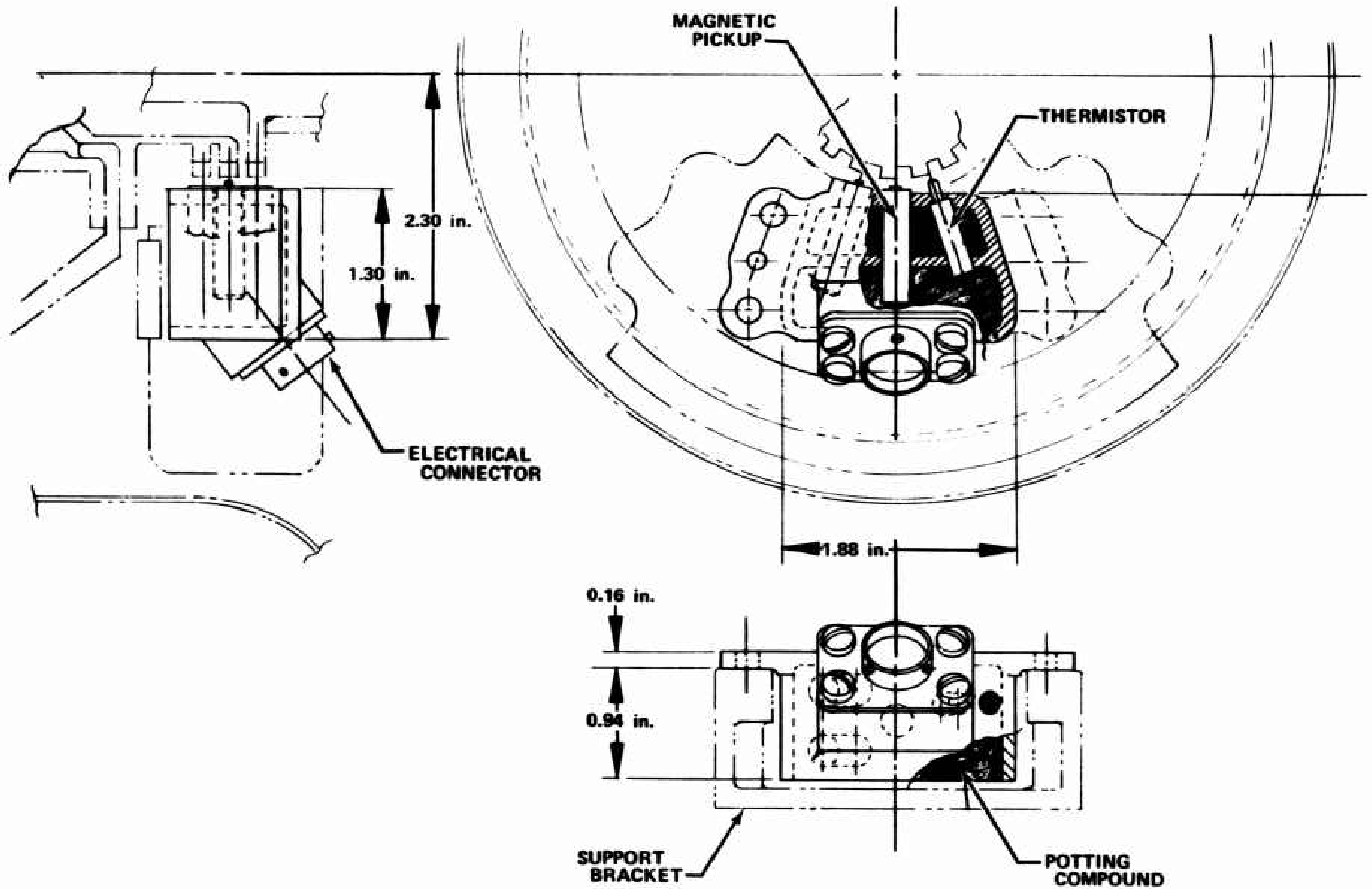


Figure 61. Torque Sensor Assembly

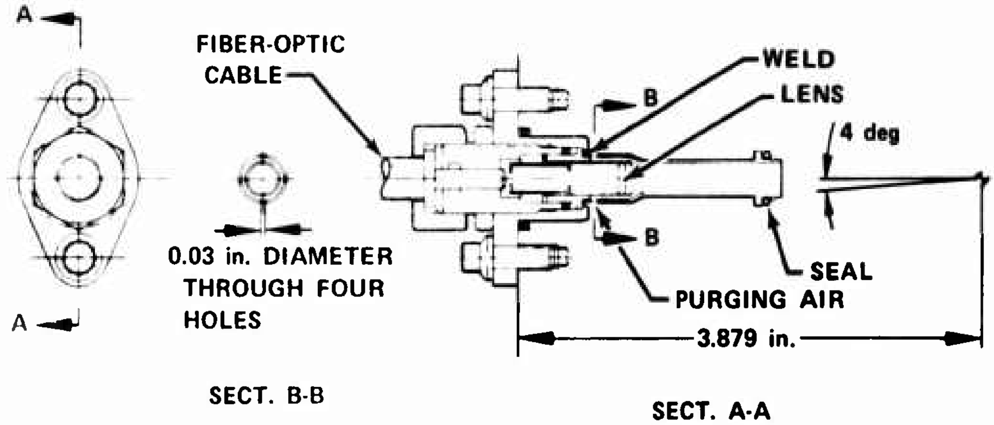


Figure 62. Optical Pyrometer

The lens focuses the light signal back onto the fiber-optic bundle, which then transmits the signal to a silicon detector, located in the electronic control. Maximum air temperature surrounding the fiber-optic bundle should not exceed 165°F, well below the 800°F capability of the cable.

Analytical estimates have been made to determine maximum gas generator turbine blade metal temperature. These estimates are based on the blade cooling scheme shown in Figure 63, where cooling air is supplied to the leading-edge root cavity and makes three radial passes through pedestal arrays before discharging at the trailing edge. The trailing-edge root cavity is cooled by a small amount of air circulating from the base root to the platform plane. Maximum metal temperatures are projected to occur along the pressure side leading edge of the airfoil and between 50 to 80% of the blade height. Design of the probe installation in the turbine case, illustrated in Figure 64, is such that the lens target diameter (0.100 in.) will sweep along all but the initial 25% of the blade length (nearest the root) on the pressure side leading edge, as the blade passes by the probe tip. Although the point of minimum life on the blade is expected to occur on the pressure side, trailing edge at approximately 50% of blade height, time at temperature on the leading edge is directly related to blade minimum life.

The expected range of gas generator turbine blade metal temperature is 1200° to 1800°F.

d. T_{t2} Sensor

The engine inlet air temperature sensor comprises a thermistor network housed in a probe (Figure 65). Around this is an outer perforated tube that allows adequate airflow to the probe and also provides a degree of mechanical protection. The sensor is mounted in the compressor inlet annulus. The range of engine inlet temperature is -65°F to +165°F.

e. P_{t3} Sensor

Burner pressure is sensed by means of a variable-capacitance-type transducer housed within the electronic control to take advantage of the close proximity of the transducer to the signal amplifier. This system results in a high degree of accuracy over the large turndown in sensed pressure, 7 to 220 psia.

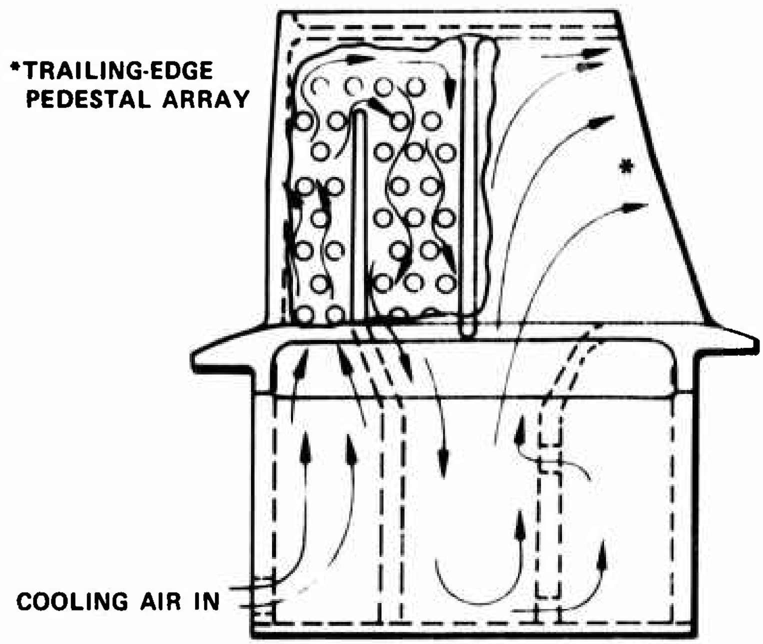


Figure 63. Turbine Blade Cooling

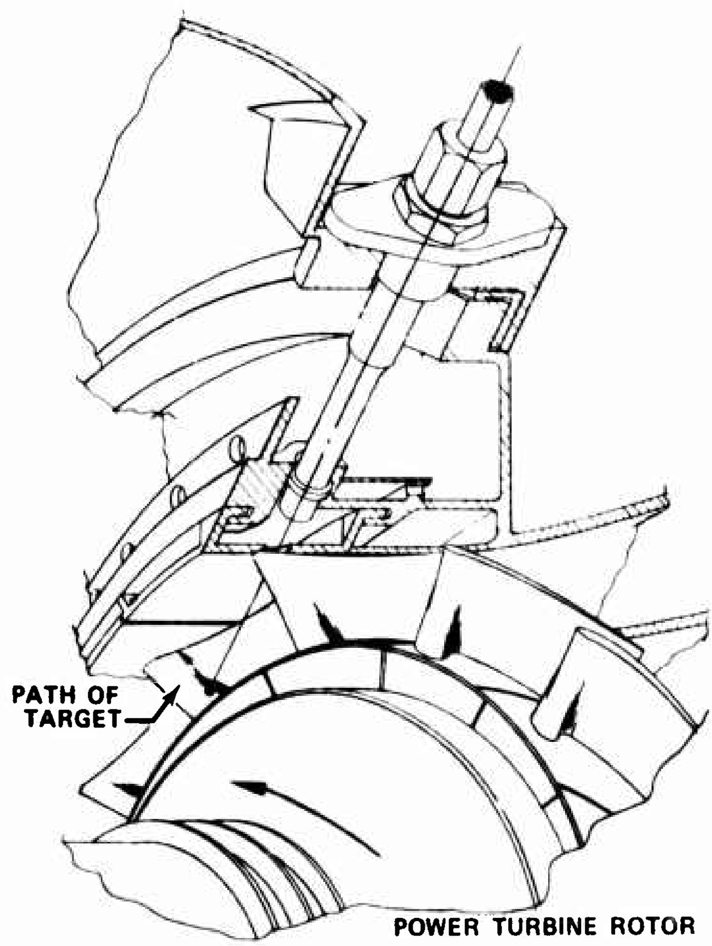


Figure 64. Optical Pyrometer Installation

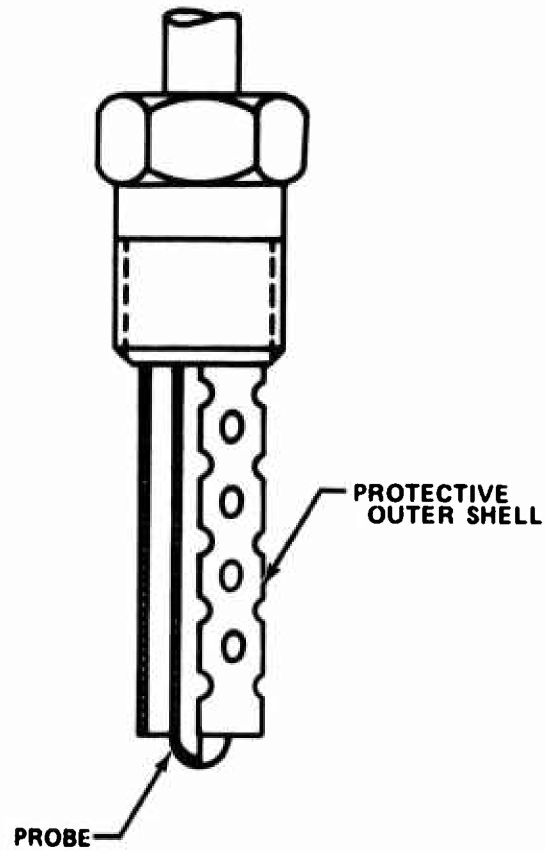
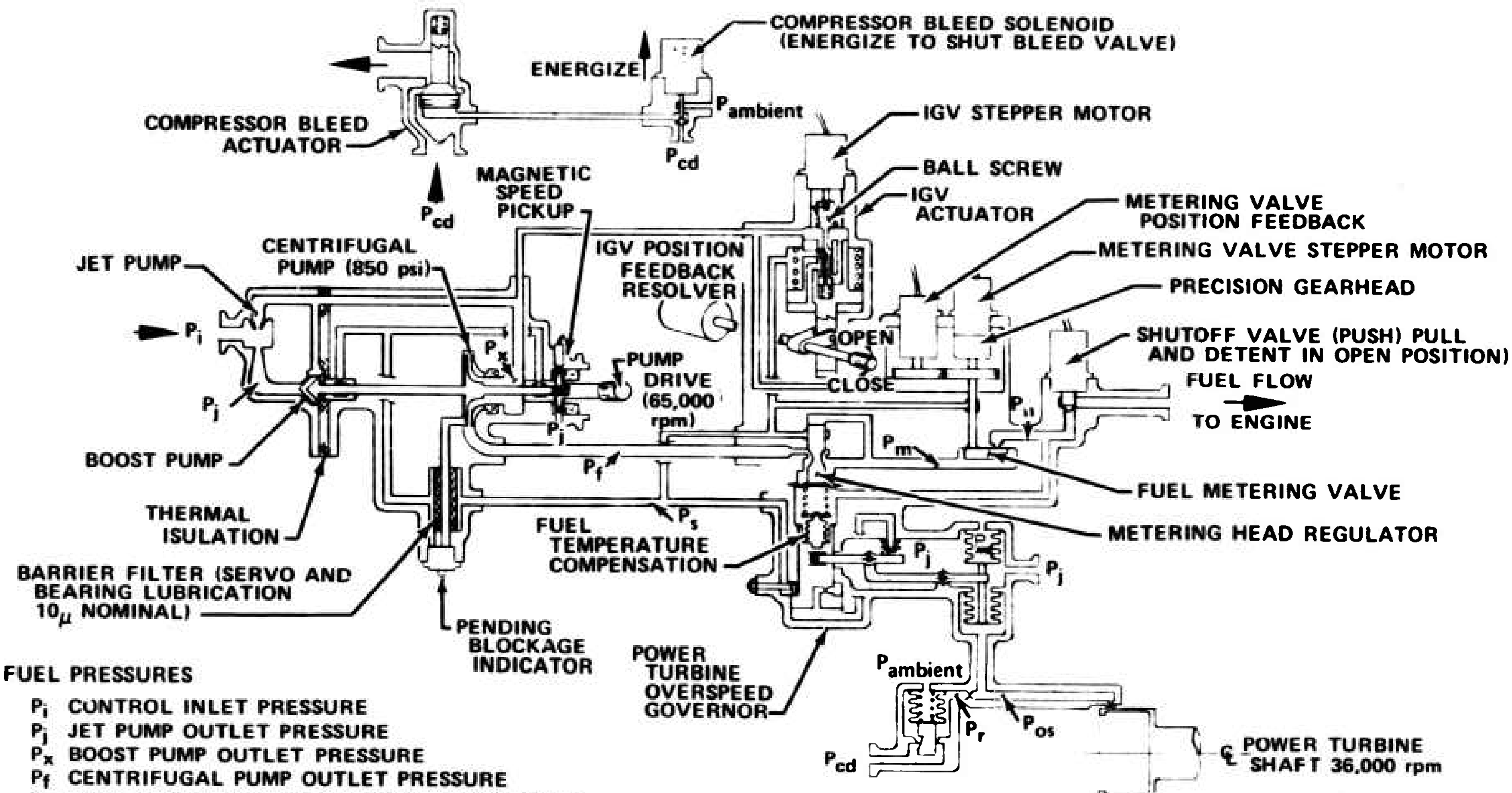


Figure 65. Engine Inlet Air Temperature Sensor

f. Fluid Controller/Overspeed Sensor

The fluid controller, shown functionally in Figure 66, meters fuel flow over the range of 20 to 500 pph to the engine and varies the compressor inlet guide vanes upon command from the electronic control. The package also houses a pneumatic regulator and logic functions for the power turbine overspeed sensor. Package implementation and cross-sectional views of the fluid controller are shown in Figures 67 and 68.

The fuel metering system incorporates a diaphragm-actuated, throttling-type metering head regulator, which regulates the pressure drop across the metering valve to approximately 9 psid. The relatively low pressure drop is used to maximize the flow area of the main metering valve. The valve is positioned by the electronic computer-controlled stepper motor. Fuel flow feedback to the computer is provided by a resolver, which indicates metering-valve position. A push-pull, solenoid-operated shutoff valve is included. A latching-type solenoid is used. Redundant solenoid coils can be included if it is necessary to operate the shutoff valve from the cockpit using airframe power.



FUEL PRESSURES

P_i CONTROL INLET PRESSURE
P_j JET PUMP OUTLET PRESSURE
P_x BOOST PUMP OUTLET PRESSURE
P_f CENTRIFUGAL PUMP OUTLET PRESSURE
P_m FUEL METERING VALVE UPSTREAM PRESSURE
P_n FUEL METERING VALVE DOWNSTEAM PRESSURE
P_s SERVO/BEARING LUBRICATION PRESSURE

AIR PRESSURES

P_{cd} ENGINE COMPRESSOR DISCHARGE PRESSURE
P_r REGULATED PRESSURE (50 psig)
P_{os} POWER TURBINE OVERSPEED SIGNAL PRESSURE

Figure 66. Fluid Controller Implementation

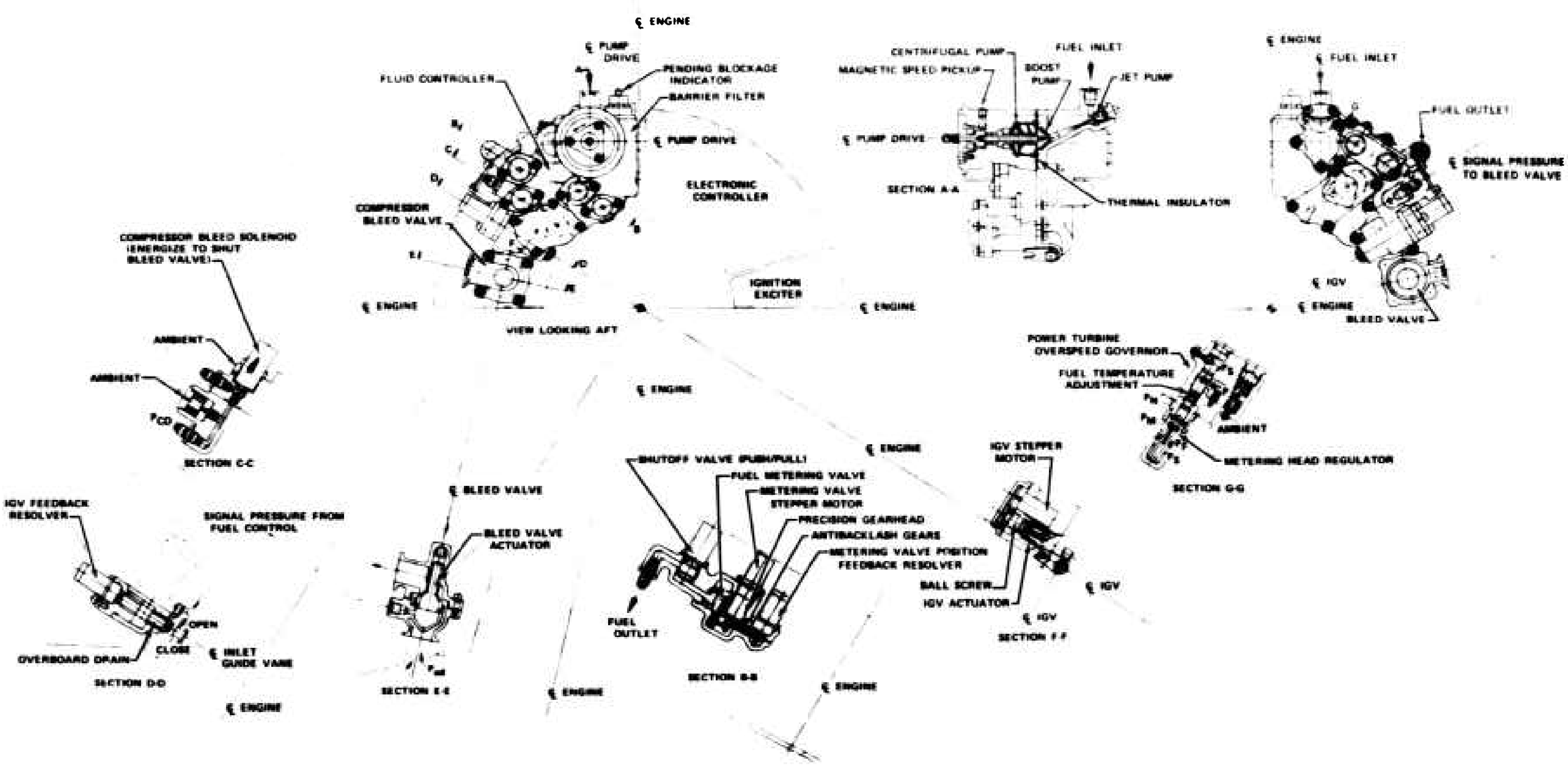


Figure 67. Engine Fluid Controller and Bleed Valve

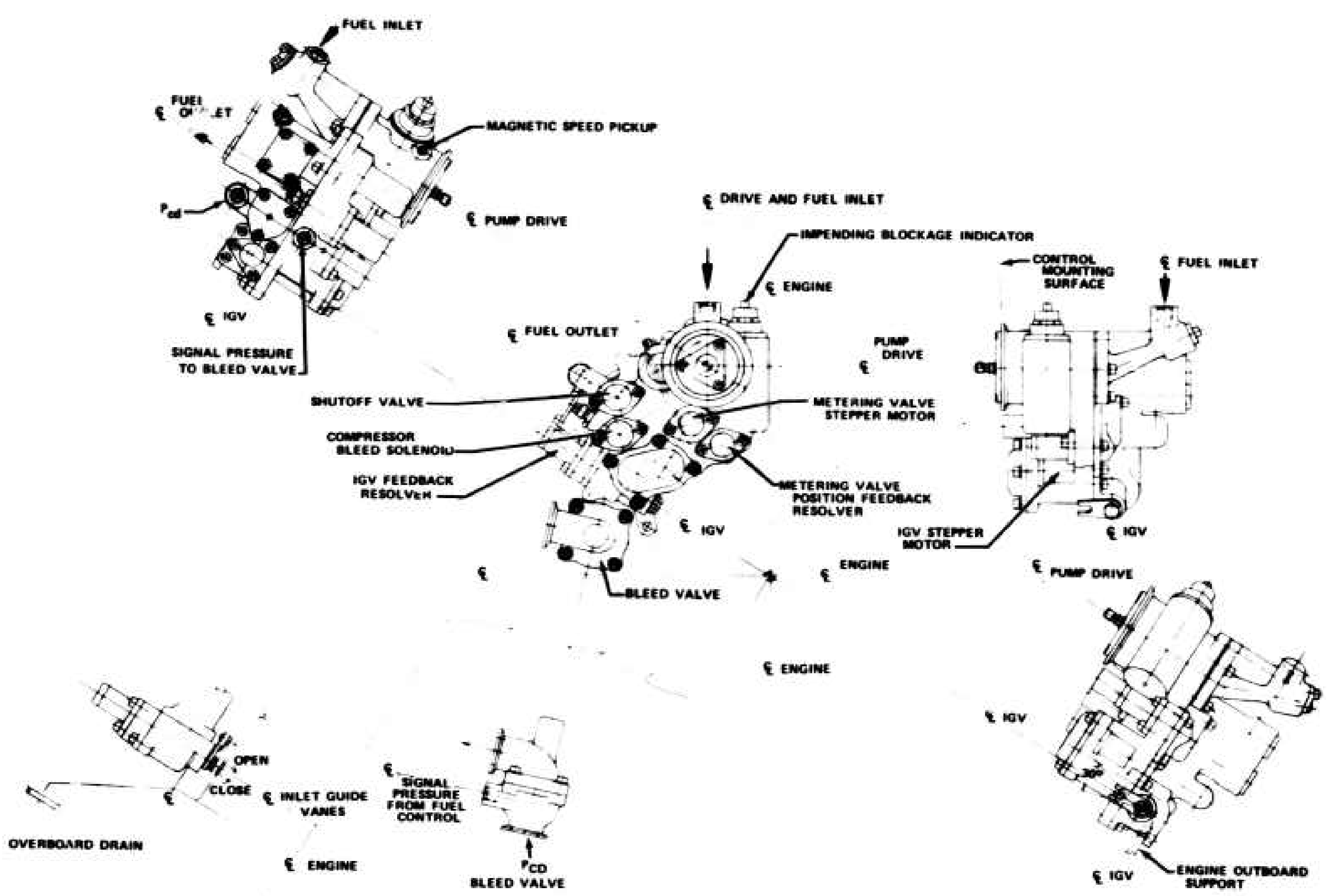


Figure 68. Engine Fluid Controller and Bleed Valve

The hydromechanical schematic also indicates a concept for providing power turbine overspeed protection using the centrifugal deflection of a cylinder of cantilevers for the overspeed signal. This deflection causes a regulated pneumatic pressure signal to be ported to a bellows actuator, which resets the metering head regulator spring load, causing the regulator to close and thereby reduce fuel flow.

The primary part of the power turbine overspeed sensor is a ring of cantilevers splined to the forward end of the power turbine shaft (Figure 69). Centrifugal deflection of the cantilever fingers changes the pressure recovery between supply and receiver ports, thus generating a pneumatic signal. The design airflow required by the sensor is 0.006 lb/sec. The range of power turbine overspeed governor operation is above the normal maximum speed range of the main electronic power turbine governor.

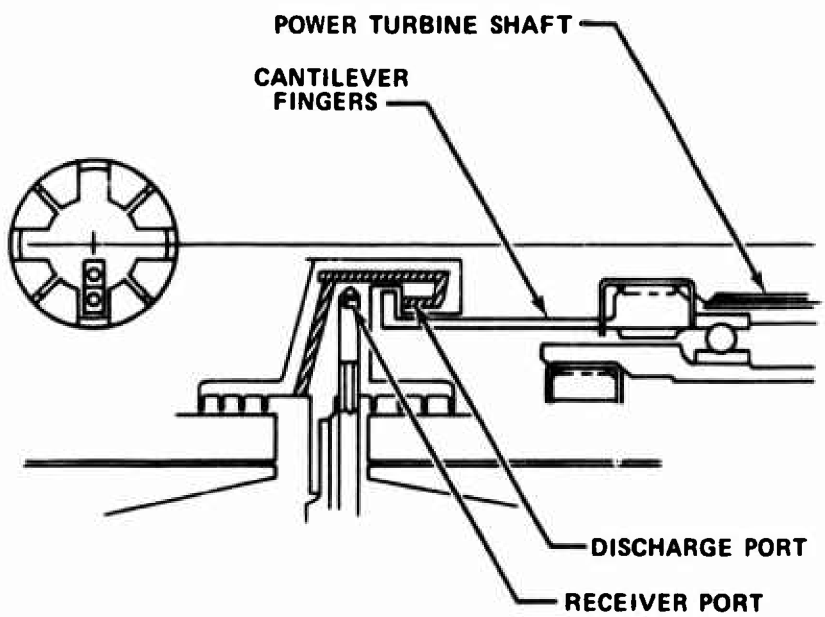


Figure 69. Power Turbine Overspeed Sensor Implementation

The IGV actuator will be incorporated as part of the hydromechanical flow control package, and inlet guide vane position will be scheduled by the electronic control as a function of corrected gas generator speed. The actuator requirement includes a maximum torque of 20 in.-lb, a 50-deg rotary movement, a 100-deg/sec response time, and a ± 1% accuracy. To eliminate vulnerable hydraulic lines, the IGV actuator is an integral part of the hydromechanical package. Output rotary motion is provided by a bellcrank. A resolver is used to feed back IGV position to the computer. Once the fluid controller has been installed on the engine, the IGV system can be indexed via the adjustment mechanism shown in Figure 70. The differential set screw is rotated to provide proper alignment of the IGV actuator output yoke with the engine IGV bellcrank.

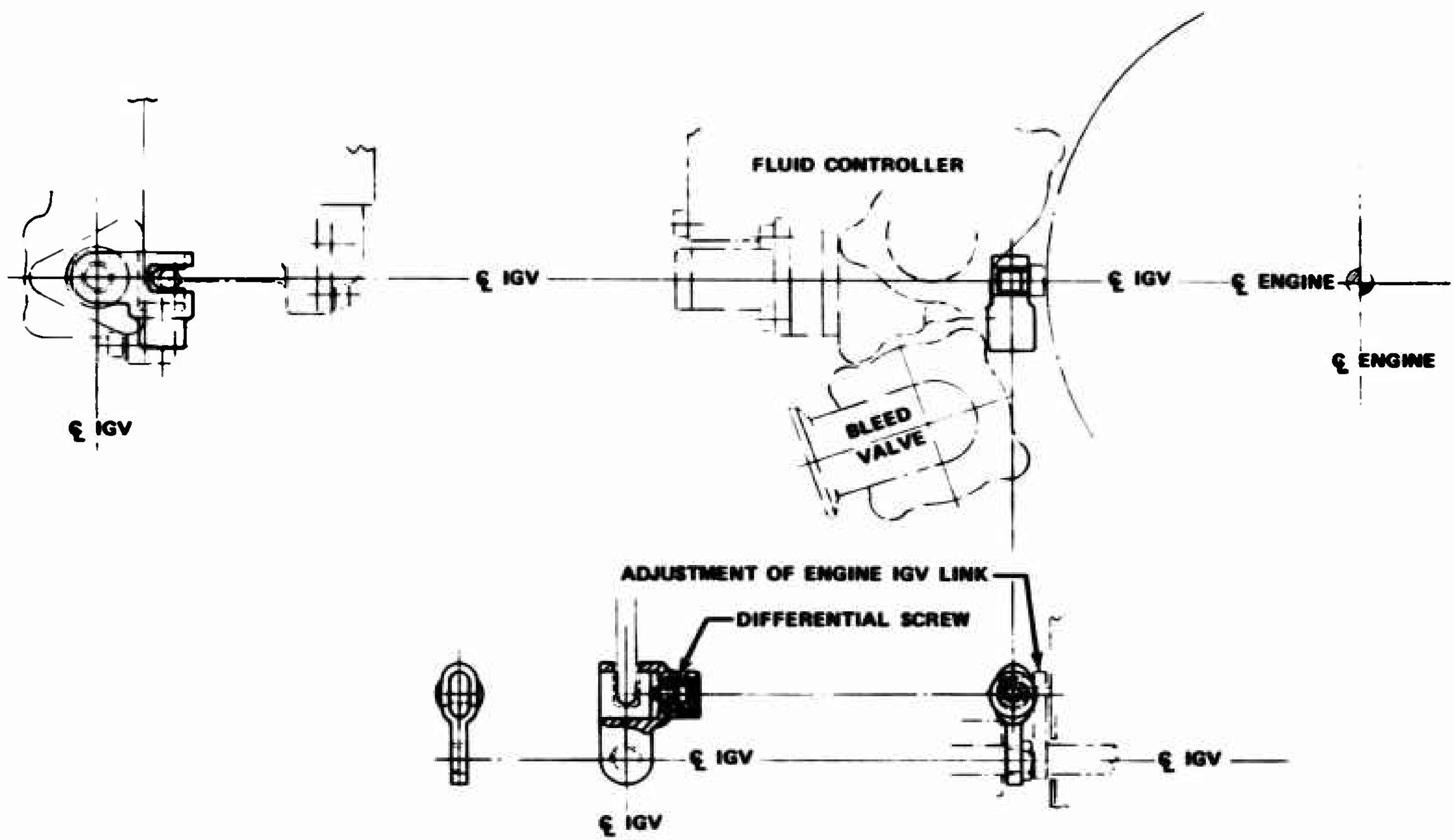


Figure 70. IGV Driving Link Adjustment

g. Fuel Pump

The fuel pump must operate with suction fuel inlet conditions equivalent to an inlet pressure of 1.0 psi above true vapor pressure and a vapor/liquid ratio of 1.0. No dry-lift requirements apply. A maximum pressure rise of 650 psi and a maximum flow of 500 lb/hr are anticipated.

The 65,000-rpm-maximum-speed centrifugal pump, shown in Figure 71, will provide the required starting flow of 10 lb/hr and deliver the required flow and pressure over the normal operating range. A separate high-speed inducer stage is used, and a jet pump is used to charge the inducer.

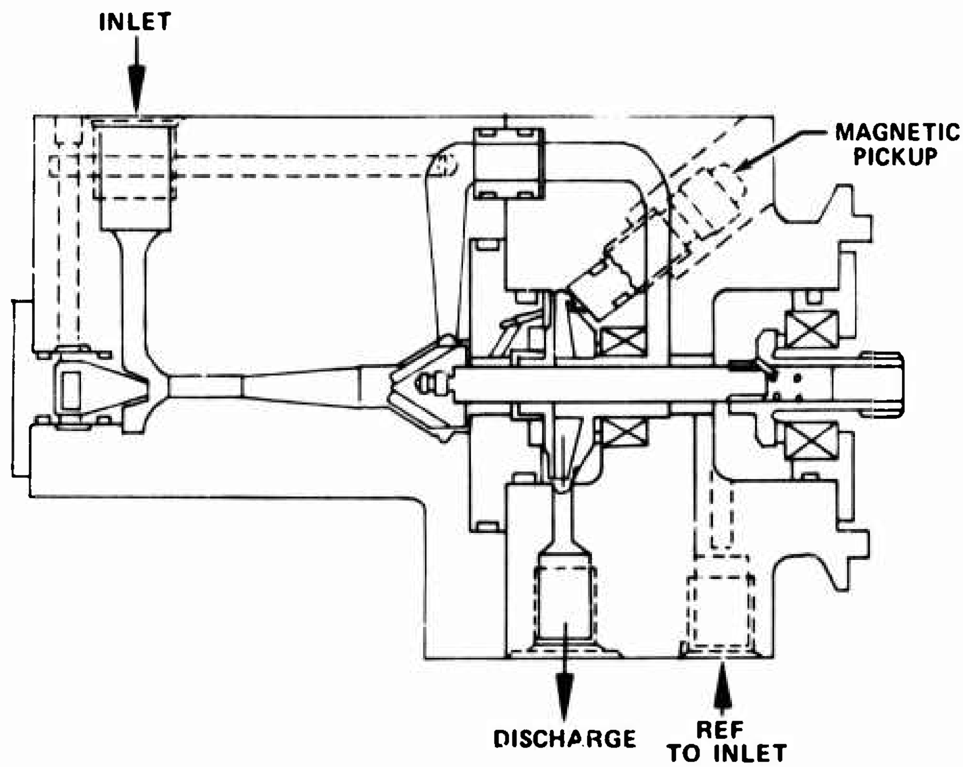


Figure 71. 65,000-rpm Centrifugal Pump

Figure 72 shows the head flow characteristics of the pump. The pump is required to provide 23 lb/hr at 21 psi rise at 10,000 rpm to meet engine starting requirements. Figures 73 and 74 illustrate temperature rise and overall efficiency characteristics, respectively. The pumping system is located in the fluid controller package and is detailed in Figures 67 and 68.

A magnetic pickup is also part of the pump housing to sense gas generator speed. The unit measures rotational speed of the centrifugal pump and provides the signal to the electronic control.

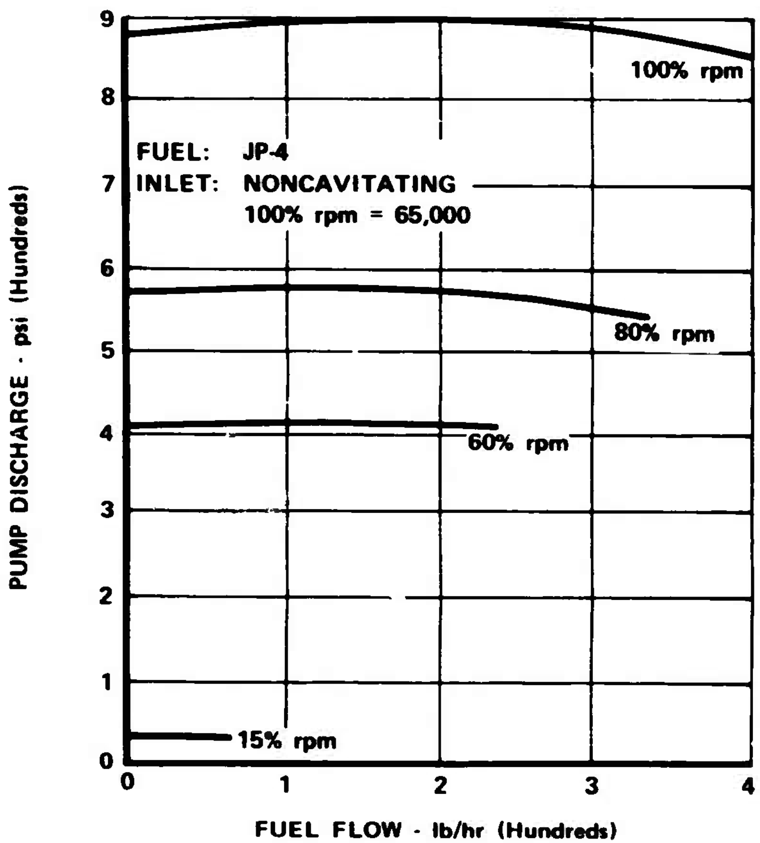


Figure 72. Estimated Discharge Pressure for 65,000–rpm Centrifugal Fuel Pump

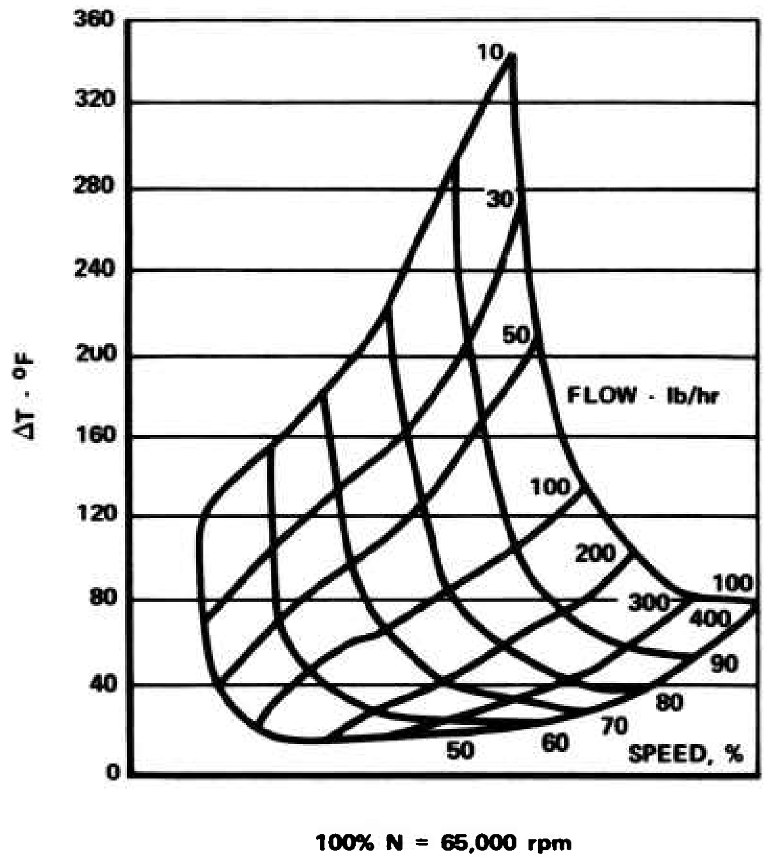


Figure 73. Estimated Temperature Rise for 65,000–rpm Centrifugal Fuel Pump

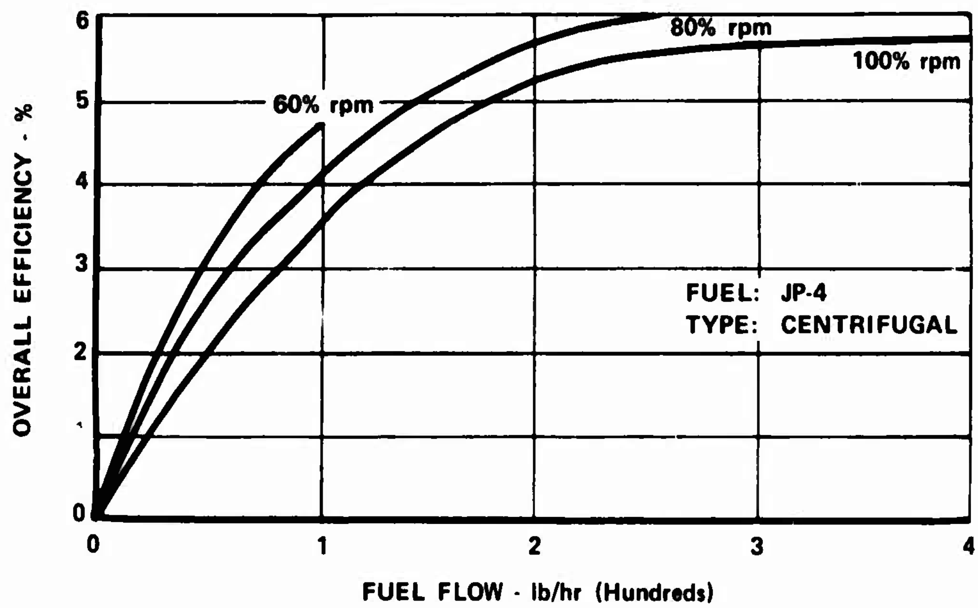


Figure 74. Estimated Efficiency for 65,000-rpm Centrifugal Fuel Pump

h. Compressor Bleed Valve

The compressor bleed valve is a two-position, solenoid-piloted, air-operated valve. The position of the valve is scheduled by the electronic control as a function of corrected gas generator speed. The valve requirements include a maximum operating pressure of 175 psi, an effective area of 0.15 in.^{2}, a 0.200-ms response time, and a maximum operating temperature of 800°F.

The layout drawing and cross-sectional views are shown in Figures 67 and 68, respectively. The solenoid for the control of the compressor bleed valve is included in the hydromechanical package, instead of mounted to the valve. This was done to keep the solenoid away from high-temperature bleed air.

i. Alternator

The alternator is a three-phase machine with a rotating permanent magnetic field. The stator assembly comprises a three-phase output winding and a single-phase control winding. Regulation is achieved by pulsing direct current through the control winding in one direction to increase the alternator output and in the reverse direction to reduce the output. Provision was made in the design for two ignition exciter power windings. Regulation for the alternator is accomplished in the electronic control. The rectified dc output voltage from the alternator is compared to a voltage proportional to the desired alternator output. The resultant error is passed through a dynamic compensation section; it then causes a switching current regulator to either "boost" or "buck" the alternator output, depending on whether the alternator output is lower or higher than required. The alternator is sized at 60w to provide electrical power for ignition and for the electronic control.

Installation of the alternator on the tower shaft is shown in Figure 75. The interface cabling for the alternator is brought out via a flange. The cable is passed through a hole in a flange extension to the alternator support housing. The hole will then be sealed with a grommet or an epoxy compound. The alternator in the tower shaft configuration will require an individual connector for its interface to the electronic control to facilitate assembly/disassembly.

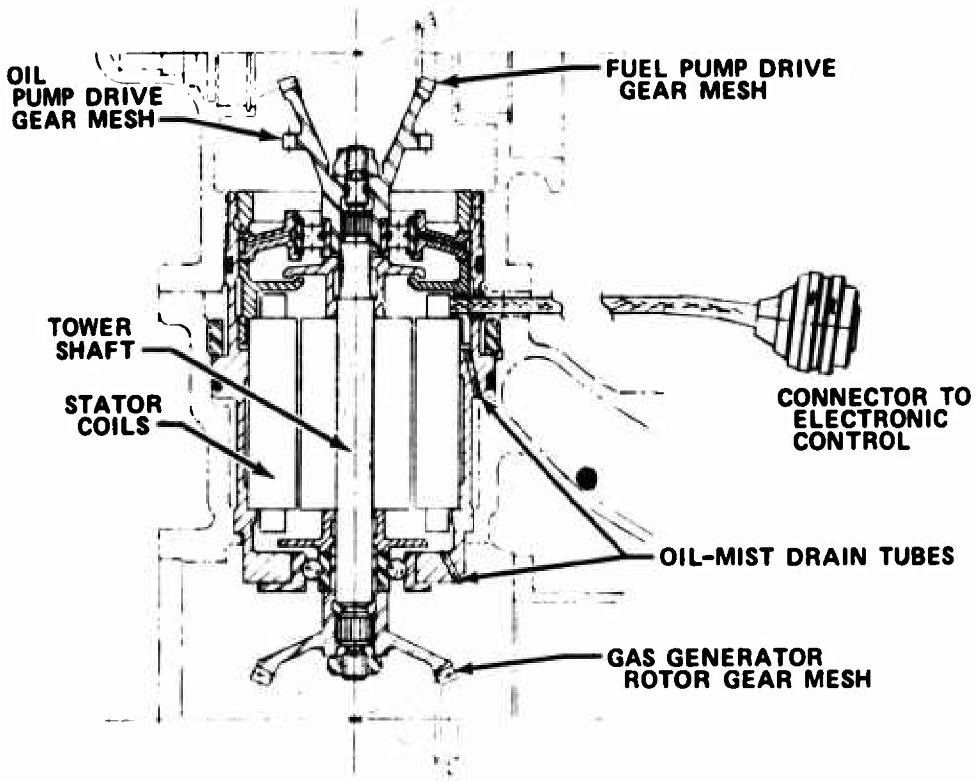


Figure 75. Engine Alternator Tower Shaft Assembly

The alternator/tower shaft assembly is designed so that all the major components are replaceable. However, the entire assembly is considered as a line replaceable unit. A review of the alternator/tower shaft concept indicated that the required spiral bevel gear alignment tolerances are attainable with a replaceable unit. Close tolerances on critical dimensions will assure the replaceability of the unit. Any shimming required would be internal to the alternator and not part of the alternator installation or engine assembly procedures.

The alternator will operate in an oil-mist environment with cooling passages in the support housing to allow draining of oil through the assembly.

j. Ignition System

The ignition system consists of dual-circuit exciters operating dual-redundant igniter plugs. The exciter is a hermetically sealed solid-state device that has two independent circuits, both of which are capable of operating the two redundant igniter plugs that are sequenced for alternate firing. The minimum exciter voltage available for ionization under a maximum demand condition is 5 kv, and the maximum exciter voltage is limited to 5.5 kv. Energy available to the igniter is 1 joule.

The ignition system is powered by a regulated three-phase synchronous alternator with a rotating permanent magnetic field. The initiation of ignition is selectable (automatic or manual), and the "off" condition is accomplished by shorting the ignition winding outputs. Ignition system operation is required throughout 15 to 100% of the rated alternator speed and has an allowable duty cycle of 10 min "on," 20 min "off."

Location of the exciter is defined on the engine cross-sectional layout drawing (Figure 95).

k. Starter System

Engine starting requirements were estimated during the front drive engine studies. As shown in Figure 76, the engine starter system is estimated to require a 5.2 ft-lb output at 16,000-rpm engine rotor speed, where ignition will occur, and operate up to 43,000-rpm engine self-sustaining speed. An air turbine starter configuration was established as a baseline starter system during the front and rear drive analysis.

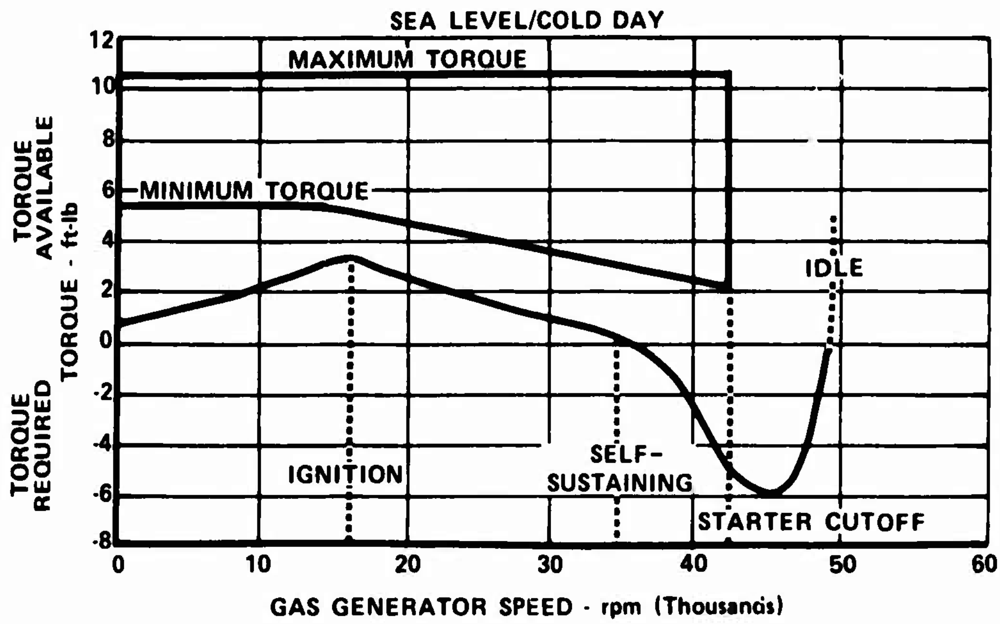


Figure 76. Estimated Starting Torque and Speed Requirements

The Garrett/AiResearch air turbine starter uses a single-stage, axial-flow turbine in a parallel spur gearing arrangement (3-to-1 ratio) with the sprag-type overrunning clutch (Figure 77). The starter is lubricated with pressurized oil from the engine. Starter output is a splined shaft with integral decoupler.

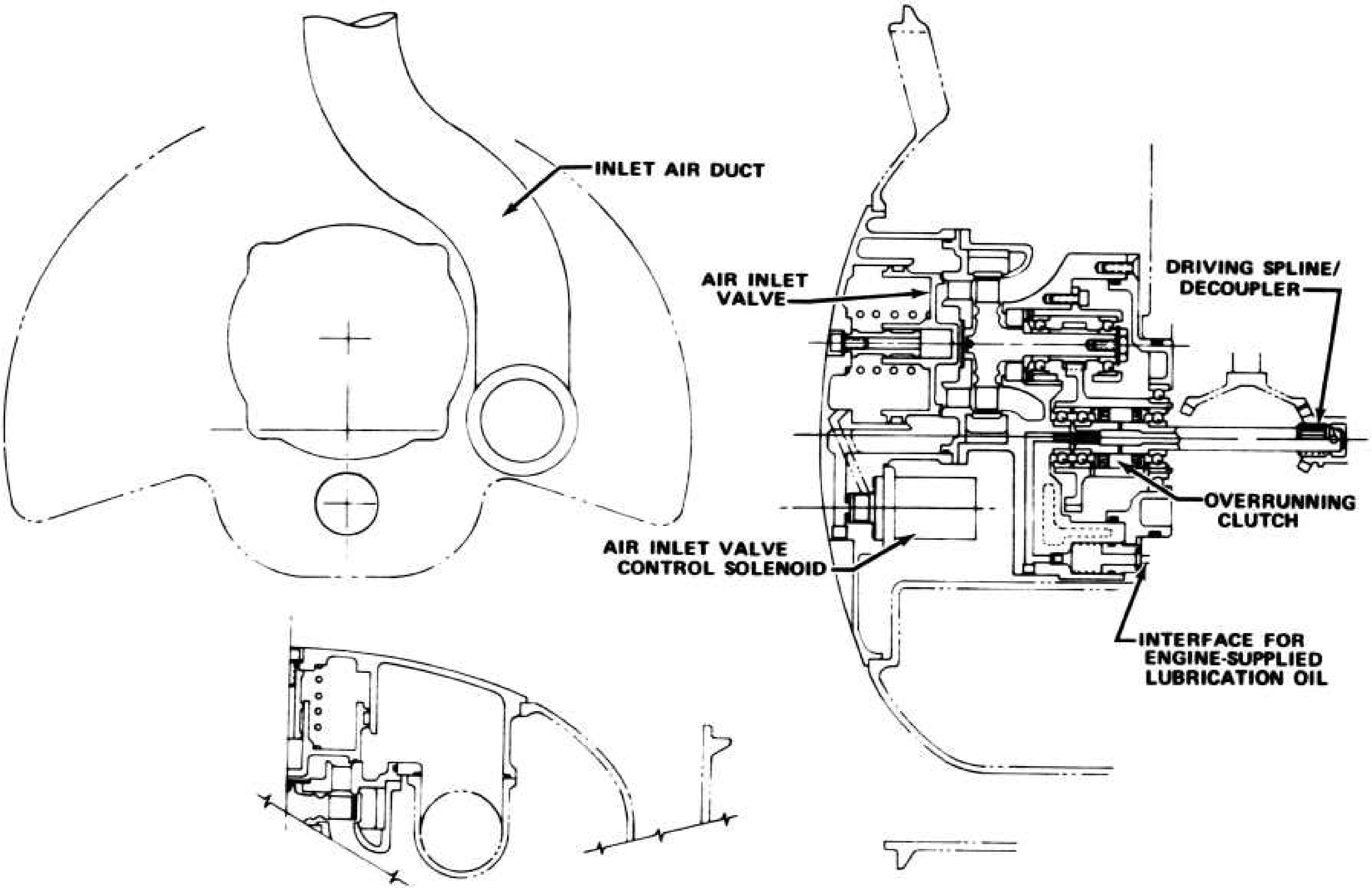


Figure 77. Air Turbine Starter

The starter unit pilots into the engine inlet case and is bolted in place around the outside housing. The air inlet shutoff valve is incorporated in the ATS package and is actuated by a close-coupled solenoid. The solenoid is actuated by a command from the electronic control when the starter cutout speed is reached. A weight and cost advantage results from having control of the shutoff valve solenoid in the electronic control, where sensed speed is available, rather than by means of a control box and sensor mounted on the starter package. The cutoff switch actuation speed is 43,000 to 45,000 rpm. Estimated performance of the ATS is shown in Figure 78.

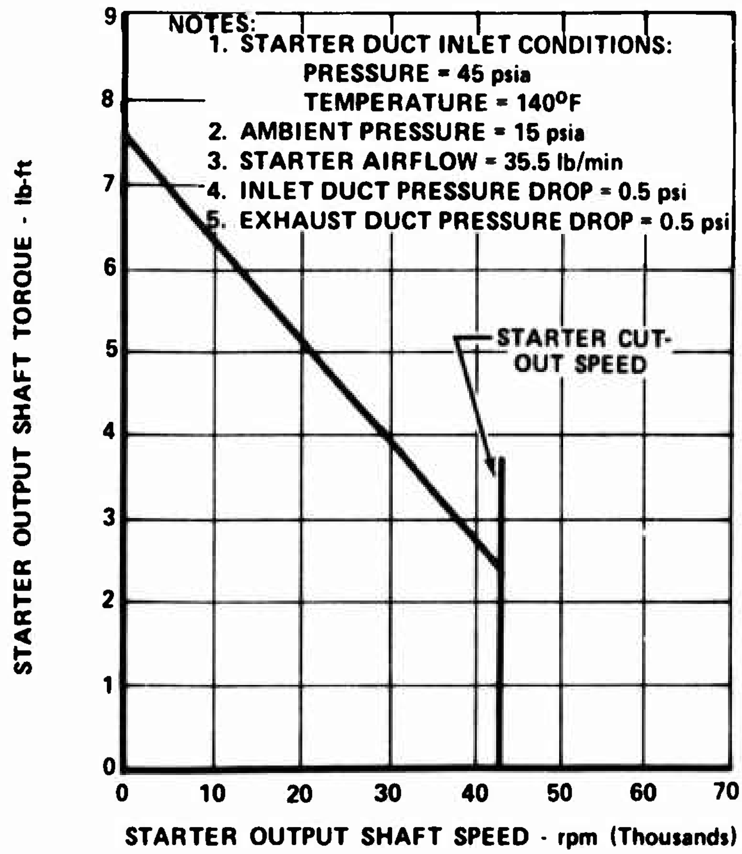


Figure 78. ATS Performance

The starter has a design life of 5000-hr overrunning clutch operation, or 10,000 start cycles.

For a single-engine installation, where an electric motor starter might better fit the overall system requirements, the impact on the study engine configuration will be minimal. An overrunning clutch arrangement would be used, and the engine oil system could potentially be used for bearing and overrunning clutch lubrication. Additionally, engine oil might be considered for cooling the field coils of the starter motor. An electric starter would be located similar to the ATS, except that a larger envelope and heavier support structure would be required.

l. Lubrication System

The engine lubrication system is a dual-compartment, pressure-fed oil system. The front bearing compartment is integral with the oil tank. A 15,000-rpm oil pump/scavenge pump was sized in coordination with Sundstrand.

(1) Lubrication System Description

The major components of the engine lubrication system are: main reservoir tank/heat exchanger, lubrication-scavenge pump, oil filter, bypass valve, scavenge flow strainer, and relief valve. These components are shown schematically in Figure 79. All components will be grouped within the gearbox in the forward part of the engine.

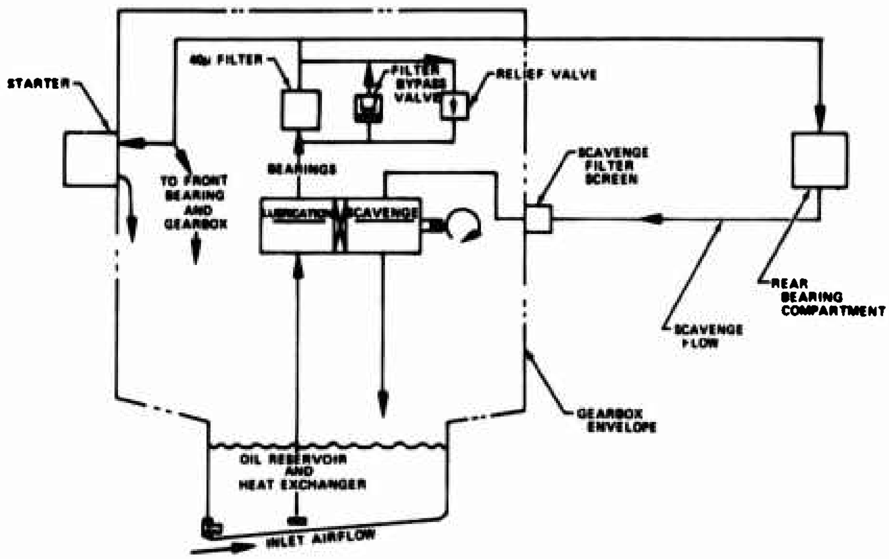


Figure 79. Engine Lubricating System Schematic

Oil from the reservoir enters the pressure element of the pump. It leaves the outlet of the pump and flows through an internal passage to the oil filter. The system uses two valves that are parallel with the filter. One is a filter bypass valve that allows oil to flow directly from the intake to the outlet side of the filter should the filter become clogged; the other is a relief valve that acts as a safety valve for the entire system, bypassing oil directly from the pump to the oil reservoir should excessive system pressure develop.

Oil flows forward and aft from the core of the filter. The forward flow exits from the filter and enters the front compartment to service the gas generator rotor bearing (front of compressor), accessory drive gears, splines, bearings, and the starter gears. After servicing these parts, the oil drains downward through the gearbox components to the primary oil supply reservoir in the accessory drive gearbox housing.

The aft flow of oil exits from the filter to the aft part of the engine to service the power turbine bearings in the rear compartment. A sump at the bottom of the rear compartment collects the scavenge oil, which is, in turn, serviced by the lubrication-pump scavenge element and returns the oil from the sump to the main reservoir.

The oil filter recommended is a 40μ screen-type filter, located in the accessory drive housing in series with the pressure pump. A conical strainer screen will be incorporated in the scavenge flow return line to prevent foreign particles from entering the scavenge pump.

The filter bypass valve is a spring-loaded poppet valve installed in series with the pressure element and parallel with the filter screen. If the filter clogs, a differential pressure of 18.3 psi will open the poppet. The relief valve is located in the accessory gearbox parallel with the oil filter. It is a spring-loaded, poppet-type valve that bypasses oil to the oil reservoir to protect the engine against excessively high pressures.

The main oil reservoir will serve as the oil cooler and will use the inlet air as the coolant. It is designed to limit oil temperature at the lubrication-pump inlet to a maximum of 250°F (120°C).

Magnetic drain plugs are installed on the lubrication system components to be used as system drains and to keep the system free of foreign materials.

Engine lubrication system design data are shown in Table 49.

(2) Oil Pump Description

The main lubrication pump is a two-element vane pump, mounted on the rear face of the accessory drive gearbox. One element supplies oil to all lubrication points within the engine, and the other element is used to scavenge oil from the rear bearing compartment sump. The elements are driven by a common drive shaft in one tandem cartridge arrangement, located in the accessory drive gearbox.

Lubrication oil from the service element will enter through an internal port on the side of the pump and exit through an internal port to the filter screen. An elastomer O-ring, incorporated between the supply and scavenge ports on the pump housing, minimizes leakage.

The oil pump configuration is based on a fixed-displacement vane pump developed by the Sundstrand Corporation. The pump element design is illustrated in Figure 80, and the final pump layout, based on the lubrication system design requirements, is shown in Figure 81.

(3) Impact of Oil Sump Heat Load on Engine Inlet Air

Design of the engine inlet particle separator forces clean air along the periphery of the oil tank and into the compressor inlet annulus. Oil in the tank can reach 250°F and thus can cause an increase in engine inlet air temperature above ambient conditions.

The engine heat loads, applicable to the lubrication system, were determined at both SLS maximum and idle speed conditions, and are summarized in Table 50. Although an idle heat load of 115.5 Btu/min was calculated, it was recommended that a 150 Btu/min heat sink capacity be designed.

A typical helicopter mission cycle includes a cruise condition at 5000 ft, 100 knots, and 60% power. The engine heat load at this operating point was obtained by scaling the SLS maximum power heat load with the cruise condition gas generator speed. The resulting increase in compressor inlet air temperature, at all three points, was determined. Engine influence coefficients relating shaft horsepower (shp) and specific fuel consumption (sfc) to changes in compressor inlet air temperature were determined from performance deck data and established the performance penalties shown in Table 51.

TABLE 49. ENGINE LUBRICATION SYSTEM DESIGN DATA

Engine Lubrication System Data

Type Pump	Recirculating, Positive-Displacement
Oil Specification	MIL-L-7808D MIL-L-23699
Oil Temperature at Pump Inlet	-65 to 250°F
Valve Settings:	
Filter Bypass Valve	15 to 20 psi
Relief Valve	TBD

Oil Pump Design Requirements

Design Criteria	Power Setting Military	Power Setting Idle
Pump Speed, rpm	15,000	11,500
Maximum Oil Temperature, °F	250	250
Minimum Oil Temperature, °F	-65	-65
Maximum Pump Environmental Temperature, °F (15 sec/continuous)	302/176	302/176
Pressure Pump		
*Oil Flowrate, lb/min	43	33
Pump Pressure Rise, psid	70	43
Minimum Pump Inlet Pressure, psia	6.75	6.75
Scavenge Pump		
**Oil Flowrate, lb/min	44	34
Pump Pressure Rise, psid	20	12

*Supply pump oil flowrate value includes 20% overcapacity for growth and aeration.

**Scavenge pump flowrate includes four times capacity for aeration plus a 20% growth factor.

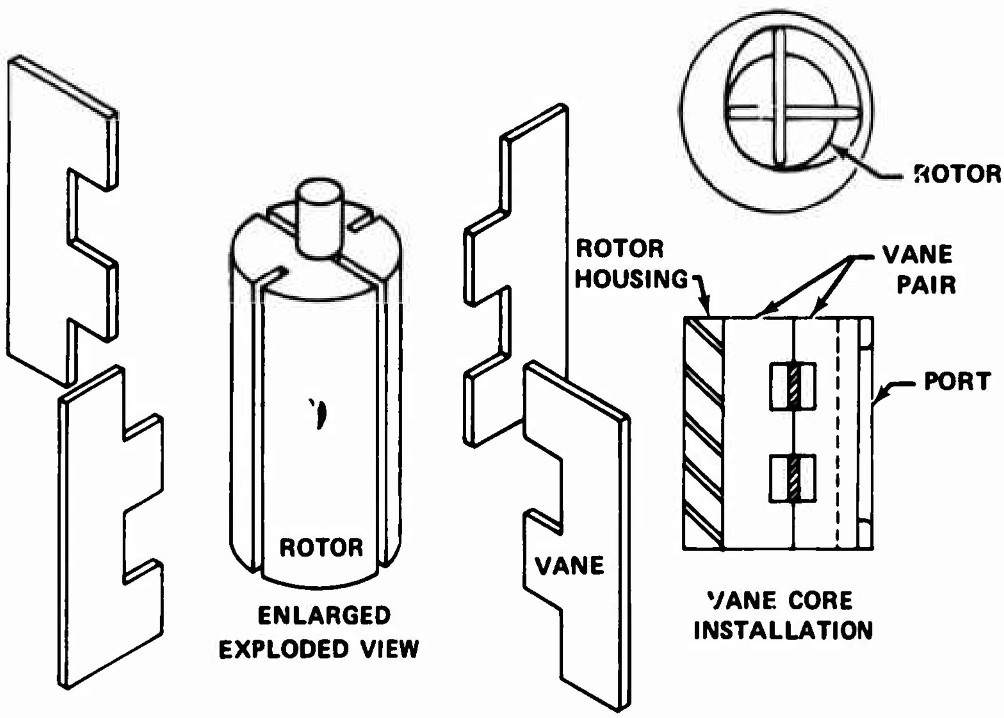


Figure 80. Lubricant/Scavenge Pump Element

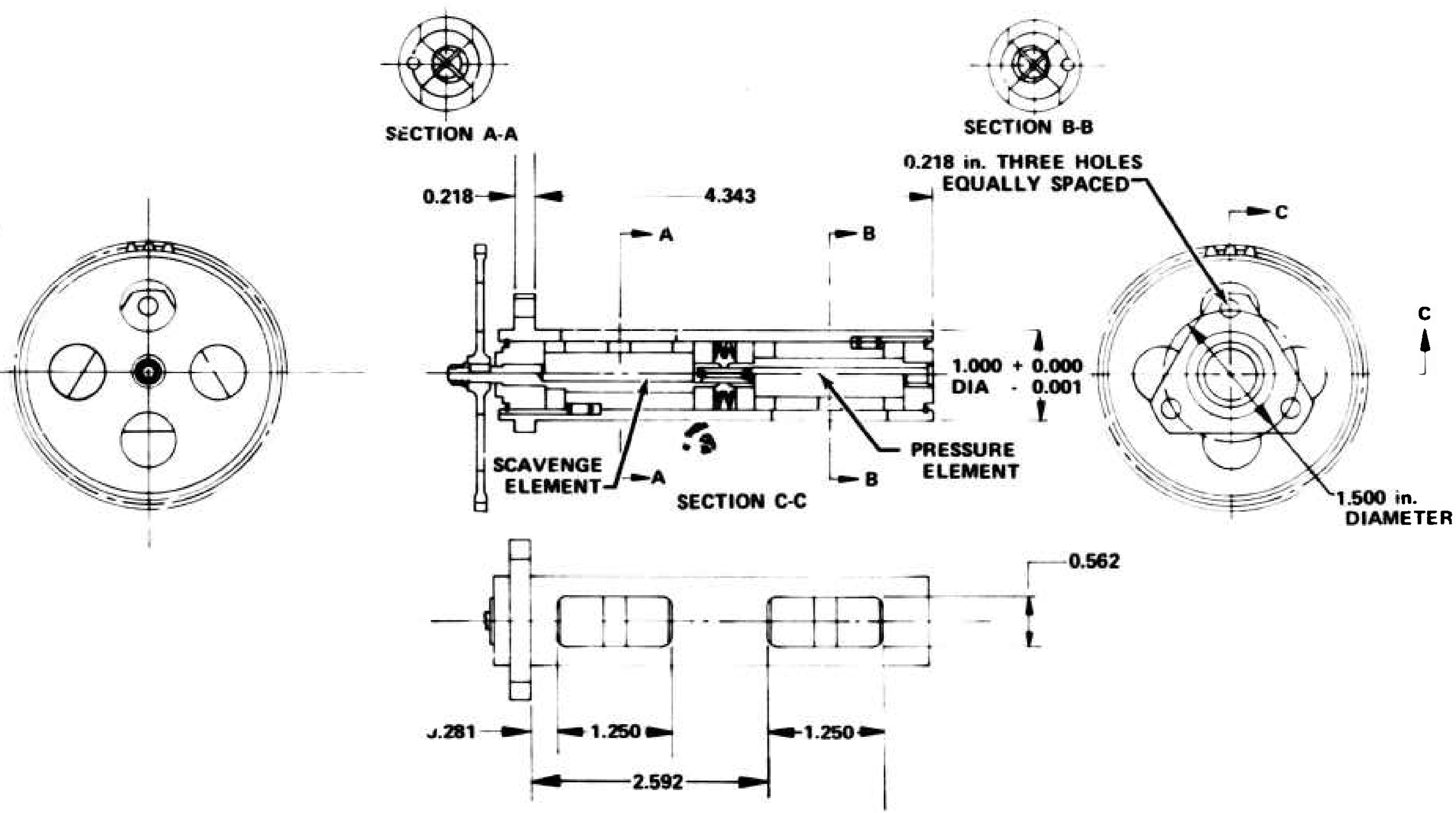


Figure 81. 15,000-rpm Oil Pump

TABLE 50. SMALL GAS TURBINE ENGINE HEAT GENERATION* SUMMARY

Compartment Location	Heat Generation - Btu/min SLS	Ground Idle
Front Compartment Ambient	-47.8	-41.1
Middle Compartment Ambient	Air Cooled	Air Cooled
Rear Compartment Ambient	+17.7	+3.1
Compartment Seal Friction	All Self-Acting ≈ 0	All Self-Acting ≈ 0
Seal Leakage	Negligible	Negligible
Gas Generator Thrust Bearing (Duplex)	132.0	85.0
Power Turbine Thrust Bearing	10.5	10.5
Gas Generator Journal	Air Cooled	Air Cooled
Power Turbine Journal	Air Cooled	Air Cooled
Gearbox	$\approx$58.0**	< 58.0
Total	+170.4	< 115.5

*Applicable to lubrication system
**Q_{gears} = 13 Btu/min, $Q_{bearings}$ 45 Btu/min

TABLE 51. PERFORMANCE PENALTY HEATING OF ENGINE INLET AIR DUE TO ENGINE HEAT GENERATION

Engine Condition	ΔT_{t2}	Increase in sfc, %	Decrease in Maximum Available shp, %
SLS Maximum Power	3.52	0.47	1.4
SLS Idle Power	5.51	0.73	NA
5000 ft/100 kt 60% Power	4.33	0.58	NA

The performance penalties are not considered excessive but, to some degree, can be compensated for. As shown in Figure 82, the oil sump/heat exchanger and IPS could be configured to remove heat from the oil by the IPS scavenge flow only. Entrance to the compressor at the bottom of the engine would be blocked, and fins would be added to the bottom of the oil sump. In addition, the upper portion of the accessory drive cavity could be insulated to minimize the possibility of adding heat to the compressor inlet airstream. Some increase in size of the airflow ducting for the modified IPS configuration is anticipated, but the impact is considered minimal.

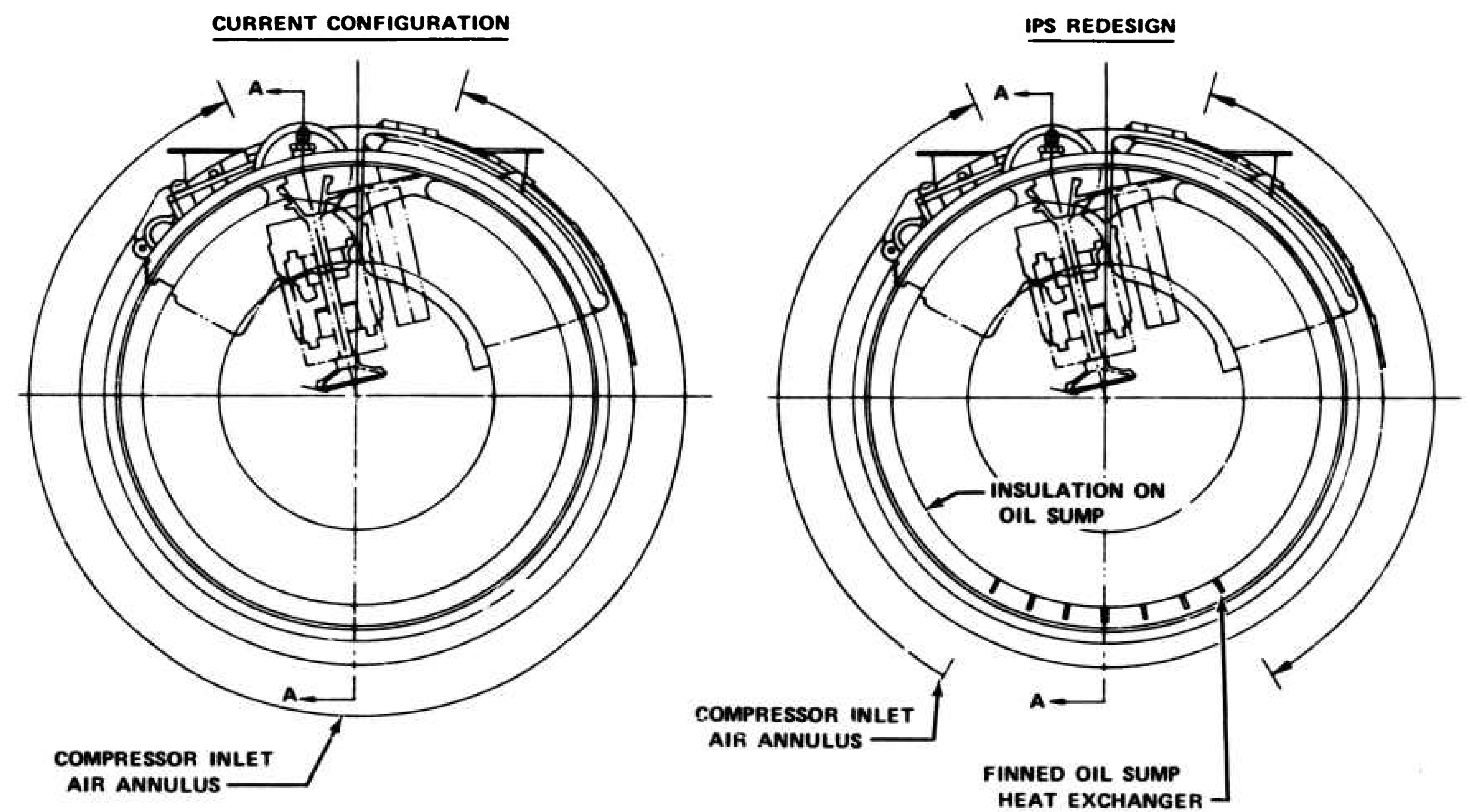


Figure 82. Redesign of IPS Can Reduce Increase in Compressor Inlet Air Temperature Resulting From Engine Lubrication System Heat Load

m. Emergency Lubrication System

The original intent of the program was to evaluate a backup emergency lubrication system with a 6-min capability. Discussions with Army program management and vulnerability personnel indicated that a 30-min backup system capability was more consistent with the airframe gearbox design goals. The study was, therefore, revised to include consideration of both a 6- and 30-min capability system. The most promising configuration for a backup oil system was identified as an air/oil-mist system using a separate oil supply, integral with the main oil tank.

Two approaches to air/oil-mist lubrication systems have been pursued. Oil-mist systems have been tested for full-time operation that use significant air-flows and very small oil flowrates. Limited tests have also been conducted on emergency systems that used much smaller air flowrates at the expense of higher oil flowrates. A conservative approach to a system design for a 30-min capability would consider the steady-state design criteria. For the purpose of this evaluation, reliable operation over a 30-min time span was considered a steady-state condition. The engine performance penalties for the bearing air supply using the steady-state design criteria were acceptable during backup system operation, and the small oil flowrates were compatible with a goal of a minimum emergency oil reservoir size. During the course of the evaluation of a system using the full-time system criteria, it was determined that the bearing cooling-air/engine-inlet-air heat exchanger size requirements were unacceptable from an engine installation standpoint. This evaluation assumed a reasonable increase in demonstrated bearing inlet temperatures.

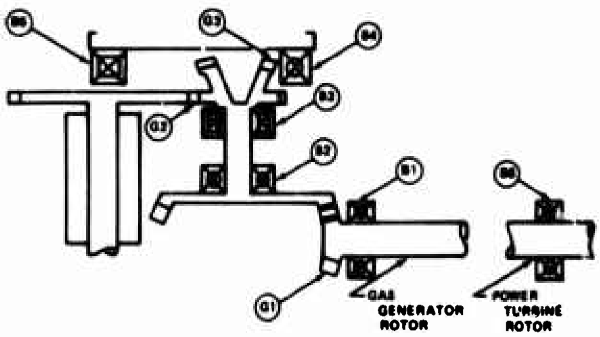
A system design using the limited test data on air and oil flowrates for an emergency system was then evaluated. The study showed that the bearing cooling-air/engine-inlet-air heat exchanger could be integrated acceptably as a part of the engine inlet particle separator bypass duct. The required emergency oil reservoir capabilities could also be incorporated without significant design compromises.

A system design that is optimized for reliability and minimum impact on the engine design will probably be somewhere between the two cases evaluated. Experimental verification of a detailed system design will be required to confirm the exact applicable design criteria.

(1) Design Requirements

The design requirements for the air/oil-mist system were established by reviewing documented test programs, by referring to design guides for commercial air/oil-mist installations, and through informal discussions with personnel now involved with experimental efforts. The most extensive testing of air/oil-mist-lubricated, high-speed, small-bore bearings is being performed by SKF Industries, Inc., under NASA contracts. Informal discussions with NASA and SKF personnel helped to establish the guidelines for this application study.

The integrated accessory systems engine bearings do not exceed size and DN values of 19 mm and 1.24×10^6, respectively. Maximum bearing speed would be 65,000 rpm. The engine bearing configuration is illustrated in Figure 83.



LOCATION	DIAMETER, mm	rpm	BEARING DN OR GEAR PITCH-LINE VELOCITY
B1	19	65,000	1.235 x 10^6
B2	12	30,000	0.36 x 10^6
B3	13	30,000	0.39 x 10^6
B4	12	65,000	0.78 x 10^6
B5	12	65,000	0.78 x 10^6
G1	25.4 56.1	65,000 30,000	17,017 ft/min
G2	36.2 73.2	30,000 15,000	11,250 ft/min
G3	42.7 19.7	30,000 65,000	13,191 ft/min
B6	19	36,000	0.69 x 10^6

Figure 83. Tower Shaft Drive System Bearing DN's and Gear Pitch-Line Velocity

(2) A System Design Based on Full-Time Design Criteria

Results of preliminary testing, based primarily on a 46-mm ball bearing running at 38,000 rpm and a 25-mm bearing running at 44,000 rpm, indicated that a 6-min or 30-min air/oil-mist lubrication system was possible for the engine size and speed range studied. The 46- and 25-mm bearing-air/oil-mist tests used 15 and 10 scfm of airflow, respectively. These data points were plotted as a function of DN and linearly extrapolated to determine air flowrates for the bearing sizes and operating speeds studied. A precise extrapolation of these data would involve determination of the relative bearing heat loads. Since prediction techniques for bearing heat loads in an oil-mist environment were not established, a linear extrapolation was considered to be a conservative approach, since the maximum system bearing DN was between the data points considered. The requirements are detailed in Table 52.

No significant distinction between the requirements for a system with a 30-min and a system with a 6-min capability was made in terms of air or oil flowrates. High-speed bearings usually fail rapidly after the lubrication or heat exchange medium is removed, or reduced to a level where the bearing fits and clearances are mismatched due to thermal distortion. Some reduction in air/oil flowrates for the lighter loaded gearbox bearings and the gearbox drive gears may be possible for a 6-min system, but would not significantly impact the total system design requirements.

TABLE 52. AIR AND OIL FLOW REQUIREMENTS

Bearing Position	Airflow, scfm	Oil Flow, cc/min
Tandem Ball Pair	28	0.28
Tower Shaft Drive Ball	11	0.11
*Power Turbine Ball	23	0.23
Tower Shaft Ball	4	0.04
Tower Shaft Roller	4	0.04
Fuel Pump Ball	7	0.07
Fuel Pump Roller	7	0.07
Gears (3)	15	0.15
Total	99	0.99

General Notes:

(a) Air in at 170°F
(b) The above schedule uses 0.01 cc/min oil flow for each (1) scfm airflow.
(c) Resulting bearing outer ring temperatures ~350°F

*Air/oil flow doubled because of hot environment

(a) System Implementation

The air/oil-mist distribution system implementation is shown in Figure 84. Static pressure is bled from the 10:1 centrifugal compressor at a selected location along the shroud. Because the airflow required (4.2% $\dot{W}_{ae}$) would cause an unacceptable performance penalty during continuous operation, the airflow is bled only upon loss of primary oil system pressure by action of an on-off, oil-pressure-operated valve. A significant part of the successful testing accomplished to date used low-temperature bearing cooling air (170°F). As a first attempt at a system design, this limit was considered. To maintain this limit, a heat exchanger would be required. The system pressure requirement was sized for an altitude (5000 ft) idle power operating condition, but must also function at a sea level maximum power point. To compensate for the variation in static pressure, a pressure regulator is required.

As mentioned previously, the original oil reservoir configuration assumed an oil supply integral with the main oil tank. However, both SKF and commercial suppliers (Alemite) of oil-mist systems have determined that to keep the oil particles from wetting out on the distribution piping, the mist velocity should not exceed 24 ft/sec. This requirement imposes large line sizes and the design difficulty of getting the mist to the various bearings.

Considering the very small total backup oil quantity required (30 cc), it was determined that a better approach would result with three oil reservoirs at the major bearing locations: the accessory gearbox, the front bearing compartment, and the rear bearing compartment. This approach will allow high supply air velocities and small tube diameters. It also results in minimum pressure loss, which will keep the system supply pressure requirement at a minimum and, consequently, the temperature rise that will size the heat exchanger.

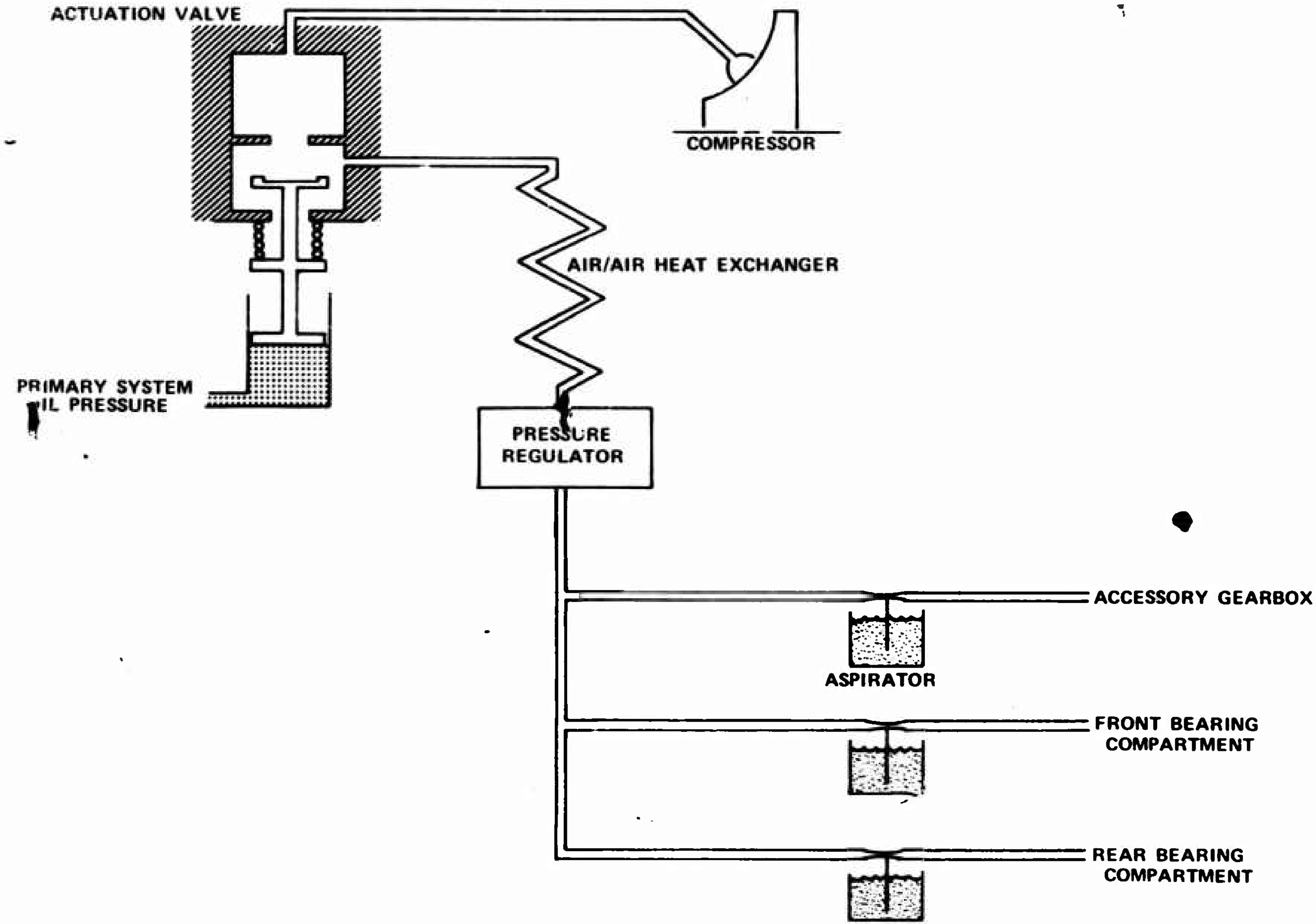


Figure 84. Air/Oil - Mist Lubrication System

Aspiration of the oil will occur at the delivery points rather than having one remote reservoir and aspirating device. The geometries of this system have been defined and are shown in Figures 85 through 87. The distribution manifold supplying mist to the various bearings is illustrated in these figures. The aspirating devices would be located in close proximity to these manifolds.

(b) Design of Distribution System

Having established a design philosophy for the air/oil-mist system and supply of the mist to the various bearings and gears, the minimum operating pressure and heat exchanger size were determined. A worst-case plumbing roadmap (rear compartment) was laid out showing line lengths and sizes, bends, and valving so that the system pressure drop could be calculated. (See Figure 88.) The system was sized for operation at a 5000-ft idle-power condition and resulted in a static pressure requirement of 23.2 psia at the compressor shroud. Figures 89 and 90 show temperature and static pressure at various locations along the compressor shroud and indicate that, at the pressure tap location required, an air temperature of 660°F would result at a sea-level, hot-day, maximum-power condition. The maximum air supply temperature sizes the required heat exchanger.

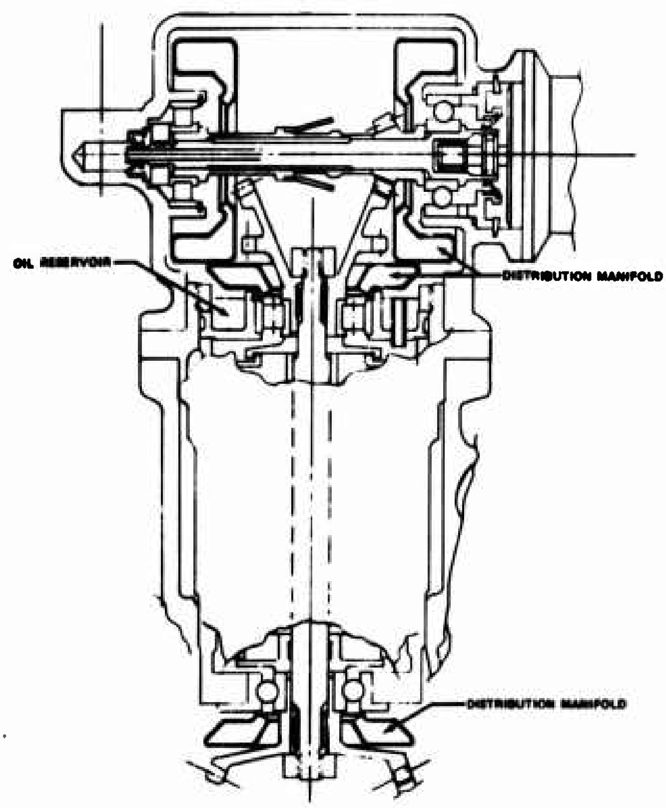


Figure 85. Control and Accessory Gearbox Bearing Compartment

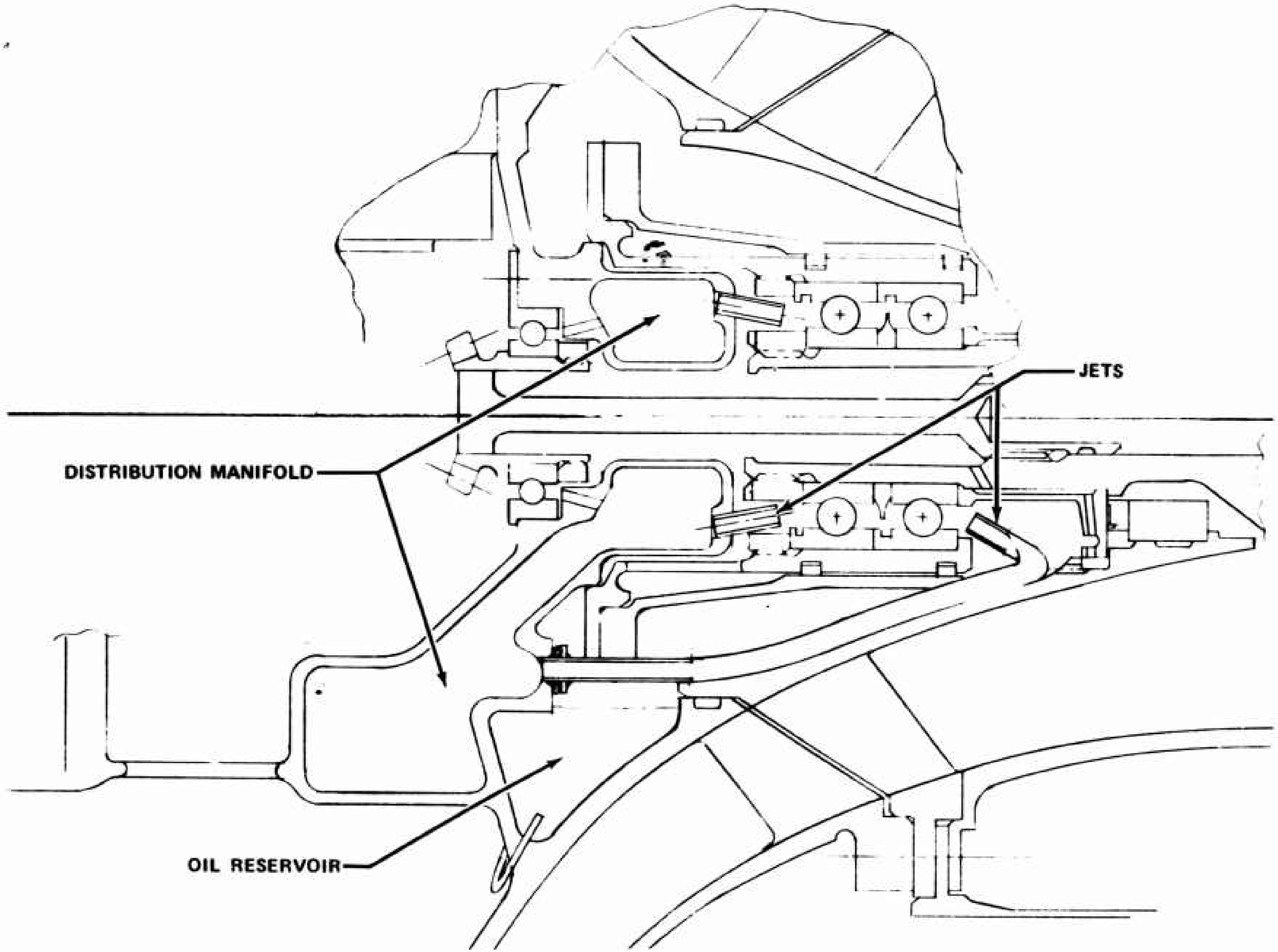


Figure 86. Front Bearing Compartment

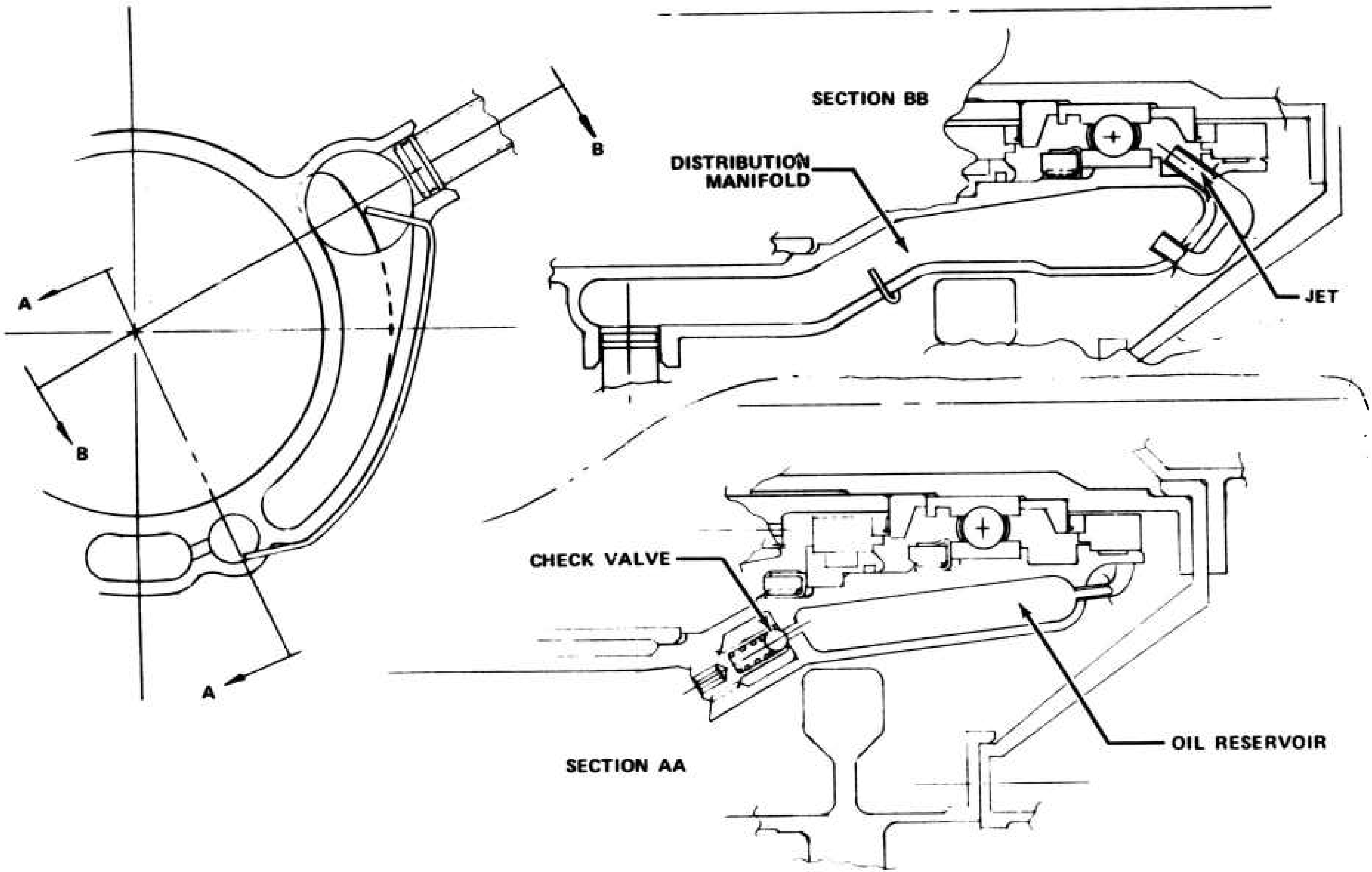
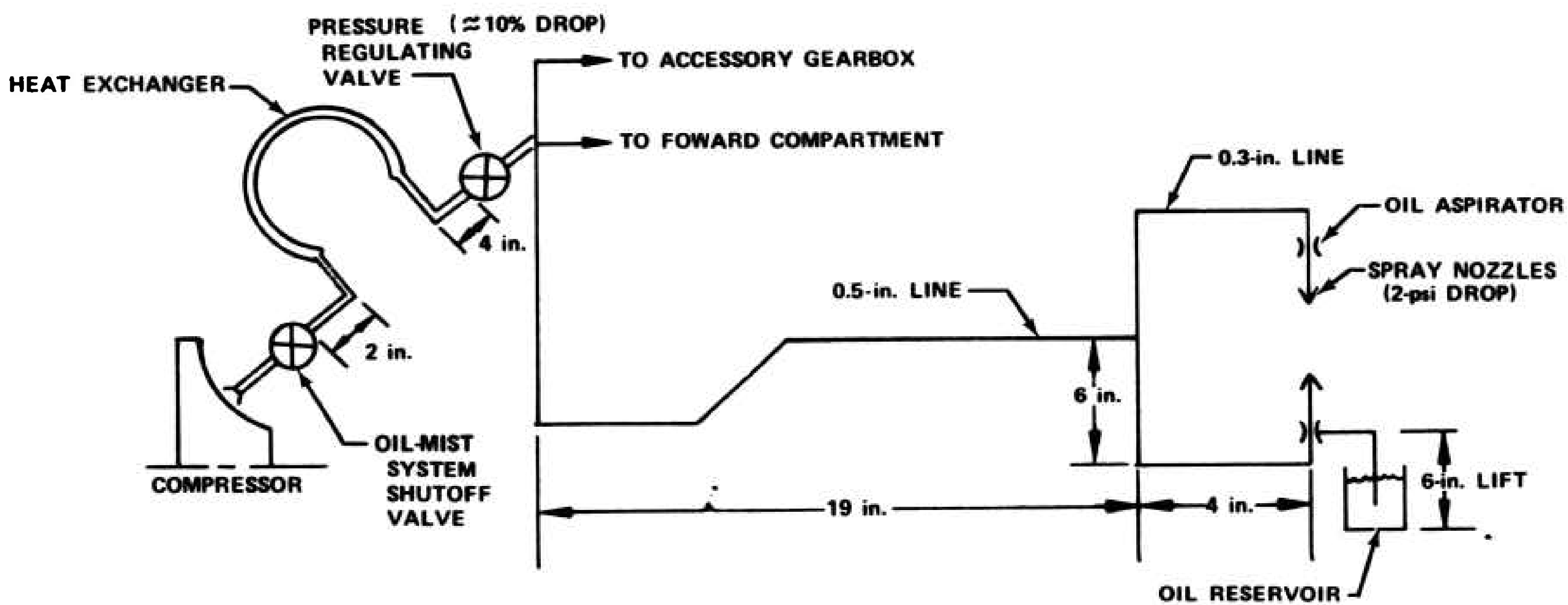


Figure 87. Rear Bearing Compartment



NOTES:

1. MAXIMUM IPS SURFACE AREA AVAILABLE FOR COOLING = 66.23 $in.^2$
2. THERE ARE SEVEN 90-deg BENDS
3. $P_{ambient}$ = 12.23 psia (5000 ft)

Figure 88. Oil-Mist System Plumbing

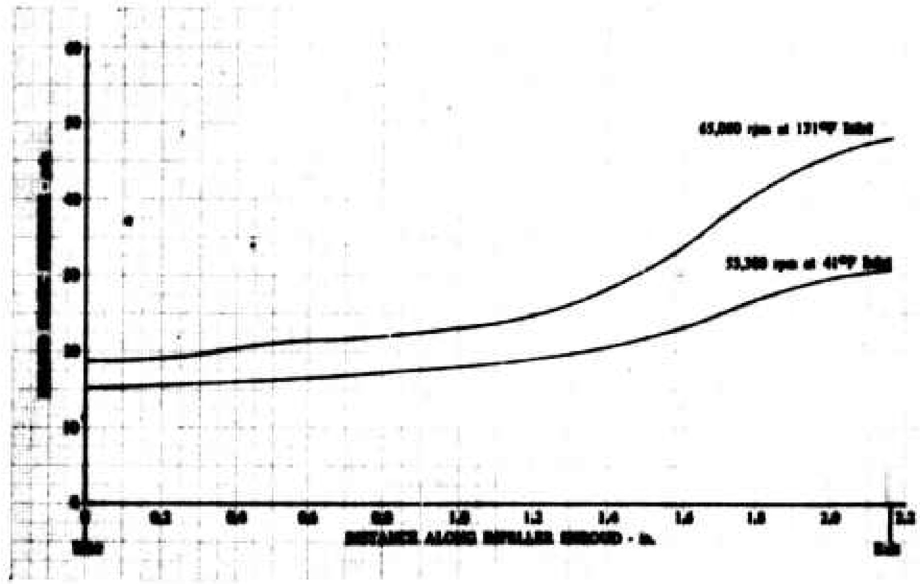


Figure 89. Estimated Shroud Static Pressure Variation

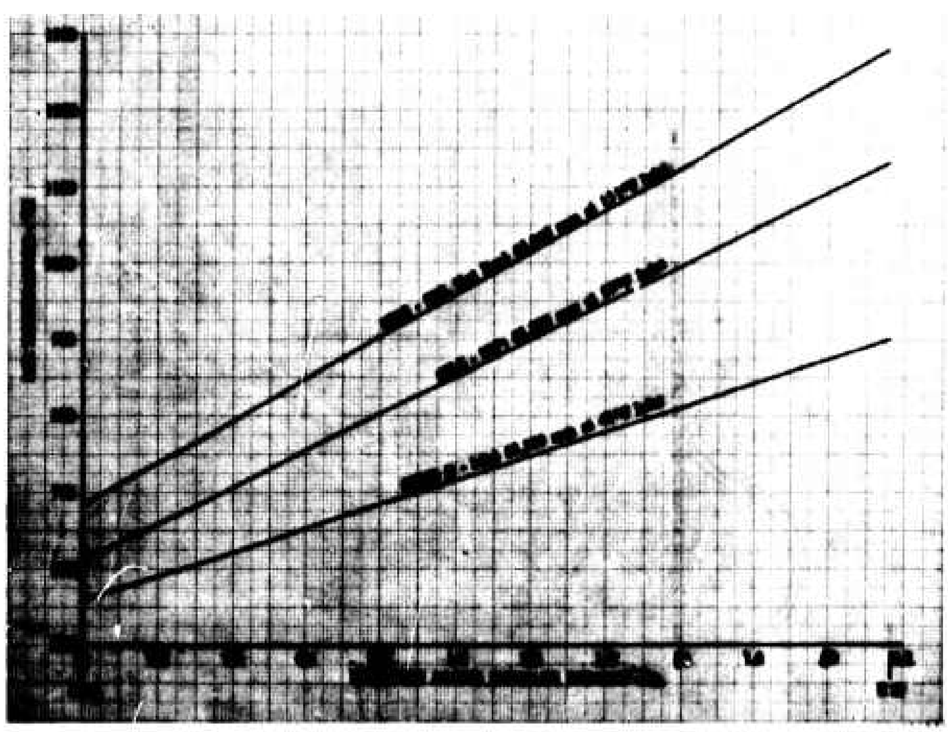


Figure 90. Estimated Compressor Shroud Air Temperature Variation

The required heat transfer surface area was found to be impracticably large due to the high heat load (890 Btu/min) and low cooling air velocities (150 to 175 ft/sec) resulting in low heat transfer film coefficients, coupled with the requirement for designing to hot-day inlet air conditions of 135°F. Therefore, the maximum air supply temperature was increased to 250°F, and the impact on heat exchanger size was determined. (See Figure 91.)

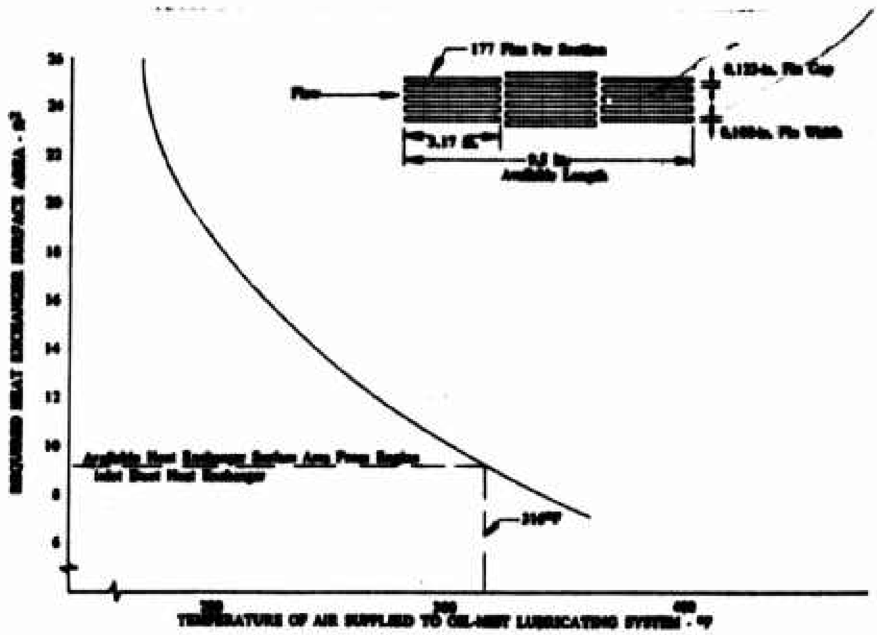


Figure 91. Finned Inlet Duct Heat Exchanger Size Required to Cool Bearing Oil-Mist Airflows

The 250°F air supply temperature is considered to be a reasonable extension of testing accomplished to date and is consistent with our maximum normal oil system supply temperature of 250°F.

The engine inlet annulus presents the largest available surface area for heat transfer purposes. With the engine diameter fixed, the only variable is duct length. Figure 92 defines the heat transfer problem; the available surface area on the engine inlet would result in a 316°F air temperature. To maintain the 250°F limit, the engine length would have to be increased by 4.8 in. Total normal engine length is approximately 36 in.

(3) System Design Based on Emergency Operation Design Criteria

Limited test data on a backup oil system designed for emergency use that utilized significantly lower air flowrates was also considered. The test point considered had demonstrated a 30-min operation of a 46-mm ball bearing operating at 38,000 rpm. The airflow used was 0.84 scfm at 200°F inlet temperature, and 4 cc/min of oil flow.

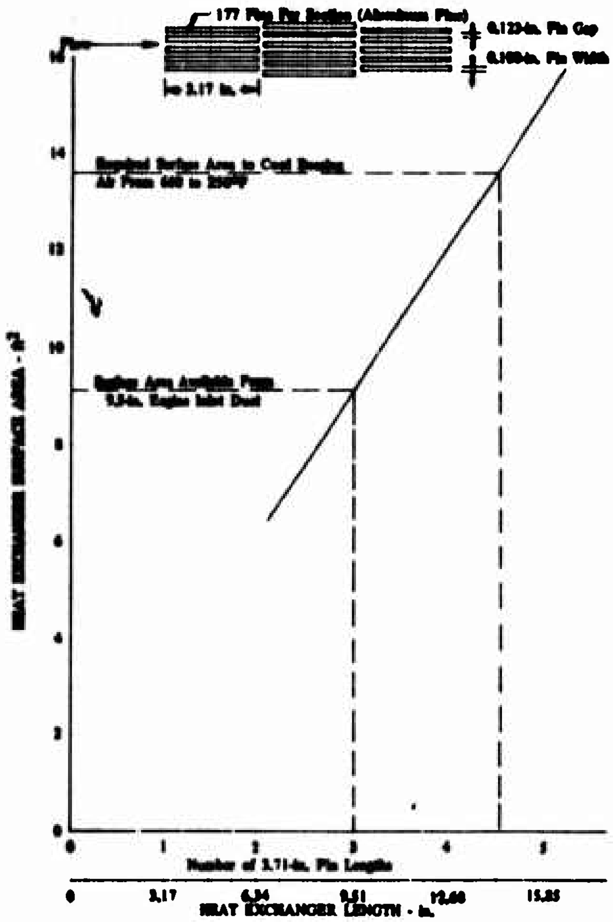


Figure 92. Finned Inlet Duct Heat Exchanger Size Required to Cool Bearing Oil-Mist Airflow From 660 to 250°F

These data were linearly extrapolated to define the air and oil flowrates for this application. The estimated required air and oil flowrates are tabulated below:

Bearing Position	Air, scfm	Oil, cc/min
Tandem Ball Pair	1.64	7.44
Tower Shaft Drive Ball	0.65	2.95
Power Turbine Ball	0.65	2.95
Tower Shaft Ball	0.2	0.9
Tower Shaft Roller	0.2	0.9
Fuel Pump Ball	0.35	1.6
Fuel Pump Roller	0.35	1.6
	4.04	18.34
Gears	1.2	6.0
Total	5.24	24.34

The required oil-mist distribution system is shown schematically in Figure 93. The lower total system air flowrates would eliminate the requirement for a system shutoff valve. The backup system would then operate continuously.

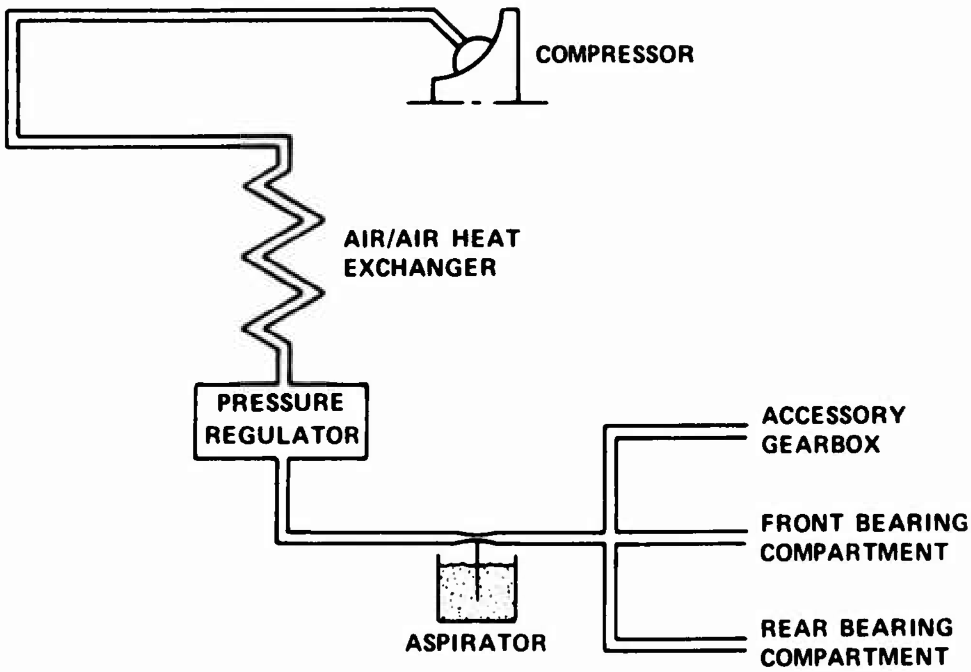


Figure 93. Emergency Operation Air/Oil-Mist Lubrication System

Due to the lower air flowrates, the use of a single aspirator and oil reservoir was considered. The required oil reservoir (720 cc) is best contained within the main oil tank. The oil-mist distribution system around the bearings and gears will be similar to that previously described.

The air/oil-mist flowrate established for the rear compartment will allow the oil mist to be transferred in a reasonable line size, thus negating the requirement for a separate rear reservoir. A required rear reservoir size of 90 cc would also be difficult to package in the rear compartment.

The required engine bleed air flowrate was established at 5.24 scfm. A bearing system supply air temperature goal of 200°F was also established. Based on the flow schematic shown in Figure 88, a total system pressure loss of 6.4 psi was estimated. This established an impeller shroud bleed location of 1.1 in. to provide the required pressure at the 5000-ft idle condition. A supply air temperature of 530°F would then be provided at sea-level, maximum-power condition.

An air/air-heat exchanger was then sized to provide the required cooling of the bearing supply air to 200°F. The IPS bypass duct was selected as the best location for the heat exchanger. A simple heat exchanger that would use the outer surface of the bypass duct would provide 1.13 ft^2 of heat exchange area without the use of fins. For a 200°F supply temperature, an area of 1.05 ft^2 would be required. The selected heat exchanger location was, therefore, determined to be acceptable. The impact of air supply temperature on heat exchanger size, for this heat exchanger location, is shown in Figure 94.

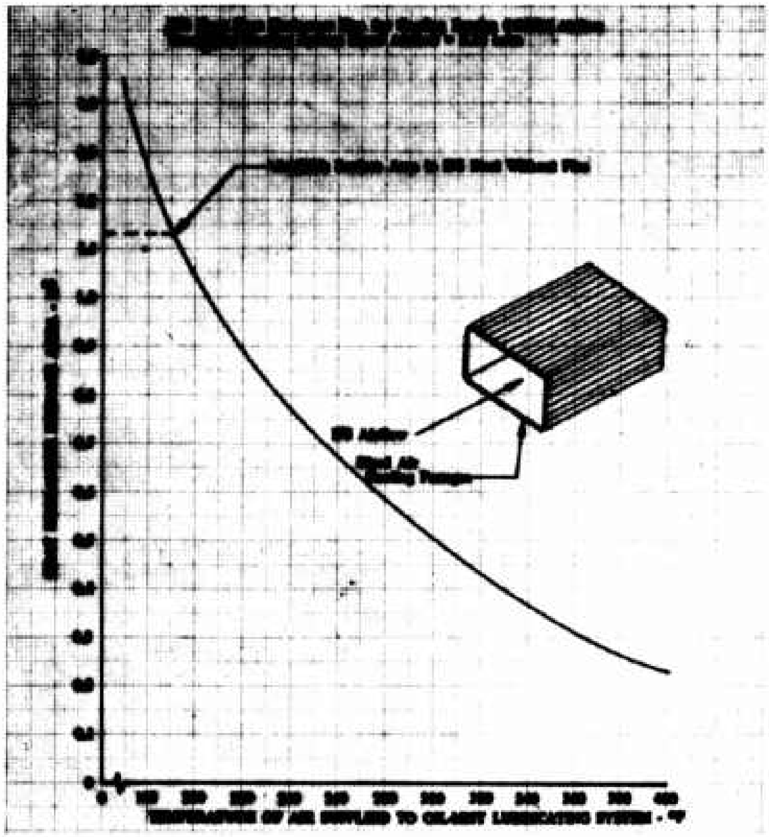


Figure 94. Air Temperature Effect on Heat Exchanger Size

A system based on the emergency operation design criteria was determined to be feasible for the engine configuration studied. However, before a final system design is established, further experimental verification of the design criteria for air and oil flowrates will be required on both the component and system levels.

(4) Conclusions and Recommendations

Industry testing of an air/oil-mist lubrication system indicates that a 6-min or 30-min system capability is possible for the engine size and speed range studied. For a system using the design criteria established for full-time, steady-state operation, the requirement for a relatively low oil-mist supply temperature, based on testing accomplished to date, imposes impractical heat exchanger requirements on the engine design.

A system using the design criteria established for emergency operation provides acceptable heat exchanger and oil reservoir designs. However, test data verifying this system approach are preliminary in nature and require further confirmation.

It is recommended that further test programs be directed toward higher air/oil-mist supply temperatures. A helicopter, with even a limited operational envelope, must be designed for standard hot-day sea level, Mn = 0 inlet temperatures of 135°F. Assuming a nominal supply pressure requirement for the oil-mist system, an engine bleed air temperature of several hundred degrees higher can be anticipated.

The oil-mist distribution system valves, pressure regulator, lines, aspirator, and oil-mist nozzle pressure drops define the system supply pressure that must be provided by compressor bleed. The location of the pressure tap is determined by the idle power conditions at the selected maximum operational altitude. The maximum air supply temperature then occurs at the sea-level, maximum-power conditions. The maximum supply temperature is, therefore, directly related to the system pressure drop requirements. An optimized system design should incorporate a minimum pressure drop oil-mist feed system.

Minimizing the required air flowrate is also a desirable goal, as that decreases the engine performance penalty and reduces the air/air heat exchanger size for a given air supply temperature. However, for emergency system operation, it is more desirable to reduce or eliminate the heat exchanger requirements at the expense of airflow. For part-time emergency operation, the airflow penalty can be tolerated, while a large heat exchanger imposes full-time cost, weight, and volume penalties.

SECTION VIII

REAR-DRIVE ENGINE WITH INTEGRATED C&A COMPONENTS

A. FINAL ENGINE DESIGN LAYOUT

Design layouts of each C&A component, component interconnects and interfaces, and sensor interfaces have been integrated in the final engine layout drawing. The final tower shaft drive/air turbine starter configuration is shown in Figure 95, which also defines the C&A system elements to show the relationship with the other engine systems.

The air turbine starter is located in the engine nose cone and is mechanically connected to the gas generator rotor through an overrunning clutch. The starter air control valve and pilot solenoid are also located in the nose cone. The starter components provide limited front vulnerability protection to the fuel and oil systems.

The oil sump is located in the front bearing compartment and also acts as the oil system heat exchanger. A vane-type positive-displacement oil pump is in the same location and is driven at a maximum speed of 15,000 rpm by a tower shaft.

The engine alternator, which has been made integral with the tower shaft, provides the required engine and electrical power for starting and control, and operates at a maximum speed of 30,000 rpm. The alternator operates in an oil-mist environment.

The main fuel pump is located in the fluid controller and is driven at a maximum speed of 65,000 rpm by the tower shaft. The pump is a two-stage centrifugal unit, with a jet pump inducer, and is designed to operate at inlet conditions of 1 psi above TVP and a V/L of 1.0.

The fluid controller is located, adjacent to the compressor case, on top of the engine. The fluid controller meters the required engine fuel flow and actuates the IGV's in response to stepper motor input commands from the electronic control. The only external fuel lines are for the fuel supply and the discharge line to the fuel manifold. A mechanical power turbine overspeed sensor interfaces directly with the fluid controller.

The full-authority electronic control is located adjacent to the fluid controller for vulnerability protection and to reduce the interconnecting cable length. The electronic computer controls all engine starting, transient, and steady-state functions (except for the redundant power turbine overspeed function) in response to electrical interface command signals from the airframe. Engine sensors are provided for compressor inlet temperature, burner pressure, rotor speeds, power turbine torque, and gas generator turbine blade temperature and interface with the electronic control. Cooling for the electronic control is provided by heat sink to the engine inlet air. Cooling for the torque sensor and cable is provided by compressor bleed air, which has been cooled by an air/air heat exchanger.

Airframe interfaces for starter air, fuel supply, and the electrical airframe input commands are on the top of the engine.

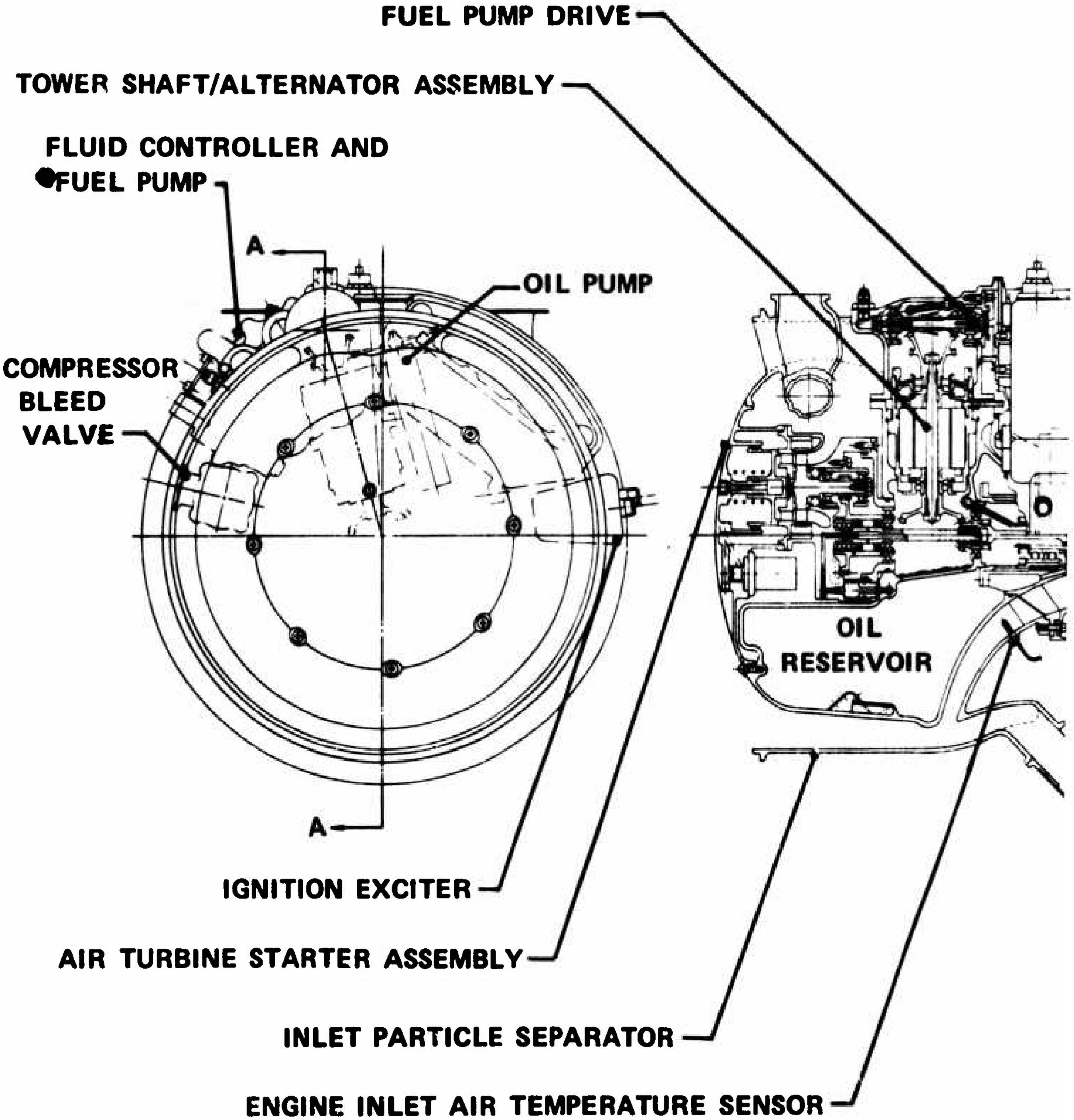
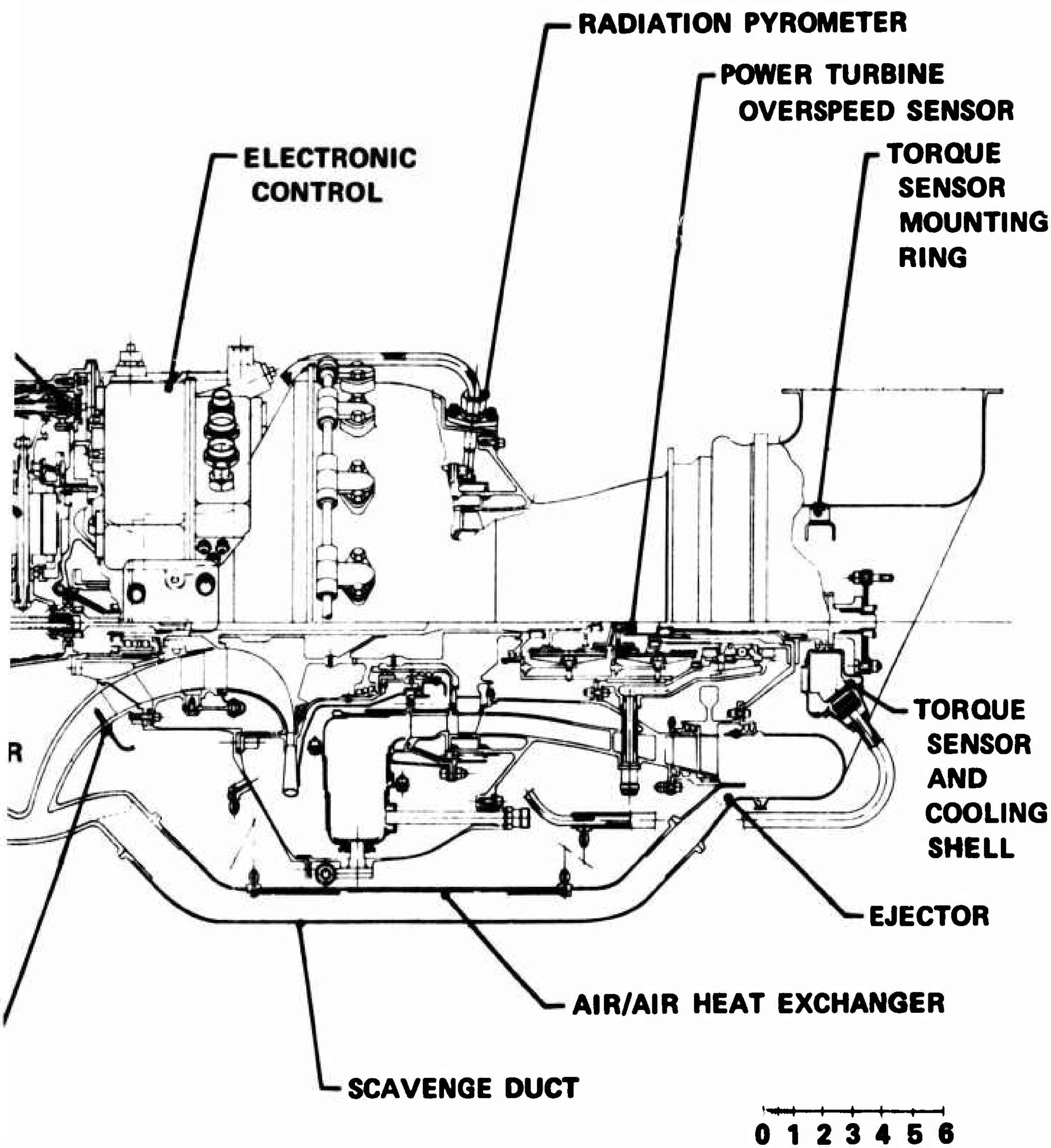


Figure 95. Final Tower Shaft Drive/Air Turbine Starter Engine Cross Section


RADIATION PYROMETER
POWER TURBINE OVERSPEED SENSOR
ELECTRONIC CONTROL
TORQUE SENSOR MOUNTING RING
R
TORQUE SENSOR AND COOLING SHELL
EJECTOR
AIR/AIR HEAT EXCHANGER
SCAVENGE DUCT
0 1 2 3 4 5 6
SCALE - in.

SECTION A-A

2

The engine layout (Figure 95) shows that the control and accessory integration techniques have resulted in a minimal impact on the engine front and side projected areas, and that a system designed with consideration for vulnerability resistance has added benefits in weight, volume, and simplicity.

B. C&A COMPONENT CRITICAL ITEMS

During the detailed design phase, several C&A component areas were identified as requiring experimental effort to verify the selected design approach for a 1977 engine development time frame. These areas are outlined below and can also be used for reference in definition of future C&A experimental development efforts.

- Fuel Pump Inducer - Based on pump development work accomplished to date, a three-stage pumping system has been proposed to meet the specified system requirements. The boost elements comprise a jet pump plus a centrifugal inducer. The main stage is a 65,000-rpm centrifugal pump.

  The pump specification for this program requires operation at inlet conditions of 1.0 V/L and 1-psi NPSP. Present technology in fuel systems is for operation at 0.45 V/L and 5-psi NPSP. While tests have demonstrated the ability of fuel boost pumps to operate at conditions higher than 0.45 V/L, sustained operation at 1.0 V/L, without a low-speed inducer, represents a significant advance of the state of the art. The high speed (65,000 rpm) and low flow (500 lb/hr) required for this pump indicate that an experimental program is required to demonstrate inducers to meet these pump inlet requirements.

- Starter Overrunning Clutch - The proposed starter design achieves minimum weight and maximum performance efficiency by locating the overrunning clutch in line with the engine rotor. This results in a clutch overrunning speed of 65,000 rpm. For this application, engine-supplied oil is available for clutch lubrication.

  High-speed sprag clutch problems generally fall into two categories. In the driving mode, fatigue and sprag rollover are the typical failure conditions. Overrunning at high speeds can lead to excessive clutch and race wear, which ultimately result in rollover or inability to drive.

  Since the proposed clutch application involves low driving torques and speeds considerably in excess of previously demonstrated capability, an experimental program was recommended to verify the 65,000-rpm overrunning capability.

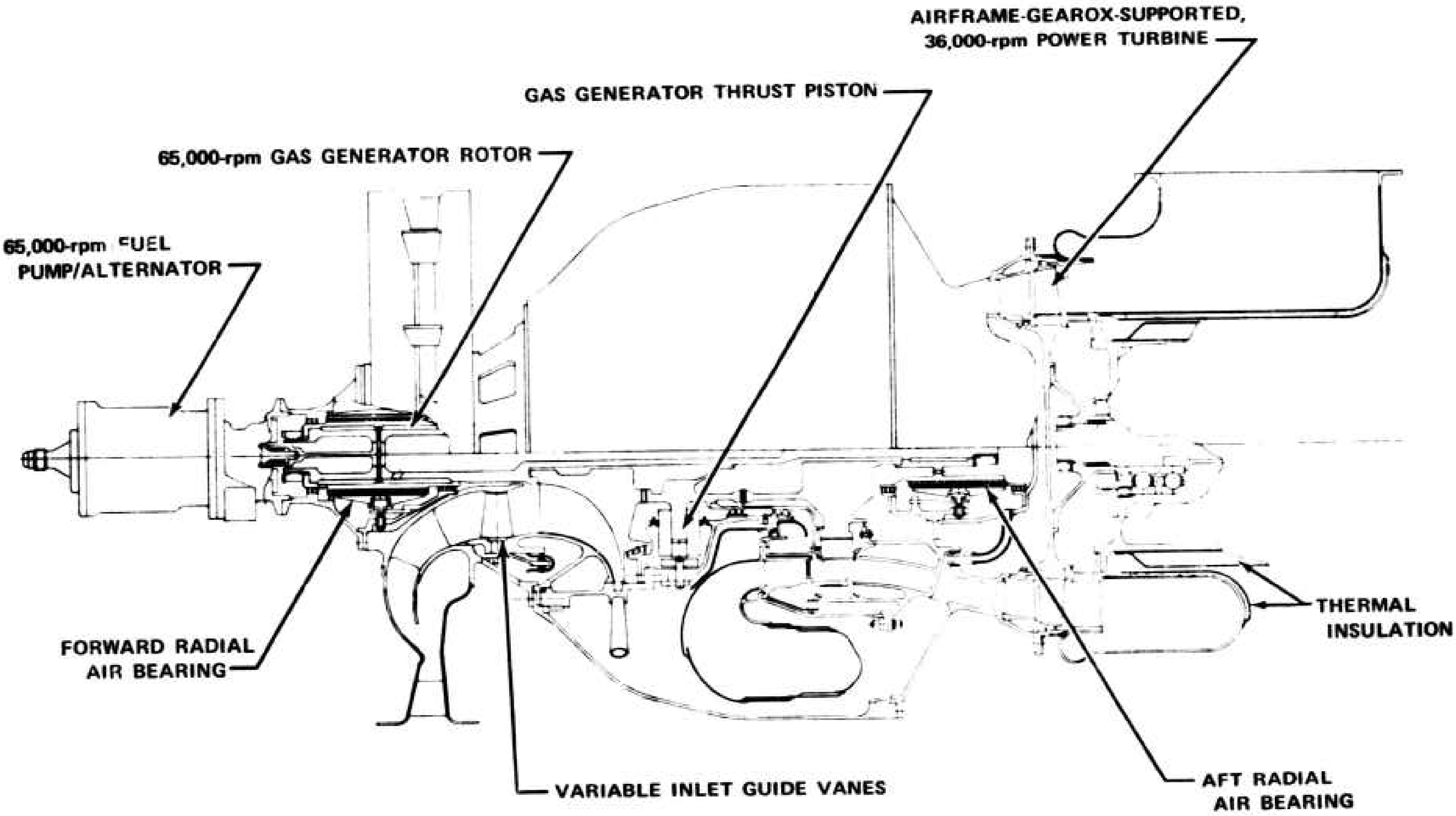


Figure 96. Gearboxless Engine (1982 Time Frame)

- Oil Pump Performance and Endurance - The limit on operating speed for a single-stage, positive-displacement oil pump is determined by the inlet cavitation effects. For the engine oil system, a vented oil tank is proposed that provides essentially ambient inlet conditions to the oil pump pressure stage. Since the engine is being designed to operate up to a 20,000-ft altitude, an inlet pressure as low as 13.75 in. absolute of mercury is anticipated. Experimental verification of the capability of an oil pump to operate at the maximum speed (15,000 rpm) and maximum altitude inlet conditions (20,000 ft) without sustaining cavitation damage was recommended.

- Radiation Pyrometer - As engines get smaller and operate at higher temperatures, accessories such as the radiation pyrometer have greater demands imposed on them. The radiation pyrometer, which is emerging as an important engine accessory, is one item, in particular, that has to operate in a hostile environment, yet provide accurate data vital to the performance of the engine.

  The temperature profile defined for the pyrometer environment on this program is as follows:

  1. CDP (for cooling and purging) = 750°F maximum
  2. Pyrometer location cavity = 800°F maximum
  3. Lower seal metal temperature = 1400°F maximum
  4. Turbine casing temperature = 1900°F maximum

  However, the limiting temperatures of the individual elements of the pyrometer, for reliable operation, are as follows:

  1. Sapphire lens and mounting flange assembly = 1400°F
  2. Solid fiber-optics light pipe = 1000°F
  3. Flexible fiber-optics cable = 800°F

  The pyrometer proposed for this application includes a sighting tube with a lens located just inside the 800°F cavity, thereby negating the need for the solid light pipe and providing a thermal environment well within the rating of the lens and flexible cable.

  A further feature of the design is the absence of a CDP purging port. Instead, the pyrometer includes an inherent purging configuration, formed by small holes, located in the pyrometer outer tube upstream of the lens. It is intended that air be drawn in through these holes from the 800°F cavity to provide purging airflow to the lens.

  This concept will reduce the pyrometer cost, provide minimal size, and ensure that the hot CDP purging air is inhibited from interfering with the flexible fiber-optics cable.

  Experimental verification of the above-mentioned purging concept was recommended.

- Electronic Control Cooling - The proposed mounting surface for the electronic computer provides heat transfer to the compressor inlet air.

  A review of installation requirements for this configuration indicates departures from previously developed cooling concepts in that forced air-convective cooling is proposed instead of using fuel as a heat sink. This concept, based on locating the control with a heat exchanger interface with the particle separator, should provide an adequate means of cooling. This approach is advantageous over its fuel counterpart from a viewpoint of cost and reliability. The anticipated primary source of heat for the computer is via radiation from the compressor; however, the magnitude is presently unknown.

  An improved conductive heat transfer concept (from components to heat sink) has also evolved for this control. This concept is based on heat transfer from the individual "chips" to the component leads and then via a special copper layer in the PCB to form a thermal conductor to the flexible interconnecting cable, which finally terminates at the heat sink. Initial analysis indicates that this concept has the potential of providing a very low thermal impedance path from all components to the heat sink. Experimental verification of this approach was also recommended.

- Electronic Control Vibration Isolation - Previous CECO efforts in designing the printed circuit boards (PCB's) for the engine vibration environment used internal damping and isolation mechanisms for the PCB's that realized the goal of minimizing the component and PCB vibration level to 30 g's with an input excitation of 5 g's (20 to 460 Hz) and 20 g's (460 to 2000 Hz). The electronic vibration environment for this application is estimated at 50 g's.

  The elements not protected by the damping and isolation mechanisms (radiation pyrometer detector and first-stage signal processing, together with the pressure transducer) are rated for 30 g's, but do not experience g levels significantly above the input excitation, as they are hard-mounted to the computer housing.

  The recommended alternative approach for this application would use external vibration isolation instead of the internal isolation/damping concept. This approach appears advantageous, in that the radiation pyrometer and its signal detector would be protected from the 50-g input. The alternative to this would be the design and qualification of components rated for 50 g's, which would be less reliable.

  It was recommended that vibration isolation techniques be experimentally verified.

- Power Turbine Overspeed Sensor - Free turbine engines require a fast-response overspeed governor for protection to prevent runaway speed. Also, in the cases where the engine control system is electronic, it is advantageous to have a power turbine overspeed governor that is redundant and operates independent of the electronic control. Conventional mechanical speed sensors operate in the 5,000-rpm speed range at 250°F, whereas, in this system, rated-power turbine speed is 36,000 rpm and ambient temperature is 800°F.

The speed sensor is formed by a set of closely spaced cantilevers that are arranged in a circle around the power turbine shaft. As the power turbine shaft speed increases, the centrifugal force causes the cantilevers to move radially outward. The cantilever motion is used to vary an orifice and thereby generate a pneumatic pressure signal. An increase in power turbine speed operates to reduce the metering head across the fuel metering valve. Hence, a turbine overspeed causes a proportional reduction in fuel flow.

Critical items in the operation of this governor are the speed sensor and the dynamic response of the governor. A speed sensor of this configuration has previously been tested to 20,000 rpm. For this system, the sensor has to be designed to operate at 36,000 rpm in surrounding temperatures up to approximately 800°F. The required response from overspeed to reduction in fuel flow is on the order of 50 ms.

An experimental program was recommended to demonstrate the performance and durability (in an engine-simulated environment) of the speed sensor and to confirm the dynamic response capability of the overspeed governor.

- Starter Overrunning Clutch and Decoupler Endurance - The selection of a sprag-type overrunning clutch necessitates incorporation of a decoupler mechanism to prevent overspeed of the starter turbine wheel in the event of a clutch or bearing failure. At the design speed of 65,000 rpm, it is important that the decoupler provide complete disengagement of the starter from the engine. Existing decouplers use some type of supporting bushing or bearing, which continues to run after decoupling occurs. It is felt that the new decoupler incorporated in the design concept approach must be experimentally verified for the proposed high-speed application.

- Starter Control Valve - A starter control valve, which is integral with the starter and incorporates a controlled opening poppet valve, has been proposed for the engine accessory program. Current air turbine starters use separate starter control valves that open between 10 to 40 psi/sec to limit the impact torque. The opening rate depends on the maximum allowable impact torque, which varies with different applications. The majority of starter control valves incorporate butterfly modulating elements, where the valve opening rate is controlled by a simple orifice circuit in the actuator head supply line. By contrast, a poppet valve tends to snap open, and the rating device required to control its opening rate is more complex, generally consisting of an incompressible fluid damper. The proposed valve will be normally spring loaded and pressure-assisted closed and will incorporate a device to control the opening rate. An experimental program was recommended to evaluate the proposed valve configuration with an incompressible fluid damper.

C. ENGINE DESIGN CRITICAL ITEMS

During the course of the program, several engine design areas that interface with the C&A systems were identified as requiring experimental effort to verify or improve the recommended baseline engine configuration. These areas are outlined below and can be used for reference in definition of future experimental development efforts.

- Air Bearings - The primary engine oil-lubricated bearing system is inherently vulnerable because of the large oil tank, heat exchanger, and fuel system. Further development effort and demonstration of radial and axial air bearings is in order to verify a near-term applicability.

- Oil-Mist Backup Lubrication System - The study of an emergency oil-mist lubrication system for this application indicates that a system design, based on the limited test data available, is feasible for a 30-min system capability. Further experimental effort to verify the required air and oil flowrates and directed toward using higher air supply temperature is in order to confirm the applicability of the backup system approach.

  Additionally, testing of the <u>total</u> air/oil-mist system is recommended to verify the system design.

- Advanced Electric Starters - For certain engine applications, an electrical starter system may be considered. Development effort for an advanced, high-speed starter that uses engine-supplied oil for lubrication and cooling is in order. The program goals would be directed toward minimizing starter weight and volume.

- Air Assist During Engine Starting - Lower engine starting speeds, with resulting lower starter horsepower requirements, can be achieved by improving fuel atomization at start. A small quantity of supplementary air for the air-assist nozzles during the start cycle will provide the desired improvement. A design study and experimental effort is recommended to determine the best means of supplying the additional air and to identify the overall impact on the engine and starter system.

- Engine/Airframe Interfaces - Engine/airframe interfaces for future applications are a fertile area for further study. Advanced engines will be much smaller and lighter in weight than current powerplants. This will allow a greater degree of freedom in establishing the optimum installation since the engine weight will have a small effect on the overall airframe CG. Vertical engine installations, nonparallel gas generators and power turbines, or two gas generators supplying one power turbine are potential configurations which can be considered.

  The power turbine interface, drive location for airframe accessories and the technique for supplying air for ECS and IRS are areas requiring engine/airframe trade-off studies to confirm the best overall system approach.

D. LONG-RANGE ENGINE DESIGN CRITICAL ITEMS

From an ultimate C&A system reliability and vulnerability standpoint, a gearbox-less engine is the ideal configuration and is shown in Figure 96. This arrangement would require experimental development in several areas but could be accomplished without excessive development risk. The areas requiring verification are outlined below:

- Development and demonstration of radial and axial air bearings
- Development and demonstration of engine integral start techniques
- Coordination of a power turbine interface that is integral with the airframe gearbox.

SECTION IX

TESTING OF C&A CRITICAL ITEM COMPONENTS

The C&A critical item components identified in Section VIII were reviewed by the Army technical program manager, and the following five programs were selected for experimental evaluation: (1) fuel pump inducer (inlet suction tests), (2) high-speed oil pump (cavitation tests), (3) electronic cooling techniques (performance tests), (4) power turbine overspeed sensor (performance tests), and (5) starter overrunning clutch (endurance tests).

The selected C&A critical item components were functionally and endurance tested, in accordance with approved test plans, to fully evaluate the high-risk aspects of the component or subsystem. Descriptions and results of the critical item testing are presented and the results evaluated relative to meeting the performance and endurance goals.

A. VANE ELEMENT OIL PUMP - SUNDSTRAND AVIATION

1. Background

The intent of the test program was to establish state-of-the-art parameters for an ultimate lubrication and scavenge system capability. State-of-the-art pump design has been advancing to the point where unboosted inlet pumps can operate at altitudes up to 20,000 ft without cavitation at speeds to 10,000 rpm. The speed limitation in an unboosted inlet pump is due to cavitation caused by the reduced inlet pressure existing at 20,000 ft altitude. Tip speeds should not exceed 33 ft/sec at this altitude, using MIL-L-23699 oil at 250°F.

The specification requirements for the pump stipulated a minimum supply flow of 43 lb/min at 70 psid differential pressure, and a minimum scavenge of 44 lb/min at 20 psid differential pressure. Both of these requirements must be met at 20,000 ft altitude or 13.75 in. HgA. To achieve the design goals without cavitation, the vane pump had to be designed so that the vane tip velocity does not exceed the velocity head conversion of the available 13.75 in. HgA. This maximum velocity is obtained from Bernoulli's equation:

$$h = \frac{V^2}{2g}$$

where

h = oil head in ft
V = fluid velocity or vane tip velocity in ft/sec
g = acceleration due to gravity 32.2 ft/sec^2

The conversion of 13.75 in. HgA to feet of oil, h, using a weight of 57.5 lb/ft^3 for 250°F MIL-L-23699 oil, is

$$h = \frac{13.75 \text{ in. HgA} \times 144 \text{ in}^2/\text{ft}^2}{2.04 \text{ in. HgA/psi} \times 57.5 \text{ lb/ft}^3} = 16.9 \text{ ft of oil available}$$

Thus, the maximum allowable tip velocity, V, is

$$V = \sqrt{2\,gh} = \sqrt{(64.4)\,(16.9)}$$

$$V_{max} = 33 \text{ ft/sec}$$

The test program obtained actual test performance enabling an engineering evaluation of Sundstrand's PBD-1 through-vane pump configuration which would be utilized as a main engine lubrication system pump to operate at 15,000 rpm and 20,000 ft altitude ambient pressure without cavitation.

A new computerized cam contour was developed by Sundstrand to minimize vane acceleration loads and incorporated into a family of single-lobe, through-vane type pumps. This new technology is incorporated into pump Model 015553-200 lubrication and scavenge pump, which is of a cartridge design and similar to the one included in the engine design study. Running this existing piece of hardware at comparable tip speeds demonstrated the flow versus altitude capability on nine different bore sizes in this family of pumps. An example of this comparison was to take the Model 015553-200, which utilizes a Sundstrand PBD-4 bore, and to compare it to the Sundstrand PBD-1 bore, which was included in the engine design study. Using the chart shown in Figure 97, which cross references the vane type radius in inches with available pressure in inches HgA, following the matrix, the maximum speed to run the pump, so that cavitation will not occur, can be determined. Example: PBD-4 equals 0.450 vane tip radius and the PBD-1 equals a vane tip radius of 0.232; thus one can see quite readily that 7,733 rpm on the PBD-4 is equal to 15,000 rpm on the PBD-1. Running the PBD-4 pump at a speed range from 6,000 rpm to 10,000 rpm demonstrated flow versus altitude performance points at high tip speeds (30 to 40 ft/sec).

The PBD-1 bore has a tip speed of 30.3 ft/sec at 15,000 rpm and 40.4 ft/sec at 20,000 rpm; therefore, 15,000 rpm would be the maximum speed the pump should be run with an inlet at 20,000 ft altitude, when pumping MIL-L-23699 oil at 250°F. An overspeed test was run to demonstrate the pump durability (no cavitation erosion while operating at 20,000 ft altitude inlet conditions).

2. Test Program

The test arrangements are shown in Figures 98 and 99.

General calibrations of the PBD-4 pump were run at speeds between 4,000 rpm and 10,000 rpm; fluid inlet temperatures of 75°F, 175°F, and 250°F; and inlet pressures between ambient and 24 in. Hg vacuum.

The discharge pressure of the lubricating pump element was maintained at 600 psi gage, and the discharge pressure of the scavenge pump element was maintained at 30 psi gage.

These data are shown in Figures 100 through 103 and did not meet the theoretical maximum performance based on tip velocity, indicated on Figure 101. The pump will have a minimum volumetric efficiency of 85% at all temperatures between 75°F and 250°F using MIL-L-23699 at inlet pressures equivalent to 20,000 ft altitude.

The calibration data show the pumping elements were not able to meet the 20,000-ft altitude requirement. At the 250°F oil temperature condition, the maximum altitude capability on the larger element, running at design speed, is approximately 9,000 ft.

REFERENCE		TIP RADIUS, in.
PBD	–1	0.232
PBD	–2	0.343
PBD	–3	0.418
PBD	–4	0.450 (TEST SIZE)
PBD	–5	0.506
PBD	–6	0.625
PBD	–7	0.682
PBD	–8	0.722
PBD	–9	0.785
PBD	–10	0.892
PBD	–11	1.000

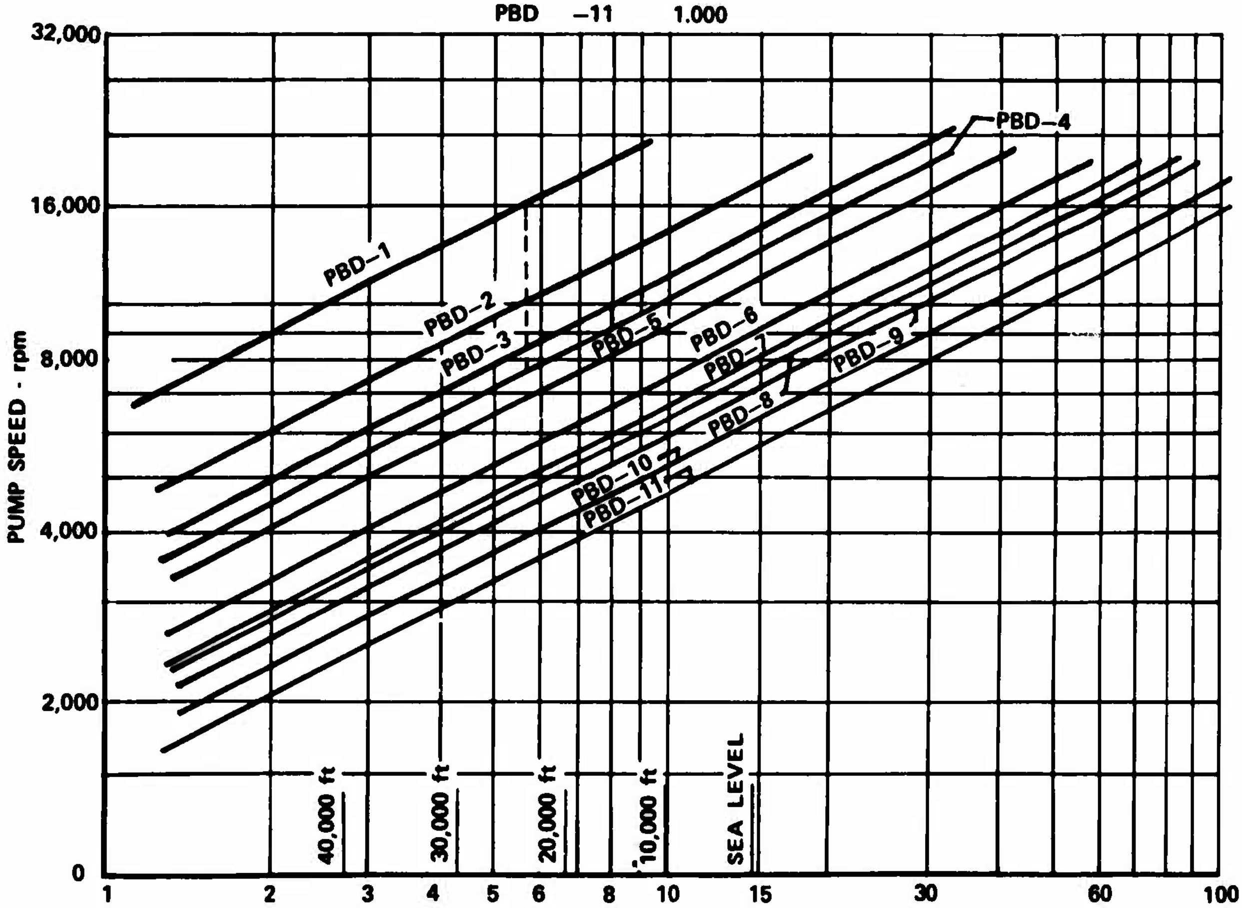
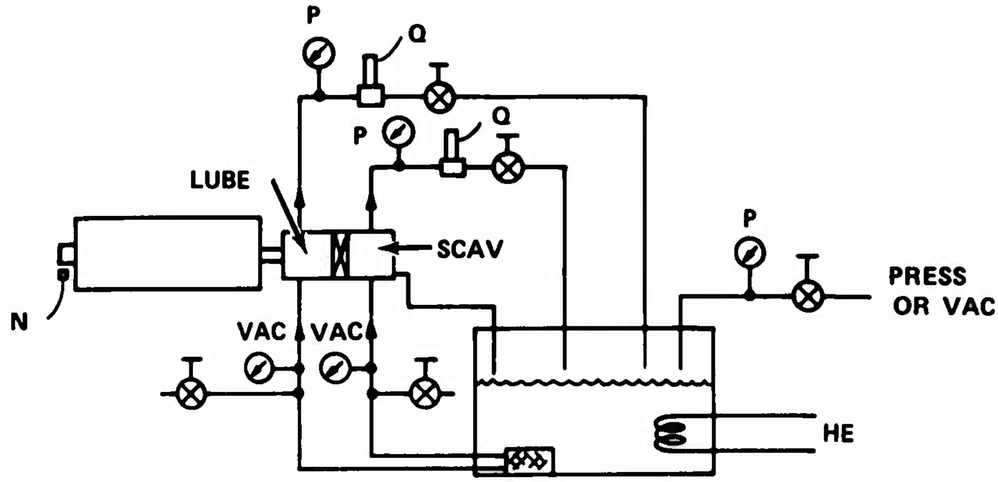


Figure 97. Maximum Pump Speed vs Minimum Inlet Pressure for Various Cross Vane Pump Designs



TEST CONDITIONS:

FLUID:	MIL-L-23699
TEMPERATURE, FLUID:	ROOM TEMP TO 250°F
SHAFT SPEED:	4000–10,000 rpm
DRIVE:	STRAIGHT SPLINE DRIVE
PUMP FIXTURE:	TO FIT 015553–200

Figure 98. Oil Pump - Test Schematic

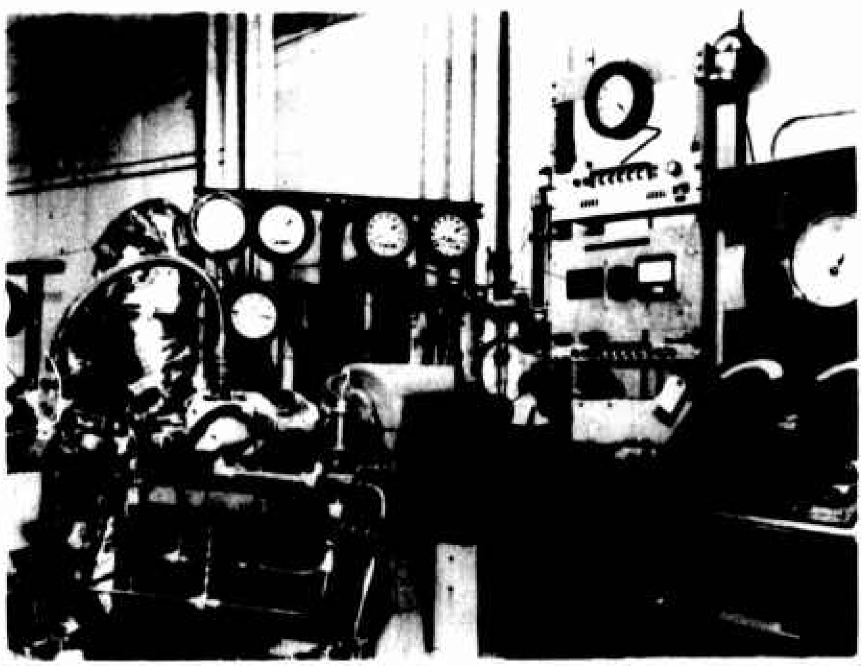


Figure 99. Oil Pump Test Arrangement

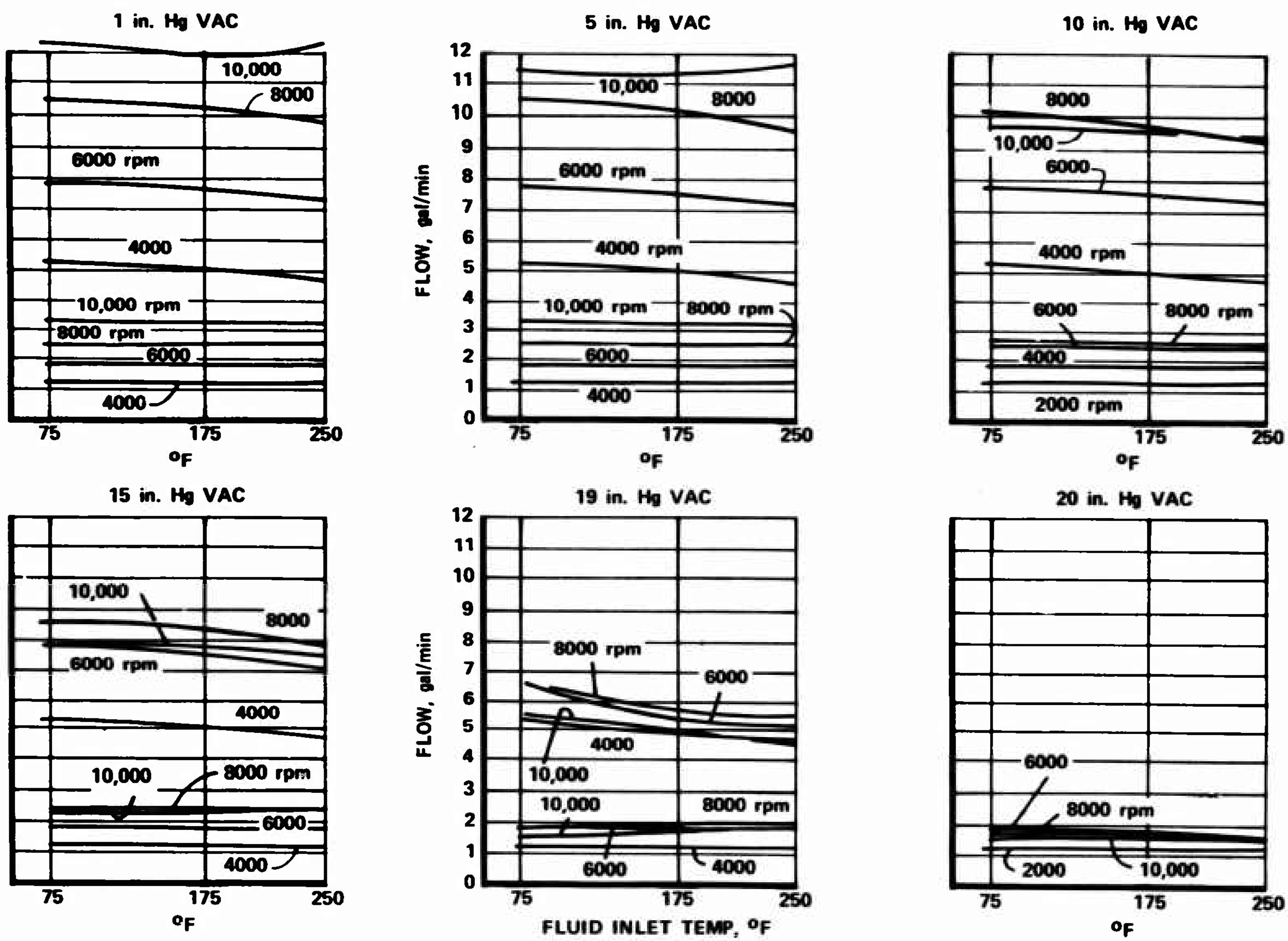


Figure 100. Pump Performance Variation With Altitude

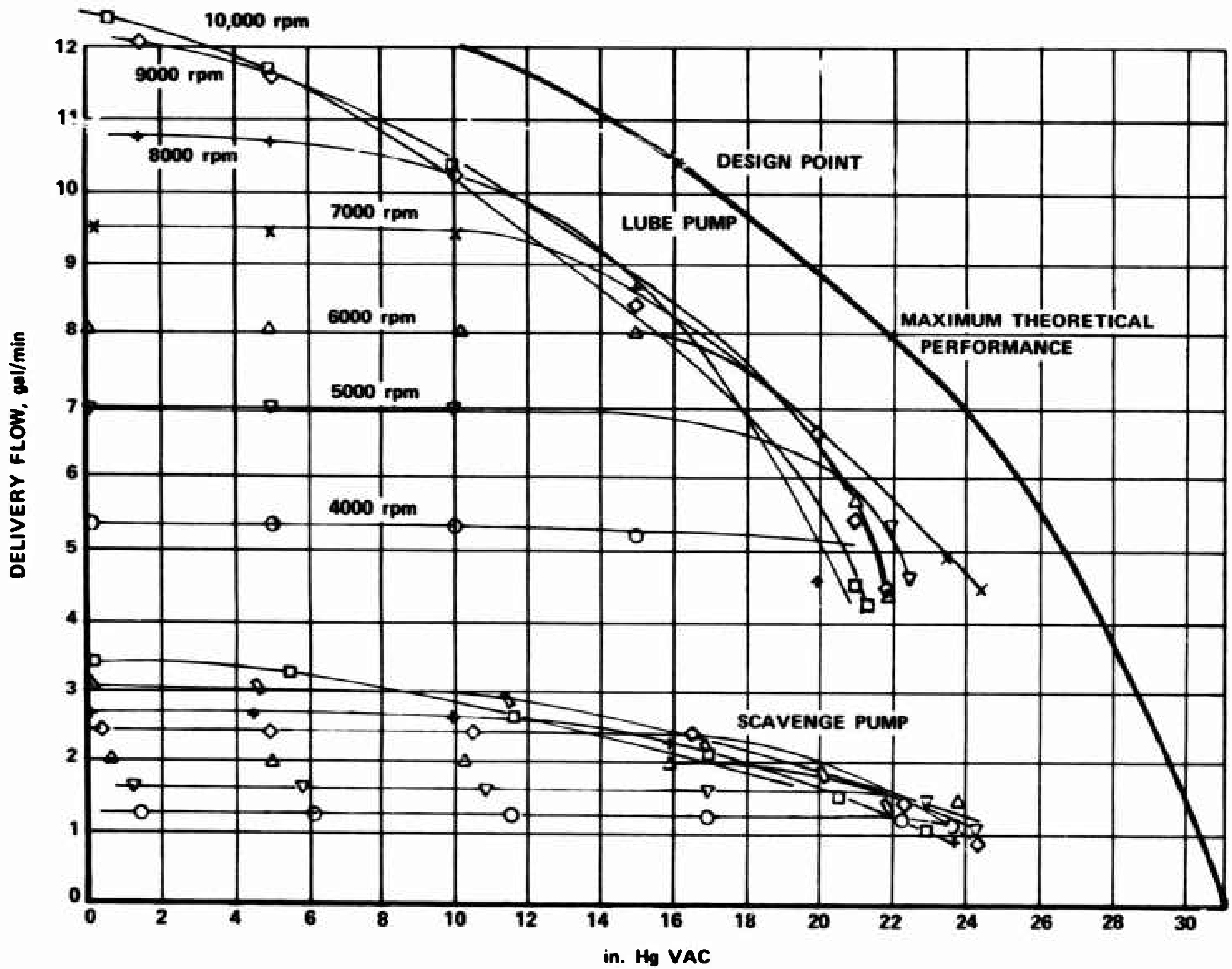


Figure 101. Oil Pump Cavitation Tests (Fluid Temperature 75–80°F)

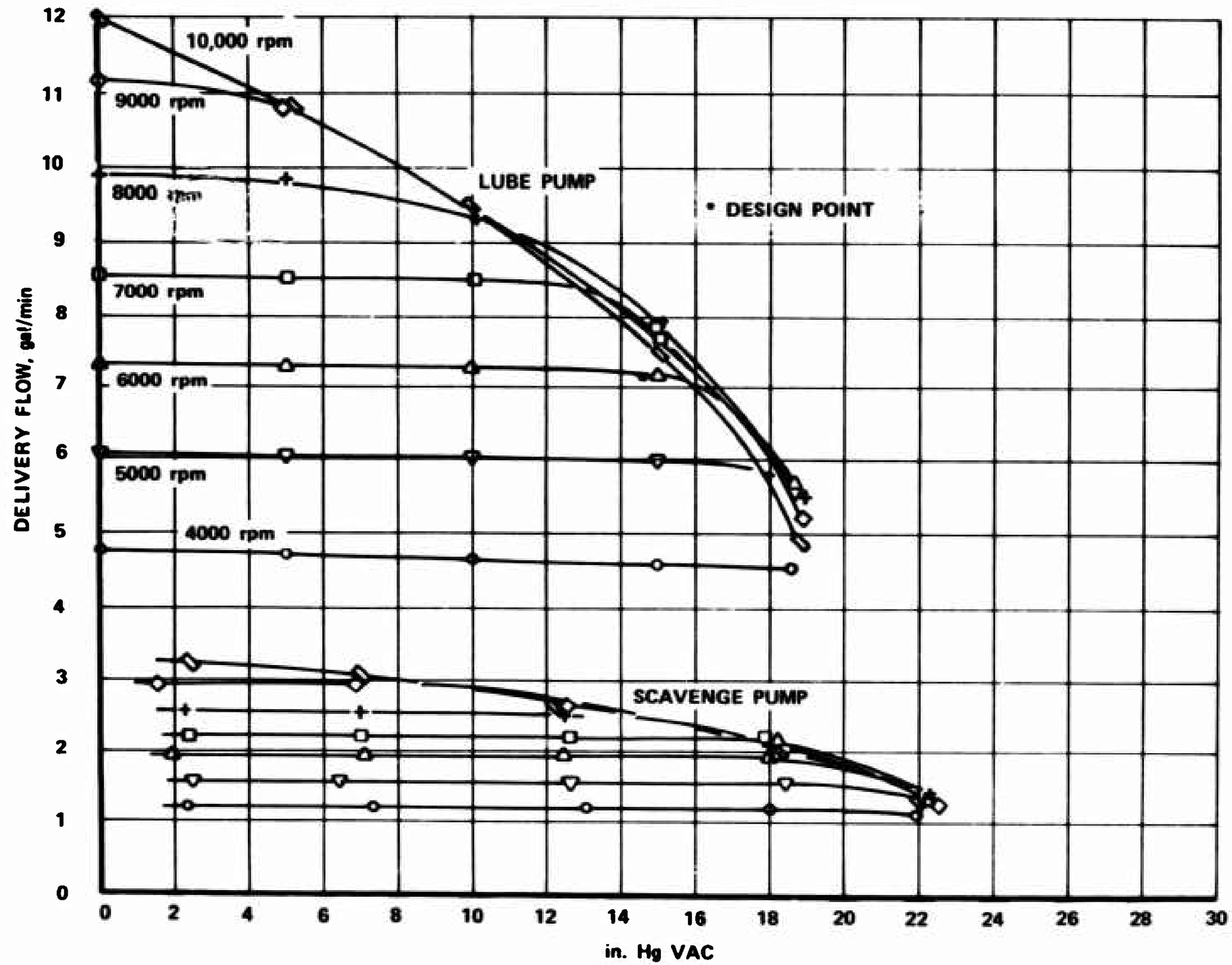


Figure 102. Oil Pump Cavitation Tests (Fluid Temperature 175°F)

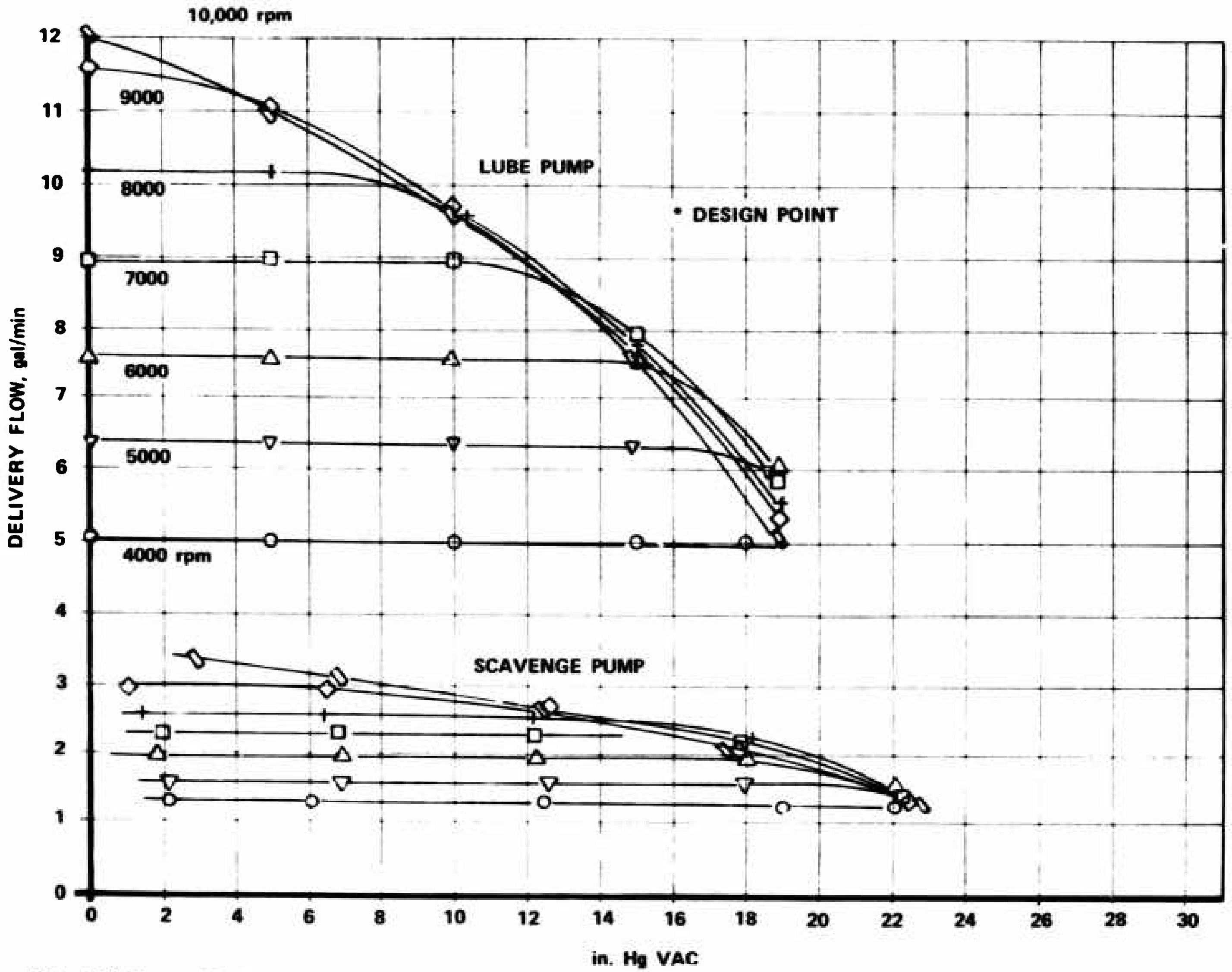


Figure 103. Oil Pump Cavitation Tests (Fluid Temperature 250°F)

To meet the high-altitude ambient inlet condition, the inlet port of the lubrication element was redesigned.

Another test series was run to establish the effect of increasing the inlet opening of the lubrication element, as shown in Figure 104, from its original 0.562 in. opening to approximately 0.900 in. The performance improvements resulting from the increased opening are shown in Figures 105 through 108, and demonstrate the ability of the pump to run cavitation free at various speeds and altitude inlet conditions.

The pump was run for a minimum of 50 hr at a speed of 8,000 rpm, a fluid inlet temperature of 250°F, inlet pressures of 16.25 through 16.75 in. Hg vacuum, and discharge pressures of 60 psig and 30 psig for the lubricating and scavenge elements, respectively. The pump performance was unchanged throughout the 50-hr endurance test, and there was no measurable or visual damage or distress to the pump as a result of the test. Photographs of the test hardware are shown in Figures 109 and 110.

With the inlet pressure at room ambient and the discharge pressures at normal conditions, ambient air was bled into the inlet lines of both elements. The pump was allowed to run 1 hour at the design speed of 7733 rpm. The housing temperature increased from room ambient to 120°F within 35 min and continued to 124°F at the end of the hour run.

No damage or distress resulted from the air bleed test. The pump was recalibrated after the dry run and showed no degradation in performance.

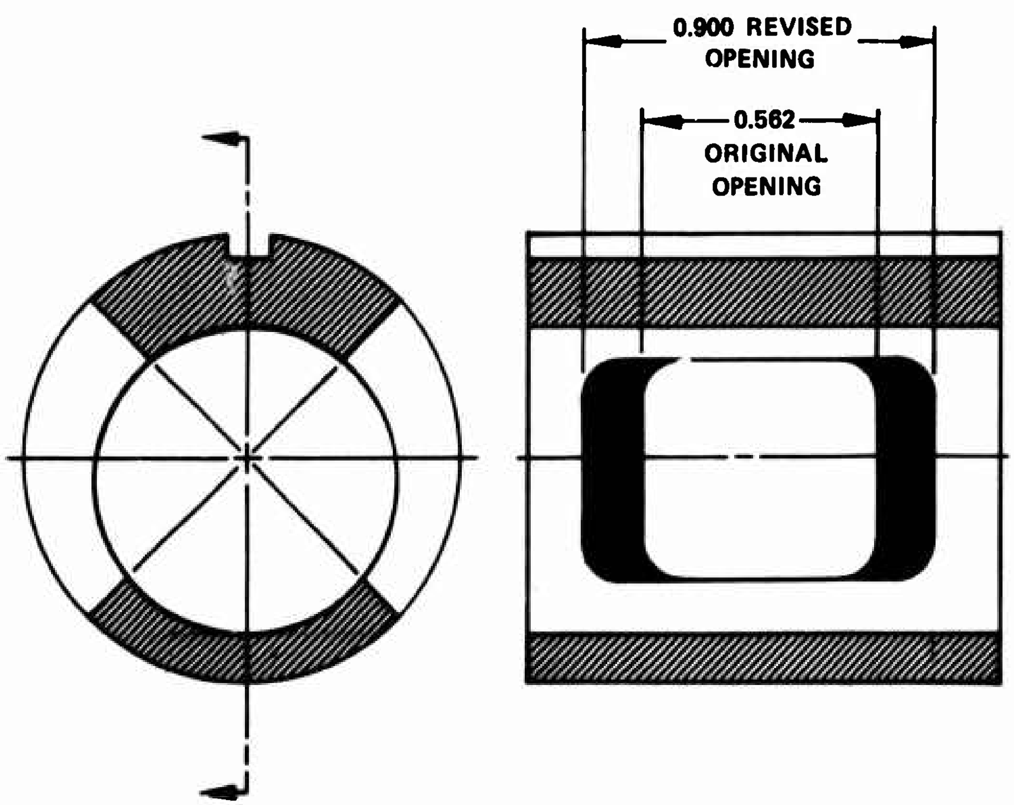


Figure 104. Oil Pump Element Liner Showing Material Removed From Inlet Opening

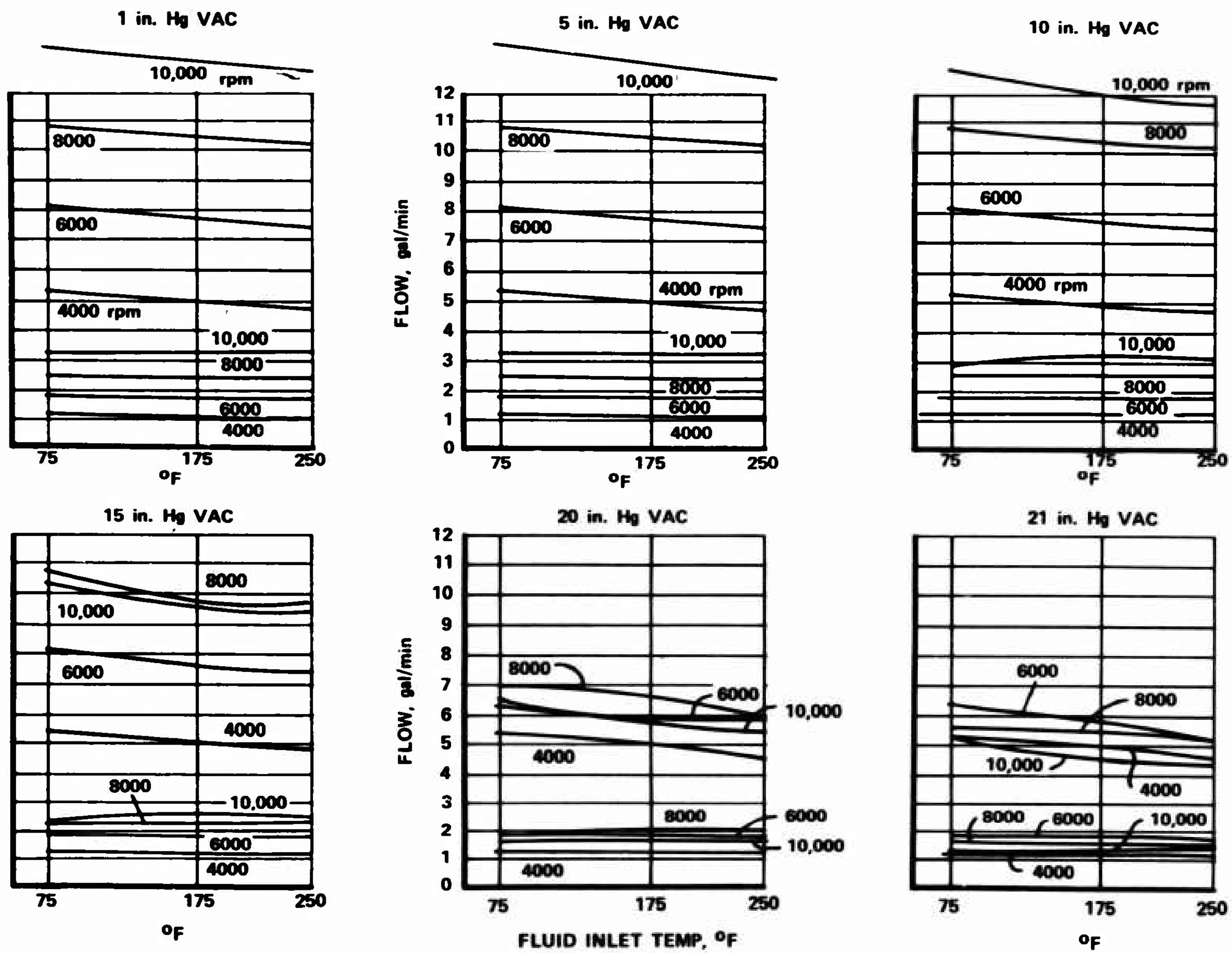


Figure 105. Pump Performance Variation With Altitude – Increased Lubrication Element Inlet Opening

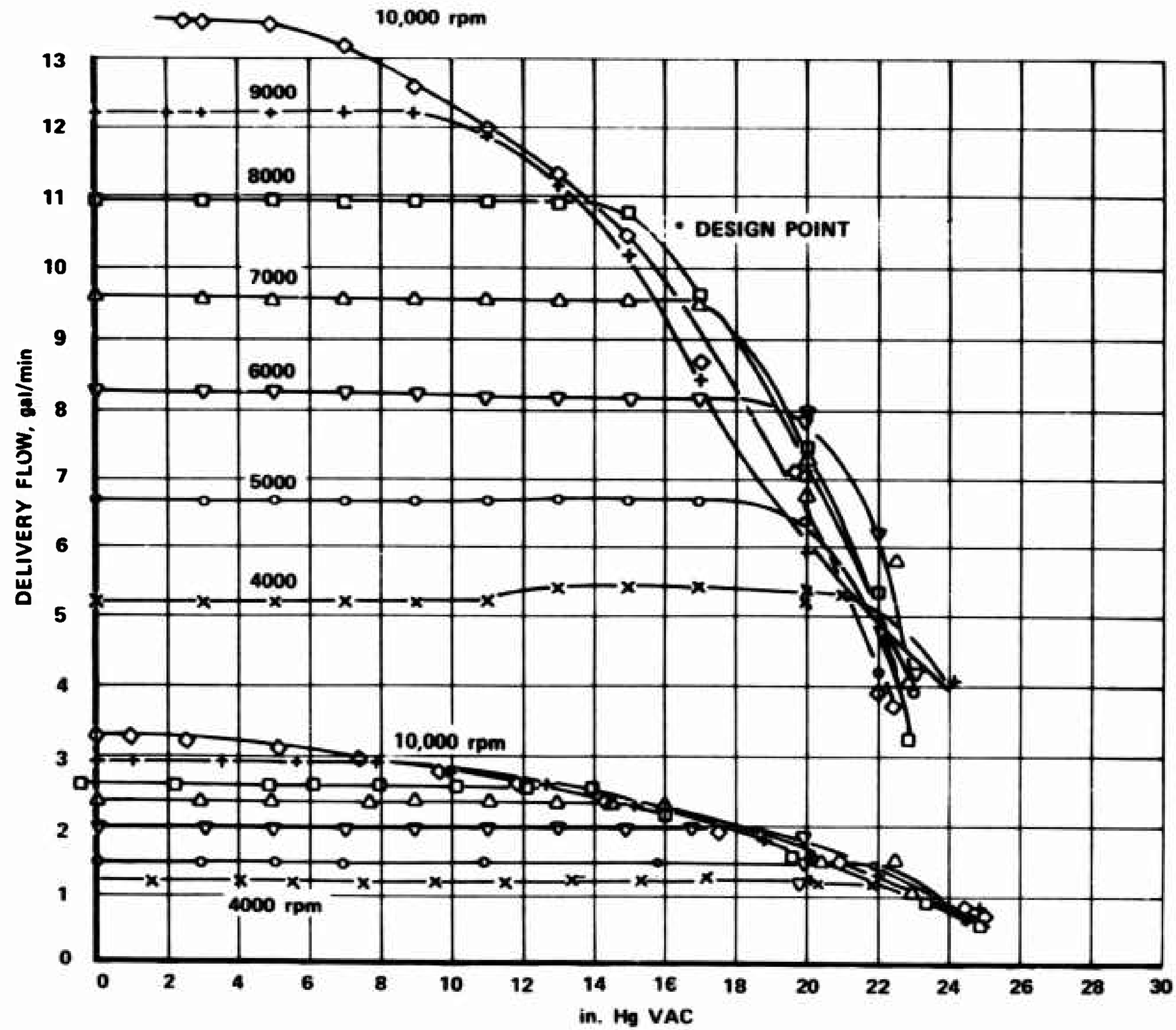


Figure 106. Oil Pump Cavitation Tests (Fluid Temperature 70–80°F)

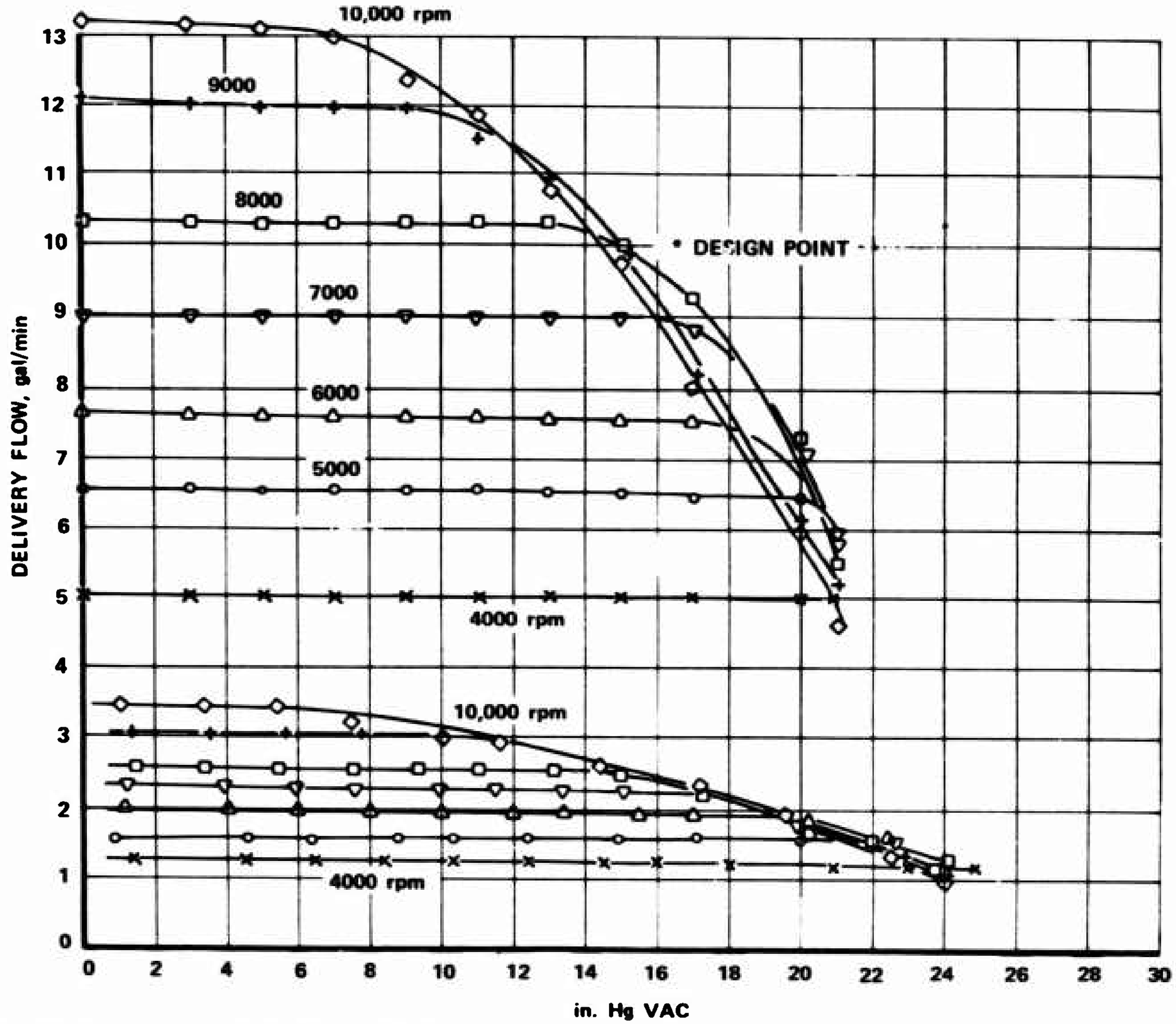


Figure 107. Oil Pump Cavitation Tests (Fluid Temperature 175°F)

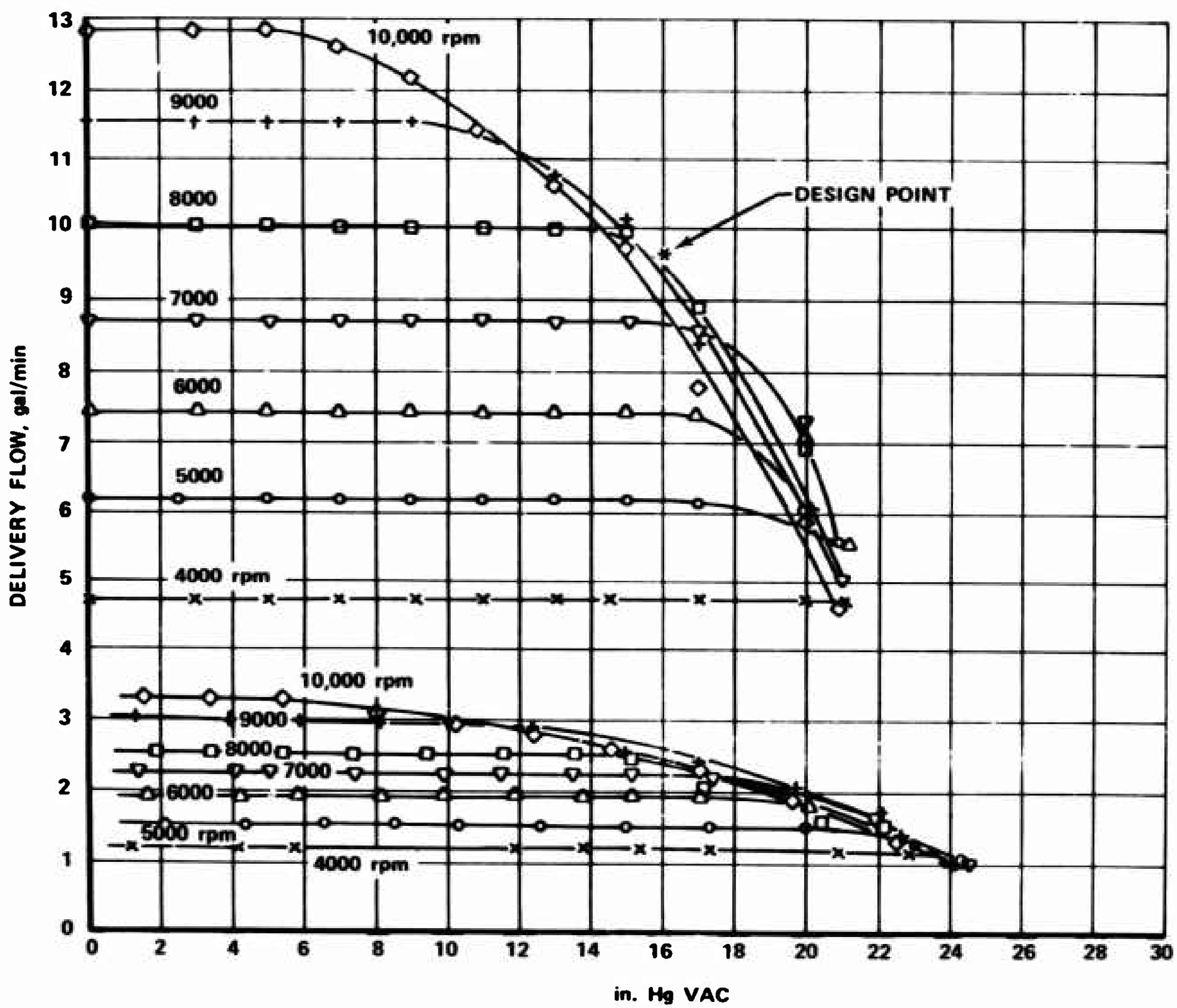


Figure 108. Oil Pump Cavitation Tests (Fluid Temperature 250°F)

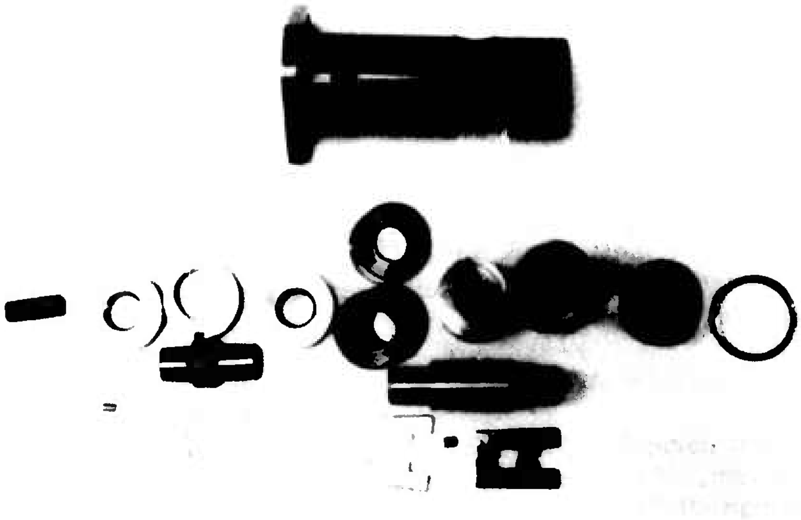


Figure 109. Oil Pump Components Before 50-hr of Endurance Testing at 250°F

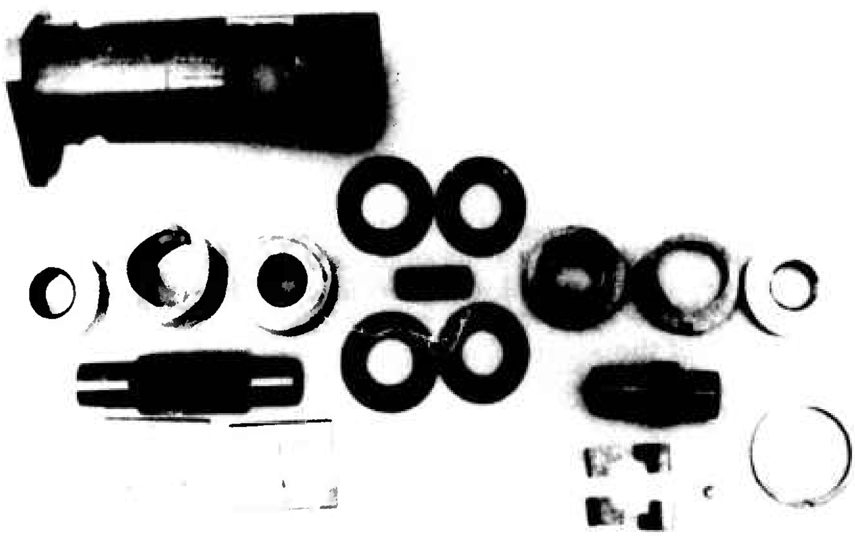


Figure 110. Oil Pump Components After 50-hr of Endurance Testing at 250°F

3. Test Results

The Sundstrand vane element oil pump ran satisfactorily throughout the test series. The altitude performance improved with increasing oil temperature, but at design speed and with 250°F oil, the lubrication element output flow began to deteriorate at an inlet condition of approximately 18,000 ft altitude (Figure 108). The lubrication and smaller, scavenge element performance was for the most part, comparable throughout the test. However, as indicated in Figure 108, the scavenge element does appear to meet the 20,000 ft altitude inlet pressure condition with negligible reduction in output flow. Performance was unchanged following a 50-hr endurance test at elevated oil temperature. Running the pump dry, simulating loss of system oil pressure, did not result in excessive housing temperature or damage to the pump.

4. Recommendations for Additional Testing

A scavenge element normally does not operate with a 100% fluid inlet, as was simulated in this test. The scavenge element is usually sized for two to four times the scavenged oil flow to pump an air/oil mixture resulting from air leakage into the bearing compartment. Additional tests are recommended to investigate and define capability of a high-speed vane element to pump a vapor/liquid mixture against various backpressures.

B. STARTER OVERRUNNING CLUTCH - AIRESEARCH

1. Background

The air turbine starter design configured for the rear-drive engine achieved minimum weight and maximum performance efficiency by locating the overrunning clutch in line with the engine rotor. This resulted in a clutch overrunning speed of 65,000 rpm that is considerably in excess of previously demonstrated capability.

High-speed sprag clutches, in the driving mode, generally fail from fatigue or sprag rollover. Overrunning at high speeds can lead to excessive clutch and race wear, which ultimately results in rollover or inability to drive. The sprag type clutch also necessitates incorporation of a decoupler mechanism to prevent overspeed of the starter turbine wheel in the event of a clutch or bearing failure. At the design speed of 65,000 rpm, it is important that the decoupler provide complete disengagement of the starter from the engine.

Two basic sprag clutch designs are currently in use, and these two designs can be arranged in two ways, that is, the inner race of the clutch can overrun or the outer race can overrun. Borg-Warner manufactures one of the basic designs. This design is a double-cage, full-phasing sprag clutch consisting of two concentric cages with a ribbon spring between. The spring provides a light energizing torque on the sprag to ensure positive engagement. Formsprag offers the second basic clutch design. This configuration is a single-cage full-complement design. The sprags contact each other to provide phasing and rollover protection. A garter spring provides engaging force.

The clutch test program was comprised of two phases. Phase I consisted of an analytical and experimental investigation of the design concepts to identify the most promising configuration. Testing enabled conclusions to be drawn with regard to feasibility, problem area definition and design changes. The Phase II effort demonstrated a measure of the operating life of the selected clutch configuration and decoupler. A 1000-hr endurance test was performed to establish a measure of the life expectancy, and decoupler tests were run to define disengagement torque levels.

2. Test Program - Phase I

Both Formsprag and Borg-Warner evaluated the clutch requirements discussed in Section VIII. Borg-Warner indicated that they favored the inner race overrunning approach due to the very high disengaging speed and close tolerance required to ensure clutch release between starter cutout speed and engine idle. In addition, Borg-Warner had an existing inner race overrunning design which they felt could be modified to meet the requirements.

Formsprag indicated that they did not have a clutch capable of meeting this design requirement, and presented a proposal for the development of an outer race overrunning configuration. After reviewing both clutch vendors' recommendations, it was the opinion of AiResearch's design and engineering departments that the outer race overrunning approach proposed by Formsprag involved greater technical risk than the inner race overrunning approach proposed by Borg-Warner. The ability to accurately predict the lift-off speed of the outer race overrunning configuration was the primary area of concern. In addition, the development program quoted for the outer race overrunning configuration was not within the scope of available time and cost.

Rather than running a 40-hr screening test on both clutch designs of each manufacturer, a 160-hr test would be run on the Borg-Warner inner race overrunning configuration. A test unit was designed around this clutch providing bearings support, lubrication system, oil dams, output shaft, drag torque measuring instrumentation and support housing (Figure 111).

The anticipated problem areas in the test unit were identified as the following:

1. Wear on the sprag clutch or the clutch inner race
2. Overrunning bearing life
3. Excessive overrunning drag torque
4. Lubrication flow and location.

The following series of tests was outlined to demonstrate the capability of the clutch test rig in achieving satisfactory performance with respect to the suspected problem areas.

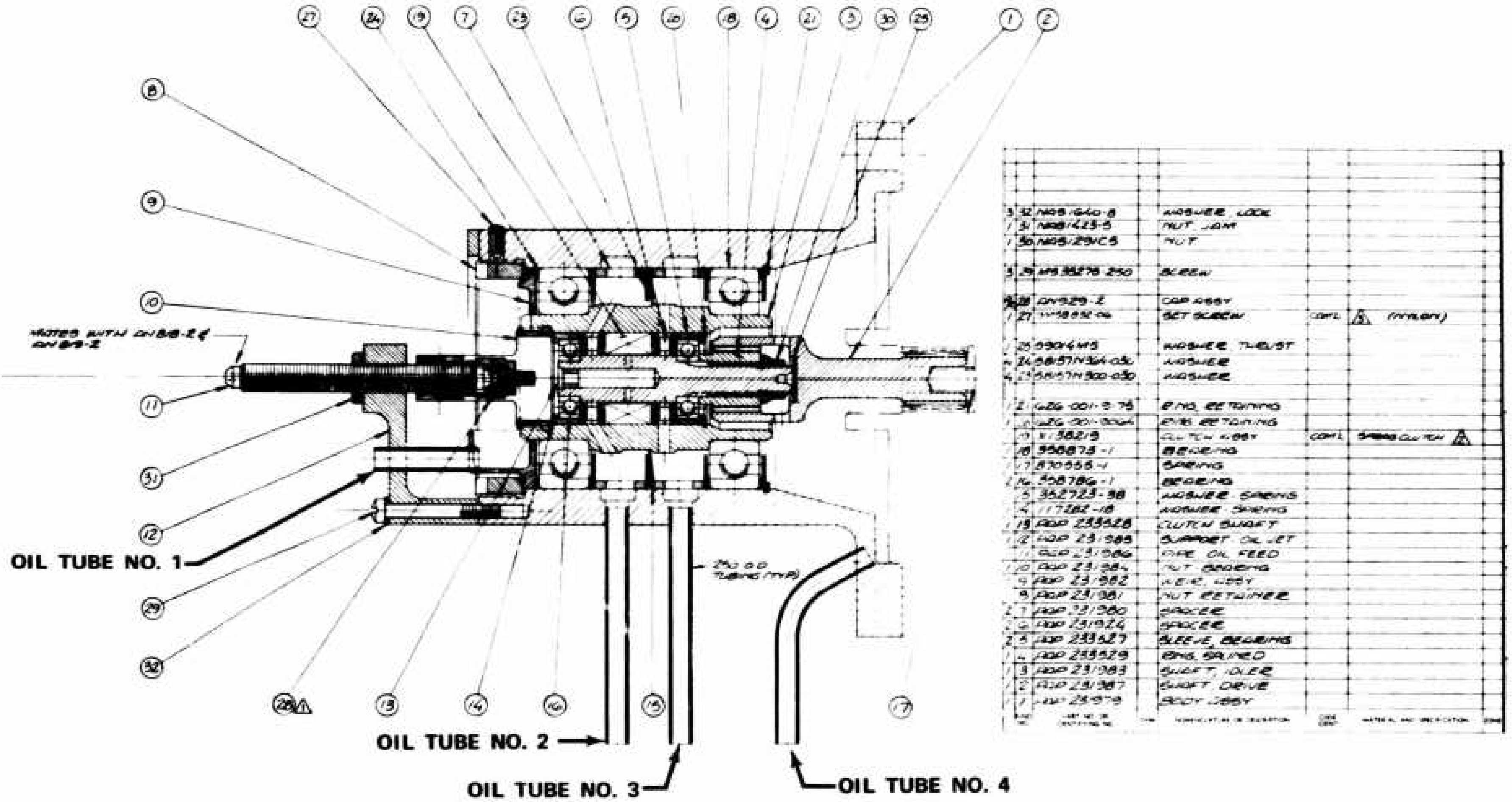


Figure 111. Phase I - Overrunning Clutch Test Rig

a. Segment I

A 40-hr test would be conducted at 65,000 rpm with predetermined values for oil jet size and pressure and for oil dam clearances. Overrunning drag torque would be monitored during the test; in the event of excessive drag torque, the oil flow to the clutch could be reduced. Upon completion of the 40-hr test, the unit would be disassembled and inspected. Critical dimensions would be measured to determine the extent of clutch or inner race wear. The overrunning bearings would be inspected for signs of deterioration. Should all parameters prove satisfactory, the test unit would be rebuilt and Segment II conducted. Should there be indications of distress in any of the components of the test unit, changes would be made, aimed at correcting the problem area, such as increasing oil flow or modification of oil dam clearances to increase oil supplies to the bearing. Following any necessary changes, the test unit would be rebuilt and subjected to Segment II.

b. Segment II

Conduct a 40-hr test similar to Segment I. Following completion of Segment II, the teardown and inspection procedure of Segment I would be performed. The test unit would again be evaluated and any necessary changes made. Upon completion of Segment II, the test unit would be subjected to Segments III and IV.

c. Segments III and IV

Segments III and IV would consist of separate 40-hr test similar to Segments I and II followed by teardown inspections. At the completion of Segment IV, a total of 160 hr of testing would be accomplished on the test unit.

Due to the high-speed operation required for this test, a special gearbox was designed to allow the use of conventional drive motors as the power source. A speed increasing gearbox with a 9.13:1 gear ratio was designed and tested at the required speed of 65,000 rpm. The gearbox presented no problems. However, upon attaching the clutch test rig and associated coupling shafts, critical speed problems were encountered above 45,000 rpm. An isolator pillow block was designed to mount between the 9.13:1 gearbox and the clutch test rig. The improved support offered by this isolator eliminated the critical speed problems. Figure 112 shows a schematic of the test stand including the clutch test rig. Figure 113 shows a photograph of the actual test stand and clutch test rig.

Figure 111 shows an assembly drawing of the clutch tested in Phase I. The outer race (Item 3) was designed to mount in antifriction roller bearings to allow accurate determination of the overrunning drag torque during testing. A strain gage was mounted to the bearing preload nut (Item 10) to provide continuous drag torque data. As previously noted, all Phase I testing was accomplished using a Borg-Warner inner race overrunning sprag clutch (Item 19). Oil dams (Item 6) were provided on either side of the clutch to control oil flow from the clutch and also to restrict the oil flow to the overrunning bearings (Item 16). The oil supply was introduced into the clutch by means of an oil jet (Item 11) which directed the oil into the center of the clutch inner race (Item 13). The oil was then discharged, using the centrifugal force created by the inner race, through three equally spaced holes located directly under the sprag clutch. Holes placed in the outer race then allowed the oil to drain to the main support housing (Item 1). Four oil return tubes were installed in this housing to monitor oil flowrates from isolated areas in the clutch. Tube No. 4 measured oil flow through the aft overrunning bearing. Tubes No. 3 and No. 2 measured oil flow through the clutch and the two oil dams. Tube No. 1 measured oil flow through the forward overrunning bearing. The oil flow data were used to evaluate the performance of the oil dams. Thermocouples were installed in Tubes Nos. 3 and 2 to monitor outlet oil temperatures and in the oil supply jet to monitor inlet oil temperature. The test rig was mounted to the drive assembly and power transmitter to the clutch through a quill shaft (Item 2). All testing was conducted using the test fixture described.

3. Testing

Initial testing of the test rig was designed to determine desirable oil flows and temperature levels and to calibrate the drag torque strain gage. A 0.040-in.-diameter orifice was installed in the oil supply tube, and oil flows and temperatures were monitored at pressures from 20 psig to 40 psig. Inlet oil pressure of 30 psig provided the most desirable results, and all further testing was accomplished using the 0.040-in.-diameter orifice and 30 ±1 psig inlet pressure. Overrunning drag torque was found to be very low, and the strain gage was calibrated to record 0-10 lb-in. This level covered all torque levels which could be expected from normal operation of the clutch. The inlet oil temperature was not controlled for any of the Phase I testing but was allowed to seek its own level.

Following the completion of these pretest evaluations, the test outlined for Segment I was initiated. The test was terminated after 18 hr of continuous operation at 65,000 rpm due to a bearing failure in the isolator pillow block. A mismatch in the pilot diameters of the isolator and clutch housing was found that led to shaft misalignment and undue load on the clutch bearings. The clutch test rig damage was repaired and the isolator pilot diameter corrected and the bearing replaced. The test was restarted at the 18-hr point of termination. At 20 hr of test time, another failure of an isolator pillow block bearing occurred, resulting in secondary damage to the clutch. It was determined that an isolator bearing of higher quality was needed to allow continuous operation at 65,000 rpm. The isolator bearings were replaced with one of a better design. The clutch test unit damage was repaired and returned for further testing. It should be noted that the strain gage indicator was disconnected after approximately 19 hr of test. The recorded torque had been consistently low, and there was no need for further data. A typical drag torque at 9 hr was 0.31 lb-in. At this time Segment II was initiated. The test rig successfully completed 40 hr of continuous overrunning at 65,000 rpm. The test was terminated and a teardown inspection performed. Figures 114, 115, and 116 show the test components. As can be seen, no sign of wear is present on the clutch or inner race. The overrunning bearings were smooth, with no sign of distress. Some wear was evident on the quill shaft spline teeth and mating clutch shaft spline. This was expected, however, since no oil was supplied to these splines in Phase I testing. In view of the excellent condition of the test hardware, the unit was reassembled with no hardware or design changes. At this time a total of 60 hr of testing had been conducted. This left 100 hr of total test time to be performed.

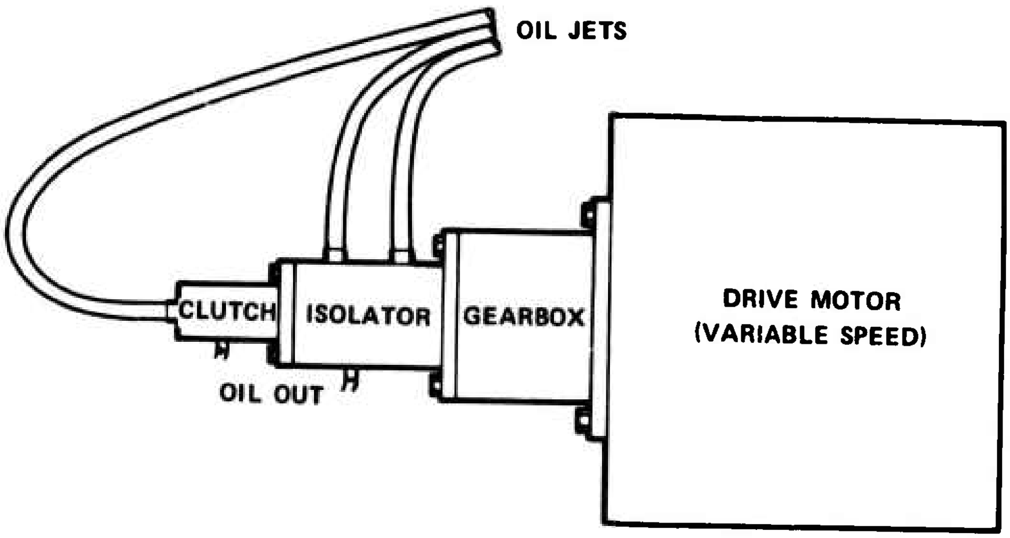


Figure 112. Overrunning Clutch Test Stand Schematic

Figure 113. Clutch Endurance Test Rig

Figure 114. Clutch Test Rig After 40-hr of Overrunning at 65,000 rpm

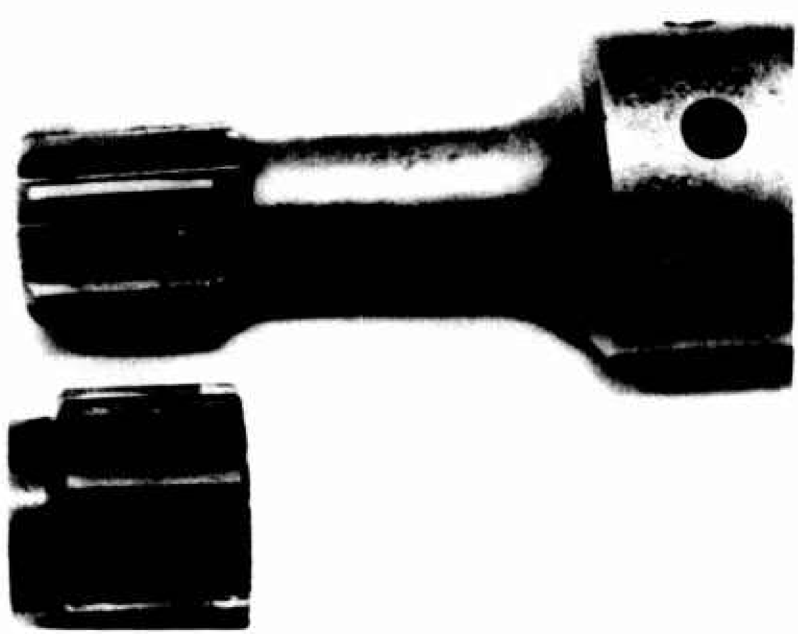


Figure 115. Quill Shaft and Spline After 40-hr Overrunning at 65,000 rpm

Figure 116. Clutch Components After 40-hr Overrunning at 65,000 rpm

It was decided, based on the results of the successful 40-hr endurance run, to attempt the remaining 100 hr without a teardown inspection. The drag torque strain gage was reconnected for this test. After 47 hr 35 min of continuous test time, a slight increase in vibration was noticed. The test was interrupted to check the test rig. It was found that the nut (Item 8 of Figure 111) was loose, allowing the clutch outer race to vibrate. The nut was tightened and secured and testing resumed. The strain gage was disconnected at this time since the torque was again very low and consistent. At 49 hr the nut (Item 10) fatigued and broke off at a spot just above the oil jet (Item 28). The broken section fell forward and struck the separator of the overrunning bearing (Item 16).

Testing was resumed and continued to 50 hr, when unusually high bearing temperatures were noted. The test was again interrupted, and a small crack was found in the bearing separator which had been struck by the broken nut. No other damage or distress was noted. The bearing was replaced and testing continued. At 97 hr 20 min, the test was terminated due to the failure of the overrunning bearing next to the quill shaft. Figures 117 and 118 are photographs of the test hardware following the 97-hr run. It should also be noted that this same set of hardware had completed 40 hr of overrunning prior to the 97 hr, for a total of 137 hr. The significant item in the photographs is the condition of the clutch inner race 180 deg away from the damage which resulted when the bearing failed. No measurable wear could be found in this area, indicating that the clutch was performing as expected prior to the bearing failure.

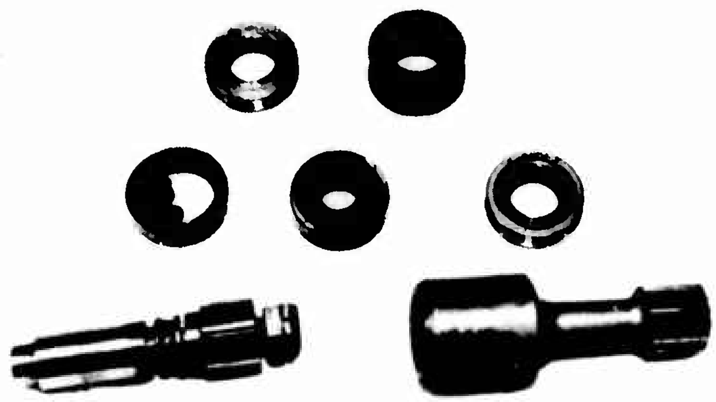


Figure 117. Clutch Components After 137.5-hr of Overrunning at 65,000 rpm (Overrunning Bearing Failure)

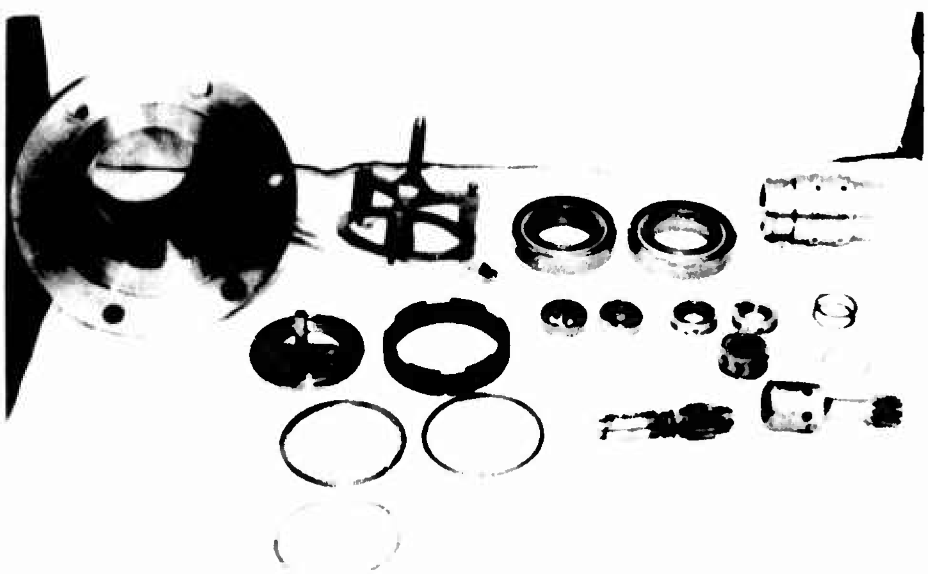


Figure 118. Clutch Test Rig After 137.5-hr Overrunning at 65,000 rpm (Overrunning Bearing Failure)

Further evaluation of the bearing failure revealed a potential design flaw which could have contributed to the bearing failure. Referring to Figure 111, a spring washer (Item 14) is employed to preload the overrunning bearings. It was noted that any reverse thrust from the quill shaft (Item 2) would tend to unload the bearing nearest this shaft. Further, a built-in thrust was inadvertently provided by the spring (Item 17) which would be added to any thrust generated by spline misalignment. The Phase II design for this test rig relocated the spring washer (Item 14) between Item 5 and Item 20. The same bearing preload will result, but reverse thrust cannot unload either bearing.

No further testing was conducted on the Phase I design. It was felt that no new information would have been gained by running of the additional 2-1/2 hr needed to complete the 100-hr test. Total accumulated test time was 157.5 hr.

4. Test Results

The results of the Phase I testing indicated a high degree of confidence in designing a reliable clutch assembly for operation at 65,000 rpm. The oil flow requirements were well within available limits. The overrunning drag torque would present no special problems in an actual engine application, and the oil temperature rise levels are not prohibitive. For a clutch oil flow of 1.25 lb/min, a nominal oil temperature rise of 10°F was observed.

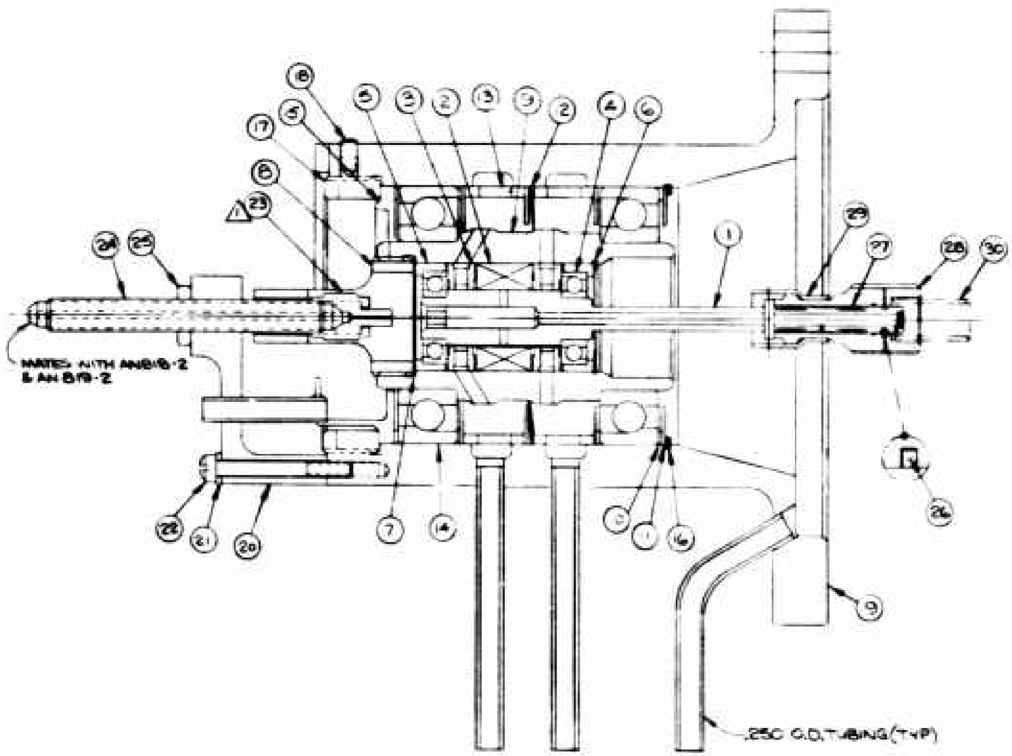
The two basic problems which were identified during Phase I testing appeared to be proper overrunning bearing preload and quill shaft unbalance and misalignment. The bearing preload problem was solved by the relocation of the preload spring washer. The quill shaft was eliminated in the Phase II design and replaced with a one-piece clutch inner race-output shaft combination.

5. Test Program - Phase II

This section summarizes the results of Phase II testing of the overrunning clutch. The purpose of this test was to incorporate the knowledge gained during Phase I testing into a design capable of 1000 hr of overrunning clutch operation at 65,000 rpm. The test stand used for this test was identical to that utilized for Phase I testing. No changes in test stand design were necessary for Phase II. However, the two design improvements discussed above were incorporated into the endurance clutch rig as described following.

a. Clutch Endurance Test Rig Design

The Phase II test program had two main goals: the first to achieve 1000 hr of 65,000 rpm clutch operation and the second to incorporate a one-piece output shaft with a spline drive and decoupler into the design. The Phase II test rig was modified to incorporate an output shaft and decoupler mechanism which simulates an actual engine application. Figure 119 is the assembly drawing for the Phase II clutch test rig. It can be seen that the test rig is identical to that tested during Phase I, except for the addition of the decoupler mechanism and the longer, one-piece, splined, output shaft. It should be noted that the spring washer (Item 7) was located between the retaining ring (Item 6) and the overrunning bearing (Item 4) when the test unit was assembled. This will prevent unloading of the overrunning bearings by the thrust resulting from the decoupler control spring (Item 30). A description of the oil feed and scavenge plumbing for the test rig was contained in the Phase I test description.



	FIND NO	PART NO. OR IDENTIFYING NO	SYM	NOMENCLATURE OR DESCRIPTION	CODE IDENT	MATERIAL AND SPECIFICATION	ZONE
1	30	PA3500938-1		SPRING			
1	29	PA3500937-1		TUBE, RETAINING			
1	28	PA3500936-1		SPLINE, DECOUPLING			
2	27	PA3500934-1		BUSHING SET			
1	26	PA3500935-1		RING, SNAP, ACTUATING			
	25	NAS1423-5		NUT, JAM			
	24	PAP 231986-1		PIPE, OIL FEED			
1	23	AN929-2		CAP ASSY			
3	22	MS35275-250		SCREW			
3	21	NAS1640-8		WASHER, LOCK			
1	20	PAP 231985-1		SUPPORT, OIL JET			
1	19	PAP 23197?-1		BODY ASSY			
1	18	MS 806?-8		SETSCREW (#8-32)			
1	17	PAP 2319?81-1		NUT, RETAINER			
1	16	626-00?-9175		RING, RETAINING			
1	15	PAP 231?82-1		WEIR ASSY			
2	14	358873-1		BEARING			
2	13	PAP 23?380-1		SPACER			
1	12	3527?3-38		WASHER, SPRING			
4	11	SB157N 00-030		WASHER			
4	10	SB157N ?4-030		WASHER			
1	9	PAP 231?83-1		SHAFT, IDLER			
1	8	PAP 231?34-1		NUT, BEARING			
	7	117282-8		WASHER, SPRING			
1	6	626-001-9C-4		RING, RETAINING			
2	5	PAP 233527		SLEEVE, BEARING			
2	4	358786-1		BEARING			
2	3	PAP 231924-1		SPACER			
1	2	X 138219	2	CLUTCH ASSY	GOAL	SPRAG CLUTCH	
1	1	PA3500333-1		SHAFT ASSY, CLUTCH			

Figure 119. Phase II - Clutch Endurance and Decoupler Test Rig

b. Clutch Endurance Testing

The data recorded during the test were identical to those for Phase I testing, except drag torque. Due to the low drag torque recorded during Phase I testing, no drag torque data were recorded during Phase II testing. The inlet oil temperature to the clutch was maintained at approximately 110°F and was allowed to seek its own level for the first 200 hr. At this time a teardown inspection was performed. Figure 120 is a photograph of the clutch components following teardown. As can be seen, there is no visible sign of distress on any of the components. The test unit was reassembled and returned for continued testing. Inlet oil temperatures to the clutch for the remainder of the test were maintained at approximately 200°F. It should be noted that some early difficulties were encountered in maintaining this temperature. However, continued efforts at proper insulation resulted in the desired inlet oil temperatures.

After 686 hr, the test was stopped for an inspection due to very erratic oil flow from oil tubes No. 3 and No. 4. A teardown inspection was initiated, revealing that the oil plug at the end of the clutch shaft (Item 1) under the decoupler had become dislodged. This plug had been tack welded in place and it is unknown what caused it to come out. When this plug is not in the system, the oil flowing down the center of the clutch shaft bypasses the clutch and overrunning bearings and is expelled through the decoupler end of the test rig.

This situation most drastically affects the bearing nearest to the oil jet and effectively causes it to run without sufficient lubrication. The decoupler mechanism had actuated, and bearing analysis indicated that the forward bearing, closest to the oil jet, was distressed due to the lack of oil. The increased bearing drag which resulted, actuated the decoupler.

Analysis of the test data showed that oil flow in oil tube No. 3, which is providing flow through the clutch, dropped almost 50% following the decoupler actuation, and oil flow in oil tube No. 4 increased by a factor of 100. This would be expected since the inner race does not rotate following decoupler actuation. Without inner race rotation, the radial holes normally expelling oil to the clutch no longer function, causing flow from the jet to back up and run into oil tube No. 4.

A photograph of the clutch components is shown in Figure 121. The output shaft was broken during disassembly when the decoupler was being removed. It is suspected that the shaft was weakened during the extended period of operation in the decoupled condition. The dislodged plug can be seen in the foreground between the shaft and the spline. The bearing nearest the spline, in the photograph located nearest the oil jet, exhibited high drag. It can be seen that the inner race does not appear damaged and detailed inspection revealed no measurable wear. The clutch and aft bearing were also undamaged.

This incident demonstrated that a failure of any overrun component in the starter that causes an additional drag torque on the starter turbine will cause the decoupler to actuate. Further, after actuating, continued operation of the engine will not cause any distress upon those critical components within the engine that connect to the starter.

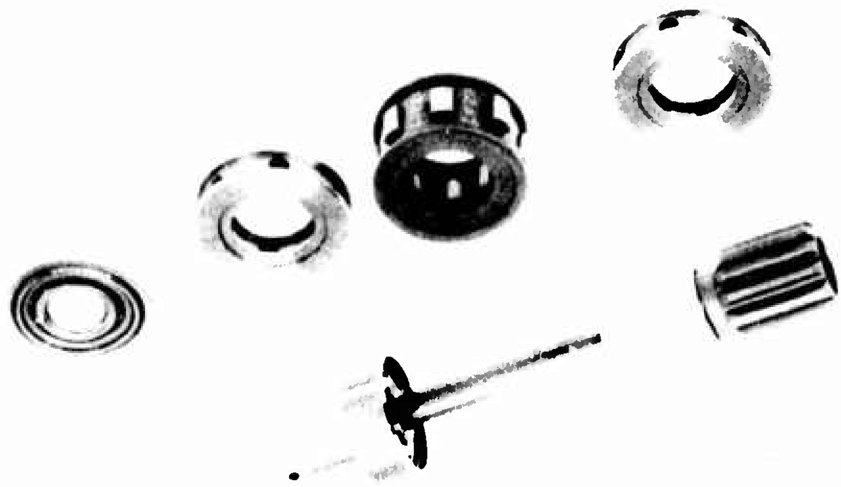


Figure 120. Clutch Components After 200-hr of Overrunning at 65,000 rpm

Figure 121. Inner Race and Bearings After 686-hr of Overrunning at 65,000 rpm (With Oil Plug Failure)

The test unit was rebuilt using the original clutch and aft bearing. The forward bearing, inner race shaft, and decoupler were replaced. Testing was resumed at 686 hr. The remaining 314 hr were completed without incident. A teardown inspection was conducted following completion of the test, with no significant signs of wear or damage to the clutch, inner race, or output shaft spline. Figures 122 through 124 show the clutch components after 1000 hr of testing.

c. Decoupler Test

Following the completion of the endurance test, the unit was rebuilt with the clutch installed so that it would drive in the normal overrunning mode, thus allowing a determination of the torque required to actuate the decoupler. The test unit was mounted to the test stand gearbox. With the output shaft of the gearbox restrained, a torque was applied to the nut (Item 8 of Figure 119) until the decoupler actuated. This procedure was repeated a total of ten times. The recorded decoupler torque was between 28 and 30 in.-lb on each occasion. The decoupler had been designed to actuate at a level of 30 in.-lb maximum. It should be noted that the level of decoupler actuation torque can be varied by changing the spring (Item 30) to result in higher or lower axial load required on the jaws of the decoupler.

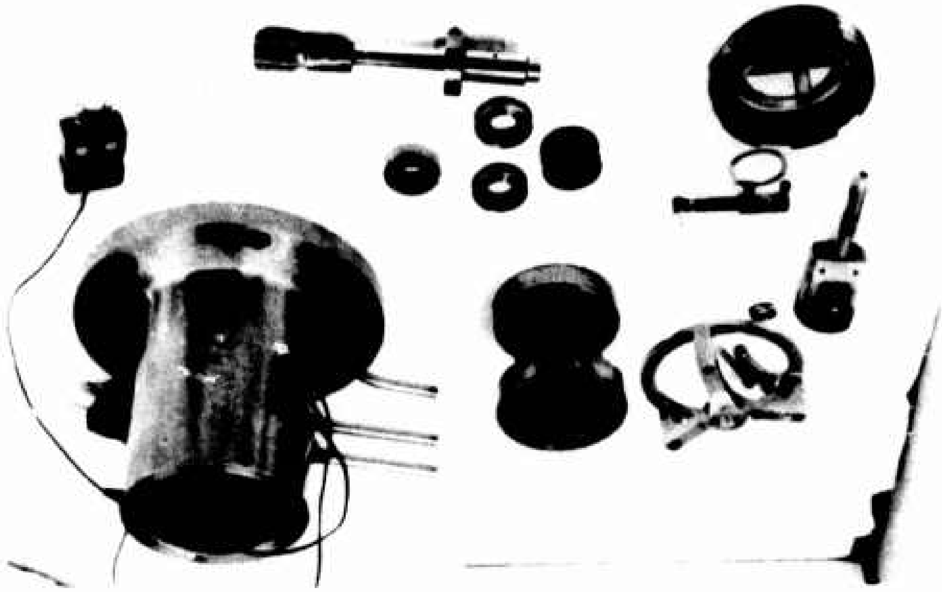


Figure 122. Clutch Test Rig After 1,000-hr of Overrunning at 65,000 rpm

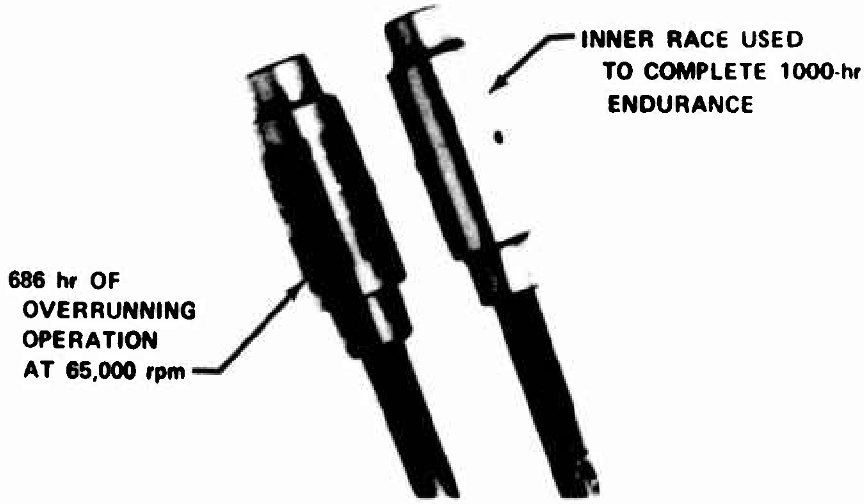


Figure 123. Inner Race at Conclusion of Endurance Test

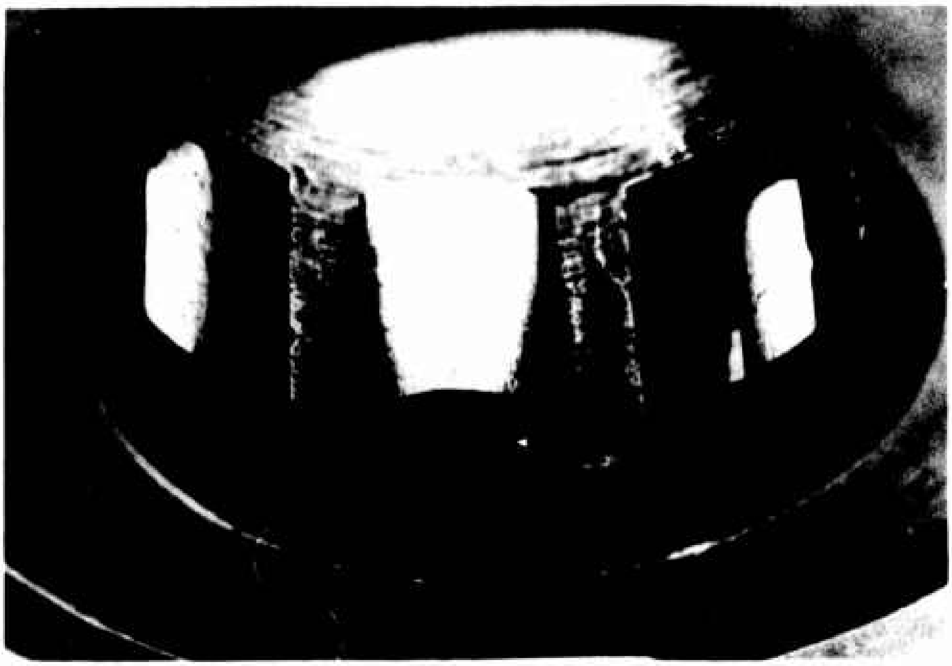


Figure 124. Clutch at Conclusion of 1,000-hr Overrunning at 65,000 rpm Showing Clutch Sprag

d. Test Results

The results of Phase II testing have indicated that the design and operation of a 65,000-rpm overrunning sprag clutch is feasible. The drag torque and resulting oil system temperature rise were well within allowable limits, and the endurance wear characteristics of the critical clutch elements were acceptable. Further, the incorporation of a decoupler mechanism has also proved to be a workable solution to the emergency disengage requirement. The spline utilized on the output shaft showed no signs of excessive wear and should present no special problem in an engine applicati n. Every effort was made in this program to simulate an actual engine installation. It is felt that the results of the test program are directly applicable to an integrated engine-starter design and will result in a high degree of confidence in a production design.

C. ELECTRONIC CONTROL COMPONENTS TO HOUSING HEAT TRANSFER - CHANDLER EVANS INC.

1. Background

Through two phases of Army-sponsored programs, development work has been conducted to formulate concepts to provide adequate cooling of the electronic computer.

A brief summary of previous milestones is as follows:

1. USAAMRDL Phase I - Electronic control hard-mounted to the fluid controller utilizing fuel at a maximum temperature = 135°F for cooling and a fluorinert fluid as a heat transfer mechanism from the components to the cold plate. The maximum ambient temperature was 250°F.

   This arrangement proved to be successful, limiting the maximum component surface temperature to 190°F, well below the maximum allowed of 250°F. However, the inclusion of the fluorinert fluid added cost and weight and proved to be very difficult to use.

2. USAAMRDL Phase II - Electronic control utilizing fuel at a maximum temperature = 135°F for cooling and conductive heat transfer via the printed circuit boards (PCB's) and interconnecting cables from the components to the heat sink.

   With an ambient temperature of 250°F, the maximum component surface temperature was 195°F; with an ambient temperature of 180°F and no fuel cooling, the maximum component case temperature was 190°F.

The basic requirements for this critical item test program were to provide an adequate thermal path from every electronic component to the computer housing heat sink, to ensure adequate heat transfer, to minimize hot spots, and thereby to enhance reliability. The heat sink in this case is the computer housing mounting surface that interfaces with the engine inlet particle separator and is thereby cooled by the compressor inlet air.

The proposed components to housing heat transfer concept is based on utilizing multilayer printed circuit boards for interconnecting the electronic components. The outer layer of the boards is an additional copper layer termed a thermal plane (Figure 125). This layer of copper covers the entire PCB except for a minimal spacing of 0.015 in. between it and the component pads. This exposed plane is connected at two adjacent corners to the computer housing by a flexible copper cable, an arrangement which provides a virtual thermal short circuit between the computer housing and the thermal plane on each board. There are two thermal paths between the heat-producing electronic components and the thermal plane. One of the paths is direct from the component housings which are in contact with the thermal plane, and the second path is from the component leads and interconnecting conductors through the PCB epoxy-glass material to the thermal plane. For the power driver transistors, a mounting clip is provided that electrically isolates the transistor, yet provides a reasonable thermal path (approximately 30°F/watt). The clip is fastened to the PCB in direct contact with the thermal plane.

The thermal plane configuration dictated a departure from previously developed concepts; thus a test program was conducted to determine the heat transfer efficiency of the new design.

The multilayer PCB, which contains a thin sheet of tinned copper on the component mounting surface, and associated mounting hardware are shown in Figure 126. The board was supported in the electronic control housing on silicone vibration isolators whose thermal conductivity is negligible compared with the flexible copper braids which coupled the thermal plane to the housing. Power transistors mounted in beryllium oxide, electrically but not thermally insulated, TO-5 clips were used as the primary heat source. Four additional resistors held in steel clips simulated the power dissipated by components normally mounted on the remaining area of the board. This method of mechanically fastening the electrical components to the board resulted in a good thermal path from component to the thermal plane which could be easily repeated from one configuration to the other.

The sealed computer module containing the PCB, associated electronic components, and temperature measuring thermistors was enclosed in an environmental chamber and dc power was applied. A schematic of the test setup is given in Figure 127. The power dissipated in each component was individually measured by reading the voltage supplied and current drawn by the device. A variable load resistor in each transistor control circuit was tuned such that each transistor dissipated the same power within a 1% variance. Thus, the heat input was accurately maintained, and any measured temperature differences could therefore be attributed to the physical configuration under test.

2. Test Program

A printed circuit board of the type to be tested on an advanced technology turboshaft engine control, containing the highest power dissipating components, the stepper motor drive transistors, was tested in two phases. In the first series of tests, various thermal plane thicknesses were evaluated by mounting the PCB into the sealed electronic computer housing, dissipating power through a few components mounted on the board, and measuring the resulting temperature gradients along the board and component operating temperatures. The second phase of testing consisted of measuring the operating temperature of selected high-power dissipating components in a completely assembled prototype electronic control which was operating at a typical engine transient condition. Both test phases were conducted at various ambient temperatures, not to exceed a maximum ambient temperature of 80°C as specified in the Appendix.

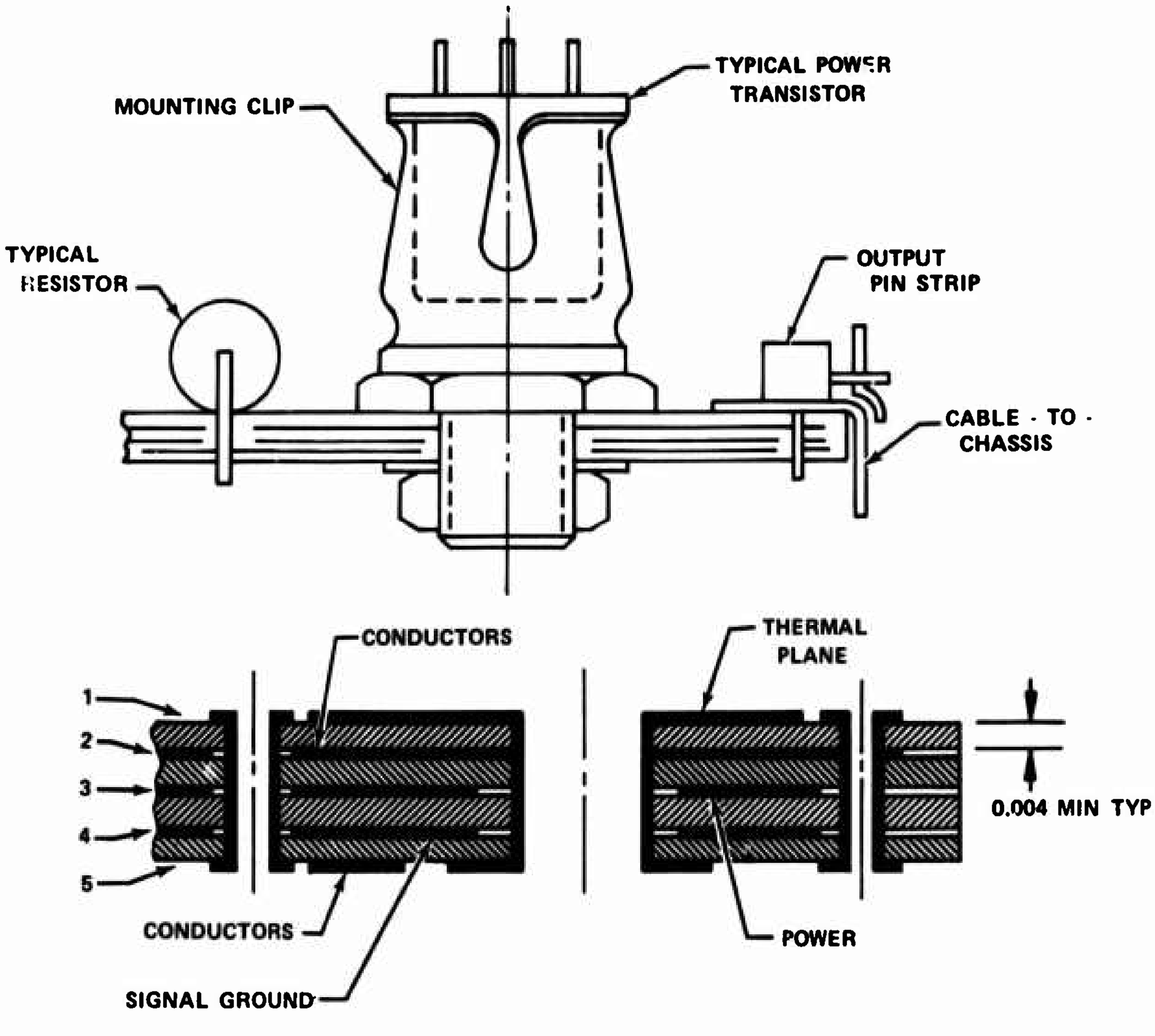


Figure 125. Thermal Plane Heat Transfer Concept

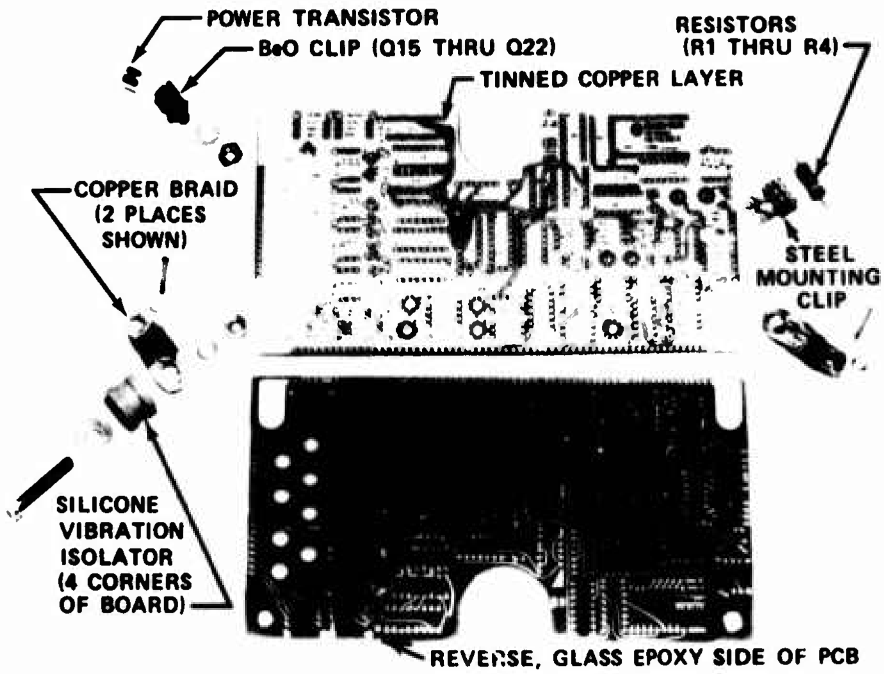


Figure 126. Thermal Plane Printed Circuit Board

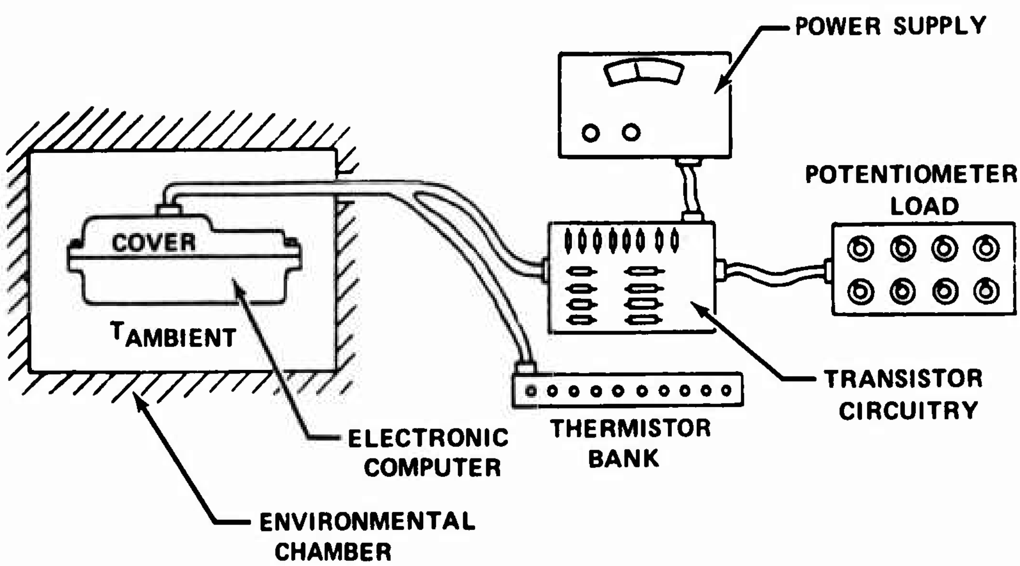


Figure 127. System Test Schematic

a. Phase I - Thermal Plane Evaluation

The first series of tests was conducted with only two power transistors bolted to the PCB through heat sinks at the Q15 and Q16 locations (Figure 128). The two transistors were operated to dissipate a considerable amount of power, 4.3 watts combined, to facilitate the measurement of temperature gradients throughout the board. Thermistors were located in positions 1, 10, 11, 12, and 13 as shown in Figure 128. The most significant effect was a 20°F decrease in the operating temperature of the components and of the board temperature in the vicinity of the heat source with a 0.009-in. thermal plane.

Ambient Temperature, °F	Thermal Plane Thickness, in.	Copper Braids, No.	Component Temperature - °F		
			Trans. at Q15	PCB_{10}	PCB_{11}
75	0.002	None	231	165	123
75	0.009	None	211	145	119

Thus, the thermal plane dissipates hot spots along the board by conducting localized heat inputs into a larger board area.

Another set of tests was conducted wherein all eight of the stepper motor drive transistors (i.e., U2T101 Darlington Power Transistors) were installed in their assigned locations, Q15 through Q22, and with 0.03-in.^{2} copper braids connecting the computer housing to the thermal plane. Each transistor was set to dissipate 3/4 watt, a power level approximately 50% greater than the individually highest dissipating operating condition which occurs when one continuously energized phase is required to hold the stepper motor in a fixed position. This power level yielded sufficient thermal gradients (10°-20°F) to negate much of the ±0.5°F accuracy of the thermistor and allow a good comparison of the various thermal plane configurations under test. Four resistors dissipating an additional total of 1 watt were spaced throughout the remainder of the board, and thermistors 1 through 8 were positioned as shown in Figure 128.

The test results are plotted in Figure 129. These data show that a nearly 30% decrease in temperature rise of the power transistors (T_Q-T_{amb}) can be achieved by utilizing a 0.009-in. thermal plane in conjunction with a 0.03-in.^{2} copper braid fastened to the housing. The components located close to the braid (Q21 is 0.5 in. away) derive approximately 7% more benefit in the reduction of temperature rise than those farthest removed from the braid (Q15 is 2 in. away). Also, the 0.009-in. thermal plane yields a 10% average decrease in temperature rise over the 0.002-in. plane.

b. Phase II - Prototype Control Component Temperatures

The completely assembled prototype electronic computer (Figure 130) employs 0.002-in.-thick thermal planes and 0.03-in.^{2} copper braid connections between the plane and the housing. The sealed unit with thermistors epoxied to the Q15 and Q21 stepper motor drive transistors was placed in an environmental chamber and operated at a typical engine transient condition with both stepper motors slewing. The operating temperature of the two transistors was monitored to determine the effect of the thermal plane with and without copper braids attached to the housing. The results are tabulated following.

Control Ambient T_{amb}, °F	Braids	Component Temperature, °F Q15	Component Temperature, °F Q21	Percent Decrease in Temperature Rise, $T_Q - T_{amb}$ Q15	Percent Decrease in Temperature Rise, $T_Q - T_{amb}$ Q21
70	No	116	106		
70	Yes	111	100	11	17
155	No	191	185		
155	Yes	187	181	11	13
180	Yes	211	205		

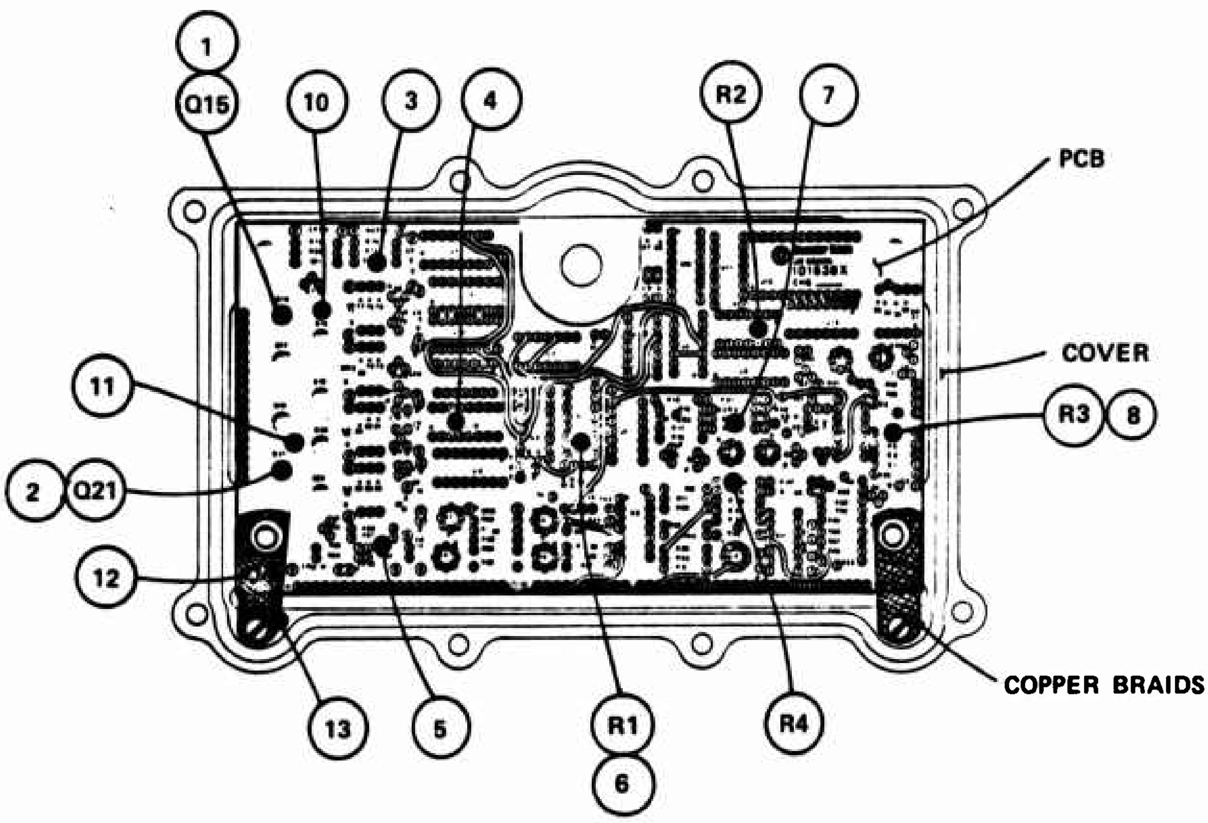


Figure 128. Printed Circuit Board Layout

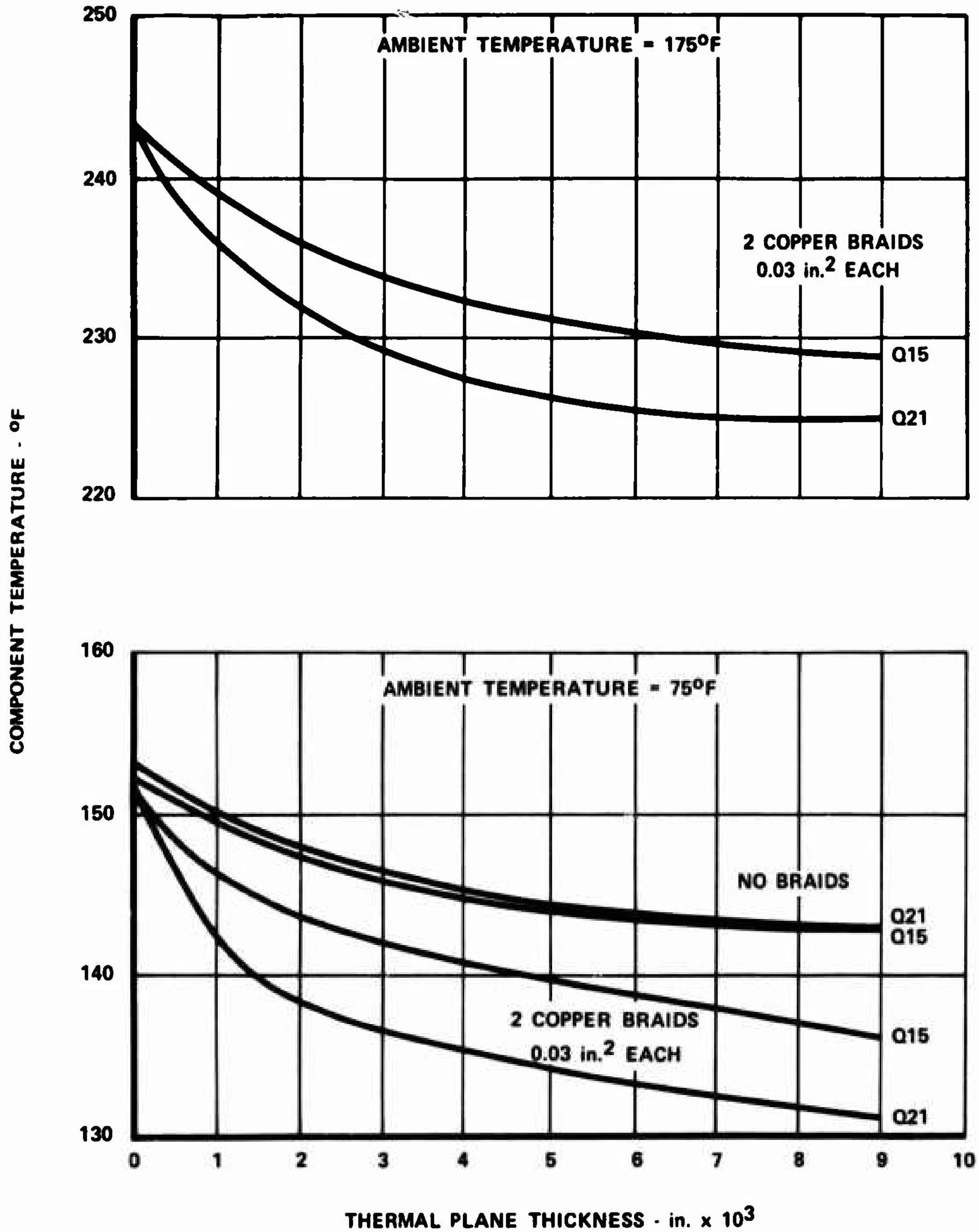


Figure 129. Component Temperatures vs Thermal Thickness

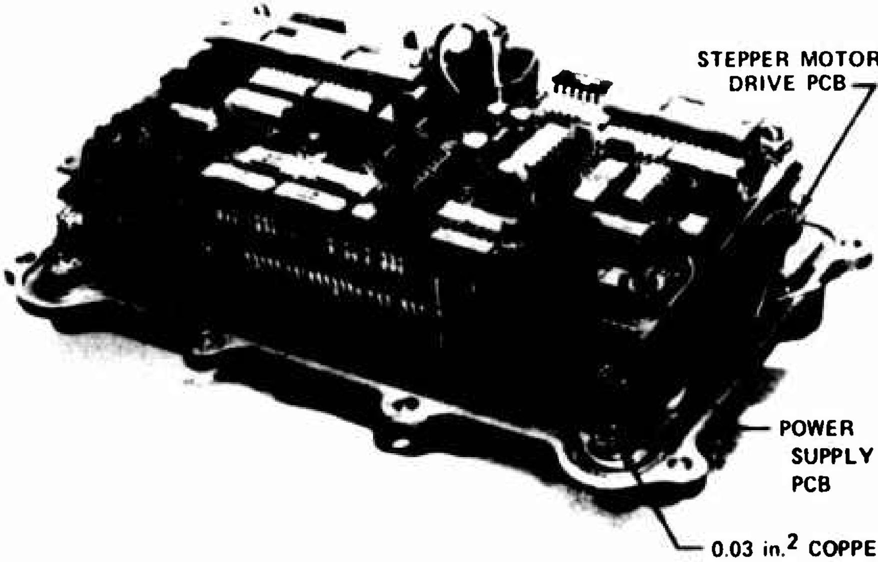


Figure 130. Electronic Computer With Thermal Plane PCB Implementation

The improvement in temperature rise of both power transistors with a braid connecting the thermal plane to the housing was better than that predicted by Phase I testing. The transistor located close to the braid, Q21, followed the previous results more closely (i.e., 13-17% decrease in temperature rise vs 12%); however, Q15, which is located 2 in. from the nearest braid, yielded a 11% decrease in temperature rise compared to the 4% measured in Phase I. It was found that with the board fully assembled and operating, components farther away from the braid were affected to a greater degree by components in their vicinity, whereas those close to a braid approach the condition of being thermally shorted to the housing. Measurement of the thermal plane near Q15 revealed a plane temperature 5°F hotter than the transistor itself, thus indicating heat flow from the rest of the board toward Q15. With the thermal braids connected, the plane temperature dropped by 5°F, thereby accounting for the total improvement in the operating temperature of the transistor mounted at Q15.

These results indicate that when heat is introduced throughout the board rather than at a few isolated locations, as was the case in Phase I testing, more of the thermal plane is utilized in the conduction path to the braids. Consequently, the thermal gradients across the board are less, the conductive heat transfer through the braids into the housing is more efficient, and component operating temperatures are lower.

The 180°F ambient test demonstrated that the drive transistors remain well within their operating temperature ceiling of 250°F at the worst-case ambient condition expected to be encountered in an advanced technology helicopter application. These component temperatures are approximately 20°F higher than those encountered on the Phase II unit due to an increase in control complexity and associated power dissipation.

3. Test Results

The results of the Phase I testing demonstrated that the thermal plane helps to spread the heat and thereby reduces the temperature rise of hot spots. Test data show a 13% improvement even without a braid connection between the thermal plane and housing. An additional 15% reduction in temperature rise was achieved with copper braids.

The Phase II testing of a fully assembled and operating system has generally confirmed the results of the PCB thermal plane evaluation testing. Figure 131, which shows the percentage improvement in component temperature rise, is also valid with a minor modification for a fully assembled control. Components which are farther removed from a braid (i.e., Q15 transistor) may approach the results of a close proximity device such as Q21 if a considerable portion of their temperature rise is attributable to other components rather than self-heating. A nearly 30% decrease in temperature rise was achieved with a 0.009 in. thermal plane in conjunction with 0.03 $in.^2$ copper braids fastened to the housing. To realize any significant additional reduction in component operating temperatures, a heat sink will be required.

4. Recommendations for Additional Testing

The tests conducted assumed a housing temperature of 175°F. Further reductions in component operating temperatures are possible by reducing the housing temperature by heat sink to the engine inlet air which is 135°F maximum. Additional tests are in order to determine the effectiveness of this heat sink approach and the overall impact on component operating temperatures.

D. FUEL PUMP INDUCER - CHANDLER EVANS INC.

1. Background

The pump specification for advanced technology helicopter engines requires operation at inlet conditions of 1.0 V/L and one psi NPSP. Current technology in fuel systems is for operation at 0.45 V/L and five psi NPSP. While tests have demonstrated the ability of fuel boost pumps to operate at conditions higher than 0.45 V/L, sustained operation at 1.0 V/L represents a significant advance of the state of the art. The additional requirements of high speed (65,000 rpm) and low flow (500 lb/hr) required for this pump indicated that a special program was required to develop inducers to meet these pump inlet requirements. The objectives of the test program were, therefore, to:

1. Develop an inducer capable of charging the high-speed main stage pump through the range of inlet temperatures and pressures required.

2. Demonstrate V/L = 1 and NPSP = 1 psi performance on a bar stock inducer test fixture.

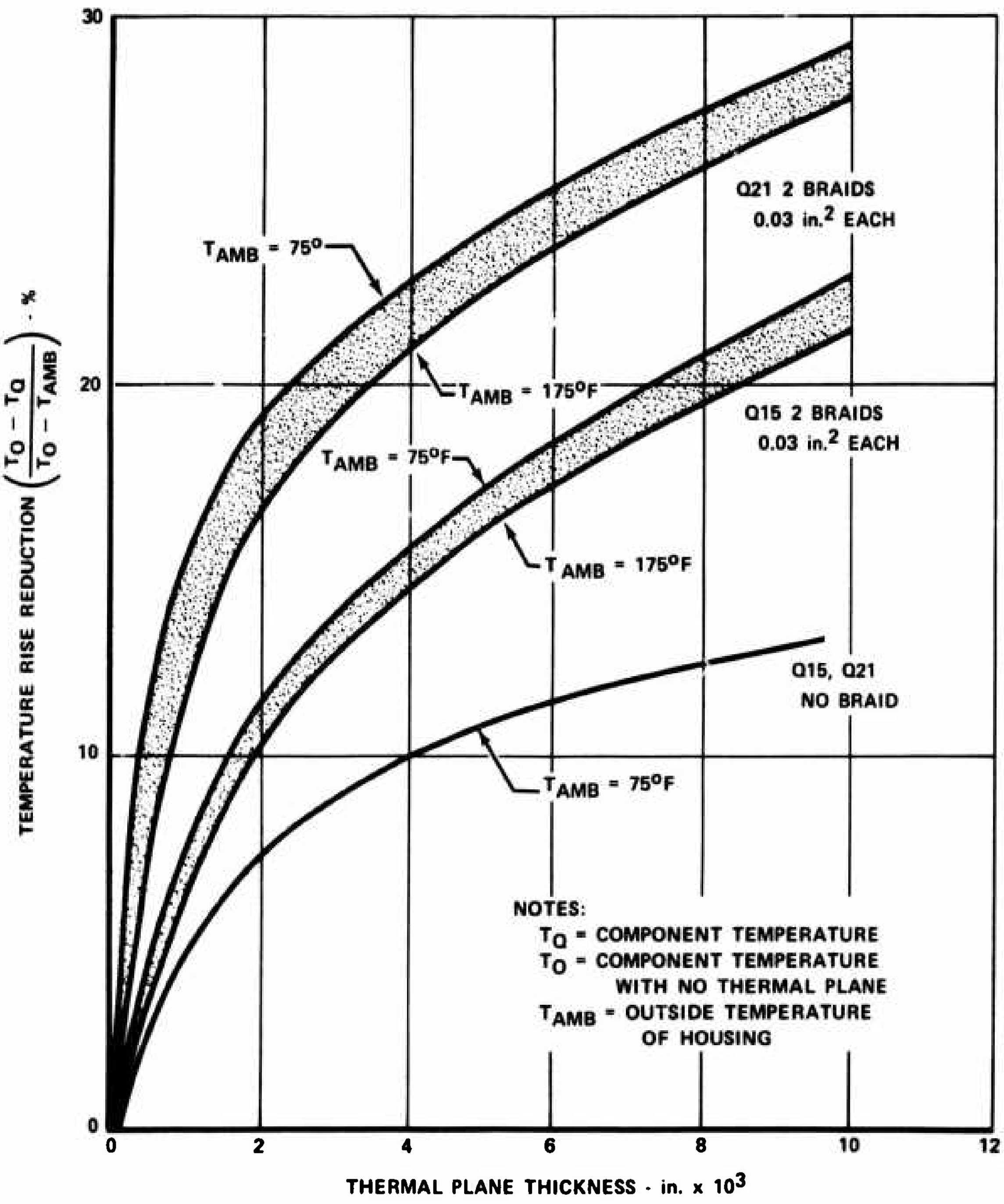


Figure 131. Percentage Reduction in Temperature
Rise vs Thermal Plane Thickness

A bar stock test fixture (Figures 132 and 133) designed to adapt to several impeller configurations was fabricated to evaluate the following:

1. A conical inducer
2. A jet inducer
3. An axial inducer
4. Combinations of the above.

Performance requirements and acceptance limits for performance capabilities were based on the main stage inlet requirements specified in Section VIII, that is, 500 lb/hr fuel flow at a pressure rise of 80 psi. The test evaluation of the hardware used an Ikor V/L meter which provides an instantaneous reading of V/L in the inlet line and does not rely on the lengthy calculations normally involved in the test procedure. The test setup conformed as nearly as possible with the ARP 492 V/L test procedure recommendations. In addition, each impeller, inducer, or combination was:

1. Calibrated to determine head-flow, ΔT, BEP, and NPSP capabilities

2. Tested to determine V/L limitation on 135°F JP-4 at sea level, 10,000 and 20,000 ft tank altitudes at three flow values.

Minor reworks to the basic designs were attempted to improve performance.

2. Test Program

The centrifugal boost impeller designed for use in an USAAMRDL pump to be tested on an advanced technology gas generator was used as a base for the tests performed. Design parameters for this impeller are summarized following.

Speed	65,000 rpm
Q	500 lb/hr
ΔP	80 psid
N_s	1,100
OD	0.583 in.
Inlet Diameter	0.234 in.
Blade Inlet Tip Angle	14.8 deg
Type	Conical
V/L Capability	V/L = 0.45
	Q = 225 lb/hr
	ΔP = 50 psid

The impeller was calibrated at a sea-level tank condition, using the test arrangement shown in Figure 134. The head/flow calibration data are illustrated in Figure 135. The V/L evaluation at various tank conditions is shown in Figures 136 through 138 and resulted in minimal V/L capability. The USAAMRDL impeller was modified as illustrated in Figure 139, which resulted in a cutback inlet. This configuration has a straight vane shape which is the standard method of treatment for conical impellers. The maximum speed head/flow calibration, with a sea-level tank, for the revised impeller is shown in Figure 140. As indicated in Figures 136 through 138, a V/L improvement does result with the cutback inlet, with a corresponding loss in available head rise.

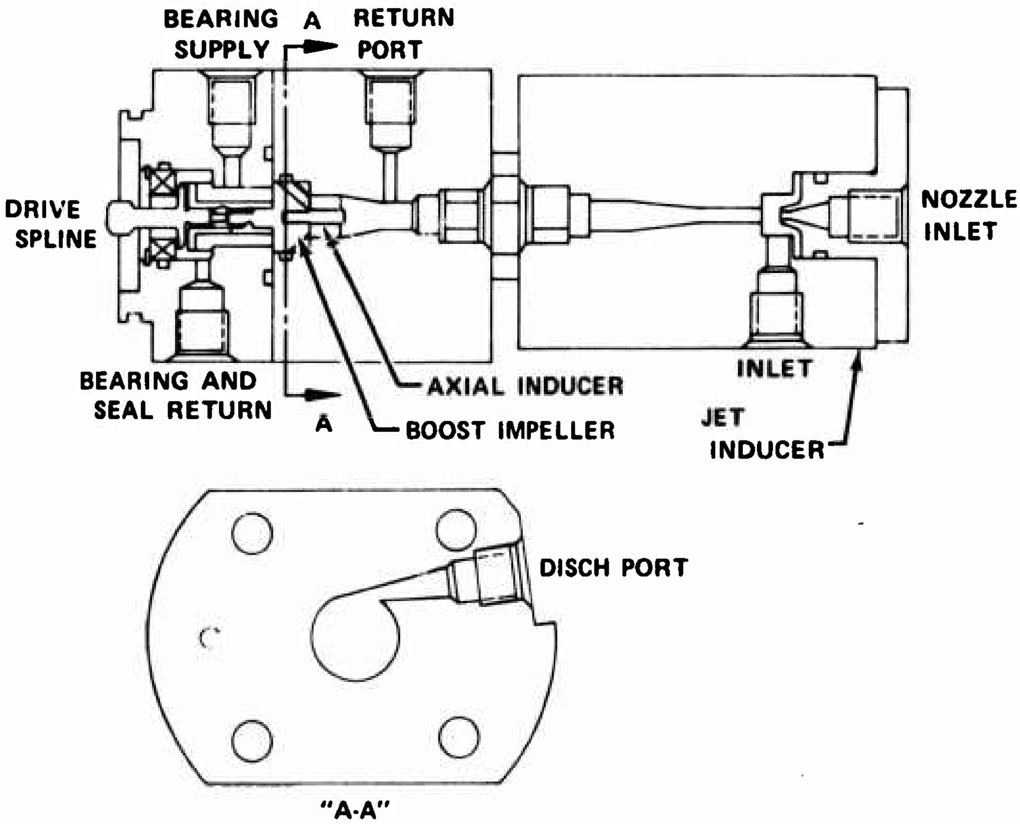


Figure 132. Inducer Test Rig Configuration

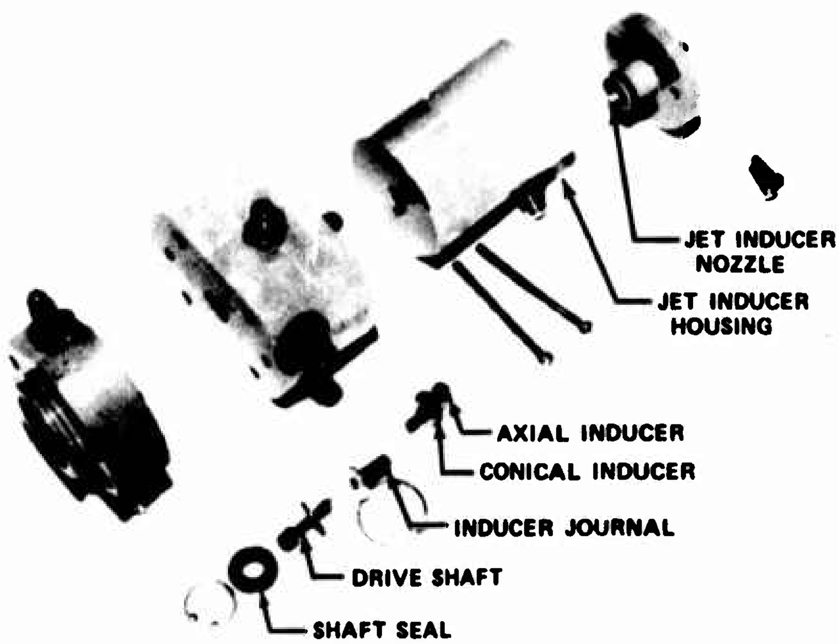
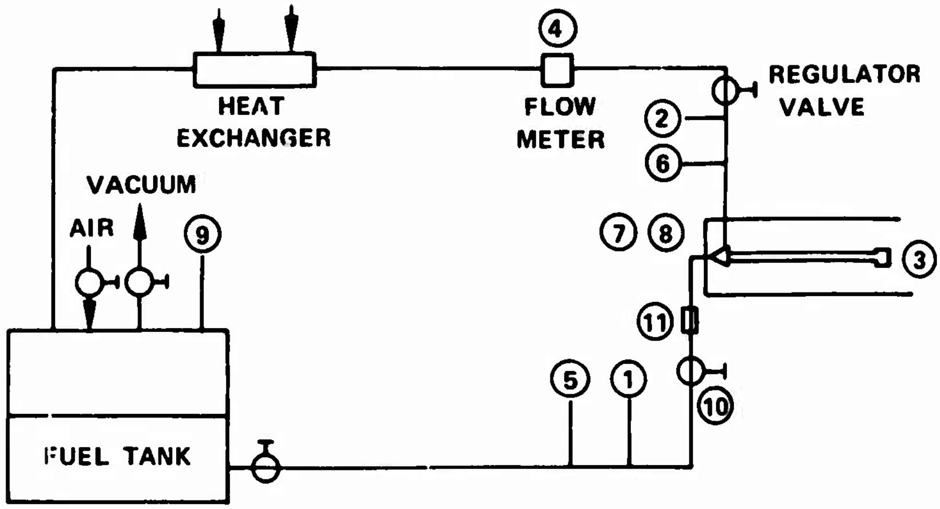


Figure 133. Inducer Pump Test Fixture



LEGEND

1. INLET PRESSURE
2. DISCHARGE PRESSURE
3. rpm
4. FUEL FLOW
5. FUEL INLET TEMPERATURE
6. FUEL DISCHARGE TEMPERATURE
7. AMBIENT TEMPERATURE
8. AMBIENT PRESSURE
9. FUEL TANK PRESSURE
10. V/L VALVE
11. V/L METER

Figure 134. Inducer Test Schematic

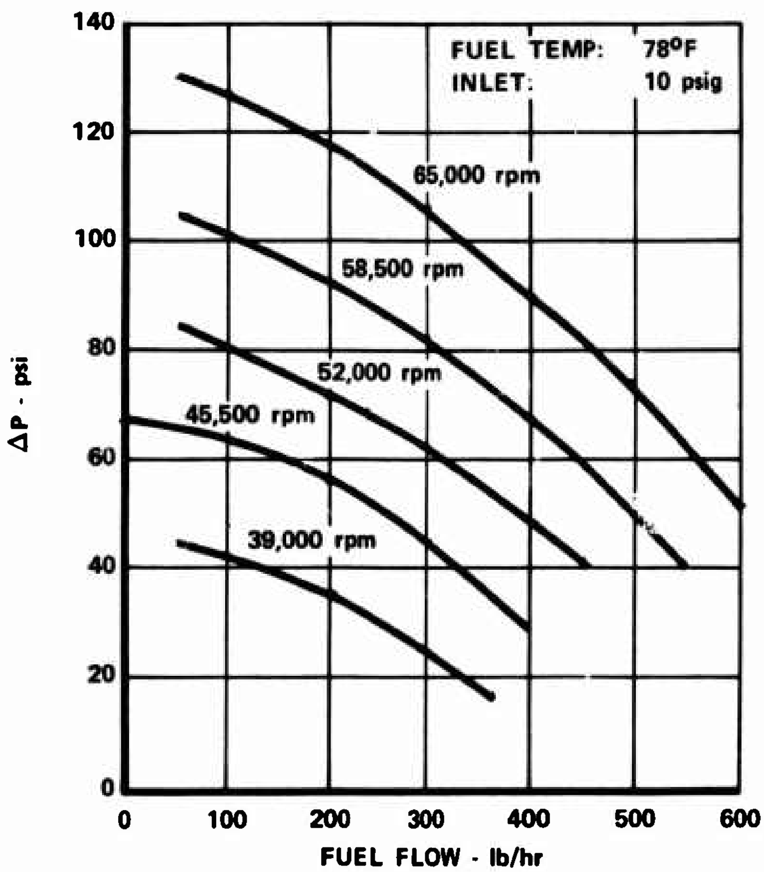


Figure 135. Inducer Head/Flow Calibration

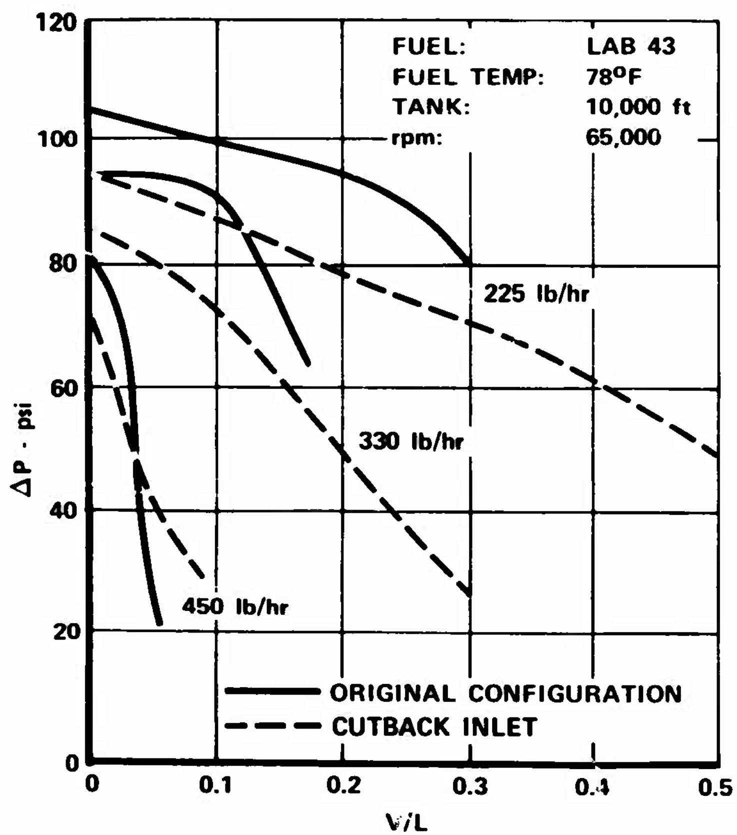


Figure 136. Inducer 10,000 ft V/L Performance

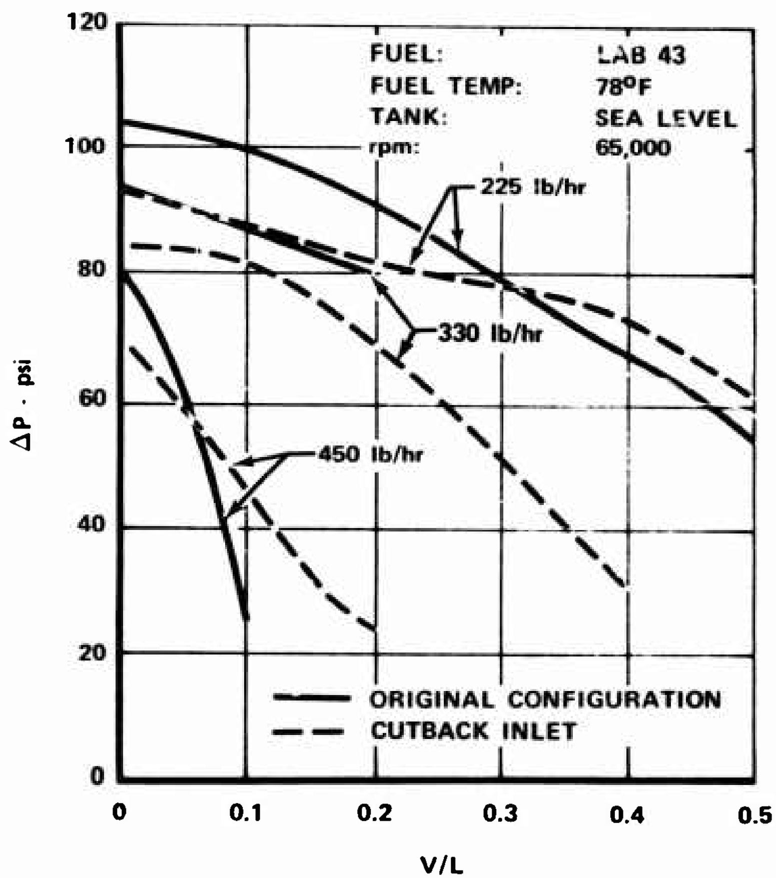


Figure 137. Inducer Sea Level V/L Performance

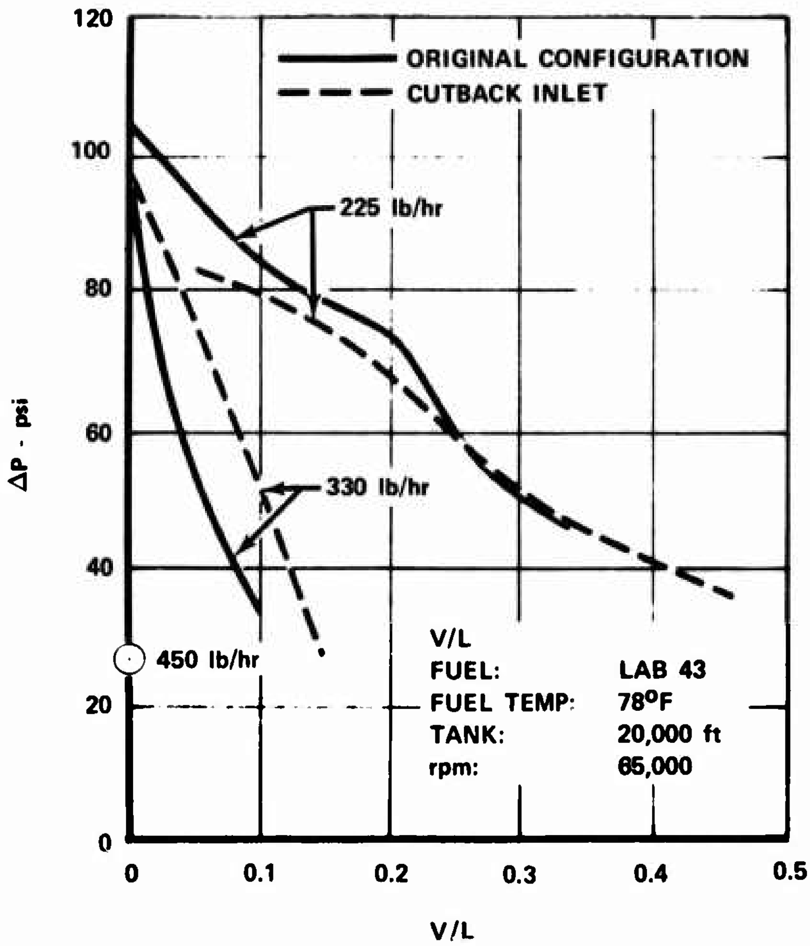


Figure 138. Inducer 20,000 ft V/L Performance

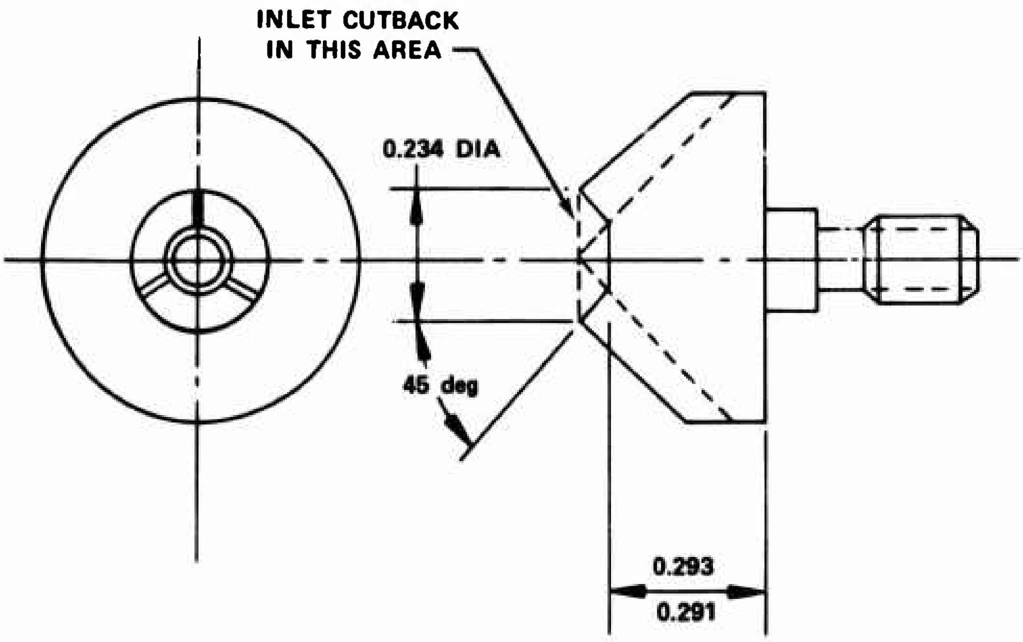


Figure 139. Cutback Inlet of the Inducer

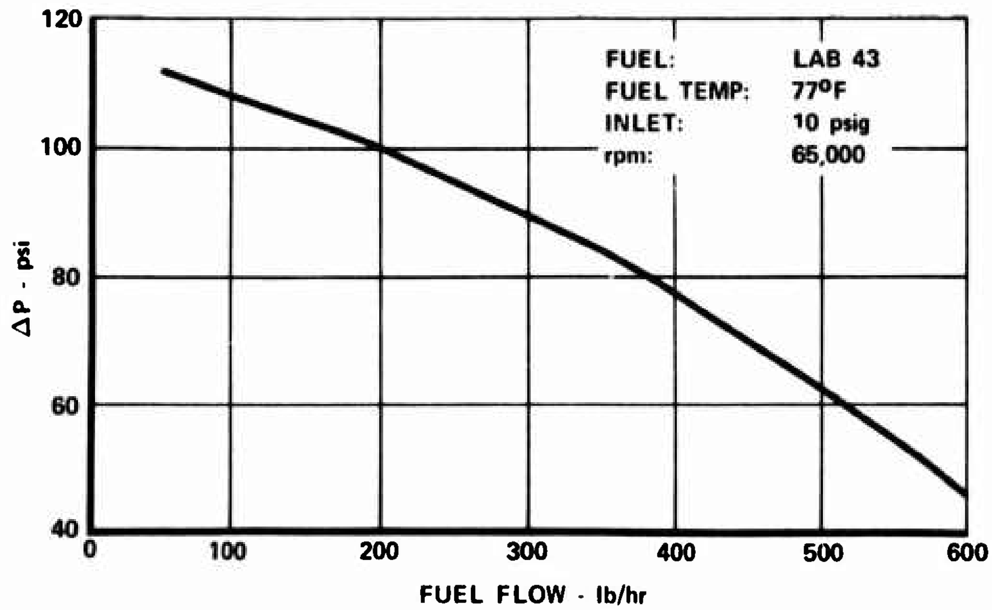


Figure 140. Inducer Head Flow Calibration With Cutback Inlet

The axial inducer tested during this program is described below:

Speed	65,000 rpm
Q	500 lb/hr
No. Blades	3
ΔP	6.2 psid
N_s	8,375
OD	0.375 in.
Inlet Angle	7.16 deg
Type	Flat Plate (Constant Lead)

Calibration of the axial inducer-conical inducer combination (Figure 141) showed a lower head rise than the uninduced conical impeller, an effect believed to be a result of increased leakage at the front shroud. Initial V/L testing of the combination, shown in Figures 142 through 144, demonstrated a loss in the performance (head rise) compared with the cutback inducer performance. Modification to the axial inducer inlet to provide sweepback to the inlet vane edge, and removal of the vanes on the trailing edge of the axial inducer, failed to provide any performance improvement of the axial/conical inducer combination. The use of an axial inducer was therefore abandoned in this program phase.

The final phase of testing incorporated a jet pump configured to run in combination with the cutback conical impeller. Design characteristics of the jet pump are defined below:

Pressure Ratio	0.46
Capacity Ratio	0.5
Nozzle Diameter	0.0559 in.
Mixing Tube Diameter	0.112 in.

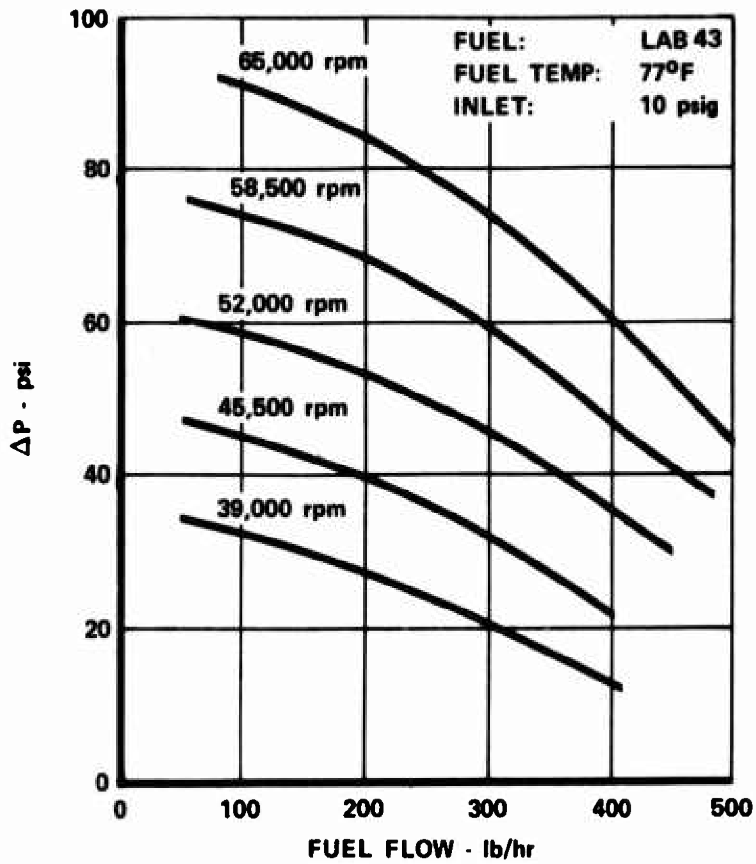


Figure 141. Axial/Conical Inducer Head vs Flow Calibration

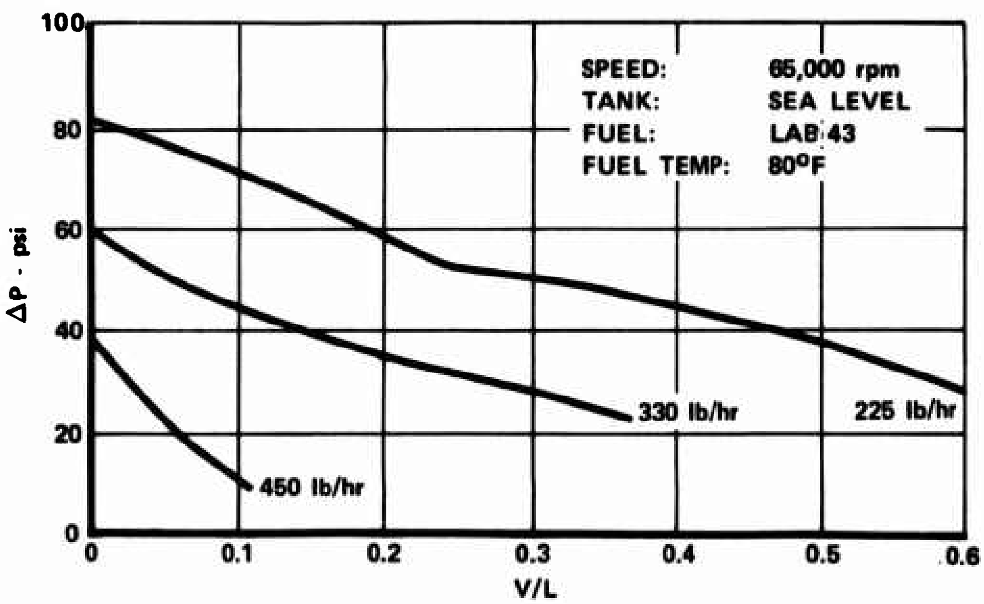


Figure 142. Axial/Conical Inducer Sea Level V/L Performance

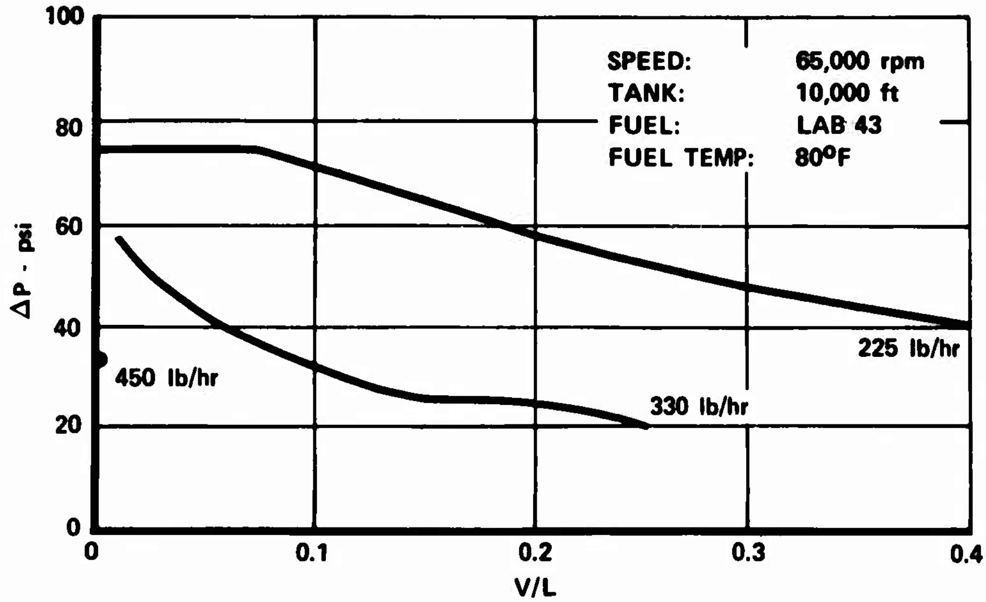


Figure 143. Axial/Conical Inducer 10,000 ft V/L Performance

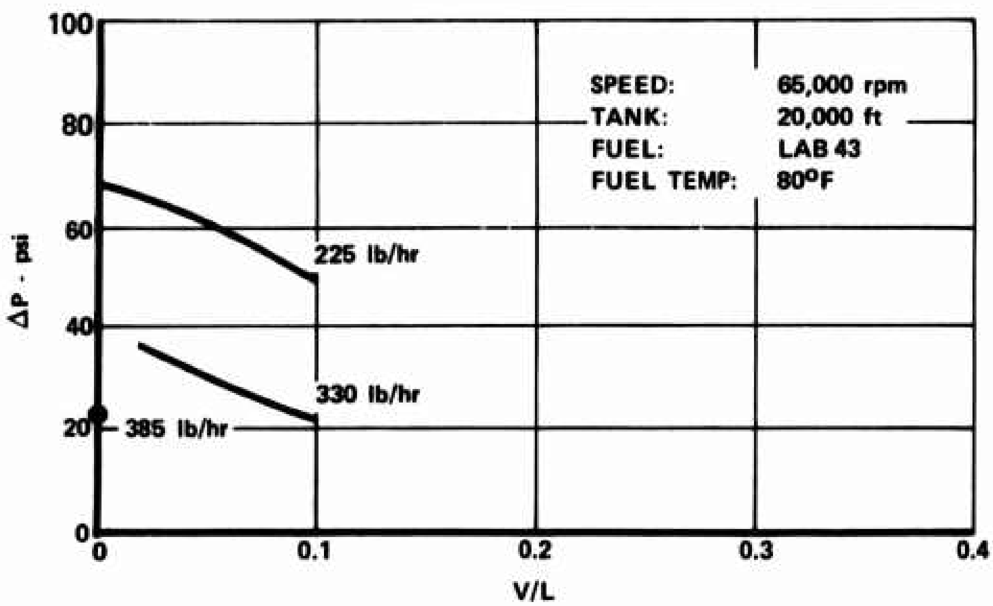


Figure 144. Axial/Conical Inducer 20,000 ft V/L Performance

A schematic of the jet-induced conical impeller test conditions at V/L = 1.0 is shown in Figure 145. Impeller flow rates and ΔP values are shown on the schematic. The V/L test results for the jet-induced conical impeller are defined in Figures 146 and 147. This combination has shown that, with the existing design configuration, the inducer arrangement is flow limited to 60 lb/hr at maximum speed and a V/L ratio of one. A V/L = 1.0 was also demonstrated at 70% speed where output flow was limited to 30 lb/hr. This limited flow capability results because the USAAMRDL conical inducer used throughout the test series was not sized for use with a jet pump and cannot provide required output flow and head while providing motive flow to the jet pump. To provide required boost stage output flow at a head rise sufficient to charge the main pumping element, a higher capacity conical impeller and a larger jet inducer would be necessary. These design considerations are discussed in subsequent paragraphs.

V/L tests are conducted by throttling the fluid below its vapor pressure to induce the formation of vapor and therefore a fluid/vapor mixture. Upon entering the pump element, the vapor is dissolved back into the liquid as the pressure is increased. These tests simulate a local restriction in the fuel line upstream of the pump inlet. Another form of V/L operation is sometimes encountered when there is an air leak into the pump inlet line. Under suction inlet fuel conditions, this results in an air/fuel mixture which behaves differently. The air remains present throughout the pumping process, although the relative V/L decreases as the fuel pump pressure increases. Since these V/L tests represent two different phenomena, air/fuel tests were also conducted on the inducer system.

Testing of the jet-induced conical impeller using an air bleed into the pump inlet resulted in sudden loss of prime in the pump inlet. Pump performance was observed to be satisfactory as V/L was increased until the point of failure, which occurred from 0.5 V/L to 0.75 V/L operating in the range of flows at which the unit had demonstrated 1.0 V/L performance using the induced line loss method of test.

A performance curve showing pump ΔP vs V/L is presented on Figure 148. These data show that the pumping system was maintaining head rise up to the point of performance breakdown.

The reasons for this sudden failure are not fully understood but are believed to be:

1. Poor control of the amount of air being introduced into the line may have resulted in instantaneous excessive V/L values.

2. The fuel may have the capacity to absorb small amounts of air but larger amounts may more severely impact pump performance.

Introduction of a jet inducer into the pumping scheme has a significant effect on the ability of the system to operate with low inlet fuel flowrates at V/L conditions. With two-phase flow at inlet velocities below 3 ft/sec, stable flow is not maintained in the inlet line. This low flow condition consists of a stratified plug, or slug, flow on which an impeller alone cannot maintain stable performance.

SCHEMATICS SHOWING FLOW CONDITIONS
AT
V/L = 1.0 CONDITIONS

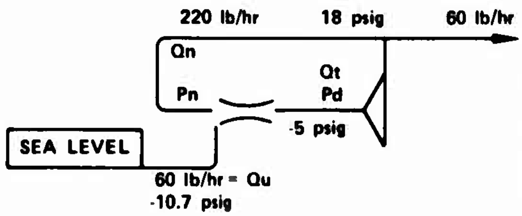
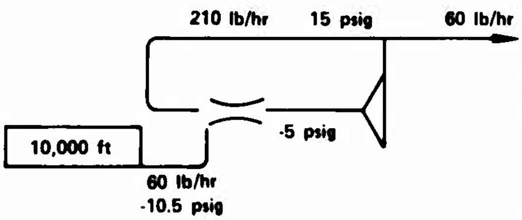
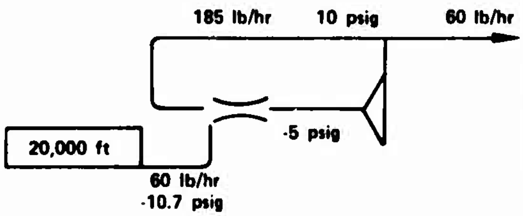
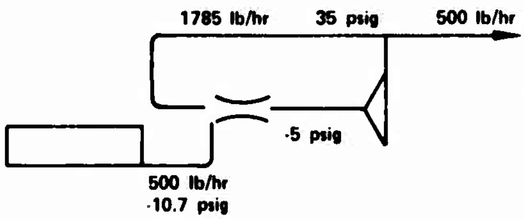
CONDITION		PRESSURE RATIO $\frac{Pd\text{-}Pin}{Pn\text{-}Pd}$	CAPACITY RATIO $\frac{Qu}{Qn}$	% NOZZLE FLOW $\frac{Qn}{Qt}$
S L TANK V/L = 1.0 65,000 rpm		0.239	0.272	78.5
10,000 ft TANK V/L = 1.0 65,000 rpm		0.275	0.285	77.7
20,000 ft TANK V/L = 0.95 65,000 rpm		0.38	0.324	75.5
DESIGN CONDITIONS		0.28	0.28	78.0

Figure 145. Conical With Jet Inducer Test Conditions

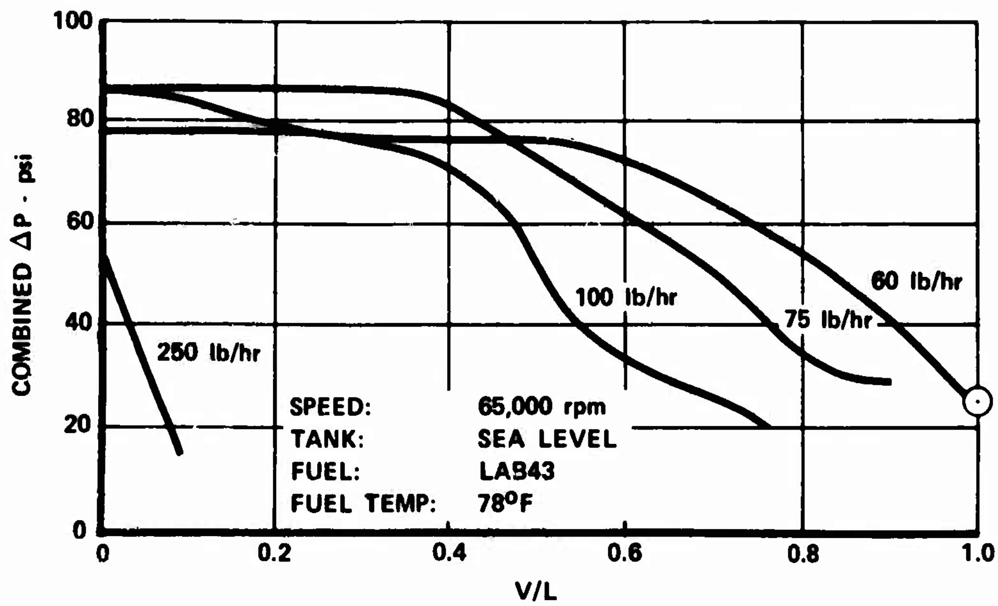


Figure 146. Conical With Jet Inducer Sea Level V/L Performance

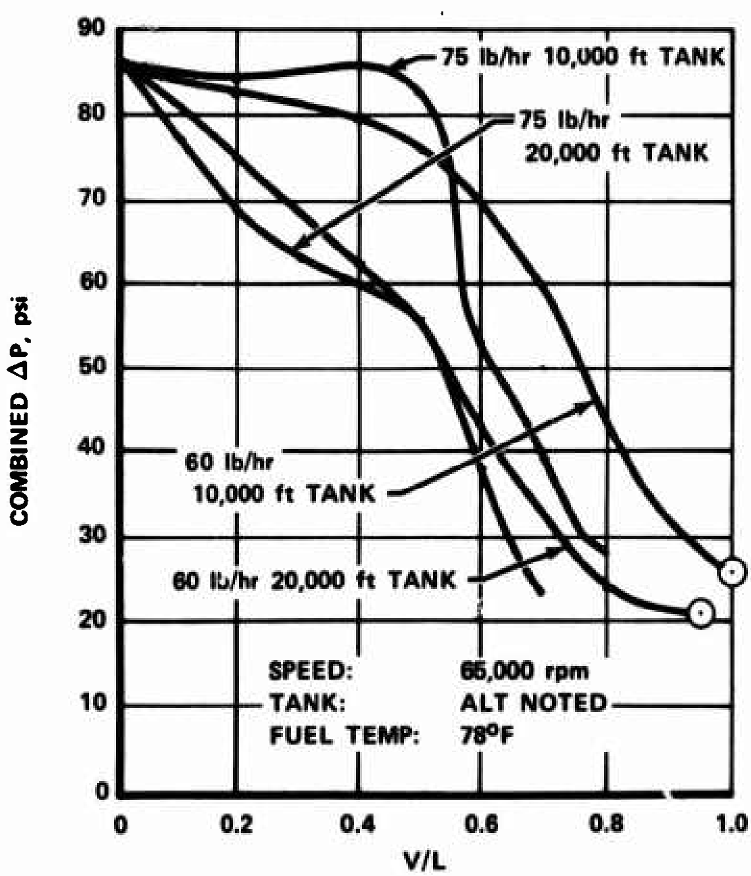


Figure 147. Conical With Jet Inducer 10,000 ft and 20,000 ft V/L Performance

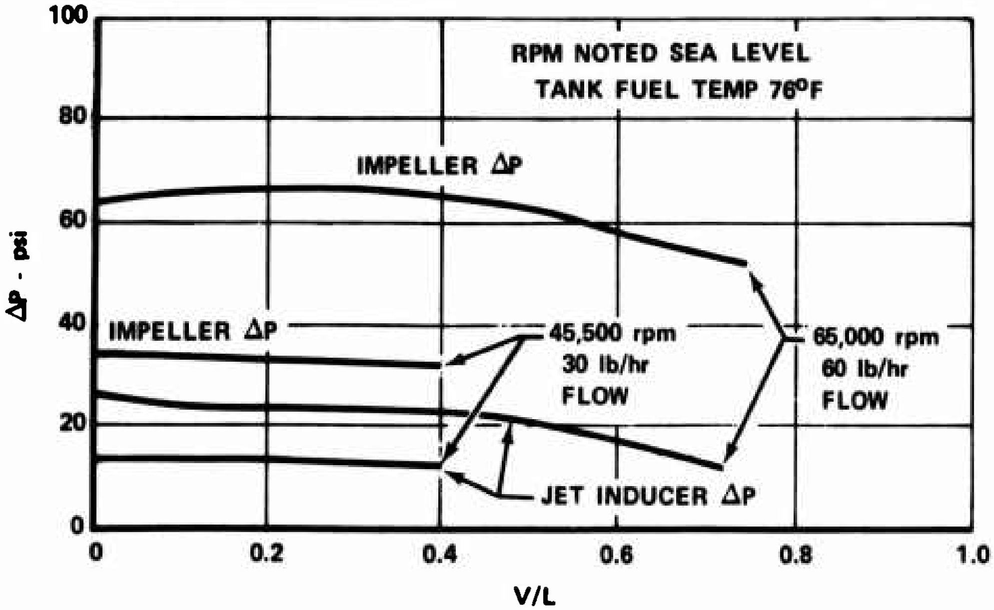


Figure 148. Conical With Jet Inducer V/L Performance With Direct Air Induction

With the jet inducer in the system, a relatively high rate of flow through the impeller and nozzle loop is maintained, and the mixing action in the jet makes the system insensitive to the type of flow in the inlet pipe.

3. Proposed Redesign

The demonstrated ability of the jet-induced conical impeller to operate with 1.0 V/L and with TVP +1 psi at low flowrates provides a baseline from which to design a larger system which will have the capability of meeting the flows required for the advanced technology helicopter engine. This redesign proposes a higher capacity mixed flow impeller and a resized jet inducer.

Test conditions at which the test hardware demonstrated V/L = 1 capability are as follows (Reference Figure 145):

Pressure Ratio	0.239 to 0.38
Capacity Ratio	0.272 to 0.324
Percent Nozzle Flow	75.5 to 78.5
Area Ratio	0.25

From curves of conical inducer at 0.45 V/L, we can conservatively say that at 0.45 V/L, ΔH = 50% noncavitating head at flows less than 50% of design flow.

To design for 1.0 V/L at 40 psi maximum nozzle ΔP and 500 lb/hr useful flow, the jet inducer modeled on the above data gives the following:

Pressure Ratio	0.28
Capacity Ratio	0.28
Percent Nozzle Flow	77
Area Ratio	0.2

Then,

$$Q_t = \frac{500}{0.28} = 1785$$

$$Q_n = 1285 \text{ lb/hr}$$

The impeller requirements then become:

ΔP	85 psid
Q	2,280 pph
rpm	65,000
N_s	2,193

Using the conical type USAAMRDL impeller as a model, this results in an impeller OD of 0.78 in. and a shutoff pressure of 220 psi. The high specific speed involved has shown the conical impeller to be a poor choice for the higher flow required due to the high shutoff head. This high shutoff head results in excessive ΔP across the inducer nozzle and, therefore, high nozzle flows at low flow conditions. To obtain a flatter head-flow curve, it is necessary to use a mixed flow type impeller which has been developed for use on the MFP 330 and other high flow CECO pumps. This impeller develops a flatter head flow curve and, therefore, will produce less pressure rise at low flows. It also has the potential of higher V/L capability due to the favorable inlet configuration.

A mixed flow impeller to meet the requirements specified above would have an OD of 0.56 in. and a shutoff pressure of 110 psi based on MFP 330 impeller performance. Performance is estimated on Figure 149 and a comparative cross section is shown in Figure 150.

Power requirements for this higher specific speed inducer (N_s = 2200) using 23% overall efficiency based on the Worthington curves calculate as follows:

$$HP = \frac{85 \times 2280}{1714 \ (0.23) \ 375} = 1.310$$

Figure 151 represents the predicted fuel pump total temperature rise with an inducer sized for V/L = 1.0. Assuming a 350°F maximum fuel temperature limit and a 135°F maximum fuel inlet temperature, a temperature rise of 215°F is allowed. An evaluation of the engine operating map showed that this temperature rise limit is approached only at the worst-case operating point of flight idle at 20,000 ft altitude. The fuel temperature rise at intermediate power level is predicted to be below 100°F over the entire operating envelope.

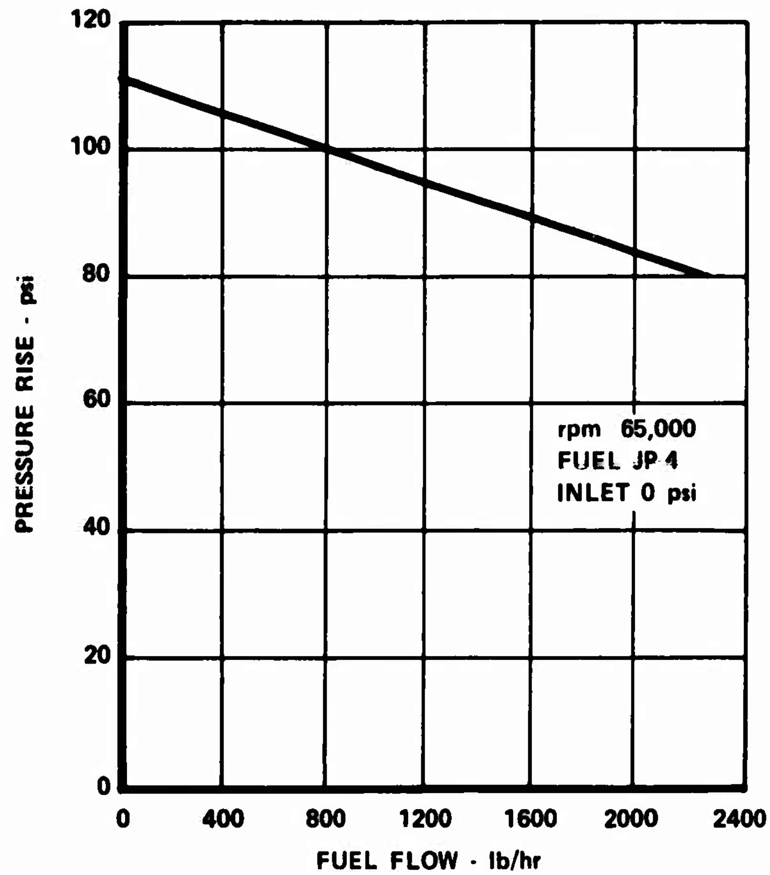
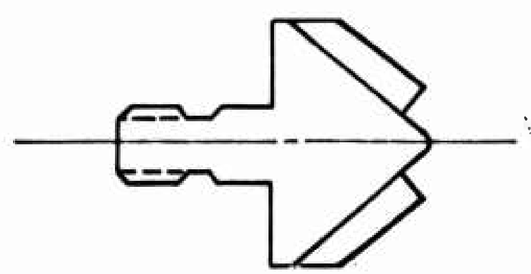
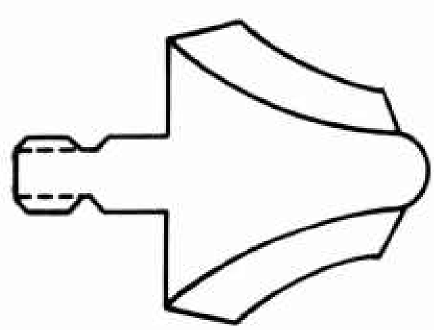


Figure 149. Mixed Flow Impeller - Estimated Head vs Flow Performance



USAAMRDL CONICAL INDUCER



PROPOSED MIXED FLOW INDUCER

Figure 150. Comparison of Boost Inducer Cross Sections

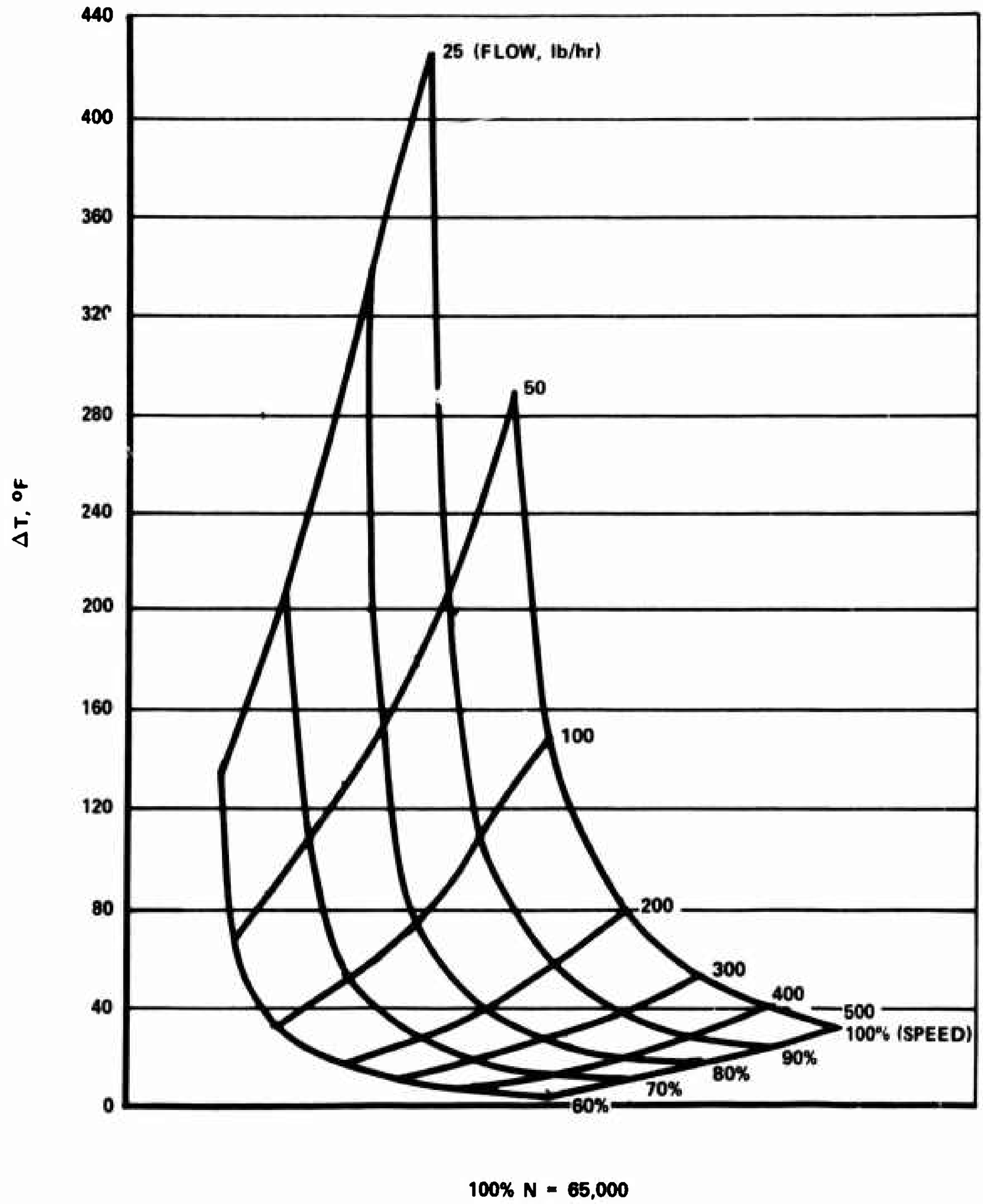


Figure 151. Predicted Fuel Pump Total Temperature Rise

4. Test Results

The test results obtained from the fuel pump inducer test program are summarized below:

1. The jet inducer and conical impeller combination was capable of low flow operation (60 lb/hr at 65,000 rpm and 30 lb/hr at 45,500 rpm) with V/L = 1.0 and/or 1.0 psi above TVP at the pump inlet. A redesign of the pumping element is required to meet the V/L specification over the entire flow range. The flowrates at which the tests were successfully completed show that an increase in capacity of 8 to 1 is required.

2. Performance of the conical inducer could be improved by redesign of the inlet area. The "cutback" rework improved performance, showing that a blockage or mismatch condition exists in this area.

3. Performance of the axial inducer was significantly less than the predicted performance based on test data for larger units. Further development in this area would be required to determine the specific design problem associated with this inducer.

4. Operation of the jet-induced conical impeller at 1.0 V/L obtained by air induction into the inlet line was not possible due to sudden loss of prime experienced during all tests of this type which were conducted. It was impossible to determine maximum V/L obtainable due to inconsistent results.

5. The increased impeller capacity required to meet the flow and pressure requirements of the jet inducer extend the capabilities of the conical type impeller beyond the normal range of usefulness. A mixed flow type of impeller would be a more suitable choice. This design has the potential advantages of increased overall efficiency and improved vapor handling capability, both of which are required for this application. Since this mixed flow impeller cannot be machined by conventional methods but must be cast from specially prepared patterns, this may require a development effort due to the small size. The cost of producing this type of impeller is many times the cost of a conical impeller in the small quantities required for a development effort.

6. The predicted fuel temperature rise characteristics with the jet pump/centrifugal element pumping configuration are within acceptable limits for the application.

7. The increased capacity impeller and jet inducer will not substantially increase the basic size of the pump package.

5. Recommendations for Additional Testing

1. Additional testing of the recommended jet pump/conical inducer configuration is in order to verify the design approach and to document the predicted fuel temperature rise.

2. The high-speed centrifugal pump elements tested have demonstrated applicability for advanced engines from a predicted performance standpoint. Endurance tests of the components on contaminated fuel are in order to further verify applicability from a mechanical design endurance standpoint.

E. POWER TURBINE OVERSPEED GOVERNOR - CHANDLER EVANS INC.

1. Background

Free turbine engines require a fast-response overspeed governor for protection to prevent runaway speed. Also, in the cases where the engine control system is electronic, it is advantageous to have a power turbine overspeed governor that is redundant and operates independent of the electronic control. Furthermore, it is desirable to sense power turbine speed directly to eliminate stepdown gearing. Conventional mechanical speed sensors operate in the 5,000-rpm speed range at 250°F, whereas in this sytem rated power turbine speed is 36,000 rpm and ambient temperature is 800°F.

The power turbine overspeed governor described in Section VII and shown schematically in Figure 152 comprises a sensor which is an integral part of the engine power turbine shaft, and a governor mounted on the fuel control. The speed sensor is formed by a set of closely spaced cantilevers arranged in a circle around the power turbine shaft. As the power turbine shaft speed increases, the centrifugal force causes the cantilevers to move radially outward. The cantilever motion is used to vary an orifice and thereby generate a pneumatic pressure signal. An increase in power turbine speed causes an increase in air pressure in the orifice line. The increase in air pressure is sensed in the fuel control by a bellows. Movement of the bellows is amplified by a follow-up flapper servo system which operates to reduce the metering head across the fuel metering valve. Hence, a turbine overspeed causes a proportional reduction in fuel flow. The point at which the overspeed governor cuts in is dependent on the spring preload acting on the sensing bellows in the fuel control.

Critical items in the operation of this governor are the speed sensor and the dynamic response of the governor. A speed sensor of this configuration has previously been tested to 20,000 rpm. For this system the sensor has to be designed to operate at 45,000 rpm (125% power turbine speed) in surrounding temperatures up to approximately 800°F. The required response from overspeed to reduction in fuel flow is on the order of 50 ms.

2. Test Program

Only the speed sensor and pressure-sensing bellows parts of the overspeed governor were developed because the remaining parts of the system are of conventional hardware and their performance can be determined from previous experience.

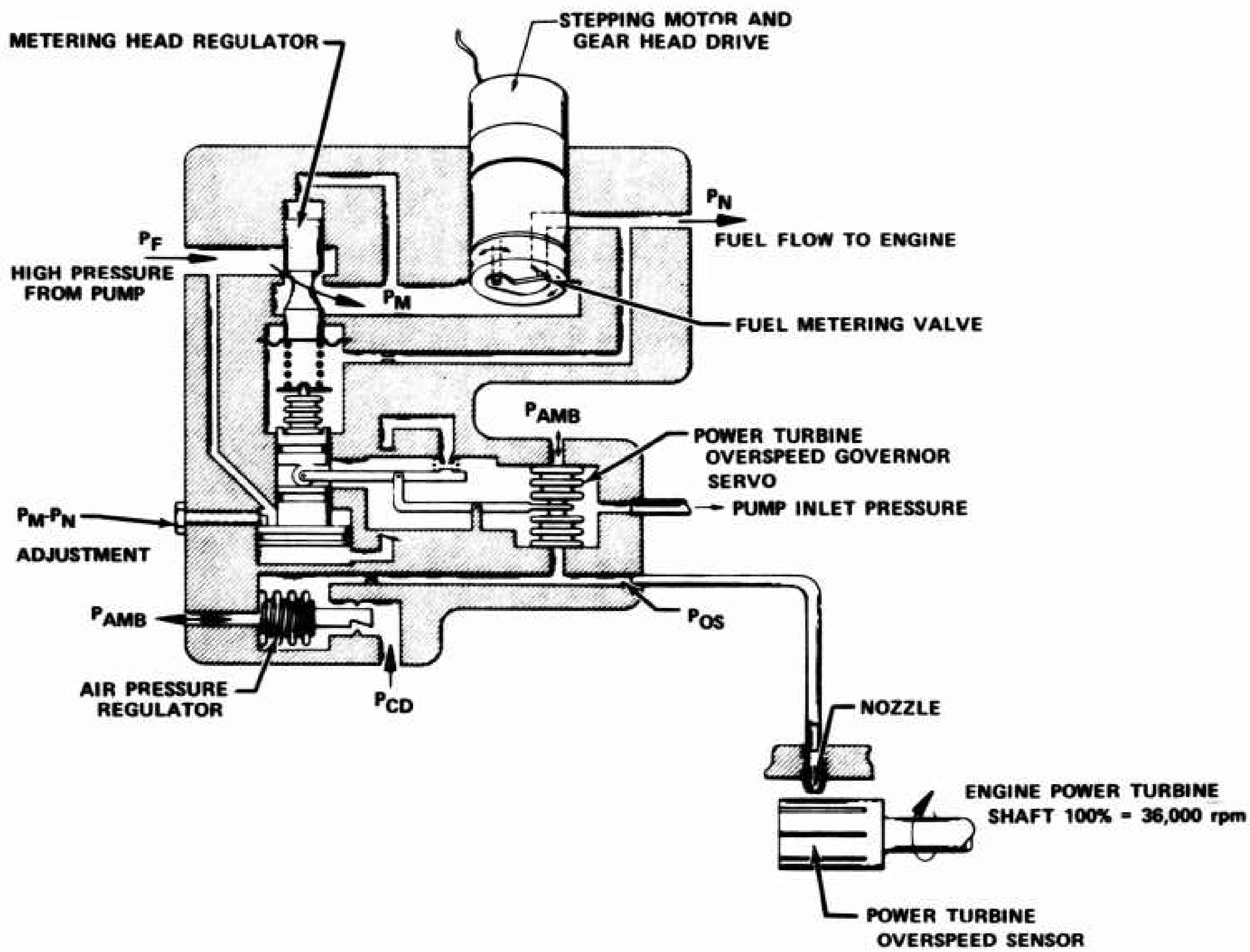


Figure 152. Power Turbine Overspeed Governor

The objectives of the program are to demonstrate the performance and durability (in an engine simulated environment) of the speed sensor and to confirm the dynamic response capability of the overspeed governor.

The test installation is shown schematically in Figure 153. The overspeed sensor is shown enclosed with a heater element in an asbestos box wherein an 800°F ambient temperature is maintained to simulate the engine environment. The test series outlined for the overspeed governor is described below.

a. Room Temperature Calibration

Sensor cantilever displacement is calibrated against drive speed and bellows stroke. Cantilever displacement is measured with a proximity indicator. Bellows displacement is measured with an LVDT.

b. Ambient Temperature Effects

Variation in ambient temperature from room temperature to 800°F is assessed during the calibration procedure described in (a) previously.

c. Transient Response

A critical requirement of the overspeed governor is that the governor be fast enough in reducing fuel flow to avoid destruction of the power turbine disk. This determines that the response from speed increase to bellows stroke be faster than 50 ms. The transient response of power turbine speed to bellows stroke is determined by providing a step change in cantilever displacement and measuring the bellows stroke with an LVDT. The length and diameter of the air line from the receiver nozzle to the sensor bellows are sized to simulate the actual engine installation.

d. Endurance Testing

The governor endurance is demonstrated by cycling the governor for 50 hr about a nominal speed of 36,000 rpm at 800°F ambient temperature. Calibration at 10-hr intervals determines degradation in performance.

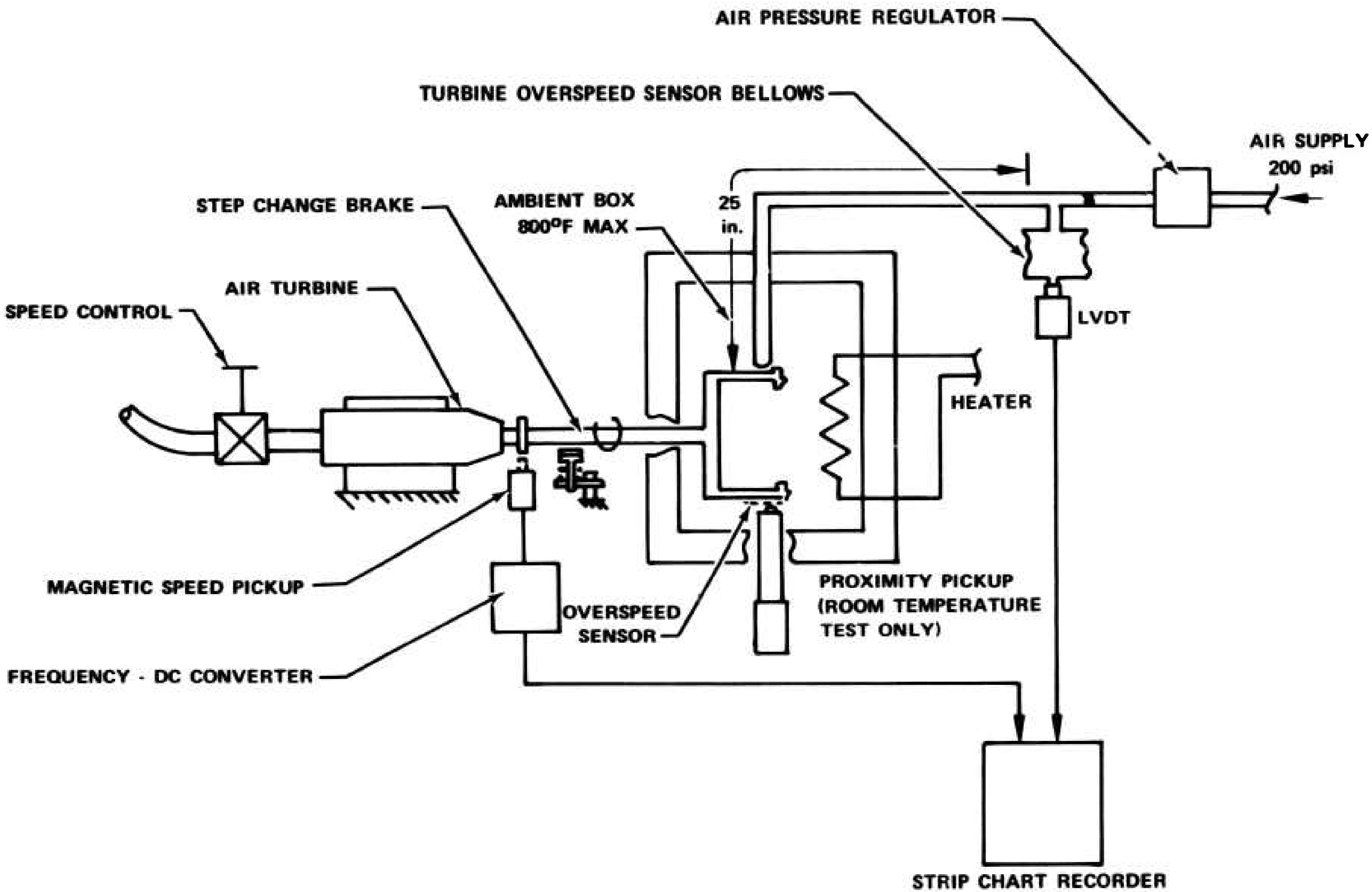


Figure 153. Power Turbine Overspeed Sensor Test Setup

3. Testing

Considerable testing was done to develop a suitable drive and sensor shaft system before a satisfactory measurement of cantilever deflection versus sensor speed could be made. A high-speed air turbine was chosen to drive the sensor, which was supported in carbon bearings lubricated with a drip oil system. The air turbine did not have sufficient power to drive this sensor to 45,000 rpm due to drive alignment problems. A second sensor was manufactured, integrating the air turbine and sensor shaft (Figure 154). This drive obtained 30,000 rpm, at which speed the shaft went into a critical vibration mode. A critical speed analysis was made on the shaft design, and it was determined that the shaft had to be shortened by 1.25 in. to avoid vibrations. A third sensor shaft was built with shortened length. The reduced length necessitated breaking into the side of the air turbine to measure the sensor speed. The third sensor failed because one of the cantilever slots was curved at the root of the cantilever. This caused this particular cantilever to deflect more than the others and to hit the sensor nozzle. The cantilever slots are difficult to machine because of their width (0.008 in. - 0.010 in.) and the poor machineability of Inconel 718 (AMS 5663).

A fourth sensor shaft was made and successfully run to 45,000 rpm. However, deflections were measured only to 43,000 rpm, at which speed another failure occurred. Deflection was measured by applying a voltage across the sensor shaft to a micrometer with a light in series. At low speeds (<30,000 rpm), as the cantilevers became very close to the micrometer (a few ten thousandths of an inch), a small spark was produced between the sensor and the micrometer, before mechanical contact was made, to light the bulb. At sensor speeds greater than 30,000 rpm, no spark was visible; the deflection was measured by observing the light. However, at 43,000 rpm, the cantilever made contact with the micrometer and caused the sensor to fail.

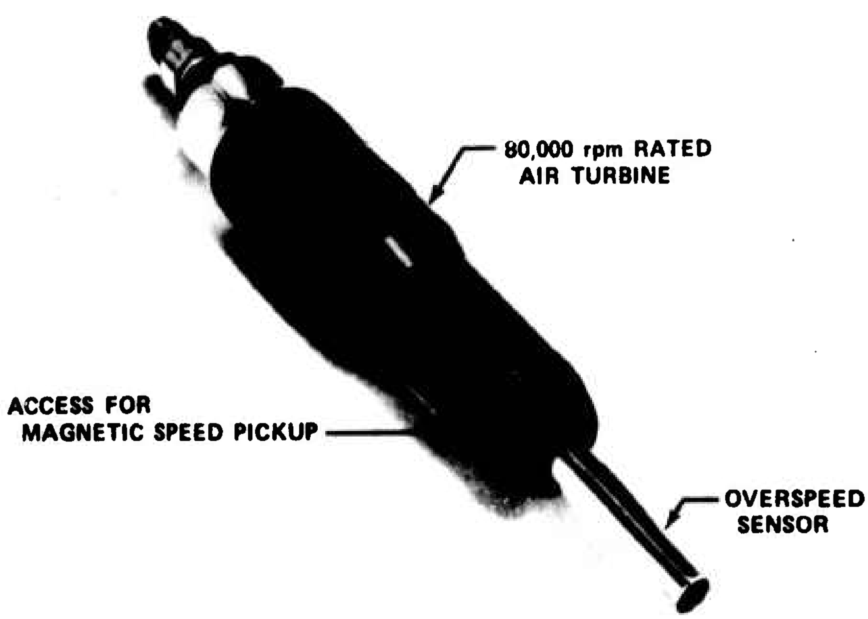


Figure 154. Power Turbine Overspeed Sensor and Test Drive

A fifth sensor shaft was fabricated, and a successful way of determining cantilever deflection was developed as follows. A shallow angled cone was forced axially into the sensor shaft, displacing the cantilevers outward. The radial deflection was measured with a dial indicator. The dial indicator was then replaced with a proximity pickup and the tests were repeated. From the relationship of cone axial position and radial displacement and cone position and proximity pickup output voltage, radial displacement versus pickup voltage was determined. Hence, the radial deflection was determined by measuring proximity pickup output voltage.

As the sensor shaft rotates, the air gap between the cantilevers passes under the proximity pickup, causing the proximity voltage to drop; therefore, a circuit was designed to pass only the highest voltage as a readout of cantilever deflection.

The test results of cantilever deflection versus sensor speed compared to theoretical predictions are shown in Figure 155. It can be seen that the test results follow theoretical predictions very closely.

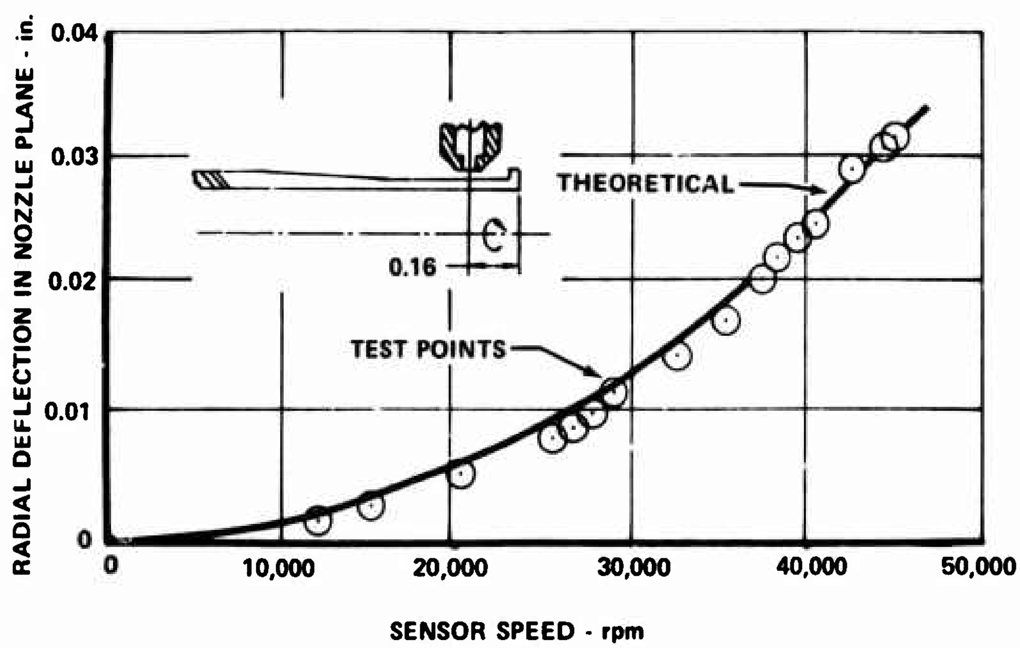


Figure 155. Overspeed Sensor Room-Temperature Calibration

A test of the pressure gain of the sensor nozzle system is shown on Figure 156. Over the operating range of the governor, there is a good correlation with the computed design performance. At $A_2/A_1 > 1.5$, actual test results give higher values of P_{os}/P_{reg} because of pressure losses in the line from the sensor bellows to the air nozzle. The line bore was minimized (0.1 dia) to provide a fast response for the governor. The performance of the governor is not affected by the higher values of P_{os}/P_{reg} because it is not operating in this range.

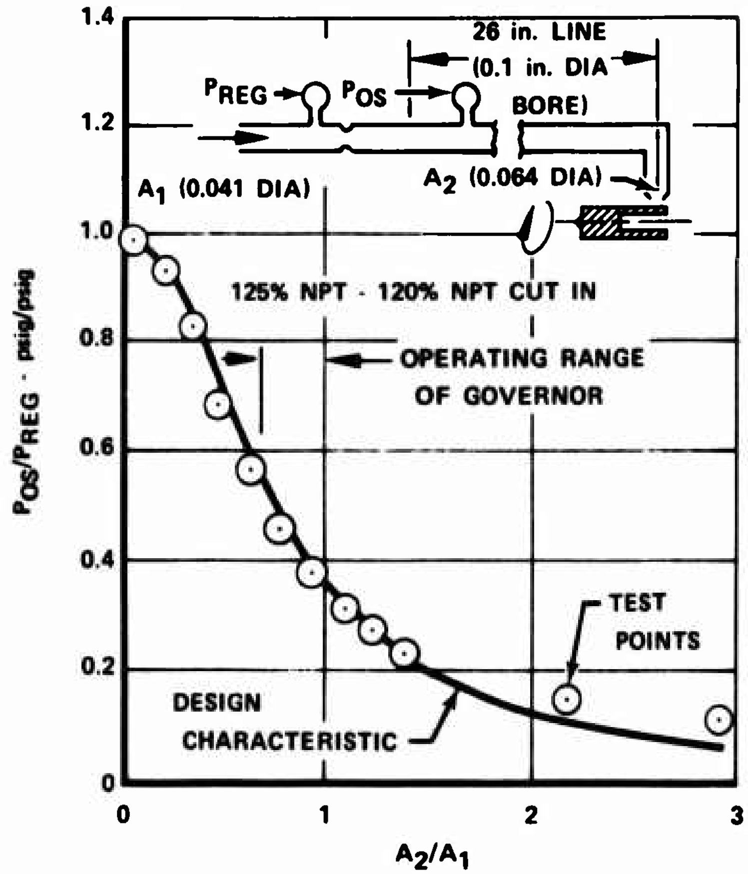


Figure 156. Pneumatic Characteristic of Power Turbine Overspeed Governor

Test data of overspeed pressure (P_{OS}) versus sensor speed is shown in Figure 157. The tests were first performed statically by mechanically forcing the cantilevers to deflect the amount required at different sensor speeds. The tests were then performed dynamically and show a close agreement with the statically determined curve, because the frequency at which the air gaps (between the cantilevers) pass the nozzle is much faster than time constant of the pneumatic system.

The effect of sensor ambient temperature on overspeed pressure (P_{OS}) is also shown in Figure 157 for an ambient temperature of 650°F. It can be seen that the effect of increasing the sensor ambient temperature is to increase the overspeed pressure P_{OS}. The reason for this shift is the reduction in Young's modulus of the sensor cantilever material; a reduction in modulus causes an increase in cantilever deflection for a given sensor speed. The change in P_{OS} would change governor cut-in speed 1% for a 100°F change in ambient temperature.

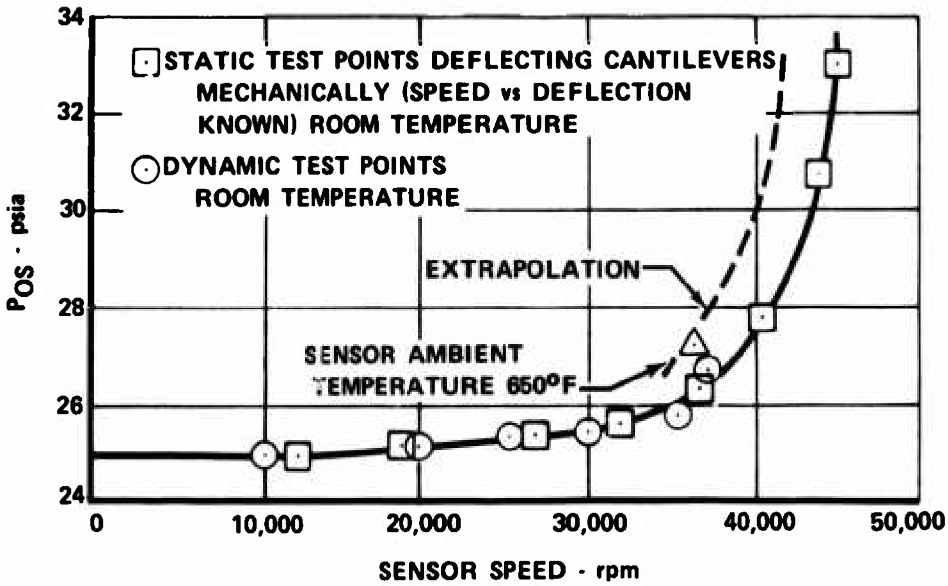


Figure 157. Variation In Power Turbine Governor Pressure Gain With Ambient Temperature

The choice of the governor cut-in point is critical because it has to be within a small margin between the maximum operating point of the engine (119%) and the rotor burst speed (137%). Therefore, it is necessary to provide sensor ambient temperature compensation. This may be possible by selective use of material thermal expansion properties in the area of the sensor nozzle.

The transient response of the overspeed governor was performed by simulating a step change in cantilever deflection and measuring the bellows stroke with an LVDT. The step change in cantilever stroke was simulated by a quick change in a micrometer axial position against the sensor nozzle. The LVDT voltage with respect to time was recorded on a strip chart recorder. The results of the transient test are shown on Figure 158. The tests were done at different levels of overspeed pressure P_{OS} to show that the time response is independent of sensor speed. The equivalent sensor speed is indicated on the traces. The measured time constant from change in cantilever position to sensor bellows stroke is on the order of 0.04 sec. The specified transient response is 0.05 sec from a turbine overspeed to reduction in fuel flow; therefore, the response of the sensor is satisfactory provided that it is incorporated in a system with fast response from bellows stroke to reduction in fuel flow.

High ambient temperature endurance testing was performed on the sensor. The test setup is shown on Figure 153. Initial tests were performed with an ambient temperature of 650°F. The sensor was run at 36,000 rpm, and after 10 hours of operation the cantilevers failed. The failed section is shown on Figure 159. The cantilever section was designed for unlimited life provided that the cycle stress was low; however, Figure 159 shows that stress concentrations existed at the corner point of the section. This problem could be eliminated by making the cantilever slots wider and providing a radius at the root of cantilevers; also, shot peening the sides of the cantilevers to remove the sharp edges of the cantilevers would increase the fatigue strength considerably.

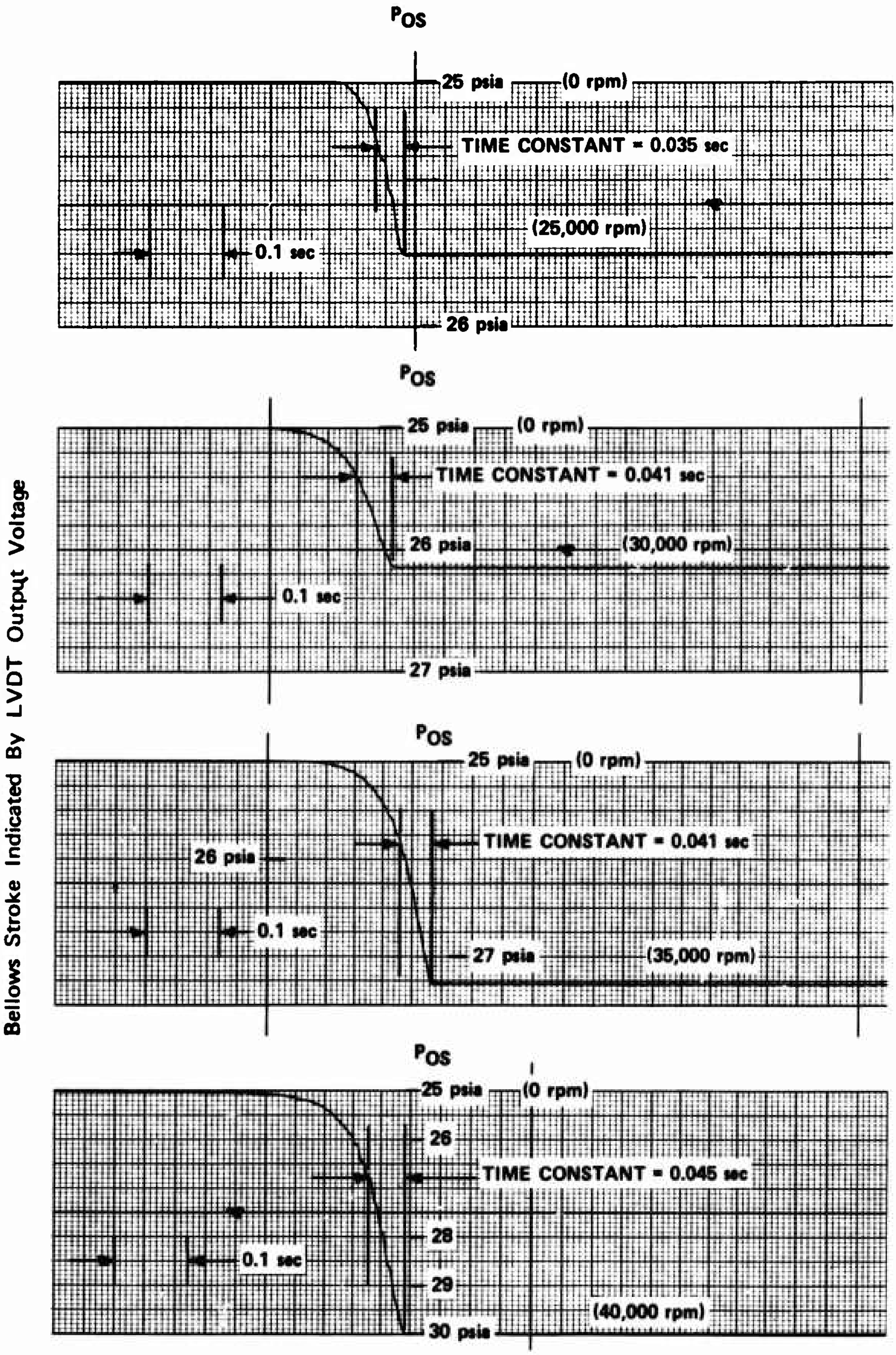


Figure 158. Transient Response Tests

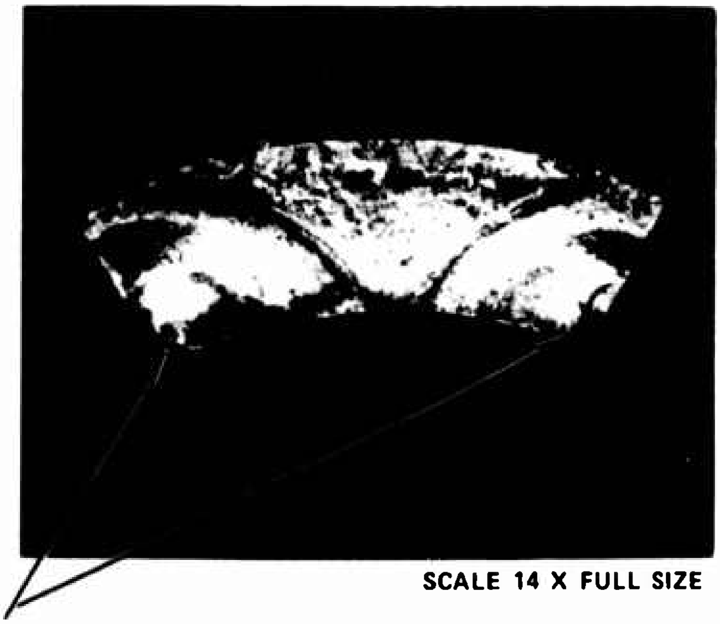


Figure 159. Failed Cantilever Section

4. Test Results

The test results obtained during this program are summarized following:

1. Tests performed on the pneumatic components show that the performance can be predicted accurately.

2. Tests of cantilever deflection versus sensor speed show that the performance can be accurately predicted.

3. The sensor tested shows a shift in calibration equivalent to a change in cut-in speed of 1% for 100°F change in sensor ambient temperature.

4. The transient response of the overspeed sensor is satisfactory.

5. The design of the cantilevers needs improvements to increase the fatigue life of the sensor.

5. Conclusions

Sufficient testing has been accomplished to demonstrate that the concept of sensing the power turbine speed with the cantilever air nozzle arrangement is feasible.

The sensitivity of the cantilever deflection to dimensional tolerances is such that it is preferable to make the sensor separable from the engine power turbine shaft so that the sensor can be calibrated at manufacture.

Experience gained during this program shows that design improvements can be made to the sensor. Figure 160 shows these improvements. The sensor is shown to be detachable from the engine power turbine shaft to allow the sensor to be calibrated at manufacture. The cantilevers would be produced so as to allow normal tolerances for the dimensions; the deflection at governed speed would be ensured by grinding the diameter against the sensor nozzle with the sensor running at governed speed. The shield around the sensor would allow setting up of the sensor nozzle by screwing the nozzle in until it touches the shield and then backing off at a predetermined angle. This would control the gap between the nozzle and the cantilevers. The shield would also stop the cantilevers from hitting the sensor nozzle in the event of an excessive overspeed.

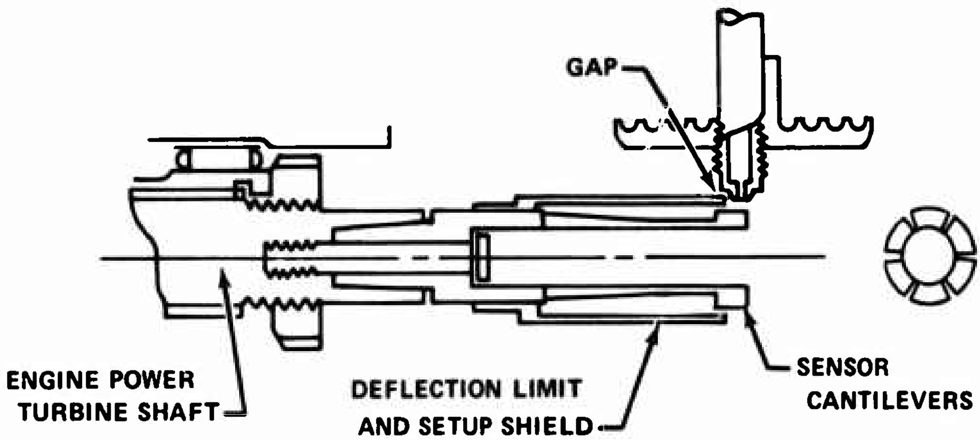


Figure 160. Improved Sensor Design

6. Recommendations for Additional Testing

1. Further evaluations of the overall system dynamic response are in order to verify applicability of the overspeed governor to the advanced technology helicopter engine.

2. Methods of compensating for sensor ambient temperature should be investigated and a solution to the problem demonstrated .

3. Experimental demonstration of the sensor installation and setup procedures is in order to assure that the system can be installed without in-place calibration.

SECTION X

CONCLUSIONS

The study has confirmed that controls and accessories for future Army turboshaft engines must utilize advanced technology to obtain a size, weight and vulnerable area consistent with the basic engine. Full-authority electronic controls, high-speed fuel and oil pumps, high-speed alternators and integral engine starters are recommended for controls and accessory implementation.

The engine configuration used in the study was optimized from a controls/accessory packaging and vulnerability standpoint. The baseline engine incorporated a rear-drive power turbine and an integral air turbine starter with no airframe accessory power takeoff. While the design of a C&A system for a front-drive engine configuration was not pursued to the detailed design stage, it is not anticipated that the technology requirement and implementation of the C&A components would vary significantly from those utilized on the rear-drive engine. In a similar light, if an engine/airframe installation favored an electrical starter or electrical starter/generator, the resulting impact on the C&A system would primarily be the requirement for additional envelope. The requirement for a relatively low speed airframe accessory drive pad would significantly impact the C&A system envelope and vulnerability resistance.

The study has also shown that an engine system designed with consideration for vulnerability has also added benefits in weight, volume and simplicity.

The detailed recommended C&A component implementation for future systems is outlined in the Recommendations section.

SECTION XI

RECOMMENDATIONS

Recommendations for additional C&A component critical item testing are included at the conclusion of the Phase III Critical Item Test Summary. The efforts on the Integrated Accessory Systems for Small Gas Turbine Engines program have indicated certain design trends for future Army helicopter engine and control systems. The projected state of the art for a 1977 development time frame has indicated that the following small, high-speed control system components and integration techniques are feasible and should be considered for small Army helicopter engines:

Flow Control - A full-authority electronic control should be used in conjunction with a hydromechanical flow metering system. Sensors would include a radiation pyrometer for turbine blade temperature limiting. Inlet guide vane actuation would be hydraulic and integral with the hydromechanical flow metering system. The power turbine would incorporate a separate mechanical/pneumatic overspeed sensor. Engine/airframe interfaces would be "fly by wire." The control system redundancy approach would be established as a function of the engine application and mission.

Fuel Pump - The fuel supply system would be designed for suction inlet conditions (1 psi above TVP, V/L = 1) and would consist of a high-speed (65,000 rpm) centrifugal pump with inducer and jet pump stages. The fuel pump would be packaged integrally with the hydromechanical flow metering system, if practical.

Oil System - The oil system remains as one of the most vulnerable of the engine systems. Vulnerability resistance can be improved by the use of air bearings to reduce the oil system heat load. Radial load capacity for air bearings is considered to be within the 1977 time frame, but axial load capacity exceeds the state of the art. Pressurized air/oil or fuel/oil heat exchangers are to be avoided for vulnerability considerations. For the system studied, an air/oil heat exchanger integral with the oil sump and inlet particle separator had sufficient heat exchanger capacity. A single-stage "vane type" oil and scavenge pump would be used. A limiting oil pump speed of 15,000 rpm was set by consideration of inlet cavitation. An air/oil mist backup lubrication system with a 30-min operation capability would also be utilized.

Electrical - Electrical power requirements for the basic engine system (computer and ignition) would be supplied by an alternator integral with the main gearbox. The projected thermal environment is well within the operational limits of alternator components, and the oil mist environment is not seen as a detriment to the alternator design. The reliability of the alternator is projected to be similar to that of other gearbox components.

Controls and Accessory Drive - A mechanical drive is recommended for the controls and accessories. A tower shaft drive off the gas generator rotor is the recommended implementation. A vertical drive to a controls and accessory location near the top of the engine is recommended for vulnerability benefits.

Accessory Drive Pads - The use of the engine gearbox for airframe accessory drive is discouraged because of the adverse impact on the gearbox design. Advanced engines will utilize small, high-speed rotating components which will be integrated within the engine structure without the conventional "hang on" gearbox. Provision of the relatively large low-speed pads for airframe accessories is possible at the expense of engine size, weight and vulnerability, but is not recommended. The use of the airframe gearbox for accessory drive will also provide drive redundancy in twin-engine installations.

Engine Starting - An air turbine starter system is recommended, if an APU is available or required for other airframe services. For certain installations where the development expense of an APU for starting may present an overwhelming cost disadvantage to the approach, electrical starters can be utilized at the expense of weight and vulnerability. Electrical starter systems can be more attractive if the -65°F start requirement is relaxed. Development of the starter as part of the engine system is recommended.

Power Turbine Interface - The packaging and resulting vulnerability resistance of controls and accessories are improved with a rear-drive engine as compared to a front drive. A rear-drive power turbine also tends to provide a better engine configuration from the standpoint of a smaller gas generator shaft diameter with the attendant lower bearing and seal velocities, improved compressor hub tip ratio, and lighter gas generator turbine disk.

APPENDIX

ENGINE DESCRIPTION

The primary features and requirements of the engine used during this program are outlined following.

A. GENERAL

The engine shall be designed for a dual-engine military helicopter application and shall reflect the technology commensurate with formal development beginning in 1977. It shall be of the direct-drive, free-power-turbine type and shall provide an output power shaft in the rearward direction.

Except as specified herein, the engine shall comply with AV-E-8593 turboshaft engine specifications unless the requirements are considered to be a hindrance to the advanced technology desired in this program.

B. CYCLE CHARACTERISTICS (Typical)

1.	Turbine Inlet Temperature	1288°C
2.	Compressor Pressure Ratio	10:1
3.	Gas Generator Speed	65,000 rpm
4.	Power Turbine Speed	36,000 rpm
5.	Air Flowrate	3 lb/sec
6.	Horsepower	600 hp

C. OPERATIONAL REQUIREMENTS

1. Environmental

1.	Ambient Temperature	-54°C to 57°C
2.	Nacelle Maximum Temperature	80°C - Continuous (150°C - 15 sec)
3.	Altitude	20,000 ft
4.	Fuel Temperature	-54°C to +57°C

2. Mission Cycle

A typical mission cycle is shown in Figure A-1 and can be used for system performance trade-off analysis.

3. Contaminated Fuel

1. The engine shall operate on fuel contaminated in accordance with AV-E-8593.

2. Fuel system components shall be designed for a life of at least 300 hr when operated on contaminated fuel.

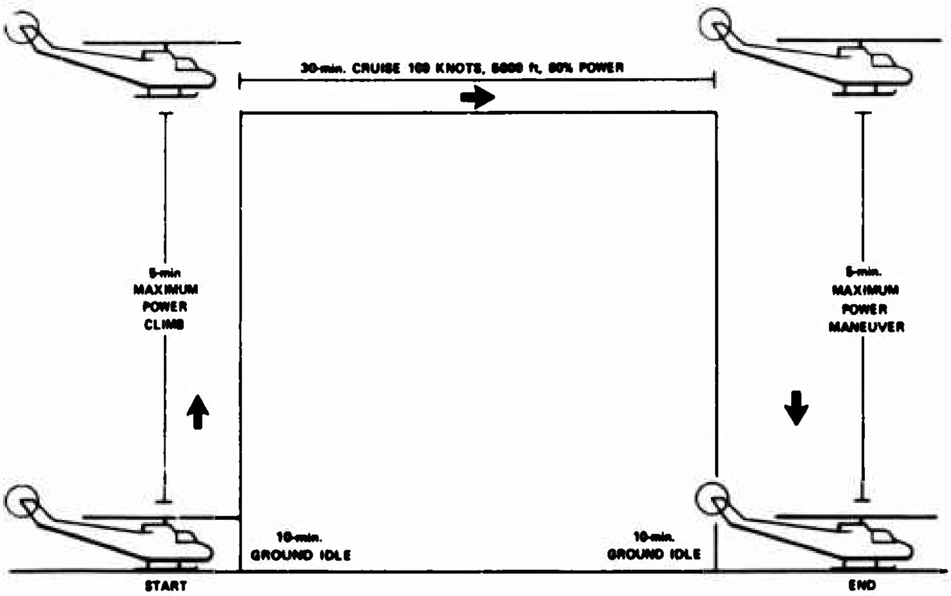


Figure A-1. Typical Mission Cycle

4. Specified Fuel

The engine shall function satisfactorily throughout its operating envelope for any steady-state and transient operating condition without requiring adjustment of the fuel system when using fuels conforming to and having any of the variations in characteristics permitted by MIL-T-5624, Grades JP-4 and JP-5, or JP-8, except that operation shall not be required using fuel having a kinematic viscosity greater than 12 cs.

5. Lubricants

The engine shall function satisfactorily throughout its operating envelope when using oil conforming to and having any of the variations in characteristics permitted by MIL-L-23699 or MIL-L-7808, except that operation shall not be required using oil having a kinematic viscosity greater than 13,000 cs.

6. Attitude

The engine shall be capable of continuous operation in nose-up or nose-down attitudes of up to 45 deg from the horizontal and roll attitudes of up to 20-deg inclination to either side. The engine shall be capable of satisfactory operation for 30 sec in nose-up or nose-down attitudes of up to 90 deg and roll attitudes of up to 45-deg inclination to either side.

7. Inlet Particle Separator

Protection from sand and dust and foreign object ingestion shall be provided by means of an inlet particle separator, which shall be self-contained and integral with the engine.

8. Condition Monitoring

The engine shall incorporate the necessary components to allow monitoring of the overall engine condition and provide fault isolation capability for all major engine modules or subsystems. The condition monitoring system shall consist of the sensors required for monitoring that will interface at a suitable location with a ground or flight diagnostic system. At least the following indications shall be provided:

1. Gas generator rotor speed
2. Power turbine rotor speed
3. Output shaft torque
4. Measured turbine temperature
5. Oil pressure
6. Oil temperature
7. Magnetic particle detector(s)
8. Oil level
9. Engine start (number) and time (hours) counter similar to MS17321.
10. Fuel filter bypass (if applicable)
11. Oil filter bypass.

The gas generator and power turbine rotor speed signals shall be an exact function of the gas generator and power turbine rotor speeds, respectively.

9. Suction Fuel System

An engine fuel pump shall be employed that shall provide fuel required to meet all engine fuel flow demands when supplied with fuel at the pump inlet at a pressure of 1.0 psi above the true vapor pressure of the fuel and with a vapor-to-liquid ratio of 1.0. The pump shall not be required to have a dry lift capability. All contaminated fuel requirements set forth in the operational requirements, outlined above, shall apply. No fuel recirculation to the vehicle tank shall be allowed.

10. Engine Starting

This subsystem shall provide for two successive starts or start attempts of the engine. For purposes of design of this subsystem, it can be assumed that sufficient hydraulic, pneumatic, or electrical energy will be available from airframe sources.

11. Fuel Control

The fuel control shall not have provisions for an emergency manual mode of control. The following features shall be incorporated into the fuel control:

1. Isochronous power turbine governing

2. Provision for selected engine operation at power turbine speeds from 85 to 115% of maximum operating speed at any power level

3. A direct (closed-loop) temperature limiting control

4. Redundant power turbine overspeed protection

5. Load sharing capability for multiengine installation.

12. Lubricating System

The lubricating system shall satisfactorily lubricate the complete engine, without change in lubricant, throughout its operating envelope, except that operation is not required if the oil viscosity is greater than 13,000 cs. The oil reservoir and cooler shall be furnished as component parts of the engine lubricant system and shall adequately cool either MIL-L-23699 or MIL-L-7808 oil throughout the entire operating envelope of the engine. The lubricating system shall require no external oil system.

13. Electrical Power

All electrical power required by the engine, including ignition, shall be provided by an integral system. There is no requirement for generation of electrical power for other than engine use.

14. Fuel Filter

If a fuel filter with finer than 1500μ openings is required, it shall be serviceable and a part of the engine. The capacity shall be sufficient to permit a cumulative fuel flow equivalent to a minimum of 25-hr engine operation at maximum continuous-rated static, sea level, standard day power with fuel contamination as specified in AV-E-8593 without being cleaned. Main flow filter shall be provided with an integral bypass, which prevents washing of the filter element or collection device, provisions for attaching instrumentation for remote indication of filter bypassing, and an accessible visual impending bypass indication.

15. Measured Turbine Temperature Sensing System

A measured turbine temperature sensing system shall be provided. The device shall be located at a point which will provide a temperature-indicating signal so related to turbine temperature that a single temperature limit may be set that will be applicable throughout the operating envelope of the engine. The temperature-indicating signal shall be suitable for an external readout.

16. Torque Sensor

The engine shall provide a signal for operation of a torque indicator throughout the complete range of the engine. The accuracy of the torque sensor, from 50% rated output shaft speed to 125% rated output shaft speed, shall be:

1. Within plus or minus 2% of the torque at maximum continuous-rated power from zero output shaft torque to the torque at maximum continuous-rated power

2. Within plus or minus 2% of the torque being measured from the torque at maximum continuous-rated power to the transient (0.2 min) torque limit.

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