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SPACE SHUTTLE AUXILIARY POWER UNIT STUDY

Phase II

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16. Abstract

A 10-month study was performed in two phases to establish the preliminary design of the space shuttle auxiliary power unit. Included in this phase II summary report are details of the analysis, optimizations, and design of the components, subsystems, and systems.

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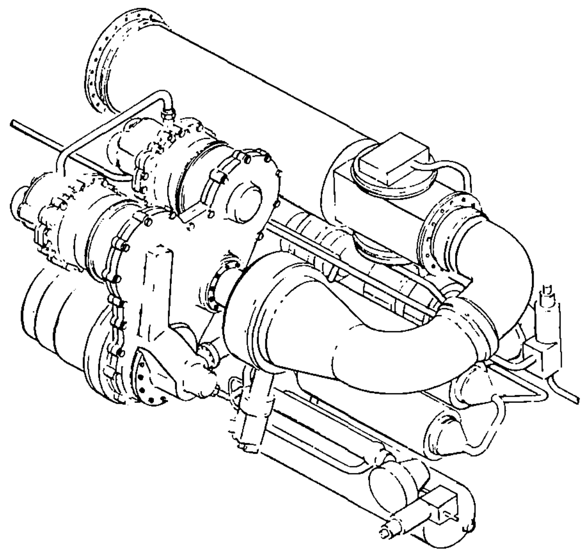
I SUMMARY OF RESULTS

The Space Shuttle Vehicle missions will encompass a broad range of operating conditions throughout launch, orbital, and landing operations. The requirements for the auxiliary power units (APU) to supply hydraulic and electrical power during critical maneuvering stages of the mission are unique. The APU must perform reliably and satisfactorily under varying environments and duty cycles even after periods of long orbital dormancy. Yet by the very nature of the application, the APU must provide the highest possible performance with a minimum weight. The Space Shuttle Auxiliary Power Unit presented in this summary report meets these requirements in the specific form as a result of primary operational and non-operational criteria applied throughout the component, subsystem, and system level analysis and design.

The Space Shuttle APU was designed with the primary criteria as listed in Fig 1, page 2. Flexibility and vehicle adaptability relate to the application of the APU in a vehicle under the various operating conditions which may be expected to be imposed under the wide range of missions anticipated. The APU must be adaptable to the various vehicle concepts under evaluation including those which may evolve from present vehicle studies. Safe operation of the APU is related to providing an APU which will never have a failure of the type which can cause damage to the vehicle itself. All APU failures must result in safe shutdowns of the APU without damaging the vehicle. The forgiving design aspect of the APU is related to the design of the components and subsystems with substantial latitude so that satisfactory operation of the APU can be expected even when the particular components or subsystems are operating outside of their basic design specifications. In addition, when unexpected conditions are imposed on a particular component, adequate design margin should have been designed into the component to assure satisfactory operation. Minimum complexity and high reliability must be designed into the APU to ensure adequate system operation throughout the entire life of the APU. Low development risk and the associated low development cost refer to selecting concepts and designs which are within the state of the art thus minimizing the amount of development associated with the APU. High performance of the system is associated with reducing the amount of propellant utilized by the APU system and minimizing the fixed weight associated with the APU, and making the maximum use of existing component technology, reinforces the reduced development costs by utilizing components which have been qualified for space application.

The APU as designed contains various features which allow it to meet the criteria discussed above and the design of the system is summarized on Fig. 2 where views of the APU complete package is illustrated. A complete design including packaging and structural elements was completed for the basic APU system. All the necessary components including turbopower unit, propellant conditioning system and integrated control system were designed and packaged as illustrated. System operation was simulated over the entire flight operating envelope through the use of a computer model. Both steady state and dynamic conditions were imposed upon the system, and successful operation was achieved over the entire flight operational envelope.

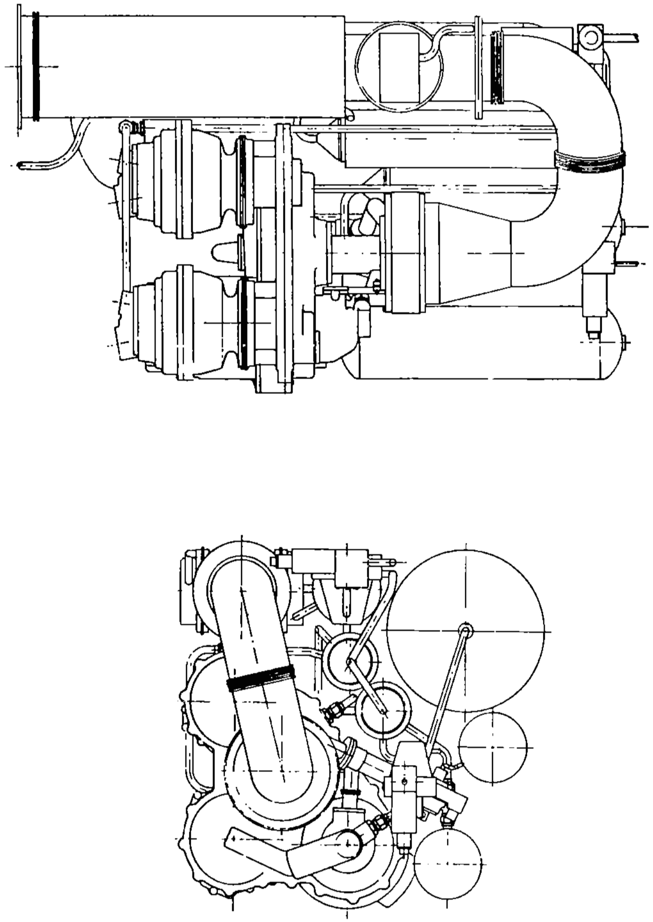
SS/APU SUMMARY CRITERIA



- FLEXIBILITY
- VEHICLE ADAPTABILITY
- SAFE OPERATION
- FORGIVING DESIGN
- MINIMUM COMPLEXITY/ HIGH RELIABILITY
- LOW DEVELOPMENT RISK
- HIGH PERFORMANCE
- MAXIMUM EXISTING COMPONENT TECHNOLOGY

Figure 1. SS/APU Summary Criteria

SYSTEM DESIGNED



OPERATION SIMULATED

ENTIRE FLIGHT OPERATING ENVELOPE

- STEADY STATE
- DYNAMIC
- START UP/SHUT DOWN
- PROPELLANT TRANSIENTS
- COMPLETE HYDRAULIC COOLING
- FAILURE MODES SYNTHESIZED

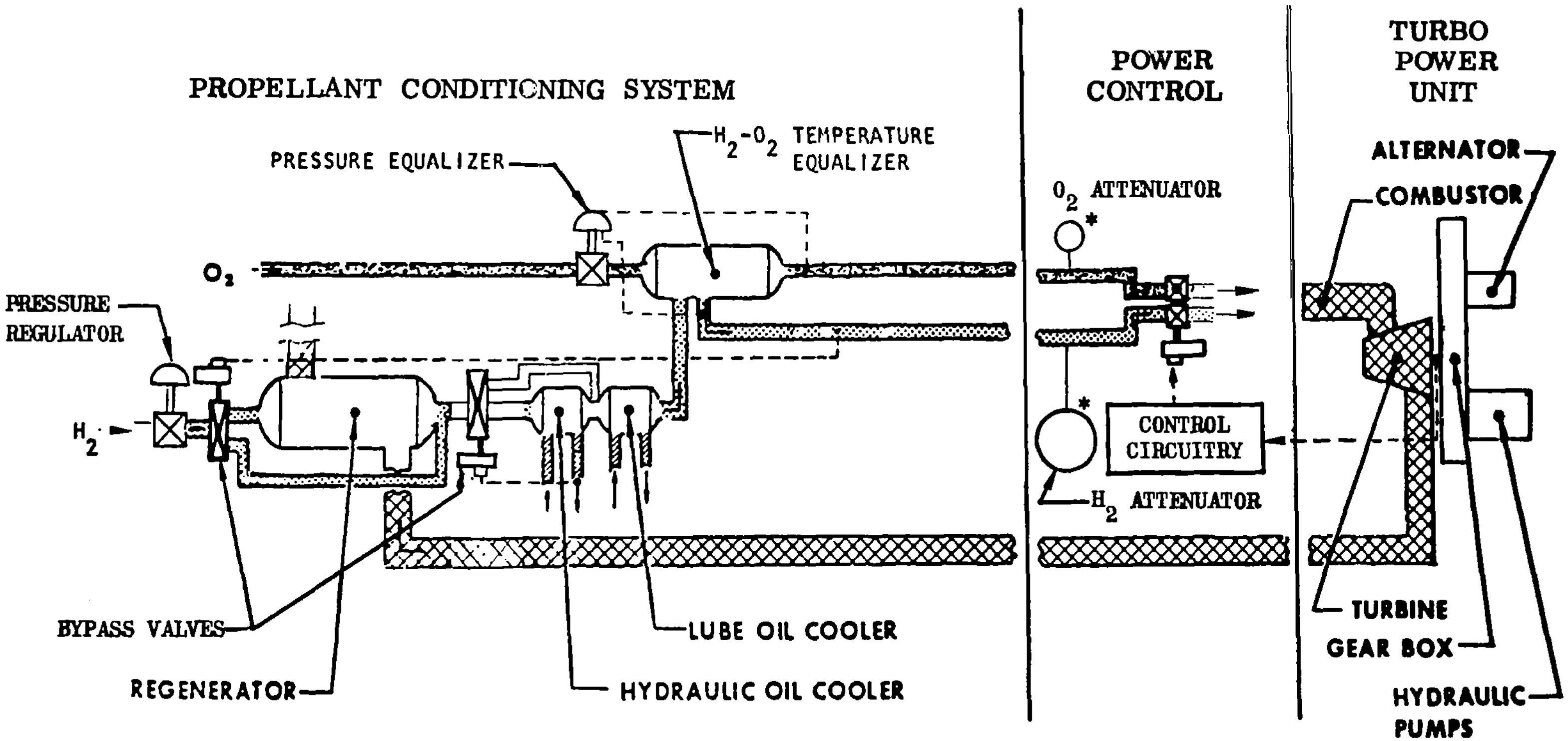
Figure 2. Accomplishments

Start up and shut down were demonstrated with start up occurring in 1.3 seconds. The system is capable of providing full power within 2 seconds from initiation of start up. All the necessary propellant transients associated with the baseline system, were successfully accommodated. Complete hydraulic cooling capability was demonstrated using the model when the necessary temperature hydrogen is provided to the system by the propellant supply system. A failure mode analysis was accomplished and for those elements deemed most likely to fail their failure mode was synthesized in the system using the computer model. All the failure modes resulted either in continued safe operation or in safe APU shut down.

The baseline APU is illustrated in Fig. 3 and consists of a propellant conditioning system, power control system, and turbopower unit. The function of the propellant conditioning system is to provide equal pressure, equal temperature, controlled propellants to the power control valve and turbopower unit combustor. The power control system provides the necessary power modulation while the turbopower unit converts the propellant to usable electrical and hydraulic power. Hydrogen is received from the vehicle propellant supply system and after flowing through the regulator, passes through the propellant conditioning system regenerator, where it is heated up by the exhaust from the turbine. An appropriate amount of hydrogen is bypassed around the regenerator to control the temperature of the hydrogen into the power control valve. The partially heated hydrogen then flows through the hydraulic oil cooler where the necessary hydraulic oil cooling is accomplished. A hydrogen bypass valve controls the temperature of the hydraulic oil to the appropriate level.

The heated hydrogen then flows through the lube oil cooler where the lube oil is cooled, and then through the hydrogen/oxygen temperature equalizer where the oxygen is brought up to a temperature equal to the hydrogen temperature. Hydrogen at 650 R then is supplied to the power control valve. On the oxygen side, the oxygen pressure is reduced from its supply level and made equal to the hydrogen pressure in a pressure equalizer. The oxygen then has its temperature brought up and equalized to the hydrogen temperature in the passive temperature equalizer.

The hydrogen and oxygen propellants which are supplied to the power control valve are of a fixed equal temperature and have a fixed equal pressure, thus ensuring that the temperature of the combustion products supplied to the turbine are determined and controlled. The power control system consists of a linked-bipropellant valve which delivers the necessary amount of propellant to the turbopower unit to provide the necessary power level from the turbine and keeping the turbopower unit within the proper speed range. For a pulse power control system attenuators are provided in the hydrogen and oxygen lines and serve to decouple the propellant feed system from the power control system assuring continuous flow in propellant conditioning system. For a pressure modulated system, the attenuators are eliminated. An isometric of the APU system is illustrated in Fig. 4 for a pulse modulated system including the 2 ft^3 hydrogen attenuator. All the necessary components are illustrated in the isometric drawing. A summary model specification appears in Table 1 page 7 for the baseline system. The pressure modulated system shown in Fig. 5,



* Required For Pulse Control Only

Figure 3. Baseline APU

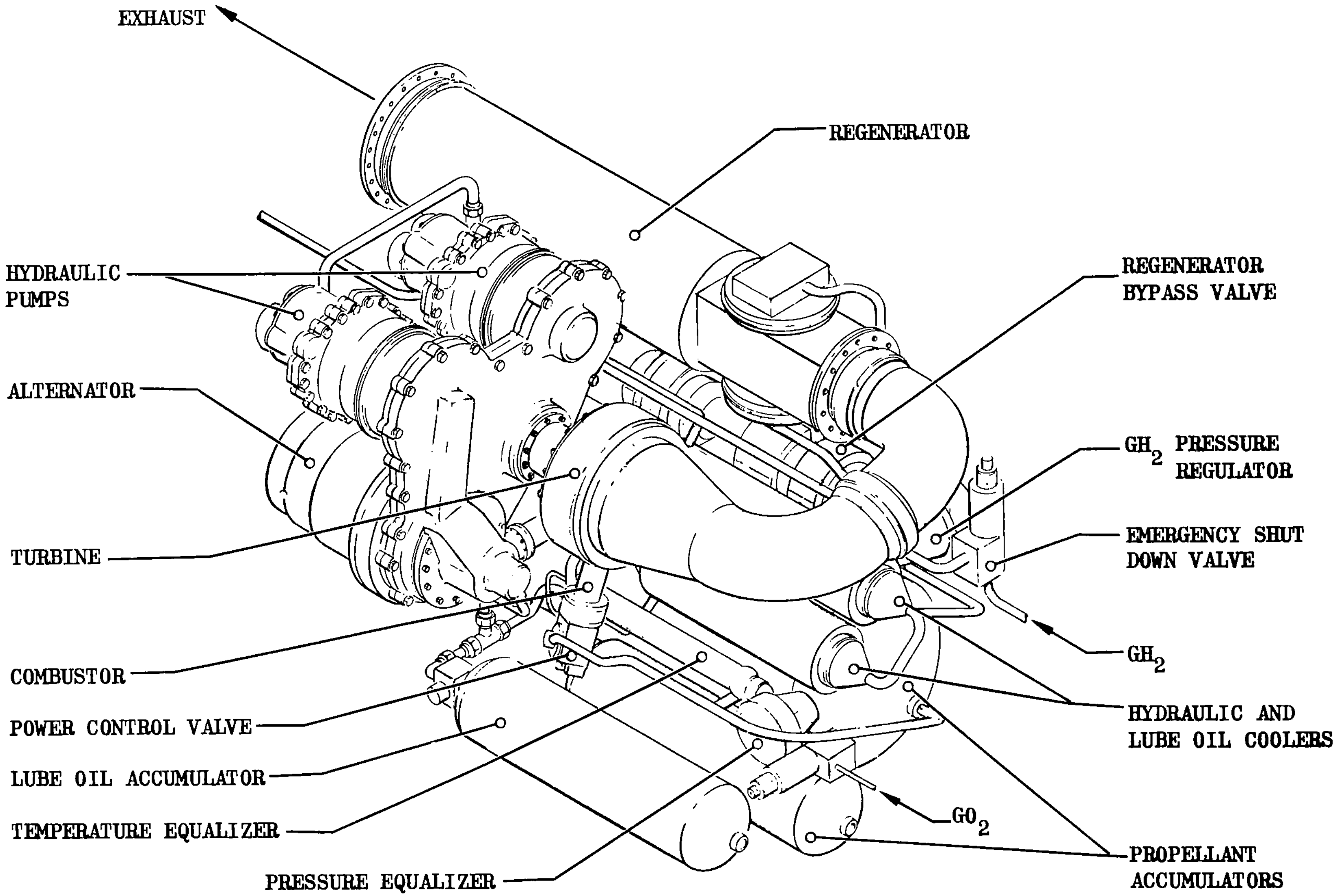


Figure 4. APU System Pulse Modulated

TABLE 1

SS/APU SYSTEM/MODEL SPECIFICATION SUMMARY

• POWER OUTPUT

400 HP (PEAK GEARBOX SEA LEVEL)
33 HP (IDLE GEARBOX SEA LEVEL)

• HYDRAULIC COOLING 100 PERCENT

• PERFORMANCE

SPECIFIC PROPELLANT CONSUMPTION, LB/HP-HR

	SEA LEVEL	ALTITUDE
PEAK	2.04	1.80
IDLE	2.90	2.35

SPECIFIC POWER 1.72 HP/LB OF APU

• ENVIRONMENT

PRESSURE 0 TO 14.7 PSIA
TEMPERATURE -65 TO 300 F

• CONTROL

TYPE: PULSE MODULATED
RESPONSE: 75 MS, 0 TO 100% POWER
STARTUP TIME: 1.3 SEC
FULL POWER: 2.0 SEC

• DESIGN

TURBINE: 60,000 RPM ±2-1/2%, 5% MAXIMUM
GEAR REDUCTION: 10 TO 1 (HYDRAULIC)
5 TO 1 (ELECTRICAL)
WEIGHT: 233 LB (LESS PUMPS AND ALTERNATORS)
VOLUME: 17 CUBIC FEET
SIZE: 27" x 25" x 43"

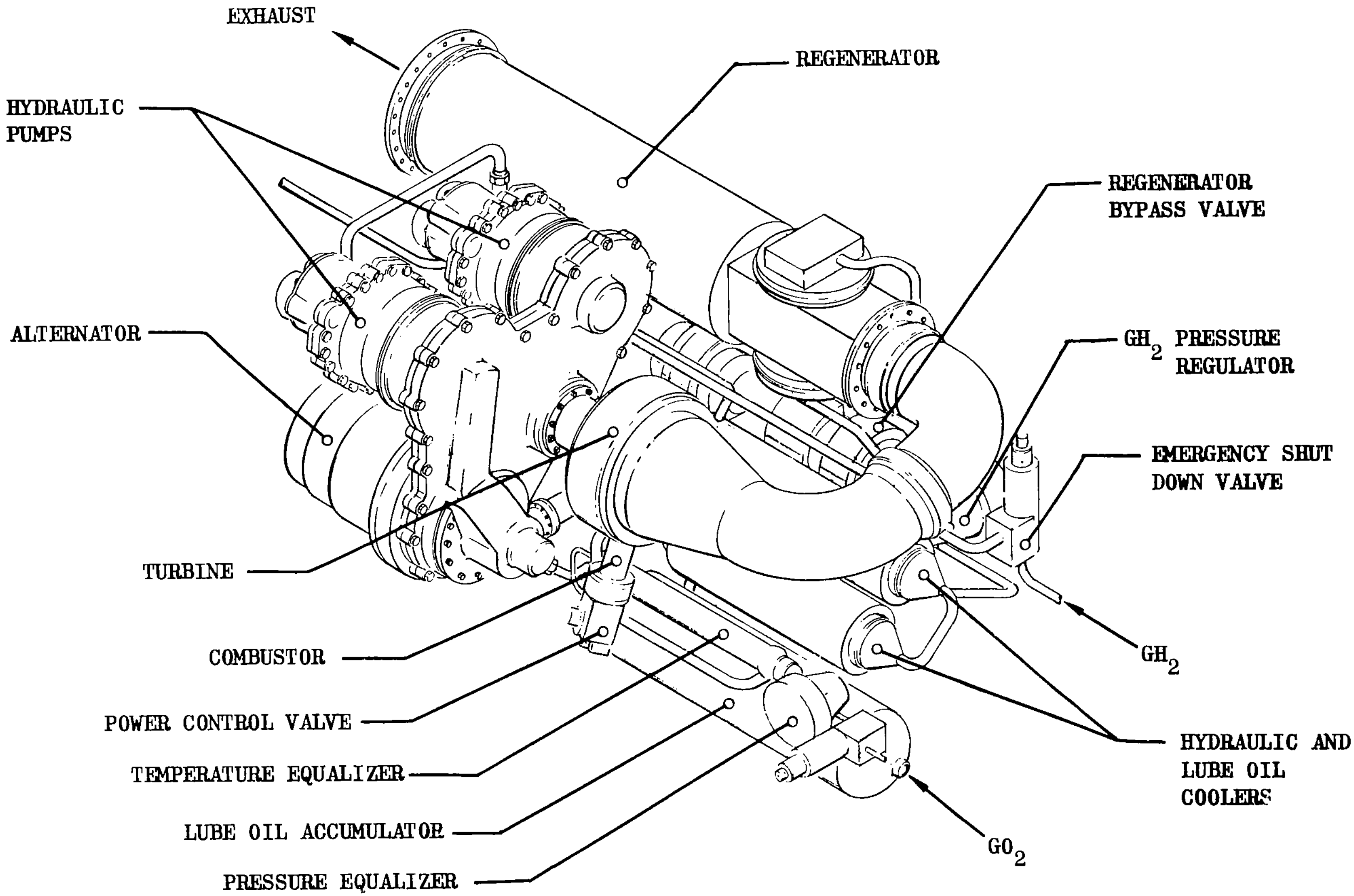


Figure 5. APU System Pressure Modulated

page 8 is packaged in a similar manner to the pulse modulated system but has the accumulators deleted.

The operational range of the system is illustrated in Fig. 6 on page 1 . The system has been demonstrated to operate over this entire operating range. The nominal conditions of 33 horsepower correspond to a booster idle condition. The necessary hydraulic cooling can be provided if hydrogen is supplied to the APU at 125 R for the orbiter vehicle and 260 R for the booster vehicle. The baseline propellant supply system as specified in the Phase II NASA specification is the Integrated ACS propellant supply system where the ACS propellants are supplied to the APU under the conditions as shown in Fig. 6 , of from 1000 to 500 psi pressure and from 75 R to 400 R on the hydrogen and 300 R to 400 R on the oxygen. A typical simulated system operation as run on the computer model is shown in Fig. 7 (on page 11). Power transients were imposed on the system with a 75 millisecond time constant from the 33 horsepower idle condition to the full peak 400 horsepower level. For both the pulse and pressure modulating power control system, speed control is maintained within ±3.5 percent which is well within the 5 percent specification. All the other important parameters in the system such as turbine inlet temperature, lube oil temperature and hydraulic oil temperature were maintained well within tolerance during these power spikes. The performance and weight of the baseline systems are shown in Fig. 8 (on page 12). SPC as a function of gearbox output power is shown for both the pressure and pulse modulated system. Gearbox power is that power actually delivered out of the gearbox to the hydraulic pumps and alternator. Weights for the APU hardware, the burned propellant and the associated propellant supply penalty are also shown on Fig. 8. The propellant supply penalty consists of the total weight of supplying the necessary propellant to the APU and is associated with conditioning and pumping up the propellant for storage in the ACS accumulators prior to usage by the APU. Also included in that weight penalty is the necessary supplemental hydrogen required to provide hydraulic cooling in the orbiter vehicle.

A detailed study was conducted to integrate the APU into the typical orbiter and booster vehicle. Information was provided from each major orbiter and vehicle prime contractor, and studies were performed on hydraulic cooling, the APU accessibility and installation, and propellant supply to the APU. Fig. 9 illustrates the APU mounted in a typical vehicle under study. Four APUs are mounted, two on each side of the vehicle. Provisions have been made for propellant supply and exhaust ducting. Emphasis has been placed on accessibility of the line replaceable items. The hydraulic pumps and alternator are on the opposite side of the gear box from the turbine assembly and are easily accessible for servicing and removal. Accessibility to such items as the spark ignition system, combustor valve assembly, and the primary control elements has been provided.

The studies conducted have resulted in the identification of several major technology areas which are illustrated in Fig. 10. It is recommended that technology programs be initiated in each of these major technology areas. Of particular importance, in the turbopower unit, is the combustor assembly. For the pulse modulating system, repeatable ignition is a primary requisite to

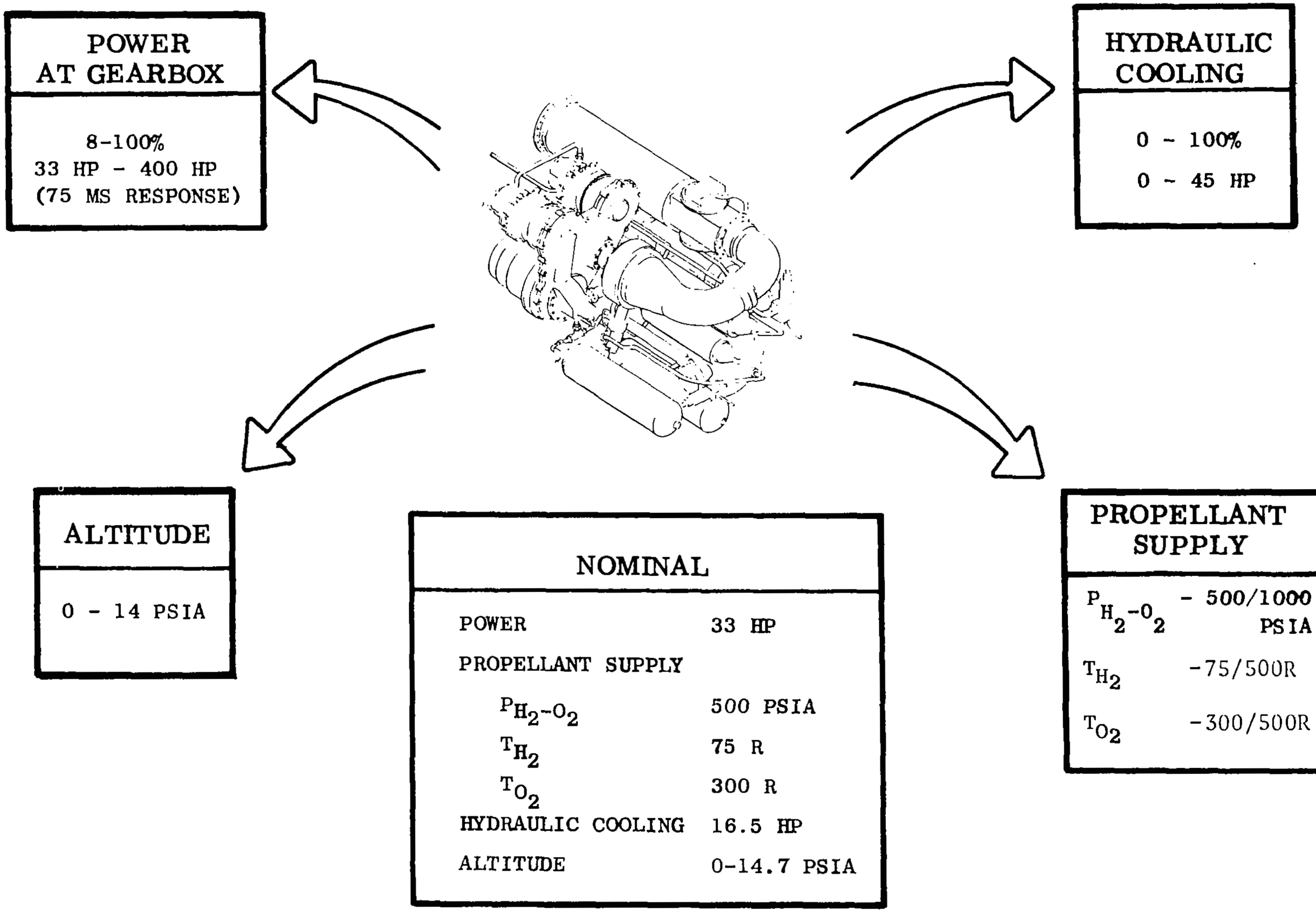


Figure 6 . Operational Range

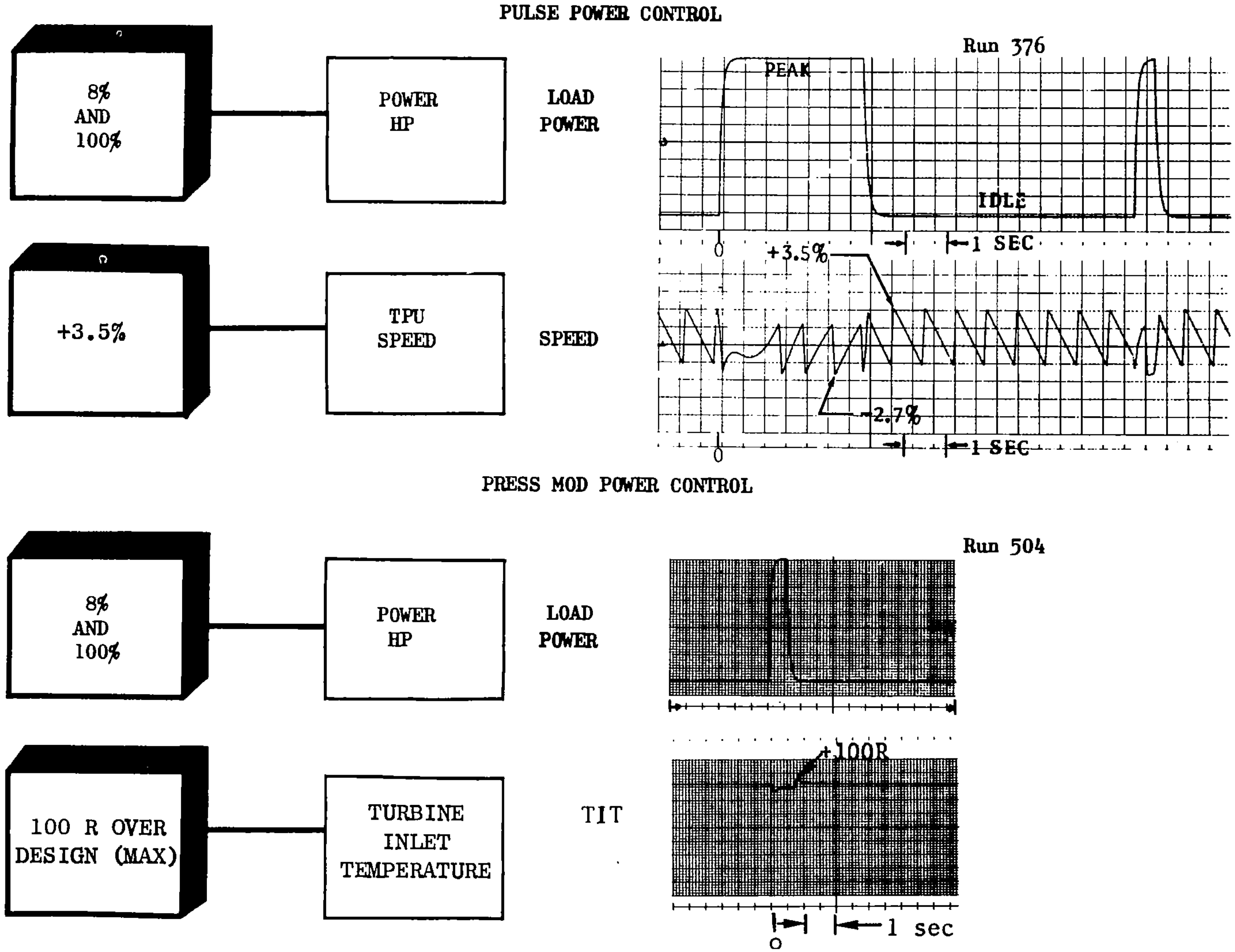


Figure 7. Simulated System Operation

SYSTEM WEIGHT*

INTEGRATED ACS PROPELLANT FEED

*Based on power-time profile furnished by NASA and shown in Appendix , Fig.

	BOOSTER		ORBITER	
	PRESSURE	PULSE	PRESSURE	PULSE
APU HARDWARE*	214	233	214	233
BURNED PROPELLANT	468	322	372	306
PROPELLANT SUPPLY PENALTY	222	153	242**	199**
TOTAL	904	708	828	738

* LESS HYDRAULIC PUMPS AND ALTERNATOR

** INCLUDING SUPPLEMENTAL H_2 FOR HYDRAULIC COOLING

SYSTEM SPC (GEARBOX)
(10 PSIA)

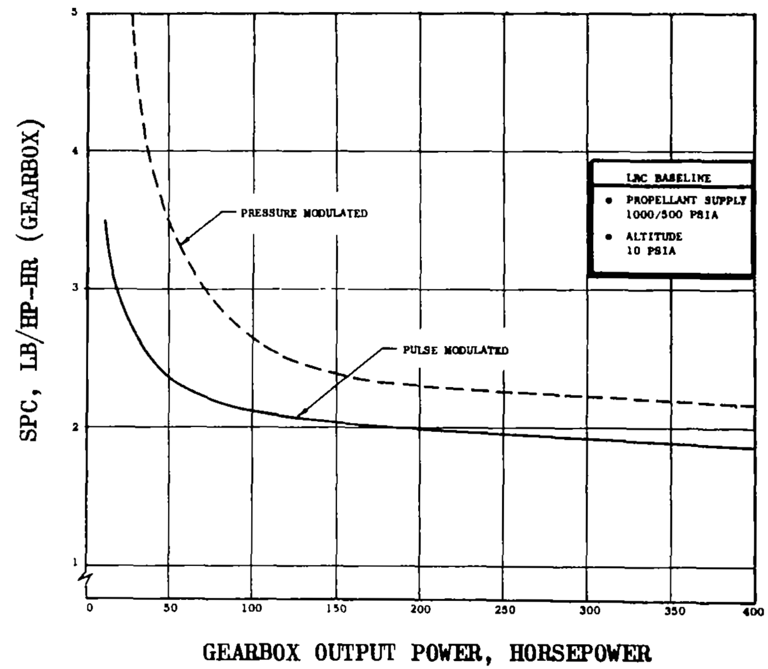
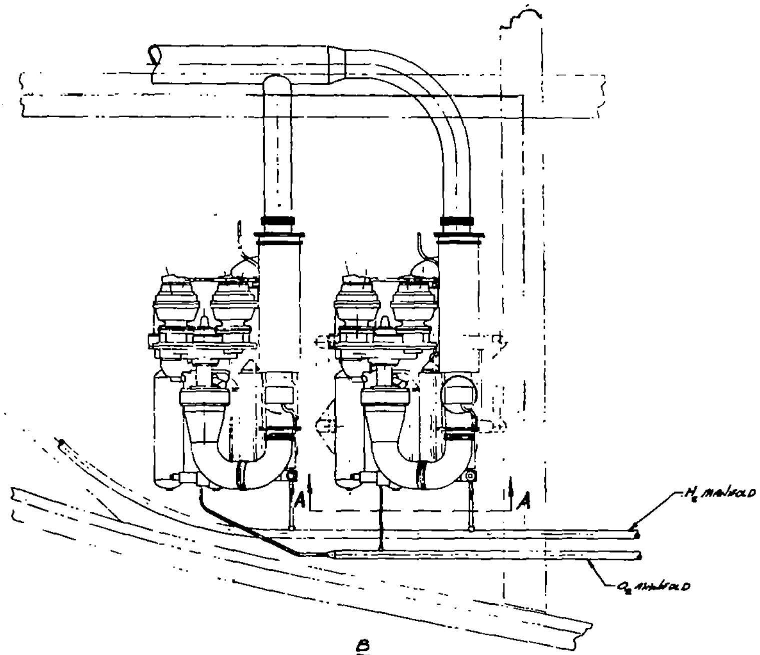


Figure 8. System Weight/Performance



- HYDRAULIC COOLING
- ACCESSIBILITY
- INSTALLATION
- PROPELLANT SUPPLY

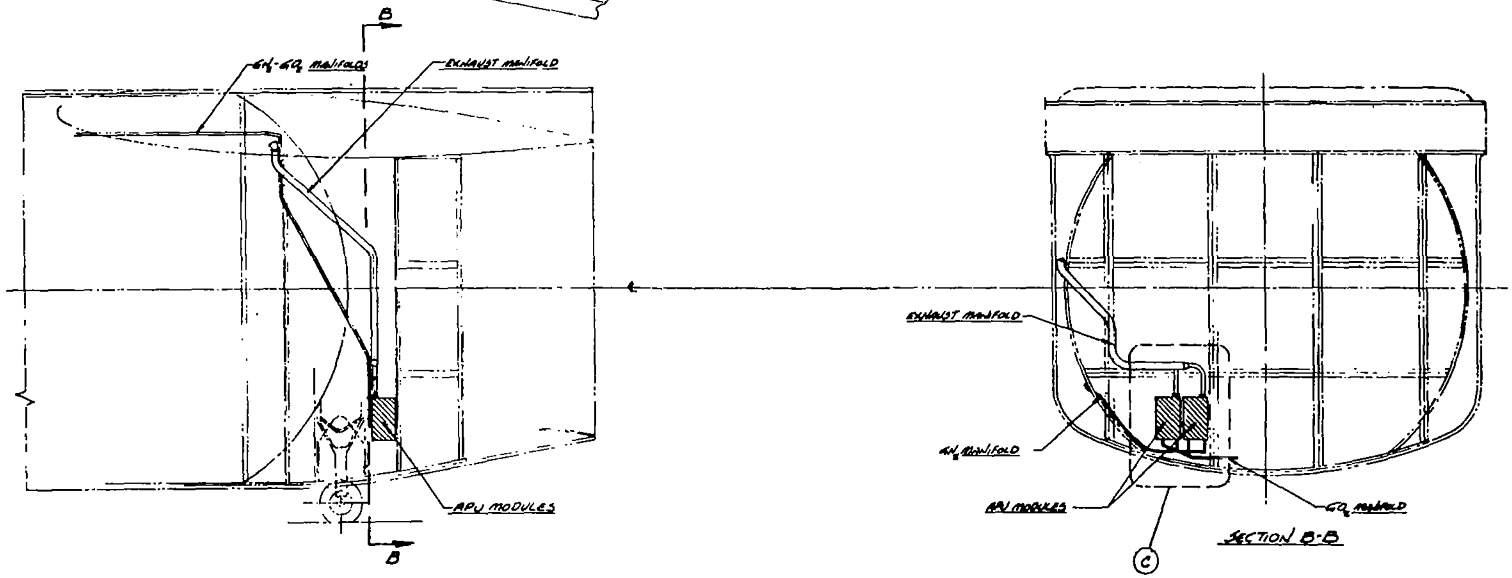


Figure 9. SS/APU Vehicle Integration

TURBO POWER UNIT

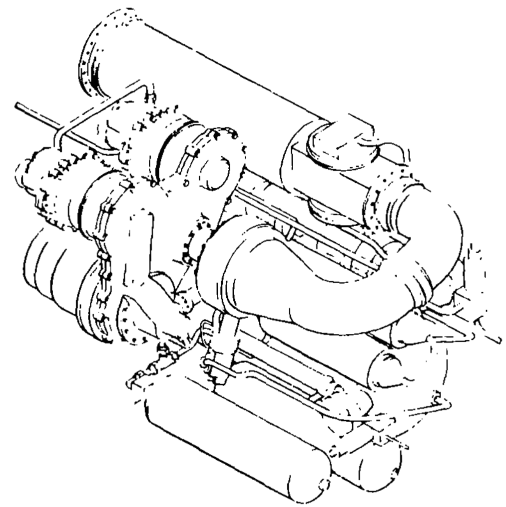
COMBUSTOR - IGNITION

COMBUSTION
(PRESS MOD)

TURBINE - STAGE TRANSITION

OFF DESIGN
(PRESS MOD)

H_2 EMBRITTLEMENT



PROPELLANT CONDITIONING

HEAT EXCHANGER - FREEZING

SAFETY

PROPELLANT CONTROL

TURBINE INLET TEMP CONTROL

POWER CONTROL

VALVE LIFE -
(PULSE CONTROL)

Figure 10. APU Major Technology Areas

to avoid the possibility of build up of unburned oxidizer in the exhaust ducting. For the pressure modulating system, performance of the combustor at part load operating conditions is vital to the achieving of acceptable specific propellant consumption. In the supersonic turbine transition between the first and second stage with minimum loss must be demonstrated. The off design performance of the pressure modulated supersonic turbine has a substantial effect on the performance of the system and a detailed experimental investigation must be carried out. Hydrogen embrittlement on the turbine for disc material selected (Astroloy) can present a problem and a detailed investigation on possible stress relieving techniques and/or on coatings should be initiated. In the power control system, the life of the linked bipropellant valve for the pulse control system must be improved and demonstrated while for the pressure modulating system, the criticality of the matching of the valve areas during part load operation requires verification. For the propellant conditioning system various aspects of the heat exchanger, including operating safety with regard to isolation of propellants from hydraulic and lubricating oil, and propellants from each other, must be demonstrated. The potential of freezing of water vapor in the exhaust of the turbine as it passes through the regenerator, heating the incoming hydrogen, requires a thorough experimental evaluation. In addition, the general problem of propellant control under the various vehicle operating conditions requires further evaluation and study.

This report presents a comprehensive review of all aspects of system design and performance as summarized above. The report has been organized on a system and subsystem basis as follows:

Section III	Performance and Design (System)
Section IV	Vehicle Integration (System)
Section V	System Operation
Section VI	Failure Mode Analysis (System)
Section VII	Turbopower Unit (Subsystem)
Section VIII	Propellant Conditioning (Subsystem)
Section IX	Control Elements (Subsystem)

Each of these sections provides details of all analysis, optimizations and design of the components, subsystems and system, and emphasizes the specific primary criteria features as discussed in this summary section.

II INTRODUCTION

The Space Shuttle APU program consisted of two phases as illustrated in Figure 11, page 18. Phase I of the program was directed towards synthesis of various candidate systems and selection of the most promising system for application to the Space Shuttle Vehicle. This work was completed and is reported in the Phase I summary report, #TMR0115-3137, December 1970. Following selection of the baseline system, a detailed study was conducted in Phase II which resulted in a preliminary design of the selected baseline system. This baseline system is described in detail in this report. The design of the system was reviewed with NASA and with the Space Shuttle vehicle contractors during the Phase II portion of the program and formal Phase II reviews were conducted midway and at the end of the Phase II effort.

The preliminary design of the APU reflects all operational and pertinent non-operational criteria including 1) flexibility to accommodate mission changes 2) adaptability to vehicle configurations, 3) safe operation over a broad range of mission conditions, 4) forgiving design characteristics tolerant to a wide latitude of conditions even beyond the basic design specifications, 5) minimum complexity and high reliability, 6) low development risk and cost through conservative design, 7) high performance across the power ranges, and 8) maximum use of existing state-of-the-art technology wherever possible.

The Phase II design analysis utilized a sophisticated system to model the design of the integrated control system, to check detailed component and subsystem analyses against system requirements and to confirm system performance across the complete spectrum of mission operating conditions. The model further provided a quantitative basis for the failure modes and effects analysis fundamental to the establishment of a safe reliable preliminary design.

The Space Shuttle APU study was conducted with Mr. S. Domokos as Program Manager and, with Mr. Robert Seigler as Principal Engineer. Mr. R. L. Binsley was responsible for the Turbopower Unit design, Mr. A. A. Krause responsible for Control Element design, Mr. J. P. Maddox responsible for System Performance and Propellant Conditioning System design, and Mr. R. D. Marcy responsible for the System Operation and Failure Mode analysis.

- DESIGN/ANALYSIS STUDY
    - PHASE I - SYSTEM SYNTHESIS AND SELECTION
    - PHASE II - PRELIMINARY DESIGN OF SELECTED SYSTEM

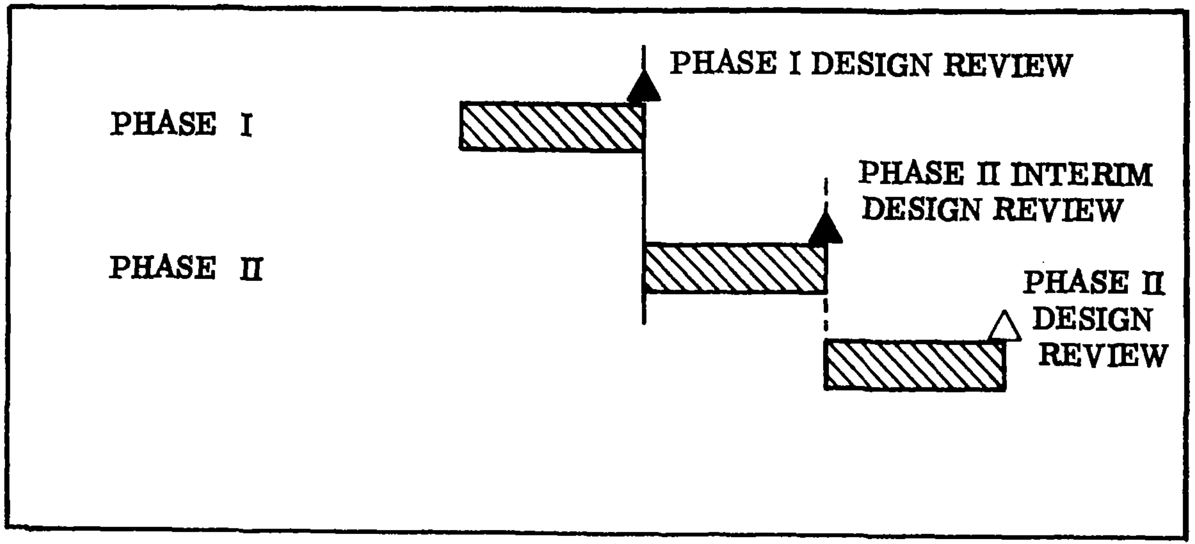


Figure 11. SS/APU Program

III APU PERFORMANCE/DESIGN

A. PERFORMANCE

The various factors considered in determining APU performance are summarized in Table 2 . The considerations include the power, altitude and duration profile of the vehicle, the turbine design and off-design performance, the power control type (pulse or pressure modulation), gear box losses at design and off-design conditions and exhaust duct effects. These factors are all provided as inputs into the digital program which calculates the burned propellant requirements.

The program used for analysis with a pressure modulated power control (SSAPU3) also includes performance degradation due to combustor off-design operation. A C-star efficiency of 98 percent is used at the design point with a degradation characteristic resulting in 95 percent at the minimum power condition. The program for analysis with a pulse power control (SSAPU4) doesn't include the degradation since the combustor is always operating at its design point. However, there is another degradation factor, namely, turbine windage occurring during the "off pulse", which is associated only with pulse power control. The program uses a loss of 5.6 horsepower at a turbine exit pressure of 10 psia, which is assumed to vary linearly with back pressure. To be somewhat conservative the turbine back pressure during the "off pulse" is taken as the average of the ambient pressure and exit pressure during the "on pulse".

An additional degradation factor common only to the pulse system is that due to degradation during "short on times". The program assumes a valve opening time of 8 milliseconds with a 16 millisecond closing time. Based upon test results with a nitrogen driven supersonic turbine, estimates of the energy utilization during the pressure rise and decay portions of the total on pulse were obtained. A minimum on time of 102 milliseconds occurs during the nominal operating condition for the baseline pulse system, resulting in a 7.0-percent spc degradation, or 10 pounds of propellant over this portion of the power-time flight profile.

Figure 12 presents the effect of the turbine design point selection on the burned propellant weight. Typical turbine specific power consumption (SPC) horsepower characteristics for low pressure ratio and high pressure ratio machines with a pressure modulated control are shown. If the major portion of an operating profile consists of low power operation, it can be seen in this characteristic that the low pressure ratio machine is favored. However, moderate to high power operation over a significant portion of the mission profile would favor the higher pressure ratio design.

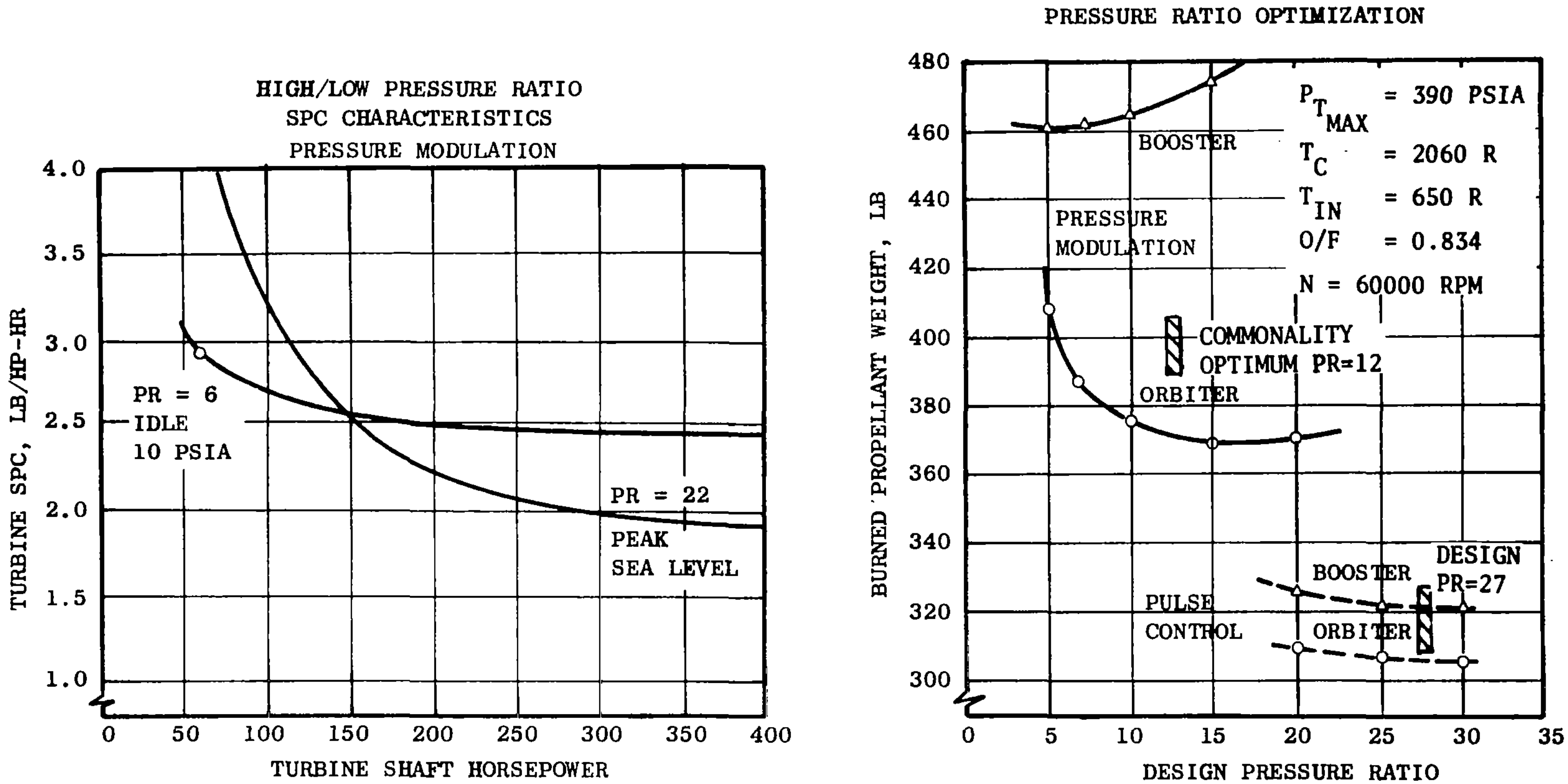
The results of the turbine design point optimization are presented in Fig. **12(b)** as a function of the design pressure ratio. All the machines evaluated provide maximum power at sea level with a maximum turbine inlet pressure of 390 psi, which provides a 5 percent power margin for controllability.

The propellant weight for the pulse control machine is relatively insensitive to design pressure ratio over the range considered. The optimum design point selected for the baseline machine is 26.9 resulting in 322 pounds of propellant for the booster and 306 pounds of propellant for the orbiter. The pressure modulated machine optimizes at a pressure ratio of about 15 for the orbiter

TABLE 2

PERFORMANCE CONSIDERATIONS

- POWER, ALTITUDE, DURATION
- TURBINE DESIGN/OFF-DESIGN
- POWER CONTROL
    - PULSE/PRESSURE MODULATION
- GEARBOX DESIGN/OFF DESIGN
- EXHAUST DUCT EFFECT



(a) (b)

Figure 12. Effect of Turbine Design Point on Burned Propellant Weight

and 7 for the booster. The commonality pressure ratio which minimizes the propellant weight corresponds to about 12, with the same machine operating in both the orbiter and the booster. The difference between using optimum APU's and the commonality optimum is only 10 pounds of propellant.

Figure 13 presents the performance of the optimum pulse and pressure modulated machines in terms of SPC versus shaft power referenced to the turbine. Three different pressure altitudes are presented - 0, 10, and 14.7 psia. The power ranges from a booster idle to orbiter peak of 33 horsepower to 400 horsepower out of the gearbox. The major difference in SPC occurs at the low power level where maximum throttling of the pressure modulated machine is required. At the booster idle point the SPC at 10 psi for the pulse machine is approximately 2.2 versus 3.7 for the pressure machine. This represents a penalty of about 60 percent for operation at idle power level. The performance difference at zero psia back pressure is considerably less since the pressure modulated machine is operating very close to its design pressure ratio even under a throttled condition. The zero psia performance is indicated at a 50 horsepower idle condition (orbiter re-entry) with a resultant penalty of 15 percent for the pressure modulated machine. The SPC differences at moderate to high power levels decrases further, resulting in a 10 percent penalty at the 400 horsepower sea level condition.

The same performance data referenced to the gearbox output power is shown in Fig. 14. Specific propellant consumption versus gearbox power are expressed over the same range from booster idle to orbiter peak power for the same ambient pressures of 0, 10, 14.7 psia. The details of the selected turbine design, pressure ratio, chamber pressure design and horsepower, are presented in the figure along with the combustion design point characteristics.

Gearbox efficiency ranges from 97.5 percent at the peak power condition to 84 percent at the booster idle condition, so that the gearbox referenced SPC is 2.5 percent higher than turbine SPC at peak and 20 percent higher at idle.

B. APU SYSTEM DESCRIPTION

The power control selected for the baseline APU is pulse modulated with the ability to respond from an idle to 100 percent step change in power within 75 milliseconds. The turbine is a 2-stage pressure compounded type operating at 60,000 rpm with a normal speed band for pulse control of ± 2.5 percent.

The APU produces a peak gearbox output requirement at sea level of 400 horsepower with a 5 percent power excess for control. The nominal sustained idle operating condition of 33 horsepower results in a power turndown ratio of 12:1. One hundred percent of the hydraulic cooling is accommodated at all power levels with 75 R hydrogen source temperature. The specific propellant consumptions are indicated for sea level and altitude and for zero psia back pressure operating conditions at the peak and idle levels. The specific power of 1.72 horsepower per pound of APU is also indicated.

The APU meets all power requirements over an ambient back pressure range of zero to 14.7 psia and an environmental temperature range of from -65 to 300 F.

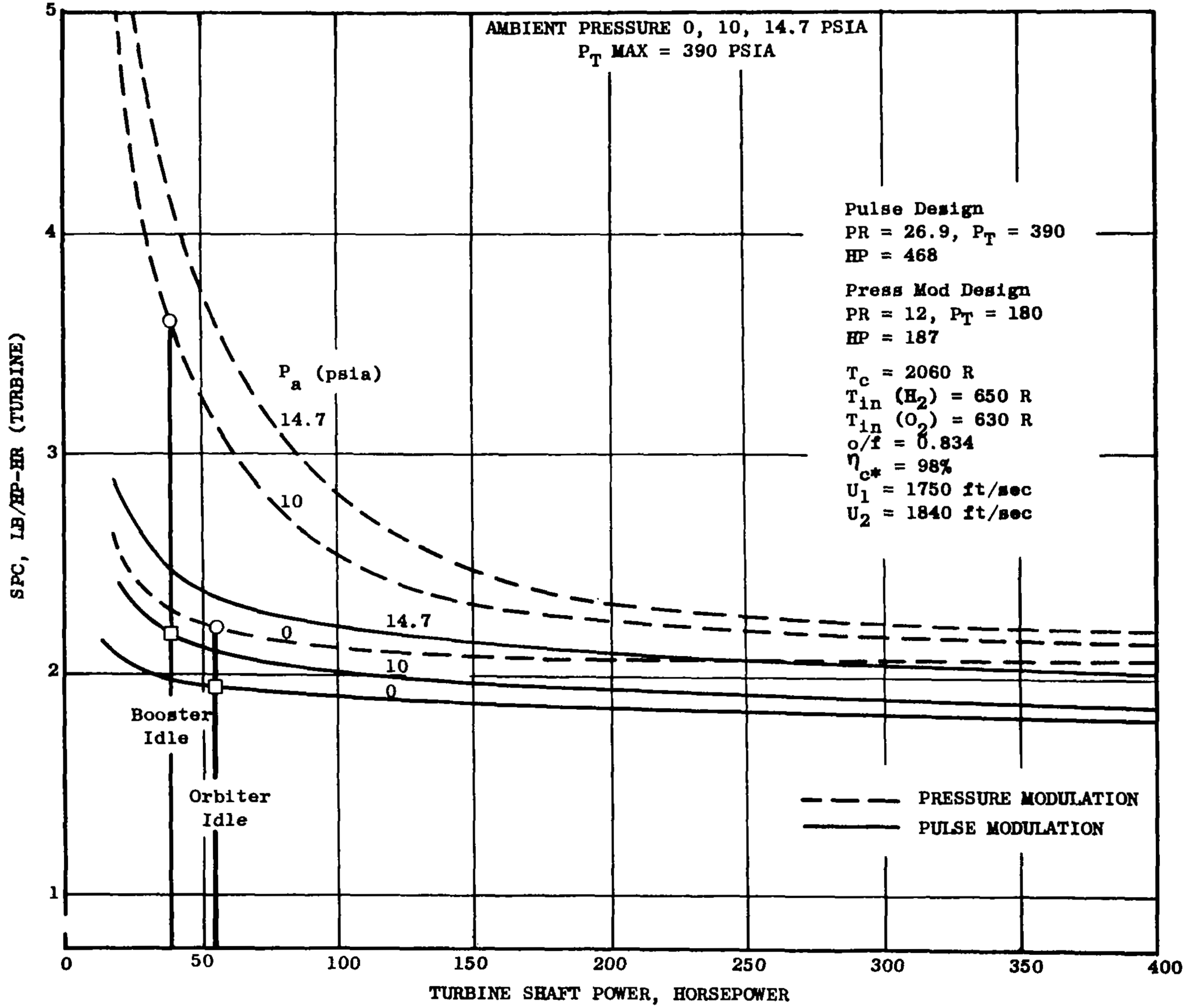


Figure 13. Specific Propellant Consumption (Turbine) Pulse and Pressure Modulation

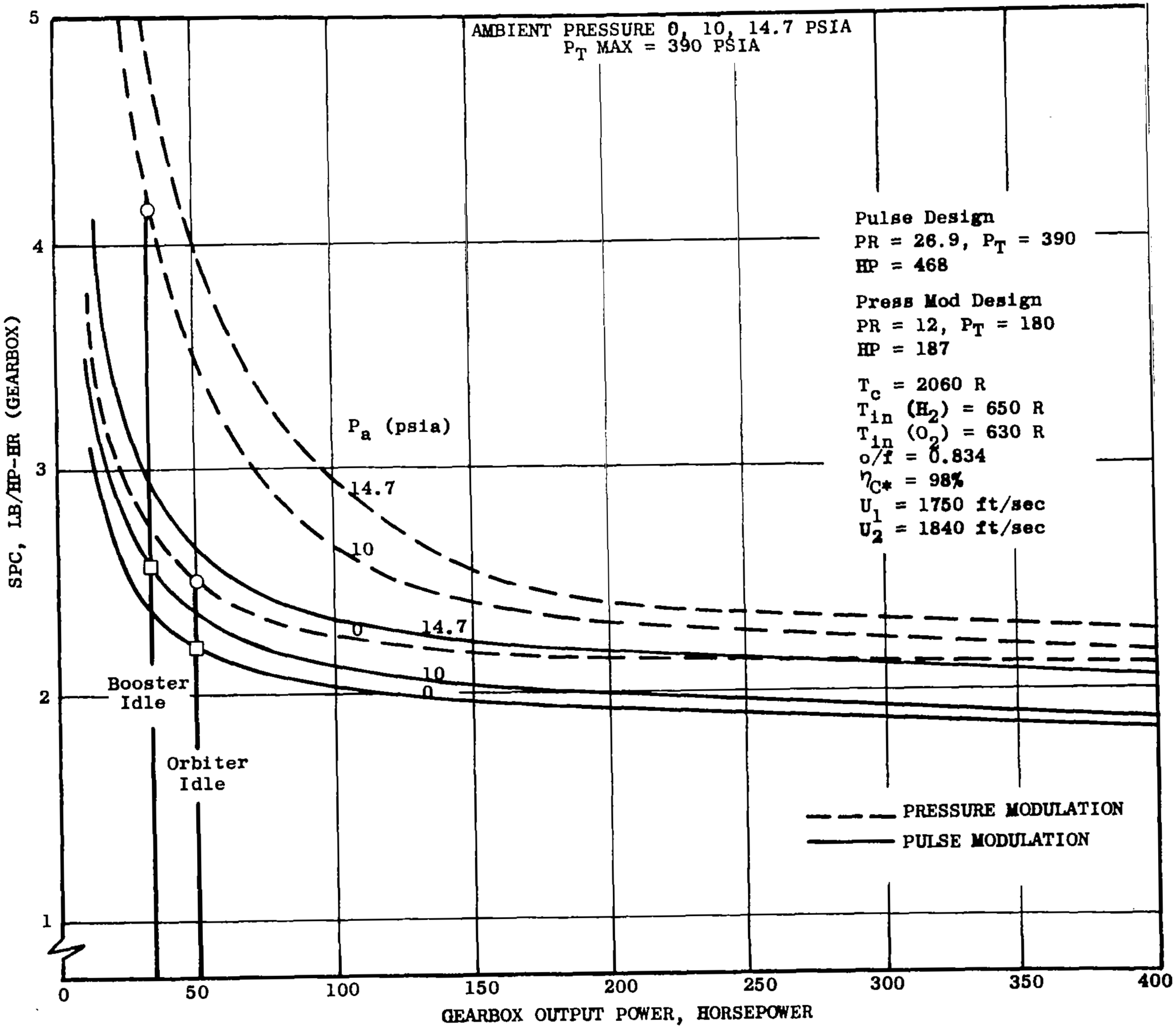


Figure 14. Specific Propellant Consumption (Gearbox) Pulse and Pressure Modulation

Figure 15 presents the APU power flow for the maximum and minimum required operating levels indicating efficiency of the various components. With 410.6 horsepower into the gearbox, 400 horsepower output is obtained and, with a hydraulic pump efficiency of 90 percent, 360 horsepower net delivered hydraulic power is achieved. This power corresponds to 212 gpm for the 3,000 psi hydraulic system. For this operating condition there is no auxiliary electrical output required for the orbiter vehicle. Sustained idle conditions occur in the booster during the cruise phase of the profile at 10 psia where 39 horsepower into the gearbox results in 2.24 KW of delivered electrical power and 3 horsepower of hydraulic power.

The nominal operating condition of the APU at the 33 horsepower gearbox output and 10 psia back pressure are presented in Fig. 16 . The source pressure for the hydrogen and oxygen is 500 psia with the oxygen inlet temperature at 300 R and a flow of 0.011 pounds per second. The hydrogen flow is 0.0129 pounds per second (a mixture ratio of 0.835).and the inlet temperature is 75 R. The hydrogen pressure regulator reduces pressure down to 461 psia at the inlet to the regenerator bypass valve. Eighty-two percent of the hydrogen flow is bypassed around the regenerator to provide a mixed temperature of 288 R with the ramaining flow which is passed through the regenerator and heated by the turbine exhaust gas. The hydrogen then enters the hydraulic cooler where it undergces a temperature rise to 609 R and then into the lube cooler with a further temperature rise to 672 R. In flowing through the temperature equalizer, the hydrogen drops from the 672 R to 650 R in heating up the oxygen from 300 R to approximately 668 R so that the oxygen is within 20 degrees of the hydrogen temperature.

The propellants then flow into the combustor where they are ignited resulting in a 2060 R combustion temperature at a pressure of 405 psia. The valve on-time fraction is approximately 10 percent (i.e., a pulse-on time of .102 seconds and an off-time of close to 1 second). The SPC (gearbox) associated with this operating condition is 2.5 lb/hp-hr.

Figure 17 presents the baseline APU package with the envelope dimensions indicated along with various components identified. The package is approximately 2 ft square by 3.5 ft long with an APU weight of 233 lb. This excludes the hydraulic pumps and alternator which add another 93 lb to the weight. The package is compact with efficient volume utilization allowing for clear accessibility to all components for inspection, maintenance, and replacement as required.

Table 3 presents the results of the APU hardware weight study exclusive of the weight of the hydraulic pumps and alternator. Both pressure and pulse modulated control systems are presented. The primary weight difference between the two systems is due to the absence of pulse attenuator tanks for the pressure modulated APU. The resultant weights are 233 pounds for the pulse system and 214 pounds for the pressure modulated system. The burned propellant weight and propellant penalty for ACS integration (turbopump and conditioner gas generator requirement and supplemental 300 R hydrogen for cooling) is shown for both the booster and orbiter resulting in the total system weight indicated in the figure. Use of pressure modulation results in a 28 percent penalty for the booster and 12 percent penalty for the orbiter.

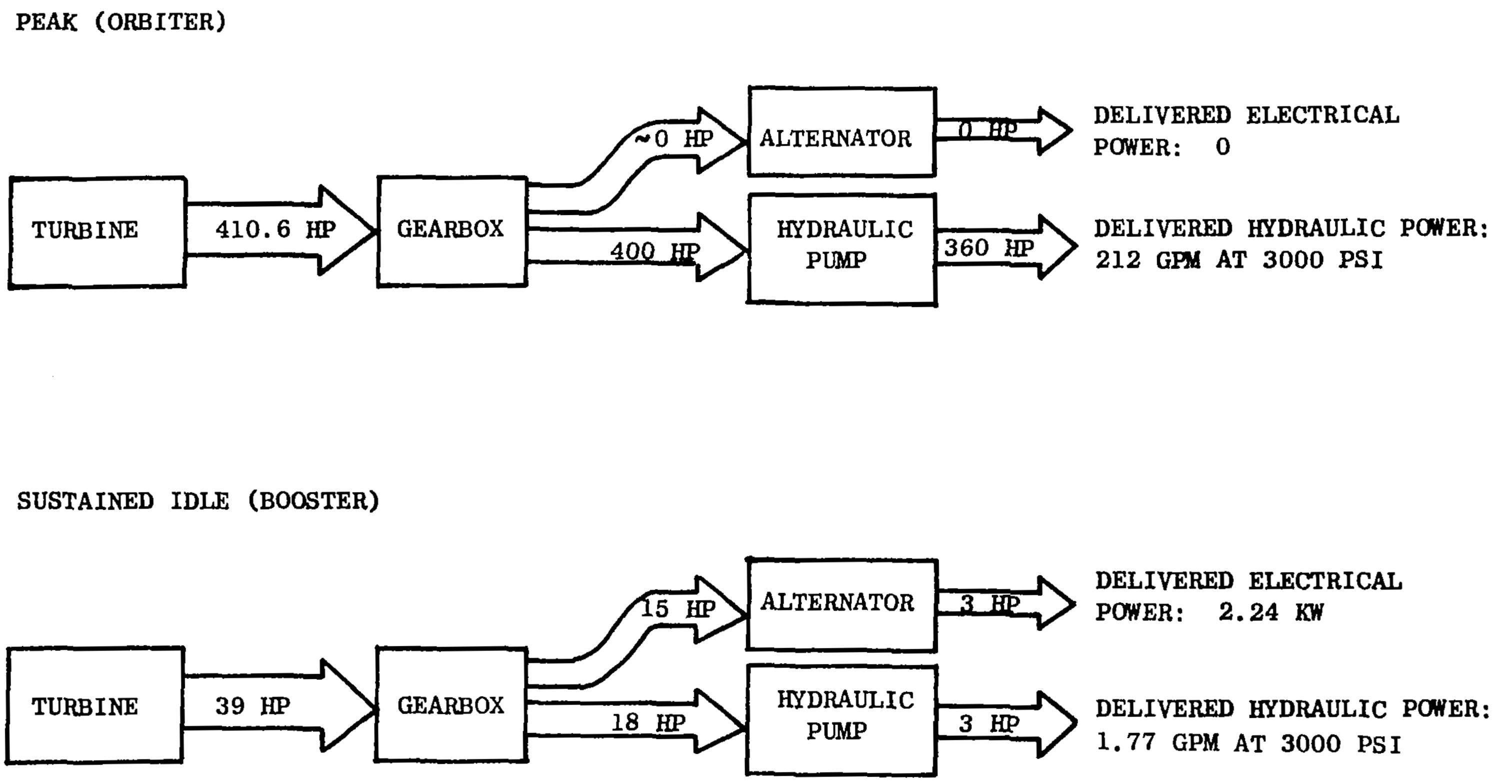


Figure 15. Typical APU Power Flow

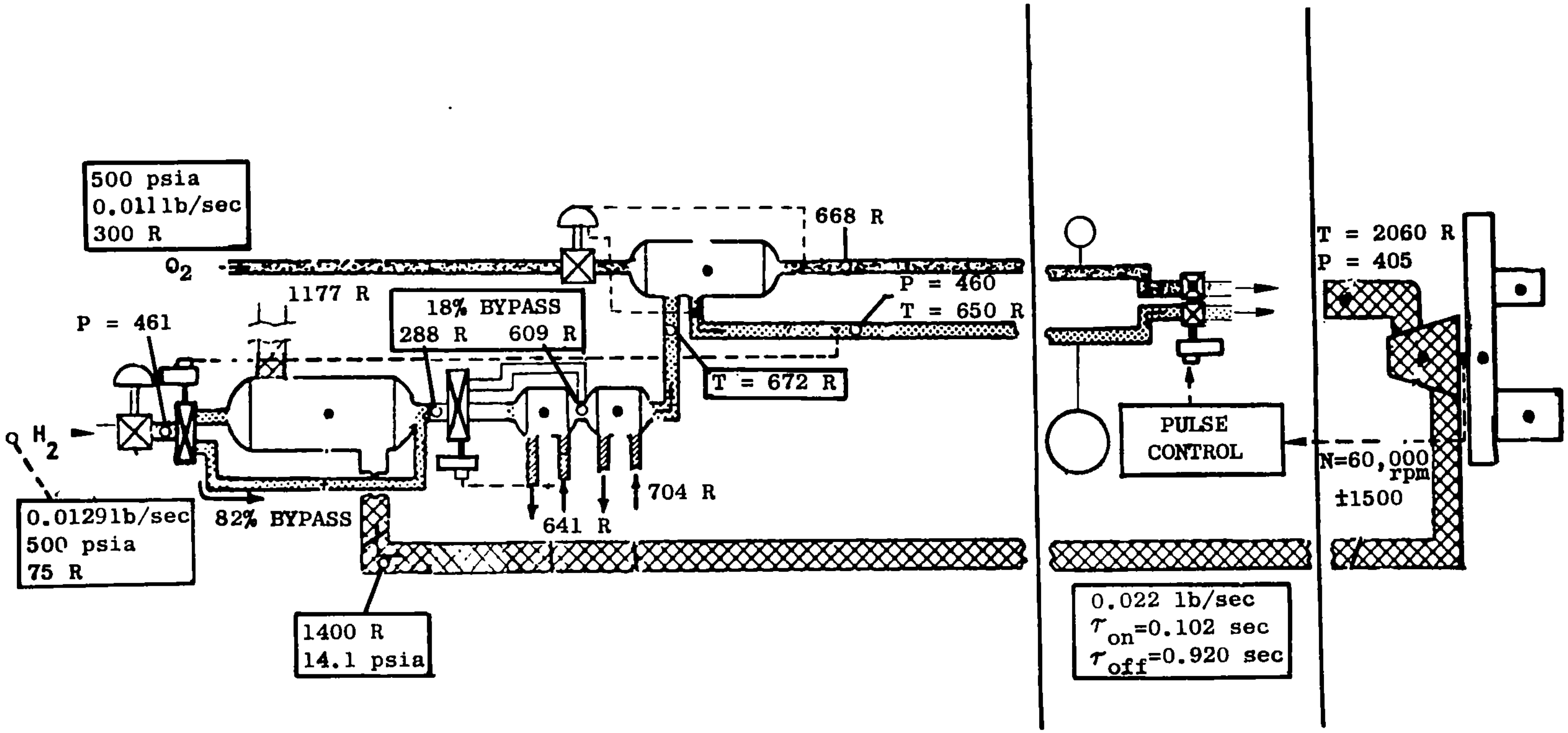


Figure 16. Nominal Operating Condition
33 HP at Gearbox/10 psia

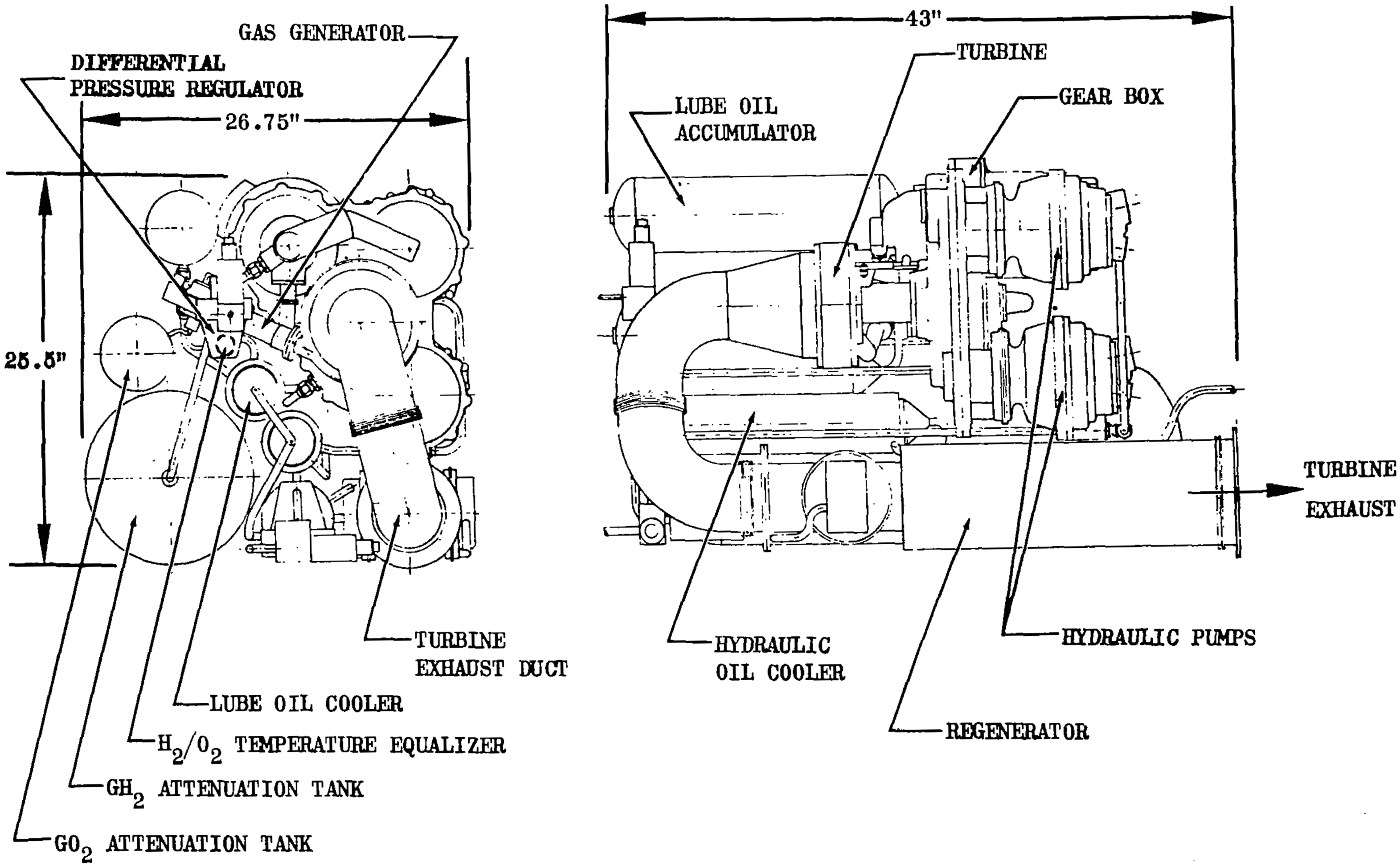


Figure 17. Baseline APU Package

TABLE 3

APU - SYSTEM WEIGHT SUMMARY (BASELINE)

COMPONENT	SYSTEM: PULSE MODULATED		SYSTEM: PRESSURE MODULATED	
TURBOPOWER UNIT				
• TURBINE AND GEARBOX ASSEMBLY	55		55	
• GAS GENERATOR AND CONTROL VALVE	10		12	
• CONTAINMENT	15		15	
PROPELLANT CONDITIONING				
• PRESSURE REGULATORS (2)	16		16	
• BYPASS VALVES (2)	17		17	
• REGENERATOR	26		26	
• HYDRAULIC COOLER	10		10	
• LUBE OIL COOLER	8		8	
• TEMPERATURE EQUALIZER	1		1	
• ATTENUATION TANKS	17		0	
LINES, DUCTS, VALVES, L.O. ACCUMULATOR	24		21	
INSTRUMENTATION/CONTROLS	12		12	
STRUCTURAL SUPPORTS	23		21	
APU HARDWARE WEIGHT	233		214	
	BOOSTER	ORBITER	BOOSTER	ORBITER
BURNED PROPELLANT	322	306	468	372
PROPELLANT SUPPLY PENALTY (INTEGRATED ACS)	153	199	222	242
TOTAL	708	738	904	828

IV VEHICLE INTEGRATION

Rocketdyne has maintained close contact with vehicle contractors in order to assess the effects on APU performance of various integration concepts. Cognizance of vehicle requirements has been a key factor in design of an APU that is flexible and readily adaptable to a variety of propellant source systems and propellant state conditions. This section discusses the effects of these conditions on APU performance (SPC) and hydraulic cooling capability. Results of an installation study are also presented covering the areas of heat exchange with the vehicle, APU accessibility and inspection, and adaptability to an air breathing drive. These integration considerations are summarized in Table 4 .

A. PROPELLANT STATE CONDITIONS

Pressure

The propellant source pressure level determines the maximum turbine pressure ratio and strongly influences the optimum design point of the TPU and the performance advantage of a pulse power control system over a pressure modulated control. Figure 18 presents the SPC (turbine) variation with the minimum supply pressure from the vehicle or pump for a 50 horsepower gearbox output at 10 psia back pressure. At the 500 psia level (APS INTEGRATION - NASA BASE-LINE) the pressure modulated SPC results in a 40% penalty compared with pulse power control. The rate of decrease of SPC with increasing pressure for pressure modulation is about three times that for pulse control so that the penalty is reduced to 15 percent at 1200 psia. The digital program was used to determine the burned propellant requirement for the entire booster and orbiter mission at the baseline pressure level of 500 psia and an increased level providing 900 psia turbine inlet pressure. The results presented in Fig. 18 show a significant reduction in pressure modulation weight penalty at the higher pressure, providing a strong incentive for selection of the inherently more reliable pressure modulated type of power control.

Temperature

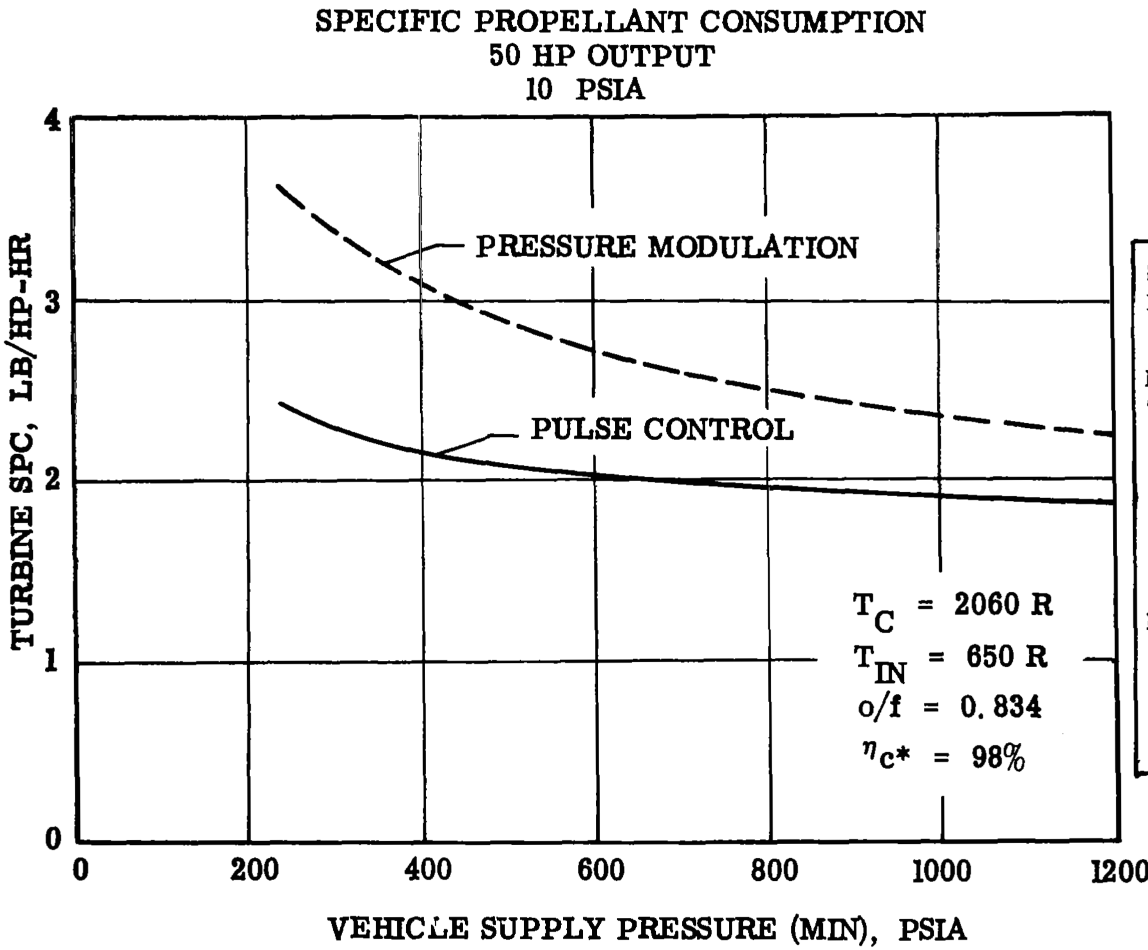
The flight operational maximum hydraulic fluid temperature is 750 R. A study was conducted to determine the hydrogen source temperature requirements to prevent over temperature of the hydraulic fluid based on heat loads associated with booster and orbiter power profiles. The hydrogen coolant flow was assumed equal to that required by the TPU. The cooler is located in the hydraulic pump case drain line so that it is receiving fluid from the maximum temperature point in the system. The fluid inventory was 321 pounds of MIL-H-5606 per NASA direction. Results of the study are presented in Table 5 The maximum heat load minimum hydrogen flow condition results on the orbiter vehicle during re-entry (the longest operating condition within the profile - 4,000 seconds). This condition requires a hydrogen source temperature of 125 R to prevent the hydraulic fluid from exceeding the 750 R maximum. The

TABLE 4

VEHICLE INTEGRATION CONSIDERATIONS

- PROPELLANT SOURCE
  - TEMPERATURE - EFFECT ON HYDRAULIC CODING
  - PRESSURE - EFFECT ON SPC
  - LIQUID/GAS
- SYSTEM SELECTION

  APU CHARGEABLE WEIGHTS
- INSTALLATION
  - HEAT EXCHANGE
  - ACCESSIBILITY/INSPECTION
  - ADAPTABILITY TO AIR BREATHING DRIVE



BURNED PROPELLANT LB PER APU

Supply Pressure (psia)	500		1150	
Maximum Turbine Inlet (psia)	390		900	
	B	O	B	O
Pressure Modulation	468	372	348	318
Pulse Control	322	306	306	290
ΔW	146	66	42	28

Figure 18. Effect of Vehicle Supply Pressure on APU Performance

TABLE 5

FLIGHT OPERATIONAL HYDRAULIC
COOLING REQUIREMENT

MAXIMUM H_2 SOURCE TEMPERATURE

- ORBITER 125R (RE-ENTRY)
- BOOSTER 260R (CRUISE)

Orbiter

- 321 LB MIL-H-5606
- CASE DRAIN EXIT 620R TO 750R in 4000 SEC (RE-ENTRY)
- BULK HYDRAULIC FLUID RISE, 540R to 670R
- 45 HP HEAT LOAD
- 6 GPM CASE DRAIN FLOW
- 584R H_2 INTO LUBE COOLER

Booster

- 321 LB MIL-H-5606
- CASE DRAIN EXIT 590R TO 750R IN 5700 SEC (CRUISE)

BULK HYDRAULIC FLUID RISE, 560R to 720R

- 16.5 HP HEAT LOAD
- 6 GPM CASE DRAIN FLOW
- 402R H_2 INTO LUBE COOLER

MAXIMUM HYDRAULIC FLUID
TEMPERATURE = 750R

temperature rise of the bulk fluid in the system is 540 R to 670 R within the 4,000 second period. The lube system heat load is 5 Btu/sec requiring 574 R hydrogen into the lube cooler. The hydrogen outlet is 672 R and it then enters the equalizer and drops to the control level of 650 R.

On the booster, the permissible hydrogen source temperature level is significantly higher because the hydraulic heat load is greatly reduced due to depressurization of the hydraulic pumps during booster cruise. This transient occurs for a 5700 second duration and because of the comparatively lower hydraulic heat load, the maximum hydrogen source temperature becomes 260 R. The resultant hydraulic fluid bulk temperature rise is 560 R to 720 R. The lube heat load is 13.5 Btu/sec requiring 402 R hydrogen into the lube cooler.

A study was also conducted to determine the requirement for steady-state cooling to prevent any temperature rise of the bulk fluid. The maximum fluid temperature of this system was held at 700 R, and the cooling requirements are shown in Table 6 . With the orbiter the resultant hydrogen source temperature is 63 R and 186 R for the booster. (Details of this study are presented in Appendix A).

It is evident from these results that for the APS baseline propellant source, which provides hydrogen at a nominal temperature of about 300 R, a cooling problem exists for the orbiter during re-entry. Supplementary hydrogen flow must be used and dumped representing a chargeable penalty to the APU. For transient cooling a flow increase of 64 percent is required assuming 300 R hydrogen representing a penalty of 41 pounds. For steady-state cooling an increase of 86 percent is needed resulting in a penalty of 54 pounds. Alternate systems providing hydrogen at an acceptable temperature level were evaluated and are discussed in the following section.

B. ALTERNATE HYDROGEN FEED SYSTEMS

The most obvious alternate system is to use a hydrogen pump supplied from a low pressure vehicle hydrogen tank to provide the required head and flow into the APU system. The pump would be an integral part of the PCS and would replace the pressure regulator located upstream of the regenerator bypass valve. The pump could be driven off the APU gearbox (must handle saturated liquid), by a hydraulic motor (close to tank), or an electric motor (in tank mounted). A reliable low specific speed pump would be required, which is beyond the present state of art thus incurring an additional development risk to the system compared with the APS baseline integrated system. An alternate approach is presented in Fig. 19 referred to as a refillable hydrogen system. The APU is still dependent upon operation of the APS turbopump, however, a small APU dedicated tank is periodically refilled to provide liquid hydrogen to the system on demand. The APU receives liquid hydrogen thus eliminating the orbiter cooling problem and also eliminates all of the conditioner gas generator propellant and some of the turbopump gas generator loss associated with the baseline APS integrated system. The operating cycle of the refillable system is best understood by referring to the T-S diagram in Fig. 19 . The pump is turned on and the APU tank is filled with liquid up to a pressure of 1400 psia at a temperature of 70 R

TABLE 6

STEADY-STATE HYDRAULIC COOLING REQUIREMENT

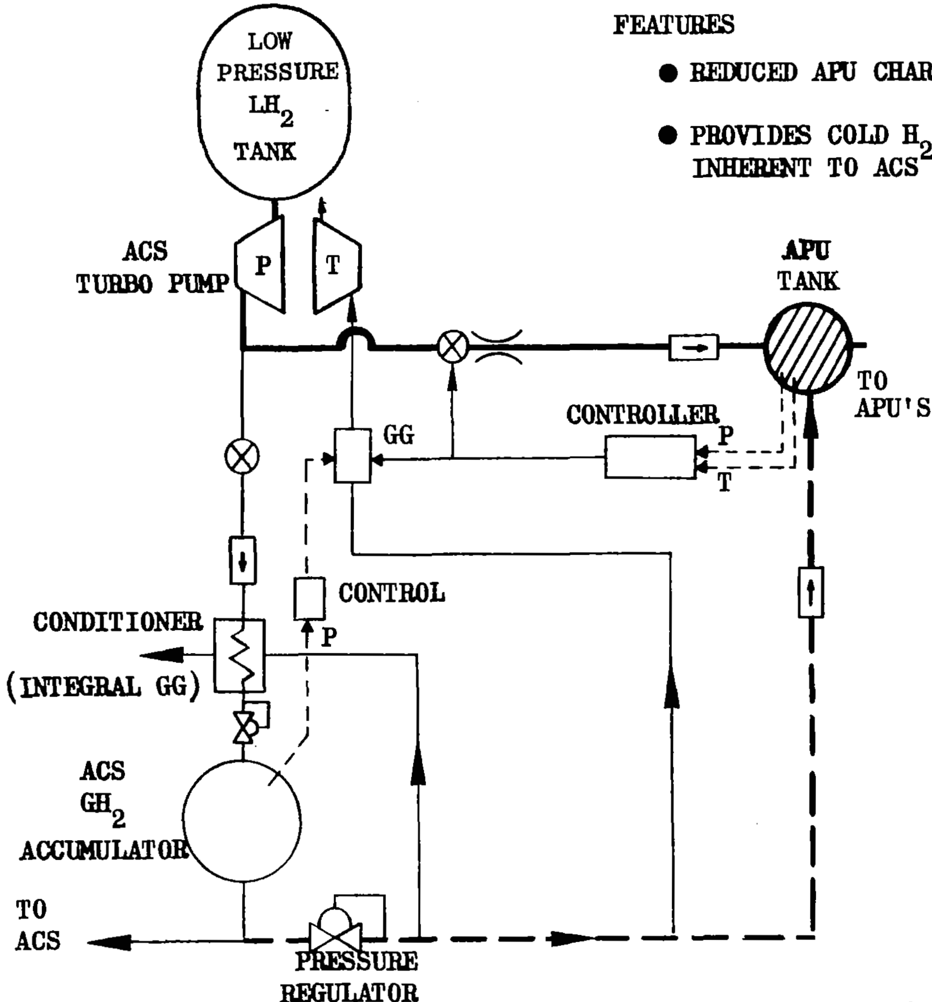
HYDROGEN SOURCE TEMPERATURE

ORBITER 63R (RE-ENTRY)	BOOSTER 186R (CRUISE)
321 LB MIL-H-5606	321 LB MIL-H-5606
CASE DRAIN EXIT 700R	CASE DRAIN EXIT 700R
BULK FLUID AT 620R	BULK FLUID AT 670R
45 HP HEAT LOAD	16.5 HP HEAT LOAD
6 GPM CASE DRAIN FLOW	6 GPM CASE DRAIN FLOW
584R H_2 INTO LUBE COOLER	402R H_2 INTO LUBE COOLER

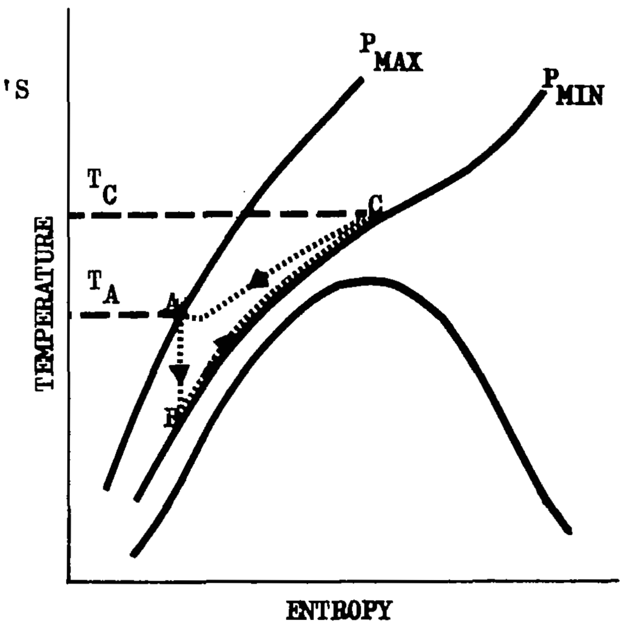
MAXIMUM HYDRAULIC FLUID TEMPERATURE = 700R

FEATURES

- REDUCED APU CHARAGABLE WEIGHT PENALTY VS ACS INTEGRATION
- PROVIDES COLD H_2 (< 125 R) SOLVING ORBITER COOLING PROBLEM INHERENT TO ACS INTEGRATION



THERMODYNAMIC CYCLE



A → B ISENTROPIC DISCHARGE OF APU TANK

B → C CONSTANT PRESSURE EXPULSION UTILIZING GH_2 FROM ACS ACCUMULATOR FOR PRESSURANT (500 PSIA 300°R GH_2)

C → A APU TANK RE-FILL USING ACS TURBOPUMP

Figure 19. Integrated Re-Fillable H_2 Feed System

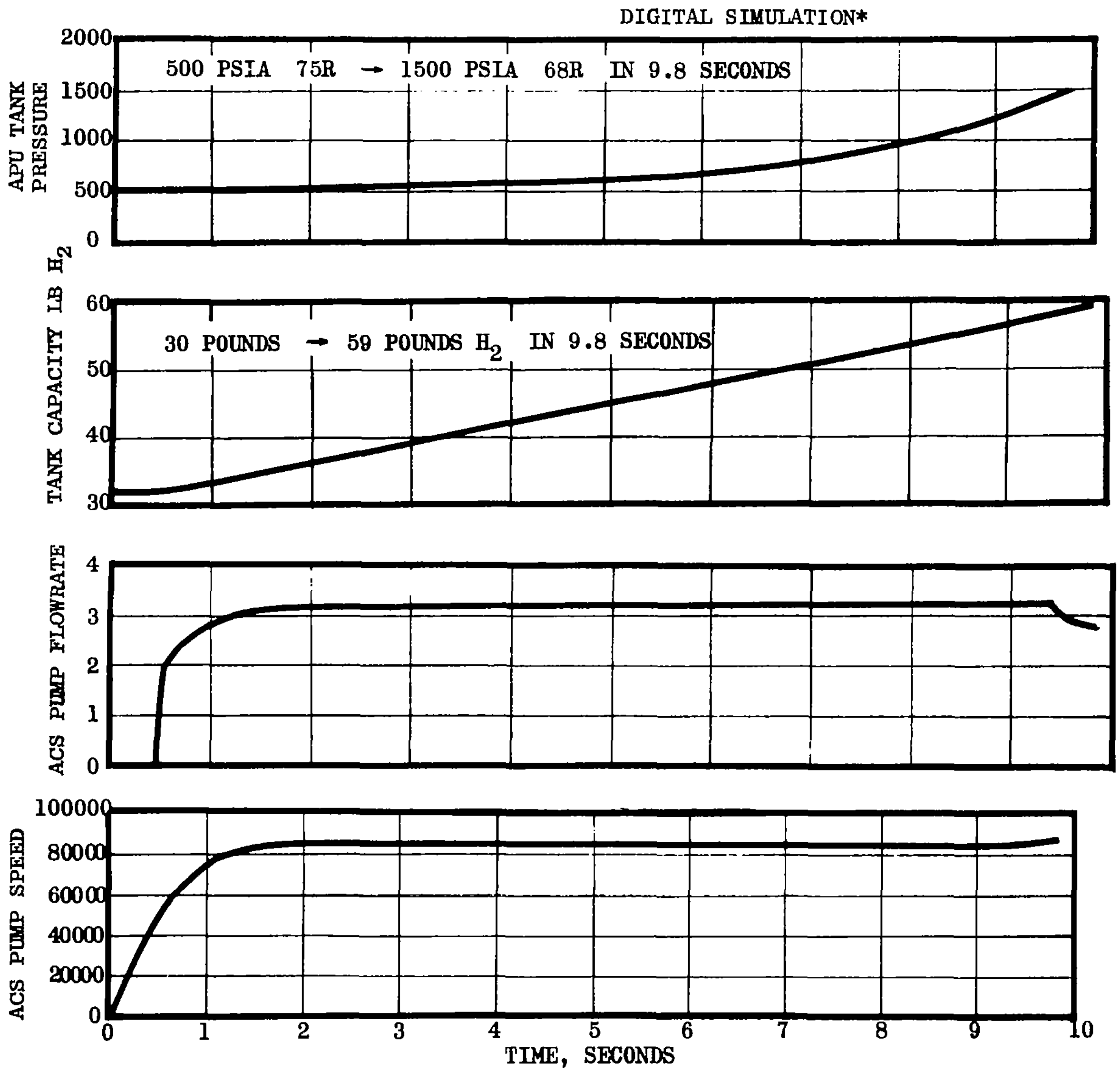
(Point A). Usage of hydrogen by the APU then drops the pressure isentropically to 500 psia at which time the check valve opens and the tank is maintained at 500 psia by the addition of pressure regulated 300 R gas from the APS accumulator (Point B). The APU continues to draw off liquid (B to C) until the temperature reaches some maximum limit T_c (100 R), at which time the APS pump is started and the tank is refilled (C to A). Some sample computer runs showing operating dynamics during refill (B to C) are shown in Fig. 20.

The digital simulation was performed by North American Rockwell's Space Division. For these runs a maximum temperature of 75 R was used and a pressure range of 500 to 1500 psia. The refill cycle is computed in about 10 seconds with a tank capacity increase from 30 to 59 pounds of hydrogen. The APS pump flow is 3.2 lb/sec at a speed of 85000 rpm. A 14 ft^3 tank (3 ft dia.) is used with 17 refill cycles being needed to provide 500 pounds of hydrogen, which would be required for 3 orbiter APU's. An optimization was performed to evaluate the sensitivity of recharge cycles and turbopump propellant penalty with the hydrogen temperature at the beginning of refill. Results (Fig. 21) indicate little incentive for going to temperatures above 100 R since improvement in expulsion efficiency is a very weak function of temperature above this point. The performance of the 100 R system is summarized in Fig. 21 indicating that 13 refill cycles are required to supply three orbiter APU's. The Space Division's estimate of the number of recharge cycles required when using 300 R gas from the APS accumulator is 7 per APU so that a significant reduction (40 percent) in the number of APS pump cycles is also a very important advantage of the refillable system.

The total weight per orbiter APU of the three systems is summarized in Table 7, which includes the baseline APU weight (233 lbs), chargeable APU weight, and burned propellant weight. The major chargeable penalty in the pump fed system is the weight of the pump (a reciprocating type of design by Cosmodyne Corporation). This pump essentially replaces the pressure regulator in the baseline system with the addition of a hydraulic motor drive. The SPC penalty for driving the hydraulic motor is indicated in Table 7 as 27 pounds which represents about 10 percent of the burned propellant weight. The refillable system represents a penalty of close to 50 percent and the ACS integrated system represents a penalty close to 70 percent of the burned propellant weight. Weight savings over APS integration is also shown, indicating a significant advantage for the refillable system (64 lbs) and a very large saving (134 lbs) for the pump fed system. Further in depth study of the systems is recommended with respect to weight, reliability, development risk, and performance in order to accurately assess the relative merits and disadvantages of each.

C. APU INSTALLATION

Figure 22 summarizes an overall heat exchange study to determine how much heat rejection from the APU can be expected in a vehicle installation. Major heat sources in the system are indicated including the regenerator, turbine, exhaust manifold, turbine housing and the combustor. The maximum surface temperature is 1600 F for the combustor wall. If all of these major heat



- **14 FT^3 TANK (3 FT. DIAMETER) 1500 PSIA MAXIMUM**
- **APU USAGE DURING RECHARGE = 0.282 LB/SEC (3 APU's AT 350 HP)**
- **PUMP DISCHARGE LH_2 TEMPERATURE AT 44 °R**
- **17 RECHARGE CYCLES FOR 500 POUNDS H_2 REQUIREMENT (3 ORBITER APU's)**

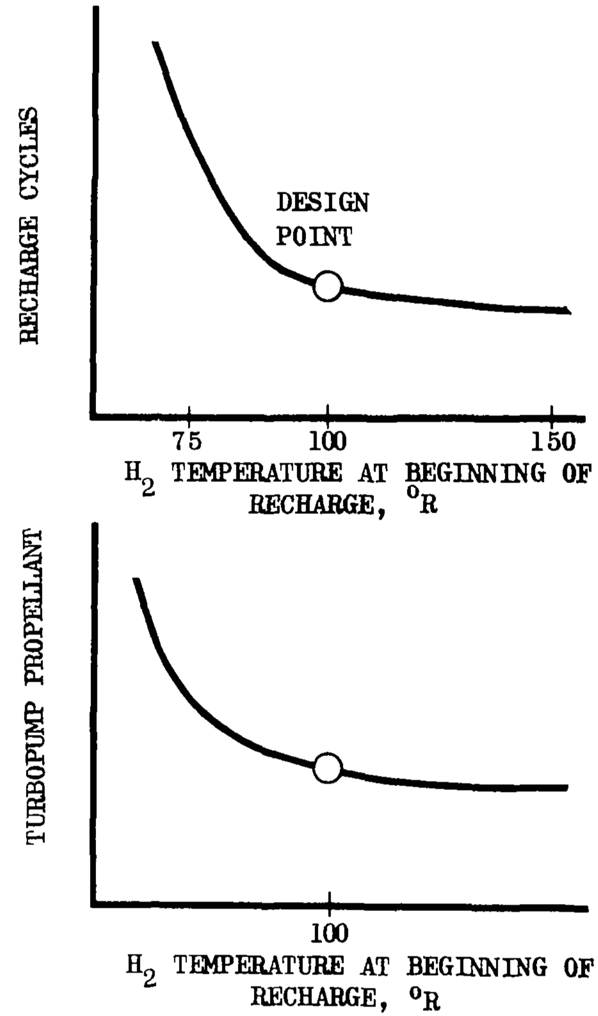
*** NORTH AMERICAN ROCKWELL SPACE DIVISION ANALYSIS**

Figure 20. Re-Fillable System Dynamic Performance

OPERATIONAL SUMMARY

APU BURNED GH_2 REQUIRED	500 LB
RECHARGE PRESSURE BAND	500-1400 PSIA
APU TANK	14.1 FT^3 (3 FT DIA.)
TANK GH_2 TEMPERATURE (BEGIN. RECHARGE)	100 R
GH_2 TEMPERATURE AT END OF RECHARGE	70 R
GH_2 TEMPERATURE AFTER BLOWDOWN TO 500 PSIA	61 R
CHARGE CYCLES REQUIRED	13
H_2 MASS PER RECHARGE	40 LB

H_2 TEMPERATURE OPTIMIZATION



* NR SPACE DIVISION

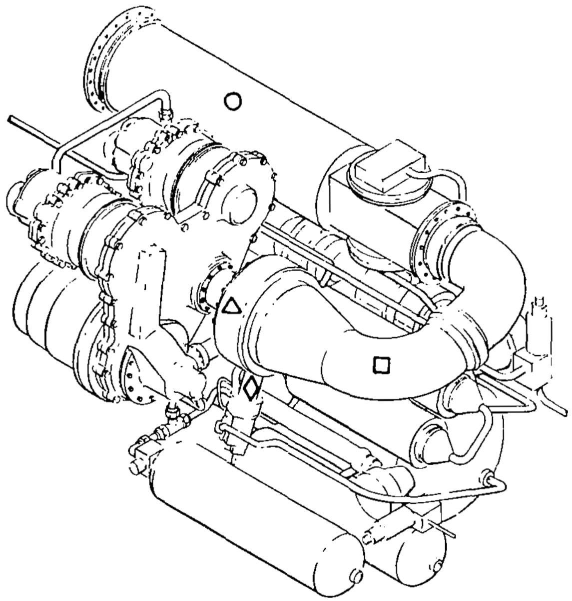
Figure 21. Re-Fillable System Performance Summary* - 3 Orbiter APU

TABLE 7

FEED SYSTEMS WEIGHT SUMMARY
ORBITER VEHICLE-WEIGHT PER APU

ACS INTEGRATED		INTEGRATED REFILLABLE		PUMP FED	
BASELINE APU	233	BASELINE APU	233	BASELINE APU	233
CHARGEABLE TO APU	199	CHARGEABLE TO APU	135	CHARGEABLE TO APU	65
H_2 CONDITIONING GG USAGE TANKAGE	145	H_2 CONDITIONING GG USAGE TANKAGE	45	REMOVE PRESS REG	10
				ADD LH_2 PUMP**	40
SUPPLEMENTAL 300R H_2 FOR COOLING	54	APU TANK	72	HYDRAULIC MOTOR DRIVE (8.5 HP)	8
		H_2 PRESSURANT	3	SPL PENALTY (H_2 PUMP HYD DRIVE)	27
		VALVES/STRUCTURE	15		
BURNED PROPELLANT (HCR ORBITER)	306	BURNED PROPELLANT	306	BURNED PROPELLANT	306
TOTAL	738 LBS		674 LBS		604 LBS
SAVINGS OVER ACS	0		64 LBS		134 LBS

*Less hydraulic pumps/alternator
**Cosmodyne



500F MAXIMUM SURFACE TEMPERATURE
0.5 INCH QUARTZ FIBER INSULATION
$3.15\ \frac{BTU}{SEC}$ REJECTION TO VEHICLE

	UNINSULATED SURFACE TEMPERATURE F	TOTAL HEAT LOSS, BTU/HR: UNINSULATED	TOTAL HEAT LOSS, BTU/HR: INSULATED FOR 500F MAXIMUM TEMPERATURE
○ REGENERATOR	940	11,600	3,740
□ TURBINE EXHAUST MANIFOLD	940	6,650	1,900
△ TURBINE HOUSING	1,270	10,380	1,740
◇ COMBUSTOR	1,600	2,900	370
ADDITIONAL COMPONENTS		1,760	1,760
		$35,120\ \frac{BTU}{HR}$	$11,340\ \frac{BTU}{HR}$
		$\boxed{9.76\ \frac{BTU}{SEC}}$	$\boxed{3.15\ \frac{BTU}{SEC}}$

Figure 22. APU Overall Heat Exchange

source surfaces are uninsulated, a heat loss of close to 10 Btu's/sec results. Providing about half an inch of quartz fiber insulation on these components yields a maximum 500 F surface temperature and reduces the heat loss by about a factor of 3 down to 3.15 Btu's/sec. For a 6000-second orbiter mission, and considering an equivalent surrounding vehicle mass of 2000 pounds of steel, the above heat loss to the vehicle would result in a temperature rise of about 100 F.

Figure 23 shows the APU as it may be installed within the vehicle indicating the accessibility for component inspection and replacement. Specifically, hydraulic pumps are easily accessible and replaceable as is the alternator. The turbine exhaust duct is also easily removed as are the electronic controls and power control valve within the vehicle installed unit. The complete APU assembly is completely contained within the structure and as a modular unit could be removed and another complete APU assembly could be installed in the vehicle.

Figure 24 depicts the APU adaptability to an air breathing turbine installation. Modifications to the gear box including a clutch would permit the APU to be coupled to a JP turbine unit for integration into a vehicle.

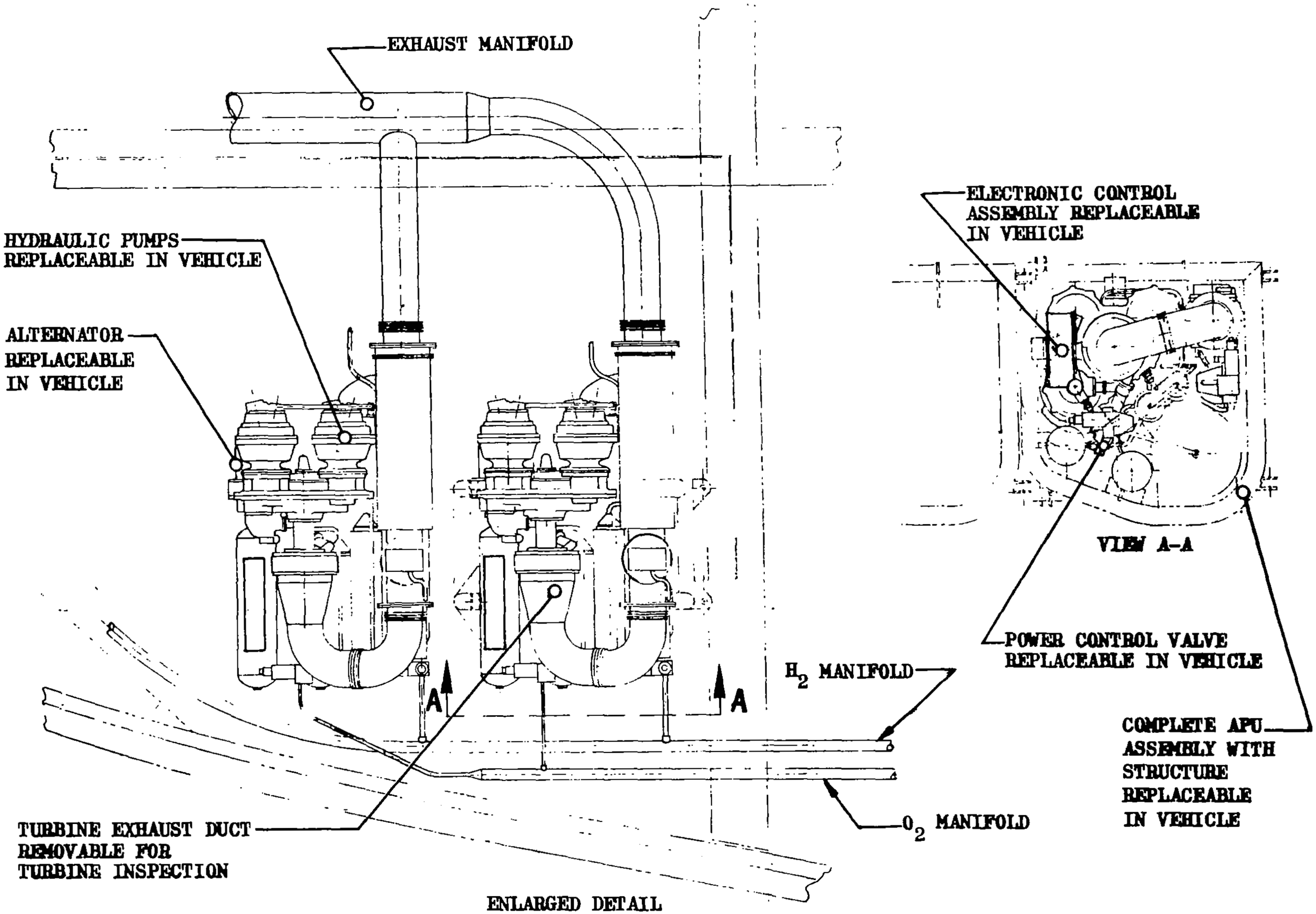


Figure 23. APU System Installation Accessibility

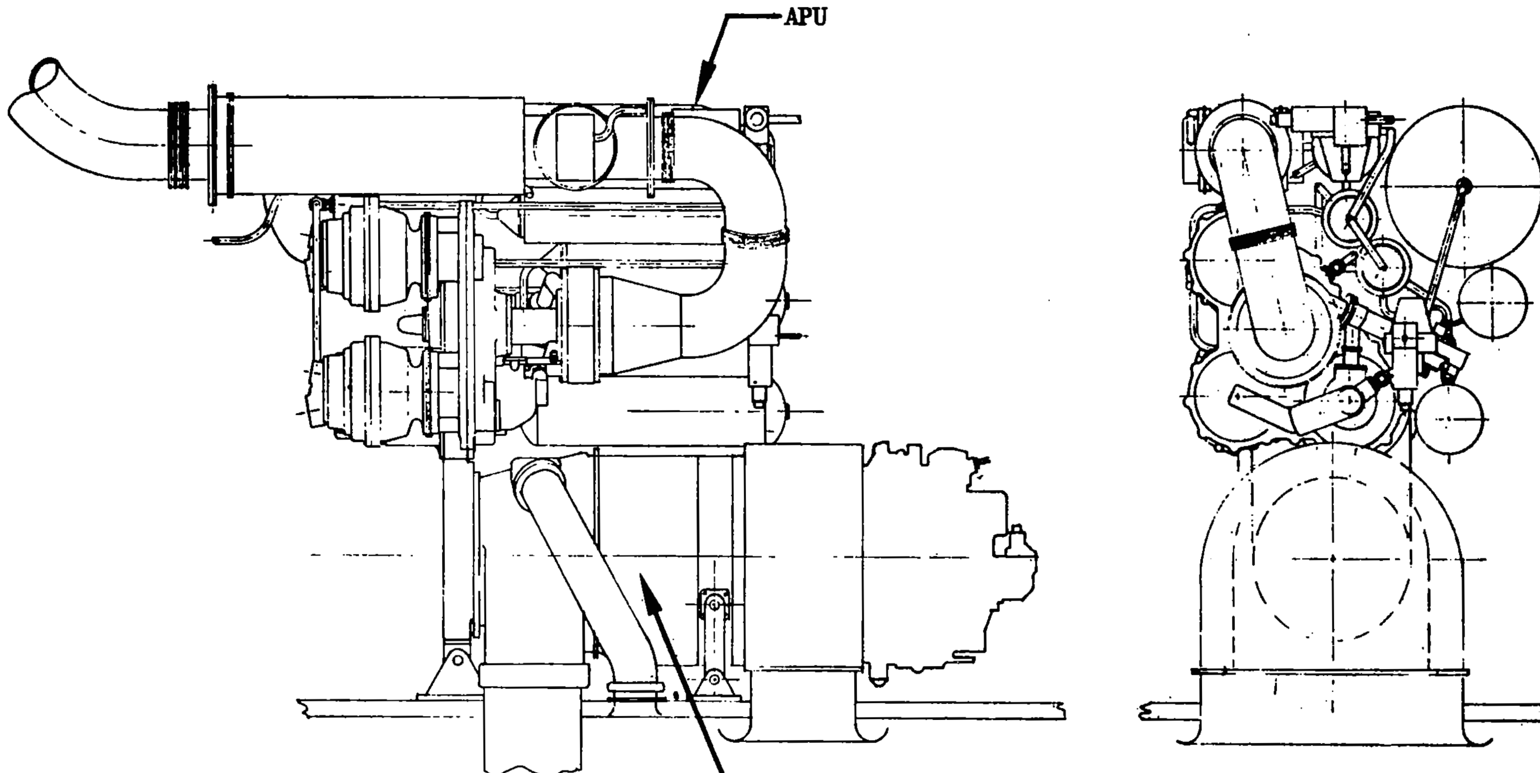


Figure 24. APU Adaptability to Air Breathing Turbine

V SYSTEM OPERATION

Figure 25 summarizes the APU operational analysis conducted during Phase II of the program. Steady-state and transient performance of the APU system was investigated over the entire flight operational envelope using an analog model as the primary analysis tool. The following boundary conditions were imposed on the system during the study:

Inlet hydrogen pressure between 500 to 1,000 psi
Inlet hydrogen temperature between 75 R and 500 R with a rate of change of 100°/sec
Inlet oxygen pressure between 500 and 1,000 psi with a rate of change of 250 psi/sec
Inlet oxygen temperature between 300 R and 500 R
Hydraulic system cooling loads of zero and 100 percent,
An ambient temperature range of from -65 F to +300 F
An ambient pressure range from zero to 14.7 psia with a rate change of 6/10ths of a psi/sec
Power demand variations between zero and 100 percent with a maximum step change from 10 to 100 percent, considering a 75 millisecond vehicle hydraulic system response time.

All critical parameters of the APU when operated in both a pulse and pressure modulated mode were satisfactorily controlled over the flight operational envelope investigated. Turbine inlet temperature was held to a maximum momentary excursion of 100 R above the design point during the most severe operating power transient. When control tolerances are included, this would represent a total momentary excursion of 125 degrees. Turbine speed was held within a band of $\pm$3-1/2 percent. Hydraulic oil temperature at the case drain exit was held within a range of ambient to 728 R. Maximum lube oil temperature was held within a range of ambient to 720 R.

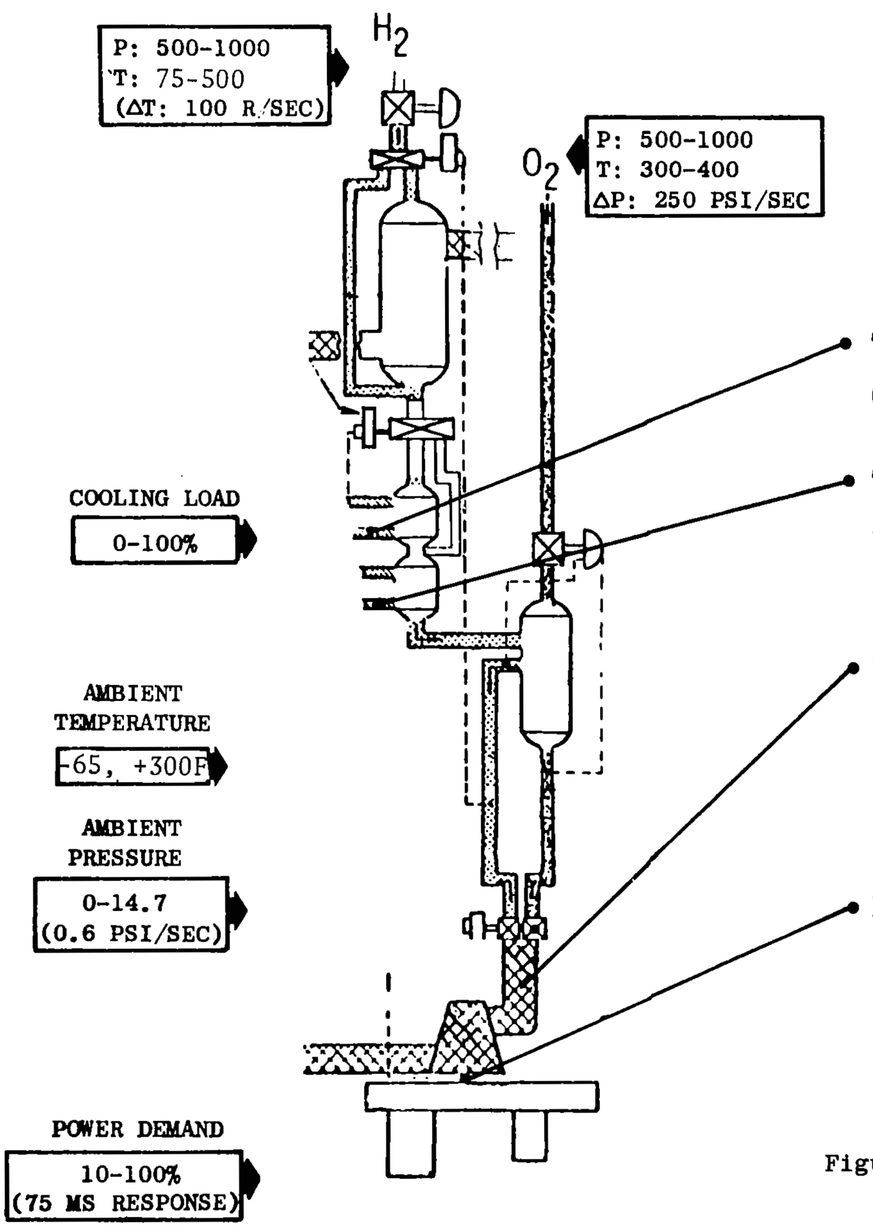
A. OPERATIONAL DESCRIPTION

Figure 26 illustrates a schematic of the APU propellant conditioning system. The major components within the hydrogen feed system are as follows:

The shutoff valves or pre-valves, in both the hydrogen and oxygen lines isolates the APU from the APS supply.

The hydrogen pressure regulator establishes a pressure level throughout the entire propellant conditioning system. This pressure which is regulated to approximately 460 psia at the valve drops to 440 psia at the TPU inlet during high power level operation.

The regenerator provides the primary propellant conditioning for the APU. The heat source for this conditioning is provided by the turbine exhaust gas. The regenerator bypass valve controls this heat flux into the hydrogen system in order to establish a controlled TPU inlet temperature of 650 R in light of



CRITICAL PARAMETERS CONTROLLED

$T_{H/O}$ (CASE DRAIN)	AMBIENT → 728 R
$T_{L/O}$ (LUBE OIL)	AMBIENT → 720 R
TIT	2060 +100 R (MOMENTARY EXCURSION) (+125 INCLUDES CONTROL ACCURACY)
N	±3.5 PERCENT

Figure 25. APU System Operation (Pulse & Pressure Modulated System)

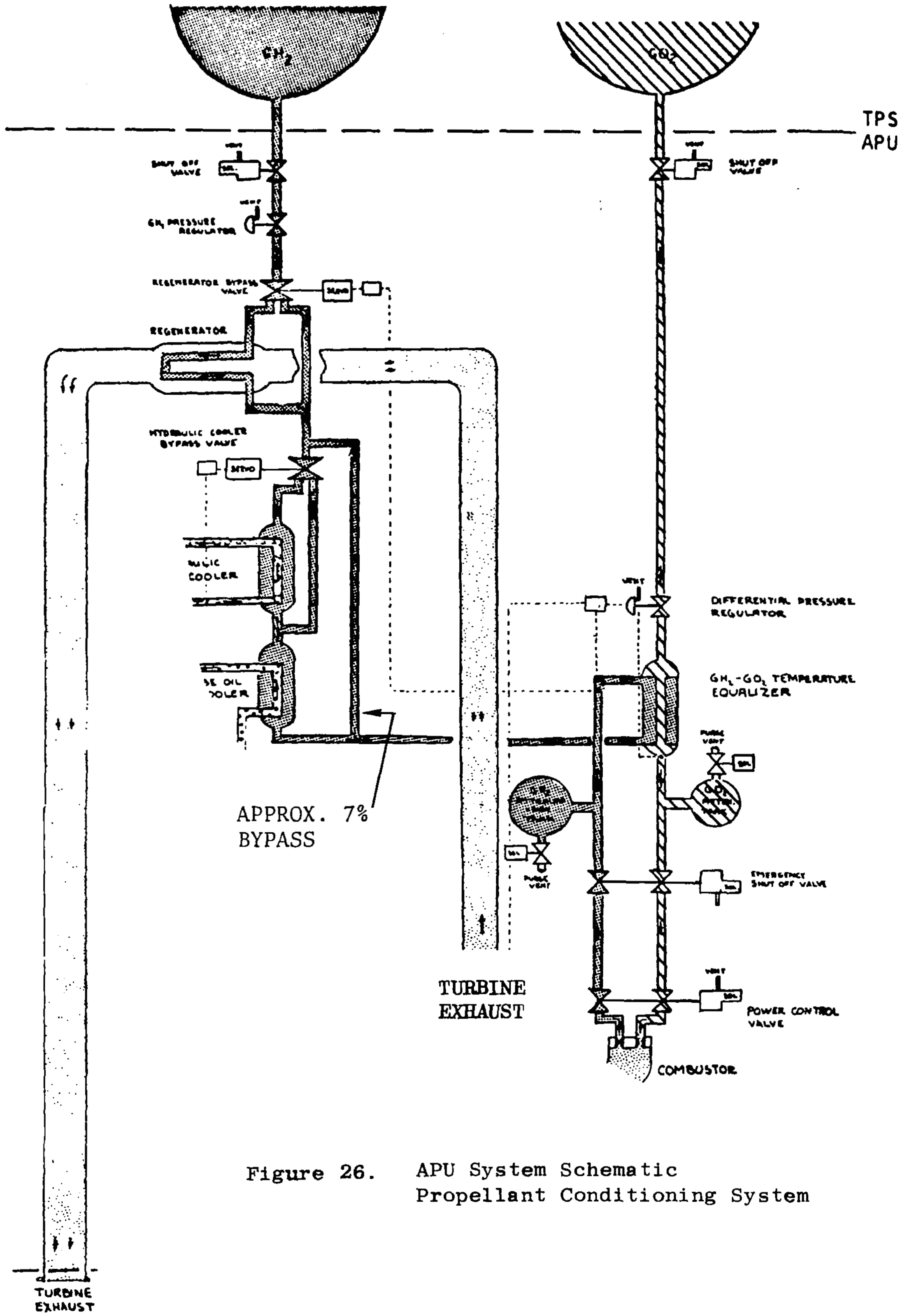


Figure 26. APU System Schematic
Propellant Conditioning System

varying hydraulic cooling loads and APS supply temperatures.

The hydraulic cooler provides cooling of the vehicle hydraulic system. The hydraulic cooler bypass valve controls the cooling rate by varying the hydrogen flow through the cooler as a function of hydraulic case drain temperature; i.e., all hydrogen flow is bypassed around the cooler until the case drain temperature reaches 600 R.

The lube cooler provides all the required cooling of the lube oil system. There are no active controls associated with the lube cooler since the hydrogen sink temperature for the cooler is controlled within ± 10 degrees over the entire operational envelope.

Heat passes from the hydrogen to the oxygen in the temperature equalizer. At the exit of the equalizer, the hydrogen and oxygen fluids are brought within a proximity of ± 25 degrees of each other.

The differential pressure regulator located in the oxygen feed system serves to equalize hydrogen and oxygen pressures at the inlet to the TPU. Hydrogen and oxygen pressure are sensed just downstream of the equalizer and directed to the differential pressure regulator. The hydrogen signal to the regulator is interrupted by a turbine inlet temperature limit control which biases the ΔP regulator hydrogen signal in such a manner as to reduce mixture ratio when turbine inlet temperature exceeds same pre set reference valve.

The attenuator tanks function is to isolate the pulsing flow characteristic developed by the "on-off" actuation of the power control valve from the propellant conditioning system, thereby allowing continuous flow within the propellant conditioning system.

In order to achieve the proper attenuation of the pulse flow, the hydrogen attenuator tank is sized at 2 cubic feet. The oxygen tank is sized at approximately 1/16th that volume, which is the ratio of the molecular weight of the two fluids.

The shutoff valve which is a mechanically linked bi-propellant valve provides both normal and emergency shut down functions for the APU.

Several modifications were incorporated in the APU system during the Phase II program.

A fixed hydrogen bleed of approximately 7 percent was placed around both hydraulic and lube oil coolers. This was done in order to improve the control of TPU inlet temperature as accomplished by the regenerator bypass valve.

The ΔP regulator which had been located downstream of the temperature equalizer was moved upstream. This was done in order to isolate the fluid capacitance and inertia of the ACS supply system from the APU propellant conditioning system. Specifically, with the ΔP regulator in its former downstream position, an oxygen supply pressure build-up of 250 psi/sec resulted in an oxygen flow surge through the equalizer which caused momentary low oxygen temperatures at the TPU inlet and a severe mixture ratio excursion. Relocation of the ΔP regulator nullified this problem.

Control of cooling load in the hydraulic cooler had previously been regulated by a thermostatic bypass valve on the hydraulic flow side of the heat exchanger. During certain APU power transients which resulted in low wall temperatures with little hydraulic through flow, a freezing condition resulted at the cooler exit. By substituting a hydrogen bypass valve in place of the hydraulic thermostatic valve, whis problem was alleviated.

The lube oil cooler bypass valve was eliminated when it was learned by investigation of APU operation over the flight and power profile that the propellant conditioning system maintained a near constant hydrogen sink temperature at the cooler exit which enabled control of lube oil temperature within an acceptably small band. During startup, heat flows from the hydrogen into the lube oil, when the oil and APU hardware are initially at both -65 F and 60 F ambient, thereby preventing congealing of the oil.

Figure 27 is used to schematically illustrate the turbopower unit and hydraulic and lube oil systems. The turbopower unit (TPU) receives hydrogen and oxygen, as supplied by the propellant conditioning system, under controlled pressure and temperature conditions. The power control valve serves to control TPU speed under varying power demands. The valve has two stable positions, open and closed. It is directed to these positions as a function of a control signal which attempts to hold speed within a pre-described band. The turbopower unit is sized to provide 5 percent excess power at sea level with a gear box power demand of 400 horsepower. Heat is generated within the lube oil system at the rate proportional to power level. At max power, this heat load is approximately 3 percent of the gear box output or 9.6 Btu's/second, and at idle power (33 horsepower at the gear box) the heat load is 4.96 Btu/sec. Approximately 8 pounds of lubricating oil is in the system with 6.5 pounds in the reservoir. The primary heat source within the hydraulic system is at the hydraulic pumps where the case drain fluid picks up 32 Btu/sec under full (3,000 psi) pressurization conditions. Under a depressurized condition of 1,000 psi, the heat load is reduced to 15 Btu/sec. Case drain fluid is directed from the pumps to the hydraulic cooler where heat is transferred to the hydrogen system. Approximately 25 gallons of hydraulic oil is cooled by each APU. Approximately 90 percent of the heat load is transferred into the hydraulic system of the case drain cooling passages which passes a small fraction of the total hydraulic flow. The remaining 10 percent is introduced into the hydraulic system at the servo actuators.

Figure 28 describes a schematic of the pressure modulated APU system. There are only three significant hardware changes in converting from a pulse to pressure modulated system: the power control valve is changed from an on-off valve to a mechanical-linked, servo actuated, modulating valve located upstream of the shut off valve. The attenuator tanks are eliminated from the propellant conditioning system. The TPU employs a turbine and combustor design which is optimized for pressure modulation of power over the flight operational envelope; this in general entails a lower turbine design pressure ratio and a deep throttling combustor design. Aside from some additional electronic and control logic changes, the pressure modulated and pulse systems are the same employing the same basic operational philosophy.

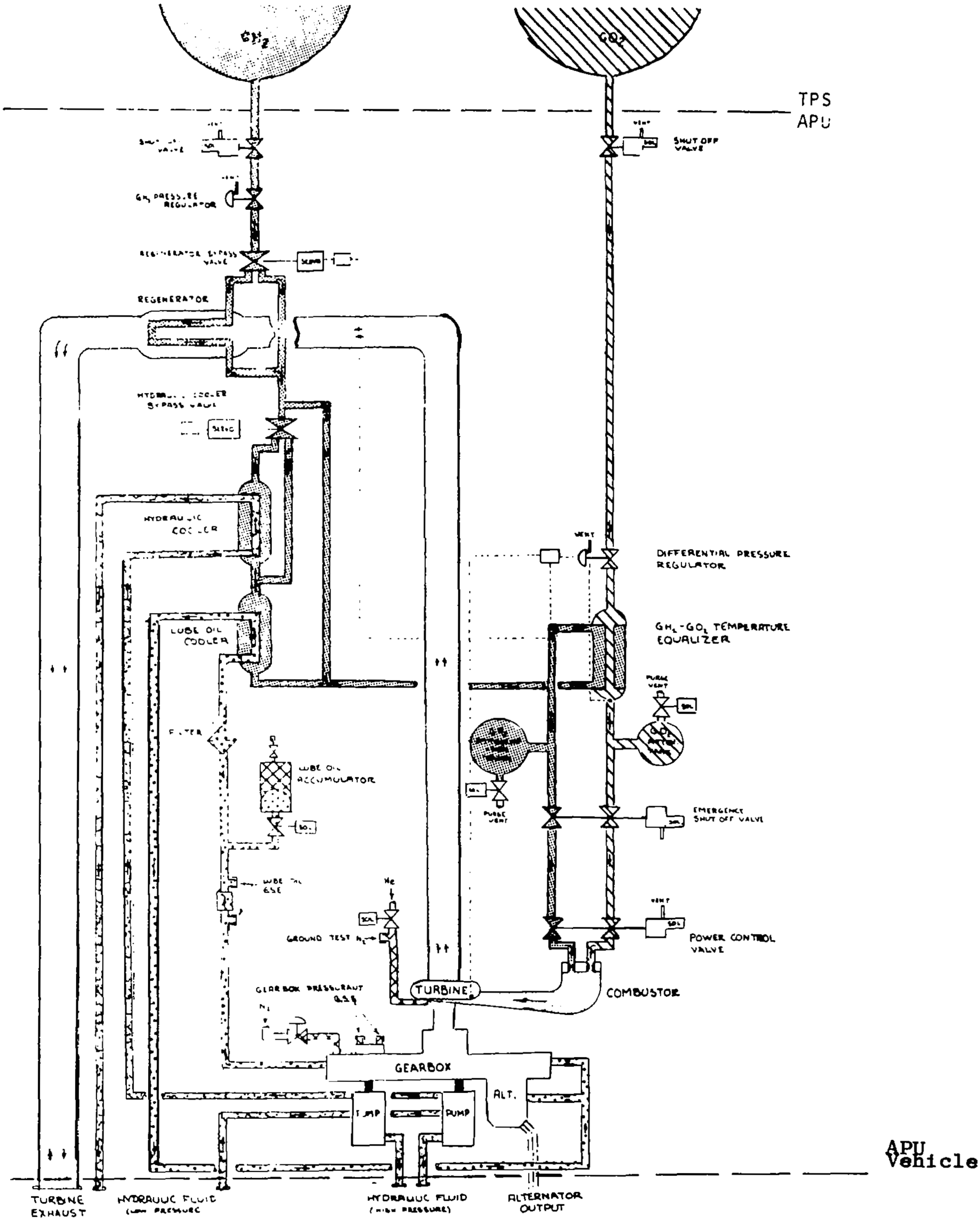


Figure 27. APU System Schematic

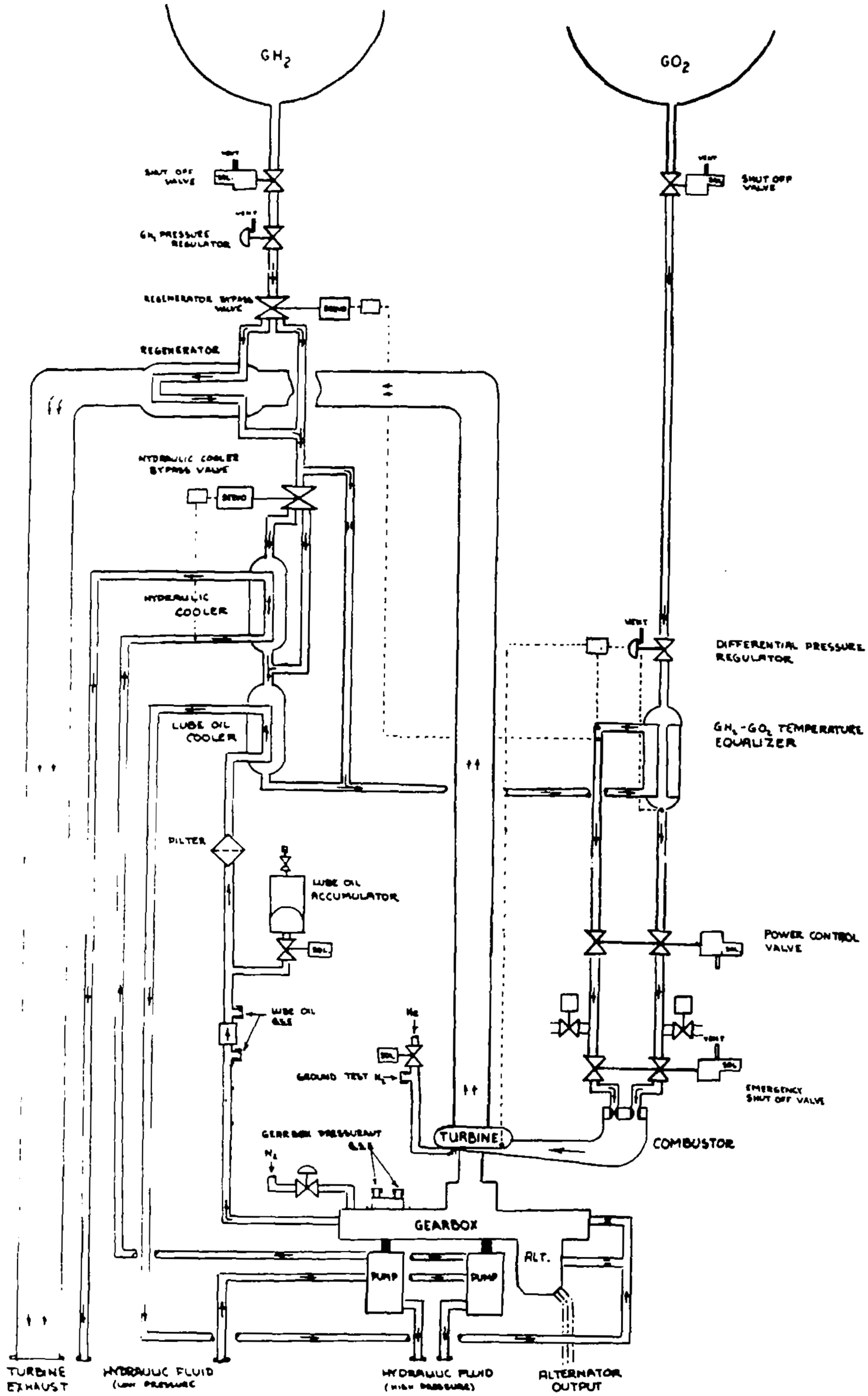


Figure 28. APU System Schematic, Pressure Modulated

B. - ANALOG MODEL SIMULATION

Figure 29 describes the construction and use of the analog model as employed in the operational analysis. The APU system was divided into approximately 60 sections and nodes for development of a comprehensive mathematical model. For those sections involving thermodynamics, heat transfer coefficients were computed as a function of fluid properties, thermal capacitance of the wall was considered as well as fluid heat transport characteristics and delays. Fluid dynamics in each section was represented by momentum equations including variable resistance, fluid capacitance and inertia (characteristics where applicable). Hydraulic pump loads on the gear box were determined as a function of power level, and based upon a variable displacement control to maintain hydraulic pressure. Heat loads into the hydraulic system computed as a function of the hydraulic system pressure level i.e., depressurized or fully pressurized. The thermal capacitance of the vehicle hydraulic system was represented and had a significant effect on transient performance. The lube oil system heat load was varied linearly as a function of power level. With respect to the control system simulation, sensor dynamics such as thermocouple response was represented together with control logic, valve flow characteristics and valve dynamics. Injector inlet temperature and mixture ratio together with combustion fluid dynamics formed the basis for simulation of the combustor. Off design performance of the turbine was represented over the entire operating range and included consideration of windage losses for pulse operation.

The analog model equations were then mechanized on two AD-256 analog computers. A PDPA digital computer was utilized to automate the analog computer set up procedure and also provide automatic readout of system parameters.

The analog model was used for the following APU system studies:

(1) synthesis of the APU control system. Using standard synthesis procedures, the controller networks and logic for each of the APU controls was developed.

(2) steady-state performance mapping over the flight operational profile

(3) transient performance under power transients, during start up and shut down of the APU, and under varying propellant supply conditions and ambient temperatures and pressures

(4) to support the overall failure mode analysis study. Component failures were simulated on the model, the effects of the failure evaluated, and compensation controls devised to either permit a "fail op" conditions or initiate a safe shut down.

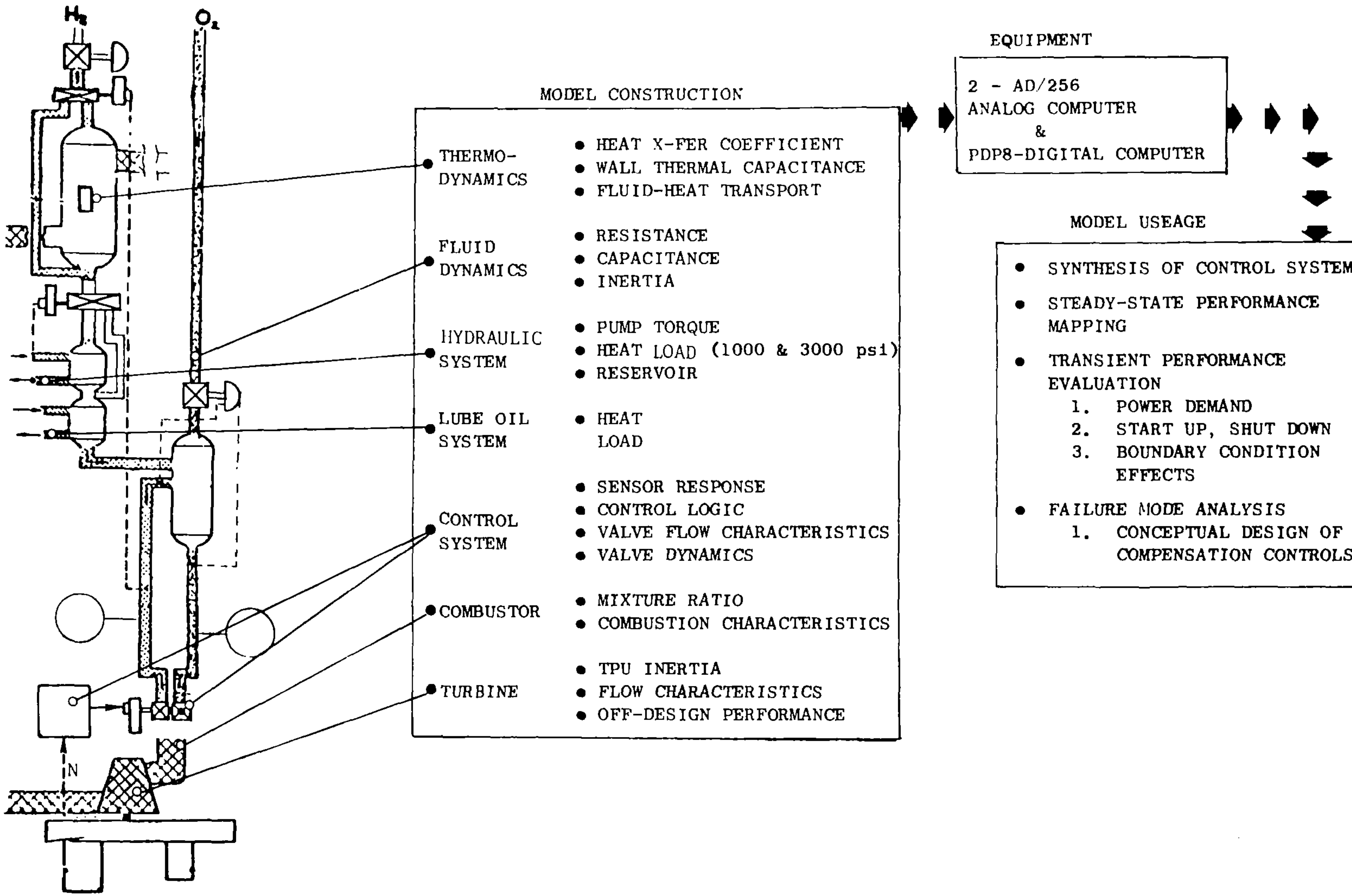


Figure 29. APU - Analog Model, Detailed System Simulation

C. CONTROL SYSTEM SYNTHESIS

Figure 30 describes a comparison of three methods of providing turbine inlet temperature control, and the selected philosophy for the baseline system.

The first method, shown at the top of the figure employs a "feed forward" control of turbine inlet temperature through control of combustion mixture ratio and injector inlet temperature. Control of mixture ratio is accomplished through pressure and temperature equalization of the oxygen and hydrogen fluids at the inlet to the turbopower unit. Control of injector inlet temperature is accomplished with the regenerator bypass control. The advantages of this approach are:

(1) turbine inlet temperature control does not rely on a high response turbine inlet temperature sensor

(2) there is little interaction with the power control system

(3) rapid control is provided to accommodate severe inlet conditions and power transients

A significant disadvantage is that control of turbine inlet temperatures is dependent upon the accuracy of the propellant conditioning system, with respect to pressure and temperature equalization and flow characteristics of the power control valve and injector.

The second method shown in Fig. 30 is a closed loop control utilizing a turbine inlet temperature sensor and feed back control to operate either propellant conditioning system valves or the power control valve. This control philosophy has the advantage of not being dependent on the accuracty of the propellant conditioning, combustor, or TPU components, but has the disadvantage of complete reliance on a high response turbine inlet temperature sensor. Furthermore, the control interacts significantly with the power control system.

The third turbine inlet temperature control philosophy and the one employed in the baseline system eliminates the disadvantages associated with each of the above control approaches. As in the first case, turbine inlet temperature is a "feed forward" control through regulation of mixture ratio and injector inlet temperature. As in the second case, it utilizes a temperature sensor and a feedback control to limit or bias the normal turbine inlet temperature control function. This method eliminates the disadvantage of the first system in that it is not entirely dependent on the accuracy of the propellant conditioning system, or turbopower unit flow characteristics; it eliminates the disadvantage of the second approach in that it does not rely entirely on the high response turbine inlet temperature sensor, and there is little interaction with the power control system.

The overall control system philosophy for both the pulse and pressure modulated APU systems is depicted in Fig. 31 . The four basic parameters to be controlled are turbine speed, turbine inlet temperature, hydraulic oil temperature, and lube oil temperature. Turbine speed is controlled by the power control

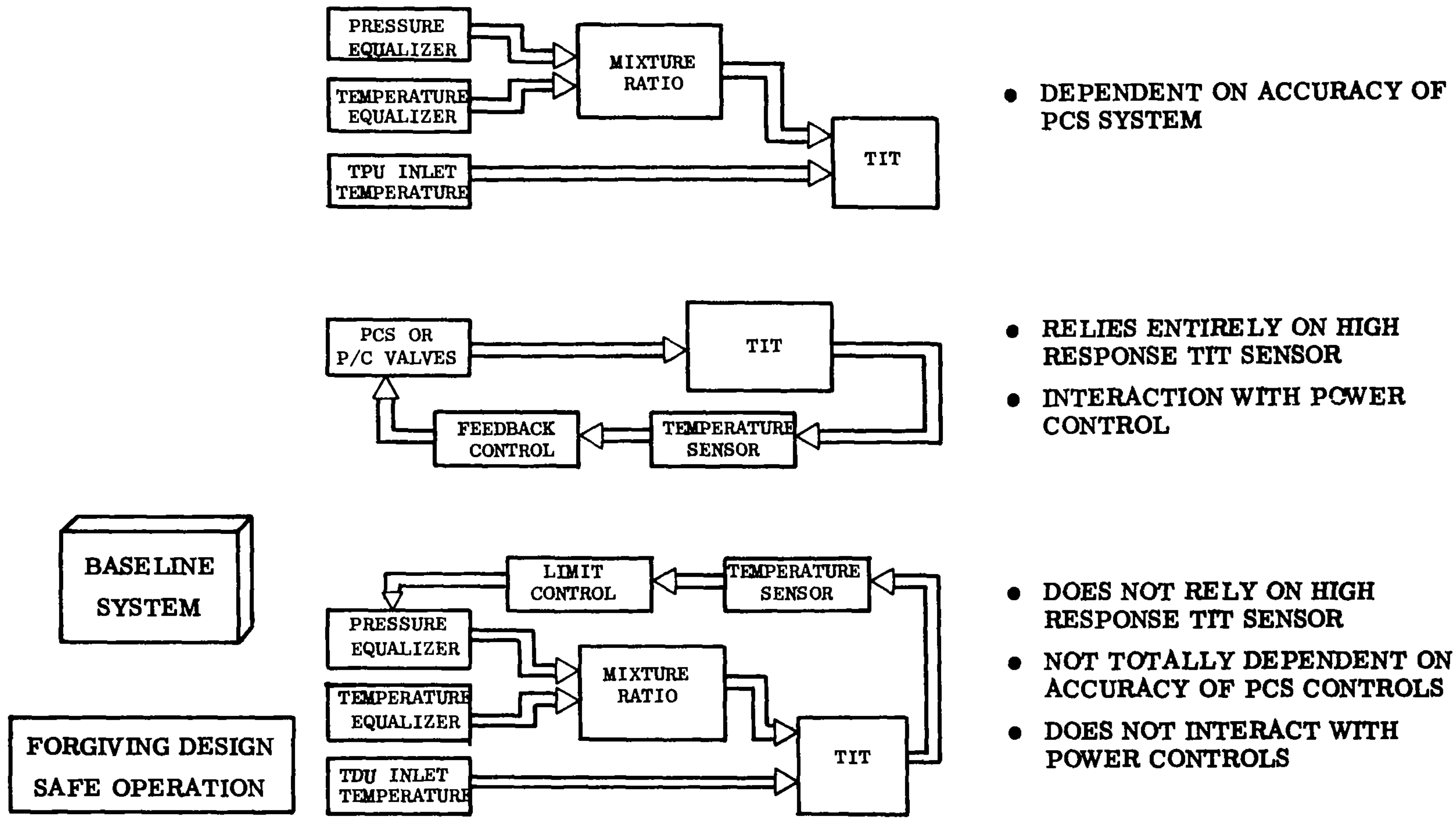


Figure 30. Turbine Inlet Temperature Control Philosophy

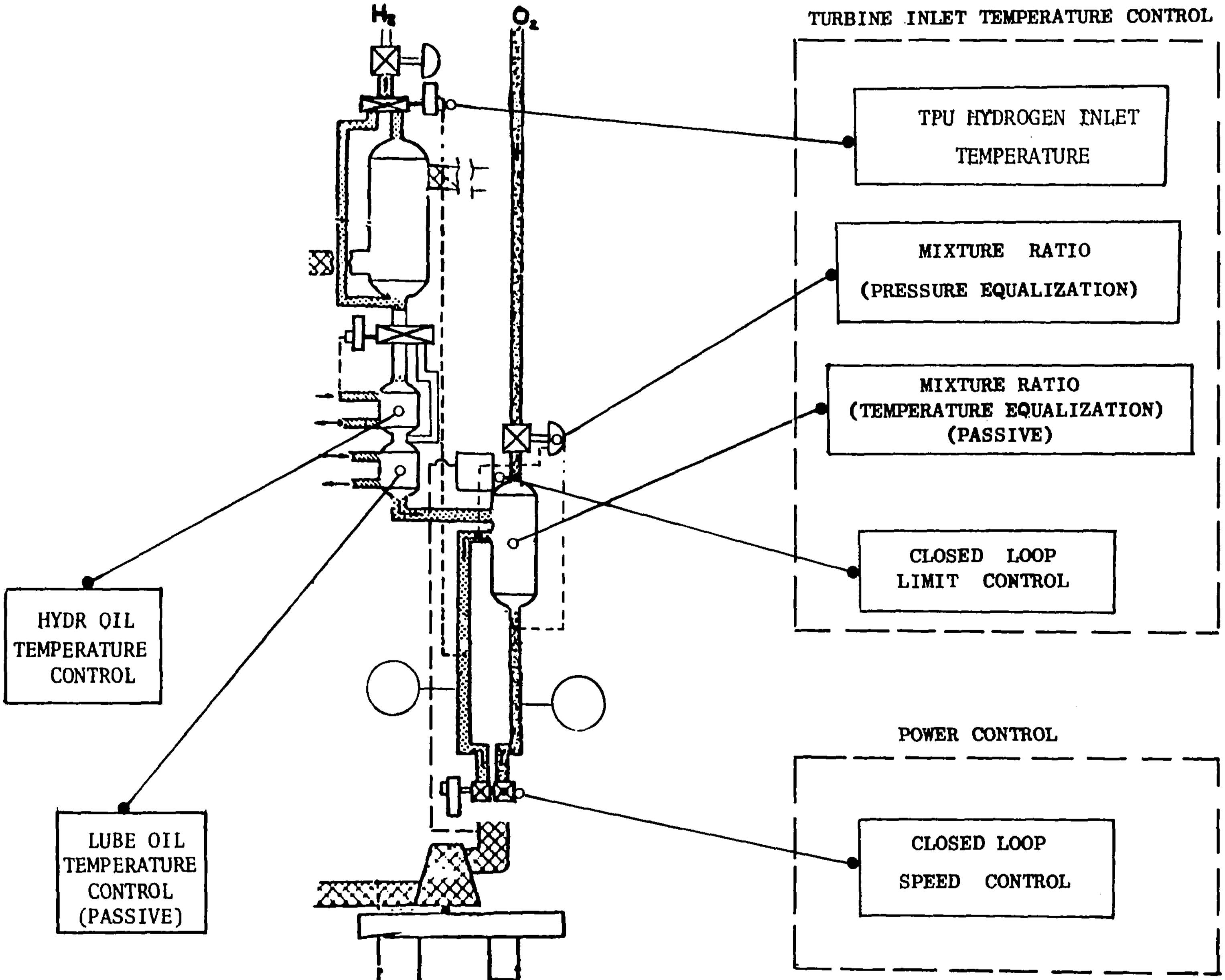


Figure 31. Control System Philosophy
(Pulse and Pressure Modulated Systems)

valve, and, utilizing a closed loop control, maintains TPU speed within a 3.5 percent speed band under all operational transients. Turbine inlet temperature is maintained within 100 R of the design value through "feed forward" control of injector inlet temperature and mixture ratio, with a feed back limit control which is inactive during most APU transients. Hydraulic oil temperature is controlled by the hydraulic cooler bypass valve, and lube oil temperature is controlled passively within the lube oil cooler; this passive control is made possible by the near constant hydrogen sink temperature.

The details of the APU primary control elements are shown in Fig. 32. Control of the hydrogen TPU inlet temperature is accomplished through actuation of the regenerator bypass valve.

A reference temperature of 650 R is compared to the sensed TPU inlet temperature. Any error, E_T is directed to a proportional + integral type controller, which positions the valve in such a manner as to eliminate the error. For example, if the hydrogen TPU inlet temperature is 10 R over temperature, the controller will generate a signal, X_{BP}, to the bypass valve, which will open the valve at 2.5 percent/sec in order to reduce the temperature. The integral portion of the controller prevents deviation from the referenced 650° under all steady-state operating conditions.

The ΔP regulator is a proportional control with a response of 1.6 cps. The regulator holds oxygen pressure approximately equal to the hydrogen pressure at the equalizer where both pressures are sensed. Small deviations of controlled oxygen pressure from the reference hydrogen pressure will exist due to unbalanced flow forces in the regulator valve, and due to the spring rate, or droop, which results in a 6 psi deviation between zero and wide open position.

The hydraulic oil temperature bypass control is a proportional control which bypasses all the hydrogen when the case drain temperature is less than 600 R and directs all the hydrogen through the cooler at a temperature of 650 R or greater. Under idle power conditions for example, the case drain temperature is 640 R and the bypass valve is at approximately 20 percent open. (A lower control setting of 530 R for initiation of coolant hydrogen would be preferable and would provide for somewhat cooler oil temperatures.)

The speed control is an on-off type. A closed signal to the power control valve occurs when speed exceeds 1117 rpm above the design speed. An open signal to the power control valve occurs when speed exceeds 1080 rpm above the design speed. Due to the delay in electrically energizing the valve solenoid (12 milliseconds) and in the valve actuation time (8 milliseconds), and the varying acceleration and deceleration rates of the TPU dependent upon power level, the actual speed and varies somewhat over the operational range.

Details of the APU secondary controls are summarized in Fig. 33. The gaseous hydrogen regulator is a proportional control with a control range or droop of 4 psi and a response of 1.6 cycles per second. The regulator spring preload is set to provide a reference pressure of 463 psi. The other secondary control elements are limit controls which are only active during some extreme transient or as the result of a failure mode.

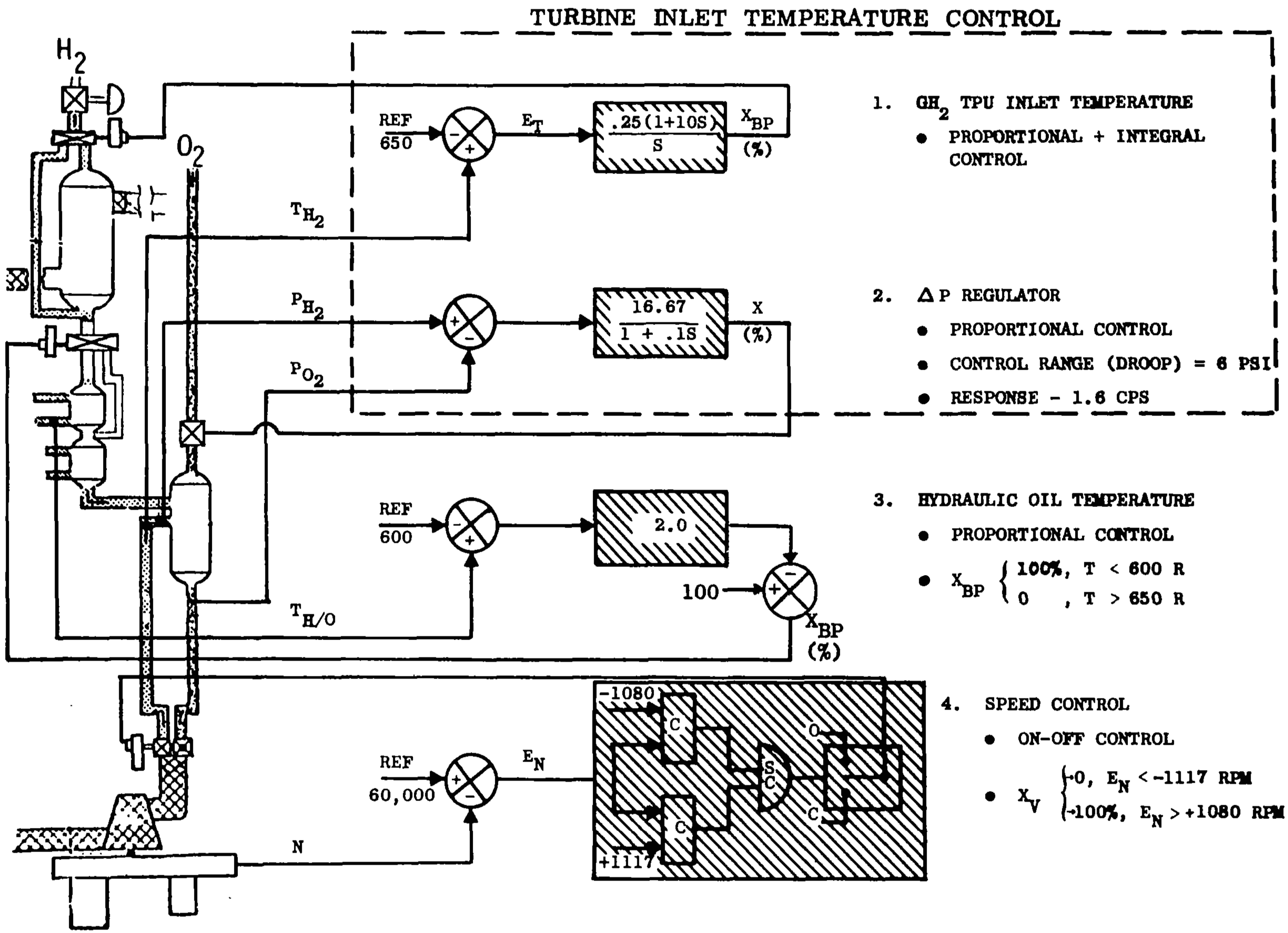
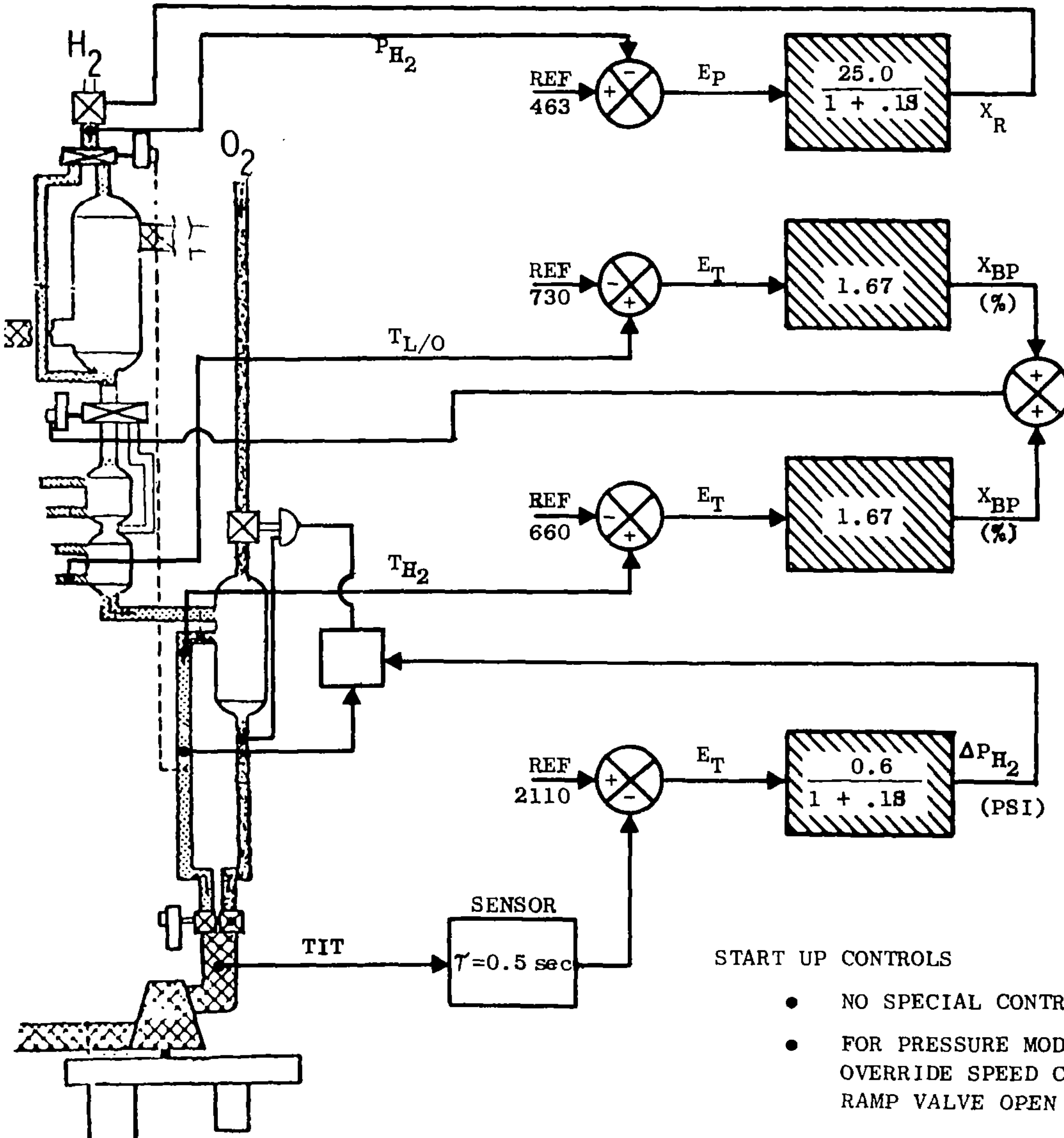


Figure 32. APU Control System Primary Controls



1. GH_2 REGULATOR
   - PROPORTIONAL CONTROL
   - CONTROL RANGE (DROOP) = 4 PSI
   - RESPONSE - 1.6 CPS

2. LUBE OIL AND TPU INLET TEMPERATURE LIMIT CONTROL
   - PROPORTIONAL CONTROL
   - $X_{BP} = 1.67 \left| (T_{L/O} - 730) + (T_{H_2} - 660) \right|$

3. TURBINE INLET TEMPERATURE LIMIT CONTROL
   - PROPORTIONAL CONTROL
   - ΔP_{H_2} = 0.6 PSI/R

START UP CONTROLS

- NO SPECIAL CONTROLS REQUIRED
- FOR PRESSURE MODULATION SYSTEM - OVERRIDE SPEED CONTROL CIRCUIT, RAMP VALVE OPEN IN 1 SEC

Figure 33. APU Control System Secondary Controls

The lube oil and TPU inlet temperature limit controls are of the proportional type. Should the lube oil temperature at the entrance to the cooler exceed 730 R, a signal is directed to the hydraulic bypass valve, which algebraically biases the normal control signal, to open the valve and provide cooler hydrogen to the lube oil cooler. This corrective action is accomplished at the expense of reducing the hydraulic system cooling rate. However, due to the large thermal capacitance of the hydraulic system complete loss of cooling would only result in a case drain temperature rise of 22 R/minute, under full hydraulic pressurization conditions.

Should the TPU hydrogen inlet temperature exceed a referenced value of 660° a similar biased signal is directed to the hydraulic cooler bypass valve to override its normal function, bypass more hydrogen and thereby reduce the TPU inlet temperature.

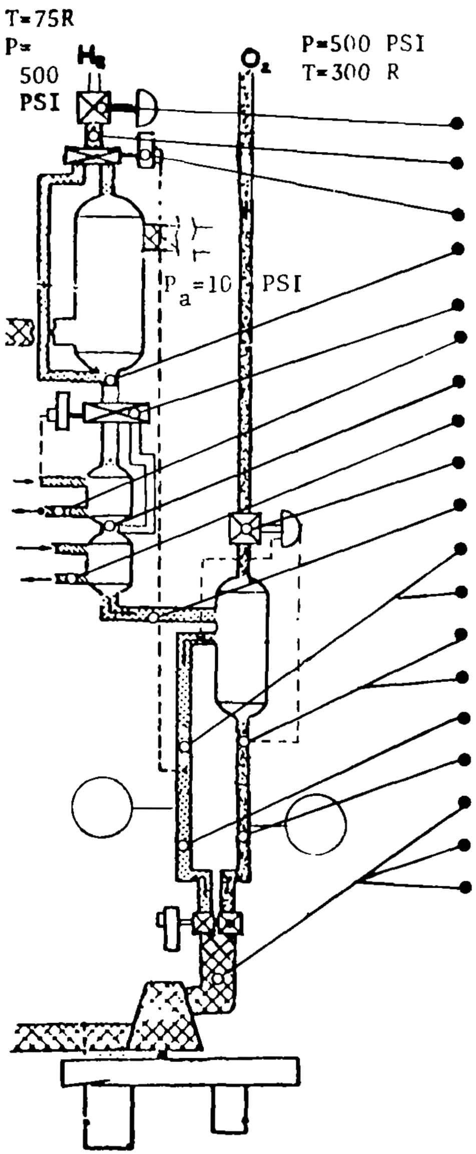
The turbine inlet temperature limit control is a proportional control with a response of 1.6 cycles per second. Should the turbine inlet temperature exceed 50° above the design point of 2060 R, the hydrogen sensing signal to the ΔP regulator is biased in such a manner as to decrease the mixture ratio to the TPU. This is mechanized by use of a variable pressure divider which bleeds down the sensed hydrogen pressure, thereby reducing the signal to the regulator. This results in a reduced regulated oxygen pressure and lower mixture ratio. For every degree above the 2110 degree reference, the pressure signal to the ΔP regulator is biased 0.6 psi.

No special start up controls are required for the pulse system. For the pressure modulation system, the normal speed control circuit is initially deactivated; the power control valve is ramped open in 1 second, and when a speed of approximately 50,000 APU is attained the speed control circuit is activated.

D. STEADY-STATE PERFORMANCE

After synthesizing the control system, the APU analog model was then used to investigate steady-state performance over the flight operational evelope. A total of 30 steady-state runs were conducted at various power levels, propellant inlet and ambient conditions. Every area of the operational envelope was probed in search of potential problem areas. A sampling of some of these tests are discussed in the following paragraphs.

Figure 34 demonstrates ability of the APU to accommodate wide ranges in power demand under steady state conditions. Operation of the APU was investigated under nominal inlet conditions at 10 psi ambient pressure at both idle and maximum power. The hydraulic oil temperature was held within a range of 87° from 641 at idle to 728 R at max. The lube oil temperature was held within a range of 16° from 704 at idle to 720 at max. The hydrogen sink temperature for the passive lube oil cooler was held constant at 672° for both idle and max power. The hydrogen TPU inlet temperature (i.e., injector inlet temperature) was controlled at 650° with no deviation between idle and max power. At the exit of the equalizer oxygen temperature was held within ±25° of the hydrogen exit temperature. The characteristics of the counter flow equalizer heat exchanger are such that at idle power (low propellant flowrates) the oxygen exit temperature



	POWER	
	IDLE (376)	MAX. (378)
X	6±3	67.5±2
P	461	459
X_{BP}	82%	26%
T	288	584
X_{BP}	18%	0
$T_{H/O}$	641	728
T	609	655
$T_{L/O}$	704	720
X	5.5±3.5	64±6.0
T	672	672
T	460±9	440±6
T	650	650
P	460±11	436±7
T	670	625
$\dot{M}$	.0119±.00615	.117±.0045
$\dot{M}$	.0098±.0045	.093±.0045
P	405	390
T	*2030	1995
MR	0.82	0.795

*Low MR Adjustment
(Injector Resistance)

NOMINAL INLET CONDITIONS

PULSE POWER CONTROL

CONCLUSIONS

- Δ TIT = 35 R
- Δ N = 3200
- $\Delta T_{H/O}$ = 641-728
- $\Delta T_{L/O}$ = 700-720
- ΔT_{inj} = ~ 0
- $\Delta T_{H/O\ sink}$ = ~ 0

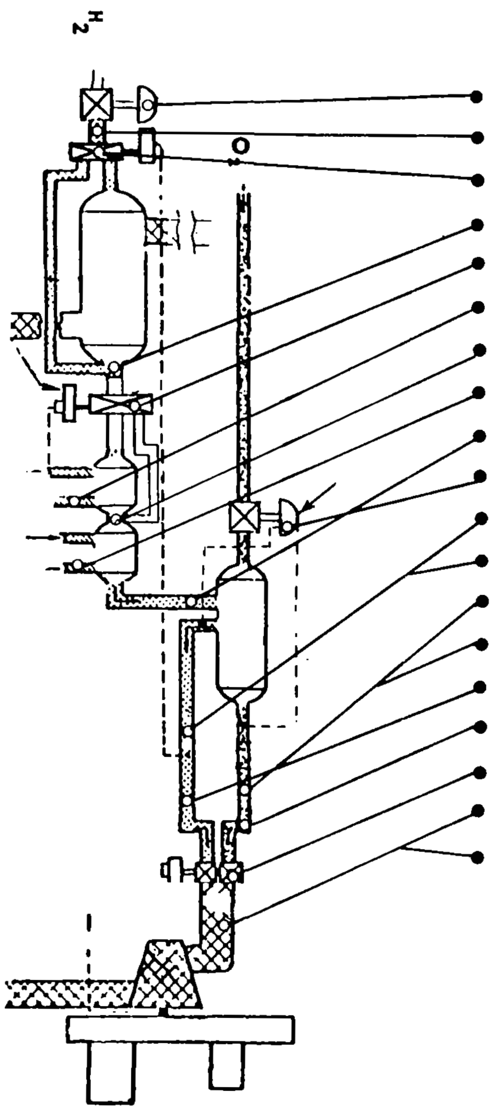
Figure 34. Steady-State Performance - Accommodation of Power Demand

is 20 R above that of the hydrogen exit temperature, and a few degrees below the hydrogen inlet temperature. At maximum power level, the oxygen exit temperature is 25 R less than the hydrogen exit temperature. This tends to increase mixture ratio with increasing power level. However, an offsetting action is provided by the pressure equalization control (ΔP regulator). Note, that the ΔP regulator equalizes TPU inlet oxygen and hydrogen pressures (460 psi) at idle power level. At maximum power the oxygen/hydrogen differential pressure is 4 psi (due to the regulator droop characteristics) in the direction to reduce mixture ratio. The overall result is that turbine inlet temperature decreases from a setting of 2030 R at idle power to 1995 R at maximum power level.

In the case of a pressure modulated system, the combustor characteristics, i.e., C star efficiency, improves with power level resulting in an increase in turbine inlet temperature. A 3-percent variation in C star efficiency, which was incorporated in the digital model for APU performance evaluation would require a 10-percent mixture ratio adjustment in order to prevent a turbine inlet temperature variation. A small C star variation (less than 1.0 percent) can be corrected by increasing the droop of the ΔP regulator thereby decreasing mixture ratio at the high power levels. Larger C star variations would make it difficult to adjust with the ΔP regulator. Two alternatives exist: (1) use a flow resistance to reduce oxygen TPU inlet pressures at high propellant flow rates and; (2) contour the mechanically linked bi-propellant throttle valve to adjust area ratio as a function of power level.

The ability of the APU to operate with both full and zero cooling loads is demonstrated in Fig. 35. Nominal propellant inlet conditions were imposed on the system at an ambient pressure of 10 psi and an idle power level with a pressure modulated power control. With full cooling the regenerator bypass valve was opened to about 82 percent. With zero cooling, it was forced to close to 55 percent in order to increase the heat flux within the regenerator and provide 590 R hydrogen exit temperature. At the entrance to the lube oil cooler, the hydrogen temperature varied only about 20 degrees (672 R) between the full and zero hydraulic cooling cases. At the exit from the cooler the temperatures were the same (672 R) for both full cooling load and zero cooling load, again indicating the constant heat sink temperature for the lube oil cooler. The hydrogen temperature at the entrance to the TPU was held at the control value of 650° under both full and zero hydraulic cooling. There was no effect in the operation of the temperature equalizer. The ΔP regulator controlled almost identically for both cases thereby providing a turbine inlet temperature within 3° under both full and zero cooling. In summary, the APU is shown to have the flexibility of operating with zero hydraulic cooling with essentially no change in speed or turbine inlet temperature, and a variation in the lube oil temperature of about 20°.

The ability of the APU to handle a wide range of hydrogen supply temperatures at idle power level, and sea level is shown in Fig. 36. Analog model run 452 represents a 75° inlet hydrogen temperature with full cooling; run 456, a 400° hydrogen supply temperature with near zero cooling. The other propellant inlet conditions were nominal and a pressure modulated power control was utilized for these runs. Position of the regenerator bypass valve was only slightly affected since the loss of cooling loads was mostly compensated for



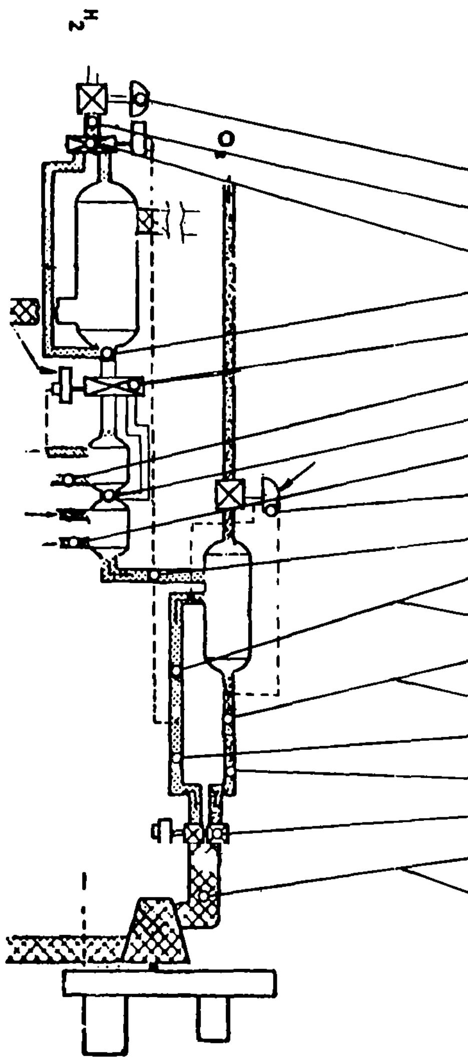
	COOLING LOAD	
	FULL (508)	ZERO (507)
X	9.46	9.45
P	482.3	482.3
X_{BP}	81.8	55.5
T	343	590
X_{bP}	10.3	100
$T_{H/O}$	645	-
T	612	590
$T_{L/O}$	706	684
T	672	672
X	8.8	8.7
P	481.6	481.7
T	650	650
P	481	481
T	667	667
$\dot{M}$	.0165	.0165
$\dot{M}$	.0137	.0137
X_v	18.0	18.0
P	52.1	52.25
T	2059	2056

- NOMINAL INLET CONDITIONS
- POWER LEVEL - IDLE
- AMBIENT PRESSURE - 10 PSIA
- PRESSURE MODULATED CONTROL

CONCLUSIONS

- Δ TiT = 0
- Δ N = 0
- $\Delta T_{L/O}$ = 684-706
- ΔT_{inj} = 0

Figure 35. Steady-State Performance - Cooling Load Flexibility



	GH_2 SUPPLY TEMPERATURE: 75 R - FULL COOLING	GH_2 SUPPLY TEMPERATURE: 400 R - NEAR ZERO COOLING
	(452)	(456)
X	11.34	11.35
P	482.2	482.2
X	78.8	84.1
T	389	594
X_{bP}	6.6	86.2
$T_{H/O}$	647	607
T	614	597
$T_{L/O}$	707	689
X	10.8	* 12.1
T	672	672
P	481.4	481.3
T	650	650
P	480.7	480.7
T	667	667
$\dot{M}$	.0198	.0198
$\dot{M}$	.0165	.0165
X_v	19.7	19.7
P	62.3	62.3
T	2060	2060
MR	.833	.833

* Supply GO_2 7 PSI LOW

- NOMINAL INLET CONDITIONS
- POWER LEVEL - IDLE
- AMBIENT PRESSURE - 14.7 PSIA
- PRESSURE MODULATED CONTROL

CONCLUSIONS

- ΔT_{IT} = ~0
- ΔN = ~0
- $\Delta T_{L/O}$ = 689-707
- ΔT_{inj} = ~0

Figure 36. Steady-State Performance - Propellant Supply Flexibility

by the high hydrogen supply temperature. The lube oil temperature was controlled within an 18 R band in both cases. The lube oil cooler hydrogen sink temperature was constant at 672 R., and the TPU inlet temperature was controlled to 650 R. There was essentially no effect on turbine inlet temperature, speed or TPU injector inlet temperatures, again demonstrating design flexibility with respect to propellant supply conditions.

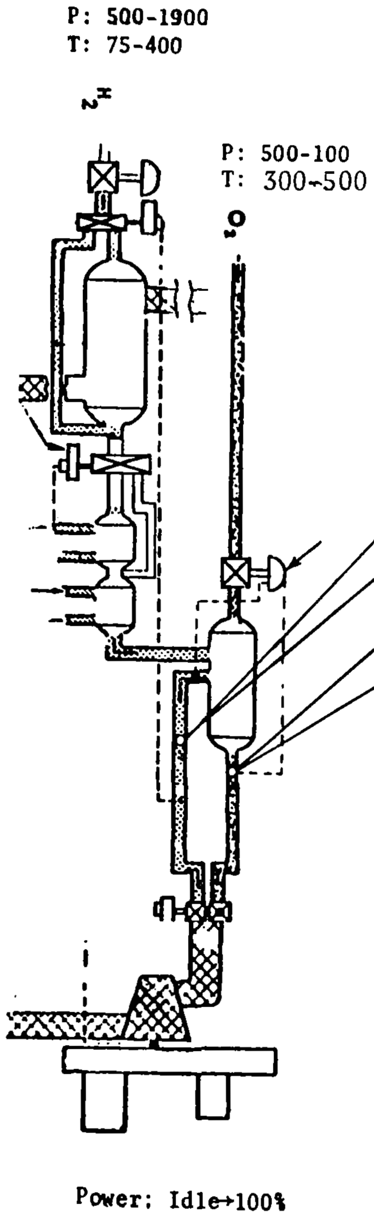
Figure 37 depicts the sensitivity of turbine inlet temperature to a variation of control parameters in the propellant conditioning system. Those parameters which effect turbine inlet temperature are hydrogen pressure and temperature at the entrance to the TPU and oxygen pressure and temperature at the entrance to the TPU. Sensitivity of turbine inlet temperature to variations in hydrogen pressure is essentially zero since the Δ P regulator equalizes hydrogen/oxygen pressure. The sensitivity of turbine inlet temperatures to variations in hydrogen injector inlet temperature is about 1 degree per degree. Turbine inlet temperature is most sensitive to variations in regulated oxygen pressure at the TPU inlet. A 1 percent variation in oxygen pressure produces a 64-1/2^{o} variation in turbine inlet temperature. This sensitivity could be substantially reduced if a larger pressure drop is taken across the injector.

For a pressure modulation system, the sensitivity of turbine inlet temperature variations in oxygen pressure is reduced to 14^{o} due to the use of choked throttle valves. The sensitivity of turbine inlet temperature to variations in oxygen injector inlet temperature is approximately 7^{o} for a one percent variation in the oxygen temperature.

The various sensitivity effects provided an input for establishment of control system specifications, in order to maintain turbine inlet temperature in an acceptable range. The specifications for the regenerator bypass control are $\pm$1 percent, providing a turbine inlet temperature range of 12 R. The ΔP regulator has an accuracy of $\pm$1.5 psi and a droop of .85 percent over the range of inlet conditions and power level. The major portion of the droop is associated with power level and is predictable, tending to reduce turbine inlet temperature with increased power level. The temperature equalizer holds oxygen temperature within a 7 percent range of hydrogen temperature from idle to maximum power. This variation is also predictable and tends to increase turbine inlet temperature with increased power level. Based on the above control element specifications and sensitivities, the expected turbine inlet temperature variation from idle to maximum power is 57^{o}.

E. TRANSIENT PERFORMANCE

After completing the steady state performance evaluation of the APU, transient performance was studied over the flight operational envelope for both the pulse and pressure modulated systems. A total of 37 different runs were made including start and shut down transients, load transients and propellant supply variations under various ambient conditions for both a pulse and pressure modulated system.



PARAMETER	EFFECT ON TIT	CONTROL ELEMENT	CONTROL RANGE
ΔP = 1%	ΔTIT = ~ 0	GH_2 PRESSURE REGULATOR	± 2.2%
ΔT = 1%	ΔTIT = +6 R	REGENERATOR BYPASS CONTROL	± 1%
*ΔP = +1%	TIT = +64.6 R	ΔP REGULATOR	0.85% DROOP (PARTLY PREDICTABLE)
ΔT = +1%	TIT = -7 R	TEMPERATURE EQUALIZER	7% RANGE (PREDICTABLE)

*For Pressure Modulation, ΔTIT=14 R

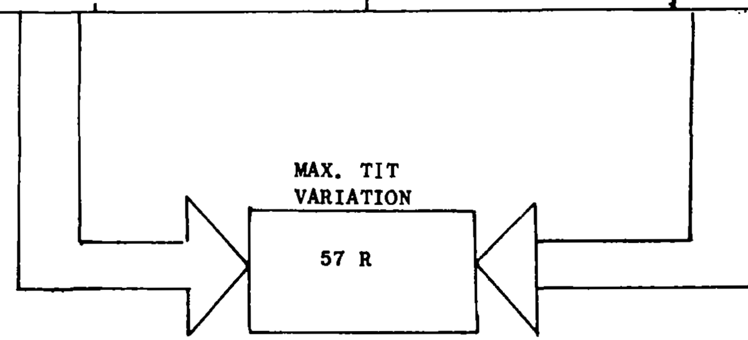


Figure 37. Turbine Inlet Temperature Sensitivity

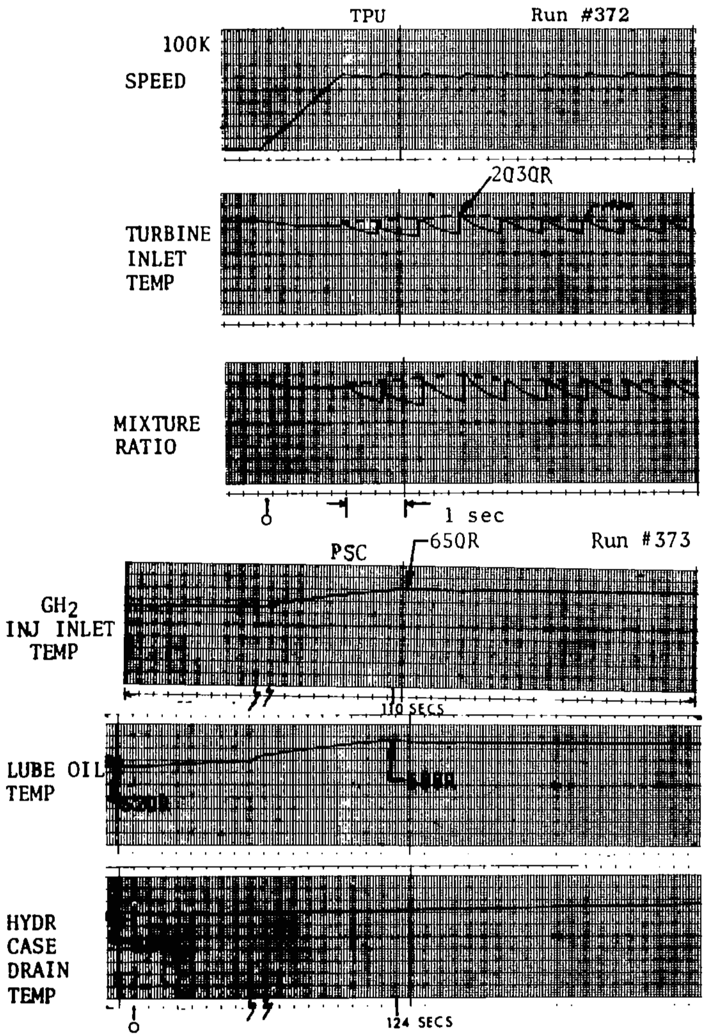
A start up transient of the pulse system is depicted in Fig. 38 . Prior to start up, all walls of the APU were initialized at 520° ambient as were the hydrogen and oxygen propellants, and the hydraulic and lube oil systems. The propellant conditioning system was pressurized up to the shut off valve. The start up was conducted at sea level with nominal inlet conditions to the APU, i.e., the hydrogen pressure of 500 psi and temperature of 75 R, and oxygen pressure of 500 psi and temperature of 300 R. The start up was initiated by activating the normal speed control system. The power control valve opened and stayed open for 1.3 seconds until speed reached the upper band limit. The power control valve then pulsed open and closed holding speed within the pre-set speed band. During the TPU acceleration mixture ratio was somewhat low due primarily to a deviation in pressure equalization by the ΔP regulator. The mixture ratio however, attained a design value when the speed reached 60,000 rpm. Turbine inlet temperature was low, at 1930 R, immediately following the start up with the exception of one combustor pulse at 2030 R. Turbine inlet temperature did not attain the design value of 2060 R until 110 seconds after initiation of the start transient. This was due to a low injector inlet temperature for that time period.

With the propellant conditioning system walls and propellant initialized at an ambient temperature of 520 R the regenerator bypass valve remained closed following the start up in an attempt to raise the injector inlet temperature to the controlled value of 650 R. This temperature was not reached until 110 seconds after initiation of the start transient at which point the regenerator bypass valve opened to maintain control. The lube oil temperature, starting at 520 R, attained its steady state value of 712°, 774 seconds after initiation of the start transient. At that same time the hydraulic oil temperature reached its steady state value of 640°. The hydraulic cooler bypass valve, which had been fully open during the start transient, started closing, and directing hydrogen through the hydraulic cooler after 264 seconds, at which time the case drain temperature reached 600°. Thus, a satisfactory APU start up was demonstrated in 1.3 seconds. It was also demonstrated that full power can be delivered in less than 2 seconds following initiation of a start up.

The effect of various APU boundary conditions on the start up transient is shown in Fig. 39 .

A high altitude start was simulated with an ambient pressure of 0.5 psia. There was virtually no effect on the start transient.

A low ambient temperature start was simulated wherein the propellant conditioning system walls, propellants and oil was initialized at -65 F. Following initiation of the start transient turbine inlet temperature was low (1850 R) and reached its design value 190 seconds later than a standard ambient start. The hydraulic case drain oil came up to the control temperature of 600 R approximately 725 seconds later than a standard ambient temperature start. There were no other significant effects on the APU.



- Max. N Attained in 1.3 Sec.
- TIT ⟶ Design Point in 110 Sec.
- TIT (Max.) = 2030 (1 Pulse)
- MR at Design When N = 60K Due to ΔP Regulator and Temperature Equalizer

- All Walls Initialized at 520 Ambient Temperature
- Propellant Pressurized to S/O Valve
- P_a = 14.7
- Nominal Inlet Conditions

Conclusions

- Satisfactory APU Startup Achieved in 1.3 Sec
- Full Power Can Be Delivered < 2 Sec. (Vehicle Adaptability)

- Regenerator Bypass Valve Opens at 110 Sec (T = 650)
- Lube Oil ⟶ 712 @ t = 774
- Cooler Bypass Valve Opens @ 264 Sec ($T_{H/O}$=600); $T_{H/O}$ ⟶ 640 @ t = 774

Figure 38. Transient Performance - Start Up (Pulse System)

RUN #374

HIGH ALTITUDE START
$P_a = 0.5$ PSIA

- VIRTUALLY NO EFFECT

RUN #391

LOW AMBIENT TEMPERATURE
$T_a = -65F$

- TIT LOWER (1850R); REACHES DESIGN @ 302 SEC (190 SEC LATER)
- $T_{H/O} \longrightarrow$ 600R AT 990 SEC (725 SEC LATER)
- NO OTHER SIGNIFICANT EFFECT

RUN #401

HIGH GO_2 SUPPLY PRESSURE
$P_{O_2} = 1000$ PSI

- MAX TIT 60R ABOVE DESIGN (1 PULSE)
- NO OTHER SIGNIFICANT EFFECT

CONCLUSION

NO DETRIMENTAL EFFECT ON START UP

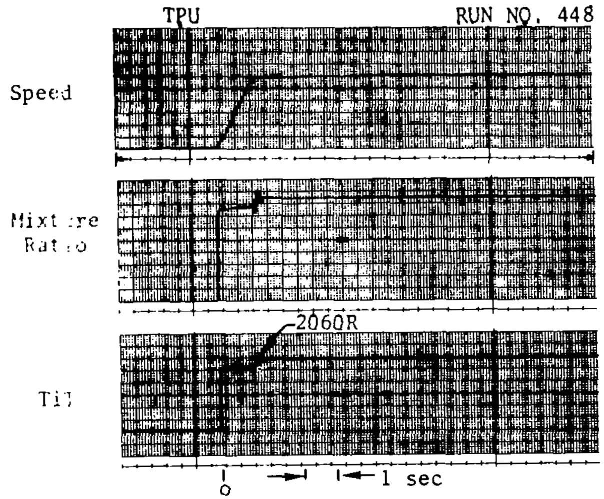
Figure 39. Transient Performance - Start Up (Pulse System)

A start transient was conducted under a high oxygen supply pressure condition of 1,000 psi to determine if this would result in a turbine inlet temperature spike during the start transient. The maximum turbine inlet temperature was 60 R above the design value for one combustor pulse during the start transient. There were no other significant effects on the APU system.

Utilizing a pressure modulated control, a start transient was conducted on the APU at sea level with nominal inlet conditions. The results of this analog are shown in Fig. 40 . All the walls were initialized at 520 R and propellant was pressurized up to the shutoff valve. The start transient was conducted by overriding the normal speed control circuit, and ramping the power control valve open in approximately 1 second. When a speed of 50,000 rpm was attained, the normal speed control loop was reactivated. Design speed was attained at 1.3 seconds as in the case of the pulse control system, with no speed overshoot. Mixture ratio was low during the TPU acceleration. This was due to the fact that a wide open throttle valve, as simulated with the model, was unchoked resulting in a reduced mixture ratio due to a lean combustor/injector setting. At 60,000 rpm, however, with the throttle valve modulated down, mixture ratio was on design. There was no over temperature spike during the start transient. For the pressure modulated system, propellant and oil temperatures throughout the APU attained their steady state value more quickly than in the case of the pulse system. The regenerator bypass valve started opening 32 seconds after initiation of the start, as the injector inlet temperature reached 650 R. At this point turbine inlet temperatures, which was initially low at 1950 R, increased to the design value. The temperature equalizer held oxygen temperature within 24^{o} of hydrogen temperature during the start. The maximum lube oil temperature reached its steady state value of 704 R in 60 seconds. The hydraulic case drain temperature reached 600^{o} in 264 seconds at which point the bypass valve started closing. As in the case of the pulse system, a completely satisfactory start was demonstrated; and full power can be delivered in less than 2 seconds.

The ability of the APU to withstand severe power demand spikes from idle is illustrated in Fig. 41 . An extreme power spike from idle to max power was imposed on the pulse system with a 70 millisecond hydraulic system lag. Nominal inlet conditions existed with an ambient pressure of 10 psia. Between idle and maximum power there is a shift in the steady state speed band which, as previously discussed results from the variable acceleration and decceleration rates of the TPU. The overall speed control range however, is -2.7 percent and +3.5 percent.

Following the power reduction after the spike, a 90^{o} excursion in turbine inlet temperature existed during one combustor pulse or approximately 65 milliseconds. In general there was a downward shift in steady state turbine inlet temperature of 40^{o} from idle to max power, due to the droop in the ΔP regulator. During these power spikes oxygen and hydrogen TPU inlet pressure was equalized by the ΔP regulator and proper sizing of the attenuator tanks. Oxygen and hydrogen temperature was equalized within a 44^{o} band. It may be concluded that the APU can accommodate the <u>most severe</u> power spikes with no detrimental effects.

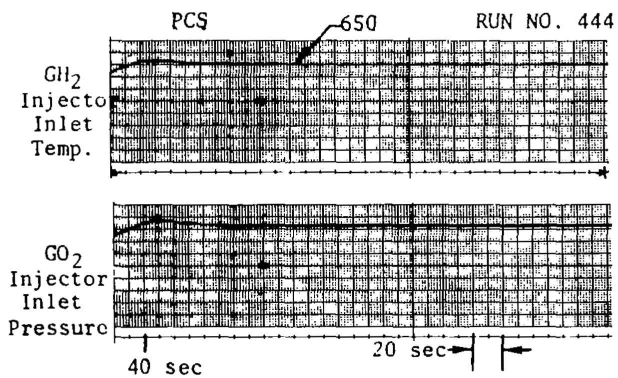


- MAX N ATTAINED IN 1.3 SEC
- NO SPEED OVERSHOOT

- M.R. LOW DURING ACCELERATION (DUE TO LEAN INJECTOR SETTING)
- M.R. AT DESIGN AT 60K

- NO OVERTEMPERATURE;
- TIT MAXIMUM = 2060

- ALL WALLS INITIALIZED AT 520R AMBIENT TEMPERATURE
- PROPELLANT PRESSURIZED TO S/O VALVE
- P_a = 14.7 PSIA
- NOMINAL INLET CONDITIONS

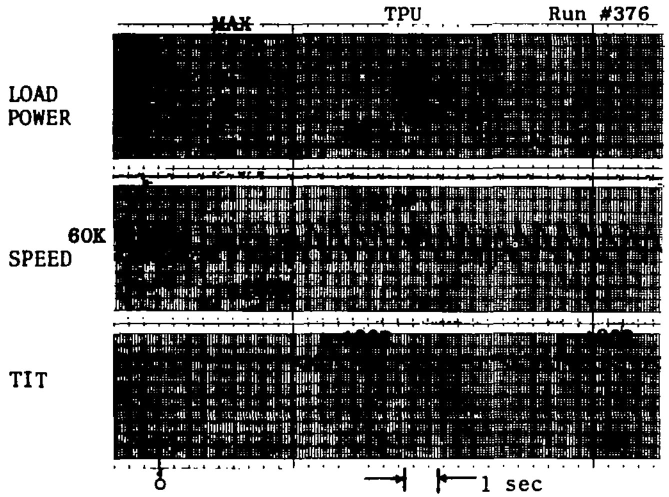


- REGEN BYPASS VALVE OPENS AT 32 SEC (T = 650)
- 24R OVERSHOOT

- TEMPERATURE EQUALIZER HOLDS GO_2 TEMPERATURE WITHIN 24R OF GH_2 TEMPERATURE
- LUBE OIL ⟶ 704R IN 60 SEC
- HYDRAULIC CASE DRAIN TEMPERATURE ⟶ 600 IN 264 SEC

- SATISFACTORY APU STARTUP ACHIEVED IN 1.3 SEC
- FULL POWER CAN BE DELIVERED < 2.0 SEC

Figure 40. Transient Performance - Start Up (Pressure Modulated System

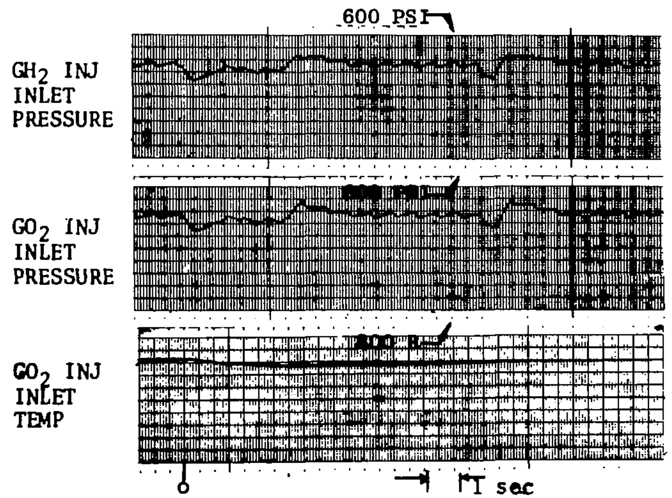


- EXTREME POWER SPIKE (MAX. TIT EXCURSION)
- PULSE DURATION=.065 SEC
- MR CONSTANT DURING PULSE
- TOTAL ΔN RANGE: -2.7%, +3.5%
- ΔT64 ~90 R ABOVE DESIGN ON STEP DOWN
- T64 SHIFT AT MAX. POWER = -40 R DUE TO ΔP REGULATOR DROOP

- NOMINAL INLET CONDITIONS
- P_a = 10 PSIA
- 70 MS HYDRAULIC SYSTEMS LAG

CONCLUSION

MAXIMUM POWER SPIKE ACCOMMODATED BY APU WITH NO DETRIMENTAL EFFECT



- GO_2 PRESSURE EQUALIZED TO GH_2 PRESSURE BY ΔP REGULATOR AND ATTENUATOR TANK. SIZING.
- GO_2 TEMPERATURE EQUALIZED TO GH_2 WITHIN 44 R BAND

Figure 41. Transient Performance
Maximum Power Spikes From Idle
(Pulse System)

Power demand variations were simulated on a pressure modulated system and the results of these transients are descibed in Fig. **42** . As in the case of the pulse system, power spikes were imposed on the APU from idle to max power with nominal propellant inlet conditions and a 10 psia ambient pressure. As a result of this load transient the speed deviation was a maximum of 3,000 rpm. The maximum turbine inlet temperature excustion, which occurred during the step down in power, was +100 R above the design value. This temperature excursion lasted for approximately 100 milliceconds. There was no deviation in hydrogen or oxygen injector inlet temperatures dueing this transient.

Two second power ramps were also executed between max and idle power, in the pressure modulated system. A maximum turbine inlet temperature excursion of 50^{o} above the design value occurred during the ramp down in power. A maximum deviation of 32^{o} resulted in the oxygen injector inlet temperature from the nominal value of 650 R. It may be concluded that all system parameters were satisfactorily controlled under the most severe load transients on the pressure modulated system.

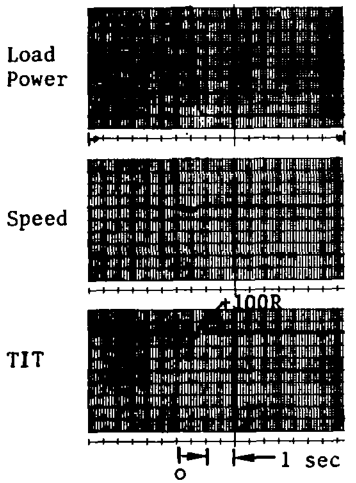
The APU system was then subjected to a simulated ACS oxygen tank pump up transient to evaluate the effects on turbine inlet temperature control. The results of this simulation are shown in Fig. **43** for a pulse system. The hydrogen inlet conditions were nominal at 500 psi and 75 R. The transient was conducted at idle power and an ambient pressure of 10 psia. The oxygen ACS tank was assumed to increase in pressure from 500 to 1000 psi and from 400 R to 300 R in 2 seconds (an extreme inlet condition transient). There was virtually no deviation in mixture ratio from the design value of .834 during the transient. There was no deviation in the hydrogen injection inlet temperature from the nominal control value of 650 R only a 3^{o} variation in oxygen exit temperature from the equalizer. It may be concluded that the APU was completely insensitive to this severe inlet condition transient.

A simulated shut down transient for the pulse system is shown in Fig. **44** Nominal inlet conditions were imposed on the APU and the transient was conducted at sea level from an idle power condition. The shut down transient was initiated by deactivization of the speed control which results in closure of the power control valve. Speed decayed from the design value to zero in 5.8 seconds. Following closure of the power control valve, both the hydrogen and ΔP regulator prevented over pressurization of the attenuator tanks by a rapid closure. No auxiliary controls were required to conduct this shut down, and all system parameters were satisfactorily controlled. A shut down transient was also performed at high altitide and resulted in a one second longer speed decay.

The analog model simulation utilized a TPU polar moment of inertia of .0079 lb-ft-sec^2 in both the pulse and pressure modulated systems. Subsequent to the dynamic analysis, a detailed evaluation of the design indicated an inertia of .01142 lb-ft-sec^2. This increased inertia will directly affect the combustor and valve on-off times for the pulse system as well as total number of mission cycles, maintaining the same speed band. For the pressure modulated

Power Spikes From Idle

Run 504

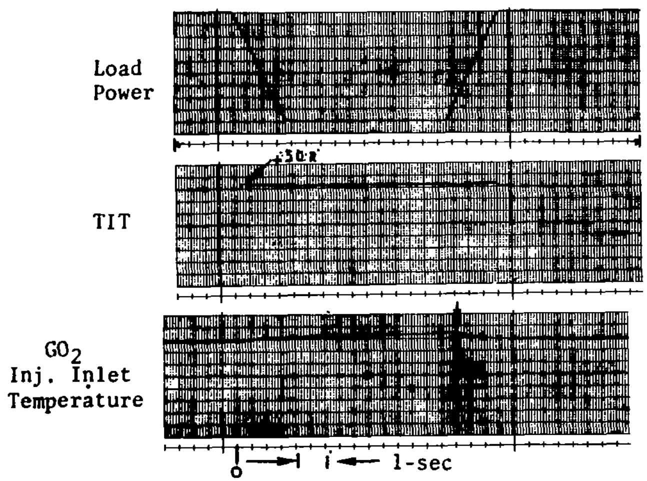


- Extreme Power Spike (max. TIT Excursion)
- Speed Deviation = -3,000 RPM
- Max. TIT Excursion + 100 R (100 MS)
- No Deviation in GH_2 or GO_2 Injector Inlet Temperature.

- Nominal Inlet Conditions
- P_a = 10 psia
- 70 MS Hydraulic System Lag

2 Sec. Power Ramp (Idle-max.)

Run 500



- Max. TIT Excursion + 50 R (Ramp Down)
- Max. GO_2 Injector Inlet Temperature Deviation From 650 R = 32 R

Conclusion

- All System Parameters Satisfactorily Controlled Under Severe Power Transients

Figure 42. Transient Performance - Power Demand Variations Pressure Modulated System

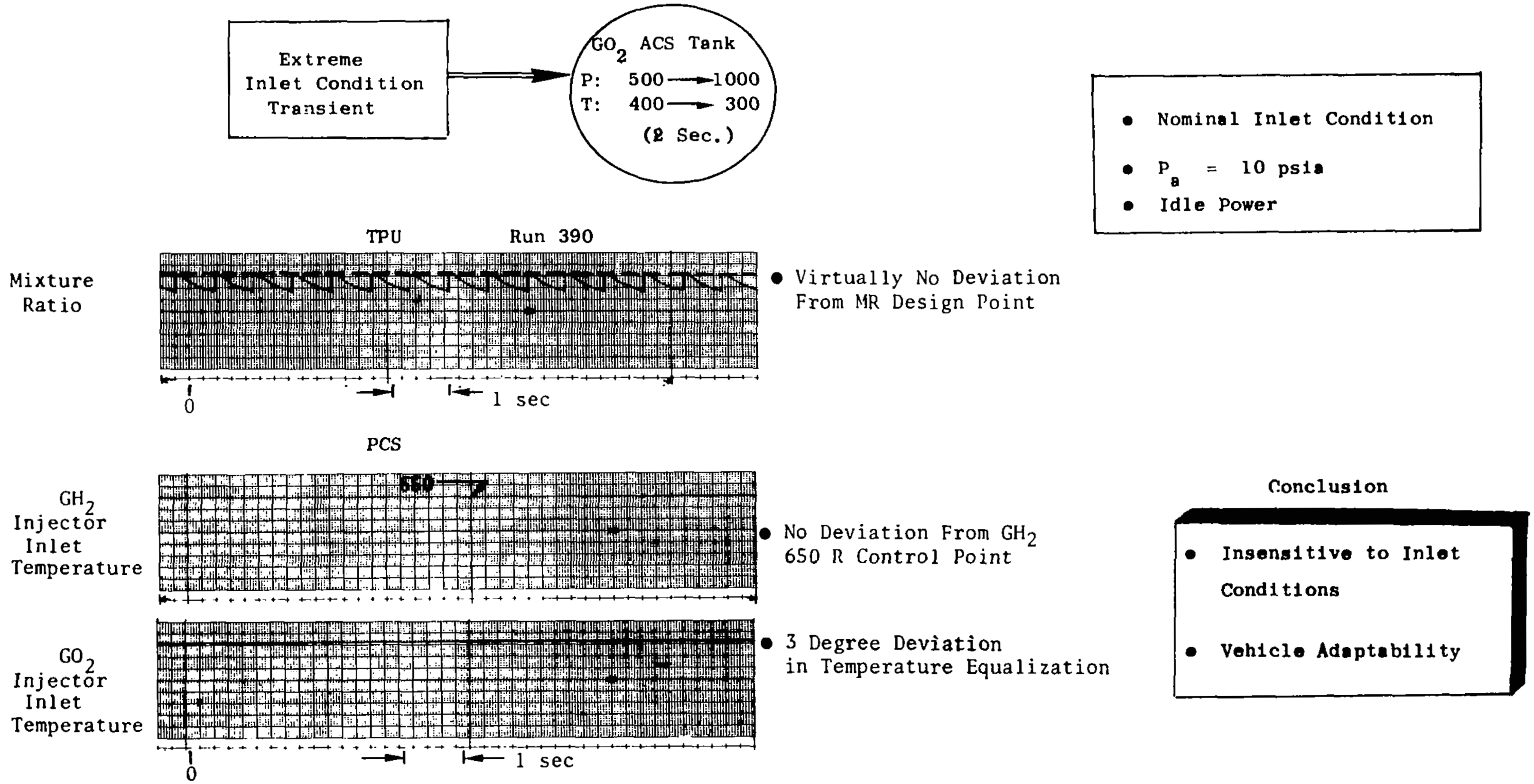
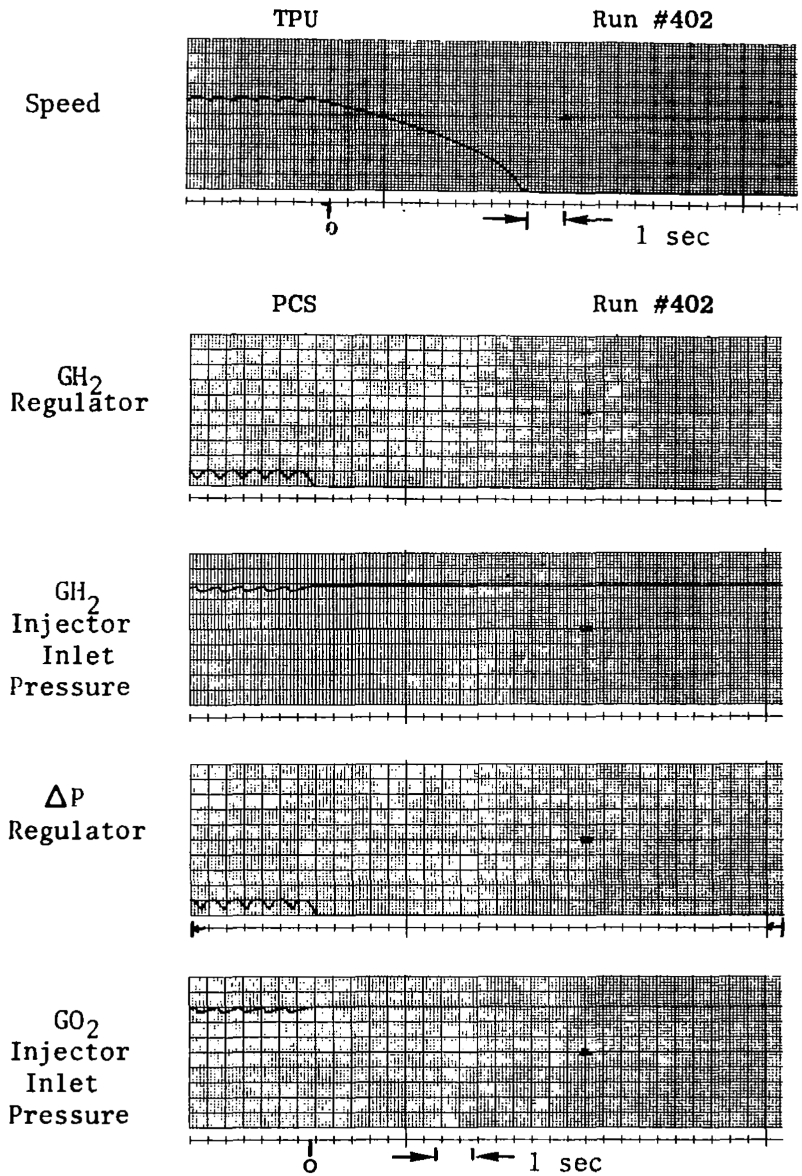


Figure 43. Transient Performance - Simulated ACS Tank Pump-Up



- SPEED DECAYS IN 5.8 SEC.

- NOMINAL INLET CONDITION
- P_a = 14.7 PSIA
- IDLE POWER CONSTANT TO 10% SPEED

- GH_2 AND Δ P REGULATOR PREVENTS OVERPRESSURIZATION OF ATTENUATOR TANKS

CONCLUSION

- SATISFACTORY APU SHUTDOWN
- NO AUXILIARY CONTROLS REQUIRED
- HIGH ALTITUDE SHUTDOWN 1 SEC. LONGER

Figure 44. Transient Performance - Shutdown (Pulse System)

system, the increased inertia will reduce the speed excursion resulting from load demand variations. These effects are quantitatively shown in Fig. **45.**

A design modification to the APU was considered during the Phase II study which involved a relocation of the regenerator downstream of the hydraulic and lube oil cooler as shown in Fig. **46** . The incentives for this modification were as follows:

A slight system weight advantage results from an improved SPC. Substantially higher regenerator exit temperatures and hence TPU inlet temperatures are possible since no cooling occurs downstream of the regenerator. A 1000 R TPU inlet temperature produces 7.0 percent reduction in SPC as compared with a 650 TPU inlet temperature; this gain is somewhat offset however, by the increased regenerator size.

This design would reduce the thermal stress and thermal shock effects in the regenerator due to increased hydrogen inlet temperatures.

Finally the operational characteristics could potentially be improved by closer proximity of the regenerator and bypass valve to the TPU thereby improving the control response of the valve to TPU inlet temperature.

The design modification was mechanized on the analog model and both steady state and transient performance evaluated. In summary, while control of TPU hydrogen inlet temperature was improved from a response standpoint, the overall APU control system became more complex. In addition, if higher (1000 R) injector inlet temperatures were utilized, precaution had to be taken against exhaust gas condensation due to increased regenerator head flux at high power levels. Figure 47 describes steady state operating characteristics between idle and maximum power level. At idle power (depressurized hydraulic pumps) the inlet hydrogen temperature to the regenerator is 395 R. The bypass valve (at 18 percent open) controls hydrogen TPU inlet temperature to 1000 R. The regenerator heat flux at this power level results in an exhaust gas temperature leaving the regenerator of 832, well above the condensation limit. Turbine inlet temperature is controlled at 2060 R with a low mixture ratio of.647, and, resulting in a 7 percent reduction in SPC as compared with the baseline APU. At maximum power level, the hydrogen TPU inlet temperature must be reduced from the 1000 R control point in order to prevent freezing of the exhaust gas within the regenerator. This would result from the high heat flux and reduced hydrogen inlet temperature to the regenerator of 170 R. Therefore a limit control is required to reduce the TPU inlet temperature to 653 R which then permits a safe regenerator exhaust gas temperature of 738 R. Along with this limit control, however, a continuous mixture ratio adjustment is also necessary since the reduced injector inlet temperature would result in approximately a 310 R reduction in turbine inlet temperature. In addition to the above controls an active lube oil bypass control should be incorporated since hydrogen temperatures in the cooler are low and vary substantially over the operating range.

POWER CONTROL CHARACTERISTICS - SUMMARY
(Pulse & Pressure Modulated Systems)

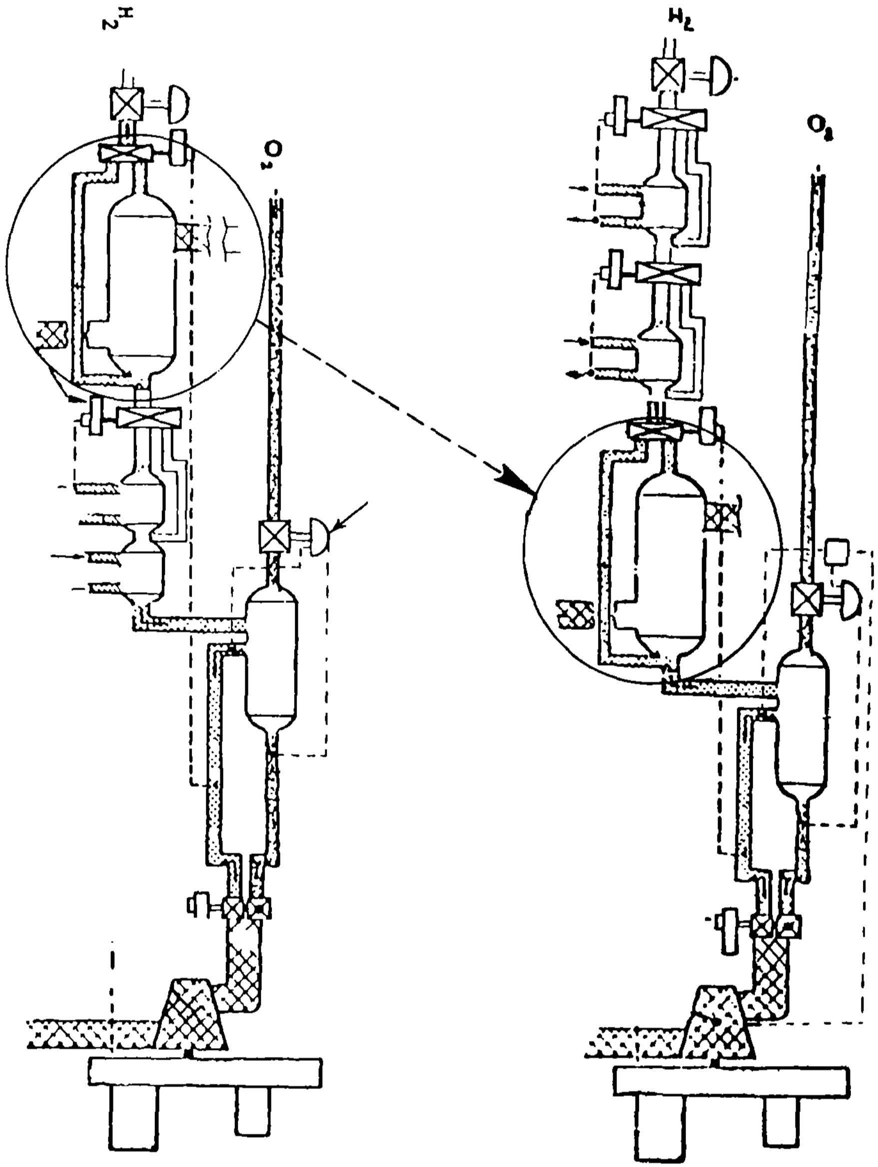
PULSE SYSTEM

MAXIMUM TRANSIENT SPEED DEVIATION	+ 3.5% - 2.7%
IDLE POWER P_a = 12 PSIA NOMINAL INLET COND.	.103 SEC ON-TIME (MINIMUM) .92 SEC OFF-TIME
TOTAL MISSION CYCLES	ORBITER: 10,380 BOOSTER: 12,370

PRESSURE MODULATED SYSTEM

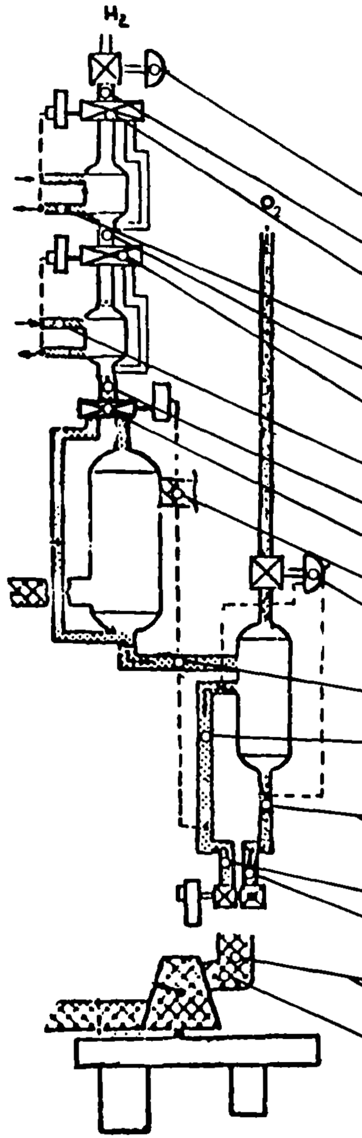
MAXIMUM TRANSIENT SPEED DEVIATION (IDLE TO MAXIMUM POWER SPIKE)	± 2070 RPM (± 3.45 PERCENT)

Figure 45. Power Control Characteristics - Summary (Pulse and Pressure Modulated Systems)



ADVANTAGES	DISADVANTAGES
• REDUCED SPC - 7% • FREEZING POSSIBILITY LOWERED • REDUCED COOLER WEIGHT	• INCREASED CONTROL COMPLEXITY • LARGER REGENERATOR • HIGH G.G. INLET TEMPERATURE (VALVE AND IGNITION SYSTEM)

Figure 46. Alternate Regenerator Placement



	POWER	
	IDLE #531	MAX. #532
X	9.84	66.88
P	482.2	476.9
X_{BP}	50.9	80
$T_{H/O}$	624.5	610
T	346	158.6
X_{BP}	73.0	96.4
$T_{L/O}$	682.5	671
T	394.9	169.8
X_{bP}	17.9	40.0
T	832	738
X	7.0	62.9
T	1035	672
P	481.7	460.3
T	1000	653
P	479.5	463.9
T	1051	635.2
$\dot{M}$	.0171	.120
$\dot{M}$	.0110	.094
P	51.25	363
T	2061	1974
MR	.647	.782

- NOMINAL INLET CONDITIONS
- PRESSURE MODULATED CONTROL
- REGENERATOR SIZE: z x
- P_a = 10 PSIA

CONCLUSIONS

- TIT LIMIT CONTROL MUST BE PRIMARY CONTROL
- REQUIRES LIMIT CONTROL ON REGENERATIVE EXHAUST GAS TEMPERATURE
- REQUIRES LUBE OIL BYPASS CONTROL

Figure 47. SS/APU - Alternate Design
Increased Injector Inlet Temperature

A power transient was conducted with the alternate design utilizing a pressure modulated power control and is demonstrated in Fig. 48 . Nominal inlet conditions were imposed on the APU at 10 psia ambient pressure. The gear box load was ramped between idle and maximum power in 1.8 seconds. Since the duration of maximum power was only about 3 seconds, the above described limit controls were not required to either reduce TPU inlet temperature or readjust mixture ratio. The maximum turbine inlet temperature excursion during the load transient was 100 R above design. The three second duration maximum power operation caused a temporary dip in TPU inlet temperature to 920 R, however the equalizer maintained oxygen temperature within 50 R of hydrogen temperature. The ΔP regulator equalized oxygen and hydrogen pressures as in the baseline APU system, and it may be concluded that all parameters were satisfactorily controlled during the simulated power transient.

It is concluded that the alternate APU design does not offer sufficient advantage (either operational or in system weight) over the baseline APU. In light of the added complexity, the baseline APU is considered superior.

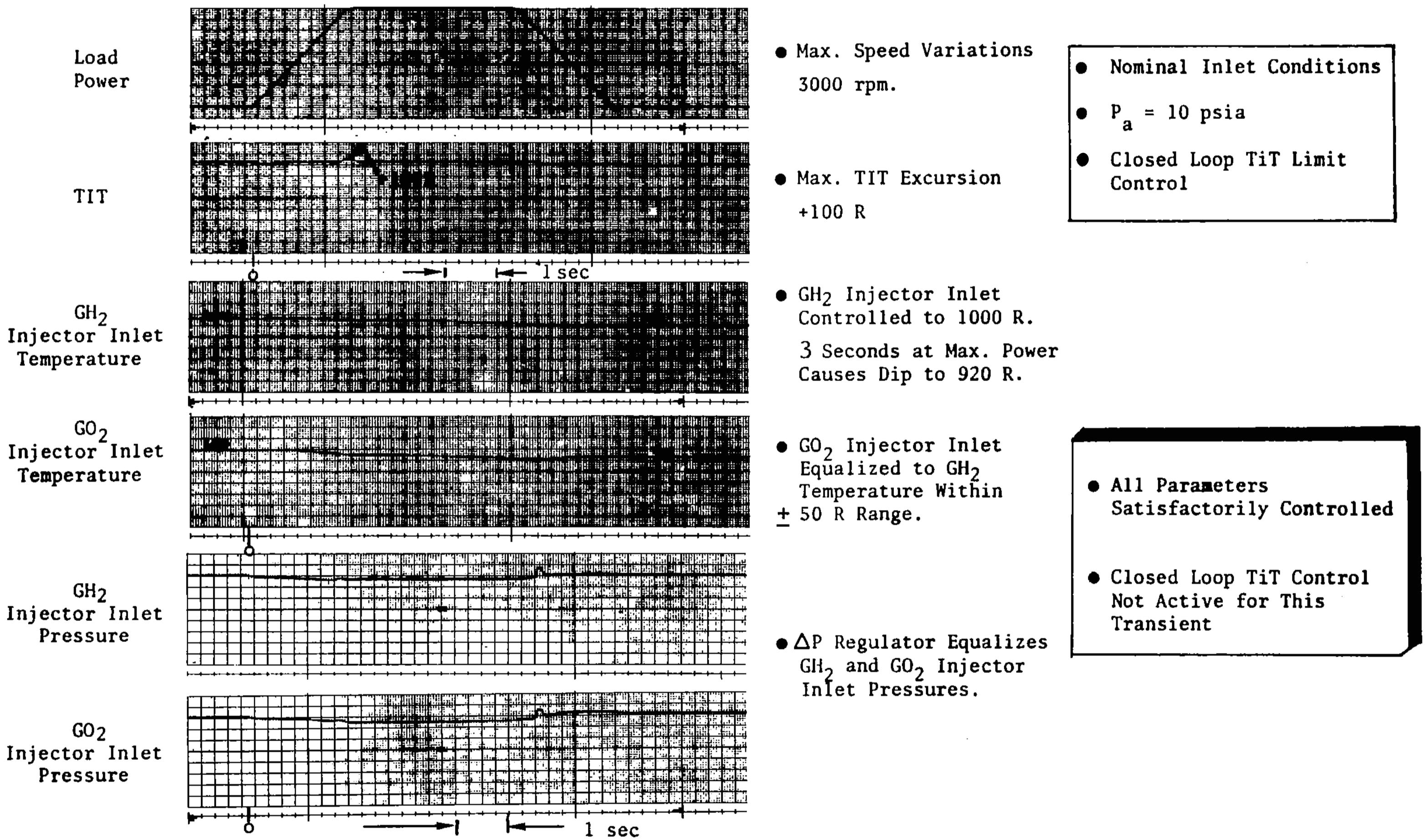


Figure 48. SS/APU Alternate Design - Power Demand Variations (1.8 Sec. Ramp: Idle-Max. Power)

VI FAILURE MODE ANALYSIS

A detailed reliability and maintainability study was conducted on the baseline APU system. This study consisted of a failure mode and effects analysis, a reliability estimate, and a maintainability analysis. The Failure Modes amd Effects Analysis (FMEA) performed on the APU resulted in the identification of over 100 separate failure modes. Each of these failure modes was considered with respect to its relative probability of occurrence as well as its possible effect on the vehicle mission. The failure effects categories ranged from those failure modes which involve possible structural damage to the vehicle to those which have no effect on vehicle or APU performance but do require a repair action between missions.

The procedure employed in the FMEA study is illustrated in Fig. 49 . Having identified a failure mode and its relative probability, the effect on system performance was evaluated. A fault detection sensor was also identified for in flight recording. This detection led either to ground maintenance or some form of fault compensation.

The fault compensation techniques fell into three categories: (1) component redesign wherein design modifications were incorporated to either reduce the probability of a failure or reduce the severity of the failure; (2) Redundancy compensation incorporated to reduce the failure probability; and (3) control compensation wherein controls were conceptually designed to permit continued operation under certain failure modes and/or to effect a safe shut down. The analog model was utilized primarily for evaluating failure effects and for evaluation of control compensation techniques.

A. DESIGN COMPENSATION TECHNIQUES

Figure 50 describes some of the failure modes which resulted in design compensation.

(1) Hydrogen pressure regulator - fails in open position due to rupture of the regulator diaphragm. This could under certain inlet conditions result in low turbine inlet temperatures and loss of power (when the oxygen APU supply pressure is less than the hydrogen supply). The design compensation incorporates an Inconel metallic diaphragm to significantly reduce the probability of failure. Should the failure occur in spite of the design compensation, fault detection would be provided by a pressure transducer located downstream of the regulator and result in initiation of a safe shut down.

(2) Hydraulic or lube oil cooler-internal leakage. This failure could result in contamination of the oil systems. The design compensation consists of a buffered zone incorporated in both the hydraulic and lube oil coolers which provides two unwelded walls between the oil and hydrogen. In the event of a failure of one of the walls, detection would be provided by a pressure transducer. A vehicle decision would then be made to continue APU operation or effect a safe shut down procedure.

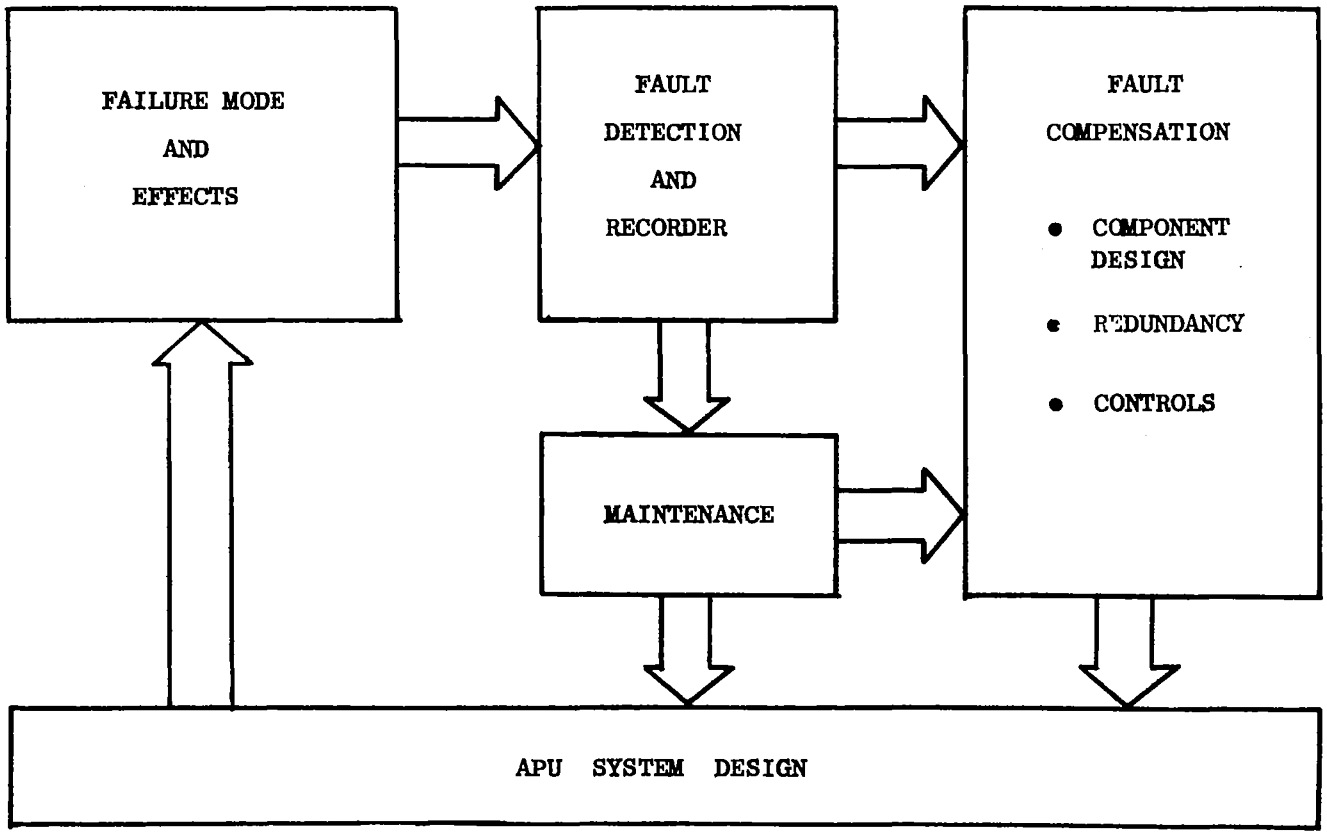


Figure 49. Failure Mode Analysis

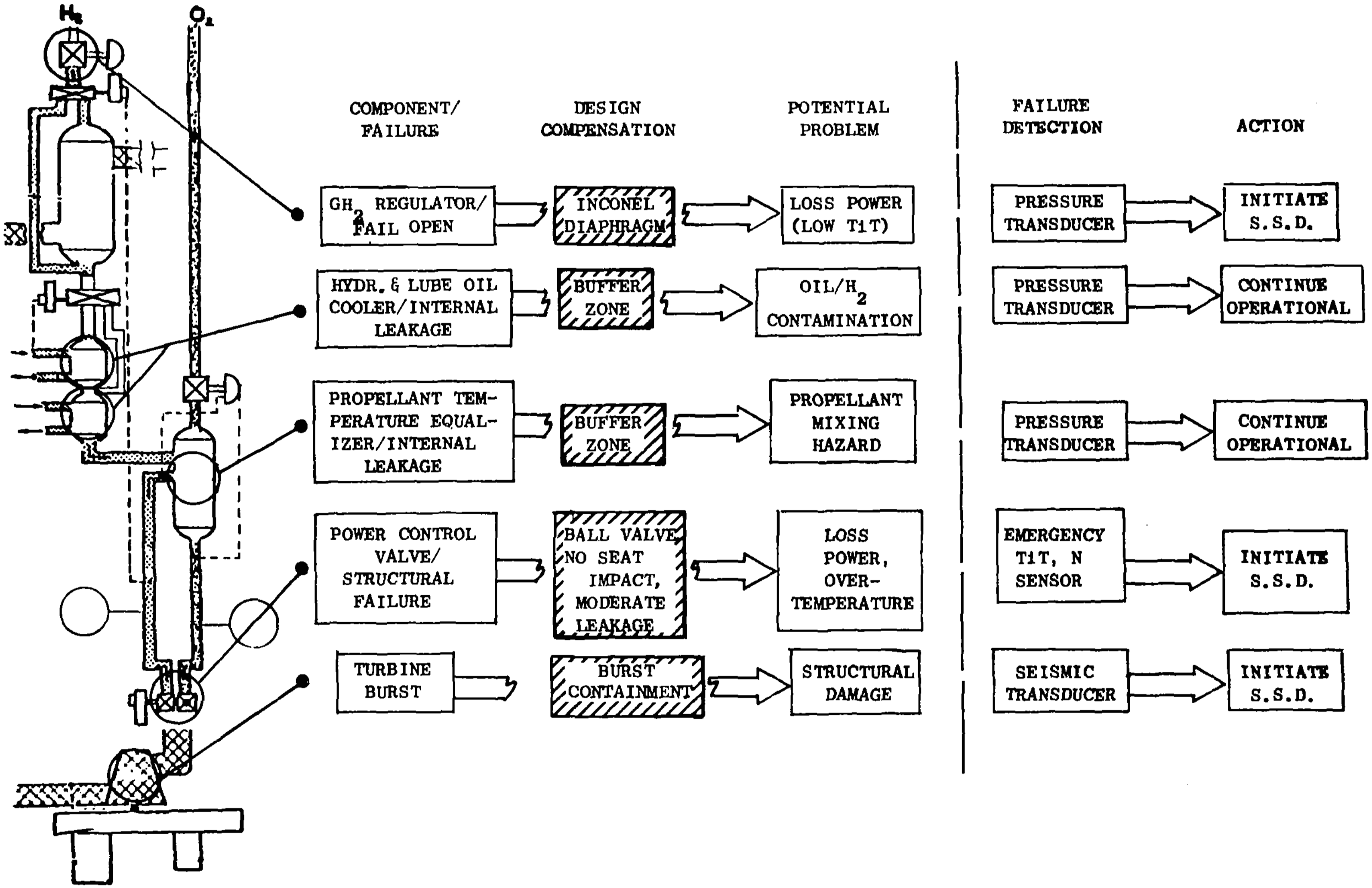


Figure 50. FMA - Design

(3) Propellant temperature equalizers - internal leakage. This failure would result in a propellant mixing hazard. The design compensation consists of a buffered zone designed to provide two unwelded walls between the hydrogen and oxygen. Should a failure of one of the walls occur, detection could be provided by a pressure transducer which would result in either continued APU operation or a safe shut down depending upon vheicle considerations.

(4) Power control valve - structural failure. This could result in loss of APU power or an over temperature condition depending on the nature of the failure. Design compensation consists of the use of a ball valve. Stops for the valve are incorporated in the actuator thereby reducing the probability of structural failure as a result of a large number of cycles involving seat impact. Should a failure occur however, protection would be provided by an emergency over temperature switch or an under speed detection which would initiate a safe shut down of the APU.

(5) Turbine burst represents a potential structural damage to the APU and surrounding components. Ultimate design compensation consists of burst containment for the turbine. A seismic transducer is employed to anticipate this failure mode through a detection of severe vibrations which would immediately initiate a safe shut down.

B. REDUNDANCY COMPENSATION TECHNIQUES

Redundancy was incorporated in the APU design as a result of certain failure modes and is illustrated in Fig. 51.

(1) Failure of the turbine inlet temperature limit control due to a sensor or circuit malfunction, could result in reduced specific power consumption or loss of power for the APU due to a reduced turbine inlet temperature. Redundant compensation is incorporated by utilizing multiple sensors, dual coils for the pressure divider servo and an averaging circuit to determine sensed turbine inlet temperature. In the event of a failure however, detection would be provided by a low emergency turbine inlet temperature circuit which would be used to first deactivate the turbine inlet temperature limit control, and allow continued operation of the APU. Sustained under temperature would result in initiation of a safe shut down.

(2) Failure of the speed controls due to either sensor or circuit malfunction could result in overspeed and structural damage to the APU. Redundant compensation is incorporated in the form of redundant sensors and a voting circuit for speed sensing and dual coils on the power control valve. Should an overspeed occur in spite of the redundant design, failure detection is provided by an emergency over speed circuit which would initiate a safe shut down to the APU.

(3) Failure of the ignition system could result in unburned propellant in the combustor and turbine inlet manifold and a potential hazard due to auto-ignition of large quantities of unburned propellant. Redundant compensation is provided by the use of dual ignition circuits. In addition, continuous ignition of both circuits is provided to simplify the curcuit logic and reduce the probability of no ignition upon valve opening.

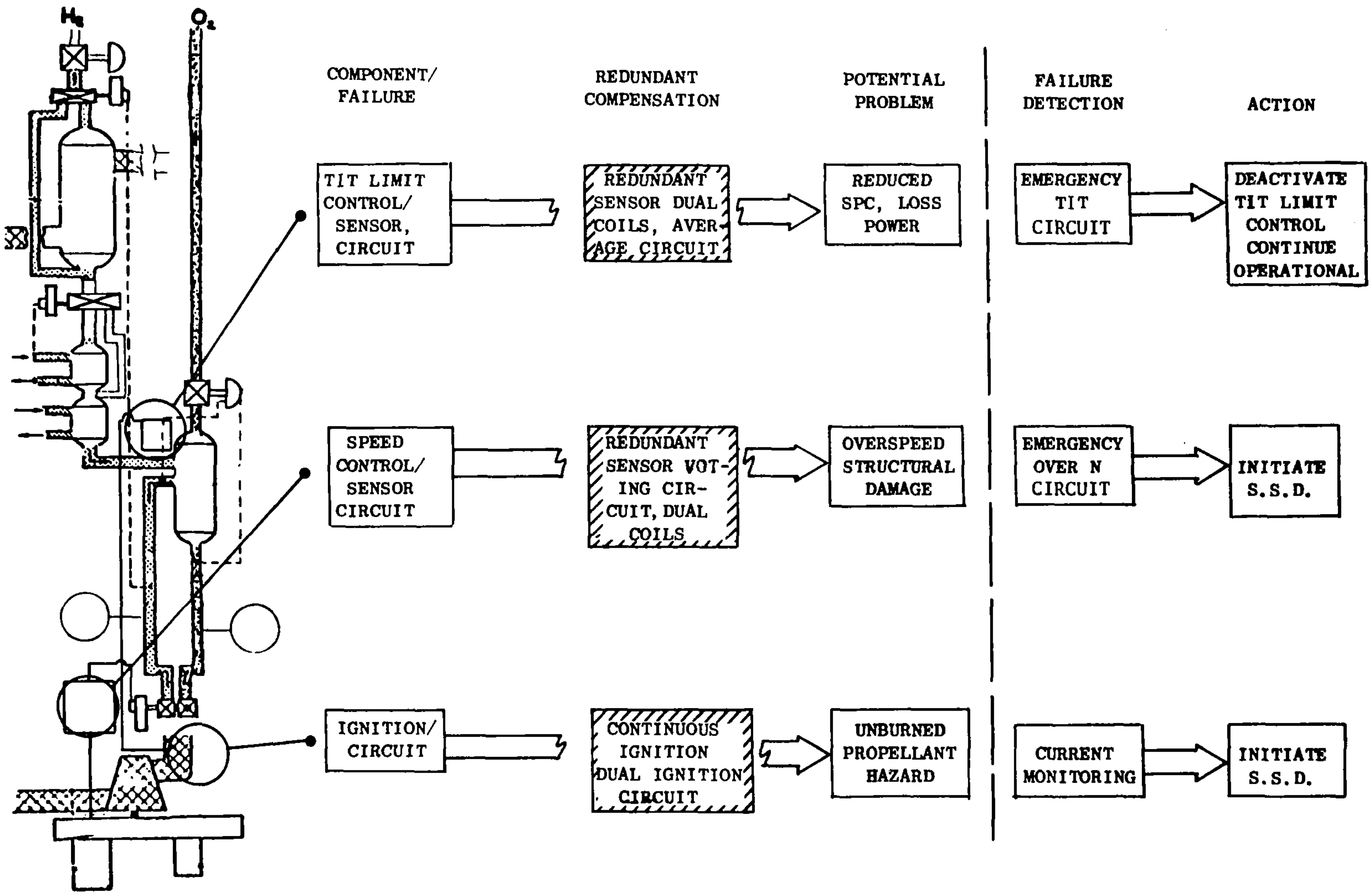


Figure 51. FMA - Redundancy

C. CONTROL COMPENSATION TECHNIQUES

Control compensation was designed into the APU to either reduce the effects of various failure modes, allow continued operation, or to provide safe shutdown of the APU. These are described in Fig. 52 . A total of 40 analog model runs were conducted during this portion of the FMEA.

(1) Drift of the hydrogen pressure regulator. This would tend to result in high or low turbine inlet temperature. Control compensation is automatically provided by the ΔP regulator which equalizes oxygen/hydrogen pressure. Detection of the failure would be sensed by a pressure signal downstream of the regulator - APU operation could continue, or a safe shut down initiated, depending on vehicle considerations.

(2) Drift high of the ΔP regulator could result in high turbine inlet temperatures. Control compensation consists of the turbine inlet temperature limit control which would bias the regulator to offset its drift. Detection of the failure would be provided by an oxygen pressure signal which would either initiate a safe shut down or allow for continued operation of the APU.

(3) Loss of load by the hydraulic pump could result in structural damage due to overspeed. Control compensation is automatically provided by the normal speed control function which prevents any overspeed condition. Detection of the failure could be provided by a low hydraulic pressure signal which would initiate a safe shut down.

(4) Contaimination of the hydrogen injector could result in high turbine inlet temperatures. Control compensation is provided by the turbine inlet temperature limit control, which would bias the ΔP regulator to offset the change in flow resistance. Failure detection would be provided by a high oxygen pressure signal at the equalizer exit, and allow continued APU operation or initiate a safe shut down depending on the vehicle considerations.

(5) Excessive heat load in the hydraulic system can result in high lube oil temperatures or a high turbine inlet temperature. Control compensation is provided by the lube oil temperature limit control which temporarily reduces the hydraulic cooling in order to protect the lube oil system. Detection of the failure would result from a lube oil temperature in excess of 730 R. Safe APU operation would continue for several minutes depending upon the exact heat load and power level.

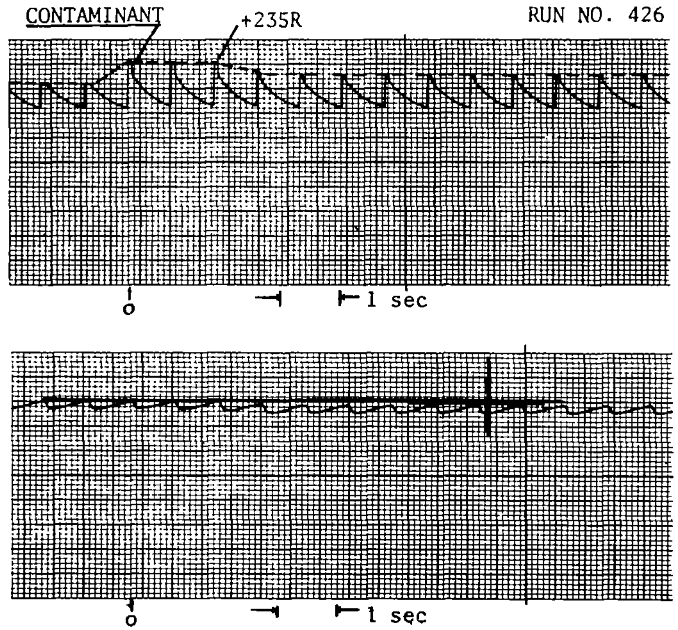
Excessive heat load in the lube system, failure of the regenerator bypass valve and failure of the hydrogen cooler bypass valve were dynamically evaluated in a similar manner. As in the previous cases, operation could either be continued for a limited or indefinite period due to incorporation of the compensation controls or a safe shut down could be initiated immediately, depending on vehicle considerations. Application of the analog model in evaluating two of the failure modes, and the effect of control compensation is demonstrated in the next two figures. Fig. 53 illustrates the effect of a 15 percent reduction in hydrogen injector area as a result of contamination. This failure was evaluated at idle power, and 10 psia ambient pressure and nominal inlet conditions. Immediately following contamination of the injector, turbine inlet temperature increased 235^{o} above the design point.

COMPONENT/FAILURE	PROBABILITY	POTENTIAL	CONTROL/COMPENSATION	DETECTION/LOGIC	ACTION
● GH_2 PRESS/REG DRIFT	LOW	LOSS POWER	ΔP REGULATOR EQUALIZES PRESSURE (WITHIN LIMITS)	LOW, HIGH PRESS	CONTINUE OPERATIONAL <u>OR</u> INITIATE SSD
● ΔP REG/DRIFT HIGH	LOW	HIGH TIT	TIT LIMIT CONTROL	HIGH O_2 PRESS	CONTINUE OPERATIONAL <u>OR</u> INITIATE SSD
● HYDRAULIC PUMP/LOSS LOAD	VERY LOW	STRUCTURAL DAMAGE (OVER SPEED)	NORMAL SPEED CONTROL FUNCTION PREVENTS OVER-SPEED	LOW HYDRAULIC PRESSURE	INITIATE SSD
● CONTAMINATION GH_2 INJECTOR	LOW	HIGH TIT	TIT LIMIT CONTROL	HIGH ΔP	CONTINUE OPERATIONAL <u>OR</u> INITIATE SSD
● HYDRAULIC SYSTEM/ EXCESSIVE HEAT LOAD	VERY LOW	HIGH LUBE OIL TEMP HIGH TIT	$T_{L/O}$ OR $T/_{H_2}$ LIMIT CONTROL	HIGH $T_{H/O}$	CONTINUE OPERATION $\sim$10 MINUTES <u>OR</u> INITIATE SSD
● LUBE SYSTEM/ EXCESSIVE HEAT LOAD	VERY LOW	HIGH LUBE OIL TEMP	$T_{L/O}$ LIMIT CONTROL	HIGH $T_{L/O}$	CONTINUE OPERATIONAL <u>OR</u> INITIATE SSD
● REGEN BYPASS VALVE /LOCKED OPEN	VERY LOW	LOSS POWER LOW $T_{L/O}$	TEMP EQUALIZER HOLDS M/R FULL POWER RANGE DELIVERED	LOW INJ. INLET T OPEN VALVE	CONTINUE OPERATIONAL 8 TO 200 SEC <u>OR</u> INITIATE SSD
/LOCKED CLOSED	VERY LOW	HIGH TIT HIGH $T_{L/O}$	TEMP EQUALIZER HOLDS M/R TIT LIMIT CONTROL	HIGH INJ. INLET T CLOSED VALVE	CONTINUE OPERATIONAL 30 SEC, <u>OR</u> INITIATE SSD
● HYDROGEN COOLER BYPASS VALVE/LOCKED OPEN	LOW	HIGH $T_{H/O}$	REGEN BYPASS CONTROL HOLDS T_{INJ} AND TIT.	$T_{H/O}$ OVER TEMP AND VALVE OPEN	CONTINUE OPERATIONAL $\sim$5 MIN (FULL POWER) <u>OR</u> INITIATE SSD
/LOCKED CLOSED	LOW	NO PROBLEM	REGEN BYPASS CONTROL HOLDS T_{INJ} AND TIT.	$T_{H/O}$ UNDER TEMP AND VALVE CLOSED	CONTINUE OPERATIONAL <u>OR</u> INITIATE SSD

Figure 52. APU Reliability Control Compensation

FAILURE: CONTAMINATION OF GH_2 INJECTOR AREA - 15% REDUCTION

TIT



GO_2
INJ INLET
PRESSURE

235R OVER TEMPERATURE

TIT SENSOR EFFECTIVE
AFTER 3 PULSES

A 12 PSI ΔP CORRECTION
EFFECTIVE AFTER 3 SEC

TIT CORRECTED TO
+75R ABOVE DESIGN

- NOMINAL INLET CONDITIONS
- IDLE POWER
- P_a = 10 PSIA

CONCLUSION

- TIT LIMIT CONTROL PROVIDES "FAIL-OP"
- INITIATE SSD IF DESIRABLE

Figure 53. APU Reliability - Control Compensation

The turbine inlet temperature sensor with a .5 second lag became effective after 3 pulses and generated a 12 psi correction, or bias to the ΔP regulator. As a result of this limit control action, turbine inlet temperature was corrected down to 75° above design. This demonstrates the effectiveness of the limit control utilizing a rugged long life sensor to provide a "fail-op" condition.

Fig. 54 describes a simulated excessive heat load in the hydraulic system resulting in a high case drain temperature. As shown in run #408 the hydraulic cooler bypass valve closes as the case drain temperature exceeds 650 R. Due to an increase in heat flux into the hydrogen the lube oil temperature exceeds 750° after 70 secs, and the hydrogen injector inlet temperature increases 80° above the control value of 650 R. In spite of maximum hydraulic cooling, the case drain temperature rises to 712° in 8 minutes. Run 409 at the bottom of the chart depicts the same failure utilizing control compensation in the form of a lube oil temperature limit control. By overriding the basic hydraulic cooler bypass control and thereby causing cooler fluid to enter the lube oil, lube oil temperatures were held to a maximum of 736°. The hydrogen injector inlet temperature was held to a maximum of 664°. The bias on the hydraulic cooler bypass valve resulted in the valve opening to 24 percent. It is interesting to note, however, that the hydraulic case drain temperature increased to a maximum of 729° in 8 minutes, or only 17 degrees higher than the previous case in which maximum hydraulic cooling existed. During that 8 minutes, the lube oil and hydrogen injector inlet temperatures were held within close proximity to their normal controlled value. It is concluded that a "fail op" situation can be provided by the control compensation network under a severe excessive heat load for several minutes. The APU is automatically returned to normal operation should the excessive heat load be eliminated.

D. CRITICAL FAILURE MODES

The most critical failure modes of those studied during the FMEA were identified in terms of: (1) their possible effects on the mission and on vehicle safety; (2) the probability of occurrence of the particular failure mode.

With respect to the first category, the two most cricical failure modes which were identified are failure to ignite in the combustion chamber resulting in explosion hazard and turbine speed control failure resulting in turbine overspeed. In each of these cases, specific compensating action was taken in the design of the APU, such that the probability of occurrence of vehicle damage is substantially reduced.

Those failure modes which have the highest probabilities of failure are the hydrogen and oxygen pressure regulators, the heat exchanger bypass valves, and the propellant control valves. Compensation actions in the event of occurrence of one of these failure modes, as previously discussed, provides for either a "fail op" condition or initiation of a safe shutdown.

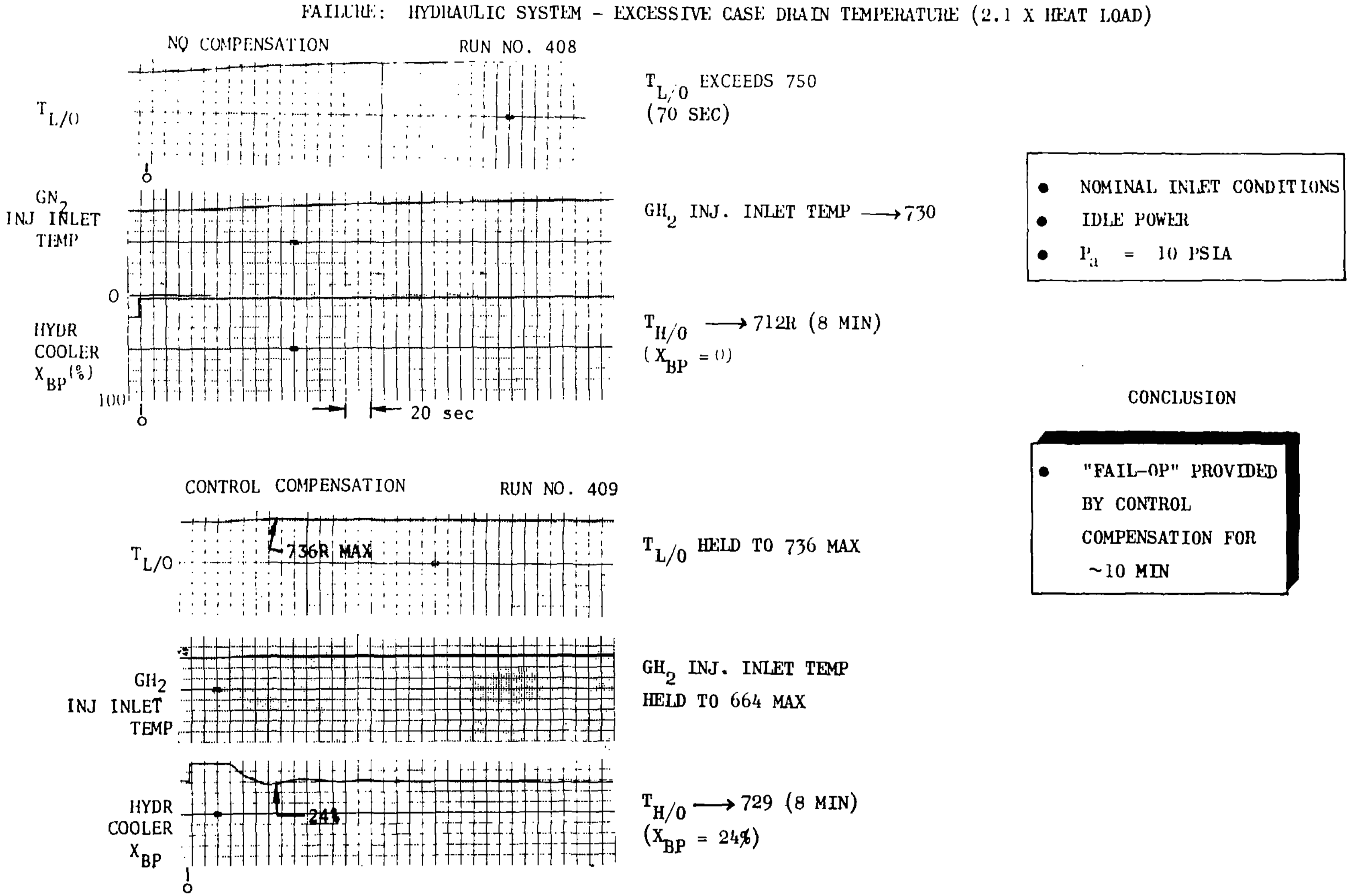


Figure 54. APU Reliability - Control Compensation

VII TURBOPOWER UNIT

A. SPECIFICATIONS

To place the turbopower unit in its proper frame of references it is appropriate to review the APU specifications as applied to the TPU. The major specifications are summarized in Fig. 55 . The output loads are assumed to be one Westinghouse 40/50 KVA alternator and two ABEX AP27V hydraulic pumps. The alternator operates at 12,000 rpm and is oil spray cooled. This cooling system is integrated with the TPU lubrication system. The hydraulic pumps produce 103 gpm at 3,000 psi while operating at 6,000 rpm.

The turbopower unit is required to produce a peak output shaft power to the loads of 400 horsepower under all conditions of altitude from sea level to space. Similarly, it is required to be capable of operating at a minimu power level of 33 horsepower under the same conditions. Further, the TPU must be designed to operate under these conditions for 1,000 hours at the design temperature conditions and an additional 2,000 hours under ground checkout conditions.

More specifically, the turbine is required to be a two-stage pressure compounded machine operating at a nominal speed of 60,000 rpm. The turbine is to operate with the nominal inlet temperature of 2060 R using uncooled Astroloy discs. The housing is to be designed to contain a burst of these discs.

The gear box transmits the power from the turbine to the loads at the design speeds. The bearings in the gear box are to be designed for a 3,000 B-1 life. The gear box lubricant as well as alternator coolant is to be MIL-L-7808. The system which circulates this lubricant and coolants must be capable of operating at any APU attitude and for extended periods of time under zero gravity conditions.

B. TPU DESIGN FEATURES

The reference turbopower unit meets all the requirements of Fig. 55 . It has additional features, some of which are described in Fig. 56 . Basic turbine hardware is usable over a wide range of output power varying from approximately 200 to 800 horsepower. With the exception of the gas flow path, the TPU is adaptable without modification to either a pulse or pressure modulated power control.

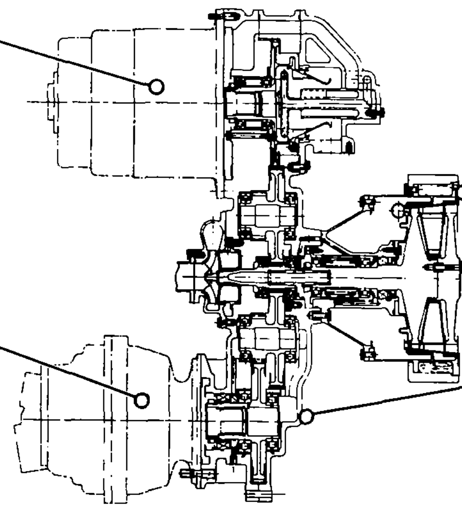
The turbine operates in the reference unit primarily at the design point. This is due to the use of the pulsed control system which operates at a fixed inlet pressure. While the turbine exhaust pressure varies somewhat over the altitude range, the pressure ratio across the turbine does not vary much. The turbine is a two-stage partial admission machine designed to have supersonic velocities relative to the rotor blades. Mechanically the turbine is designed with sufficient flexibility in the housing elements to minimize the possible thermal distortion due to the combination of high temperatures and partial admission.

ALTERNATOR

- WESTINGHOUSE 40/50 KVA
- OIL SPRAY COOLED
- 12,000 RPM

HYDRAULIC PUMPS (2)

- ABEX AP 27V
- 103 GPM, 3000 PSI
- 6000 RPM



TURBINE

- 2 STAGE, PRESSURE COMPOUNDED
- 60,000 RPM
- 2060 R INLET TEMPERATURE
- ASTROLOY DISKS, UNCOOLED
- BURST CONTAINMENT
- ROTOR INERTIA--0.011 LB-FT-SEC^2

GEARBOX

- 3000 HR B-1 BEARING LIFE
- MIL-L-7808 LUBRICANT
- ZERO G AND ALL ATTITUDE LUBRICATION--DESIGN BASED ON BRAYTON ENGINE TECHNOLOGY

TURBOPOWER UNIT

- DESIGN LIFE
  - 1000 HR AT DESIGN TEMPERATURE
  - 2000 HR AT GROUND CHECKOUT CONDITIONS
- ALTITUDES - SEA LEVEL TO SPACE
- OUTPUT SHAFT POWER
  - PEAK - 400 HP
  - MINIMUM - 33 HP

Figure 55. Reference Turbo Power Unit Specifications

TURBOPOWER UNIT

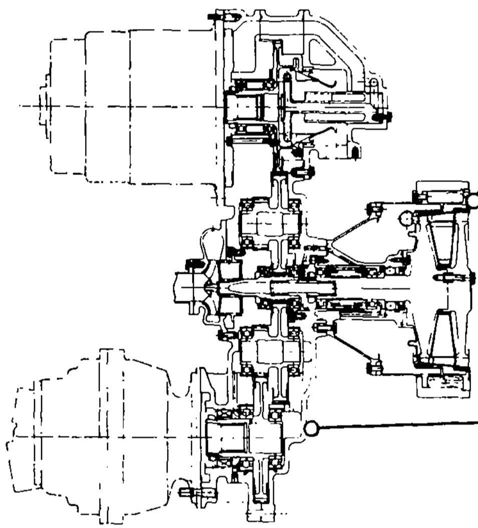
- BASIC TURBINE HARDWARE USABLE FROM 200-800+HP
- ADAPTABLE TO PULSE OR PRESSURE MODULATION

TURBINE

- INSENSITIVE TO INLET TEMPERATURE SPIKES
- DESIGN POINT OPERATION
- SUPERSONIC RELATIVE VELOCITIES
- MINIMUM THERMAL DISTORTION
- TWO-STAGE, PARTIAL ADMISSION

COMBUSTOR

- HIGH PERFORMANCE
- CONSTANT OPERATING CONDITION



GEARBOX

- CONVENTIONAL LONG LIFE, LOW LOSS DESIGN
- ZERO GRAVITY LUBRICATION SYSTEM
  - POSITIVE CIRCULATION
  - BASED ON PROVEN TECHNOLOGY
  - INTEGRATED WITH SPRAY COOLED ALTERNATOR

Figure 56. Reference Turbo Power Unit Features

One other major feature of the turbine is its insensitivity to potential spikes in inlet temperature. Both blade and disk cyclic life are limited in normal operation by the temperature gradients and corresponding strain range associated with a TPU start. An abnormally high temperature for several pulses, such as might be produced during large system load transients, results in temperature gradients and strains much lower than those associated with a start. Thus, the short-duration overtemperature spikes will not limit turbine life. (Continued overtemperature would cause ultimate turbine failure, but this is prevented by an overtemperature limit control.)

The combustor also functions at a constant operating condition, that is, fixed flow and pressure level. Since mixture ratios are held constant, the combustor temperature is also constant. The combustor is thus capable of giving consistently high performance over a wide range of TPU operating conditions.

The gear box is a conventional long life, low power loss design. One unique aspect of the gear box is the inclusion of a zero gravity lubrication system. This system which is based on proven technology utilizes positive circulation of a carrier gas to carry the oil mist through the system. It has also been integrated with the spray cooled alternator.

In Fig. 57 important features of the mechanical configuration of the TPU are pointed out. The alternator and hydraulic pumps are located on the same side of the gear box to give maximum ease of installation and permit easy field replacement of these devices.

Power is transmitted from the turbine shaft into the gear box through a quill shaft. This quill shaft tends to prevent any radial forces caused by turbine rotor dynamics from being transmitted into the pinion of the gear box, thereby minimizing gear misalignment problems.

The turbine housings have been designed to have sufficient flexibility to allow them to withstand radial, axial, and circumferential thermal gradients. All of these gradients will exist to some extent on a transient and steady-state basis. This housing flexibility is attained by using thin sections in cylindrical and conical surfaces.

One other feature unique to this TPU is the placing of the first stage disc inboard, that is, next to the turbine bearing. There are several reasons for doing this. First, it places the heaviest disc next to the bearing, thereby minimizing the overhung moment. Second, it places the maximum torque next to the bearing. (Normally in partial admission machines such as this, one of the major bearing loads is that due to the torque on a small fraction of the turbine arc). Third, the first stage runs somewhat cooler than if it were placed outboard, thus allowing a higher tip speed to be utilized for the same disc weight. Higher tip speed of course, means higher performance.

This configuration, with first stage disc inboard, has further advantages with respect to the turbine housings. Placing the inlet nozzles on the bearing side of the unit allows closer control of the nozzle location. Since the first stage is the smallest nozzle and therefore most sensitive to location, this is desirable. Having the exhaust flow outboard of the turbines allows considerable flexibility in the design of the exhaust passages. An exhaust diffuser is feasible. Also, it is much easier to make the flow turn toward the regenerator.

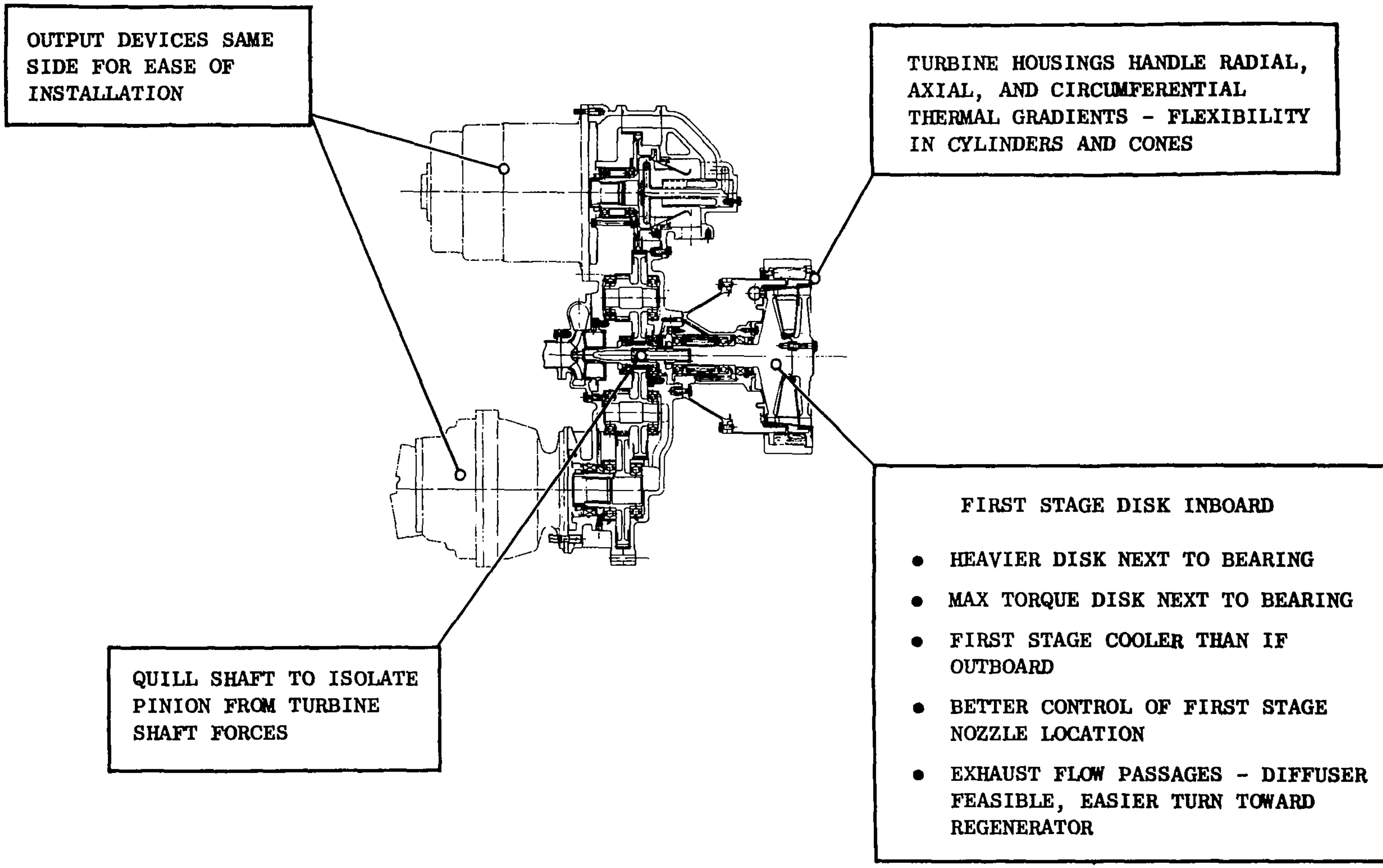


Figure 57. Turbo Power Unit Configuration Features

The layout of the machine is shown in Fig. 58 . In addition to those features discussed above, others will be brought out in the next sections as the components are discussed in more detail.

C. TURBINES

C 1. Design Procedure

The procedure used in design of the turbines for the reference TPU is summarized in Fig. 59 . It should be noted that, while the referenced TPU is a pulse modulated turbine, the design procedure is equally applicable to a pressure modulated turbine with supersonic relative velocities.

Figure 59 shows the step by step procedure followed in going from the turbine design conditions to design point performance, as well as those involved in the calculation of off-design performance and the thermal and stress limitations on the turbines. Each block in Fig. 59 represents a particular type of calculation. In most cases the calculation was performed using a computer program. The name or number of the computer program used is indicated in the block in parenthesis. For example, the nozzle design program is Rocketdyne's program No. 9R313, which uses the method of characteristics to design two-dimensional nozzles. The boundary layer program used next, BLAYER, is that of McNally. (1) The nozzle loss program, ZETAP, is based on the compressible form of loss equations given by Stewart. (2)

The rotor inlet program XINLET, is based on the use of method 1 of Boxer, et. al, (3). The rotor design program applied to the supersonic passages of the rotor is that of Goldman (4). Results of the application of these various programs and calculations in the design procedure will be presented next.

C2. Performance

The performance of the reference turbine design is summarized in Fig. 60 . The reference turbine is designed for a back pressure of 14.5 psia. This value corresponds to the back pressure expected to exist under full flow conditions with an 11 psia ambient pressure. The ambient pressure of 11 psia was selected because typical mission profiles show a large fraction of the

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(1) McNally, W. D., Fortran Program for Calculating Compressible Laminar and Turbulent Boundary Layers in Arbitrary Pressure Gradients, NASA TN D-5681, May 1970.
(2) Stewart, W. L., Analysis of Two-Dimensional Compressible-Flow Loss Characteristics Downstream of Turbomachine Blade Rows in Terms of Basic Boundary Layer Characteristics, NACA TN 3515, 1955.
(3) Boxer, E., Sterrett, J. R., and Wlodarski, J., Application of Supersonic Vortex-Flow Theory to the Design of Supersonic Impulse Compressor-Turbine Blade Sections, NASA RM L52B06, 1952.
(4) Goldman, L. F., and Scullin, V. J., Analytical Investigation of Supersonic Turbomachine Blading, I-Computer Program for Blading Design, NASA TN D-4421, March 1968.

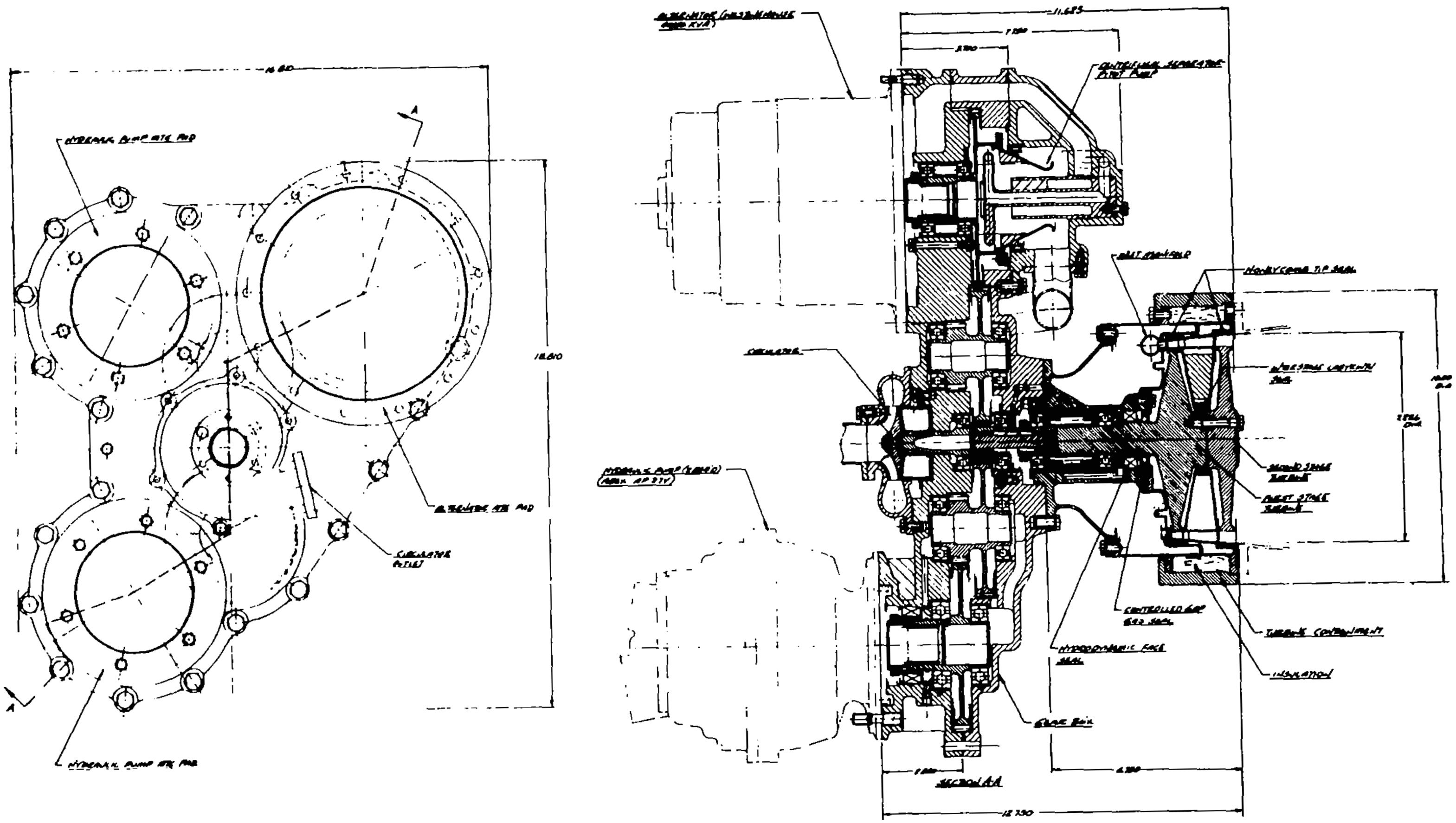

Figure 58. Turbopower Unit Layout

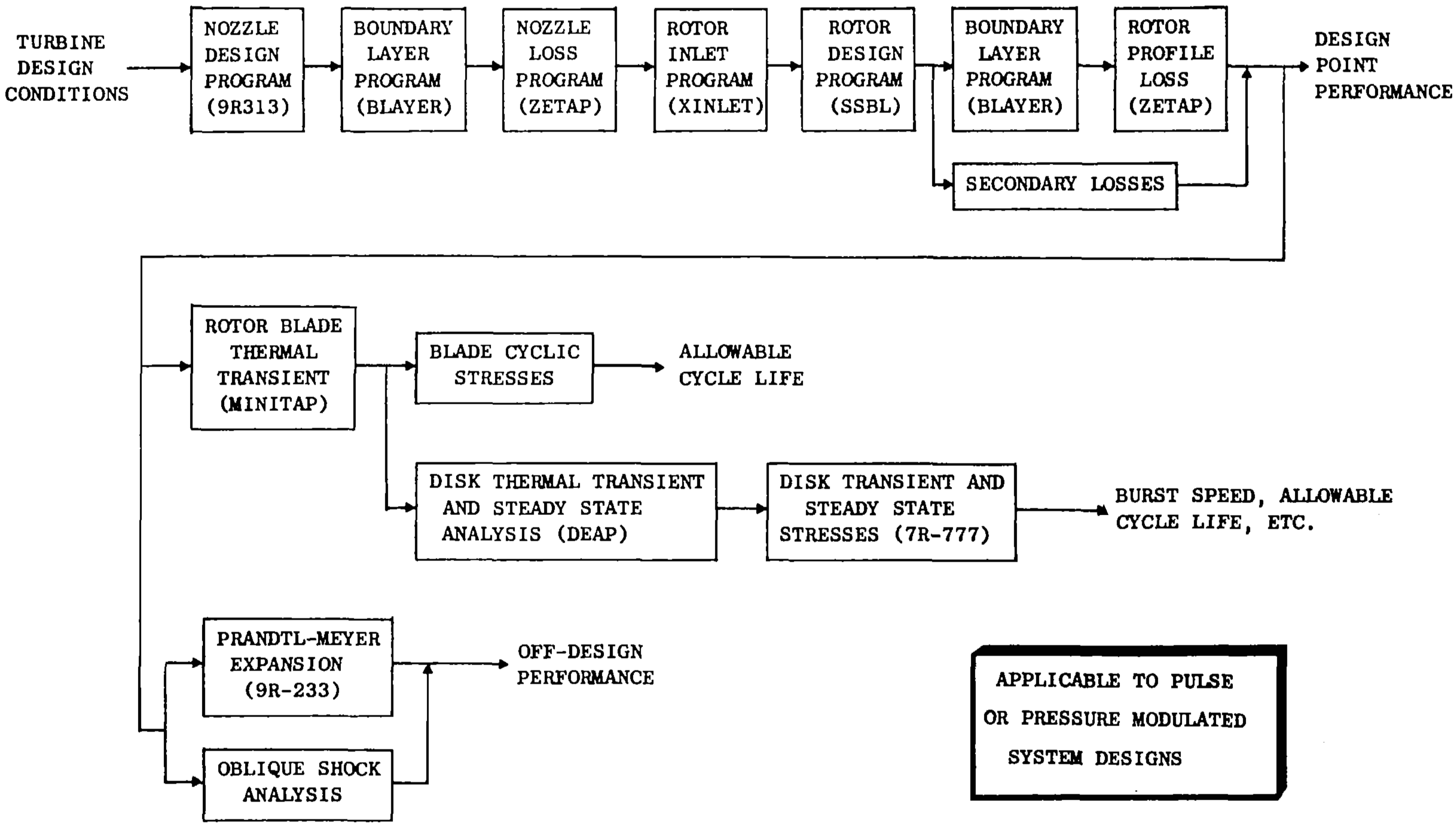

Figure 59. Supersonic Turbine Design Procedure and Computer Programs

APU operation at that level. The inlet pressure level of 390 psia was derived from a careful analysis of the minimum feasible pressure drops expected to exist in the propellant conditioning and combustion systems. At these design conditions the reference turbine will produce an output shaft power of 468.9 horsepower with a flow of 0.24 lbs/sec. This corresponds to an efficiency of 57.2 percent.

When the reference turbine is operated at other than design pressure ratios, the performance expected is that shown in Fig. 60 . The independent variable is the ratio of actual operating pressure ratio to design pressure ratio; the dependent variable is the ratio of work output, (foot-pounds per pound) to design work output of the machine. For a fully supersonic turbine such as the reference design, the calculation of off-design performance in the vicinity of the design pressure ratio is comparatively simple. So long as all four flow elements, the two stators and the two rotors, remain choked, the continuity equations remain satisfied through the machine. With supersonic flow maintained through the machine, the only place at which the effect of changing back pressures can be felt is in the vicinity of the trailing edge of the second stage rotor. The additional torque available due to reduced back pressure comes from a Prandtl-Meyer expansion in the vicinity of the trailing edge of the rotor. This expansion makes itself felt as reduced pressures along a portion of the suction surface at the trailing edge. As the back pressure is reduced, a point is reached where additional Prandtl-Meyer expansion takes place outside the blade passage. Thus, no additional pressure changes take place on the suction surface and no further torque increase is in evidence. This situation corresponds to the maximum work output.

On the other side of the design pressure ratio, as back pressure is increased, oblique shocks tend to form in the vicinity of the trailing edge, thus raising the pressure on the suction machine. As pressure continues to rise, a point is reached at which supersonic flow can no longer be maintained. The reference machine, at its highest pressure ratio (i.e., at sea level), will maintain supersonic flows through the rotor passage. Some oblique shocks on the suction surface trailing edge will reduce the torque output as shown in Fig. 60 .

Some of the details of the designs of the reference turbine are shown in Fig. 61 . The nozzle and rotor (1) blade configurations for both stages are shown. The individual stage efficiency and power numbers are quoted along with some of the key geometric details for the two stages. Comment is in order about the shapes of the nozzles for the two stages.

The first stage nozzles were selected as circular cross section nozzles with their attendant additional wake mixing losses in order to provide a proper match between the flow leaving the first stage and that entering the second stage, that is, to carefully pick up and guide the flow through both stages.

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(1) A final iteration on the rotor blade shape will be required to insure adequate trailing edge thickness when the boundary layer displacement thickness is removed from the ideal blade surface.

DESIGN POINT

INLET PRESSURE, PSIA	390
EXHAUST PRESSURE, PSIA	14. 5*
PRESSURE RATIO	26. 9
FLOW RATE, LB/SEC.	0. 24
EFFICIENCY, PERCENT	0. 572
SHAFT POWER, HP	469

*AMBIENT PRESSURE = 11 PSIA

OFF - DESIGN

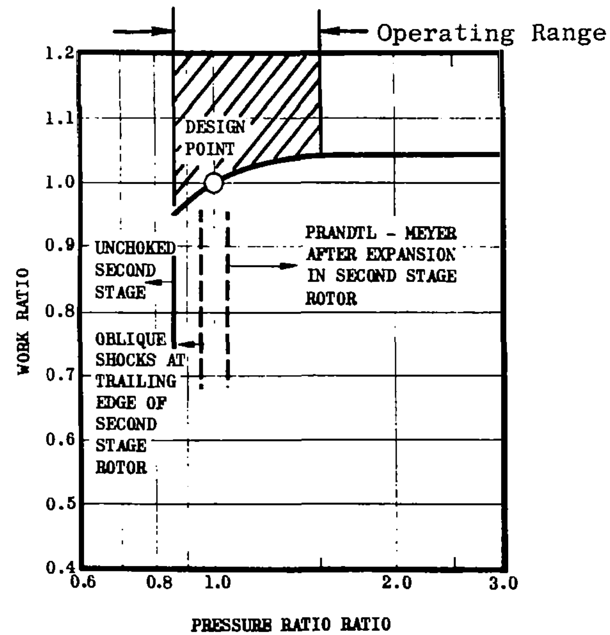


Figure 60. Reference Turbine Performance Summary

FLOW PATHS

	FIRST STAGE	SECOND STAGE	Overall
DESIGN PRESSURE RATIO	7.0	4.72	26.9
EFFICIENCY	.444	.556	.572
POWER OUTPUT, HP	254.1	214.8	468.9
NOZZLES			
NUMBER	7	11	
TYPE	Circular Cross-Section	Rectangular Cross-Section	
DEGREE OF ADMISSION	.234	.383	
THROAT AREA, IN^2	0.149	0.755	
ROTOR			
INLET RELATIVE MACH NUMBER	1.485	1.243	
BLADE NUMBER	89	89	
EXIT TIP SPEED, FT/SEC	1782.8	1891.8	
EXIT TIP DIAMETER, IN.	6.810	7.226	
EXIT BLADE HEIGHT, IN.	.31	.51	
CHORD, IN.	.421	.634	

Figure 61. Reference Turbine - Selected Details

If two dimensional cross-section nozzles had been used in the first stage, the resulting arc of admission would have been very small. In order to match the arc of admission in the second stage it would have been necessary to have extremely high blade height in the second stage to provide the full flow area. This additional blade height would cause additional stress difficulties and additional partial admission losses in the second stage. Thus the compromise of circular first stage nozzles and two- dimensional second stage nozzles was selected.

Additional details of the flow through the first and second stage nozzles are shown in Table 8 . The complete velocity diagrams may be constructed from these data.

C3. Thermal and Mechanical Limitations

Transient heat transfer analyses of the turbine blades and turbine rotor system were conducted along with, as a limit, steady state analysis of the rotor system. Some of the results of the analysis are shown in Fig. 62 . Fig. 62a shows the gas blade heat transfer coefficients in the first stage blade as a function of distance along the surface for both the suction and pressure surfaces of the blade. These instantaneous coefficients are derived from the boundary layer analysis conducted on the blade. Appropriately modified for arc of admission and fractional on-time, these coefficients are then applied to the calculation of transient temperature distributions in the blade during a start cycle from room temperature. The adiabatic wall temperature relative to the blade which causes the heat transfer was assumed to have a square wave characteristic, i.e., on instantaneous rise. The worst transient condition with respect to total strain in the blade occurred at 0.15 seconds. The isotherms for this situation are shown in Fig. 62b. The rapid blade response results from their small size and the large fraction of hydrogen in the working fluid.

Figure 62c represents the temperature as a function of radius at selected times during the transient period. The temperature distribution in the disc at 6 seconds after start is determined to be the worst condition from a thermal stress point of view. Figure 62 presents the same disc data plotted as a function of time for selected radii. It will be noted that thermal steady state is actually achieved after 5 minutes of operation.

With the temperature gradients known, the steady-state and transient blade and disc stresses can be studied. Figure 63 presents some of the pertinent results for the blades. Figure 63a shows modified Goodman diagrams for the blade root stresses for both stages. It is clear that a more than adequate safety factor exists. The cyclic stress situation due to the transient temperature distribution in the blade is shown at Fig. 63b. The total strain corresponding to the transient temperature distribution 0.15 seconds after a start is seen to correspond to 10^5 stop-start cycles. This compares with a required 1500 stop-start cycles for the TPU.

TABLE 8

VELOCITY DIAGRAM DATA

ITEM	FIRST STAGE	SECOND STAGE
ISENTROPIC VELOCITY, C_0, FT/SEC	9185	7600
NOZZLE EXIT VELOCITY, C_2, FT/SEC	8775.60	7375
NOZZLE EXIT FLOW ANGLE, α_2, DEGREES	12.84	16.48
NOZZLE EXIT TANGENTIAL VELOCITY, C_{U_2}, FT/SEC	8556.31	7070
NOZZLE EXIT AXIAL VELOCITY, C_{M_2}, FT/SEC	1949.54	2090
ROTOR INLET PITCH LINE VELOCITY, U_{P_2}, FT/SEC	1680.75	1748.29
ROTOR INLET RELATIVE TANGENTIAL VELOCITY, W_{U_2}, FT/SEC	6875.56	5321.71
ROTOR INLET RELATIVE FLOW ANGLE, β_2, DEGREES	15.83	21.44
ROTOR INLET RELATIVE VELOCITY, W_2, FT/SEC	7146.62	5720
ROTOR EXIT RELATIVE VELOCITY, W_3, FT/SEC	4963.72	4414.27
ROTOR EXIT RELATIVE FLOW ANGLE, β_3, DEGREES	18.89	24.50
ROTOR EXIT RELATIVE TANGENTIAL VELOCITY, W_{U_3}, FT/SEC	4696.38	4016.81
ROTOR EXIT AXIAL VELOCITY, W_{M_3}, FT/SEC	1607.02	1830.57
ROTOR EXIT TANGENTIAL VELOCITY, C_{U_3}, FT/SEC	2994.69	2258.57
ROTOR EXIT PITCH LINE VELOCITY, U_{P_3}, FT/SEC	1701.69	1758.24
ROTOR EXIT ABSOLUTE VELOCITY, C_3, FT/SEC	3398.52	2907.25
ROTOR EXIT ABSOLUTE FLOW ANGLE, α_3, DEGREES	28.22	39.02
NOZZLE EXIT MACH NUMBER, M_2	1.823	1.605
ROTOR INLET RELATIVE MACH NUMBER, M_{W_2}	1.4848	1.243
EXIT MACH NUMBER, M_3	0.660	0.599

(ALL ANGLES MEASURED FROM PLANE OF WHEEL)

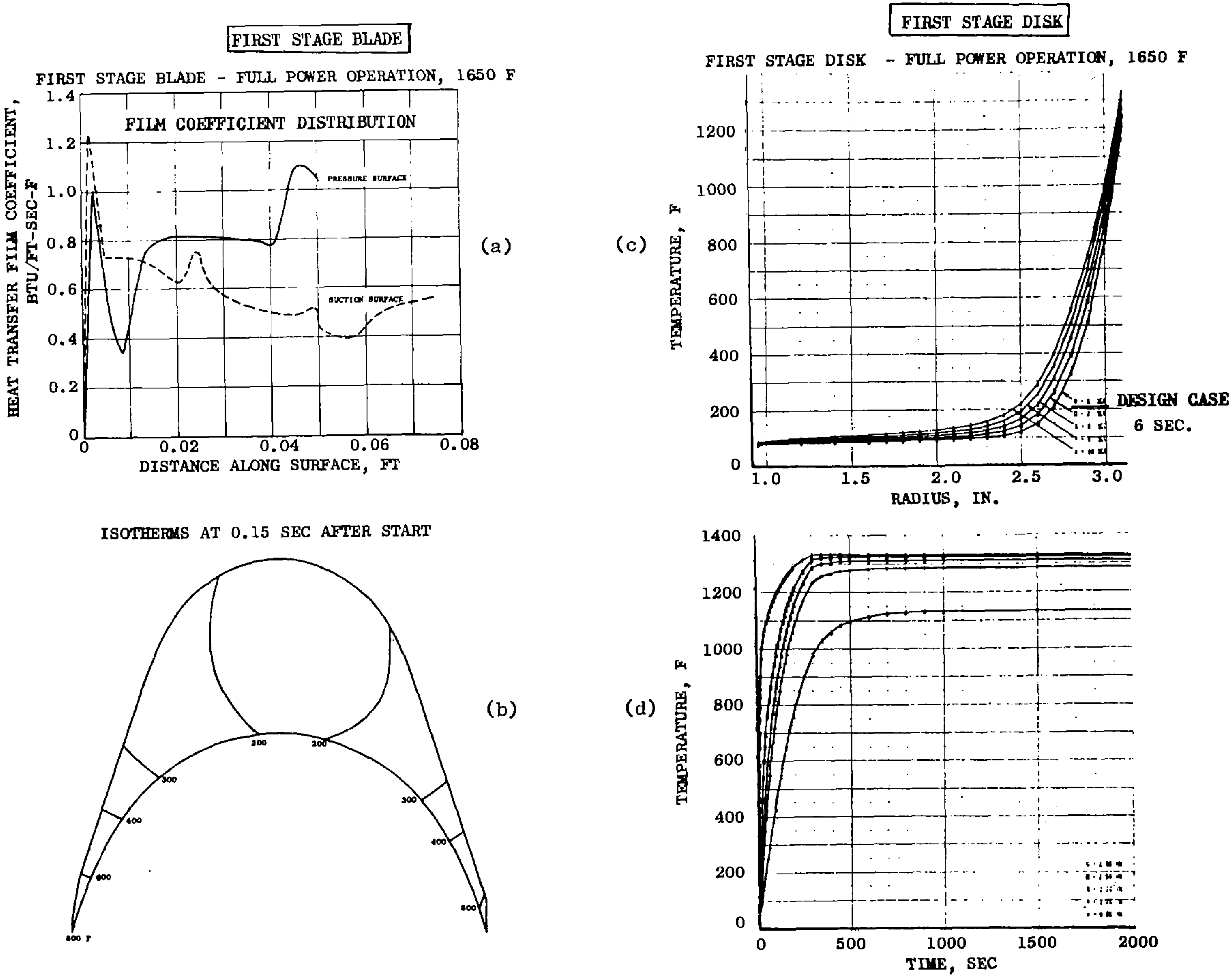
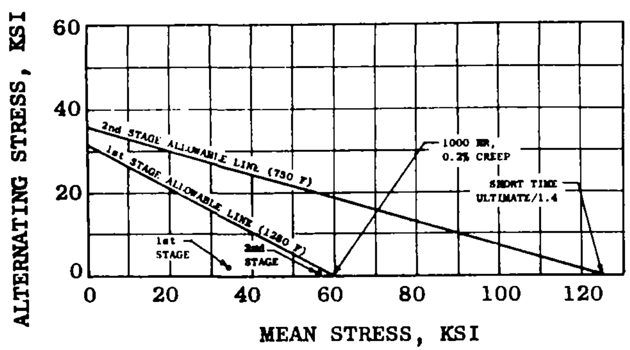


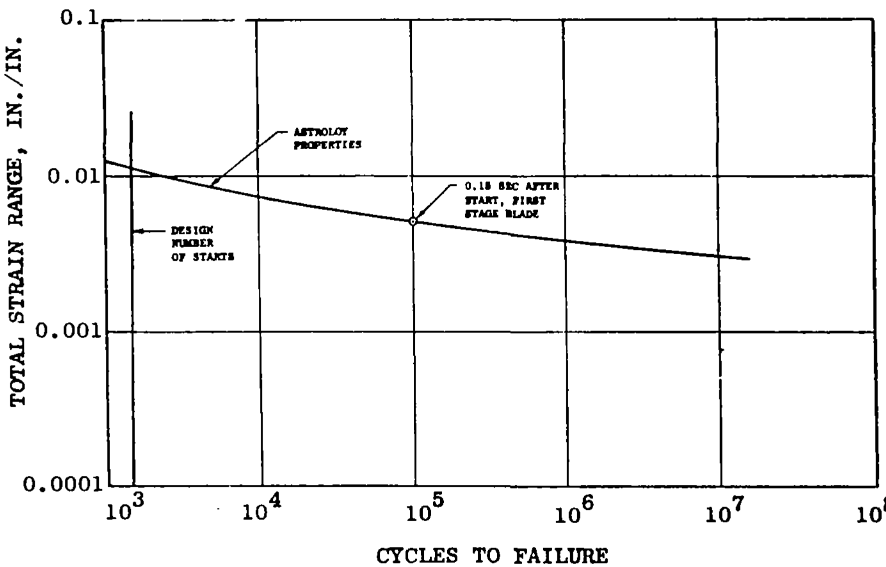
Figure 62. Transient Heat Transfer Analysis

THERMAL STEADY STATE



(a)

THERMAL TRANSIENTS



(b)

Figure 63. Turbine Blade Stresses

Following the start transient, the rotor blades are still subjected to cyclic strain due to the pulse mode of combustor operation. The strain range associated with these pulses is very small. Heat flows into the blades by conduction when the combustor is on, and flows out of the blades by conduction all of the time. The conduction is the limiting process. Thus, the blade temperature is very close to the gas temperature while the combustor is off as well as while it is on. The temperature gradient in the blades under both conditions is very much smaller than during the start cycle. Thus, the cycle life for normal pulse operation is essentially unlimited. While the margin of safety on cycle life for stop-start cycles is more than adequate, another question to be considered is the cycle life when the unit is subjected to temperature transients during operation. This sort of situation can arise, when, for example, the TPU is running at full load with thermal steady state attained and the external load is suddenly dropped to the minimum value. Under these circumstances, two or three pulses of operation with the reference propellant conditioning system, can produce temperatures in excess of the nominal temperature by say, 100°F. The effect of temperature spikes, 500 and 1000° F above the nominal values were calculated in order to get a measure of the severity of the problem. As will be seen in Fig. 64 expected cycle life under these conditions is approximately 10^6 cycles. This means that almost every pulse during the 100 mission life of the unit could be an over-temperature pulse from the point of view of transient stresses without causing difficulty. Of course, if every pulse were in fact an over-temperature pulse the average temperature of the blade would creep up to an unacceptable level. Thus, while the transient temperature response is quite adequate from the point of view of cycle life, extended exposure to over-temperature conditions would cause blade failures. For this reason an over-temperature control has been included in the system as described elseqhere in this report.

The design of both turbine discs was based on steady-state temperature distributions. These designs are summarized in Fig. 65. The first stage disc being hotter, was designed to have an effective stress well below the 0.2 percent 1,000 hour creep strength, as shown in Fig. 65. The second stage disc being much cooler was limited not by a long time creep but by short time ultimate properties. This disc was therefore designed to have an adequate strength compared to the short time ultimate strength with a factor of safety of 1.4. Both disc designs are conservatively based on continuous operation at maximum rated speed of 63,000 rpm. The temperature distributions used are based on maximum power operation. This is also conservative since most operation in the typical power profile is at reduced power levels which cause the average disc temperature to be reduced a few degrees. Use of the full power levels gives a capability to adapt to changed mission requirements.

The cyclic life of the turbine discs based on the transient temperature distribution at 6 seconds after start is shown in Fig. 65. The first stage disc is seen to be capable of 10^4 start-stop cycles compared to the required 1500. The second stage has a much higher cycle lite. Both discs are more than adequate with respect to cyclic life.

The various rotor dynamics calculations which have been made on the referenced TPU are summarized in Fig. 66. Figure 66a shows the synchronous critical speeds for the turbine rotor as a function of bearing spring rate. At the selected bearing spring rate of $1x10^5$ lb/in the rotor operates between the first and second critical speeds. In this region room exists for some variation in the bearing spring rates without causing difficulty with respect to critical speeds. Figure 66b shows the rotor mode shape for the second critical speed. This is seen to correspond to rotation about the center of gravity of the turbine rotors.

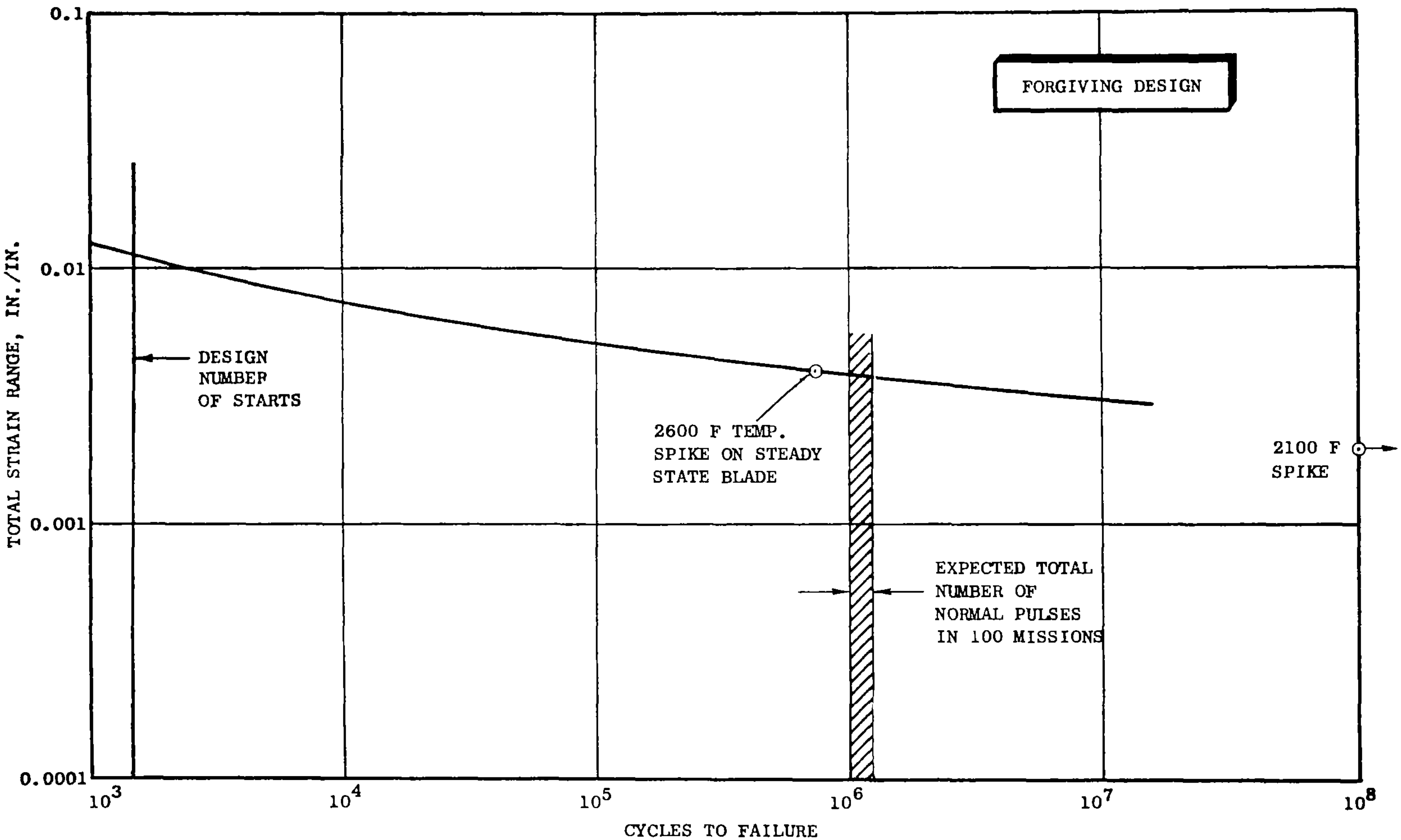
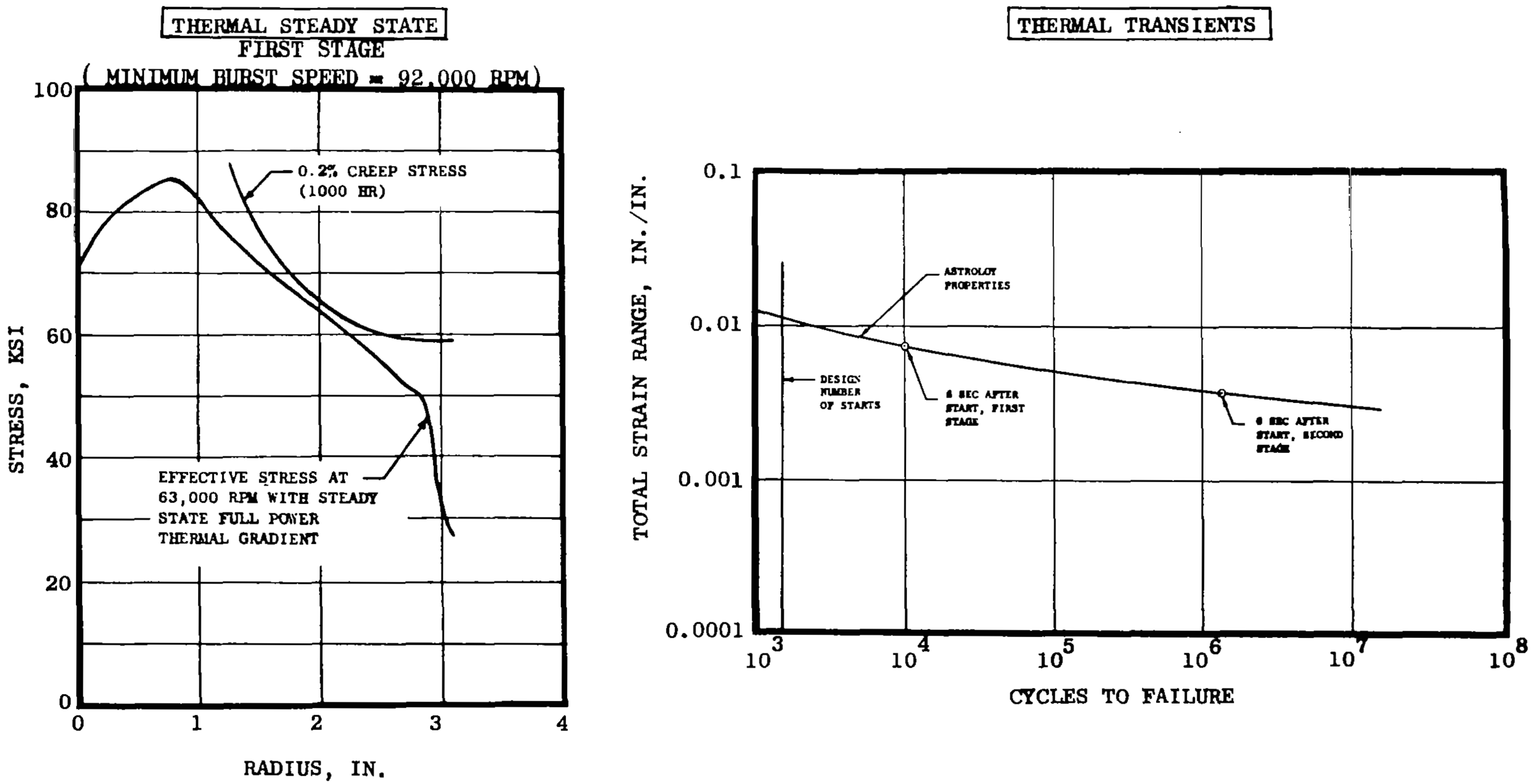


Figure 64. Temperature Spike Insensitivity

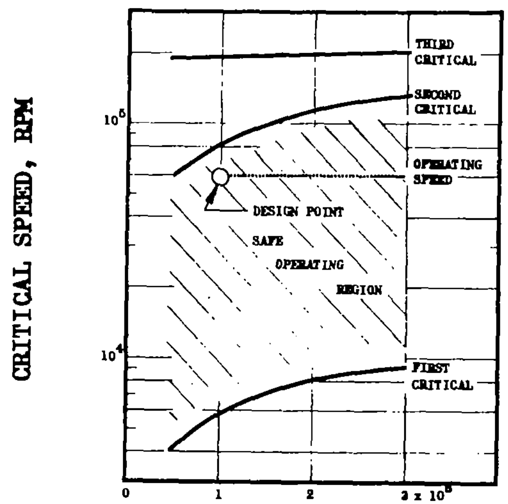


SECOND STAGE

- SIZED FOR SHORT TIME ULTIMATE/1.4
- MINIMUM BURST SPEED = 74,800 RPM

Figure 65. Turbine Disk Stresses

SYNCHRONOUS CRITICAL SPEEDS

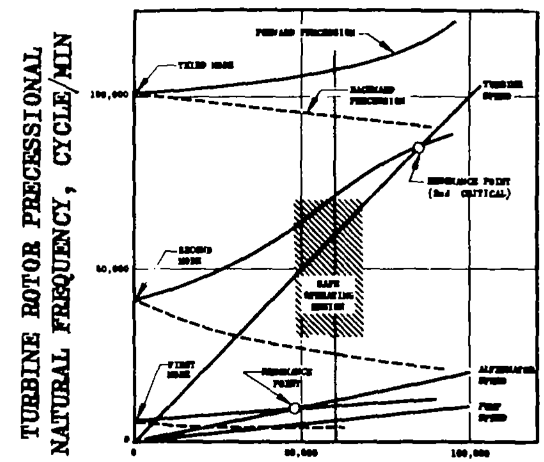


(a)

BEARING SPRING RATE, K, LB/IN

FORGIVING DESIGN

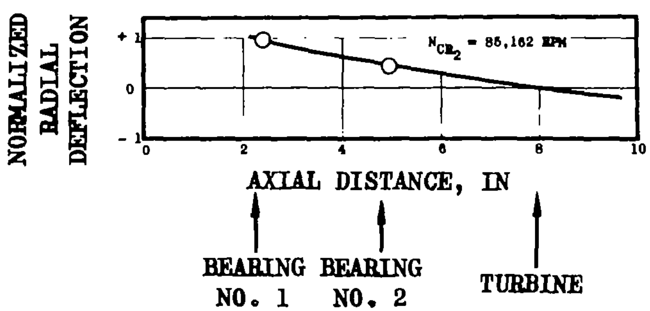
PRECESSIONAL NATURAL FREQUENCIES



TURBINE SPEED, RPM

(c)

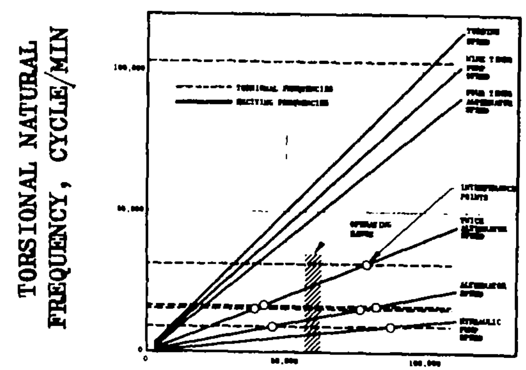
ROTOR MODE SHAPE



(b)

TORSIONAL NATURAL FREQUENCIES

With Recommended Stiffnesses



TURBINE SPEED, RPM

(d)

DISK DIAMETRAL CRITICAL SPEEDS

BOTH > 100,000 RPM

Figure 66. Rotor Dynamics Summary

Figure 66c presents the precessional natural frequencies of the turbine rotor as a function of turbine speed. The natural frequencies for forward (solid lines) and backward (dotted lines) precession are shown for the first, second and third modes. Since all the rotating shafts in theis TPU rotate in the same direction only forward precession need by condidered. The straight lines radiating from the origin represent the possible exciting frequencies due to unbalance of the labeled rotating shafts. For example, the intersection of the turbine speed line with the forward precession natural frequencies correspond to the synchronous critical speeds as shown in Fig. 66c. Intersection of the alternator speed excitation curve with the first mode forward precessional natural frequency curve represents a possible excitation of that first mode by alternator unbalance. Such excitation or resonance would occur at a turbine speed of 48,000 rpm. A resonance point at least 20% away from the normal operating speed is generally considered safe even with little or no damping in the system. It can be seen therefore that precessional natural frequencies are not a problem in this TPU.

Figure 66d presents the torsional natural frequencies as a function of turbine speed. The horizontal dotted lines represent the various torsional natural frequencies while the solid lines radiating from the origin represent the possible exciting frequencies. Intersections of the exciting frequencies curves with the natural frequency curves represent interference points or potential resonances. Again it will be seen that the various intersections lie at least 20% away from the normal operating speed. It should be noted that in order to attain the margin shown in Fig. 66d, it is necessary to make some slight modifications to the TPU. Stiffnesses of the alternator shaft and hydraulic pump shafts will have to be increased slightly while the stiffness of the quill shaft will have to be reduced somewhat from that shown in the drawings of Fig. 58 . This latter can be accomplished by increasing the length of the quill shaft.

The final vibration charcteristic studies on the referenced TPU was the disc diametral vibration characteristics. All diametral critical speeds for all modes for both discs were above 100,000 rpm and thus not a problem.

D. TPU SEALS

Several seals are used in the TPU, both on the high speed rotor and on the hydraulic pump pads of the gear box. Some of the major charcteristics of these seals are described in Fig. 67. Fig. 67a shows a sketch of the turbine hot gas seal and the turbine and oil seal. The turbine hot gas seal is a floating ring type seal with .002 inch diametral clearance. The turbine oil seal is a hydrodynamic face-seal selected to provide negligible leakage during operation combined with long life. As seen in Fig. 67c, this hydrodynamic seal requires slightly more power than would be required of a more common rubbing contact seal. The hydrodynamic oil film which prevents rubbing contact greatly extends the life of the seal. Fig. 67b shows a sketch of the hydraulic pump pad seals. These are conventional rubbing contact welded bellows type seals. Power loss for these rubbing contact seals is also shown in Fig. 67c.

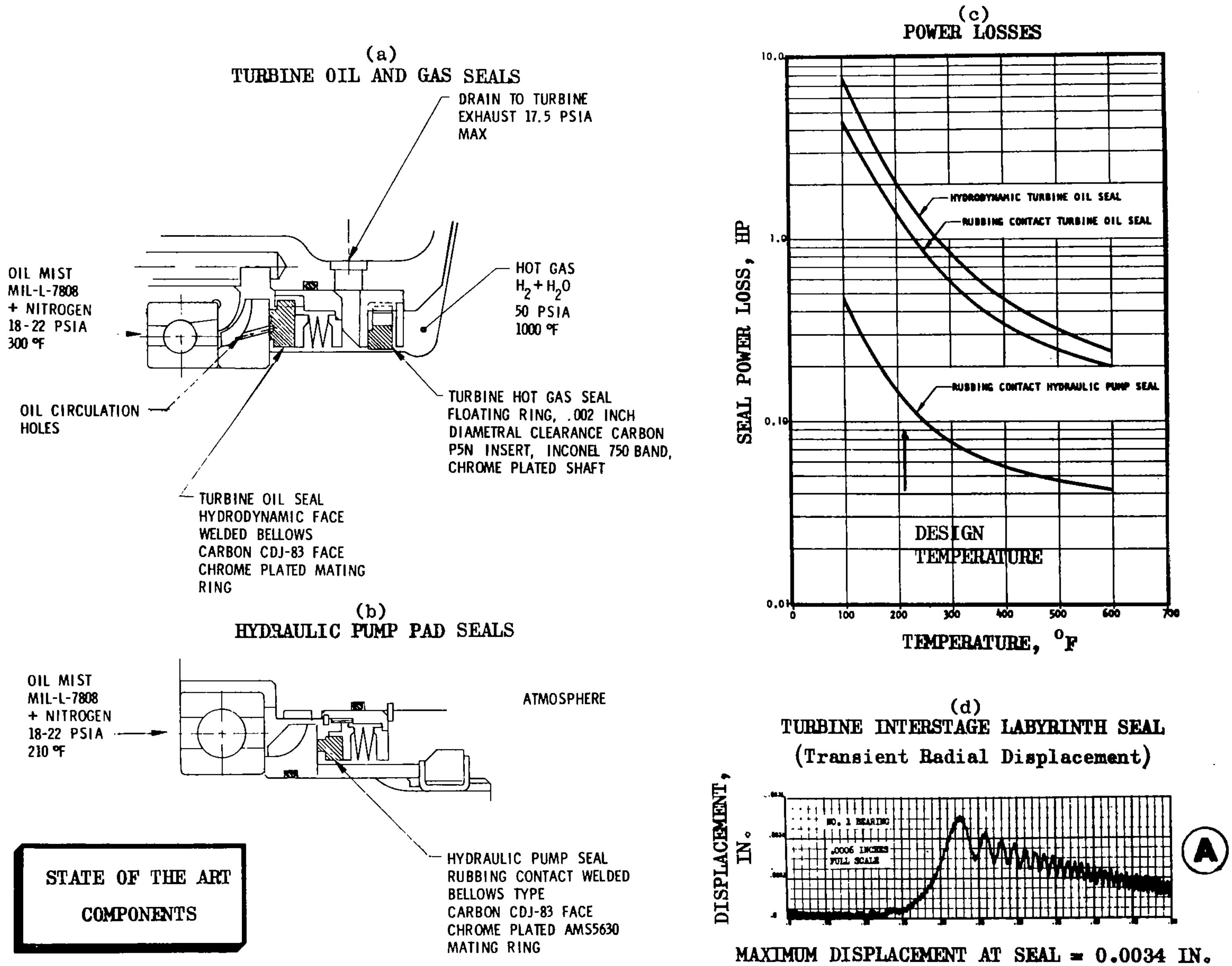


Figure 67. Seals

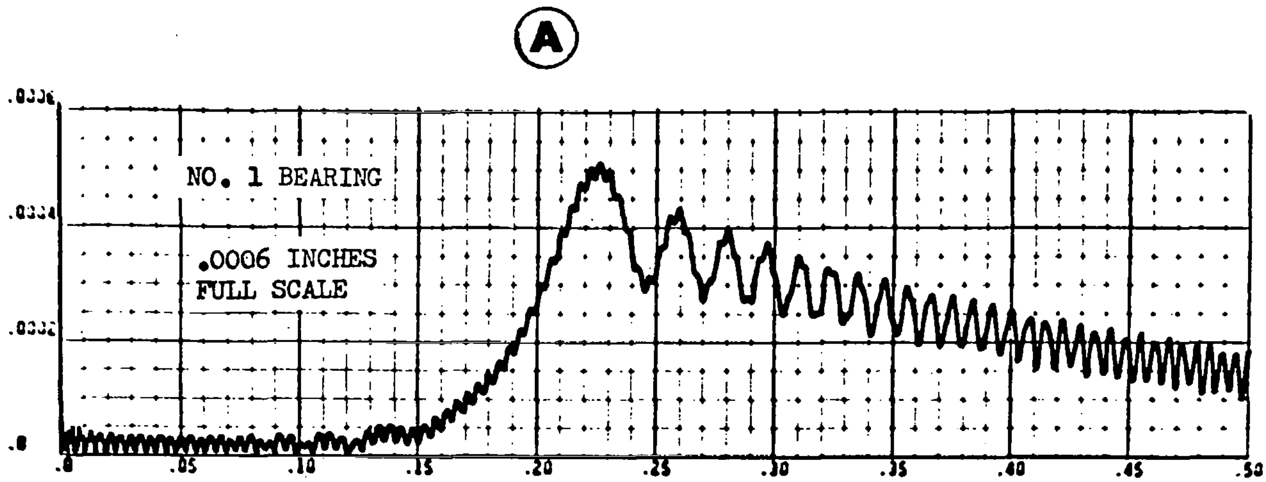


Figure 67. (Concluded)

The other seal on the high speed shaft is the inter-stage labyrinth seal between the two turbine stages. It is required that this seal be operated with absolute minimum clearance. This clearance is determined essentially by the maximum transient displacement of the rotor during passage through the first critical speed. Fig.67d, shows the typical transient displacment of the shaft at the number 1 (a drive end) bearing. Similar data for the number 2 (turbine end) bearing allowed the determination of the maximum transient displacement at any location along the shaft. Maximum displacement at the interstage labyrinth is thereby determined to be .0034 inches. Interstage leakage for the performance calculations was based on this clearance.

E. TPU BEARINGS

All the bearings in the TPU are expected to have a B-1 life in excess of 3,000 hours. Some of the characteristics associated with the high speed angular contact bearings are shown in Fig. 68 . The effect of axial preload on radial stiffness is shown in Fig. 68a . An axial preload of 25 pounds on the high speed turbine rotor will provide the desired radial stiffness of 10^5 lbs/inch. The high speed pinion rotor required a radial stiffness of $2x10^5$ lbs/in to avoid critical speed problems thus requiring an axial preload on the pinion bearings of 55 lbs.

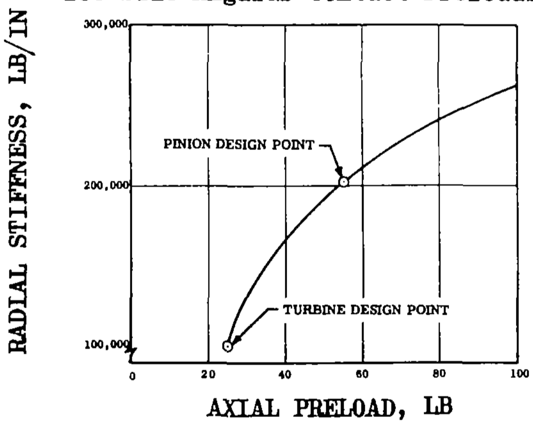
Figure 68b shows a typical power loss characteristic for pinion bearings as a function of both load and oil temperature. The lower curve represents the power loss due to the applied load while the upper curves represents the viscous power loss due to the presence of oil in the bearings. The curves represent three different forms of lubricating varying from flooded operation to the selected oil mist operation. Total power loss is the sum of the viscous and load losses. Similar calculations were made for the other bearing sizes used in the gear box. The total bearing power loss then is shown in Fig.68c for both the maximum power and minimum power conditions.

F. TPU GEARBOX

F1. Gear Design

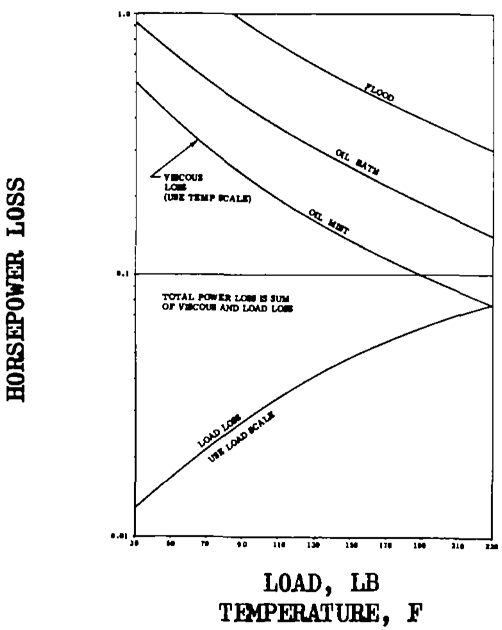
The gear box itself was shown in some detail in Fig. 58 . Some of the features of this gearbox are pointed out in Fig. 69 . The general gear arrangement was selected based on the speeds, which required a double reduction from the turbine shaft to the pump shaft, and a desire to obtain balanced loads on the pinion. Idler gears were used between the pump and pinion shafts to provide center line spacing and to allow location of the idler gears on opposite sides of the pinion thus balancing the torque loads and minimizing the pinion bearing load. The main bearing load applied to the pinion is due to the alternator load which is applied through one of the idler gears. As long as the two hydraulic pumps operate together, that is, absorb the same torque, the load on the pinion is nearly balanced and the bearing life is greatly enhanced.

HIGH SPEED BEARINGS
105 Size Angular Contact Preloads



(a)

PINION BEARING POWER LOSSES



(b)

(c)

OTHER BEARINGS

106, 108, 109 SIZES

TOTAL BEARING POWER LOSSES

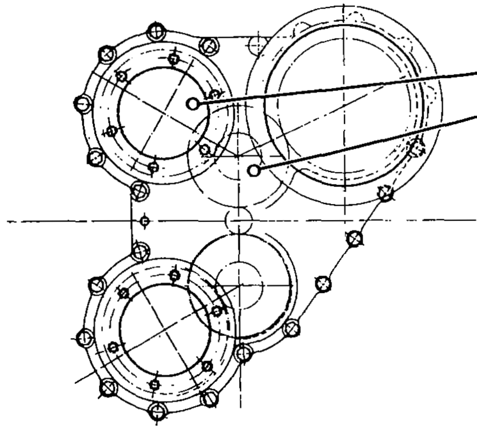
	Max Power	Min Power
DUE TO LOAD	1.32	0.13
DUE TO OIL	0.36	0.36
TOTAL	1.68 HP	0.49 HP

EXPECTED B1 LIFE 3000 HR FOR ALL BEARINGS

MATERIAL: HIGH-SPEED BEARINGS--
VACUUM MELTED M50
OTHER BEARINGS--SAE 52100

STATE OF THE ART COMPONENTS

Figure 68. Bearings



GEAR ARRANGEMENT

- PUMPS - DOUBLE REDUCTION REQUIRED
- BALANCED LOADS ON PINION
- IDLERS TO GET CENTERLINE SPACING

GEARS

- CONVENTIONAL INVOLUTE SPUR
- FULL DEPTH TOOTH
- 25° PRESSURE ANGLE
- AGMA QUALITY GRADE 2 MATERIALS
- AGMA TOLERANCE CLASS 13
- FULL HYDRODYNAMIC OIL FILMS

POWER LOSS

- INCLUDING LUBRICATION SYSTEM, BEARINGS, GEARS, AND SEALS
- FULL LOAD - 10.6 HP
- ZERO LOAD - 5.9 HP

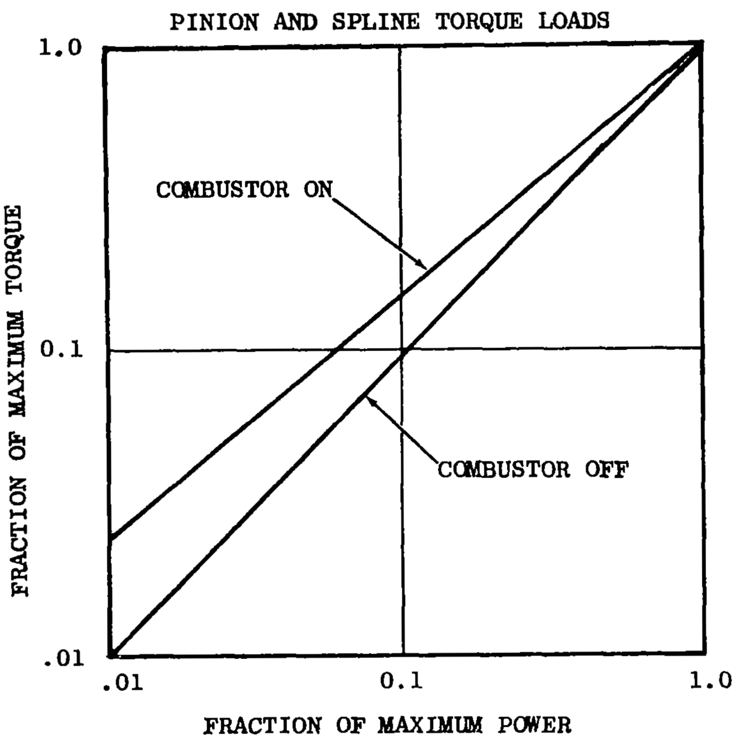


Figure 69. Gearbox Features

It might be thought that the torque loads on the pinion and spline would be in the nature of a completely released torque load with a pulse modulated control system. The curve in Fig. 69 shows that the actual transmitted torque has a much greater variation as a function of the steady power output of the gear box. The small variation between the combustor on-off situations results from the fact that most of the inertia of the rotating assembly is in the turbine discs. Thus, when the combustor is on a significant portion of the energy generated goes into accelerating the turbine discs. The remainder is transmitted and provides load torque and acceleration torque for the load inertias. Then the combustor is off the same amount of load torque is extracted from the total system inertia in proportion to the inertia of each element. Thus most of the torque comes from turbine discs which store most of the energy during acceleration.

With this type of loading the gears can be conventional involute spur gears with full depth teeth and 25 degree pressure angles. The materials are AGMA quality grade 2, while the dimensions will be AGMA tolerance class 13. This combination is well within the state of the art for aircraft quality gears. Further studies have shown that, with the tooth loads and speeds used in this gear box, full hydrodynamic oil film will be maintained continuously on all gears.

The total power lost in the gear box is 10.6 horsepower at full load and 5.9 horsepower at zero load. These powers include the power absorbed by the gears, the power absorbed by the bearings and seals as previously discussed, and the power absorbed by the separator/pump and circulator of the lubrication system. The total power loss at full load for this gear box thus is less than 2.7 percent of the transmitted power.

<u>F2 Zero Gravity Lubrication System</u>

The zero gravity lubrication system designed for the reference TPU is shown schematically in Fig. 70 . This system is based on the use of an oil mist lubrication with nitrogen as a carrier providing positive circulation of the oil mist. The design is based on NASA sponsored technology (developed in the Brayton cycle program). The result is a low-loss, attitude insensitive lubrication system.

Operation of the system will be described by reference to Fig. 70 . Oil and nitrogen gas from the gear box, turbine housing, and alternator are manifolded together into a lube scavenge manifold. Two phase flow from the manifold passes into the centrifugal separator which is rotating on the 12,000 rpm shaft of the gear box. The liquid is thrown to the outside of the separator and rotates at separator tip speeds. The gas passes out of the separator along the central portion of the separator and is carried to the circulator inlet where the pressure is raised. It is then fed back into the gear box at the appropriate locations to pick up oil mist and return it to the lube scavenge manifold.

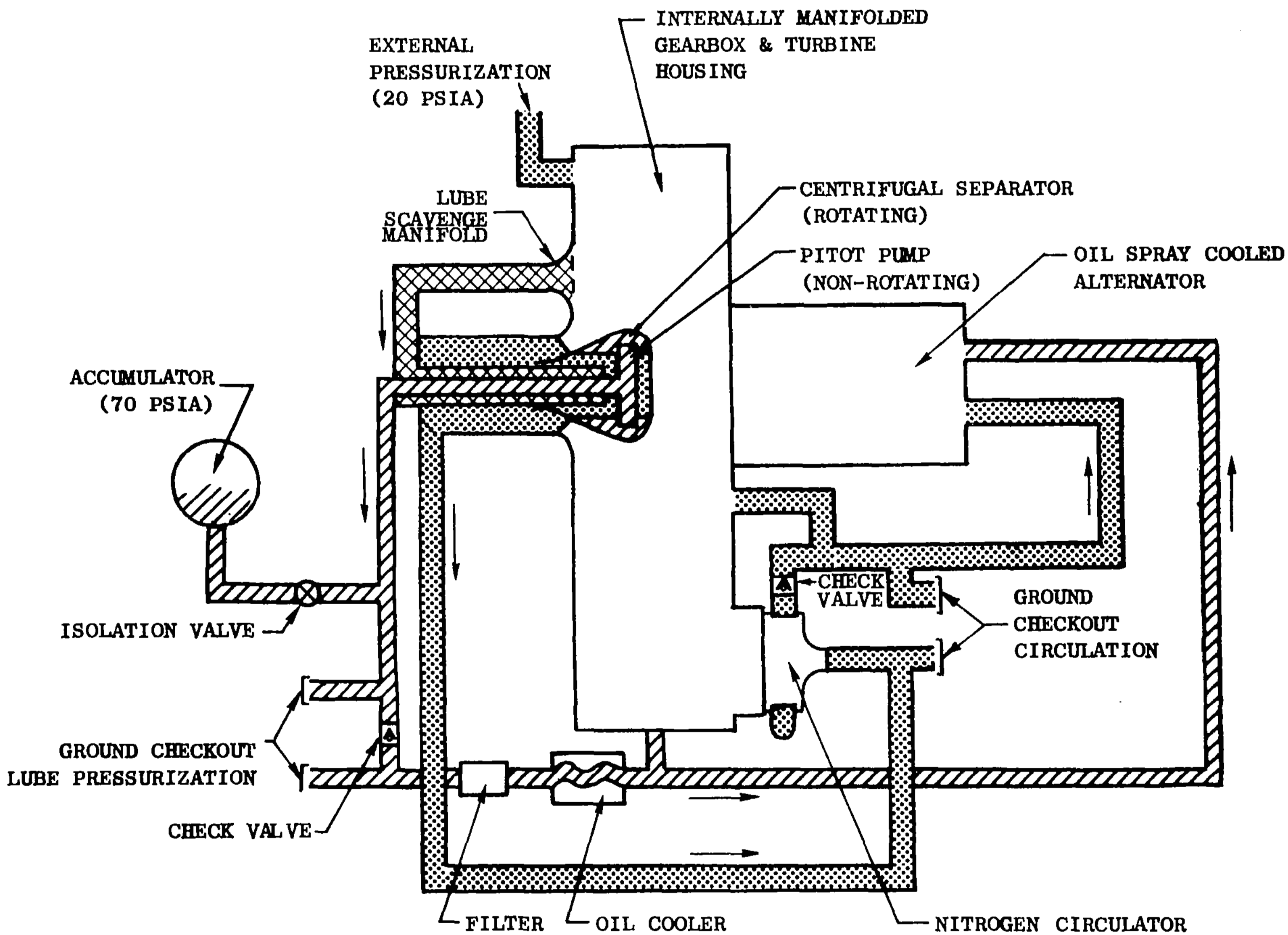


Figure 70. Zero Gravity Lubrication System Schematic

The oil in the separator, rotating at separator velocities, generates a head. This oil is picked up and removed from the separator on a continuous basis by the pitot pump which is a non-rotating element submerged in the rotating pool of oil. The pressure level of the oil coming out of the pitot pump is a function of the rotative speed and the differential radial distance between the pitot pump pickups and the inner surface of the rotating pool of oil.

The level of this rotating pool is controlled by the accumulator which serves as a high pressure storage device for oil. If the pressure leaving the pitot pump is low, oil flows out of the accumulator into the system, thereby raising the oil level in the separator. If excess oil gets into the separator and the pitot pump pressure rises above 70 psi, oil flows back into the accumulator. The main flow from the pitot pump passes through a filter and a hydraulic oil cooler before being sent back into the gear box and alternator to perform its cooling and lubricating functions. The whole gear box and lubrication system is held at a minimum pressure of 20 psia by external pressurization with nitrogen.

This system has several features worthy of note. It is readily integrated with the existing design oil spray-cooled alternator by simply providing a port close to the oil inlet to allow carrier nitrogen to be injected simultaneously. The system can also be operated without an alternator being present. Since this system has positive circulation it is attitude insensitive. The level control in the system is passive, i.e., an accumulator. Further, in the event that the system should for some reason momentarily fail to pump, the pitot pump mechanism is self restoring.

The circulator and pitot pump have been scaled from results of the previously mentioned Brayton cycle developed technology. For example, the oil droplet drag is maintained the same as that in the tested system by maintaining a system gas pressure drop of 3.2 psi. At the same time the carrying capacity of the gas is obtained by having a volume flow of gas 125 times that of the volume flow of liquid. The geometry of the pitot pump itself is also scaled.

This lubrication system is also easily adapted to the ground checkout system. As will be described later the recommended ground checkout system operates at greatly reduced speed. The lubrication system can be made to function adequately under these circumstances by utilizing a ground source of oil pressure and a ground source of nitrogen pressure. These two devices would essentially be plugged into the circuit in parallel with the circulator and in series with the pitot pump. Such an external system also permits the use of an external oil cooler thereby allowing extended duration ground checkout runs.

G. COMBUSTOR

The combustor used with the referenced TPU is shown in section in Fig. 71 . The basic combustor uses regenerative dump cooling to maintain low wall temperatures and low injector temperatures. This cooling system also minimizes the soakback which might occur during the off-time between pulses. Further,

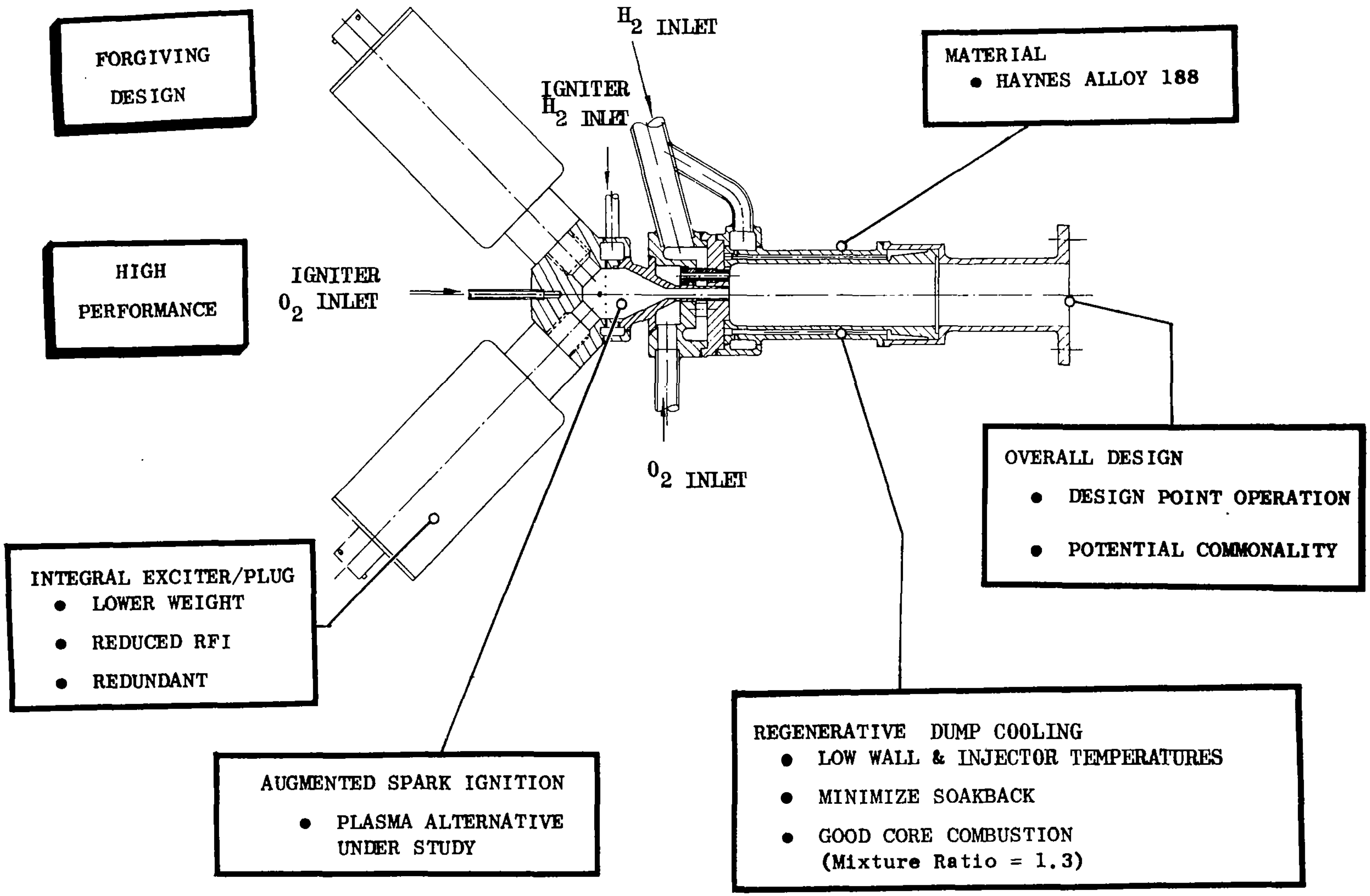


Figure 71. Combustor

since the design mixture ratio is .835, use of the dump cooling allows for combustion to take place at a mixture ratio of about 1.3 thereby providing good combustion performance. The combustor, of course, always operates at the design point in the pulse modulated system. It might be noted that this dump cooled combustion chamber, combined with coaxial injection has great potentiality for commonality with combustor designs in other elements of the space shuttle auxiliary propulsion system.

The combustor efficiency is 98 percent as demonstrated in a related program. Successful ignition has been repeatedly demonstrated in the same program.

For the pressure modulated system design of the combustor is more critical with regard to performance due to the large flow modulation requirement (approximately 10 to 1 turn down ratio). The reference pressure modulation system performance is based on attaining a combustion efficiency of 95 percent at the idle power setting and 98 percent from the 187 horsepower design point to the maximum power point. Attainment of this combustion efficiency at the idle point requires utilization of special injector and combustor design techniques developed in related programs and may require use of higher pressure drop injectors then designed for the pulse power control system.

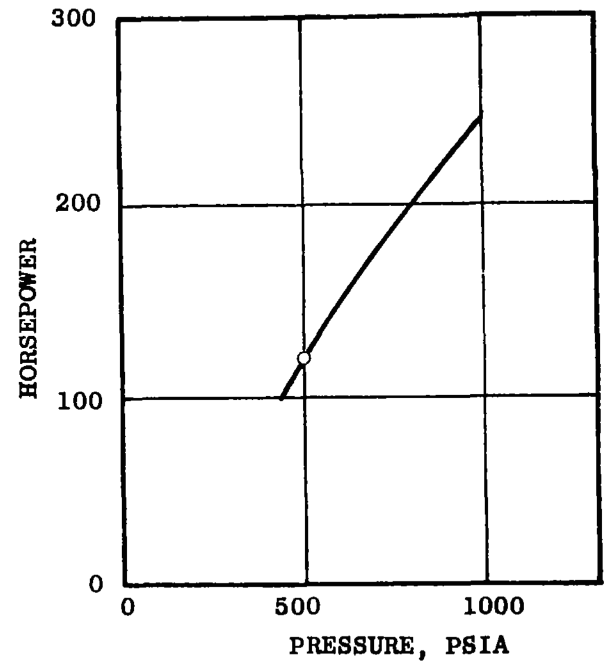
The ignition system presently planned for the combustor is the augmented spark ignition system shown in Fig. 71. This system utilizes an antechamber where combustion is initiated at relatively high mixture ratio with a small percentage of the propellants, while the major propellant flow is introduced downstream in the main chamber. While normal spark ignition in the combustor has been demonstrated to be satisfactory, the augmented spark ignition system has been utilized for a number of years under a wide variety of conditions in the J-2 rocket engine. A fairly recent development, the plasma ignition system, is also under study as an alternative. The plugs themselves are made integral with the exciter in order to provide a lower weight system and, since there is no cabling connecting the exciter and plugs, reduced radio interference. The exciter and plugs are made redundant for increased reliability.

H. GROUND CHECKOUT

Ground checkout using the TPU will be accomplished by using an external source of pressurized heated nitrogen. This nitrogen is connected directly to the turbine inlet manifold without going through any control valves. With this approach, 500 psia nitrogen heated to 300 F (to avoid icing in the turbine and exhaust), will produce an output power of 120 horsepower. Because of the lower spouting velocity of the nitrogen compared to hydrogen, which is the main constituent of the normal exhaust gases, the TPU will operate at a significantly reduced speed. Typically it will produce the 120 horsepower at 13,000 rpm. The actual rpm at which the TPU runs will be the function of the inlet conditions and the actual load applied to the TPU. Maximum power as a function of inlet pressure level is shown in Fig. 72.

A major advantage to this system as presently conceived is that no speed control is required. The necessity for speed control is eliminated because the ground test runaway speed of the turbine at no load is less than 50,000 rpm compared to the normal operating speed of 60,000 rpm. Thus with these low gas temperatures and self limiting speeds, no controls are required. A further advantage of this

- GROUND NITROGEN SOURCE
  - HEATED (300 F) TO AVOID ICING
  - PRESSURIZED (500 PSIA)
  - CONNECTED TO TURBINE INLET MANIFOLD
- PART POWER (120 HP)
- REDUCED SPEED (13,000 RPM NOMINAL)
- NO SPEED CONTROL REQUIRED (50,000 RPM MAXIMUM)
- SIMPLE GROUND LUBE CIRCULATION AND COOLING SYSTEM



SAFE OPERATION

LOW DEVELOPMENT RISK/LOW COST

VEHICLE ADAPTABILITY

Figure 72. Ground Checkout Capability

approach is that it allows lube system operation, including cooling of the lube oil, by integration with a simple ground system. There is no significant weight penalty in the flight equipment for this ground checkout capability.

I. POWER RANGE CAPABILITY

In summary, the TPU will meet all the specifications for which it was designed. It will further provide additional power range capability as shown in Fig. 73 . For example, the TPU can operate either with or without an alternator as required by the particular vehicle system. The TPU as designed, can be operated with only one hydraulic pump although the life of the gear box would be reduced because of the additional bearing load on the pinion.

The gear box itself as designed, can operate at maximum output power ratings from 200 to 400 horsepower. It can operate at ratings above 400 horsepower with reduced life (due to increased bearing and gear loads).

The turbine can produce rated powers from 200 horsepower to 800 plus horsepower. This can be accomplished by an admission arc change. Such a change in rating in general would have a negligible effect on turbine life.

The TPU using a pulse modulated power control can also operate at extreme turn-down ratios and can even operate with zero shaft power output. This feature provides a very forgiving design. It also makes it much easier to use the same TPU for both the booster and orbiter thus providing considerable savings in development and production costs.

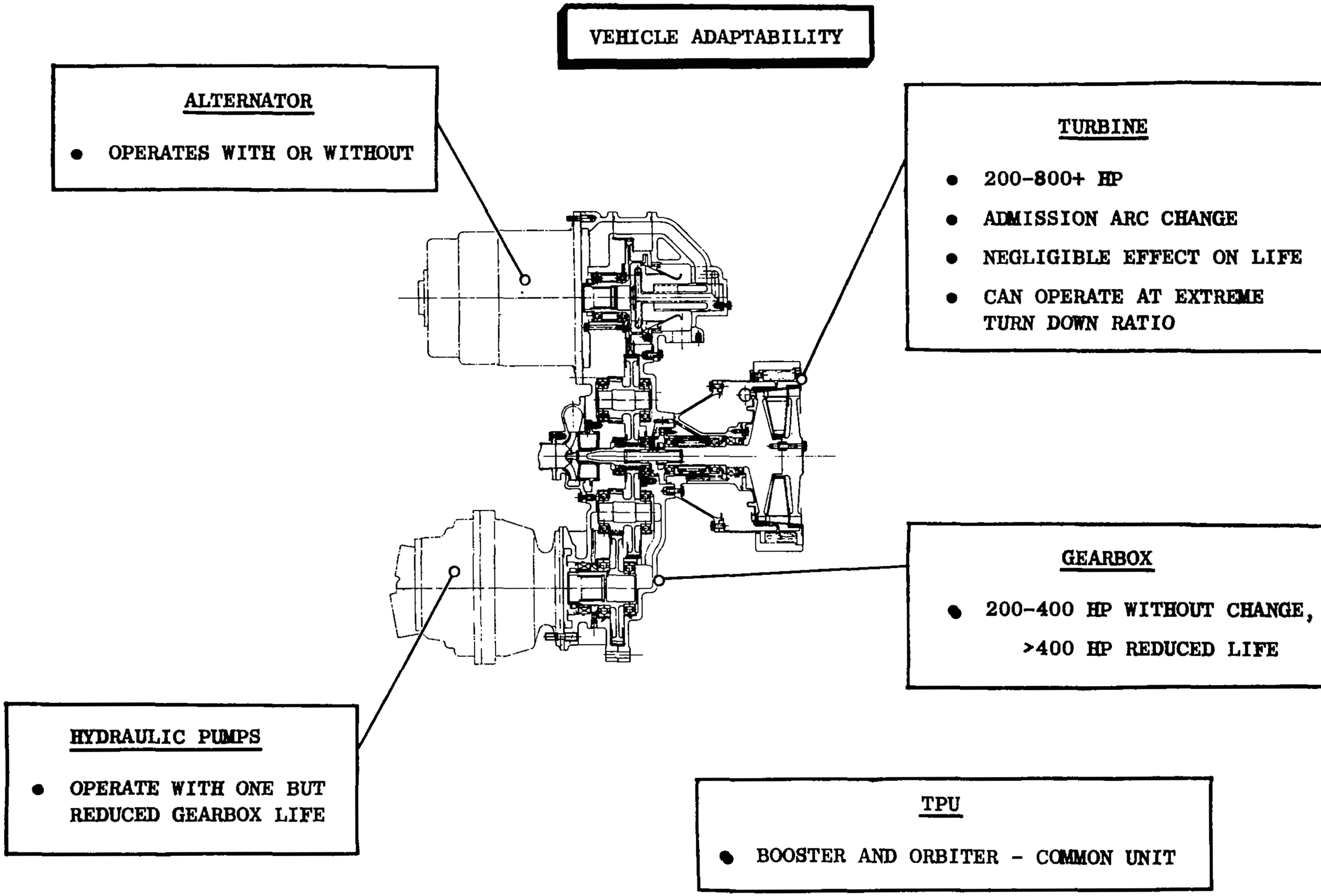


Figure 73. TPU Power Range Capacity

VIII PROPELLANT CONDITIONING SYSTEM

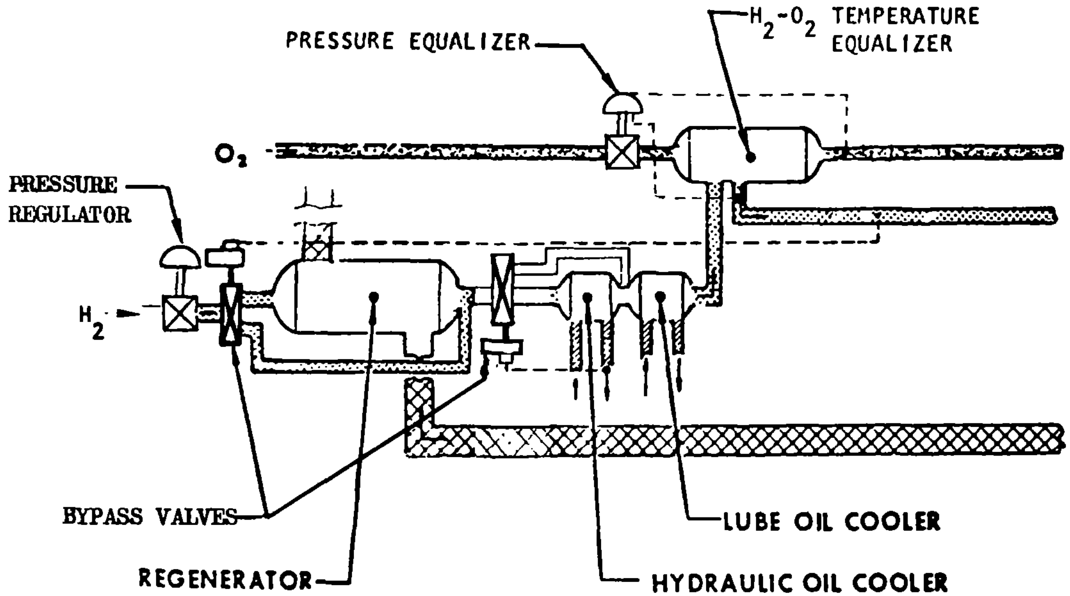
The propellant conditioning system (PCS) must provide hydrogen and oxygen to the TPU power control valves at constant and equal pressure and temperature over the entire operating envelope of the Booster and Orbiter. This is accomplished by means of a pressure equalizer (ΔP regulator) and temperature equalizer so that the proper mixture ratio is achieved in the PCS and the power control valves function only to hold TPU speed. The purpose of this section is to discuss the performance and design details of the four heat exchanger elements in the PCS that provide for thermal conditioning of the propellant. Discussion of the steady-state and dynamic characteristics of the system and its required controls is covered in the section on system operation. A schematic of the PCS is shown in Fig. 74 indicating the placement of the heat exchanger elements.

The hydrogen is conditioned to its design injector inlet temperature of 650 R by absorbing all the waste heat generated in the hydraulic and lube systems and some fraction of the waste heat available in the turbine exhaust. At low hydrogen flow the major portion of the conditioning is provided by the coolers and at high flow rates the regenerator is the major heat source. Temperature control is accomplished by the modulating three-way valve upstream of the regenerator which bypasses some fraction of the hydrogen flow to obtain the desired inlet temperature. The oxygen is heated by the warm hydrogen from its minimum temperature of 300 R to within 20 F of the hydrogen temperature (650 $\pm$20 R).

The major features of the PCS are summarized in Fig. 74 . Safe operation is achieved over the entire operating profile, which is accomplished through conservative thermal design of the heat exchangers coupled with double wall (buffered) construction to preclude fluid mixing should a leak occur. The system has operational flexibility since it can accept propellants from either a liquid or gas source and performs essentially independent of the type of TPU power control utilized -- either pulse or pressure modulated control. It is also adaptable to variations in vehicle cooling requirements since the regenerator is sized to provide 100 percent of the thermal conditioning requirement at peak hydrogen flow. Although lube cooling will always be required, there is some question as to whether the APU will have to provide cooling on the Booster vehicle. The following sections discuss the design and performance details of the heat exchangers, sequentially as they occur in the PCS, i.e., regenerator, hydraulic cooler, lube cooler, and temperature equalizer.

A REGENERATOR DESIGN

Figure 75 shows the APU regenerator in a cutaway view indicating the header and tube sheet design, hairpin tube bundle, and baffling required for proper flow distribution within the heat exchanger. The regenerator has been fabricated as part of a company-sponsored program and is undergoing performance evaluation at this time.



FEATURES

SAFE OPERATION

- BUFFERED HEAT EXCHANGER DESIGNS
- NO ICING IN REGENERATOR
- NO CONGEALING OF HYDRAULIC FLUID WITH LIQUID HYDROGEN INTO COOLER
- SAFE STARTUP/SHUTDOWN

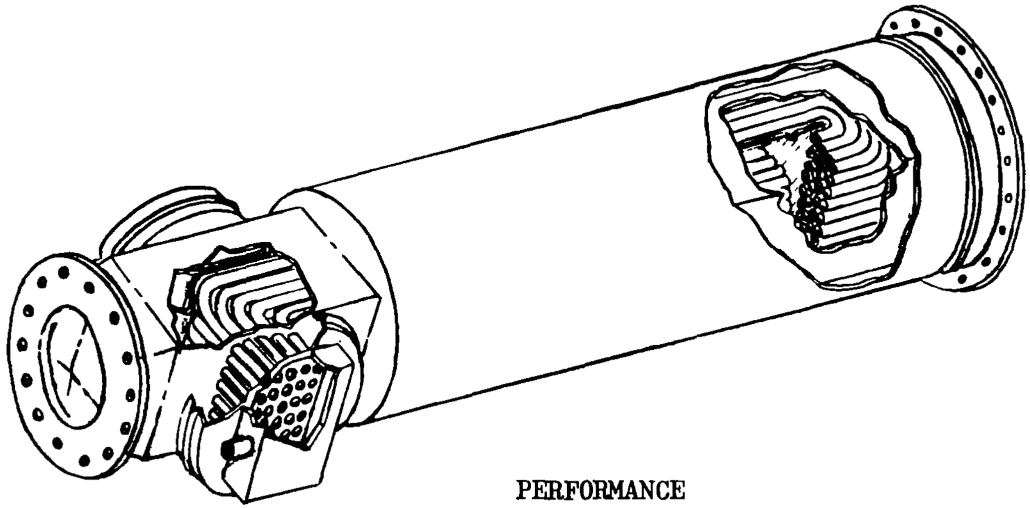
FLEXIBILITY

- PULSE OR PRESSURE MODULATION CONTROL
- LIQUID OR GASEOUS PROPELLANTS

VEHICLE ADAPTABILITY

- CONSTANT INJECTOR INLET TEMP INDEPENDENT - OF HYDRAULIC COOLING REQUIREMENT

Figure 74. Propellant Conditioning System



PERFORMANCE

- LOW EXHAUST Δ P

SAFE OPERATION

- WALL TEMPERATURES ABOVE CONDENSATION TEMPERATURES AT ALL OPERATING CONDITIONS
- TUBE SHEETS BUFFERED TO PREVENT EXTERNAL LEAKAGE
- NO THERMAL SHOCK PROBLEM

FLEXIBILITY

- PULSE OR PRESSURE MODULATED FLOW CONDITIONS ACCEPTABLE

Figure 75. SS/APU Regenerator

A shell and tube design was selected due to the high pressure requirements associated with integration of the feed system with the APS. This configuration is relatively compact and lightweight without excessive hot gas ΔP, which would penalize the turbine performance. The tube bundle is arranged in a hairpin configuration minimizing stresses due to thermal expansion commonly associated with straight-through designs.

Particular emphasis was placed on the thermal design to assure that the wall temperatures remain significantly above the condensation temperature of the water vapor in the exhaust gas. Condensation of the water could lead to freezing which would increase turbine exhaust ΔP and, consequently, incur an SPC penalty. The critical points for icing in the regenerator are the places at which the hydrogen temperature is coldest (inlet) and the hot gas temperature is the lowest (exit). The regenerator is designed such that the hot turbine exhaust gas (1400 R) flows over the hydrogen inlet tubes providing a high hot gas side heat transfer coefficient maintaining the tube walls well above condensation.

A design layout of the regenerator is shown in Fig. 76 . The hot gas flows in a single pass through the tube bundle, with 67 percent of the flow passing over the hydrogen inlet section moving parallel with the hydrogen flow to the hot gas exit. The hydrogen then makes the 180-degree turn at the exit and flows counter to the 33 percent hot gas flow. The 2:1 hot gas flow ratio is achieved by use of an orifice (1.7 in. dia.) located at the hot gas inlet on the counterflow side. A baffle is located at the center of the unit and runs the entire length to separate the two hot gas flow sections.

The hydrogen inlet and exit headers are isolated from the wall of the shell and are closed out by a compliant tube to prevent any external leakage which may occur due to failure of a brazed joint. This region is then connected with a line to a dry hydrogen vent. The reason for isolating the cold hydrogen header from the hot shell wall is to minimize the thermal stresses which would be induced in the header with hot gas flowing on one side and the cold hydrogen on the other.

The header design utilizes A286 age hardenable stainless steel which is insensitive to hydrogen embrittlement reaction. 347 stainless steel was selected for material throughout the rest of the regenerator with a working stress well below the yield strength of the material to avoid phase transformation to martensite, which would result in a severe hydrogen embrittlement reaction. The tube bundle consists of 98 1/4-inch tubes with a 0.010-inch wall thickness. The total length of the hairpin is 50 inches resulting in a heat transfer area of 24.6 square feet. The core weight is about 10 pounds and total weight of the regenerator is 26 pounds. The tubes are arranged in an aligned pattern with equal transverse and longitudinal spacing of 0.068 inch ($X_t = X_L = 1.27$). The spacing is maintained with the use of 0.062-inch rods that are staggered vertically and horizontally along the length of the bundle. The tubes are free to move axially so that no stresses are induced due to axial temperature gradients.

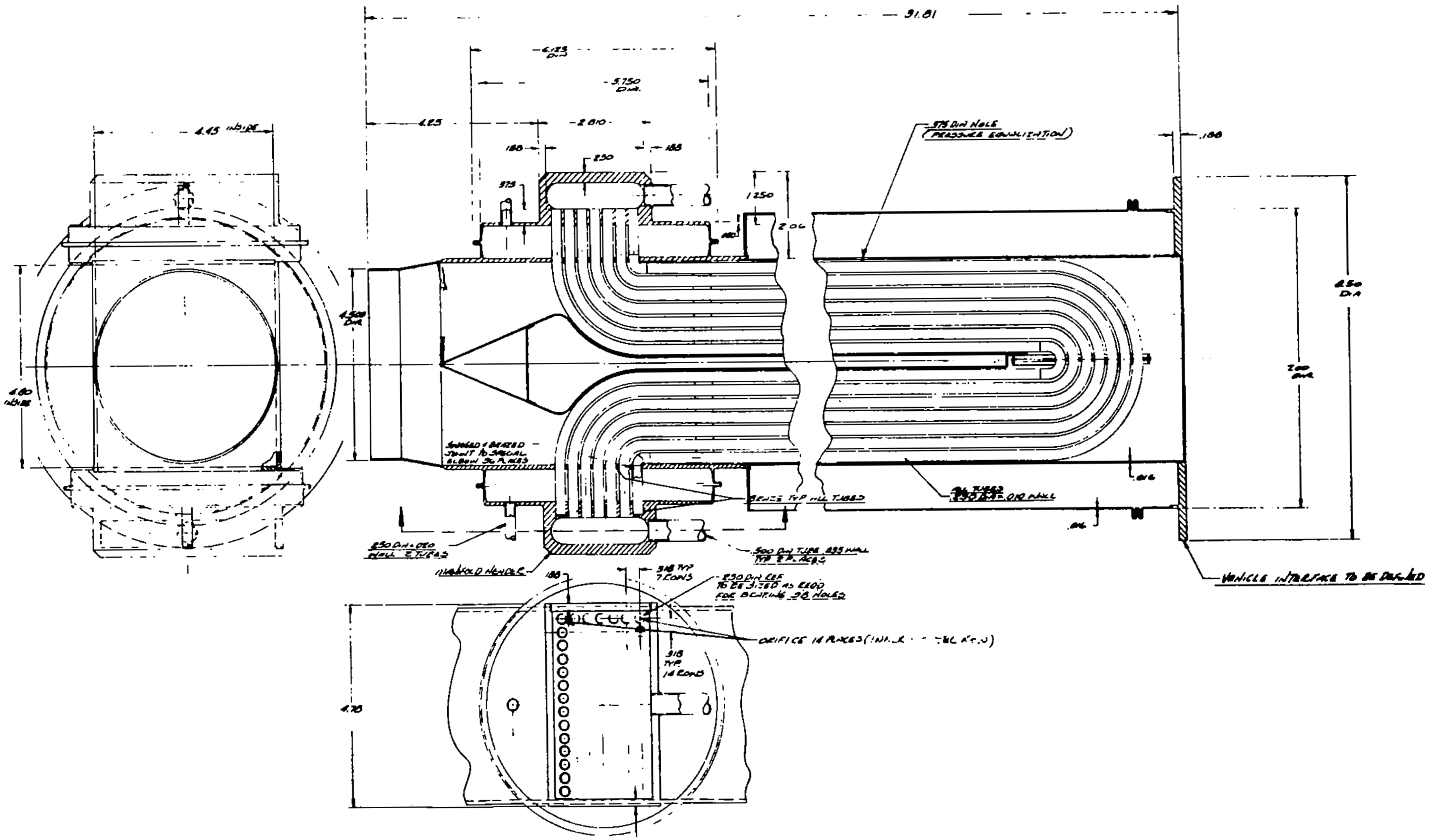


Figure 76. Regenerator Design

A tubular enclosure is used over the major length of the heat exchanger to support the maximum operating pressure differential on the shell side of 10 psi. Consequently, the high corner stresses which can result in thin wall box structures are eliminated and the "box" can be made of thin gage stainless steel serving only as a hot gas flow baffle. Results of the stress analysis are presented in Table 9. The resultant stress levels at the 460 psia working pressure in the tubes and header provide acceptable factors of safety on the yield stress.

Maximum stress in the regenerator is in the header front plate and exists as localized stress around the edges of the holes, resulting in a yield safety factor of 1.3. This is considered acceptable based on criteria established for design of the main engine components. The only life-limited structure in the regenerator is the box section at the hot gas inlet since it must accommodate the ΔP resulting during each "on" pulse if a pulse power control is utilized. No significant pressure oscillation occurs on the hydrogen side since the attenuation tanks de-couple the combustor pressure pulses from the feed system. The maximum stress of 14,200 psi results in a life of 10^7 cycles or about 1100 orbiter flights and 800 booster flights, which exceeds the 1000-life requirement by about 2:1.

The performance of the regenerator at its design point is shown in Fig. 77 . The design point was selected at the peak power sea level condition where the maximum thermal conditioning heat load is required. The regenerator was sized to do the entire conditioning job so that system design point operation is obtained independent of vehicle hydraulic cooling demand. A total of 10 percent hydrogen bypass flow is used to provide margin for control. The hydrogen inlet temperature condition is 75 R and hot gas inlet is 1400 R. The total heat transferred is 276 Btu/sec with 67 percent in the parallel flow section and 33 percent in the counterflow section. The hydrogen outlet temperature is 740 R and, when mixed with the 10-percent bypass flow, results in 671 R as the inlet temperature into the equalizer. The hot gas ΔP is 2.3 psi, resulting in a 5 percent SPC penalty at the peak power sea level condition. For pulse power control, this degradation occurs at all power levels during sea level operation since flow is constant (peak flow) with pulse duration and frequency varying to meet the load demand. The weight penalty for the orbiter due to the regenerator ΔP is 2.4 pounds of propellant with 4.8 pounds on the booster. The penalty at 10 psia back pressure (booster cruise) is 2 percent, resulting in 3.6 pounds. The penalty on the orbiter is negligible since the major portion of operation is at pressures significantly less than 10 psia. The overall effectiveness of the heat exchanger is 50 percent and the weight is 26 pounds. Other details such as area density, core density, and flows are indicated on the figure.

The results of a detailed flow analysis and thermal heat transfer analysis to determine the temperature distribution across the tubes and the effects of local variation in the gas side heat transfer coefficients are shown in Fig. 78 . These results were obtained by a computer program used extensively at Rocketdyne in the design of regeneratively cooled thrust chambers and analysis of tube and gas side heat transfer. The nominal operating condition

TABLE 9

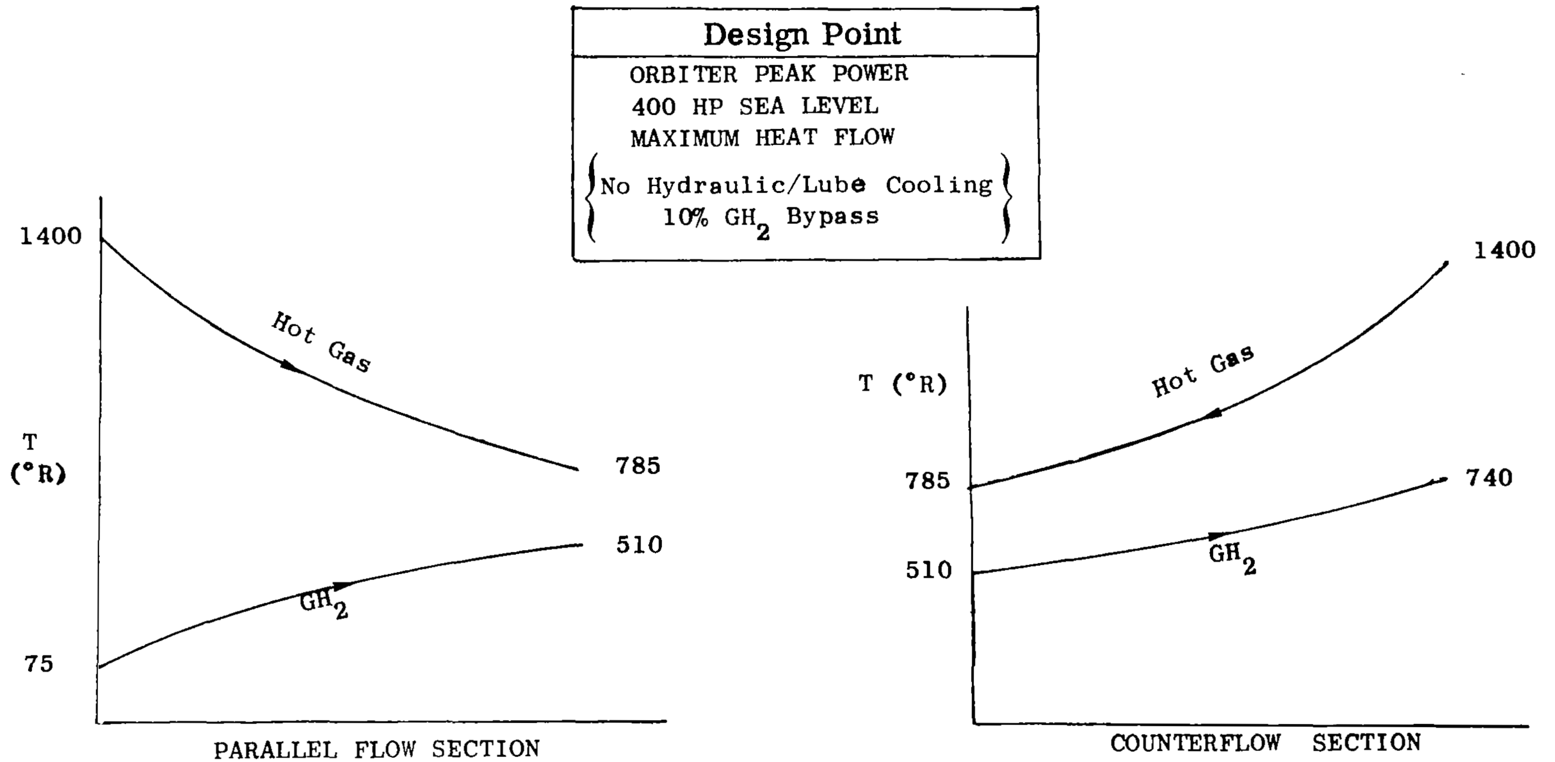
REGENERATOR STRESS ANALYSIS

	TUBES	HEADER	SHELL
MATERIAL	347 CRES	A-286	347 CRES
WORKING PRESSURE	460 PSIA*	460 PSIA*	10 PSI ΔP MAX. (500 HP AT 0 PSIA)
WORKING STRESS	5760 PSI	27,000 PSI (BACK PLATE)** 12,350 PSI (FRONT PLATE)**	14,200 PSI (BOX)*** 2190 PSI (TUBE)
YIELD STRESS	21,000 PSI (940 F)	80,000 PSI (940 F)	21,000 PSI (940 F)
ULTIMATE STRESS	57,000 PSI (940 F)	120,000 PSI (940 F)	57,000 PSI (940 F)
SAFETY FACTOR (YIELD)	3.7	3 (BACK PLATE) 1.3 (FRONT PLATE)	1.5 (BOX) 9.6 (TUBE)
SAFETY FACTOR (ULTIMATE)	10	4.4 (BACK PLATE) 2 (FRONT PLATE)	4 (BOX) 26 (TUBE)

*MAXIMUM = 1000 PSIA IF REGULATOR FAILURE; TUBE $S.F._u$ = 4.6
HEADER - LOCAL YIELDING AT HOLE EDGES

**PERFORATED FOR TUBES-NOMINAL STRESS, K_t=5 FOR HOLES SO THAT STRESS AT HOLE EDGES = 61,000 PSI

***14,500 PSI ENDURANCE AT 940 F; LIFE = 10^7 CYCLES, 800 BOOSTER FLIGHTS; 1100 ORBITER FLIGHTS FOR PULSE POWER CONTROL



Parallel Flow Section

GH_2 FLOW	0.1125	lb/sec
HOT GAS FLOW	0.154	lb/sec
HOT GAS INLET PRESSURE	17.5	psia
HOT GAS ΔP	2.3	psi
GH_2 INLET PRESSURE	460	psia
GH_2 ΔP	0.05	psi
HEAT FLOW	185	Btu/sec
AREA	12.3	ft^2

Counterflow Section

GH_2 FLOW	0.1125	lb/sec
HOT GAS FLOW	0.076	lb/sec
HOT GAS INLET PRESSURE	17.5	psia
HOT GAS ΔP	2.3	psi
GH_2 INLET PRESSURE	460	psia
GH_2 ΔP	0.09	psi
HEAT FLOW	91	Btu/sec
AREA	12.3	ft^2

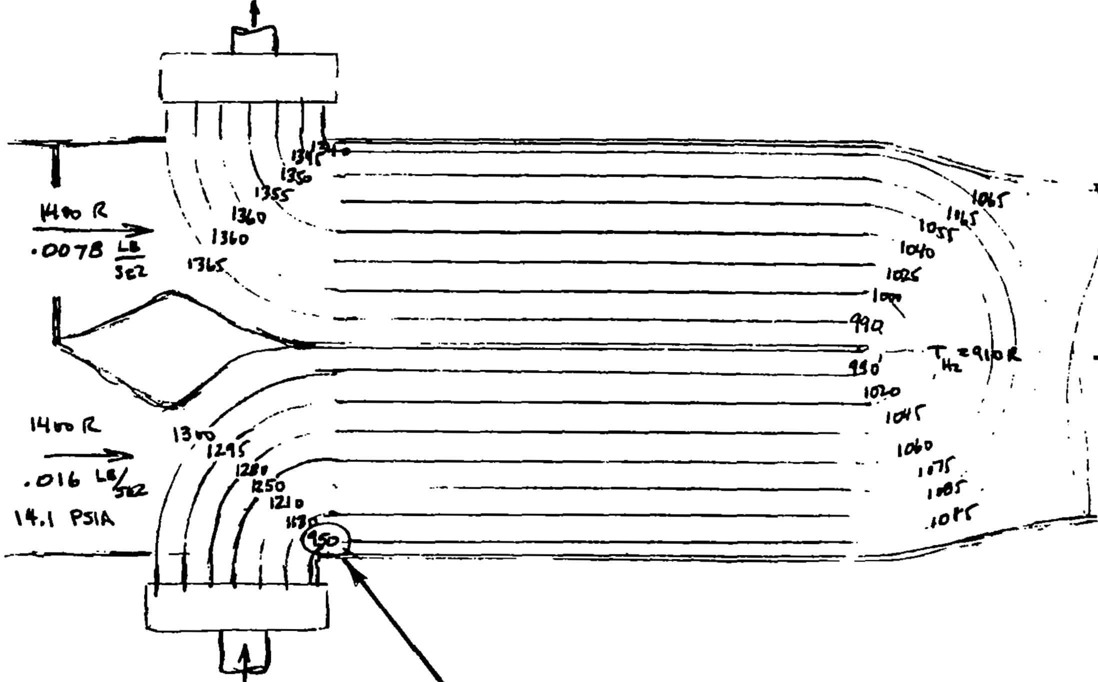
EFFECTIVENESS	50%
WEIGHT	26 lbs
AREA DENSITY	85 ft^2/ft^3
CORE DENSITY	35.2 lb/ft^3

Figure 77. Regenerator Performance

NOMINAL OPERATING CONDITION - 5700 SEC
33 HP 10 PSIA
WITH HYDRAULIC/LUBE COOLING (82% GH_2 BYPASS)

DIGITAL COMPUTER ANALYSIS
(9R-243)

1320 R



1177 R
0.0238 LB/SEC (AVG.)
10.7 PSIA
(10.1 PSIA)
AMBIENT

DUCT ΔP = 0.6 PSI
(DURING ON PULSE)

H_2
75 R
460 PSIA
0.00234 LB/SEC

SAFE OPERATION
MINIMUM WALL TEMPERATURE (R)
NO FREEZING PROBLEM

Q = 10.4 BTU/SEC

MIXED GH_2 TEMPERATURE = 288 R

Figure 78. Regenerator Temperature Distribution

shown corresponds to a gearbox output of 33 horsepower at a 10-psia back pressure, (10,000 feet altitude) for a duration of 5700 seconds. At this operating condition, due to hydrualic and lube cooling, the majority of the hydrogen (82 percent) is bypassed around the regenerator. Only 18 percent of the hydrogen flows through the regenerator tubes so that there is a high ratio of hot gas flow to the hydrogen flow. The heat addition is 10.4 Btu/sec, resulting in a mixed hydrogen temperature of 288 R. The results of the wall temperature distribution analysis are shown with the minimum of 950 R occurring at the tightest bend (outside hairpin) at the hydrogen inlet. Obviously, there is no freezing problem at this nominal operating condition. Even if the hot gas heat transfer coefficient prediction was high by 100 percent and the hydrogen side low by 100 percent for the 950 R wall temperature, the resultant wall temperature for the real coefficients is 600 R or 110 F above freezing.

The temperature distribution corresponding to the most extreme operating condition for the regenerator is presented in Fig. 79. This is the APU peak power sea level condition corresponding to the orbiter maximum power point. The duration of the operation is from 2 to 20 seconds. Assuming that 90 percent of the flow is going through the tube bundle (no hydraulic or lube cooling), a steady-state minimum temperature is indicated by this analysis of 580 R. Dynamic analysis indicates that it takes approximately 15 seconds to reach a steady-state operating condition at 580 R after changing from idle to peak power. While 580 R is close to the condensation temperature, this remains a safe operating condition even should the duration be longer than the 15- or 20-second period since there is still a 90 F margin over freezing.

Testing conducted under a Rocketdyne-funded experimental program* have confirmed the satisfactory operation of a hydrogen-hot gas (H_2-H_2O) heat exchanger under similar adverse conditions. Even with 100 percent condensation with an exhaust gas exit temperature (525 R) below the condensation limit (570 R), there was no evidence of icing (no hot gas ΔP increase).

The final complication regarding the possible formation of ice in the regenerator is that due to use of the pulse power control. In Fig. 80 the conditions resulting from a short on-time hot gas pulse and a long off time with hydrogen flowing through the regenerator tubes is shown. The specific conditions are a step-change in power from a 33 horsepower idle condition up to a 400 horsepower peak condition with the hot gas flow (as shown in the second figure from the top) off for a period of 920 milliseconds and on for about 100 milliseconds at the 33 horsepower condition. In stepping to 400 horsepower, the hot gas flow is on 95 percent of the time. Results of the transient analysis supported through the analog tests are presented in the lower figure, indicating only a 10 F drop in wall temperature within this 920-millisecond idle power off period. When the power shift occurs, 15 seconds are required to reach the minimum steady-state temperature (580 R) and a 6 F drop in the tube wall temperature occurs from this nominal value. The conclusions from this analysis are that adequate margin exists above the freezing point of the water and there should be no problem with pulse control in terms of regenerator operation.

*Space Shuttle Cryogenic Propellant Conditioning Evaluation Program.

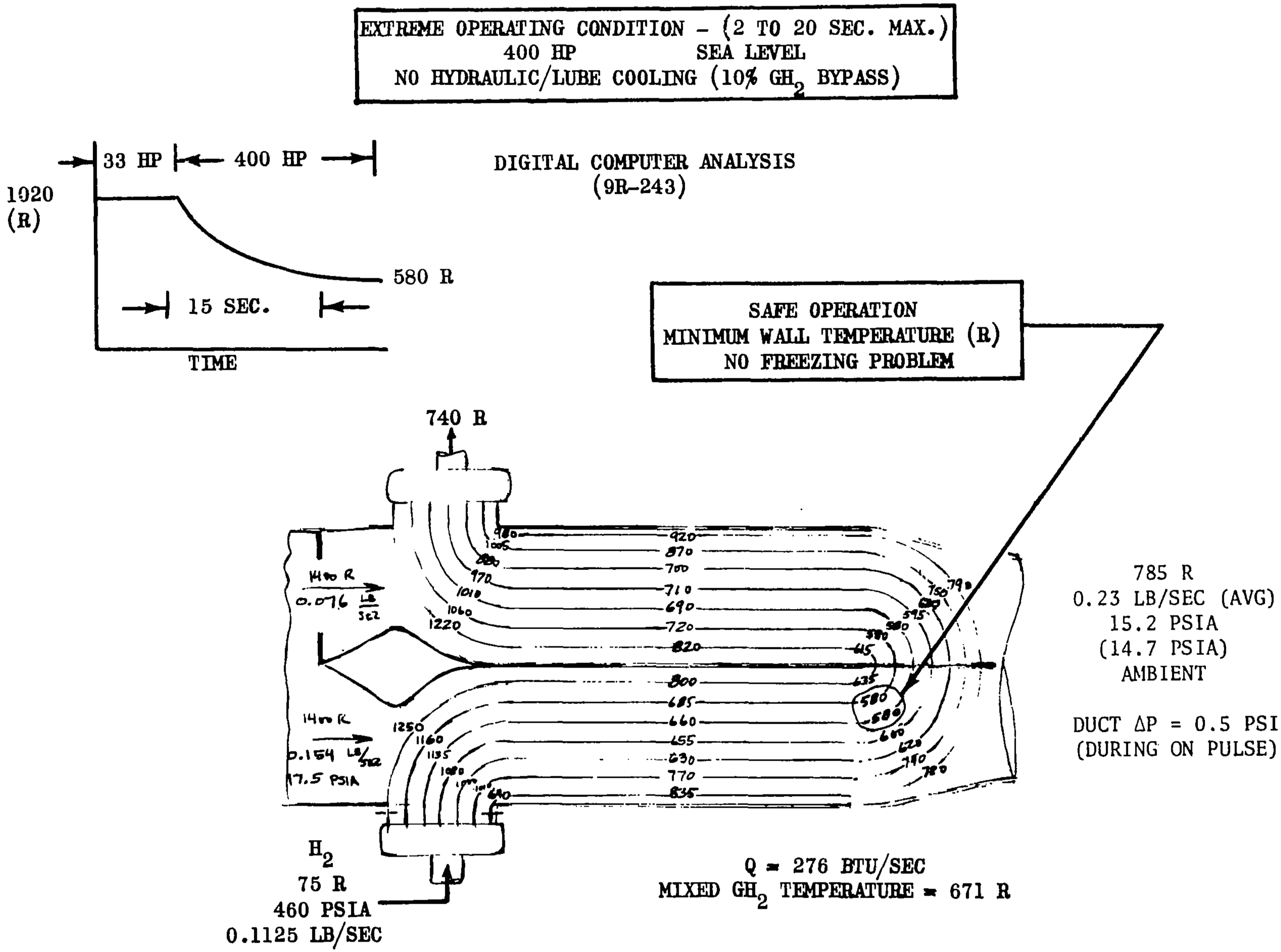


Figure 79. Regenerator Temperature Distribution

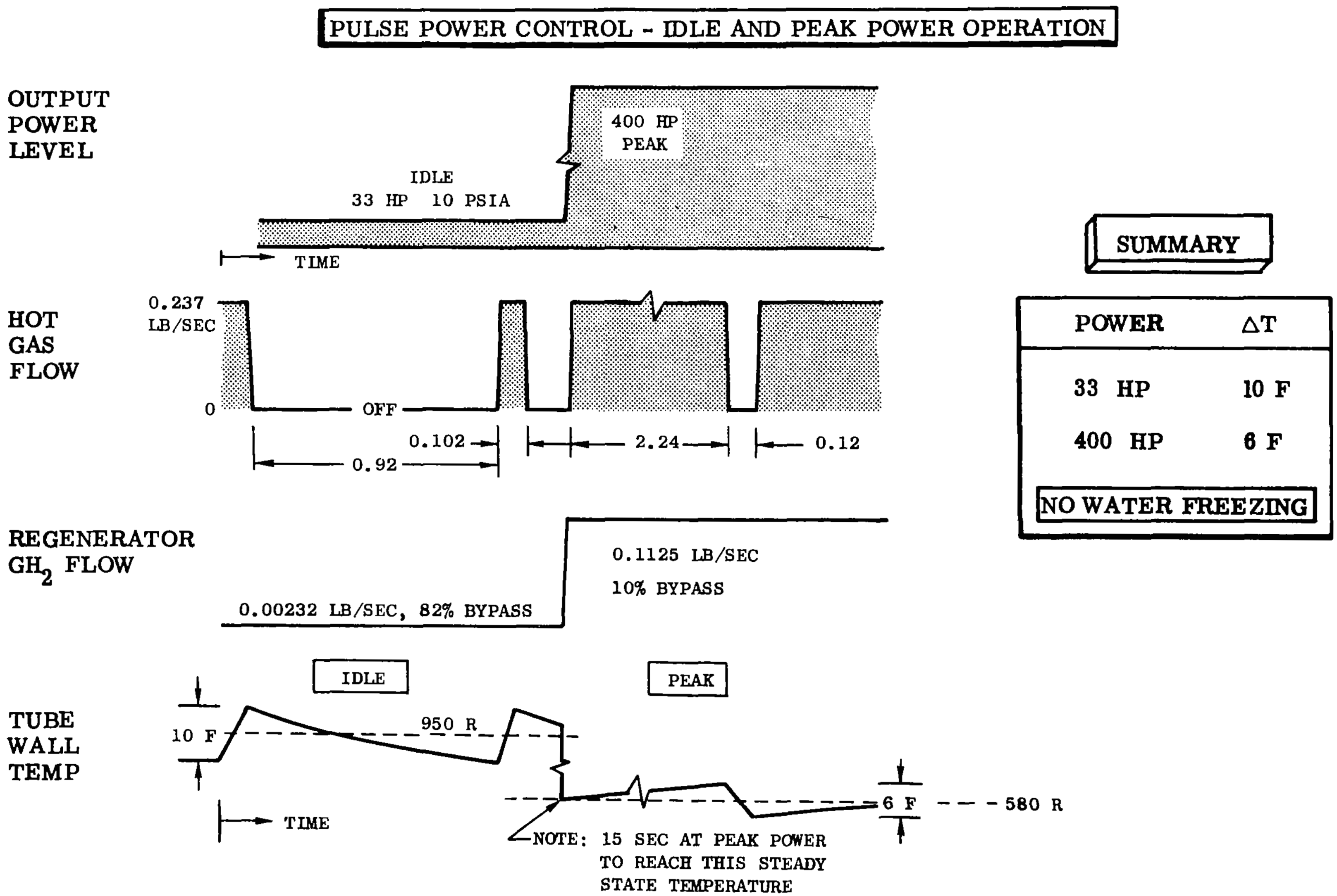


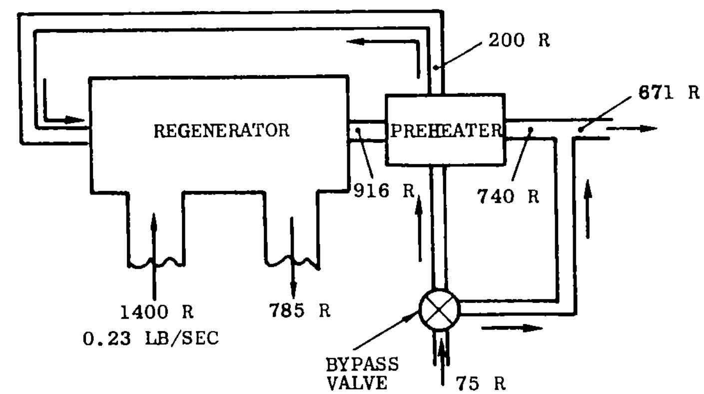
Figure 80. Regenerator Temperature Response

In order to provide more margin against condensation, a hydrogen-hydrogen preheater could be used upstream of the regenerator as shown in Fig. 81 . The preheater would function to heat the incoming 75 R hydrogen to a higher temperature, thus raising the wall temperature at the inlet section of the bundle and the hot gas exit section. To provide the same thermal conditioning (276 Btu/sec) the regenerator must be increased in size by about 60 percent, incurring a 15-pound weight penalty when the small pre-heater (5 pounds) is added to the system. The corresponding temperature profiles for the system with and without the pre-heater are shown in the figure. This weight penalty must be traded for increased margin against freezing and represents a backup scheme should problems be encountered in the development of the baseline concept.

A thermal shock analysis was performed on the regenerator at the tube inlet location in the front plate of the hydrogen inlet header. A hypothetical condition was analyzed which will never occur during actual operation; however, the resulting strain range of 0.0011 in./in gives a low cycle thermal fatigue life in excess of 100,000 cycles (for 20 shocks per mission - 5000 missions). Consequently, since the hypothetical case does not result in a life-limiting strain range, the more realistic operating transient effect was not analyzed. Referring to Fig. 82 , it is assumed that all of the hydrogen is being bypassed around the regenerator and that the entire regenerator is at 1400 R, the hot gas inlet temperature. This is a condition which would occur during orbiter re-entry with full hydraulic and lube cooling. It is then assumed that a step-change to 400 horsepower occurs, the 650 R hydrogen injector inlet temperature drops and 100 percent of the hydrogen flow at 75 R is "stepped" into the 1400 R tubes. The maximum hydrogen heat transfer coefficient occurs at the tube inlet since the thermal boundary layer is not fully developed. This value was calculated as 970 Btu/hr ft^2 F. Utilizing a Schmidt graphical analysis technique, the maximum radial temperature gradient of 165 F occurs after 50 milliseconds. The strain range was then calculated and fatigue life estimated based upon experimental data for 347 stainless steel.

Another potential problem area associated with shell and tube heat exchangers is that of tube vibration due to flow disturbances. If a tube resonant frequency is excited by hot gas flow, high cycle structural fatigue failure could occur under severe loading conditions. Testing was conducted in support of another program to evaluate vibration of injector tubes when subjected to crossflow of high velocity gas. Vortex shedding frequency, Strouhal number, and lift and drag coefficients were determined for a variety of tube designs and spacings. These results were utilized to predict the vortex shedding frequency for the 1000 ft/sec hot gas velocity across the hydrogen inlet tube bank. Based upon a Strouhal characteristic for aligned tube banks and fully developed vortex sheets, a shedding frequency of 20,200 cps was calculated (Fig. 83). This is far above the structural resonant frequency of the tube bank (144 cps). The vibration frequency due to pulsing operating ranges from 0.43 to 2.9 cps. Consequently, there should be no excitation of structural resonances and high cycle fatigue should be no problem.

OPERATION AT PEAK POWER, SEA LEVEL



- VEHICLE ADAPTABILITY
  - FOR EXTENDED DURATION FULL POWER OPERATION - INCREASED PROTECTION AGAINST CONDENSATION
- WEIGHT INCREASE OVER BASELINE
  - REGENERATOR 15 LB (60%)
  - PRE-HEATER 5 LB

TOTAL 20 LB

TEMPERATURE PROFILES

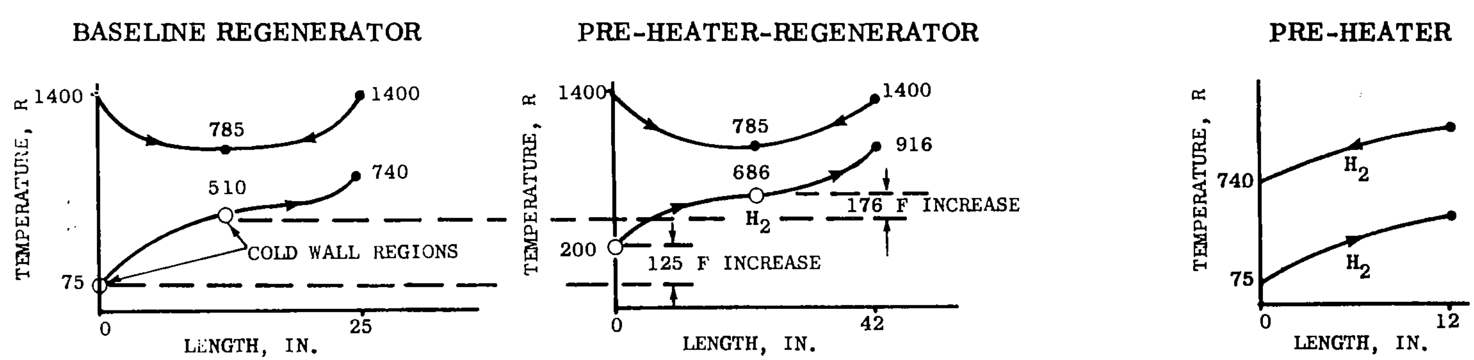


Figure 81. Regenerator with Hydrogen Pre-Heater

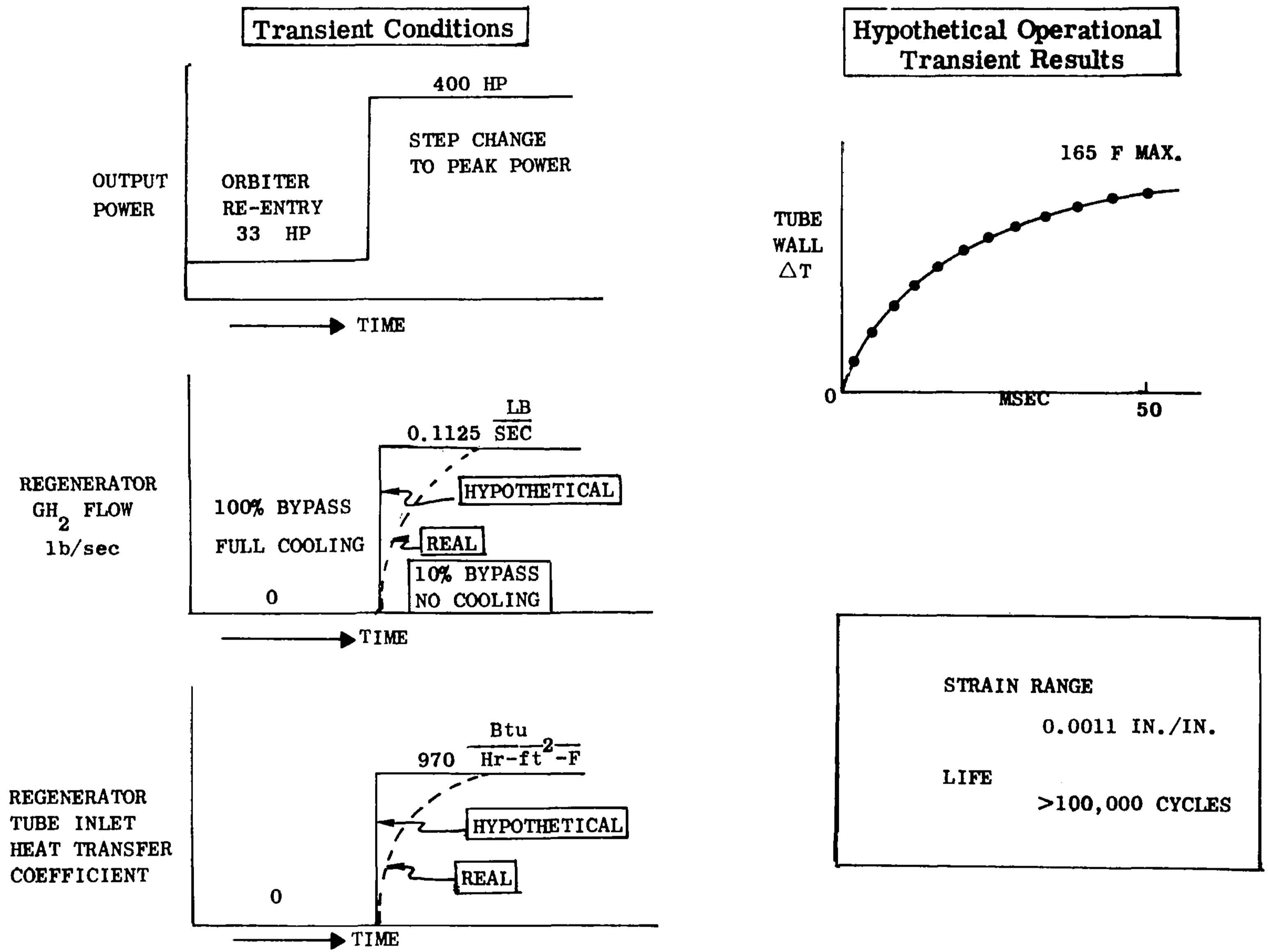
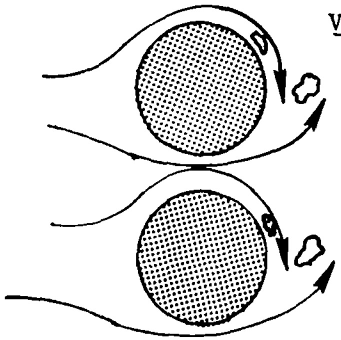


Figure 82. Regenerator Thermal Shock Analysis

FORCING FUNCTIONS



VORTEX SHEDDING

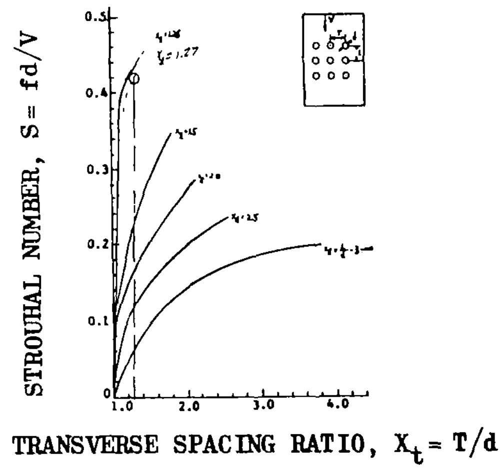
FULLY TURBULENT

VORTEX SHEET

(Re = 10,000)

CROSS FLOW REGIONS

OF REGENERATOR



INTERMITTENT FLOW

PULSE POWER CONTROL

FREQUENCY RANGE

SUMMARY

		Hz
TUBE NATURAL FREQUENCY		144
VORTEX SHEDDING		20,200
PULSING FLOW	0.43	2.9

NO RESONANT VIBRATION PROBLEM

REGENERATOR DESIGN

$X_t = X_L = 0.318/0.25 = 1.27$

V = 1000 FT/SEC (ON PULSE)

S = 0.42

VORTEX SHEDDING FREQUENCY = 20,200 CPS

PULSE CONTROL FREQUENCY RANGE:
0.43 Hz MINIMUM (PEAK)
2.9 Hz MAXIMUM (222 HP)

Figure 83. Regenerator Tube Vibration

B. HYDRAULIC COOLER DESIGN

The excellent cooling capability of hydrogen due to its high heat capacity, low temperature, and favorable transport properties is well known. The hydrogen is utilized in the baseline APU as a heat sink for both the hydraulic and lube system heat loads. When hydrogen cooling flow is limited to that required only for operating the TPU, the hydrogen inlet temperature and maximum oil/hydraulic fluid temperature uniquely determines the maximum heat load which can be accommodated.

This is strictly a thermodynamic limit as demonstrated in Fig. 84 . The cooling capability is independent of the number of hydrogen passes (direct-single pass, pre-heat, multiple pass) with multiple passes increasing the hydrogen inlet temperature to provide extra margin against locally freezing the oil. If further increases in hydrogen inlet temperature are required, an augmented flow method may be utilized whereby a blower or jet pump recirculates some portion of the main flow back to the hydrogen inlet. The lower schematic in Fig. 84 shows results for 50 percent flow augmentation using 1200 R hydrogen which raises the inlet temperature to 450 R, or 62 F higher than the dual pass approach. The resultant cooling is reduced by 40 percent. It is obvious that careful cooler design to handle low temperature hydrogen is necessary to obtain the maximum cooling benefits.

The baseline system hydraulic cooler design is presented with cut-away views in Fig. 85 . (The hydraulic cooler and lubricating oil cooler are very similar in design). It is a single pass counterflow tubular type design with fins on both the hydraulic side and hydrogen side. The hydraulic side finning is denser than the hydrogen side to maintain the higher wall temperatures needed to avoid prohibitive pressure drop due to the high shear stress which would result from an excessively cold wall. The primary mechanical design feature of the heat exchanger is the buffered region between the fluids. The buffer region is formed by a double wall from which any leakage, if it did develop through a crack in the wall, can be vented or contained without contaminating the opposite side. Furthermore, there are no mechanical or weld joints across the heat transfer surfaces greatly reducing the probability of wall crack failures, which would allow the fluids to mix if there were no buffer zone.

The detailed design of the hydraulic cooler (again also representative of the lube oil cooler) is shown in Fig. 86 . The overall dimensions of the cooler are 19 inches long and 4 inches in diameter. There are 101 fins on the hydrogen inlet side for an initial length of approximately 8 inches. This fin density reduces the heat transfer coefficient in the area where the hydrogen temperature is low (75 R), thus providing a wall temperature on the hydraulic side which is well above the pour point of the MIL-H-5606. The hydrogen side fins(202) are closely spaced approximately 0.020 inch apart and are 0.020 inch thick. Finning is increased to 202 fins on the hydrogen side after the initial 8-inch length section.

The design of the cooler was established with a digital computer program that performed analysis to determine wall temperature distributions both axially and radially. The axial temperature profile from such analysis and operational

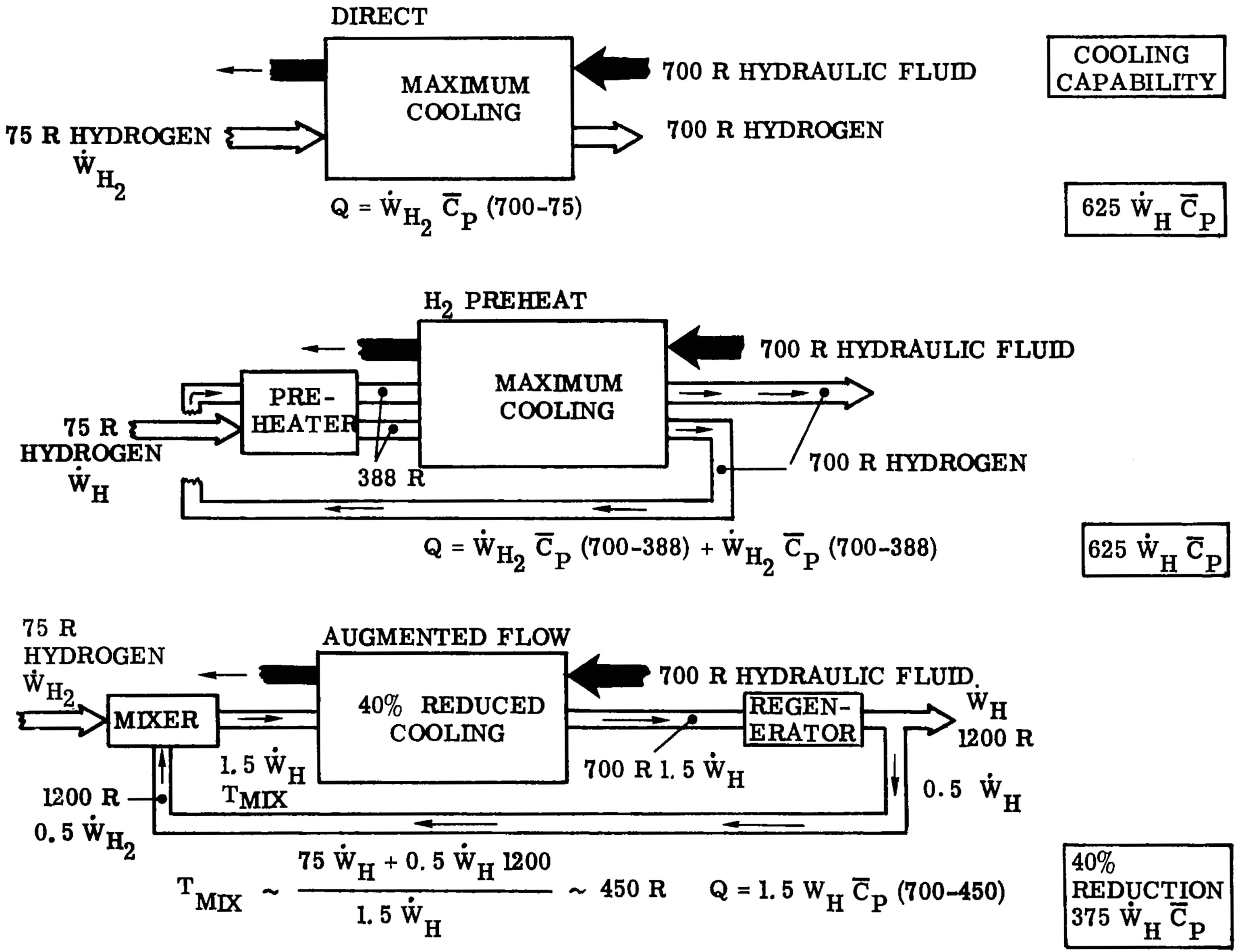
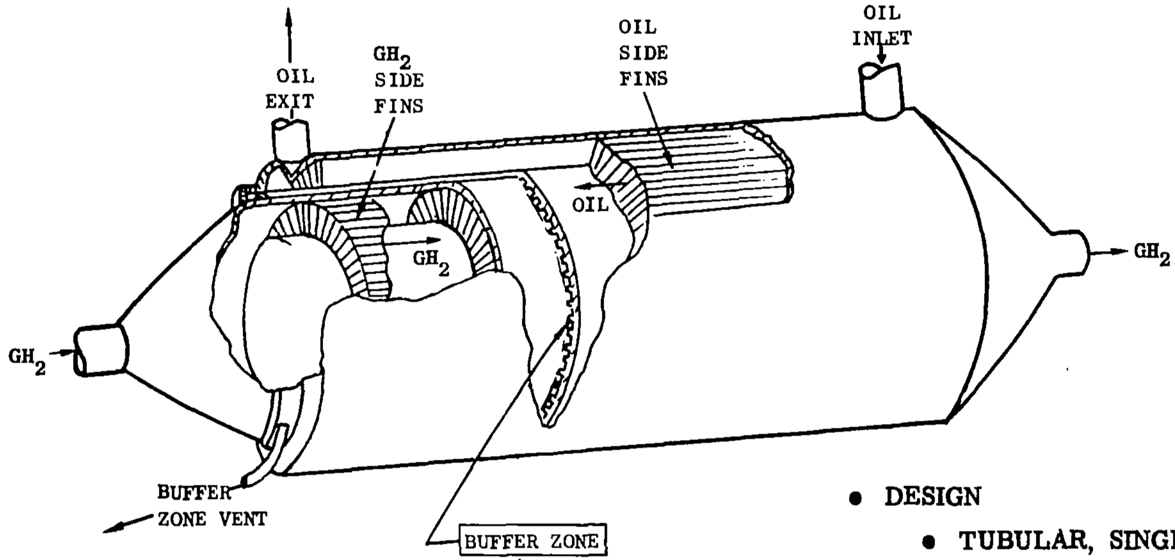
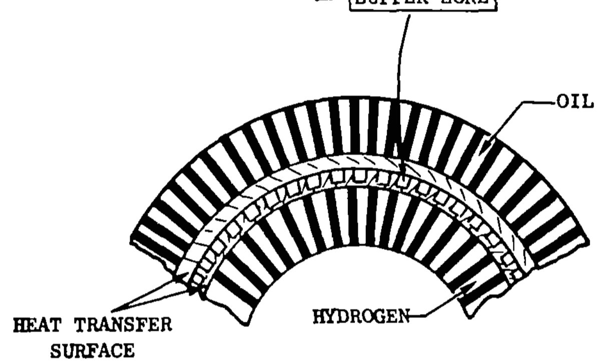


Figure 84. Thermodynamic Hydraulic Cooling Limitation





- DESIGN
  - TUBULAR, SINGLE PASS
- <u>SAFE OPERATION</u>
  - LOW PRESSURE BUFFER ZONE BETWEEN HYDROGEN (HGIH PRESSURE) AND OIL (LOW PRESSURE)
  - NO JOINTS BETWEEN FLUIDS (VIRGIN METAL)
- UTILIZES LIQUID HYDROGEN WITH NO OIL CONGEALING (I. E., LOW ΔP)
- LIGHT WEIGHT, COMPACT (BRAZED ALUMINUM CONSTRUCTION)

Figure 85. SS/APU Hydraulic/Lube Cooler

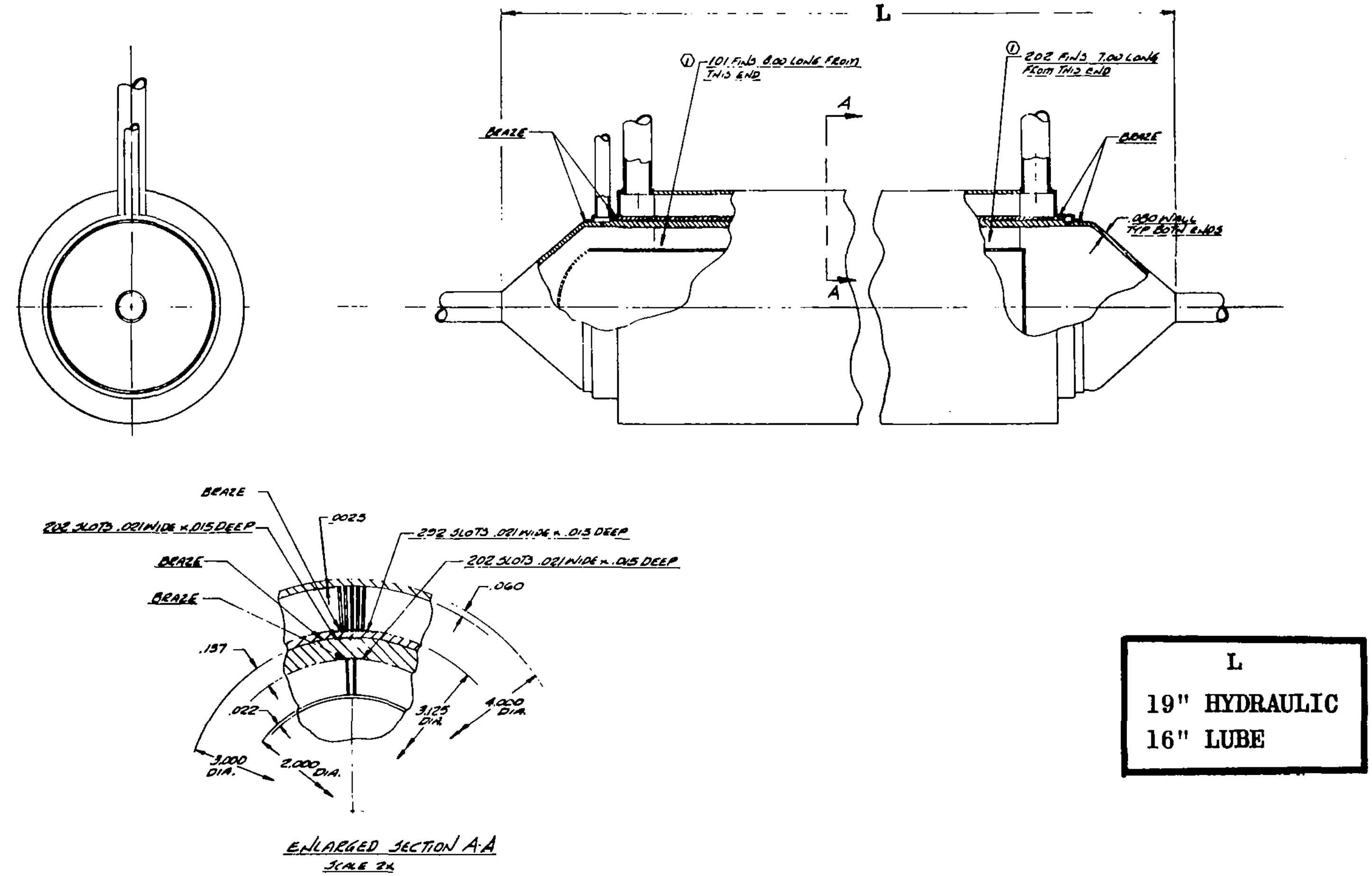


Figure 86. Hydraulic/Lube Cooler Design

conditions at the cooler design point is shown in Fig. 87 . This design point corresponds to the orbiter re-entry 50 horsepower condition at zero psia where the maximum heat load occurs. This low power point requires a minimum hydrogen flow, which, when coupled with the maximum hydraulic heat load, results in a maximum effectiveness of almost 95 percent for the cooler. This heat load is 45 horsepower with a total hydraulic flow of 6 gpm, representing case drain flow for two pumps. The hydraulic pressure is nominally 100 psi, although the cooler was designed to withstand 500 psi in the hydraulic system reservoir. The hydraulic side flow ΔP is 10 psi. Hydrogen side pressure is nominally 460 psi and the ΔP is less than 0.1 psi. Weight of the cooler is 10 pounds.

The fluid temperatureprofiles are indicated in the figures with the minimum wall temperature on the hydraulic side of the cooler at 510 R and occurring at the 137 R hydrogen inlet section.

A computer analysis determined the radial temperature distribution across the fins and buffer wall region of the hydraulic cooler for the lowest possible hydrogen inlet temperature to ensure that there would be no congealing of the oil on the hydraulic side. The results are presented in Fig. 88 . The operating conditions correspond to the 50 horsepower/zero psia conditions of orbiter re-entry with lube cooling. The lowest hydrogen temperature into the hydraulic cooler is 63 R, representing the steady-state cooling condition. The lowest hydraulic side wall temperature at these conditions based on the computer analysis is 475 R. The bulk hydraulic fluid temperature is 608 R. This is the most severe condition from the standpoint of low hydrogen temperature for hydraulic cooling. The steady-state hydrogen inlet condition without lube cooling is 137 R and the transient cooling condition is 125 R. Consequently, safe cooling exists under all hydrogen inlet temperature conditions.

If the hydraulic cooling load should be significantly increased to the point where considerably more heat must be removed out of the case drain flow, the wall temperature could be reduced significantly along with the hydraulic side wall temperature and a higher ΔP would result. A solution to this potential problem is to utilize a double pass hydrogen configuration as presented in Fig. 89 . Essentially, an additional hydrogen flow passage is added to the present design such that the hydrogen flows from the inlet end of the cooler over the entire length and then doubles back upon itself so that the hydraulic fluid will see an increased hydrogen temperature. The advantage of counter flow heat transfer is still maintained in this design and the hydraulic fluid is essentially isolated from the excessively cold hydrogen gas by the intermediate layer of warmed hydrogen. The temperature profile flow path is indicated in the plot underneath the flow schematic. With the hydrogen entering at 137 R (re-entry operating condition) it is heated to about 450 R in the first passage and then is conditioned to its final temperature of 671 in the counterflow passage.

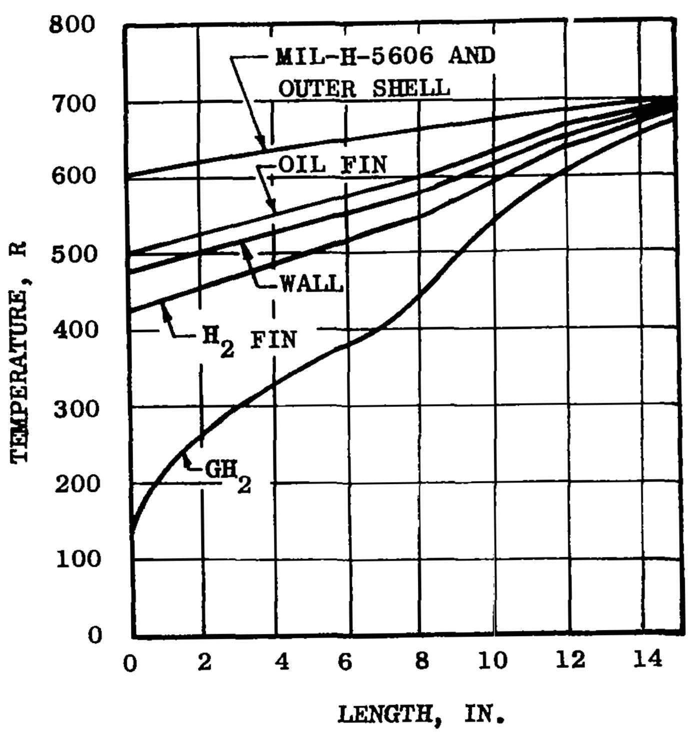
While the same amount of heat is transferred with this design there is a penalty incurred since length must be increased due to the higher wall temperature over the reference design. An additional weight penaly also occurs in the inner passages which must also be finned with 202 fins as in the

DESIGN POINT

ORBITER RE-ENTRY
50 HP, 0 PSIA
MAXIMUM HEAT LOAD
NO LUBE COOLING

*NOTE: GH_2 INLET TEMPERATURE IS 137 R UNDER CONDITION OF NO LUBE COOLING; AND LOG MEAN TEMPERATURE DIFFERENCE IS MINIMUM

AXIAL TEMPERATURE PROFILE



DESIGN: COUNTERFLOW, SINGLE PASS FINNED TUBULAR

MATERIAL: G061-T6 ALUMINUM BRAZED CONSTRUCTION

HEAT LOAD: 45 HP ($31.8 \frac{BTU}{SEC}$)

HYDRAULIC FLOW: 6 GPM (3 GPM/PUMP)

HYDRAULIC PRESSURE: 100 PSIA

HYDRAULIC ΔP: 10 PSI

HYDROGEN FLOW: $0.0162 \frac{LB}{SEC}$

HYDROGEN PRESSURE: 459 PSIA

HYDROGEN ΔP: 0.036 PSI (1.1 PSI @ $.125 \frac{LB}{SEC}$)

SURFACE AREA:

H_2 SIDE - 12 FT^2

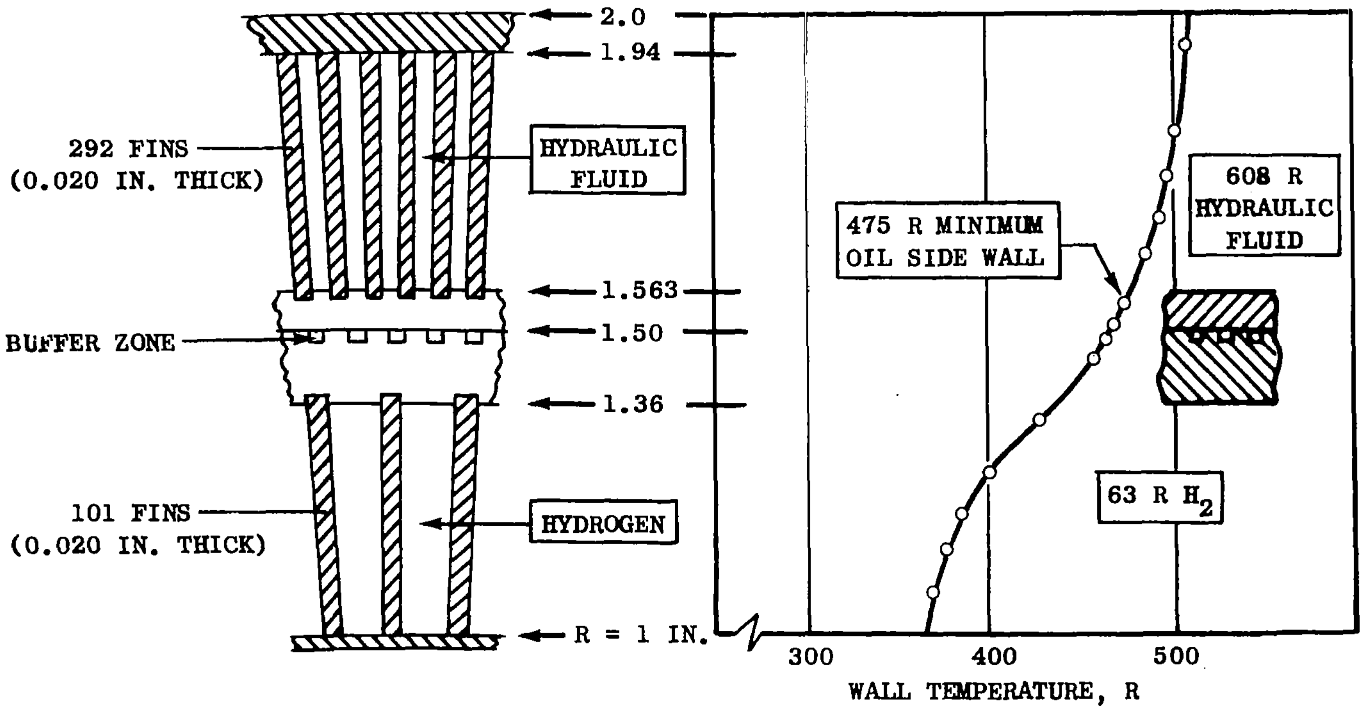
5606 SIDE - 21.1 FT^2

WEIGHT: 10 LBS

EFFECTIVENESS: 94.8%

Figure 87. Hydraulic Cooler Performance

ORBITER RE-ENTRY
50 HP 0 PSIA
WITH LUBE COOLING

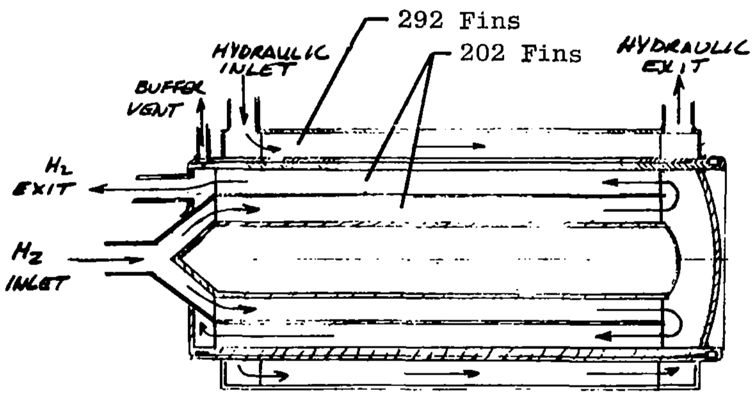


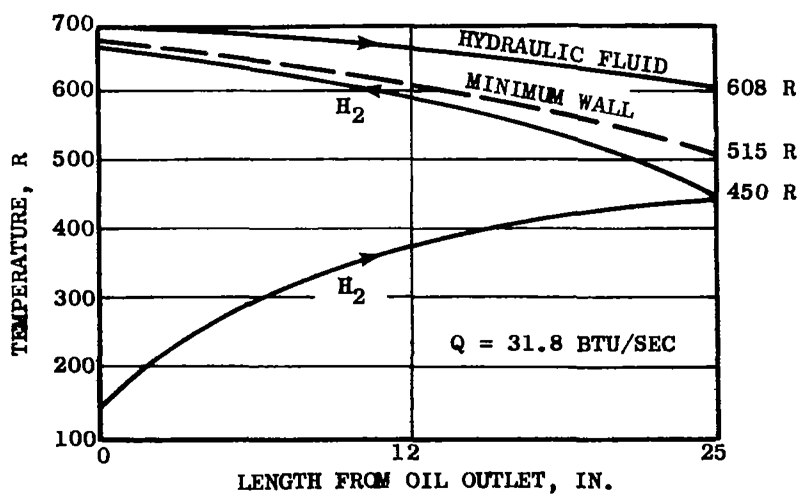
REQUIRED H_2 INLET TEMPERATURE
ORBITER RE-ENTRY

STEADY STATE HYDRAULIC COOLING	63 R	(WITH LUBE COOLING)
STEADY STATE HYDRAULIC COOLING	137 R	(NO LUBE COOLING)
TRANSIENT HYDRAULIC COOLING	125 R	(WITH LUBE COOLING)

Figure 88. Hydraulic Cooler Radial Temperature Distribution

H_2 PRE-HEAT DESIGN
ORBITER RE-ENTRY
50 HP 0 PSIA





VEHICLE ADAPTABILITY

- ACCOMMODATES INCREASED HYDRAULIC COOLING REQUIREMENTS (RE-ENTRY HEATING)

SAFE OPERATION

- NO LOW TEMPERATURE HYDROGEN IN CONTACT WITH OIL COOLING SURFACE
- WEIGHT PENALTY - 11 LB
- LENGTH INCREASE - 15 → 25 IN.

Figure 89. Alternate (H_2) Hydraulic Cooler

reference design. The total penalty incurred is 11 pounds. The length increase is estimated to be from 15 up to 25 inches.

Results of the hydraulic cooler stress anslysis are presented in Table 10 . All stress levels are below the endurance limit so that the cooler is not a life limited element. Stress due to axial thermal gradients are low with the outer tube in compression (6150 psi) and the inner tube in tension (2360 psi), so that a bellows expansion joint is not needed. Fin stress is also very low with the maximum occurring at the tip of fin on the hydrogen side (5200 psi tension). The cooler is of a forgiving design since it can easily accommodate a 1000 psi H_2 pressure and up to 500 psia on the hydraulic side without yielding the 6061-T6 aluminum.

C. LUBE OIL COOLER

The lube oil cooler preformance at its design point (booster cruise) of 33 horsepower and 10 psia is summarized in Fig. 90. This is where the maximum heat of 13.4 Btu/sec occurs with the minimum hydrogen flow, thus maximum effectiveness of the cooler is required at this condition. The axial temperature profile is indicated along with the operating conditions and design characteristics. The minimum wall temperature for this cooler is 651 R. A ΔP of 5 psi exists on the oil side for a flow of approximately 0.6 lbs/sec. It should be noted that the oil heat load of 19 horsepower is approximately 3 times that required to be accommodated on the orbiter, which has a much smaller alternator.

The counter flow single pass tubular design is essentially identical to that previously described for the baseline hydraulic cooler with the exception that the hydrogen fin density is constant over the entire hydrogen pass, since the minimum temperature inlet requirement is not as severe as in the hydraulic cooler.

This heat exchanger design is also buffered to provide protection against hydrogen leakage into the lubrication system in the event of a wall failure. It is also constructed with no mechanical or weld joints within the heat transfer surface separating the fluids. The weight of this cooler is approximately 8 pounds with an effectiveness of 91 percent.

D. HYDROGEN OXYGEN TEMPERATURE EQUALIZER

A cutaway view of the hydrogen/oxygen equalizer is shown in Fig. 91. It is a high effectiveness single pass tubular counterflow heat exchanger that equalizes temperature of the oxygen within $\pm$20 F of the hydrogen temperature.

The heat exchanger provides fast response to meet changes in inlet temperature thus assuring maintenance of mixture ratio within acceptable limits by satisfactory temperature equalization at the exit. This fast response is obtained by utilization of moderate ΔP on both the hydrogen and oxygen side, resulting in high heat transfer coefficients coupled with a lightweight wall to give excellent accomodation to temperature transients. Safe operation is assured through use of a buffered design (double wall) with no joints within the heat transfer surface separating the two propellants.

TABLE 10

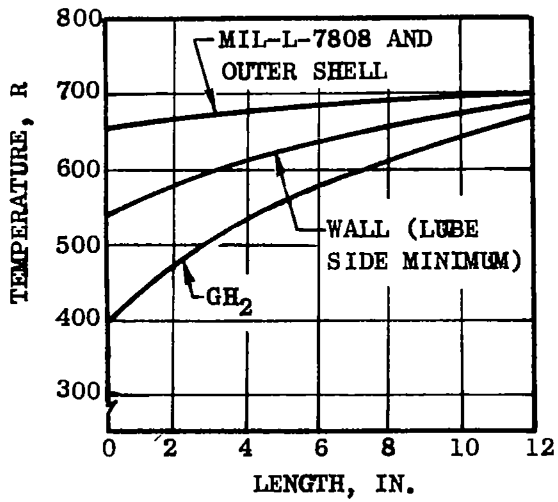
HYDRAULIC COOLER STRESS ANALYSIS

	Outer Tube (Hydraulic Side)	Middle Tube (Hydrogen Side)	Fins
Working Pressure	100 psia*	460 psia**	
Radial Stress	3320 psi	6450 psi	
Axial Stress $\Delta T = 65$ F	6150 psi	2360 psi	3240 psi - Oil Side (Compression)
Combined Stress	7000 psi***	6000 psi***	5200 psi - H_2 Side (Tension)
Yield Stress (700 R)	31000	31000	31000
Yield Safety Factor @ Working Pressure	4.43	4.56	
Yield Safety Factor @ Maximum Pressure	1.75	2.36	

* 500 psia maximum reservoir pressure
** 1000 psia maximum
*** Endurance limit = 12600 psi; life $> 10^7$ cycles

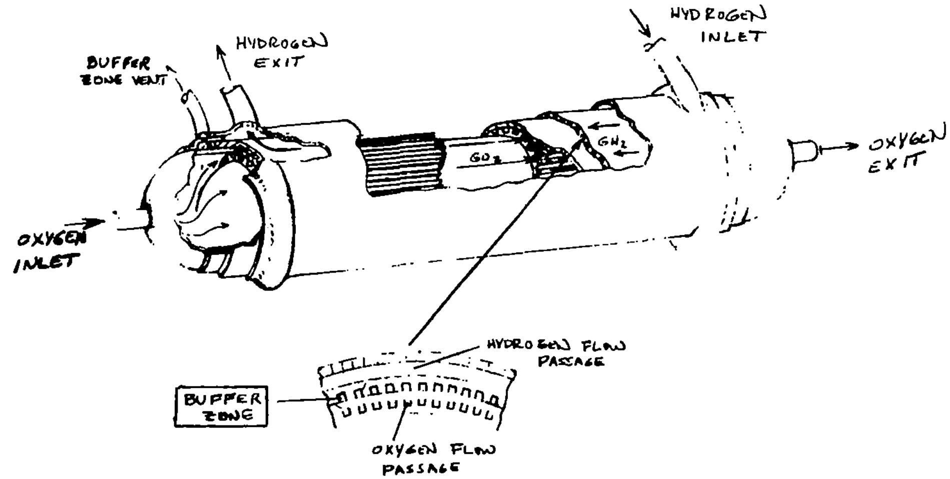
Design Point
BOOSTER CRUISE 33 HP, 10 PSIA MAX. HEAT LOAD

AXIAL TEMPERATURE PROFILE



DESIGN:	COUNTERFLOW, SINGLE PASS, FINNED TUBULAR
MATERIAL:	6061-T6 ALUMINUM, BRAZED
WEIGHT:	8 LBS
HEAT LOAD:	19 HP (13.4 BTU)
EFFECTIVENESS:	91%
OIL FLOW:	0.59 LB/SEC
OIL PRESSURE:	100 PSIA
OIL ΔP	5 PSI
HYDROGEN FLOW:	0.013 LB/SEC
HYDROGEN ΔP	0.02 PSI
SURFACE AREA:	H_2 SIDE - 13 FT^2
	OIL - 11 FT^2

Figure 90. Lube Cooler Performance



- SAFE OPERATION
  - BUFFERED DESIGN
  - NO JOINTS BETWEEN PROPELLANTS
- PERFORMANCE FAIL-SAFE/HIGH PRESSURE CAPABILITY
  - SINGLE PASS/TUBULAR DESIGN
  - TEMPERATURE EQUALIZED WITHIN ± 20 F
  - FAST RESPONSE
  - COMPACT/LIGHTWEIGHT
- FLEXIBILITY
  - ACCOMMODATES RAPID PROPELLANT INLET TEMPERATURE CHANGES

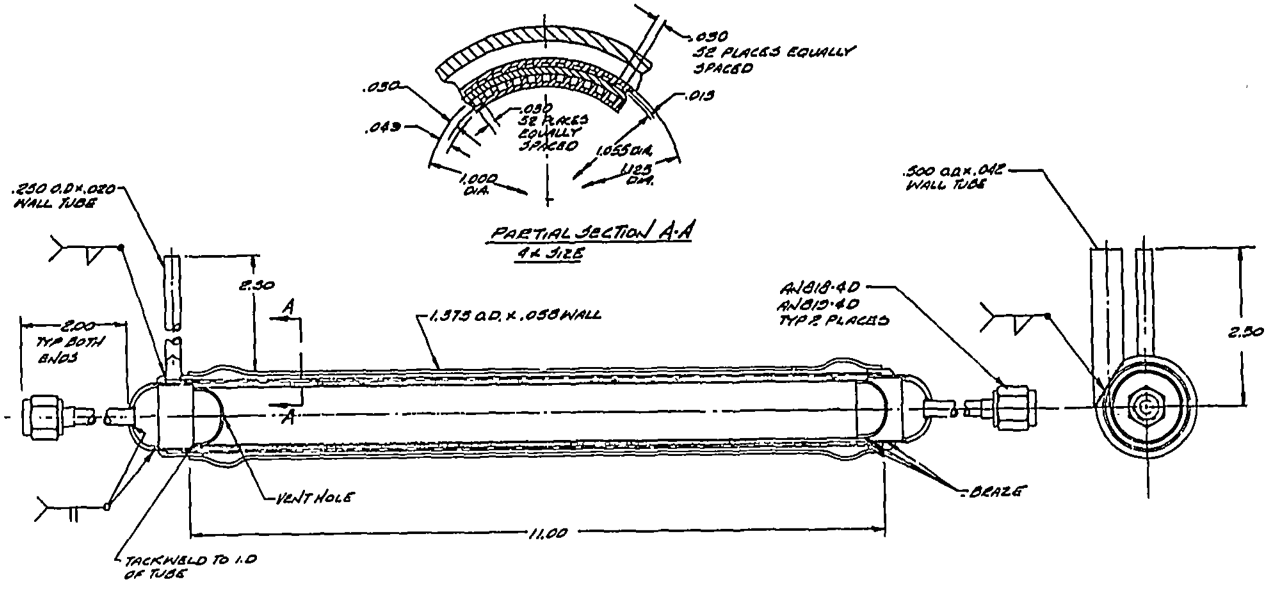
Figure 91. H_2/O_2 Temperature Equalizer

The heat exchanger is a forgiving design in that it can easily accommodate 1,000 psi maximum system pressure should the upstream hydrogen pressure regulator fail. It can also accommodate 1,000 psi pressure differential across either side without collapsing the tubes.

Details of the equalizer design are presented in Fig. 92 . The weight of the heat exchanger is less than 1 pound. Construction material is 6061-T6 aluminum throughout, which can be obtained as a post brazed stress condition. The brazing alloy that is used in fabrication is an Alcoa 718 designation.

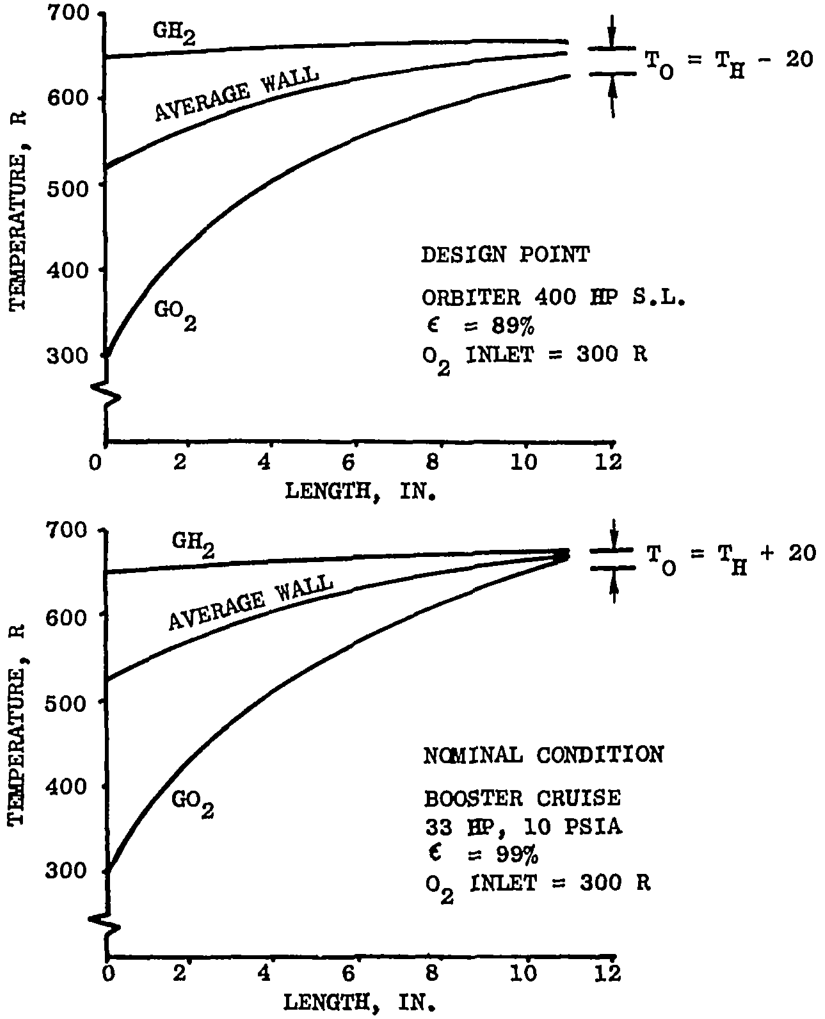
The axial temperature profile is presented in Fig. 93 along with the operating conditions at the equalizer design point, which is the same as the regenerator design point (400 horsepower at sea level). This is the maximum heat transfer condition for the equalizer occurring with an oxygen ΔP of 23 psi and a hydrogen Δ P of 11 psi. The oxygen outlet temperature of 630 R is within 20 degrees of the hydrogen outlet temperature of 650 R. The effectiveness of the equalizer at this opeating condition is 89 percent. At the low power or idle condition, the heat exchanger becomes more effective reaching a level of 99 percent, with the oxygen temperature exceeding the hydrogen outlet temperature by about 20 degrees.

Results of the equalizer stress analysis are presented in Table 11 . All stresses are below the endurance limit so that there is no life limitation to the element. The maximum stress of 9880 psi occurs in the oxygen header resulting in a safety factor on yield of 3.27. No expansion joint is needed in the outer tube since the middle wall is only about 30 F cooler than the outer tube due to the high hydrogen side heat transfer coefficient. The equalizer is a forgiving design since it can support a side to side pressure differential of 1000 psia without exceeding the 32000 psi yield stress.



MATERIAL:	6061-T6 ALUMINUM
BRAZE:	ALCOA 718

Figure 92. H_2/O_2 Equalizer Design



DESIGN: COUNTERFLOW, SINGLE PASS, TUBULAR

MATERIAL: 6061-T6 ALUMINUM

WEIGHT: 0.9 LBS

HEAT FLOW: 8.7 $\frac{BTU}{SEC}$

GH_2 FLOW: 0.125 LB/SEC

GO_2 FLOW: 0.104 LB/SEC

GH_2 INLET PRESSURE: 451 PSIA

GH_2 ΔP: 11 PSI

GO_2 INLET PRESSURE: 459 PSIA

GO_2 ΔP: 23 PSI

SURFACE AREA:

GH_2 - 33 IN^2

GO_2 - 66 IN^2

HEAT TRANSFER COEFFICIENTS:

GH_2 - 2560 $\frac{BTU}{HR\ FT^2}$ (AVG)

GO_2 - 723 $\frac{BTU}{HR\ FT^2\ F}$ (INLET)

1000 $\frac{BTU}{HR\ FT^2\ F}$ (EXIT)

TIME CONSTANTS:

GH_2 - 0.514 SEC

GO_2 - 0.760 SEC

Figure 93. H_2/O_2 Equalizer Performance

TABLE 11

H_2/O_2 EQUALIZER STRESS ANALYSIS

MATERIAL:
- 6061 ALUMINUM THROUGHOUT
- T6 CONDITION AFTER BRAZE
- MAXIMUM PRESSURE 1000 PSIA**
- ENDURANCE LIMIT 11700 PSIA***

	OUTER TUBE	MIDDLE TUBE	O_2 HEADER	H_2 HEADER
WORKING PRESSURE	460 PSIA	460 PSIA		
RADIAL STRESS	5450 PSI (T)	0	1630 PSI TENSION 8250 PSI BENDING	1070 PSI
AXIAL STRESS*	3110 PSI (C)	2330 (T)	-	3110 PSI
COMBINED STRESS	6250 PSI	2330	9880	4180 PSI
YIELD STRESS (671R)	32000 PSI	32000 PSI	32000	32000
YIELD SAFETY FACTOR	5.1	13.7	3.27	7.7

*Outer-inner wall ΔT = 30F.

**If failure of upstream regulator P $\longrightarrow$ 1000 psi
middle tube collapsing pressure P = 5050 psi - no problem
minimum yield safety factor = 1.5 (O_2 header) @ 1000 psi.

***All stresses $<$ endurance limit; life $> 10^7$ cycles.

IX CONTROL COMPONENTS

Primary control elements are the elements associated with the control of the major function of the Space Shuttle APU System, namely, flow control of hydrogen and oxygen at the proper mixture ratio to the APU turbines. The five valves listed in Table 12 control those functions listed beside each component. Each of the two GH_2 bypass valves perform single functions: that of regenerator temperature control, and hydraulic cooler temperature control. The gaseous hydrogen pressure regulator controls the hydrogen supply pressure. The oxygen pressure is also regulated as a function of the hydrogen supply pressure in that the differential regulator uses the hydrogen pressure as a reference. And, the power control valve is the final control element for controlling turbine power. The power control valve accomplishes this by pulse modulation of the hydrogen and oxygen flow. The system also contains a variety of auxiliary control elements listed in Table 13 with their associated functions. The auxiliary control elements provide means for starting and stopping the system under normal and emergency circumstances and also consist of a number of sensors which are not active control devices but monitoring devices for limit control.

In selecting component configurations for each of the primary and auxiliary components, a design philosophy shown in Table 14 was established consistent with the requirements of the Space Shuttle APU System. The basis of this philosophy is to avoid expensive and long-term development programs associated with any component and emphasizes state-of-the-art, off-the-shelf configurations. Any new design that was required for the APU system used design elements taken from qualified, proven and reliable products.

A. HYDROGEN PRESSURE REGULATOR

Table 15 shows a list of the major requirements for the hydrogen pressure regulator. Significant among these requirements are the variation in supply pressure, 495 to 1,000 psig, the regulated pressure and the required accuracy, 460 ±10 psig, and the flow requirement of .125 lb-sec under nominal conditions. These requirements size the regulator and determine the complexity of the component that is required to meet the regulation accuracy requirement. The relief function of limiting the system pressure to 485 ±10 psig is achieved with an integral exhaust feature built into the regulator rather than requiring an additional complete relief valve assembly. The required accuracy and flow for the gaseous hydrogen regulator is very close to the requirements for the J-2 pneumatic control assembly regulator. The J-2 pneumatic control assembly is shown in Fig. 94 in its production configuration as used on the J-2 engines for the Saturn application.

Figure 95 shows the J-2 control assembly in schematic form with the regulation portion outlined. These same lements will be used in the gaseous hydrogen regulator for the Space Shuttle APU System.

TABLE 12

SS/APU PRIMARY CONTROL ELEMENTS

COMPONENT		FUNCTION
GH_2 PRESSURE REGULATOR AND RELIEF VALVE	-	HYDROGEN SUPPLY PRESSURE REGULATION - OXYGEN SYSTEM PRESSURE REFERENCE
DIFFERENTIAL PRESSURE REGULATOR AND RELIEF VALVE	-	OXYGEN PRESSURE REGULATION
GH_2 BYPASS VALVES	-	REGENERATOR TEMPERATURE CONTROL AND HYDRAULIC COOLER TEMPERATURE CONTROL
POWER CONTROL VALVE	-	TURBINE POWER CONTROL

TABLE 13

SS/APU AUXILIARY CONTROL ELEMENTS

COMPONENT		FUNCTION
TANK SHUTOFF VALVES	-	PROPELLANT SHUTOFF
EMERGENCY SHUTOFF VALVE	-	TURBINE POWER SHUTOFF
ATTENUATOR TANK VENT VALVE	-	TANK VENTING
TURBINE SPEED SENSORS	-	TURBINE SPEED SENSING
TURBINE INLET TEMPERATURE SENSORS	-	TURBINE GAS TEMPERATURE SENSING
EQUALIZER GAS TEMPERATURE SENSOR	-	EQUALIZER HYDROGEN OUTLET TEMPERATURE
PROPELLANT INLET FILTERS	-	PROPELLANT FILTRATION
HYDRAULIC AND LUBE OIL TEMPERATURE SENSORS	-	HYDRAULIC AND LUBE OIL TEMPERATURE
SIGNAL PRESSURE BIAS VALVE	-	TIT LIMIT CONTROL

TABLE 14

VALVE AND REGULATOR DESIGN PHILOSOPHY EMPHASIS

- ALL CONCEPTS WITHIN CURRENT STATE-OF-THE-ART
- OFF-THE-SHELF QUALIFIED HARDWARE USED WHERE AVAILABLE
- CONSERVATIVE DESIGN FOR RELIABILITY AND LIFE
- NEW DESIGNS BASED ON QUALIFIED COMPONENT CONCEPTS

TABLE 15

GH_2 PRESSURE REGULATOR AND RELIEF VALVE REQUIREMENTS

<u>OPERATIONAL</u>: REGULATE PRESSURE OF GASEOUS HYDROGEN SUPPLIED TO SYSTEM

<u>COMPONENT CONDITIONS</u>:

3/4 INCH LINE	460 $\pm$ 10 PSIG REGULATED OUTLET
495 TO 1000 PSIA INLET	0.125 LB/SEC NOMINAL FLOW
-60 TO -385F FLUID	485 $\pm$ 10 PSIG RELIEF SETTING
-60 TO 130F AMBIENT	

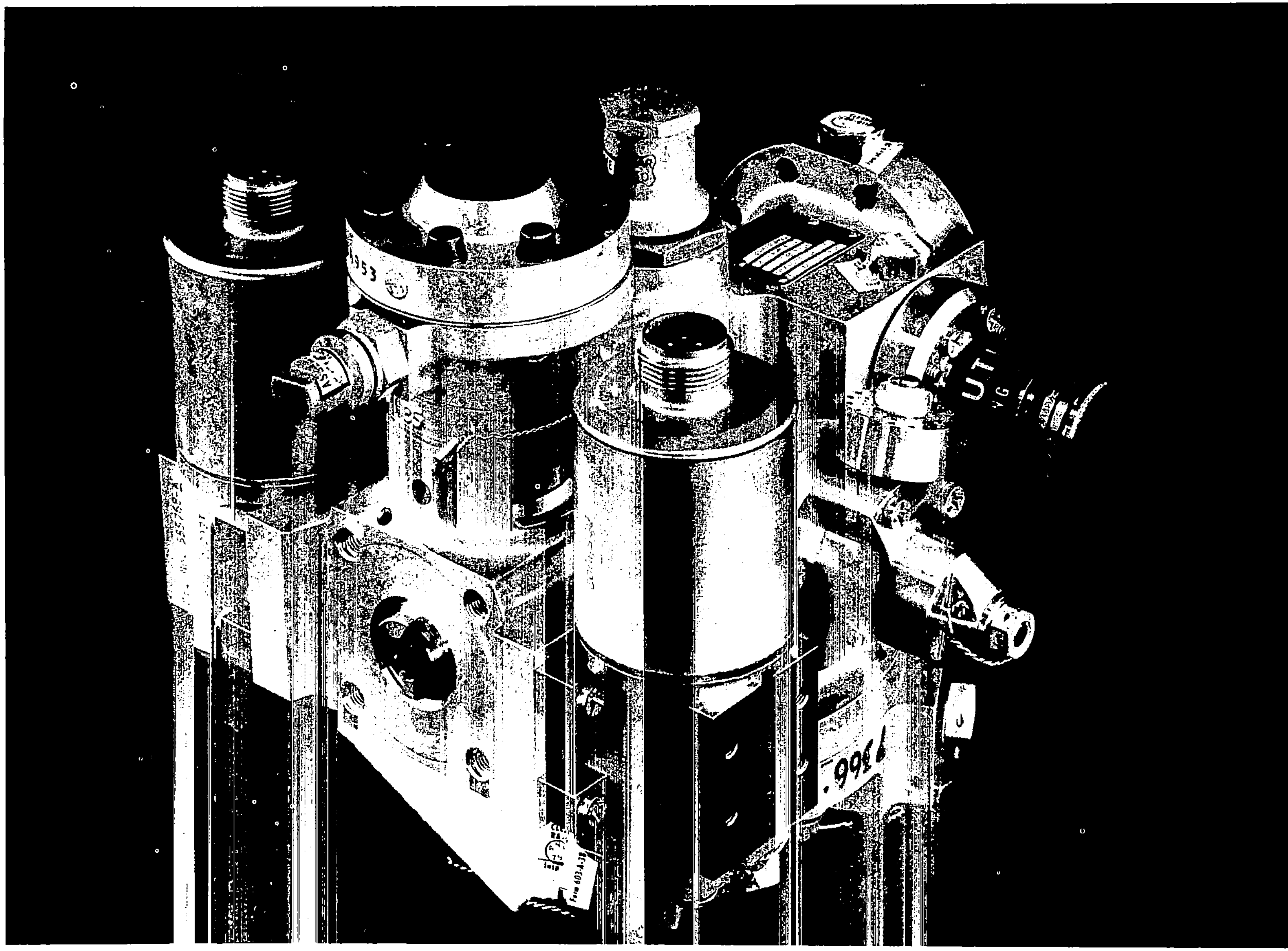


Figure 94. J-2 Pneumatic Control Assembly

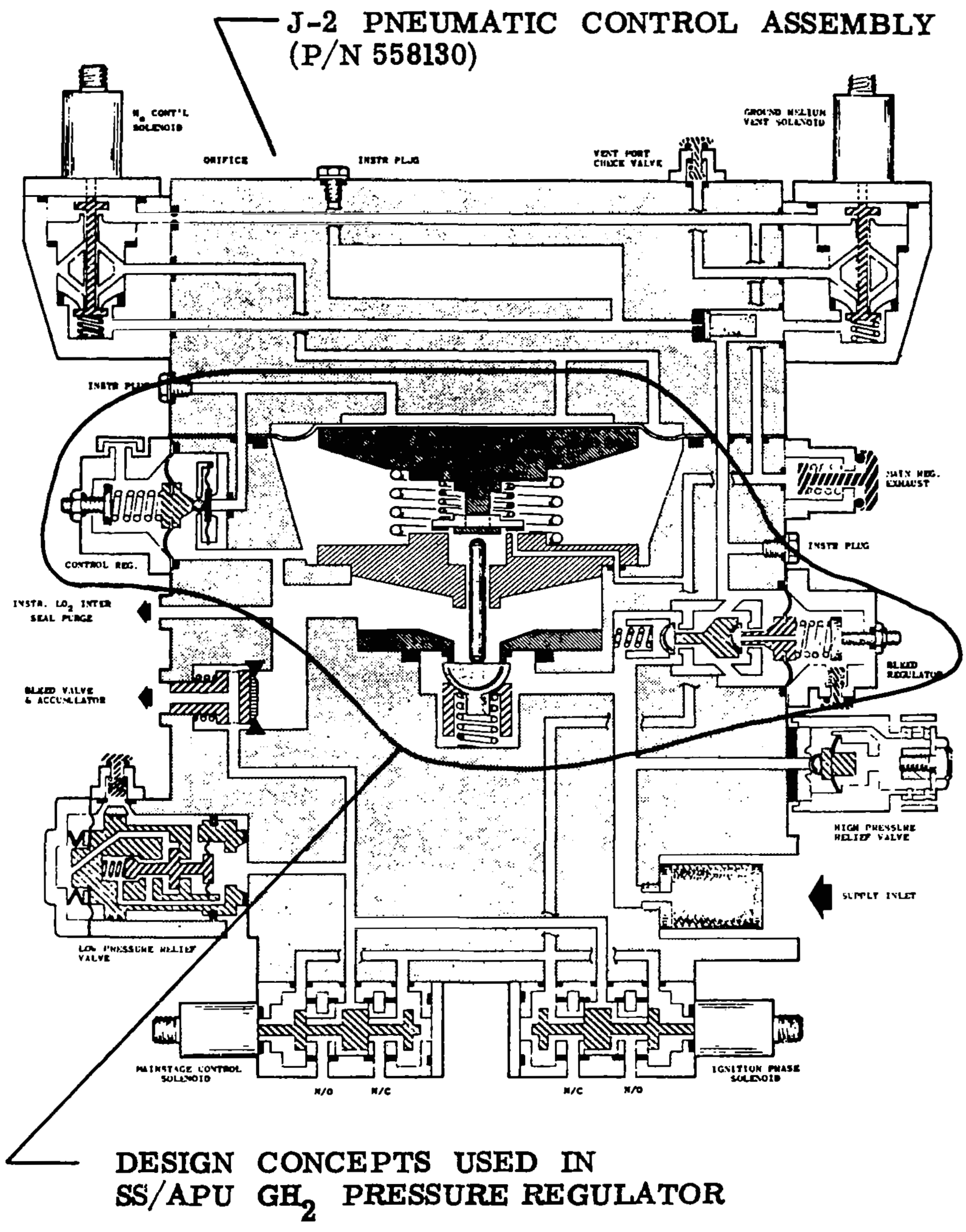


Figure 95. Proven Design Concept for GH_2 Pressure Regulator

Figure 96 shows these elements arranged in a schematic form that illustrates the operation of the component. The component consists of three main sections. The actuator section or the main regulation portion of the device is shown in the center. The bleed regulator which supplies the pressure to the dome of the actuator is on the left while the controller which senses the regulator pressure and adjusts the dome pressure is required to maintain the regulation accuracy is shown on the right. The actuator also contains the relief function with an arrangement termed a 3-way non=interflow valve. With this arrangement the main inlet can be closed before the exhaust or relief function opens. Similarly, the exhaust function, if starting from the open position, can be closed before opening the inlet.

With this arrengement either the inlet valve is open and the regulator is flowing, or the exhaust valve is open exhausting excess pressure to atmosphere. At no time however are both the exhaust valve and the inlet open. Accordingly, there is a period of dead-band travel by the actuator when neither the exhaust valve nor the inlet valve is open.

This regulator can be categorized as a dome-loaded, pilot-operated device. The dome-loader, or the bleed regulator on this schematic, provides a reference pressure to the dome of the actuator, and the regulator operates very similar to a directly actuated regulator in which a spring device is used as a reference. In the dome-loaded configuration, the pressure takes the place of the spring reference which results in a significantly lighter assembly and more accurate regulation. The accuracy requirements for the APU System are sufficiently tight to preclude the use of a dome-loaded regulator alone. Pilot operation is introduced by interposing an orifice between the bleed regulator and the actuator and adding a pilot valve downstream of the actuator dome. The bleed regulator is set to regulate a low flow of hydrogen at a setting somewhat above the regulated pressure required from the main regulator.

If the controller is closed and seated as shown in the schematic, the pressure in the actuator area would rise to the setting of the bleed regulator eventually since there is no flow out of the regulator dome. On the other hand, if the controller is positioned slightly open, the flow that is passing through the orifice will eventually be established through the controller pilot and the pressure that remains in the actuator dome will be a function of these two resistances.

In practice, the controller senses the gaseous hydrogen regulator pressure and positions the pilot valve against the reference coil spring, such that the flow through the orifice equals the flow through the pilot valve and the pressure in the regulator actuator dome becomes the reference pressure for gaseous hydrogen regulation. A reduction in gaseous hydrogen pressure will cause the controller to position the pilot valve closer to the seat increasing the resistance of the pilot valve and the pressure in the actuator dome will increase. This actuates the main valve inlet farther open to return the hydrogen pressure to the desired set point. If, on the other hand, the pressure in the downstream side of the regulator is above the desired set point, the controller

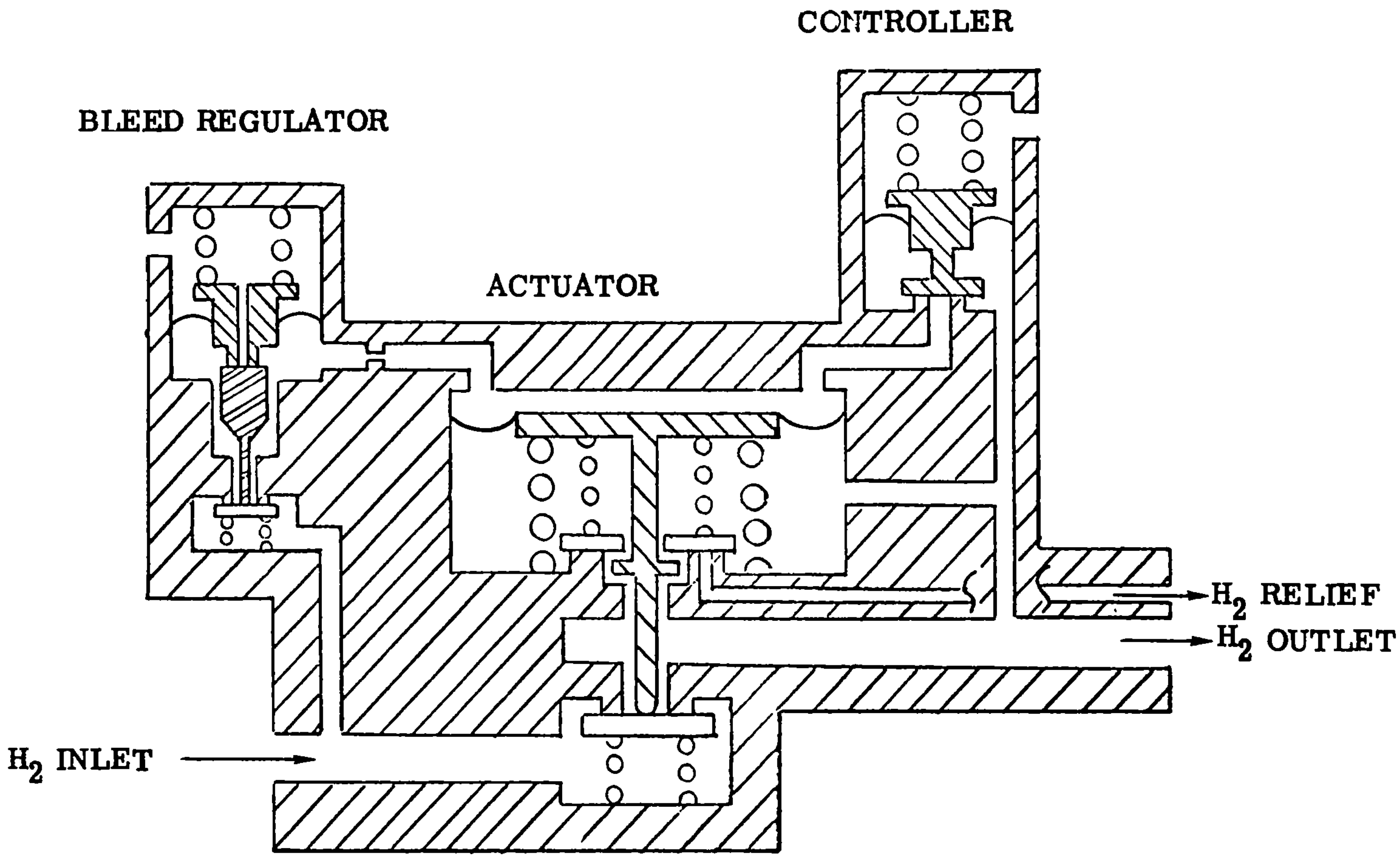


Figure 96. GH_2 Pressure Regulator and Relief Valve Schematic

will be positioned to a larger pilot valve flow area, decreasing the resistance of the controller pilot valve and reducing the pressure in the actuator dome to allow the inlet seat to move toward the closed position. If the downstream pressure is sufficiently high, the actuator will first close the main inlet then will stroke through a small deadband to lift the actuator exhaust seat and exhaust the excess pressure to atmosphere.

As stated previously, the gaseous hydrogen pressure regulator is an adaptation of the J-2 regulator design. Design of the regulator for the APU is shown in cross-section in Fig. 97 using those same detail elements. In fact, this design is very similar to an early version of the J-2 regulator assembly where discrete components were used.

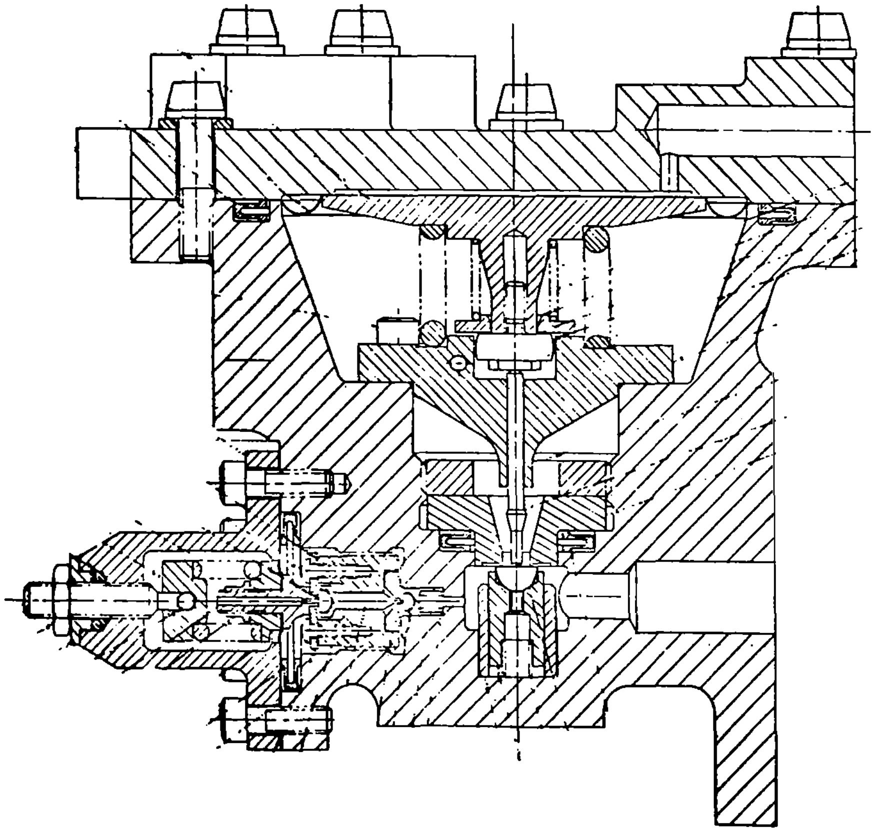
B. DIFFERENTIAL PRESSURE REGULATOR

The differential pressure regulator is a device very similar to the gaseous pressure regulator. The requirements for this element are shown in Table 16 . The device regulates gaseous oxygen pressure using the regulated gaseous hydrogen pressure as a reference. Figure 98 shows the basic J-2 regulator assembly with that portion of the regulator used for the differential regulator outlined. In this application multiple diaphragms must be provided in order to safely isolate the gaseous oxygen system from the gaseous hydrogen system.

Actually, three diaphragms are used; two buffered double diaphragms and a centrally located vent separator membrane. The upper buffered diaphragm senses gaseous hydrogen pressure. The lower buffered diaphragm senses gaseous oxygen pressure. Any leakage occurring across the diaphragm layer in contact with the hydrogen or oxygen would result in a pressure rise in the buffered zone as a leakage detection. Leakage across either the entire upper or lower buffered diaphragm into the vent area is separated from the other propellant by both the middle membrane and the other buffered diaphragm, and vents to atmosphere. Figure 99 illustrates a schematic of the APU ΔP regulator with the multiple diaphragms.

A 3-way non-interflow inlet valve arrangement is used, such that the regulator will not only provide regulated flow in response to gaseous oxygen flow demands, but also will provide a relief function which can vent excess oxygen pressure safely overboard.

The detail design of the differential pressure regulator is shown in Fig. 100 where most of the actuator elements again are the same as the detail parts used in the gaseous hydrogen regulator. The differential regulator configuration is somewhat more simplified than the gaseous hydrogen pressure regulator in that a ready reference source is available and the bleed regulator and controller sections are not needed for regulation accuracy. The accuracy of the differential pressure regulator is equivalent to the accuracy of the gaseous hydrogen pressure regulator since the gaseous hydrogen is the reference.



FEATURES

- METAL DIAPHRAGM
- DOME LOADED
- PILOT CONTROLLED
- QUALIFIED J-2 COMPONENT
- NO DYNAMIC SEALS

Figure 97. GH_2 Pressure Regulator

TABLE 16

DIFFERENTIAL PRESSURE REGULATOR AND RELIEF VALVE REQUIREMENTS

OPERATIONAL: REGULATE OXYGEN PRESSURE AT EQUALIZER OUTLET TO EQUAL HYDROGEN PRESSURE

COMPONENT CONDITIONS:

3/8 INCH LINE	440 PSIG NOMINAL SENSED GH_2 PRESSURE
495 TO 1000 PSIG INLET	+ 1 1/2-7 PSI REGULATION AT ALL CONDITIONS
-60 TO -160F FLUID	0.105 LB/SEC NOMINAL FLOW
-60 TO +130F AMBIENT	NORMALLY CLOSED
RELIEF SETTING 5 TO 10 PSI ABOVE SENSED PRESSURE	

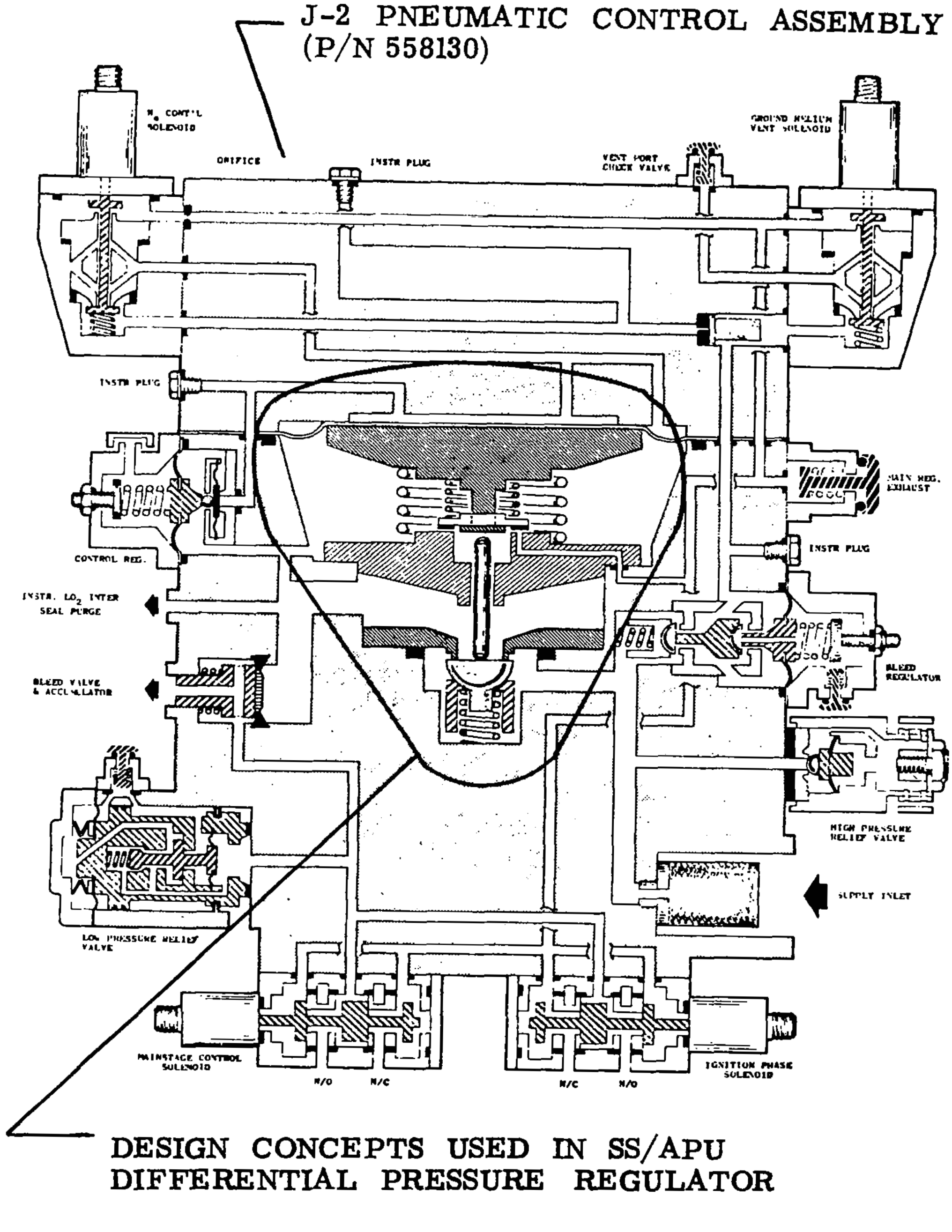


Figure 98. Proven Design Concept for Differential Pressure Regulator

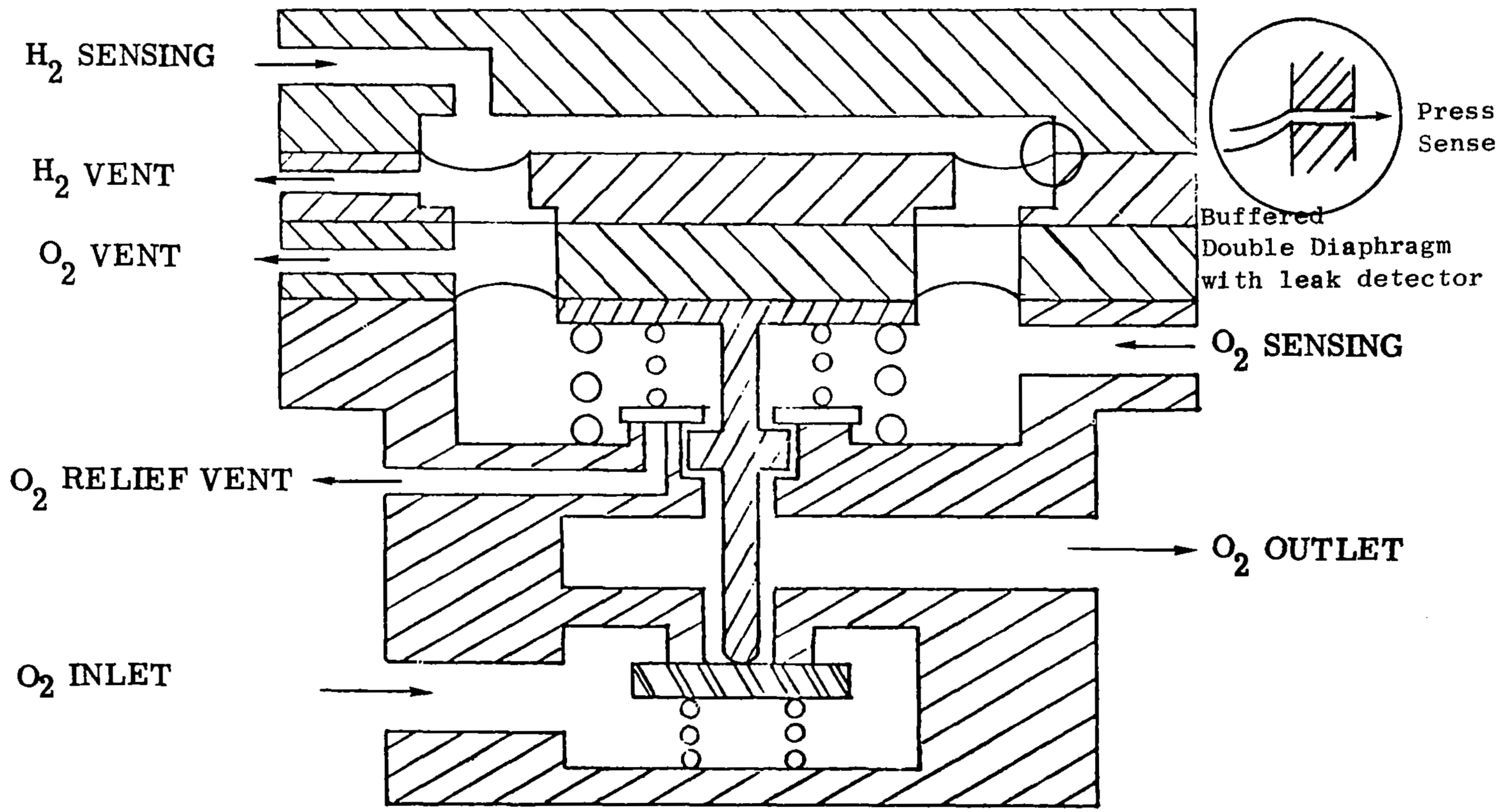
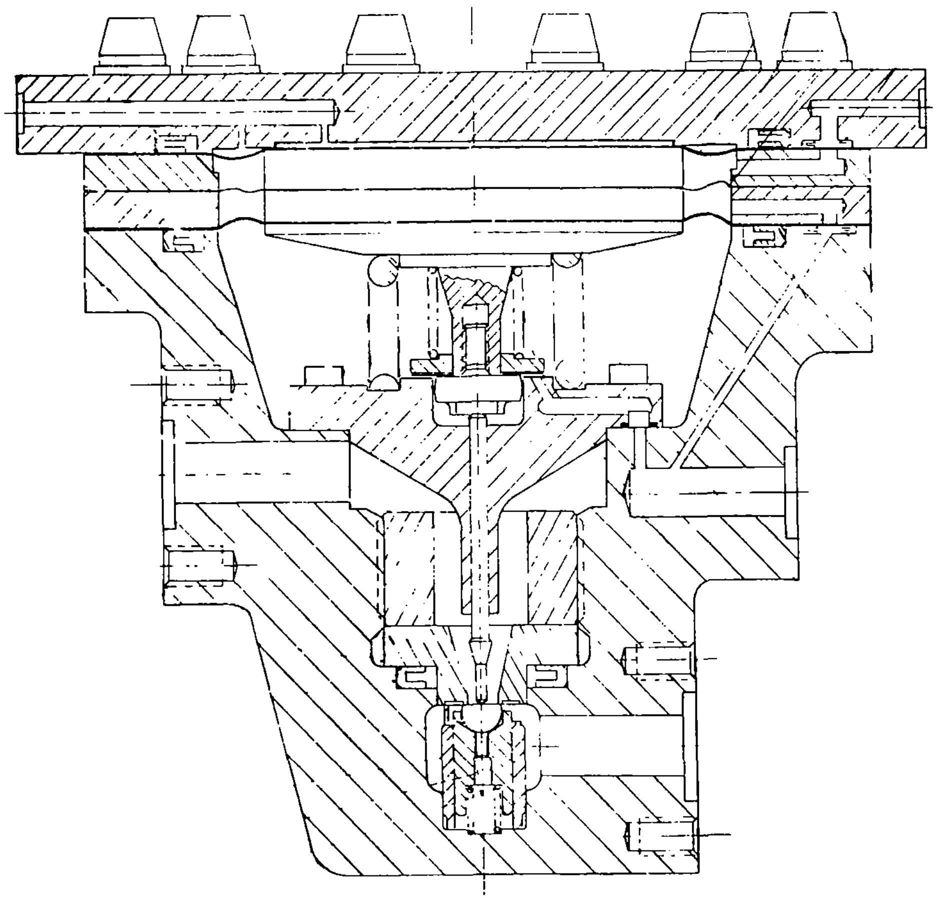


Figure 99. Differential Pressure Regulator Schematic



FEATURES

- DUAL METAL DIAPHRAGMS FOR PROPELLANT ISOLATION
- MEMBRANE SEPARATION OF ANY DIAPHRAGM OR STATIC SEAL LEAKAGE
- DASHPOT DAMPING ON HYDROGEN SIDE OF ACTUATOR

Figure 100. Differential Pressure Regulator and Relief Valve Design

C. HYDROGEN BYPASS VALVE

The gaseous hydrogen bypass valve design has two applications in the APU, each of which satisfies a different control requirement, as listed in Table 17. The first is associated with the hydrogen regenerator and the second is related to the hydraulic cooler control. These functions both require a servo position actuator device with a 3-way final control element for flow control. This device is very similar to the current device used on the J-2 engine for propellant utilization control. The propellant utilization valve is a 2-way valve which bypasses the J-2 liquid oxygen turbopump. However, the concept is easily adaptable to a 3-way configuration.

J-2 propellant utilization valve and the requirements that it currently satisfies are shown in Fig. 101 . The propellant utilization valve with its AC-driven actuator was selected primarily as a result of the philosophy of using existing qualified hardware. Additional investigation has indicated, however, that the current J-2 configuration could be modified as shown in Fig.102 to significantly reduce weight and add certain simplification by replacing the basic servo motor and gear-drive actuator with a DC torque motor. A rather wide variety of DC torque motors which are currently being used in space applications similar to the requirements for the Space Shuttle APU bypass valve.

Existing J-2 propellant utilization valve actuators could be used to advantage in the next phase of the Space Shuttle APU System. While the actuator for the basic configuration uses 400 cycle AC power the conversion from the DC supply to the 400 cycle AC can be easily provided in the electronic control assembly which is part of the APU system. Nevertheless further exploration of available hardware on the DC torque motor device for the bypass valve application is warranted.

D. POWER CONTROL VALVE (PULSE MODULATION)

D1. Control Valve Technology Considerations

Rocketdyne is currently involved in a variety of shuttle systems requiring propellant control valves. For example, the APS thrusters require low leakage valves with high cycle life. Similar conditions exist for the APU system (pulse modulated or continuously modulating) systems and for the shuttle propellant conditioner system. With respect to allowable leakage and cycle life requirements, virtually all combinations of low and high allowable leakage and low and high cycle life occur. These differences in requirements lead to somewhat different configurations in the power control valve area for each of the systems. Table 18 summarizes these combinations.

The critical design elements related to these requirements are shown in the central portion of Table 18 for each of the systems. A critical design element, means to imply elements that are the most influential in establishing the design configuration, and where important state-of-the-art considerations and trades must be made. When considering a low allowable leakage, the seat or closure seal becomes a critical design element and studies have shown that

TABLE 17

GH_2 BYPASS VALVE REQUIREMENTS
(TWO APPLICATIONS)

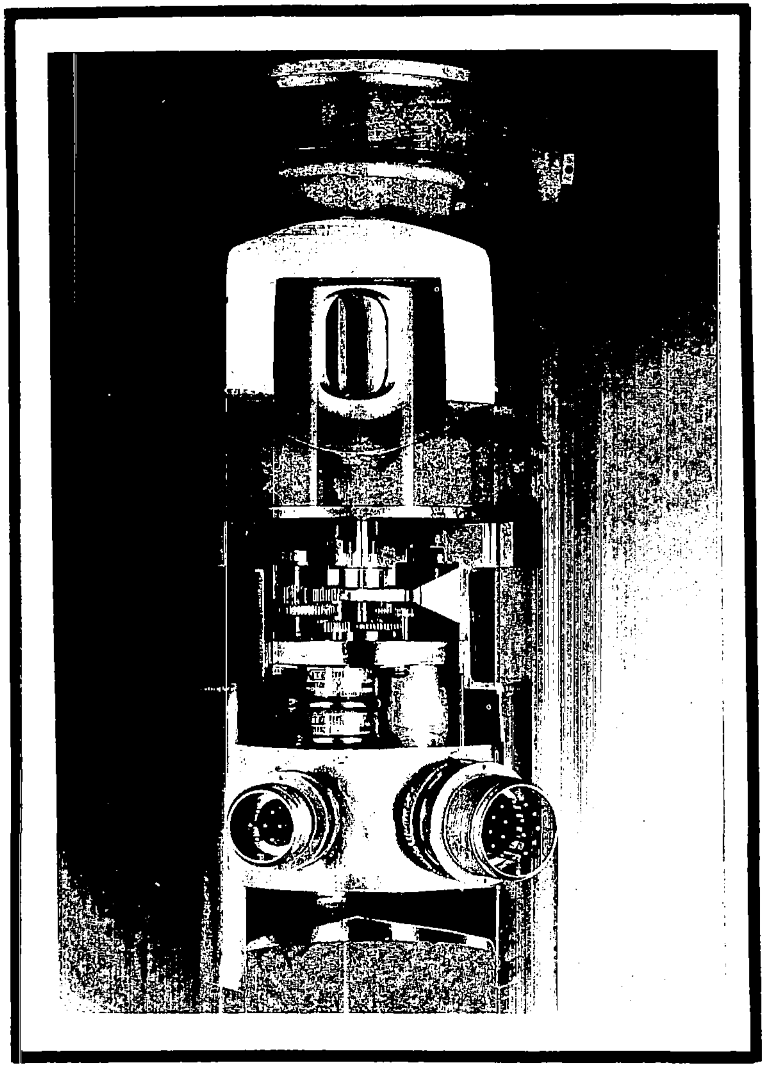
OPERATIONAL: THREE WAY HYDROGEN FLOW SPLITTER VALVE AROUND AND THROUGH (1) REGENERATOR, (2) HYDRAULIC COOLER IN CLOSED LOOP TEMPERATURE CONTROL OPERATION

VALVE CONDITIONS:

3/4 INCH LINE	DESIGN FOR MINIMUM PRESSURE DROP
460 PSIG INLET	LINEAR CONSTANT TOTAL FLOW AREA
-60 TO -385F FLUID	MINIMUM LEAKAGE CONSISTENT WITH NO SLIDING SEALS
-60 TO +130F AMBIENT	0 TO 0.126 LB/SEC FLOW

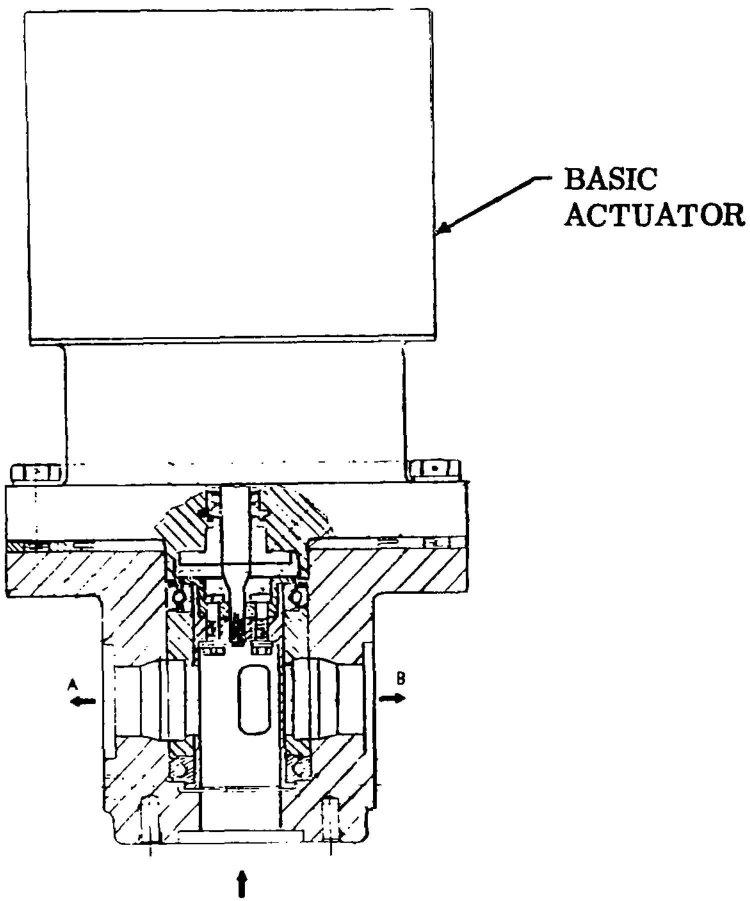
ACTUATOR:

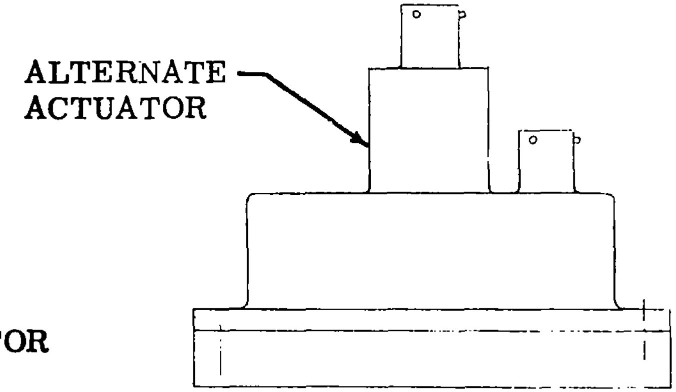
SERVO POSITIONED	ONE SECOND FULL STROKE



• FLUID:	LIQUID OXYGEN
• FLUID PRESSURE:	1100 PSIG
• OPERATING TEMPERATURE:	-298 TO +140 F
• TYPE:	PLUG, METERING
• ACTUATION:	MOTOR, 115 VAC
• LEAKAGE:	NOT A SHUTOFF VALVE
• RESPONSE:	5 CPS (1 SECOND FULL STROKE)
• APPLICATION:	MIXTURE RATIO CONTROL: 20,094 SECONDS AND 100 ENGINE RUNS MAXIMUM SINGLE EXPOSURE; OVER 400,000 SECONDS AND 4000 ENGINE RUNS OVERALL

Figure 101. 1-3/4 inch Propellant Utilization Valve





VALVE DESIGN FEATURES

- ALL METAL GATE AND HOUSING ASSEMBLY
- NO DYNAMIC GATE SEALS

ACTUATOR DESIGN FEATURES

BASIC:

- 400 CYCLE AC SERVO ACTUATOR QUALIFIED IN J-2 PROGRAM

ALTERNATE:

- DC TORQUE MOTOR LOW WEIGHT

Figure 102. GH_2 Bypass Valve Design

TABLE 18

POWER CONTROL VALVES
RELATED TECHNOLOGY CONSIDERATIONS

	SYSTEM			
	APS THRUSTERS	AUXILIARY POWER UNIT PULSE	AUXILIARY POWER UNIT MODULATING	PROPELLANT CONDITIONER
<u>**REQUIREMENTS**</u>				
ALLOWABLE LEAKAGE	**LOW**	**HIGH**	**HIGH**	**LOW**
CYCLE LIFE	**HIGH**	**HIGH**	**LOW**	**LOW**
	CRITICAL DESIGN ELEMENT			
ALLOWABLE LEAKAGE	**CLOSURE SEAL**	**NONE**	**NONE**	**CLOSURE SEAL**
CYCLE LIFE	**DYNAMIC SEALS**	**DYNAMIC SEALS**	**NONE**	**NONE**

poppet configuration will most probably satisfy this requirement. Similarly, when high cycle life is required, the dynamic seals become the critical design element.

A mechanically-linked poppet type bipropellant valve would most probably use a bellows to isolate the propellants and to partially balance poppet forces. Bellows life to-date has been demonstrated up to 10,000 cycles; however, this was for a specific application using a relatively long stroke bellows. Shorter stroke applications have successfully achieved 1 million cycles.

Sliding dynamic seals can take various forms where a low leakage dynamic seal is required as would be the case when attempting to isolate propellants. The seal can be expected to be rather sensitive to cycle life. On the other hand, certain valve configurations limited to individual propellant valves as opposed to bipropellant valves can use high leakage sliding seals to advantage.

In the case where the allowable leakage is high, a variety of configurations is acceptable. For example, in the pulse mode and modulating type auxiliary power unit systems, either a ball configuration or a poppet configuration could be used. Here functional simplicity rather than seat leakage is the criteria. The cycle life requirements in the pulse mode system are high which again poses the same question regarding dynamic seals. In the case of the ball configuration, a rotary sliding seal can be used to advantage, whereas in the poppet configuration the bellows dynamic seal is a candidate. A bellows seal configuration was selected in the case of the poppet valve because a linked bi-propellant valve is projected for use in the APU system for its power control valve.

D2 Control Valve Requirements

The specific requirements for the power control valve as used on the APU system as listed in Table 19 requirements include the allowable leakage of two-tenths of a percent of the maximum flowrate which significantly reduces the design problem and life problems on the seat and the three million on-off cycles required for the pulse mode operation. Also significant is the twenty millisecond maximum operating time, and the relatively low 10 psi maximum ΔP.

In selecting a configuration for the power control valve, all power sources were considered, as shown in Table 20 namely hydraulic, pneumatic, and electrical. Among the pneumatic power sources, both high pressure helium supply and the system propellant itself were considered. The hydrogen, or propellant, operated means of control was selected because it provided a low weight actuation system. It has capability for the required response and does not depend upon any external suppies as would be the case with the pneumatic, helium, hydraulic or electrical systems.

TABLE 19

POWER CONTROL VALVE REQUIREMENTS FOR PULSE MODE

OPERATIONAL: ON-OFF CONTROL OF HYDROGEN AND OXYGEN FLOW TO GAS GENERATOR AS PART OF CLOSED LOOP SYSTEM

VALVE CONDITIONS:

	GH_2	GO_2	BOTH SIDES
LINE SIZE	3/4 IN	3/8 IN	ON-OFF LINKED BIPROP VALVE
INLET PRESSURE	440 PSIG	436 PSIG	-60 TO +130F AMBIENT
FLUID TEMPERATURE	190F	170F	0.2 PERCENT ALLOWABLE LEAKAGE
FLOWRATE	0.125 LB/SEC	0.105 LB/SEC	NORMALLY CLOSED
			.02 SEC MAXIMUM OPERATING TIME
			3,000,000 ON-OFF CYCLES
			10 PSI MAXIMUM ΔP

TABLE 20

ACTUATION POWER SOURCES
FOR PULSE MODE POWER CONTROL VALVE

- POWER SOURCES CONSIDERED
  - HYDRAULIC
  - PNEUMATIC

    HELIUM

    PROPELLANT
  - ELECTRICAL
- HYDROGEN OPERATING FLUID SELECTED
  - LOWEST WEIGHT ACTUATION SYSTEM
  - ADEQUATE RESPONSE CAPABILITY
  - NO EXTERNAL SUPPLY REQUIRED

Two ways of arranging the power control valve actuator are available and shown in Fig 103 . The actuating fluid may be returned to the fluid system as shown or dumping the actuating fluid to the turbine exhaust or hydrogen vent. Dumping the fluid does not represent a significant performance disadvantage and somewhat simplifies the actuation system. To illustrate, note that the system which returns the fluid to the fluid stream requires a venturi section downstream of the hydrogen valve to provide a local low-pressure area to exhaust the actuation fluid. This arrangement is required to meet the dynamic response requirements. The venturi section is needed since the 10 psi maximum Δ P does not provide sufficient differential pressure to vent the actuating fluid and allow the actuator to move.

In this system, a ball type valve is virtually mandatory since a certain amount of the ΔP is lost across the venturi, recovery of the venturi being about 80 percent on a realistic basis. The balance of the differential pressure is sufficiently low such that a ball type valve with its inherent low ΔP is the only configuration which can meet the maximum ΔP requirements.

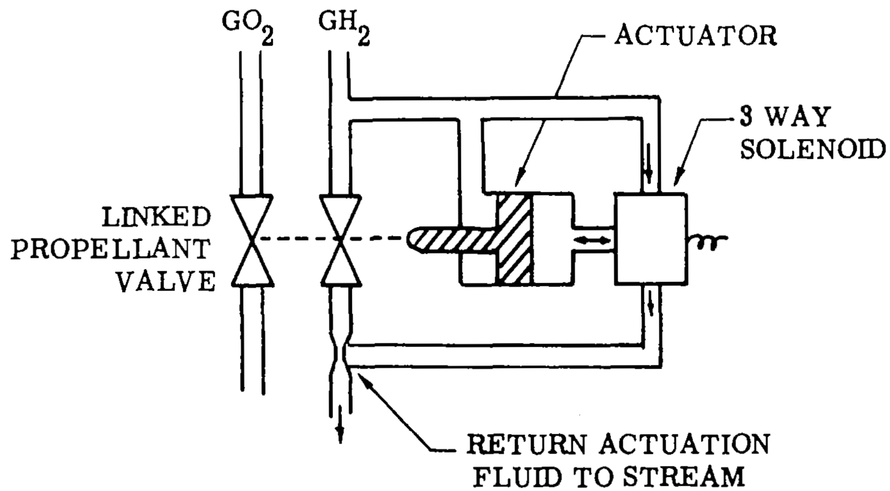
In the case where the fluid is dumped to the turbine exhaust, or to a hydrogen vent significantly greater ΔP is available and response is not a problem. Either a poppet or a ball configuration could be used. However, the allowable leakage requirement is such that a ball configuration can be used with a greater design margin on ΔP and somewhat simpler construction in that the valves are more easily linked to a common actuator. A candidate configuration is currently in production configuration used on our Atlas Vernier Systems. This valve controls the flow of liquid oxygen and RP fuel to the Atlas Vernier engines. The actual configuration as used in the Atlas Program is a 3-way type valve is shown in Fig. 104 . Figure 105 shows the one-half of the valve modified from a 3-way to a 2-way and also serves to illustrate a simple seat design which should be adequate to maintain the allowable leakage level over the required 3 million cycle life.

Experience with this seat design includes some work done as part of the APS Valve Technology Program. A Vespel 21 seat and a virgin teflon seat were cycled 1 million cycles in a modified Hydromatics ball valve. Initially, the leakage through the teflon seat was significantly lower than that of the Vespel 21. However, after abour 5,000 cycles the teflon seat deteriorated and the leakage increased to an uncontrolled level. Vespel 21 leakage, on the other hand, was initially higher than the teflon but the valve leakage throughout the 1 million cycle test period remained essentially constant.

Additional tests have been conducted with Vespel 21 on a larger ball valve configuration with an approximate 3 inch seat diameter. This seat was successfully cycled up to 50,000 cycles with the results again showing consistent leakage. Tests indicate that the Vespel 21 seat can be designed to provide a wide range of design leakage rates, and indicate that whatever the leakage initially, the seat quality is sufficiently constant over a large number of cycles.

RETURN TO FLUID STREAM

- NO FLUID DUMP
- ADDED ΔP REQUIRED FOR ACTUATION FORCE
- LARGER ACTUATOR



DUMP FLUID TO TURBINE EXHAUST

- HIGH FORCE MARGIN
- FAST RESPONSE
- ACCEPTABLE ACTUATION FLUID CONSUMPTION
- GREATER DESIGN MARGIN

SELECTED METHOD

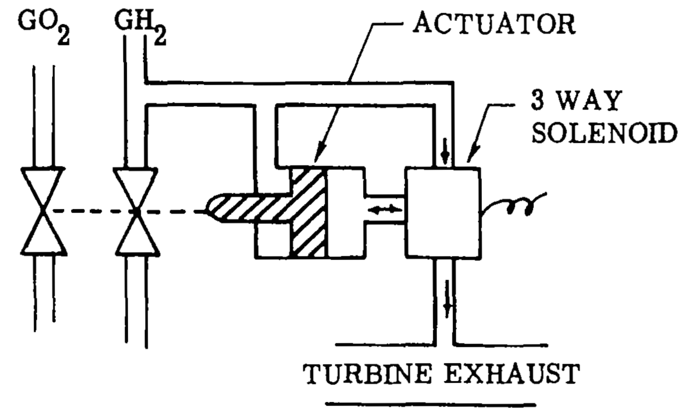


Figure 103. Methods of Actuation Fluid Disposal for Pulse Mode Power Control Valve

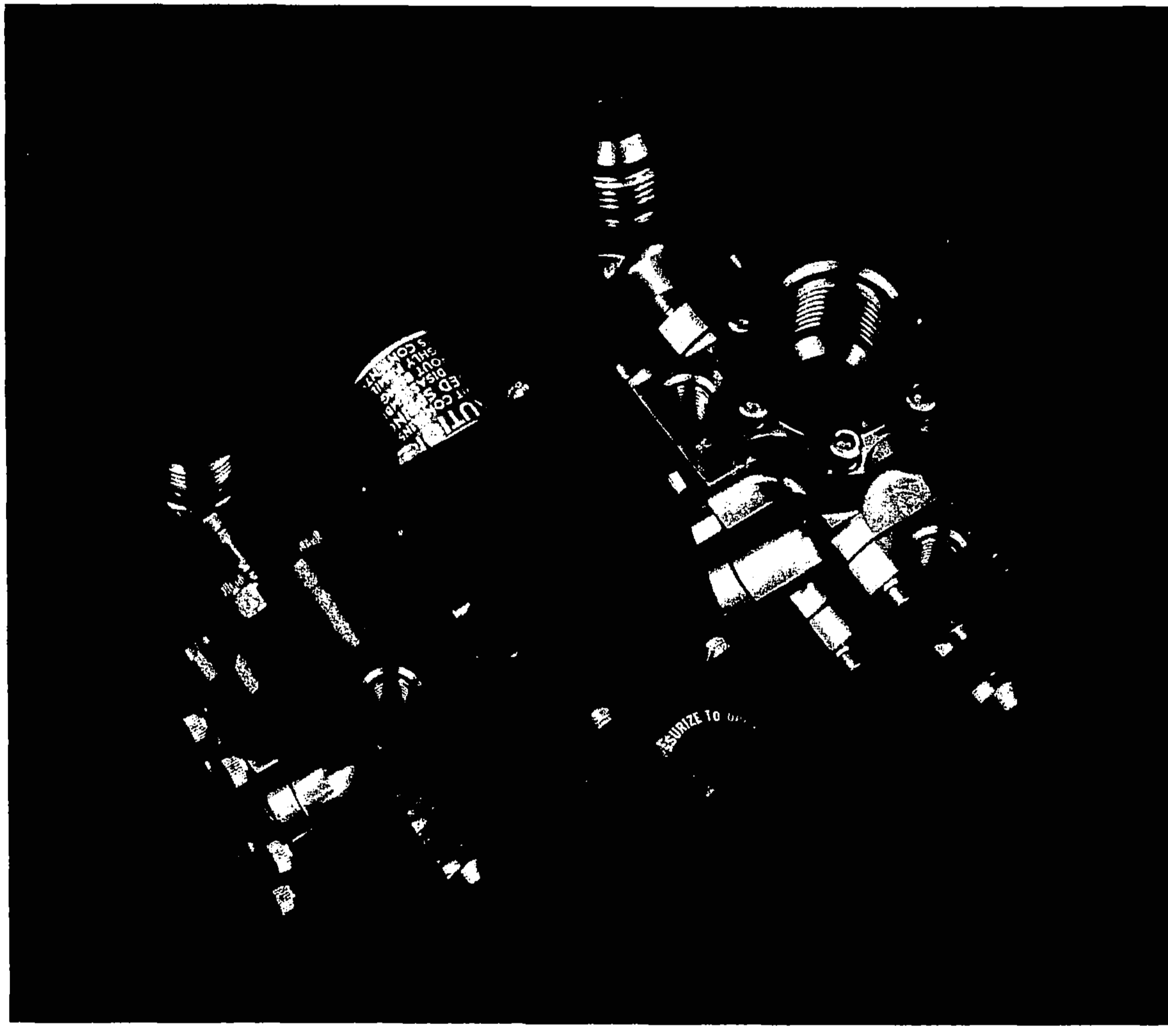


Figure 104. Hydromatics Bipropellant Valve

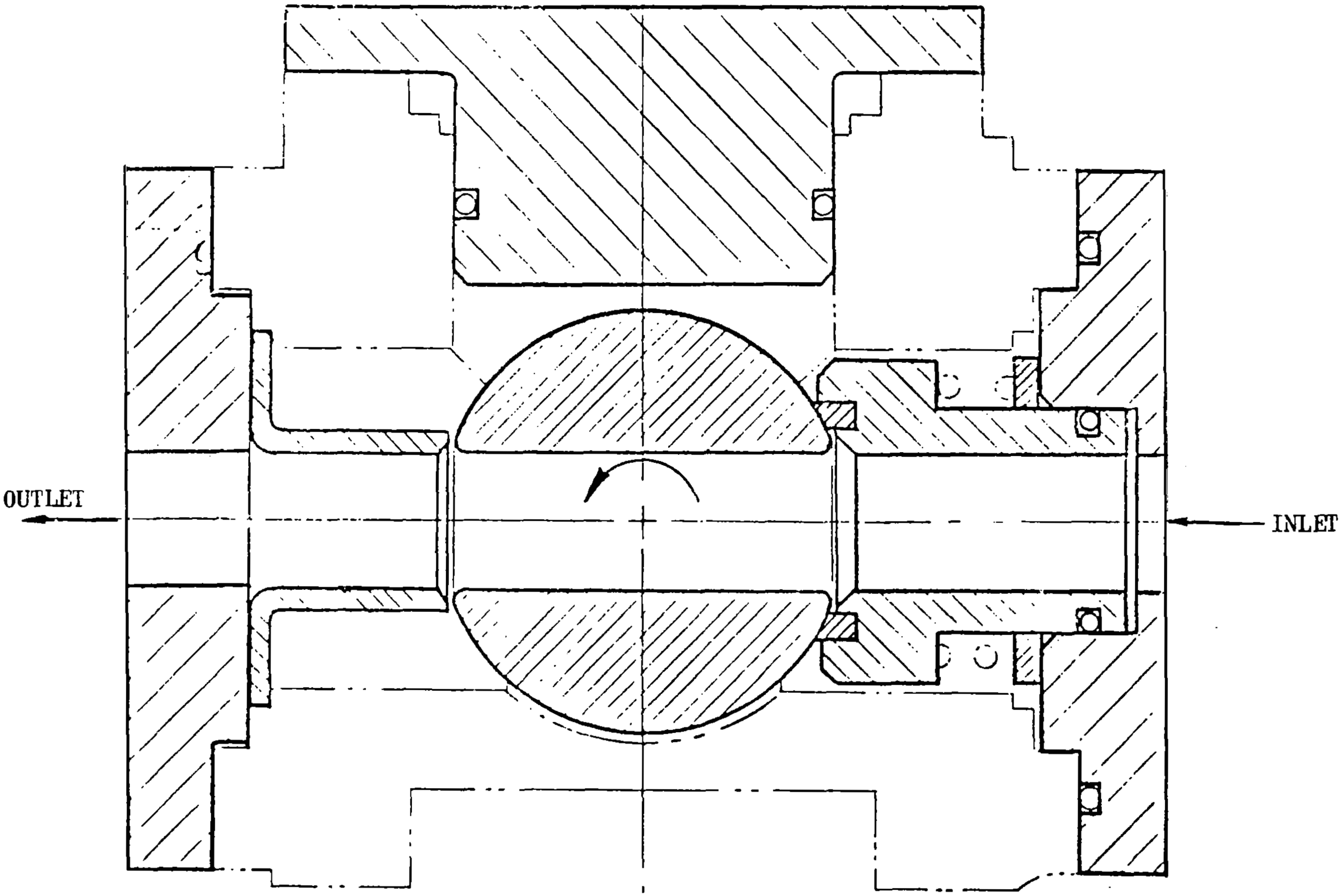


Figure 105. Power Control Valve - Modification of Hydromatics Ball and Housing Configuration

An alternate bipropellant configuration is shown in Fig. 106 utilizing bellows isolated, flat seated poppets. Bellows, in addition to providing a hermetic isolation of the oxygen and hydrogen system also provide a means for pressure balancing the valve seats against inlet pressure and thereby reducing the valve actuation forces.

This configuration is similar to the bipropellant poppet configuration considered for the APS System. Some of the more specific features of this design are that the valve is normally closed with a bias spring in the actuator, closing both mechanically-linked poppets. The bellows balances the poppet forces which reduces the required actuation forces. Poppets and seats are metal-to-metal and the seats are flexure mounted for better alignment of the poppets with their respective seats. In the closed position only the seat seals themselves are exposed to the propellant thereby minimizing the number of leakage paths through dynamic seals or closures.

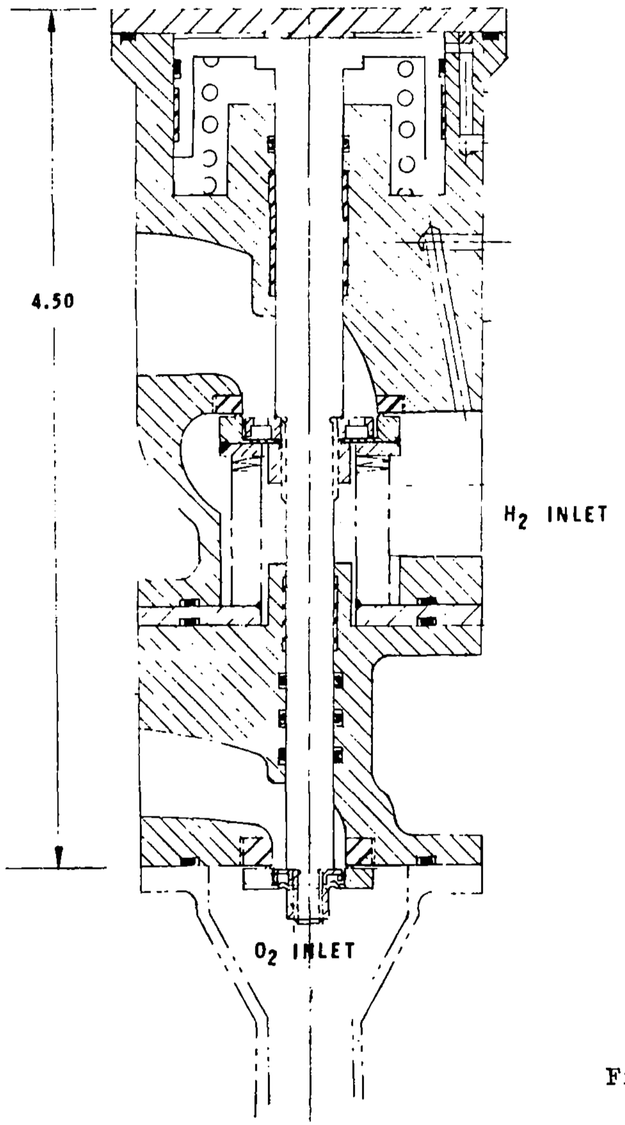
E. POWER CONTROL VALVE (PRESSURE MODULATED)

The throttle valve concepts presented in the discussion of the pulse mode power control valve may also be used for the modulating control application. As shown in Table 21 the electrical method of actuation is preferable for a continuously modulating power control valve. The actuator would be similar to the actuators considered for the GH_2 bypass valve. Hydraulic actuation is not as attractive in that an external supply is required for start which may or may not be available. A pneumatic system utilizing either high pressure gas or the propellant system has certain disadvantages. There is a fairly high gas usage rate associated with a pneumatic servovalve which would be required for positioning the power control valve. If this source of pneumatic pressure came from a stored helium system, additional weight would be required to provide such a system to the APU system. The propellant pressure itself, namely the hydrogen pressure, is not sufficiently high to operate such a servovalve system.

F. CONTROL ELEMENT STATUS

Tables 22, 23 and 24 summarize the primary and auxiliary control element status, with respect to similar design status and certain pertinent remarks.

As has been previously discussed the gaseous pressure regulator and relief valve is a design very similar to an existing J-2 production design using the regulator portion of the J-2 control assemblies, but substituting metal diaphragms for the currently used mylar diaphragms. The differential pressure regulator and relief valve is also quite similar to the J-2 production regulator except that it uses only the main actuator portion of that assembly. The GH_2 bypass valve has an actuator section identical to a present AC operated actuator used in the J-2 production program though it would appear that a DC torque motor may prove to be a more suitable selection. The valve portion of the GH_2 bypass valve is a 3-way configuration similar to the presently used 2-way construction and the power control valve currently projected for use in the next phase of the program is a linked-bipropellant ball configuration with



FEATURES

- MECHANICALLY LINKED SPRING RETURN NORMALLY CLOSED POPPETS
- METAL TO METAL SEATS
- SEATS FLEXURE MOUNTED
- ONLY SEAT SEALS EXPOSED TO PROPELLANT WHEN CLOSED
- REDUNDANT SEALS BETWEEN PROPELLANTS

Figure 106. Power Control Valve - Alternate Design

TABLE 21

MODULATING POWER CONTROL
VALVE CONSIDERATIONS

<u>ACTUATION</u>

ELECTRICAL - PRACTICAL USING ACTUATOR CONCEPTS SIMILAR TO THOSE FOR GH_2 BYPASS VALVE

PREFERRED SELECTION

HYDRAULIC - EXTERNAL SUPPLY REQUIRED FOR START

PNEUMATIC - HIGH GAS USAGE; HYDROGEN ACTUATION NOT STIFF; EXTERNAL SUPPLY REQUIRED FOR HIGH PRESSURE HELIUM

<u>THROTTLE VALVE TYPE</u>

- PULSE MODE VALVE CONCEPTS MAY BE USED FOR MODULATING CONTROL

TABLE 22

SUMMARY - PRIMARY ELEMENT STATUS

COMPONENT	SIMILAR DESIGN	DESIGN STATUS	REMARKS
GH_2 PRESSURE REGULATOR AND RELIEF	558130	J-2 PRODUCTION	USE REGULATOR PORTION OF J-2 CONTROL ASSEMBLY WITH METAL DIAPHRAGM
DIFFERENTIAL PRESSURE REGULATOR AND RELIEF VALVE	558130	J-2 PRODUCTION	USE MAIN ACTUATOR PORTION OF J-2 CONTROL ASSEMBLY WITH DUAL METAL DIAPHRAGMS FOR PROPELLANT ISOLATION
GH_2 BYPASS VALVE			
ACTUATOR	NA5-26726	J-2 PRODUCTION	ELECTRIC SERVO MOTOR WITH GEAR REDUCTION
	OR TORQUE MOTOR	COMMERCIAL	DIRECT DRIVE
VALVE	251351	J-2 PRODUCTION	CONVERTING EXISTING 2-WAY DESIGN TO 3-WAY DESIGN (2 REQUIRED)
POWER CONTROL VALVE	NA5-26312	ATLAS PRODUCTION	MODIFY SEATS TO "SEMI-SEAL" FOR LONG LIFE
	OR APS VALVE	STUDY	TECHNOLOGY PROGRAM IN PROGRESS

TABLE 23

SUMMARY - AUXILIARY ELEMENT STATUS

COMPONENT	SIMILAR DESIGN	DESIGN STATUS	REMARKS
EMERGENCY SHUTOFF VALVE	SAME AS POWER CONTROL VALVE	SAME AS POWER CONTROL VALVE	MODIFY SEATS FOR SEALING ABILITY; LOW CYCLE LIFE
ATTENUATOR TANK VENT VALVES	558301	J-2 PRODUCTION	MAY REQUIRE BIFILAR WINDINGS AND NEW CONNECTOR FOR SPACE SHUTTLE COMMONALITY (2 REQUIRED)
OXYGEN TANK SHUTOFF VALVE	SAME AS H_2 TANK SHUTOFF	SAME AS H_2 TANK SHUTOFF	SAME AS H_2 VALVE WITH DIFFERENT LINE SIZE ADAPTER
HYDRAULIC FLUID COOLER GH_2 BYPASS VALVE	SAME AS GH_2 BYPASS VALVE	SAME AS GH_2 BYPASS VALVE	SAME VALVE IN TWO APPLICATIONS
HYDROGEN TANK SHUTOFF VALVE	R5003679L	DESIGNED FOR SSME	MODIFY 3-WAY DESIGN TO 2-WAY DESIGN

TABLE 24

SUMMARY - AUXILIARY ELEMENT STATUS COMPLETED

COMPONENT	SIMILAR DESIGN	DESIGN STATUS	REMARKS
FILTERS	NA5-260264-T1	UNITS QUALIFIED	MFG. BY WINTEC INC.
TURBINE SPEED SENSORS	RC7005	PROCUREMENT SPEC ISSUED	DESIGNED FOR SSME (2 REQUIRED)
TURBINE INLET GAS TEMPERATURE SENSORS	RC7004	PROCUREMENT SPEC ISSUED	DESIGNED FOR SSME (2 REQUIRED)
EQUALIZER GAS TEMPERATURE SENSOR	RC7002	PROCUREMENT SPEC ISSUED	DESIGNED FOR SSME
HYDRAULIC AND LUBE OIL TEMPERATURE SENSORS	RC7002	PROCUREMENT SPEC ISSUED	DESIGNED FOR SSME
SIGNAL PRESSURE BIAS VALVE	--	--	COMMERCIAL

Vespel 21 seat seals. The recommendation, of course, could be modified pending the results of the APS Valve Technology Program, currently in progress.

All of the auxiliary elements are either existing designs, qualified and in production, or existing designs that have been created specifically for the Space Shuttle application for units to be procured, based on established procurement specifications.

APPENDIX A

APU COOLING CAPABILITY

A study was made to evaluate the cooling capability of the APU using the hydrogen as the heat sink fluid. The maximum heat load which can be accommodated by the hydrogen is a function of the flowrate, source temperature and maximum continuous oil temperature assuming a 100 percent effective oil cooler. The booster and orbiter profiles were utilized to determine the GH_2 flow at each "slice" and the corresponding hydraulic and lube (alternator, gearbox, bearings, seals) heat loads. The GH_2 source temperature required for steady-state cooling and the effect of increasing source temperature on transient heating of the bulk hydraulic fluid was determined.

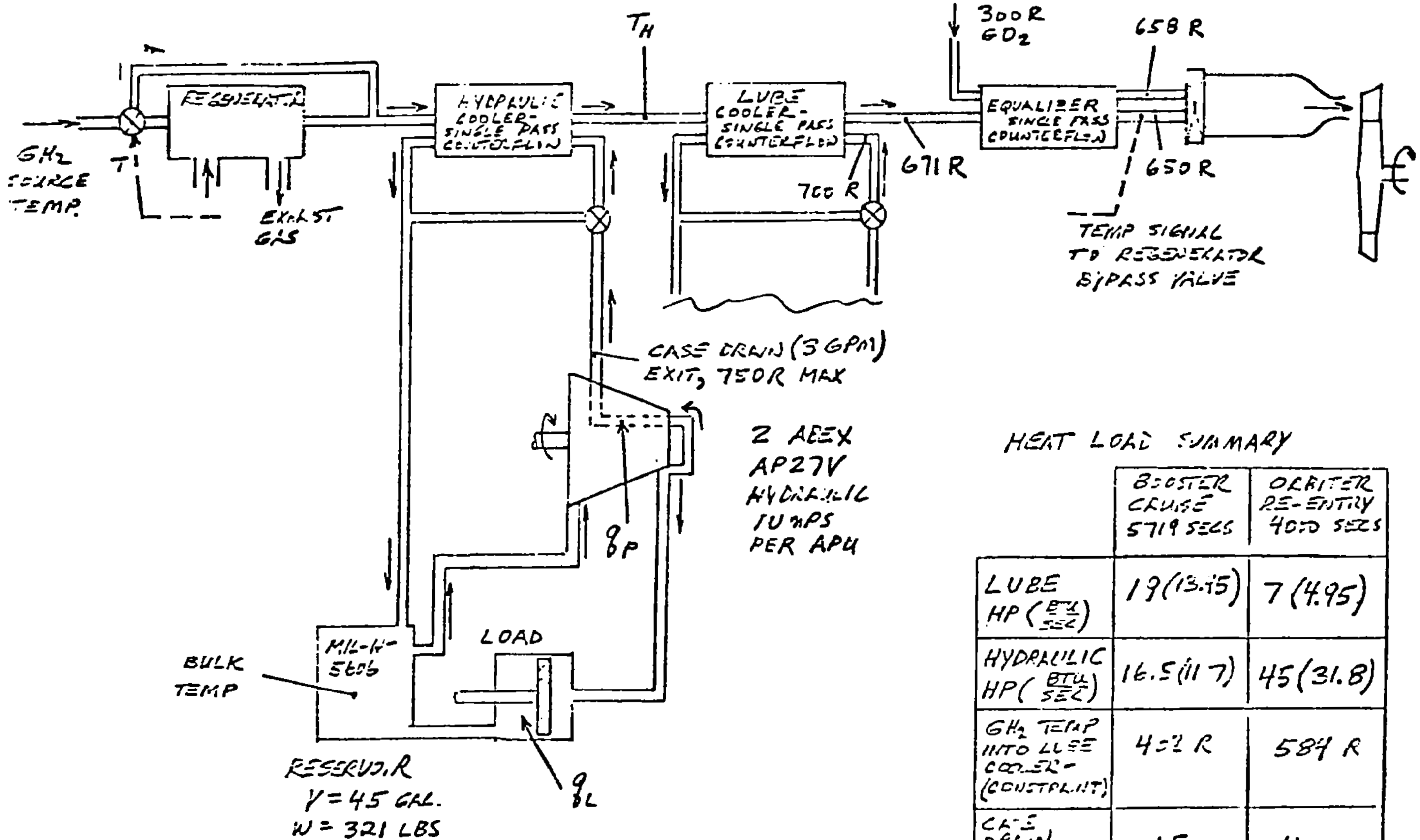
A schematic of the system is shown in Fig. 107, indicating use of a hydraulic cooler in the pump case drain line. Total case drain flow was assumed to be 6 gpm (3 gpm/pump) independent of hydraulic power level. The fluid (MIL-H-5606) total system inventory per APU is 321 pounds, assumed for both booster and orbiter. The lube oil (MIL-L-7808) inlet temperature to the cooler is 700 R and the GH_2 exit temperature is 671 R. The resulting oil cooler effectiveness is 88 percent and 75 percent during booster cruise and orbiter re-entry, respectively. These operating conditions represent the highest GH_2 enthalpy rise required for steady-state cooling and also the maximum operating duration within each profile. The heat loads for these two operating conditions are shown on Fig. 107. Results of the study are shown in Fig. 108 through 111. The effect of varying GH_2 source temperature and initial hydraulic fluid bulk temperature on the allowable operating period such that the case drain exit temperature equals 750 R is shown in Fig. 108 and 109. The GH_2 source temperature limiting the case drain exit to 750 R at the end of booster cruise (5719 secs) and orbiter re-entry (4000 secs) is shown in Fig. 110 and 111 as a function of initial hydraulic fluid temperature. Initial case drain exit temperature is also shown. Assuming the booster hydraulic fluid at launch is 540 R, a temperature rise of about 20 F prior to the start of cruise results, so that a maximum GH_2 source temperature of 260 R (Fig. 110) is required to prevent the case drain exit from going over temperature. With 540 R initial temperature at the start of orbiter re-entry, 125 R GH_2 (Fig. 111) is required to prevent case drain over temperature.

CONCLUSIONS

1. Integration of the APU with the APS system on the booster should not result in hydraulic system over temperature so that a Type II system (750 R max) could be retained.

2. Use of an integrated APU on the orbiter would require a higher temperature rated hydraulic system and/or a significant increase in maximum continuous lube oil temperature since the 125 R GH_2 is well below the 200 R to 300 R APS system nominal temperature. Increasing the maximum oil temperature by about 125 F (825 R) would make integration feasible still retaining a Type II hydraulic system.

3. Steady state cooling can be provided by 63 R GH_2 and 186 R GH_2 on the orbiter and booster, respectively. This necessitates use of a pump fed system for the orbiter and either a pump fed system or an integrated LH_2 (small APU supercritical GH_2 tank fed periodically by APS pump) system for the booster.



HEAT LOAD SUMMARY

	BOOSTER CRUISE 5719 SECS	ORBITER RE-ENTRY 4000 SECS
LUBE HP (BTU/SEC)	19(13.45)	7(4.95)
HYDRAULIC HP (BTU/SEC)	16.5(11.7)	45(31.8)
GH2 TEMP INTO LUBE COOLER - (CONSTRAINT)	402 R	584 R
CASE DRAIN HEAT LOAD HP	15	40
SYSTEM HEAT LOAD HP	1.5	5

Figure 107. Hydraulic Cooling System

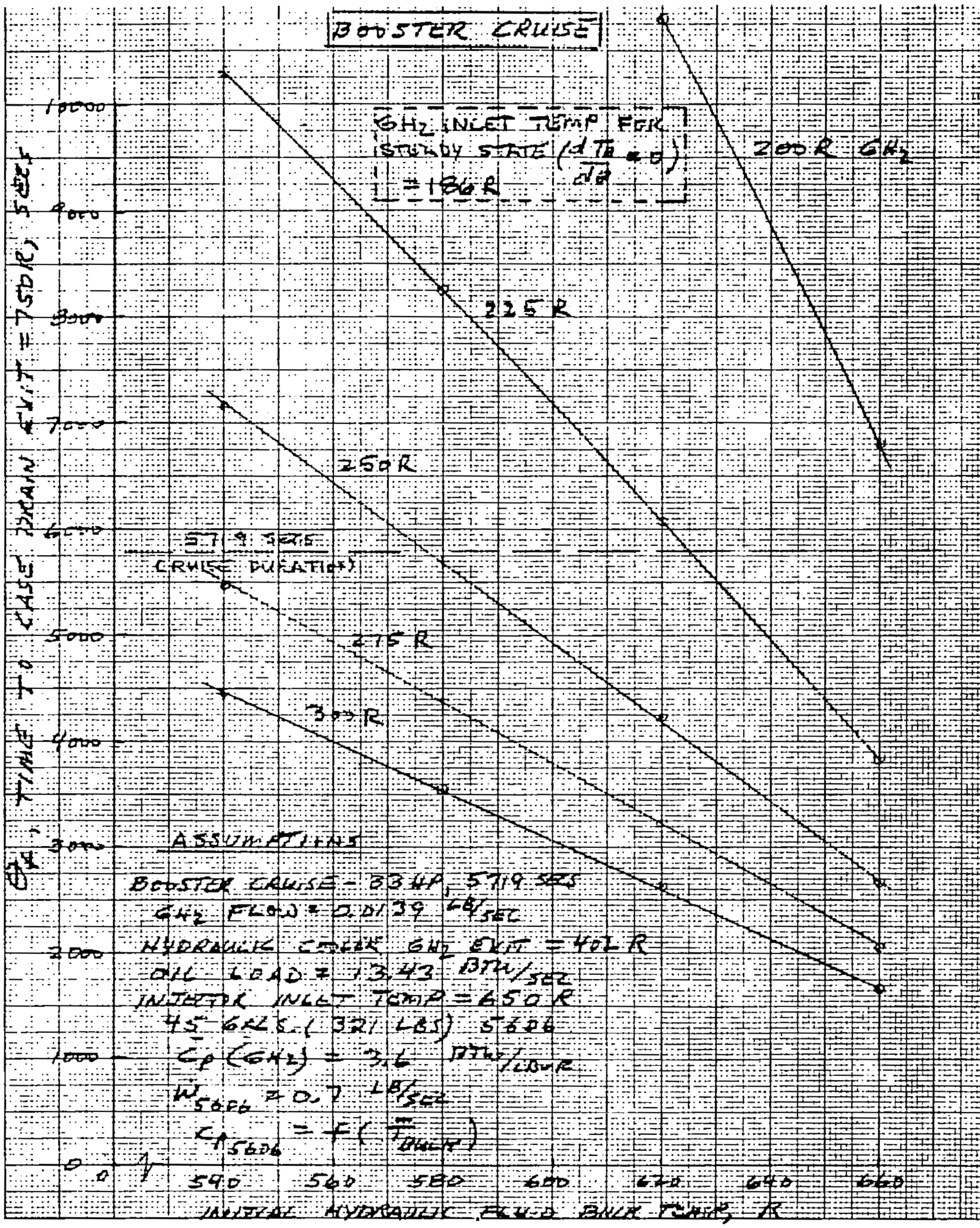


Figure 108. Effect of Hydrogen Source Temperature and MIL-H-5606 Initial Bulk Temperature on Transient Heating Duration Booster Cruise

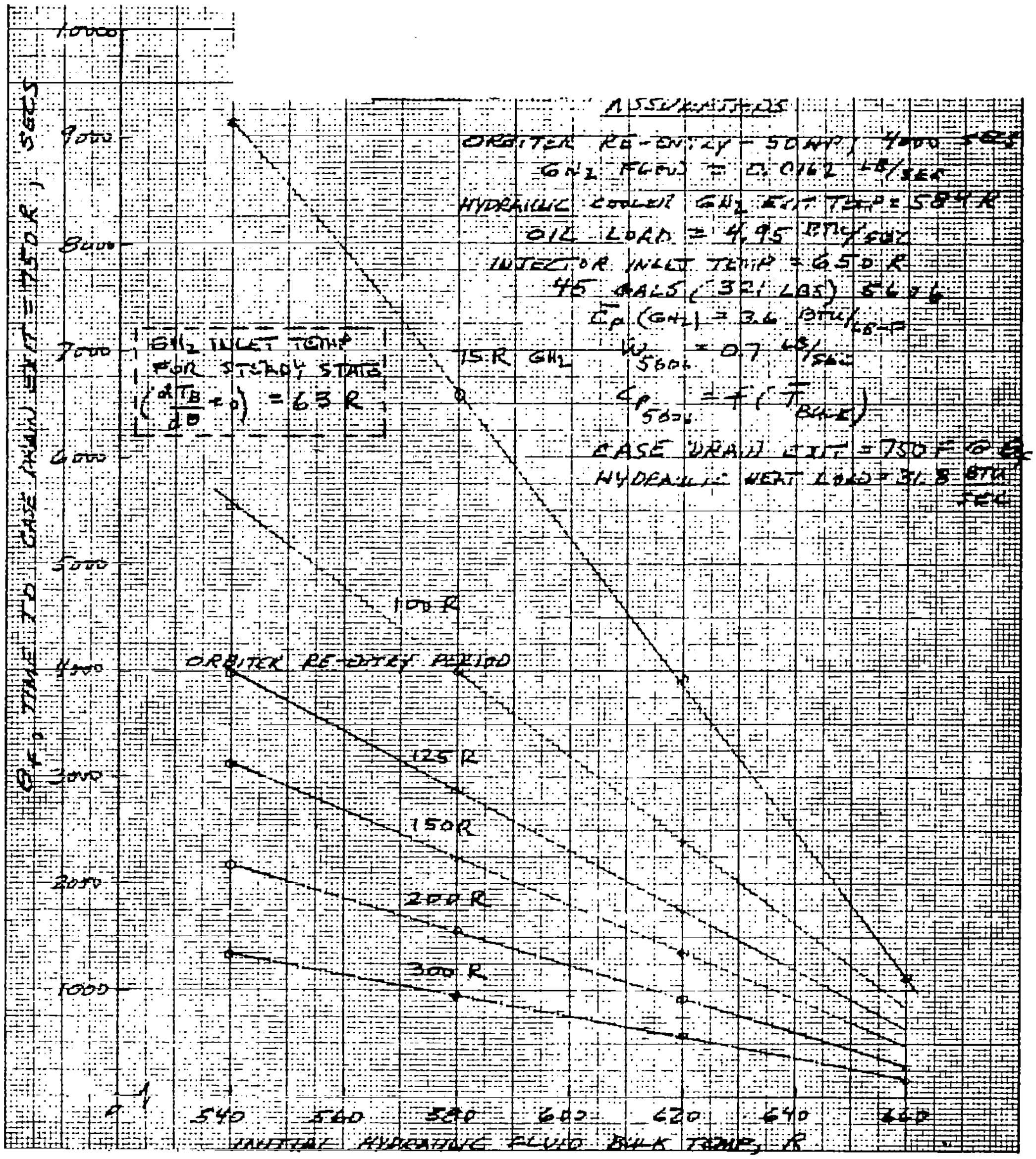


Figure 109. Effect of Hydrogen Source Temperature and MIL-H-5606 Initial Bulk Temperature on Transient Heating Duration Orbiter Re-Entry

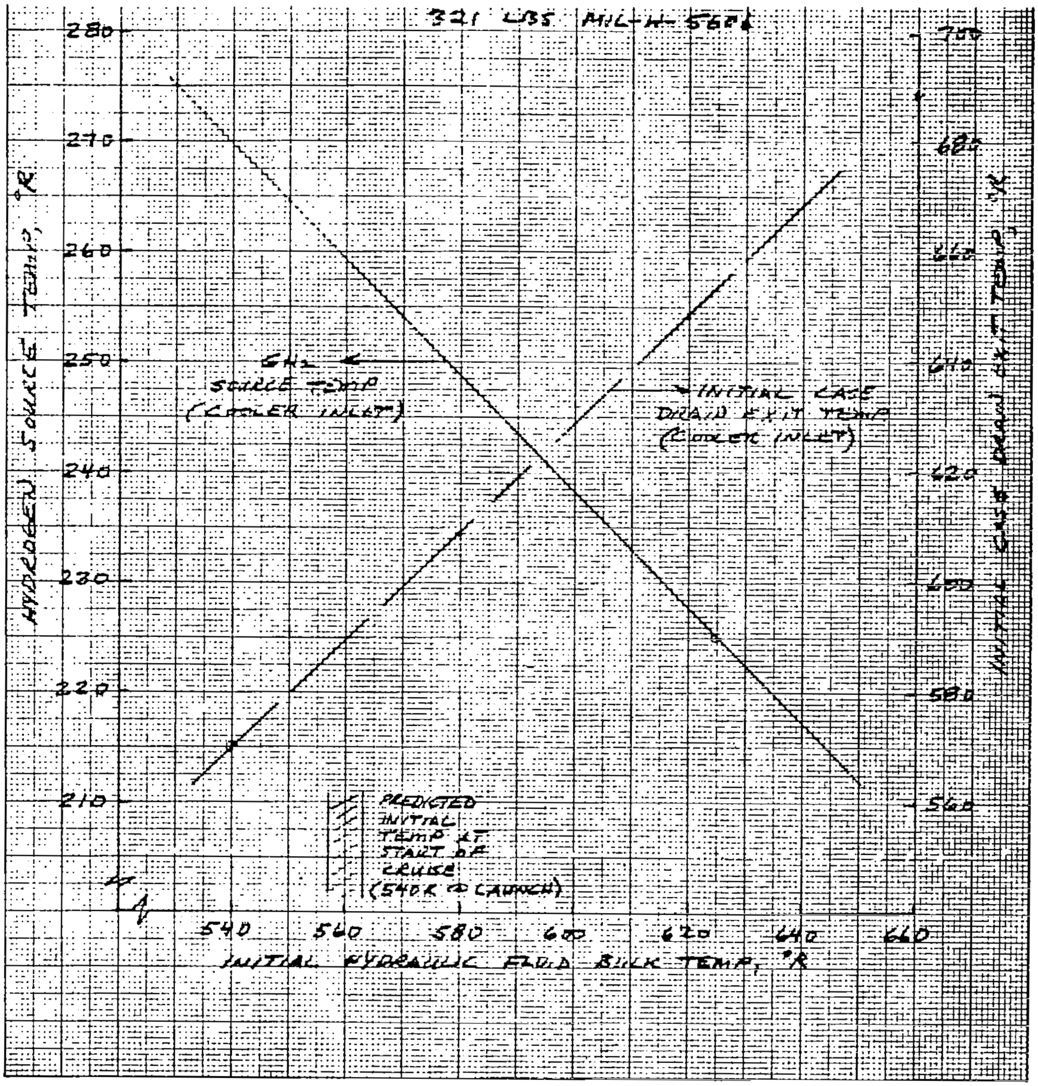


Figure 110. **Hydrogen Source Temperature Required to Limit Case Drain Exit To 750^{0}R at End of Booster Cruise**

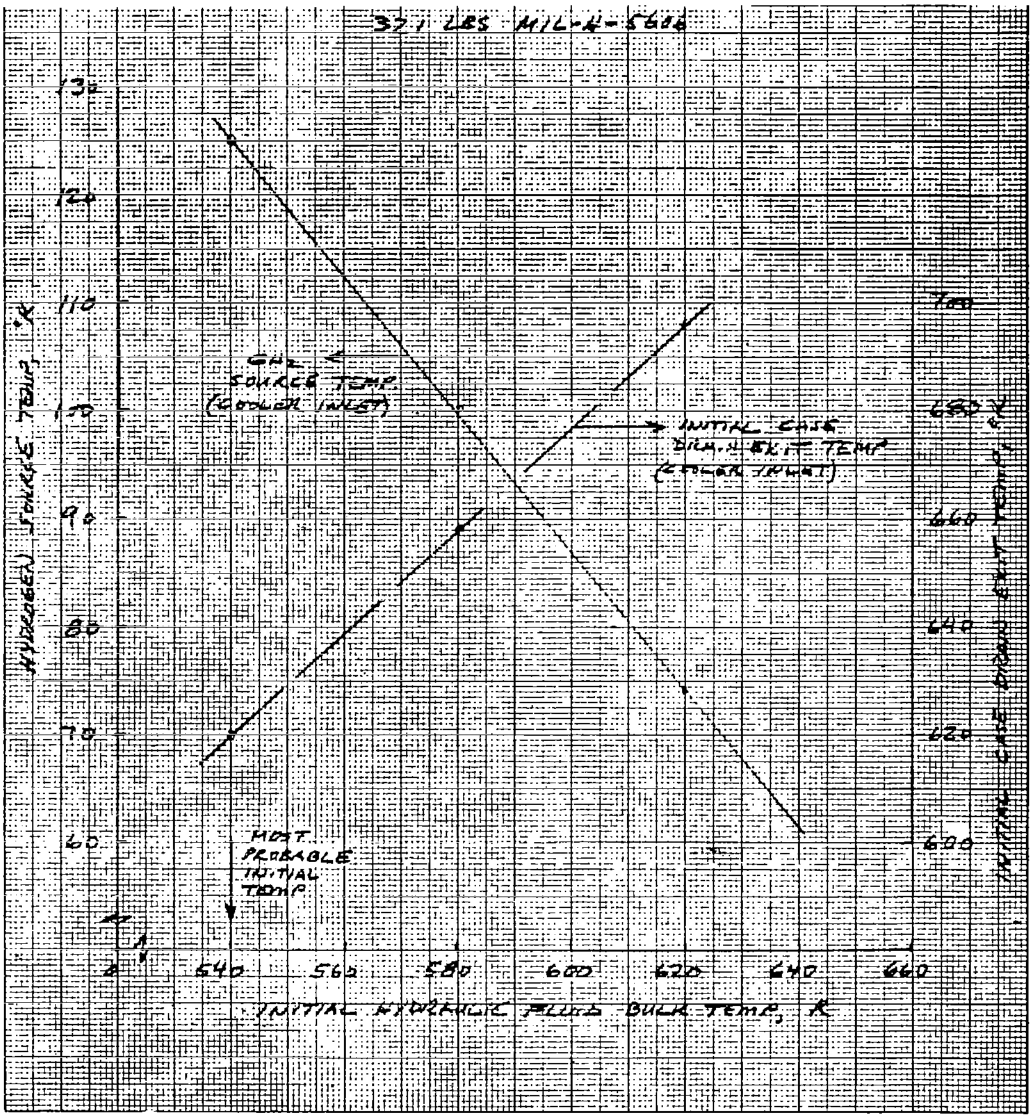


Figure 111. Hydrogen Source Temperature Required To Limit Case Drain Exit to 750°R at End of Orbiter Re-Entry

APPENDIX B

APU POWER-TIME PROFILE FOR ORBITER AND BOOSTER VEHICLES

The power-time profiles, furnished by NASA, formed the basis for Rocketdyne's digital APU performance program. These profiles, for both the booster and orbiter vehicles, are shown in Fig. 112 and 113.

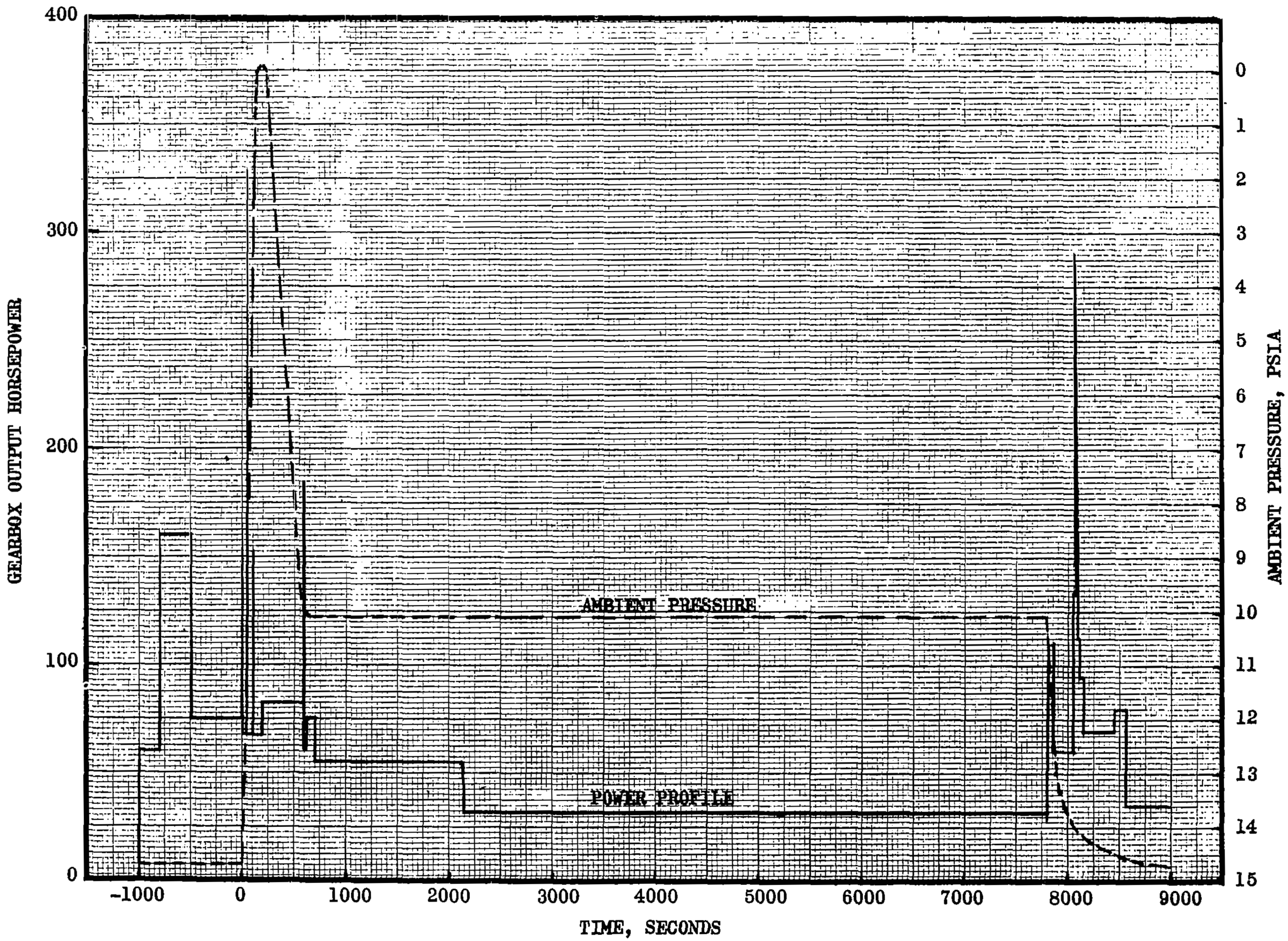


Figure 112. Booster Power Profile--Phase II

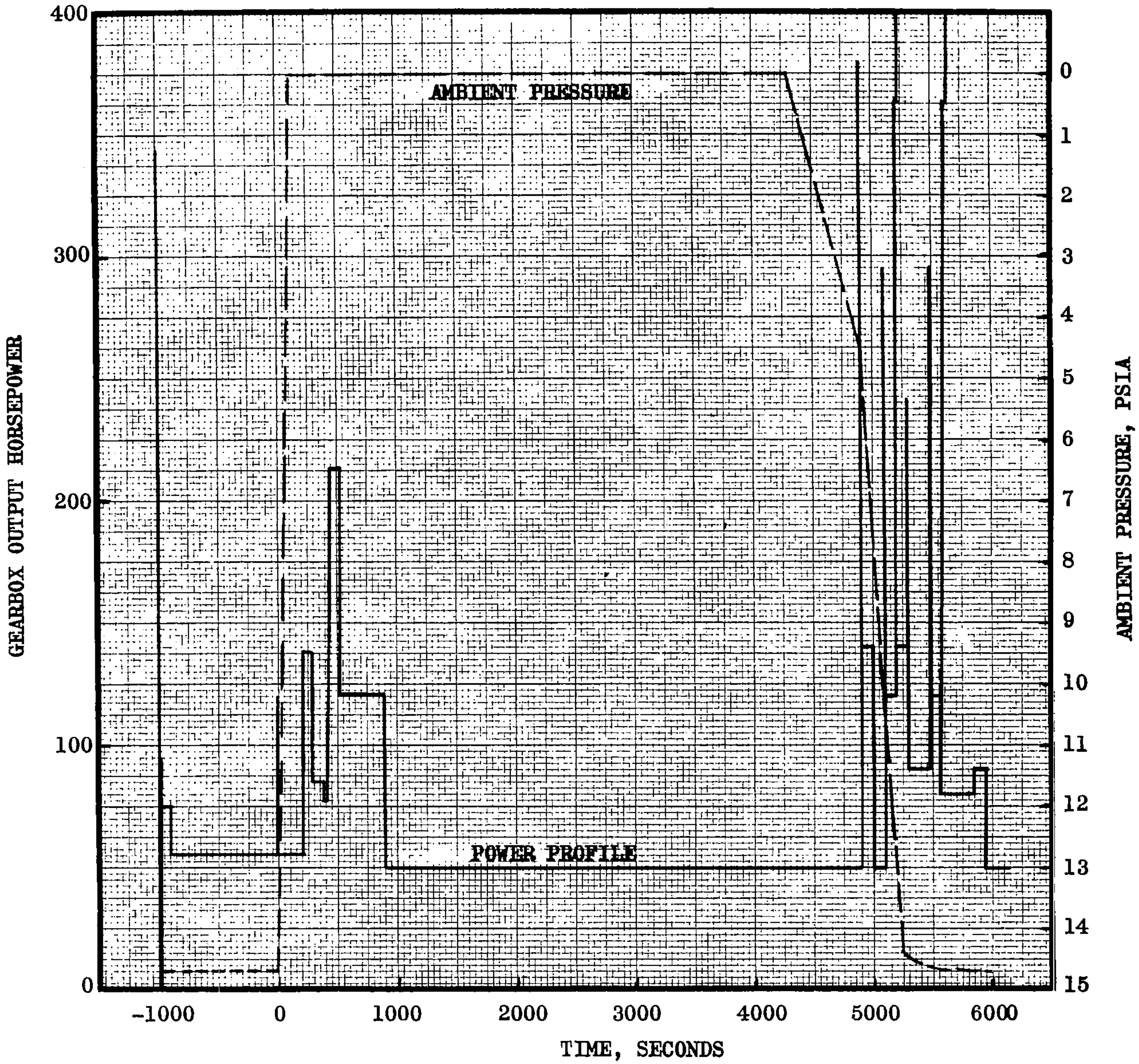


Figure 113. Orbiter Power Profile--Phase II

APPENDIX C

HYBRID POWER CONTROL APU SYSTEMS

Early in the Phase II program, prior to the selection of a baseline design, an analytical study was conducted of hybrid APU systems. A hybrid system utilizes two combustors: a continuously operated, pressure-modulated sustainer combustor and either a pulse- or pressure-modulated peaking combustor. The sustainer combustor is capable of meeting power demands up to 30 percent power level. The peaking combustor is ignited to meet higher power demands. The objectives of a hybrid system are improved system S.P.C. and elimination of a deep-throttling (10:1 flow range) combustor. A description of the hybrid system operational characteristics, design estimates of a hybrid, two-stage turbine, and a performance evaluation are presented in the Sixth Monthly Technical Progress Narrative, which is enclosed herein. It was concluded that while the hybrid system could functionally achieve its objectives, the physical designs of both the turbopower unit and control system were excessively complex.

NAS3-14407

SIXTH MONTHLY TECHNICAL PROGRESS NARRATIVE

FEBRUARY 1971

INTRODUCTION

The program objective is to provide analysis and design information for gaseous oxygen/hydrogen auxiliary power units for the Space Shuttle booster and orbiter elements, with consideration for recoverable, fully reusable designs. Input of this information into the APU tradeoff studies and configuration decisions will ensure that APU specifications are realistic and achievable, and will provide a basis for space shuttle APU development programs.

The program consists of two phases. Phase I encompassed component screening/system synthesis and evaluation with the objective of selecting a system with the most potential for meeting the requirements. Phase II allows an in depth design/analysis of the chosen system.

DESCRIPTION OF PROGRESS

The overall Phase II program progress is displayed in Figure 1. All planned work has been completed on schedule.

The Hybrid System Evaluation has been completed. Design estimates of a hybrid, two-stage pressure and velocity compounded turbine have been completed. Control systems were synthesized for both a hybrid-pressure modulated and hybrid-pulse system and an operational analysis was performed through use of analog models. A system performance analysis was completed to compare propellant related weights of the hybrid, pressure modulated and pulse systems using the Phase II reference flight profile.

The hybrid system evaluation effort, which was initiated to explore the potential system weight advantage over a pressure modulated system, has been completed. The evaluation consisted of three areas of activity:

1. Synthesis of control systems and an operational analysis of both the hybrid-pressure modulated and hybrid-pulse power control system through use of the analog model.

2. Design estimates of a hybrid, two stage, pressure and velocity compounded turbine.

3. Performance analysis to compare propellant related weights of the hybrid, pressure modulated and pulse power control systems.

CONTROL SYSTEM SYNTHESIS AND OPERATIONAL ANALYSIS

Hybrid Pulse Peaking Combustor

This control utilizes a pressure modulated sustainer combustor and pulse mode peaking combustor. A block-logic diagram of the control system is shown in Figs. 2a and 2b, and a turbine and gear box load torque-speed relationship is described in Fig. 3. Consider operation of the sustaining combustor at point P_1, (Fig. 3) holding design speed with the bipropellant valve modulating at approximately 60 percent open. As the gear box load is stepped up to 100 percent power, the sustaining combustor bipropellant valve opens wide as speed drops (P_1-P_2). Speed continues to drop with the valves wide open (P_2-P_3), until the minimum speed limit is reached (98 percent). At this point, speed control switches over to the peaking combustor bipropellant valves, which are signalled open (P_3-P_4) and the sustaining combustor valves are positioned at a fixed value of 100 percent. Since the peaking combustor is sized to provide 100 percent of maximum power, the combined power of the sustainer plus peaking combustor is approximately 140 percent. The TPU starts accelerating after the peaking valves open, and speed increases to the upper speed limit of 102 percent (P_4-P_5) at which point the valves are signalled closed, and total power is provided only by the sustainer combustor.

With reference to the logic diagram of Fig. 2a, when gear box load demand is below 30 percent, speed control operates through the sustainer combustor bipropellant valves, i.e., switch #1 to position "0". The peaking combustor valves are closed, i.e., switch #2 at "0". When load demand exceeds 30 percent, the sustainer combustor can no longer maintain design speed, and speed drops to the lower speed band of 98 percent, control then switches over to the peaking combustor (switch #2 pulses the valve open and closed to hold speed between ± 2 percent). The sustainer combustor bipropellant valves are placed on position control (switch #1 to position "1") with the valves set at 100 percent.

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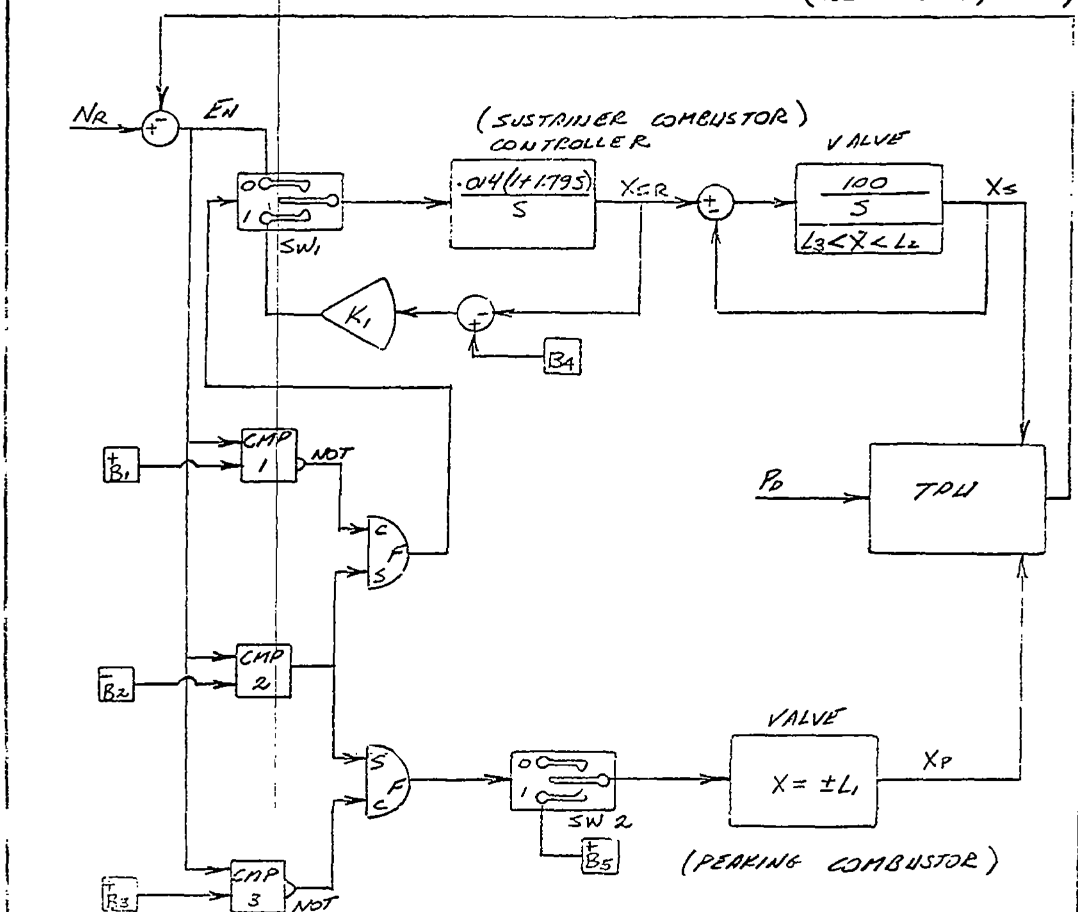
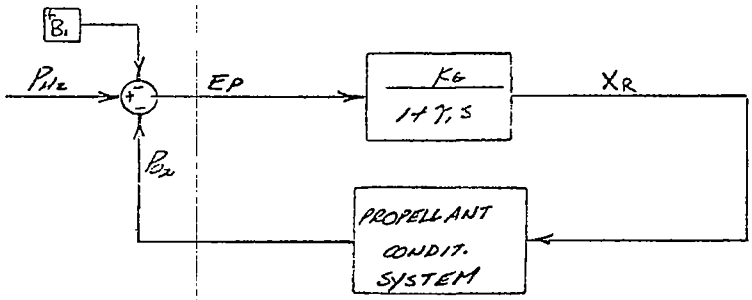
SPEED CONTROL (RUNS 232, 233)



LIST OF VARIABLES	NOMENCLATURE	
B_1 = 2400 RPM	N_R 60,000 RPM	X VALVE VELOCITY, %/SEC
B_2 = 1200 RPM	N SPEED, RPM	CMP COMPARATOR OUTPUT
B_3 = 1200 RPM	E_N SPEED ERROR, RPM	0 FOR NEG. INPUT
B_4 = 100 %	X_S SUST COMB VALVE	1 FOR POS. INPUT
B_5 = SOLENOID ACT. VOLT.	POSIT, %	F FLIP FLOP OUTPUT:
L_1 = 4000 %/SEC	X_P PEAKING COMB VALVE	0 FOR LOGIC 1 TO C
(.025 SEC FULL TRAVEL)	POSIT, %	1 FOR LOGIC 1 TO S
L_2 = +1000 %/SEC		TPU TURBO POWER UNIT
L_3 = −200 %/SEC	X_{SR} VALVE POSIT. REF, %	P_D POWER DEMAND, H.P.
τ_N = .01 SEC	X_{PR} VALVE POSIT. REF, %	
K_1 = HIGH GAIN	SW SWITCH	

FIGURE 2a

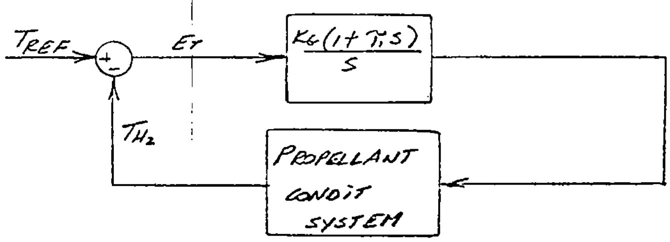
PRESSURE REGULATOR (RUNS 232-239)



H_2 ACCUMULATOR VOL. = $2\ FT^3$
O_2 ACCUMULATOR VOL. = $1/8\ FT^3$

LIST OF VARIABLES	NOMENCLATURE	
K_G = 4.35 %/PSI	P_{H_2}	TPU H_2 INLET PRESS., PSIA
τ_1 = 0.87 SEC	P_{O_2}	TPU O_2 INLET PRESS, PSIA
	X_R	REGULATOR VALVE POSIT, %
	E_P	PRESSURE ERROR, PSIA.

BY PASS CONTROL (RUNS 232-239)



LIST OF VARIABLES	NOMENCLATURE	
K_G = .356	T_{REF}	TPU H_2 INLET TEMP. REF., R
τ_1 = .0475	T_{H_2}	TPU H_2 INLET TEMP, R
	E_T	TEMP. ERROR, R
	X_{BP}	BY PASS VALVE POSIT., %

FIGURE 2b

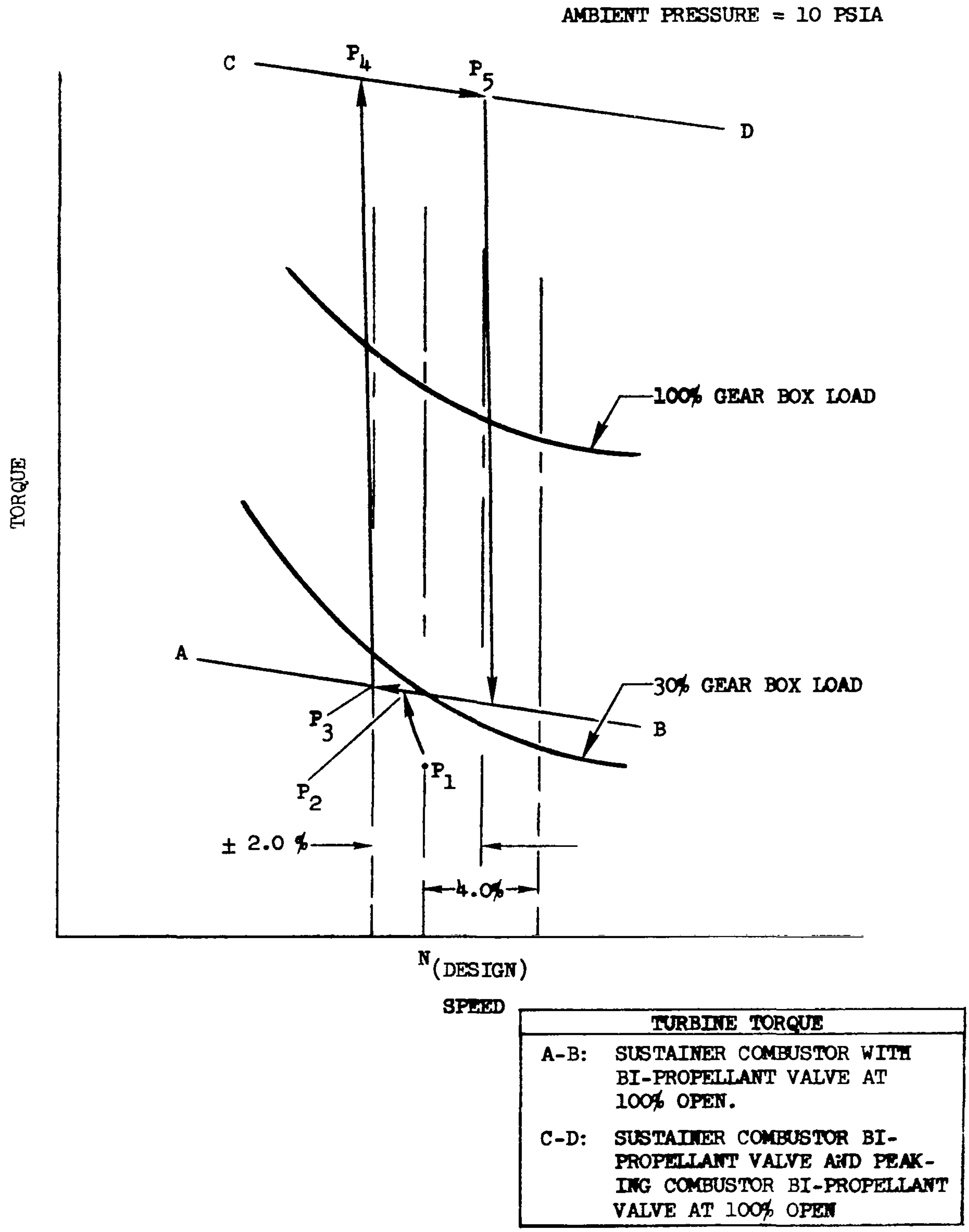


Figure 3. Hybrid Power Control (Pulse Peaking Combustor)

Hybrid-Pressure Modulated Peaking Combustor

This hybrid control utilizes a pressure modulated sustaining and peaking combustor. A block-logic diagram of the control system is shown in Figs. 4a and 4b and a turbine and gear box load torque-speed relationship is described in Fig. 5. Consider operation of the sustaining combustor at point P_1 (Fig. 5) holding design speed with the bipropellant valve modulating at approximately 60 percent open. As the gear box load is stepped up to 100 percent power, the sustaining combustor bipropellant valve opens wide as speed drops (P_1-P_2). Speed continues to drop with the valves wide open (P_2-P_3), until the minimum speed limit is reached (97 percent). At this point, speed control switches over to the peaking combustor bipropellant valves, which are driven open by the speed error signal, and the sustaining combustor valves are positioned at a fixed value of 70 percent. Speed continues to decrease until turbine power exceeds the 100 percent load demand; the valves then modulate turbine power to return the TPU to the design speed at point P_4.

It is not practical to design the peaking combustor bipropellant valves with a requirement for modulating below 2 to 3 percent, due to the difficulty of smooth combustion and maintaining proper mixture ratio. When the peaking combustor is "on active speed control," a 3.0 percent minimum valve position is therefore established. In order to prevent speed control from cycling between the two combustors when power demands just exceed the sustainer, the sustainer bipropellant valves are positioned to 70 percent, thereby forcing the peaking combustor valves to a position greater than its minimum of 3.0 percent.

With reference to the logic diagram of Fig. 4a, when gear box load demand is below approximately 30 percent, speed control operates through the sustainer combustor bipropellant valves i.e., switch #1 on "0", and the valves are closed. When load demand exceeds 30 percent, the sustainer combustor can no longer maintain design speed and speed drops to the lower speed band of 97 percent; control then switches over to the peaking combustor (switch #2 to position "1"). The sustainer valves are placed on position control (switch #1 to position "1") with the valves set at 70 percent.

TURBINE DESIGN ESTIMATE

Several factors have been studied to determine their effects on efficiency of the hybrid turbine. The effect of stage pressure ratio split is shown in Fig. 6. It will be seen that best sustainer flow path efficiency occurs when the first stage pressure ratio is approximately twice that of the second stage. This condition results, however, in a second stage relative mach number of only 1.24 compared to the first stage value of 1.66. The low second stage value presents a more severe design challenge than does the first stage. It is also much more severe than would be associated with a more equal pressure split.

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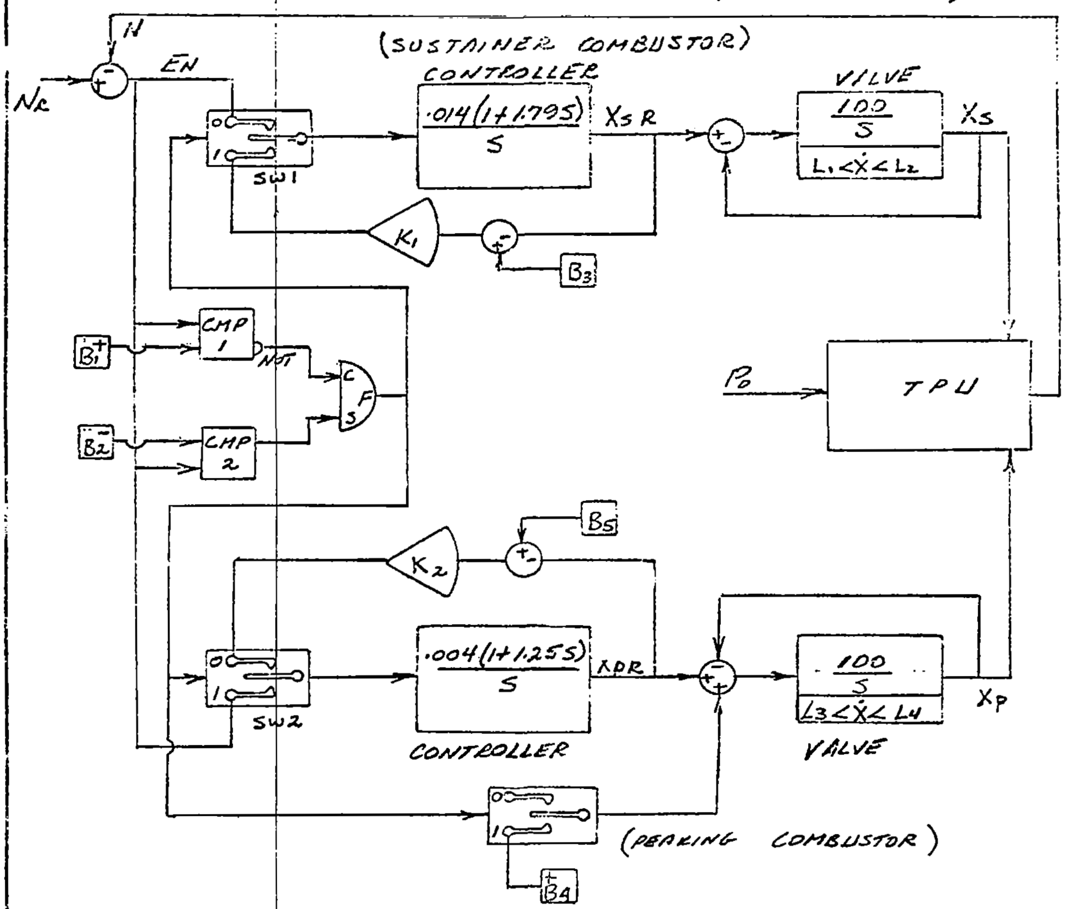
DATE:	PEAKING COMBUSTOR:	MODEL NO.

SPEED CONTROL (RUNS 216-219)

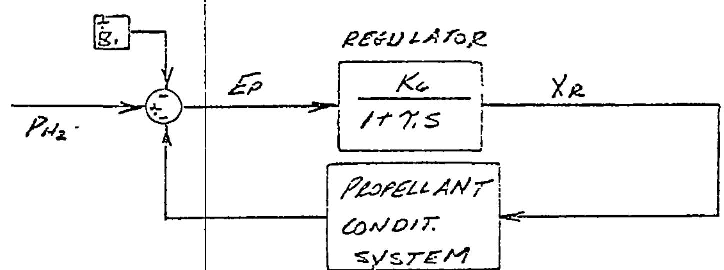


LIST OF VARIABLES	
B_1	= 1800 RPM
B_2	= 1800 RPM
B_3	= 70 %
B_4	= 3.0 %
L_1	= -100 %/SEC
L_2	= +1000 %/SEC
L_3	= -100 %/SEC
L_4	= +1000 %/SEC
K_1, K_2	= high gain
N_R	= 60,000 RPM

NOMENCLATURE			
N	SPEED, RPM	X_{SR}	VALVE POSIT. REF., %
EN	SPEED ERROR, RPM	X_{PR}	VALVE POSIT. REF., %
X_S	SUST. COMB VALVE POSITION, %	SW	SWITCH
X_P	PEAKING COMB. VALVE POSITION, %	$\dot{X}$	VALVE VELOCITY, %/SEC
CMP	COMPARITOR OUTPUT: 0 FOR NEG. INPUT, 1 FOR POS. INPUT	F	FLIP FLOP OUTPUT: 0 FOR LOGIC 1 TO C, 1 FOR LOGIC 1 TO S
		TPU	TURBO POWER UNIT
		P_o	POWER DEMAND, HP

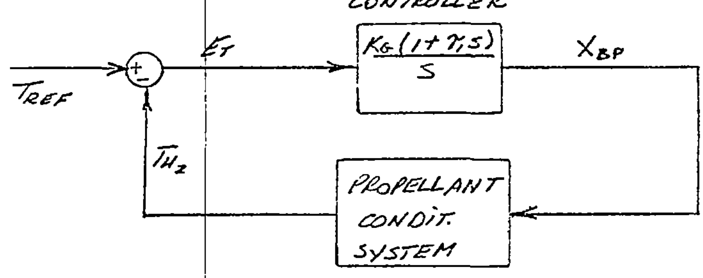
FIGURE 4a

PRESSURE REGULATOR (RUNS: 216 - 219)



LIST OF VARIABLES	NOMENCLATURE
$K_G = 4.35$ %/psi	P_{H_2} TPU H_2 INLET PRESS., PSIA
$\tau_1 = 0.77$ SECS	P_{O_2} TPU O_2 INLET PRESS, PSIA
S_1 = ADJ. TO PROVIDE TURB. INLET TEMP = 2005R AT DESIGN LEVEL.	X_R REGULATOR VALVE POSIT., %
	E_P PRESSURE ERROR, PSIA

BY PASS CONTROL



LIST OF VARIABLES	NOMENCLATURE
$K_G = 0.356$ %/R	T_{REF} - TPU H_2 INLET TEMP. REFERENCE, R
$\tau_1 = .0475$ SECS	T_{H_2} TPU H_2 INLET TEMP, R
	E_T TEMP. ERROR, R
	X_{BP} BY PASS VALVE POSITION, %.

FIGURE 4b

TURBINE TORQUE	
A-B:	SUSTAINER COMBUSTOR WITH BI-PROPELLANT VALVE AT 70% OPEN.
E-F:	SUSTAINER COMBUSTOR WITH BI-PROPELLANT VALVE AT 100% OPEN.
C-D:	SUSTAINER COMBUSTOR BI-PROPELLANT VALVE AT 70% PLUS PEAKING COMBUSTOR BI-PROPELLANT VALVE AT MINIMUM POSITION.

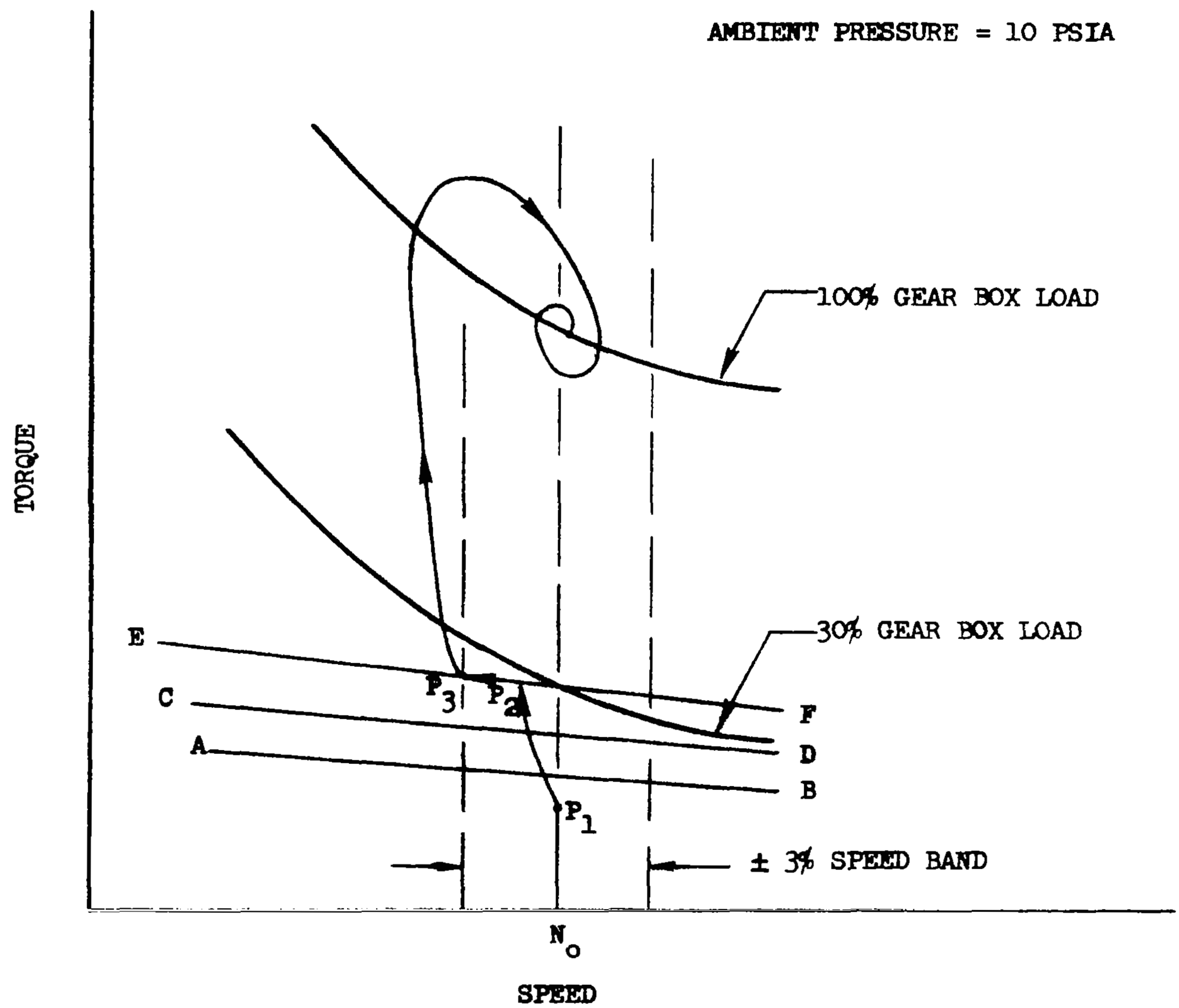


Figure 5. Hybrid Power Control
(Pressure Modulated Peaking Combustor)

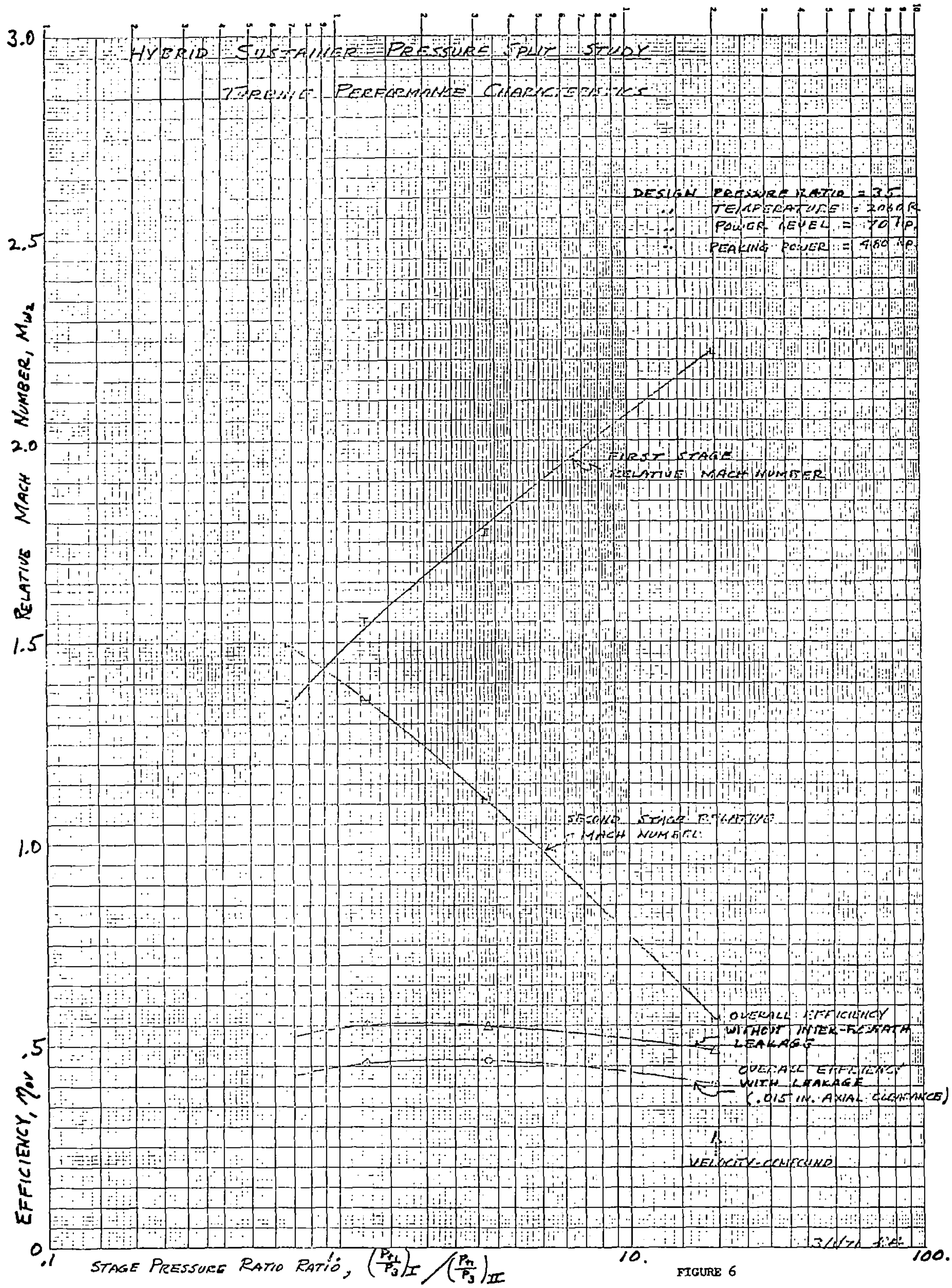
Another factor shown in Fig. 6 as an extreme value is use of a velocity-compound machine. This corresponds to a first stage pressure ratio about 20 times that of the second stage. The velocity-compound machine has an efficiency lower than any of the pressure-compound machines investigated.

The factors discussed above apply both to turbines where no inter-flow path leakage (i.e., into the peaking combustor flow path) is assumed as well as to a case where leakage is assumed. The amount of leakage assumed in Fig. 6 may be somewhat optimistic due to the unknown effect of tip leakage and leakage on the downstream side of the rotor.

The worst possible leakage case is seen in Fig. 7 as a function of axial and radial clearance values. The worst case at a generous 0.015 in. clearance is over six efficiency points lower than the more optimistic case cited in Fig. 6. This case in turn was nearly ten points lower in efficiency than the case with no leakage. Thus, even the relatively optimistic leakage assumptions resulted in greatly reduced efficiency while more pessimistic assumptions result in unacceptably low efficiencies.

PERFORMANCE ANALYSIS

The effect of the hybrid sustaining combustor power level on burned propellant weight is shown in Fig. 8. This was done for both the hybrid-pressure modulated and hybrid-pulse power controls. Turbine



HYBRID SUSTAINER PRESSURE SPLIT STUDY
PERFORMANCE
DESIGN PRESSURE RATIO = 35
TEMPERATURE = 2060 R
POWER LEVEL = 70
PEAKING POWER = 480
FIRST STAGE
RELATIVE MACH NUMBER
SECOND STAGE
MACH NUMBER
OVERALL EFFICIENCY
WITHOUT
LEAKAGE
OVERALL
WITH LEAKAGE
(.015 IN. AXIAL
VELOCITY-
3.0
2.5
2.0
1.5
1.0
.5
0
.1
10.
100.
EFFICIENCY, η_{ov}
RELATIVE MACH NUMBER, M_{W2}
STAGE PRESSURE RATIO RATIO, $\left(\frac{P_{t1}}{P_3}\right)_I / \left(\frac{P_{t1}}{P_3}\right)_{II}$

FIGURE 6

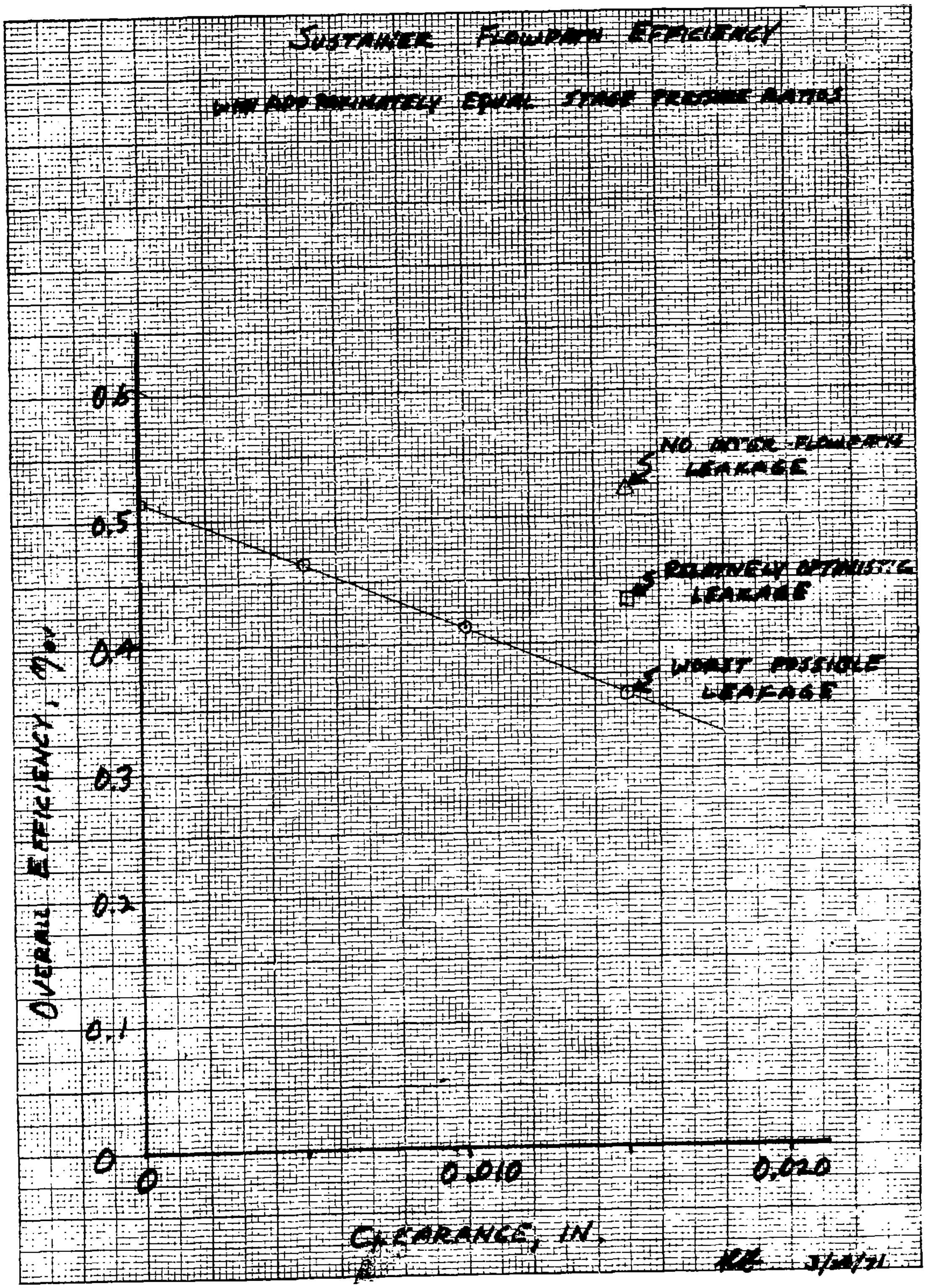

SUSTAINER FLOWPATH EFFICIENCY
WITH APPROXIMATELY EQUAL STAGE PRESSURE RATIOS
NO INTER-FLOWPATH LEAKAGE
RELATIVELY OPTIMISTIC LEAKAGE
WORST POSSIBLE LEAKAGE
OVERALL EFFICIENCY, η_{o}
0.6
0.5
0.4
0.3
0.2
0.1
0
0
0.010
0.020
CLEARANCE, IN.

FIGURE 7

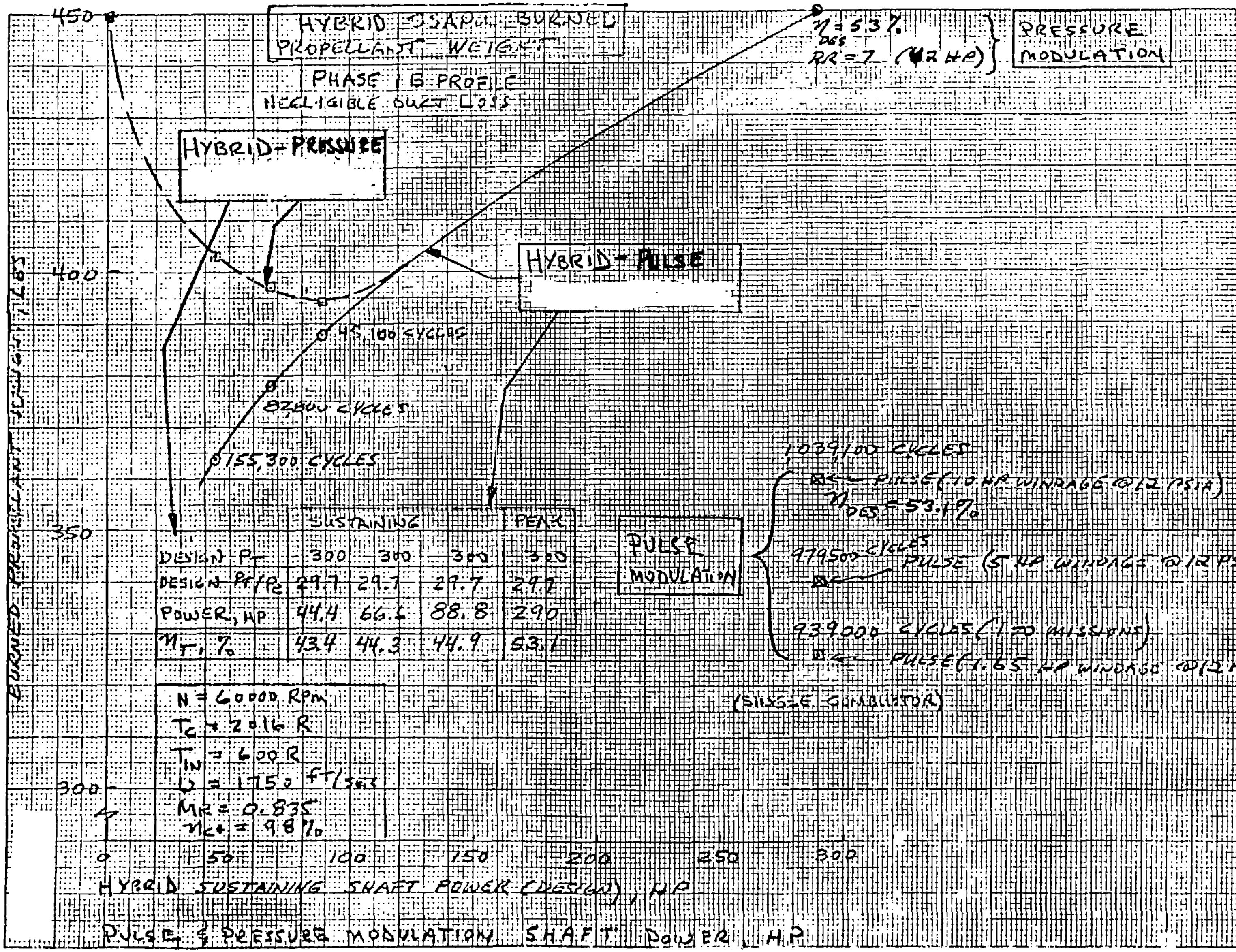

HYBRID 3SARLL BURNED PROPELLANT WEIGHT
PHASE I B PROFILE
NEGLIGIBLE DUCT LOSS
HYBRID - PRESSURE
HYBRID - PULSE
η_{DES} = 53%
RR = 7 (42 HP)
PRESSURE MODULATION
45,100 CYCLES
83,800 CYCLES
755,300 CYCLES
1039,100 CYCLES
PULSE (10 HP WINDAGE @ 12 PSIA)
η_{DES} = 53.1%
979500 CYCLES
PULSE (5 HP WINDAGE @ 12 PSIA)
939000 CYCLES (1 MISSIONS)
PULSE (1.65 HP WINDAGE @ 12 PSIA)
(SINGLE COMBUSTOR)
PULSE MODULATION
SUSTAINING
PEAK
DESIGN P 300 300 300 300
DESIGN Pt/P2 29.7 29.7 29.7 29.7
POWER, HP 44.4 66.6 88.8 290
η_T, % 43.4 44.3 44.9 53.1
N = 60000 RPM
T_C = 2016 R
T_{IN} = 600 R
U = 1750 ft/sec
M_R = 0.835
η_{c+} = 98%
450
400
350
300
0
50
100
150
200
250
300
BURNED PROPELLANT WEIGHT, LBS
HYBRID SUSTAINING SHAFT POWER (DESIGN), HP
PULSE & PRESSURE MODULATION SHAFT POWER, HP

FIGURE 8

efficiencies corresponding to a relatively optimistic leakage assumption (Fig. 7) were used in the performance analysis. Also shown in Fig. 8 for comparison is burned propellant weight for the pulse and pressure modulated control systems.

The Phase 1B booster power profile was used for the duty cycle. The design points and operating conditions of the various hybrid combinations are also shown in the figure. An optimum power level of 90 HP result for the hybrid-pressure control. A power level of 106 HP for the hybrid-pulse sustainer combustor would provide the same amount of burned propellant (390 lbs), and result in a 96% reduction in required pulses when compared with a "pure" pulse power control.

For a pure pulse system, a windage loss of approximately 6 HP was estimated at 12 psia turbine exhaust pressure, based upon a correlation with Rocketdyne test data. The effect of windage on propellant assumption is shown in Fig. 8.

APPENDIX D

PROPELLANT UTILIZATION DIGITAL PROGRAM ANALYSIS

Digital computer programs were written to determine the burned propellant required for a given TPU design operating over the booster and orbiter mission profiles. One program (SSAPU3) utilizes pressure modulation control and the other (SSAPU4) pulse modulation control. The profiles were subdivided into many slices (28 orbiter, 26 booster) so that the program could handle the transient duty cycle on a steady-state basis, greatly simplifying the calculations. In addition to propellant utilization estimates, the programs were also utilized in support of system and TPU optimization studies and to determine the SPC characteristic with power and altitude for the resulting baseline design. Samples of program output for the optimum pulse and pressure modulated machines are presented in the following tabulation for the orbiter re-entry, booster cruise, and peak power condition.

PRESSURE MODULATION CONTROL

DESIGN INPUT

SSAPJ3 08:13 NR T/S JULY 21,1971

SSAPJ BURNED PROPELLANT REQUIREMENT - PRESS MOD CONTROL
EXHAUST DUCT - 50 FEET LONG, 4 INCH DIAMETER, 3 ELBOWS
REGENERATOR DELTA P=710*FLOW SQUARED/AVE. PRESS
DESIGN EFFICIENCY = .555
DESIGN PRESSURE RATIO = 12
DESIGN PT (PSIA) = 139.8
DESIGN POWER (HP) = 135.2
DESIGN PE (PSIA) = 11.65

ADIABATIC SPC (LB/HP-HR)= 1.262
MIXTURE RATIO = .834
TURBINE SPEED (RPM) = 60000
TIP SPEED (FT/SEC) = 1750
COMBUSTION TEMP (R) = 2060
COMBUSTOR INLET TEMP (R) = ~~800~~ 650
C-STAR EFFICIENCY (%) = .98

BOOSTER MAXIMUM POWER

PROPELLANT FLOW - LBS/SEC	= .16775
PROPELLANT WEIGHT - LBS	= 2.68416
SPC LB/HP-HR (TURBINE)	= 1.78785
SPC LB/HP-HR (GEARBOX)	= 1.84116
C-STAR EFFICIENCY	= .98
DEGRADATION	= 1.2715
POWER LEVEL (HP)	= 337.8
GEARBOX OUTPUT POWER(HP)	= 328.019
TURBINE INLET PRESS (PSIA)	= 271.793
TURBINE EXIT PRESS (PSIA)	= 8.63445
AMBIENT PRESSURE (PSIA)	= 5.5
DURATION (SECONDS)	= 16

BOOSTER CRUISE 55 HP

PROPELLANT FLOW - LBS/SEC	= .048986
PROPELLANT WEIGHT - LBS	= 64.8575
SPC LB/HP-HR (TURBINE)	= 2.86747
SPC LB/HP-HR (GEARBOX)	= 3.20976
C-STAR EFFICIENCY	= .973247
DEGRADATION	= .802568
POWER LEVEL (HP)	= 61.5
GEARBOX OUTPUT POWER(HP)	= 54.9417
TURBINE INLET PRESS (PSIA)	= 78.5925
TURBINE EXIT PRESS (PSIA)	= 10.5213
AMBIENT PRESSURE (PSIA)	= 10.1
DURATION (SECONDS)	= 1324

BOOSTER CRUISE 33 HP

PROPELLANT FLOW - LBS/SEC	= 3.809388E-2
PROPELLANT WEIGHT - LBS	= 202.24
SPC LB/HP-HR (TURBINE)	= 3.48951
SPC LB/HP-HR (GEARBOX)	= 4.15561
C-STAR EFFICIENCY	= .967129
DEGRADATION	= .567442
POWER LEVEL (HP)	= 39.3
GEARBOX OUTPUT POWER(HP)	= 33.0006
TURBINE INLET PRESS (PSIA)	= 61.1371
TURBINE EXIT PRESS (PSIA)	= 10.4276
AMBIENT PRESSURE (PSIA)	= 10.1
DURATION (SECONDS)	= 5309

PRESSURE MODULATION CONTROL

(Continued)

ORBITER MAXIMUM POWER

PROPELLANT FLOW - LBS/SEC	=	.220272
PROPELLANT WEIGHT - LBS	=	5.28653
SPC LB/HP-HR (TURBINE)	=	1.93127
SPC LB/HP-HR (GEARBOX)	=	1.98259
C-STAR EFFICIENCY	=	.98
DEGRADATION	=	1.17701
POWER LEVEL (HP)	=	410.6
GEARBOX OUTPUT POWER(HP)	=	399.37
TURBINE INLET PRESS (PSIA)	=	357.07
TURBINE EXIT PRESS (PSIA)	=	17.3663
AMBIENT PRESSURE (PSIA)	=	14.7
DURATION (SECONDS)	=	24

ORBITER RE-ENTRY

PROPELLANT FLOW - LBS/SEC	=	.02817
PROPELLANT WEIGHT - LBS	=	95.3556
SPC LB/HP-HR (TURBINE)	=	1.79491
SPC LB/HP-HR (GEARBOX)	=	2.02824
C-STAR EFFICIENCY	=	.95339
DEGRADATION	=	1.30458
POWER LEVEL (HP)	=	56.5
GEARBOX OUTPUT POWER(HP)	=	50
TURBINE INLET PRESS (PSIA)	=	45.9223
TURBINE EXIT PRESS (PSIA)	=	1.18314
AMBIENT PRESSURE (PSIA)	=	0
DURATION (SECONDS)	=	3385

PULSE POWER CONTROL

DESIGN INPUT

SSAPU4 08:07 NAR JUN 08, 1971

SSAPU BURNED PRØPELLANT REQUIREMENT-PULSE MØDULATED CØNTRØL
EXHAUST DUCT-50 FEET LØNG, 4 INCH DIA., 3 ELBØWS
REGENERATØR DELTA P=710*FLØW SQUARED/AVE. PRESS

DESIGN EFFICIENCY (%/100)	=	.5716
DESIGN PT/PE	=	26.9
DESIGN PT (MAXIMUM PSIA)	=	390
DESIGN PØWER (HP)	=	468.9
TPU WEIGHT (LBS)	=	110
TPU INERTIA (LB-FT-SEC,SQUARED)	=	.01142
ADIABATIC SPC (LB/HP-HR)	=	1.041
MIXTURE RATIØ	=	.834
TURBINE SPEED (RPM)	=	60000
TIP SPEED (FT/SEC)	=	1750
CØMBUSTIØN TEMPERATURE (DEG R)	=	2060
CØMBUSTØR INLET TEMPERATURE (R)	=	650
C-STAR EFFICIENCY (%)	=	.98

PULSE POWER CONTROL

(Continued)

BOOSTER MAXIMUM POWER

ØN PULSE FLØW - LBS/SEC	=	.237212
AVERAGE FLØW - LBS/ SEC	=	.170212
PRØPELLANT WEIGHT - LBS	=	2.7234
SPC LB/HP-HR	=	1.81399
TURBINE SHAFT PØWER (HP)	=	472.189
LØAD SHAFT PØWER (HP)	=	337.8
TURBINE INLET PRESSURE (PSIA)	=	390
TURBINE EXIT PRESSURE (PSIA)	=	11.3569
AMBIENT PRESSURE (PSIA)	=	5.5
DUCT + REGENERATØR DELTA P	=	5.85688
DURATIØN - SECØNDS	=	16
PULSE CYCLES	=	37.7178
PULSE ØN TIME (SECØNDS)	=	.304389
PULSE ØFF TIME (SECØNDS)	=	.119814
PULSE PERIØD (SECØNDS)	=	.424203

BOOSTER CRUISE 33 HP

ØN PULSE FLØW - LBS/SEC	=	.237212
AVERAGE FLØW - LBS/ SEC	=	2.37913E-2
PRØPELLANT WEIGHT - LBS	=	126.308
SPC LB/HP-HR	=	2.17936
TURBINE SHAFT PØWER (HP)	=	438.479
LØAD SHAFT PØWER (HP)	=	39.3
TURBINE INLET PRESSURE (PSIA)	=	390
TURBINE EXIT PRESSURE (PSIA)	=	14.1377
AMBIENT PRESSURE (PSIA)	=	10.1
DUCT + REGENERATØR DELTA P	=	4.03775
DURATIØN - SECØNDS	=	5309
PULSE CYCLES	=	5196.02
PULSE ØN TIME (SECØNDS)	=	.102477
PULSE ØFF TIME (SECØNDS)	=	.919267
PULSE PERIØD (SECØNDS)	=	1.02174

ORBITER MAXIMUM POWER

ØN PULSE FLØW - LBS/SEC	=	.237212
AVERAGE FLØW - LBS/ SEC	=	.227279
PRØPELLANT WEIGHT - LBS	=	5.45469
SPC LB/HP-HR	=	1.9927
TURBINE SHAFT PØWER (HP)	=	428.849
LØAD SHAFT PØWER (HP)	=	410.6
TURBINE INLET PRESSURE (PSIA)	=	390
TURBINE EXIT PRESSURE (PSIA)	=	17.7559
AMBIENT PRESSURE (PSIA)	=	14.7
DUCT + REGENERATØR DELTA P	=	3.05587
DURATIØN - SECØNDS	=	24
PULSE CYCLES	=	10.2587
PULSE ØN TIME (SECØNDS)	=	2.24151
PULSE ØFF TIME (SECØNDS)	=	.097965
PULSE PERIØD (SECØNDS)	=	2.33948

ORBITER RE-ENTRY

ØN PULSE FLØW - LBS/SEC	=	.237212
AVERAGE FLØW - LBS/ SEC	=	3.05124E-2
PRØPELLANT WEIGHT - LBS	=	103.285
SPC LB/HP-HR	=	1.94415
TURBINE SHAFT PØWER (HP)	=	453.976
LØAD SHAFT PØWER (HP)	=	56.5
TURBINE INLET PRESSURE (PSIA)	=	390
TURBINE EXIT PRESSURE (PSIA)	=	10.1373
AMBIENT PRESSURE (PSIA)	=	0
DUCT + REGENERATØR DELTA P	=	10.1373
DURATIØN - SECØNDS	=	3385
PULSE CYCLES	=	4230.77
PULSE ØN TIME (SECØNDS)	=	.102915
PULSE ØFF TIME (SECØNDS)	=	.697176
PULSE PERIØD (SECØNDS)	=	.800092

APPENDIX E

TRANSIENT PERFORMANCE EVALUATION-ANALOG MODEL ANALYSIS

Transient performance evaluation of the APU over the flight operational envelope was performed with the aid of an analog model constructed during the Phase II study.

Startup evaluation of a pulse system under varying environmental conditions is presented in Fig. 114 through 119.

Fig. No.	Run No.	Condition
114	372	Nominal inlet conditions; ambient pressure = 14.7 psia
115	373	Same as above (compressed time scale to view long time effects)
116	401	Nominal inlet conditions except inlet O_2 pressure = 1000 psi; ambient pressure = 10 psia
117	374	Nominal inlet conditions; pressure = 0.5 psi
118	416	Nominal inlet conditions; ambient pressure = 14.7 psia. APU brought "on line" in less than 2.0 seconds to 100-percent power
119	391	Nominal inlet conditions; ambient pressure = 14.7 psia. APU hardware and hydraulic-lube oil initialized at 400 R

Power transients of a pulse system are presented in Fig. 120 and 121.

Fig. No.	Run No.	Condition
120	376	Power spikes 40 to 428 horsepower; P_a = 10 psia; nominal inlet conditions
121	386	Power spikes 40 to 200 horsepower; P_a = 10 psia; nominal inlet conditions

The effect of inlet condition variations on a pulse system is presented in Fig. 122 and 123.

Fig. No.	Run No.	Condition
122	390	ACS oxygen tank pump up, i.e., a 2-second ramp in GO_2 supply temperature from 400 to 200 R and GO_2 pressure from 500 to 1000 psi; other inlet conditions nominal
123	380	Hydrogen supply temperature range increase from 75 to 375 R at 100 R/sec with full hydraulic cooling; other inlet conditions nominal

A pulse system shutdown is presented in Fig. 124

Fig. No.	Run No.	Condition
124	402	Shutdown, nominal inlet conditions; P_a = 14.7 psia

Some selected results of the failure mode analysis conducted on a pulse system arepresented in Fig. 125 through 127.

Fig. No.	Run No.	Condition
125	426	Simulated 15-percent reduction in GH_2 injector flow area with turbine inlet temperature limit control at 40 horsepower; nominal inlet conditions; P_a = 10 psia
126	409	Excessive hydraulic system heat load (approximately 69 Btu/sec) with lube oil temperature limit control at 40 horsepower; nominal inlet conditions; P_a = 10 psia
127	419	Regenerator bypass valve stuck closed at 40 horsepower; nominal inlet conditions; P_a = 14.7 psia

Startup evaluation: power transients; and inlet condition variations of a pressure modulated system are presented in Fig. 128 through 132.

Fig. No.	Run No.	Condition
128	448	Nominal inlet conditions; P_a = 14.7 psia
129	444	Nominal inlet conditions; P_a = 14.7 psia
130	504	Nominal inlet conditions; P_a = 10 psia; power spikes, 40 to 428 horsepower
131	500	Nominal inlet conditions; P_a = 12 psia; power ramp 40 to 428 horsepower
132	450	Inlet oxygen pressure ramp from 520 to 1000 psia at 250 psi/sec; other inlet conditions nominal; P_a = 14.7 psia

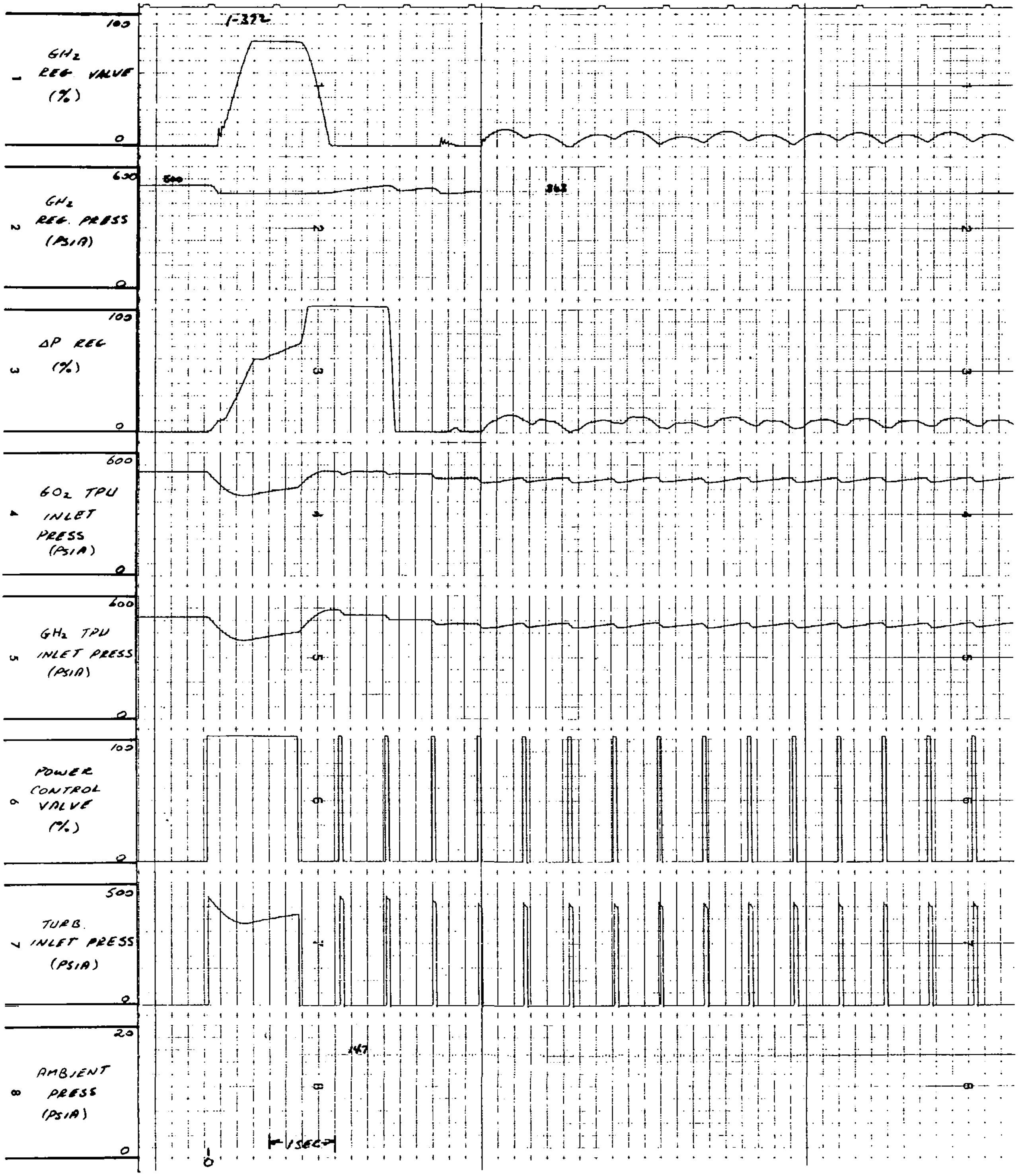


Figure 114. Sea Level Startup--Pulse System

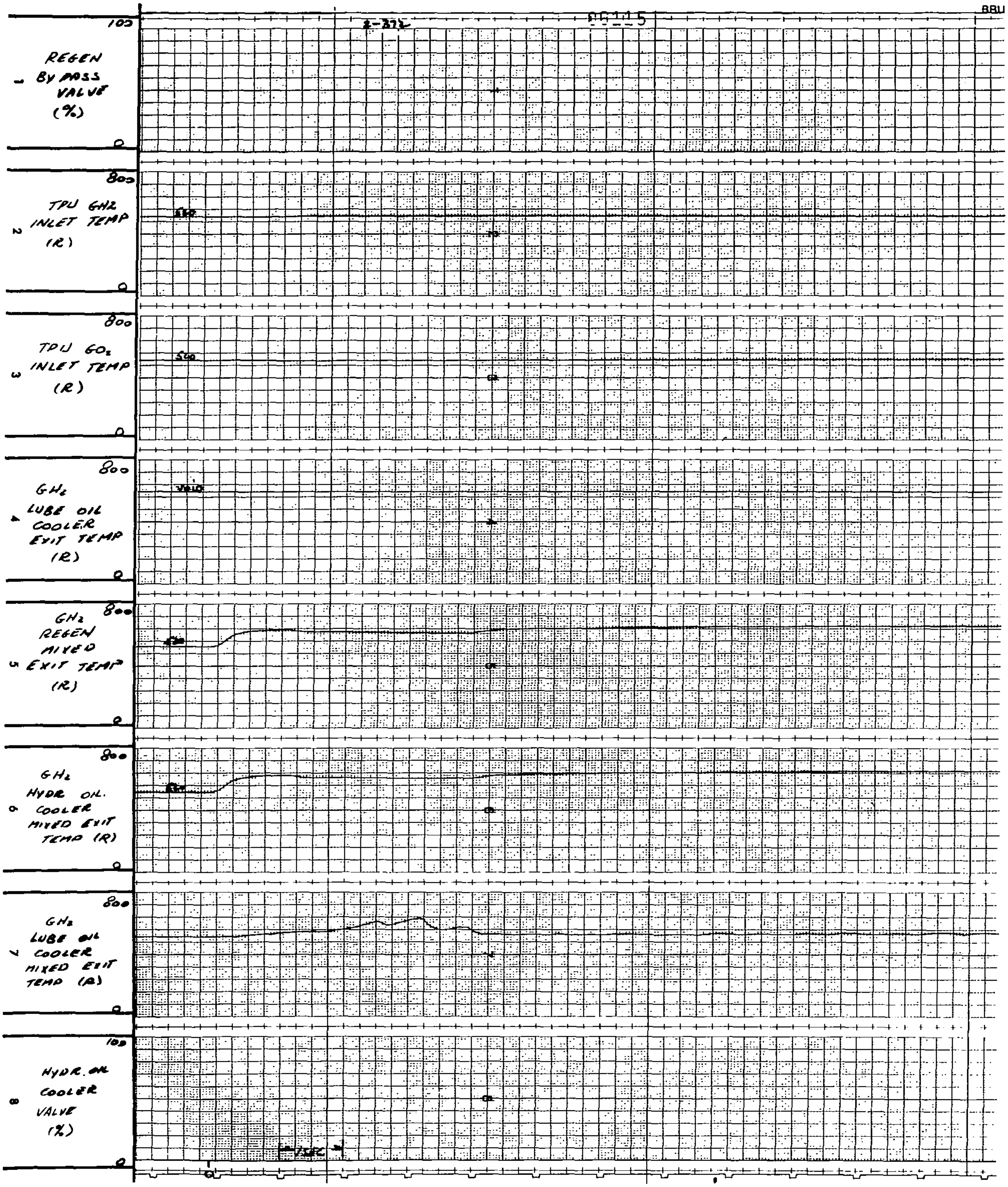


Figure 114. (Continued)

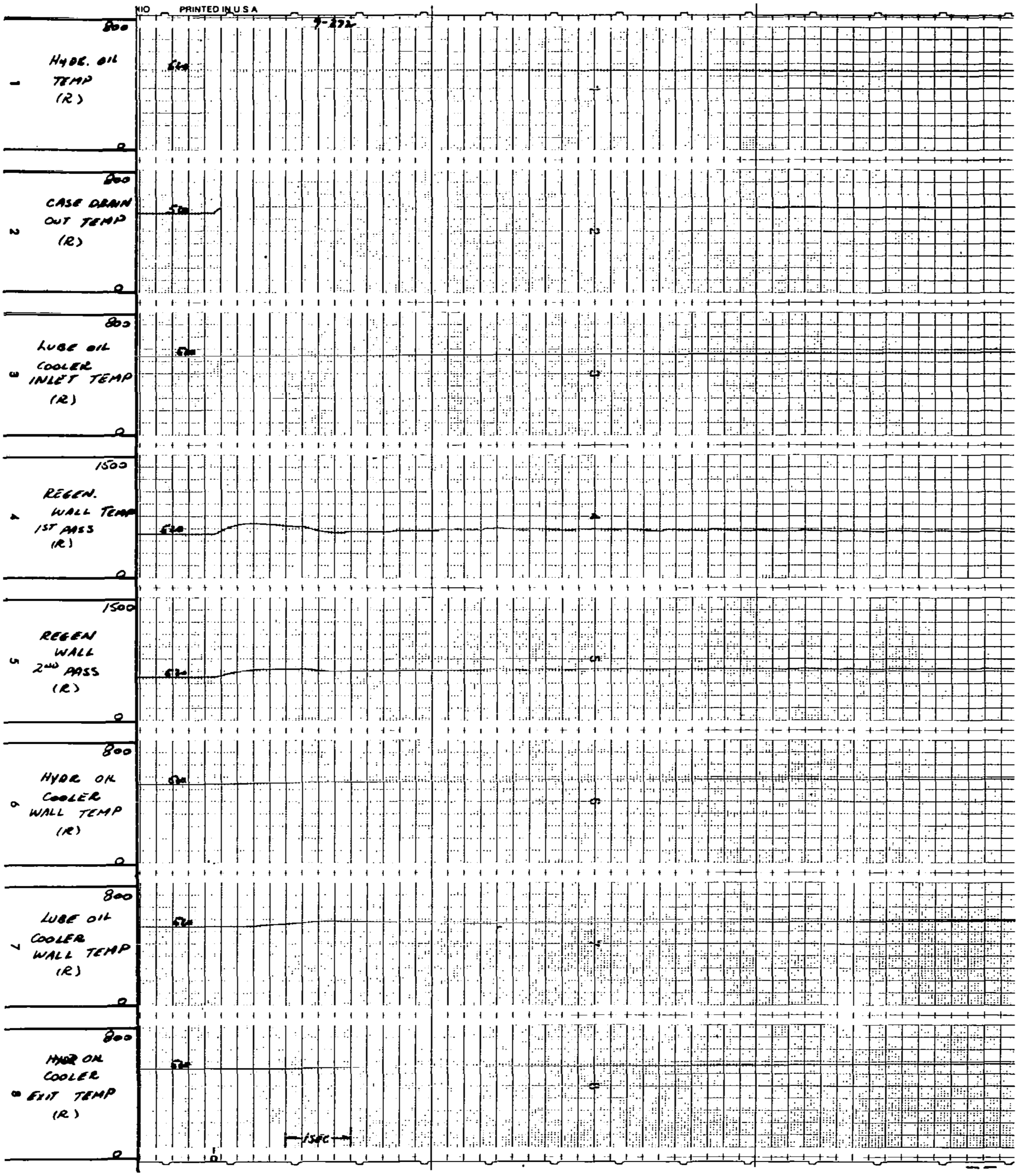


Figure 114. (Continued)

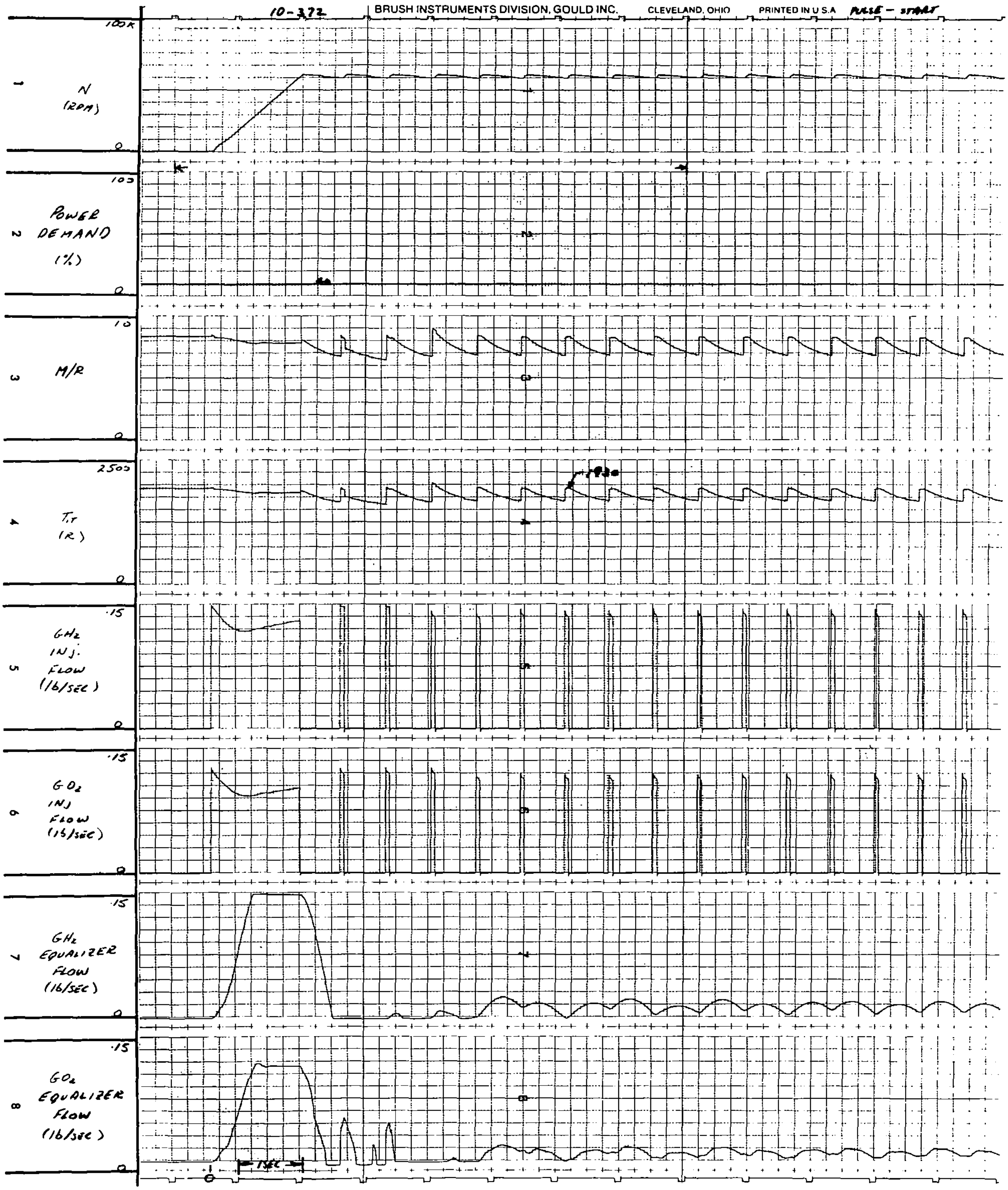


Figure 114. (Concluded)

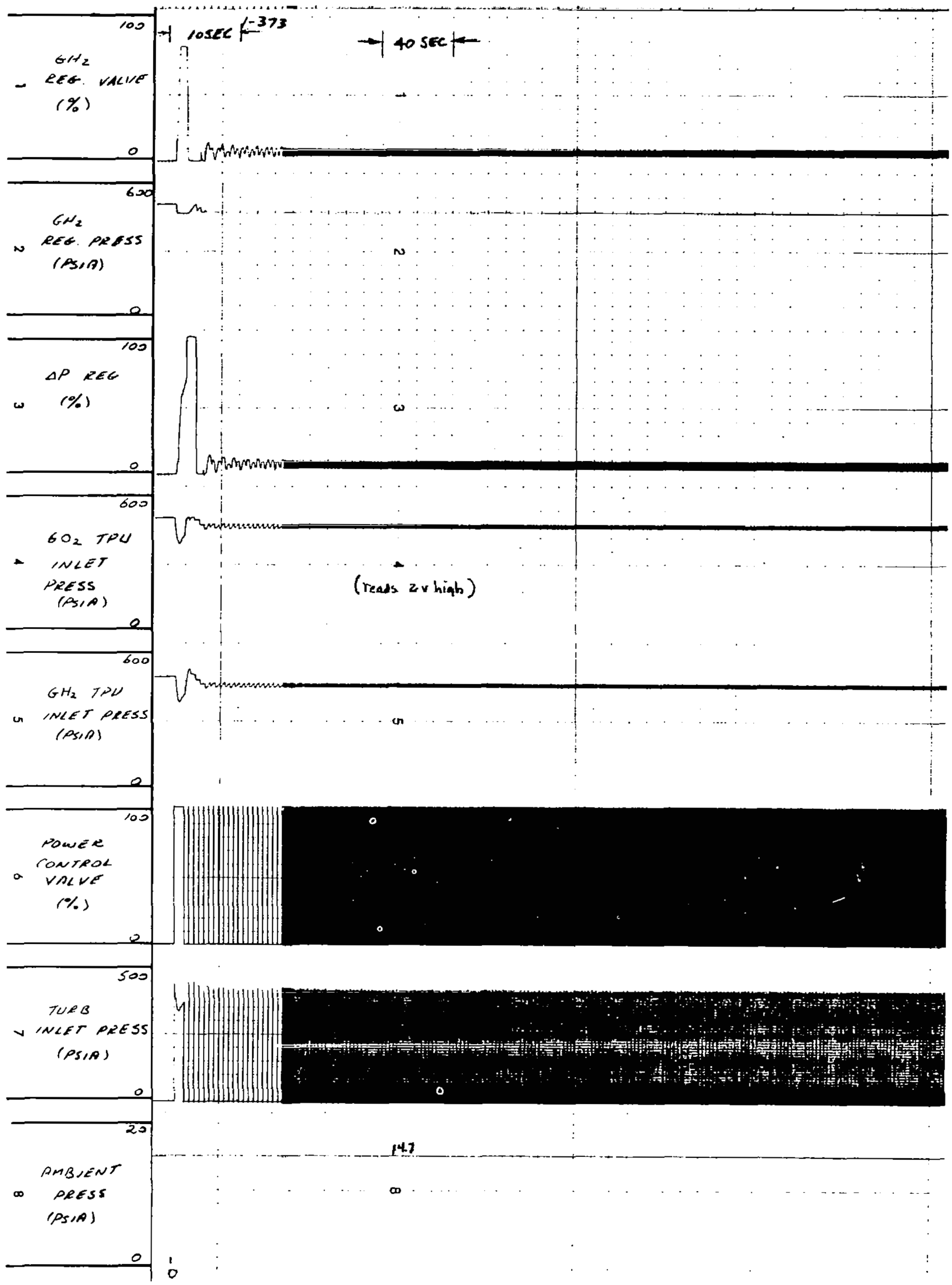


Figure 115. Sea Level Startup, Thermal Effects--Pulse System

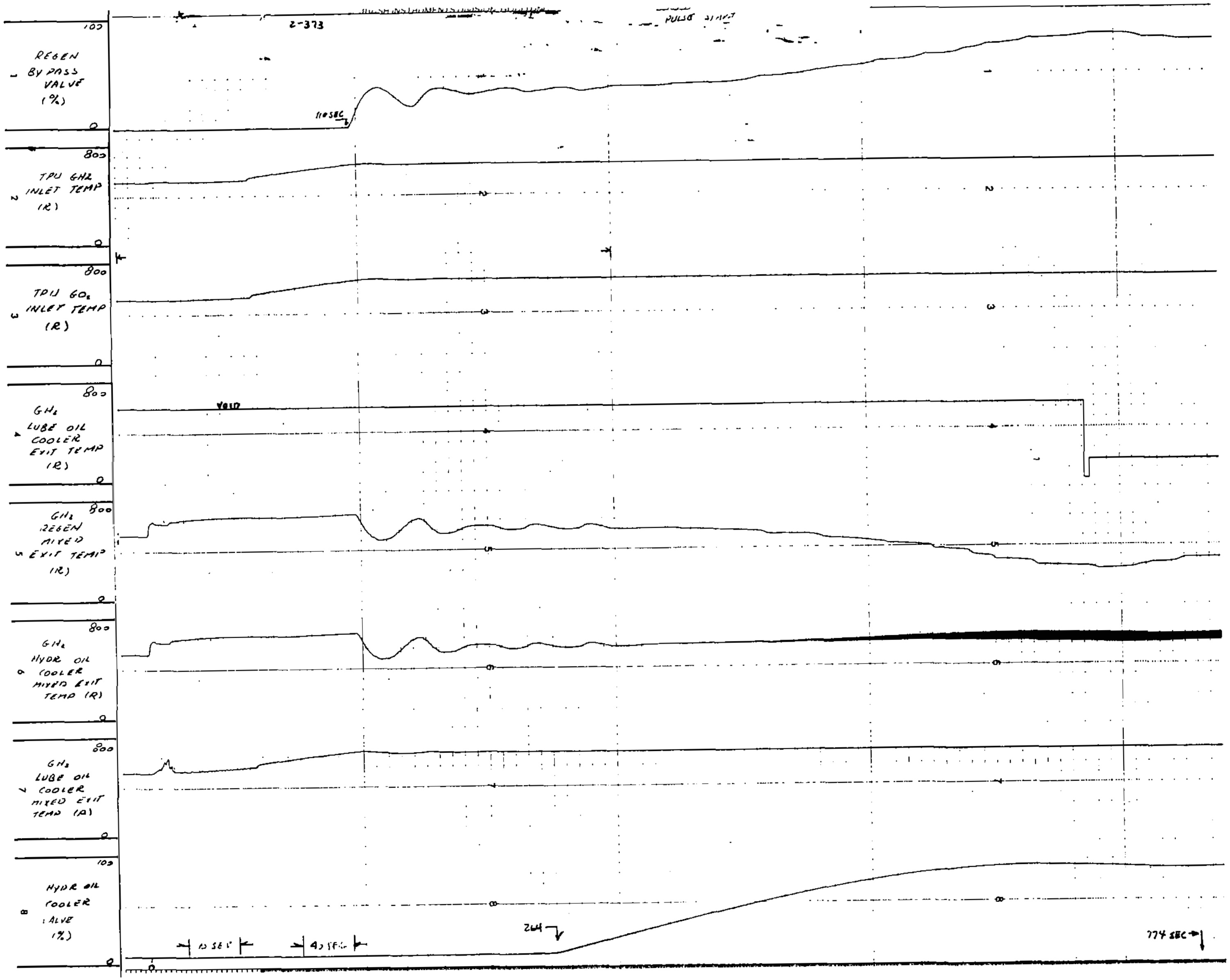


Figure 115. (Continued)

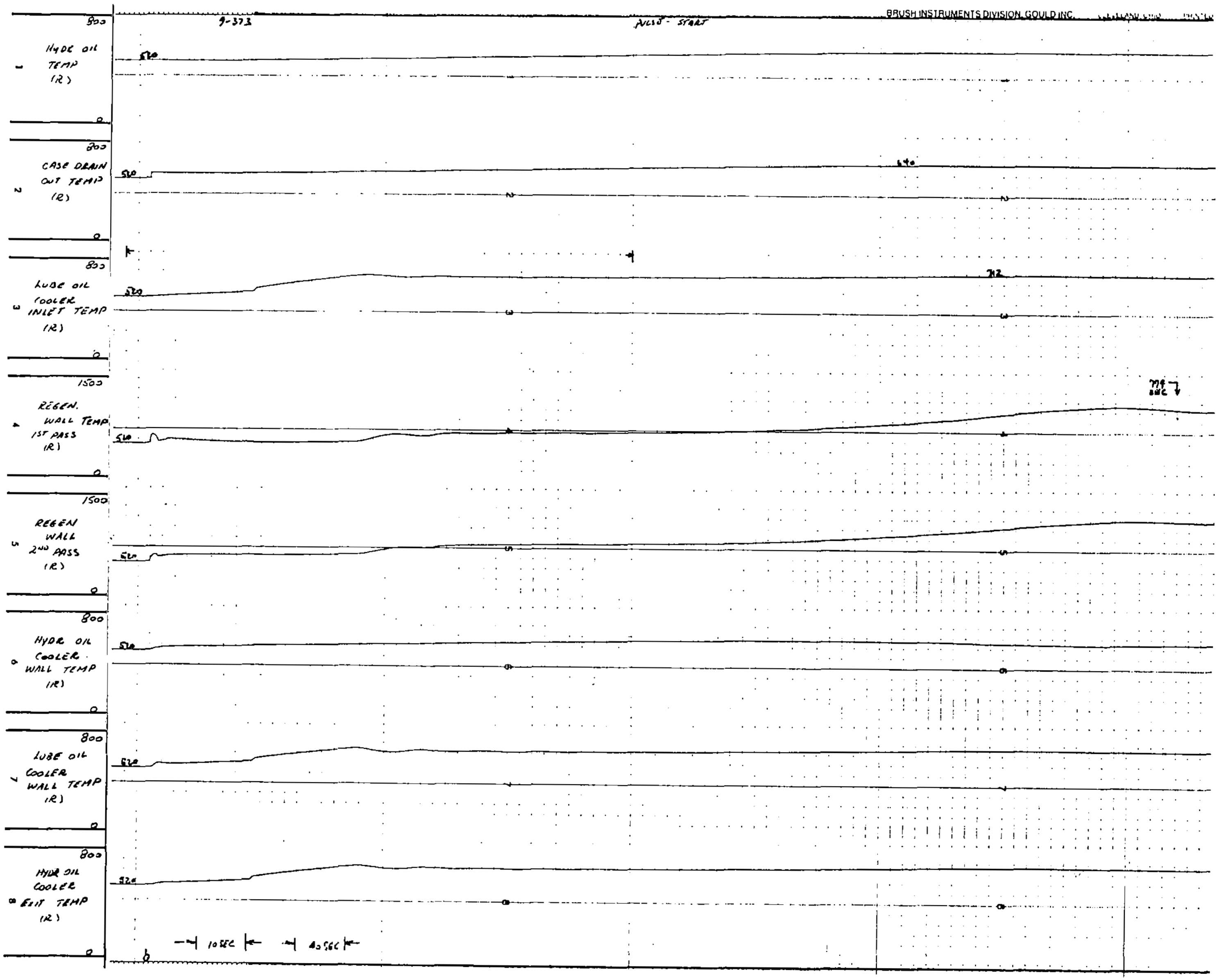


Figure 115. (Continued)

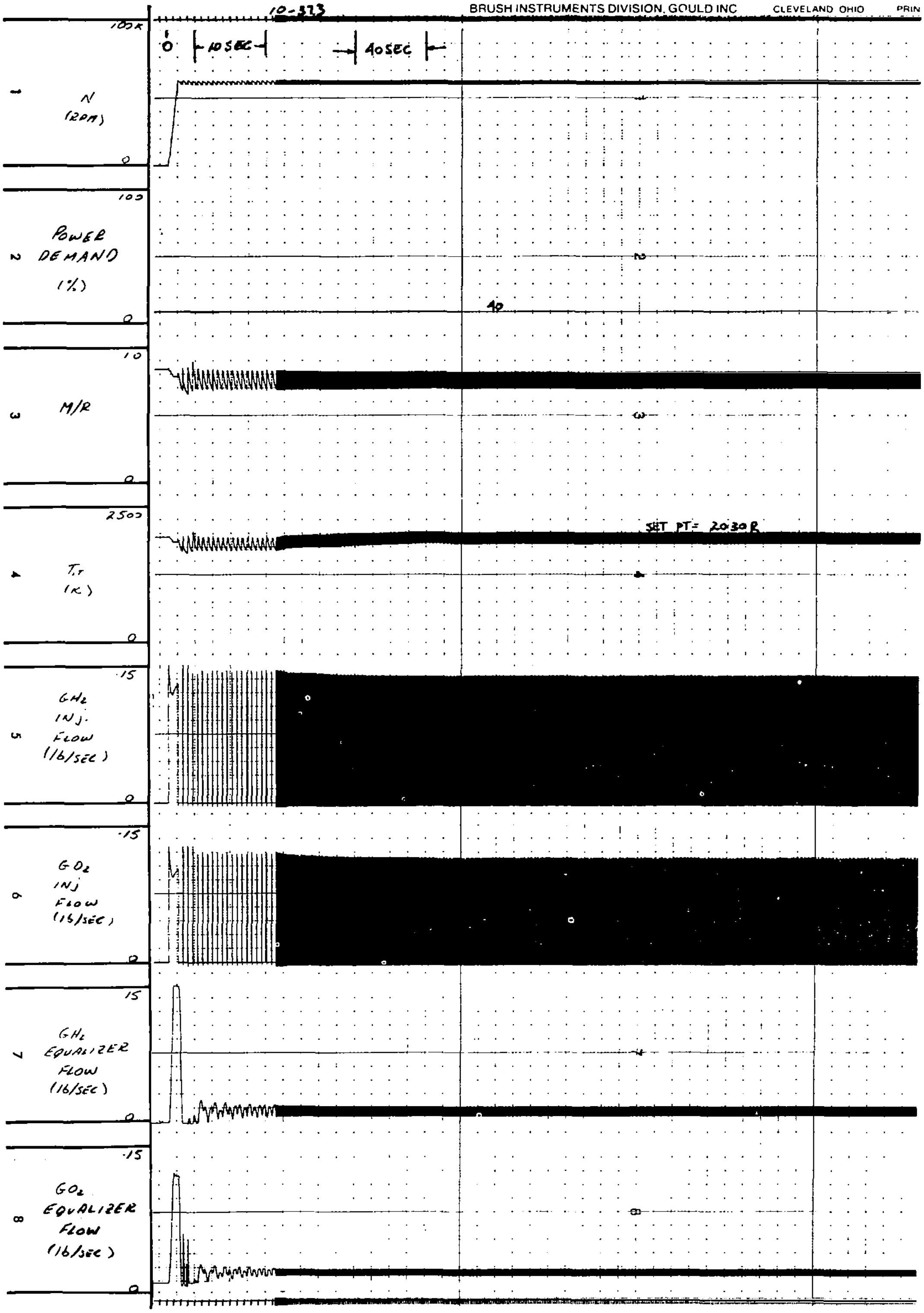


Figure 115. (Concluded

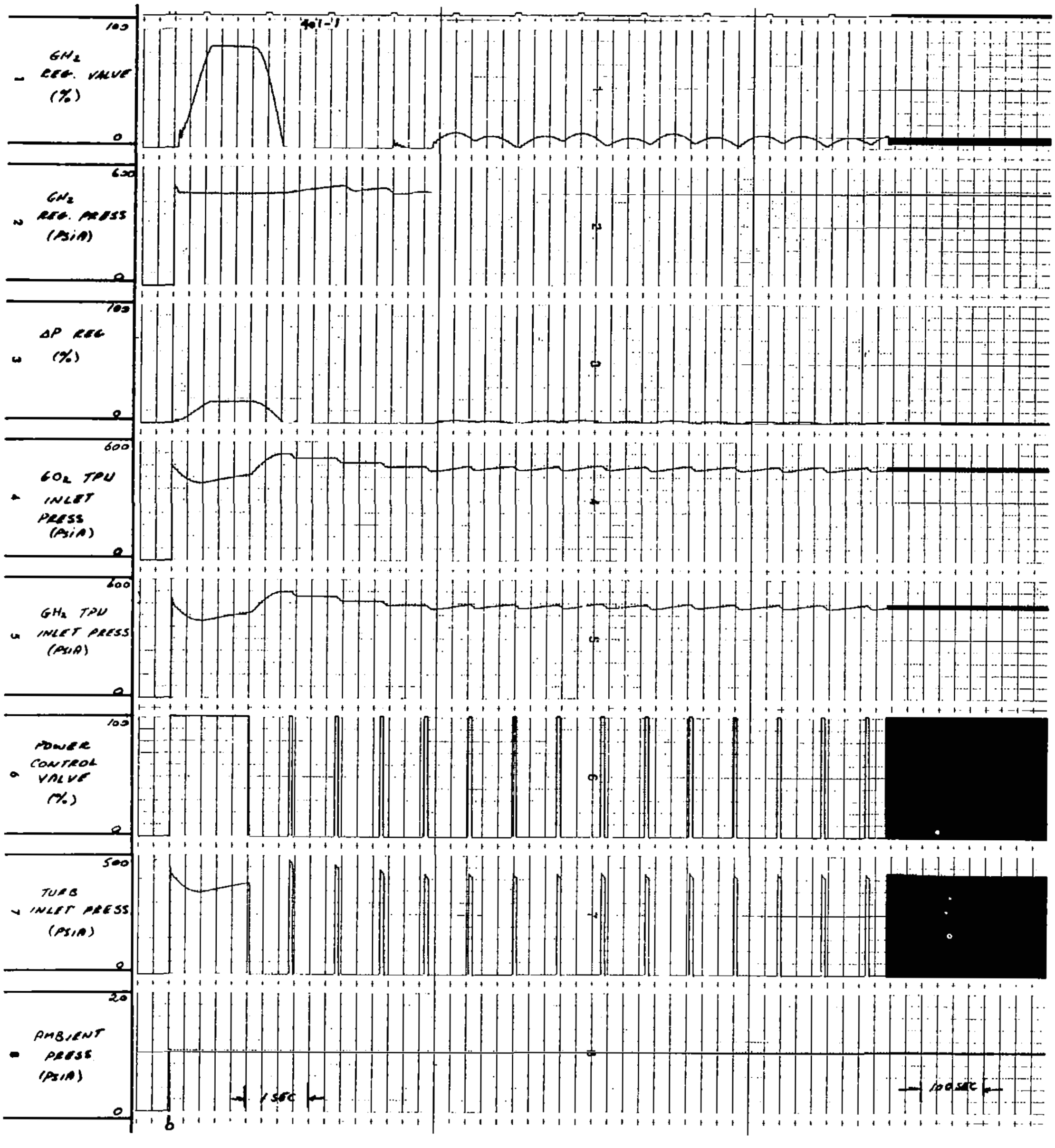


Figure 116. Startup, High GO_2 Supply Pressure--Pulse System

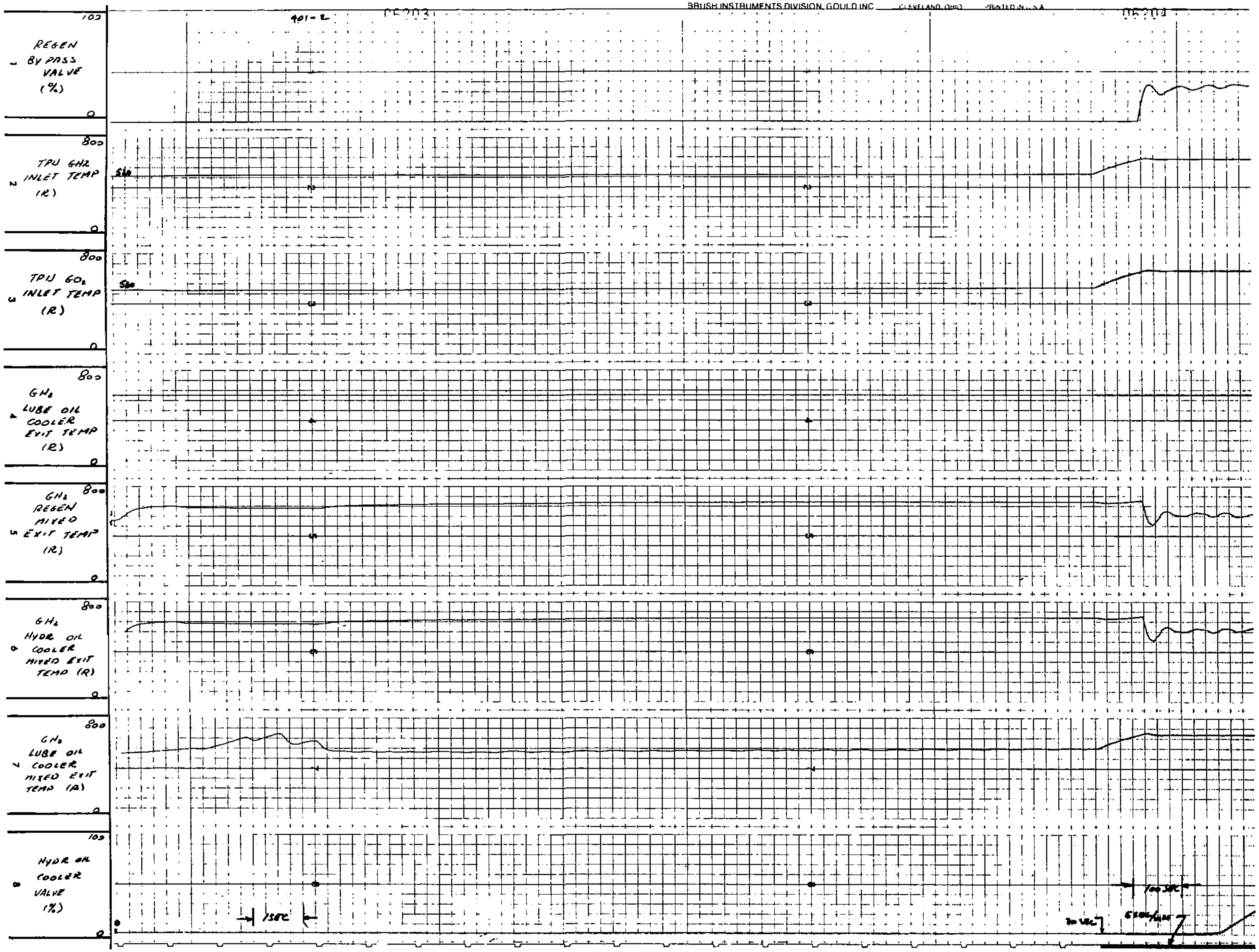


Figure 116. (Continued)

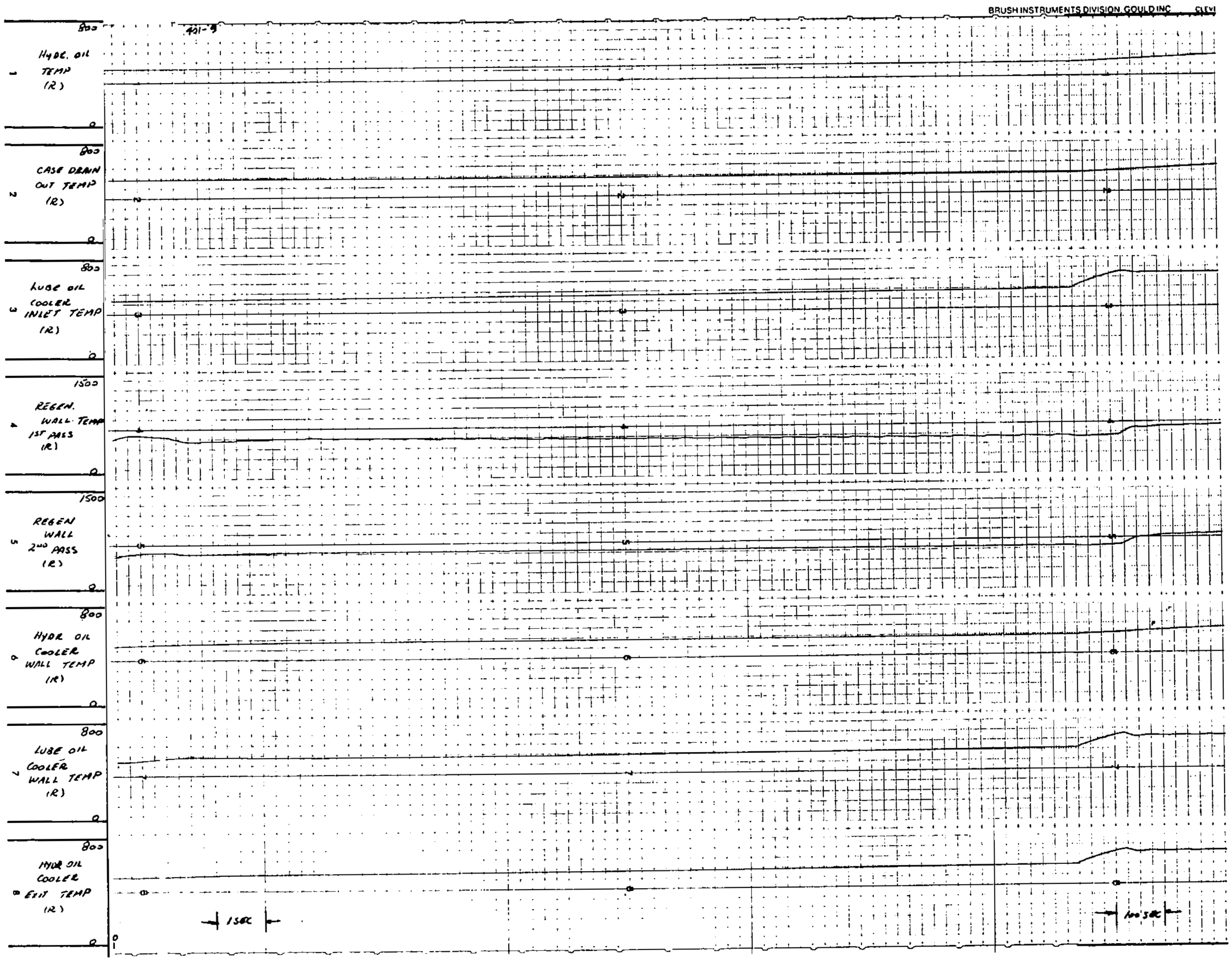


Figure 116. (Continued)

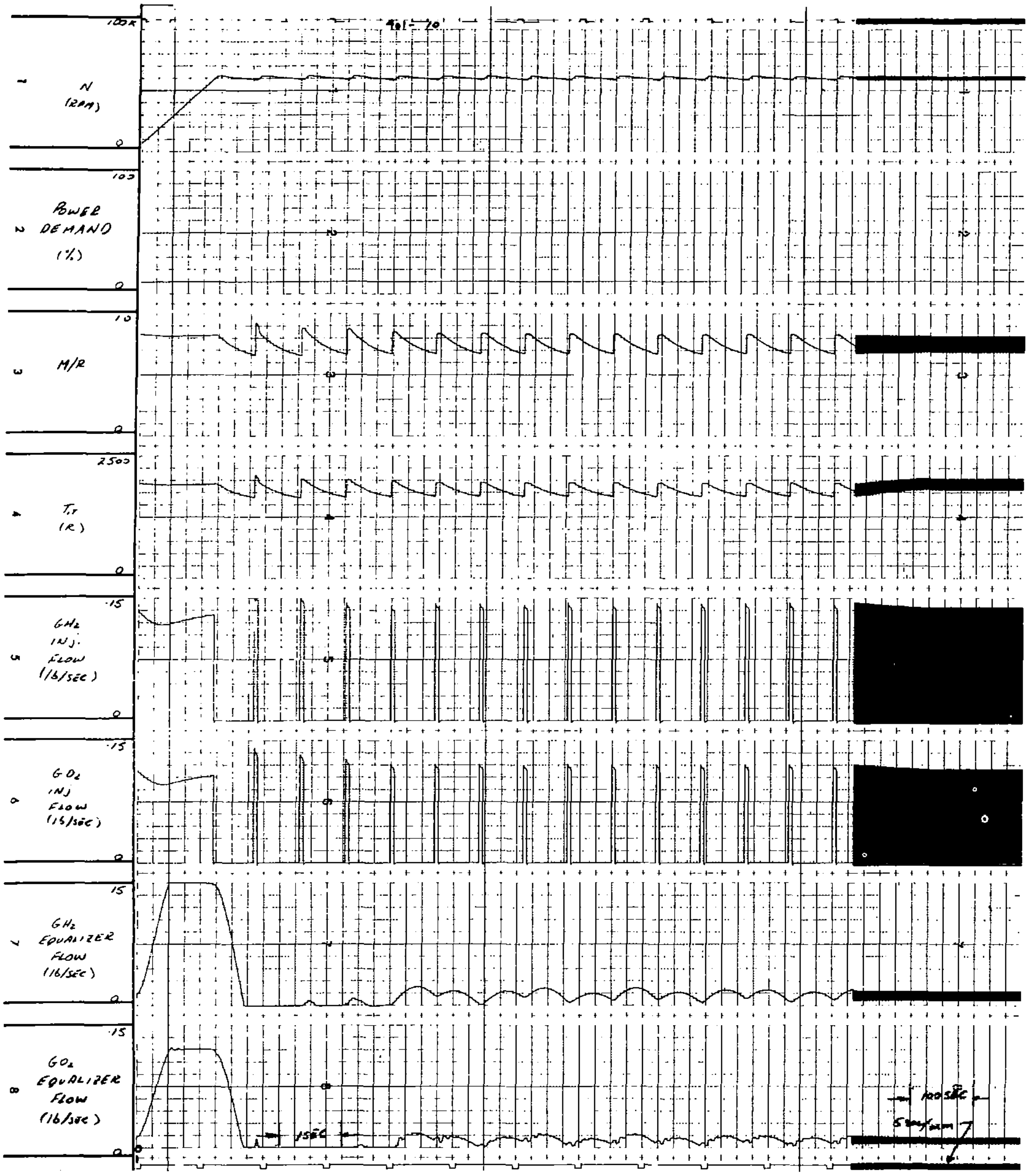


Figure 116. (Concluded)

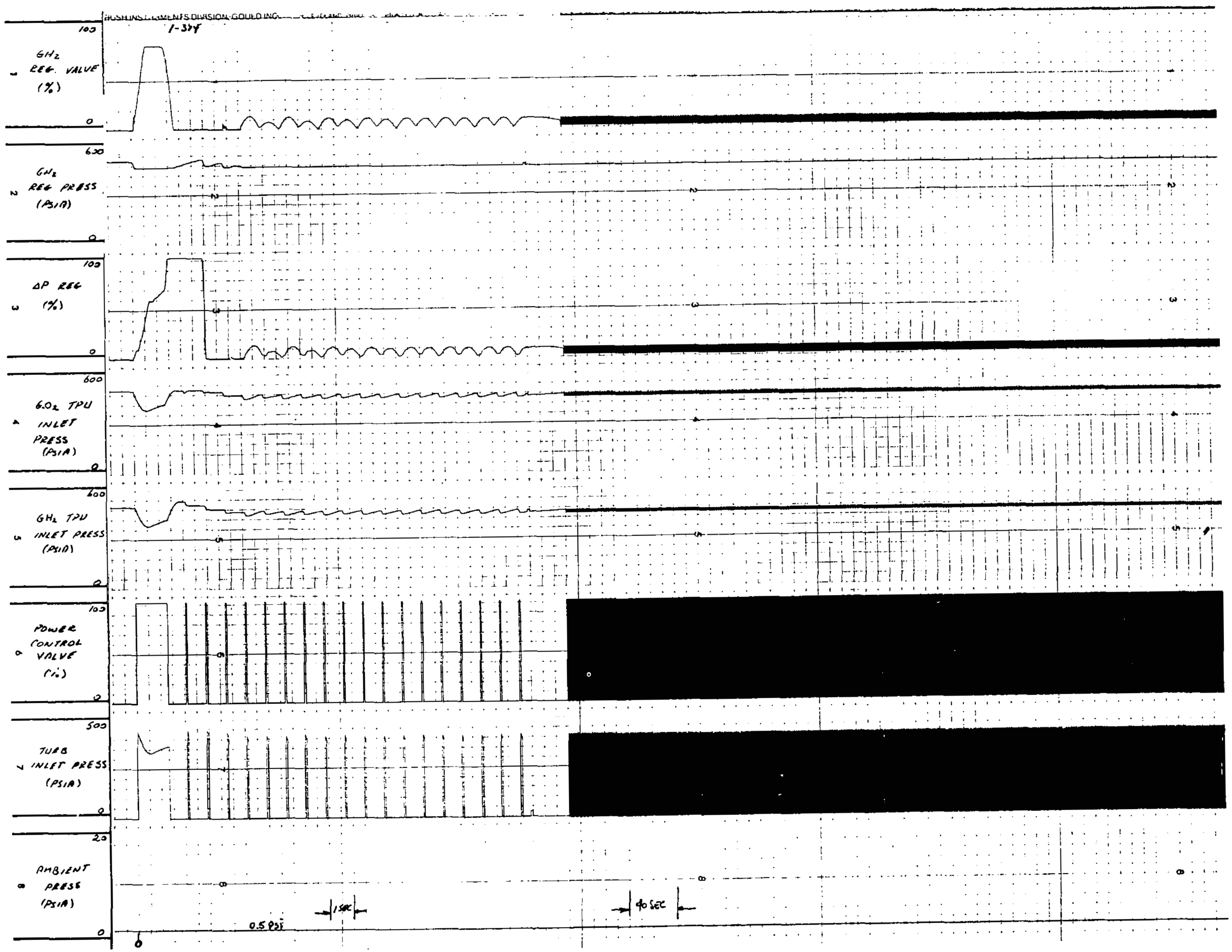


Figure 117. High-Altitude Startup--Pulse System

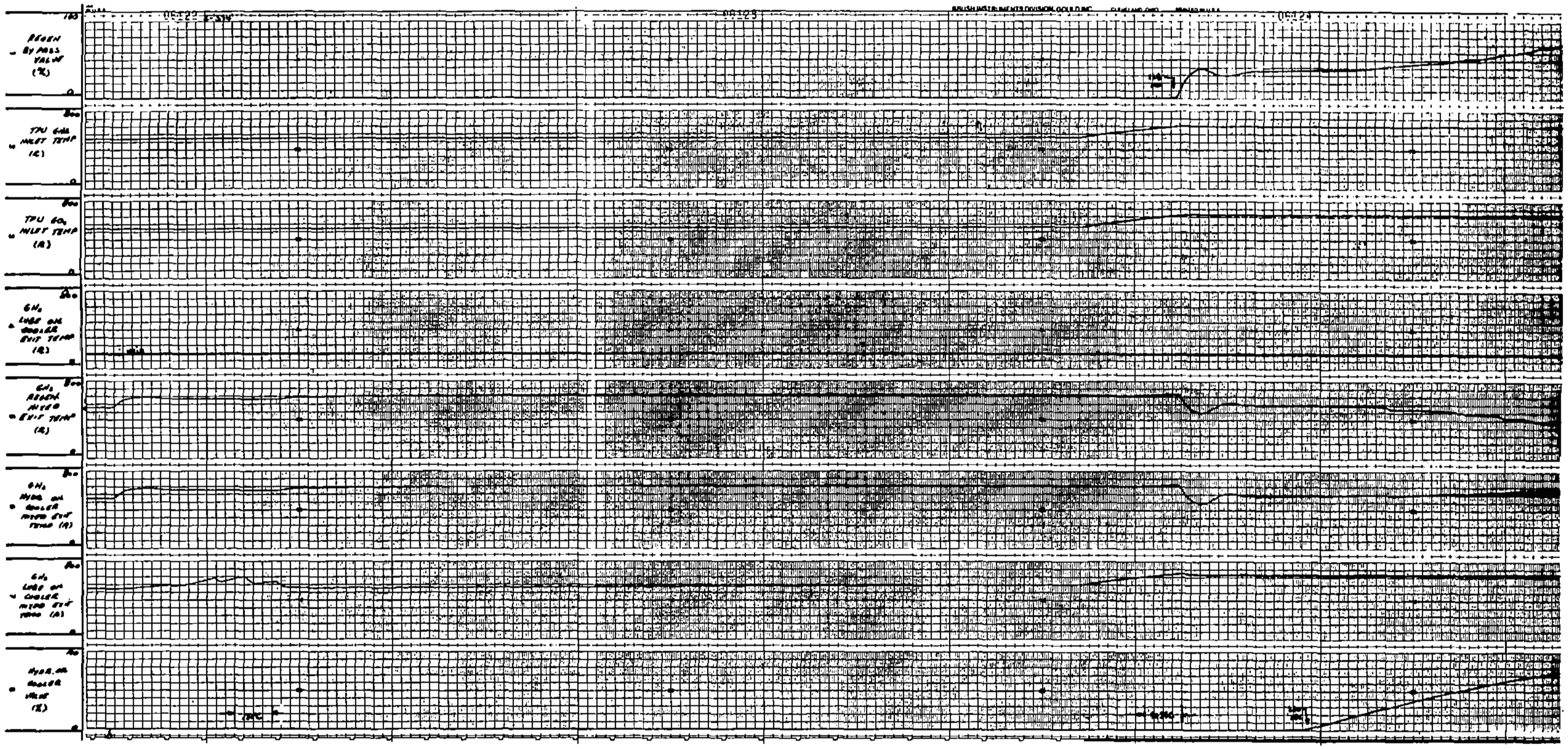


Figure 117. (Continued)

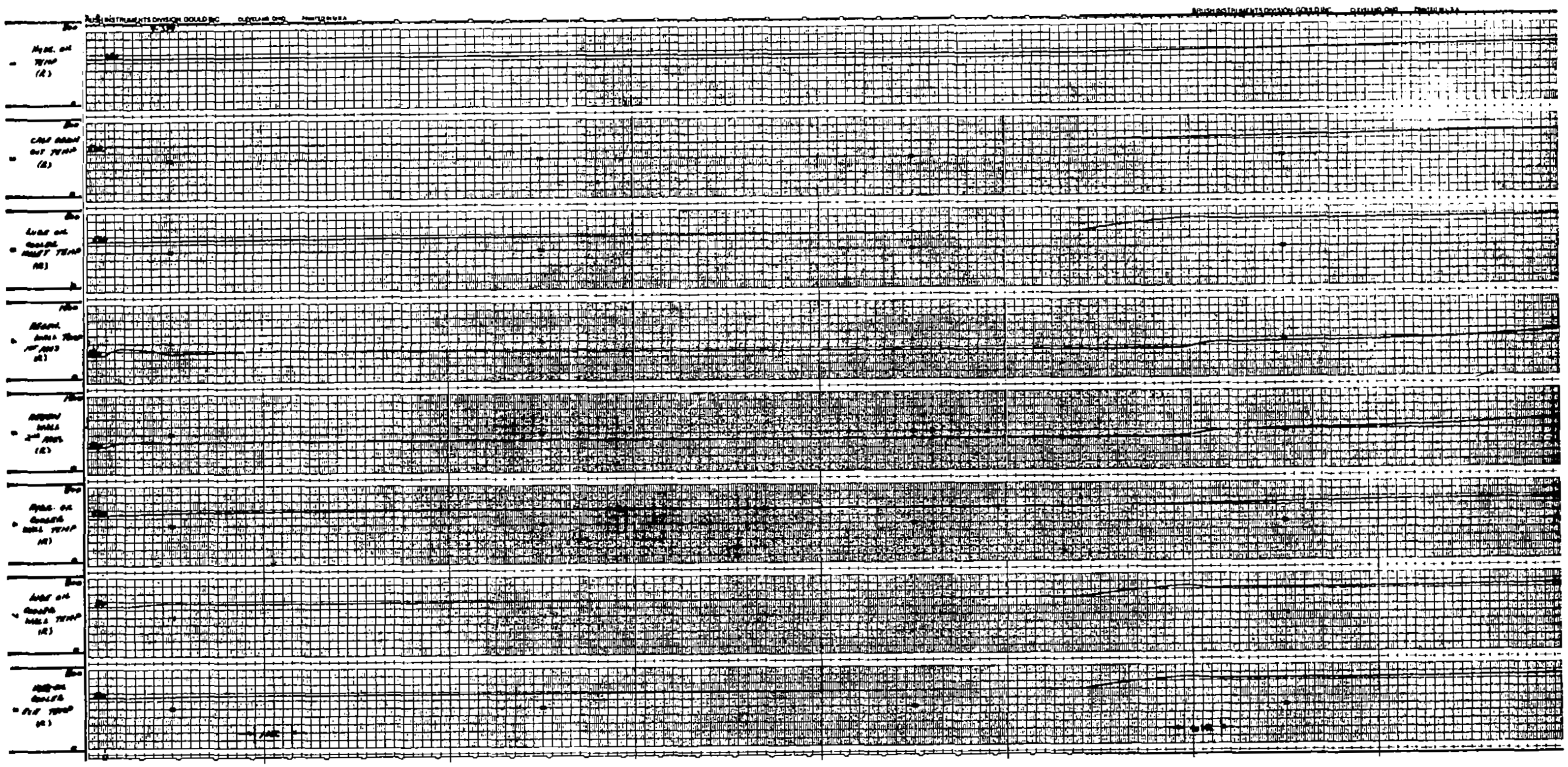


Figure 117. (Continued)

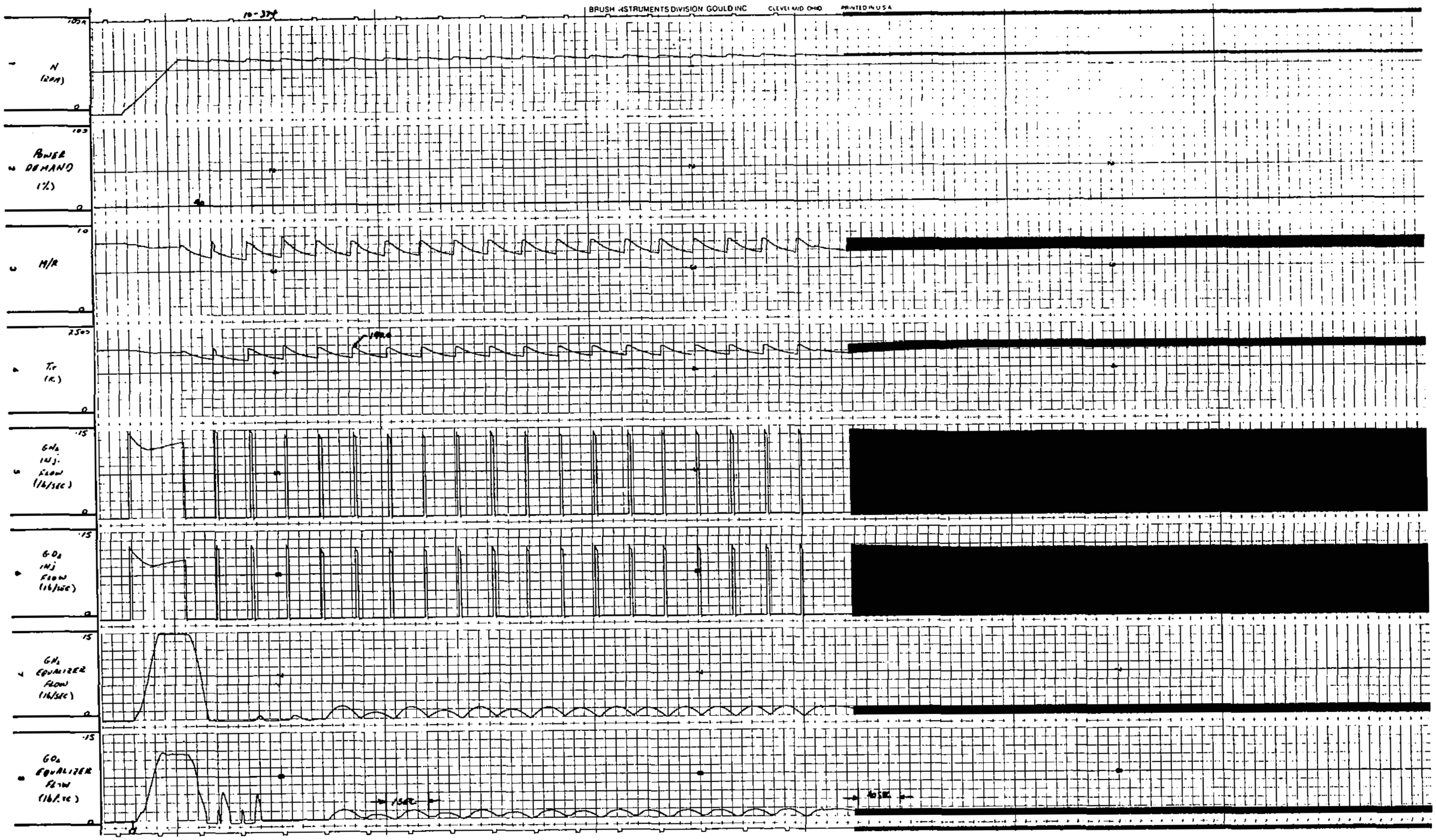


Figure 117. (Concluded)

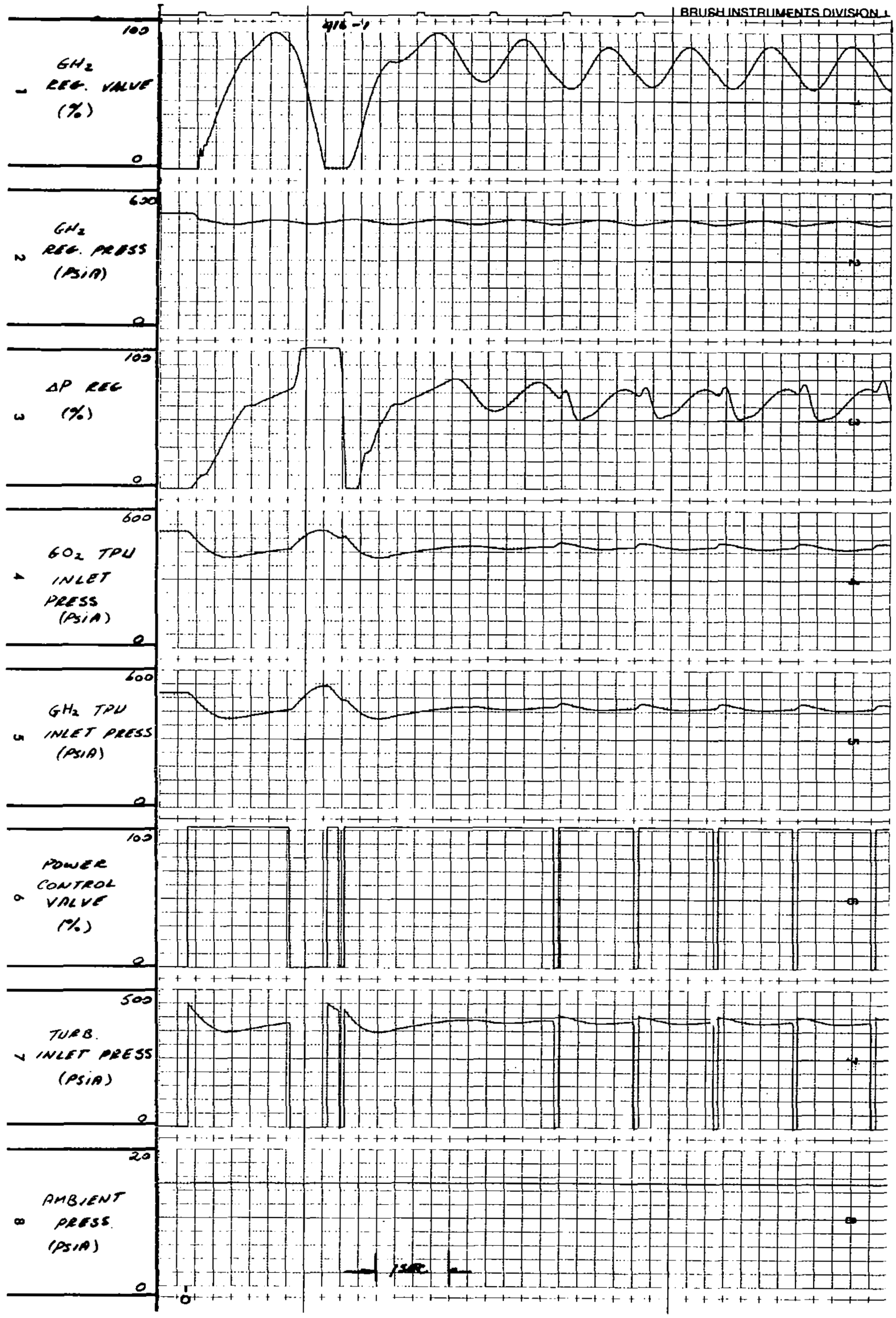


Figure 118. Sea Level Startup, 2 Seconds to Maximum Power--Pulse System

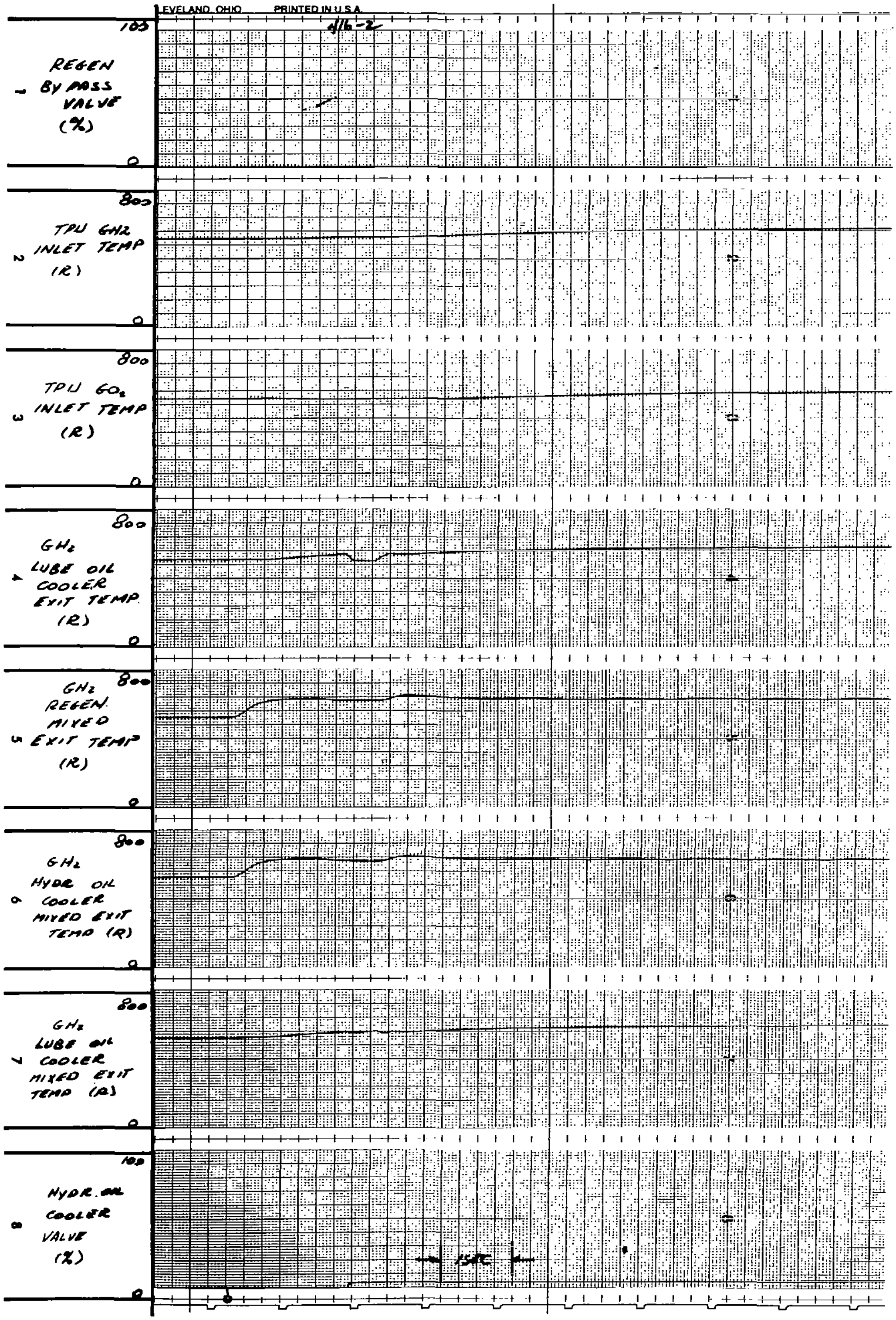


Figure 118. (Continued)

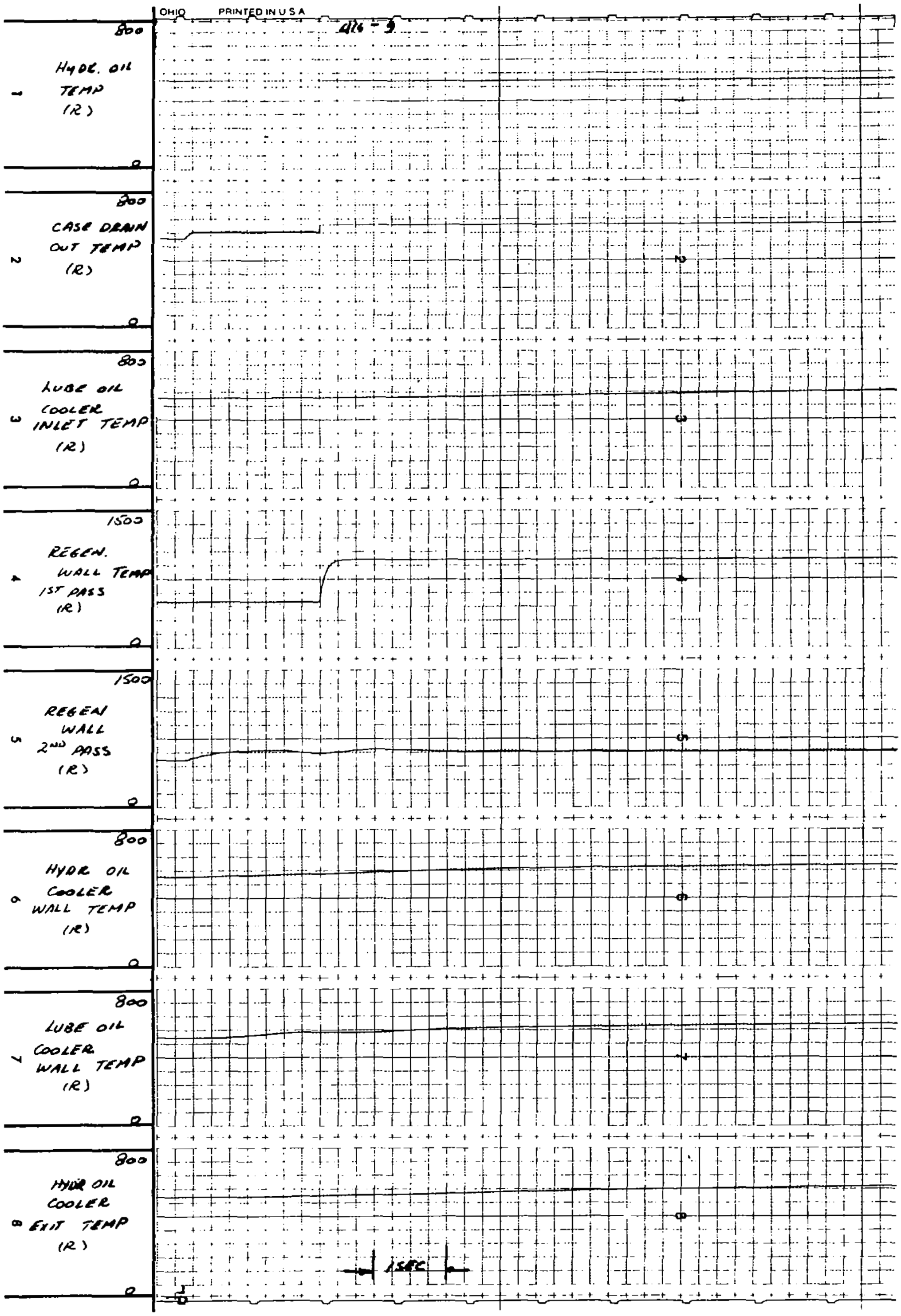


Figure 118. (Continued)

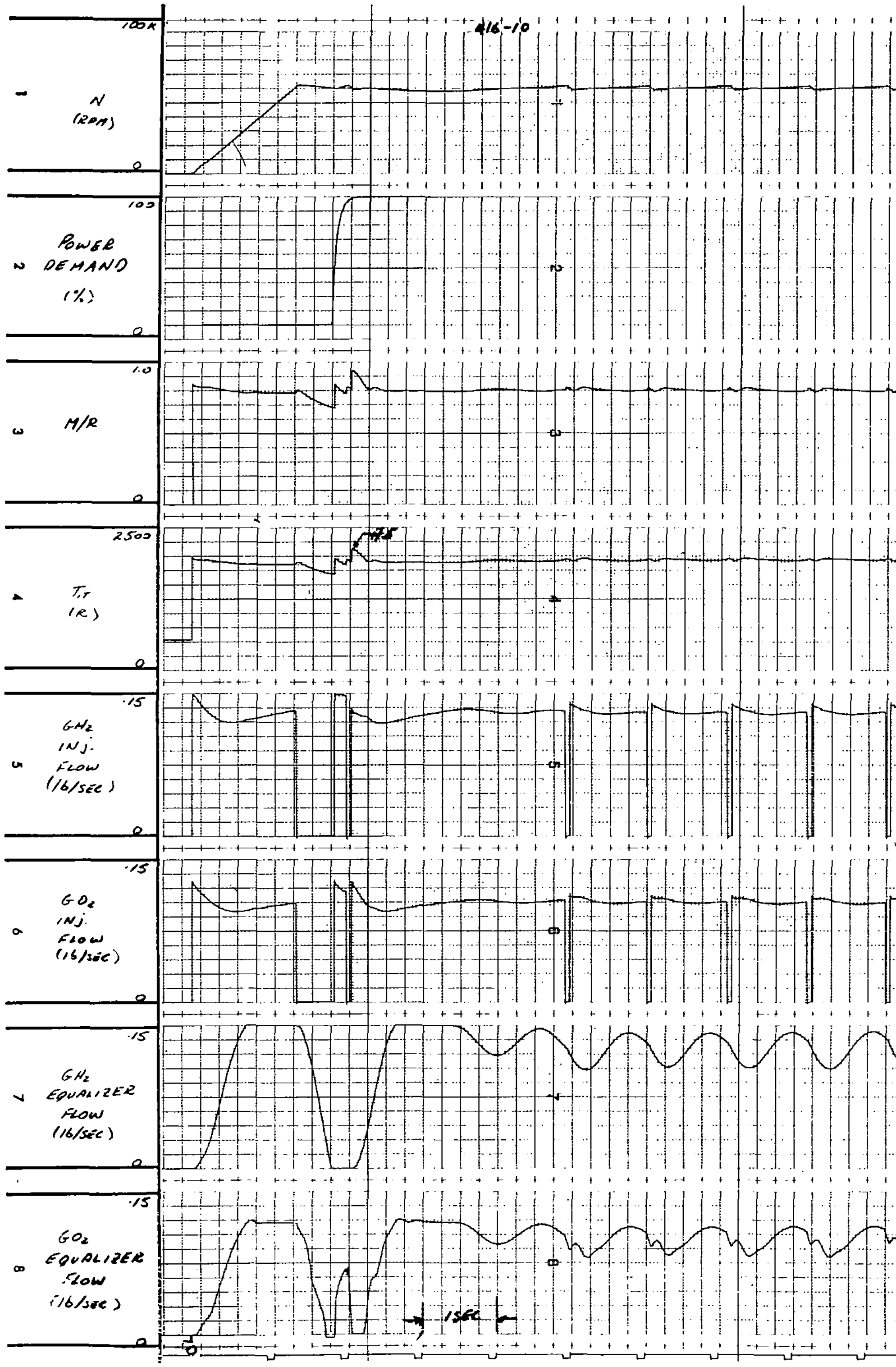


Figure 118. (Concluded)

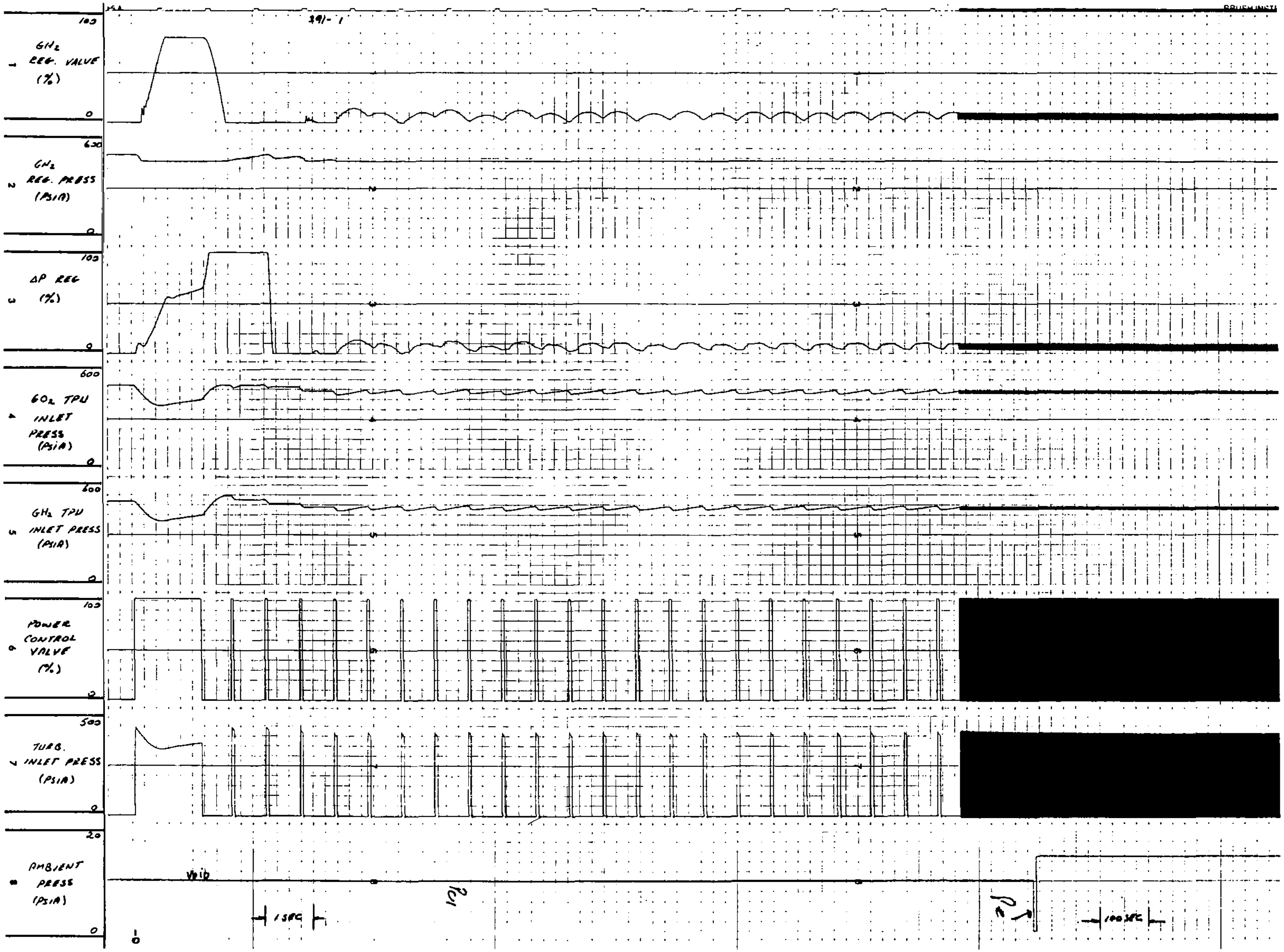


Figure 119. -60 F, Sea Level Startup--Pulse System

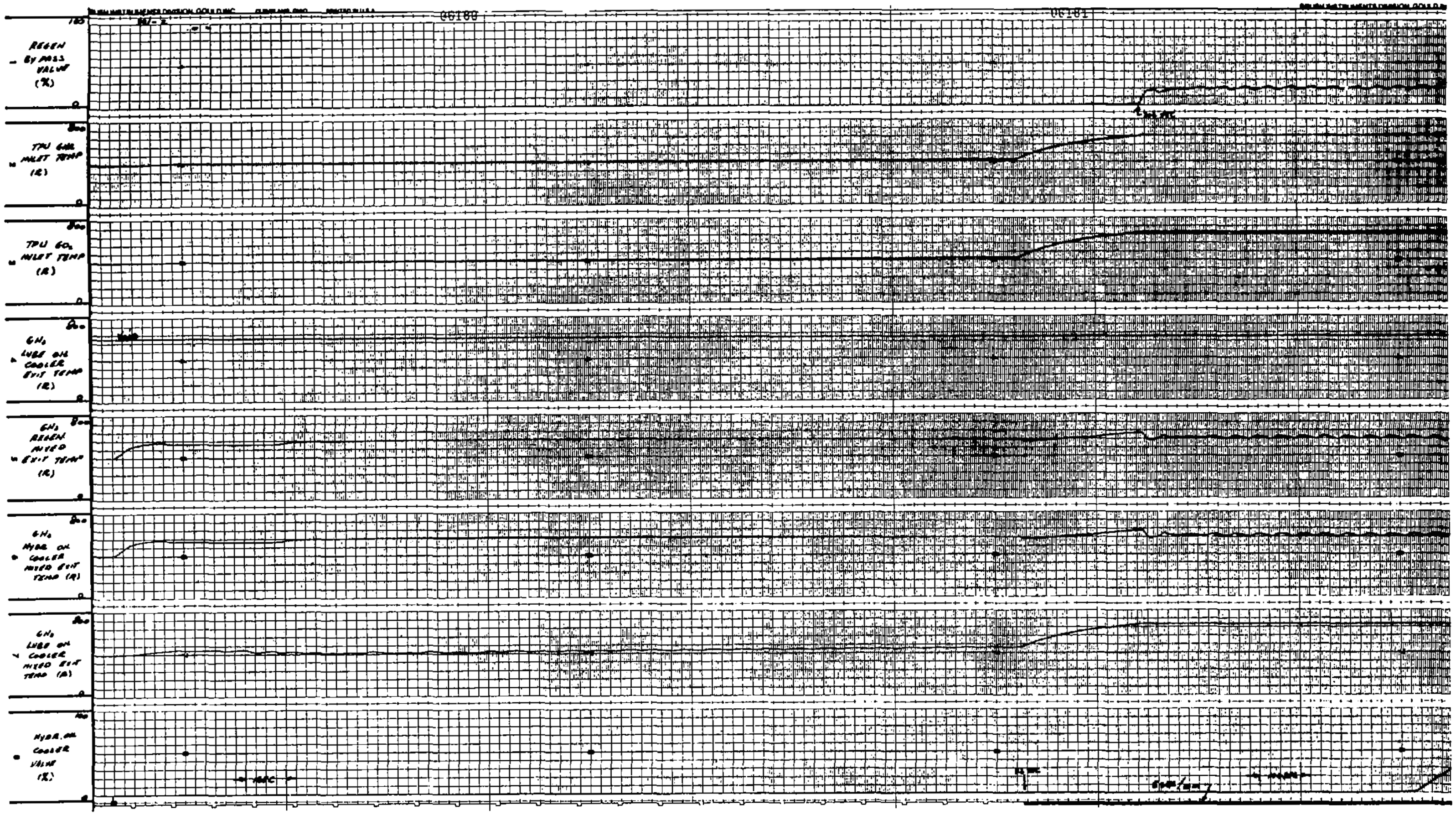


Figure 119. (Continued)

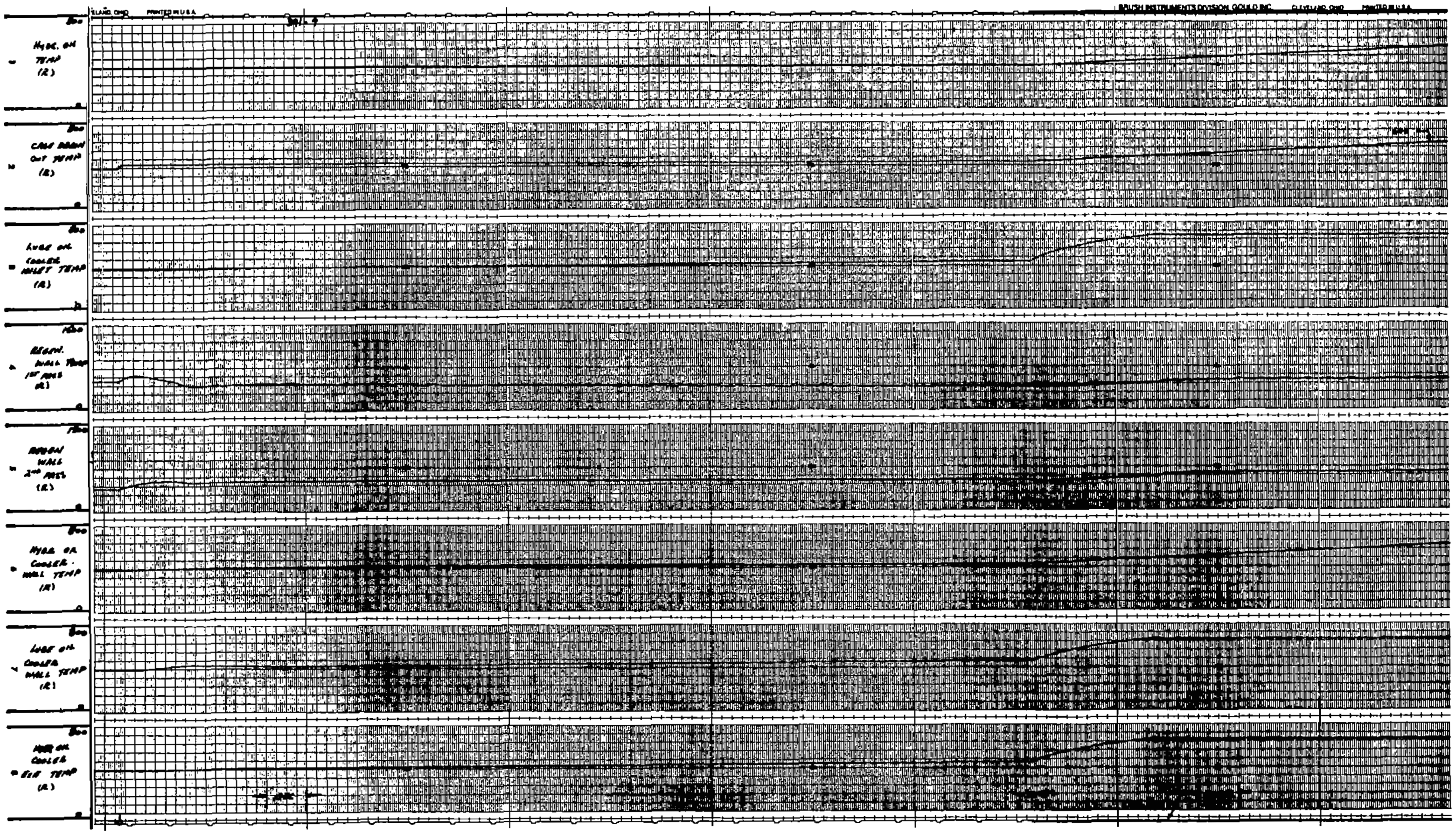


Figure 119. (Continued)

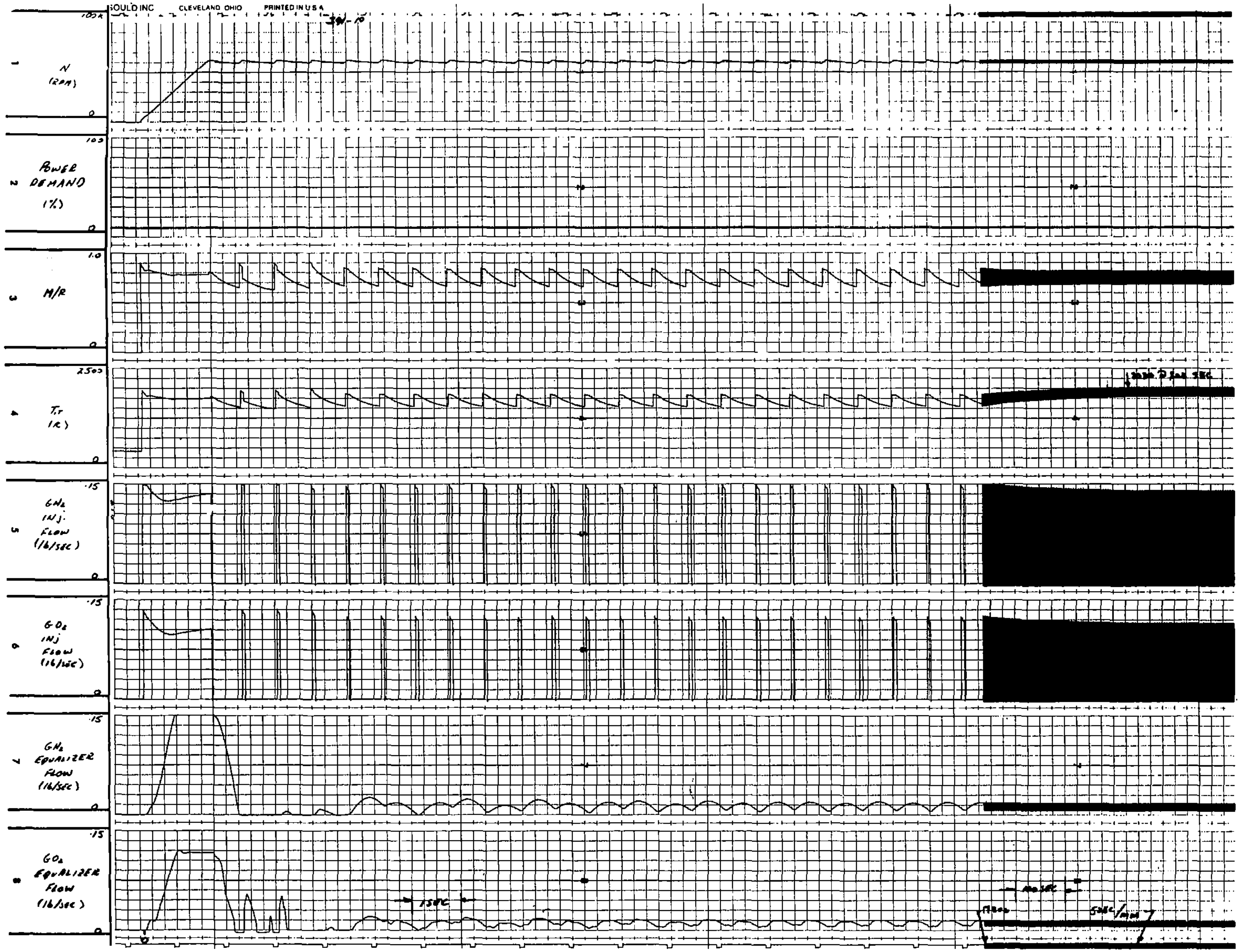


Figure 119. (Concluded)

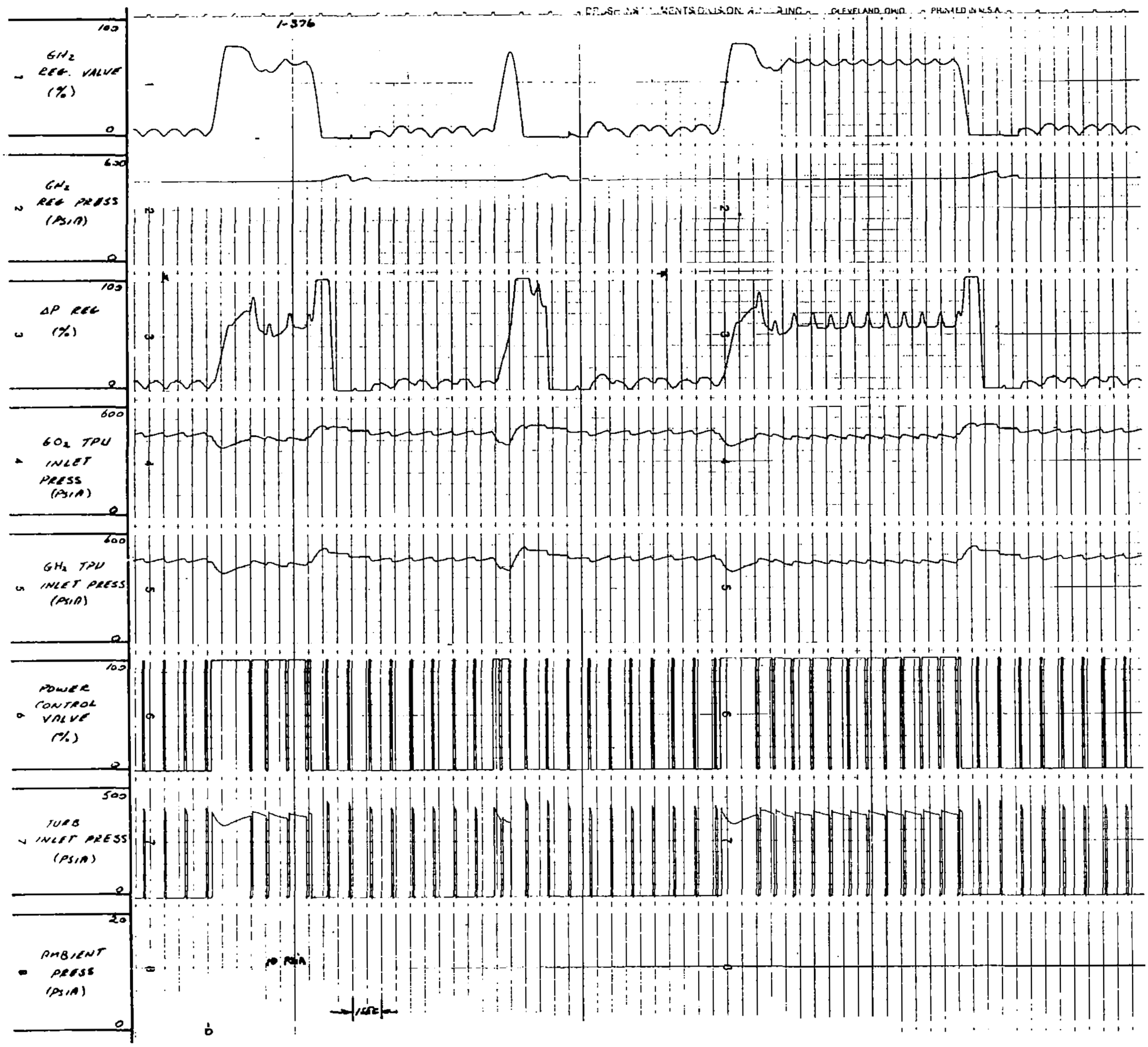


Figure 120. Power Spikes, Idle to Maximum--Pulse System

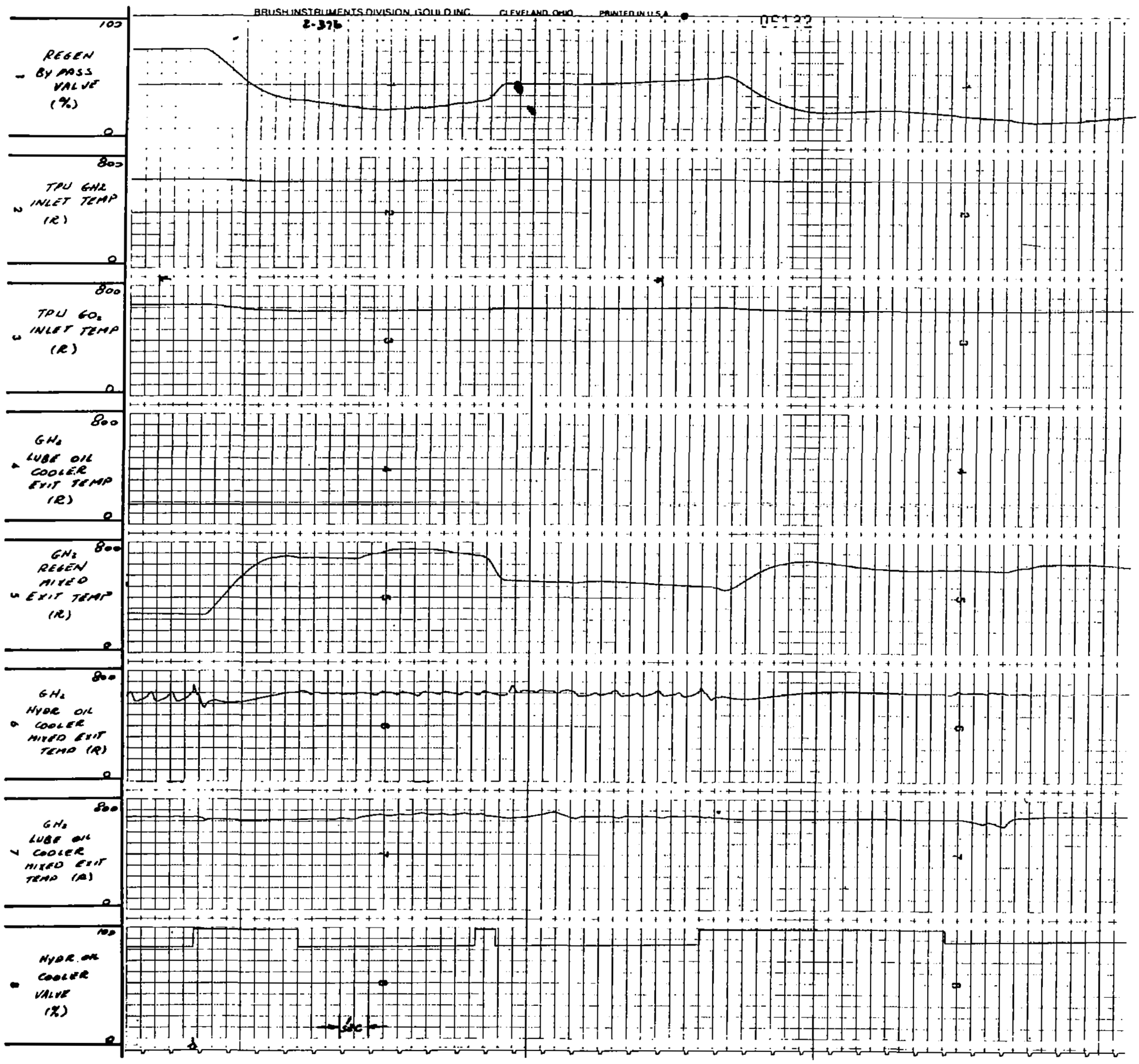


Figure 120. (Continued)

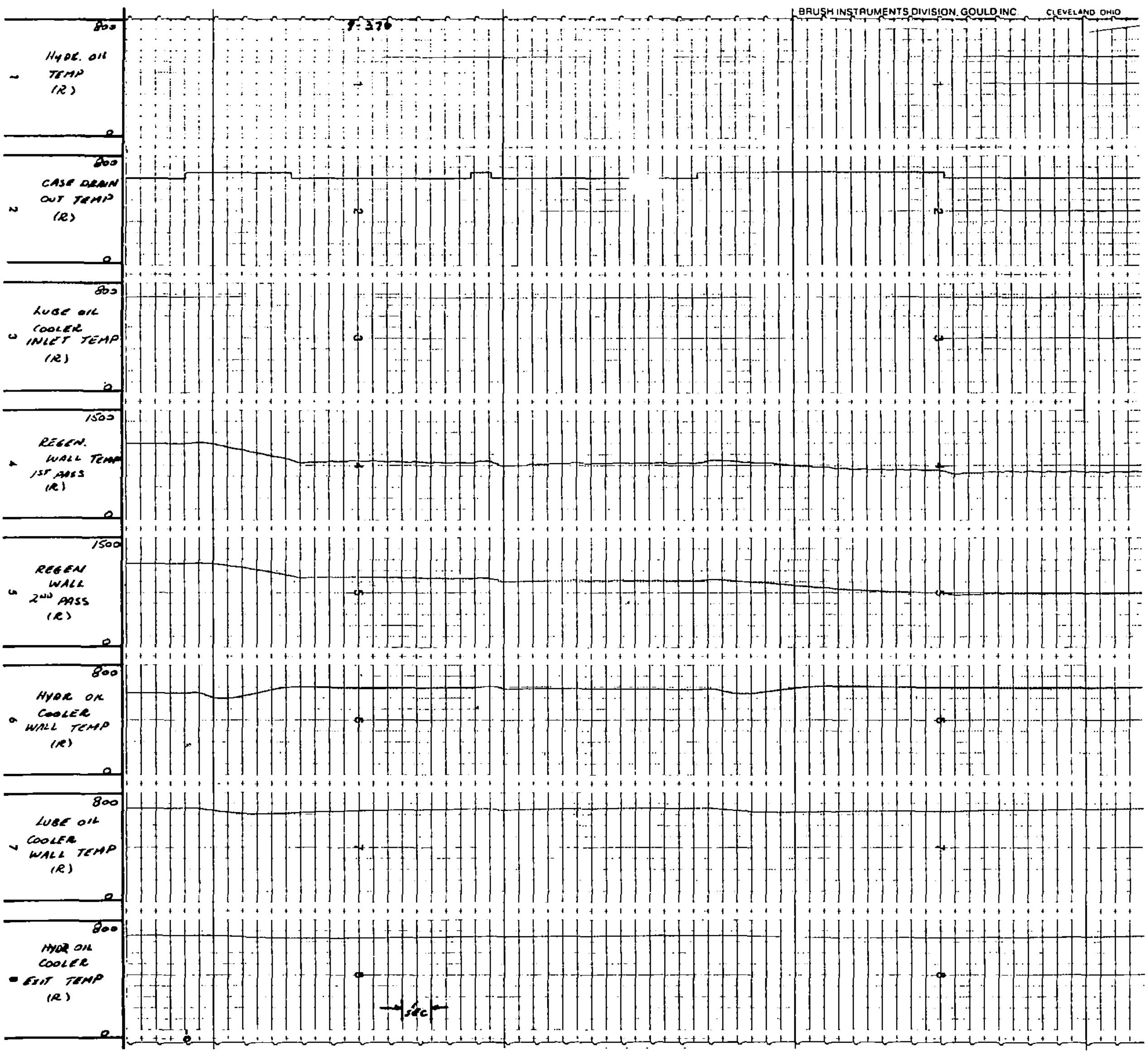


Figure 120. (Continued)

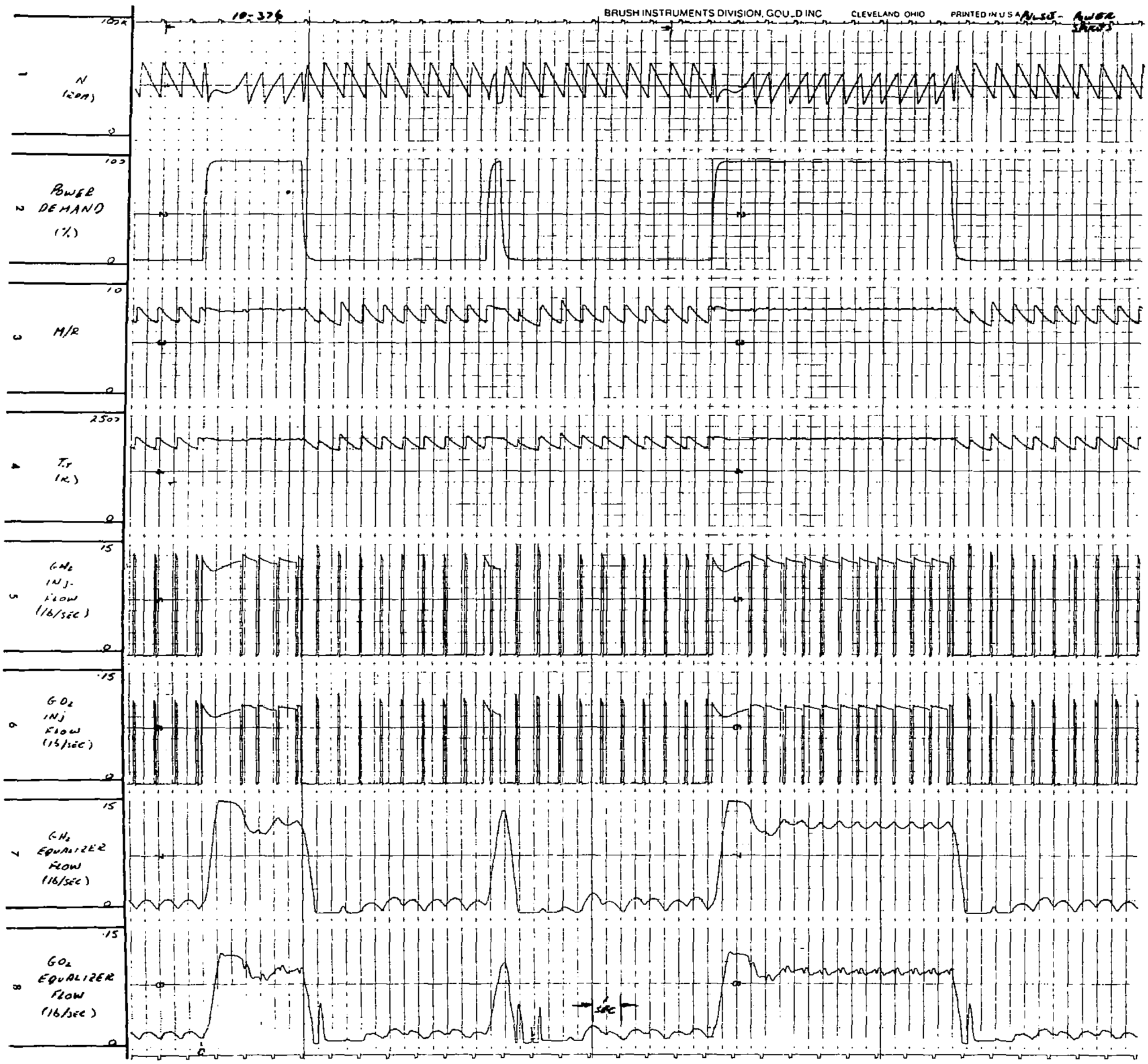


Figure 120. (Concluded)

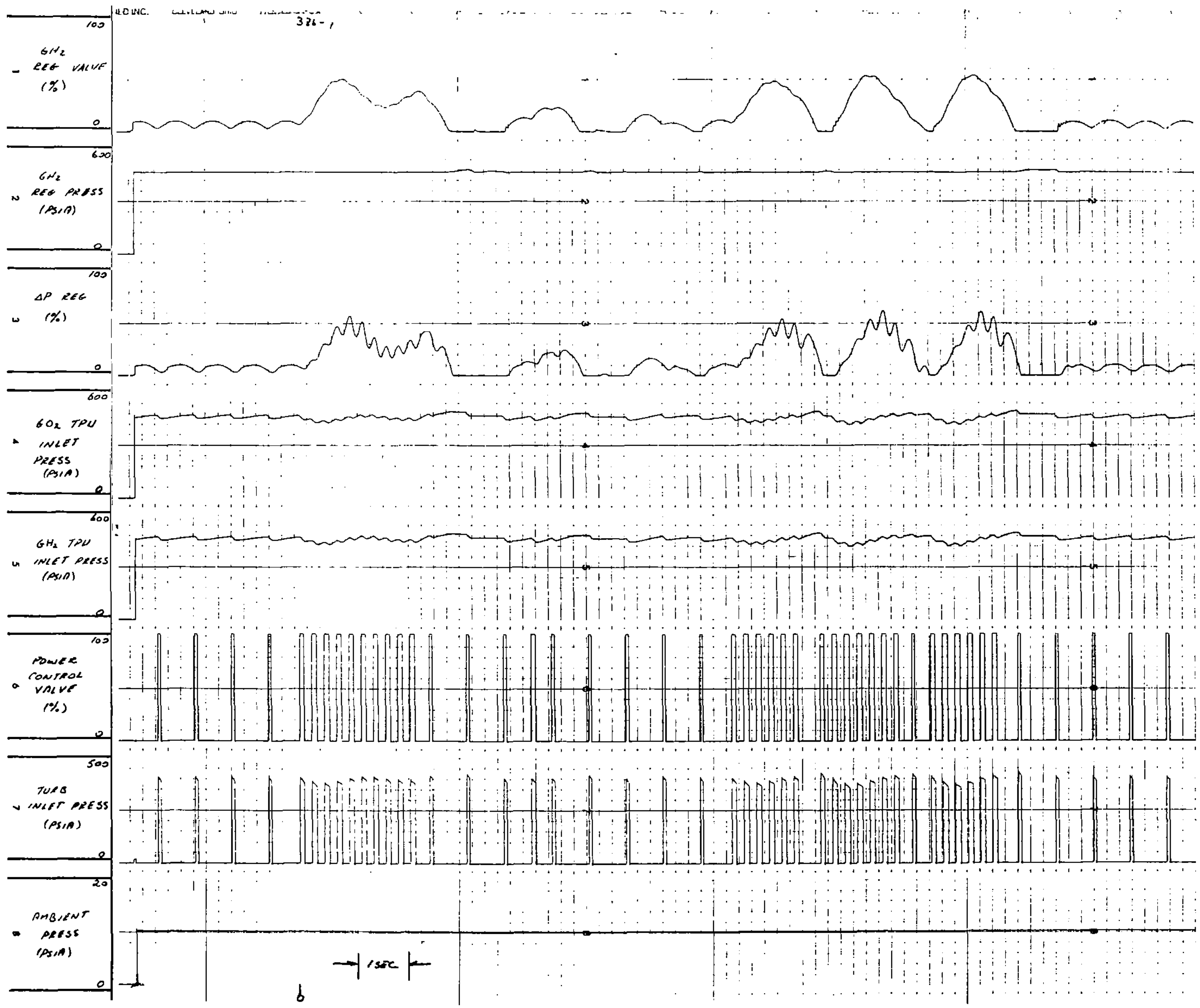


Figure 121. Power Spikes, Idle to 200 Horsepower--Pulse System

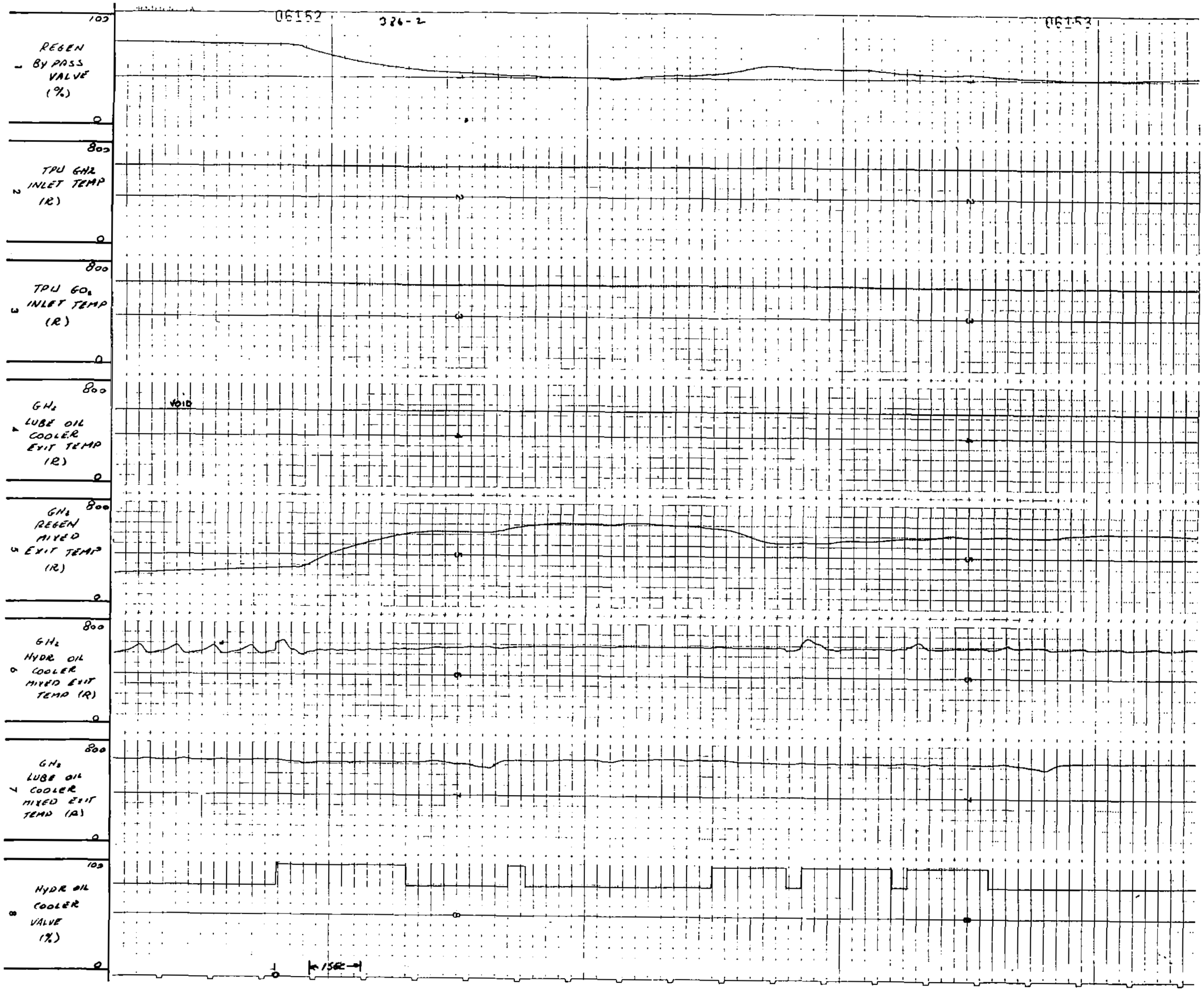


Figure 121. (Continued)

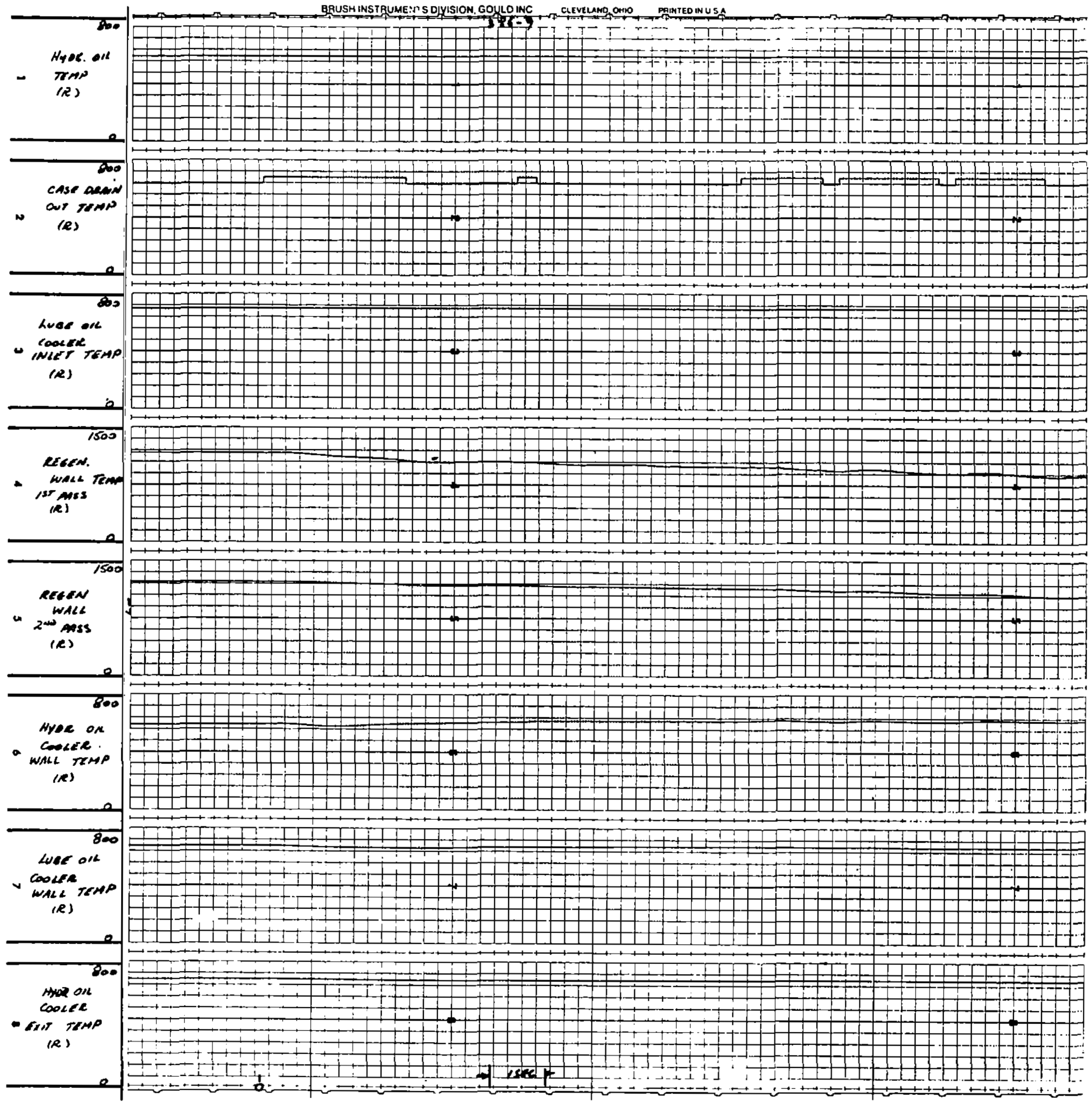
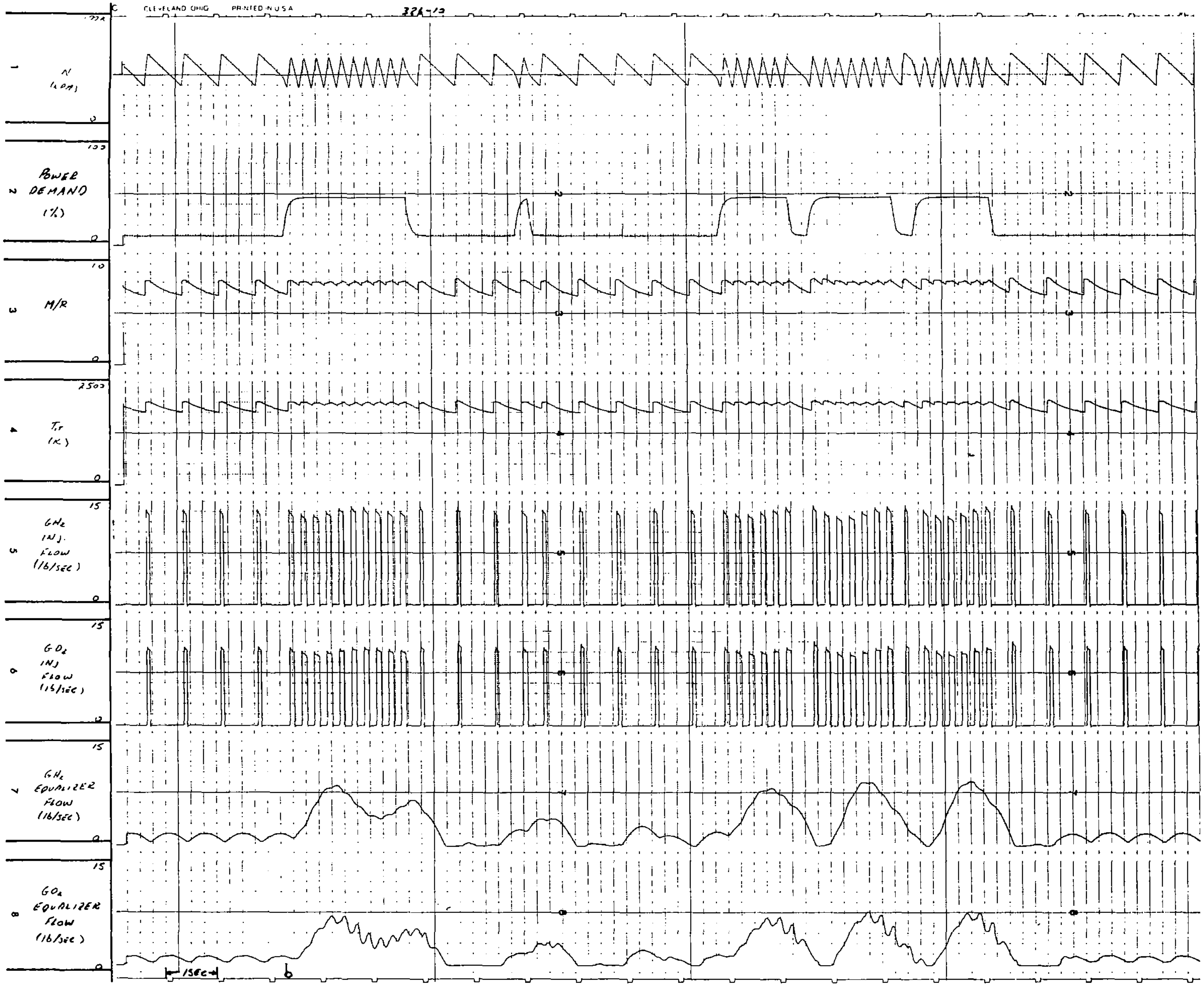


Figure 121. (Continued)



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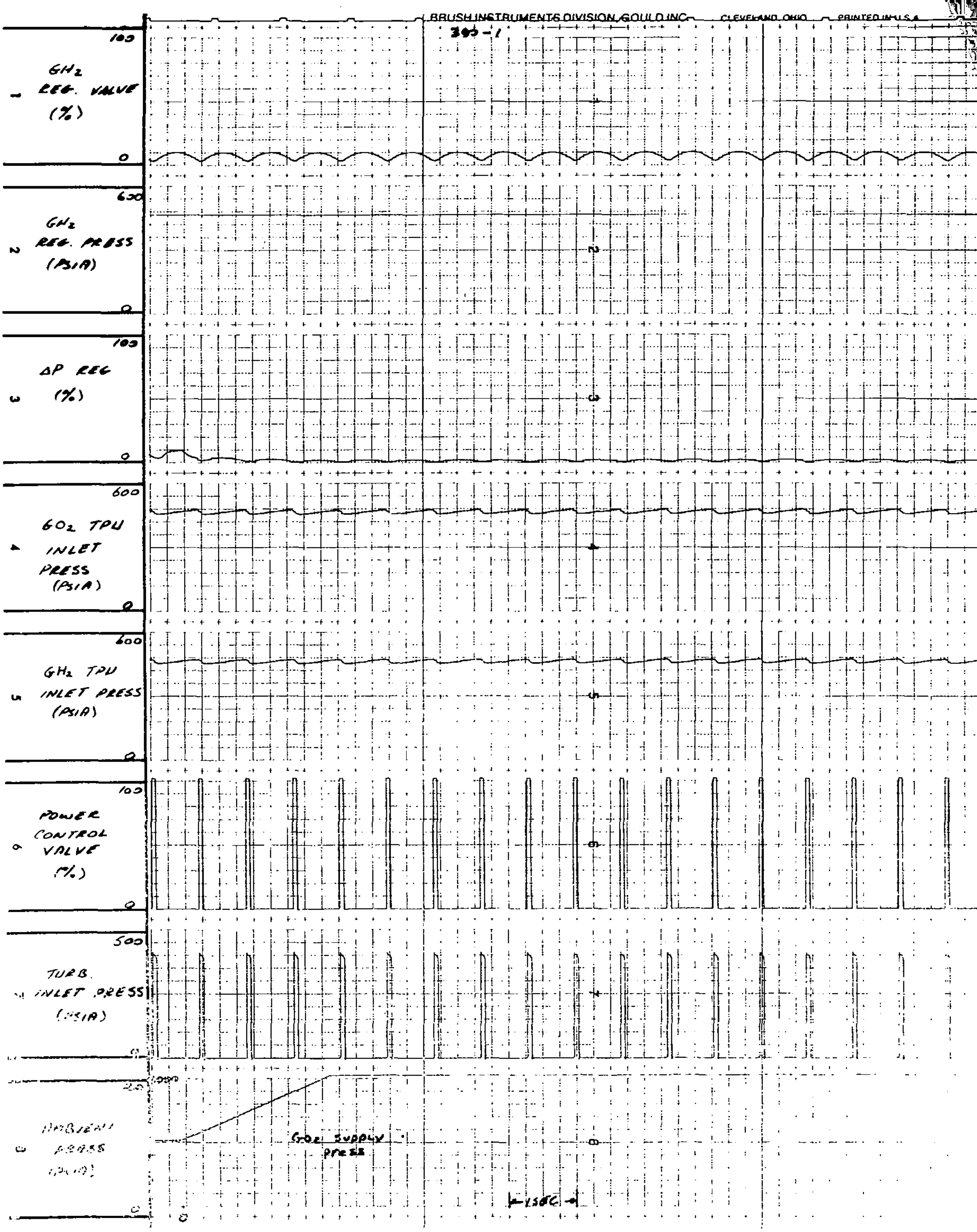


Figure 122. Response to ACS GO_2 Tank Pumpup--Pulse System

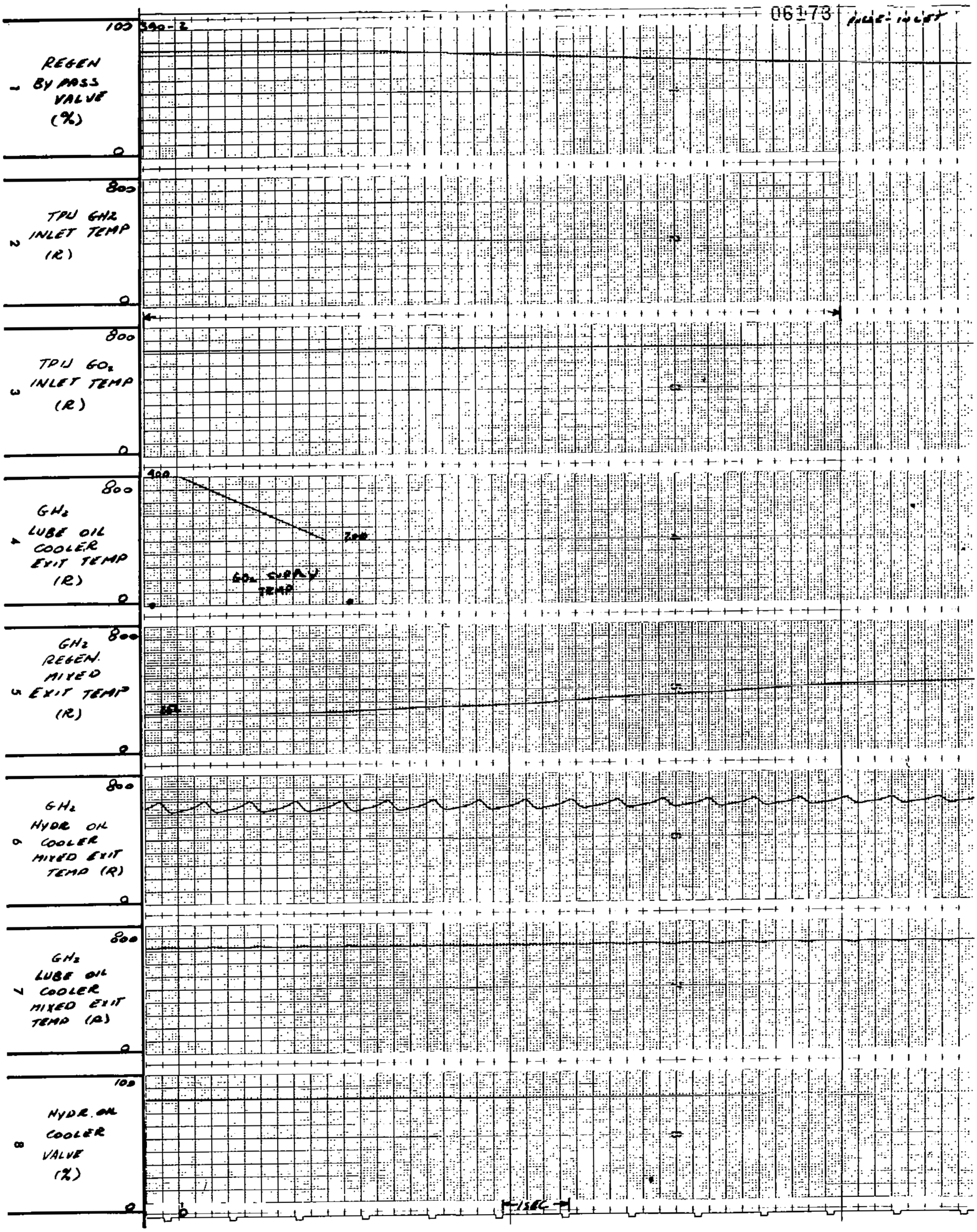


Figure 122. (Continued)

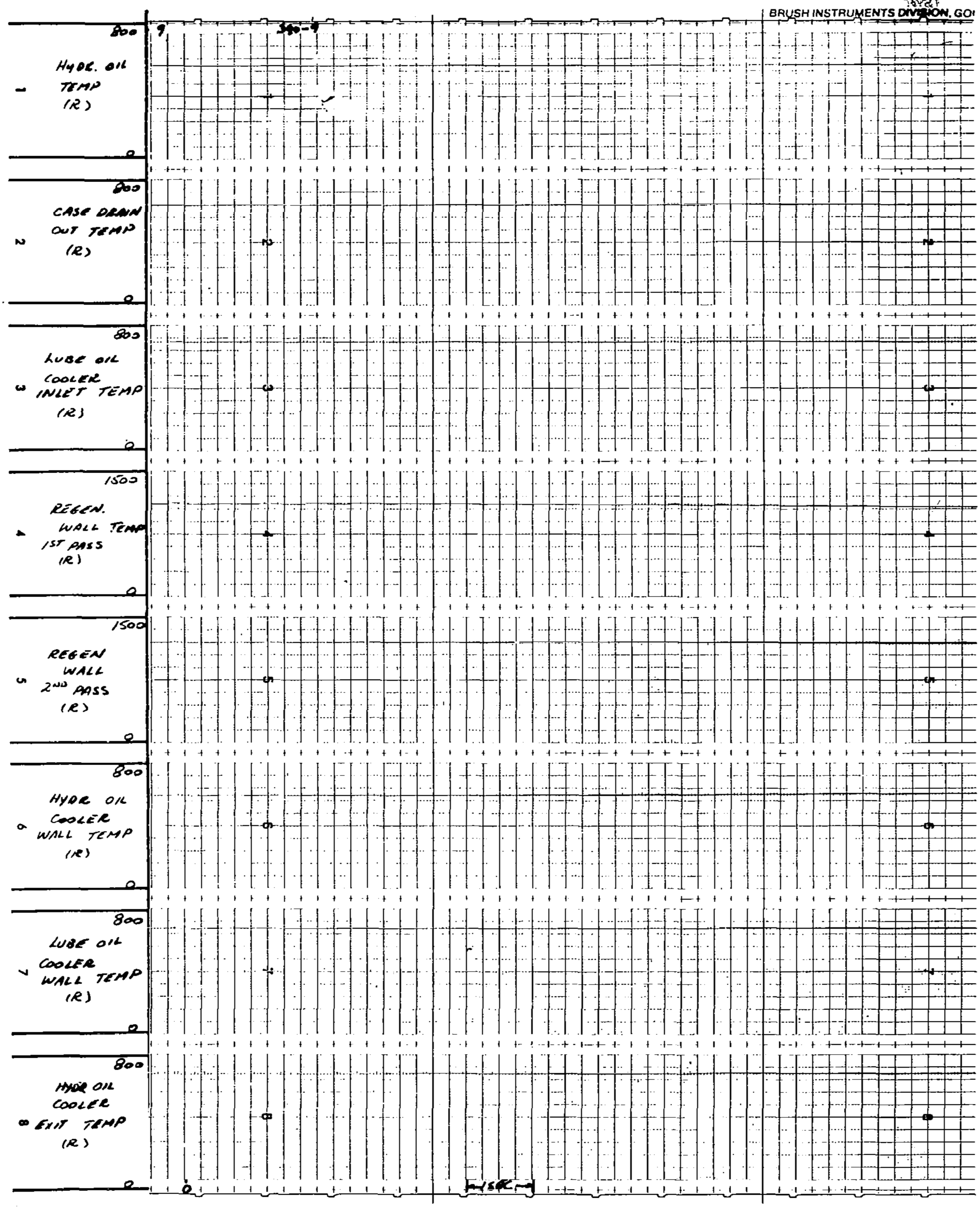
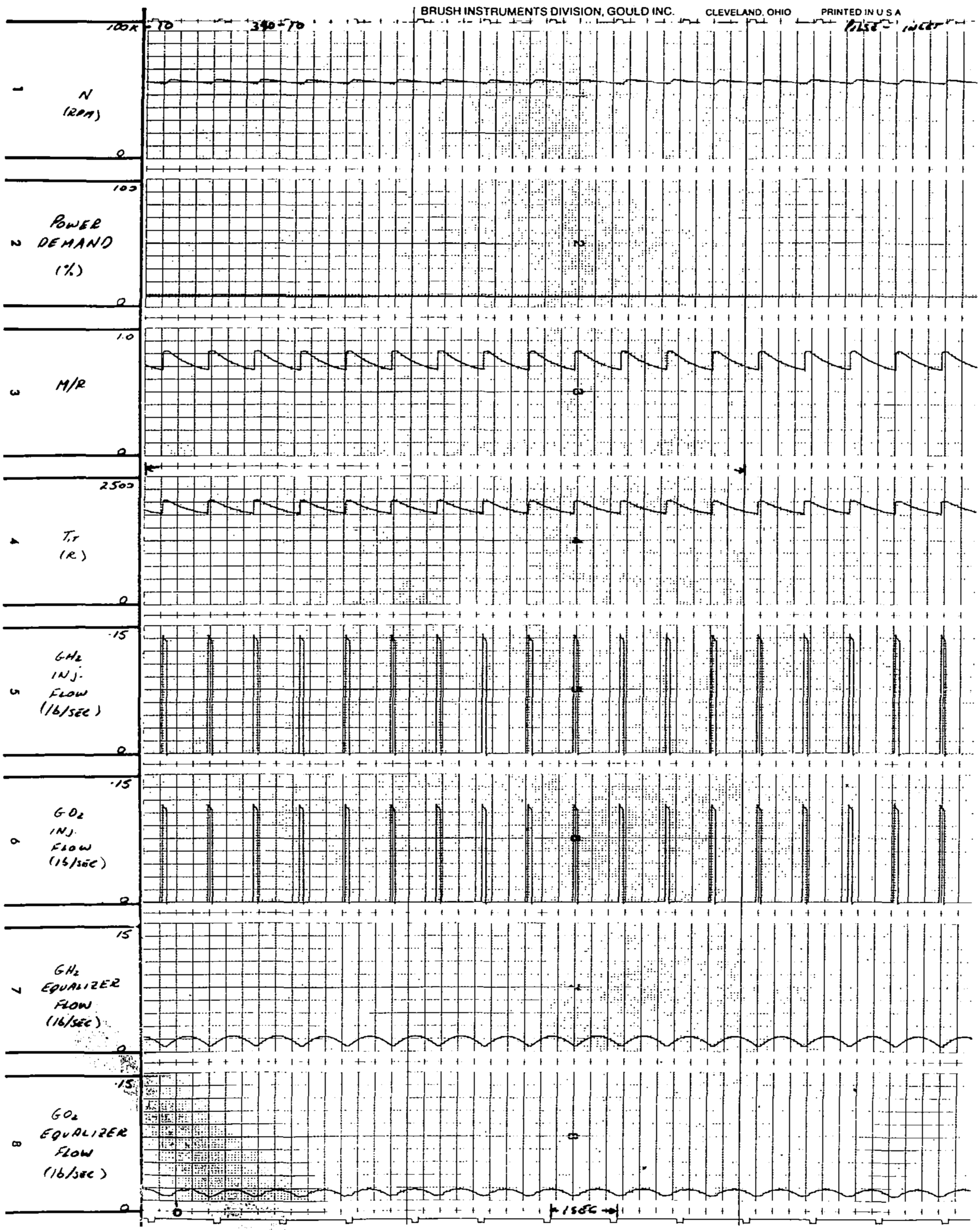


Figure 122. (Continued)



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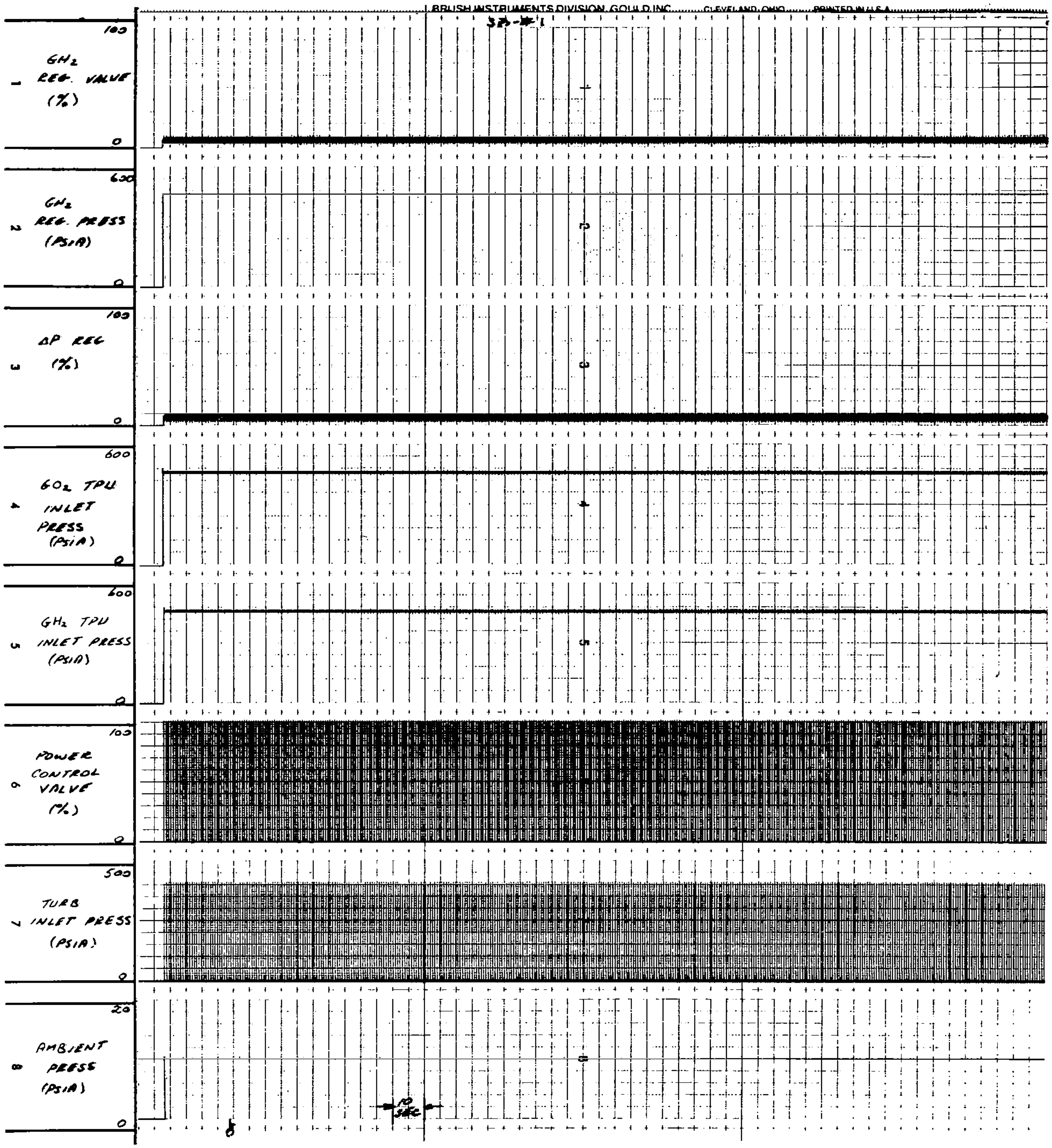


Figure 123. Response to ACS GH_2 Temperature Change--Pulse System

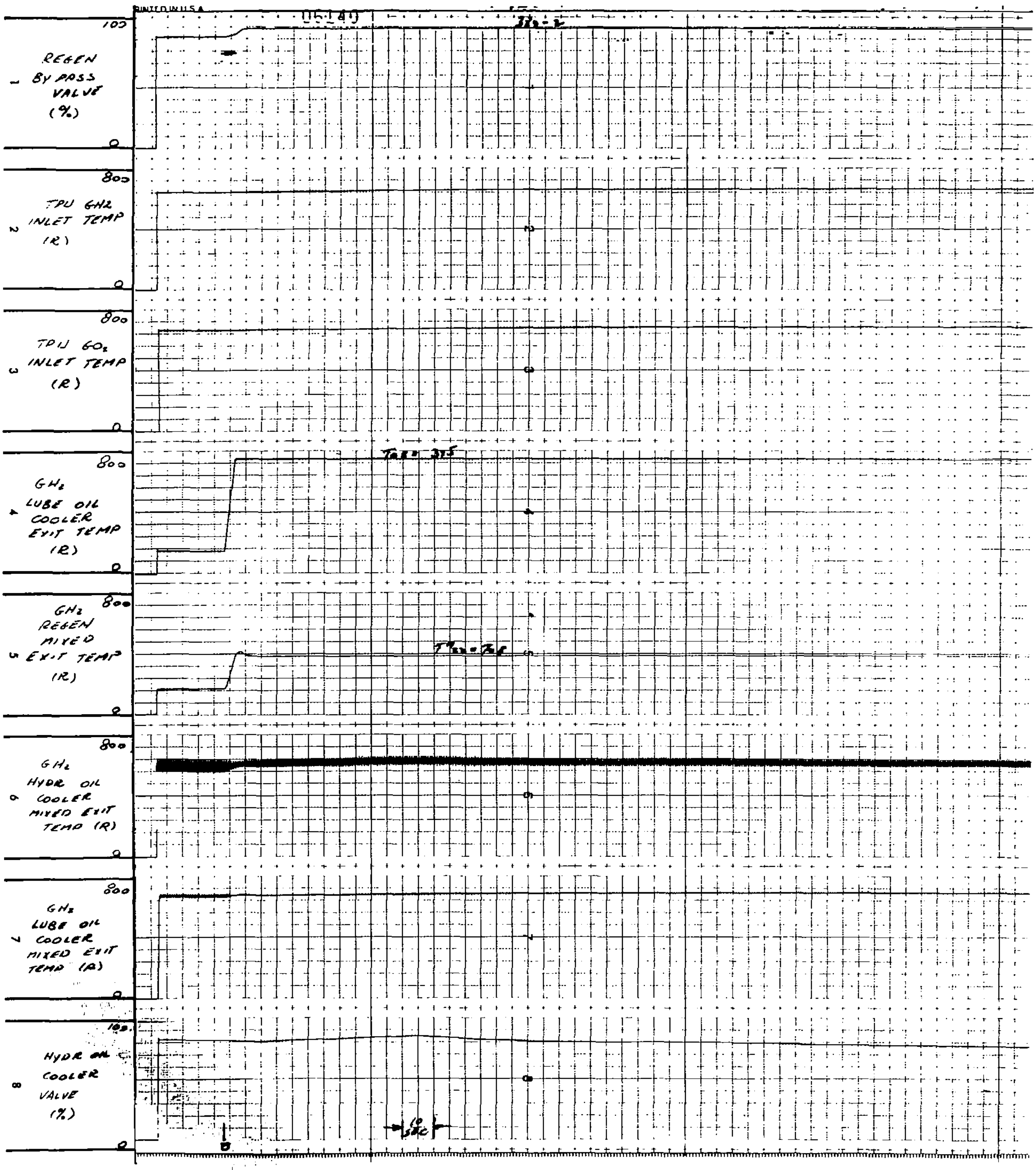


Figure 123. (Continued)

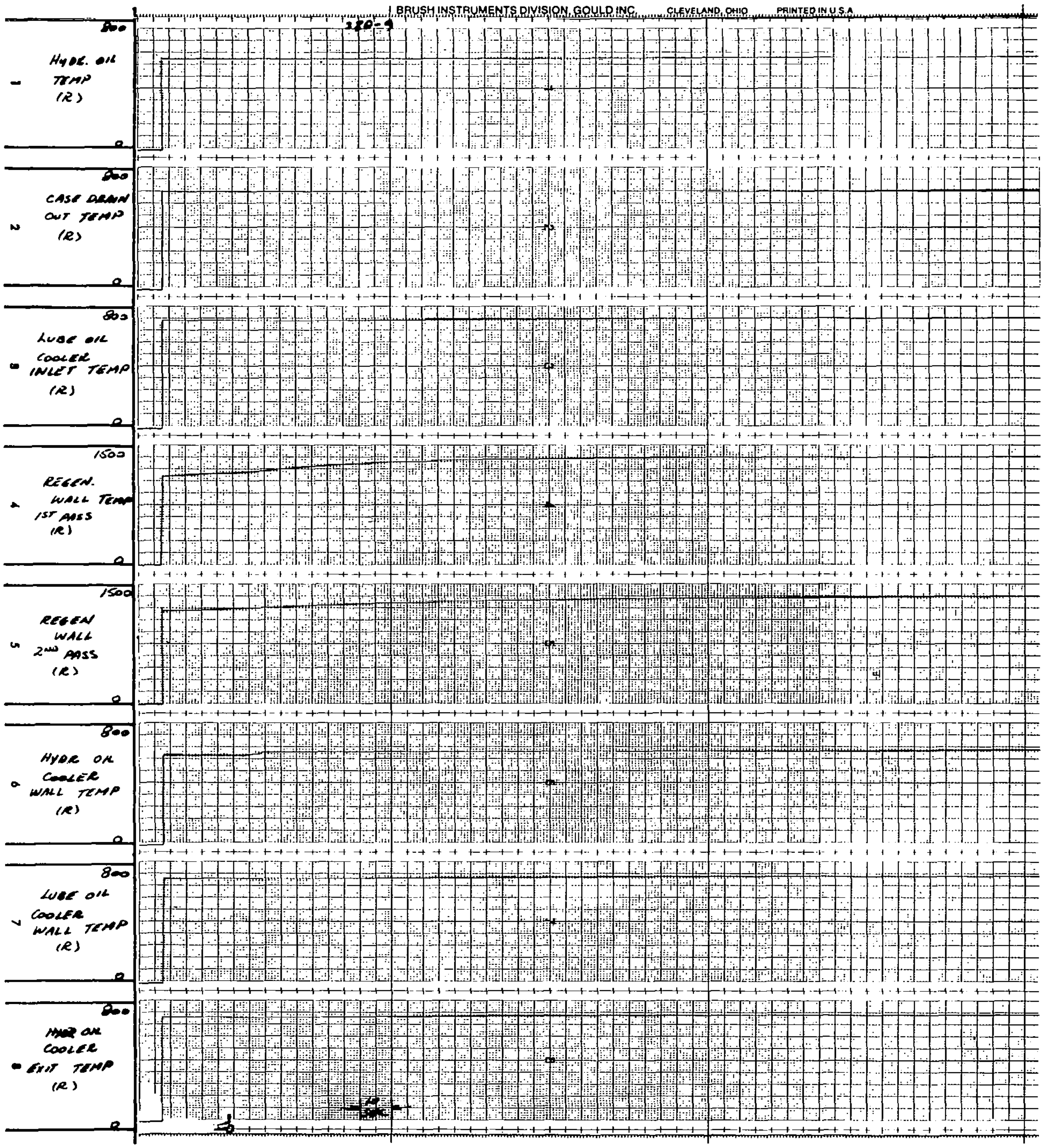


Figure 123. (Continued)

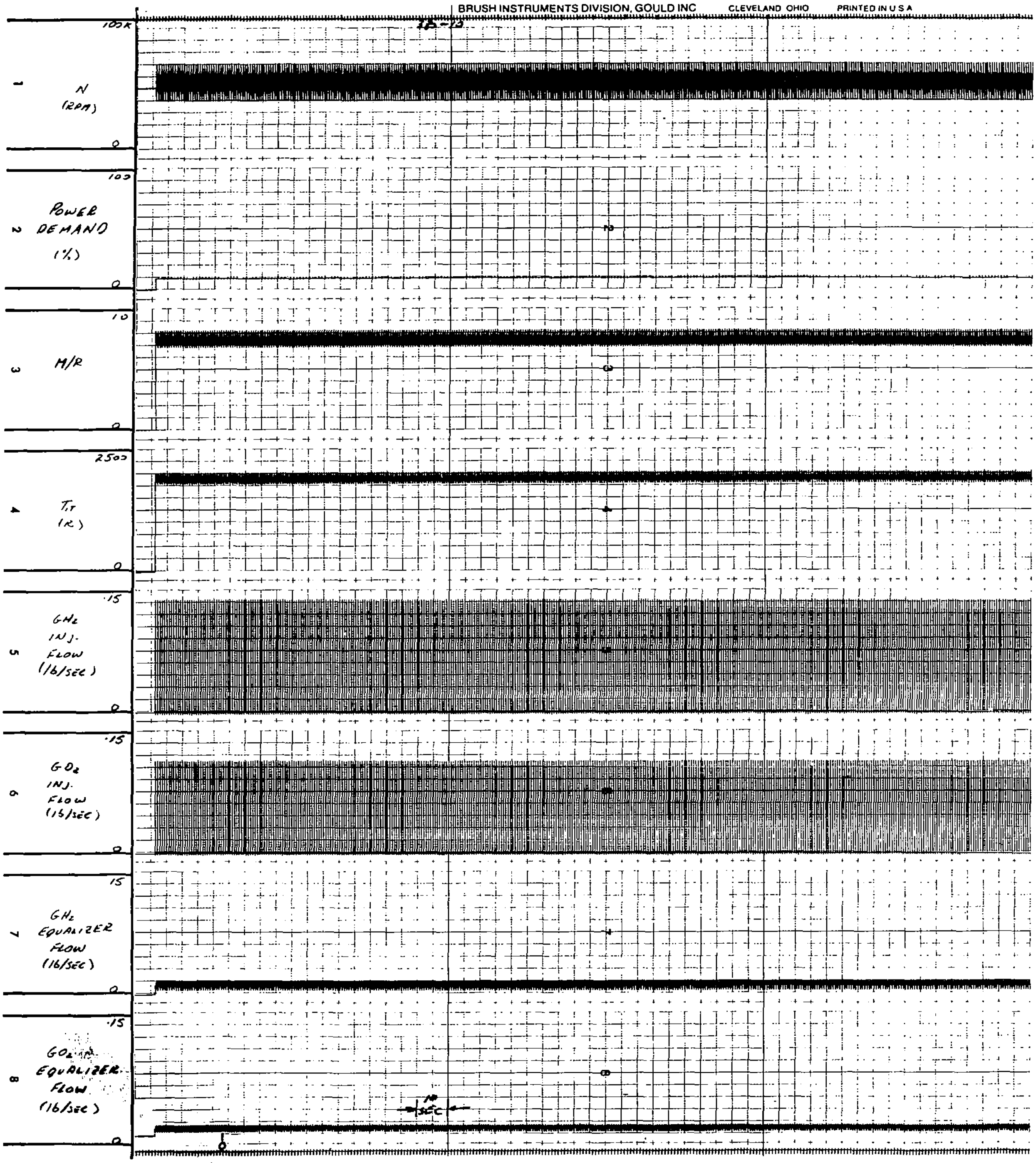


Figure 123. (Concluded)

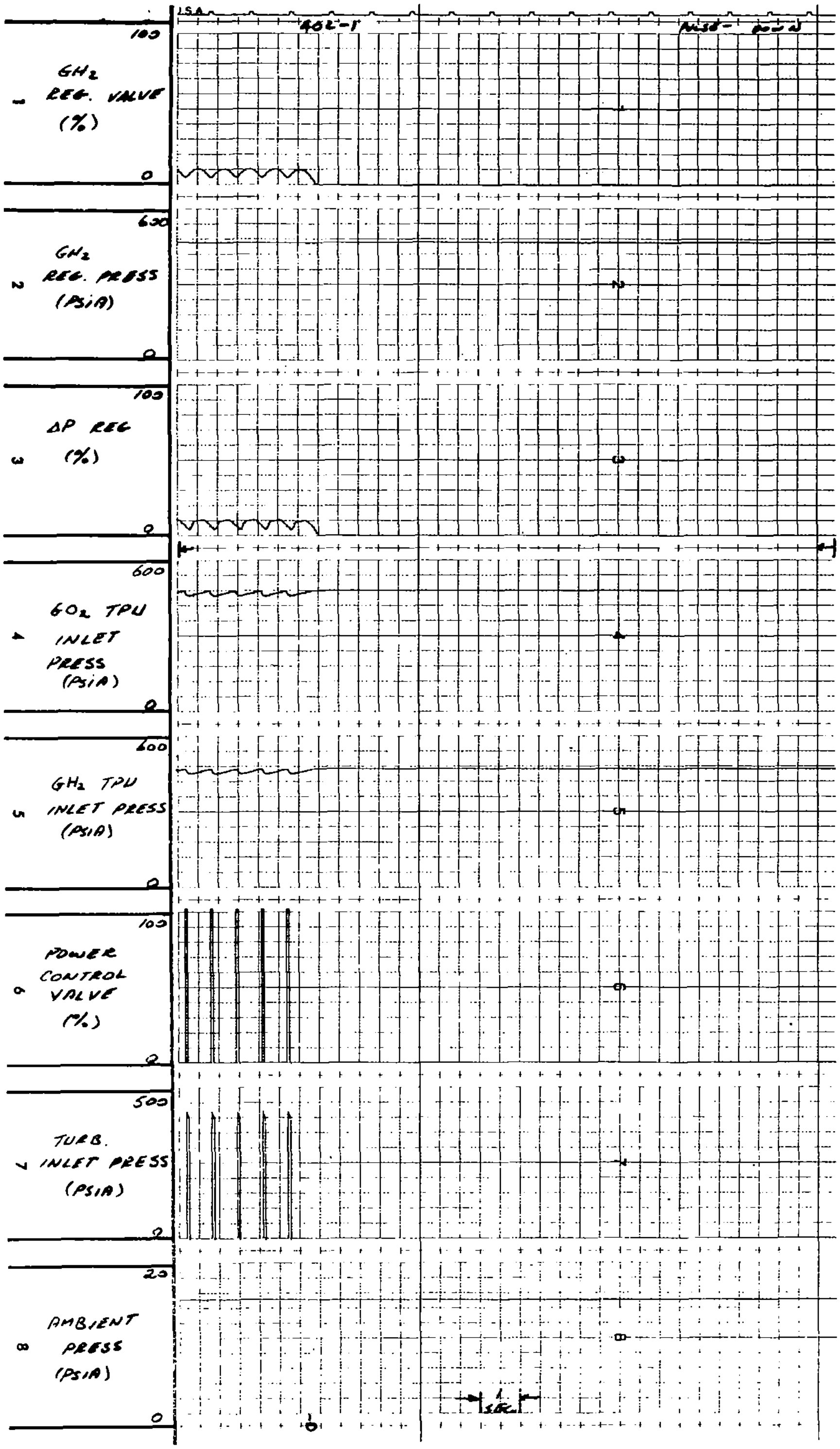


Figure 124. Sea Level Shutdown--Pulse System

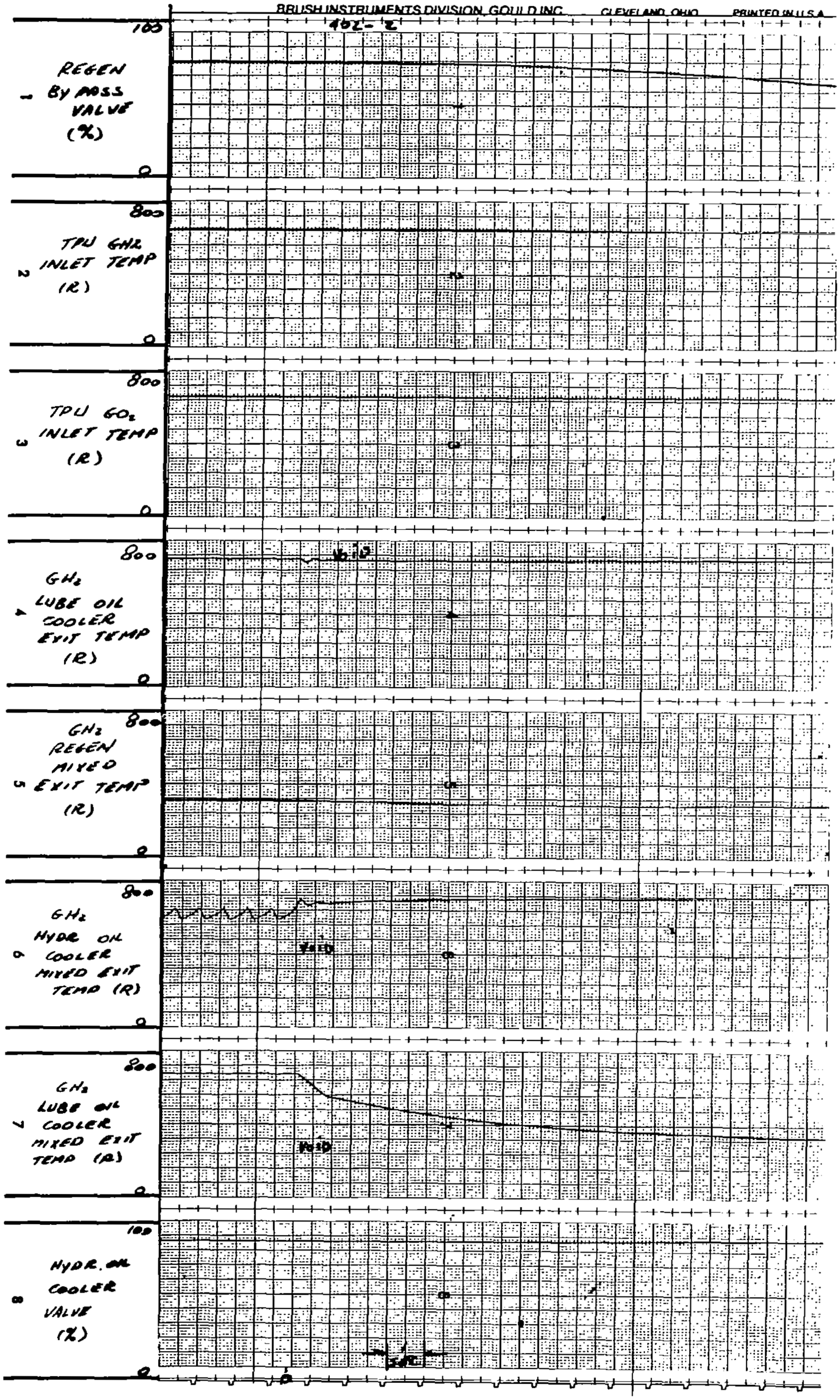


Figure 124. (Continued)

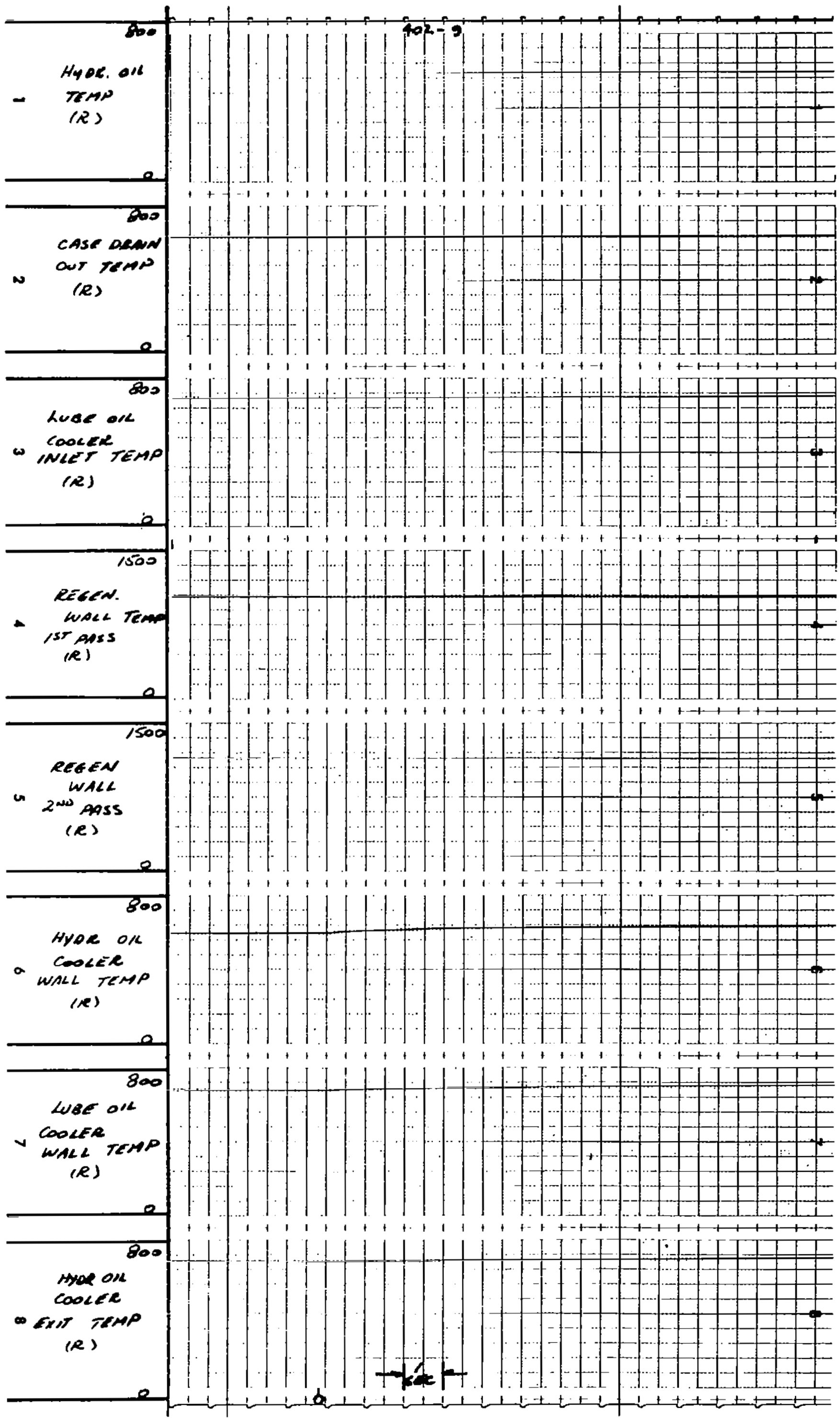
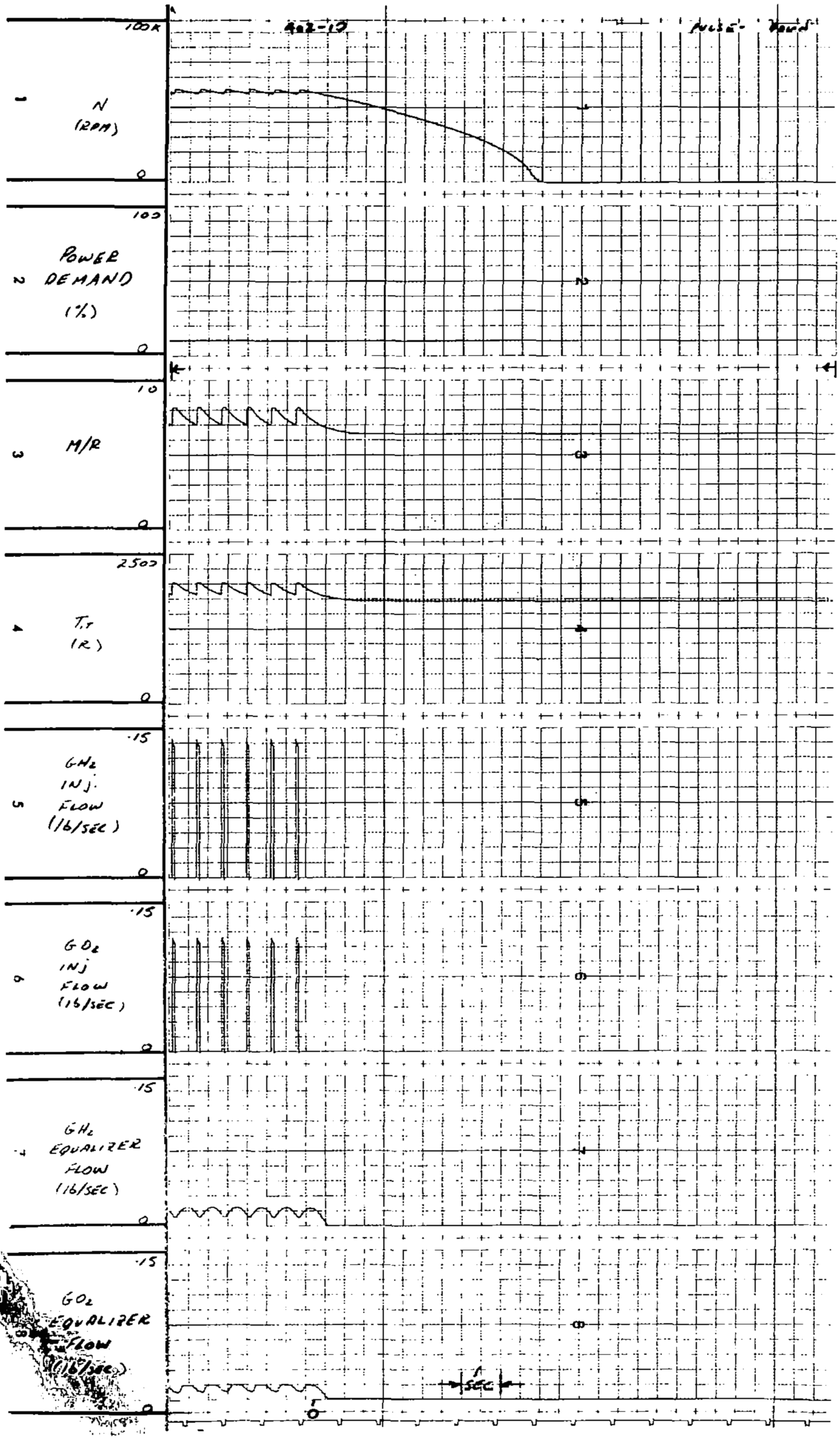


Figure 124. (Continued)



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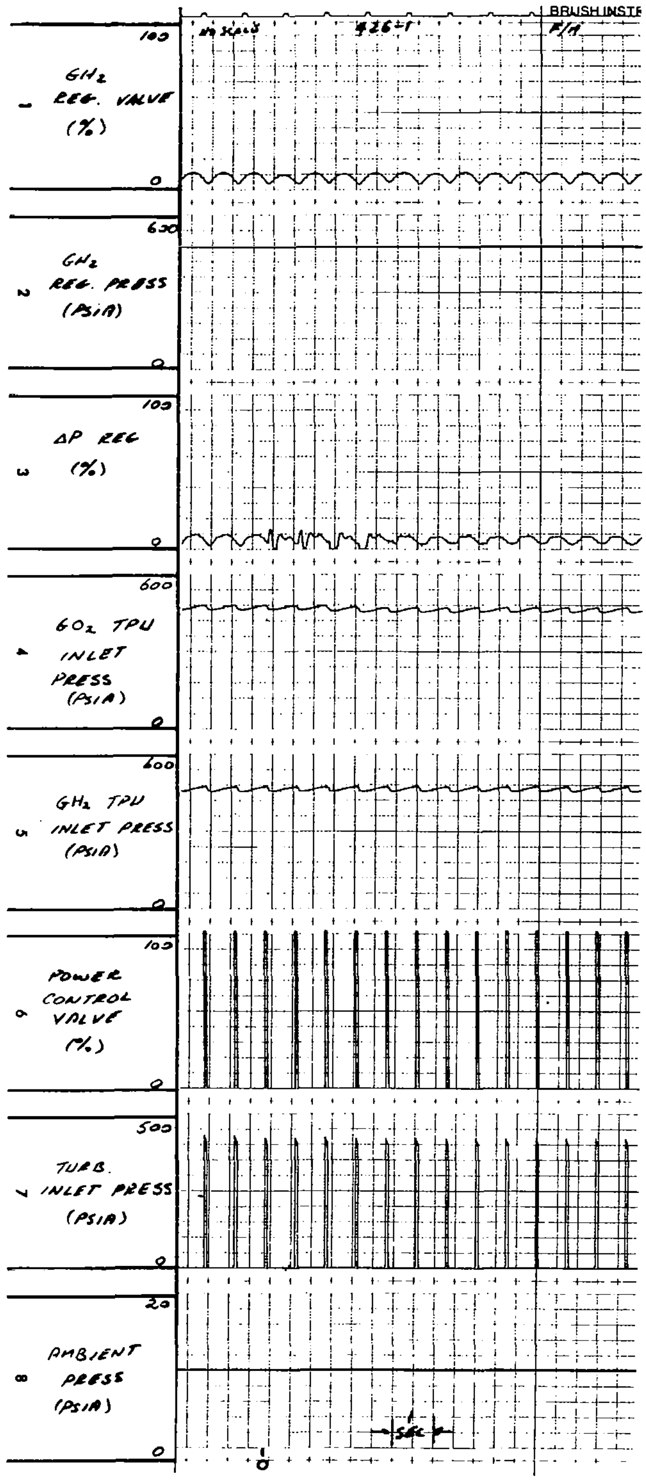

Figure 125. FMA, 15 Percent Contamination of GH_2 Injector--Pulse System

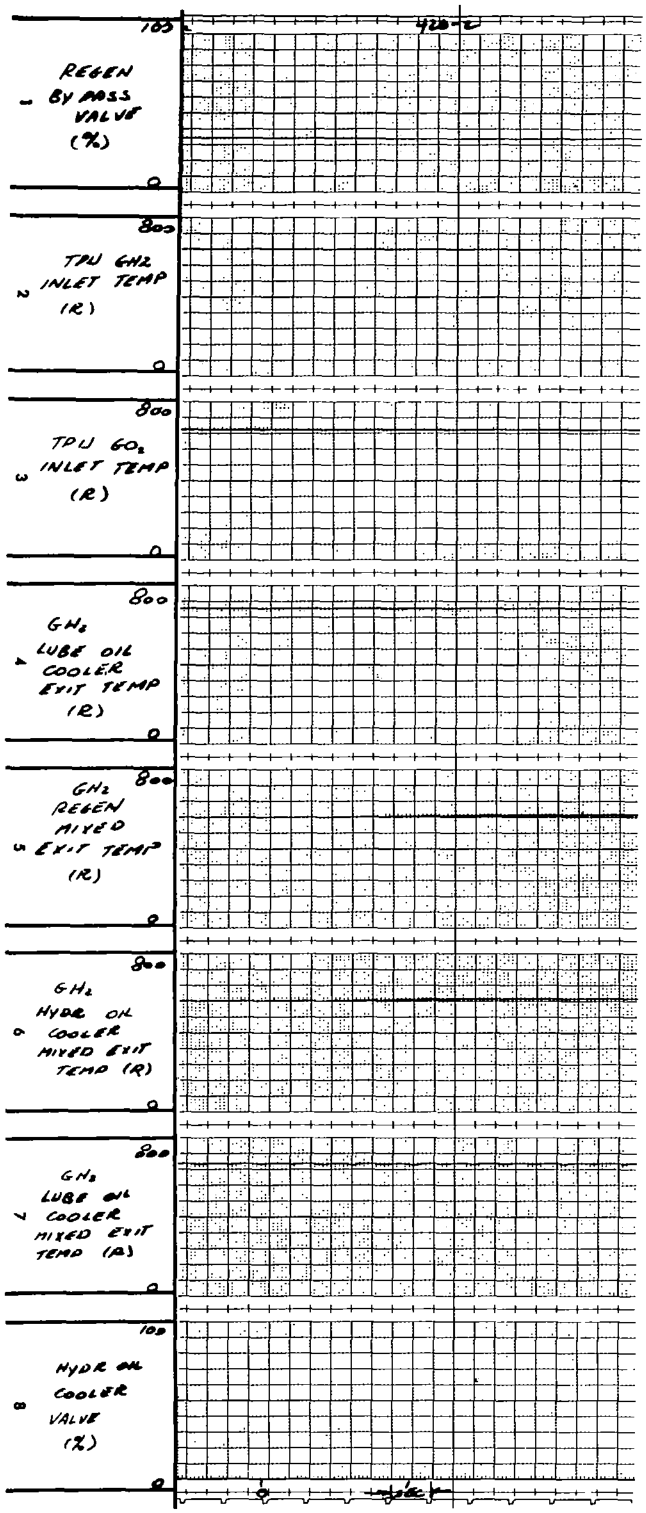

Figure 125. (Continued)

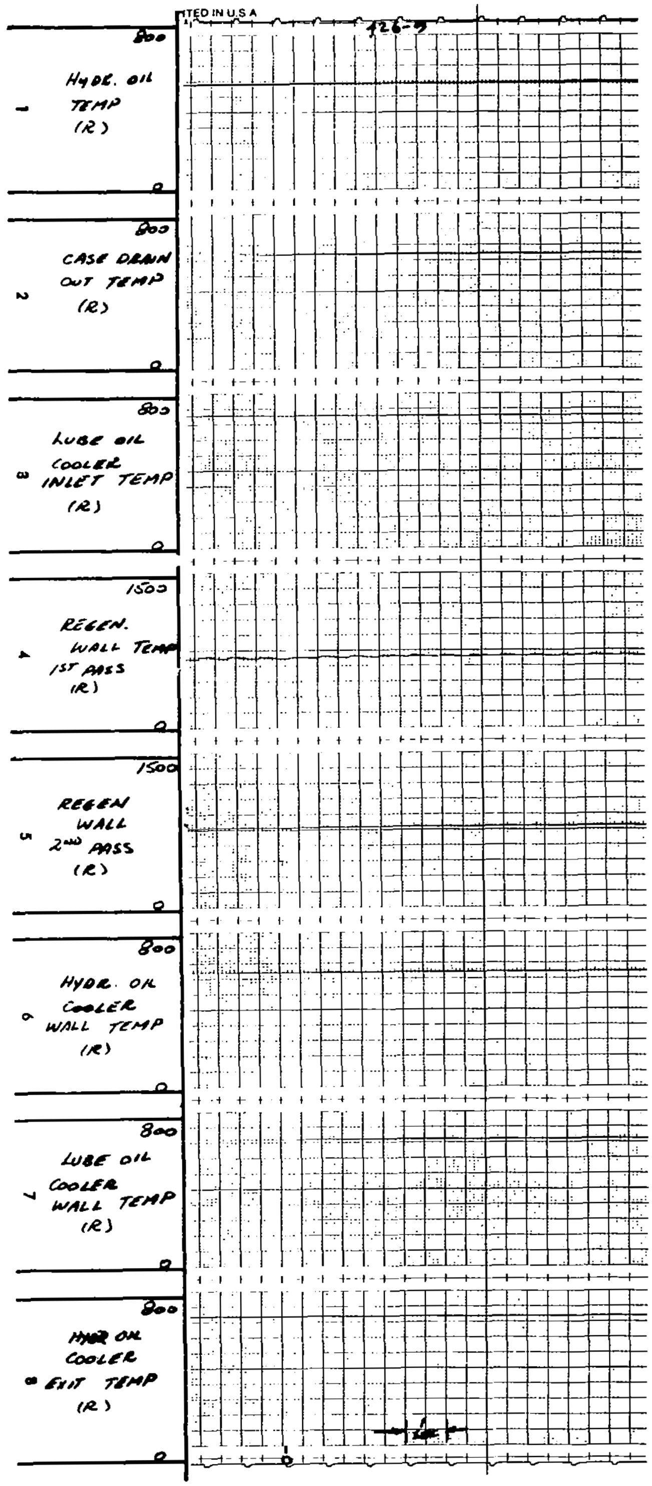

Figure 125. (Continued)

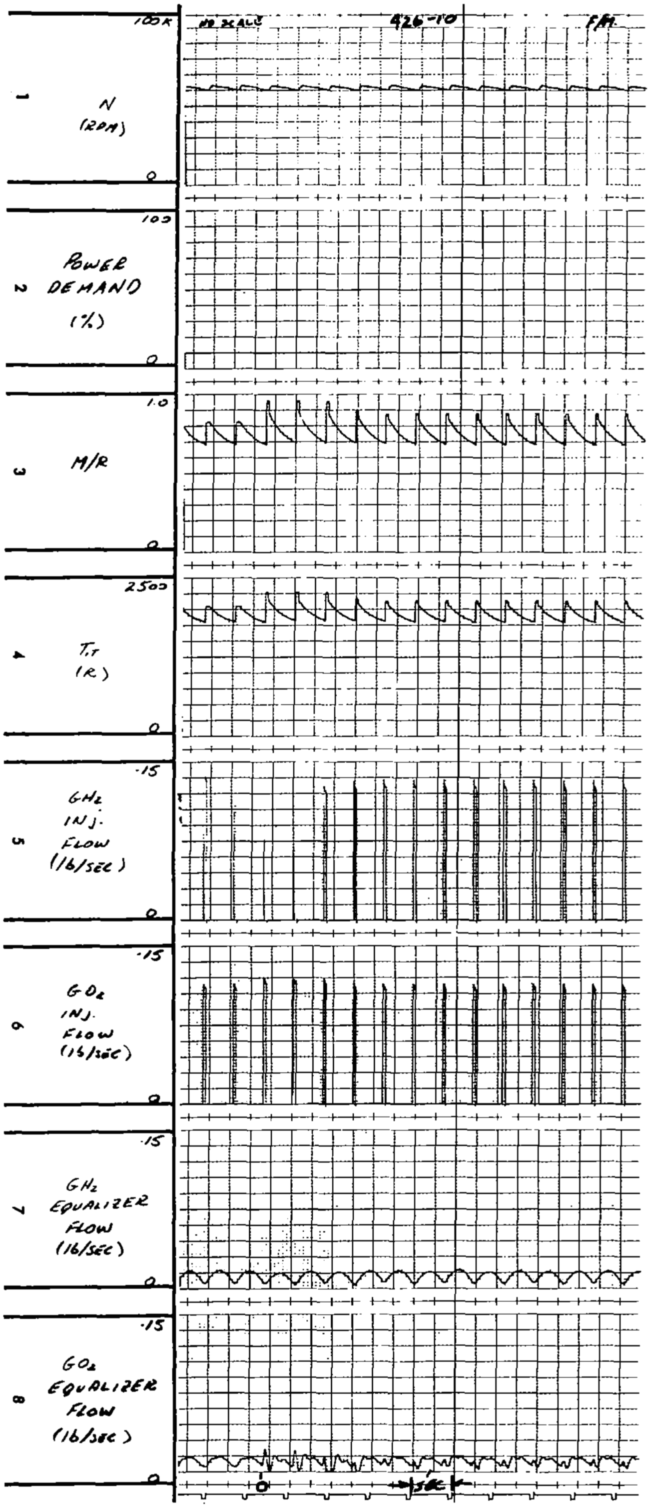

Figure 125. (Concluded)

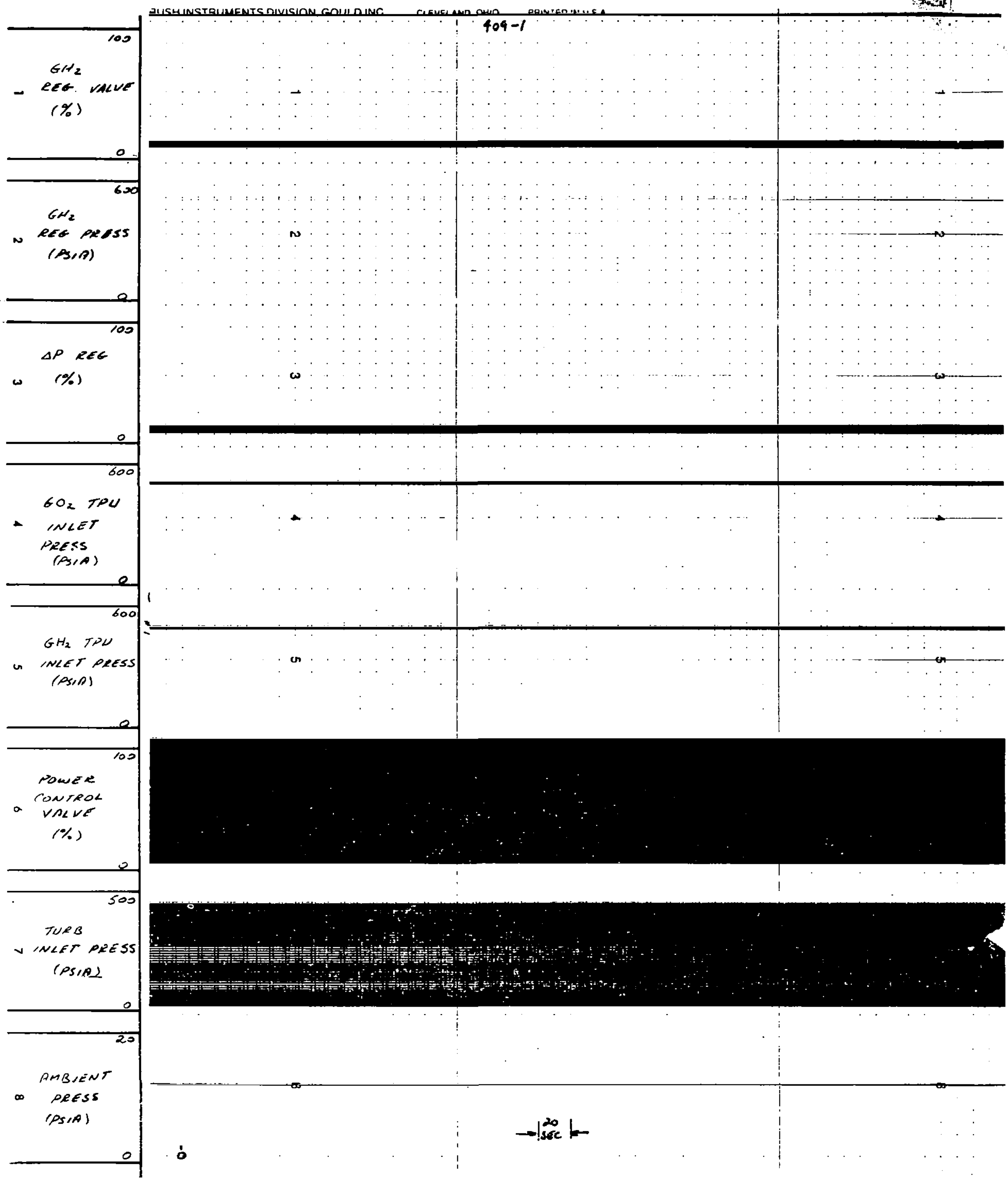

Figure 126. FMA, Hydraulic System Excessive Heat Load--Pulse System

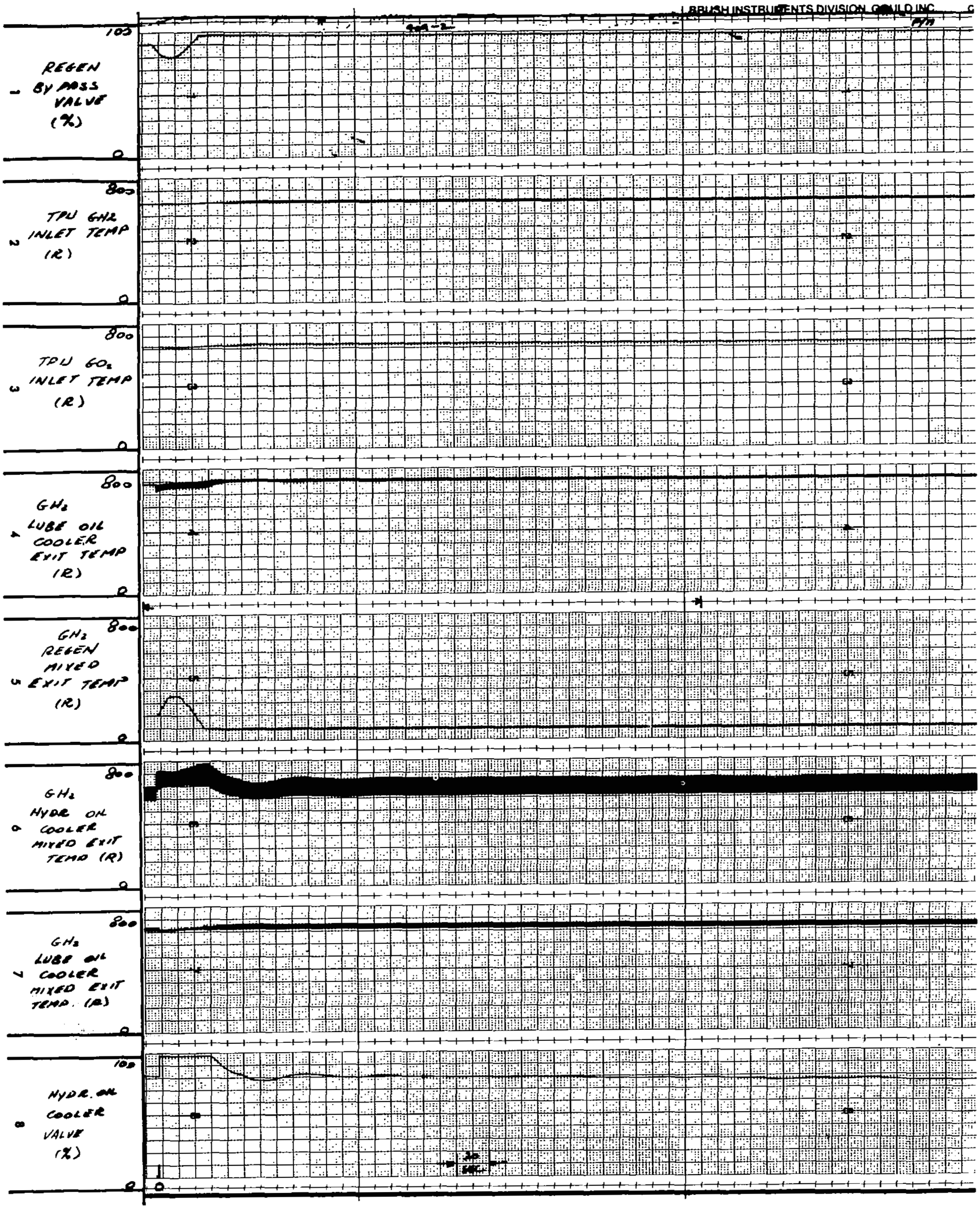

Figure 126. (Continued)

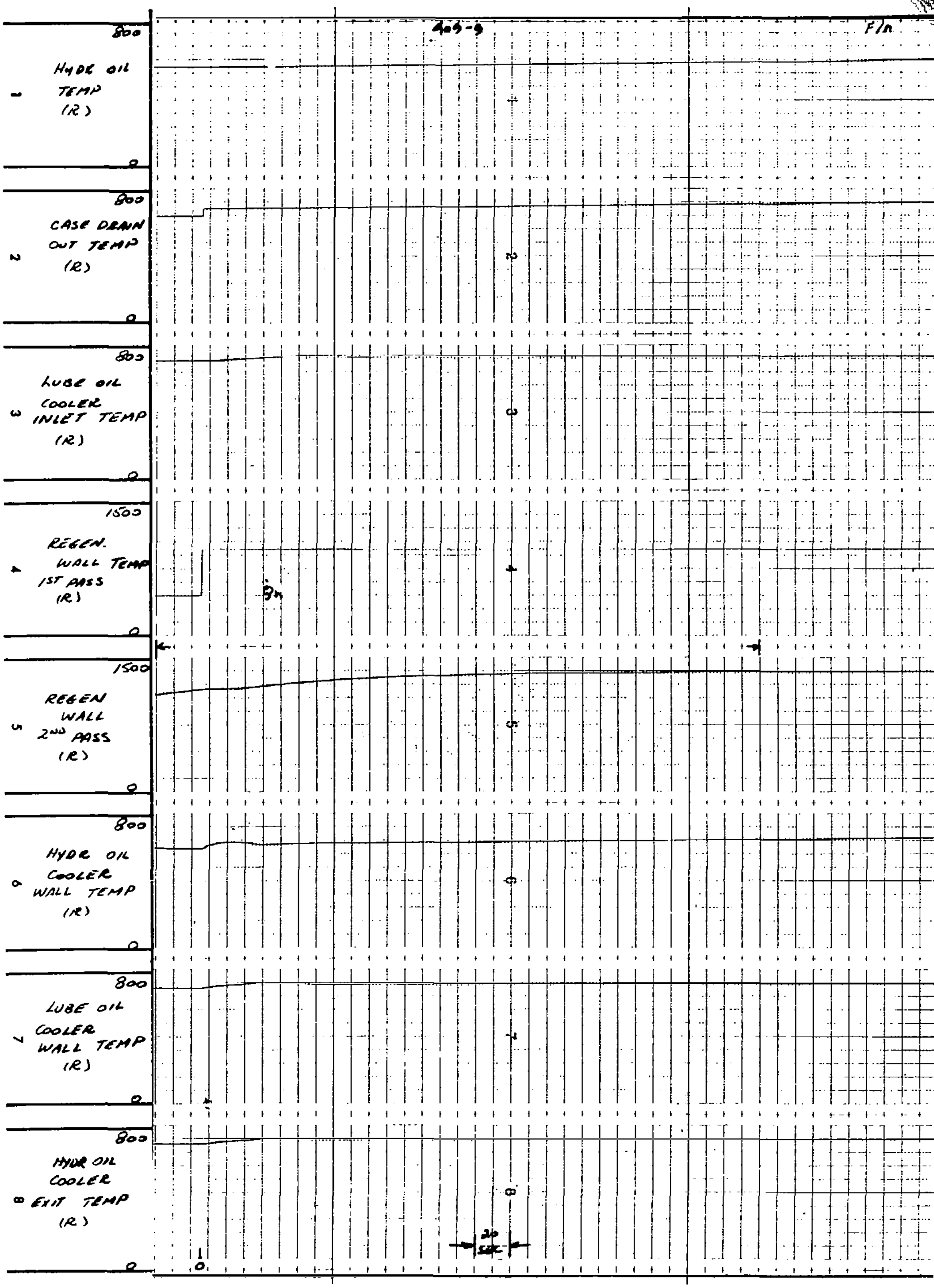

Figure 126. (Continued)

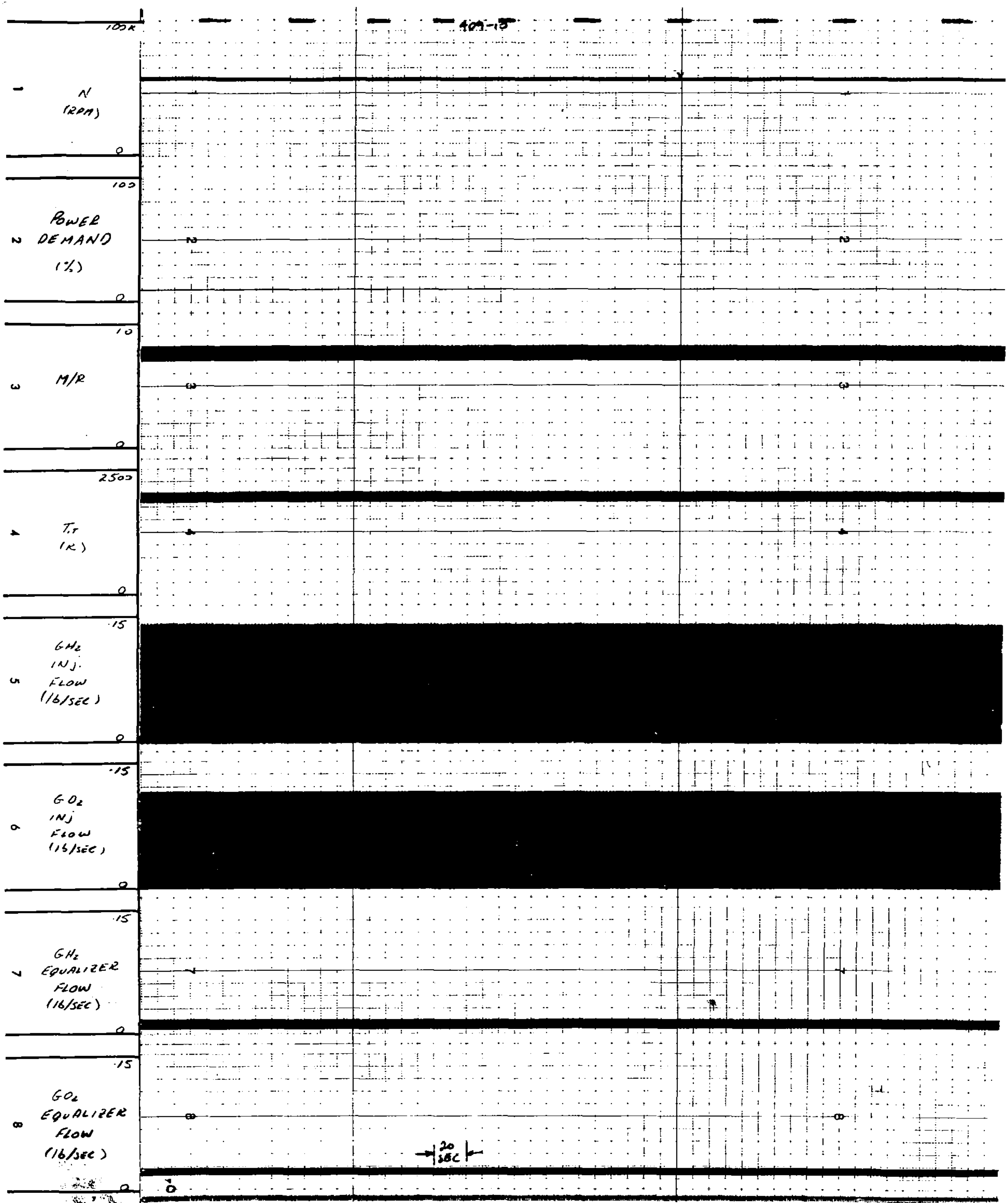

Figure 126. (Concluded)

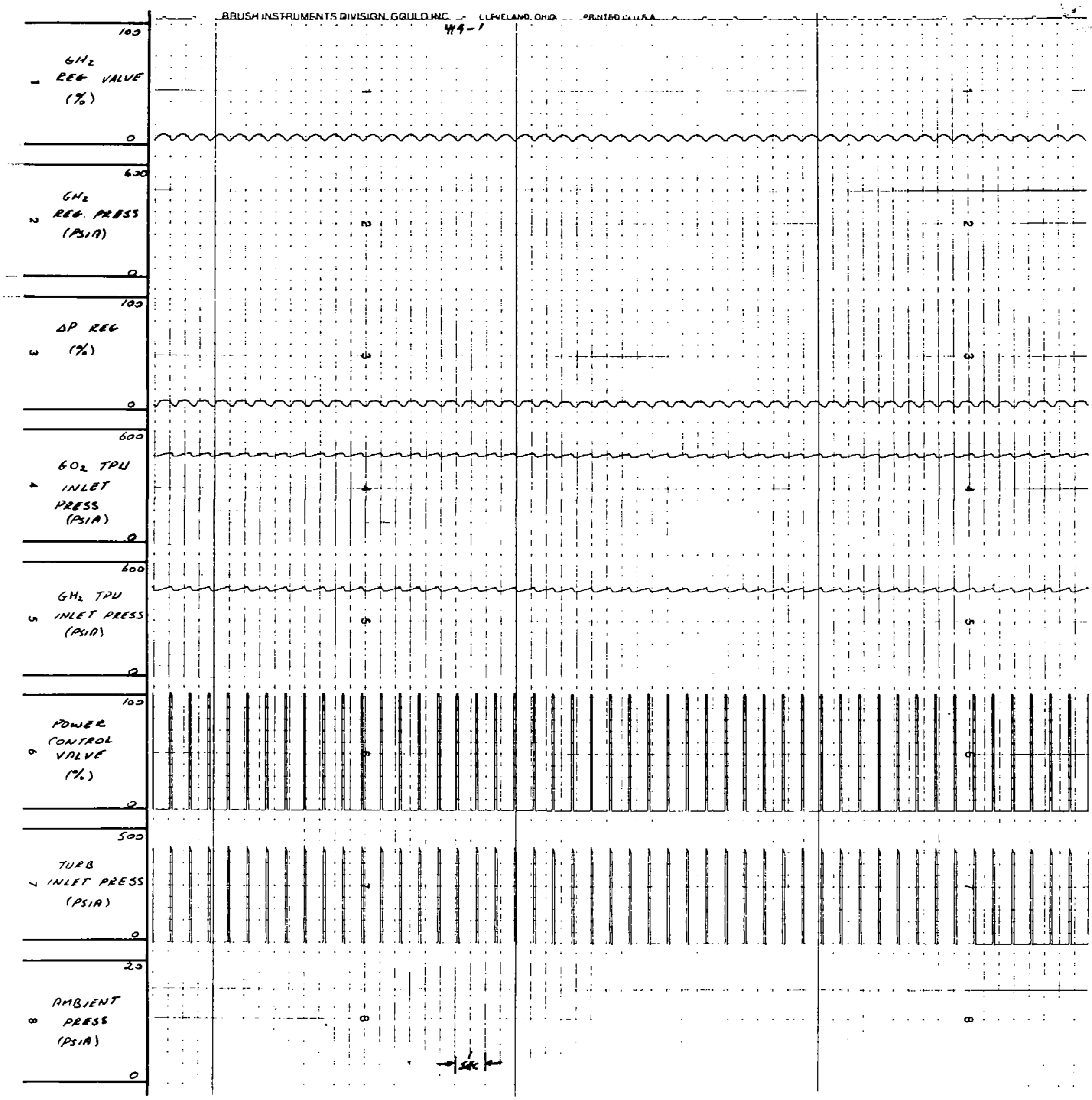

Figure 127. FMA, Regenerator Bypass Valve Closure--Pulse System

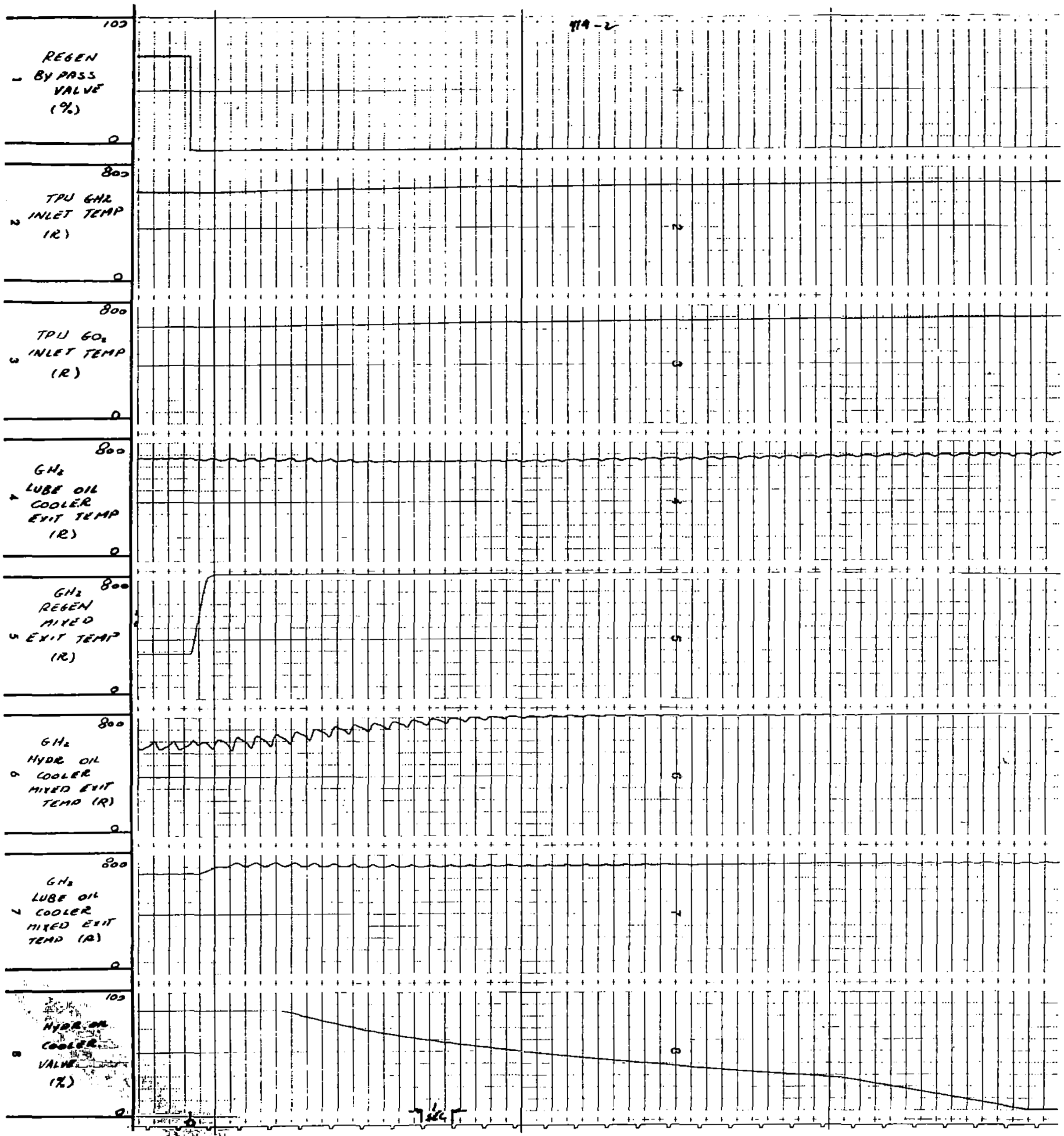

Figure 127. (Continued)

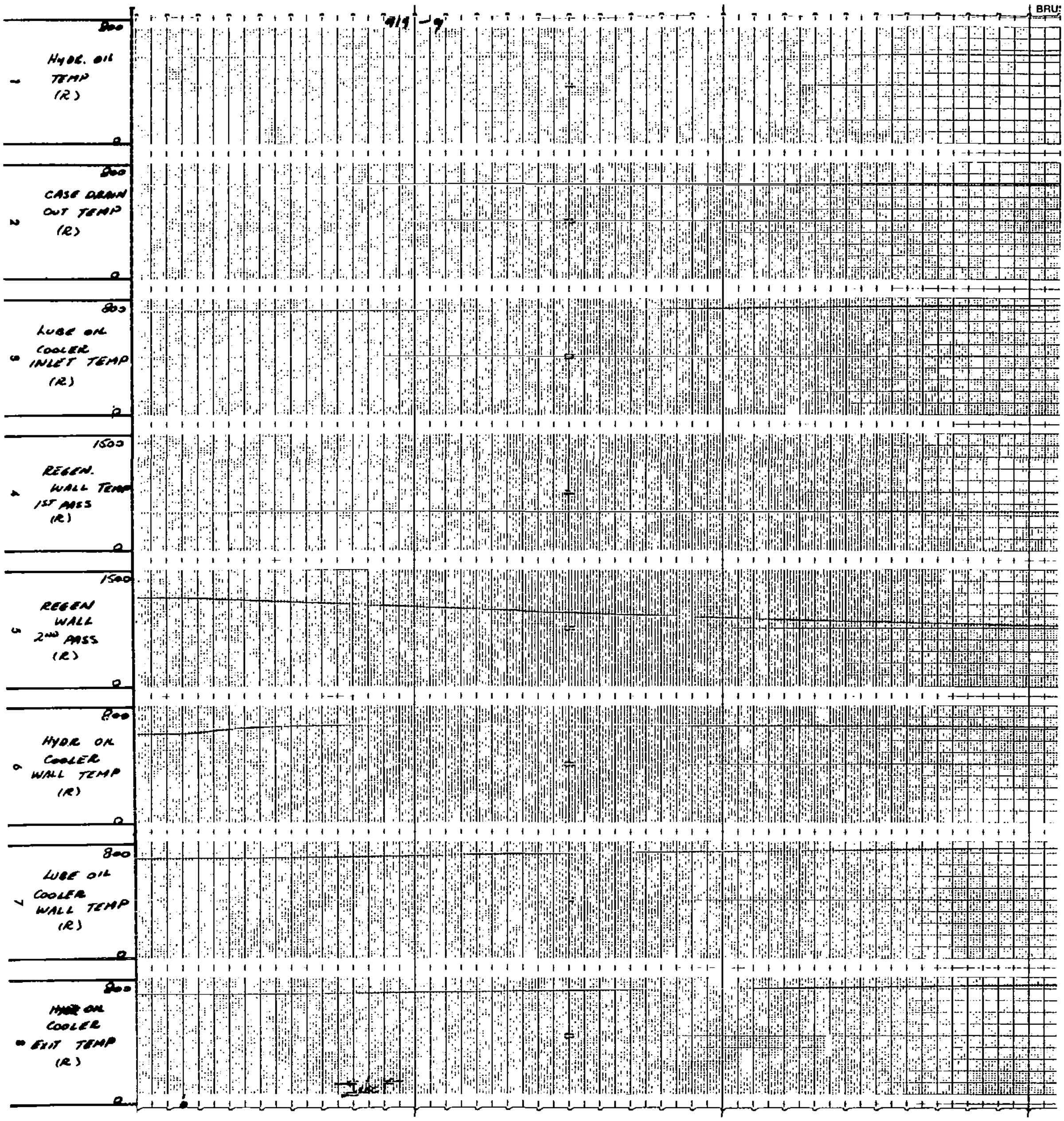

Figure 127. (Continued)

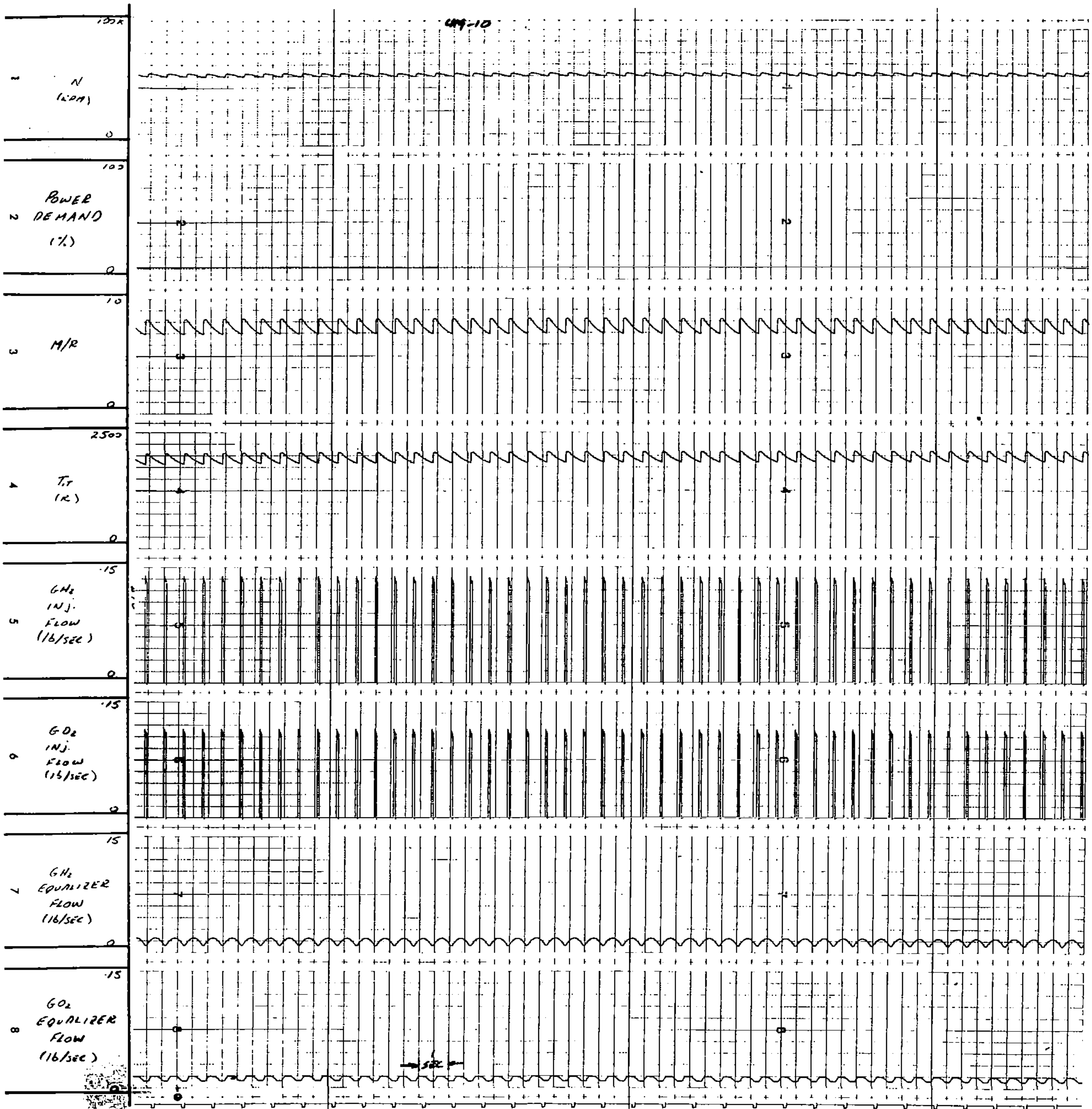

Figure 127. (Concluded)

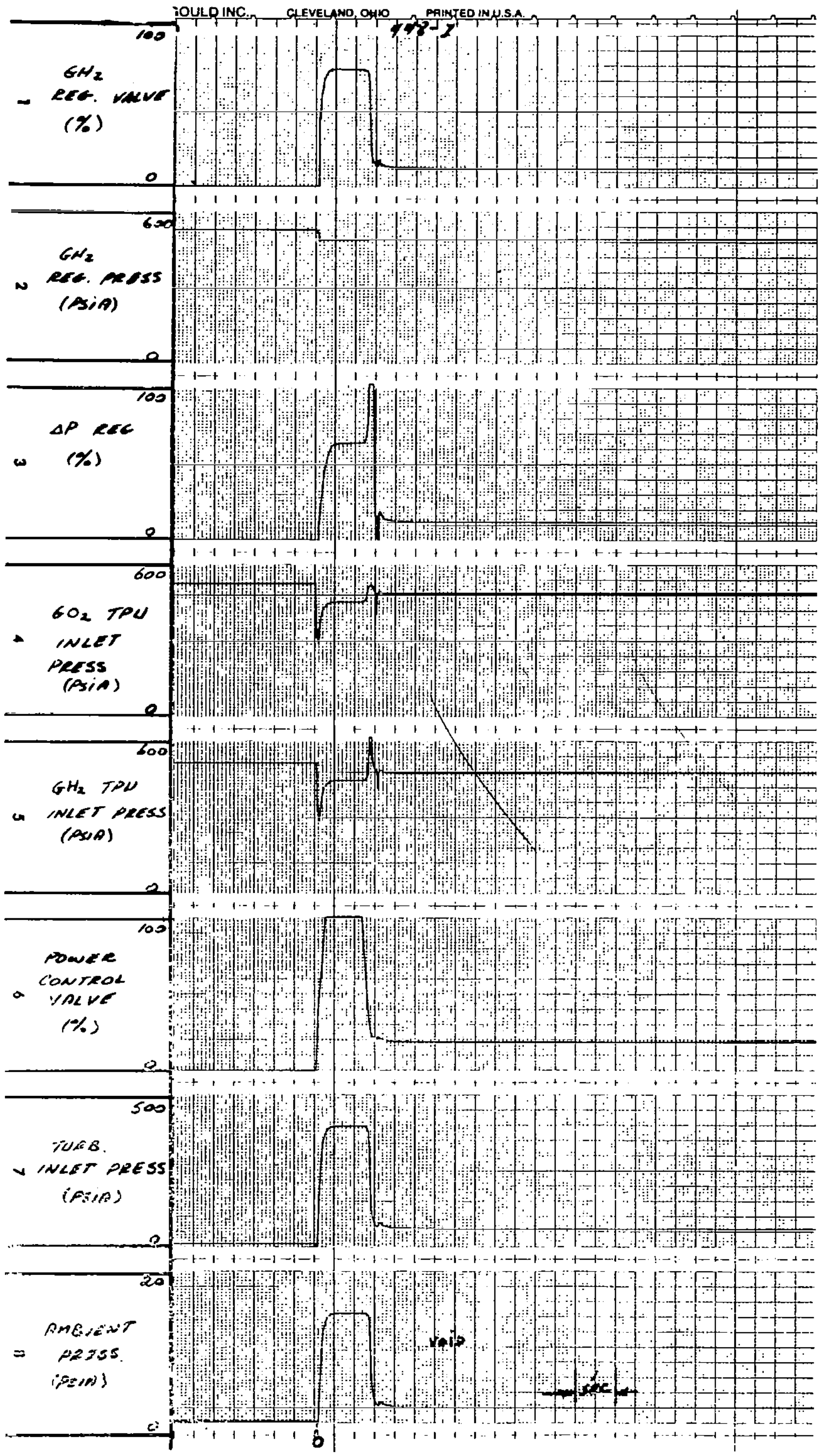

Figure 128. Sea Level Startup--Pressure Modulation System

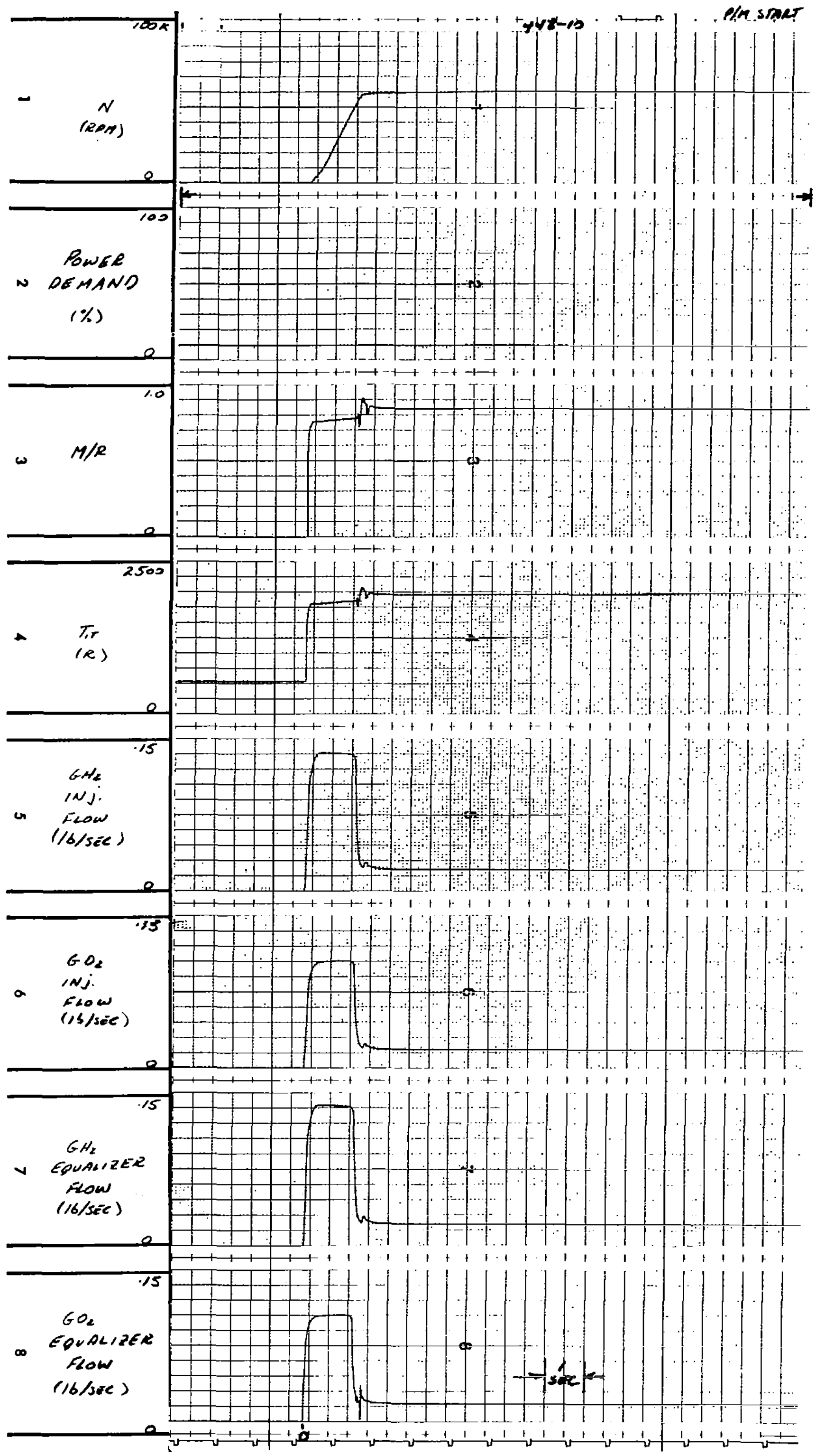

Figure 128. (Concluded)

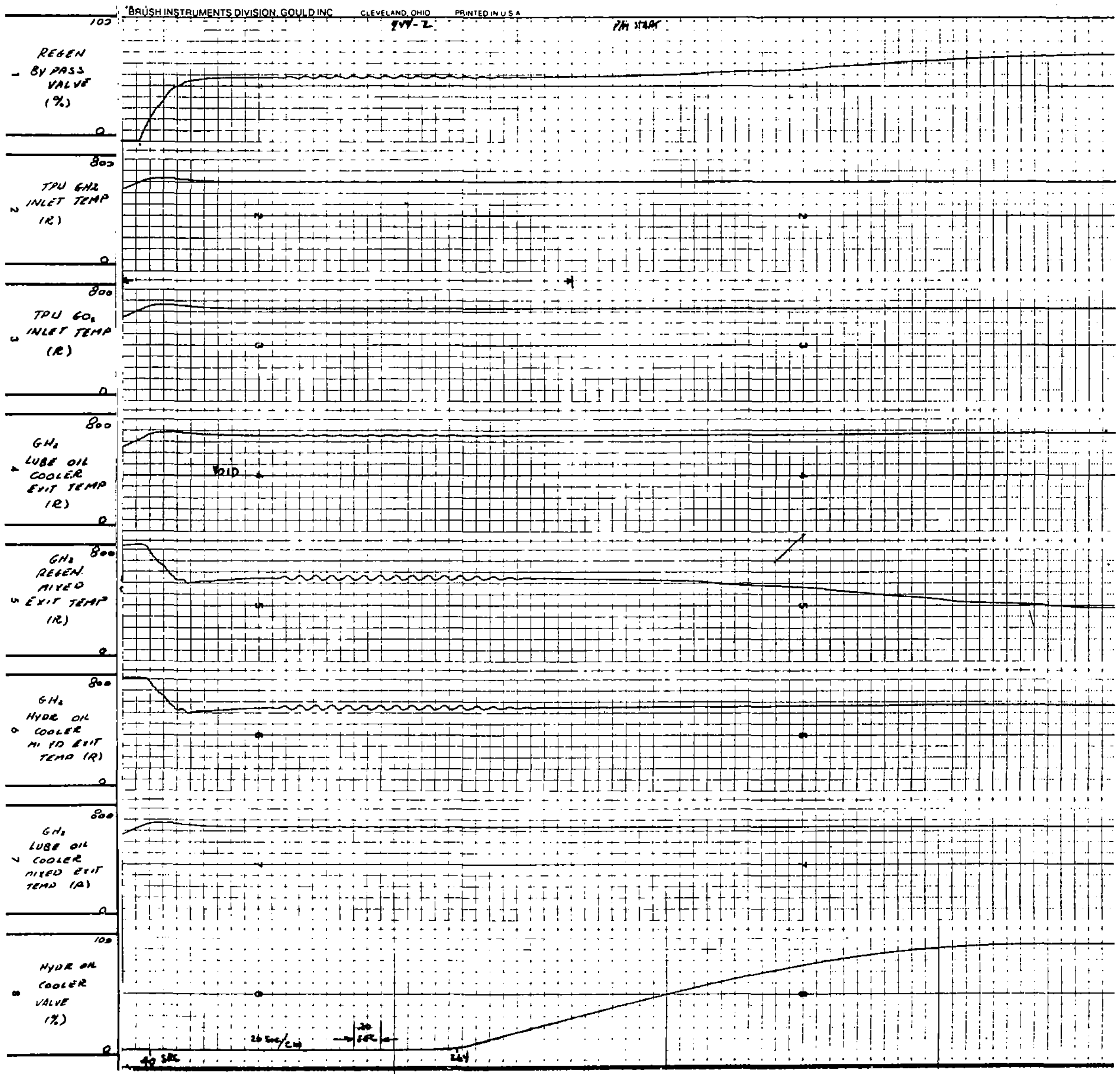

Figure 129. Sea Level Startup, Thermal Effects--Pressure Modulation System

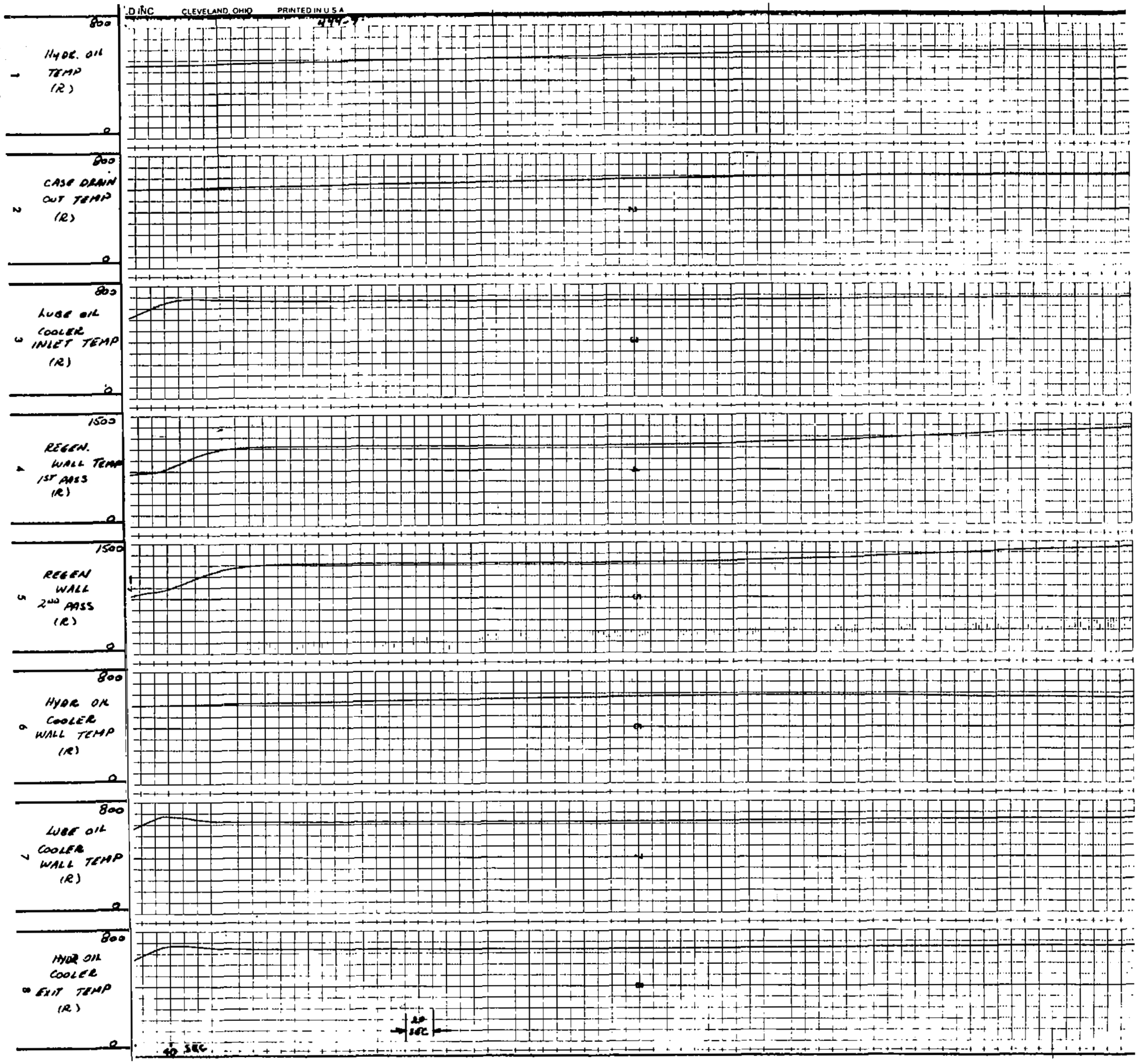

Figure 129 . (Concluded)

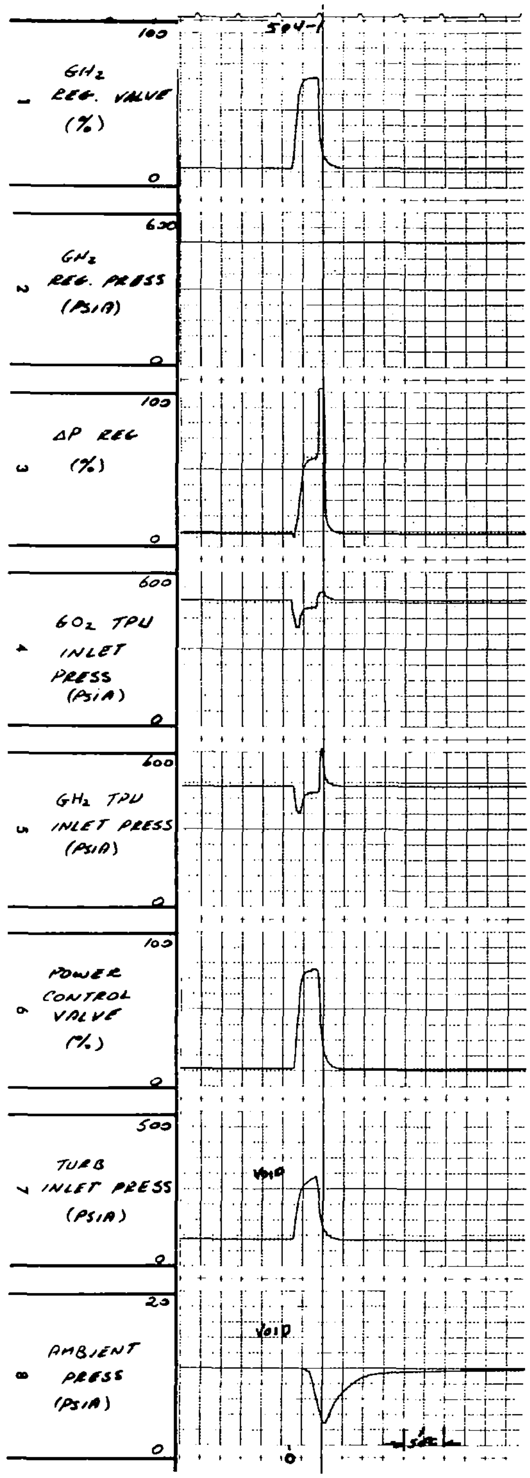

Figure 130. Power Spikes, Idle to Maximum--Thermal Pressure Modulation System

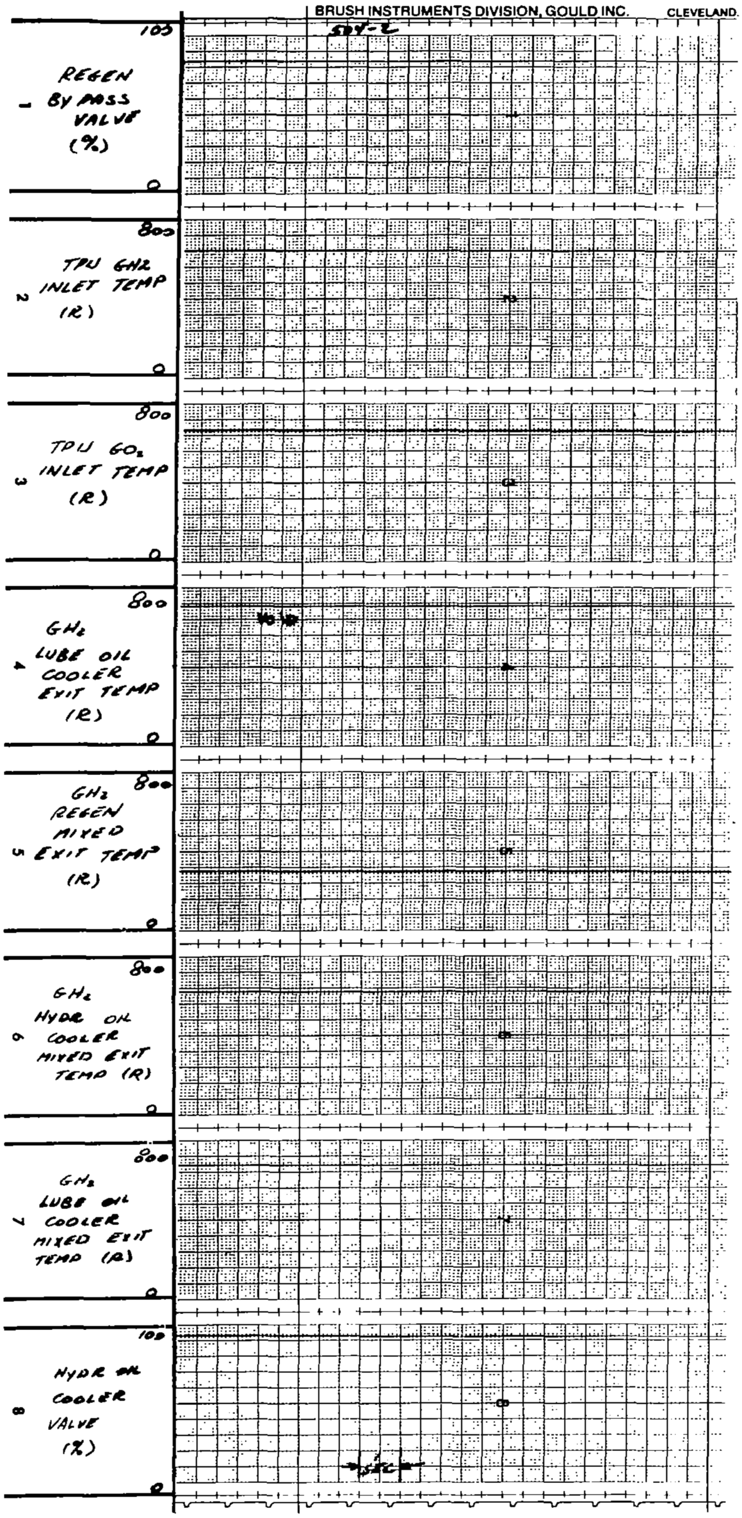

Figure 130. (Continued)

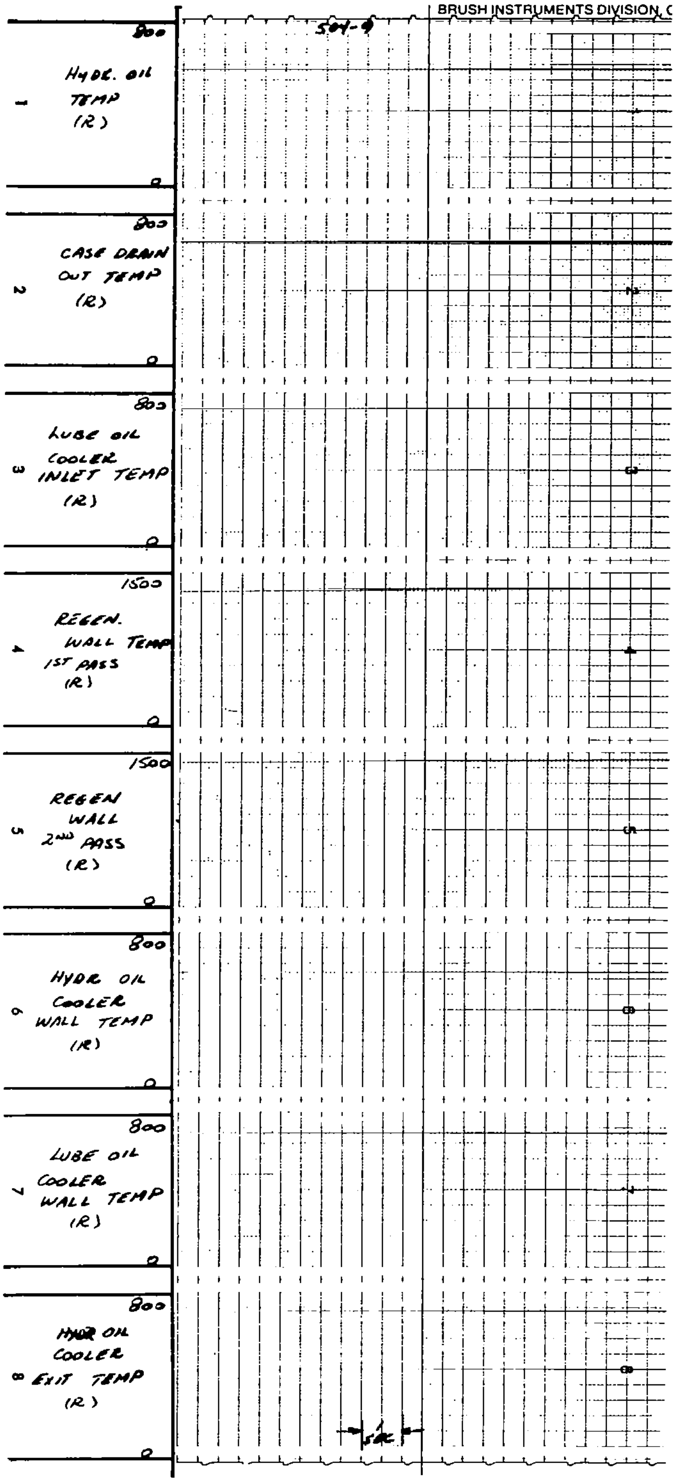

Figure 130. (Continued)

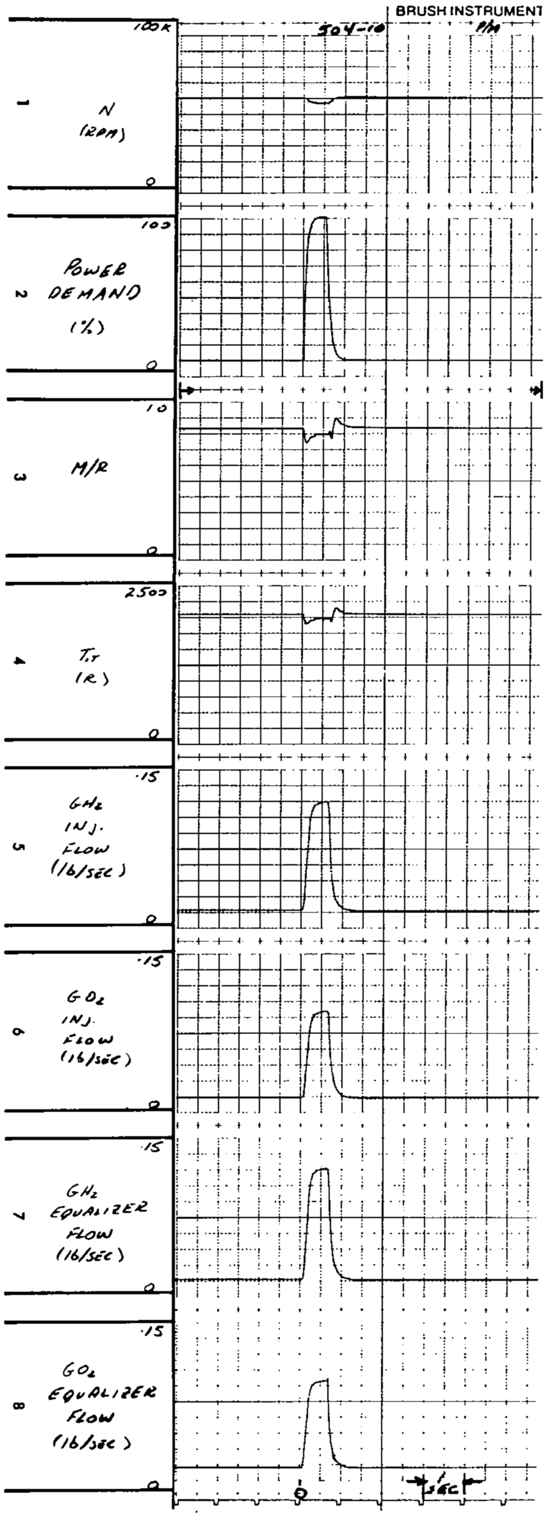

Figure 130. (Concluded)

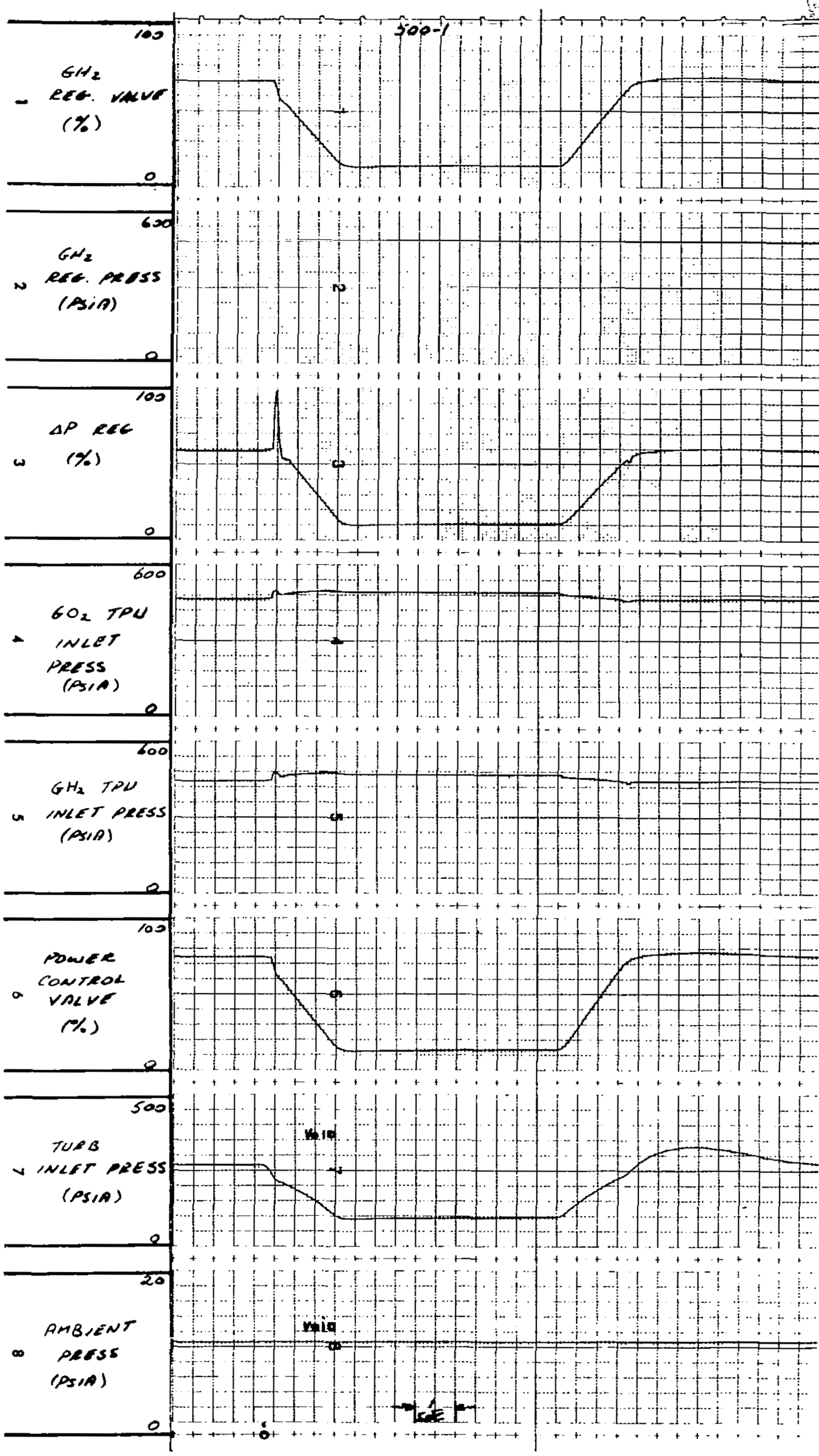

Figure 131. Power Ramps, Idle to Maximum--Pressure Modulation System

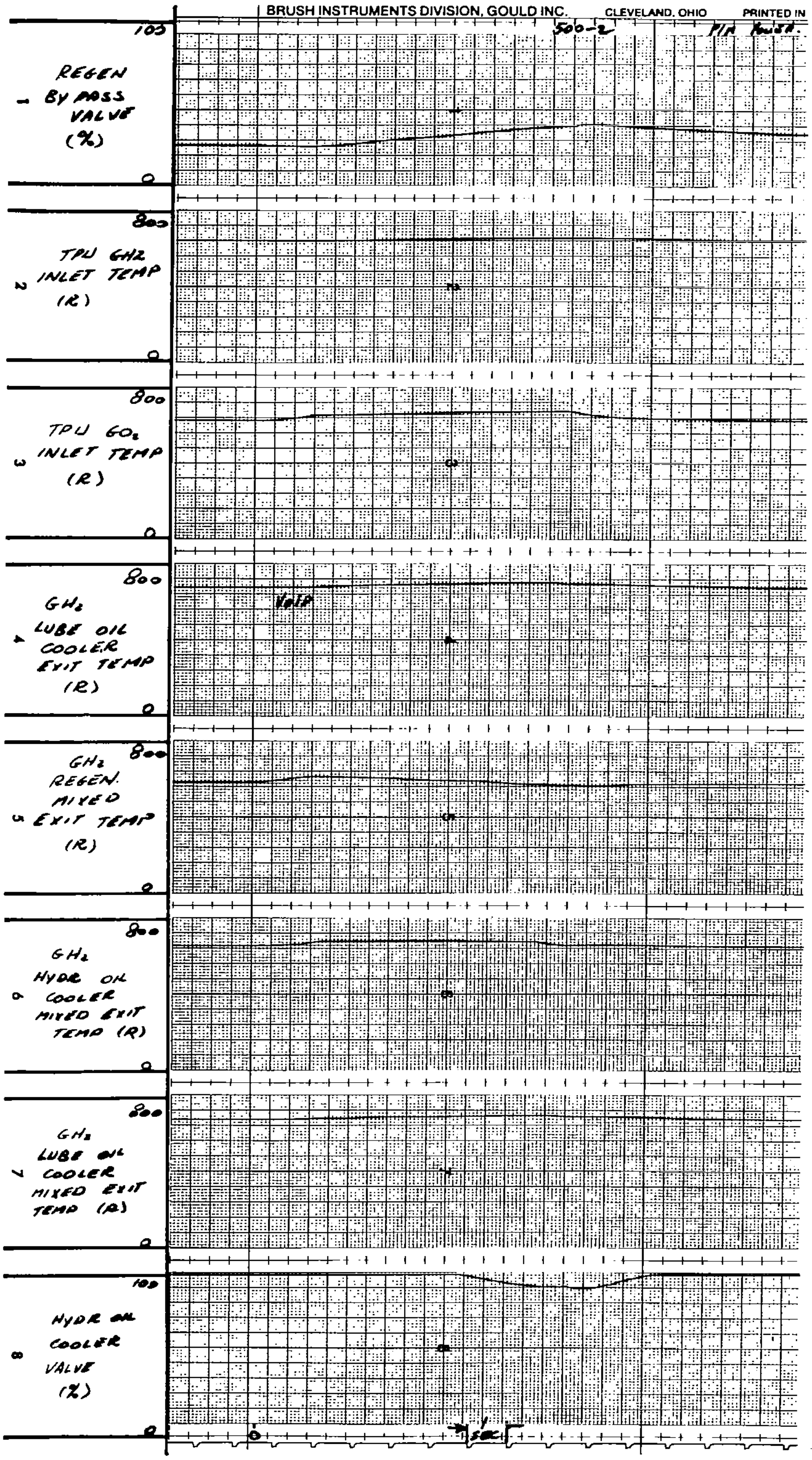

Figure 131. (Continued)

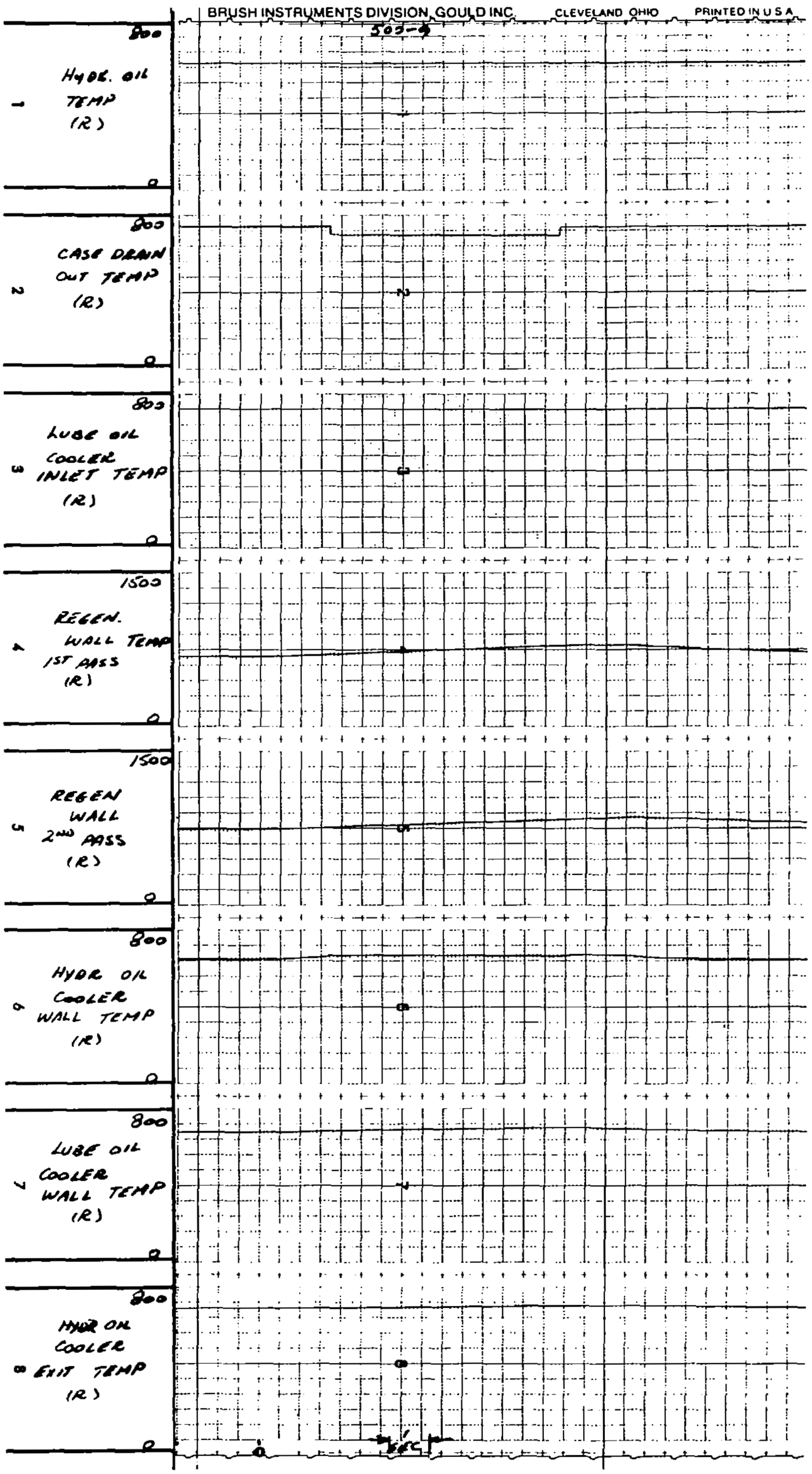

Figure 131. (Continued)

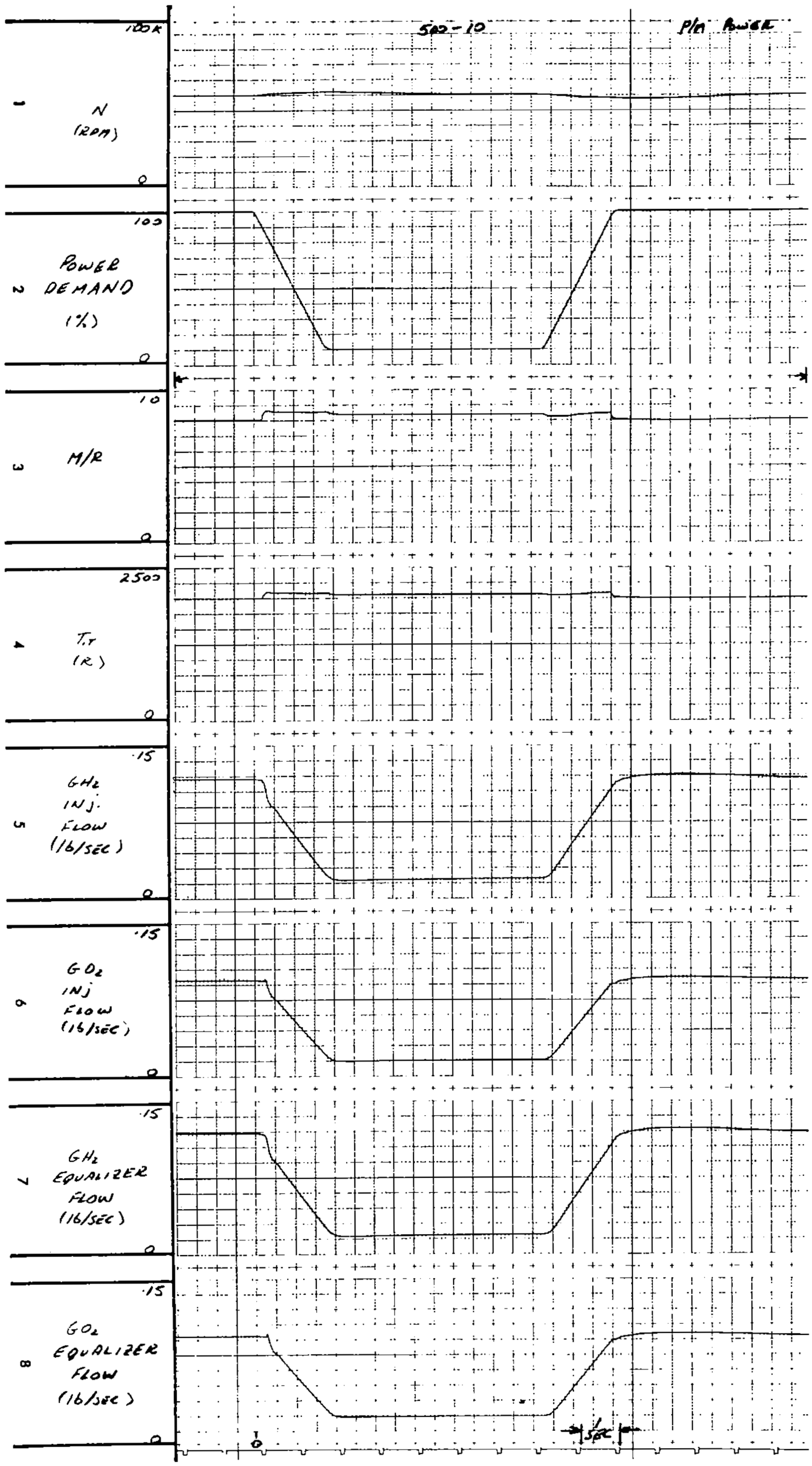

Figure 131. (Concluded)

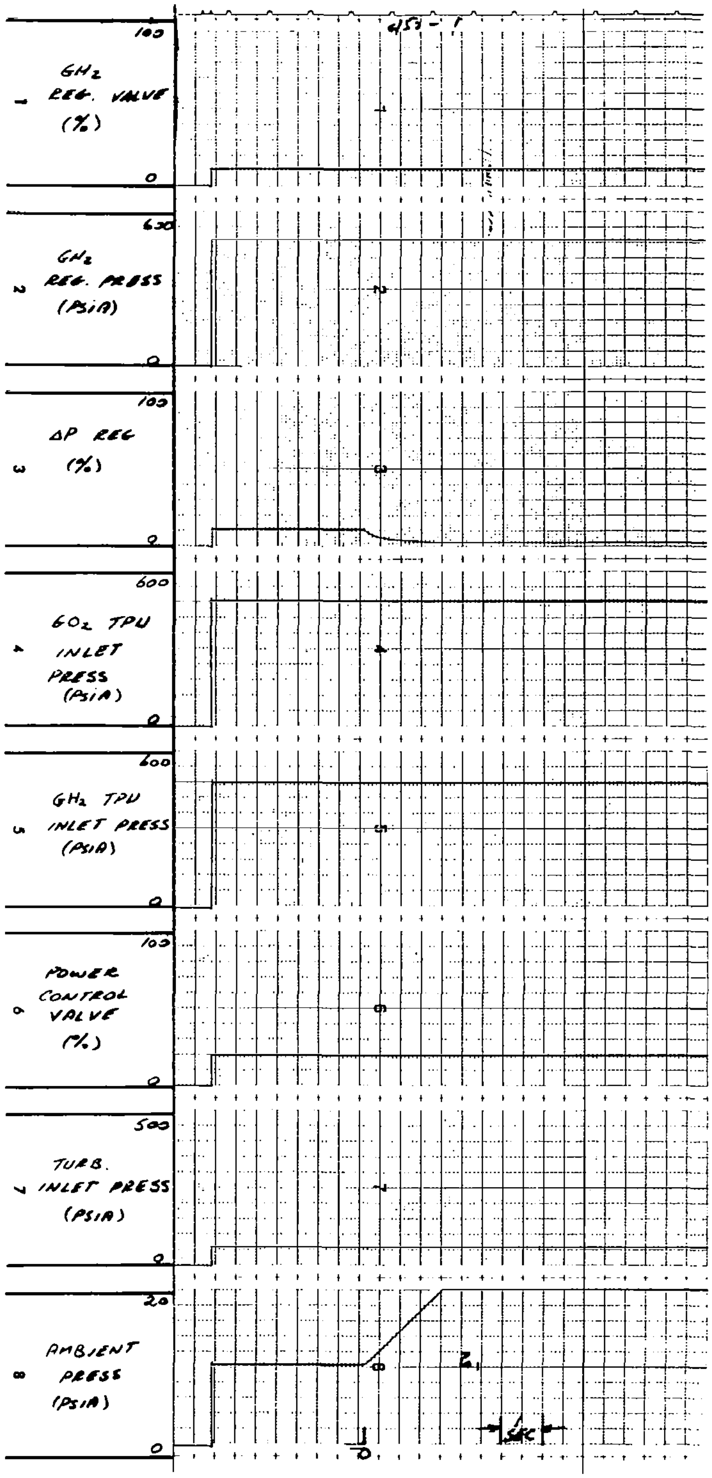

Figure 132. Response to ACS GO_2 Pressure Change--Pressure Modulation System

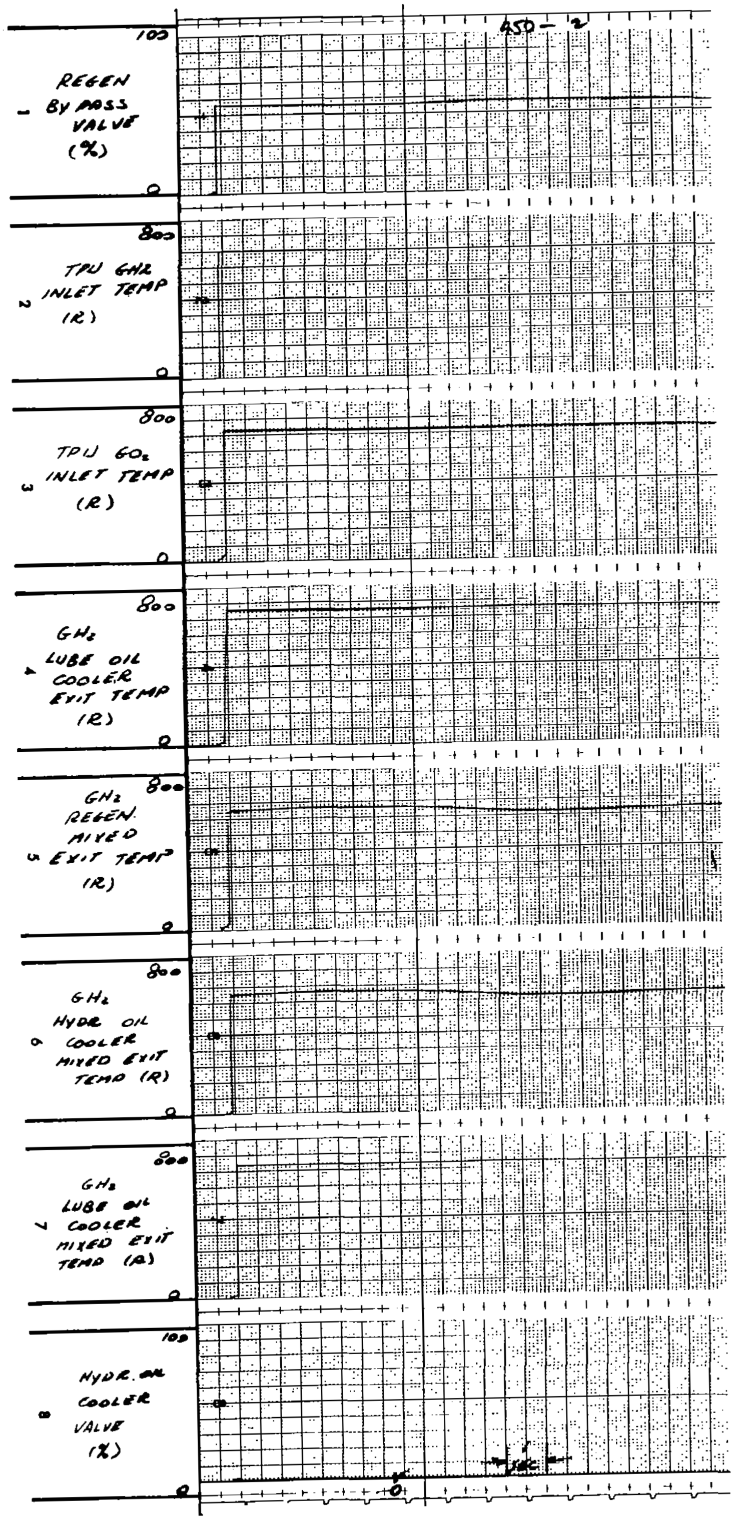

Figure 132. (Continued)

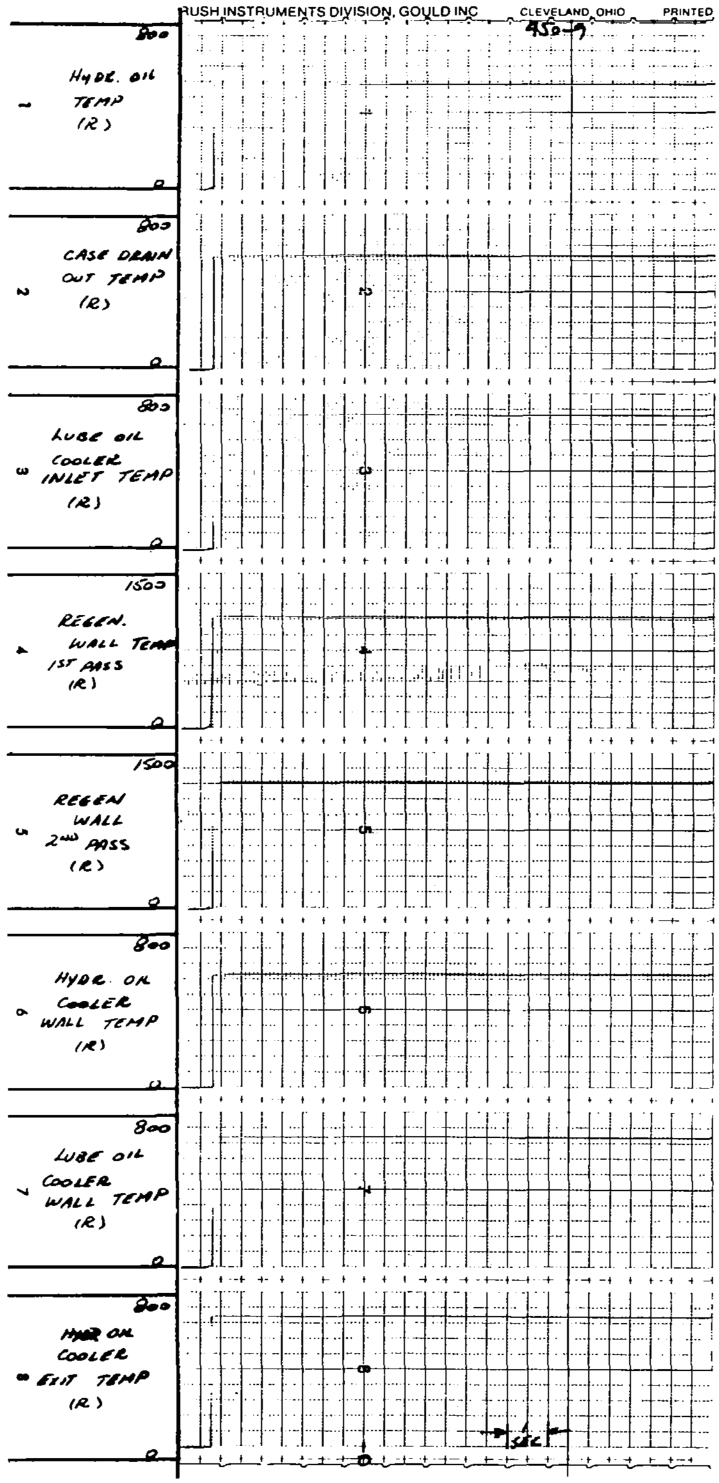

Figure 132. (Continued)

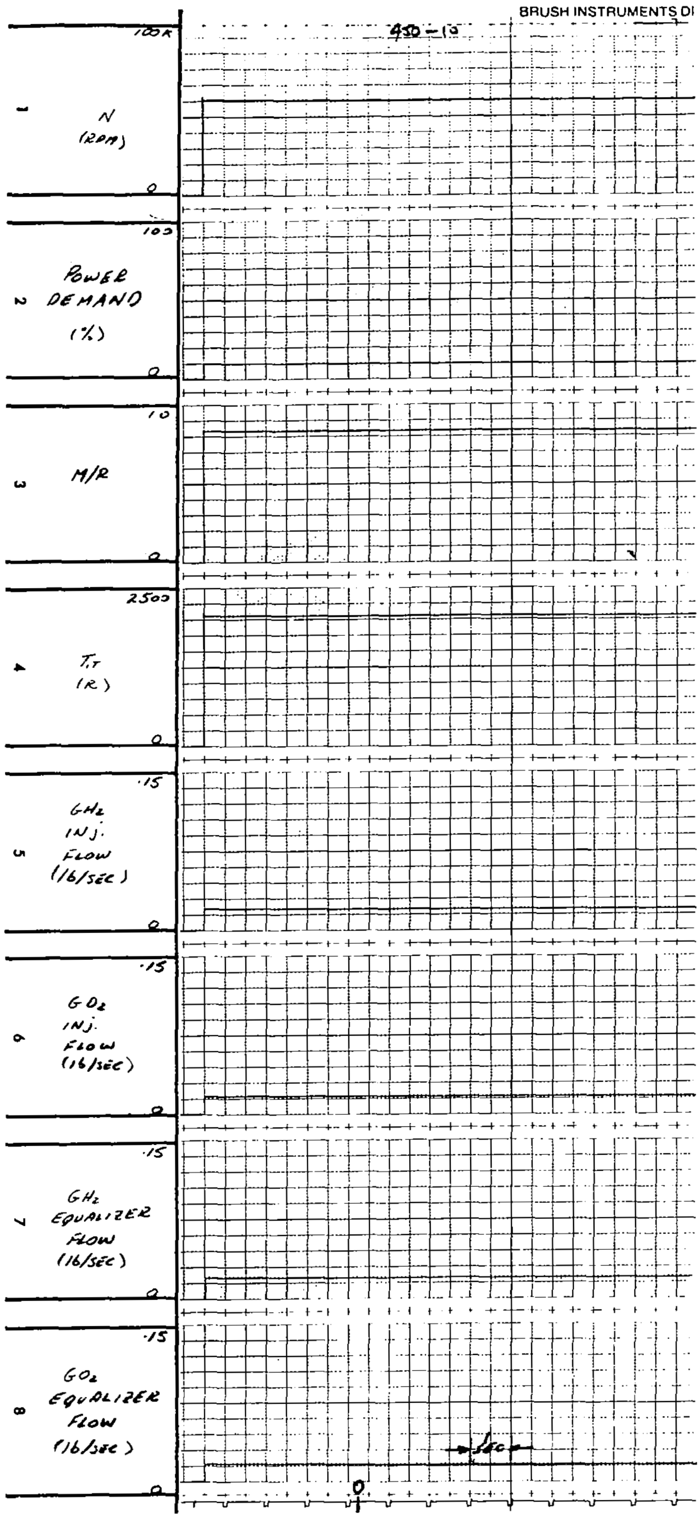

Figure 132. (Concluded)