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**PRELIMINARY DESIGN OF  
AN AUXILIARY POWER UNIT  
FOR THE SPACE SHUTTLE**

**Volume III - Details of System Analysis,  
Engineering, and Design for Selected System**

*by M. L. Hamilton and W. L. Burriss*

*Prepared by*

**AIRESEARCH MANUFACTURING COMPANY**

**Los Angeles, Calif.**

*for Lewis Research Center*





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16. Abstract This study has considered numerous candidate APU concepts, each meeting the Space Shuttle APU problem statement. Evaluation of these concepts indicates that the optimum concept is a hydrogen-oxygen APU incorporating a recuperator to utilize the exhaust energy and using the cycle hydrogen flow as a means of cooling the component heat loads. The initial portion of the study (Phase I) was concerned with evaluation of the candidate concepts; this information is presented in Volume II. The Phase II work accomplished preliminary design of the selected APU concept, placing primary emphasis on the cycle thermal management and the controls (to maintain desired turbine inlet temperature and rotational speed). The Phase II work is presented in Volumes III, IV, and V. Volumes III, IV, and V also present results for both steady-state and transient APU performance, based on digital computer programs developed during the study. The selected APU provides up to 400 hp out of the gearbox, has a fixed weight of about 277 lb, and requires about 2 lb/shp-hr of propellants.					
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## FOREWORD

This report is the third volume of a series that comprises the following:

- Volume I - Summary
- Volume II - Component and System Configuration Screening Analysis
- Volume III - Details of System Analysis, Engineering, and Design for Selected System
- Volume IV - Selected System Supporting Studies
- Volume V - Selected System Cycle Performance Data

Volume II summarizes the Phase I portion of the program in which the various component and system concepts were compared and evaluated. Volumes III, IV, and V contain the Phase II work in which preliminary design of the selected APU system concept was performed.



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## SECTION I

### INTRODUCTION AND SUMMARY

#### INTRODUCTION

The Phase II work performed under Contract NAS3-14408, "Preliminary Design of an Auxiliary Power Unit (APU) for the Space Shuttle," was primarily concerned with detail system analysis, engineering, and design of an APU system concept selected during Phase I. The Phase II work is reported in three volumes:

Volume III Details of System Analysis, Engineering, and Design  
for Selected System

Volume IV Selected System Supporting Studies

Volume V Selected System Cycle Performance Data

Table 1-1 summarizes the ground rules used for Phase II.

#### SUMMARY

Following are abstracts summarizing each of the sections in this volume.

#### Section 2. Evolution of System Configuration

Section 2 reviews the considerations leading to selection of the final system concept shown in Figure 1-1 in terms of evolution from the Phase I concept. Some of these changes (elimination of supplemental heat sink and turbine shroud cooling provisions, for example) resulted from NASA direction and evaluation of system interfaces. Other modifications to the original system concept resulted from more detailed studies performed during Phase II.

#### Section 3. System Configuration and Packaging

This section summarizes the system concept in terms of its overall characteristics (space envelope, weight, etc.). The APU consists of three principal subsystems: propellant conditioning/thermal control subsystem; turbine power unit; and controls as shown in Figure 1-2. (These subsystems are described to a component level in Sections 5, 6, and 7).

#### Section 4. System Performance

Section 4 gives steady-state performance parameters as a function of output power, ambient pressure, hydraulic fluid temperature, and propellant supply temperature. (Detailed steady-state performance data are given in Volume V.) Transient-state performance is given for startup, shutdown, and steady-state operation with changes in load and propellant inlet temperature. Adequate control system performance is shown for the entire envelope of possible steady-state and transient-state conditions.



TABLE 1-1

PRINCIPAL PHASE II GROUND RULES FOR H<sub>2</sub>-O<sub>2</sub> APU

<p>DESIGN PHILOSOPHY</p> <p>Maximize proven design concepts Monitor for failure detection</p> <p>LIFE</p> <p>1000 hr on H<sub>2</sub>-O<sub>2</sub> plus 2000 hr on inert gas 900 H<sub>2</sub>-O<sub>2</sub> starts plus 600 inert gas starts</p> <p>AMBIENT ENVIRONMENT</p> <p>Temperature = 400 to 760°R Pressure = sea level to vacuum</p> <p>POWER OUTPUT</p> <p>400 shp out of gearbox 100 shp out of gearbox for ground checkout with inert gas Output pads for 2, 90-120 gpm hydraulic pumps, 1 60/75 kw alternator Power turndown ratio = 16:1</p>	<p>PROPELLANT SUPPLY CONDITIONS</p> <p>Hydrogen = 75°R and 200 to 500°R, 500 to 1000 psia Oxygen = 300 to 500°R, 500 to 1000 psia Transients in propellant temperatures or pressures can cover entire range in 2 sec at APS accumulators</p> <p>TURBINE DESIGN REQUIREMENTS</p> <p>Material = Udimet 700 Type = 2 stage axial, pressure-compounded Inlet temperature = 2060°R Rotational speed = 70,000 rpm Speed control = ±5 percent Design for containment with tri-hub burst</p> <p>LUBE AND HYDRAULIC FLUID REQUIREMENTS</p> <p>MIL-L-7808 lube oil, 750°R maximum M2V hydraulic oil, 530°R minimum, 750°R maximum, 850°R several seconds duration maximum</p>
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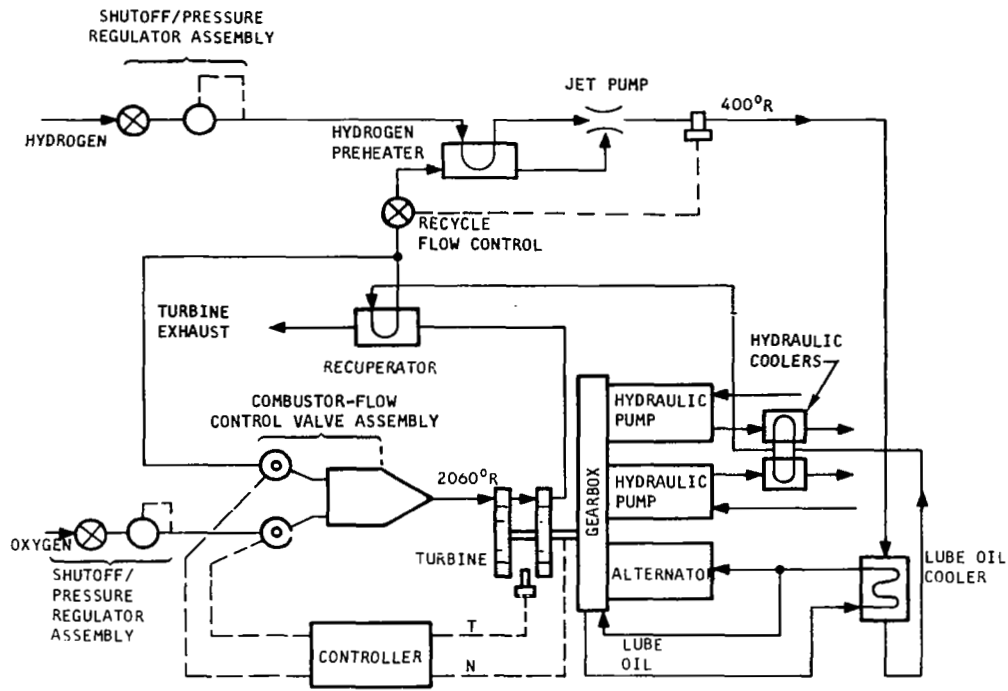
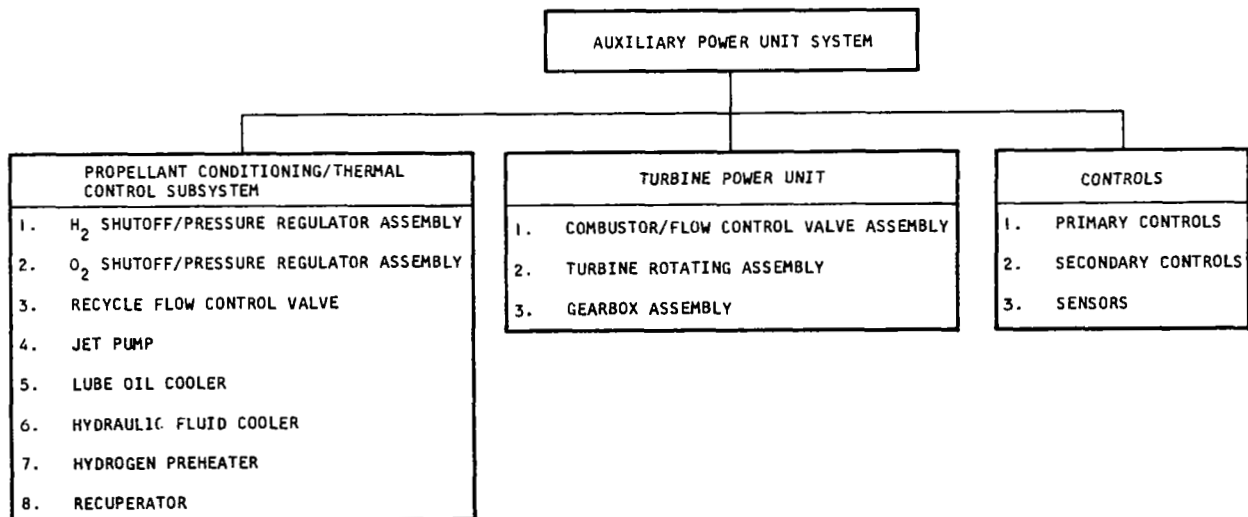


Figure 1-1. Final System Concept Schematic



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Figure 1-2. Subsystems and Components Comprising APU System

## Section 5. Propellant Conditioning/Thermal Control Subsystem

The components comprising the propellant conditioning/thermal control subsystem are the heat exchangers, propellant pressure regulator and shutoff valves, the jet pump and recycle flow control, and the interconnecting ducting. The design and performance characteristics of these components are described and compared with the state-of-the-art.

## Section 6. Turbine Power Unit

Section 6 describes the components forming the turbine power unit. These components are contained in three major assemblies: combustor flow control valve assembly; turbine rotating assembly; and gearbox assembly. The design and performance of these assemblies are described together with the design basis.

## Section 7. Controls

In Section 7, the controls subsystem is described in terms of the primary functions (turbine speed control, turbine inlet temperature control, and hydrogen loop temperature control), the secondary functions (APU startup/shutdown, fault detection, and emergency shutdown), and the sensors required to perform the various control functions. In addition, the digital transient computer program used in assessing controls performance is described.

## Section 8. Liquid Pumped Cycles

Section 8 is concerned with the components and cycle modifications necessary to convert the baseline system concept into one utilizing low-pressure liquid cryogenics. Pumps capable of pumping liquid hydrogen and liquid oxygen are the primary additional components required for this conversion. A positive-displacement hydrogen pump and a centrifugal oxygen pump are recommended.

## SECTION 2

### SYSTEM CONCEPT EVOLUTION

#### INTRODUCTION

The baseline system concept presented here was evolved during the course of the study program after a number of different concepts were studied in some detail. During the Phase I studies reported in Volume II of this final report, the following five types of APU system were evaluated and compared:

- Low-Pressure Cryogenic Hydrogen-Oxygen Supplied System
- High-Pressure Cryogenic Hydrogen-Oxygen Supplied System
- High-Pressure Gaseous Hydrogen-Oxygen Supplied System
- Dual-Mode Airbreathing/Cryogenic Hydrogen-Oxygen System
- Monopropellant System

At the completion of Phase I, in concurrence with study findings, NASA selected the high-pressure gaseous supplied system, in which the APU shares the tankage, pumping, and propellant conditioning functions with the Auxiliary Propulsion System (APS). Subsequently, because of questions concerning the impact of APU operation on APS turbopump life (by greatly increasing the number of operational cycles), NASA directed that the study include consideration of APU systems with integral propellant pumping provisions.

#### Phase I Gaseous Supplied APU System Configuration

Figure 2-1 shows the system schematic for the gaseous supplied APU system as defined during Phase I. At that time it was assumed that the APU cooling load would include the vehicle hydraulic system and that it was necessary for the APU system to be self-contained with respect to dissipating the system heat load. As a consequence of these assumptions and the requirement for the system to function with propellant inlet temperatures ranging from 200° to 500°R, a supplemental heat sink (in the form of a water boiler) was included in the system to provide the necessary supplemental cooling required at high propellant-inlet temperatures (in excess of 300°R) and low output power. The Phase I system concept also incorporated provisions for turbine disk cooling by radiation to a hydrogen-cooled housing. With this type of cooling, turbine stress and thermal studies showed that the turbine could be designed for reliable operation at a pitch line velocity of 1800 fps at a turbine inlet temperature of 2260°R. Table 2-1 summarizes the major configuration changes that were made during Phase II to establish the final configuration (shown in Figure 2-2). These will be discussed in the paragraphs following.

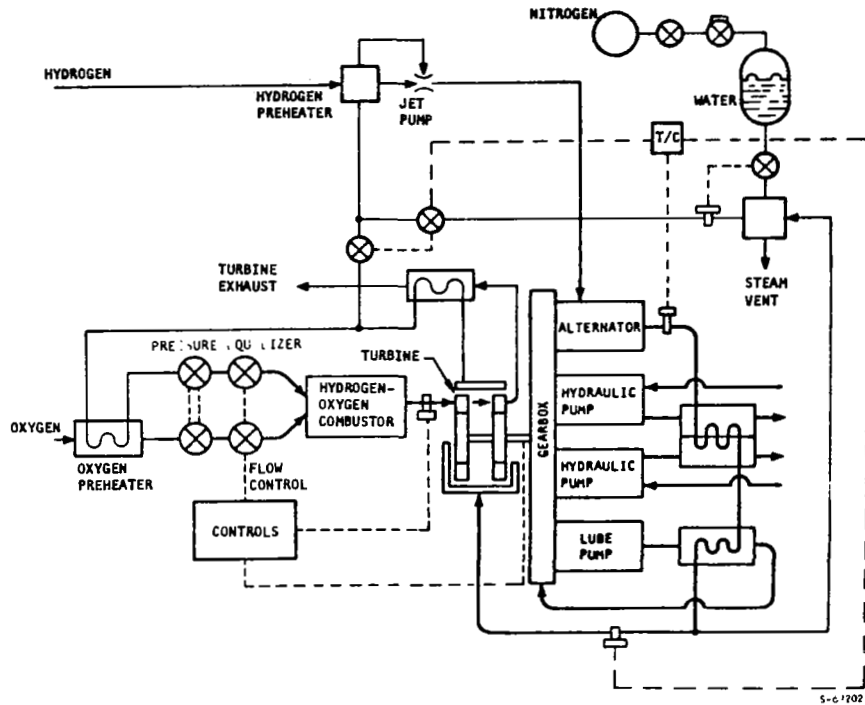


Figure 2-1. High-Pressure Gaseous Hydrogen-Oxygen Supplied System Configuration at End of Phase I

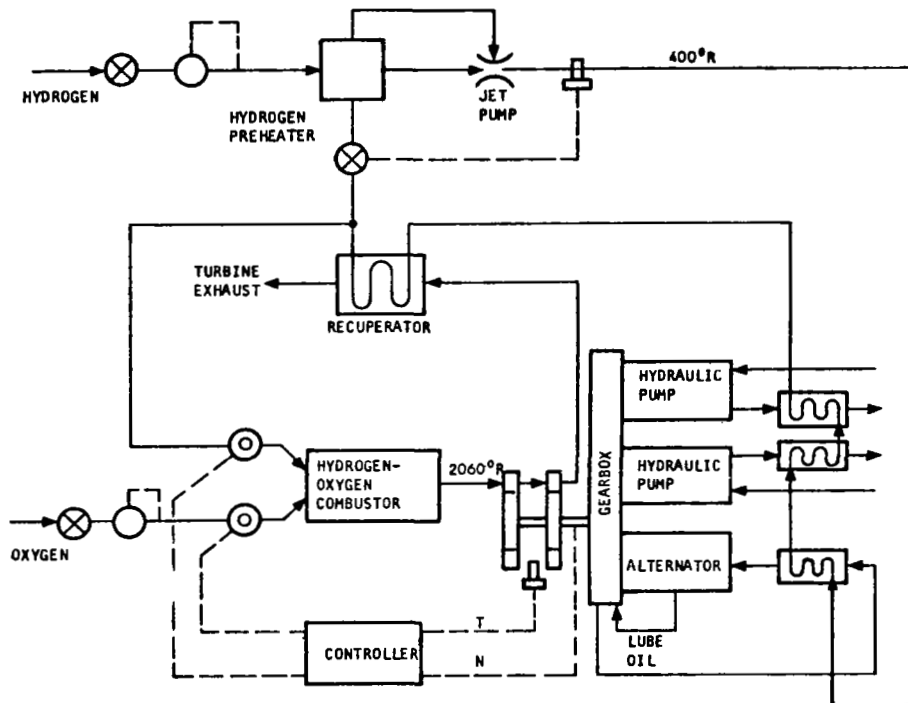


Figure 2-2. Final APU System Schematic

TABLE 2-1

## SUMMARY OF MAJOR SYSTEM CONFIGURATION CHANGES FROM PHASE I

Configuration Change	Reason
1. Supplemental cooling provisions (water boiler) eliminated	NASA directive
2. Turbine cooling provisions eliminated	NASA directive
3. Turbine inlet temperature = 2060°R	NASA directive
4. Use of a lube-oil-cooled alternator specified	NASA directive
5. Propellant pressure regulators and shutoff valves added	NASA directive
6. Lube oil cooler upstream from hydraulic cooler	Consequence of changes 1 and 2
7. Low-temperature recycle loop eliminated	Consequence of change 2
8. Temperature and pressure equalizers eliminated	Shown unnecessary by controls studies
9. Turbine inlet temperature sensor relocated to interstage location	Favored by response, packaging, and life considerations
10. Lube pump second stage eliminated	Pitot scavenge pump provides sufficient $\Delta P$

Configuration Changes Resulting from NASA Directives at End of Phase I

The following configuration changes were made as a result of NASA directives following completion of Phase I:

Supplemental cooling provisions eliminated.

Turbine cooling provisions eliminated and turbine inlet temperature reduced to 2060°R.

Use of a lube-oil-cooled alternator specified.

Propellant pressure regulators and shutoff valves added to system

Location of hydraulic cooler in high-pressure pump discharge line

## CONFIGURATION CHANGES RESULTING FROM PHASE II SYSTEM STUDIES

### Oil Cooler Rearrangement

Because of the NASA-directed changes above, it was found necessary to rearrange the oil coolers in the hydrogen circuit and locate the lube oil cooler upstream from the hydraulic fluid cooler. As discussed in detail in Section 2 of Volume IV, this arrangement is necessary to insure proper cooling of the lubricant. Since the hydrogen flow is the only heat sink available to the lubricant and the thermal time constant in that circuit is relatively low, priority must be given to lubricant cooling to insure proper system operation under conditions where total cooling capacity is insufficient to absorb all of the waste heat generated internally within the APU.

### Elimination of Low-Temperature Recycle Loop

The low-temperature recycle loop (which recycles hydrogen flow from the oil coolers to the hydrogen preheater) in the original concept provides hydrogen preheating with no loss in cooling capacity at low APU output power. The high-temperature recycle loop (which recycles hydrogen flow from the recuperator discharge to the hydrogen preheater) is necessary to provide the required preheating with low inlet hydrogen temperature and high output power. In the Phase I system configuration shown in Figure 2-1, both loops were used, one using a water boiler for supplemental cooling. With elimination of the supplemental cooling feature by NASA directive, as mentioned previously, there is less need for the low-temperature recycle. At the cost of a small loss in cooling capacity at minimum APU output conditions, the system was simplified by elimination of the low-temperature recycle loop.

### Elimination of Temperature and Pressure Equalizers

The oxygen preheater and pressure equalizing valve were included in the Phase I system concept of Figure 2-1 to provide essentially equal pressure and temperature between the oxygen and hydrogen flow to the propellant flow control valves. This feature was included in the system to reduce the flow control valve turndown ratio, the operating range for the O/F control function, and make the inlet conditions satisfactory for pulse modulated as well as pressure modulated control. In the system controls studies performed during Phase II, these functions were found to be unnecessary. Therefore, additional improvements in system simplicity were obtained with elimination of the pressure and temperature equalizing devices. Elimination of the oxygen preheater is additionally desirable from a safety standpoint, since the possibility of leakage of one fluid into another is always present, even with buffered or single integral tube designs.

### Relocation of Turbine Inlet Temperature Sensor

The first stage of the turbine operates at constant pressure ratio and, as a consequence, constant efficiency, regardless of turbine output power or ambient pressure. Therefore, turbine interstage temperature remains essentially constant at  $1705^{\circ}\text{R}$  for a turbine inlet temperature of  $2060^{\circ}\text{R}$ . Since turbine interstage temperature is proportional to turbine inlet temperature, interstage

temperature can be sensed and used to control turbine inlet temperature without compensation for power level or ambient pressure. This location is particularly advantageous for the following reasons:

The turbine interstate temperature sensor operates at a temperature level 355°R lower than the turbine inlet temperature, which favors longer sensor life.

High gas velocities are obtained in the interstage location, resulting in good sensor response.

The interstage sensor location provides convenient packaging.

#### Lube Pump Second Stage Eliminated

The Phase I system concept had a two-stage lubricant pumping system using a pitot pump for the first stage and an internal gear pump for the second stage. First-stage pumping was provided by a pitot pickup in a rotating lubricant sump contained on the alternator drive gear. The rotating sump also served to provide zero-gravity scavenging and deaeration. During Phase II system studies, it was determined that the pitot pump would be sufficient and, as a consequence, the second pumping stage would not be required.

#### PHASE II STUDIES SUPPORTING BASELINE CONCEPT SELECTION

Figure 2-2 shows the final gas-supplied APU system concept. During Phase II, a number of studies (summarized in Volume IV) of alternative cycles and concepts were performed that supported the selected baseline concept. Results of these studies will be summarized in the paragraphs following.

#### System Cycle Studies

There are three basic functional elements in the system thermal cycle: the equipment heat loads (hydraulic cooler and lube oil cooler), the recuperator, and the turbine-combustor. These three elements can be arranged in two ways, one with the recuperator upstream from the oil coolers, the other with the recuperator downstream from the oil coolers. As discussed in Section 2 of Volume IV, the upstream recuperator concept leads to heat exchangers with unacceptably high design problems and development risk because of the low inlet hydrogen temperatures.

#### Turbine Speed Control Studies

##### 1. Pulse-Modulation Control vs Pressure-Modulation Control

In Section 3 of Volume IV, detailed comparisons are made of pulse-modulation control and pressure-modulation control. The performance advantage of the pulse-modulation control is largely offset by increased system fixed weight. More important factors that led to selection of pressure-modulation control for the baseline concept are:

Possible cyclic life problems with flow control valves and other components as a result of large number of on-off cycles with pulse-modulation control



Greater flexibility of pressure-modulation control in meeting alternator synchronization MIL-STD-704A electrical power quality, and/or close frequency control requirements.

Better present technology status for pressure-modulation control; pulse-modulation control utilization is limited to a few short-life missile APU's.

## 2. Integrating vs Droop Controller

Section 7 of Volume IV summarizes the results of fluid dynamic analog studies concerning droop (valve position change proportional to speed error) and integrating type turbine speed controls. These studies simulated the volumetric and pressure drop effects and assessed system response to step changes in load and ramp changes in propellant inlet pressure and temperature. With the droop type control, system performance is limited by control valve response; superior performance is attainable with the integrating type control, which was selected for the Space Shuttle APU. These studies also show that flow continuity can be assumed through the APU in the transient state, if the controls concept is correctly chosen. This work led to the more detailed digital transient-state computer program studies that are reported in this book.

## Hydrogen Loop Temperature Control

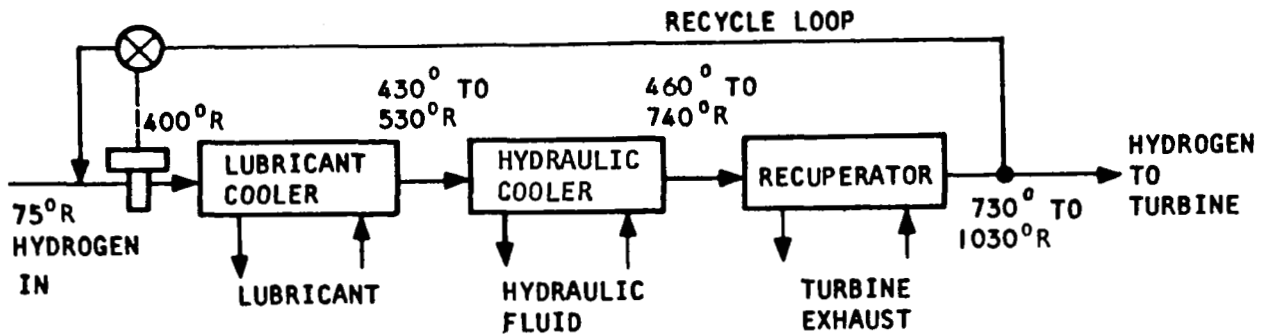
In Section 5 of Volume IV, the upstream recuperator cycle (which uses recuperator bypass for temperature control in the hydrogen loop) is compared with the downstream recuperator cycle (which uses recuperator discharge recycle for temperature control in the hydrogen loop). These two arrangements are illustrated in Figure 2-3.

### 1. Upstream Recuperator with Bypass Temperature Control

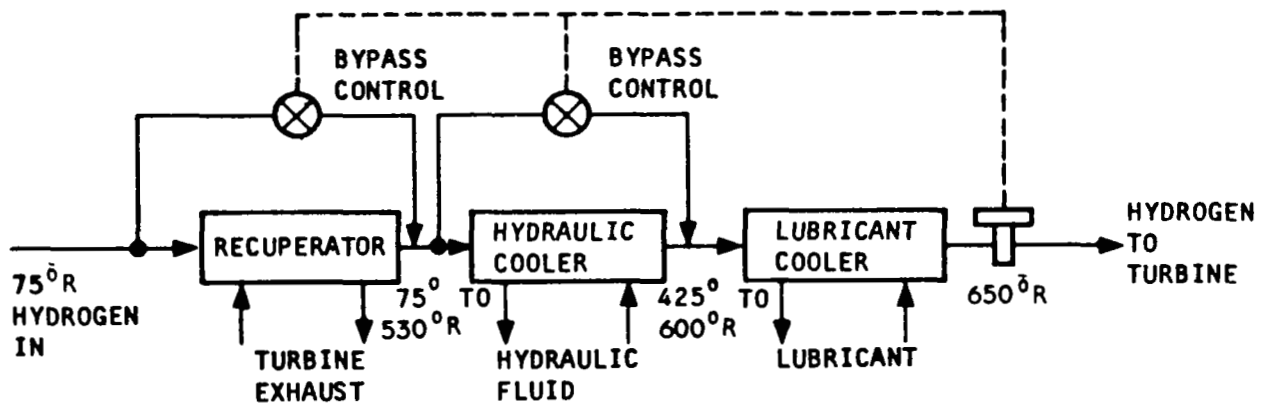
With this cycle, bypassing occurs at reduced APU output power, exposing the heat exchangers to very low inlet-hydrogen temperatures. In addition, because of the variations in hydraulic fluid temperature ( $530^{\circ}$  to  $750^{\circ}$ R), it is necessary to provide a bypass on the hydraulic heat exchanger to insure adequate lubricant cooling (or to provide constant outlet temperature, if an open-loop turbine inlet temperature control approach is used).

### 2. Downstream Recuperator with Recycle Temperature Control

Flow recycling in the downstream recuperator cycle is simply and efficiently performed by a jet pump using inlet hydrogen flow to the system as the primary (pumping) fluid. The jet pump  $\Delta P$  capability increases with hydrogen throughflow, which provides a degree of inherent temperature regulation at the inlet. Where the system operates at constant inlet hydrogen temperature (for example, in a liquid-fed pumped system), the recycle flow control is unnecessary for controlling the temperature in the hydrogen loop because of the inherent characteristics of the jet pump. The downstream recuperator with recycle control avoids low-temperature problems in the heat exchangers and consequently is recommended for the Space Shuttle APU.



a. DOWNSTREAM RECUPERATOR CYCLE (SELECTED)



b. UPSTREAM RECUPERATOR CYCLE (REJECTED)

S-67280

Figure 2-3. Recuperator Cycle Arrangements

## Turbine Inlet Temperature Control Studies

As discussed in Section 4 of Volume IV, there are a number of different active and passive methods for directly and indirectly maintaining turbine inlet temperature below a specified limit (in this case 2060°R). Figure 2-4 shows the two most competitive concepts, one using open-loop control of turbine inlet temperature, the other using closed-loop control.

### 1. Open-Loop Control of Turbine Inlet Temperature

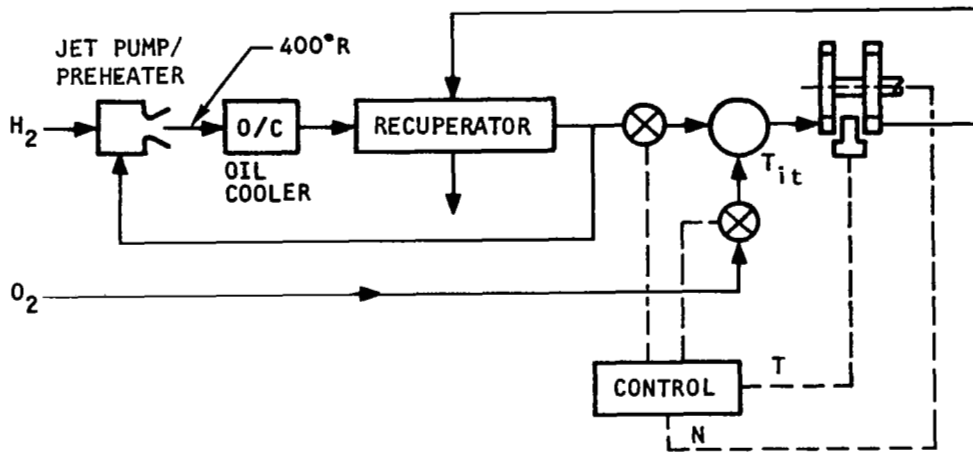
Open-loop systems depend upon maintaining essentially constant O/F ratio and constant inlet hydrogen temperature to the combustor. O/F ratio control requires essentially equal temperatures and pressures for the hydrogen and oxygen and constant metering orifice area relationships between the oxygen and hydrogen flow control valves. From this, it can be seen that the open-loop approach is sensitive to pressure-equalizing valve/flow-control valve performance and manufacturing tolerances. Small variations in flow control valve leakage or orifice areas will be reflected in significant variations in turbine inlet temperature.

### 2. Closed-Loop Control of Turbine Inlet Temperature

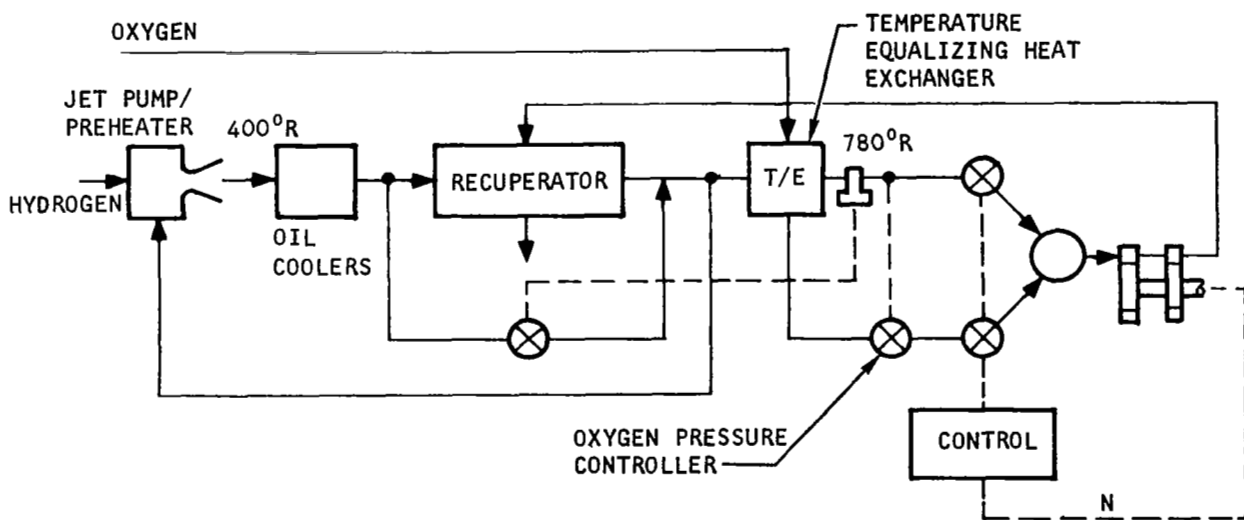
With the closed-loop control, oxygen flow is trimmed to maintain constant turbine interstage temperature and, hence, constant turbine inlet temperature. No great development problems are expected with the temperature sensor using tungsten-rhenium (or, alternatively, Geminol) thermoelectric materials. Closed-loop control is not highly sensitive to control valve or pressure regulator characteristics. As a consequence of its superior performance and ability to accommodate control valve tolerances (which should be reflected in high system reliability), closed-loop control of turbine inlet temperature is recommended for the present application.

## Reliability Studies

Reliability studies (summarized in Section 8 of Volume IV) were conducted at system and component levels. System level studies defined instrumentation requirements for operational monitoring of system status, ground checkout, and fail detection for subsystem shutdown. The individual components were analyzed to determine potential failure modes. Results of the component analysis indicate that the component designs lead to fail-safe APU operation.



a. CLOSED LOOP  $T_{it}$  CONTROL (SELECTED)



b. OPEN LOOP  $T_{it}$  CONTROL (REJECTED)

S-67272

Figure 2-4.  $T_{it}$  Control Loop Arrangements

## SECTION 3

### SYSTEM CONFIGURATION AND PACKAGING

#### INTRODUCTION

This section is concerned with the overall configuration and packaging of the APU system. Figures 1-1 and 1-2 (in Section 1) contain the system schematic and functional grouping of components used in this report to describe the system.

#### APU SYSTEM PACKAGING

The turbine power unit and propellant conditioning/thermal control subsystem have been packaged into an integrated assembly shown on Drawing SK39903. The controls will probably be mounted separate from the mechanical portion of the system in an adjacent electronics equipment bay on the vehicle. Controls are compatible with vehicle multiplexing to permit remote location or integration with the vehicle data system. (If desired, the controls can be mounted on the APU.) The controls are packaged in a standard 3/8-ATR short rack (with approximate dimensions of 4 in. by 8 in. by 14 in.).

The mechanical and thermal components have been integrated into a compact design with a minimum of component to component mounting. Flow losses are therefore minimized and the overall package made lighter and more rigid. All of the connections within this package are tungsten inert gas welded joints, except for the low pressure exhaust flange. Thus, leakage problems within the package components are virtually eliminated. Each of the joints are repairable and weldable to permit duct cutting for disassembly or maintenance.

Mounting provisions can be provided at any location around the main gearbox housing. The mounts have not been shown on the drawing because of the undefined nature of the mounting requirements.

#### FIXED WEIGHT SUMMARY

Following is a listing of the APU fixed weight:

Turbine-gearbox assembly	71.7 lb
Hydraulic pumps and alternator	100.0
Ducting	34.4
Lube and hydraulic coolers	31.6
Recuperator	11.8
Valving	8.6
Controls	7.0
Hydrogen preheater	6.1
Combustor/flow control assembly	5.6
Total fixed weight	<u>276.8 lb</u>

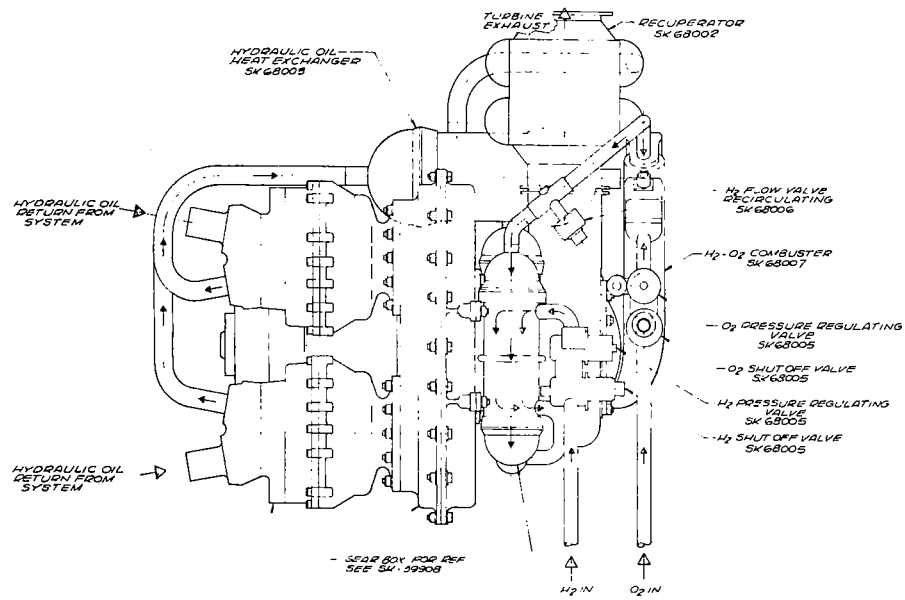
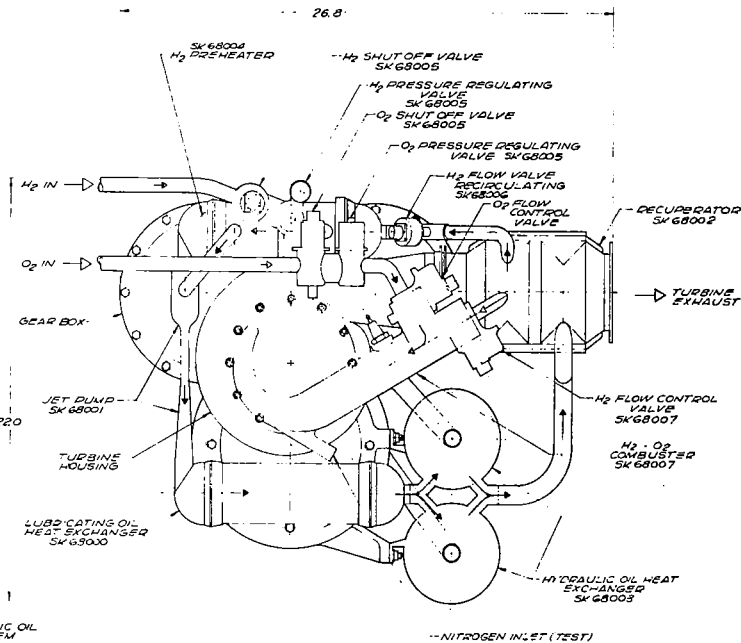
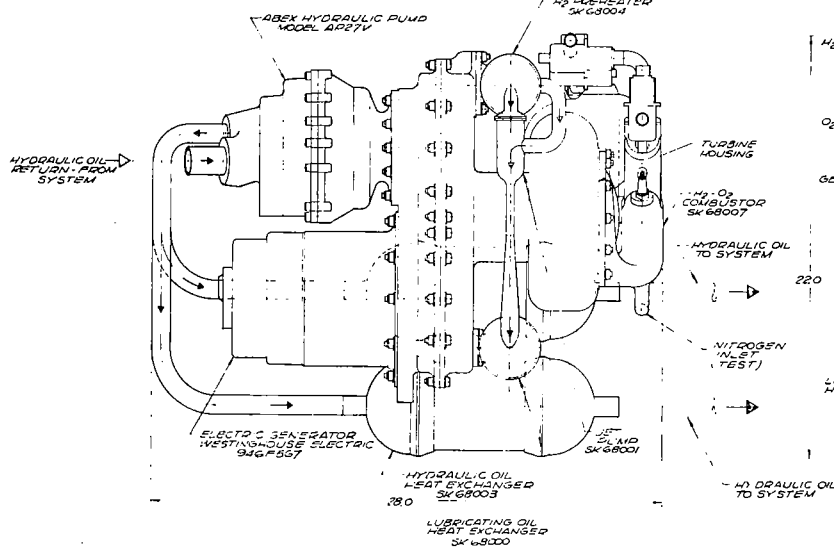


TABLE OF WEIGHTS: LBS

TURBINE & GEAR BOX ASSEMBLY	67.73
LUBRICATING OIL-HE	5.01
RECUPERATOR	11.80
H <sub>2</sub> PREHEATER	6.09
HYDRAULIC OIL-HE	24.50
JET PUMP	4.10
H <sub>2</sub> SHUT OFF VALVE & PRESS REG	4.10
H <sub>2</sub> FLOW VALVE RECIRCULATING	3.80
O <sub>2</sub> SHUT OFF VALVE & PRESS. REG	3.60
COMBUSTOR WITH H <sub>2</sub> & O <sub>2</sub> FLOW CONT. VALVES	5.80
SENSORS & CONTROLS	5.80
MISCELLANEOUS DUCTING	1.10
TOTAL, POWER UNIT LESS DUMPS & ELEC GENERATOR	138.73 LB - DRY WEIGHT



1 MOUNTING PROVISIONS ARE NOT SHOWN - DADS, ARMS AND OR BUCKETS TO MEET THE PACKAGE REQUIREMENTS CAN BE ACCOMMODATED

AUXILIARY POWER UNIT FOR THE SPACE SHUTTLE

SK-39903

## PROPELLANT CONSUMPTION

Volume V and Section 4 of this volume give detailed propellant consumption data as a function of all controlling parameters (output power, ambient pressure, inlet hydrogen temperature, and inlet hydraulic fluid temperature). For the NASA-prescribed booster and orbiter vehicle missions, the following overall propellant consumptions are obtained (for 300°R hydrogen inlet temperature):

	Booster	Orbiter
Average SPC, lb/shp-hr	2.08	1.82
Average O/F	0.575	0.614
Total hydrogen-oxygen weight, lb	308.6	286.5

## SYSTEM DESCRIPTION

The APU system consists of three major subsystems, each representing a grouping of functional elements:

Propellant conditioning/thermal control subsystem

Turbine power unit

Controls

### Propellant Conditioning/Thermal Control Subsystem

This subsystem, described in detail in Section 5 of this volume, provides the following functions:

Supplies propellants at proper temperature and pressure to the turbine power unit

Dissipates waste heat generated internally in the turbine power unit at suitable temperature levels for the various system components

The propellant conditioning/thermal control concept uses waste heat from the lubricant, hydraulic fluid, and turbine exhaust gas to preheat the incoming hydrogen (which may vary in temperature from 75 to 500°R). A recycle loop is used in the system to maintain proper hydrogen inlet temperatures to the various heat exchangers. In this way, heat exchanger design problems with flow instability and maldistribution leading to fluid freezing or congealing are avoided (as discussed in Section 2 of Volume IV). Also avoided, at the same time, are heat exchanger designs which depend upon accurate predictions of heat transfer coefficients to maintain wall temperatures at acceptable values (to avoid freezing, congealing, or moisture condensation). AiResearch experience has shown that in practice such designs, however attractive they may be on paper, are extremely difficult to effect.

Following are brief discussions of the component functions for this subsystem:

1. Hydrogen and Oxygen Shutoff/Pressure Regulator Valve Assemblies

The shutoff/pressure regulator valve assemblies provide the APU isolation and propellant pressure regulation functions. Because of the control system selected for the APU, these pressure regulation functions are not critical with respect to accuracy.

2. Recycle Flow Control Valve

This modulating valve is controlled by the jet pump exhaust temperature transducer. A sensed temperature less than 400°R results in a command to open the recycle valve, thereby increasing the hot gas recycle flow. If the APU system is supplied with hydrogen at a relatively constant temperature (as with a liquid-fed system, for example), the recycle flow control valve can be eliminated (with a reduction in available hydraulic cooling capacity, as discussed in Section 5 of Volume IV).

3. Jet Pump

The jet pump uses a portion of the pressure head of the inlet hydrogen to the system as an energy source to recirculate hot hydrogen flow from the recuperator.

4. Lube Oil Cooler

This heat exchanger provides cooling of the lubricant by heat exchange with the hydrogen loop. The lubricant serves to cool the alternator, gearbox, and bearings. The lubricant heat load is primarily a function of APU output power.

5. Hydraulic Fluid Cooler

The hydraulic fluid cooler is a dual heat exchanger assembly which cools hydraulic fluid flow from the two hydraulic pumps which are mounted on the APU gearbox. Hydrogen flowing through the heat exchanger on the cold side serves as the heat sink for hydraulic cooling. The available hydraulic cooling capacity depends upon the hydraulic fluid temperature and APU operating conditions (output power level, ambient pressure, and inlet hydrogen temperature). As discussed in Section 6 of Volume IV, excess cooling capacity (beyond that represented by the hydraulic pump heat load) will be available at high-power-output and low hydrogen inlet temperature. Conversely, the cooling capacity will not be adequate under other conditions to meet the pump heat load and the system hydraulic fluid temperature will rise.

6. Hydrogen Preheater

This hydrogen-to-hydrogen heat exchanger serves to equalize the temperatures of the primary and secondary hydrogen inputs to the jet pump to achieve uniformly high pumping performance.



## 7. Recuperator

This heat exchanger operates with hydrogen on the cold side and turbine exhaust gas on the hot side. It serves to provide sufficient heat input into the cycle for propellant thermal conditioning with low inlet hydrogen temperatures and to improve cycle thermal efficiency by recovering waste heat from the turbine exhaust. Overall cycle considerations in recuperator location and design are discussed in Section 2 of Volume IV.

### Turbine Power Unit

The turbine power unit, which is described in detail in Section 6, consists of three major subassemblies, each containing a number of functionally related components, as follows:

Combustor-flow control valve assembly

Turbine rotating assembly

Gearbox assembly

#### 1. Combustor-Flow Control Valve Assembly

This assembly delivers a controlled flow of hot gas to the turbine, using as inputs gaseous oxygen and hydrogen from the propellant conditioning subsystem and signals from the controller for actuation of the propellant flow control valves. The hydrogen and oxygen flows are controlled by separate electrically-driven (torquemotor) flow modulating valves. Each valve is provided with a linear variable differential transformer position sensor which provides position feedback to the controller thereby insuring proper control system performance under all conditions without dependence on control amplifier gain characteristics. The diffusion-flame type combustor is designed for spark ignition and is self-sustaining during operation after the initial ignition. The combustor has low hydrogen pressure drop and low wall temperature (because of the hydrogen film cooling effect).

#### 2. Turbine Rotating Assembly

The turbine is a two-stage pressure-compounded axial-flow impulse design which operates at a nominal inlet temperature of 2060°R, a rotational speed of 70,000 rpm, and a pitch-line velocity of 1700 fps. The turbine rotating assembly incorporates a fusible-wax heat sink material to prevent deleterious effects to the bearings because of heat soakback after shutdown. The turbine is designed to provide high efficiency over a wide range of pressure ratio with a pressure-modulating type of turbine control.

#### 3. Gearbox Assembly

The gearbox assembly consists of the gearing, mounting pads for the various components and an integral pressure lubrication system with provisions for zero-gravity scavenging and deaeration. The gearbox mounting pads are as follows:

Turbine rotating assembly	70,000 rpm
Alternator	12,000 rpm
Hydraulic pumps (2)	6,000 rpm

The gearbox assembly also provides the main structural mounting points for the APU system package.

## Controls

System controls, described in Section 7, consist of primary controls, secondary controls, and sensors.

### 1. Primary Controls

Three primary control functions are provided:

- Turbine speed control
- Turbine inlet temperature control
- Hydrogen loop (jet pump discharge) temperature control

Turbine speed control is accomplished by sensing turbine shaft speed and providing proportional control of the propellant flow (for both hydrogen and oxygen) by means of an integrating type of control. Turbine inlet temperature is controlled by sensing turbine interstage temperature (which is proportional to inlet temperature) and trimming oxygen flow. Hydrogen loop temperature is controlled by means of a recycle flow control valve which regulates the hot hydrogen (from the recuperator) to maintain jet pump discharge temperature at a nominal 400°R.

### 2. Secondary Controls

Secondary control functions are those required for APU startup, shutdown, fault detection, and emergency shutdown. The startup/shutdown controls provide proper sequencing of valve operations, combustor ignition, and emergency shutdown overrides (during startup, it will be necessary to override some emergency shutdown functions). The fault detection provisions involve monitoring system parameters which indicate proper functioning of the components. The parameters to be monitored include:

- Turbine inlet (interstage) temperature
- Regulated hydrogen and oxygen pressures
- Recycle flow control valve position
- Lube oil pressure and temperature
- Hydrogen and oxygen flow control valve positions

The emergency shutdown functions protect the APU to avoid potentially hazardous conditions in event of malfunctions. The emergency shutdown functions selected for the space shuttle APU are:

Turbine overspeed/underspeed (+10%, -5%)

Turbine overtemperature (2200°R)

Lube oil overtemperature/underpressure (800°R, 150 psia)

Hydrogen underpressure (450 psia)

### 3. Sensors

The sensors provide the signals required for the primary and secondary control functions. As a consequence, the sensors required by the control subsystem correspond to the previous listings of primary and secondary control functions. Conventional aircraft-type sensors are, for the most part, suitable for the space shuttle APU. Shielded thermocouples using tungsten-rhenium alloys have been selected for the turbine interstage temperature sensor.

## SECTION 4

### SYSTEM PERFORMANCE

#### INTRODUCTION

This section summarizes the Phase II system performance studies. Determination of system performance was accomplished primarily through use of two digital programs. The steady-state program uses nested iterative loops to converge to a solution for any specified operating point. (Steady-state performance data as output by this program are presented in Volume V.) The transient performance program uses a hyperspace error vector method of determining the route to converge at each time increment. (This program is summarized at the end of Section 7 of this volume.)

The material in this section is organized as follows:

- Steady-state performance
- Transient performance
- APU operation from inert gas
- Integrated mission performance

#### STEADY-STATE PERFORMANCE

The performance data in Volume V provides complete details of the system operating conditions at 144 different points within the APU operational envelope. The material here summarizes those data, indicating typical cycle state points at key locations in the APU system. The primary variables determining the APU operating conditions are as follows:

- Useful output power (since the hydraulic pump and alternator losses are assumed to be independent of output power, the useful power split into hydraulic and electric power is not a variable)
- Ambient pressure--affects the turbine discharge pressure, and hence the turbine performance
- Hydrogen inlet temperature--establishes the amount of recycle flow needed for component temperature control, and influences the O/F ratio required to obtain the desired turbine inlet temperature
- Hydraulic fluid heat exchanger inlet temperature--combined with the useful power output and the hydrogen inlet temperature, the hydraulic fluid heat exchanger inlet temperature establishes the amount of heat removed from or added to the hydraulic fluid

Unlike the hydraulic fluid, the lube oil thermal inertia is sufficiently small so that it can be assumed that the lube oil temperature will rise or fall to obtain heat rejection to the hydrogen equal to the heat added by the gearbox and output component heat generation.

Propellant Flows, SPC, and O/F Ratio

Figure 4-1 shows the APU propellant flow rates as a function of the net useful hydraulic and electric output power. Figure 4-2 plots SPC vs turbine shaft power (turbine shaft power equals the net useful output power plus the power losses in the hydraulic pumps, alternator, gearbox, and lube pump). Both curves are for a 300°R hydrogen inlet temperature and a 550°R hydraulic fluid heat exchanger inlet temperature.

It is possible to generalize on the specific performance of Figures 4-1 and 4-2 by determining the relationship between the required propellant flow and the four primary variables of useful output power, ambient pressure, hydrogen inlet temperature, and hydraulic fluid heat exchanger inlet temperature. Table 4-1 lists the equations used in this performance simplification. Figure 4-3 presents the curves required by the equations. This performance correlation is accurate to within about 2 percent of the actual data at all points within the APU operating envelope.

TABLE 4-1

EQUATIONS FOR APU SPC AND O/F DETERMINATION

$\text{Turbine shp} = 1.05 (\text{net useful output hp}) + 55.2$ $\text{SPC} = \left[ \text{SPC}^* + \delta(\text{SFC})_1 + \delta(\text{SFC})_2 \right] \left[ 1 + (\text{O/F})^* + \delta(\text{O/F})_1 + \delta(\text{O/F})_2 \right]$ $\text{O/F} = (\text{O/F})^* + \delta(\text{O/F})_1 + \delta(\text{O/F})_2$
---

Typical Cycle Operating Conditions

Figures 4-4 through 4-9 are plots of various cycle parameters as a function of the cycle operating condition.

1. O/F Ratio

Figure 4-4 shows the relation between the hydrogen inlet temperature to the APU and the O/F required to obtain the desired turbine inlet temperature. The O/F ratio at low power output is greater than that at high power levels since most of the power losses are fixed. Thus, at low power levels, the waste heat added per unit of hydrogen flow is considerably larger than at high power levels. At a constant power level, the O/F variation with temperature is nonlinear because of the varying specific heat of hydrogen at these cryogenic temperatures.

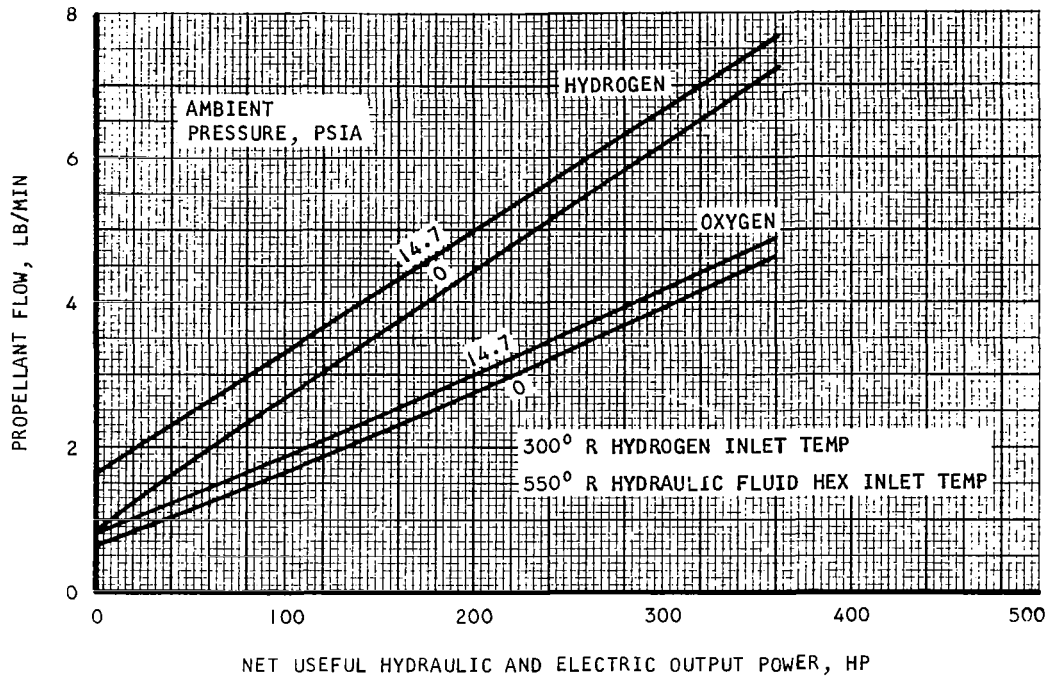


Figure 4-1. Typical APU Propellant Flow Required

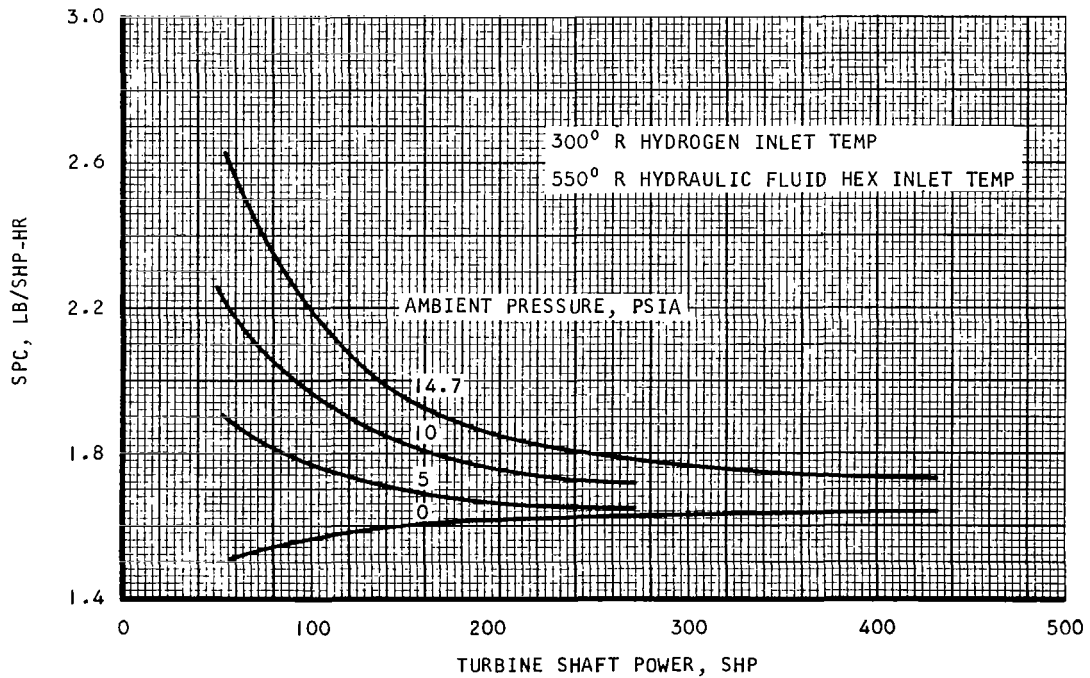


Figure 4-2. Typical APU SPC

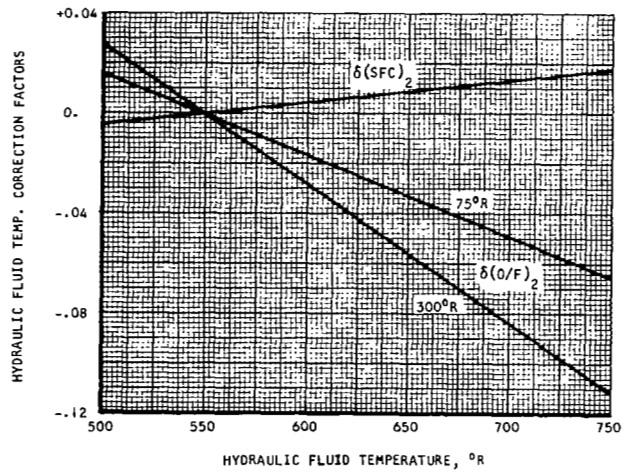
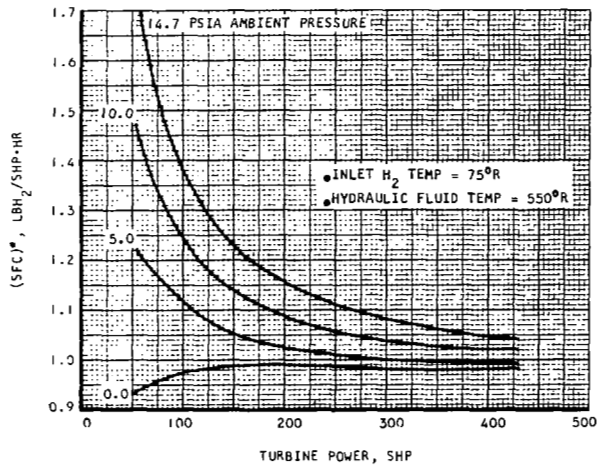
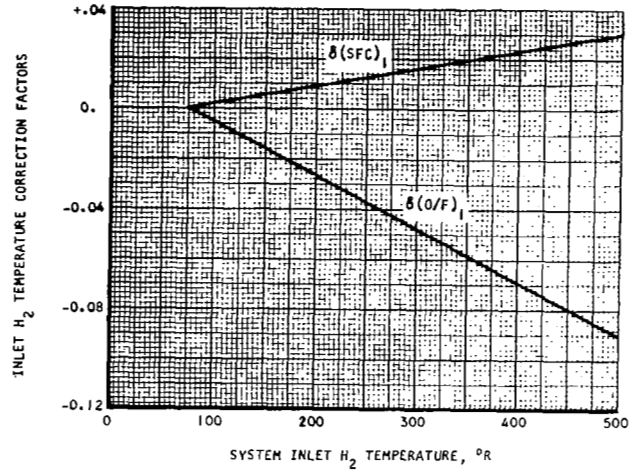
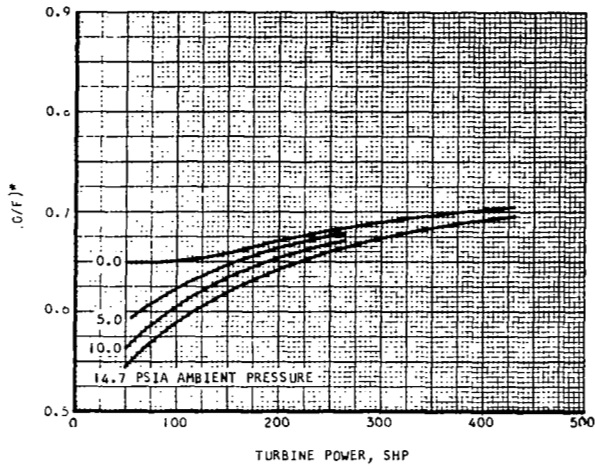


Figure 4-3. Data for SFC and O/F Determination

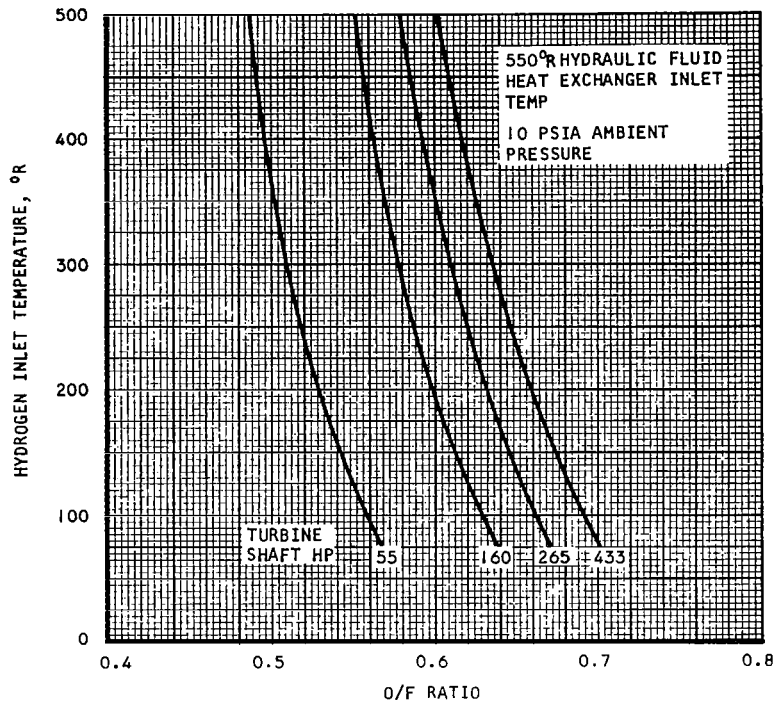


Figure 4-4. Typical Hydrogen Inlet Temperature vs O/F Ratio

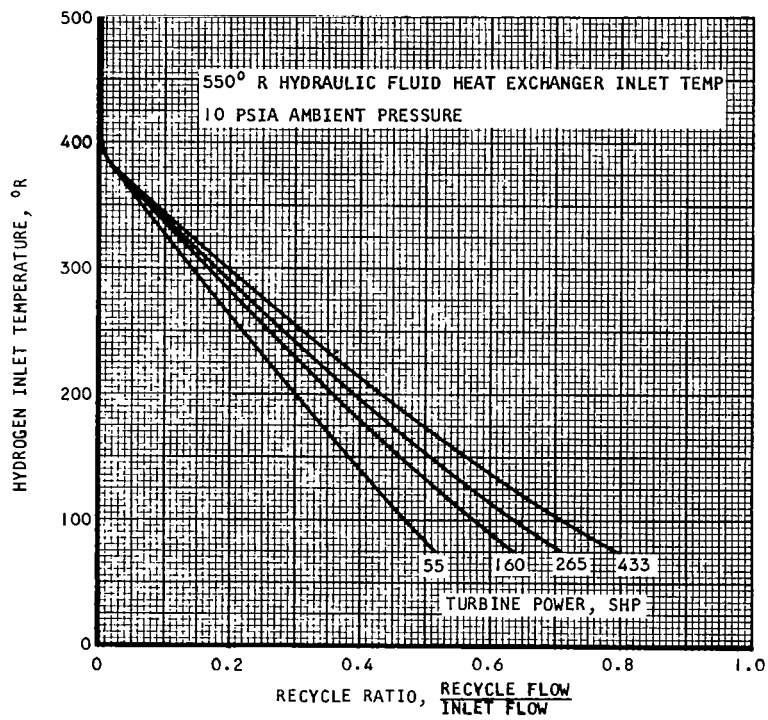


Figure 4-5. Typical Hydrogen Inlet Temperature vs Recycle Flow



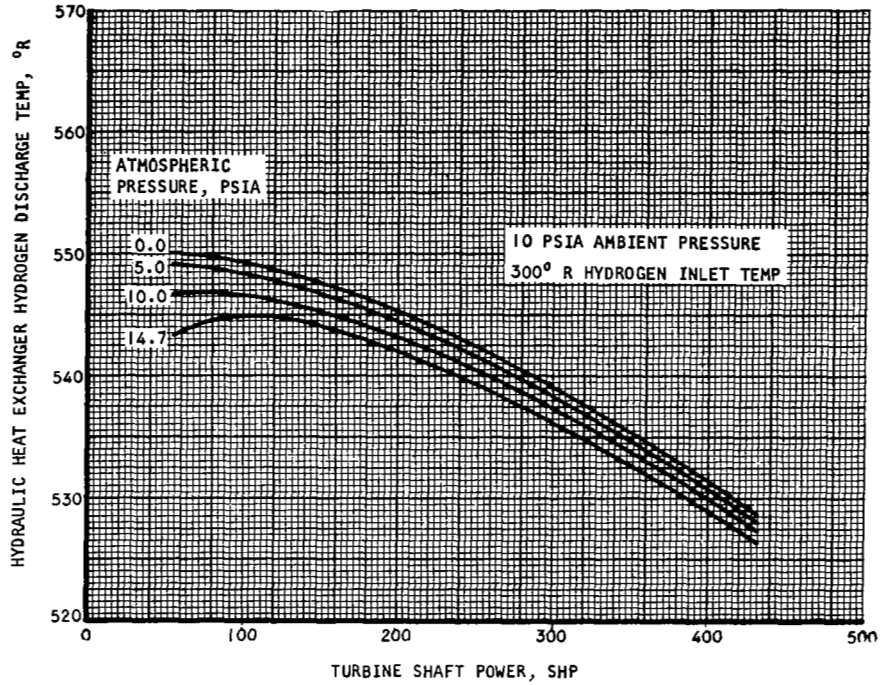


Figure 4-6. Hydraulic Heat Exchanger Hydrogen Discharge Temperature vs Turbine Shaft Power

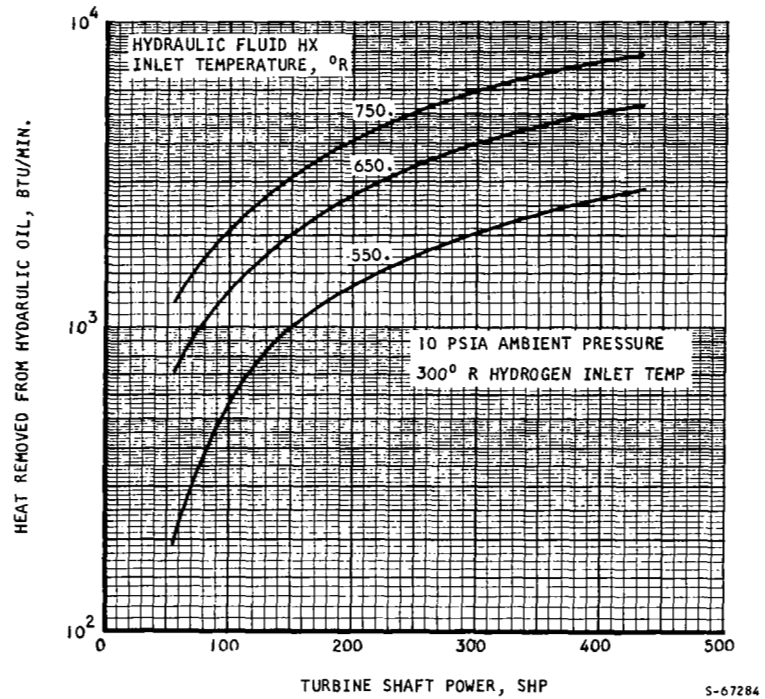


Figure 4-7. Hydraulic Fluid Heat Rejection vs Turbine Shaft Power

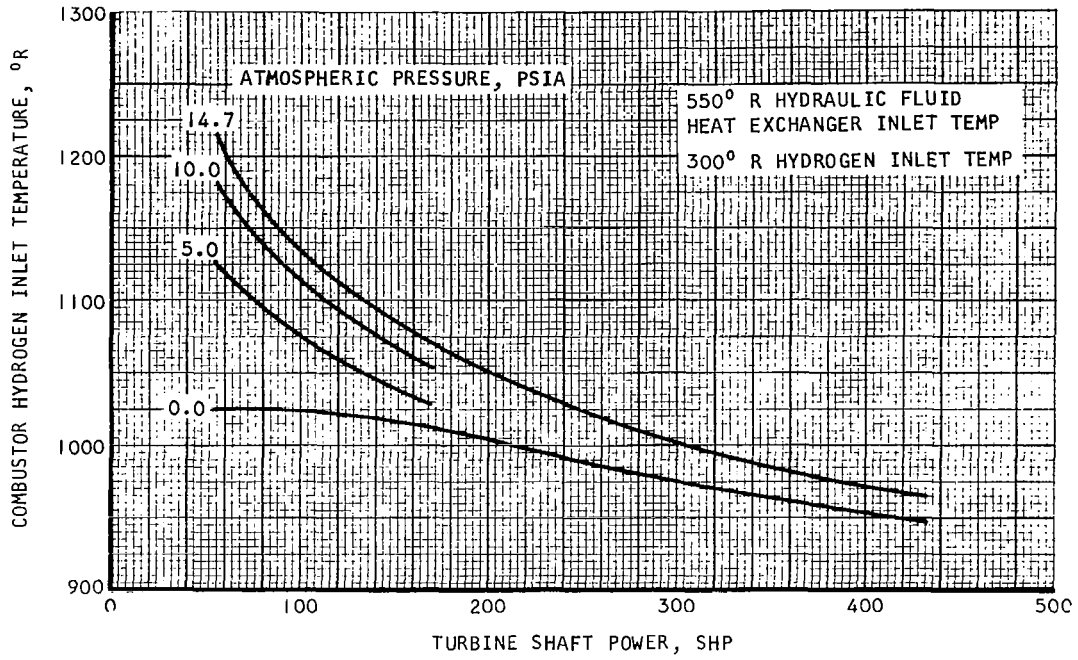


Figure 4-8. Combustor Hydrogen Inlet Temperature vs Turbine Shaft Power

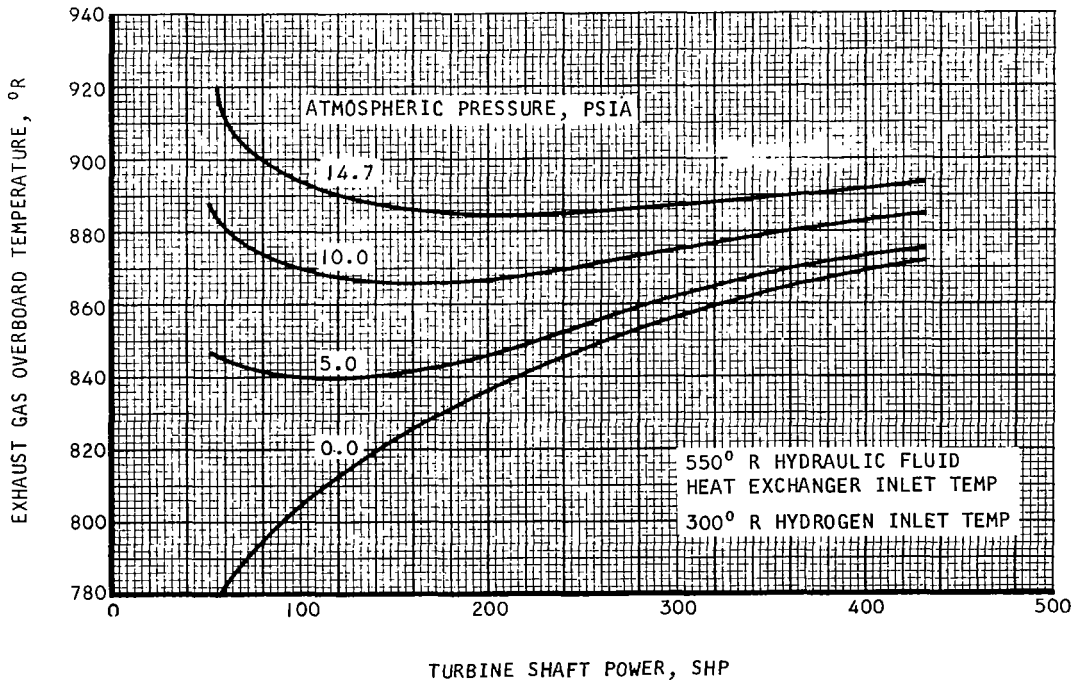


Figure 4-9. Exhaust Gas Overboard Temperature vs Turbine Shaft Power

## 2. Recycle Flow

Figure 4-5 plots the hydrogen inlet temperature-hydrogen recycle flow relationship. The hot recycle flow is used to control the jet pump discharge temperature to  $400^{\circ}\text{R}$ . When the inlet hydrogen temperature exceeds  $400^{\circ}\text{R}$ , no recycle flow is required. Again, there is a slight variation in the amount of recycle flow with power level. This is due to the following reasons:

- Heat exchanger effectiveness is improved as the throughflow (and hence output power) is reduced.
- Hydrogen inlet temperature to the recuperator at low power levels is higher than that at high outputs because the heat added in the oil coolers per unit of hydrogen flow is greater.

## 3. Hydraulic Fluid Heat Removal

Figure 4-6 gives the hydrogen temperature leaving the hydraulic fluid heat exchanger as a function of the turbine shaft power and ambient pressure for a fixed hydraulic fluid inlet temperature to the hydraulic oil cooler. The data indicate that the hydrogen temperature closely approaches the hydraulic fluid temperature at low power levels and that there is about a  $20^{\circ}\text{R}$  difference at high outputs.

Figure 4-7 plots the heat rejected to the hydrogen flow by the hydraulic fluid as a function of the hydraulic fluid inlet temperature and the power output. The heat generated by the hydraulic pumps is about 1700 Btu/min so that heat rejections below this level will result in an increase in the hydraulic fluid temperature. The maximum hydraulic fluid temperature is limited to  $750^{\circ}\text{R}$  by NASA direction. The APU can maintain the hydraulic fluid below this temperature under all operating conditions except prolonged operation at power levels below 80 shp. The selected APU cycle provides better hydraulic fluid cooling than any of the other APU cycles considered in Phase II (described in Volume IV).

## 4. Combustor Hydrogen Inlet Temperature

Figure 4-8 shows the hydrogen temperature at the combustor inlet as a function of the turbine shaft power. The difference between the hydrogen temperature at the combustor inlet and that at the hydraulic oil cooler discharge (Figure 4-6) is the temperature rise in the recuperator. The high hydrogen combustor inlet temperatures indicate that the recuperator is located in such a place in the cycle so as to maximize the exhaust heat utilization. An advantage of the selected cycle is that it always operates to maximize exhaust heat utilization, regardless of hydrogen inlet temperature to the APU. Thus, at high hydrogen APU inlet temperatures, the combustor inlet temperature is significantly higher than at low hydrogen inlet temperatures. This feature is a direct result of selecting an APU cycle capable of operating at a variable O/F ratio. A cycle operating at a fixed O/F would have to waste almost all of the turbine exhaust heat at high hydrogen inlet temperatures, while only using the exhaust effectively at the minimum hydrogen inlet temperature.

## 5. Exhaust Gas Overboard Temperature

Figure 4-9 shows the exhaust gas overboard temperature as a function of the turbine shaft power output. At a low ambient pressure, the turbine pressure ratio remains high over the full range of output power so that the turbine exhaust temperature is about constant; consequently, the exhaust gas overboard temperature continues to decrease with power output due to the improved recuperator effectiveness and reduced recycle flow obtained at low power output. At higher ambient pressures, the turbine pressure ratio starts to decrease as the power is reduced; hence, the turbine efficiency decreases and the turbine exhaust temperature increases. The turbine exhaust temperature increases more than offsets the improved recuperator effectiveness and the reduced recycle flow so that the exhaust gas overboard temperature increases.

### TRANSIENT PERFORMANCE

The APU transient performance was evaluated by using both analog and digital transient performance programs. The analog model (described in Section 7 of Volume IV) was confined to simulation of the line accumulator and pressure drop effects to investigate the relationships between the control concept and the pressure regulator/flow control valve stability. That study indicated that continuous flow can be assumed throughout the APU if the controls concept is correctly chosen. This conclusion is consistent with AiResearch findings on transient performance in both aircraft APU's and aircraft environmental control systems, both of which have components similar to those of the Space Shuttle APU.

The transient digital performance program (which assumes continuous flow throughout the APU) was then used to establish the system controls concepts and performance. The selected controls are described in Section 7 of this volume. The resulting system transient performance is described in the following pages.

The transient performance can be divided into five classes of conditions:

- Startup
- Response to inlet propellant pressure changes
- Response to inlet propellant temperature changes
- Response to output power demand changes
- Shutdown

If the APU is supplied from pumps integral to the APU system, there will be no significant changes in the inlet propellant conditions. However, if the APU is supplied with high-pressure gas from the APS accumulators rapid changes in propellant pressure and temperature are possible.

Most of the transient performance graphs use greatly expanded scales to accurately reflect the data provided by the transient digital program. Thus, spikes or droops in a parameter that appear large on the graph are actually very small percentage changes in the parameter.

In fact, the overall change in control parameters over the range of transient conditions investigated was about 1.7 percent in speed, 2.5 percent in turbine inlet temperature, and 2.5 percent in jet pump discharge temperature. And, as discussed in Section 7, improved speed control is easily possible by incorporating load anticipating in the APU controller.

### Hydraulic Pump and Turbine Torque Relationship

Figure 4-10 shows the torque vs speed curves for both the hydraulic pump and the turbine. It should be noted that a variable-displacement pump will adjust its stroke as speed changes in order to maintain the demanded hydraulic output power. Thus, the torque speed curves for the pump appear as hyperbolas bounded by the minimum torque (loss, or drive inefficiency) and the maximum torque. The turbine torque curves are approximately straight lines with a negative slope.

At intermediate power levels, a prolonged torque mismatch between the pump and the turbine will cause a speed runaway. Thus, a torque-balance speed control concept, such as used in turbopumps, is not possible. Therefore, maximum system stability will be obtained by sensing both speed and speed rate of change. Such a concept has been selected for the APU and is described in Section 7; the design is similar to controls used on existing AiResearch turbine generator sets.

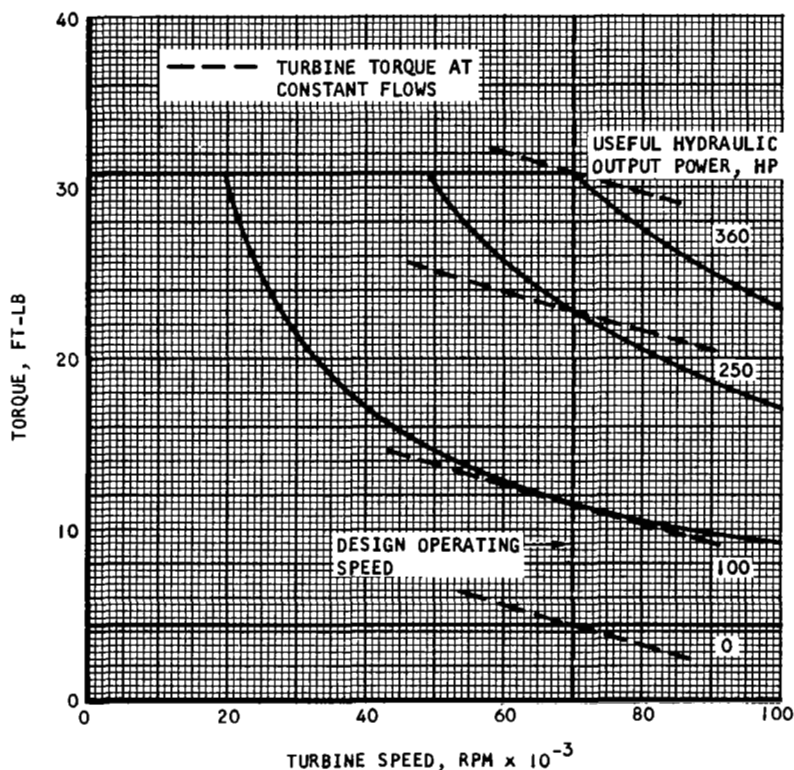


Figure 4-10. Hydraulic Pump and Turbine Torque Relationship

## Startup

Three different startup conditions have been investigated:

- Startup with ambient hydrogen and cold oxygen (300°F)
- Startup with both propellants at ambient (500°F) temperature
- Startup with cold hydrogen (75°F) and cold oxygen (300°R)

In all cases, it is assumed that the APU itself is initially at an ambient temperature of 500°R.

### 1. Ideal Control Startup

To allow comparison of the actual performance with that obtained by an ideal control system (one that can instantly adjust the system operating parameters to the desired values), Figure 4-11 shows the system performance under ideal control for a startup with ambient hydrogen and cold oxygen propellants. The sudden change in output power occurring when the turbine reaches 70,000 rpm is not possible with actual controls. In an actual control system, the controller must start reducing the output power as the speed approaches the desired value in order to avoid an excessive overshoot.

### 2. Actual Control Startup with Ambient Hydrogen and Cold Oxygen

Figure 4-12 shows the actual system performance obtainable with realistic control components for the same conditions. The data indicate that full speed can be reached in about 1.4 sec. Such acceleration capability is possible because the APU is started under no load (other than its internal losses and the fixed torque losses of the hydraulic pumps and the alternator) and because the inertia of the rotating assembly is quite small. Normally in air-breathing APU where a compressor is required, much of the startup torque is required to offset the compressor loading and the significantly higher rotating assembly inertia.

Because of the high density associated with the cryogenic oxygen, the oxygen flow through its flow control valve is maximized for a given valve position. And, as discussed in more detail in Section 7, the oxygen valve position is fixed relative to the hydrogen valve position during that portion of startup at speeds below 20,000 rpm. Thus, the initial oxygen valve position is selected to provide a relatively cold startup to minimize the temperature transients at the turbine disks. With the 300°R oxygen inlet temperature, the initial turbine inlet temperature is about 1650°R.

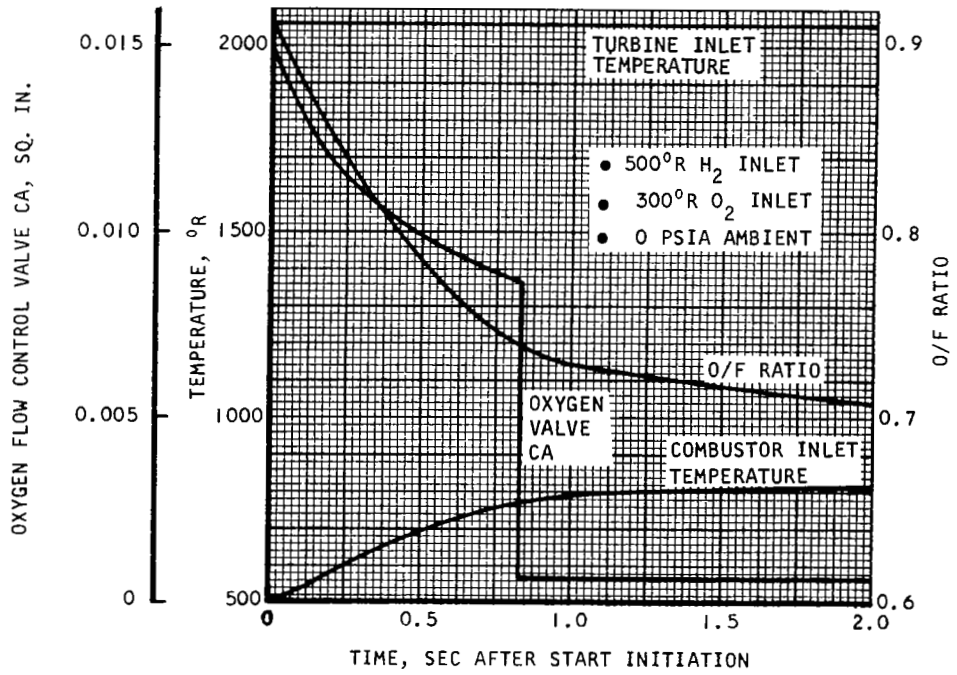
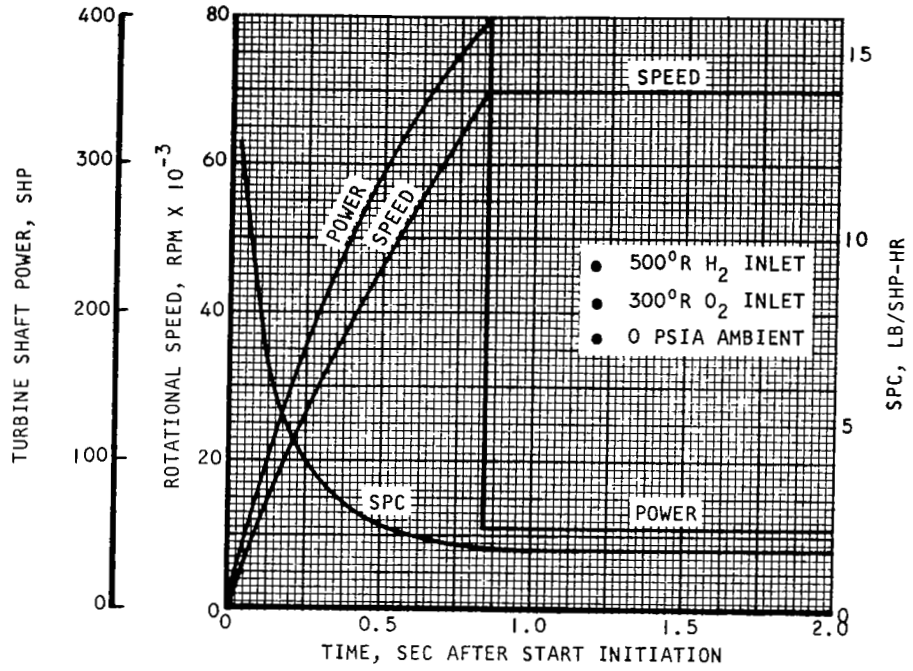


Figure 4-11. Ideal Performance, Ambient Startup

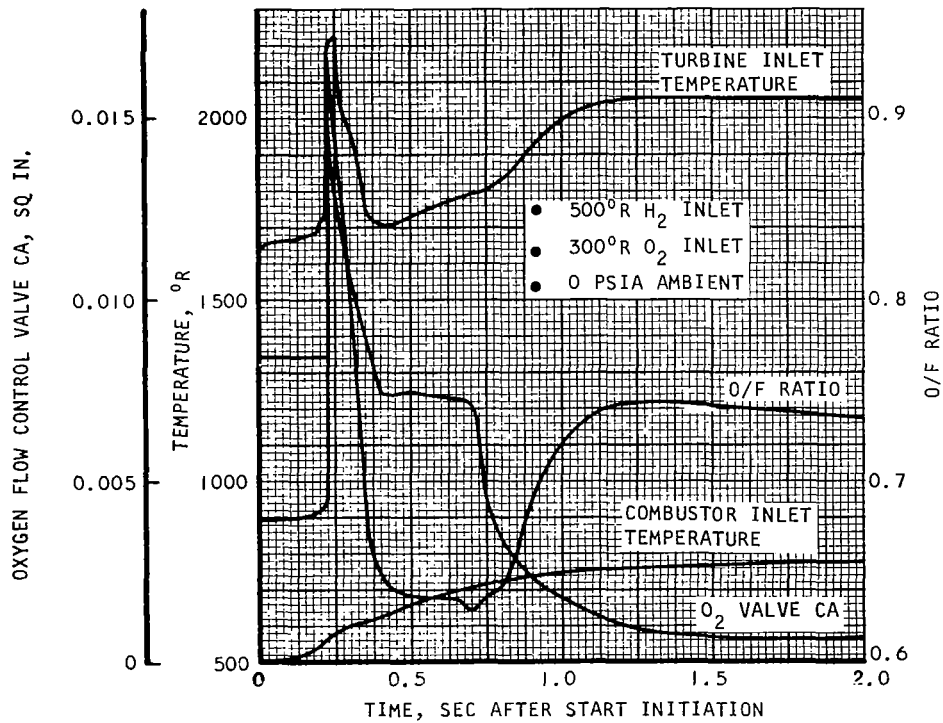
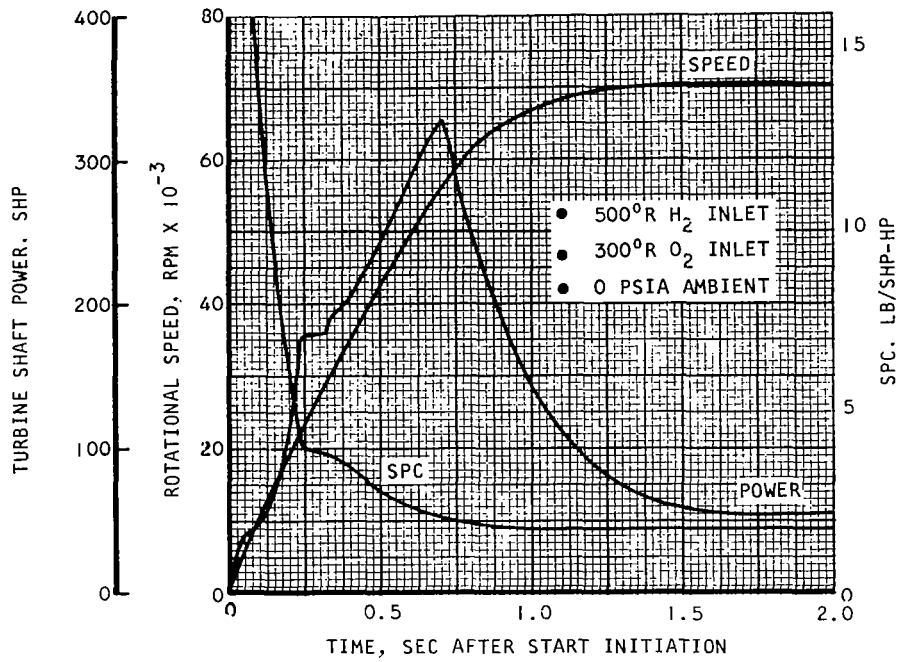


Figure 4-12. Actual Performance, 500°R Hydrogen Startup



### 3. Actual Control Startup with Both Propellants at Ambient Temperatures

Figure 4-13 shows the system startup transient when both the oxygen and hydrogen are supplied to the APU at 500°R throughout the startup. The data indicate a lower initial turbine inlet temperature (1350°R) than in the above case with 300°R oxygen on startup. The temperature reduction is due to the reduced flow obtainable through the valve for a fixed position.

### 4. Actual Control Startup with Cold Hydrogen (75°R) and Cold Oxygen (300°R)

If the APU propellants are supplied by the APS accumulators, the lines from the accumulators to the APU will probably approach ambient temperatures so that a cold propellant startup is not probable. However, if the APU propellants are provided by pumps integral with the APU, then the initial propellant temperatures will approach the pump discharge temperatures.

Figure 4-14 shows the system performance in such a case. The thermal inertia of the hydrogen preheater causes the initial jet pump discharge temperature to be equal to the ambient temperature of 500°R. As the incoming hydrogen flow cools the preheater, the temperature at which recycle hydrogen flow for heating the preheater is available is sufficient to maintain continuous control of the jet pump discharge temperature. Since this temperature can be maintained at its steady-state value throughout this most extreme of startup transients, it can be concluded that the temperatures of the lube oil and hydraulic fluid will remain at satisfactory levels throughout startup.

It should be noted that accommodation of this startup transient is a feature peculiar to the selected cycle configuration since it places a thermal inertia (preheater) in front of the hydraulic and lube oil coolers. Thus, hydrogen at temperatures lower than about 390°R is never introduced into the oil coolers.

### Response to Inlet Propellant Pressure Changes

The effects of propellant pressure changes were assessed using the analog program, which considers compressibility effects in the APU ducting. The study indicated that correct selection of control concept will provide continuous flow with no pressure variations.

### Response to Inlet Propellant Temperature Changes

NASA has specified that the hydrogen temperature may vary at the APS accumulator from 75° to 500°R in 2 sec. The effect of such a temperature transient is not as severe as the startup case with 75°R hydrogen. Consequently, the hydrogen temperature transient has not been analyzed by itself.

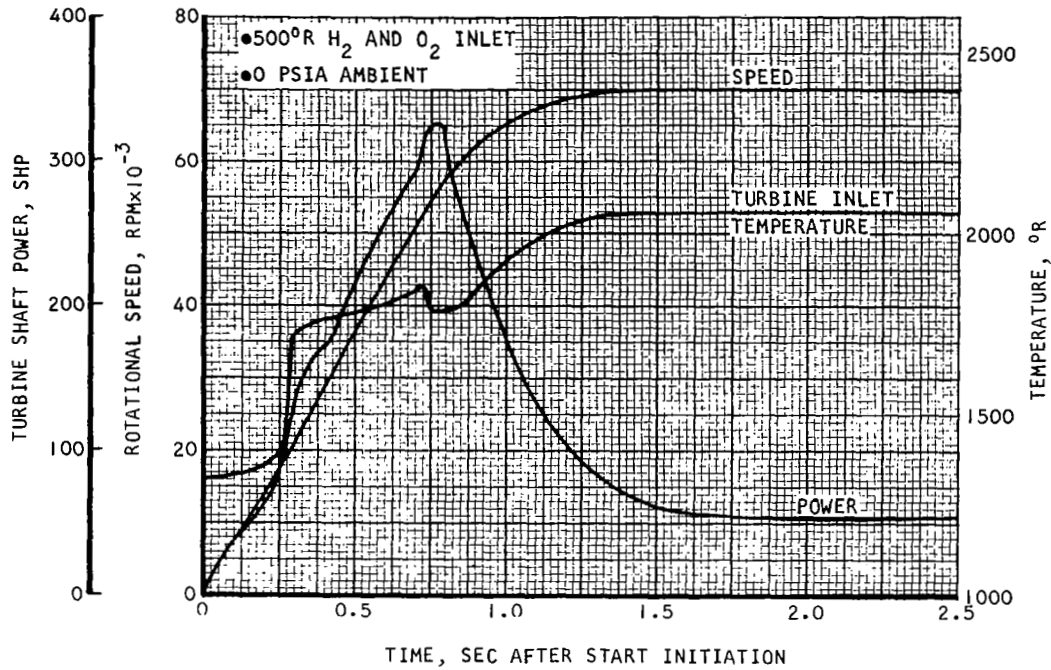


Figure 4-13. Actual Performance. Ambient Propellants

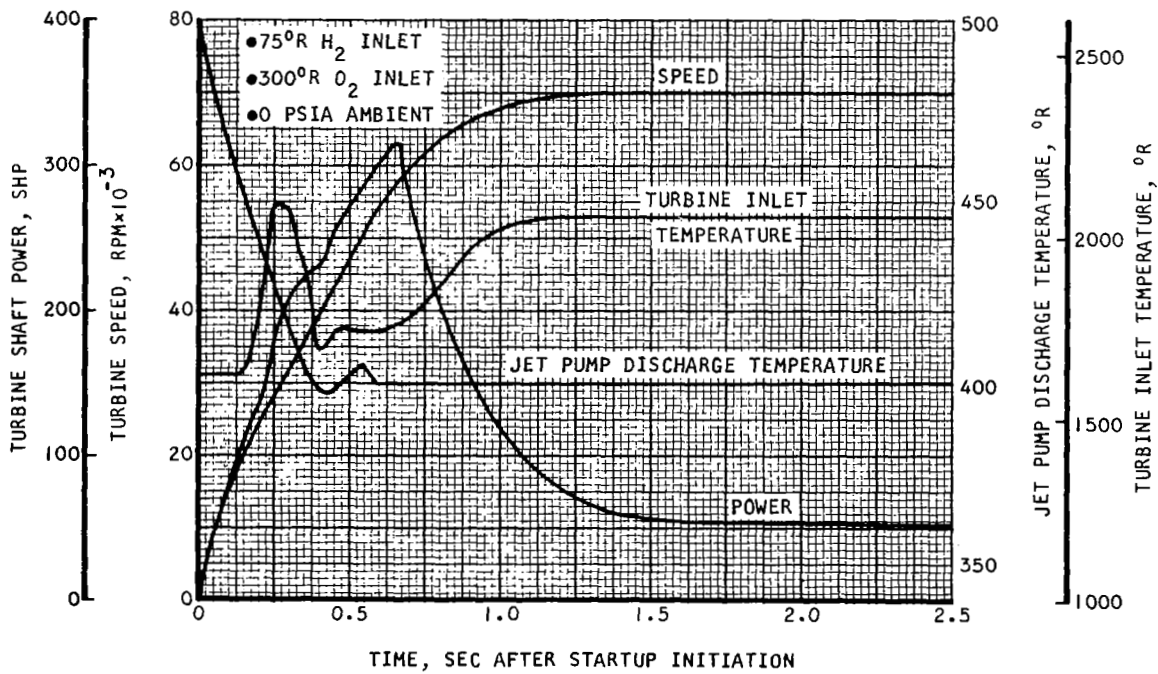


Figure 4-14. Actual Performance. 75°R Hydrogen Startup

The oxygen APS accumulator temperature may vary from 300 to 500°R in 2 sec. However, unlike the hydrogen lines, the oxygen line and the oxygen valving at the APU represent a large thermal inertia so that the temperature transient at the APU combustor will not be more than about 20°R/sec, or 20 percent of that at the APS accumulators. However, to assess the effects of the oxygen temperature transient, it was assumed that the oxygen temperature could vary from 300° to 500°R in 1 sec, or twice the APS accumulator rate of change, and 10 times the anticipated APU combustor inlet rate of change. Figure 4-15 shows the system response to such a condition for 180 shp useful output.

#### Response to Output Power Demand Changes

Several different load steps have been considered:

- Stepup from 0 to 100 hp useful output
- Stepup from 0 to 180 hp useful output
- Stepup from 0 to 300 hp output with simultaneous oxygen temperature ramp change

It will be noted that the system response to the speed change occurring with the load step does not result exactly in valve repositioning at a constant O/F ratio (the turbine inlet temperature changes slightly). This is because the controller translates the commanded hydrogen and oxygen flows into valve position commands by using a single curve relating valve flow to valve area. Complete decoupling of the speed command from an effect on turbine inlet temperature would require a family of valve flow/area curves for different O/F ratios and turbine inlet temperatures. But, the simplicity offered by the selected control concept appears justified. The temperature is controlled to within about 3 percent, which is more than adequate.

Figures 4-16 through 4-18 show the system response to each of these conditions. The data indicate tight speed and temperature control can be maintained throughout these step load changes (the actual APU will only see ramp power changes, dictated by the response capability of the hydraulic pump).

#### Shutdown

Figure 4-19 shows the APU speed vs time relationship occurring during shutdown. The relation is approximately a straight line, since it was assumed that the hydraulic pump and alternator losses are constant torque losses, independent of speed.

Table 4-2 shows the equilibrium temperatures for each of the system heat exchangers after a typical shutdown condition. The data indicate that the temperatures in the lube and hydraulic oil coolers are well above the congealing point of either fluid. Further, the recuperator temperature is over 300°R above the point at which steam might be condensed from the turbine exhaust. The low temperature of the hydrogen preheater is also quite acceptable, since this unit is a hydrogen-to-hydrogen heat exchanger.

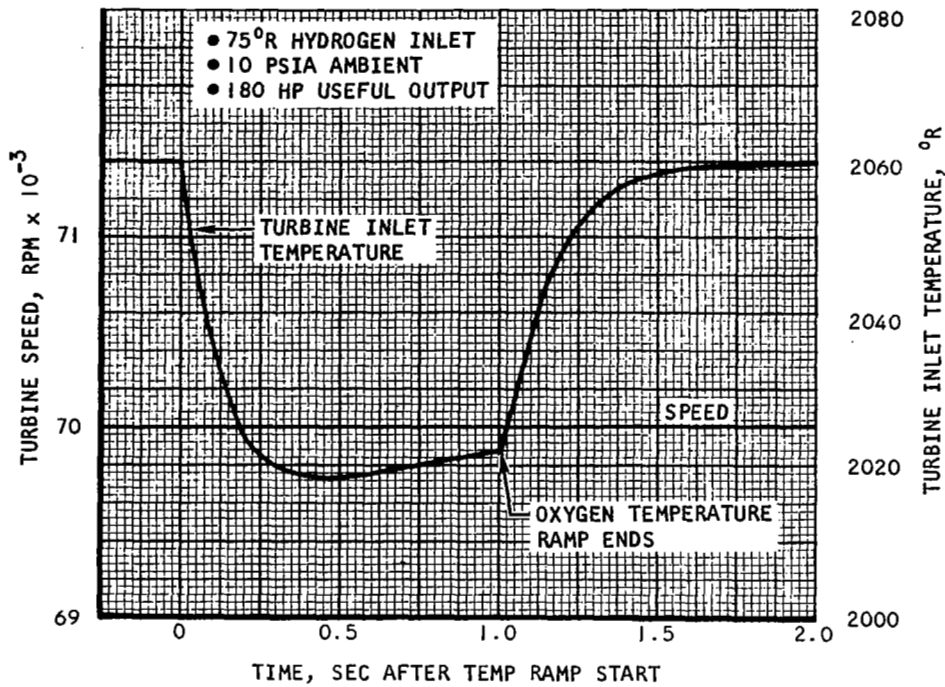


Figure 4-15. Actual Performance Ramp Change in Oxygen Inlet Temperature from 300° to 500°R in 1 Sec

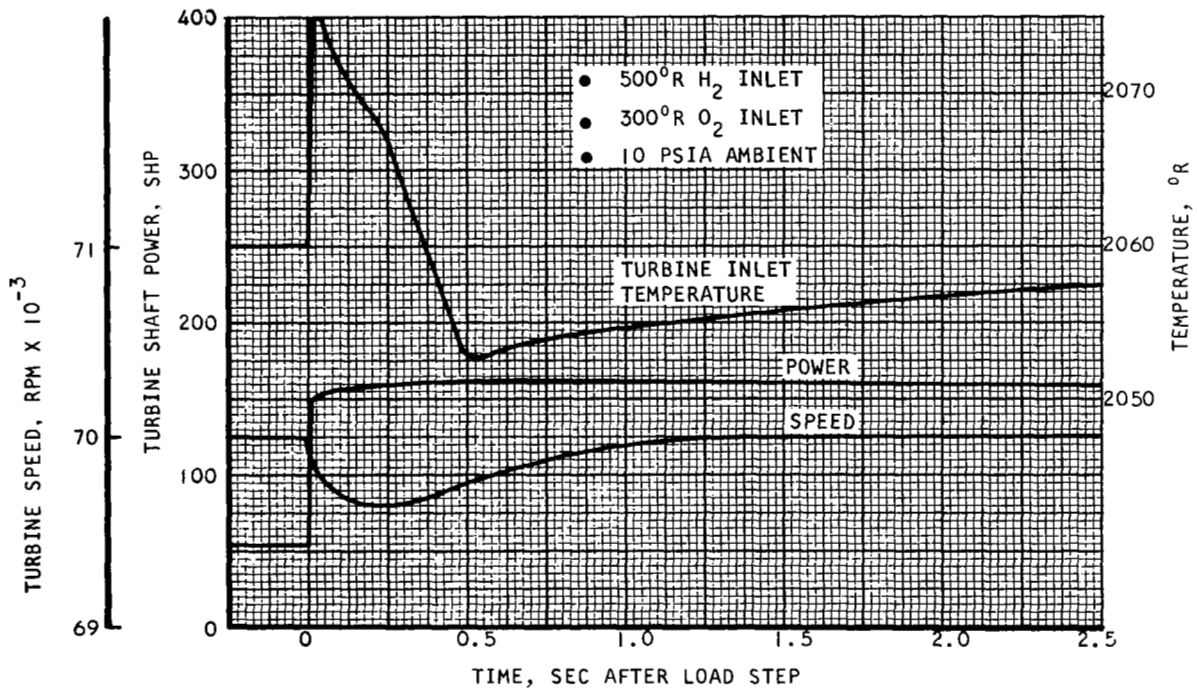


Figure 4-16. Actual Performance, Load Stepup from 0 to 100 Hp Useful Output

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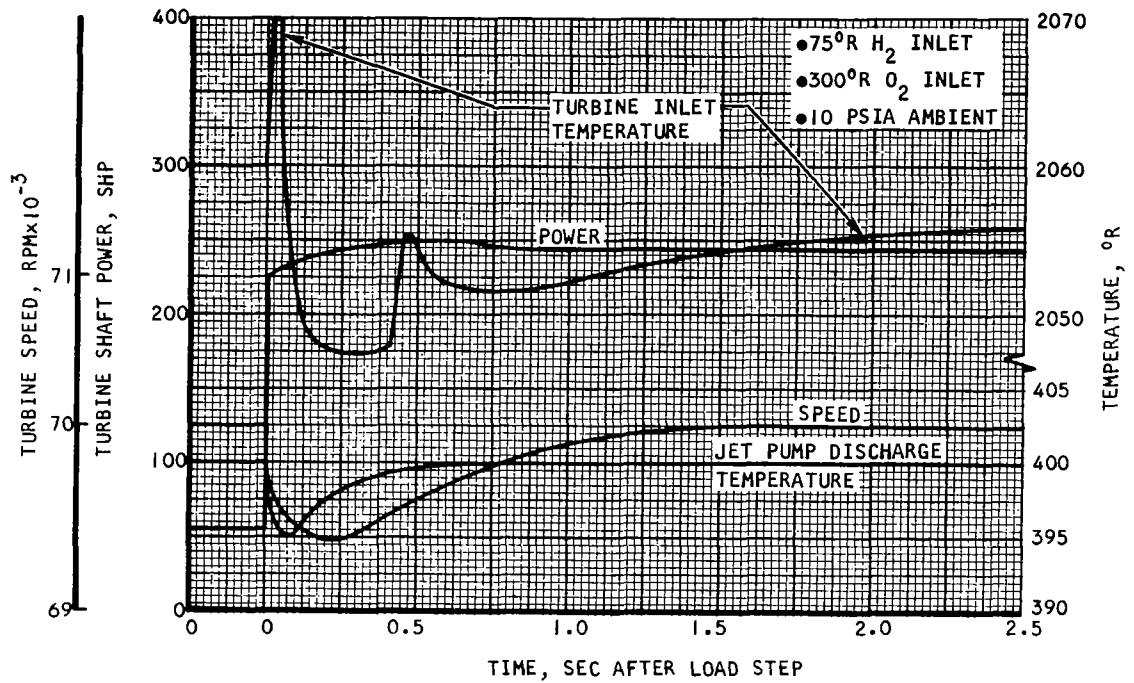


Figure 4-17. Actual Performance, Load Stepup from 0 to 180 Hp Useful Output

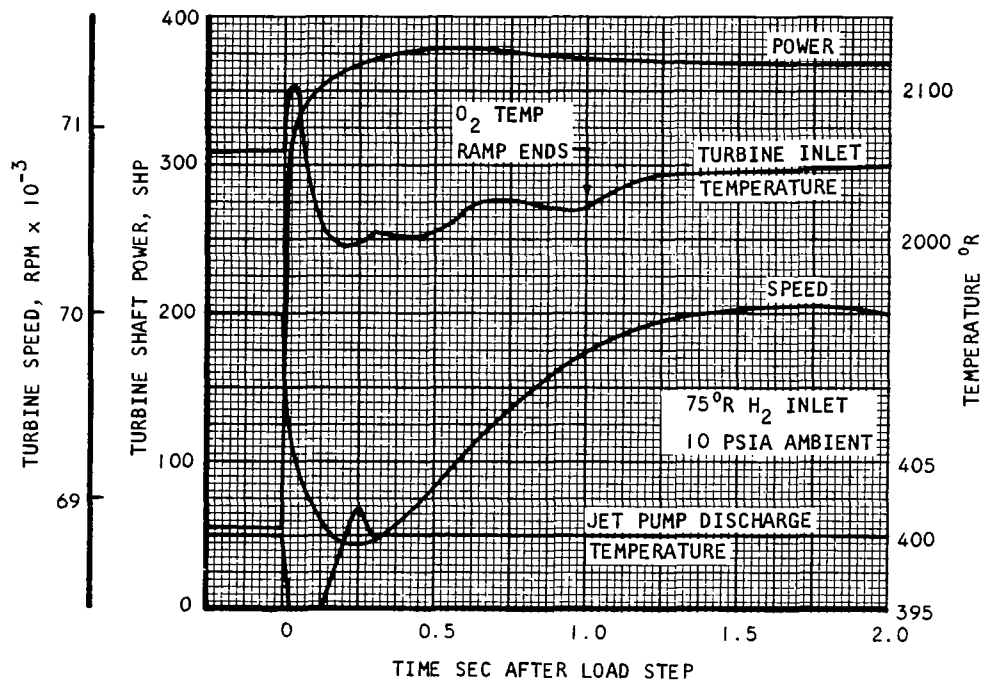


Figure 4-18. Actual Performance, Load Stepup from 0 to 300 Hp Useful Output with Simultaneous 1-Sec Ramp of Oxygen Temperature from 300° to 500°R

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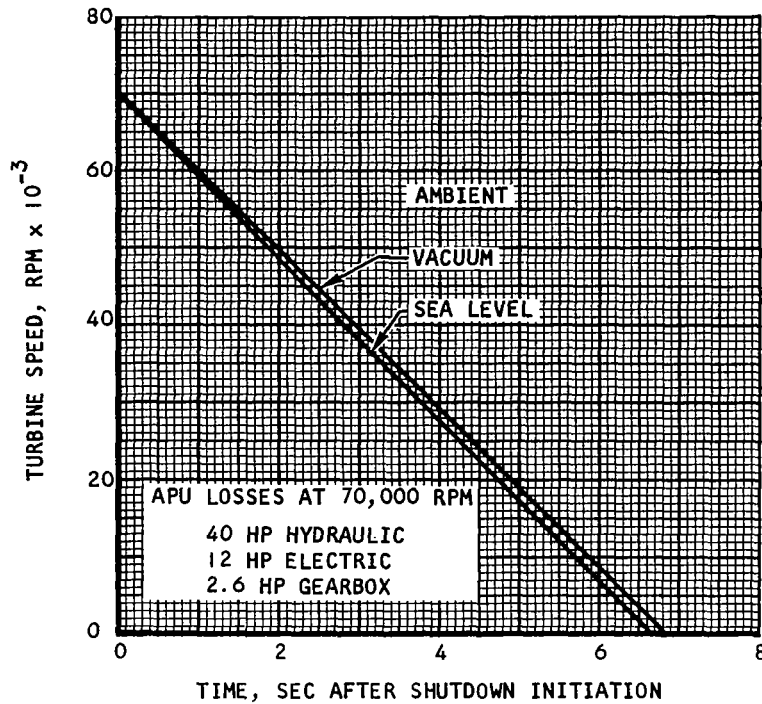


Figure 4-19. Turbine Speed/Time Relation During Shutdown

Summary

Table 4-3 summarizes system transient performance. The data indicate that the selected system can provide close speed and temperature control over the entire operating regime. Of particular importance is the system capability of starting with 75°R hydrogen inlet temperature. Such a feature eliminates the need for a separate start bottle.

TABLE 4-2

HEAT EXCHANGER EQUILIBRIUM TEMPERATURES AFTER SHUTODWN

Heat Exchanger	Temperature, °R
Hydrogen preheater	383
Lube oil cooler	521
Hydraulic oil cooler	520
Recuperator	962

Shutdown conditions: Steady-state at 0 hp useful output with 75°R hydrogen and 300°R oxygen inlet temperatures, 550°R hydraulic fluid heat exchanger inlet temperature, 10 psia ambient

TABLE 4-3  
SYSTEM TRANSIENT PERFORMANCE SUMMARY

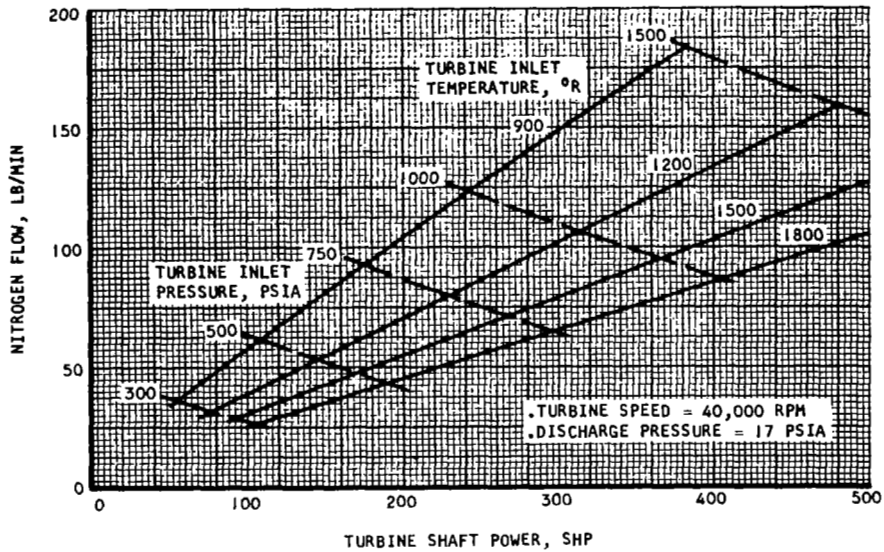
Transient	Maximum Error Range, Percent		
	Turbine Inlet	Turbine Speed	Jet Pump Discharge Temperature
Startup with Ambient Propellants	+0 -35.6*	+0.35	-
Startup with Cold Propellants	+1.94 -11.4*	+0.35	+1.5 -1.25
Inlet Oxygen Temperature Ramp (10 times maximum anticipated rate)	±2.04	±0.0013	-
Load Steps	±3.15	±1.71	±2.25

\* Controls intentionally designed for low turbine inlet temperature at speeds below 20,000 rpm

#### APU OPERATION FROM INERT GAS

NASA has specified that the APU be designed to operate from inert gas (nitrogen is assumed) while the vehicle is on the ground. During this period, the vehicle will only require hydraulic power. Consequently, it is not necessary to operate at the turbine design speed (where 400-Hz electric power would be available), nor is it necessary to provide close speed control. Therefore, the APU will be operated at 40,000 rpm, instead of 70,000 rpm, while on inert gas. The 40,000 rpm speed is selected to provide good turbine performance, while still operating at a speed well above the rotating assembly rigid body criticals.

The inert gas will be supplied to the turbine through a valve placed in the combustor discharge line. This shutoff valve is the only additional piece of equipment required to allow ground operation. It is assumed that turbine speed control will be accomplished by monitoring the APU speed sensor (a special connection line can be placed on the APU controller front for this purpose) and throttling the inert flow at the ground gas supply cart. Figure 4-20 shows the required inert gas flow vs the developed turbine shaft power for a wide range of inert gas supply temperatures and pressures. These data indicate that 100 hp of useful power (about 160 shp) can be provided by using inert gas at a pressure of 600 psia and a temperature of 1200°R. The low temperature and rotational speed while on the ground indicate that prolonged operation can be sustained without affecting the turbine disk material design criteria for hydrogen-oxygen operation.



5-67219

Figure 4-20. APU Ground Inert Gas Flow vs Developed Power

#### INTEGRATED MISSION PERFORMANCE

NASA has specified typical booster and orbiter missions to be used as baselines for establishing mission propellant requirements and APU operating conditions. AiResearch developed during Phase I of this study a computer program for integrating the APU propellant requirements over any specified mission. Figures 4-21 and 4-22 show the outputs from this program for the booster and orbiter missions. Figure 4-21 assumes that the APU is supplied with 300°R hydrogen and Figure 4-22 is for 75°R hydrogen. The total APU propellant requirements are summarized in Table 4-4.

TABLE 4-4

#### APU MISSION PROPELLANT REQUIREMENTS

	APU Hydrogen Inlet Temperature			
	300°R		75°R	
	Hydrogen, lb	Oxygen, lb	Hydrogen, lb	Oxygen, lb
Booster mission	195.9	112.7	194.9	116.4
Orbiter mission	177.5	109.0	177.1	111.0



\* BOOSTER MISSION \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE		
1	-1000.0	-800.0	8.0	-0	-0	-0	0	0	0	5.75	0.00	3.16	0.00	3800.00
2	-800.0	-500.0	103.0	-0	-0	-0	0	0	0	16.53	0.00	10.50	0.00	3800.00
3	-500.0	0	33.0	-0	-0	-0	0	0	0	17.88	0.00	10.42	0.00	3800.00
4	0	40.0	19.0	104.0	8.0	1.0	5925.9	0	0	1.95	0.43	1.59	0.27	3800.00
5	40.0	120.0	19.0	279.0	8.0	1.0	40929.9	0	0	1.60	0.77	1.01	0.54	3800.00
6	120.0	200.0	19.0	104.0	16.0	1.0	125000.0	0	0	1.30	0.73	0.85	0.60	3800.00
7	200.0	595.0	34.0	-0	-0	-0	106000.0	0	0	9.89	0.00	6.46	0.00	3800.00
8	595.0	800.0	135.0	-0	-0	-0	31000.0	0	0	1.29	0.00	1.19	0.00	3800.00
9	800.0	808.0	35.0	-0	-0	-0	30000.0	0	0	1.13	0.00	1.08	0.00	3800.00
10	808.0	896.0	50.0	-0	-0	-0	30000.0	0	0	1.30	0.00	1.00	0.00	3800.00
11	896.0	7800.0	6.0	28.0	1489.0	1.0	30000.0	0	0	75.94	27.94	44.38	16.89	3800.00
12	7800.0	8090.0	10.0	57.0	80.0	1.0	20000.0	0	0	5.25	2.22	3.12	1.59	3800.00
13	8090.0	8100.0	82.0	-0	-0	-0	10000.0	0	0	1.46	0.00	1.29	0.00	3800.00
14	8100.0	8105.0	237.0	-0	-0	-0	10000.0	0	0	1.45	0.00	1.30	0.00	3800.00
15	8105.0	8115.0	67.0	-0	-0	-0	10000.0	0	0	1.42	0.00	1.26	0.00	3800.00
16	8115.0	8145.0	42.0	-0	-0	-0	5000.0	0	0	1.82	0.00	1.09	0.00	3800.00
17	8145.0	8465.0	42.0	-0	-0	-0	0	0	0	9.35	0.00	3.63	0.00	3800.00
18	8465.0	9000.0	9.0	53.0	180.0	1.0	0	0	0	10.14	3.38	9.62	8.07	3800.00

TOTAL ENERGY OUTPUT 3064.3  
 TOTAL AVERAGE POWER OUTPUT 18.0  
 TOTAL HYDROGEN REQUIRED 194.90  
 TOTAL OXYGEN REQUIRED 116.39  
 TOTAL VALVE CYCLES REQUIRED 0

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

\* ORBIT MISSION ASCENT PHASE \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE		
1	-1000.0	-995.0	302.0	-0	-0	-0	0	0	0	.54	0.00	.37	0.00	3800.00
2	-995.0	-990.0	92.0	-0	-0	-0	0	0	0	1.19	0.00	1.11	0.00	3800.00
3	-990.0	-985.0	32.0	-0	-0	-0	0	0	0	2.97	0.00	1.69	0.00	3800.00
4	-985.0	-20.0	14.0	-0	-0	-0	0	0	0	24.55	0.00	13.34	0.00	3800.00
5	-20.0	0	77.0	-0	-0	-0	0	0	0	.91	0.00	.54	0.00	3800.00
6	0	40.0	17.0	-0	-0	-0	8000.0	0	0	1.03	0.00	.58	0.00	3800.00
7	40.0	80.0	17.0	-0	-0	-0	27000.0	0	0	1.83	0.00	.90	0.00	3800.00
8	80.0	120.0	17.0	-0	-0	-0	54000.0	0	0	1.70	0.00	.84	0.00	3800.00
9	120.0	160.0	17.0	-0	-0	-0	93000.0	0	0	1.68	0.00	.84	0.00	3800.00
10	160.0	190.0	17.0	-0	-0	-0	143000.0	0	0	1.50	0.00	.73	0.00	3800.00
11	190.0	220.0	17.0	-0	-0	-0	312000.0	0	0	1.50	0.00	.73	0.00	3800.00
12	220.0	285.0	45.0	95.0	16.0	1.0	464729.7	0	0	1.49	.65	.32	.43	3800.00
13	285.0	295.0	45.0	95.0	16.0	1.0	474864.9	0	0	1.61	.65	.40	.43	3800.00
14	295.0	335.0	45.0	95.0	16.0	1.0	485675.7	0	0	1.61	.65	.40	.43	3800.00
15	335.0	375.0	45.0	95.0	16.0	1.0	496486.5	0	0	1.61	.65	.40	.43	3800.00
16	375.0	415.0	45.0	95.0	16.0	1.0	510949.9	0	0	1.61	.65	.40	.43	3800.00

TOTAL ENERGY OUTPUT 583.7  
 TOTAL AVERAGE POWER OUTPUT 24.8  
 TOTAL HYDROGEN REQUIRED 35.99  
 TOTAL OXYGEN REQUIRED 22.82  
 TOTAL VALVE CYCLES REQUIRED 0

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

\* ORBIT MISSION DESCENT PHASE \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE		
1	-0	5.0	37.0	-0	-0	-0	517500.0	0	0	1.12	0.00	1.08	0.00	3800.00
2	5.0	100.0	174.0	-0	-0	-0	487500.0	0	0	5.98	0.00	4.11	0.00	3800.00
3	100.0	500.0	81.0	-0	-0	-0	340000.0	0	0	14.63	0.00	9.65	0.00	3800.00
4	500.0	1000.0	10.0	-0	-0	-0	235000.0	0	0	7.24	0.00	4.77	0.00	3800.00
5	1000.0	2000.0	10.0	-0	-0	-0	193333.3	0	0	14.49	0.00	9.54	0.00	3800.00
6	2000.0	3000.0	10.0	-0	-0	-0	153333.3	0	0	14.50	0.00	9.55	0.00	3800.00
7	3000.0	4000.0	10.0	-0	-0	-0	101000.0	0	0	14.61	0.00	9.59	0.00	3800.00
8	4000.0	4500.0	10.0	-0	-0	-0	50700.0	0	0	7.94	0.00	5.09	0.00	3800.00
9	4500.0	4523.0	340.0	-0	-0	-0	30000.0	0	0	2.58	0.00	1.84	0.00	3800.00
10	4523.0	4623.0	101.0	-0	-0	-0	30000.0	0	0	4.49	0.00	2.94	0.00	3800.00
11	4623.0	4730.0	10.0	-0	-0	-0	24250.0	0	0	2.05	0.00	1.22	0.00	3800.00
12	4730.0	4750.0	254.0	-0	-0	-0	14666.7	0	0	1.81	0.00	1.24	0.00	3800.00
13	4750.0	4825.0	82.0	-0	-0	-0	8918.7	0	0	3.28	0.00	2.04	0.00	3800.00
14	4825.0	4830.0	325.0	-0	-0	-0	7000.0	0	0	1.56	0.00	1.39	0.00	3800.00
15	4830.0	4840.0	300.0	-0	-0	-0	7000.0	0	0	1.05	0.00	1.72	0.00	3800.00
16	4840.0	4842.0	360.0	-0	-0	-0	7000.0	0	0	1.24	0.00	1.17	0.00	3800.00

TOTAL ENERGY OUTPUT 2073.8  
 TOTAL AVERAGE POWER OUTPUT 29.7  
 TOTAL HYDROGEN REQUIRED 95.55  
 TOTAL OXYGEN REQUIRED 62.85  
 TOTAL VALVE CYCLES REQUIRED 0

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

\* ORBIT GO-AROUND \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE		
1	-0	20.0	360.0	-0	-0	-0	0	0	0	2.46	0.00	1.70	0.00	3800.00
2	20.0	120.0	100.0	200.0	80.0	1.0	0	0	0	4.14	1.58	2.61	1.06	3800.00
3	120.0	300.0	90.0	-0	-0	-0	0	0	0	6.84	0.00	4.04	0.00	3800.00
4	300.0	320.0	254.0	-0	-0	-0	0	0	0	1.86	0.00	1.27	0.00	3800.00
5	320.0	395.0	80.0	-0	-0	-0	0	0	0	3.48	0.00	2.14	0.00	3800.00
6	395.0	400.0	325.0	-0	-0	-0	0	0	0	1.57	0.00	1.39	0.00	3800.00
7	400.0	410.0	300.0	-0	-0	-0	0	0	0	1.06	0.00	1.73	0.00	3800.00
8	410.0	412.0	360.0	-0	-0	-0	0	0	0	1.25	0.00	1.17	0.00	3800.00
9	412.0	712.0	40.0	-0	-0	-0	0	0	0	10.56	0.00	6.13	0.00	3800.00
10	712.0	812.0	50.0	-0	-0	-0	0	0	0	3.80	0.00	2.25	0.00	3800.00
11	812.0	1012.0	10.0	-0	-0	-0	0	0	0	9.34	0.00	2.87	0.00	3800.00

TOTAL ENERGY OUTPUT 1660.4  
 TOTAL AVERAGE POWER OUTPUT 62.9  
 TOTAL HYDROGEN REQUIRED 41.97  
 TOTAL OXYGEN REQUIRED 25.37  
 TOTAL VALVE CYCLES REQUIRED 0

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

Figure 4-21. APU Mission Propellant Requirements for 300°R Hydrogen Inlet Temperature to APU

\* BOOSTER MISSION \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE
1	-1700.0	-400.0	8.0	-0	-0	-0	.0	.0	.0	5.81 .00	2.95 .00	3000.00
2	-870.0	-500.0	103.0	-0	-0	-0	.0	.0	.0	16.63 .00	9.42 .00	3000.00
3	-500.0	.0	33.0	-0	-0	-0	.0	.0	.0	17.95 .00	9.56 .00	3000.00
4	.0	40.0	19.0	104.0	8.0	1.0	5925.9	.0	.0	.95 .43	.51 .25	3000.00
5	40.0	120.0	19.0	279.0	8.0	1.0	40925.9	.0	.0	1.59 .78	1.02 .50	3000.00
6	120.0	200.0	19.0	104.0	16.0	1.0	125000.0	.0	.0	1.28 .73	.88 .46	3000.00
7	200.0	595.0	34.0	-0	-0	-0	100000.0	.0	.0	9.74 .00	4.44 .00	3000.00
8	595.0	600.0	45.0	-0	-0	-0	31300.0	.0	.0	.29 .00	.18 .00	3000.00
9	600.0	604.0	35.0	-0	-0	-0	30900.0	.0	.0	.13 .00	.08 .00	1000.00
10	604.0	696.0	50.0	-0	-0	-0	30000.0	.0	.0	1.28 .00	.77 .00	1000.00
11	696.0	780.0	6.0	28.0	1485.0	1.0	30000.0	.0	.0	76.29 27.79	44.88 16.97	1000.00
12	780.0	870.0	10.0	57.0	80.0	1.0	20000.0	.0	.0	5.26 2.20	3.10 1.30	3000.00
13	870.0	8100.0	82.0	-0	-0	-0	10000.0	.0	.0	.46 .00	.27 .00	3000.00
14	8100.0	8105.0	237.0	-0	-0	-0	10000.0	.0	.0	.45 .00	.28 .00	3000.00
15	8105.0	8135.1	67.0	-0	-0	-0	10000.0	.0	.0	.42 .00	.24 .00	3000.00
16	8135.0	8185.0	42.0	-0	-0	-0	5000.0	.0	.0	1.82 .00	1.01 .00	3000.00
17	8185.0	8445.0	42.0	-0	-0	-0	.0	.0	.0	9.75 .00	5.07 .00	1000.00
18	8445.0	9000.0	9.0	55.0	100.0	1.0	.0	.0	.0	10.38 3.94	9.11 1.84	1000.00
TOTAL ENERGY OUTPUT						3364.3						
TOTAL AVERAGE POWER OUTPUT						18.0						
TOTAL HYDROGEN REQUIRED						195.92						
TOTAL OXYGEN REQUIRED						112.49						
TOTAL VALVE CYCLES REQUIRED						.0						

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

\* ORBIT MISSION ASCENT PHASE \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE
1	-1000.0	-995.0	302.0	-0	-0	-0	.0	.0	.0	.54 .00	.34 .00	3000.00
2	-995.0	-990.0	57.0	-0	-0	-0	.0	.0	.0	.19 .00	.10 .00	3000.00
3	-990.0	-900.0	32.0	-0	-0	-0	.0	.0	.0	2.98 .00	1.36 .00	3000.00
4	-900.0	-20.0	14.0	-0	-0	-0	.0	.0	.0	24.82 .00	12.80 .00	3000.00
5	-20.0	.0	77.0	-0	-0	-0	.0	.0	.0	.81 .00	.90 .00	3000.00
6	.0	40.0	17.0	-0	-0	-0	8000.0	.0	.0	1.03 .00	.56 .00	3000.00
7	40.0	80.0	17.0	-0	-0	-0	27000.0	.0	.0	.83 .00	.51 .00	3000.00
8	80.0	120.0	17.0	-0	-0	-0	54000.0	.0	.0	.71 .00	.48 .00	3000.00
9	120.0	160.0	17.0	-0	-0	-0	93000.0	.0	.0	.67 .00	.47 .00	3000.00
10	160.0	190.0	17.0	-0	-0	-0	140000.0	.0	.0	.50 .00	.35 .00	3000.00
11	190.0	220.0	17.0	-0	-0	-0	312000.0	.0	.0	.30 .00	.25 .00	3000.00
12	220.0	295.0	45.0	95.0	16.0	1.0	464729.7	.0	.0	.48 .64	.32 .41	3000.00
13	295.0	295.0	45.0	95.0	16.0	1.0	474864.9	.0	.0	.60 .64	.40 .41	3000.00
14	295.0	335.0	45.0	95.0	16.0	1.0	485675.7	.0	.0	.60 .64	.40 .41	3000.00
15	335.0	375.0	45.0	95.0	16.0	1.0	496486.5	.0	.0	.60 .64	.40 .41	3000.00
16	375.0	415.1	45.0	95.0	16.0	1.0	510945.9	.0	.0	.60 .64	.40 .41	3000.00
TOTAL ENERGY OUTPUT						583.7						
TOTAL AVERAGE POWER OUTPUT						24.4						
TOTAL HYDROGEN REQUIRED						39.81						
TOTAL OXYGEN REQUIRED						21.68						
TOTAL VALVE CYCLES REQUIRED						.0						

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

\* ORBIT MISSION DESCENT PHASE \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE
1	-0	5.0	37.0	-0	-0	-0	517500.0	.0	.0	.11 .00	.08 .00	3000.00
2	5.0	100.0	174.0	-0	-0	-0	487500.0	.0	.0	6.00 .00	3.76 .00	3000.00
3	100.0	700.0	81.0	-0	-0	-0	340.000.0	.0	.0	14.40 .00	9.17 .00	3000.00
4	700.0	1000.0	17.0	-0	-0	-0	215000.0	.0	.0	7.24 .00	5.25 .00	3000.00
5	1000.0	2000.0	10.0	-0	-0	-0	193333.3	.0	.0	14.48 .00	10.51 .00	3000.00
6	2000.0	3000.0	11.0	-0	-0	-0	153333.3	.0	.0	14.49 .00	10.51 .00	3000.00
7	3000.0	4000.0	11.0	-0	-0	-0	101000.0	.0	.0	14.40 .00	10.34 .00	3000.00
8	4000.0	4500.0	10.0	-0	-0	-0	50100.0	.0	.0	7.96 .00	5.43 .00	3000.00
9	4500.0	4923.0	340.0	-0	-0	-0	30100.0	.0	.0	2.62 .00	1.66 .00	3000.00
10	4923.0	4623.0	107.0	-0	-0	-0	30300.0	.0	.0	4.47 .00	2.71 .00	3000.00
11	4623.0	4123.0	10.0	-0	-0	-0	24250.0	.0	.0	2.07 .00	1.25 .00	3000.00
12	4123.0	4750.0	254.0	-0	-0	-0	14666.7	.0	.0	1.83 .00	1.14 .00	3000.00
13	4750.0	4825.0	80.0	-0	-0	-0	8916.7	.0	.0	3.26 .00	1.87 .00	3000.00
14	4825.0	4630.0	325.0	-0	-0	-0	7000.0	.0	.0	5.37 .00	3.36 .00	3000.00
15	4630.0	4442.0	300.0	-0	-0	-0	7000.0	.0	.0	1.06 .00	.66 .00	3000.00
16	4442.0	4442.0	367.0	-0	-0	-0	7000.0	.0	.0	1.25 .00	.16 .00	3000.00
TOTAL ENERGY OUTPUT						2073.8						
TOTAL AVERAGE POWER OUTPUT						75.7						
TOTAL HYDROGEN REQUIRED						95.39						
TOTAL OXYGEN REQUIRED						61.05						
TOTAL VALVE CYCLES REQUIRED						.0						

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

\* ORBIT GO-AROUND \*

SEC	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE CYCLES	SPIKE CYCLES	H2 CONSUMED BASE SPIKE	O2 CONSUMED BASE SPIKE	HYDRAULIC PRESSURE
1	-0	20.0	360.0	-0	-0	-0	.0	.0	.0	2.50 .00	1.97 .00	3000.00
2	20.0	120.0	100.0	200.0	20.0	1.0	.0	.0	.0	4.16 1.40	2.34 .97	3000.00
3	120.0	300.0	50.0	-0	-0	-0	.0	.0	.0	4.88 .00	3.69 .00	3000.00
4	300.0	320.0	254.0	-0	-0	-0	.0	.0	.0	1.91 .00	1.17 .00	3000.00
5	320.0	395.0	80.0	-0	-0	-0	.0	.0	.0	3.48 .00	1.93 .00	3000.00
6	395.0	400.0	325.0	-0	-0	-0	.0	.0	.0	.88 .00	.36 .00	3000.00
7	400.0	410.0	300.0	-0	-0	-0	.0	.0	.0	1.08 .00	.67 .00	3000.00
8	410.0	412.0	360.0	-0	-0	-0	.0	.0	.0	3.25 .00	1.16 .00	3000.00
9	412.0	712.0	40.0	-0	-0	-0	.0	.0	.0	10.41 .00	5.43 .00	3000.00
10	712.0	812.0	50.0	-0	-0	-0	.0	.0	.0	8.21 .00	2.05 .00	3000.00
11	812.0	1012.0	13.0	-0	-0	-0	.0	.0	.0	5.42 .00	2.70 .00	3000.00
TOTAL ENERGY OUTPUT						1060.4						
TOTAL AVERAGE POWER OUTPUT						62.9						
TOTAL HYDROGEN REQUIRED						42.27						
TOTAL OXYGEN REQUIRED						33.24						
TOTAL VALVE CYCLES REQUIRED						.0						

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC, PROPELLANT CONSUMED IN LB., PRESSURE IN PSI

Figure 4-22. APU Mission Propellant Requirements for 75<sup>0</sup>R Hydrogen Inlet Temperature to APU

## SECTION 5

### PROPELLANT CONDITIONING/THERMAL CONTROL SUBSYSTEM

#### INTRODUCTION

The propellant conditioning/thermal control subsystem has the following functions:

- (a) Supply propellants at proper temperature and pressure to the turbine power assembly
- (b) Dissipate waste heat generated internally in the turbine power unit at suitable temperature levels for the various system components

The propellant conditioning/thermal management concept, shown schematically in Figure 5-1, uses waste heat from the lubricant, hydraulic fluid, and turbine exhaust gas to preheat the incoming hydrogen. A recycle loop is used in the system to maintain proper hydrogen inlet temperatures to the various heat exchangers. As discussed in Sections 2 and 5 of Volume IV, in this way the heat exchanger design problems and development risk are minimized.

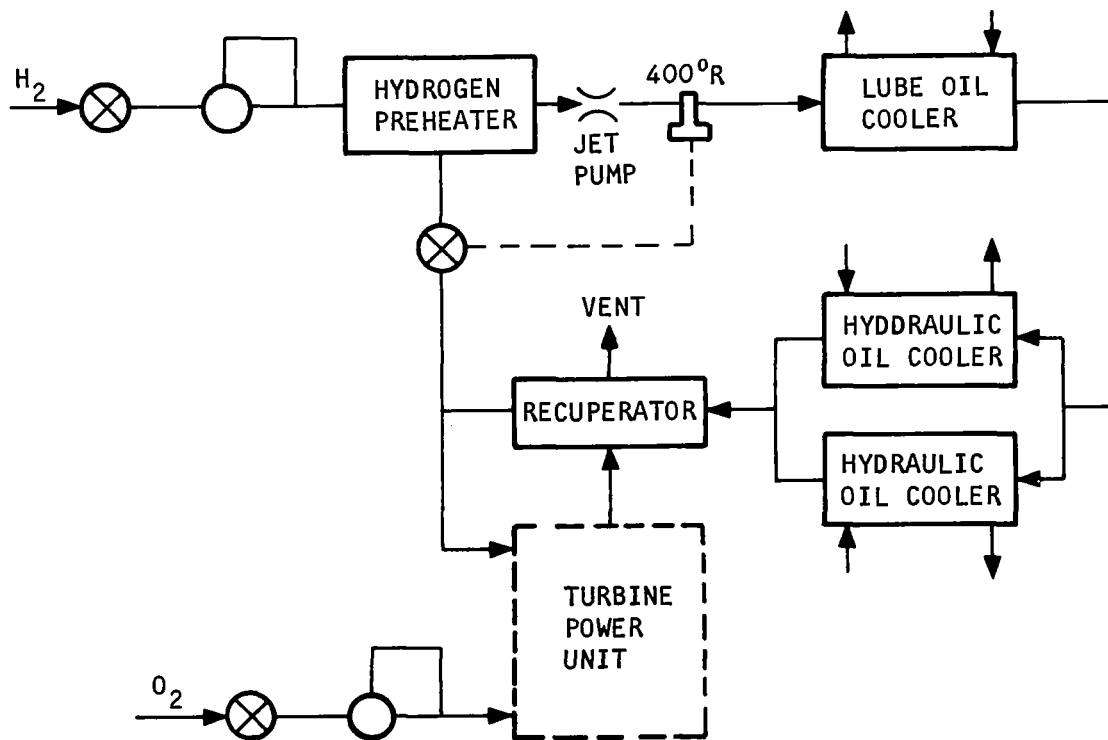


Figure 5-1. Propellant Conditioning/Thermal Management Subsystem

## Components

For the purposes of describing the components forming the subsystem, it will be divided into the following:

- Oxygen shutoff/pressure regulator valve assembly
- Hydrogen shutoff/pressure regulator valve assembly
- Hydrogen recycle flow control valve
- Hydrogen recirculation jet pump
- Ducting
- Lube oil cooler
- Hydraulic fluid cooler
- Recuperator
- Hydrogen preheater

## Design Justifications

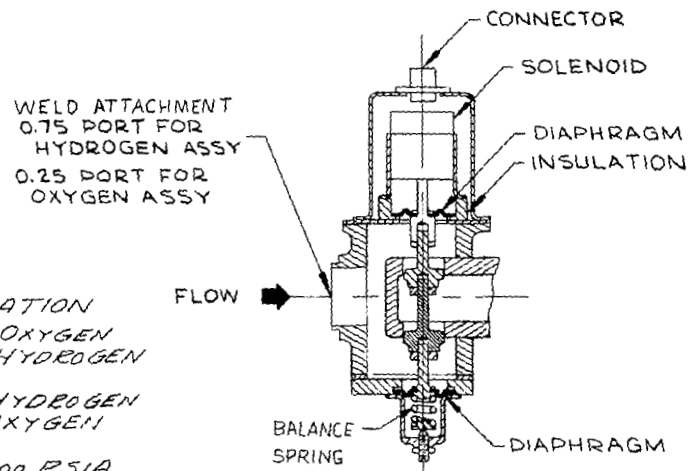
Descriptions of the system components, above, will be followed by a summary of related experience with comparable hardware to indicate the status of present technology in meeting the Space Shuttle APU system requirements.

### SHUTOFF/PRESSURE REGULATOR VALVE ASSEMBLY

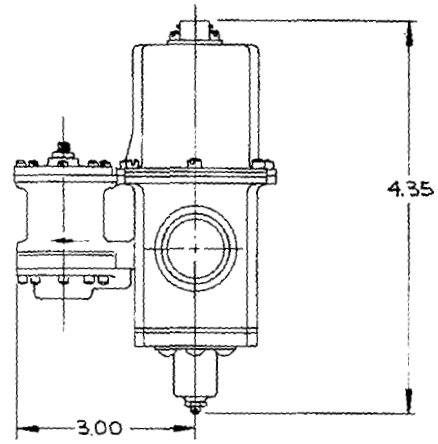
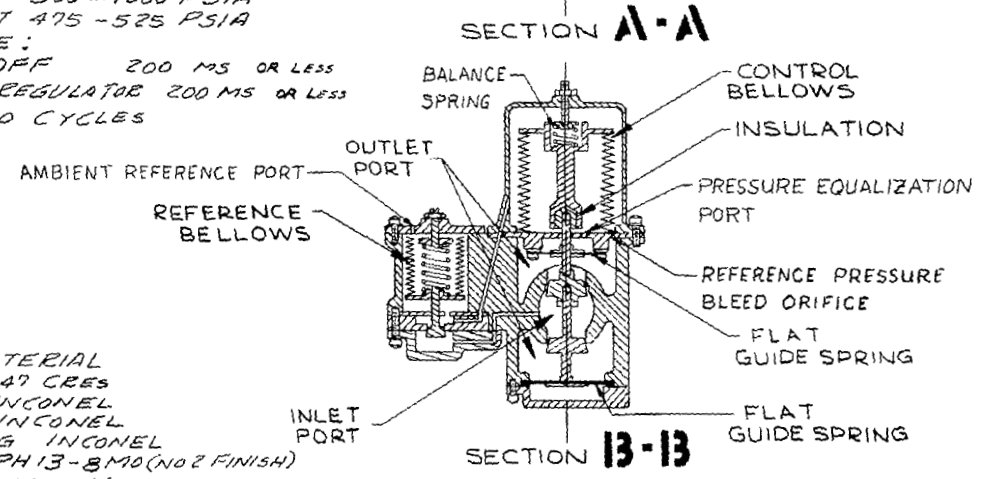
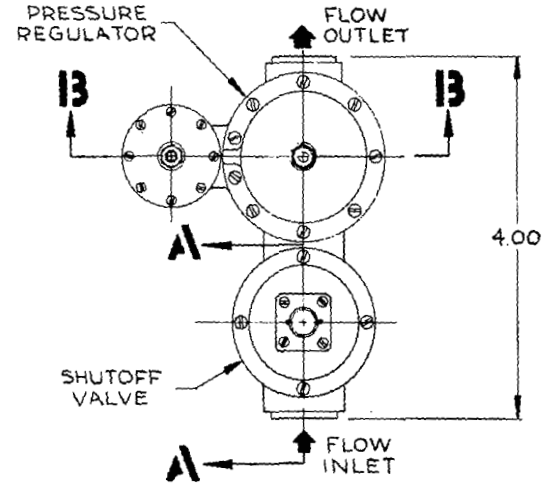
The shutoff/pressure regulator valve assembly, shown in Drawing SK68005, provides propellant shutoff and pressure regulation functions.

#### Shutoff Valve Element

The shutoff valve is solenoid actuated. The poppet assembly is spring-loaded to close when the solenoid is deenergized. In the event of any electrical power interruption, therefore, the valve poppets will close and remain closed in a fail safe condition. The poppet assembly consists of dual, pressure balanced poppets. The pressure at the inlet cavity of the valve operates on equal and opposing areas of the poppets and the convoluted diaphragms. This pressure balancing offers two important advantages, (1) lower solenoid power requirements, and (2) smaller solenoid envelope requirements. The convoluted diaphragms perform several functions in the valve design. First the balanced and opposing areas of the diaphragms maintain the pressure balanced configuration of the poppets. Second, the diaphragms provide a guide function for the poppet to maintain the poppet centered in the orifices. The spring rate of the diaphragm in the radial direction is very high compared to the transverse direction. Overall balancing of the spring forces is accomplished by adjusting the balance spring assembly at the bottom of the valve.



**SPECIFICATION**  
 FLOW: 5.1 LB/MIN OXYGEN  
 7.1 LB/MIN HYDROGEN  
 TEMP RANGE:  
 75°R - 500°R HYDROGEN  
 300°R - 500°R OXYGEN  
 PRESSURE:  
 INLET 500 - 1000 PSIA  
 OUTLET 475 - 525 PSIA  
 RESPONSE:  
 SHUTOFF 200 MS OR LESS  
 PRESS REGULATOR 200 MS OR LESS  
 LIFE: 10000 CYCLES



**MATERIAL**  
 HOUSING 347 CRES  
 BELLOWS INCONEL  
 CONVOLUTE INCONEL  
 GUIDE SPRING INCONEL  
 PORPET PH 13-BMD (NO 2 FINISH)  
 INSULATION MIN K  
 BALANCE SPRING INCONEL

VALVE, PRESSURE  
REGULATOR, SHUT OFF

SK68005

The listing below presents the design requirements of the valve assembly.

Flow, lb/min	Maximum	Oxygen; 5.089;	Hydrogen; 7.07
	Minimum	Oxygen 0.259;	Hydrogen; 0.434
Temperature, °R	Maximum	Oxygen 500	Hydrogen 500
	Minimum	Oxygen 300	Hydrogen 75
Pressure, psia	Maximum	1000	
	Minimum	500	
Response, time, MS	200		
Pressure drop, psid	5 at 500 psia inlet		
Life, cycles	10,000		
Leakage, sccm	50		
Line size, in.	Oxygen	0.25	
	Hydrogen	1.125	

#### Pressure Regulator Valve Element

The pressure regulator accepts the pressure of the propellant input line and provides regulated pressure to the APU components. The method of accomplishing this is by throttling the flow to a rate which corresponds to the use rate in the combustor based on a pressure of approximately 500 psia. Sketch SK 68005 shows a cross section of the regulator assembly.

The regulator operates from a pilot control, in the following manner. High pressure gas at the inlet port is applied to the poppet of the reference bellows cavity. The poppet is stroked open until the pressure in the reference bellows cavity is at a pressure of approximately 525 psia. At this pressure, the reference bellows in loaded in compression and the poppet closes to maintain the 525 psia. A pressure communication passage is built into the valve body which applies the same 525 psia to the control bellows cavity. Pressure difference across the control bellows causes the main poppet to stroke open when the down stream pressure falls, and to stroke closed when the down stream pressure rises.

The principal design features of the regulator include adjustable balance spring assemblies for the reference and control poppets. Final poppet seat force adjustments are made using this provision.

Flat guide springs at each end of the poppet assembly provide the positive poppet centering action while serving as low rate springs in the transverse force direction. The flat guide springs are multipointed stars, which allow free pressure communication across the spring to adjacent volumes.

The listing below presents the design requirements of this assembly.

Parameter	Oxygen	Hydrogen
Flow lb/min	5.09	7.07
Temperature °R		
Maximum	500	500
Minimum	300	75
Inlet pressure, psia	500 to 1000	500 to 1000
Response time, MS	200	200
Controlled outlet pressure, psia	500	500
Control pressure tolerance, psi	25	25
Duct diameter, in.	0.25	1.125
Leakage, sccm	100	100

A study was conducted to determine the effects of expansion of the hydrogen and oxygen across the regulator, specifically the possibility of liquification caused by sudden expansion of the gas from 500 to 1000 to 500 psia. Assuming hydrogen at 75°R and 1000 psia, and expansion to 500 psia, results in a hydrogen temperature approaching 65°R, a value well above the 2-phase region. Therefore, no liquifaction of hydrogen should occur through expansion in the regulator.

A similar evaluation of oxygen expansion from 1000 psia and 300°R to a pressure of 500 psia results in some liquification of the propellant. No such liquification will occur over this range of expansion if the initial temperature is above 308°R however. In the event some oxygen liquification does occur, it will be a fine mist in the moving gas stream. Upon contact with duct walls and components down stream, these fine particles will vaporize so that no slugging or flow instability is expected.

## Mechanical Design

### 1. Mechanical Linkages

Mechanical linkage between the actuator and the valve control element is required in all valves. The solenoid valve is driven by a solenoid direct connected to the poppet assembly. Backlash is a consideration in the design of this assembly only to the extent that total motion of the solenoid be minimized. The direct connection eliminates backlash in this valve.

The regulator pilot and control poppet assemblies are both direct driven from the respective bellows assemblies. In this application backlash is important, and the direct connection of actuator-poppet assemblies eliminates this mechanical design consideration.

## 2. Seals

Valve seals are important to the shutoff and the regulator assemblies. Here, the shutoff must seal to prevent excessive leakage during extended periods of quiescent storage. The regulator seal is important from the standpoint of the pilot valve seal only, since the main regulator stage can leak significantly (an amount equal to the minimum system flow requirements) and the system pressure level will still be maintained at operational limits.

Seals for these applications are accomplished using metal to metal surfaces. The valve bodies are fabricated of relatively soft 347 stainless steel. The valve body seats will be manufactured with square edges on cylindrical ports. The mating poppets will be fabricated of PH 13-8MO hardened to approximately Rockwell C 60 to 62, and will have a No. 2 finish. After assembly and alignment, the poppet will be overdriven into the body seat, to form a leak tight seal. Soft seats are not considered in this application because of the problems of cold flow and creep over the extreme range of temperatures.

Dynamic seals are avoided in the design of all valves. The method of achieving a seal between the reactant and the ambient is by use of welding, or by use of diaphragm/bellows designs. Convolute diaphragms are used for sealing around the poppet actuation mechanism, and also to serve as poppet guides. These devices are a single piece component.

### RECYCLE FLOW CONTROL

This modulating valve is controlled by the jet pump exhaust temperature transducer. A sensed temperature less than 400°R results in a command to open the recycle valve. A sensed temperature greater than 400°R results in a closing of the valve.

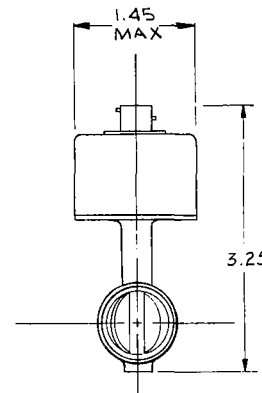
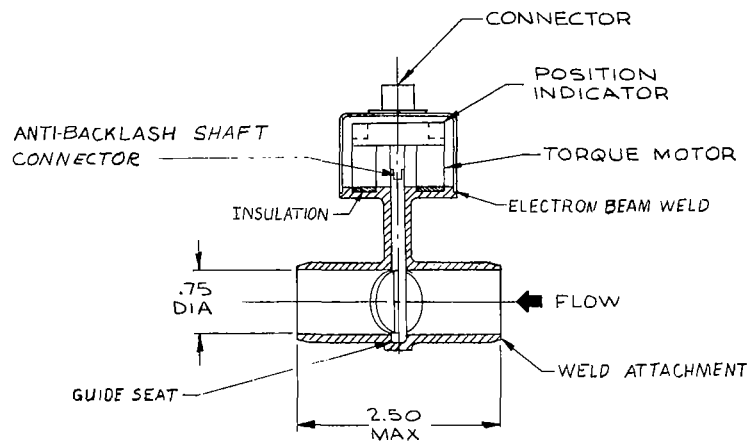
#### Description

The valve modulates flow by rotation of a butterfly in the flow stream. Drawing SK 68006 shows the valve outline and characteristics. A torque motor actuates the valve element. The motor has a rotation capacity of 80° total rotation which is sufficient to modulate the flow from a full flow (full open) to an off condition (closed) position. The valve does not need to seal leak tight upon closure. The requirement for the valve is to accomplish flow modulation and a leakage value of 2 to 3 percent at the closed position is acceptable.

The position sensor for the valve is a rotary differential transformer. Shaping of the position indicator output is accomplished in the control.

The packaging of the valve reflects the thermal design of this assembly. The relatively hot recirculating hydrogen gas flowing through the valve must be thermally separated from the electrical components. This is accomplished by extension of the actuator shaft and by addition of an insulation barrier under the torque motor. Both these features are shown on the drawing. The temperature of the torquemotor windings, the critical parameter, is therefore limited to 155°C.





SPECIFICATIONS  
 FLOW, 7.0 lb/MIN  
 PRESSURE DROP AT FULL FLOW, 0.30 PSI  
 OPERATING PRESSURE, 500 PSIA  
 TEMPERATURE RANGE, 700 TO 1100° R  
 RESPONSE TIME, 200 MS OR LESS

TORQUE MOTOR  
 TORQUE, 15 OZ. IN.  
 STALL POWER, 63 W  
 MAX VOLTAGE, 63 V  
 TIME CONSTANT:  
 MECHANICAL, 24 MS  
 ELECTRICAL, 0.4 MS

POSITION INDICATOR  
 ROTARY VARIABLE DIFFERENTIAL  
 TRANSFORMER RANGE,  $\pm 40^\circ$   
 SENSITIVITY, 1.5 MV/DEGREE  
 LINEARITY,  $\pm 1\%$   
 OUTPUT LOAD, 10K OHMS

MATERIAL:  
 HOUSING 347 CRES  
 BUTTERFLY & SHAFT 347 CRES  
 TORQUE MOTOR:  
 INLAND MOTOR CORPORATION  
 TYPE NO. T-1218  
 POSITION INDICATOR  
 SCHAEVITZ ENGINEERING  
 R 13 MLS-A1

VALVE,  
 RECYCLE

SK68006

## Performance

The valve design requirements are presented in the listing below.

Fluid	Hydrogen
Flow, lb/min	7.0
Pressure drop at full flow, psi	0.02
Operating pressure, psia	500
Temperature range, °R	700 to 1075
Response time, MS	200
Duct diameter, in.	1.5
Leakage, sccm	1000

The characteristics of the torque motor and the position sensor are presented on the drawing.

## JET PUMP

The jet pump provides hydrogen circulation in the thermal control loop. Hydrogen added to the APU provides the primary energy to cause circulation of the pump secondary flow. The use of this static eliminates the requirement for a rotating fan or gas compressor for hydrogen circulation.

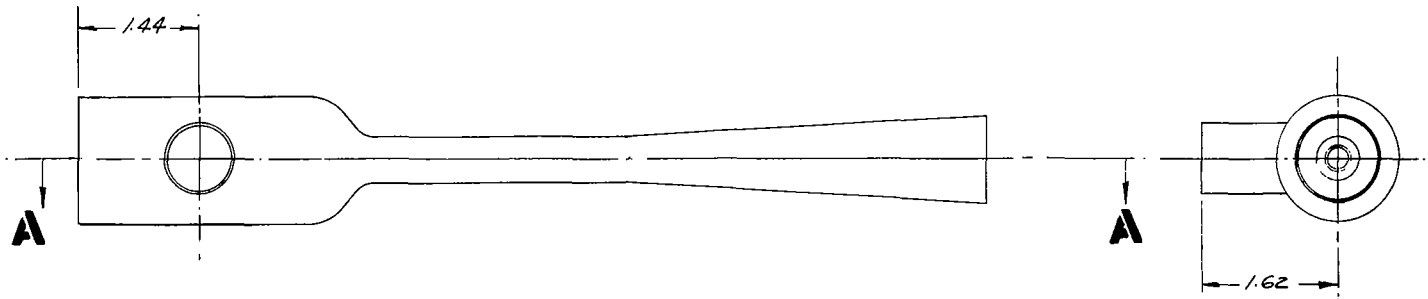
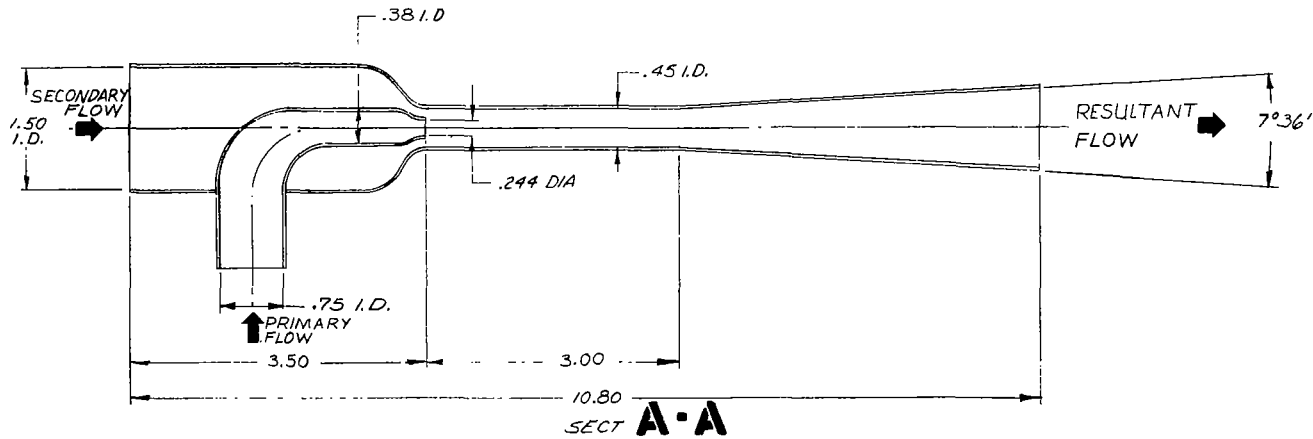
## Description

The jetpump (ejector) uses the 500 psia hydrogen supplied to the APU in a converging nozzle where a portion of its internal energy is converted to kinetic energy. The high velocity gas entrains the lower pressure, low velocity, recirculation gas flowing coaxially with the nozzle. The combined stream enters a mixing section and diffusing section which increases the static pressure of the secondary flow stream by reducing the velocity. The pressure differential across the secondary flow portion of the ejector provides the pressure for flow in the subsystem components and ducting. The secondary flow may be throttled by the recycle flow control valve to yield a mixed temperature of 400°R at the jet pump exhaust.

The jet pump configuration is shown in Drawing No. SK68001. The jet pump is fabricated of stainless steel and welded into the APU package as an integral portion of the ducting, as shown in the packaging drawing. The length of the mixing zone is equivalent to approximately 7 times the mixing zone diameter, and the diffuser cone angle (7°36') is selected to yield maximum static pressure recovery within practical duct lengths.

## Performance

The performance requirements of the jet pump are presented in the listing below. The detailed performance maps are presented in Figure 5-2. Here performance in terms of pressure ratios is presented for various values of corrected flow factor.



52

*SPECIFICATION*

*FLOW RATE:*

*PRIMARY 6.93 LB/MIN*

*SECONDARY 6.33 LB/MIN*

*PRESSURE:*

*PRIMARY 500 PSIA*

*SECONDARY 440 PSIA MIN*

*OUTLET 450 PSIA MIN*

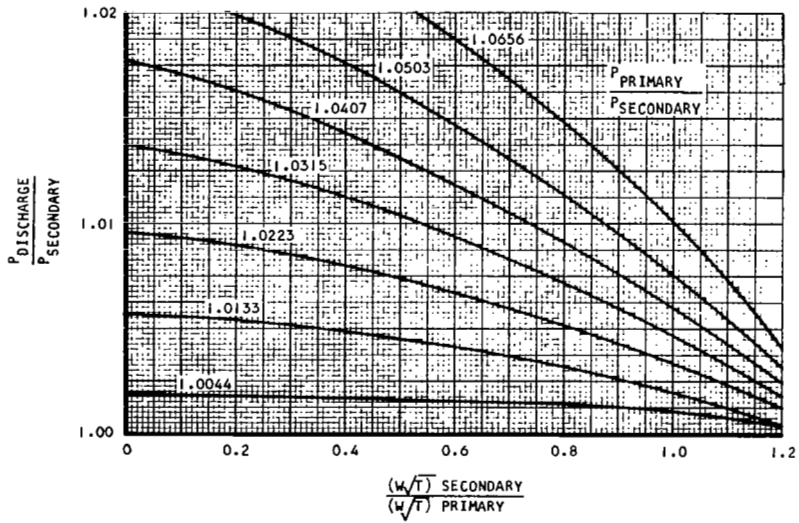
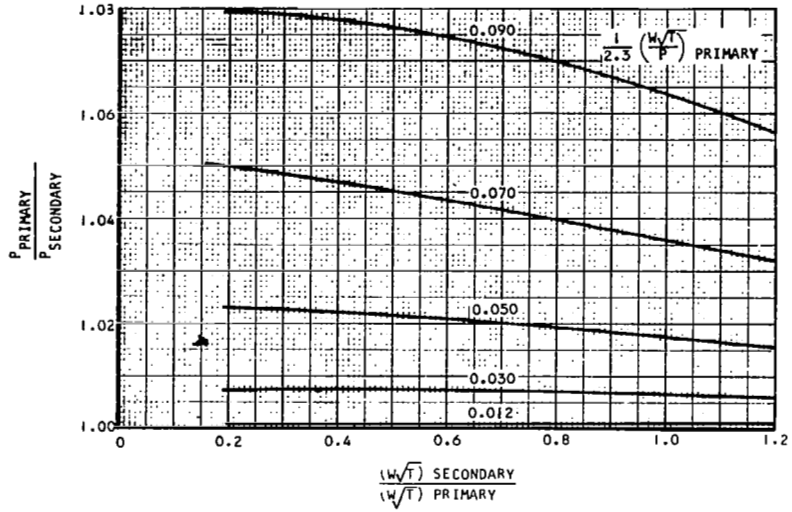
*TEMP:*

*400°E Nom*

*MATERIAL: 347 CRES 0.020 WALL*

*JET PUMP*

*SK68001*



S-67216

Figure 5-2. Jet Pump Performance Maps

	Primary Side	Secondary Side	Outlet
Flow rate, lb/min	6.93	6.33	
Inlet total pressure, psia	500	452	465
Inlet temperature, °R	392	408	
Duct diameter, in.	0.75	1.5	1.0
Working fluid	Hydrogen	Hydrogen	Hydrogen

## DUCTING

Ducting for the APU is considered in two parts; the ducting which is an integral part of the APU packaged assembly, and the propellant and exhaust ducts which are considered to be external to the package.

### External Ducting

Propellant supply lines are not included in the weight estimate for ducting. The propellant supply lines are considered to interface with the APU at a terminal point on the package where other interfaces will be accommodated, which has electrical power and control provisions. Service from the interface (mounted on the APU) to the propellant shutoff valves is 0.5 dia for the oxygen and 0.75 dia for the hydrogen. The line length for each reactant is 10 in.

The turbine exhaust duct, connected to the recuperator outlet, is sized for a 4 in. diameter duct 600 in. long. The duct is steel and weighs 32.8 lb. Any modification in the duct outlet location, or ducting route, will change the duct weight in direct proportion to any resulting change in duct length.

### APU Package Ducting

The detailed ducting arrangement is shown in the packaging drawing of Section 3. Figure 5-3 shows a simplified ducting schematic of the APU. The numbers on the schematic correspond to the duct numbers shown on Table 5-1. The table presents the duct diameter, length, number of 90° bends, and the calculated pressure drop factor, Z. The Z factor is defined as follows:

$$Z = \frac{\rho \Delta P}{W^2}$$

and

$\rho$  = density, lb/cu ft

$\Delta P$  = pressure drop, psi

W = flowrate, lb/min

The weight of the ducting is based on the use of stainless steel and a duct wall thickness of 0.020 in.

TABLE 5-1

APU DUCTING DESIGN PARAMETERS

Duct No.	Duct Diameter, in.	Duct Length, in.	No. of 90° Bends	Z Factor x 10 <sup>4</sup>
4	0.25	3	1	1110
5	0.25	3	1	70000
8	0.75	2	1	10
9	1.0	6	1	3.64
11	1.0	2	0	1.18
12	1.0	14	2	1.14
15	1.0	5	1	3.5
16	0.75	5	2	20.8
17	1.5	2	2	19
19	0.5	2	1	54
22	0.75	3	0	50
23	0.75	4	0	2.74
24	4.0	8	2	0.023
25	4.0	600	3	0.1185

APU assembly ducting 1.4 lb

APU exhaust ducting 32.8 lb

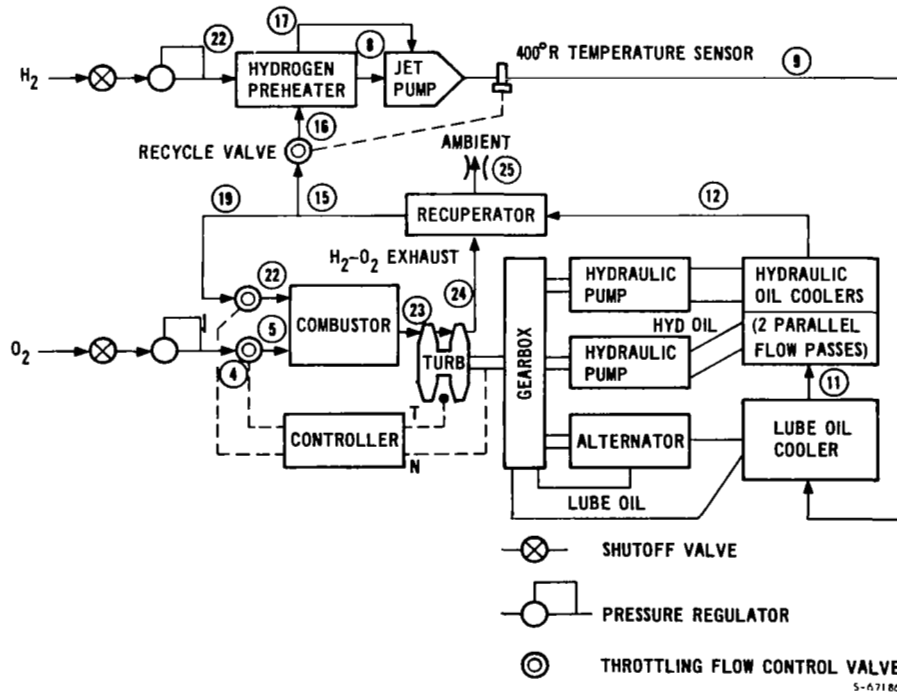


Figure 5-3. APU Ducting Schematic

## HEAT EXCHANGER DESIGN CONSIDERATIONS

As indicated previously, the system has four heat exchangers:

- Lube oil cooler
- Hydraulic fluid cooler
- Hydrogen preheater
- Recuperator

Although clearly the detailed design considerations and problem areas will be somewhat different with each of these heat exchangers, there is sufficient commonality to permit a general discussion of the design considerations which were involved. Consequently, the present discussion will be concerned with the thermal and structural design considerations. (The problems associated with flow maldistribution, flow instability, fluid congealing freezing, etc., are discussed in Section 2 of Volume IV--these problems have been solved by avoiding designs and conditions where they would occur and consequently do not require discussion here.)

### Thermal Design Considerations

Parametric data was developed for each of the heat exchangers using a computer design program, H0 424A. The computer program uses the physical property data of the fluids, the tested friction factor and the Colburn modulus (F and J curves) data, and the problem statements, and iterates to a solution by using the fluid properties corresponding to the average film temperatures in the heat exchanger. Solutions from this analysis are then used as inputs to the heat exchanger design program H0 415P, which determines the performance of the selected heat exchanger core assembly. General results of the performance programs include the following comments:

- (a) Tubular heat exchanger designs are used to satisfy containment of high fluid pressures. Pressure and thermal stress considerations favor tubular configurations.
- (b) Bare (unfinned) tubes yield satisfactory performance, are simple to fabricate, and are easily assembled with internal baffles for multipass configurations.
- (c) Stainless steel construction is used for three of the heat exchangers for compatibility, long material life under thermal and pressure cycling loads, and ease of assembly (welding). (Because of the stress problems, the hydraulic cooler uses Inconel 718.)

## Structural Design Considerations

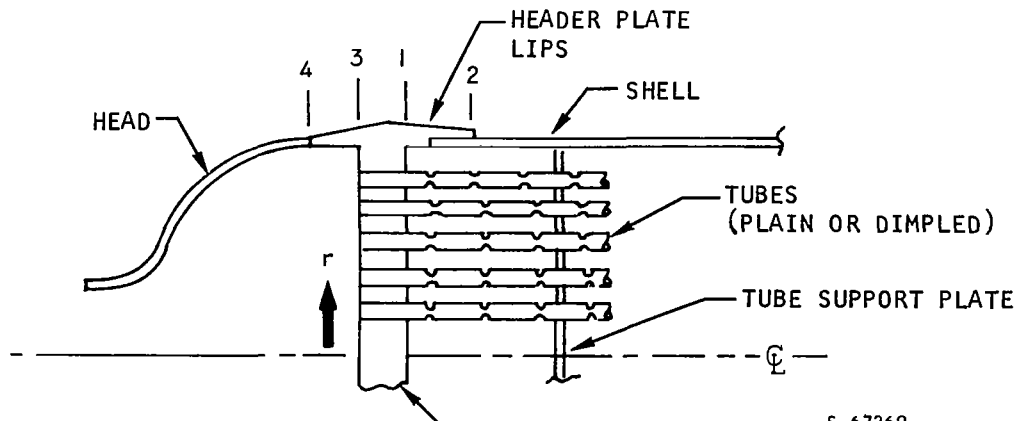
### 1. General

The design of the heat exchangers must consider the internal pressure and temperature conditions, and also an allowance for external environmental loadings. The internal loadings include the specified proof and burst-test conditions, (1.5 proof, 2.0 burst), the thermal-pressure cycling requirements, and the most severe combinations of the operating conditions. The external loadings include vibration (mechanical and acoustic), shock, steady accelerations and temperature. In each case the design of the unit must demonstrate adequate life and structural integrity under all practical loading combinations.

The preliminary analysis has considered the proof and burst pressure conditions and also the design point pressure-thermal conditions for the units which have appreciable thermal gradients. The method of analysis used to arrive at an optimum design is different for the cylindrical tubular units and the rectangular tubular recuperator. In each case however, in order to minimize weight, the apparent elastic local stress levels are permitted to exceed the yield strength of the material. In such instances an elastic-plastic analysis is conducted to demonstrate adequate cyclic life. The specific methods employed for the determination of the apparent elastic stress levels in the cylindrical and rectangular type units are briefly discussed below.

### 2. Cylindrical Tubular Units

The stresses and deflections due to pressure and temperature loadings are evaluated and the designs optimized using AiResearch X0560 digital computer program. The stiffness and strength characteristics of the ring dimpled tubes are measured by test and these data input to the program and used in establishing allowable stresses. The program computes the stresses in the perforated header plates, the shell, the heads, and in the tubes. The stresses in the header plate and in the tubes are obtained at several radial locations while the stresses in the shell and the head are obtained at the header plate. Special attention is given to the stresses that exist in the tubes and at the tube-header brazed interface. Specifically, the outputs from the program with internal pressures and/or thermal loadings are as follows:



S-67269



<u>Tubes</u>	Axial and bending stresses at each tube now described by a radial location, r. The tube bending stress is at the joint to the header plate.
<u>Header Plate</u>	The axial deflection, slope, moment, and bending stress at each radial location, r.  The axial and bending stress in the header plate lip at Location 1.  The axial and bending stress in the header plate lip at Location 3.
<u>Shell</u>	The axial and bending stress in the shell at Location 2.
<u>Head</u>	The axial and bending stress in the head at Location 4.

The units incorporate a head, header plate, and shell joint configuration as shown above. The head is welded to the header plate lip on the head side while the shell is brazed to the header plate lip on the shell side. The reason for this type of joint is to remove the weld and braze areas from the junction areas of the head and shell with the header plate. The internal pressure and thermal loadings incur considerable bending moments at the junction areas which in the subject designs are taken by the parent metal of the integral header-lip configuration without imposing high stresses near the joints.

The preheater and the lube oil cooler are made of Type 347 stainless steel while the high pressure hydraulic oil cooler is made of Inconel 718. The parallel flow preheater has considerable thermal gradients and incorporates a convolution in the shell to accommodate the metal temperature differential between the tubes and the shell and also a radial flexible joint between the inlet header plate and the shell to accommodate the radial temperature differential between the header plate and the shell. Without these flexible joints the local plastic stresses incurred by the design point pressures and temperatures are too high to be commensurate with the required life.

Considerable care must be taken in the internal design of the tube bundle support system in light of the vibratory and acoustic environment. Previous experience on similar units in high vibratory and acoustic environments has shown that the tubes can suffer considerable fretting damage by vibrating against thin gage support or baffle plates. The subject designs use thick support plates with close tolerance holes to eliminate any tube fretting problem. Sufficient tube support plates must be used so that the fundamental tube natural frequencies are above the predominant excitation frequencies in the installation.

## 2. Rectangular Tubular Recuperator

The design incorporates flexible side plates and exhaust gas ducting to allow differential thermal movement between the tube bundles and the outer box structure. The flexibility is obtained by incorporating formed beads

in the plates in the appropriate directions. As a result of this flexibility the internal pressure loads from the hydrogen manifolds are reacted by the tubes alone. The design also incorporates a flexible strip between the header plates on the inlet-outlet side so as to permit relative thermal expansion between the two tube bundles.

The steady state and transient stresses in the header plates and the tubes are obtained by considering a strip of the header plate as a beam on multisupports. Each tube now is represented as a linear and torsional spring supporting the header plate. Both pressure and thermal loadings are imposed on the system and the model is analyzed using the AiResearch V0 245 beam program. The model enables evaluation of the stress to be made under any given set of internal pressure and temperature distribution conditions through the tube bundle.

Local pressure and temperature stresses at the tube and header joint area are obtained using the X0815 triangular element program. The program considers axi-symmetric elements of triangular cross section.

### 3. Allowable Stress Levels and Elastic-Plastic Analysis

In general, the stress levels at the proof and burst pressure conditions are kept below the yield and ultimate strengths respectively of the material at the relevant temperature. In some cases highly localized yielding at the proof pressure condition is permitted as long as there is no measurable permanent set. The steady and transient stress levels at the most severe operating conditions must be evaluated in terms of creep and fatigue. Where these stresses go into the plastic range an elastic-plastic analysis must be conducted to show sufficient cyclic life capability. There is no creep problem for these units since the temperatures levels are not high enough to make creep critical, however the design point stresses do go plastic in some locations. The elastic-plastic analysis is accomplished using the Wetzel-Morrow method with Neuber hyperbolas. The method gives the stabilized total strain range that the material attains during the load cycle. The number of cycles to failure is then computed using the Manson-Halford formula. This elastic-plastic analysis procedure is computerized (AiResearch Program X0870) and a theoretical S/N curve can rapidly be generated for any material and for any given type of cyclic loading and stress concentration level.

#### LUBE OIL COOLER

This heat exchanger provides cooling of the lubricant by heat exchange with the hydrogen loop. The lubricant heat load is primarily a function of APU output power. As a consequence, the temperature of the lubricant will approach an equilibrium value for steady state performance which will be a function of APU output power, ambient pressure, and hydrogen inlet temperature to the system. Volume V gives the equilibrium temperatures reached by the lubricant fluid under steady state conditions.

## Description

The lube oil cooler is a four-pass, counter-flow, shell-and-tube heat exchanger. The hydrogen flows inside the tubes in a single pass. The lube oil flows outside the tubes in four passes as dictated by the presence of flow baffles within the heat exchanger shell. SK68000 is a drawing of the unit showing the flow paths. The drawing shows a flow baffle at the periphery of the tube bundle which prevents lube oil from bypassing the tubes along the shell wall. The unit is assembled by brazing the tubes into the header, and the shell is closed by welding.

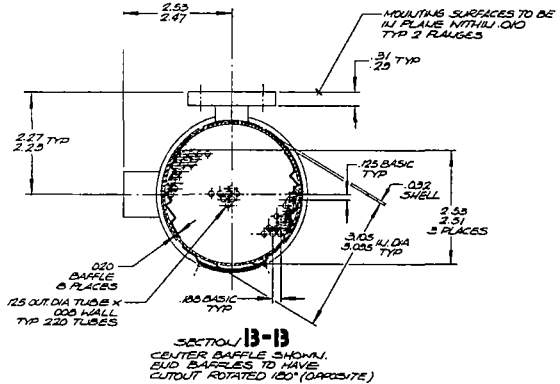
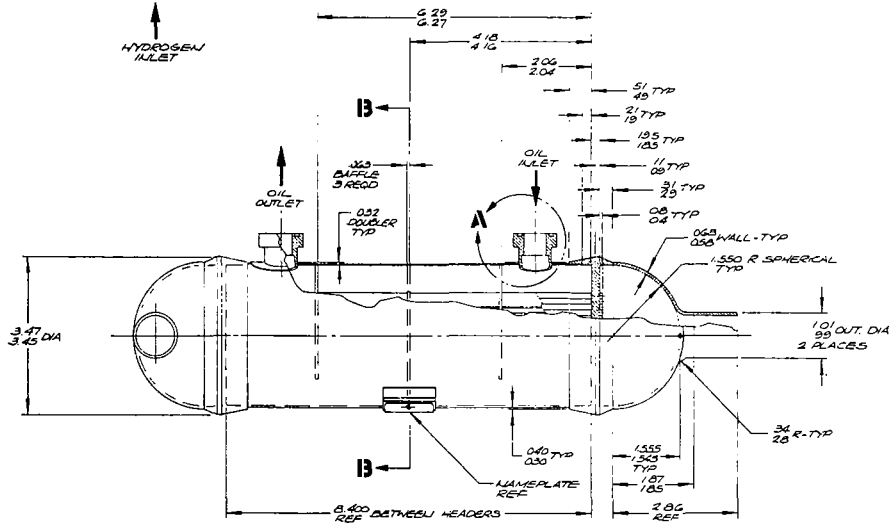
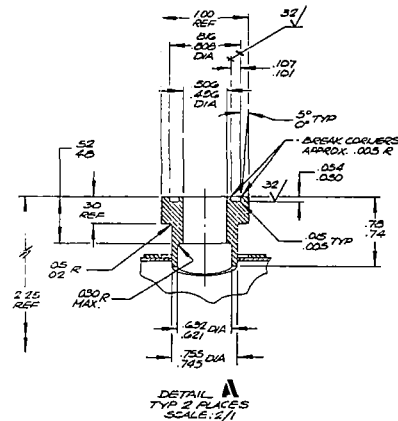
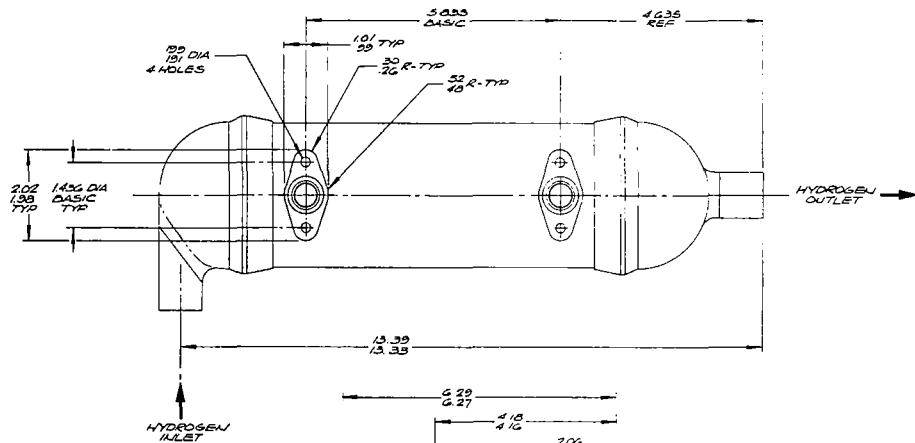
Mounting of the unit is accomplished by the attachments shown for the lube oil inlet and outlet. The flanges attach directly to the gearbox, thereby eliminating interconnecting ducting.

## Performance

The lube oil cooler is designed for a total heat transfer rate of 1303 Btu/min. The hydrogen inlet temperature is controlled by the recycle loop flow control to a minimum of 400°R. This limit is established to prevent congealing or freezing of the lube oil in the lube oil cooler as discussed in Section 2 of Volume IV. The heat exchanger design characteristics are listed below. Figures 5-4 through 5-6 present the performance characteristics of the unit in terms of heat transfer, pressure drop and effectiveness as functions of various values of flow rate.

	<u>Cold Side</u>	<u>Hot Side</u>
Fluid	Hydrogen	MIL-L-7808
Flow rate, lb/min	13.26	30
Inlet temperature, °R	400	543
Outlet temperature, °R	425.7	443
Inlet pressure, psia	500	100
Total pressure drop, psi	1.9	0.1
Effectiveness	0.181	0.694
Duct diameter, in.	1.125	0.25
Number of tubes	220	
Tube diameter	0.125 in. 00	
Tube wall thickness	0.008 in.	
Tube designation		
Inside	PLNTD	
Outside	SB 150100	

T9



3. UNIT SPECIFICATIONS ARE AS FOLLOWS:

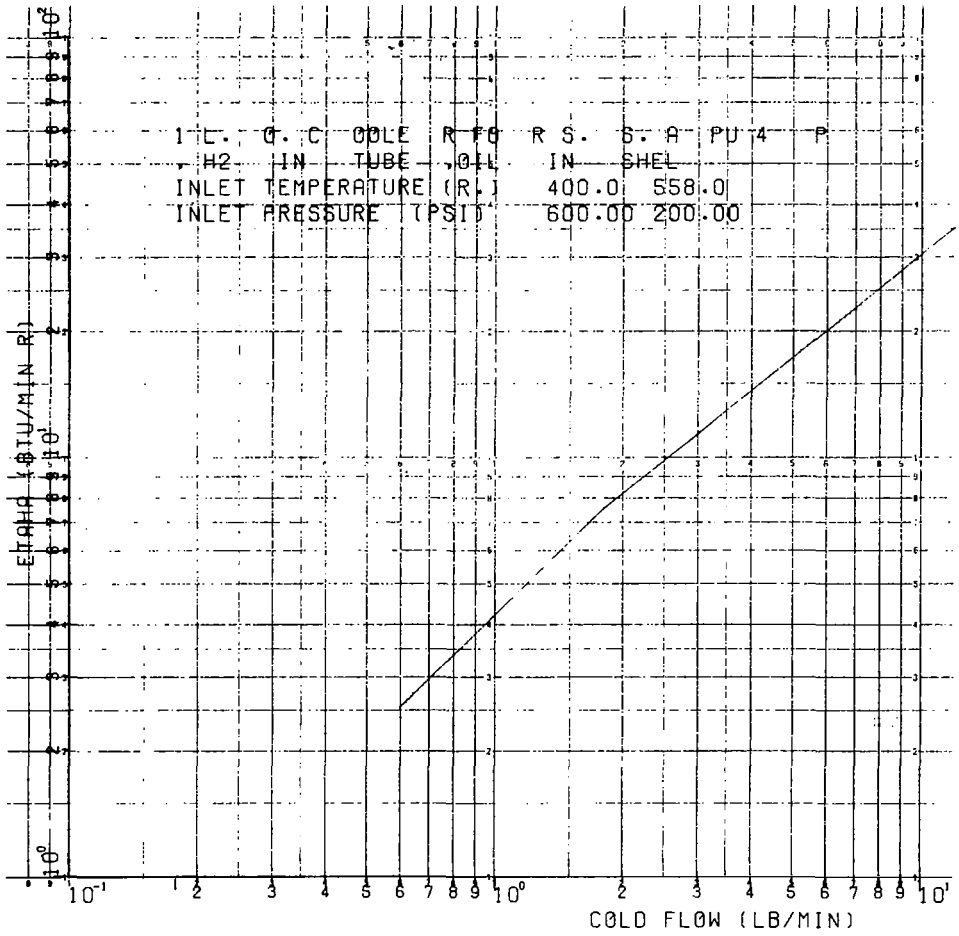
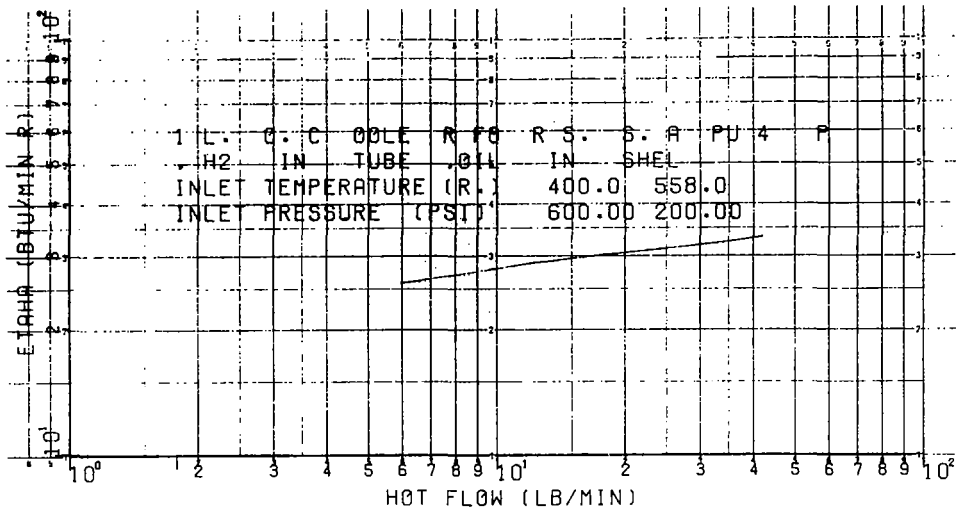
FLOW RATE HYDROGEN 13.26 LB/MIN  
MIL-L-7800 30.0 LB/MIN  
INLET TEMPERATURE HYDROGEN 400°F  
MIL-L-7800 343°R  
OUTLET TEMPERATURE HYDROGEN 422°R  
MIL-L-7800 443°R  
INLET PRESSURE HYDROGEN 300 PSIA  
MIL-L-7800 100 PSIA  
PRESSURE DROP HYDROGEN 2.1 PSI  
MIL-L-7800 3.1 PSI

2. MOUNTING PROVISIONS ARE NOT DEFINED. BRACKETS, ETC. TO MEET PACKAGE REQUIREMENTS CAN BE ACCOMMODATED.

1. MATERIAL PART TO BE MADE OF 06G5, TYPE 347.

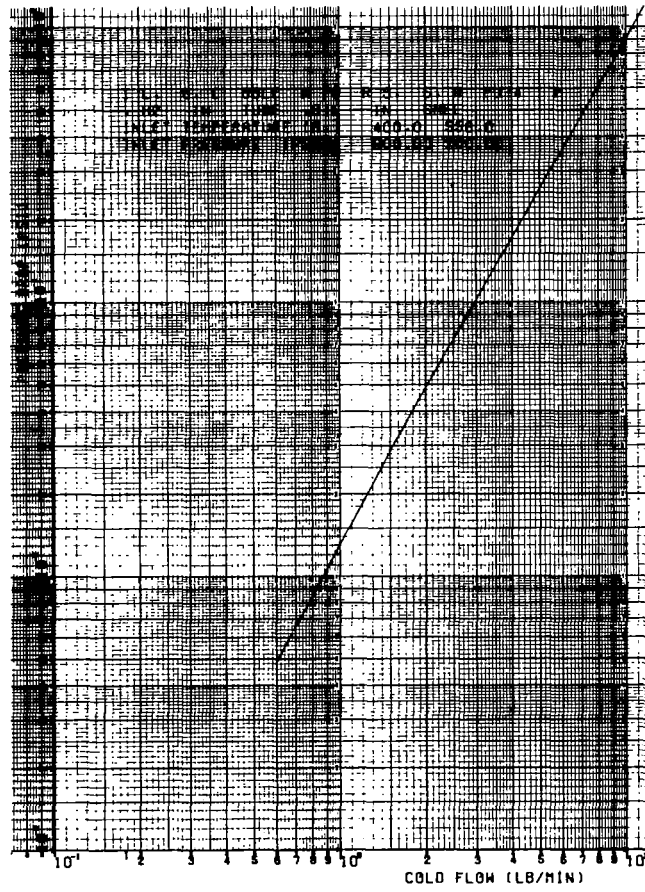
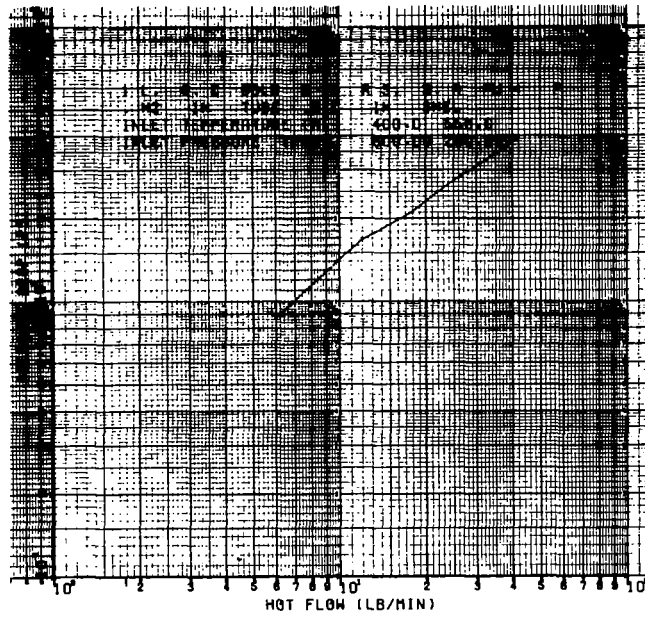
COOLER,  
LUBRICATING OIL

70210 SK68000



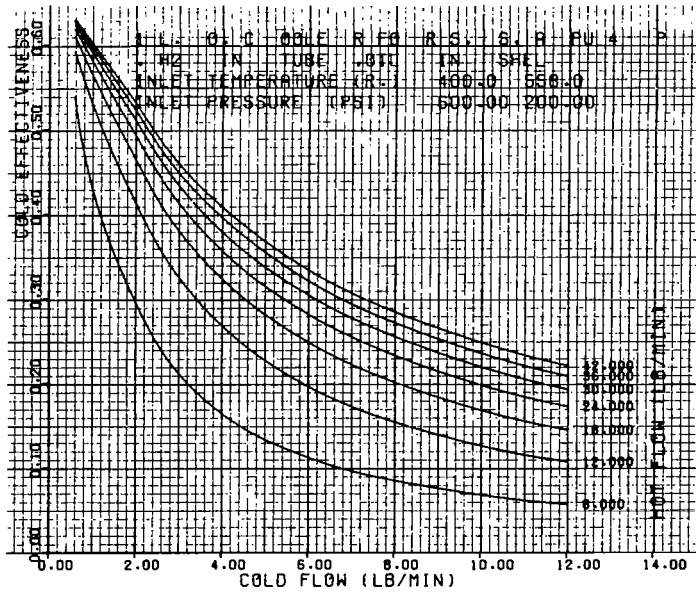
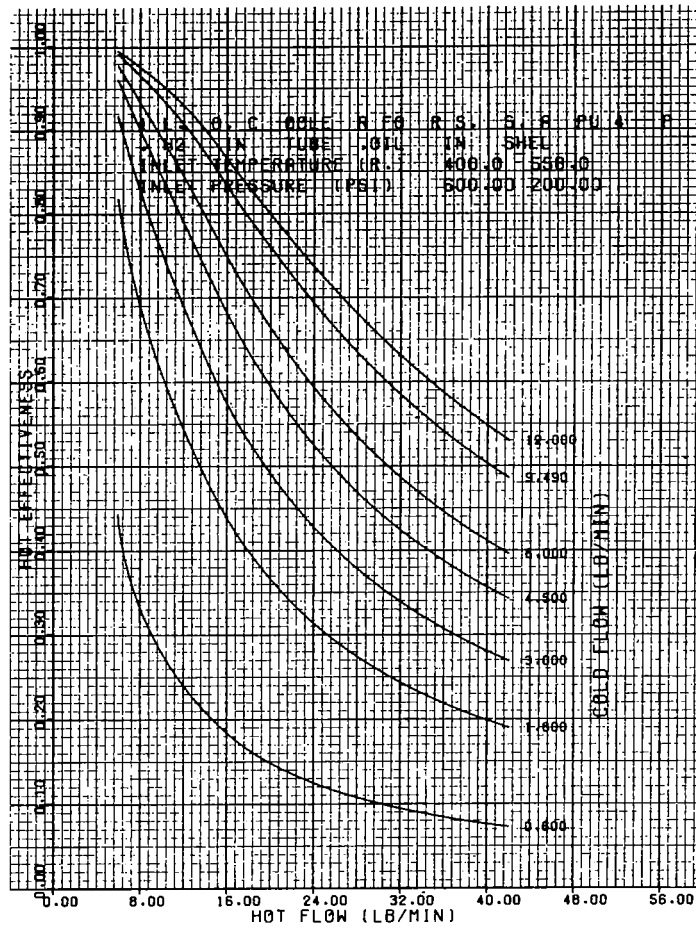
S-67263

Figure 5-4. Lube Oil Heat Exchanger Heat Transfer



S-67292

Figure 5-5. Lube Oil Heat Exchanger Pressure Drop



S-67291

Figure 5-6. Lube Oil Heat-Exchanger Effectiveness

## HYDRAULIC FLUID COOLER

The hydraulic fluid cooler is a dual heat exchanger assembly which cools hydraulic fluid flow from the two hydraulic pumps which are mounted on the APU gearbox. Hydrogen flowing through the heat exchanger on the cold side serves as the heat sink for hydraulic cooling. The available hydraulic cooling capacity depends upon the hydraulic fluid temperature and the APU operating conditions (output power level, ambient pressure, and hydrogen inlet temperature). Section 6 of Volume IV discusses the hydraulic cooling problems.

### Description

The hydraulic fluid cooler is a shell-and-tube unit shown in the Drawing SK 68003. Hydrogen flows on the outside of the tubes in two passes in a cross counter flow direction with respect to the hydraulic oil. The hydraulic oil (4100 psia) flows on the inside of the tubes. This results in a weight saving because the heat exchanger shell is not exposed to the high pressure except in the two head areas.

A circumferential baffle is shown in the design which eliminates flow of hydrogen between the tube bundle and the shell of the unit. This offers increased performance at negligible weight penalty.

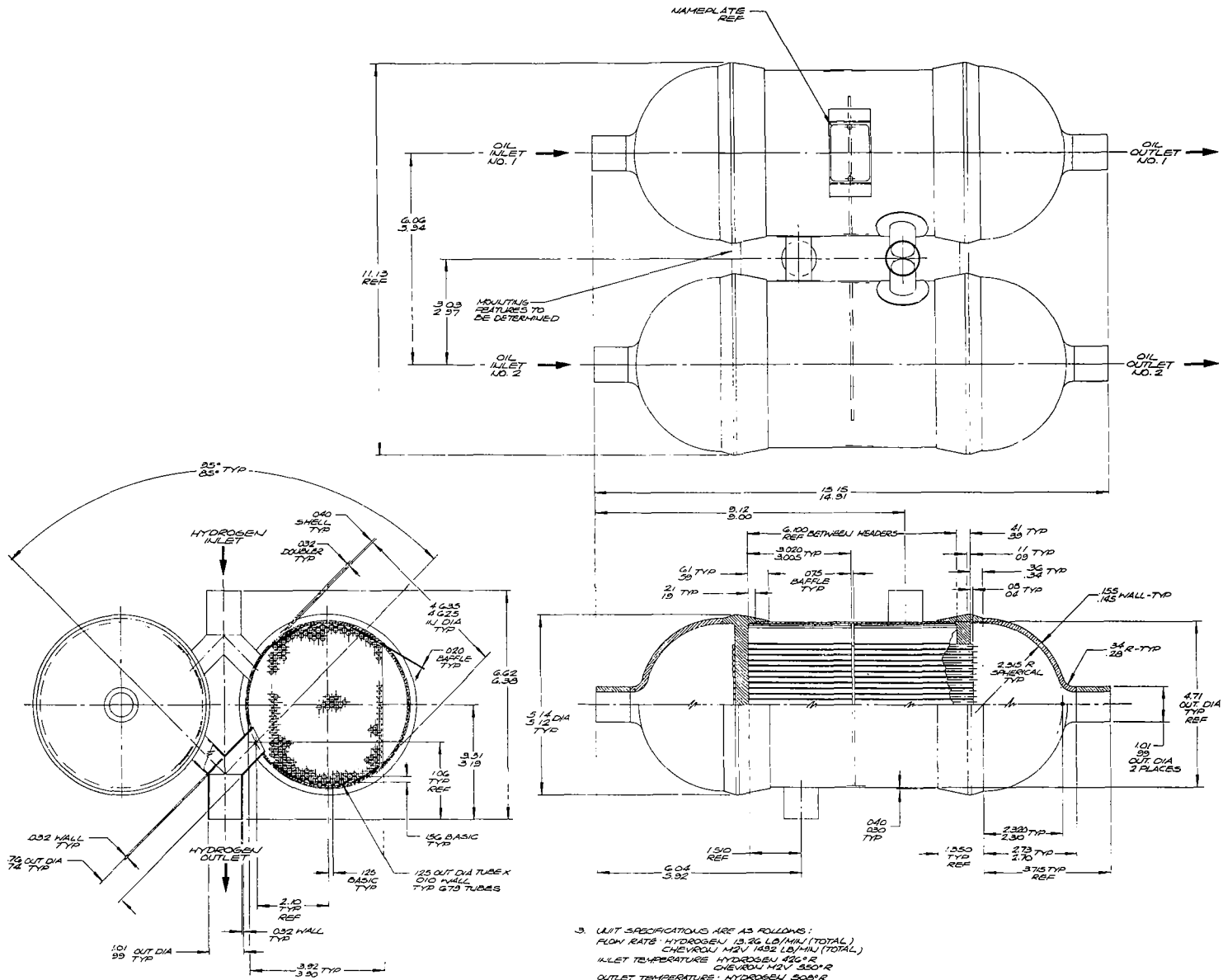
The unit is assembled by brazing the tubes into the header plates. The outer shell and head assemblies are welded.

### Performance

The design requirements of the hydraulic oil cooler are listed below. The values of flow and heat transfer are noted to correspond to total quantities for both hydraulic fluid loops. The values for each of the dual units is, therefore, one-half of the listed quantities.

Figures 5-7 through 5-9 present the detailed operational characteristics of the designed unit. The variations of heat transfer, pressure drop and effectiveness are presented as functions of combinations of gas and liquid flow rates. The unit is designed to provide a cold side effectiveness of approximately 0.65 at maximum flow. This prevents the possibility of supplying hydrogen to the recuperator at a temperature so low that freezing of turbine exhaust gas could occur.

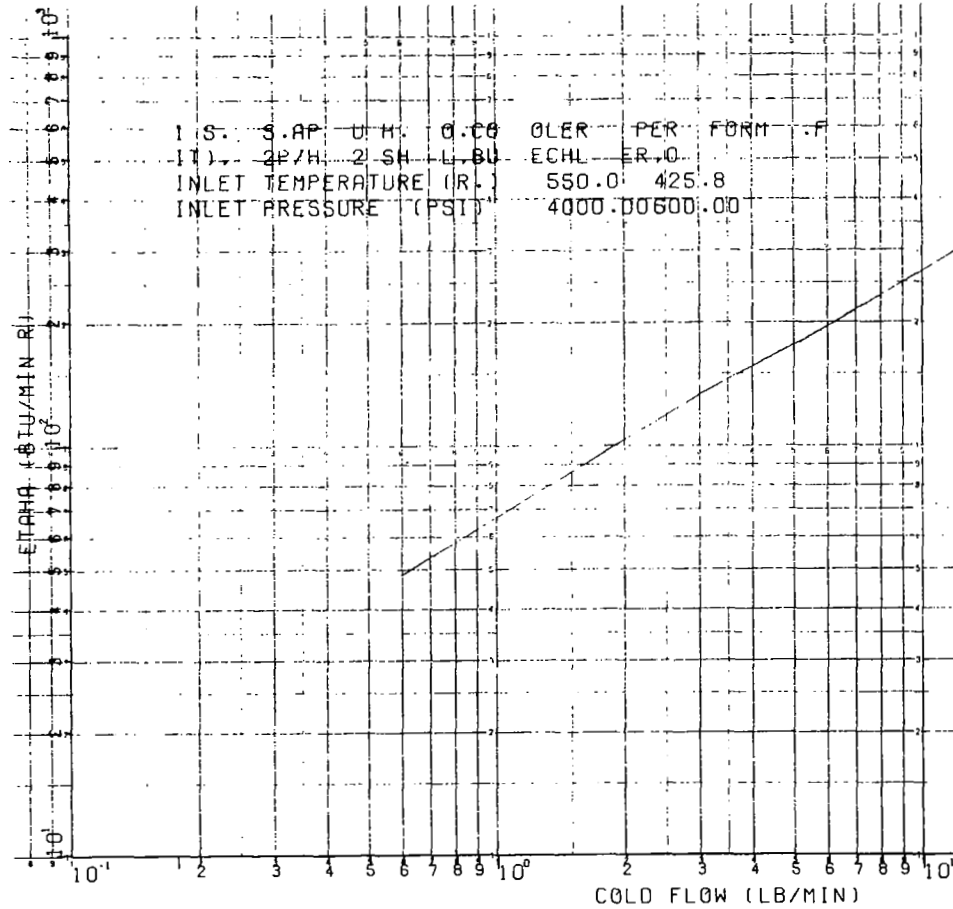
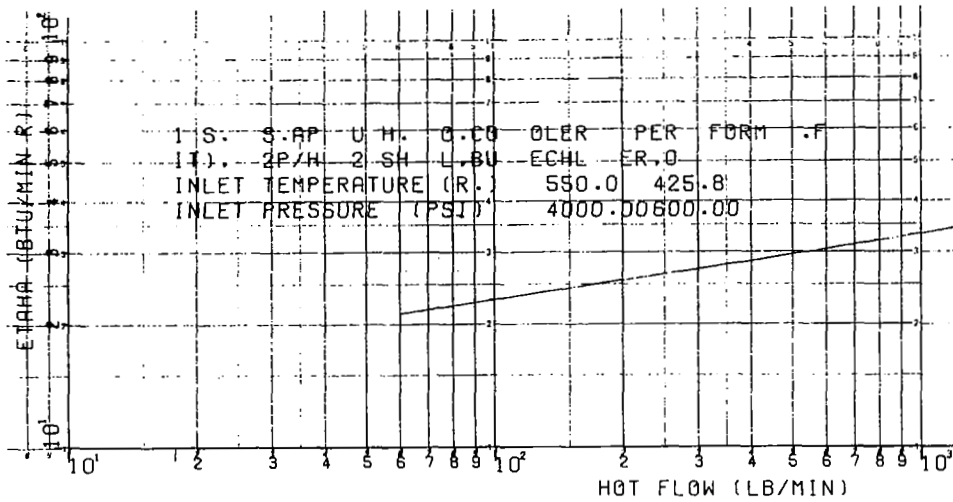




3. UNIT SPECIFICATIONS ARE AS FOLLOWS:  
 FLOW RATE: HYDROGEN 13.26 L/MIN (TOTAL)  
 CHEVRON H2V 1432 LB/MIN (TOTAL)  
 INLET TEMPERATURE: HYDROGEN 420°R  
 CHEVRON H2V 350°R  
 OUTLET TEMPERATURE: HYDROGEN 300°R  
 CHEVRON H2V 344°R  
 PRESSURE: HYDROGEN 500 PSIA  
 CHEVRON H2V 4000 PSIA  
 PRESSURE DROP: HYDROGEN 2.1 PSI  
 CHEVRON H2V 3.1 PSI
2. MOUNTING PROVISIONS ARE NOT DRAWN. BRACKETS, ETC. TO MEET PACKAGE REQUIREMENTS CAN BE ACCOMMODATED.
1. MATERIAL: PART TO BE MADE OF INCONEL 718.

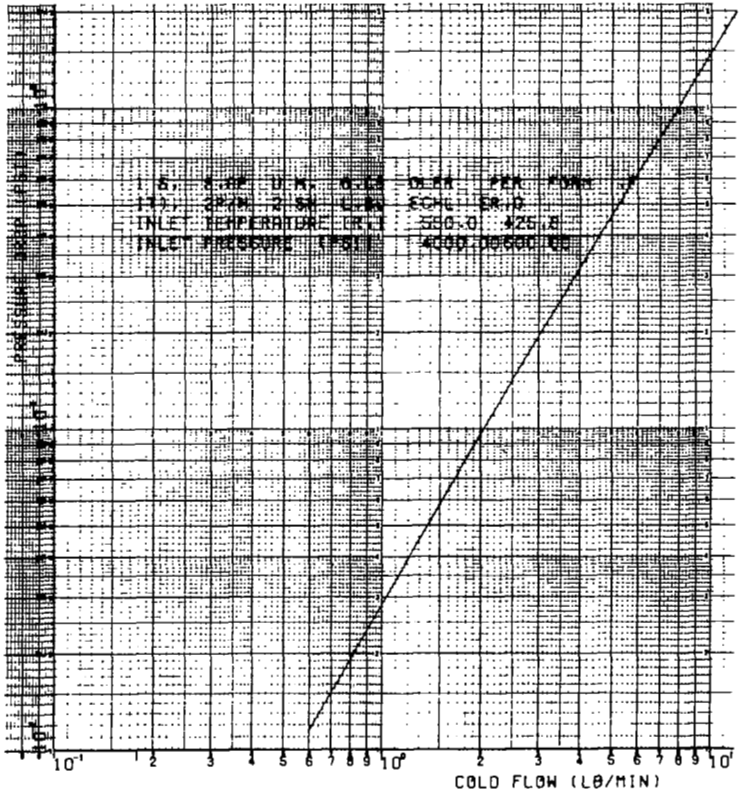
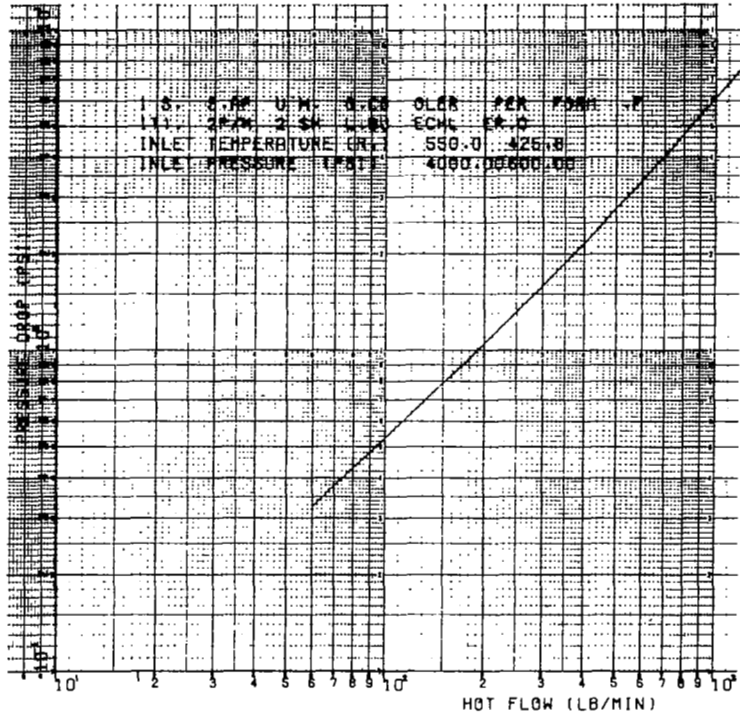
COOLER,  
HYDRAULIC OIL

SKG8003



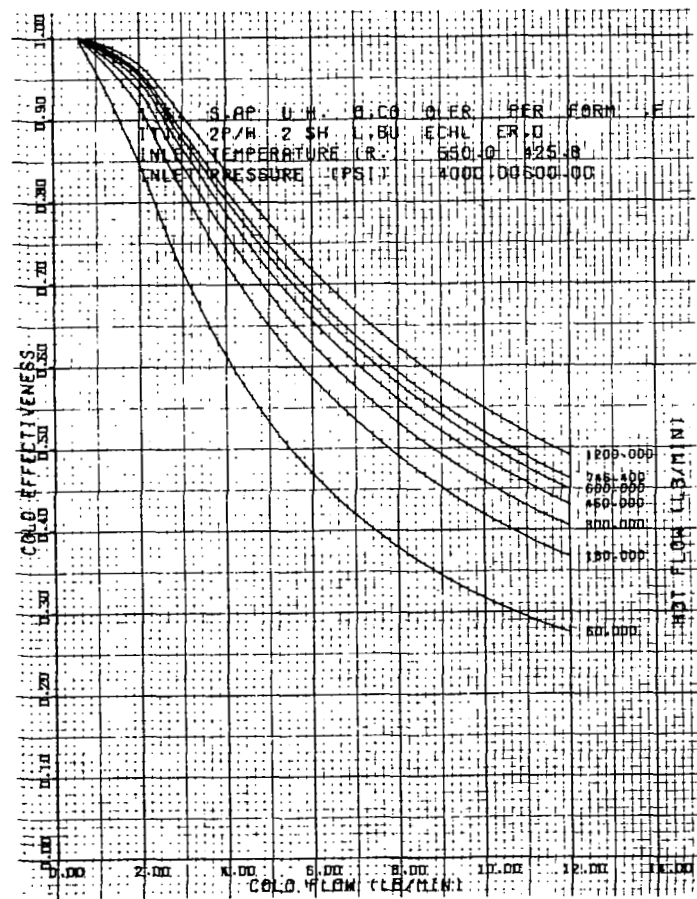
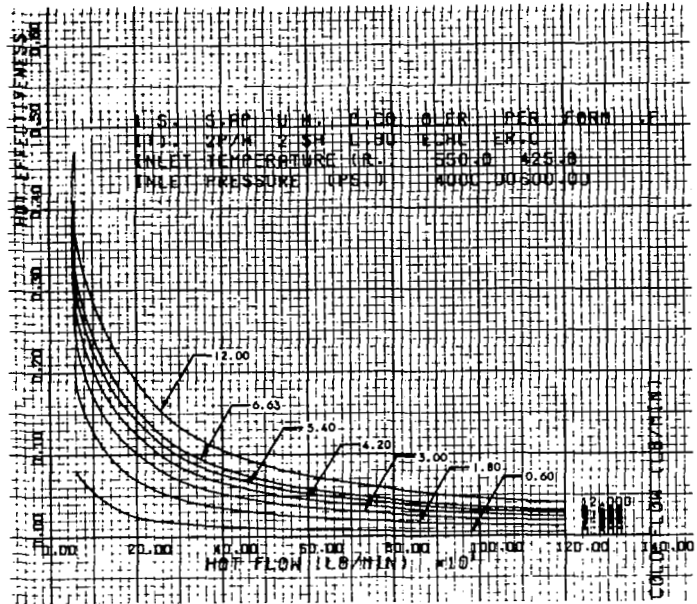
S-67264

Figure 5-7. Hydraulic Oil Cooler Heat Exchanger Heat Transfer



S-67290

Figure 5-8. Hydraulic Oil Cooler Heat Exchanger Pressure Drop



S-67289

Figure 5-9. Hydraulic Oil Cooler Heat Exchanger Effectiveness

	<u>Cold Side</u>	<u>Hot Side</u>
Fluid	Hydrogen	Chevron M2V
Flow rate, lb/min	13.26**	1492.5*
Inlet temperature, °R	425.8	550
Outlet temperature, °R	508.1	544.1
Inlet pressure, psia	500	4000
Total pressure drop, psi	0.6	0.1
Effectiveness	0.663	0.047
Duct diameter, in.	2.0	0.5
Total heat transferred, Btu/min	1697	

\*  $\frac{1492.5}{2} = \dot{w}$  for each of 2 separate hydraulic oil passes.

\*\*  $\frac{13.26}{2} = \dot{w}$  for each of 2 separate hydrogen passes.

Number of tubes 679 each unit

Tube diameter 0.125 in. OD

Tube wall thickness 0.010 in.

Tube designation

Inside PLNTD

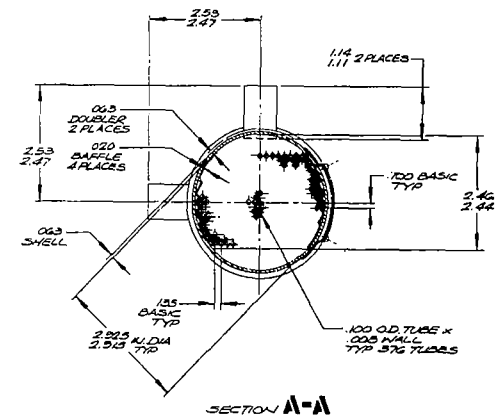
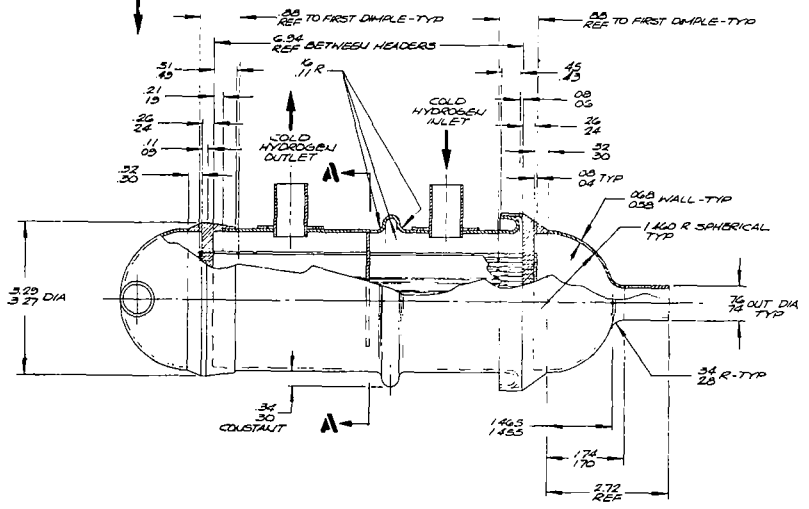
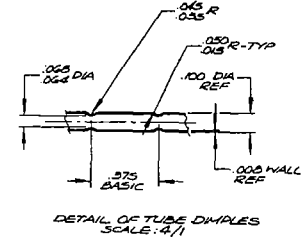
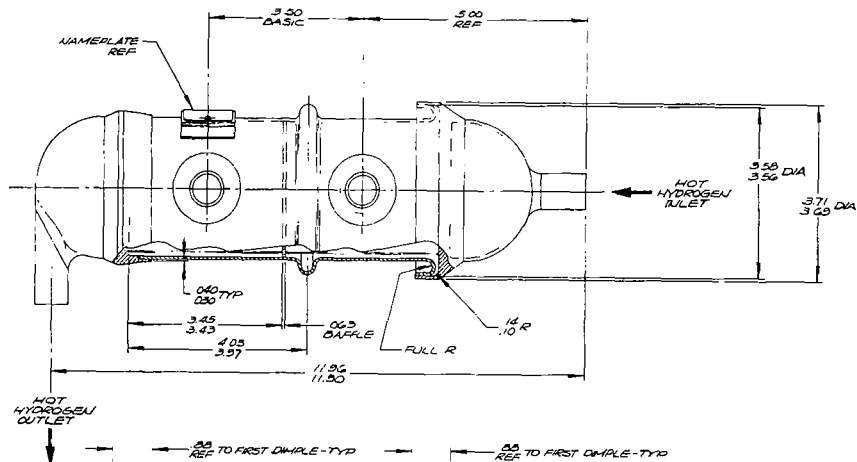
Outside SB 125100

## HYDROGEN PREHEATER

This hydrogen-to-hydrogen heat exchanger serves to equalize the temperatures of the primary and secondary hydrogen inputs to the jet pump to achieve uniformly high pumping performance (particularly for very low hydrogen inlet temperatures to the system). The hot recycled hydrogen passing through the preheater is controlled by the recycle flow control valve to maintain the jet pump discharge temperature of 400°R for the mixed primary and recycled hydrogen flows.

### Description

The preheater is a two-pass, cross-parallel flow unit. Hydrogen from the recirculating coolant (hot side) flows in a single pass axially along the heat exchanger inside the tubes. The hydrogen supplied to the APU (cold side) flows in the shell side in a two-pass parallel flow direction. The shell (shown in Drawing SK 68004) includes a convolute located in the center near the flow baffle. This convolute allows axial differential expansion of the shell with respect to the tube bundle. The header plate on the hot hydrogen inlet end is attached to the shell using a J type welded seam. This configuration allows radial growth of the header and/or the shell while maintaining the induced stress levels to workable levels.



71

3. UNIT SPECIFICATIONS ARE AS FOLLOWS  
 FLOW RATE: COLD HYDROGEN 6.33 LB/MIN  
 HOT HYDROGEN 6.33 LB/MIN  
 INLET TEMPERATURE: COLD HYDROGEN 75°R  
 HOT HYDROGEN 782°R  
 OUTLET TEMPERATURE: COLD HYDROGEN 352°R  
 HOT HYDROGEN 408°R  
 EQUAL OPERATING PRESSURE: COLD HYDROGEN 500 PSIA  
 HOT HYDROGEN 500 PSIA  
 PRESSURE DROP: COLD HYDROGEN 0.1 PSI  
 HOT HYDROGEN 0.4 PSI
2. MOUNTING PROVISIONS ARE NOT DEFINED. BRACKETS, ETC., TO MEET PACKAGE REQUIREMENTS CAN BE ACCOMMODATED.
1. MATERIAL PART TO BE MADE OF CR52, TYPE 347.

PREHEATER,  
HYDROGEN

SK68004

The heat exchanger is assembled using brazed tube/header plate assemblies. The shell and head closures are welded.

Performance

The listing below presents the design requirements for the hydrogen preheater heat exchanger. Figures 5-10 through 5-12 present the detailed performance characteristics of the unit shown in drawing SK 68004. The curves give relationships of heat transfer, pressure drop and effectiveness for combinations of hot and cold flow.

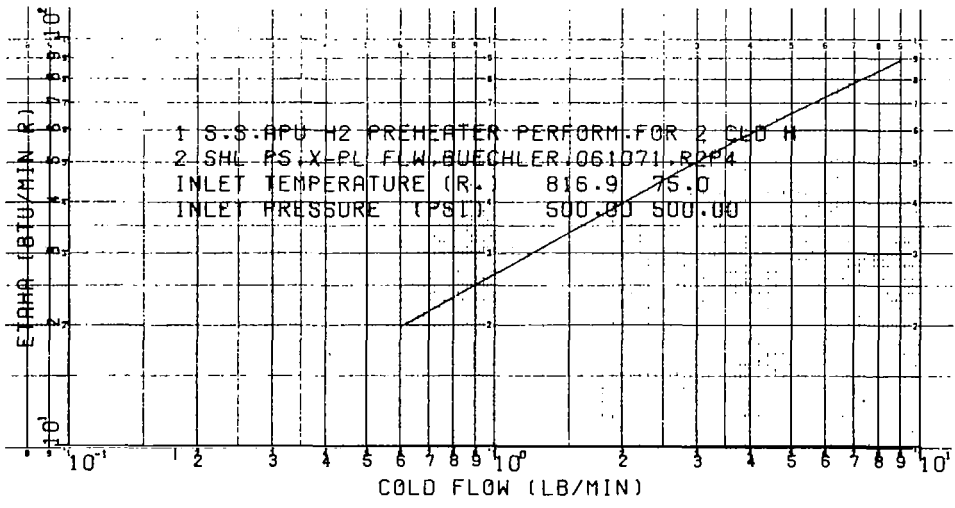
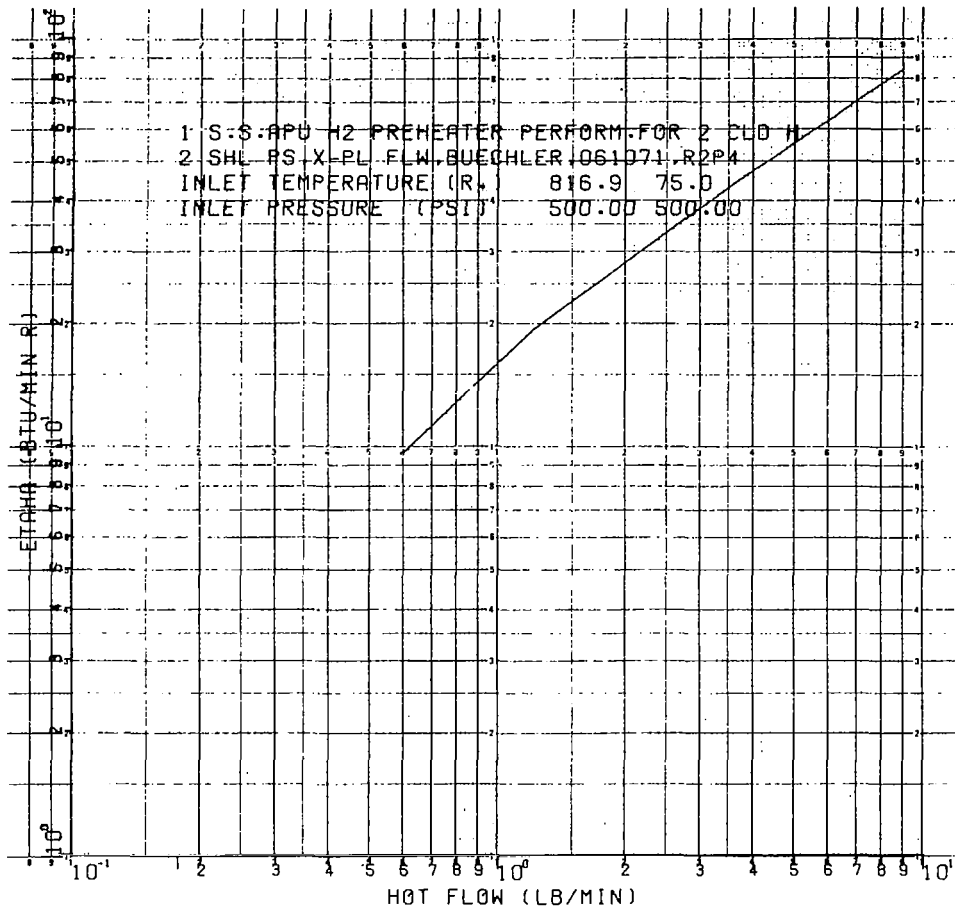
	<u>Cold Side</u>	<u>Hot Side</u>
Fluid	Hydrogen	Hydrogen
Flow rate, lb/min	6.93	6.33
Inlet temperature, °R	75	782
Outlet temperature, °R	392	408.2
Inlet pressure, psia	500	452.8
Total pressure drop, psi	0.1	0.4
Effectiveness	0.448	0.529
Duct diameter, in.	1.125	1.5
Total heat transferred, Btu/min 8410		
Number of tubes	380	
Tube diameter	0.100 in. OD	
Tube wall thickness	0.010 in.	
Tube designation		
Inside	DMPO5	
Outside	SB 135100	

RECUPERATOR HEAT EXCHANGER

This heat exchanger operates with hydrogen on the cold side and turbine exhaust gas on the hot side. It serves to provide sufficient heat input into the cycle for propellant thermal conditioning (with low inlet hydrogen temperatures) and to improve cycle thermal efficiency by recovering waste heat from the turbine exhaust. Overall cycle considerations in recuperator location and design are discussed in Section 2 of Volume IV.

Description

The recuperator is a box and tube design as shown in Drawing SK 68002. The exhaust gas from the turbine flows in a single pass through the shell side of the heat exchanger. This minimizes the pressure drop in the exhaust gas stream. The hydrogen flows in cross counter flow through the tubes of the

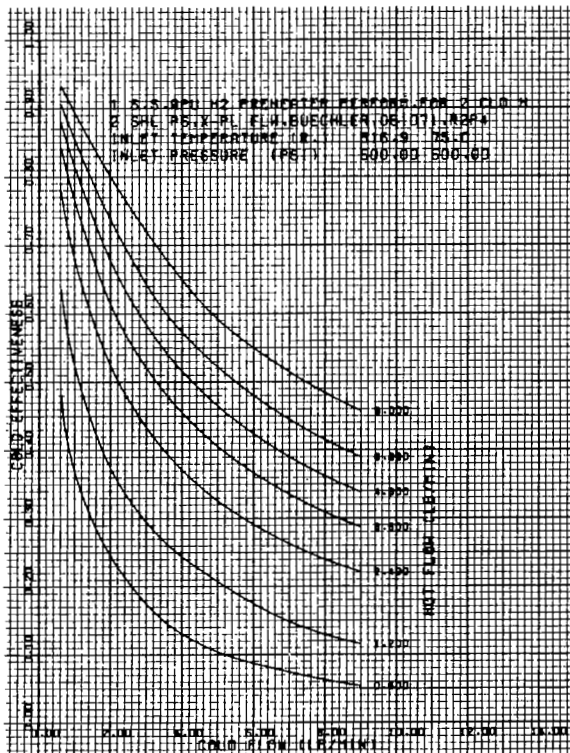
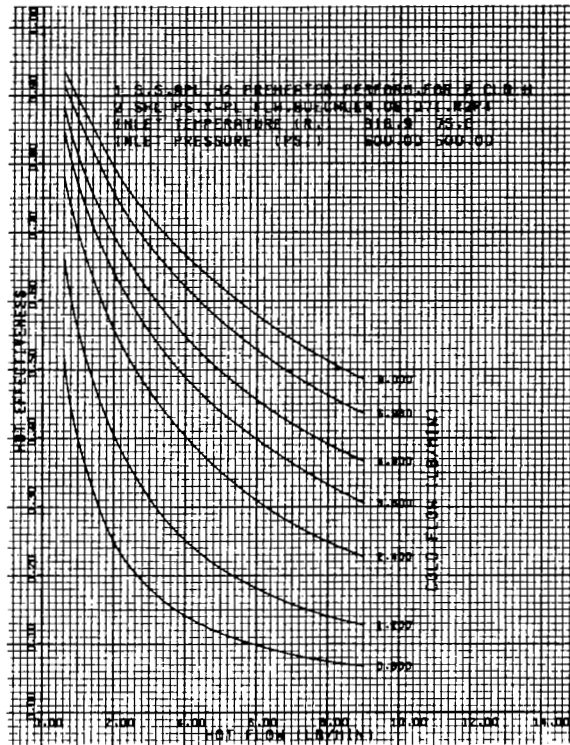


S-67262

Figure 5-10. Preheater Heat Exchanger Heat Transfer







S-67287

Figure 5-12. Preheater Heat Exchanger Effectiveness



unit. This flow arrangement allows the box structure to be lightly pressure loaded by the exhaust gas (less than 10 psi), and the high pressure hydrogen (450 to 500 psia) is contained within the tubes of the heat exchanger.

An expansion joint is included in the flat shell surface between the two hydrogen passes. This flexible joint relieves stresses which would otherwise occur due to temperature differentials approaching 600°R. As discussed in the structural design considerations for this unit, dimples are included in sections of the flat box sides to prevent stress buildup which could lead to shortened component life.

Performance

The listing below presents the design requirements for the recuperator. The detailed performance for the heat exchanger is presented in Figures 5-13 through 5-15. The performance characteristics include relationships for heat transfer, pressure drop, and effectiveness for various values of hot and cold side flow. The performance of this unit is a trade off between high effectiveness on the hot side to increase system performance, as opposed to exhaust gas cooling to the extent that freezing of the exhaust products could occur. A minimum exhaust gas outlet temperature of 700°R is used to satisfy both requirements.

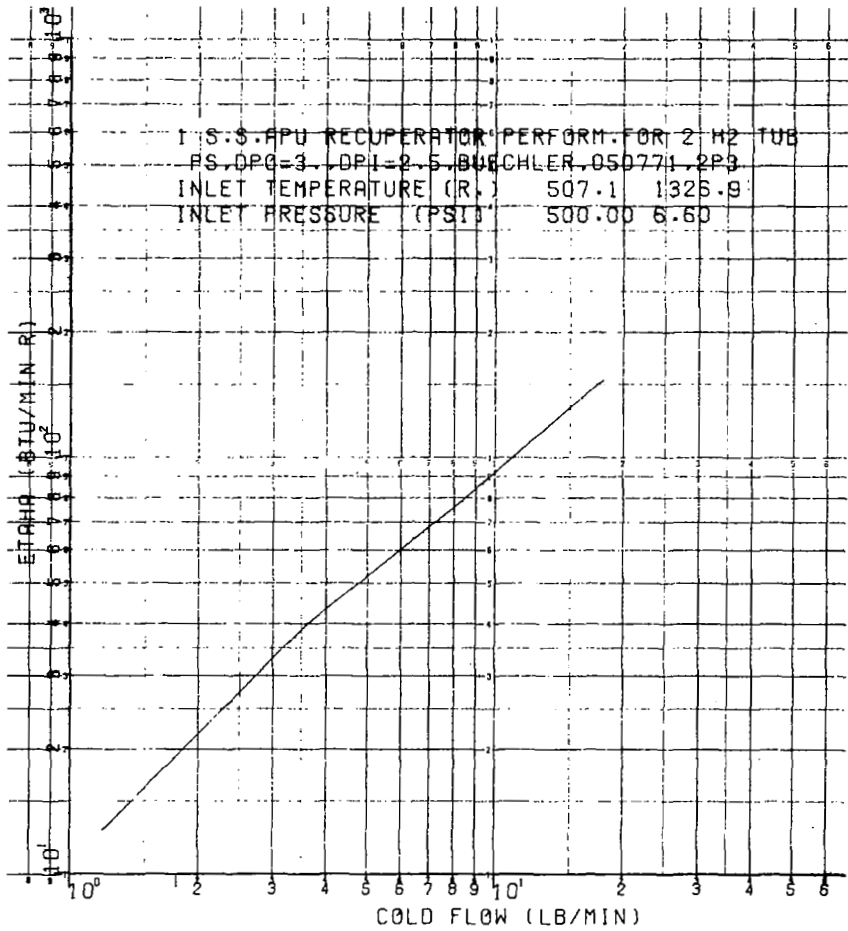
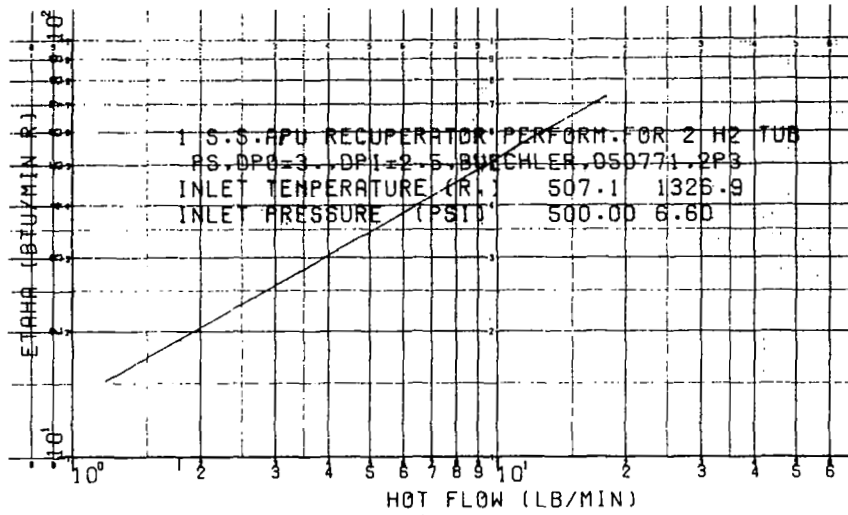
	<u>Cold Side</u>	<u>Hot Side</u>
Fluid	Hydrogen	Hydrogen-Steam (60-40 by Mass)
Flow rate, lb/min	13.26	12
Inlet temperature	508.1	1326.9 <sup>††</sup>
Outlet temperature, °R	781.8	810.4 <sup>†</sup>
Inlet pressure, psia	456.7	6.6
Total pressure drop, psi	3.8	3.0
Effectiveness	0.334	0.631
Duct diameter, in.	1.0	4.0

Total heat transferred, Btu/min 12750

<sup>†</sup>Minimum allowable outlet temperature = 700°R

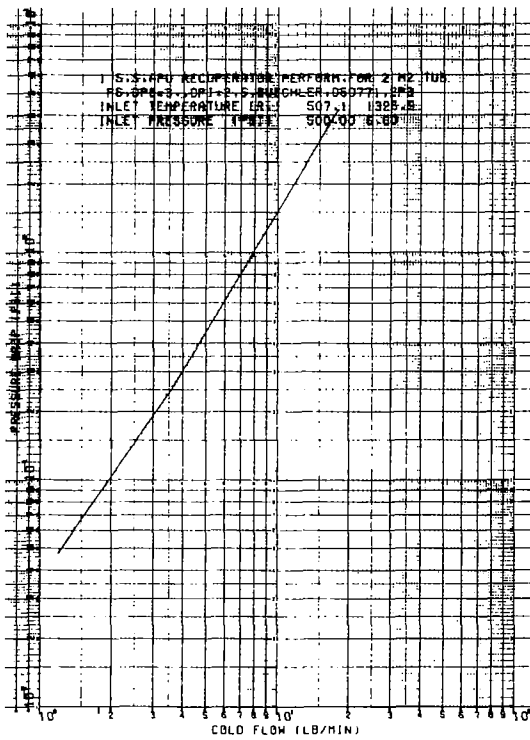
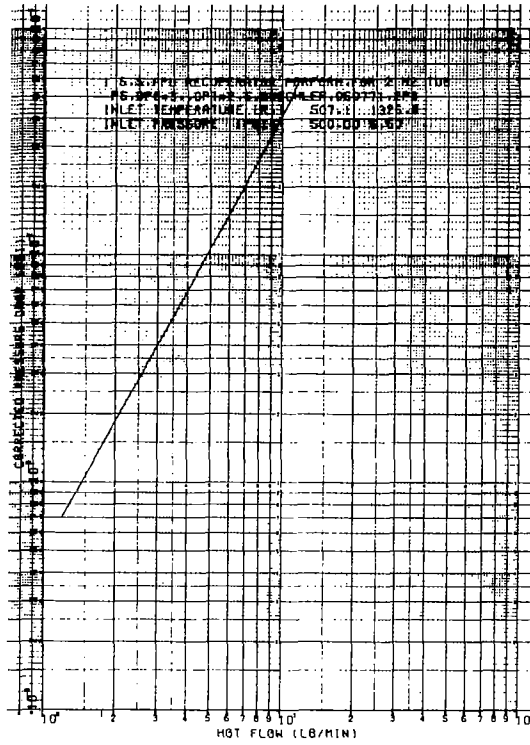
<sup>††</sup>Maximum inlet temperature 1388°R

Number of tubes        770  
 Tube diameter        0.100 in. OD  
 Tube wall thickness 0.008 in.  
 Tube designation  
     Inside        DMP 03  
     Outside      SB 300100



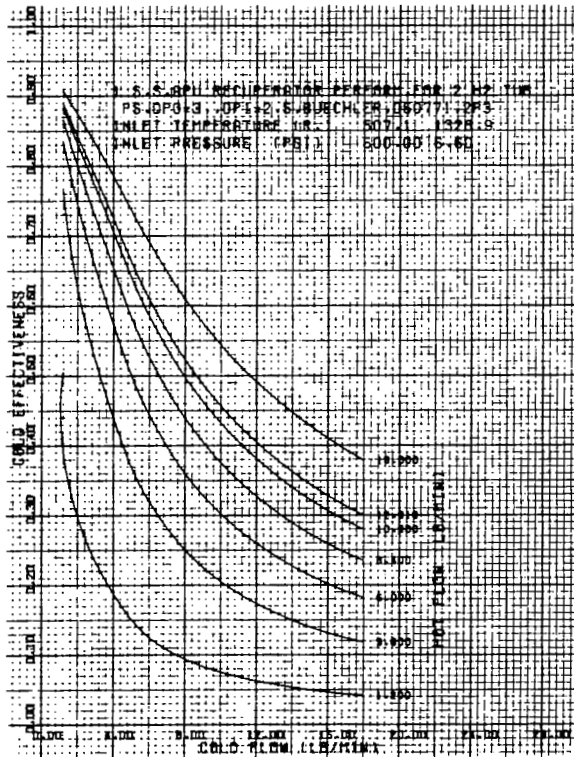
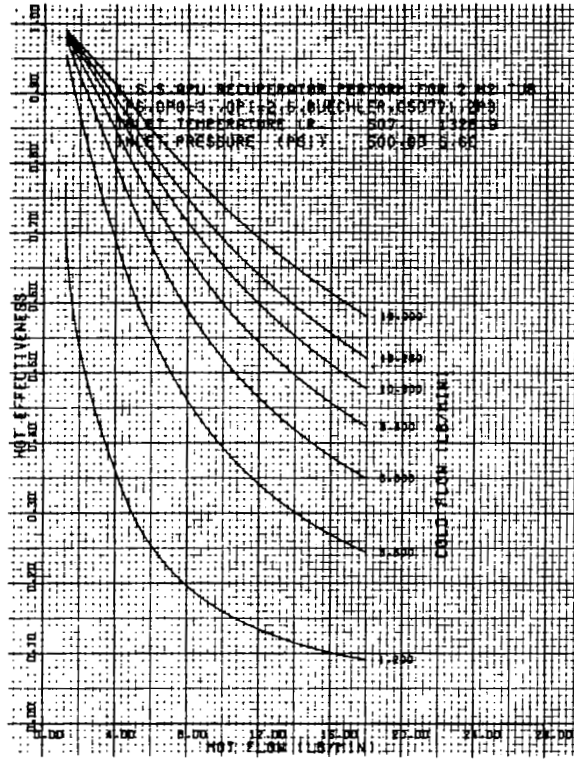
S-67261

Figure 5-13. Recuperator Heat Exchanger Heat Transfer



S-67286

Figure 5-14. Recuperator Heat Exchanger Pressure Drop



S-67285

Figure 5-15. Reciprocator Heat Exchanger Effectiveness

## DESIGN BASES

### Valving

The APU valves described earlier are based on designs of valves in comparable applications. Figure 5-16 contains two drawings of specific valves which illustrate some of the design features employed in the APU valving. The pressure relief valve, P393298-1, uses a flat guide spring to assure alignment of internal parts which have relative motion. This method of parts alignment is used in the APU shutoff, regulator and the flow control valves. The valve assembly shown also includes a bellows assembly for positive sealing between moving parts, and for pressure balancing of internal actuation forces. This approach is used in the APU regulator, and to a lesser degree by the balanced convoluted diaphragms of the shut off and flow control valves.

The solenoid shut off valve shown in Drawing P393720-2, includes features of solenoid operation (similar to the shut off valve) bellows sealing of moving parts (similar to shut off, regulator and flow control valves), hardened poppet in a soft body seat (similar to shut off, regulator, and flow control valves), provision for valve position indication, (similar to flow control valve and recycle valve). Primary and secondary seals are shown in the design to eliminate external leakage through static seal areas.

### Jet Pump

Many AiResearch air cycle refrigeration systems for aircraft utilize jet pumps. These vary greatly in size, depending upon the refrigeration capacity of the air cycle system with primary air flow ranging from about 20 to over 1000 lb per min. AiResearch has developed a number of smaller jet pumps for cryogenic systems and for portable life support systems (using cryogenic or high pressure breathing gas supply). Figure 5-17 shows representative test data for an application of this type, which involved testing of a number of different designs.

### Heat Exchangers

Figure 5-18 presents several tubular shell and tube cryogenic heat exchangers that have been produced in quantity, and which operate over the temperature range of interest for the APU hydrogen preheater. The unit shown in Figure A for the primary oxygen supply heater includes provisions for prevention of coolant freezing problems. Similar design considerations are included in drawings shown earlier for the lube oil and hydraulic oil coolers. Figure 5-19 shows stainless steel and aluminum oil coolers built for a number of aerospace applications. All of the units shown use shell and tube construction.

Conventional gas turbine engine recuperators have many design requirements in common with the regenerator for this application. Some of the tubular recuperators, designed fabricated and proven in service are shown in Figure 5-20. Recuperators have been developed and produced for such engines as the T-56, T-78 and the T-53.



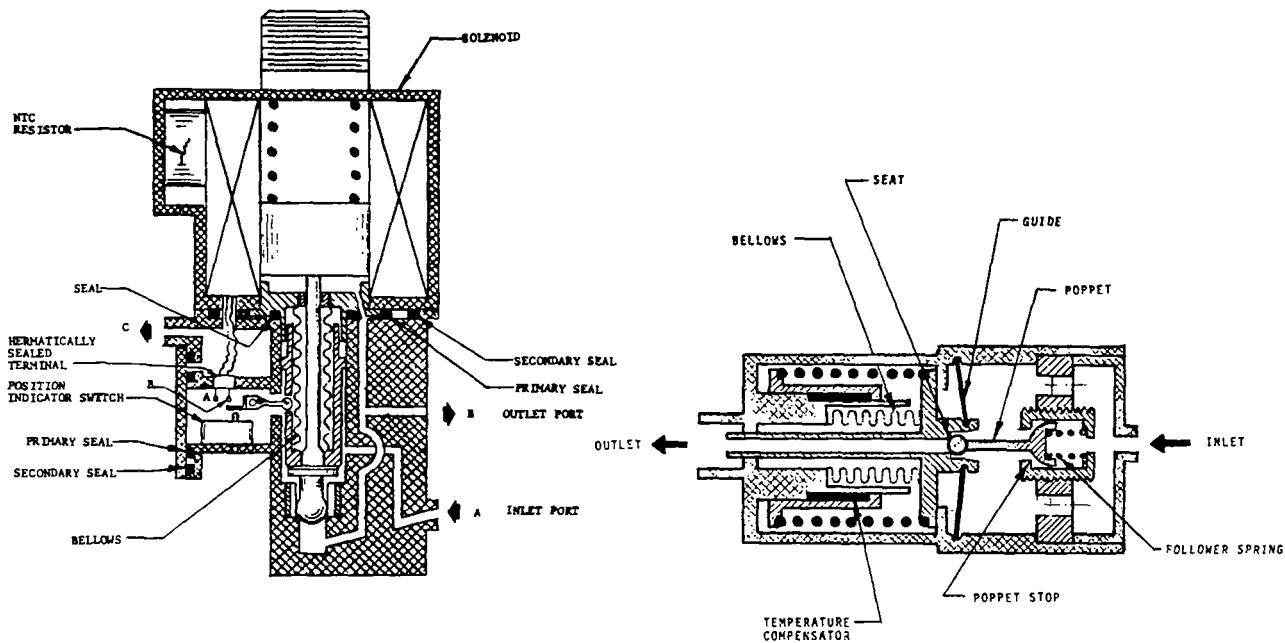


Figure 5-16. Applicable Valve Designs

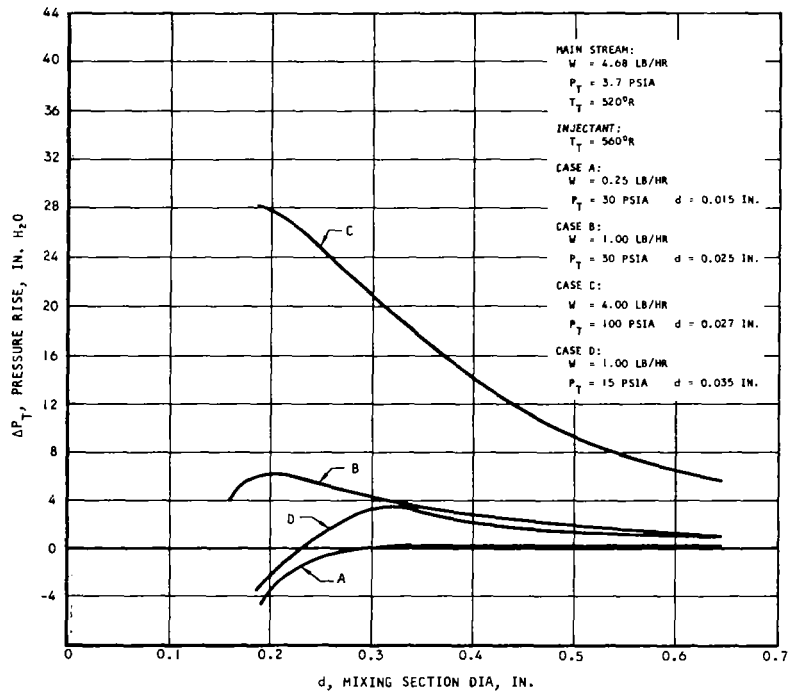


Figure 5-17. Pressure Rise vs Mixing Section Diameter

S-67293

## CRYOGENIC HEAT EXCHANGERS

Among the cryogenic heat exchangers designed and fabricated by AiResearch are two cryogenic heaters used in the Gemini spacecraft. One unit, shown in the upper photograph, is used for two separate applications: to heat cryogenic hydrogen and to heat cryogenic oxygen. The heat exchanger is of the shell-and-tube type, fabricated entirely from stainless steel using brazed and welded construction. The complete unit shown has two separate coolant circuits for redundancy. The flow configuration in each section is cross counterflow, with the oxygen making one pass through the tubes and the MCS198 coolant flowing over the tubes. The division between the two units is a plate across the shell that separates the coolant circuits. The oxygen tubes are continuous through both units.

The cryogenic heat exchanger shown in the center photograph is the primary ECS oxygen supply heater for the Gemini capsule. The fluids, flow configuration, and construction are identical to those of the unit previously described. An unusual feature in the design of this unit was the use of a special sleeve inside the shell to minimize bypassing of liquid. A manufacturing limitation of normal shell and tube designs is that the space between tubes and shell is greater than that between tubes; this permits a small amount of fluid bypassing. In the original design of this unit, the bypassing incurred some MCS198 freezing problems. The use of the special sleeve to reduce bypassing eliminated the freezing problem completely.

A prototype heat exchanger for the LEM helium system also was fabricated and tested. This unit, shown in the lower photograph, was delivered to Grumman for use in R&D work on the LEM Helium System. The heat exchanger is a shell-and-tube unit, fabricated entirely of stainless steel. The unit was designed for a parallel flow configuration, with supercritical helium flowing inside the tubes and a water-glycol mixture flowing in the shell. The critical design-point operating condition for this heat exchanger was to heat 2.34 lb per min of 2500-psia helium from 55°R to 460°R with 399 lb per min of water-glycol at 530°R.

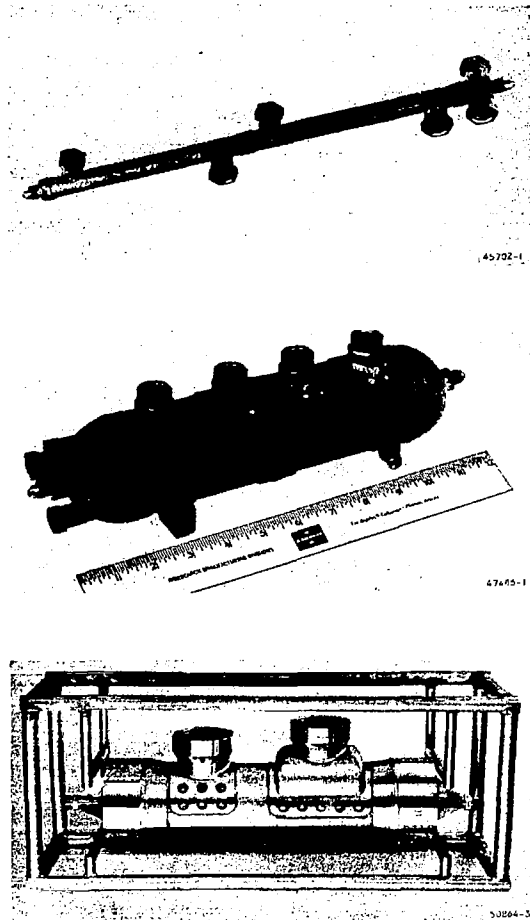
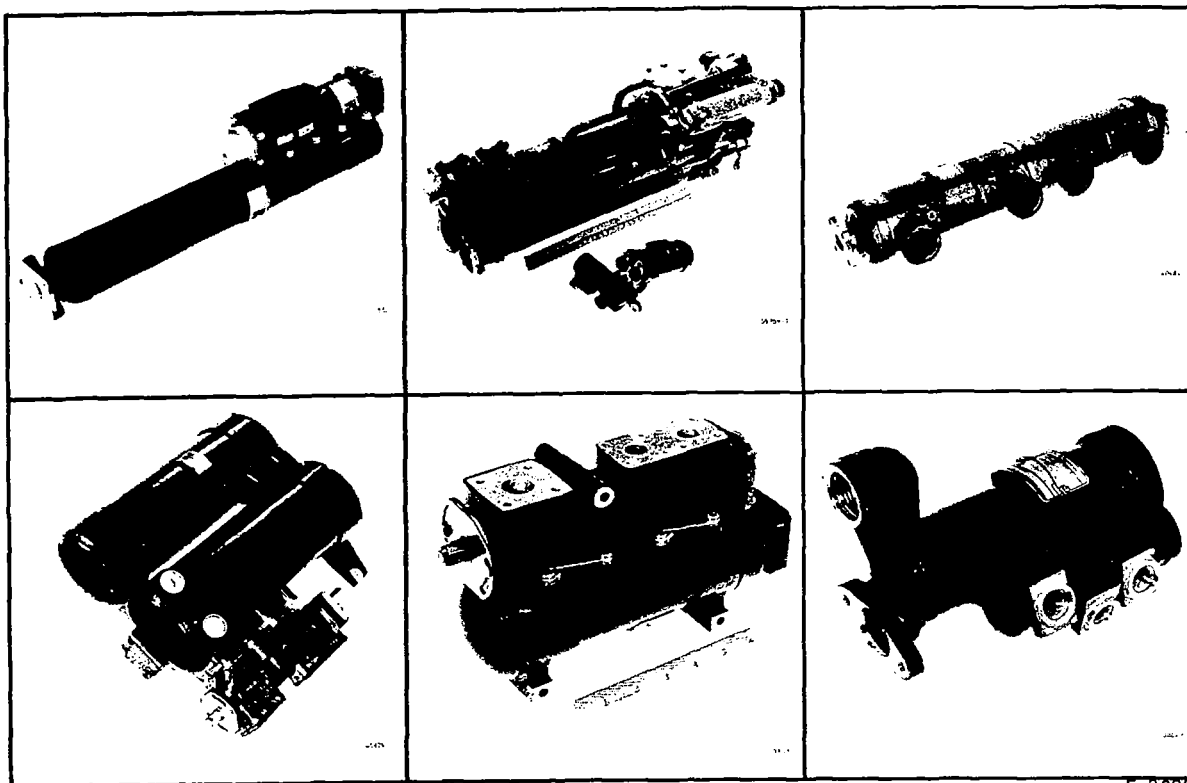


Figure 5-18. Cryogenic Fluid Heat Exchangers



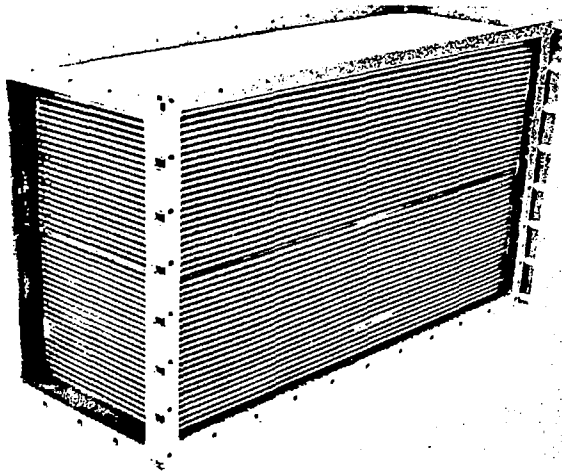
F-2800

### TUBULAR OIL COOLERS

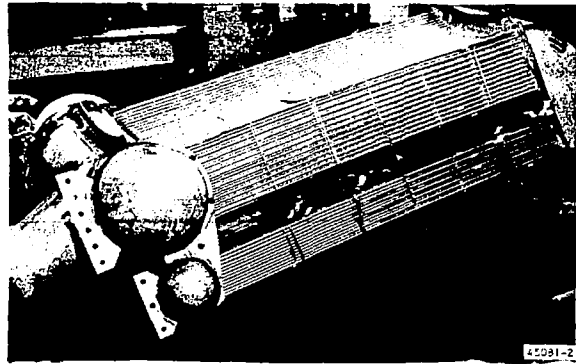
AiResearch has designed and built several thousand stainless steel and aluminum oil coolers of shell-and-tube construction. One type of unit, built for the General Electric J79-5 engine, had an internal fuel bypass tube in the core. Another unit, built for the McDonnell F-101A aircraft, had an internal fuel bypass tube that used the same type of fuel pressure relief valve.

Other tubular fuel-to-oil coolers have been produced by AiResearch for the GE J79-9 main engine and afterburner fuel circuits, GE J85, J93-3, CF700 and X211 engines, the North American Hound Dog Missile, the Douglas A3D, the Lockheed Electra hydraulic oil system, and the Northrop SM-62 missile. More than 3000 units have been delivered for the GE J79-8 engine, and several thousand more will be built. Typical tubular units manufactured by AiResearch are shown above.

Figure 5-19. Oil Cooler Units



45002-2  
PLATE-FIN RECUPERATOR MODULE

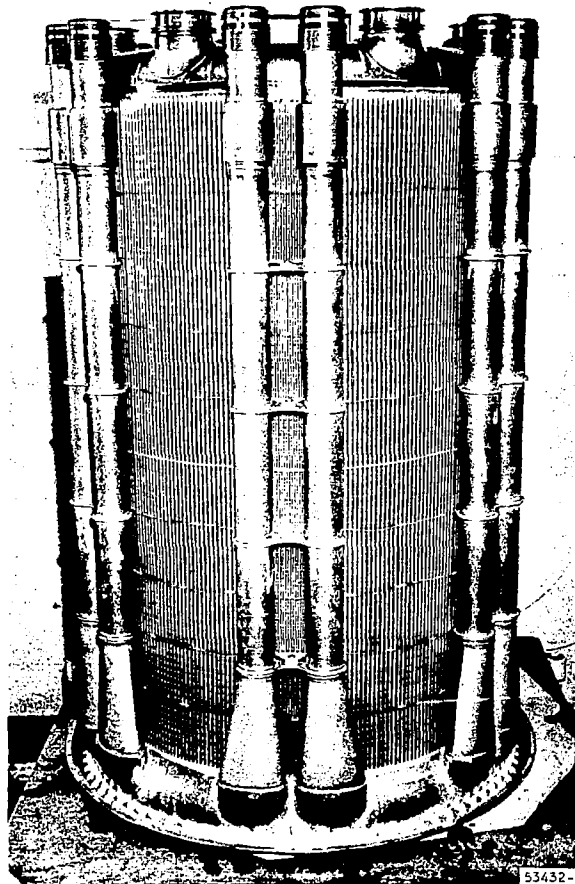


45031-2  
CURVED-TUBE RECUPERATOR MODULE

## RECUPERATORS

AiResearch has developed and produced recuperators for gas turbine and turboprop engines since 1958. Fuel control studies, fuel fouling investigations, and endurance testing of a 1-lb-per-sec recuperator fabricated at that time provided data required to develop recuperators for larger engines. Compact lightweight recuperators have since been developed for high horsepower aircraft engines where long endurance at partial-load engine operation is required. Developments also are under way for recuperators for use with the small gas turbines produced by the AiResearch Phoenix Division.

AiResearch has produced recuperators under Air Force, Army, and Navy contracts, and as a subcontractor to a number of engine manufacturers. Techniques for fabricating the modules, and for joining the modules to form a recuperator, have been developed. The various types of modules developed and produced are illustrated by the plate-fin, straight-tube, and curved-tube modules shown above.



53432-  
STRAIGHT-TUBE RECUPERATOR MODULE

F-2344

Computer programs that account for such factors as fin-joint resistance, material properties, brazing material weight, flow configuration, nonuniformities, heat leakage, and interpass mixing assure optimum-design, high-efficiency recuperators with the lowest penalties in weight, size, power, and cost.

Figure 5-20. Recuperator Experience

## SECTION 6

### TURBINE POWER UNIT

#### INTRODUCTION

The turbine power unit consists of three major subassemblies, each containing a number of functionally-related components, as follows:

- (a) Hydrogen-oxygen combustor assemblies with propellant flow control valves, hot gas manifold, and turbine inlet nozzles.
- (b) Turbine rotating assembly with rotors, bearings, housing, inter-stage nozzles, and exhaust ducting
- (c) Gearbox assembly with gearing, accessory mounting pads, and integral lubricant scavenging and pumping system.

#### Functional Interfaces

Figure 6-1 shows the functional interfaces between the turbine power unit assemblies and the other two APU subsystems (controls and propellant conditioning/thermal control). It should be noted that as presently defined, the alternator and hydraulic pumps are not a part of this subsystem, although the shaft power and thermal control requirements have been considered in detail in system design.

#### Design Justifications

This section will first contain descriptions of the three major assemblies which will be followed by design justifications (in terms of state-of-the-art experience) for the various concepts selected.

#### HYDROGEN-OXYGEN COMBUSTOR ASSEMBLY

The hydrogen-oxygen combustor assembly delivers a controlled flow of hot gas to the turbine, using as inputs, gaseous hydrogen and oxygen from the propellant conditioning subsystem and signals from the controls subsystem for actuation of the propellant flow control valves. Drawing SK68007 shows the combustor-flow control valve assembly, less the hot gas manifold which is welded to the open end of the combustor shown in the drawing. The combustor assembly has the following functions:

- control of total propellant flow to match power demand of the turbine
- control of oxygen to hydrogen mixture ratio to control turbine inlet temperature
- mixing of hydrogen and oxygen for uniform combustion

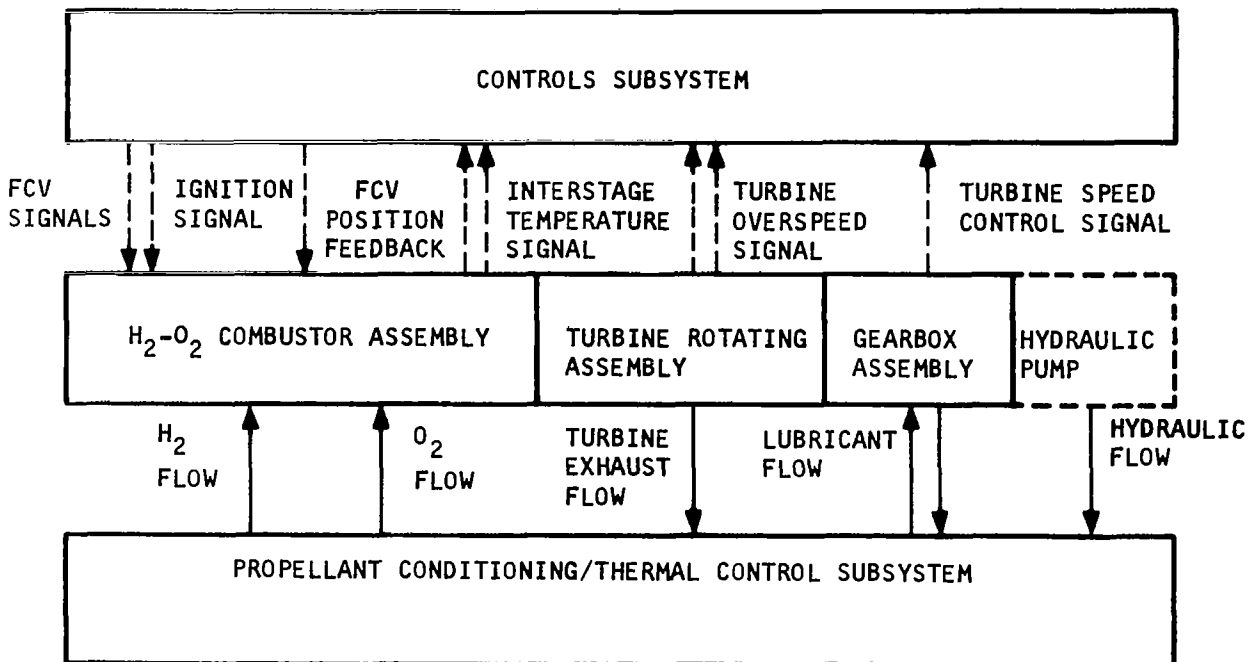


Figure 6-1. Turbine Power Unit System Interfaces

- ignition of the combustible mixture
- provide to the turbine a gas stream at uniform temperature, pressure, and flow for each power level required

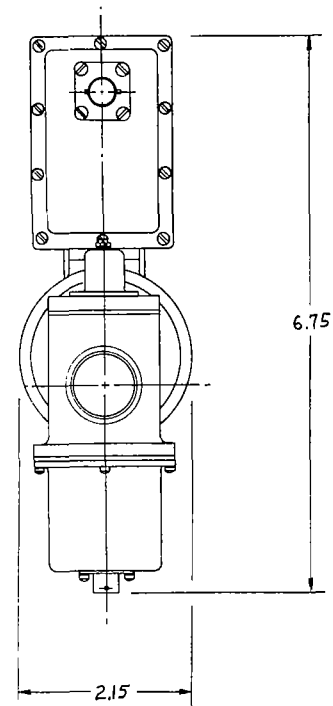
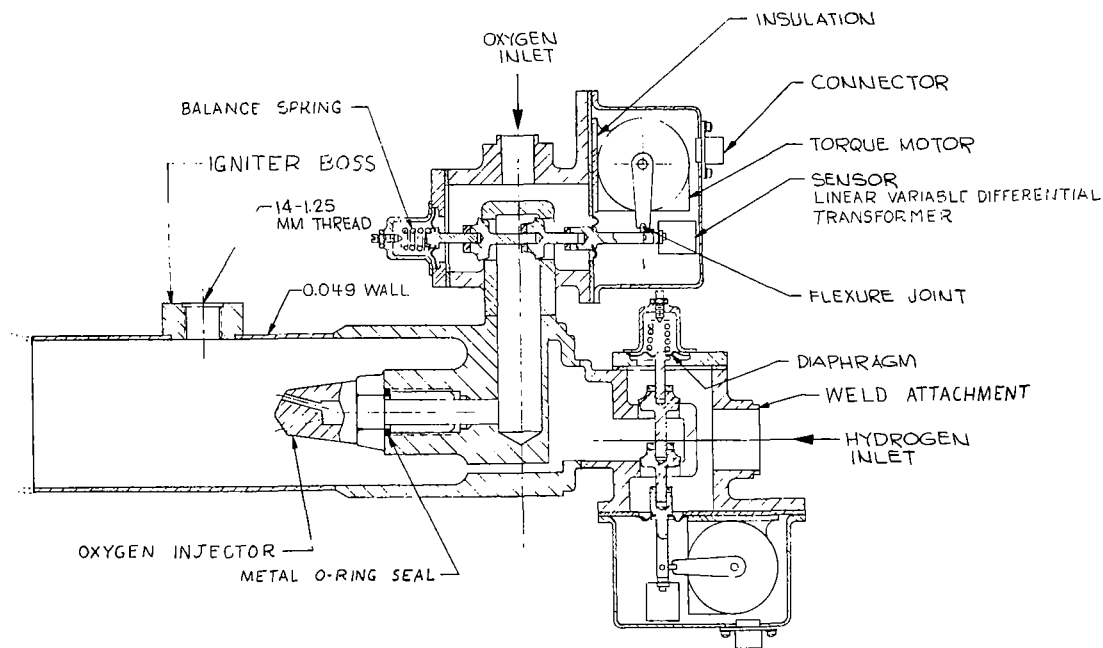
The combustor assembly consists of three major elements: (1) hydrogen flow control valve; (2) oxygen flow control valve; and (3) the combustor.

#### Flow Control Valves

Hydrogen and oxygen flow are controlled by separate electrically-driven flow modulating valves using balanced poppet designs to limit the force requirement of the torque motor actuator (thereby minimizing weight and power consumption for these components).

#### 1. Performance Characteristics

The performance requirements of the two propellant valve assemblies are presented in Table 6-1. Specifications of the individual components are included in the drawing. The torque motor selected exhibits a linear characteristic of poppet stroke (CA) as a function of the maximum control supplied to the torque motor. The relationship is shown in Figure 6-2.



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MATERIAL LIST

VALVE BODY 347 CRES  
 DIAPHRAGM INCONNEL  
 POPPET PH13-8 MO  
 INSULATION MIN-K  
 BALANCE SPRING INCONNEL  
 SENSOR SCHAEVITZ MB SERIES DIOMS-L  
 TORQUE MOTOR SERVONICS MODEL 20-2

SPECIFICATION

FLOW : OXYGEN 5.0 lb/MIN.  
 HYDROGEN 7.07 lb/MIN.  
 INLET PRESSURE 500 PSIA  
 TEMPERATURE : OXYGEN 300 TO 500 °R  
 HYDROGEN 750 TO 905 °R  
 FLOW MODULATION : 20 TO 1

POWER INPUT NOMINAL 2.65 WATTS

SENSOR SPECIFICATION:

STROKE ± 0.01 IN.  
 SENSITIVITY @ 400 CPS EXCITATION  
 2.47 MV/0.001 IN./VOLT

ASSY, FLOW CONTROL AND  
 COMBUSTOR INJECTOR

SK68007

TABLE 6-1  
FLOW CONTROL VALVE REQUIREMENTS

		Oxygen	Hydrogen
Flow, lb/min	Maximum	5	7.07
	Minimum	0.259	0.434
Temperature °R	Maximum	500	905
	Minimum	300	750
Inlet pressure, psia		500	500
Response time, MS		200	200
Leakage, sccm		50	50
Flow control tolerance, percent of flow		5	5
Duct diameter, in.		0.25	0.75
Valve position feed back required		yes	yes

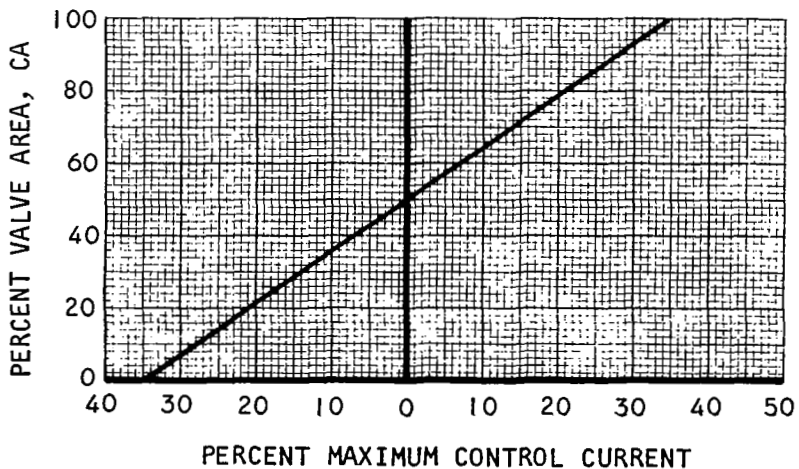


Figure 6-2. Flow Control Valve Area vs Torque Motor Current



A principal feature of the valve design is the individual electric drive of each valve using torquemotors. The torque motor provides the high spring rate in the drive direction (similar to a gear train) and also offers fast response to system control commands. The reliability of the torque motor is superior to a motor and gear train assembly. The valve response of 100 cps is calculated. The individual electrical signals to the torque motor and the individual feed back sensor output complete the control loop.

The poppet position sensors are linear differential transformers. The poppets are straight conical elements, shaping of the flow-position command relationship to a logarithmic function is accomplished within the controller.

## 2. Packaging

The arrangement of the valves with respect to the combustor injector affords a relatively flat assembly outline which lends itself to the package configuration shown in Section 3.

## 3. Seals

The propellant flow control valves need not seal at the low flow condition since the upstream shutoff valves perform this function. The flow control valves are designed to provide a flow modulation ratio of approximately 14:1, which does not include sealing.

The pressure forces of the dual, pressure balanced poppet assembly are equalized at the seal area by the opposing dual-convoluted diaphragms. These diaphragms serve to center the poppets in the bore of the seat because of high spring rate in the radial direction, while exhibiting a low spring rate in the transverse direction.

Dynamic seals are avoided in the design of the valves. The method of achieving a seal between the propellant and the ambient is by use of diaphragm/bellows designs. Convoluted diaphragms are used for sealing around the poppet actuation mechanism, and also to serve as poppet guides. These devices are a single piece component. Since they contain no weld joints, the possibility of leakage occurring at a weld is eliminated.

## 4. Adjustment

The balance spring assembly allows final adjustment of actuation torque requirements after assembly. The balance spring on the oxygen valve will be adjusted to drive the valve to the most closed position in the event of loss of electrical power. The hydrogen valve balance spring will be set to drive the valve to the full open position in the event of an electrical failure. Use of this approach to fail safe operation eliminates the possibility of a combustor outlet temperature higher than the desired 2060°R.

## 5. Thermal Considerations

Thermal protection of electrical drive components is provided by the combination of resistance to heat transfer to the vicinity of the components, insulation of the component mounting from direct contact with the metal structure, and radiant heat loss to an environment at an average temperature of 500°R. Implementation of these considerations in the design minimizes the thermal stress imposed on the electrical components of the valve assemblies.

## 6. Position Feedback Sensor

The flow control valve requires accurate valve element position feedback control. The poppet assembly is tracked using a linear variable differential transformer (LVDT) sensor. Backlash between the torquemotor, poppet assembly, and the LVDT is eliminated by the use of flexure joints. This method of joining parts allows minor relative angular motion by flexing of thin members while maintaining positive linear relationships.

## 7. Alternative Mechanically Linked Valve Concept

An analysis was performed to compare the individual electrically driven valves with the mechanically linked propellant valve approach. Both approaches use the same electronic control concept (assuming both are pressure modulated turbines, and operate from the same control sensors). Either valving approach requires two actuators, one for valve element position control for total flow of propellants, and one for relative control of one valve element with respect to the other to yield mixture ratio control. Signal feed back (used on electrical or mechanical valve assemblies) insures accurate positioning of the electrically driven valve even if the electronics tend to drift. The same is true of the mechanically linked valves, based upon individual valve element position sensor information. The mechanically linked valves exhibit one characteristic which will require additional design and development effort and that is alignment and backlash. Mechanically linked propellant valves require that the propellants be brought into proximity prior to the valves and well ahead of the combustor injector. The separate electrically controlled valves eliminate this potential compromise to system safety.

The comparison summarized above results in the conclusion that the use of mechanically linked valving offers no advantage over the individual electrically driven valves. The electrically-driven valves with position feedback provide separation of the propellants without compromise of safety goals, afford greater packaging flexibility and offer the potential for simplified assembly, adjustment, and checkout.

## Hydrogen-Oxygen Combustor

The function of the combustor is (1) to provide uniform mixing of the hydrogen and oxygen for uniform combustion, (2) to provide for ignition of the resultant mixture, and (3) to provide to the turbine a gas stream at uniform temperature, pressure, and flow for each power level required.

## 1. Combustor Performance Requirements

The principal design requirements of the combustor are to (1) provide efficient, reliable performance throughout the operational range of flow and pressure, (2) provide a uniform temperature profile at the turbine nozzle inlet, and (3) to avoid high wall temperatures and localized hot spots on the combustor exterior.

The listing below presents the design requirements of the combustor.

TABLE 6-2

DESIGN CRITERIA FOR APU HYDROGEN-OXYGEN COMBUSTOR

Propellants	Hydrogen	Oxygen
Feed Pressure, psia	min 26	26
	max 500	500
Inlet temperature °R	700 to 1100	300 - 500
Pressure drop at high flow, psia	13	35
Chamber psia	450	
Chamber °R	2060	
Weight flow lb/min	0.7 to 13.5	
Mixture ratio lb O <sub>2</sub> /lb H <sub>2</sub>	0.5 to 0.8	
Life cycles	1000	

## 2. Combustor Description

The combustor is shown in sectional view in Drawing SK68007. The combustor head is machined from a stainless steel casting and contains the hydrogen and oxygen manifolds including the cavities for the propellant control valves. The hydrogen injection holes are drilled directly into this casting, while the oxygen injector is a separate part threaded into the combustor head. The body tubular portion of the combustor is welded to the combustor head to assure leak tight assembly. The combustor head is never subjected to combustion temperatures, therefore, high-temperature seals are not required.

The body of the combustor is of CRES 347 steel which combines low cost, availability, ease of fabrication (machining, forming, and welding) with high strength at the operating temperature of 2060°R. The combustor head is an integral part of the turbine inlet scroll, therefore, high temperature seals are not required within the pressurized regions.

### 3. Hydrogen Injection

In this combustor design the hydrogen is introduced axially along the inner chamber wall through a series of holes from the hydrogen inlet manifold. Although the injector pressure drop is small, the gas is introduced at a velocity of several hundred feet per second velocity. This produces a cooling gas layer along the chamber wall. Thus, cool combustor chamber walls are obtained without the use of additional devices such as sleeves, cooling jackets, or other devices and the hot combustion core is maintained in the center of the chamber by hydrogen gas flow along the walls.

### 4. Oxygen Injection

The oxygen is introduced through a group of small holes near the chamber center line in a direction nearly parallel with the hydrogen flow. Hydrogen flow in the center of the chamber is well-mixed as the average velocity at the point of oxygen injection is only one-tenth of that when it leaves the injector. This swirling, plus the velocity of the oxygen jet, causes rapid mixing in the center of the chamber and ensures rapid and essentially complete combustion.

As mixing occurs in the combustor between (1) the combustion products, (2) the heated but uncombusted hydrogen in the core region, and (3) the wall-cooling hydrogen, the resultant temperature is maintained at the desired 2060°R at the turbine inlet.

### 5. Ignition

Ignition is accomplished by means of a spark plug located adjacent to the chamber wall and outside of the hot combustion area. This location promotes long plug life since it is never in the combustion zone except at the moment of ignition. The ignition sequence allows for a slight oxygen lead (approximately 50 ms) which fills the chamber with oxygen and when the hydrogen flow is started a flammable mixture occurs momentarily at the spark plug. Once the combustor is ignited, the spark plug can be turned off.

## TURBINE CONFIGURATION/MECHANICAL DESIGN

The APU turbine is a 2-stage pressure-compounded axial flow turbine rated at 450 shp at 70,000 rpm. Drawing SK39908 shows the turbine, gearbox, and the hydraulic pumps and alternator. The following pages describe the selected turbine configuration and the mechanical design studies conducted to insure integrity.

### Description of Configuration

The turbine assembly includes the turbine disks, shaft, bearings, bearing carrier and pre-load spring, the interstage nozzle, the exhaust manifold, and the overspeed sensor. The turbine stages are cantilevered with the second stage inboard. The disk shapes are identical, the only difference being the blade height increase from the first to second stage. This commonality of





disks is attractive from a manufacturing cost standpoint while concurrently satisfying the thermodynamic performance requirements. Attachment of the wheels is by means of curvic couplings and off-centerline bolts. This concept was selected because it offers assembly ease and also because of the low degree of disk material weldability. Additionally, the conductive contact resistance of the curvic coupling interfaces acts as a thermal barrier.

The turbine shaft bearings are 25 mm, angular contact bearings of M-50 tool steel (60 Rc or greater) which have been extensively used by AiResearch in similar applications at the same speed and loading conditions. They are spring loaded to 125 lb nominal preload force and spray lubricated and cooled. The hot end bearing has special cooling provisions provided by incorporating grooved inner bore for additional oil flow around the shaft. Further, there is a positive flow of oil inside the drilled shaft that is circulated by the hot end oil slinger.

The turbine shaft seal is a critical design component, forming the only potential path for oil migration from the turbine and gearbox. Such oil losses might be incurred during APU operation and/or during standby mode while the vehicle is in orbit. Every effort has been concentrated on a design that minimizes and negates the loss of lubricant. The selected design solution is a mechanical carbon face seal that is bellows actuated. The anticipated wear rate of the seal face is minimal since it is both lubricated and cooled by oil vapor during APU operation. An alternate seal design to give greater wear life has been considered and would consist of a pressurized lift-off seal assembly.

### Mechanical Design

The Phase II mechanical design studies represent an extension of those performed during Phase I (described in Appendix B of Volume II). However, NASA-directed design changes have necessitated reanalysis of the turbine. NASA has specified that the turbine inlet temperature be reduced from the 2260°R used during Phase I to 2060°R. Further, NASA has eliminated use of hydrogen passages in the turbine casing as a means of cooling the turbine disks.

The mechanical design analyses are as follows:

- turbine disk steady-state and transient temperature distributions
- turbine disk allowable stresses
- turbine disk steady-state and transient stress distributions
- turbine disk containment
- rotating assembly critical speeds
- rotating assembly bearing operating conditions

## 1. Turbine Disk Steady-State and Transient Temperature Distributions

Approximately 100 nodes were used to model the rotating assembly with a transient thermal analyzer program based on a modification of the Runge-Kutta relaxation technique. Figures 6-3 and 6-4 show the resulting steady-state and transient turbine temperature profiles. The transient profiles assume that the initial turbine inlet temperature is 2060<sup>o</sup>R; however, the controls have been designed to reduce the temperature below this level during startup. These data were computed using the assumption of no case cooling. A consequence of the lack of case cooling is that the turbine disks are necessarily sized to lower allowable stresses, and are therefore relatively heavier. The data indicate first stage turbine blade temperatures of approximately 1300<sup>o</sup>F and a hub temperature of about 1190<sup>o</sup>F. Second stage temperatures are less stringent and are of primary consequence for the transient condition.

## 2. Turbine Disk Allowable Stresses

Table 6-3 shows the four stress criteria considered in selecting the turbine disk steady-state and transient allowable stresses. The lowest values for steady-state and transient conditions have been used in obtaining the selected disk designs. Figures 6-5 and 6-6 plot the allowable stresses of the disks as a function of disk radius, using the steady-state temperature profiles of Figures 6-3 and 6-4.

TABLE 6-3

### CRITERIA FOR SELECTING ALLOWABLE DISK STRESSES

Overspeed	Design Overspeed 1.3 (Nominal design speed) 91,000 rpm Uses 80 Percent of ultimate stress as a limit
Disk growth	Limits creep effects to 0.1 percent overall allowable growth; blade tip/casing clearance maintained
Low cycle fatigue	Stress strain hysteresis fatigue effects considered using an extension of Neuber's rule and the Manson-Halford equation
Maximum allowable yield and ultimate stresses	Uses safety margin applied to yield and ultimate stress of disk material



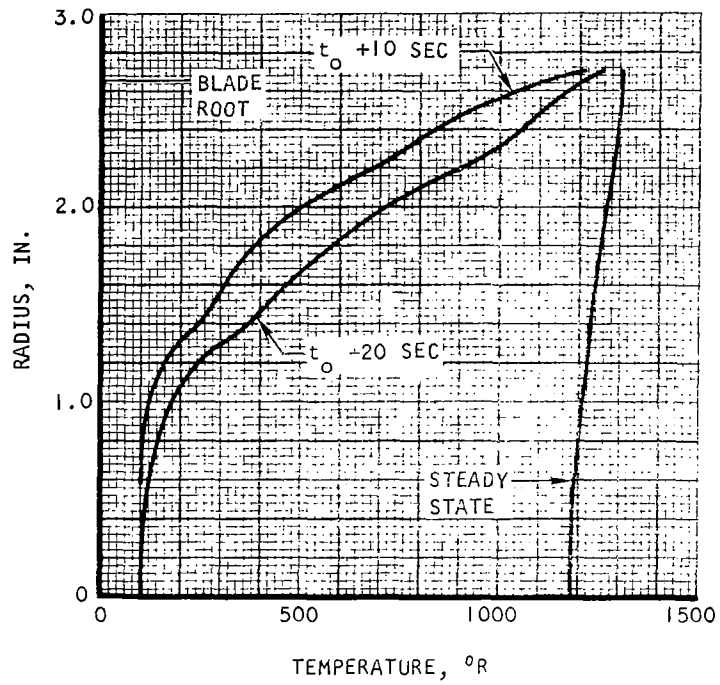


Figure 6-3. First-Stage Temperature Profiles

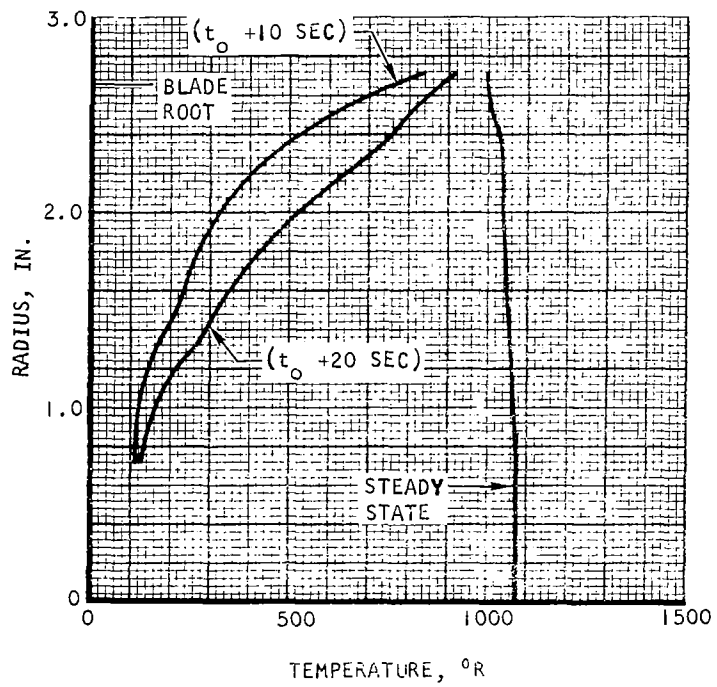
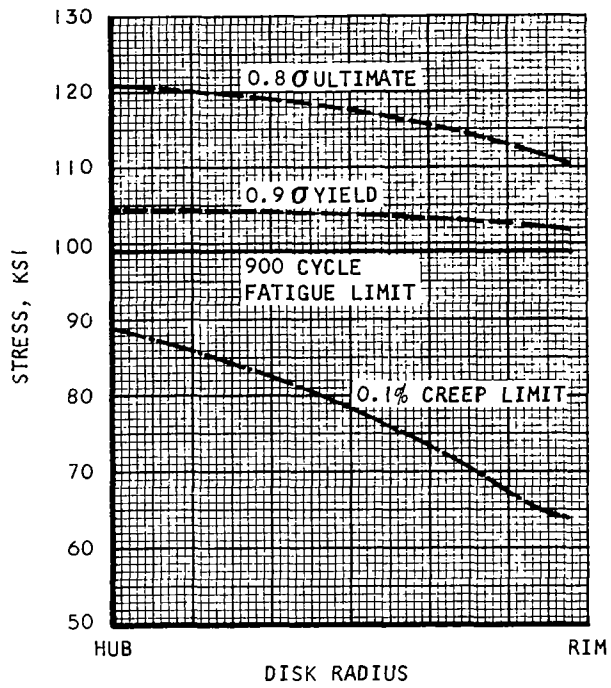
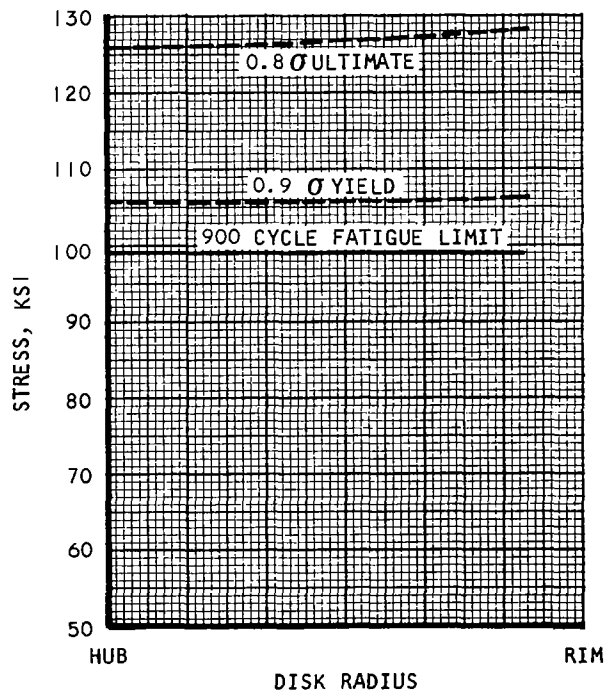


Figure 6-4. Second-Stage Temperature Profiles



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Figure 6-5. First-Stage Stress Limit Criteria



S-67119

Figure 6-6. Second-Stage Stress Limit Criteria

The low cycle fatigue allowable stress was obtained using a computerized elastic-plastic stress-strain analysis. Figure 6-7 shows the relationship between the number of cycles and the allowable stress used in this study. The disk areas adjacent to the bolt holes were a focal point of this analysis because of the stress concentration factor associated with the holes.

The limiting stresses in steady-state are the 0.1 percent creep criterion on the first stage and the fatigue limit on the second stage. In the transient, the limiting stresses on both stages are 90 percent of the allowable yield stress.

### 3. Turbine Disk Steady-State and Transient Stress Distributions

Figures 6-8 and 6-9 show the radial and tangential components of the combined centrifugal and thermal stresses on the first and second state disks during steady-state operation. The figures also show the allowable stress for each disk. The selected designs are within the allowable stress envelope.

Figures 6-10 and 6-11 show the components of the combined stresses on the disks during the startup transient (at 10 sec after startup with the disk at speed). Again, the designs are within the allowable stress envelope.

In both steady-state and transient stress distributions, the discontinuity in stress at mid-radius is due to the influence of the curvic coupling platform.

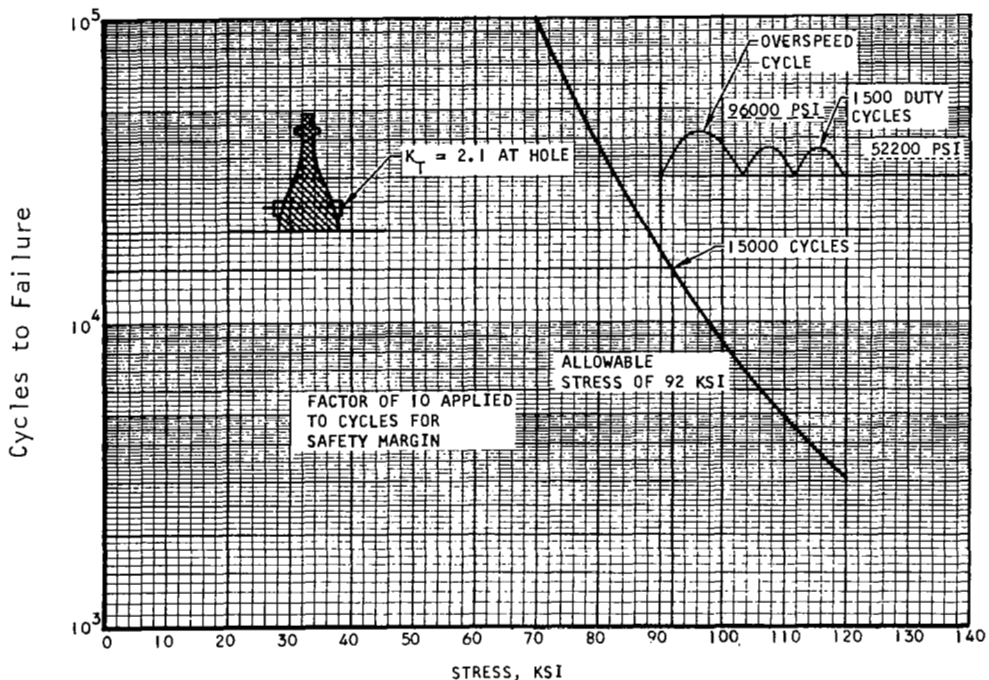


Figure 6-7. Stress vs Cycles to Failure for Udimet 700

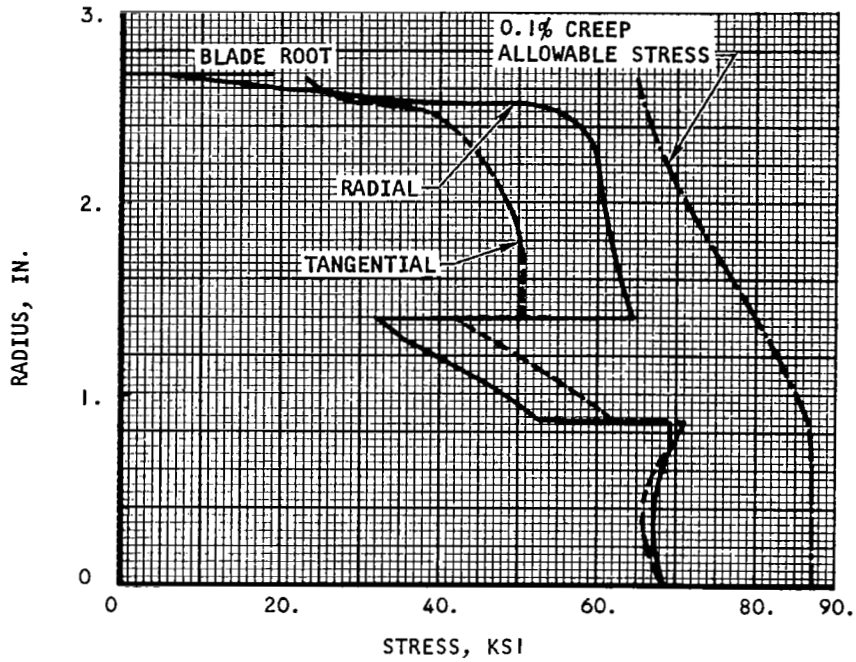


Figure 6-8. First-Stage Steady-State Stresses at 73,500 Rpm

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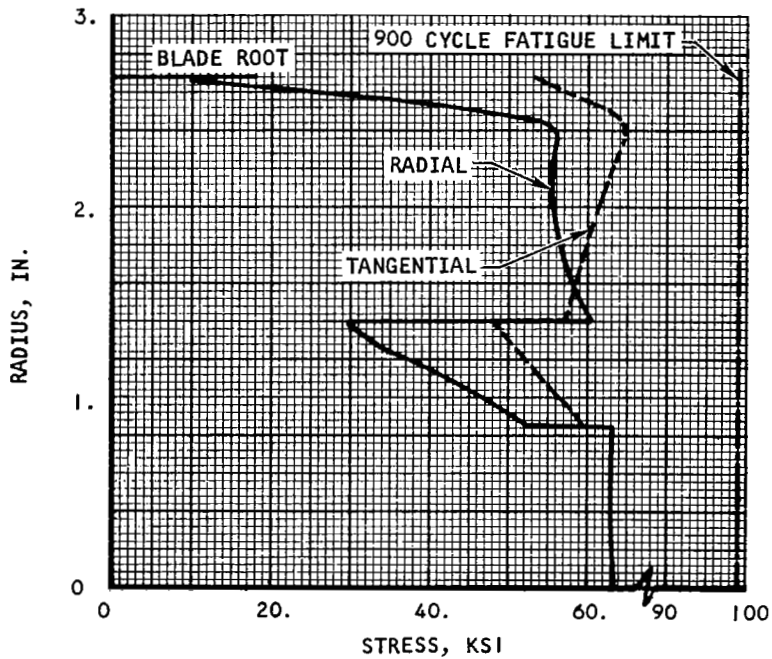


Figure 6-9. Second-Stage Steady-State Stresses 73,500 Rpm

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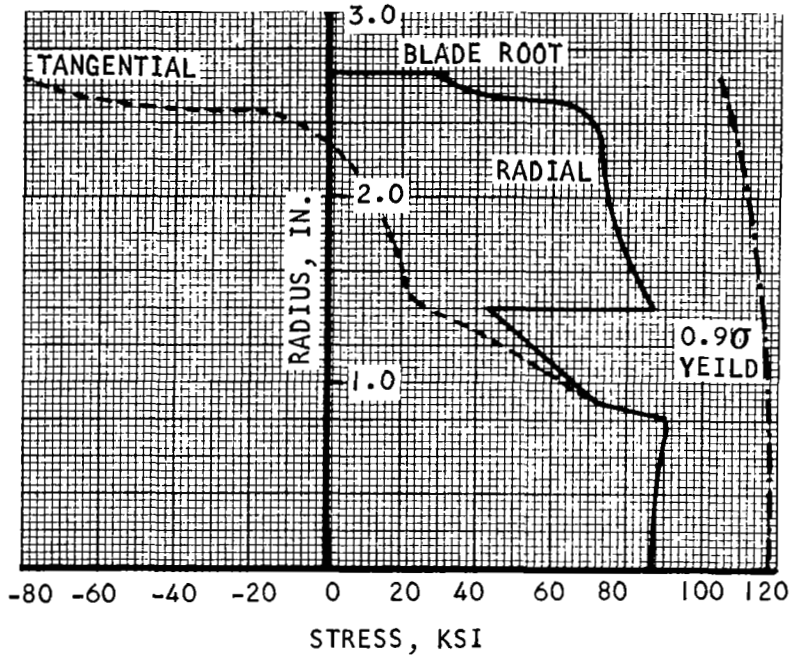


Figure 6-10. First-Stage Transient Stresses at 10 Sec after Startup and 73,500 Rpm

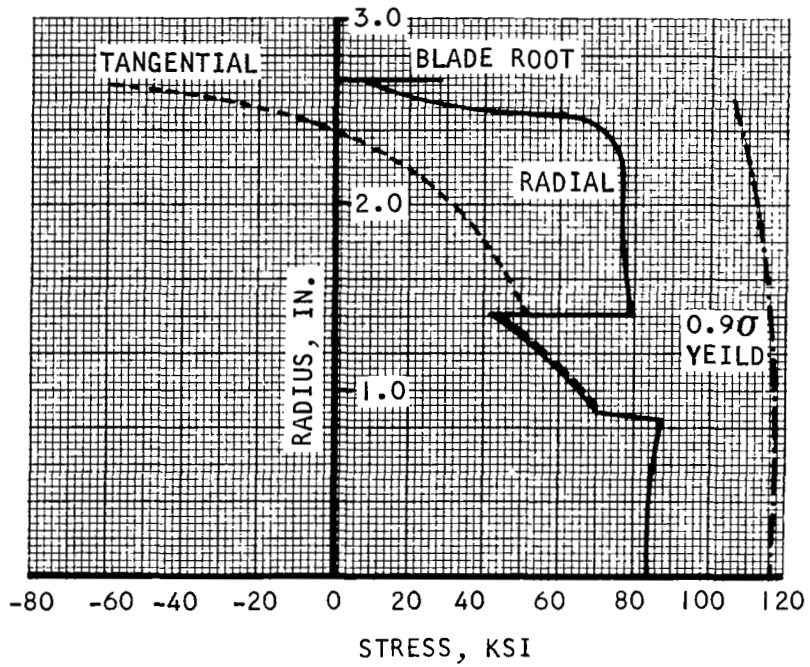


Figure 6-11. Second-Stage Transient Stresses at 10 Sec after Startup and 73,500 Rpm

#### 4. Turbine Disk Containment

Based on the multi-node thermal analysis of the rotating assembly, the turbine containment must be designed to operate at 1000 to 1300°F. The containment must protect against a tri-hub burst at 130 percent of normal speed. Waspaloy has been selected for the containment material because of its high ultimate strength and elongation capability.

The containment design will contain against a tri-hub burst of either turbine disk, as well as containing the unfractured disk. The structure is designed using an empirical equation relating the armor thickness to the disk kinetic energy (mainly translational energy). The analysis has a safety factor of 1.5. The weight penalty for containment is 7 lb (an additional 2-3 lb of containment material is available, being required for the casing).

#### 5. Rotating Assembly Critical Speeds

A multi-nodal computer analysis was used to determine the critical speeds and the shaft whirl amplitudes as a function of the speed of rotation. Figure 6-12 shows the analytical model and Figure 6-13 presents the variation in both rigid and flexural resonant frequencies as a function of the bearing mount spring rate and turbine rotational speed. These data indicate that a suspension spring rate of 15,000 lb/in. will allow turbine operation at both the 70,000 rpm normal speed and the 40,000 rpm speed as used when running from inert gas on the ground.

Figure 6-14 depicts the vibrational amplitude vs speed for the assembly and the modal shapes for a representative bearing mount spring rate. The radial deflections are based on a balancing accuracy of 0.18 gm-in. These deflections are within the design blade tip-to-casing clearance of 0.012 in.

#### 6. Rotating Assembly Bearing Operating Conditions

The thermal analyzer program (described under turbine disk steady-state and transient temperature distributions) was used to determine the hot-end bearing temperatures during both steady-state running and after shutdown when the thermal energy in the disks is dissipated during thermal soakback into the bearing carrier and other surrounding metal. The steady-state bearing temperature is primarily dependent on the lube oil temperature. In shutdown, it is dependent on the thermal energy of the disks, and the thermal capacity of the surrounding materials.

In operation, the hot-end bearing is cooled by 60 lb/hr of oil sprayed into the bearing case and by an additional 120 lb/hr of oil passed through the turbine shaft and out under the bearing inner race. In steady-state operation, the bearing is at 115°F.

After shutdown, the bearing oil flow stops and the bearing temperature increases until an equilibrium state is reached. Figure 6-15 shows the bearing temperature as a function of time after shutdown for two different conditions; one in which the APU is immediately shutdown, and the other in which

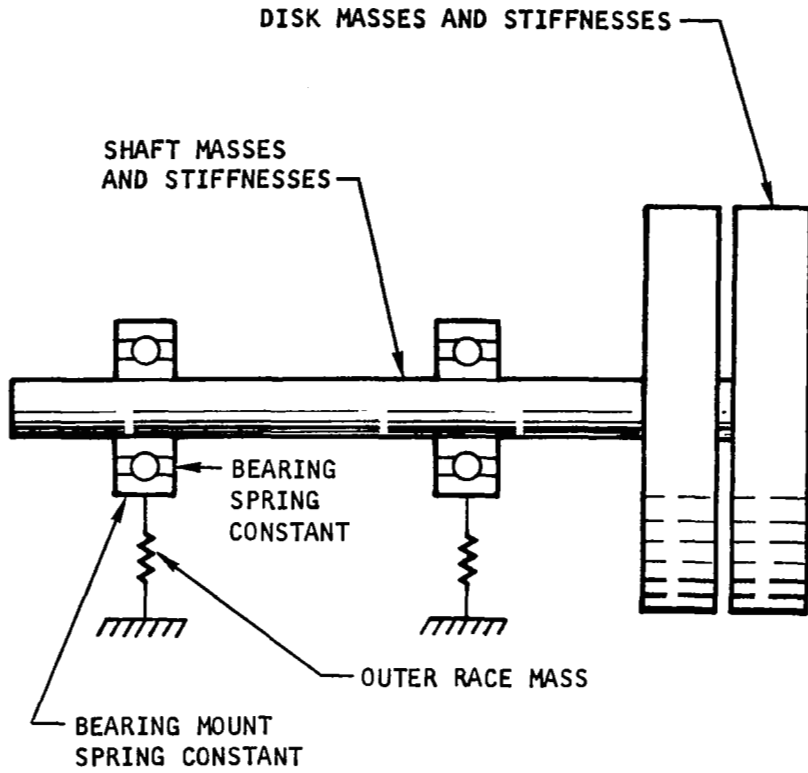


Figure 6-12. Rotating Assembly Critical Speed Analytical Model

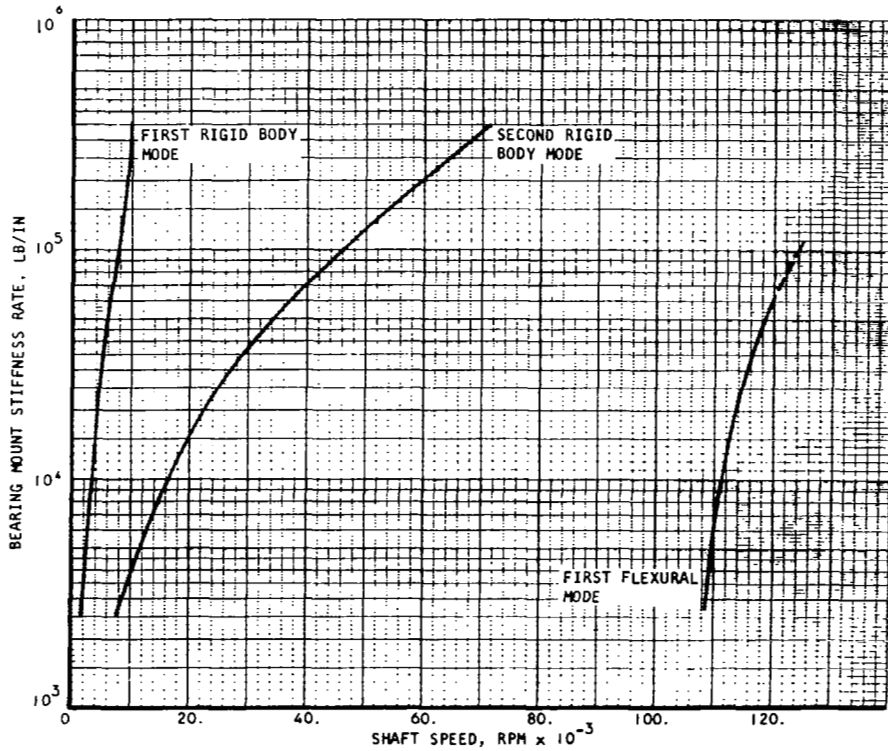
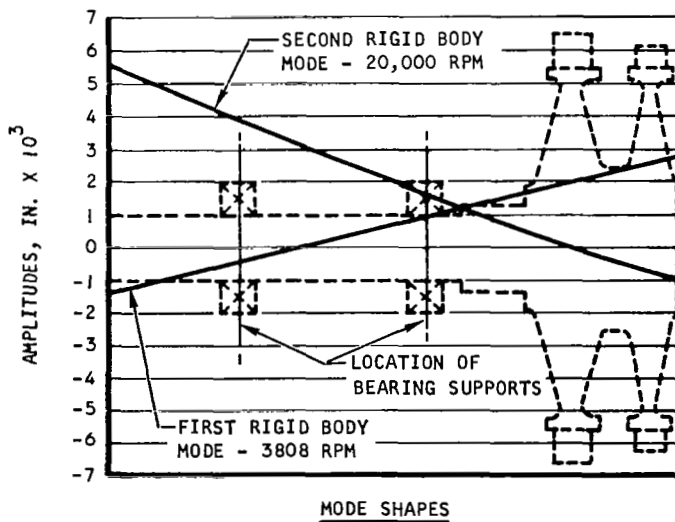
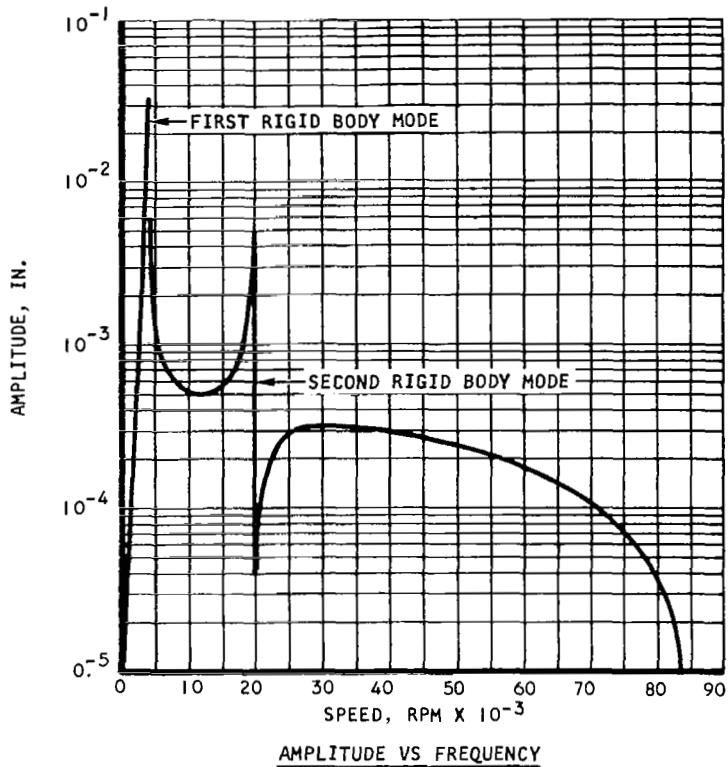


Figure 6-13. Rotating Assembly Resonant Frequencies

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- BEARING MOUNT SPRING CONSTANT = 15,000 LB/IN.
- BEARING SPRING CONSTANT = 800,000 LB/IN.

S-67283

Figure 6-14. Rotating Assembly Vibrational Amplitude and Mode Shapes



the APU operates for 3 minutes after initiation of shutdown at a reduced turbine inlet temperature ( $1600^{\circ}\text{R}$ ). Preshutdown reduction of the inlet temperature for this time interval allows the disks to equalize to the new inlet conditions; thus, reducing the temperature difference between the disks and the bearing.

Immediate shutdown without inlet temperature reduction has been selected since this allows the APU to be shutdown in emergencies without harming the bearings. This does, however, require addition of a high-capacity thermal inertia to reduce equilibrium temperature. A Resorcinol wax has been selected for this purpose. The wax has a melting point of  $230^{\circ}\text{F}$ , a specific heat of about  $0.4 \text{ Btu/lb}$  and a heat of fusion of  $83 \text{ Btu/lb}$ . The weight penalty incurred by adding the wax is about  $1 \text{ lb}$ .

It should be noted that the final bearing temperature after shutdown exceeds the NASA-specified maximum temperature of  $290^{\circ}\text{F}$  for the lube oil. However, the shutdown transient will cover only a small portion of the APU life and is confined to an extremely small amount of oil. AiResearch has operated various lubricants (notably MIL-L-23699 and MIL-7808) at temperatures up to  $350^{\circ}\text{F}$  for extended periods of time.

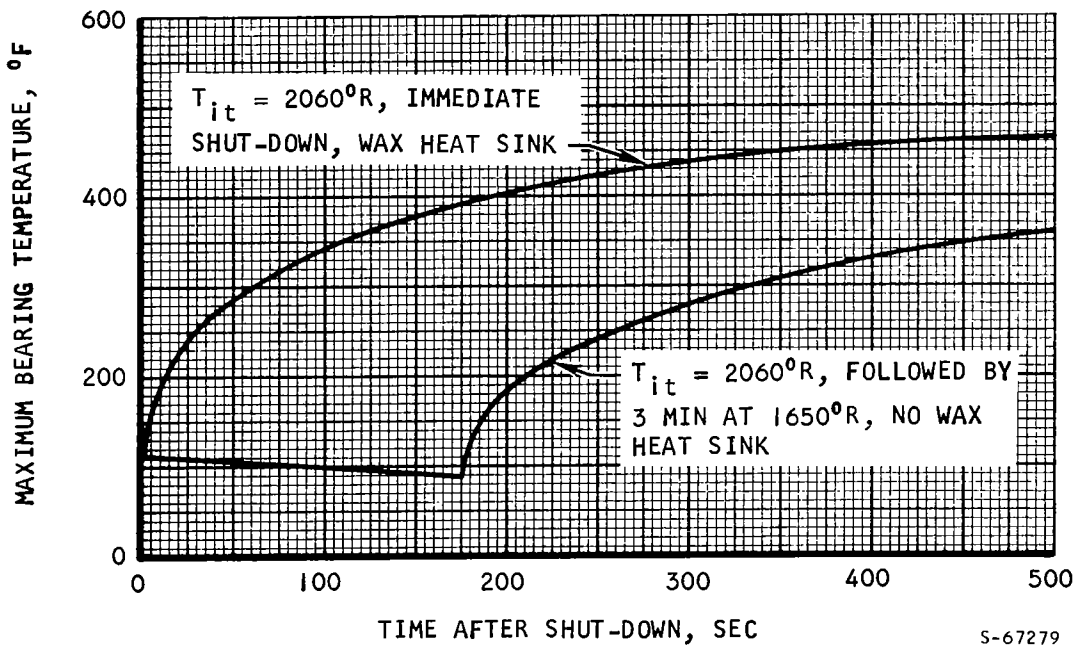


Figure 6-15. Hot-End Bearing Temperature during Shutdown Heat Soakback

## TURBINE AERODYNAMIC DESIGN

### Design Requirements

The turbine is required to operate with a high-energy working fluid (with adiabatic heads in the range from 1.5 to  $2.5 \times 10^6$  ft-lb per lb) and varying pressure ratio (which depends upon output power and ambient pressure, as shown in Figure 6-16). Although the turbine can conceivably operate over a wide range of pressure ratio (from 6 to approximately 70), not all of this range is of equal interest. Figure 6-17, for example, shows the range of pressure ratio for the NASA Booster mission profile. It can be seen from this that operation at moderate pressure ratios (in the range from 7 to 13) will be particularly important. In addition, any turbine design is constrained by the requirement for delivery of full power output (400 hp at the gearbox) at sea-level ambient pressure with a maximum turbine inlet pressure of 450 psia.

### Aerodynamic Design Analytical Procedure

Several computer programs were used to assist in turbine design and evaluation of performance. First, the design program (AiResearch A-1205) was used to establish turbine design parameters such as nozzle area ratios, arc of admission, and blade height for selected input parameters such as rotational speed, number of stages, rotor diameter, inlet pressure, discharge pressure, working fluid properties, chord width, and tip clearance. Then, the off-design-point performance analysis program (AiResearch TMAP) was used to establish turbine performance maps which give turbine efficiency as a function of output power and turbine discharge pressure (turbine inlet pressure was contained in the output power parameter). Finally, the turbine performance maps were fed into the cycle steady state performance program together with other component off-design performance data to establish turbine performance at the system level. It was necessary to perform much of the turbine design analysis at the system level to (1) insure performance compatibility over the entire envelope of operating conditions and (2) obtain proper optimization of internal design parameters (such as pressure ratio split between stages, pitch line velocity, turbine design point pressure ratio and power level, etc.).

### General Considerations

The general design requirements listed above have led to selection of a two-stage pressure-compounded axial-flow impulse turbine design. Both stages are partial admission. Integrated mission propellant requirement studies have indicated that minimum propellant consumption is obtained by designing the turbine nozzle at an altitude, part power condition. The blading is designed for optimum performance at the high overall pressure ratios and velocities obtained at the maximum power point. This concept results in a turbine having a high efficiency over a wide range of operating conditions and utilizes well established principles as will be discussed.

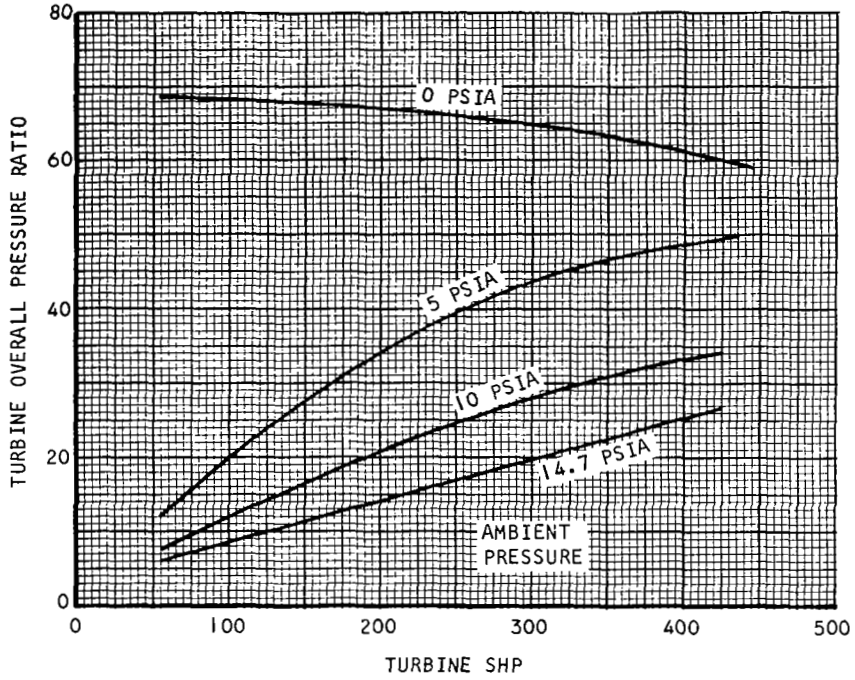


Figure 6-16. APU Turbine Pressure Ratio Envelope

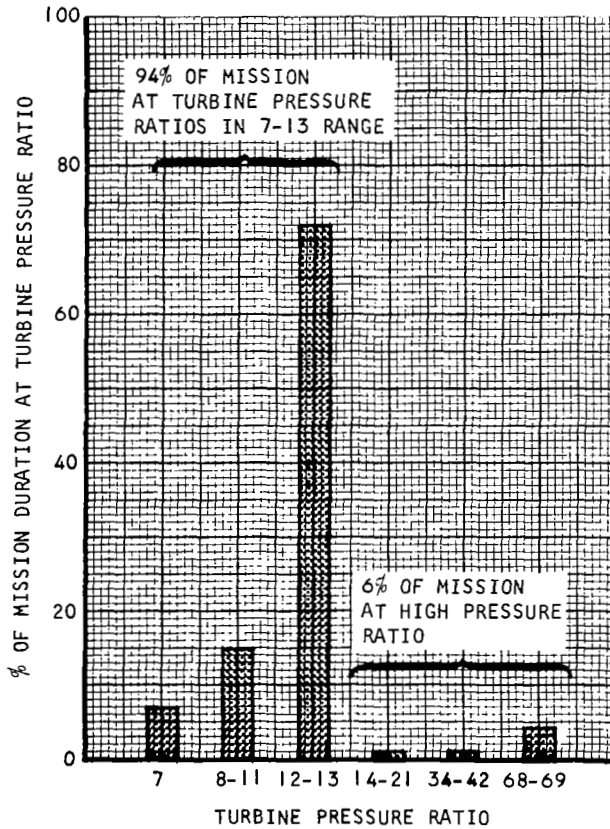


Figure 6-17. Turbine Pressure Ratio Operating Time for Typical Space Shuttle Vehicle Mission

S-70505

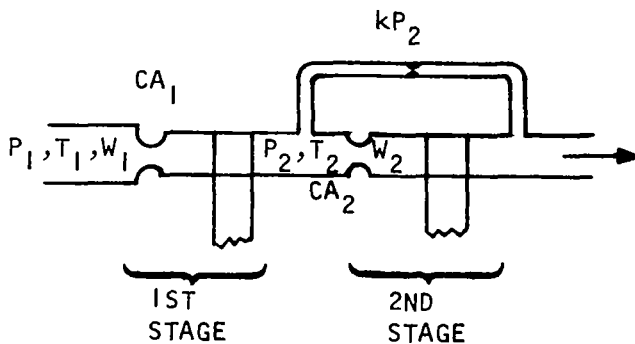
The problem of designing a turbine to operate over a range of inlet Mach number from subsonic to supersonic conditions is not new. Partial-admission supersonic stages are conventionally used in steam turbines which show high turbine efficiency over a wide range of inlet Mach number. Aircraft accessory turbines frequently operate at high pressure ratios corresponding to supersonic conditions. Considerable data are available from this source showing high performance over a wide range of inlet Mach number. Another important area requiring supersonic turbine design has been missile APU's using liquid monopropellants (such as hydrazine or ethylene oxide) and solid propellants (principally low-temperature ammonium nitrate composite type solid propellants). Again, test data are available showing good performance over a wide range of inlet Mach number.

Since the expansion (and hence exit plane velocity) across the turbine nozzle is fixed by the geometry, the question arises concerning the mechanism by which wide-range efficient performance is obtained. Clearly what occurs between the turbine inlet and the nozzle outlet (at the discharge area plane) is important. The experimental data are reasonably well correlated by the assumption of a free-stream expansion between the nozzle outlet and the rotor blades. Sufficient area is available for this expansion to occur as a result of the clearances required for installation of the nozzle geometry and for mechanical separation of the rotating and stationary parts. Increasing the expansion area has not resulted in any significant improvement in performance for the range of inlet Mach number of interest here.

### Turbine Nozzle Design

It was established in the Phase I studies and confirmed during Phase II that optimum turbine performance is obtained with unequal split in pressure ratio between the two stages. The first stage is designed for a higher pressure ratio which, as will be discussed, remains constant with varying overall pressure ratio.

#### 1. First-Stage Nozzle



The flow through the first-stage nozzle is given by:

$$W_1 \sqrt{T_1} = P_1 CA_1$$

and the flow through the second-stage nozzle (for a supercritical pressure ratio) is given by:

$$W_2 \sqrt{T_2} = P_2 CA_2$$

The difference between the first and second stage flows,  $W_1 - W_2$ , is represented by a leakage flow which is proportional to interstage pressure,

$$W_1 - W_2 = kP_2$$

By combining the three previous equations, the following expression for first-stage pressure ratio can be obtained:

$$\frac{P_1}{P_2} = \frac{k + \frac{CA_2}{\sqrt{T_2}}}{\frac{CA_1}{\sqrt{T_1}}}$$

It can, therefore, be seen that for constant  $CA_1$ ,  $CA_2$ ,  $T_1$ , and  $T_2$ ,

$$\frac{P_1}{P_2} = \text{constant}$$

Therefore, since the first stage pressure ratio remains constant (varying slightly with O/F ratio because of its effects on specific heat ratio), the first stage will operate at constant conditions and constant specific work output from the working fluid (Figure 6-18) with constant inlet Mach number (Figure 6-19).

Consequently, variations in overall pressure ratio will be reflected in variations in the second-stage pressure ratio and efficiency. As indicated by Figure 6-20, it is expected that, with increasing pressure ratio, the turbine will be able to convert a portion of the increased adiabatic head into useful output work. In other words, reductions in turbine efficiency resulting from off-design operation will be more than offset by increased head, and the product of turbine efficiency and adiabatic head will increase with pressure ratio over the range of primary interest.

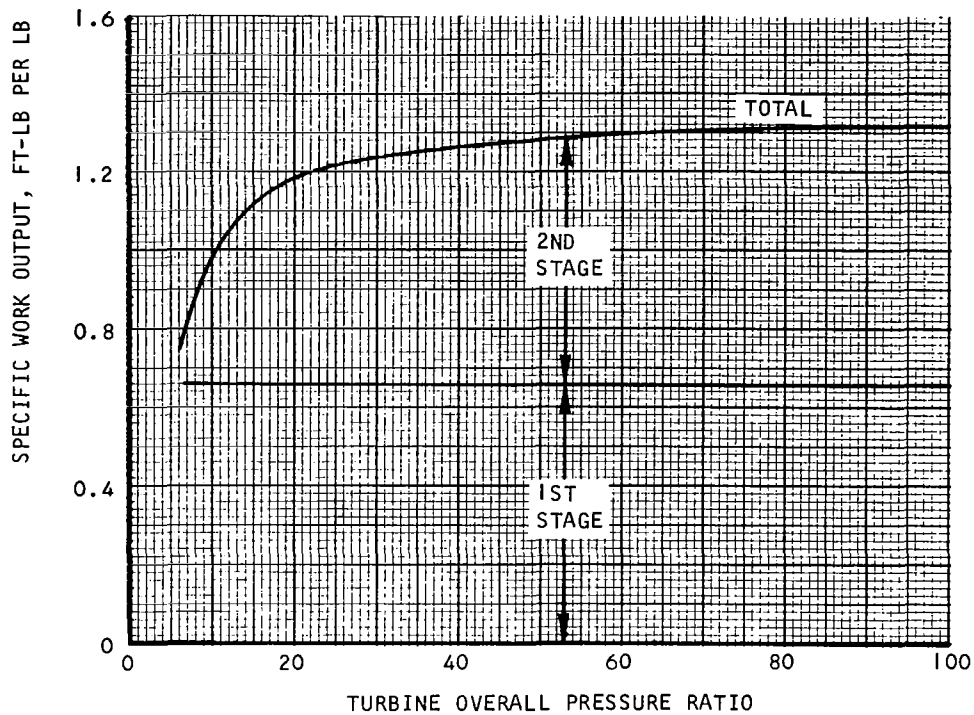


Figure 6-18. Specific Work Output

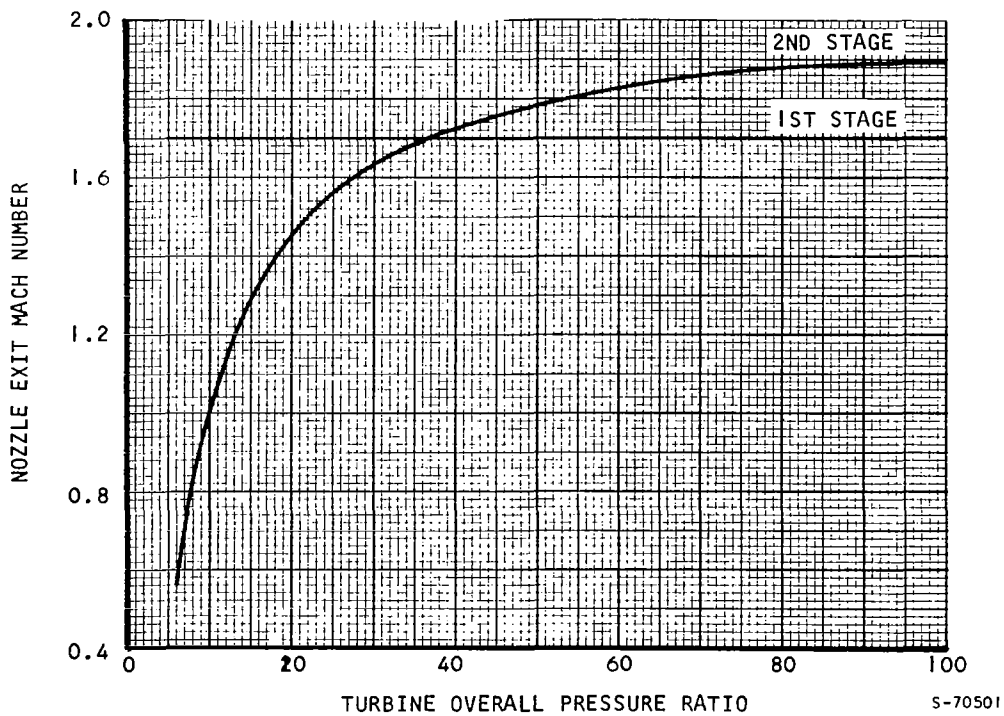


Figure 6-19. Rotor Inlet Absolute Mach Number

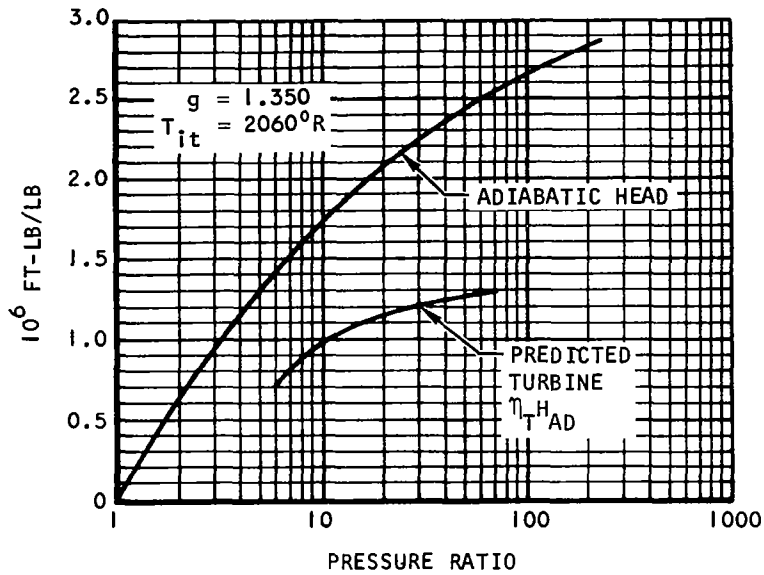


Figure 6-20. Conversion of Adiabatic Head Into Useful Work Output

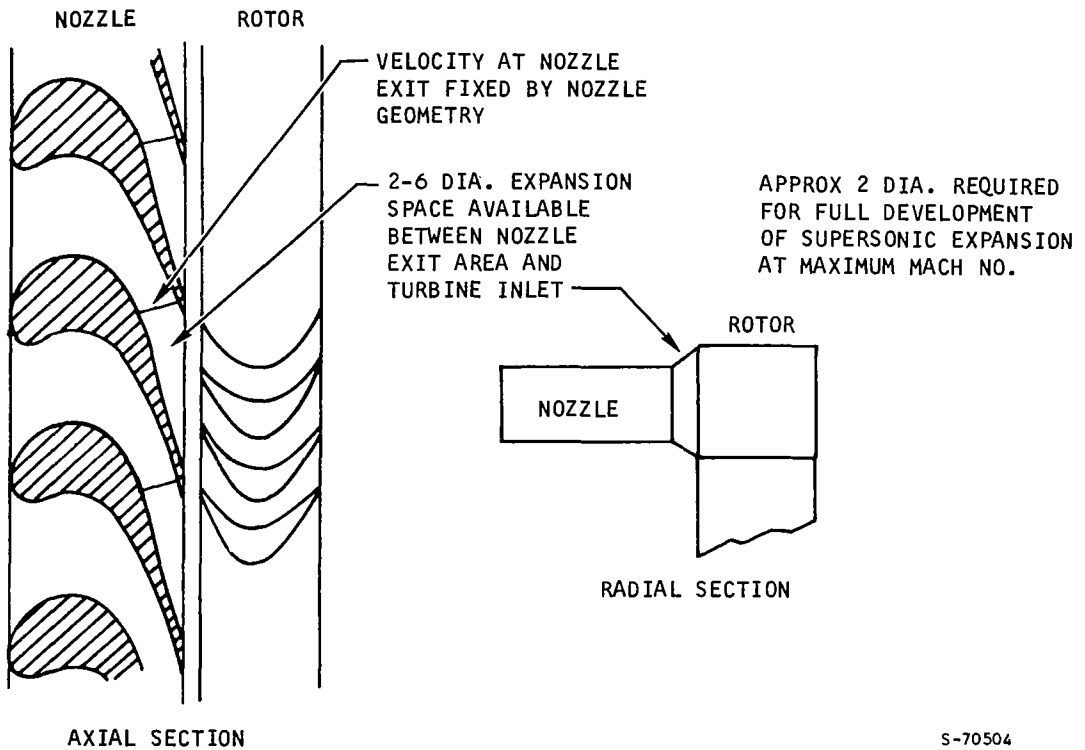


Figure 6-21. Space Available For After-Expansion in Second Stage

## 2. Second-Stage Nozzle

It has been well established that nozzle discharge gas velocity increases with pressure ratio beyond the design point for both converging and supersonic nozzles by expansion outside the nozzle. This type of behavior is also experienced with turbine nozzles where there is no choking or high shock losses in the rotor blading to limit nozzle pressure ratio. (Incidentally, downstream choking is sometimes intentionally used to provide a type of inherent speed control.) Assuming careful design of the rotor blading to minimize flow losses, the after-expansion leads to increased useful work output with increasing pressure ratio beyond the design point. As mentioned previously, test data are correlated by the assumption of free-stream expansion between the nozzle outlet and the rotor inlet. This will be illustrated subsequently in this section. This expansion takes place in the wedge-shaped segment between the nozzle exit plane and the rotor inlet plane (Figure 2-21). The radial expansion provided by the blade inlet area overlapping the nozzle discharge area is important to off-design performance of the second stage.

### Turbine Design Parameters

Table 6-4 summarizes the design characteristics of the turbine design in terms of the major geometrical parameters for the two stages. Use of identical symmetrical rotor blading is assumed and reaction is neglected. It is probably possible to improve performance through use of nonsymmetrical blading and different blading design for the two stages. Detailed evaluation of flow distribution and reaction effects may also be important to attainment of maximum performance. These factors should be considered during the detailed design phase of APU development.

TABLE 6-4

TURBINE DESIGN PARAMETERS

	First Stage	Second Stage
Nozzle effective throat area, sq in.	0.1170	0.4270
Nozzle exit area, sq in.	0.1800	0.4700
Nozzle type	Axisymmetric	Two-dimensional
Number of nozzles	4	16
Percent admission	21.35	55.00
Blade height, in.	0.2650	0.3300
Axial chord length, in.	0.3500	0.3500
Number of blades	85	85
Pitch diameter, in.	5.566	5.631
Nozzle exit angle, degrees	16	16
Blade inlet angle, degrees	23	23
Blade exit angle, degrees	23	23



## Rotor Blading

Figure 6-19 shows that the first stage rotor operates with a constant inlet absolute Mach number of 1.7 and the second stage operates with a variable inlet absolute Mach number ranging up to 1.85 at the maximum pressure ratio (this corresponds to a maximum inlet relative Mach number of approximately 1.60, which serves as the design point for the rotor blading). The impulse blading design for higher Mach numbers is different from that optimally used for low or transonic Mach numbers. First, small blade angles (22-23 degrees) are used with supersonic turbines (as compared with 30 degree blades for typical transonic turbine designs). Second, to minimize shock losses, the buckets have sharp inlet edges, close spacing between blades, and flow channel curvature designed for cancellation of the shocks while turning the flow. Finally, the blades extend above and below the nozzle exit area to allow after-expansion in the radial direction.

## Stage Performance

Table 6-5 lists the performance of the two turbine stages for the following three typical operating points (which correspond to the referenced case numbers for overall system performance in Volume V):

Space (0 psia ambient) zero net output power: Case 1

Mode power (100 hp net output power, 10 psia ambient pressure):  
Case 10

Sea level maximum output power: Case 16

The varying O/F ratios shown for these three conditions result for cycle energy balances for the system. Turbine performance variations due to differences in working fluid molecular weight and specific heat ratio (vs a result of O/F ratio variations) are taken into account in the analytical procedures used here for turbine performance prediction.

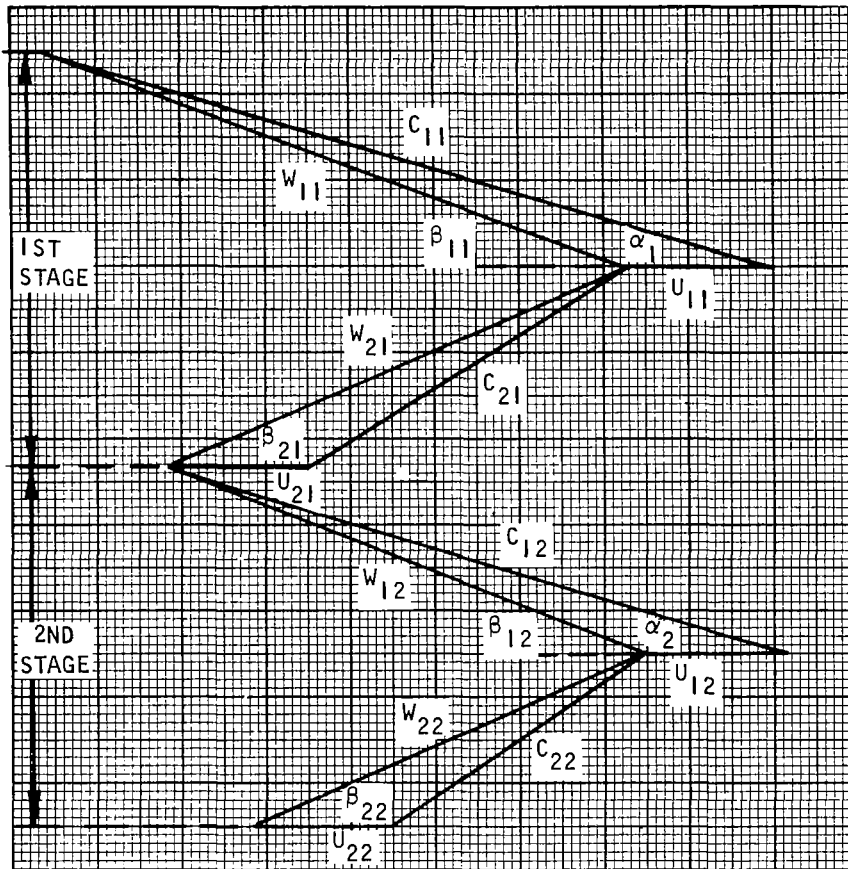
It will be noted that the product of the stage pressure ratio exceeds the overall pressure ratio for the turbine. This is due to the interstage pressure recovery in partial conversion of the kinetic head from the first stage into static pressure rise.

## Velocity Triangles

Figure 6-22 gives the velocity triangles for the three cases listed previously. The differences shown for the first stage are due to variations in thermodynamic properties resulting from the O/F ratio required to satisfy cycle energy balances.

TABLE 6-5  
TYPICAL STAGE PERFORMANCE

	Space, Zero Power (Case 1)	Mode Power (Case 10)	Sea-Level, Maximum Power (Case 16)
<b>Overall performance</b>			
$\dot{w}$ (lb/sec)	0.0229	0.0794	0.2049
PR	71.14	16.81	26.94
Output SHP	51.3	156.3	423.3
Efficiency	0.506	0.567	0.557
O/F	0.649	0.638	0.696
<b>First stage</b>			
$H_{AD}$ ( $10^6$ ft)	1.352	1.363	1.312
$U/C_o$	0.1824	0.1816	0.1852
Efficiency	0.4562	0.4558	0.4626
$P_{1T}$ (psia)	49.80	173.1	439.1
$P_{2S}$ (psia)	8.44	29.3	75.0
<b>Second stage</b>			
$H_{AD}$ ( $10^6$ ft)	1.508	0.832	1.051
$U/C_o$	0.1747	0.2352	0.2093
Efficiency	0.4158	0.5659	0.5147
$P_{3T}$ (psia)	10.08	35.00	89.2
$P_{4S}$ (psia)	0.700	10.30	16.30



$C$  = ABSOLUTE VELOCITY  
 $W$  = RELATIVE VELOCITY  
 $U$  = ROTOR PITCH LINE VELOCITY  
 $\alpha$  = NOZZLE ANGLE  
 $\beta$  = RELATIVE VELOCITY  
 TANGENTIAL VECTOR ANGLE

OUTPUT POWER 0      MODE      MAX  
 AMB. PRESS. 0      10      14.7  
 CASE NO. 1      10      16

## FIRST STAGE

$C_{11}$	8950	8988	8818
$W_{11}$	7330	7367	7197
$U_{11}$	1701	1701	1701
$\beta_{11}$	19.7	19.7	19.7
$\alpha_1$	16.0	16.0	16.0
$C_{21}$	4452	4481	4349
$W_{21}$	5968	5998	5864
$U_{21}$	1701	1701	1701
$\beta_{11}$	23.0	23.0	23.0

## SECOND STAGE

$C_{12}$	8710	6987	7739
$W_{12}$	7071	5354	6103
$U_{12}$	1721	1721	1721
$\beta_{12}$	19.8	21.1	20.5
$\alpha_2$	16.0	16.0	16.0
$C_{22}$	4288	2984	3574
$W_{22}$	5819	4500	5095
$U_{22}$	1721	1721	1721
$\beta_{22}$	23.0	23.0	23.0

Figure 6-22. Typical Turbine Velocity Triangles

## Thermodynamic Paths

Figure 6-23 depicts the thermodynamic paths obtained in the two turbine stages for the three operating conditions given previously. As indicated previously, first-stage performance remains essentially constant. As indicated by the increased entropy in expansion, the second-stage is less efficient at higher pressure ratios.

## Reduced Backpressure

Figure 6-24 gives overall and stage performance and a partial breakdown of the losses as functions of the turbine discharge pressure (for constant working fluid throughflow = 0.2049 lb per sec). As before, first-stage performance remains constant, second-stage incidence and nozzle losses increase with decreasing backpressure. Blade disk friction and scavenging losses decrease with backpressure. The decreased turbine efficiency is more than offset by the increased adiabatic heat and increased turbine output power is obtained with reduced backpressure.

## Overall Performance

Figures 6-25 and 6-26 show turbine efficiency as functions of velocity ratio and overall pressure ratio. Figure 6-27 shows the turbine performance map which was input to the cycle analysis program to determine the system performance data given in Volume V.

## Substantiation of Performance Predictions

### 1. Nozzle Performance Correlation

Because of the importance of predicting off-design-point performance of the second-stage nozzles, a check was made of the performance prediction given by the TMAP (turbine off-design analysis) program. Figure 6-28 shows satisfactory agreement with published nozzle data.

### 2. Turbine Performance Correlation

To substantiate the performance predictions established for the Space Shuttle APU, the TMAP program was used to assess the performance of a Zeus turbine wheel\* for which detailed turbine design and dynamometer test data were available. Figure 6-29 shows the results of the performance correlation at two off-design pressure ratios corresponding to overexpansion and underexpansion (design pressure ratio = 10). The Zeus turbine is a partial-admission, single-stage supersonic, axial-flow impulse design similar to that selected for the Space Shuttle APU turbine stages.

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\* Originally developed at AiResearch for a propellant APU for the Nike-Zeus missile.

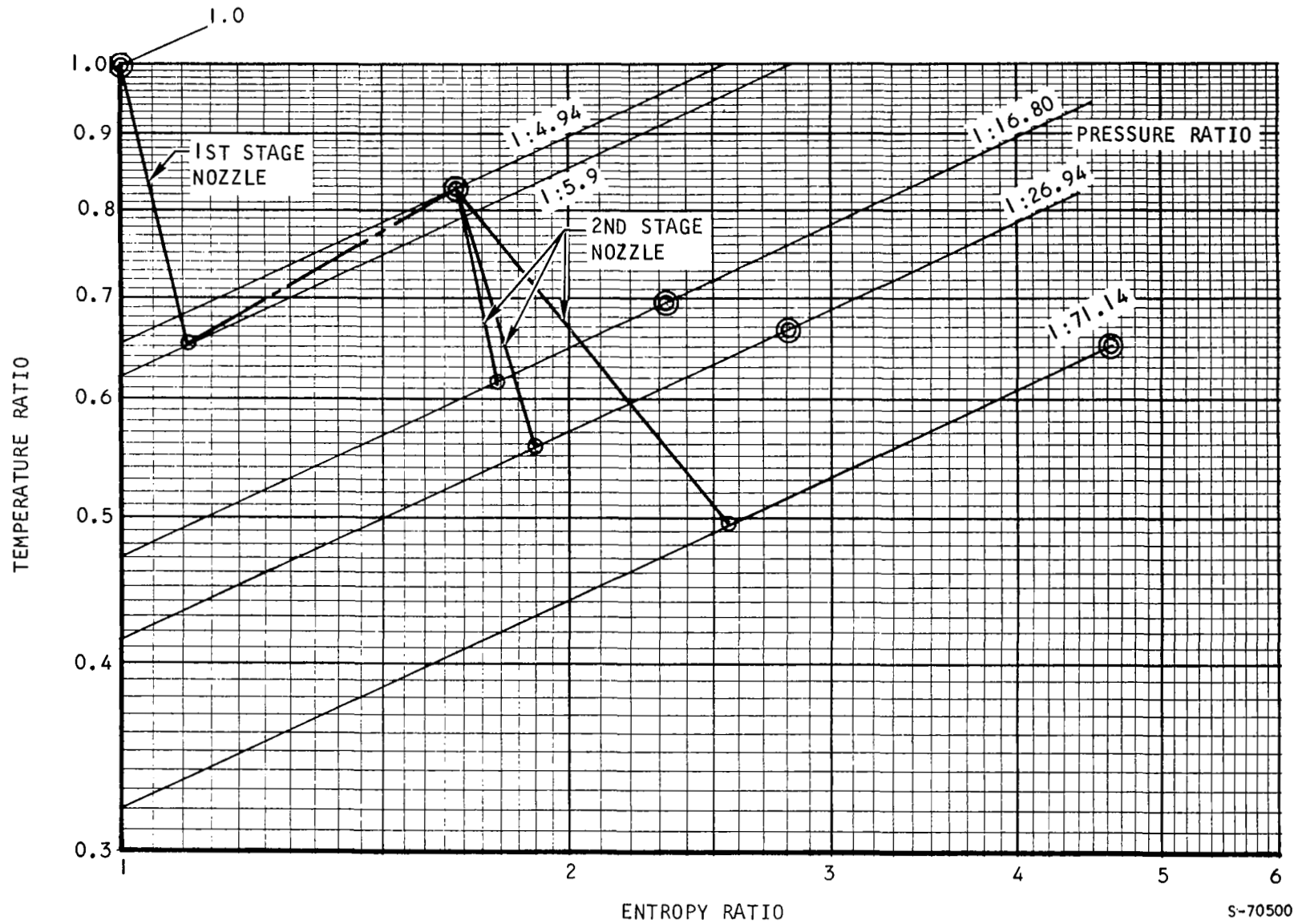


Figure 6-23. Temperature-Entropy Relationships

TURBINE PERFORMANCE  
 NOZZLE INLET 439.1 PSIA - 2060°R  
 ROTATIONAL SPEED 70,000 RPM  
 MIXTURE RATIO O/F = 0.696  
 WEIGHT FLOW = 0.2049 LB/SEC

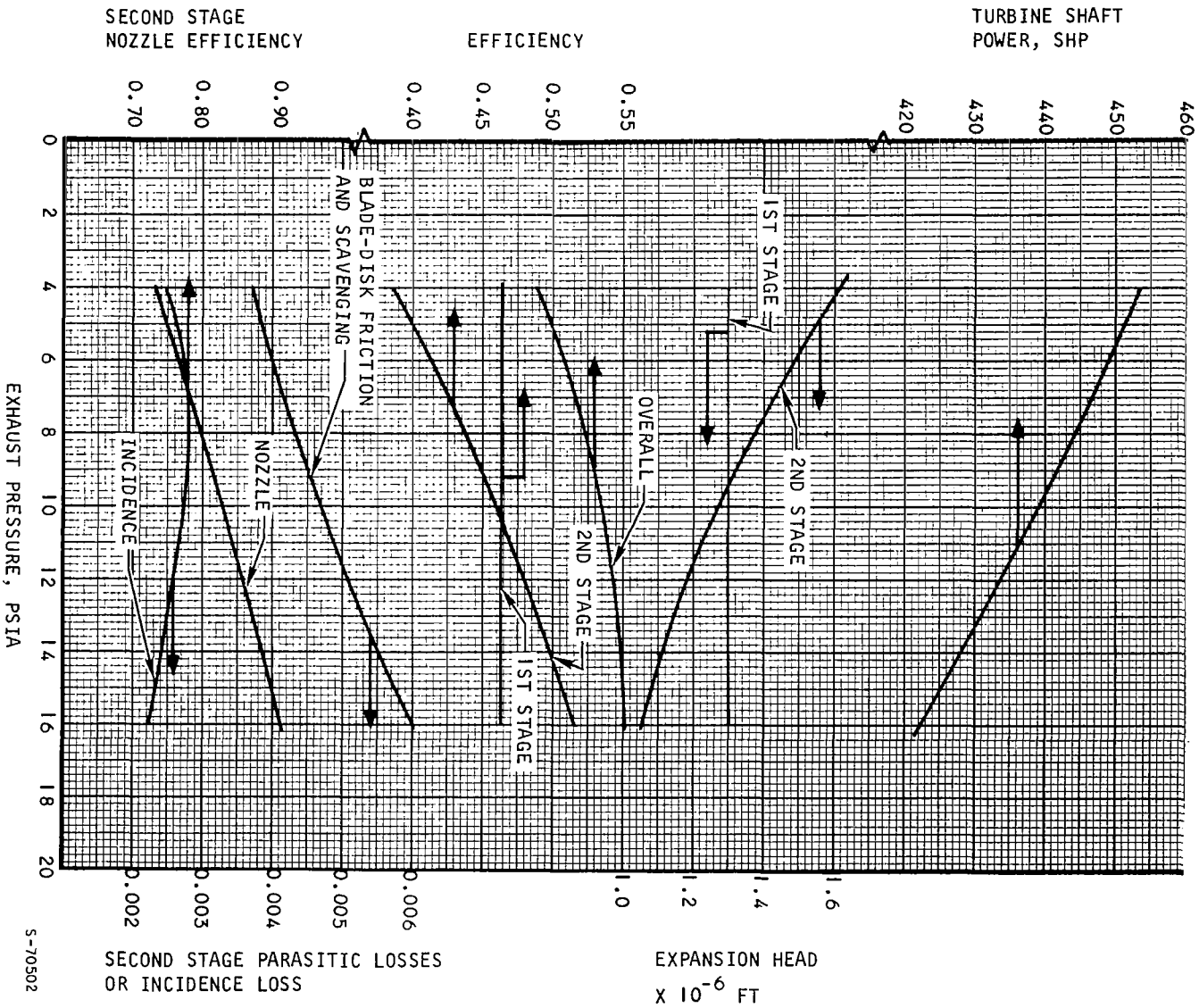


Figure 6-24. Turbine Overall and Stage Performance

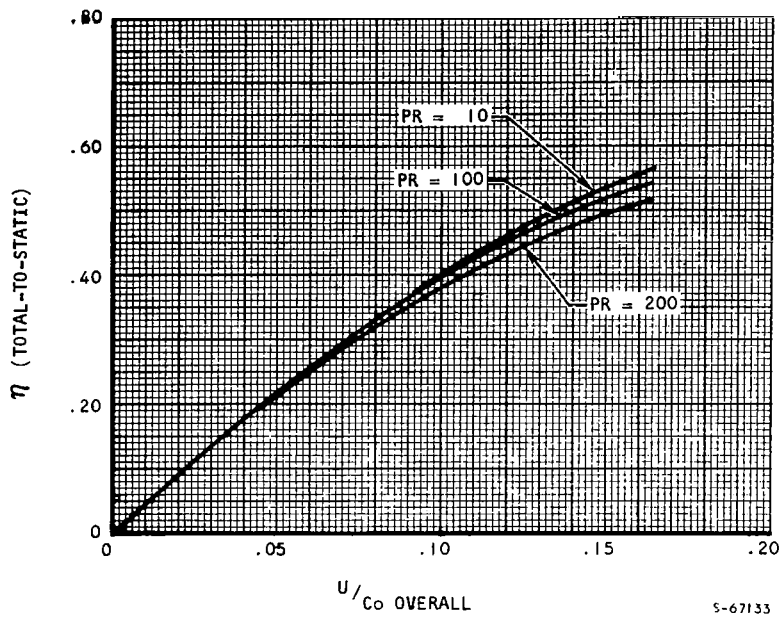


Figure 6-25. Turbine Efficiency vs Velocity Ratio

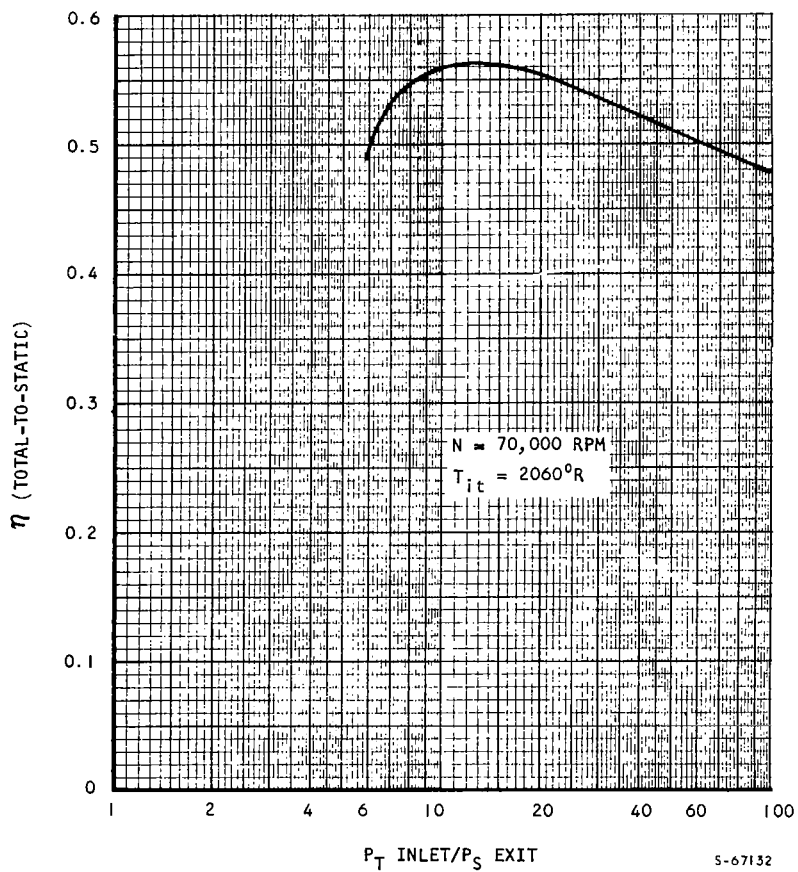


Figure 6-26. Turbine Efficiency vs Pressure Ratio at Operating Speed

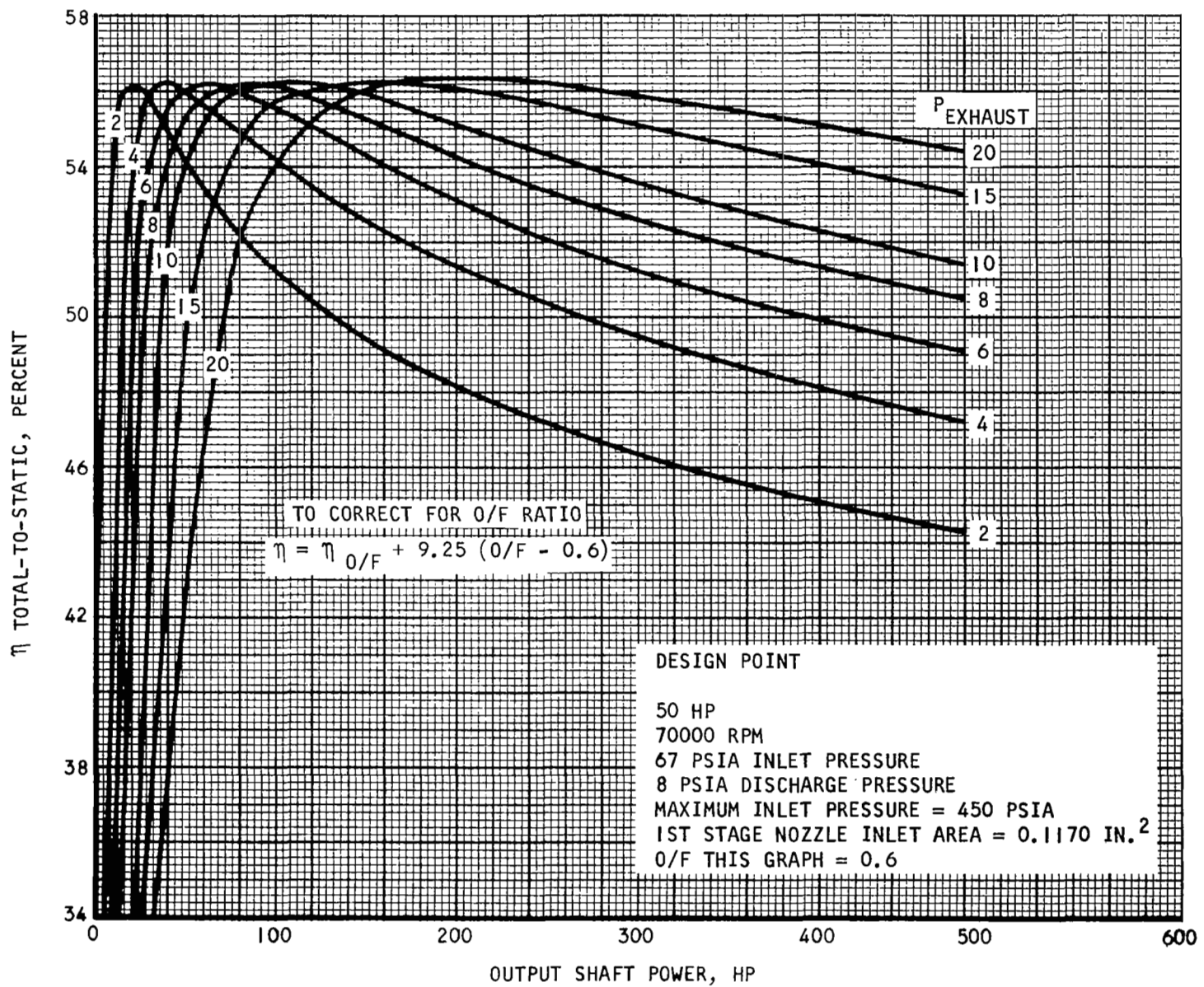


Figure 6-27. Turbine Efficiency vs Developed Power



KRAFT "REACTION TEST OF TURBINES FOR SUBSONIC VELOCITIES."  $\square M_D = 1.0$

KEENAN "REACTION TEST OF TURBINE NOZZLES FOR SUPERSONIC VELOCITIES."

$\circ M_D = 1.0$   $\diamond M_D = 1.6$   $\triangle H_D = 2.2$   $\nabla H_D = 2.3$

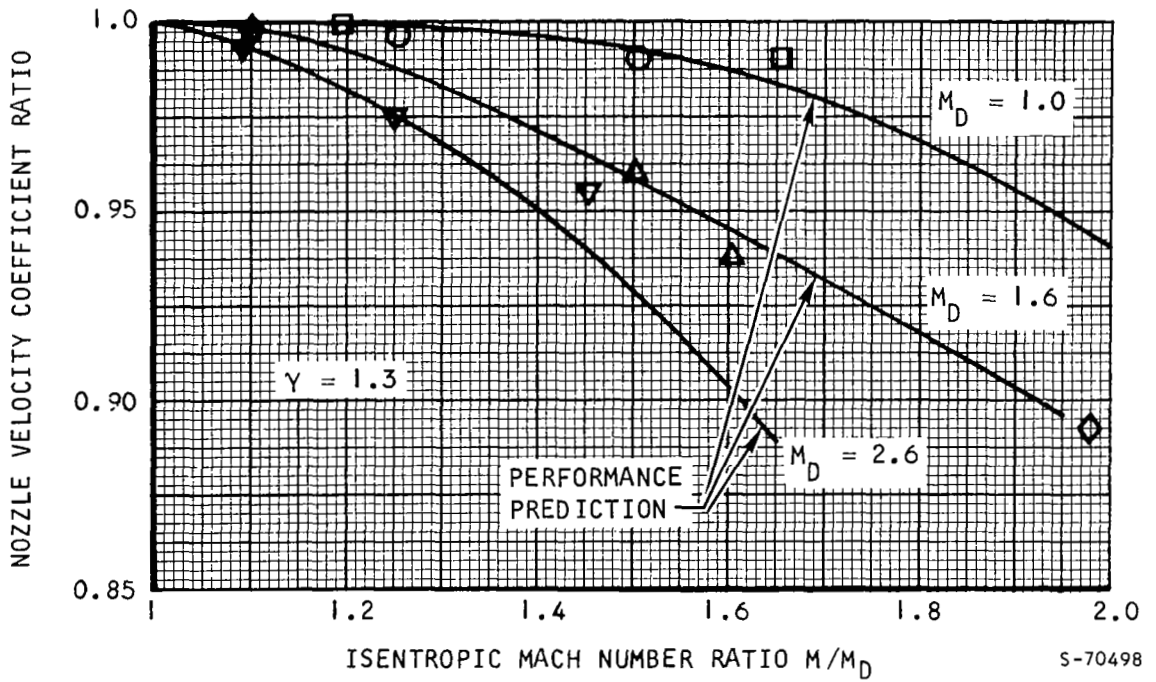


Figure 6-28. Correlation of Nozzle Predictions

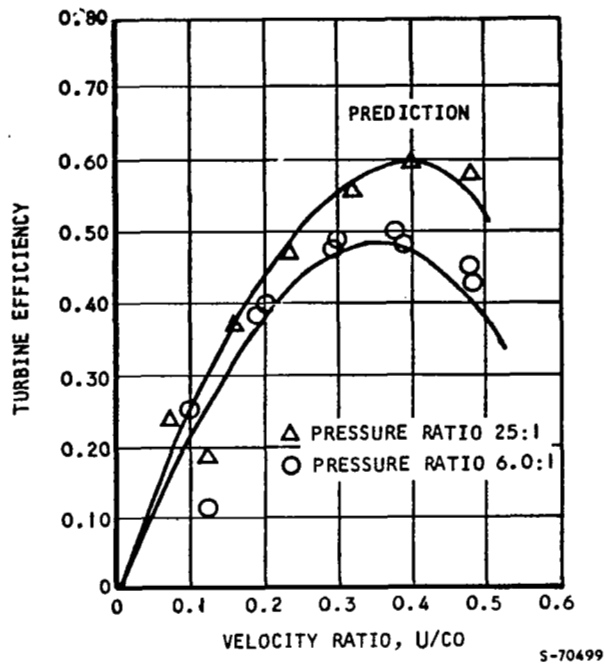


Figure 6-29. Zeus Turbine Off-Design Computer Program Performance Correlation

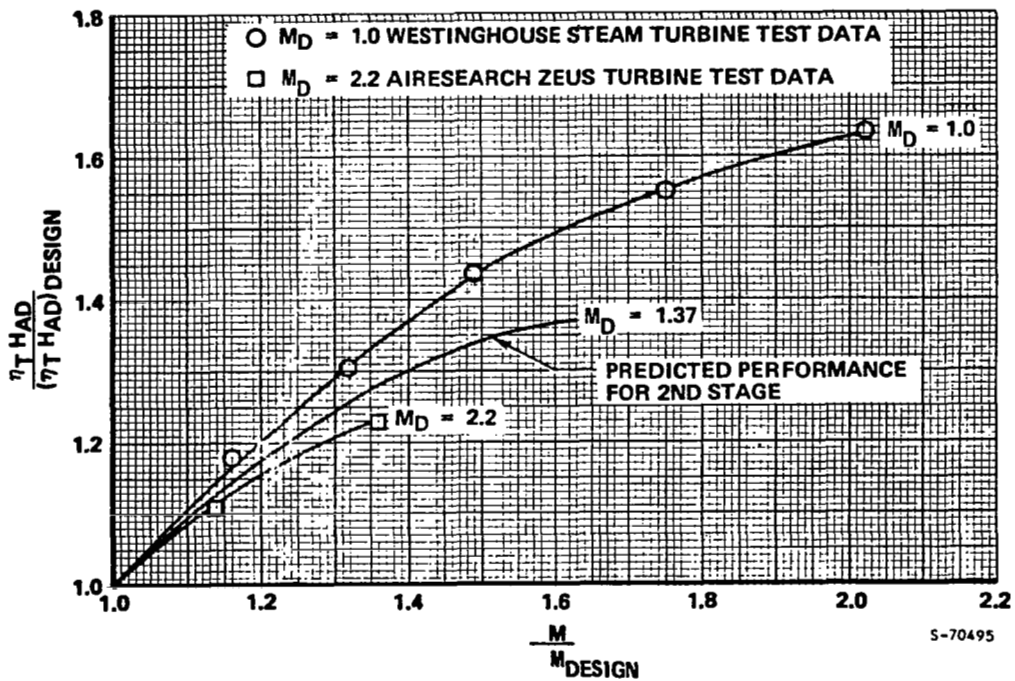


Figure 6-30. Second Stage Performance Correlation

### 3. Efficiency-Head Product Correlation

As further verification of the increase in usable specific work output with pressure ratio, a correlation was made of the normalized  $\eta_T H_{AD}$  and isentropic Mach number parameters for the second-stage. This correlation depends upon the design Mach number. As shown by Figure 6-30, the predicted performance for the second-stage falls between test data for two different design Mach numbers. This correlation appears to establish reasonable confidence in the validity of the analytical techniques used to predict off-design performance.

#### GEARBOX

The other major component of the turbine power unit, in addition to the rotating assembly, is the gearbox. The gearbox provides the following functions:

- Output pad for electrical alternator
- Output pad for hydraulic pumps
- Means of circulating oil through the gearbox and the rotating assembly
- Input pad for the rotating assembly
- Structural support for the entire turbine power unit

It should be noted that the design of the hydraulic pump and the alternator is outside the scope of this study contract. However, it was necessary to briefly consider these units when designing the gearbox. For this study, the units considered were 99-120 gpm Abex Ap27V hydraulic pump units while the alternator (generator) was a 60-75 KVA Westinghouse 946F501-1. These components are representative designs that are compatible with the basic requirements of the APU problem statement.

#### Gearbox Configuration

The gearbox shown in SK39908 is a straight spur gear configuration with a single mesh reduction to the alternator and a double mesh reduction to each of the hydraulic pumps. This arrangement results in the minimum number of gears and a lightweight, close coupled, rigid housing assembly. Gear sizes and proportions have been computer designed and are based on a 20 degree tooth pressure angle. The tooth loading on the high speed pinion and the pinion bearing load are the most critical parts of the entire assembly. AiResearch experience in similar designs indicate that conservative design values should be used in such cases particularly where the pinion is driving a pulsating load such as a multi-cylinder, high pressure pump. Accordingly, the pinion tooth Hertz stresses have been limited to 150 ksi.

Input drive to the gearbox is provided by a quill shaft connecting the turbine shaft and gearbox input shaft. Use of a quill shaft permits considerable freedom in the choice of the bearing mount flexibility and also minimizes the effects of alignment variances and torque pulsations.

The gearbox housing consists of a lightweight assembly of two halves, each of cast, ribbed construction. Housing material is 355 aluminum alloy. The requirements for lightweight, high rigidity, and minimum internal free volume for the oil are satisfied by this design.

### Lubrication and Cooling

The design of the oil lubrication and cooling system is based on a proven concept previously demonstrated with a working model. Each of the gears operates as a drag pump, scavenging the oil-vapor mixture and pumping it through glands in the housing to the central portion of the alternator drive gear. The oil is then centrifuged to the cup shaped rim where the peripheral speed is about 400 ft/sec. This speed creates the necessary pressure head to dual oil scoops where the oil is picked up for distribution. The oil circuit resistances are carefully designed so that the oil will be properly distributed. A total flow of 6 gpm at 200 psid is provided, half of this flow acting as coolant for the alternator. The alternator outlet cavities as shown require minor modifications to provide a positive oil scavenging back to the gearbox but this is not a problem of consequence.

### Alternate Gearbox

Both spur and planetary gearing were considered for this application. However, the speed reductions are such as to allow driving the alternator with only a single stage of spur gearing. Thus, spur gears show a decided weight advantage over planetary gearing for the turbine speed of 70,000 rpm. An alternate gearbox incorporating planetary gearing is shown by drawing SK 39906. This planetary gearbox is approximately 25 percent heavier than its spur gear counterpart and is presented to show an alternative arrangement which might have advantage in packaging. It is frequently necessary for packaging reasons to select gearbox configurations which are not optimum from the standpoint of minimum weight.

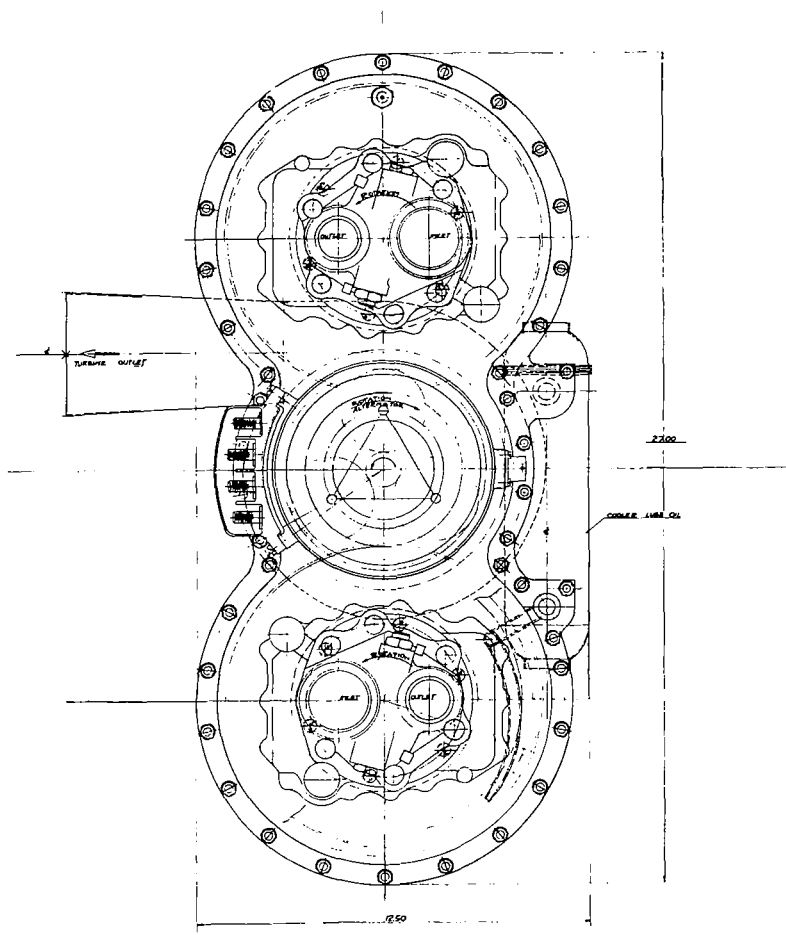
## DESIGN BASES

The paragraphs following summarize the state-of-the-art experience, which, in part, led to selection of the combustor assembly configuration described previously in this section. Since a major objective of the program was to establish a system configuration with acceptable development risk, it is important to relate the proposed concepts to the present technology base. In summary, it can be stated that previously-described combustor-control valve concept is attainable at low development risk.

### Hydrogen-Oxygen Combustor Technology

#### I. IPECS Combustor

Ten years ago, as a part of a company-funded program for development of an Integrated Power Environmental Control System (IPECS) concept, AiResearch built and tested a gas generator design similar to that proposed here. The unique features of this design were as follows:



5K-39706



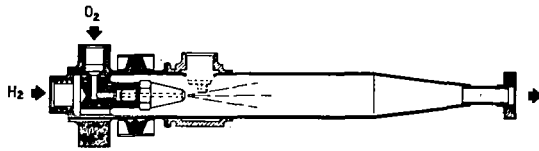
- (a) Very low pressure drop in the hydrogen flow, conserving the available head for driving a turbine and minimizing the hydrogen supply pressure requirement.
- (b) Diffusion flame design, with combustor walls cooled by the inlet hydrogen flow.

Table 6-6 lists the design characteristics of the test combustor. Figure 6-31 shows hardware and general performance characteristics which met all of the design requirements for the system. Successful integrated system tests (with active O/F control) were performed with no problems with combustor stability or efficiency over a reasonably wide range of pressure and O/F ratio.

TABLE 6-6

IPECS COMBUSTOR DESIGN FEATURES

<u>Hydrogen Inlet</u>	
Injector hole diameter, in.	0.189
Number of holes	8
Total area, in. <sup>2</sup>	0.224
Nominal pressure drop, psi	Chamber pressure/40
<u>Oxygen Inlet</u>	
Injector hole diameter, in.	0.0645
Number of holes	8
Total area, in. <sup>2</sup>	0.0262
Nominal pressure drop, psi	Chamber pressure/8
<u>Chamber</u>	
Inside diameter, in.	1.90
Length (to turbine scroll), in.	4.1
Volume (including turbine scroll), in. <sup>3</sup>	25.7
Turbine nozzle area, in. <sup>2</sup>	0.112
Chamber L <sup>†</sup> , in.	230



HYDROGEN	OXYGEN
0.15 - 0.85	0.15 - 0.35 LB/MIN
400 - 1000	400 - 1000 °R

#### PERFORMANCE CHARACTERISTICS

- 0.4 - 0.6 : 1 O/F
- 1700 °R
- 50 - 220 PSIA
- 0.30 - 1.20 LB/MIN
- 140 L\*

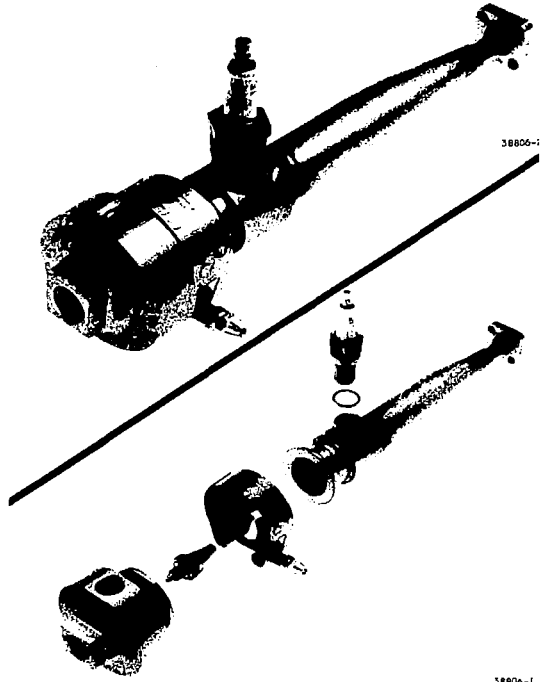
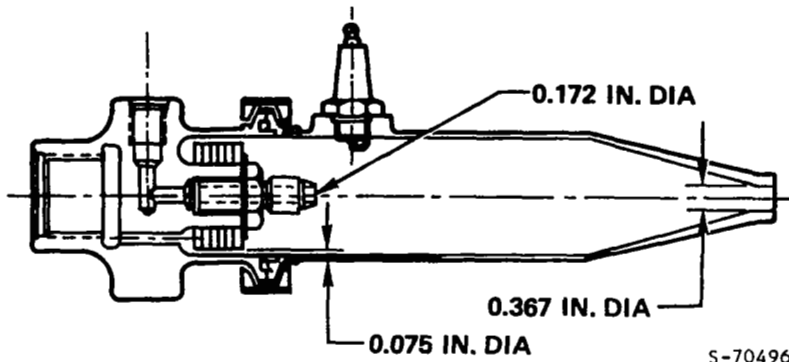


Figure 6-31. IPECS Hydrogen-Oxygen Combustor

## 2. Space Shuttle APU Combustor

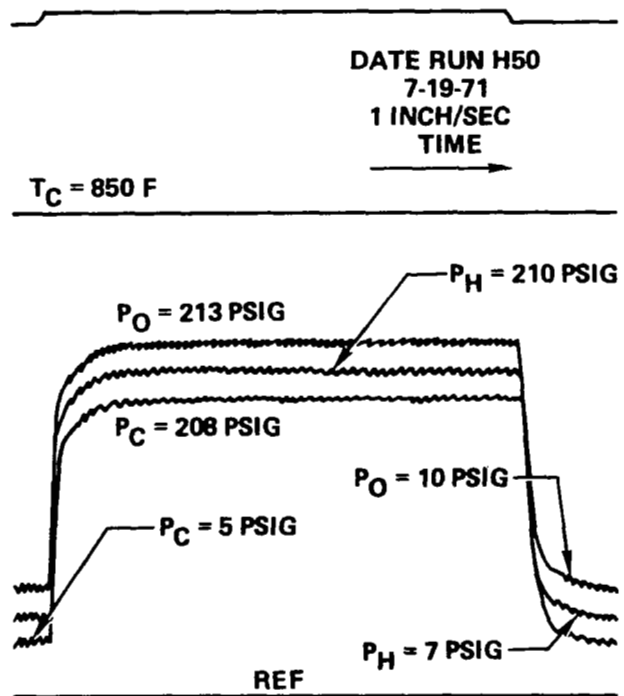
Recently, as a part of a company-funded program in support of the Space Shuttle APU study, AiResearch has built and tested a full-scale gas generator based upon the IPECS design described previously. Figure 6-32 shows this unit. It has demonstrated reliable ignition and stable combustion over a range of chamber pressure varying from 90 to 400 psia and of O/F ratio varying from 0.24 to 1.0. Typical transient performance test data are shown in Figure 6-33.





S-70496

Figure 6-32. Space Shuttle APU Test Combustor



S-70480

Figure 6-33. Space Shuttle APU Combustor Transient Test Performance

## Flow Control Valve Technology

Section 5, Propellant Conditioning Components, contains a summary of directly related hardware experience for the valves used in the APU. The related experience section points out design similarities of the APU components with performance proven components used in other aerospace applications. A table is included which lists directly applicable designs for the flow control valves described here.

The selected propellant flow control valve concept is individual electrically driven modulating valves with valve element position feedback. The selected method of flow control is derived from proven designs of cabin pressure control systems used currently on commercial passenger aircraft. The basis for the APU flow control is the electronic-pneumatic cabin pressure control system shown in Figures 6-34 and 6-35. The system pictured has been installed on the Boeing 707/300 series aircraft and successfully operated for over 2 years. It represents a fourth generation evolution in both valve and control design for this application.

Figure 6-36 shows the linearity of the feedback sensor in response to an altitude change of only 100 ft. The signal generated for this small change in pressure is 6 volts which provides ample resolution within the controller for valve actuation.

## Turbine

Figure 6-37 shows representative AiResearch supersonic turbines whose designs form a basis for the APU turbine design. Test data for the Zeus turbine is shown in Figure 6-38. The Zeus turbine blading is used on the Spartan turbine and is similar to that proposed for the Space Shuttle APU.

## Gearbox

AiResearch designs and manufactures almost all of the gearboxes used on its gas turbines and APU's. These designs form a basis for the selected gearbox for the Space Shuttle APU. Some of the most applicable gearboxes are:

<u>Application</u>	<u>Power, Hp</u>	<u>Speed, Rpm</u>
Spartan APU	70-140	70,000
SST Boost Compressor	280-400	73,800
TSE 231 Helicopter Engine	400-500	60,000
Air Force Advanced Technology Aircraft APU	400-600	80,000

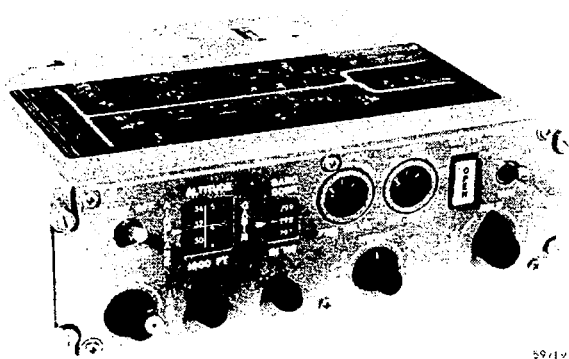


Figure 6-34. Cabin Pressure Altitude Selector

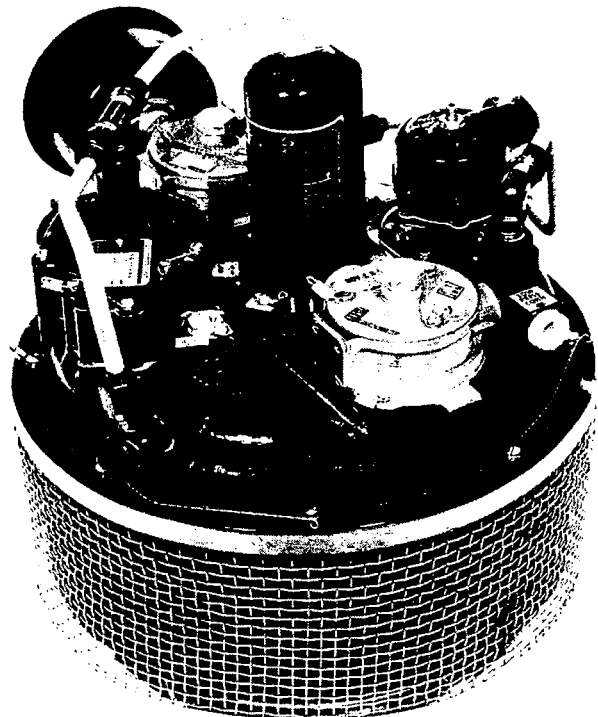


Figure 6-35. Cabin Pressure Controller and Valve System

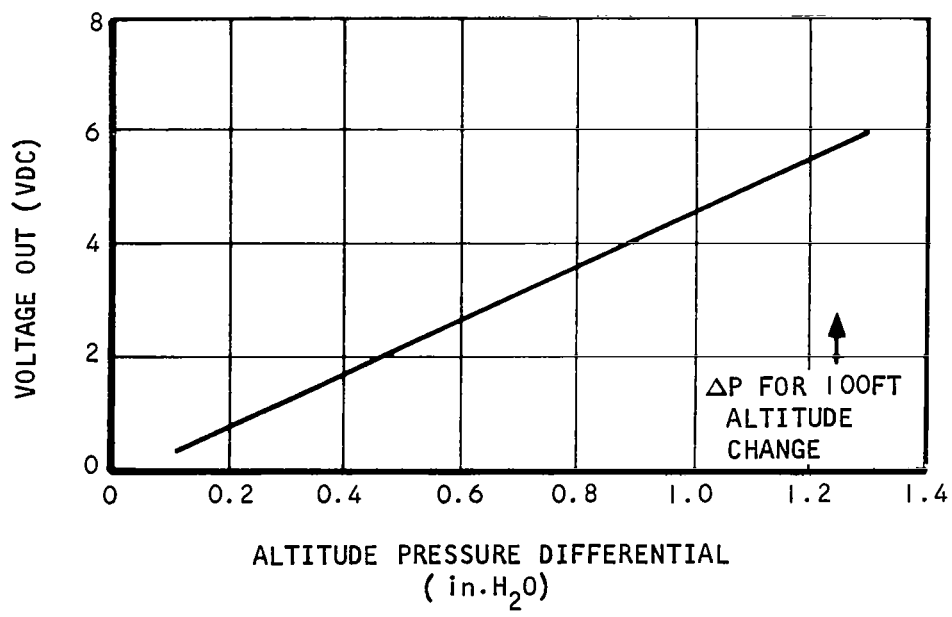


Figure 6-36. Feedback Sensor Response Characteristic



Figure 6-37. Axial Flow Impulse Turbine Wheels

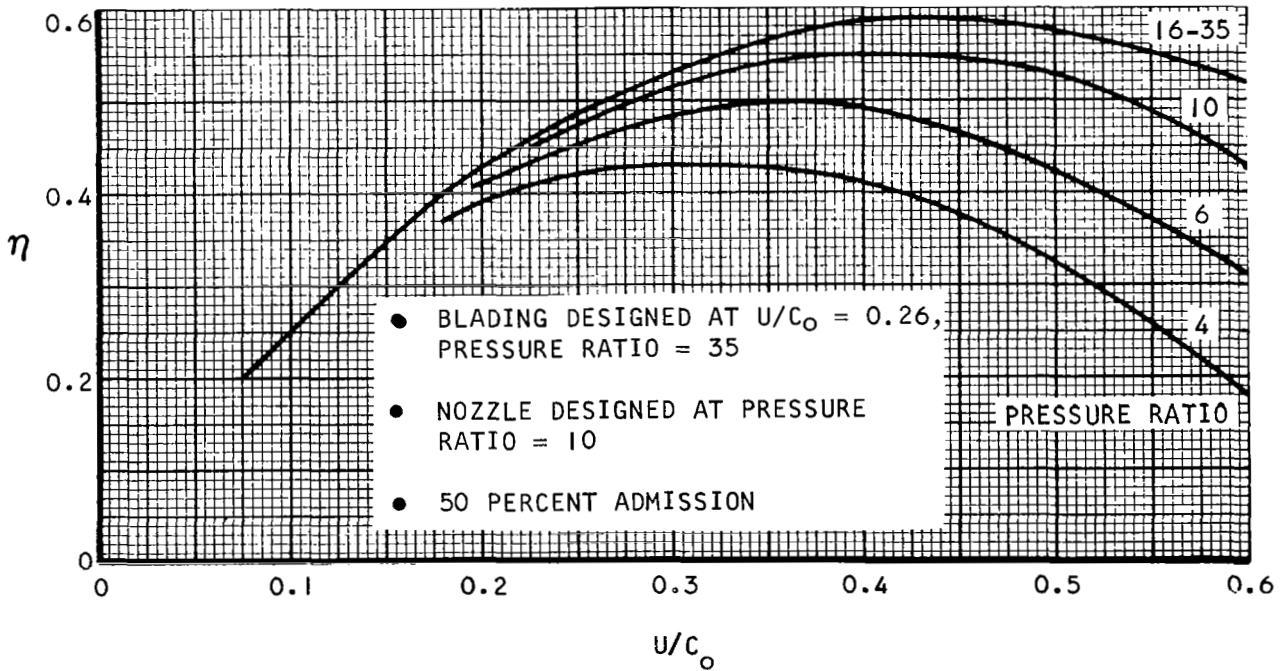


Figure 6-38. Zeus Turbine Dynamometer Test Data

## SECTION 7

### SYSTEM CONTROLS

#### INTRODUCTION

This section presents the selected control system design and the basis of the design. Additionally, it briefly summarizes the digital transient performance program that has been used as the primary means for assessing controls performance. The controls are divided into three areas:

- primary controls - those used by the APU to maintain speed and temperature control during operation
- secondary controls - those controls required to provide startup/shutdown, component monitoring, and emergency shutdown capability
- sensors - to provide the signals required for the primary and secondary control functions.

The bulk of the control design effort has been placed on optimizing the primary control circuitry. The supporting studies of Volume IV indicate that the desired primary controls for the APU are as follows:

- turbine interstage temperature (which is proportional to turbine inlet temperature)
- turbine speed
- jet pump discharge temperature (temperature at which hydrogen is provided to the lube oil cooler)

To assess the circuitry required to implement these functions, both analog and digital transient analyses have been performed. The analog studies are described in Section 7 of Volume IV. The digital studies used the analytical methods previously used for design of the complete engine control systems for AiResearch-produced propulsion engines (ATF-3 and TFE 731) and aircraft APU's (for 747 and DC-10). Both studies indicated that the desired controls should be integrating controls, as compared with droop controls.

The primary objective of the analog studies was to establish the relation between the control type and any pressure/flow fluctuations in lines between the pressure regulators and the flow control valves, and to determine the required control component response times. The digital studies were used to determine the system transient temperatures and component operating conditions throughout the transients. The digital program was also used to establish the optimum control gains, compensation, and coupling.

## PRIMARY CONTROLS DESCRIPTION

The system performance studies conducted during Phase II indicated that the optimum method of utilizing the selected primary control signals is as follows:

- turbine inlet temperature - adjust the oxygen flow (increase or decrease) while simultaneously slightly changing (decreasing or increasing) the hydrogen flow so that the total mass flow (and hence turbine power) remains constant
- turbine speed - simultaneously adjust both the hydrogen and oxygen flows maintaining a constant turbine temperature so that power is controlled
- jet pump discharge temperature - alter the hydrogen recycle flow to the hydrogen preheater

When the transient digital program was used to assess the performance of such an optimum control, it became apparent that it was possible to reduce the controls complexity and the intercoupling of the control signals by adjusting only the oxygen flow control valve in response to the turbine temperature signal. The resulting slight speed change will be controlled by the speed control loop, which will adjust both the oxygen and hydrogen flow control valve position. Thus, with this simplification, the selected primary control circuits are as shown in Figure 7-1.

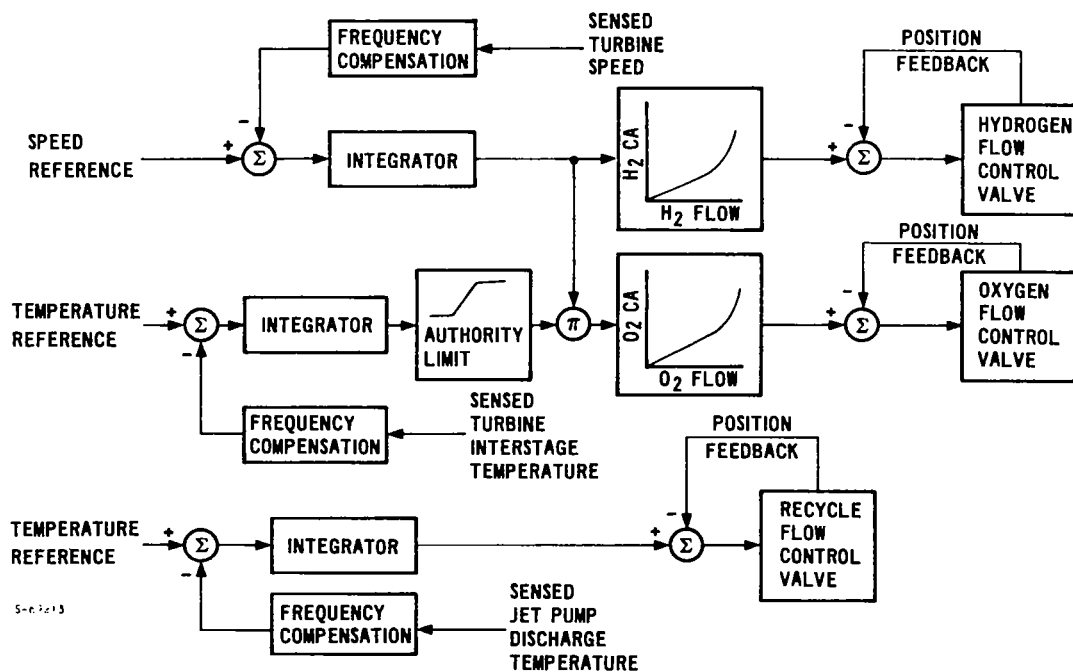


Figure 7-1. Primary Control Circuits - Block Diagram

All three primary control outputs (position signals to the hydrogen and oxygen flow control valves, and to the recycle flow control valve) are generated by integrated circuits with frequency compensation applied to the incoming sensed signals. The drive signal to the oxygen valve consists of the product of the conditioned speed and turbine temperature signals; thus, a change in turbine temperature will only drive the oxygen valve, whereas a change in speed will drive both valves.

The transfer functions, or relations between the sensed input, the reference input, and the resulting integrated output signal are given in Table 7-1. These relations were determined using the assumed sensor and valve response times shown in Table 7-2. The method of determining the transfer function was to use the transient digital program to determine system response when one of the control valves was perturbed from its equilibrium position. The resulting errors in the sensed signals provided by the program gave a basis for setting the gain and response required by the control circuits. Bode plots indicating the relation between the gain and the frequency for each control for four system perturbations were used to generate the desired circuits.

AiResearch has provided both gain and phase margins to insure controls stability over the anticipated variations in individual APU systems and in the controller component performance variations over the life/temperature profile anticipated for the APU.

Figure 7-2 shows how the circuit transfer functions can be implemented. Such an implementation is used in the high and low spool speed governor control circuits of the fuel controls for the flight-proven ATF-3 and TFE 731 engines.

Because of the nonlinearity between the oxygen and hydrogen flow valve position, and the resulting flow through the valve, it is necessary to provide a relation to compensate for this in the control logic. Consequently, the oxygen and hydrogen circuits use the flow/effective area relation shown in Figure 7-3 as a basis for control. At low flows, the valve area is directly proportional to the throughflow since the pressure ratio across the valve is greater than the critical pressure ratio. As the flow is increased, the valve area must become significantly greater to offset the reduced pressure drop across the valve. Implementing the function of Figure 7-3 will require four amplifiers. Multi-amplifier function generators are frequently used at AiResearch in such diverse applications as jet engine controls, cabin pressure controls, auxiliary power unit controls, and uninterrupted power sources.

Figure 7-4 shows the control performance obtained during a typical transient condition, a load step.

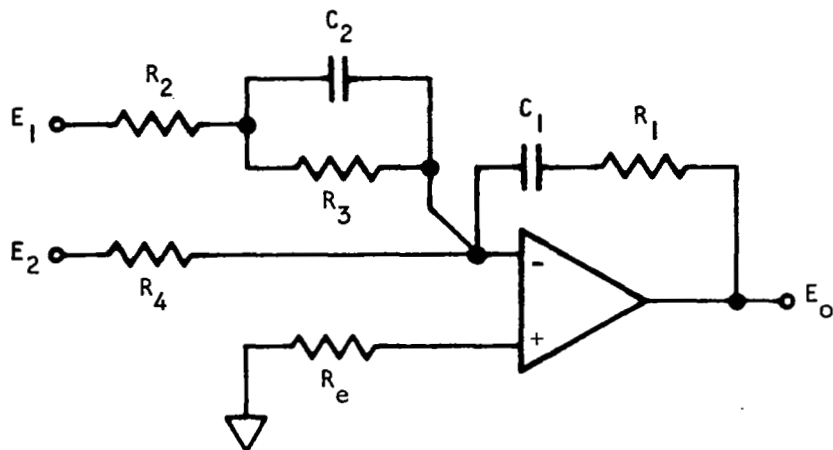
Table 7-1  
PRIMARY CONTROLS TRANSFER FUNCTIONS

CONTROL CIRCUIT	TRANSFER FUNCTION
Turbine Inter-stage Temperature	$G(S) = \frac{0.00526(0.15S+1)(1.5S+1)}{S(0.15S+1)} \frac{H_2-O_2 \text{ Flow Ratio}}{^{\circ}R - \text{Sec}}$
Turbine Speed	$G(S) = \frac{0.001 (0.25S + 1) (0.25S+1)}{S (0.05S+1)} \frac{H_2 \text{ Flow}}{RPM-Sec}$
Jet Pump Discharge Temperature	$G(S) = \frac{0.046 (1.5S+1)(S+1)}{S(0.05S+1)} \frac{Sq.In.}{^{\circ}R-Sec}$

Table 7-2  
ASSUMED CONTROLS RESPONSE TIMES

ITEM	RESPONSE TIME
<b>SENSORS</b>	
Turbine Speed	0.002 sec
Turbine Interstage Temperature	1.5 sec at 6 lb/min throughflow 1.0 sec at 12 lb/min throughflow
Jet Pump Discharge Temperature	1.5 sec at 5 lb/min throughflow 1.0 sec at 10 lb/min throughflow
<b>VALVING</b>	
Flow Control Valves	0.005 sec.
Pressure Regulators	0.03-0.05 sec.



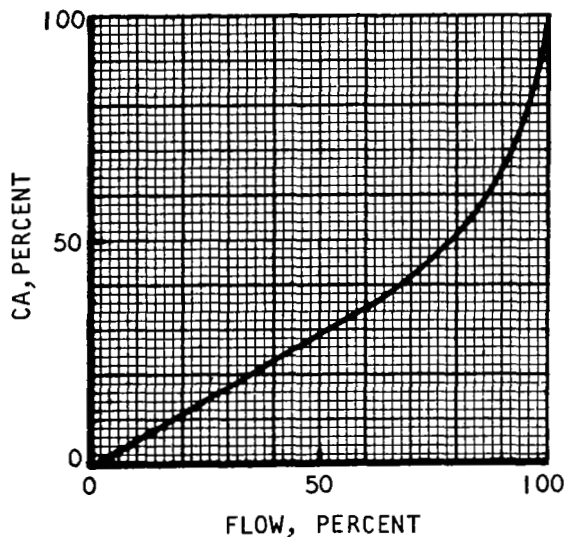


$$E_o = - \frac{R_1 C_1 S + 1}{C_1 S} \left[ \frac{E_2}{R_4} + \frac{E_1 (R_3 C_2 S + 1)}{R_2 + R_3 \left( \frac{R_2 R_3}{R_2 + R_3} C_2 S + 1 \right)} \right]$$

$$\frac{1}{R_e} = \frac{1}{R_4} + \frac{1}{R_2 + R_3}$$

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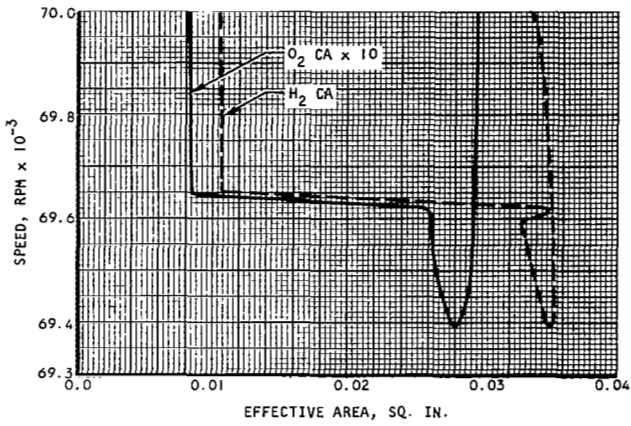
Figure 7-2. Typical Control Mechanization



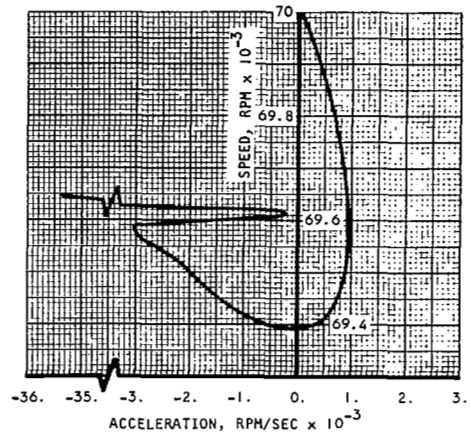
- VALVES SET FOR LINEAR  $H_2$  CA -  $O_2$  CA RELATION AT  $T_{it} = 2060^\circ R$ ,  $O/F = 0.6$
- NONLINEAR FLOW-CA RELATION COMPENSATES FOR PRESSURE RATIO CHANGE

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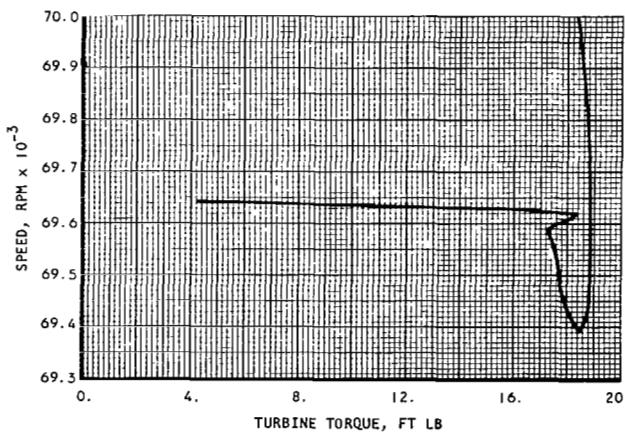
Figure 7-3. Hydrogen and Oxygen Flow Control Valve Compensation Curve



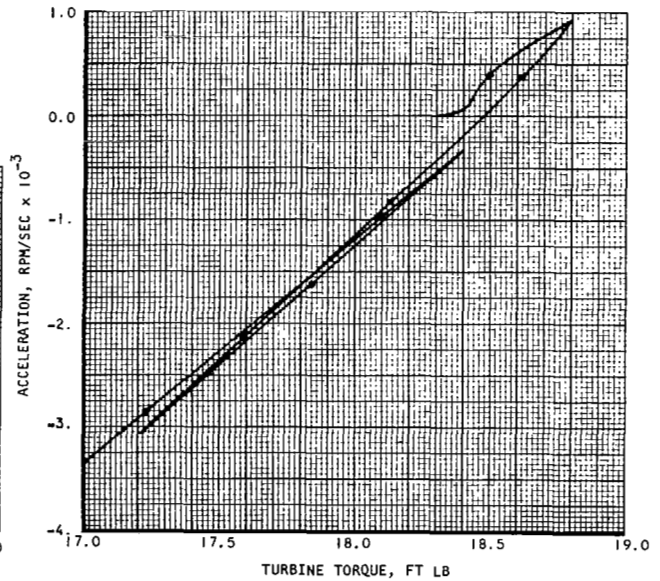
a. TURBINE SPEED VS FLOW CONTROL VALVE AREAS



b. TURBINE SPEED VS ACCELERATION



c. TURBINE SPEED VS TURBINE TORQUE



d. TURBINE ACCELERATION VS TURBINE TORQUE

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Figure 7-4. Controls Response to Load Step from 0-to-180 hp Useful Output; 75°R Hydrogen and 300°R Oxygen Inlet Temperature. 10 psia Ambient

## SECONDARY CONTROLS DESCRIPTION

The secondary controls fulfill the functions of startup, shutdown, component status monitoring, and fault detection.

### Startup/Shutdown

Table 7-3 shows the sequence of operations required during startup and shutdown. Consistent with normal aircraft practice, it is assumed that the APU controller will be directly connected to a 28 vdc power bus so that the controller is energized when the bus is energized. Thus, it is only necessary to open the hydrogen and oxygen shutoff valves and to energize the combustor ignitor to startup. To insure low initial ignition temperatures, it will probably be desirable to initiate hydrogen flow slightly prior to oxygen flow, although AiResearch has repeatedly started combustors using an oxygen lead as well. To start with a relatively low turbine temperature, the oxygen valve position is fixed relative to the hydrogen valve position until the turbine reaches 20,000 rpm. This concept has several advantages:

- thermal shock to the turbine disks is minimized (note that at low rpm, the turbine efficiency is low enough so that there is essentially no temperature drop across the turbine, thus a cold start is desirable)
- a short duration of heating by the turbine gas prior to assuming control, assures that the turbine interstage temperature has been brought well above its initial, ambient temperature

Shutdown is accomplished by closing the oxygen and hydrogen shutoff valves.

### Component Status Monitoring

The APU controller will provide several different signals for status monitoring by a centralized vehicle data monitor. These signals are as follows:

- turbine interstage temperature
- turbine speed
- jet pump discharge temperature
- regulated hydrogen pressure
- regulated oxygen pressure
- lube oil temperature

- lube oil pressure
- recycle valve position
- oxygen flow control valve position

The signals have been selected to maximize the amount of information obtained by monitoring, while still limiting the number of monitored signals to a reasonable level. Since most of the signals are used within the APU controller itself, the amount of complexity added to provide signals for monitoring is slight. Only the regulated oxygen pressure is not used directly within the controller.

#### Fault Detection

Figure 7-5 presents a summary fault logic diagram based on the study presented as Section 8 of Volume IV. The diagram indicates that the following parameters should be used to provide emergency shutdown signals:

- controller internal monitoring (continuous checking of controller circuits at key points defined by the control or detailed design)
- turbine over-temperature
- turbine over-speed
- turbine under-speed
- regulated hydrogen pressure
- lube oil temperature
- lube oil pressure

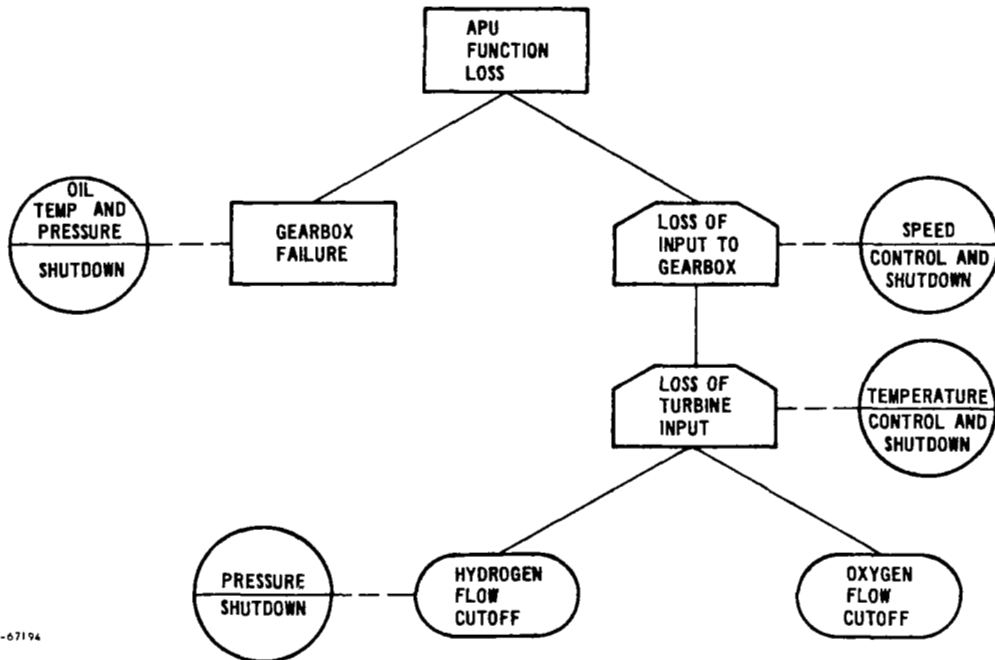
Additionally, it may be desirable to incorporate signals from related subsystems, such as the alternator and the hydraulic pumps in the APU controller.

Controller internal monitoring is a standard feature on AiResearch engine controllers. Typically, a propulsion engine controller will have monitoring at about 80 internal points. The APU controller is considerably simpler than an engine controller so that it is estimated that 40 to 50 points might require monitoring.

It should be noted that it is necessary to monitor the regulated hydrogen pressure since loss of that pressure would result in turbine over temperature. Although the overtemperature sensors would indicate such a condition, the temperature transient could be so abrupt that the sensing lag might be unacceptable.

**Table 7-3**  
**STARTUP/SHUTDOWN PROCEDURES**

<div style="border: 1px solid black; display: inline-block; padding: 2px 5px; margin-bottom: 10px;"><b>STARTUP</b></div> <ul style="list-style-type: none"> <li>● Drive H<sub>2</sub> and O<sub>2</sub> flow valves to startup positions (automatic with application of power because sensed speed is below 20,000 RPM)</li> <li>● Open hydrogen shutoff valve</li> <li>● Open oxygen shutoff valve</li> <li>● Turn on combustor ignitor power</li> <li>● Switch to turbine temperature control at 20,000 RPM</li> </ul>
<div style="border: 1px solid black; display: inline-block; padding: 2px 5px; margin-bottom: 10px;"><b>SHUTDOWN</b></div> <ul style="list-style-type: none"> <li>● Close oxygen shutoff valve</li> <li>● Close hydrogen shutoff valve</li> </ul>



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Figure 7-5. Fault Logic Summary

## Secondary Control Implementation

Figure 7-6 shows a block diagram of the secondary controls.

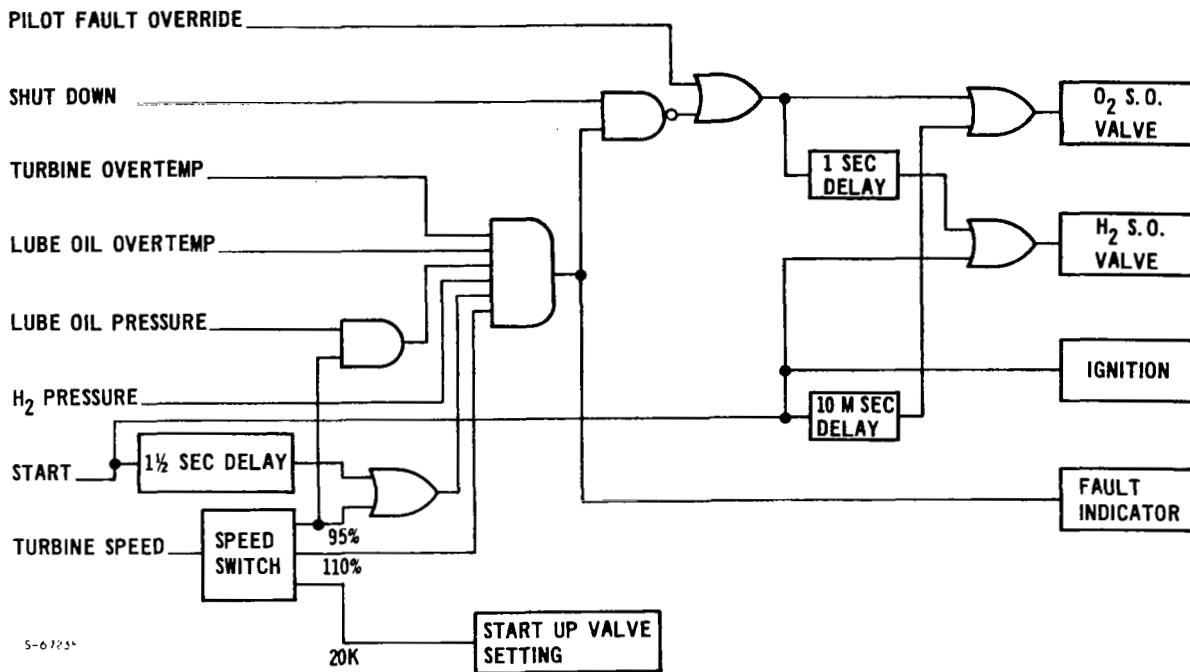


Figure 7-6. Secondary Control Circuits - Block Diagram

### SENSORS

Table 7-4 lists the sensors used in the system together with their function and normal operating ranges. Brief descriptions of these sensors follow.

#### Turbine Interstage Temperature Sensor

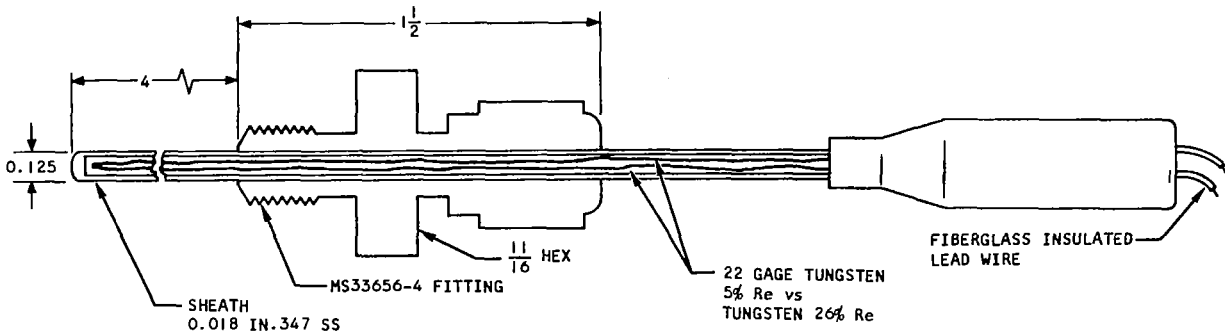
Turbine interstage temperature sensing is accomplished with shielded thermocouple using a tungsten-26% rhenium and tungsten-5% rhenium ungrounded junction. These materials are suitable for hydrogen environments, but require protection from oxygen or steam; hence, shielding is essential. AiResearch studies of thermocouple response (performed with a multi-nodal thermal analyzer program) indicate that the desired response can readily be attained with a shielded thermocouple. As a consequence, since shielding does provide protection, a shielded thermocouple is used. Figure 7-7 shows the dimensions and lists the general characteristics of the sensor. Each temperature sensor probe contains two independent thermocouple elements, one for primary control purposes (of turbine inlet temperature), the other for secondary monitoring functions. The thermoelectric output provides an estimated resolution of 20°F in turbine inlet temperature control.

TABLE 7-4  
APU SENSORS

SENSOR	FUNCTION	NORMAL OPERATING RANGE	SENSOR TYPE
Turbine Inter-stage Temperature	P,S	1705 ± 42 <sup>0</sup> R	Thermocouple (dual element)
Jet Pump Discharge Temperature	P,S	390-500 <sup>0</sup> R	Thermocouple (dual element)
Regulated Hydrogen Pressure	S	500 ± 25 psia	Linear variable-differential transformer
Regulated Oxygen Pressure	S	500 ± 25 psia	Linear variable-differential transformer
Lube Oil Temperature	S	460-760 <sup>0</sup> R	Thermocouple
Lube Oil Pressure	S	175-225 psia	Linear variable-differential transformer
Recycle Valve Position	S	0-80 degrees	Rotary variable-differential transformer
Hydrogen Flow Control Valve Position	P	0.001-0.015 in.	Linear variable-differential transformer
Oxygen Flow Control Valve Position	P	0.001-0.015 in.	Linear Variable-differential transformer
Turbine speed (overspeed)	S	70,000 ± 1200rpm	Reluctance pickup
Turbine speed (control)	P	6000 ± 100 rpm	Reluctance pickup

P = Primary control function  
S = Secondary control function

Considerable experience exists with the selected thermocouple materials in high-temperature hydrogen environments in both commercial and aerospace application. These materials are readily available at low cost.



<u>DESIGN</u>																	
FLUID	COMBUSTION PRODUCTS OF HYDROGEN AND OXYGEN AT AN O/F OF APPROXIMATELY 0.6																
OPERATING TEMPERATURE, °R RESPONSE, FLOW, LB/MIN	1705 1.5 SEC AT 5 LB/MIN; 1.0 SEC AT 10, LB/MIN 0.65 TO 12																
<u>OUTPUT</u>																	
TYPE	THERMOCOUPLE, TUNGSTEN 26 PERCENT Re VS TUNGSTEN 5 PERCENT Re, SHIELDED, UNGROUNDED JUNCTION																
PROBE DIAMETER, IN.	0.125																
PROBE IMMersed LENGTH	4 IN. FROM FITTING, 0.5 IN. IN INTERSTAGE FLOW																
FITTING	MS33656-4																
MATERIAL	CRES 347																
TRANSITION	TO FIBERGLASS INSULATED LEAD WIRE, 3 FT																
OUTPUT	<table> <tbody> <tr> <td>1560°R</td> <td>10.379 MV</td> </tr> <tr> <td>1660</td> <td>11.458</td> </tr> <tr> <td>1760</td> <td>12.536</td> </tr> <tr> <td>1860</td> <td>13.608</td> </tr> <tr> <td>1960</td> <td>14.671</td> </tr> <tr> <td>2060</td> <td>15.723</td> </tr> <tr> <td>2160</td> <td>16.764</td> </tr> <tr> <td>2260</td> <td>17.793</td> </tr> </tbody> </table>	1560°R	10.379 MV	1660	11.458	1760	12.536	1860	13.608	1960	14.671	2060	15.723	2160	16.764	2260	17.793
1560°R	10.379 MV																
1660	11.458																
1760	12.536																
1860	13.608																
1960	14.671																
2060	15.723																
2160	16.764																
2260	17.793																

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Figure 7-7. Turbine Interstage Temperature Sensor



### Recycle Valve Position Sensor

Recycle flow control valve position will be monitored by a rotary variable differential transformer type sensor which is discussed in more detail in the description of the control valve.

### Hydrogen Flow Control Valve Position Sensor

Position of the hydrogen flow control valve is measured by a conventional linear variable differential transformer type sensor. As discussed in this section, the signal from the sensor is used in the primary turbine controls. The sensor is discussed in more detail in the description of the hydrogen flow control valves.

### Oxygen Flow Control Valve Position Sensor

The oxygen flow control valve position sensor is similar to that used on the hydrogen flow control valve.

### Turbine Overspeed Sensor

The turbine overspeed sensor is located on the turbine shaft to sense turbine speed in event failure in the power train which would cause a loss of signal from the normal speed control sensor (which senses of a drive gear in the gearbox). The turbine overspeed sensor is a conventional reluctance type pickup of the type used on most AiResearch turbines for this purpose.

### Turbine Speed Control Sensor

To provide high resolution capability, the turbine speed control sensor supplies an output frequency of 10 kHz. To supply this high frequency, the reluctance pickup is used in conjunction with a 6000 rpm gear (which has 100 holes to provide the desired frequency).

### Jet Pump Discharge Temperature Sensor

This temperature sensor operates at moderate temperatures (390 to 500°R) in pure hydrogen. Conventional chromel-alumel thermocouples will be suitable for this application and will provide higher resolution for this control than the tungsten-rhenium thermocouple used for turbine interstage temperature sensing. As before, each sensor probe will contain two thermocouple elements, one for primary control (of jet pump discharge temperature), the other for secondary monitoring functions.

### Regulated Hydrogen Pressure Sensor

A conventional aircraft type linear variable differential transformer type of pressure transducer will be used for sensing regulated hydrogen pressure. Experience with this type of sensor has shown that mounting of the sensor may be important for proper performance and reliability in environments with high acoustic noise levels or high vibration.

### Regulated Oxygen Pressure Sensor

The regulated oxygen pressure sensor will be similar in design to the regulated hydrogen pressure sensor, described previously.

### Lube Oil Temperature Sensor

A chromel-alumel thermocouple will be used to measure lube oil temperature, which can vary over a range of approximately 460 to 760<sup>o</sup>R. Since this will be used for monitoring purposes only, a single element thermocouple will suffice for this application.

### Lube Oil Pressure Sensor

The lube oil pressure sensor will consist of a linear variable differential transformer type pressure transducer similar to those used for monitoring regulated hydrogen and oxygen pressures.

## PERFORMANCE

The primary controls performance is presented in Section 3 of this volume. The data there indicate that the control can maintain tight speed tolerances during step changes in the APU output power (in actuality, the output power will not have a step change due to the slow - 0.05 to 0.075 sec - response of the hydraulic pump). The Section 3 data also show that the turbine temperature can be controlled to within about 35<sup>o</sup>R during the step load changes and to within 42<sup>o</sup>R during 200<sup>o</sup>R/sec changes in the inlet oxygen temperature. The jet pump discharge temperature is controlled to within 10<sup>o</sup>R during the worst step load condition and is essentially constant during inlet oxygen temperature transients (because the hydrogen flow is not adjusted by turbine temperature changes).

In steady state operation, the turbine speed will be controlled to  $\pm 0.3$  percent, with most of this error being associated with the accuracy of the speed reference signal (in which a timing clock is necessary). The turbine temperature resolution will be about  $\pm 20^{\circ}\text{R}$ , based on AiResearch experience with high temperature thermocouples in similar applications.

Greatly improved turbine speed control during transients can be obtained by incorporating some form of load anticipation into the speed control circuit. Such a feature is a standard part of AiResearch-manufactured control systems for commercial and military turbine-driven generator sets. In these generator sets, 0.25 percent speed control can be maintained for load steps from 0 to 100 percent of rated output. In the APU application, load anticipation could be obtained by providing the APU controller with the conditioned hydraulic pump discharge pressure signal used by the pump as the means of controlling the pump displacement, and hence pump drive torque.

If such anticipation is provided, then the APU controller can be designed to alter the hydrogen and oxygen flow control valve positions at the same time that the pump displacement is being changed. At present, the controller only alters the valve positions after a load change causes a change in the turbine speed and speed rate of change.

## CONTROLS PACKAGING

Figure 7-8 shows the controller inputs and outputs. Figures 7-1 and 7-6 indicate the circuitry required for the primary and secondary controls relating the input and output signals. The electronics will be packaged in a standard ATR package used on both commercial and military aircraft. Figure 7-9 shows a typical package, (a 3/8 ATR short) a turbofan engine fuel control currently in commercial aircraft service. The electronics for the APU are estimated to require approximately 60 percent of the volume required by the turbofan engine fuel control. However, if additional control functions required by the alternator and the hydraulic pumps are incorporated into the APU controller, the package would probably require a 3/8 ATR short.

### Internal Packaging

Internal packaging of the ATR will be based on current AiResearch methods used for aircraft engine fuel controls, as shown in Figure 7-9. A two-sided printed-circuit mother board interconnects the edge card connectors of the individual circuit boards. The circuits are placed on two-sided printed circuit boards which are wave-soldered for high reliability. Heat dissipation is provided by regulating the electronics density, relying on conduction and radiation to the surrounding box.

### External Arrangement

Externally, the package will have an input/output connector and a test connector. The test connector allows complete checkout of the controller with the input/output connector in place. The controller packaged weight will be about 7 lb.

### Compatibility with Vehicle Multiplexing System

A key feature of the selected control concept in which continuous control (as opposed to pulsed, or bang-bang control) is used is its compatibility with the vehicle multiplexing system. This facilitates locating the controller remotely from the APU. Normally, the multiplexing system operates at about 100 Hz sampling frequency; AiResearch has evaluated such a sampling rate with its digital program and found complete control compatibility.

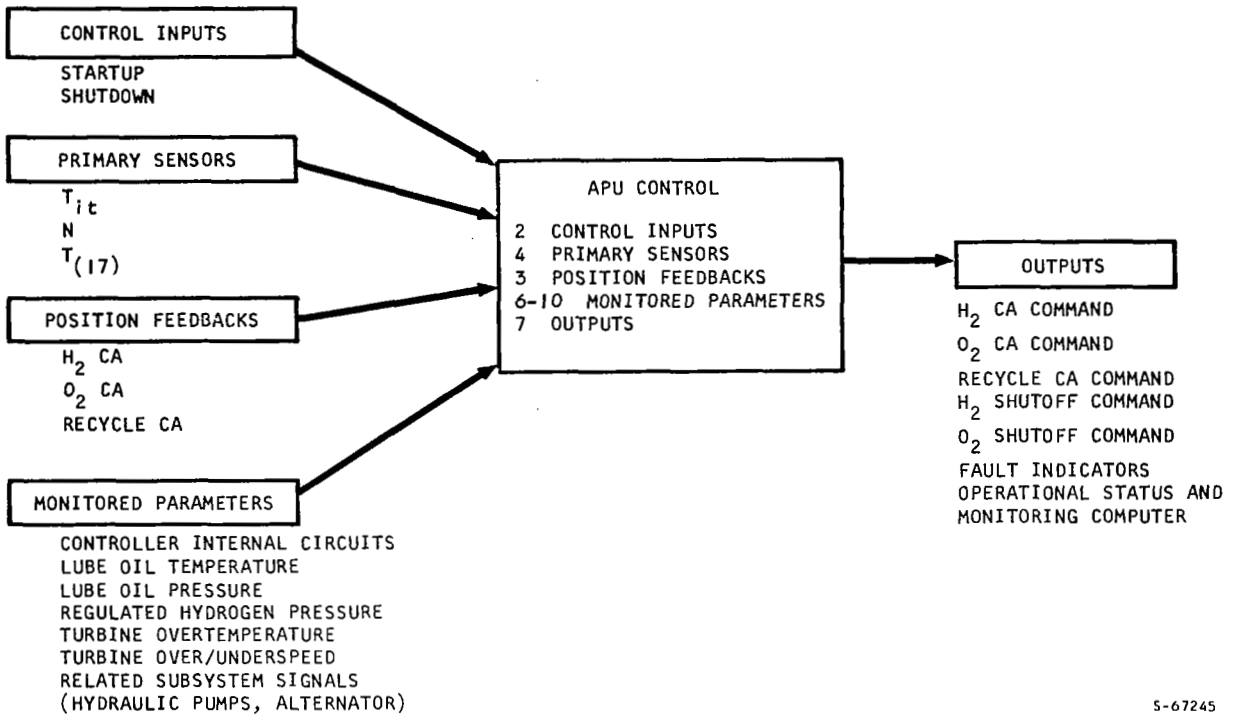


Figure 7-8. APU Control System Inputs and Outputs

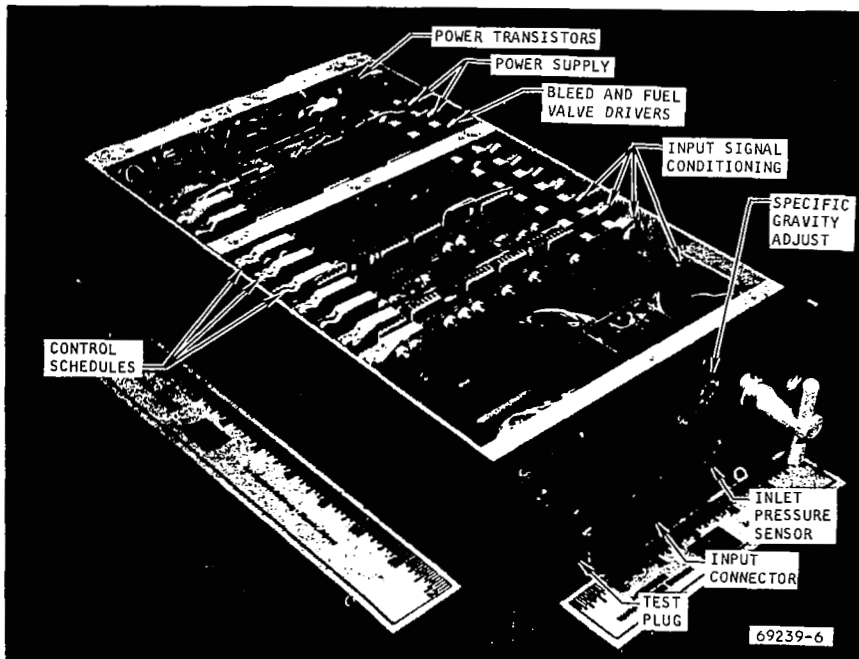


Figure 7-9. TFE 731 Turbofan Engine Controller

## DESIGN BASIS

The controller design selected for the Space Shuttle APU is based on AiResearch experience summarized in Table 7-5.

Table 7-5  
SHUTTLE APU DESIGN BASIS

SHUTTLE APU REQUIREMENT	APPLICABLE AIRESEARCH EXPERIENCE
Long Life	20,000 hr Gas Turbine Generator Controls 5,000 hr MTBF propulsion engine controls
Closed Loop Control	IPECS H <sub>2</sub> -O <sub>2</sub> APU Controls Gas Turbine Generator Sets Propulsion Engine Controllers Aircraft APU Controllers Air Inlet Control Systems
Biopropellant Flow Controls	IPECS H <sub>2</sub> -O <sub>2</sub> APU Controls
Simultaneously-positioned Electrically-actuated valves	IPECS H <sub>2</sub> -O <sub>2</sub> APU Controls Air Inlet Control Systems
Speed Control at Constant O/F Ratio	IPECS H <sub>2</sub> -O <sub>2</sub> APU Controls
Temperature Control by Adjusting Oxygen Flow Only	IPECS H <sub>2</sub> -O <sub>2</sub> APU Controls
Fault Detection and Monitoring of Both System and Controller	Propulsion Engine Controllers Aircraft APU Controllers Air Inlet Control Systems
Aircraft Compatibility (environment, packaging, multiplexing, etc)	Propulsion Engine Controllers Aircraft APU Controllers Air Inlet Control Systems

## IPECS Control System

In 1961, as part of an in-house development program, AiResearch built and tested a cryogenically-supplied hydrogen/oxygen APU in which turbine inlet temperature and turbine speed were controlled by coupled, closed-loop circuits similar to those evolved during the present study. Figure 7-10 shows a block diagram of the control circuitry. The speed signal is from the alternator output (APU load was an alternator only), and the temperature signal is from a sensor located in the combustor discharge flow line.

Figure 7-11 shows representative system test performance data. The speed control loop provides speed regulation to within  $\pm 1$  percent over a 15:1 load step change. The temperature control loop provides inlet temperature control to  $\pm 12$  percent. These controls were deemed adequate for the intended application so tighter accuracy (which is easily obtainable with present technology) was not attempted.

## Gas Turbine Generator Sets

The gas turbine generator sets manufactured by AiResearch use control concepts similar to those for the APU. Figure 7-12 shows a block diagram of the generator set control circuitry and Figure 7-13 shows test performance of a set during a step in output power from 0 to 100 percent of rated load. Because the controls incorporate load anticipation, the turbine speed change during the load step is less than 0.25 percent. These controls are designed for 20,000 hr life in a commercial installation.

## Propulsion Engine/Aircraft APU Controls

AiResearch manufactures most of the world's aircraft APU's, and many of the propulsion engines primarily for smaller, private aircraft). The ATF-3 spool turbofan propulsion engine controls are typical of the current controls technology selected for the Space Shuttle APU. The ATF-3 engine controller uses a wins technology control for each of the three primary control parameters, high pressure spool speed, fan spool speed, and turbine inlet temperature. The controls selects the lowest fuel flow required by any of the primary control parameters. This flow is compared with the deceleration fuel flow (acceleration and deceleration control is required to prevent surge) and the higher flow is selected. The selected value in turn is compared with the flow required for acceleration control with the lower value being the commended fuel flow for the engine. Figure 7-14 shows a block diagram of the control system. The controller has a 5000 hr MTBF. The packaging and controls design of both the ATF-3 and TFE-731 (controls shown in Figure 7-9) are similar.

## Inlet Duct Controls

The F-14 fighter incorporates a double ramp and bleed door inlet. This yields a system with 3 actuators per inlet. The six inlet control actuator on the F-14 are electronic liked to maintain desired inlet geometry.

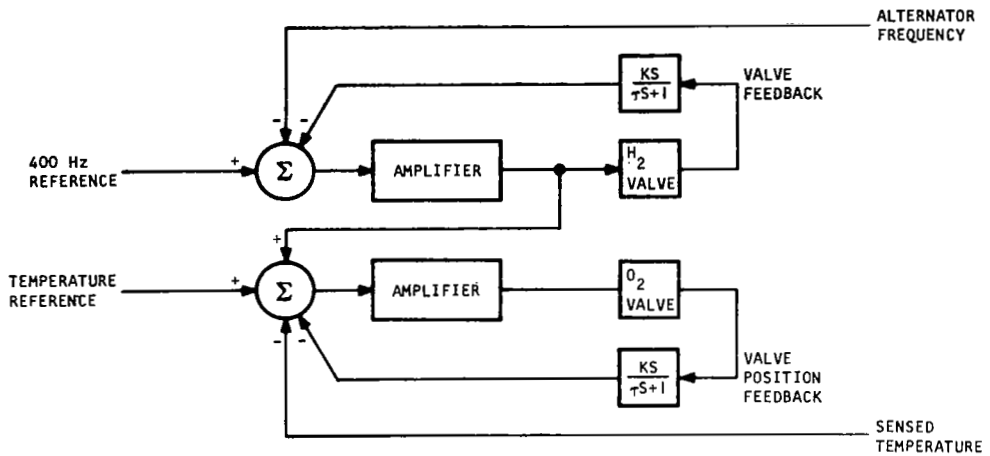
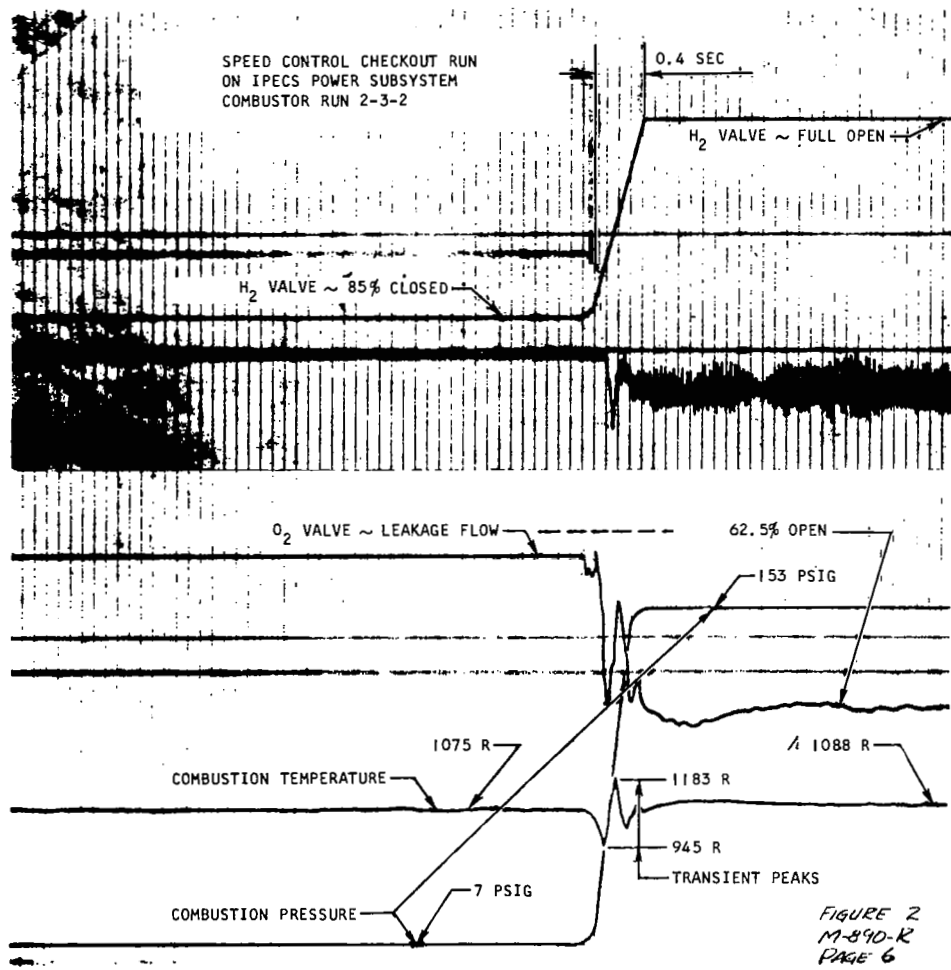


Figure 7-10. IPECS Closed Loop Speed and Temperature Control



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Figure 7-11. Typical IPECS Transient Performance Test Data

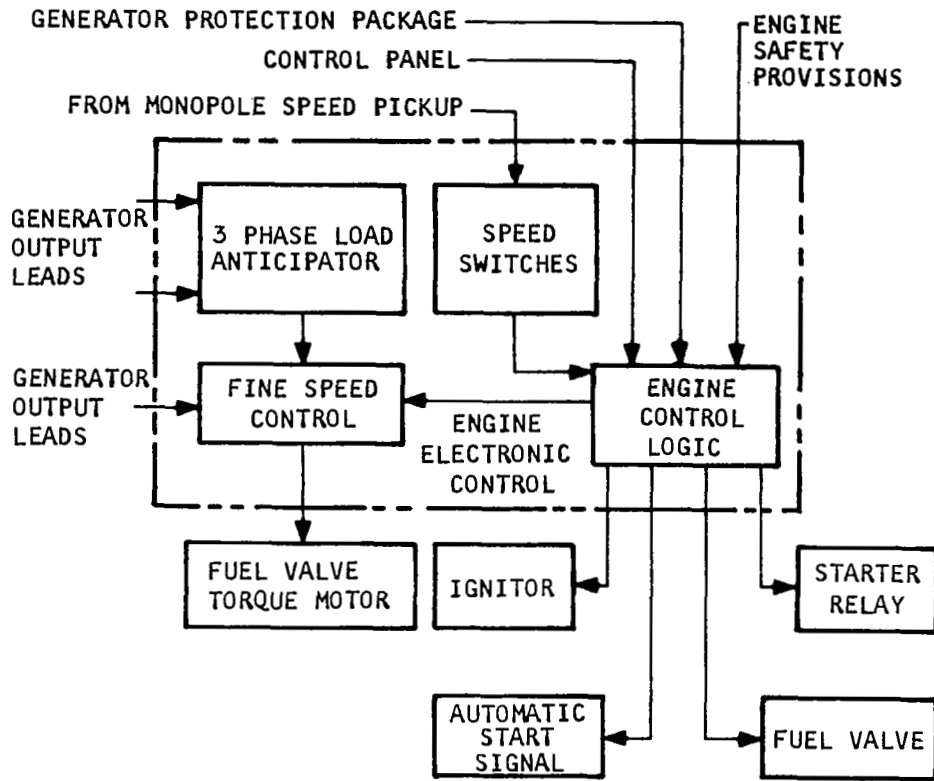
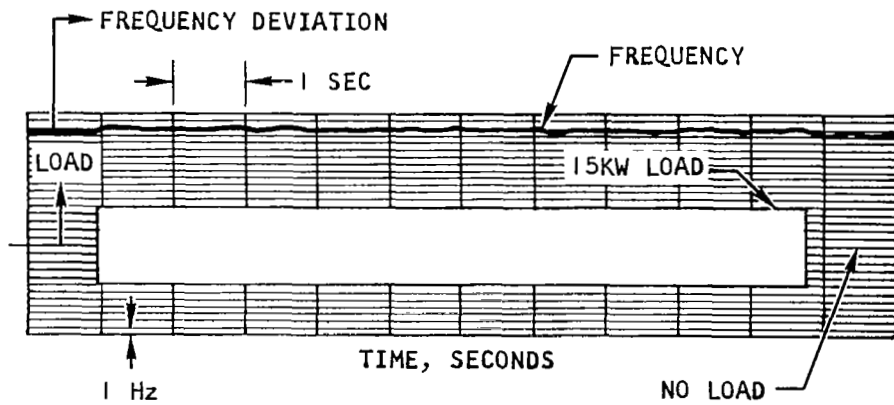


Figure 7-12. Typical Turbine Generator Controller - Block Diagram



ERDL 15-KW GENERATOR SET S/N E 127

S-67180

Figure 7-13. Typical Turbine Generator Step Load Response



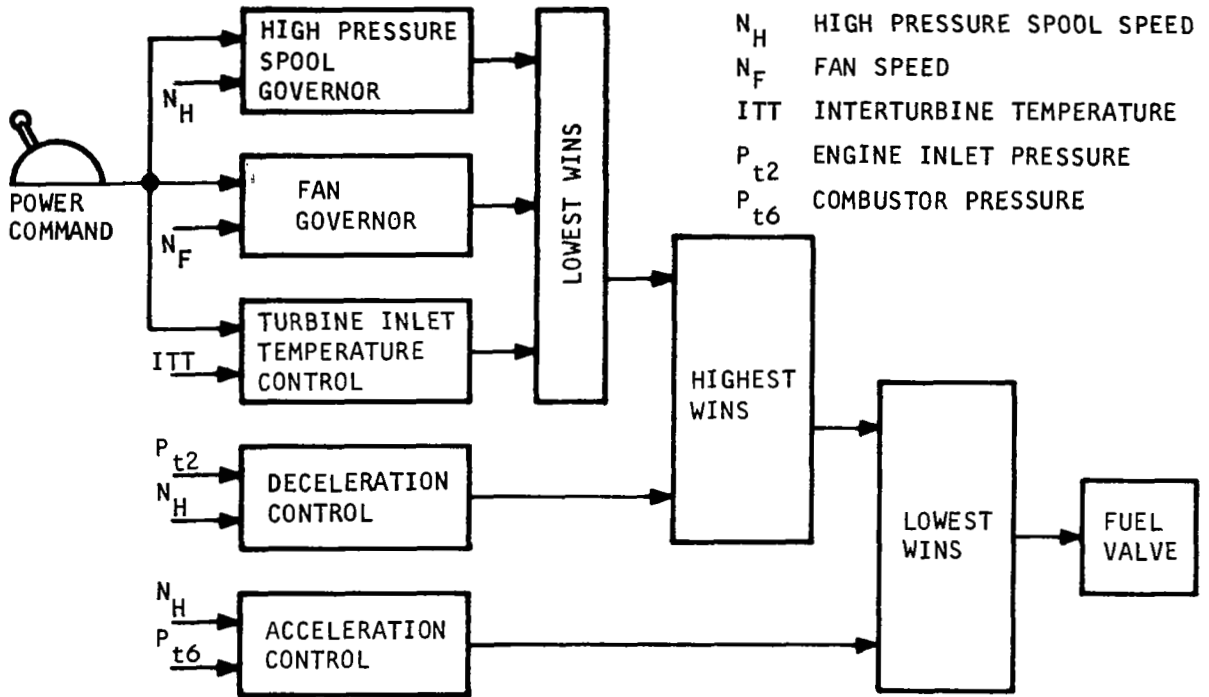
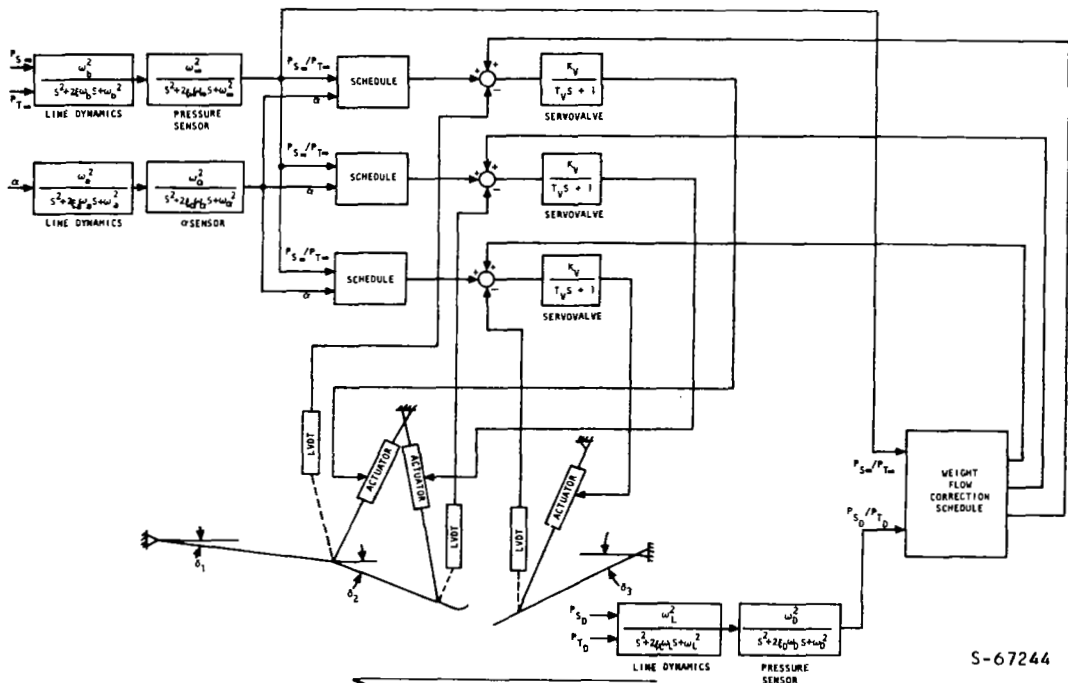


Figure 7-14. AFT3 Engine Controller - Block Diagram



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Figure 7-15. F-14A Air Inlet Control System Schematic

Such a concept eliminates the high maintenance (rigging, backlash elimination, lubricating, etc) associated with mechanical valve sequencing. Figure 7-15 shows the control system.

#### DIGITAL TRANSIENT PERFORMANCE PROGRAM

The bulk of the controls design work has been performed using a digital transient performance program written for the APU application. Table 7-6 summarizes the system model, the digital transient study scope and the objective. The procedure used for evaluating the controls is shown in Figure 7-16.

The digital program is modeled after existing digital transient programs used for design of AiResearch-manufactured propulsion engines and aircraft APU's. The Space Shuttle APU program consists of about 10,000 program cards (divided into 34 subroutines), and 3000 data input cards. The program is written in Fortran V for the Univac 1108 computer.

Table 7-6  
TRANSIENT STUDIES--SCOPE/OBJECTIVE

<b>SYSTEM MODEL</b>
<ul style="list-style-type: none"><li>● Use steady-state performance curves for jet pump, valves, combustor, and gearbox</li><li>● Use turbine <math>\eta</math> vs speed for pressure ratio lines to establish turbine performance</li><li>● Use newly-developed transient HX model which includes performance correction associated with HX construction</li></ul>
<b>SCOPE</b>
<ul style="list-style-type: none"><li>● Evaluate "ideal" and "best actual" control systems performance</li><li>● Assess response during startup, step load application, shutdown, and inlet propellant temperature transients</li></ul>
<b>OBJECTIVE</b>
<ul style="list-style-type: none"><li>● Generate all required transfer functions to be simulated by controls</li></ul>

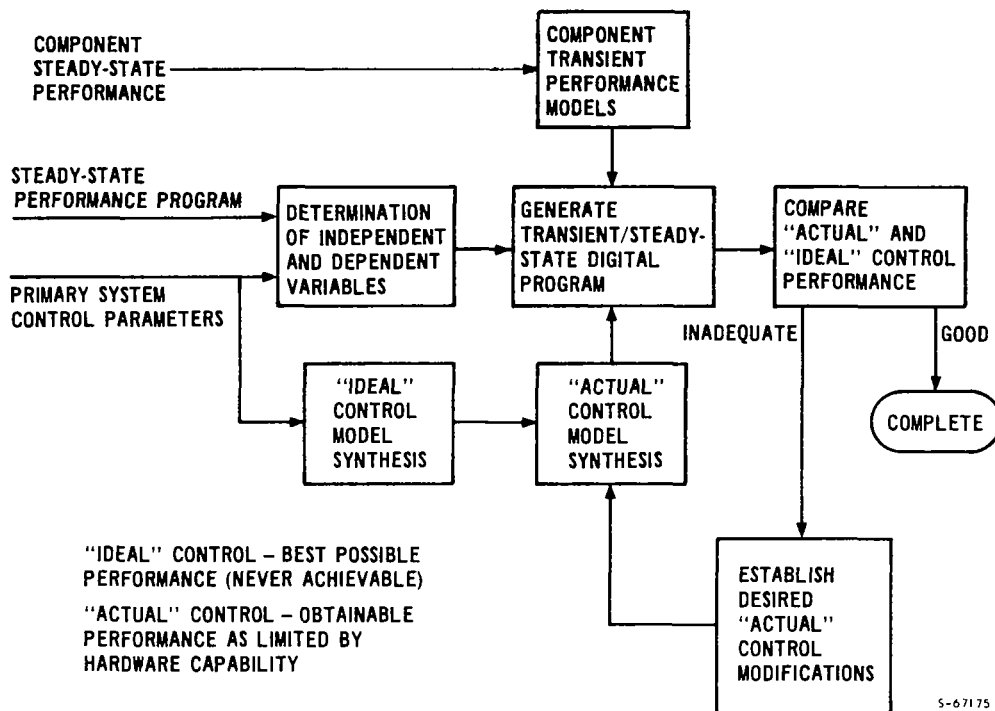


Figure 7-16. Digital Control Study Procedure - Block Diagram

Figure 7-17 summarizes the program method of solution at each time increment during the imposed transient. At each time, the program converges to a solution in which the sum of the squares of the normalized independent variable ( a total of 7 independent variables are required) errors is less than 1 part in 1,000,000.

AiResearch developed a new method of estimating heat exchanger transient performance on a 3 mode basis as part of this contract. Figure 7-18 shows the possible heat exchanger configurations handled by the model, and displays the logic process used for performance prediction. Normally, 3 mode models are not usable for transient performance prediction because the results violate the second law of thermodynamics. However, by incorporating a construction factor (which is an indication of the average temperature gradient available for heat transfer) in the thermal model, it is possible to satisfy the second law. The selected model predictions have been compared with those obtained from a 42 model heat exchanger transient performance analysis. The comparison indicates that the APU heat exchanger model is in good agreement with the more sophisticated multi-nodal analysis.

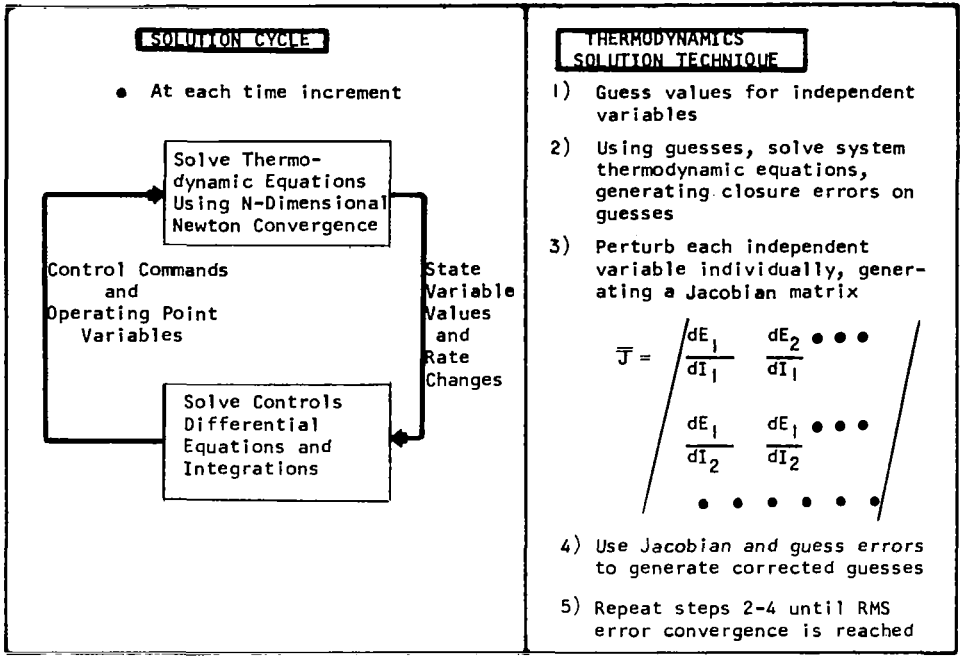
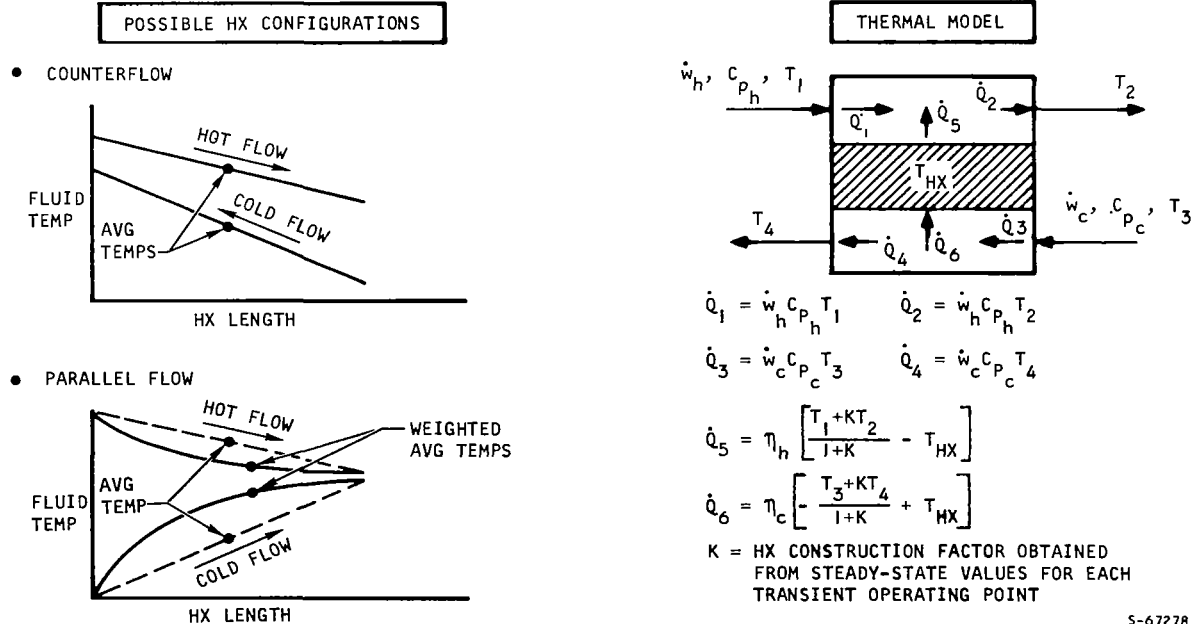


Figure 7-17. Digital Transient Program Solution Methodology



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Figure 7-18. Heat Exchanger Transient/Steady-State Thermal Model

SECTION 8  
PUMPED LIQUID CYCLES

INTRODUCTION

As indicated previously, the Phase II baseline system concept was based upon use of propellants supplied as high-pressure gas (on the order of 500 psia minimum) from the Auxiliary Propulsion System (APS). Because of the questions concerning the impact of APU operation on APS turbopump life (by greatly increasing the number of operational cycles), NASA directed that the study include consideration of APU systems with integral pumping provisions.

Since the range of propellant inlet conditions specified by NASA for the baseline system includes that obtained in a liquid-supplied system with integral propellant pumps, the baseline system data are applicable to the pumped cycles. Specifically, the propellant pumping power must be considered as output power in calculating propellant consumption. The other cycle performance parameters are determined for the propellant temperature and pressure delivered by the pumps. Figure 8-1 illustrates the functional interfaces between the propellant pumps and the baseline APU cycle. From this, it can be seen that the propellant pumps represent the major new element. As discussed in Section 5 of Volume IV the system can be simplified by eliminating the recycle flow control where inlet propellant temperature remains relatively constant. Since this will be the case for the liquid-fed system, the baseline cycle can be modified in this way (shown in Figure 8-2) for pumped liquid operation.

PUMP DESIGN CRITERIA

Optimum cycle performance will be obtained with liquid-fed systems for pump delivery pressures on the order of 1100 psia. Table 8-1 summarizes the design requirements for the hydrogen and oxygen pumps.

TABLE 8-1  
PUMP DESIGN REQUIREMENTS

Parameter	Oxygen	Hydrogen
Flow, lb/min		
maximum	5	7.07
minimum	0.25	0.43
Output pressure, psi		
maximum	1100	1100
minimum	900	900
Fluid		
State	Subcooled liquid	Subcooled liquid
Pressure, psi	50	50
Net positive suction head, psi	10	10
Life, hr	3000	3000

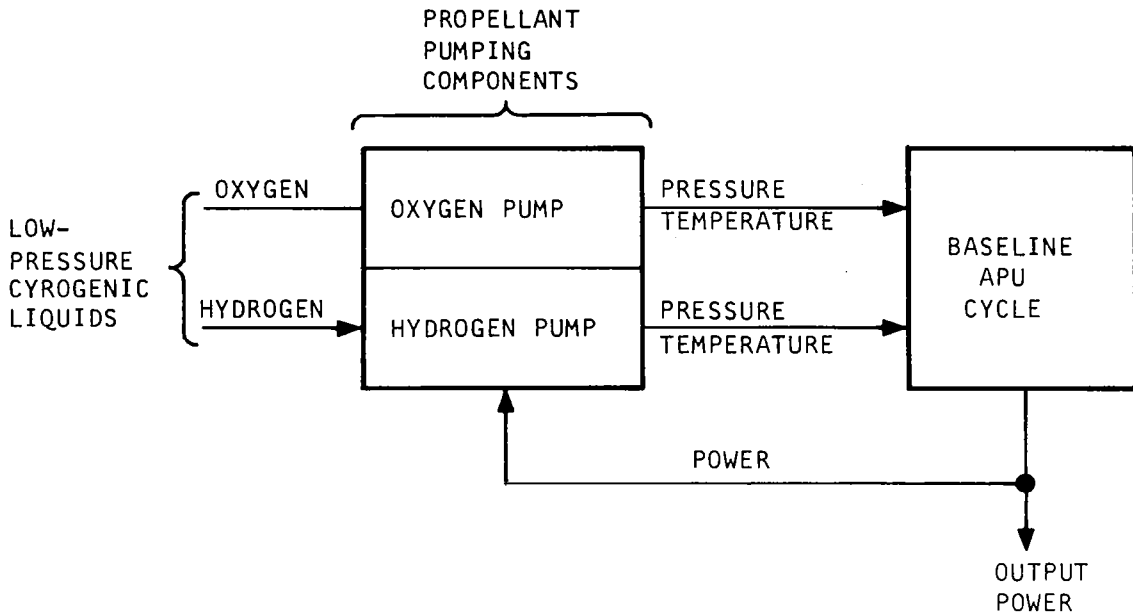


Figure 8-1. Liquid-Fed System/Baseline APU Cycle Interfaces

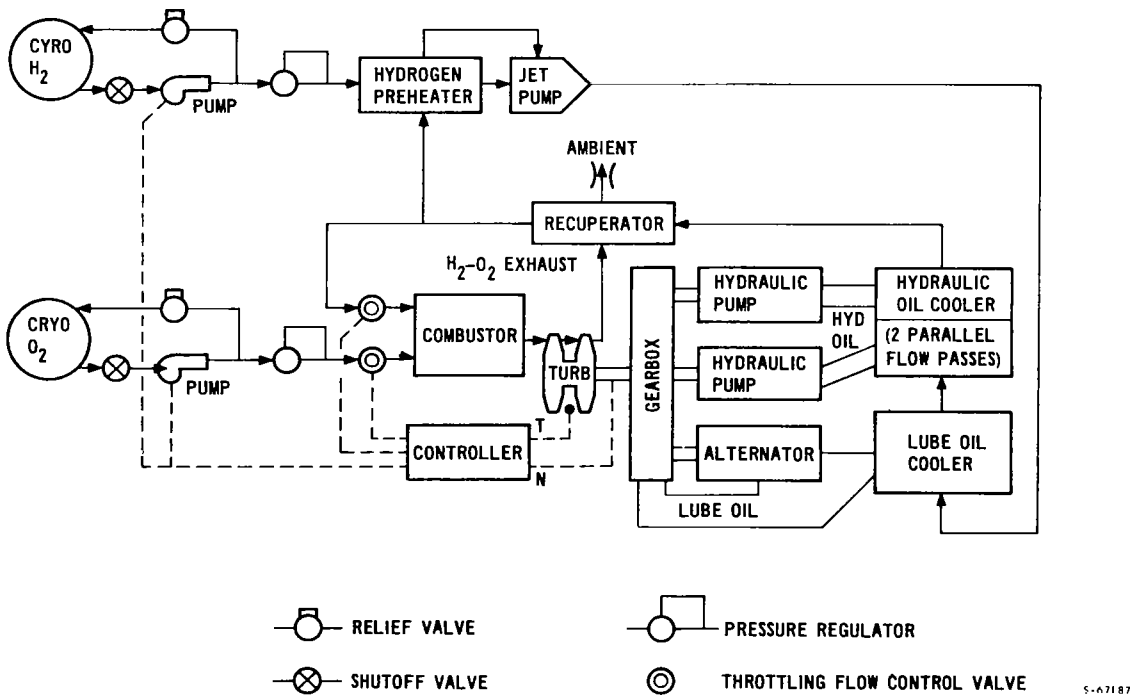


Figure 8-2. Low-Pressure Cryogenic Liquid Supplied System Schematic

## TYPES OF PUMP CONSIDERED

Consideration was given to positive-displacement and centrifugal pumps. Cosmodyne was solicited for preliminary information concerning positive-displacement cryogenic pumps and responded with a technical proposal, a portion of which is included in this section.

The Cosmodyne approach centered on the use of constant and variable flow positive displacement pumps. The high delivery pressure was the principal reason for selecting this approach. The Cosmodyne analysis concluded that the positive displacement pump best suits the hydrogen problem statement; a firm recommendation of a pump preliminary design was not received for the oxygen application.

AiResearch evaluated the same problem statements by using high-speed centrifugal pumps. This analysis indicated that the oxygen centrifugal pump best suits the oxygen problem statement, and a centrifugal pump will not appear reasonable for the liquid hydrogen application.

The results of the study are therefore as follows:

Liquid oxygen pump, centrifugal type, AiResearch design  
Liquid hydrogen pump, positive displacement type, Cosmodyne design

## OXYGEN LIQUID PUMP

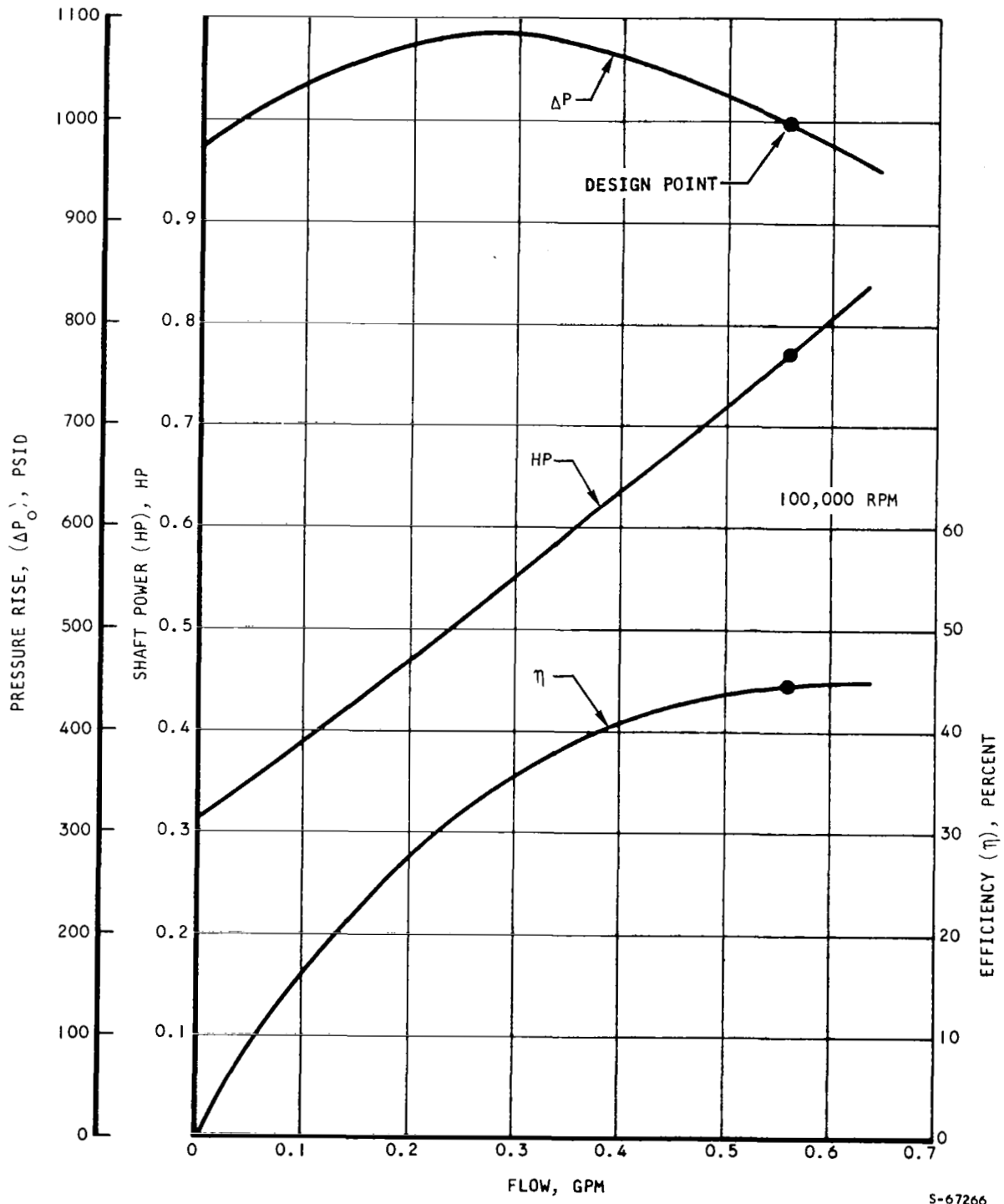
### Positive Displacement Pump Concept

Analysis of the performance requirements based upon a positive displacement pump revealed the following design considerations. First, the extremely low flow rates require only a single-cylinder pump; second, an accumulator is required in the pump discharge line to smooth pressure fluctuations. Third, it is necessary to include provisions for bypassing the pumped fluid to the storage tank to obtain the high turndown ratio while maintaining adequate net positive suction head (NPSH) at the pump inlet. Therefore, it is concluded that this type of positive displacement is not practical for the present application.

### Centrifugal Pump Design

Analysis of the performance requirements based upon the centrifugal pump approach reveals the necessity for a small, high-speed pump to match the low flow, high head requirements. The unit studied is a two-stage centrifugal pump. The two impellers are mounted on opposite ends of the electric motor shaft. Table 8-2 shows the pertinent data of the design.

The speed of 100,000 rpm has been chosen close to the optimum efficiency point. The small size and narrow channel width of the impellers are within the state of the art. The performance losses due to increased inaccuracy are included in the estimation of efficiency. Part-load performance is shown in Figure 8-3. The pressure rise, pump efficiency, and shaft power are given



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Figure 8-3. Liquid Oxygen Pump



TABLE 8-2

## LIQUID OXYGEN PUMP DESIGN DATA

Fluid	Subcooled liquid oxygen
Inlet temperature (maximum)	186.777°R
Inlet pressure (maximum)	50 psia
NPSP	10 psi
Pressure rise (design)	1000 psid
Weight flow (design)	5 lb/min maximum
Pump design	Centrifugal
Number of stages	2
Speed	100,000 rpm
Specific speed of stage	398
Suction specific speed of first stage	7483
Impeller Reynolds No.	$1.1 \times 10^6$
Efficiency	44.5 percent
Shaft power	.772 hp
Dimensions	
Impeller tip diameter	0.551 in.
Impeller tip width	0.015 in.
Impeller inlet eye diameter	0.200 in.
Diffuser design: Single outlet conical	
Overall length	6.6 in.
Overall diameter	3.5 in.
Total weight	6 lb

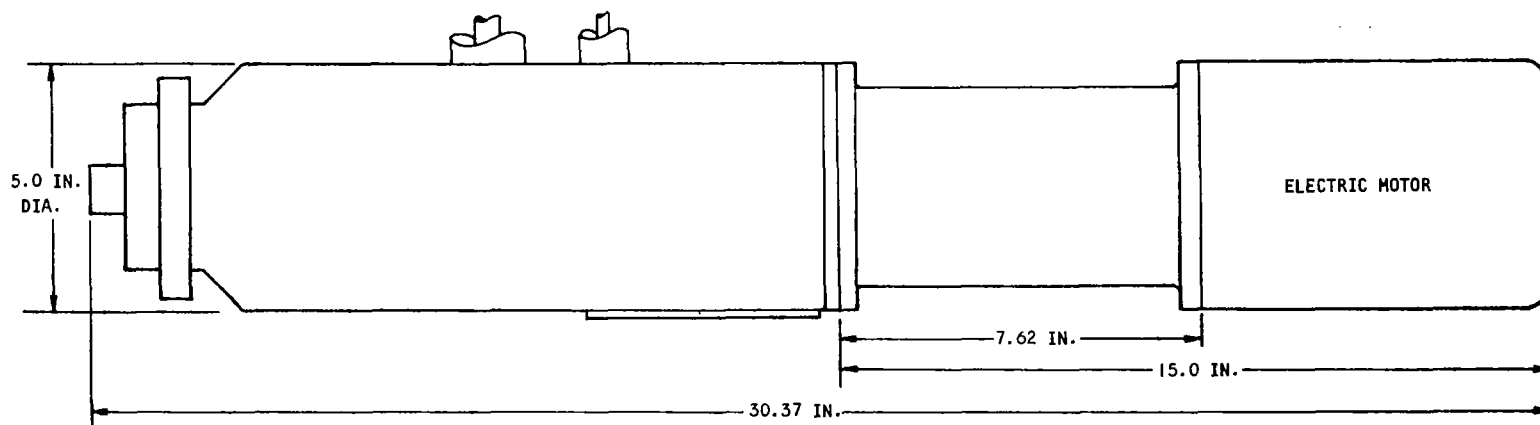
as functions of flow rate at constant speed. The shaft power includes the ball bearing losses, but the efficiency values do not.

## HYDROGEN LIQUID PUMP

Positive Displacement Pump Concept

The pump selected for the hydrogen application is a 5-cylinder, axial-piston type and is designed to be directly driven from a constant speed electrical motor. It contains both a cryogenic pumping portion and a drive portion which is actually an integral electric motor. The pump is compact, lightweight, and has been designed for a 3000-hour life. A drawing of the pump is shown in Figure 8-4. Leading characteristics of the pump are listed in Table 8-3.

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Figure 8-4. Constant Flow Pump Outline of Cosmodyne SK-8K-843

d. The cryogenic piston always travels to the same point at top-dead-center of its stroke, thus fixing the clearance volume and therefore the compression ratio. This is extremely important in a cryogenic pump. It is likely that gas bubbles will be present in the cylinder and if the compression ratio is too small, the pressure in the cylinder will not rise to the point which will open the discharge valve. The gas will then remain in the cylinder and the pump will cavitate. Maintaining the clearance volume at an absolute minimum assures good volumetric efficiency and minimizes the net positive suction pressure requirements.

e. The nutating plate directly drives the hydraulic sequencing valve. While the sequencing valve could be driven independently, driving it with the nutating plate positively relates the hydraulic valve timing to the stroke of the pistons. Leading characteristics of this hydraulically driven pump assembly are shown in Table 8-4 below:

TABLE 8-4

HYDRAULICALLY DRIVEN PUMP ASSEMBLY LEADING CHARACTERISTICS

Flow	0.43 to 7.07 lb/min 0.73 to 12.0 gpm
Discharge pressure	1000 psig maximum
Pump inlet	
Pressure	50 psia maximum
NPSH	10 psi minimum
Hydraulic drive	
Hydraulic supply pressure	3000 psig
Hydraulic flow	0.3 to 6.0 gpm
Weight	25 lb
Size:	
Diameter	5 in.
Length	20-3/8 in.
Servo valve electrical supply	24 vdc, 80 ma

Centrifugal Pump Design

The conventional centrifugal, or Barske-type pump design of the liquid hydrogen pump represents extensions of current design practices. To reach an acceptable level of specific speed, high rotative speed is required, (well over 100,000 rpm) even when using two stage design. The power requirement is high (over 15.7 hp) with 50% efficiency. The high speed and high power represents a feasibility problem for the electric motor design.

Reduction of the speed is possible using a two-stage regenerative pump (vortex-pump) design. This pump would run at 60,000 rpm having 37.6 percent efficiency and requiring a 20.9 hp electric motor. Pertinent data for this pump is shown in Table 8-5.

TABLE 8-5

## LIQUID HYDROGEN PUMP DESIGN DATA

Fluid	Subcooled liquid hydrogen
Inlet temperature	45.7°R
Inlet pressure	50 psia
NPSP	2.8 psi
Pressure rise (design)	1000 psid
Weight flow (design)	7.07 lb/minimum
Pump design	Regenerative (vortex)
Number of stages	2
Speed	60,000 rpm
Specific speed of stage	139.2
Suction specific speed of first stage	2617
Rotor Reynolds No.	$5.07 \times 10^6$
Efficiency	37.6 percent
Shaft power	20.9 hp
<u>DIMENSIONS</u>	
Impeller tip diameter	2.156 in.
Impeller tip width	0.250 in.
Intake duct diameter	0.350 in.
Motor type	4-POLE induction
Rotor diameter	2.0 in.
Rotor length	2.8 in.
Stator outside diameter	4.0 in.
Stator length with windings	4.3 in.
Motor weight	12.05 lb
Pump unit outside diameter	4.5 in.
Pump unit overall length	7.5 in.
Pump unit weight	19.8 lb

An off-design performance map is presented in Figure 8-6 which illustrates a major disadvantage of this design: the power requirement increases with decreasing flow rate. Because of this characteristic, the pump would evaporate the fluid at low flow rates. A control would be required to regulate a liquid bypass, to assure a minimum flow through the pump that would exceed the minimum value for evaporation.

The electric motor would be an induction motor having 4 inch overall diameter and 4.3 inch overall length and weighing 12.05 lb. The relatively small motor size is achieved at the expense of the converter (if required). Based upon a motor input voltage of 270 vdc, the converter value is estimated to be 3.5 cu ft and to weigh 130 lb. These facts, together with the basic pump design problems, preclude further consideration of the centrifugal pump approach to meeting the requirements for cryogenic liquid hydrogen pumping.