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SPACE SHUTTLE
AUXILIARY PROPULSION SYSTEM
DESIGN STUDY

PHASE D REPORT
OXYGEN - HYDROGEN
SPECIAL RCS STUDIES

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY • EAST

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15 JUNE 1972

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SPECIAL RCS STUDIES

PREPARED BY: T.L. Baumann
T.C. Patten
H.B. McKee

APPROVED BY: P.J. Kelly, Study Manager

CONTRACT NO. NAS 9-12013

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY - EAST

Saint Louis, Missouri 63166 (314) 232-0232

MCDONNELL DOUGLAS CORPORATION

ABSTRACT

This report describes one phase of a Space Shuttle Auxiliary Propulsion System Design Study under NASA Contract NAS 9-12013. The objective of this program was to fully define competing auxiliary propulsion concepts and to compare them on the basis of weight, reliability, and technology requirements. Propulsion systems using both cryogenic oxygen-hydrogen and earth storable propellants were considered. The main thrust of the cryogenic effort was focused on detailed design and operating analyses for gaseous, oxygen-hydrogen systems, using heat exchangers to thermally condition the propellants, and turbopumps to provide system operating pressure. The effort described in this report complemented this primary design effort by exploring the potential of two alternate, oxygen-hydrogen system concepts. The two fundamental concepts considered in this phase were:

(1) Gaseous oxygen-hydrogen systems, with conditioners similar to those of the primary candidates, but using alternate means of providing system pressure, e.g., electric or hydraulic motor driven pumps or pneumatic bellows pumps.

(2) Liquid oxygen-hydrogen systems, which eliminated conditioning equipment entirely and delivered the propellants to the engines in a liquid rather than a gaseous state.

For these two basic system concepts, this report provides the results of system design analyses and compares various means of implementing each of the concepts on the basis of weight, technology requirements and operational considerations. Additionally, weight comparisons are made between cryogenic oxygen-hydrogen system concepts and earth storable propellant systems for parallel propulsion system requirements. These show that the liquid, oxygen-hydrogen system concepts have the potential to effect very marked weight reductions in the Space Shuttle orbiter total impulse range.

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1. INTRODUCTION

To provide the technology base necessary for design of the Space Shuttle, the National Aeronautics and Space Administration has sponsored a number of technology programs related to the Auxiliary Propulsion System. Among these has been a series of studies aimed at providing the system design necessary for selection of preferred system concepts and for delineation of requirements for complementing component design and test programs. The first of these system study programs considered a broad spectrum of system concepts, but because of high vehicle impulse requirements, coupled with safety, reuse, and logistics considerations, only cryogenic oxygen and hydrogen were considered as a propellant combination. Additionally, unknowns in engine pulse mode ignition and concerns with the distribution of cryogenic liquids served to eliminate liquid-liquid feed systems from the list of candidate concepts. Therefore, only systems which delivered propellants to the engines in a gaseous state were considered for the Reaction Control System (RCS). The results of these initial studies, reported in References (a) through (d), indicated that among the many options for design of a gaseous, oxygen-hydrogen system, an approach using heat exchangers to thermally condition the propellants and turbopumps to provide system operating pressure, would best satisfy requirements for a fully reusable Space Shuttle. These study programs focused attention to this general system type, but did not examine in depth several viable approaches to turbopump system design and control. To fill this need the NASA contracted with McDonnell Douglas Astronautics Company-Eastern Division in July 1971 for additional study of the Shuttle Auxiliary Propulsion Systems. This contract (NAS 9-12013) titled "Space Shuttle Auxiliary Propulsion System Design Study" was under the technical direction of Mr. Darrell Kendrick, Propulsion and Power Division, Manned Spacecraft Center, Houston, Texas.

As originally defined, this design study was a five phase program considering only oxygen-hydrogen propellants. Reference (e) provides the Executive Summary of program results and Reference (f) provides a detailed description of the program plan for each of the five program phases listed below:

1. Phase A - Requirements Definition
2. Phase B - Candidate RCS concept comparisons
3. Phase C - RCS/OMS Integration
4. Phase D - Special RCS Studies
5. Phase E - System Dynamic Performance Analyses

Phase A, which defined all design and operating requirements for the Auxiliary Propulsion Systems is documented in Reference (g). Phase A results showed that requirements for the booster and orbiter stages were sufficiently similar to allow concentration of all design effort on the orbiter stage as results obtained would be applicable to a "fly-back" type booster. In Phase B, detailed design and control analyses for the three most attractive gaseous oxygen-hydrogen Reaction Control System (RCS) concepts using turbopumps were conducted. Reference (h) provides a discussion of results from this phase of the study. Phase C was aimed at defining the potential for integration of the RCS with the Orbit Maneuvering System (OMS). As defined by the original contract, only oxygen and hydrogen were considered in Phase C. However, concurrent vehicle design studies showed that smaller shuttle orbiters with external, expendable main engine tankage would provide a far more cost effective vehicle approach. This change in vehicle design resulted in a significant reduction in APS requirements and, coupled with a companion Shuttle program decision to allow scheduled system refurbishment, allowed consideration of systems using earth storable propellants for auxiliary propulsion. Thus, in November of 1971, the NASA issued a contract change order that extended the scope of Phase C to include earth storable monopropellant and bipropellant systems and redirected Phase E to provide final performance analyses on storable propellant systems. Reference (i) provides documentation of Phase C effort on oxygen-hydrogen and Reference (j), reports the results of both Phase C and E effort on earth storable propellant systems. In addition to the principal contract effort in Phases B and C, the study included an exploratory effort to evaluate two alternatives to gaseous oxygen-hydrogen RCS using turbopumps. The system concepts investigated in Phase D had not been evaluated during the early conceptual definition studies and had the potential to both improve performance and reduce technology risk. These systems are the subject of this report.

Two basic alternatives to the "turbopump", gaseous oxygen-hydrogen system were originally defined, as shown in Figure 1-1. The first alternative maintained the same basic system approach used for turbopump systems but used a high pressure liquid accumulator with a small pump thereby reducing pump power requirements to level where power sources other than hot gas turbines could be considered. Eliminating the turbopump avoided the technology concerns associated with turbopump bearing life in an environment requiring many rapid startups in each mission.

SPECIAL RCS STUDY CONCEPTS

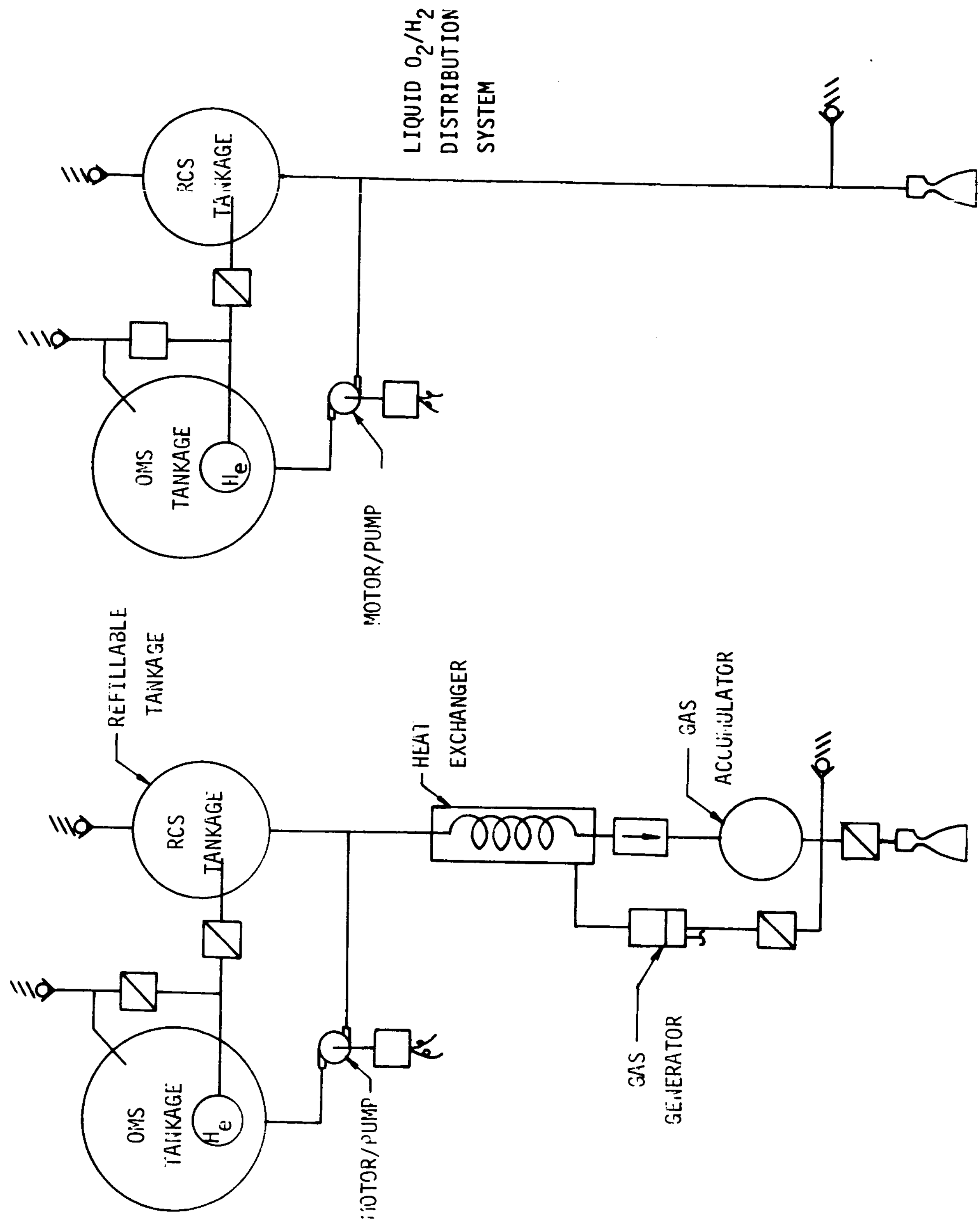


Figure 1-1

The second alternative also used a liquid accumulator in conjunction with a low power pump but considered distribution of the cryogenic oxygen-hydrogen as liquids to engines which had the capability for liquid propellant ignition. The "liquid" concept thereby eliminated the need for gaseous accumulators and thermal conditioning equipment.

Study results show that designs for the first approach can be developed which are weight competitive with turbopump systems. However, the most striking study results were obtained with the liquid system concept. This approach was found to provide very significant weight savings and a marked reduction in system complexity.

This report is organized to provide a summary of technical effort and results in the report body. Substantiating technical detail and data is provided in the attached appendices. Based on its uniqueness, the liquid RCS concept is given major emphasis in the body of the report.

2. REQUIREMENTS

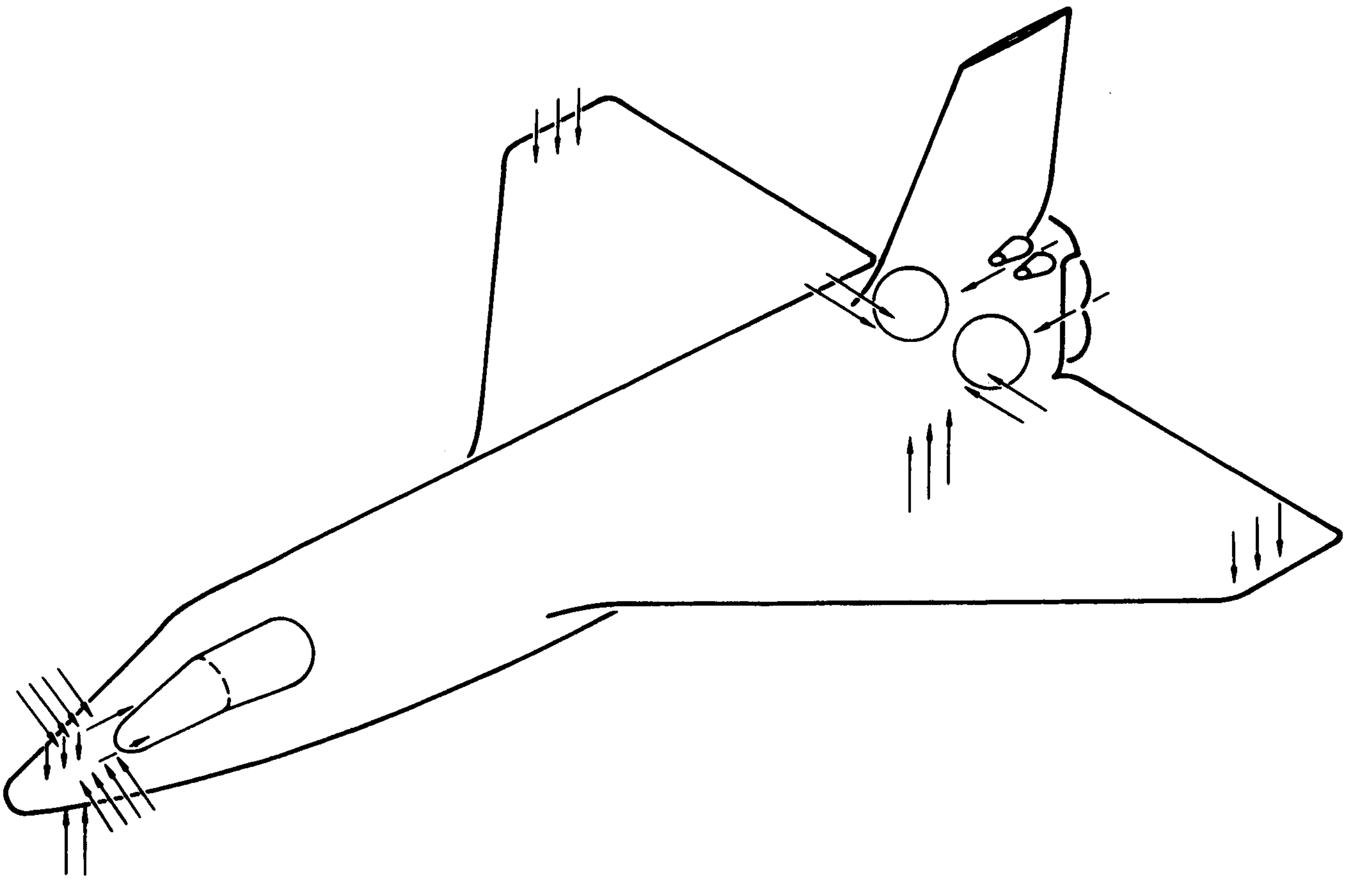
The orbiter stage, for which the oxygen-hydrogen RCS studies were conducted is illustrated in Figure 2-1. The characteristics of this vehicle were based primarily on the results of MDAC-East studies of fully reusable orbiters and boosters as defined in Reference (k). The most distinguishing feature of this orbiter configuration was that the main engine propellant tanks were internal to the vehicle, resulting in a relatively large orbiter stage. Most of the design studies described herein used this orbiter as a reference configuration. The exceptions are in the system weight comparisons of Section 4, which show RCS weight at design requirements corresponding to smaller orbiter configurations of the type designed to use external, main engine tankage.

Reference (g) provides the detailed analyses and rationale used to develop the RCS requirements tabulated in Figure 2-1. The RCS requires 33 engines at 1150 lbf each to provide three axis attitude control. The thrust level and engine arrangement are designed such that, with the failure of any two control engines, the system will still provide torque levels sufficient for safe vehicle entry. The total impulse of the system is 2.25 million lb-sec. This includes total impulse for both attitude control and vernier translation maneuvers of ± 20 ft/sec. These requirements serve to define the basic system design parameters, viz., engine size and storage tank capacity. However, for the systems to be considered in this report, two additional, interrelated requirements affect the supply system design. These are: (1) system thrust level, in terms of the maximum number of engines firing simultaneously; and (2) the maximum system impulse level expended during any single maneuver. These are important because they affect pump, pressurization and liquid accumulator design within the RCS.

As shown in Reference (g), the system must be capable of sustaining a maximum thrust of 5750 lbf, or five engines firing simultaneously. This corresponds to the use of four control engines for a translation maneuver and the equivalent of one additional engine for vehicle attitude control during the maneuver. The maximum total impulse during any single maneuver is shown in Reference (g) to be 166,000 lb-sec, based on coelliptic ΔV requirements for a resupply mission. These added constraints set the design criteria for the propellant supply system and establish an envelope for tradeoffs between pump flowrates and the storage capacities of the high pressure liquid accumulators. For example, in a system designed without liquid accumulators, the pumps must be capable of satisfying flow demands

APS DESIGN STUDY ORBITER VEHICLE

MDC Phase B Orbiter



WEIGHT (INSERTION)	331,780 LB
PAYLOAD	65,000 LB
LENGTH	174.7 FT
NUMBER OF THRUSTERS	33
THRUST LEVEL	1,150 LB
IMPULSE REQUIRED	
LIMIT CYCLE	150,000 LB-SEC
ATTITUDE MANEUVER AND DAMPING	830,000 LB-SEC
MANEUVERS (– 20 FT/SEC)	1,270,000 LB-SEC
TOTAL	2,250,000 LB-SEC

Figure 2-1

for full system thrust. Conversely, if the high pressure liquid accumulators are sized to provide 166,000 lb-sec, the pump flow rates required during the maneuver would be zero. However, minimum pump flow requirements are set by the average system thrust level during entry which is shown by Reference (g) to be 250 lb. Single burn total impulse and system thrust requirements can then be satisfied with a pump flow equivalent to 250 lb thrust and accumulator capacities equivalent to 157,000 lb-sec total impulse. After a maneuver, the pumps would continue to operate, recharging the accumulators to full capacity in approximately 17 minutes. Increases in pump flow capacity result in proportional reductions in accumulator capacity. Figure 2-2(a) provides the relationships between pump flow and accumulator capacity necessary to satisfy both system thrust and maximum single burn impulse requirements.

A final vehicle requirement affecting design of the RCS supply is the total impulse expended for attitude control. As defined in Reference (g), 982,000 lb-sec of total impulse is consumed for attitude control. This impulse expenditure and the capacity of the high pressure liquid accumulator dictate the number of operating cycles required for the pumps and accumulators per mission. In the extreme, with no liquid accumulator, pump operation would be required for each engine firing, imposing severe demands on pump cycle life. However, even small liquid accumulators effect marked reductions in the number of cycles. Figure 2-2(b) relates the number of pump-accumulator operating cycles in each mission to the storage capacity of the liquid accumulator.

EFFECT OF ACCUMULATOR CAPACITY ON
PUMP REQUIREMENTS

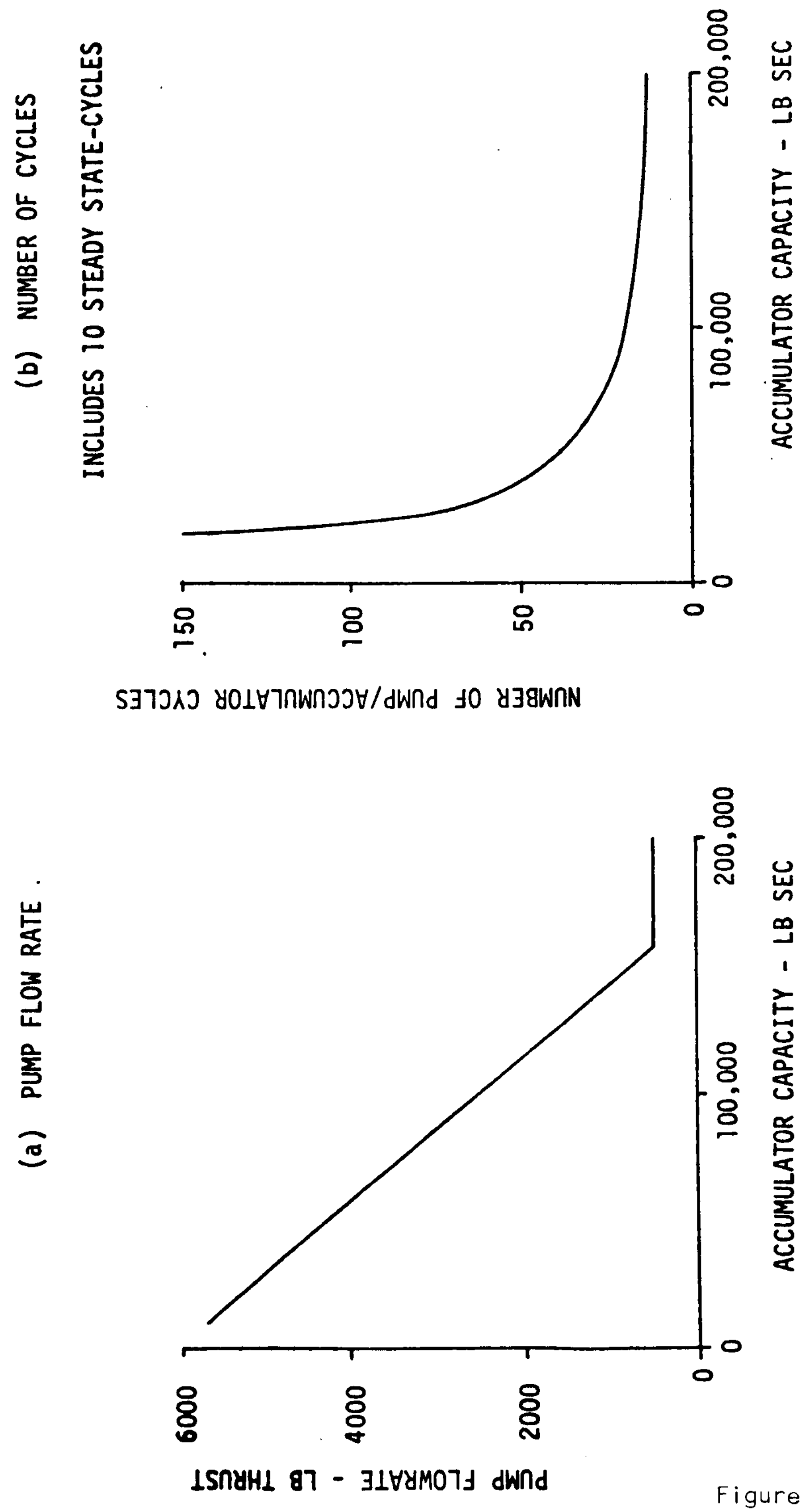


Figure 2-2

3. SUMMARY OF EFFORT

The following paragraphs provide summary results for both the gaseous and the liquid RCS concepts considered in this phase of the study. Only summary data, comparing alternate means of gaseous RCS implementation,are provided in Paragraph 3.1 below. Details on the gaseous designs, weight breakdowns,and optimization studies are presented in Appendix B. More detail is provided for the liquid RCS (Paragraph 3.2) because its propellant distribution features are unique and because it proved to be the more attractive of the two system concepts but, again, only an overview of the results is provided. Liquid RCS distribution system thermal analyses are covered in Appendix C and liquid system design optimizations are provided in Appendix D.

Component weight and performance models and the subassembly trade studies used to develop system weights are provided in Appendix A. The gaseous propellant distribution system features (lines and insulation) used for the gaseous RCS were identical to those used in Reference (h), and are therefore not duplicated in this report.

Details of hydrogen pump preliminary design effort provided by the Pesco Products Division of Borg-Warner Corporation (now a division of Sunstrand Corporation) are given in Appendix E.

3.1 Gaseous Oxygen-Hydrogen System Results - The motivation to explore alternate gaseous system approaches stemmed from the technology concerns associated with turbopumps for a reusable system application requiring many restarts during each mission. Turbopump life requirements for the Shuttle (Reference (h)) were placed at approximately 50 starts per mission, making a total life requirement of 5000 cycles for the 100 mission shuttle vehicle. This requirement is approximately two orders of magnitude greater than demonstrated by any turbopump in an aerospace application. The basic concept posed to eliminate this problem was illustrated in Figure 1-1. The approach postulated a small motor driven pump, powered by the Shuttle auxiliary power unit (APU), in combination with a liquid accumulator to satisfy thrust and impulse requirements during translation maneuvers. However, the concept illustrated is but one of a generic series derived from the basic "turbopump" RCS. The most fundamental approach being a fully pressurized system which was used as the starting point for study of gaseous systems.

Simplified schematics of the alternate gaseous systems are shown in Figure 3-1, which also illustrates their evolution from the fully pressurized concept. Complete schematics were developed for each of these systems with all of the

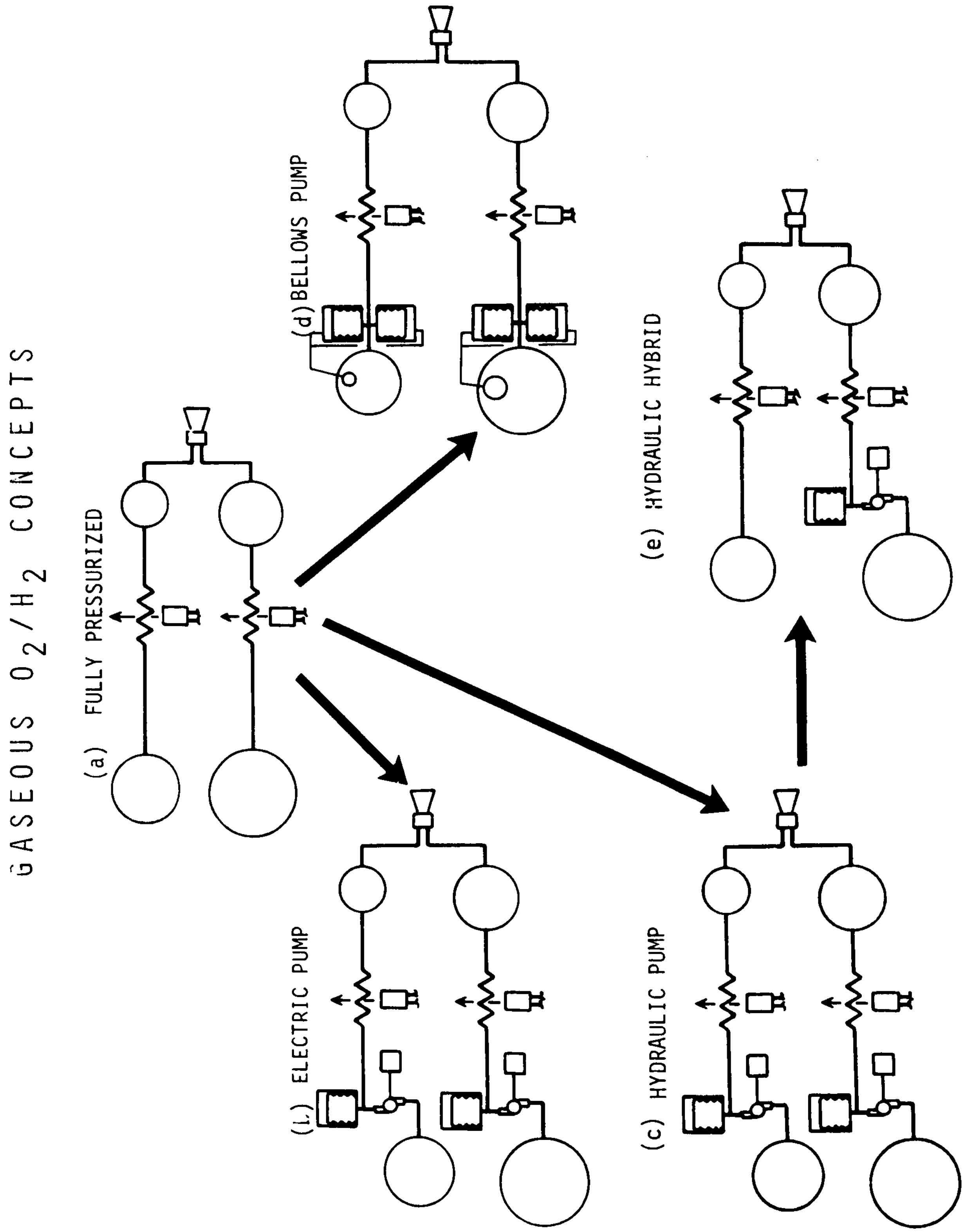

GASEOUS O_2/H_2 CONCEPTS
(a) FULLY PRESSURIZED
(b) ELECTRIC PUMP
(c) HYDRAULIC PUMP
(d) BELLOWS PUMP
(e) HYDRAULIC HYBRID

Figure 3-1

components necessary to satisfy Shuttle fail-safe, fail-safe criteria. Design analyses were then performed to evaluate weight sensitivities and to define the system design points which would provide minimum weight. These analyses are contained in Appendix C and the resulting system weights, at the selected design points, are shown in Figure 3-2. For reference purposes, Figure 3-2 also shows the weight of a turbopump system with perfect controls as defined in Reference (h).

As shown in Figure 3-2, the weight penalty for full pressurization of both the hydrogen and oxygen is severe (5,000 lb). In a system of this type the tankage and pressurization system weight penalties are compounded by the blowdown pressure ratio required in the gaseous accumulators. The allowable number of heat exchanger operating cycles controls the capacity of the gas accumulators. Recent technology programs (Reference (1)) indicate that heat exchanger designs with a thermal cycle life of 20,000 cyles (200 accumulator recharge cycles per mission) are feasible. This value, while significantly greater than the 50 cycle per mission design life used for turbopump systems, still requires a significant accumulator volume and blowdown pressure ratio. Thus, in contrast to conventional pressure fed systems, where tank pressures exceed chamber pressure only by line/component pressure loss, this system design must also directly amplify tank pressure by the accumulator blowdown ratio. At the design point shown, this effect has been minimized by optimizing the accumulator design (following the procedures of Reference (h)). The design weight shown is at a very low chamber pressure of 50 psia.

The severe weight penalties in tankage and pressurization, that occur with full pressurization, can be reduced by incorporating a pump in the system as shown by schematics (b) and (c) of Figure 3-1. Either electric motor or hydraulic motor drives can be considered. With this approach the gas accumulators are designed, as before, for 200 cycles per mission, and the heat exchangers operate during each recharge cycle. The pump systems can be designed with or without liquid accumulators. Without liquid accumulators, the pumps and their power supply must operate each time the heat exchanger is used, i.e., 200 times per mission. However, if a liquid accumulator is also used, it can provide sufficient liquid for several gas accumulator recharge cycles. This would reduce cycle life requirements on the pump and its power supply and also allow a significant relaxation in pump response requirements. The resulting system weights for the two design approaches are shown in Figure 3-2. These weights include power supply penalties

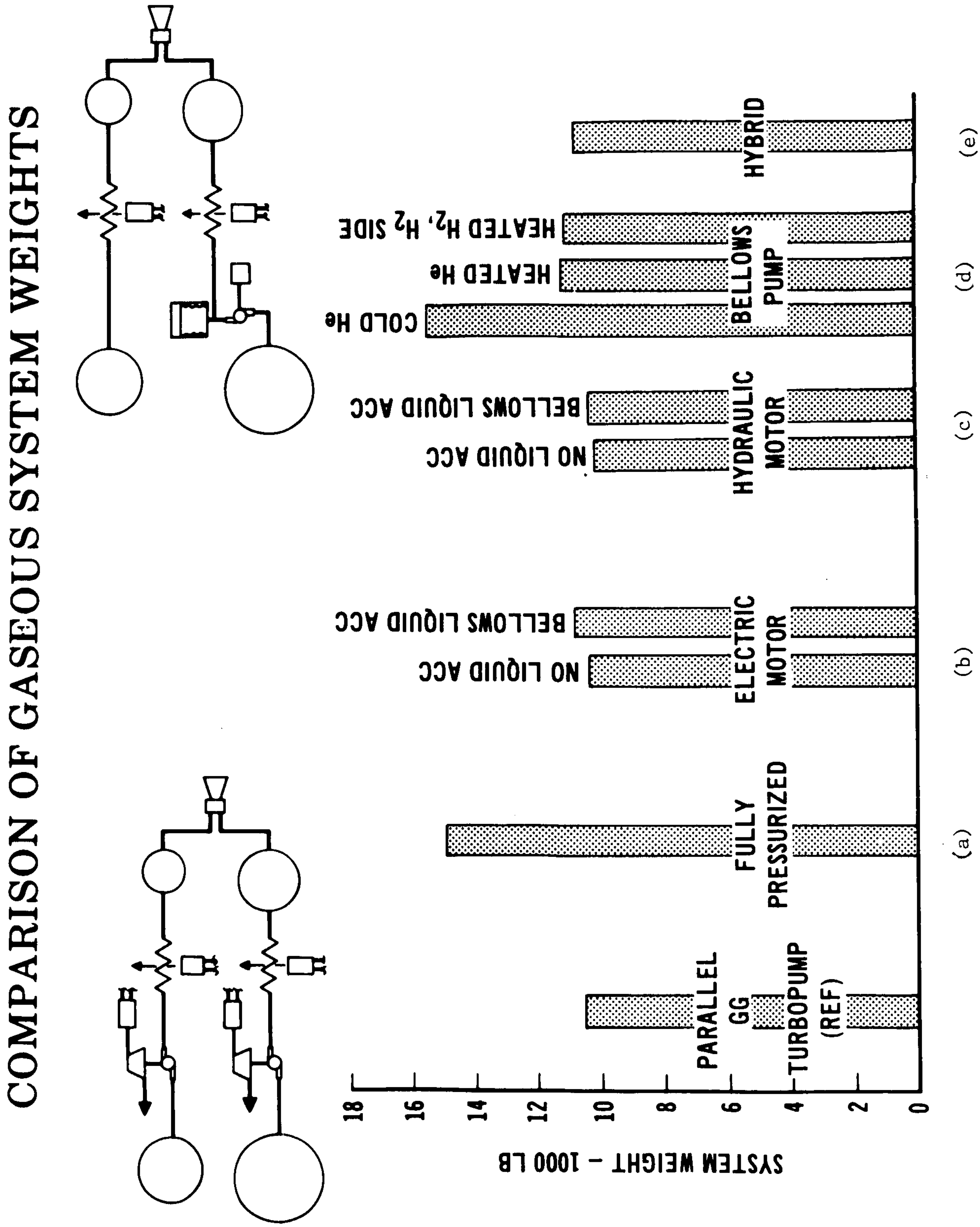

COMPARISON OF GASEOUS SYSTEM WEIGHTS
SYSTEM WEIGHT – 1000 LB
0
2
4
6
8
10
12
14
16
18
PARALLEL GG TURBOPUMP (REF)
FULLY PRESSURIZED
ELECTRIC MOTOR
NO LIQUID ACC
BELLOWS LIQUID ACC
HYDRAULIC MOTOR
NO LIQUID ACC
BELLOWS LIQUID ACC
BELLOWS PUMP
COLD He
HEATED He
HEATED H_2, H_2 SIDE
HYBRID
(a)
(b)
(c)
(d)
(e)

Figure 3-2

associated with both APU propellant and power rating increases, where required. The available APU hydraulic power was 20 horsepower; any power in excess of this level incurred appropriate weight penalties. Comparison of the motor driven pump systems shows that their weights are competitive with the reference turbopump system. Their principal drawback is their dependence on APU operation and the increased number of APU starts incurred.

The undesirable feature of pump system dependence on APU operation is eliminated by concept (d) of Figure 3-1. This system uses two pressurized, bellows type liquid accumulators for each propellant. System controls are such that when one accumulator is depleted its pressurant is vented and refill is provided under the storage tank pressure head. During vent and refill system propellant requirements are met by the alternate accumulator. When it is depleted, the supply is shifted back to the full accumulator and the same sequence of operations is repeated. Weights for this system concept are shown in Figure 3-2 for three alternate pressurants. Use of cold helium for the pressurant results in weight penalties almost identical to those for a fully pressurized system because the gas pressure-volume requirements for expulsion are almost equal. With this system approach, however, it is entirely feasible to use a heated pressurant because: (1) the liquid accumulators and heat exchangers always operate in unison, and (2) the accumulators are at high pressure and some heat transfer to the cryogenic liquids is tolerable. Both heated helium and heated hydrogen (H_2 accumulators only) were evaluated and the resulting system weights are shown in Figure 3-2. As shown, both systems are weight competitive because, by heating the pressurant, vent losses are greatly reduced.

The final gaseous system concept investigated was a hybrid approach that introduced some compromise in weight in the interest of minimizing technology risk. This approach is illustrated by schematic (e) of Figure 3-1. The oxygen side of the system operates fully pressurized and a hydraulic motor pump-liquid accumulator combination is used for the hydrogen. Use of fully pressurized oxygen results in some weight penalty, but since the oxygen is less than 20% of the total propellant volume, this penalty is small and allows simplification of system design and development. As shown by Figure 3-2 the weight penalty with the hybrid approach is small and it is competitive with the other low weight system concepts.

The above gaseous system concepts are compared in Figure 3-3. Relative system weights for design points at both 40 and 200 cycles per mission are shown as are a qualitative identification of APU interactions and features of significance

COMPARISON OF GASEOUS SYSTEM CONCEPTS

SYSTEM	RELATIVE WEIGHT - LB 40 HX CYCLES	 200 HX CYCLES	APU INTERACTIONS	REMARKS
TURBOPUMP (PARALLEL GG)	0	-	NO APU INTERACTIONS	LIGHTEST, MOST EFFICIENT GASEOUS SYSTEM
HYBRID HYDRAULIC	+650	+220	225 HORSEPOWER REQUIRED WHICH IS WITHIN CAPABILITY OF 1 PHASE B SIZE APU UNIT OR TWO CURRENT SHUTTLE APU UNITS	USES FULLY PRESSURIZED O_2 SYSTEM, REQUIRES DEVELOPMENT OF POSITIVE DISPLACEMENT HYDRAULIC MOTOR DRIVEN PUMP
FULLY HYDRAULIC H_2 AND O_2	+173	-520	WOULD UTILIZE FULL HYDRAULIC POWER CAPABILITY OF ONE APU UNIT (PHASE B SIZE) OR TWO APU UNITS (CURRENT SHUTTLE SIZE)	INCREASED COMPLEXITY ASSOCIATED WITH PUMPING O_2 DOES NOT SEEM TO JUSTIFY THE POTENTIAL WEIGHT SAVINGS INDICATED
ELECTRIC MOTOR OPERATED	+616	-233	POWER REQUIREMENTS EXCEED CURRENT APU CAPABILITY BY NEARLY 10 TIMES.	POWER REQUIREMENTS ARE VERY HIGH FOR ELECTRICAL POWERED SYSTEM, WOULD REQUIRE DEVELOPMENT OF 275 HP ALTERNATORS
BELLOWS PUMP (HEATED He)	——	+660	NO APU INTERACTIONS	UNIQUE TYPE OF PUMP APPEARS RELATIVELY SIMPLE. WOULD REQUIRE THERMAL ISOLATION OF HOT HELIUM AND PROPELLANT
FULLY PRESSURIZED	——	+4355	NO APU INTERACTIONS	SIMPLE SYSTEM, BUT HEAVY EVEN FOR LOW (50 PSI) CHAMBER PRESSURES. CONTRIBUTING FACTOR IS GASEOUS ACCUMULATOR P-V-WT RELATIONSHIPS.

Figure 3-3

to the systems. In general, several system approaches are seen to be weight competitive with the turbopump concept and potentially offer less development risk. Of all the systems, lowest weight is realized with hydraulic driven motor-pumps for both hydrogen and oxygen. The pumps in this system could be designed for low acceleration rates during start up with small weight penalties, since liquid accumulators are less weight sensitive to capacity than the earlier gaseous accumulators. In addition, the hydraulic motor-pump system can be simplified and its development cost reduced by fully pressurizing the oxygen supply, thus eliminating the oxidizer pump and accumulator. Further simplification in terms of operation can be achieved by using the bellows pump approach to eliminate dependency and interaction with the APU.

3.2 Liquid Oxygen-Hydrogen System Results - The remaining study effort was directed toward a liquid oxygen/hydrogen system. Liquid oxygen/hydrogen APS concepts were explored to determine the feasibility of using a simplified RCS in which propellants are distributed to the engines as liquids. This approach would allow the gas generators, propellant heat exchangers, and gaseous accumulators to be removed, as depicted in Figure 3-4. In addition to advantages in system simplicity, the liquid concept offered a large system weight reduction by eliminating the heavy gaseous accumulators and by avoiding gaseous propellant conditioning losses.

Liquid cryogenic systems were not considered in previous studies due to concerns associated with propellant heating in a large, relatively complex distribution system and because of concerns with engine pulse mode ignition. Operating regimes for the liquid concept are compared to other concepts considered in previous studies (Figure 3-5). Previous studies restricted engine inlet temperatures to approximately 200°R, while the liquid systems deliver liquids at cryogenic temperatures to the engines. Concerns with the use of liquids stemmed largely from inexperience with liquid cryogenic O_2/H_2 ignition and postulated problems during development and qualification. Cryogenic systems developed to date have used propellants which were stored and distributed near saturation conditions. Thus, any heating resulted in propellant vaporization and large propellant density changes. These latter problems are avoided in the liquid system by operating with highly subcooled liquid propellants, which can absorb a large heat input without propellant vaporization. In fact, the hydrogen would be delivered supercritically, ensuring that two phase flow would not occur. Thus, the liquid system design differed from that for the gaseous systems in that a large

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LIQUID SYSTEM STUDIES

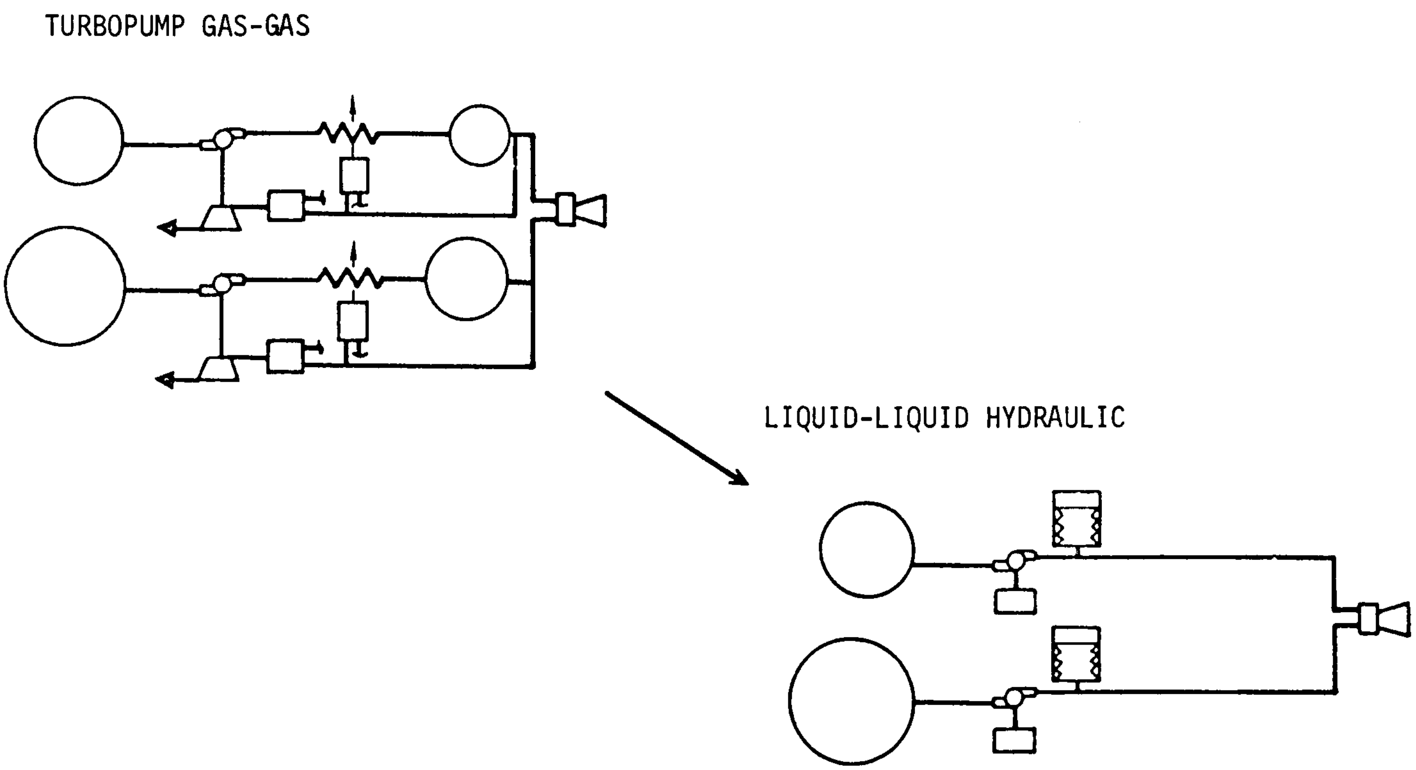


Figure 3-4

O_2/H_2 CONCEPT OPERATING REGIMES

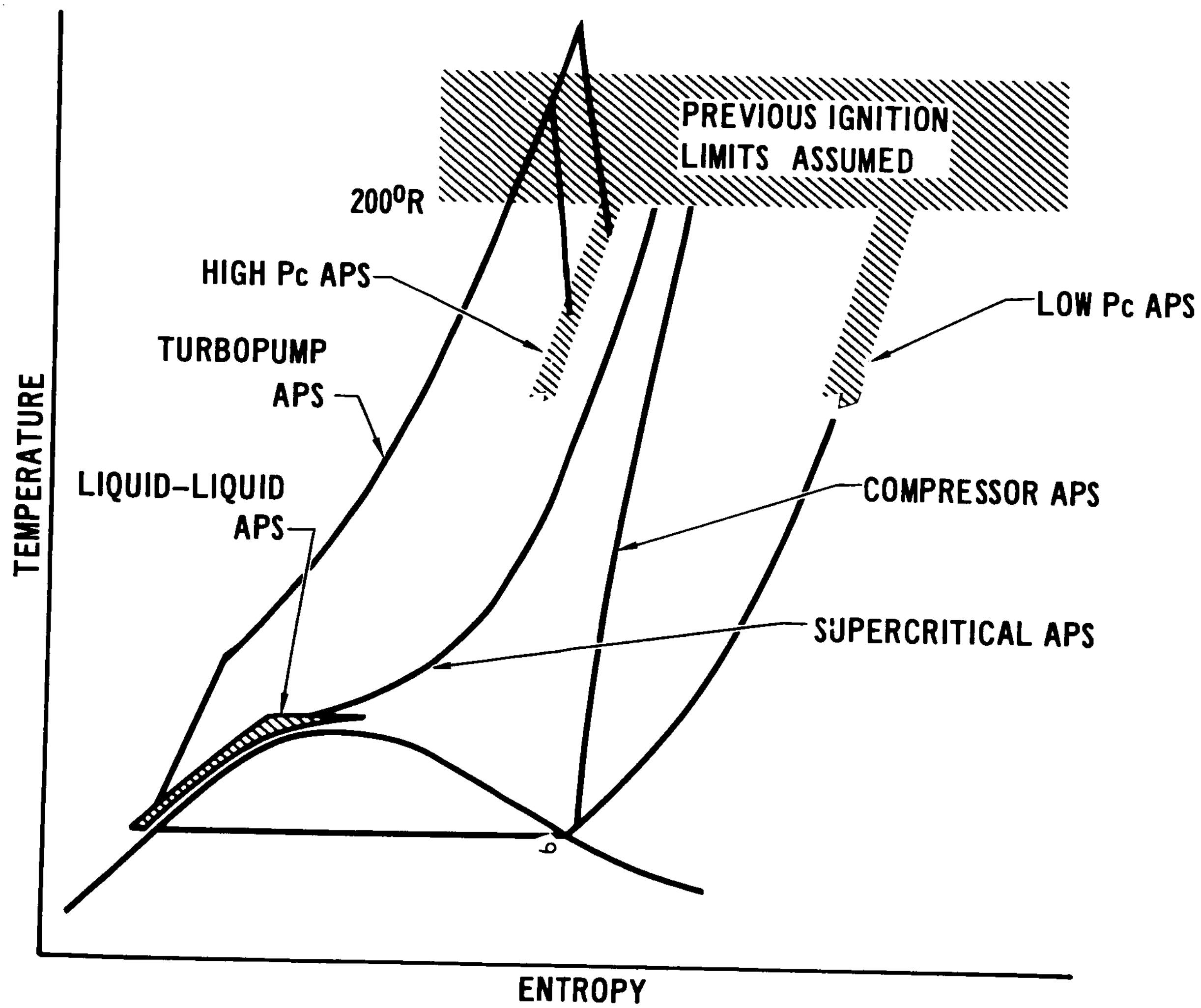


Figure 3-5

portion of the effort was directed toward detailed thermal analyses of the propellant distribution. To ensure validity of study results, a detailed thermal model, described in Appendix C, was developed.

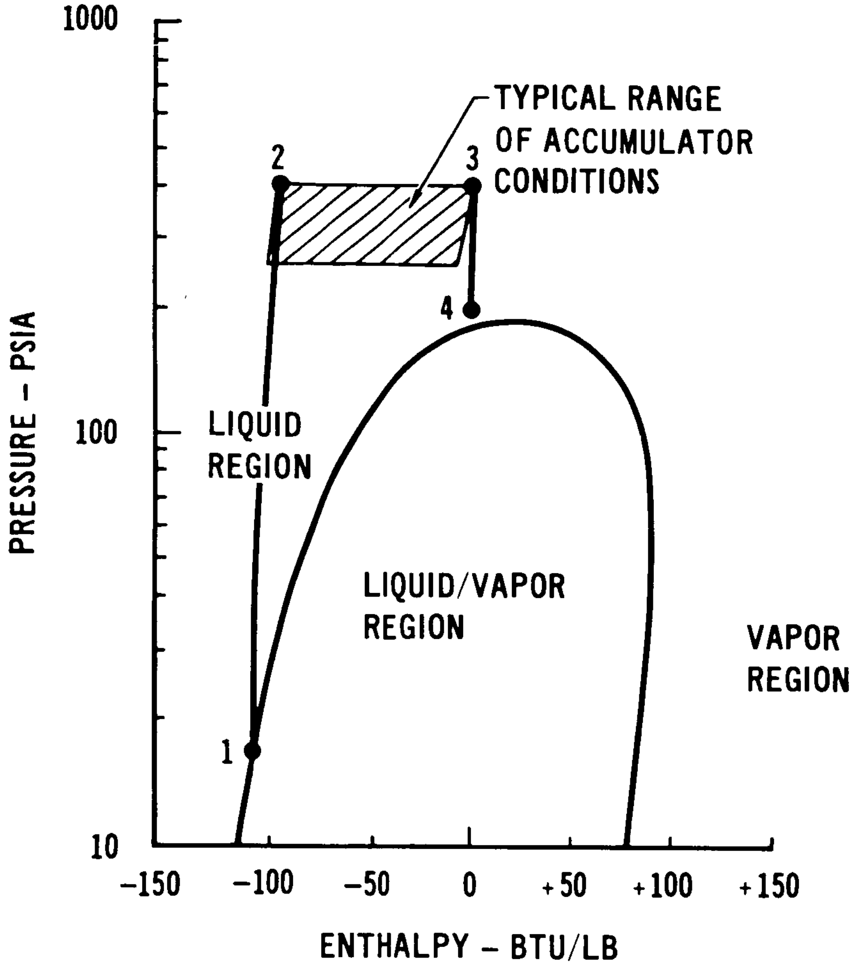
Complete schematics, including redundancy were developed and detailed design analyses were performed to evaluate system weight sensitivities and to define design points for minimum system weight. The significant study findings and rationale for the selected system are given below. The detailed design studies and trades are presented in Appendix D. In addition, a subcontract was issued to Pesco Products for comparison of hydrogen pump types and definition of pump weights and performance. The hydrogen pump study results are presented in Appendix E.

3.2.1 Liquid System Operation - The liquid hydrogen storage conditions are shown in Figure 3-6. Hydrogen is stored at saturated conditions and pumped into a liquid accumulator at supercritical pressures. The accumulators operate in a blowdown mode and their pressure levels are selected to provide minimum system weight, considering the pressure margins required to ensure that liquids will be delivered to the engines when the propellant is throttled to pressures as low as 200 psia (minimum chamber pressure). For these conditions, the hydrogen density changes from 4.4 to 3 lb/ft^3. Liquid oxygen storage, shown in Figure 3-7 uses subcooled propellant provided by either pumping or helium pressurization. Although the oxygen is stored subcritically, as much as 25 BTU/lb could be absorbed without two phase operation. For these conditions, the oxygen density could vary from 72 to 60 lbf/ft^3.

Two primary questions arise with the use of liquid propellants. These are shown in Figure 3-8. The first question is: "Can a liquid distribution system be designed to provide sufficiently low propellant heating such that density changes are held to levels low enough for satisfactory engine operation?" It can be seen from the inset figure that at all pressure levels of interest significant hydrogen density changes will occur if hydrogen temperature changes are not controlled. This question was addressed fully in the study and will be described in the following paragraphs. The second question is: "Is liquid ignition feasible in a pulse mode engine?" This question was beyond the scope of this study and will not be covered in this report. However, although the ignition temperature limits are significantly lower than those previously considered, a review of ignition phenomena with engine manufacturers revealed no fundamental reasons that would make liquid igntion doubtful. Additionally, NASA comtemplates technology programs by engine manufacturers to fully define engine ignition aspects related to system design.

LIQUID HYDROGEN STORAGE CONDITIONS

ΔH = 100 BTU/LB
DENSITY RANGE: 4.4 TO 3 LB/FT^3



1 – LIQUID STORAGE CONDITIONS

2 – LIQUID PUMPED TO 400 PSIA AND STORED IN ACCUMULATOR

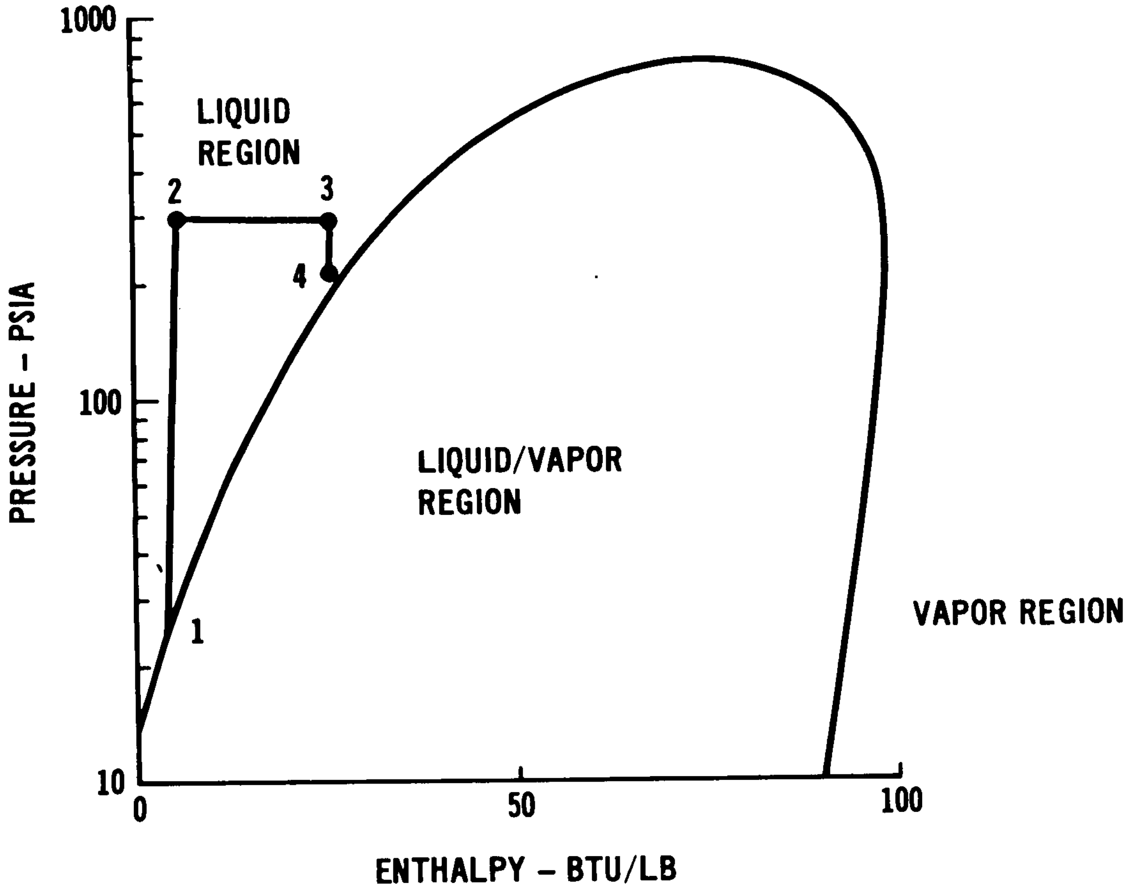
3 – HEAT ADDITION FROM ENVIRONMENT RAISES TEMPERATURE FROM 40 TO 65°R

4 – PROPELLANT USED IN ENGINE AT CHAMBER PRESSURE OF 200 PSIA

Figure 3-6

LIQUID OXYGEN STORAGE CONDITIONS

ΔH = 25 BTU/LB
DENSITY RANGE: 72 TO 60 LB/FT^3



1 – SATURATED PROPELLANT

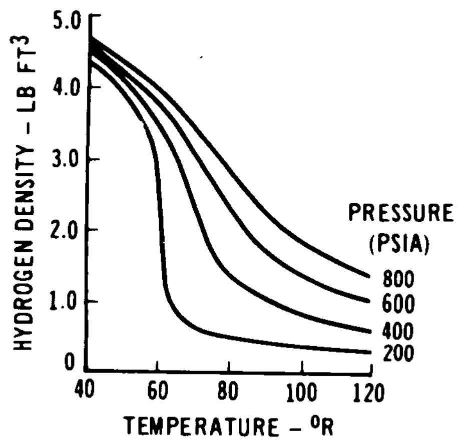
2 – OXYGEN PRESSURIZED TO 300 PSIA USING REGULATED COLD HELIUM PRESSURIZATION OR MOTOR DRIVEN PUMP

3 – HEAT ADDITION FROM ENVIRONMENT RAISES TEMPERATURE FROM 168°R TO 225°R

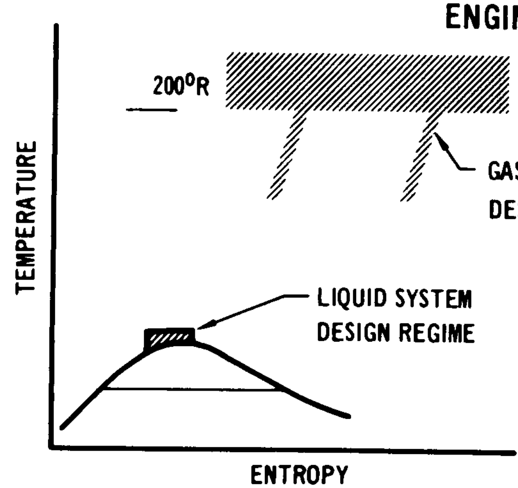
4 – PROPELLANT USED IN ENGINE AT CHAMBER PRESSURE OF 200 PSIA

Figure 3-7

PREDOMINATE LIQUID SYSTEM QUESTIONS



FEED SYSTEM DESIGN – COULD A PRACTICAL DISTRIBUTION SYSTEM BE DESIGNED AND WOULD THAT DESIGN PROVIDE SUFFICIENTLY LOW PROPELLANT HEATING LEVELS FOR ENGINE OPERATION?



ENGINE IGNITION – WAS LIQUID IGNITION FEASIBLE IN A PULSE MODE ENGINE?

Figure 3-8

3.2.2 Liquid System Thermal Analyses - The first step in the analysis was to establish the limits on propellant temperature rise and then to examine the practicality of the limits with respect to the allowable line heat leak with varying rates of propellant usage. The effect of propellant temperature changes on engine operating characteristics are shown in Figure 3-9. The major effect is an increase in engine mixture ratio as hydrogen temperature increases. The effects of oxygen temperature, for the range of interest, are minor. To maintain engine mixture ratios below 5.0, the hydrogen temperature must be limited to approximately 65°R. At this temperature, both thrust level and specific impulse are reduced 3% below the design values, primarily due to mixture ratio changes. The only other parameter found to be affected by temperature changes was the injection velocity. As propellant temperatures increase, the injection velocity increases, but again, if the hydrogen temperature is limited to 65°R, these changes are minimal.

The practicality of this temperature limit was examined by determining equilibrium temperatures as a function of heating rate and hydrogen usage rate. These results are shown in Figure 3-10. The smallest hydrogen usage rate is associated with a $\pm 20°$ limit cycle. This would result in temperatures above 65°R for anticipated heating rates in excess of 25 BTU/hr. In order to limit the hydrogen temperature to 65°R, the usage rate could be increased. For example, increasing the usage rate by a factor of 3 (by decreasing the deadband to $\pm 6°$) would result in an equilibrium hydrogen temperature less than 60°R. This higher propellant utilization introduces a weight penalty on the order of 1/2 pound per hour. Shuttle vehicle studies have shown that a limit cycle deadband of $\pm 5°$ is desirable. At this design point,no weight penalty would be incurred. All other operating conditions, such as fine attitude control or attitude control maneuvers, require much larger usage and would remove sufficient propellant to carry away the incoming heat, thereby maintaining chilled propellant and lines.

The data shown in Figure 3-10 corresponds to an equilibrium condition where the entire heating rate was uniformly distributed through the total propellant mass in the feed system. In order to determine actual temperature gradients and peak heating rates, the manifold shown in Figure 3-11 was designed. The manifold consists of vacuum jacketed propellant lines with high performance (HPI) insulation between the inner and outer lines. One manifold is provided forward and one manifold aft in the vehicle. A complete ring manifold was used to eliminate trapped or stagnant propellant regions. Usage of any engine will result in propellant flow through both sides of the manifold thus circulating all propellant in the

EFFECT OF PROPELLANT DENSITY ON ENGINE PERFORMANCE

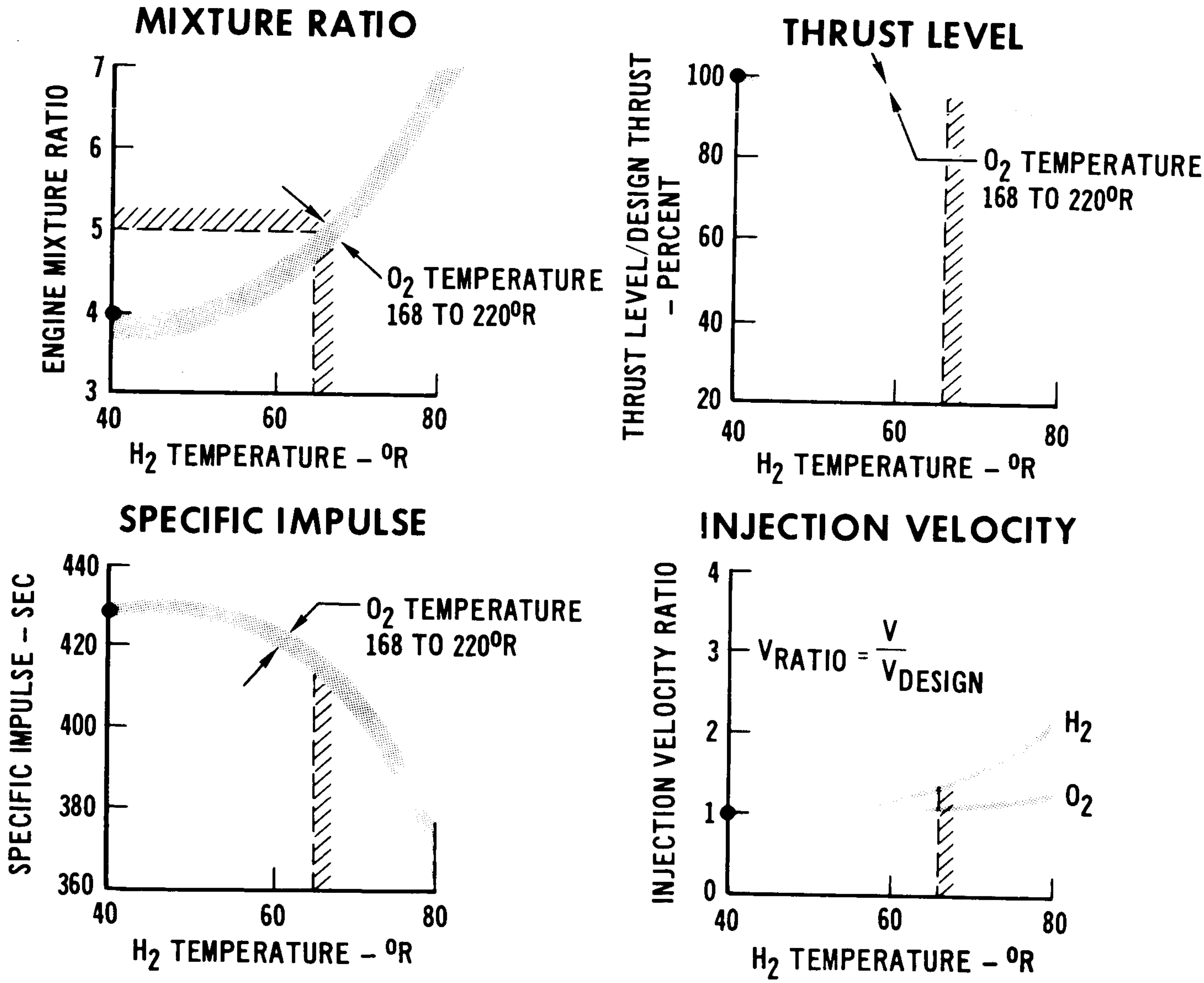


Figure 3-9

EFFECT OF PROPELLANT CONSUMPTION ON HYDROGEN TEMPERATURE

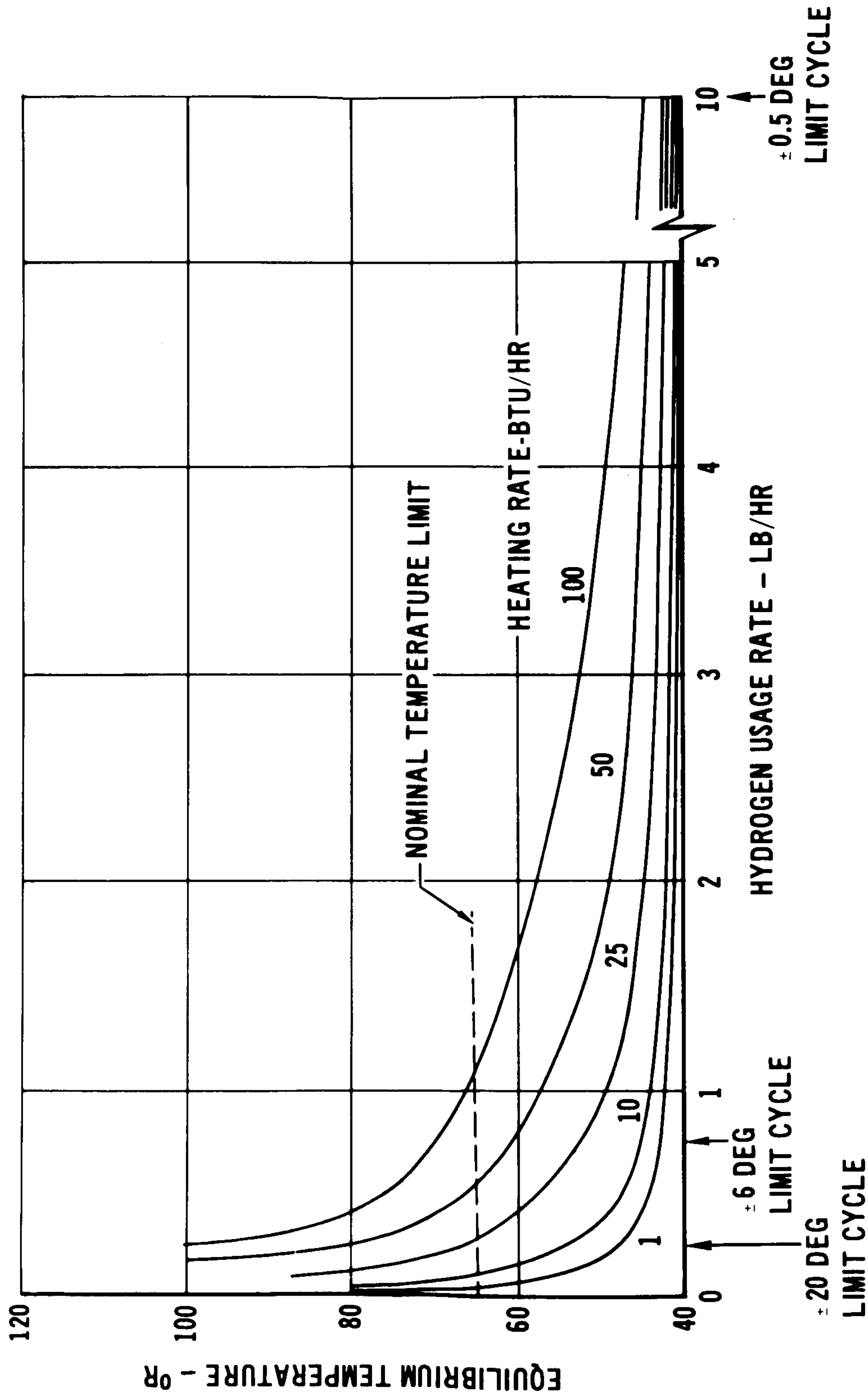


Figure 3-10

LIQUID H2 FORWARD MANIFOLD INSTALLATION

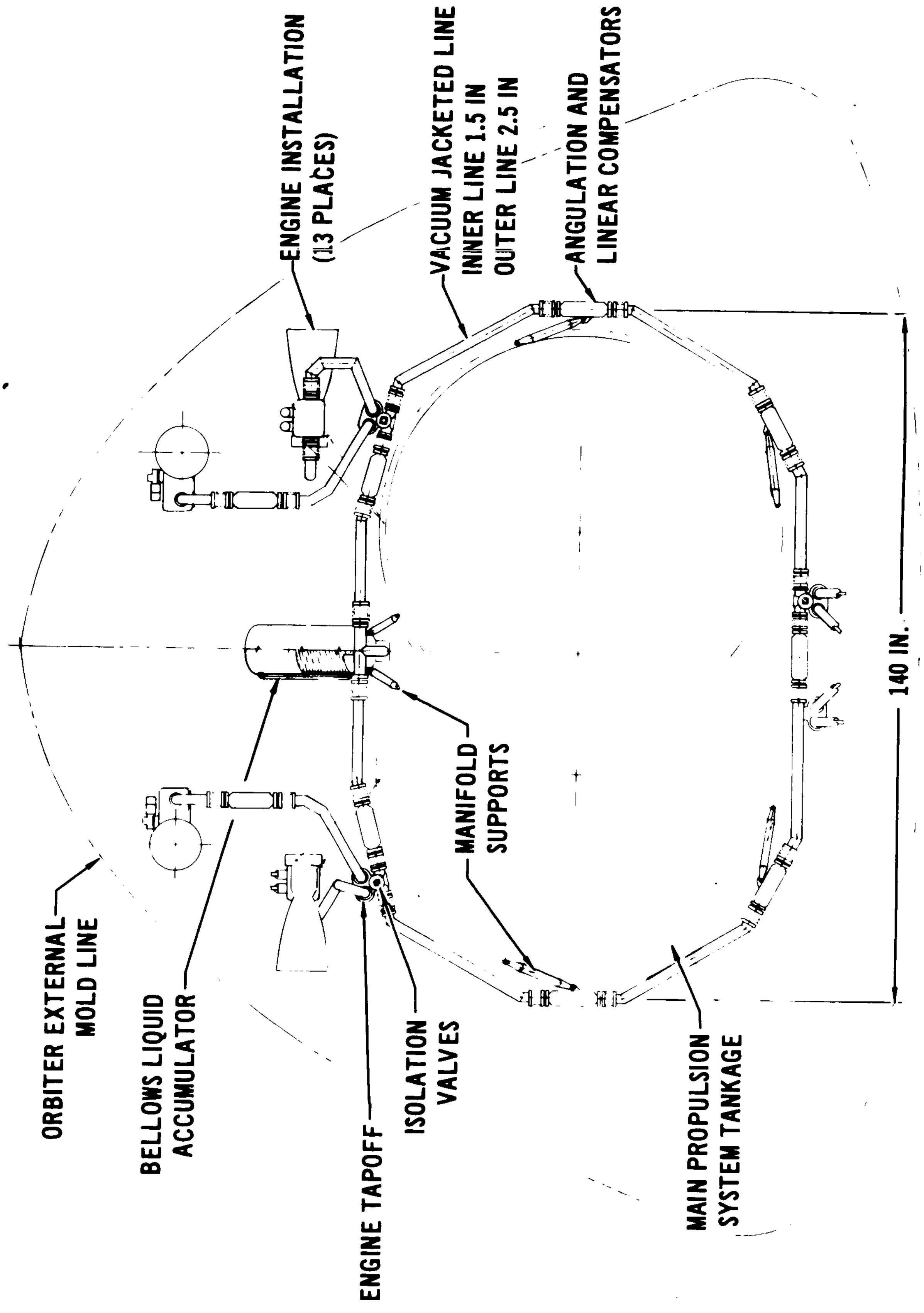


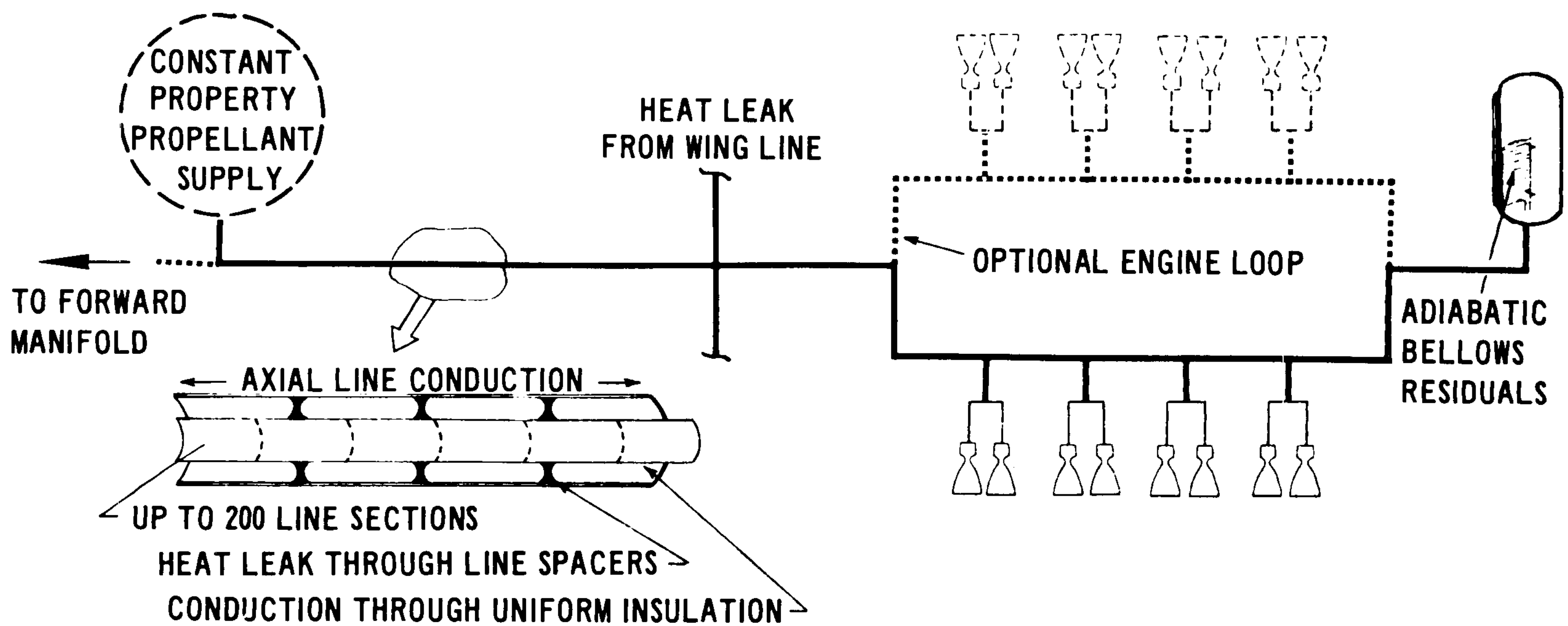
Figure 3-11

manifold with each engine firing. The outer line is maintained at approximately 520° by the vehicle surroundings. The inner line, however, is chilled to 40°R during filling. Both angular and linear compensators are provided to accommodate thermal contraction.

A thermal model was set up to evaluate, in detail, all of the significant heat transfer effects. The thermal model is summarized in Figure 3-12 and described in detail in Appendix C. As shown, up to 200 individual line sections or nodes could be analyzed. Heat input considerations included heat leak through inner line spacers (required to prevent HPI crushing) and axial conduction down the lines. Aluminum feed lines were used in order to conduct the incoming heat away as rapidly as possible from heat short locations and minimize local peak temperatures. Conduction through the liquid was neglected as this was found to contribute little to the thermal characteristics. Both real fluid and real line properties were utilized. The distribution system model considered the main supply line from the propellant tank to the manifold and included heat shorts at junctions with branch feed lines to each wing tip and to each engine group. A bellows tank was included to provide for fluid thermal expansion that resulted from heating. The initial analyses of the propellant distribution system were exploratory in nature to determine what, if any, modifications should be made in the feed system design in order to achieve low heating rates.

The hydrogen temperatures from these initial studies obtained are shown in Figure 3-13. It was assumed that no engines were firing and no propellant was used. This conservative assumption results in high localized heating rates. As shown, significant thermal spikes were obtained. However, the maximum temperature at the end of one hour was 60°R. This temperature was much lower than originally anticipated for a condition in which the propellant was stagnant. One of the reasons for the low temperatures is that any heat input, near the tank or upstream in the line, locally expands the propellant, moving it away from the heat source and promoting migration of fresh, cooler propellant into heat short areas downstream. Significant heating is also evident at the wing lines which are tied directly to the main line, and from the inner line supports which, in this case, were spaced at 10 ft intervals along the line. Closer line support spacing was investigated to determine if there would be any significant changes in the heat profile. As shown in Figure 3-14, decreasing the feed line spacing from 10 to 2-1/2 feet does not significantly change the maximum temperature encountered, although it does slightly increase the bulk propellant temperature. Thus, closer line supports could be utilized with little temperature effect.

FEED LINE THERMAL MODEL



THERMAL MODEL

- ISOBARIC HEATING
- SOLUTION OF MASS AND ENERGY CONSERVATION EQUATIONS FOR EACH LINE SECTION USING IMPLICIT DIFFERENCE TECHNIQUE
- REAL FLUID AND LINE PROPERTIES
- HEATING
  - a) UNIFORM INSULATION
  - b) LINE AXIAL CONDUCTION
  - c) LOCALIZED HEAT LEAKS

ENGINE FIRING CAPABILITY

- RESUPPLY FROM PROPELLANT TANK OR BELLOWS TANK IS AVAILABLE
- SINGLE OR MULTIPLE FIRING ON EITHER OR BOTH SIDES OF LOOP
- LINE-VAPOR THERMAL EQUALIZATION AFTER FIRING

Figure 3-12

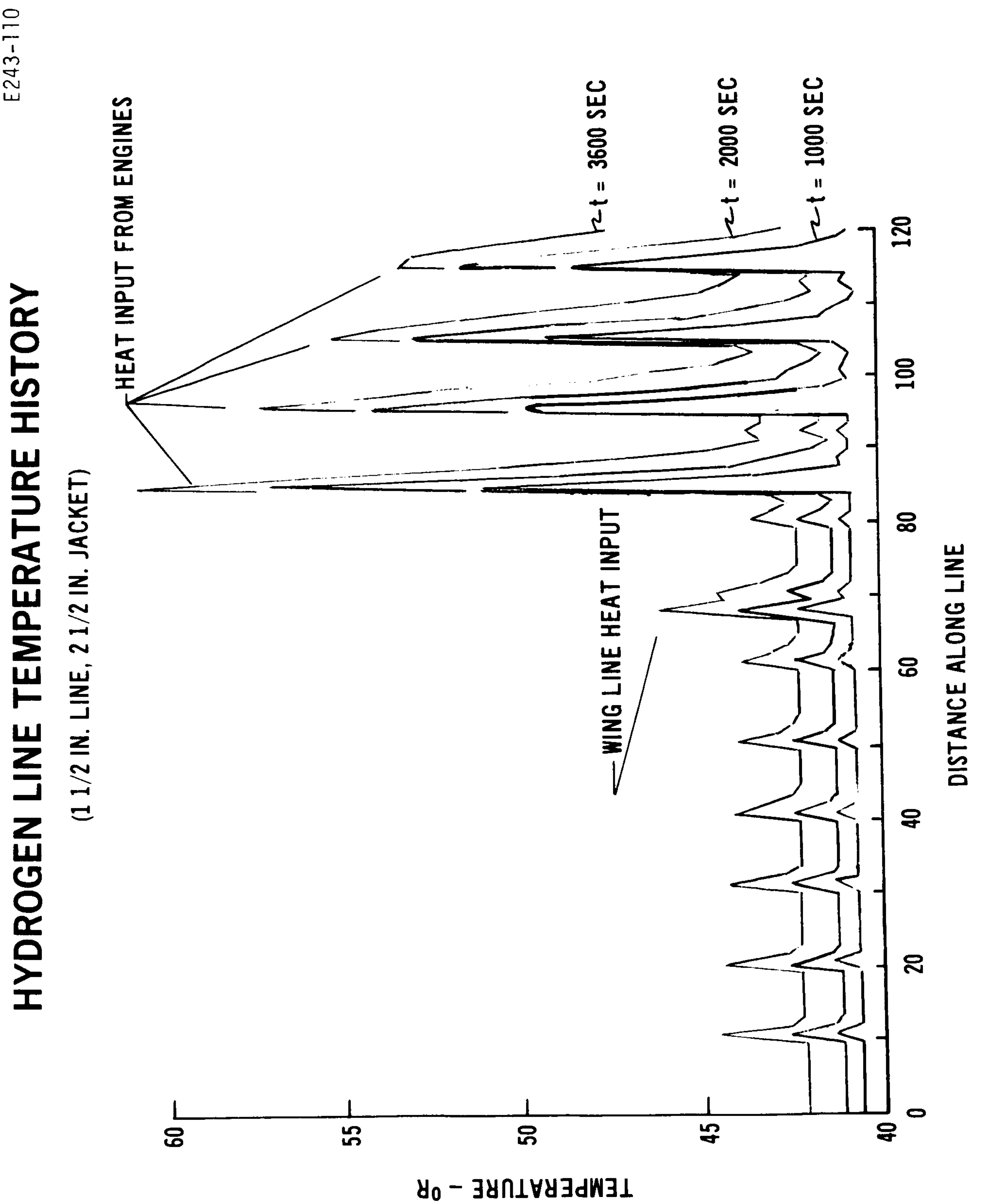

E243-110
HYDROGEN LINE TEMPERATURE HISTORY
(1 1/2 IN. LINE, 2 1/2 IN. JACKET)
HEAT INPUT FROM ENGINES
WING LINE HEAT INPUT
t = 3600 SEC
t = 2000 SEC
t = 1000 SEC
TEMPERATURE – °R
60
55
50
45
40
0
20
40
60
80
100
120
DISTANCE ALONG LINE

Figure 3-13

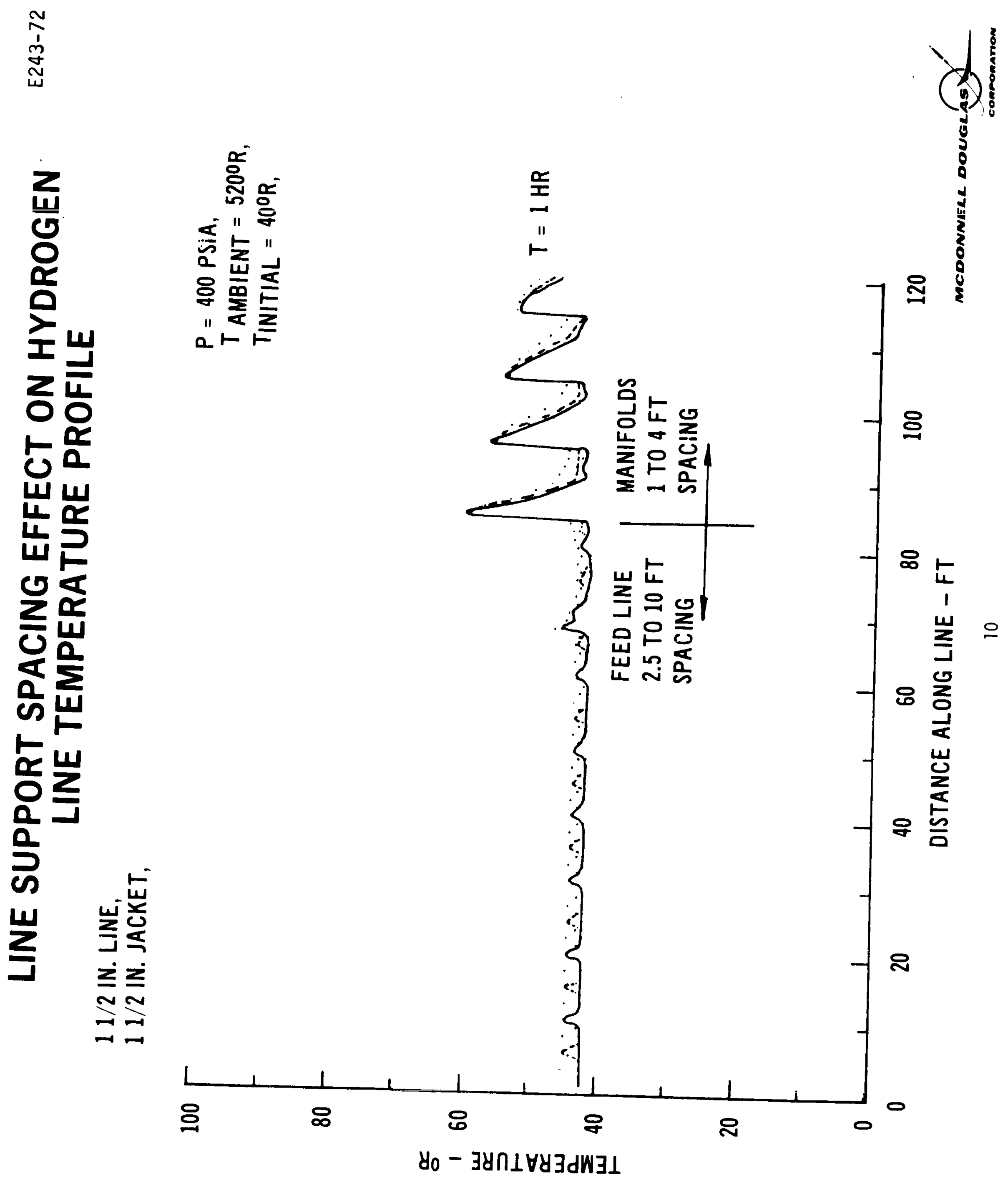

E243-72
LINE SUPPORT SPACING EFFECT ON HYDROGEN LINE TEMPERATURE PROFILE
1 1/2 IN. LINE,
1 1/2 IN. JACKET,
P = 400 PSIA,
T AMBIENT = 520°R,
T INITIAL = 40°R,
T = 1 HR
MANIFOLDS
1 TO 4 FT
SPACING
FEED LINE
2.5 TO 10 FT
SPACING
TEMPERATURE – °R
100
80
60
40
20
0
DISTANCE ALONG LINE – FT
0
20
40
60
80
100
120
10
MCDONNELL DOUGLAS CORPORATION

Figure 3-14

The oxygen temperatures, shown in Figure 3-15, are similar to those of the hydrogen except that the nominal temperature limit (225°R) is not encountered for 20 hours. Again, temperature spikes are seen at each engine line junction and smaller spikes are evident where the wing line joins the main feed line and at each inner line spacer.

The preceeding data indicated that some control of engine heat input was mandatory if satisfactory hydrogen temperature limits were to be achieved. A simple tubular thermal standoff, similar to those commonly employed for hydrazine and hydrogen peroxide engines, was evaluated to determine its effectiveness. The standoff selected was a stainless steel tube 1/4 inch in diameter and 6 inches long. Pressure drop through this tube is on the order of 20 psi. Propellant heating at the engine valve junction with this type thermal standoff is shown in Figure 3-16. As shown, the nominal temperature limit is not achieved until approximately seven hours as opposed to approximately one hour without the thermal standoff. The hydrogen distribution system temperature profile with thermal standoffs employed and simulated propellant usage are shown in Figure 3-17. As shown, the large temperature spikes associated with the engine line junctions have been completely removed and the only significant heating is from the wing line input and inner line supports. Maximum temperatures from these and similar data are shown as a function of time in Figure 3-18. The hydrogen temperature limit of 65° would not be achieved until 10 hours after the start of the mission. However, for a vehicle attitude deadband of approximately 8°, essentially steady-state conditions are reached at 12 to 14 hours into the mission at a hydrogen temperature below the maximum 65°R condition. Beyond this point, no further temperature increase is encountered as the propellant heat input is balanced by the heat removed through propellant usage. The oxygen temperature with thermal standoffs and engine usage are shown in Figure 3-19. These temperatures are much below the 225°R limit assigned.

3.2.3 Liquid System Design Comparison - The preceeding analyses showed that thermal management of liquid propellants in the APS distribution system is feasible if proper attention is given to thermal insulation and isolation of major heat inputs such as thruster heat soakback. The remainder of the study effort was directed toward system design and sizing considerations. The details of system sizing and design trades are presented in Appendix D and pertinent results associated with system sizing and alternate design options are discussed below. A hybrid system, using fully pressurized oxygen and pumped hydrogen, was selected as

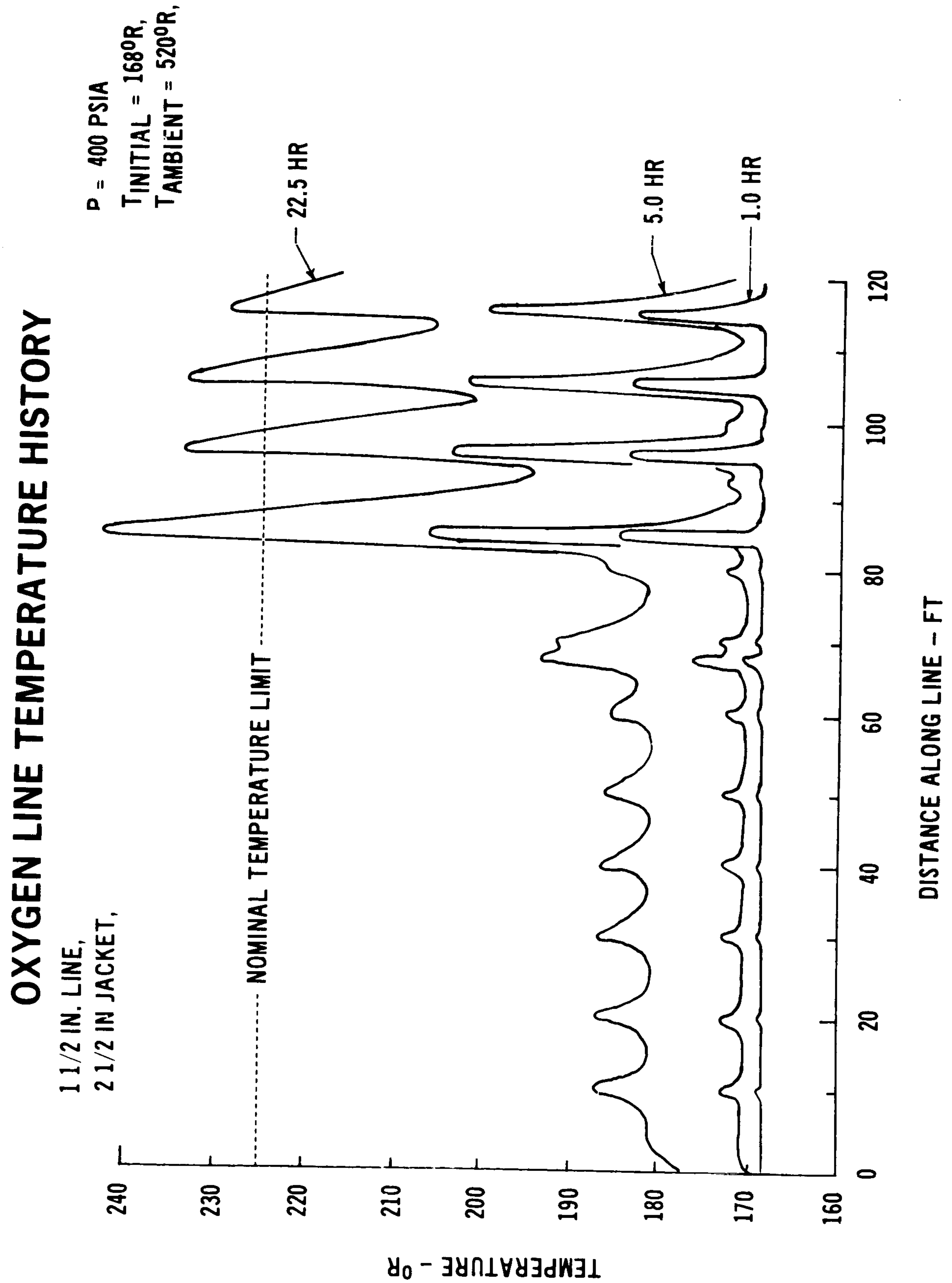

OXYGEN LINE TEMPERATURE HISTORY
1 1/2 IN. LINE,
2 1/2 IN JACKET,
P = 400 PSIA
T_INITIAL = 168°R,
T_AMBIENT = 520°R,
NOMINAL TEMPERATURE LIMIT
22.5 HR
5.0 HR
1.0 HR
TEMPERATURE – °R
240
230
220
210
200
190
180
170
160
DISTANCE ALONG LINE – FT
0
20
40
60
80
100
120

Figure 3-15

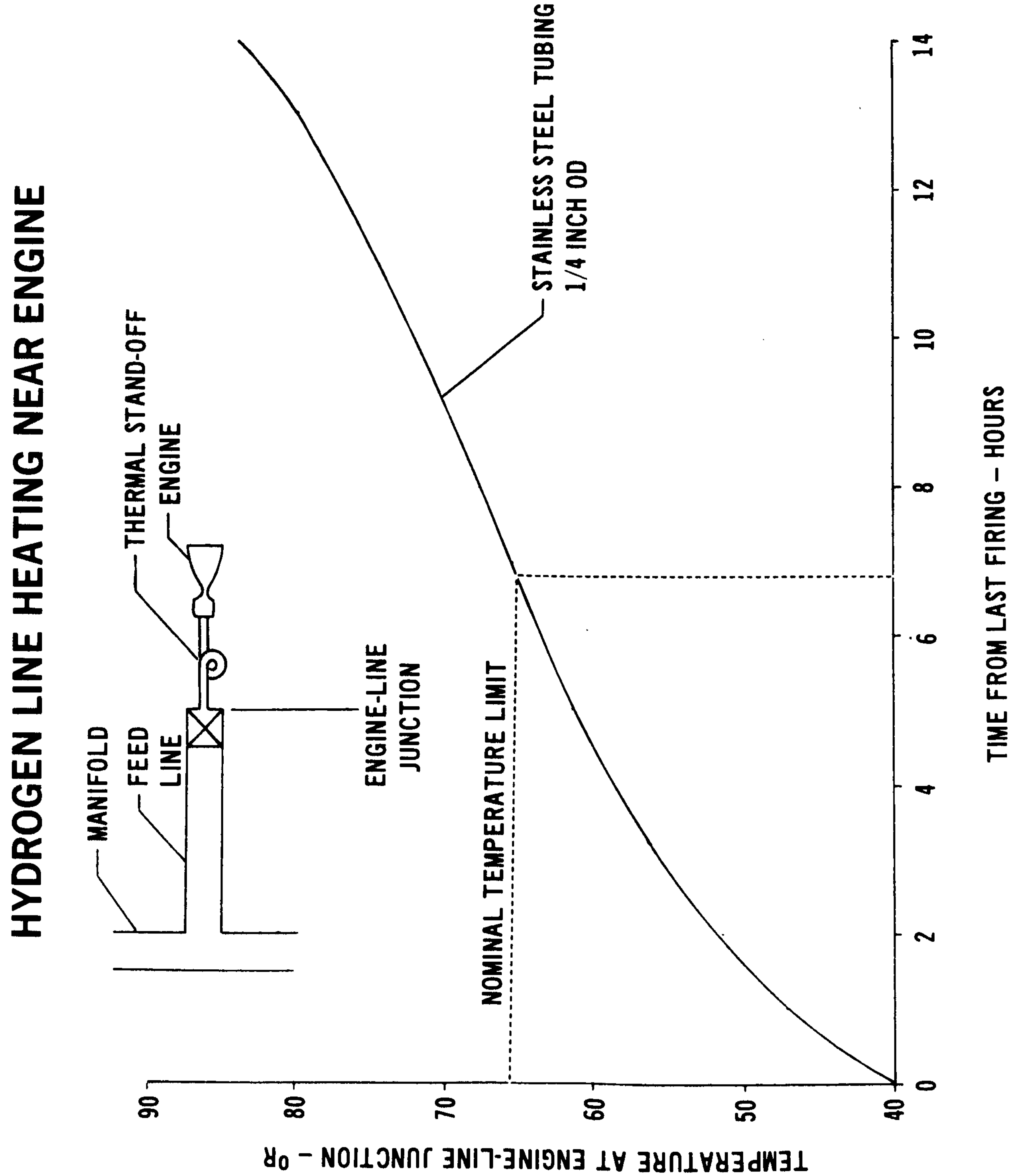

HYDROGEN LINE HEATING NEAR ENGINE
MANIFOLD
FEED LINE
ENGINE-LINE JUNCTION
THERMAL STAND-OFF
ENGINE
NOMINAL TEMPERATURE LIMIT
STAINLESS STEEL TUBING
1/4 INCH OD
TEMPERATURE AT ENGINE-LINE JUNCTION – °R
90
80
70
60
50
40
0
2
4
6
8
10
12
14
TIME FROM LAST FIRING – HOURS

Figure 3-16

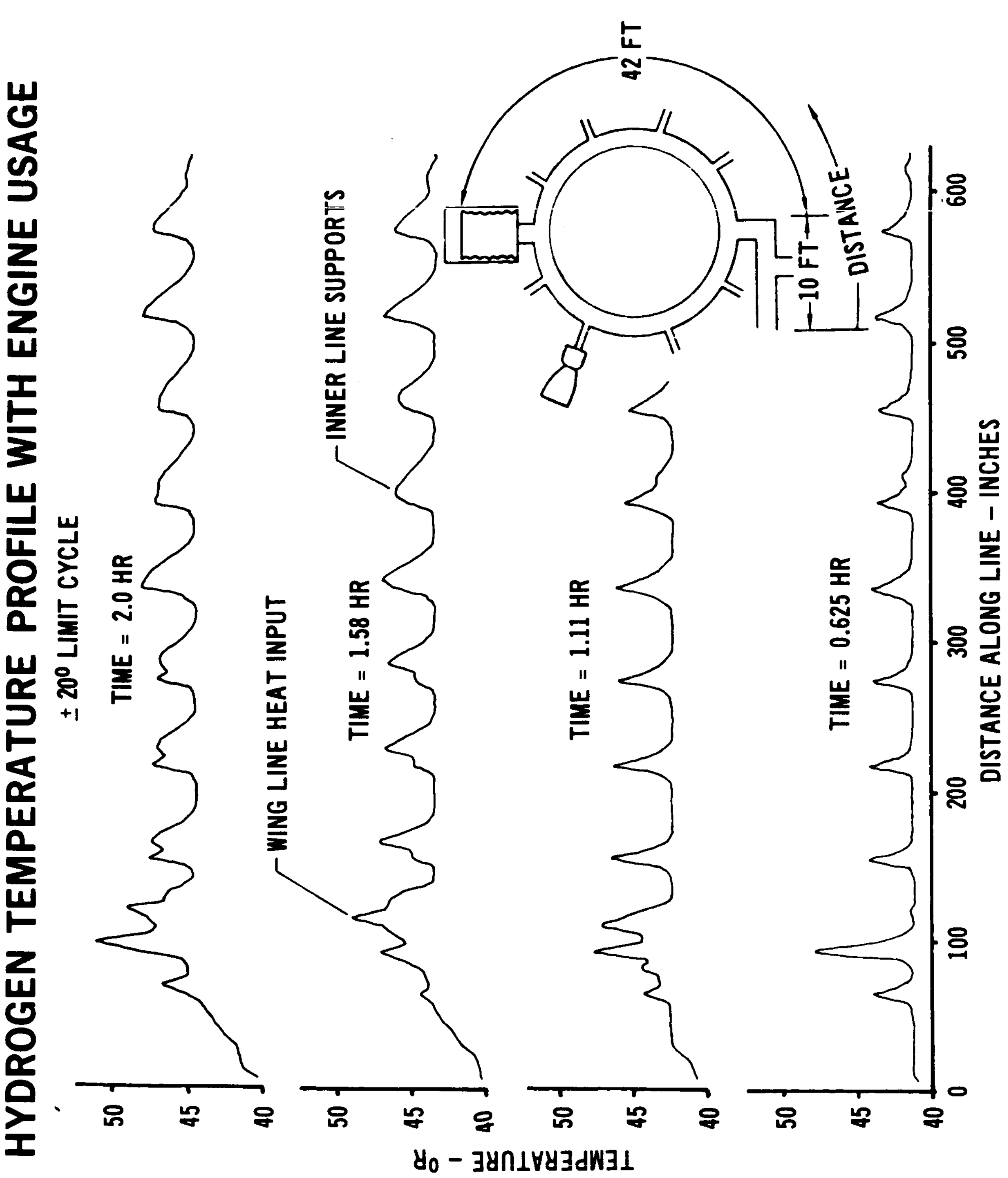

HYDROGEN TEMPERATURE PROFILE WITH ENGINE USAGE
± 20° LIMIT CYCLE
TIME = 2.0 HR
WING LINE HEAT INPUT
TIME = 1.58 HR
INNER LINE SUPPORTS
TIME = 1.11 HR
TIME = 0.625 HR
42 FT
10 FT
DISTANCE
TEMPERATURE – °R
50
45
40
0
100
200
300
400
500
600
DISTANCE ALONG LINE – INCHES

Figure 3-17

MCDONNELL DOUGLAS CORPORATION

MAXIMUM HYDROGEN TEMPERATURE VS TIME

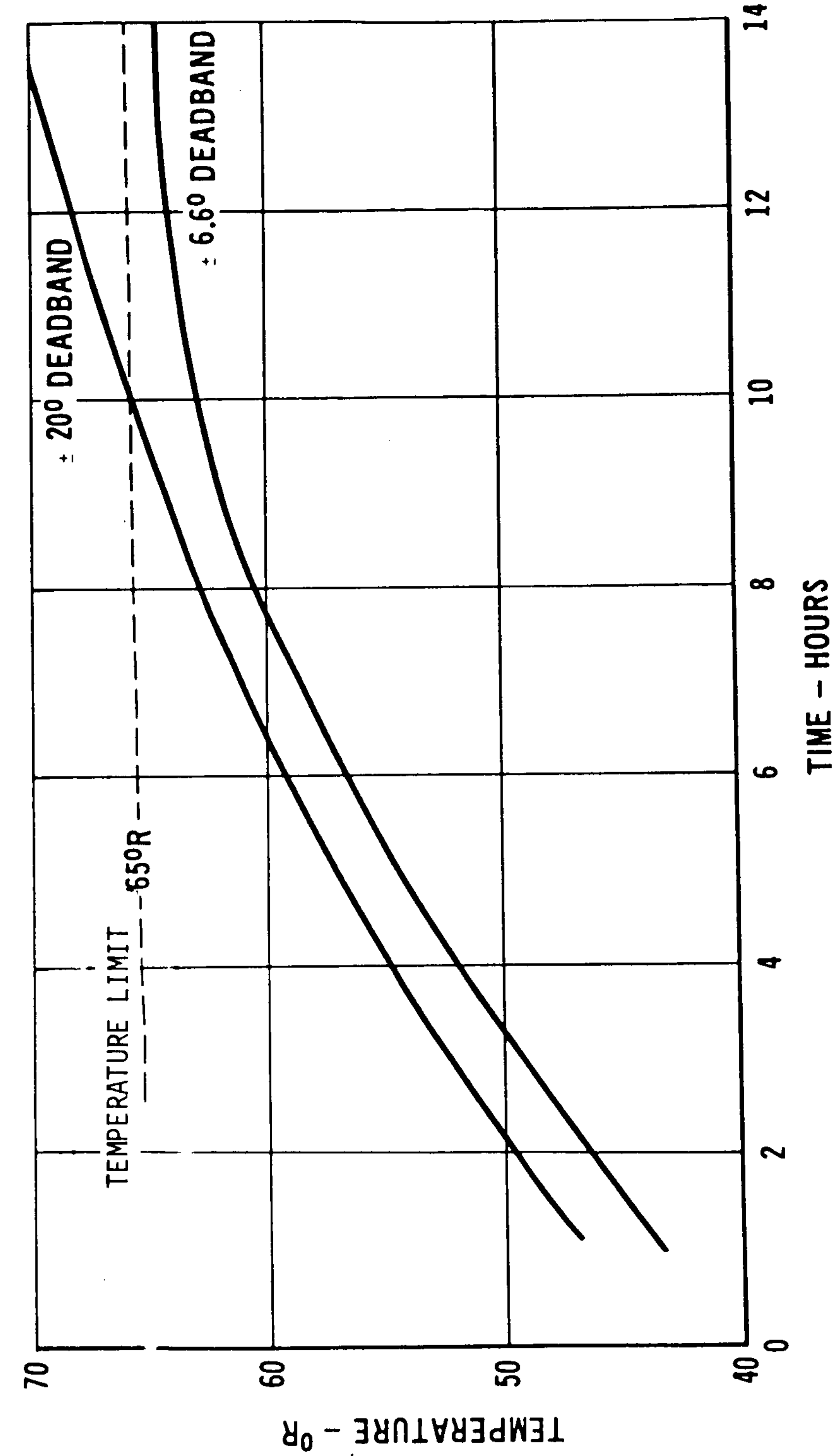


Figure 3-18

MAXIMUM OXYGEN TEMPERATURES VS TIME

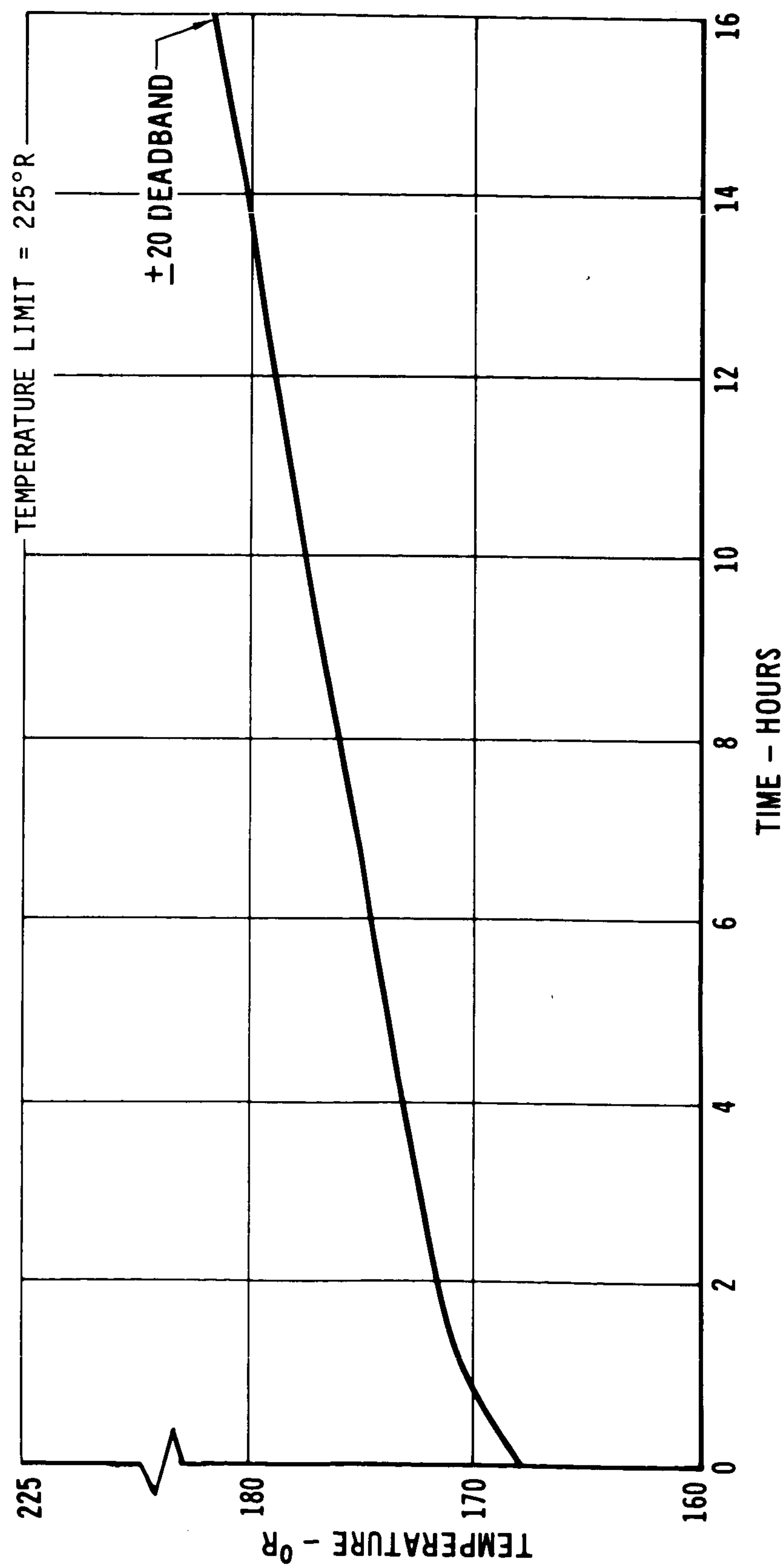


Figure 3-19

the baseline system, based on the gaseous system results presented in Paragraph 3.1. This system is shown schematically in Figure 3-20. A weight breakdown of the baseline system is shown in Figure 3-21.

In the liquid system, the weight penalty associated with deletion of the oxidizer pump in favor of a fully pressurized pump is 127 lb if the pressurant is stored in the hydrogen tank. By comparison, using pressurized oxygen in the gaseous system incurred a weight penalty of 740 lb. The difference is primarily due to deletion of the gaseous accumulators which amplify pressurization requirements.

Several alternate design approaches to the baseline were investigated to reduce system weight and/or simplify system interactions. These included alternate pressurization, feed lines, and pumping options. A summary of the weights and a comparison of feed line and pressurization options is given in Figure 3-22. The vacuum jacketed feed lines represent a large (23%) portion of the system inert weight. The use of non-jacketed lines could reduce system weight by 365 lb. However, without jacketed lines, the HPI would be exposed to potential handling and atmospheric damage and would at least require a soft purge bag for protection. Further investigation of the technology risks involved with non-jacketed lines is warranted before selection of this option. Fully pumped systems will save 355 lb.

The remaining option, a fully pressurized system, would result in an excessive weight penalty of 1500 lb. However, although this weight penalty is prohibitive for operational use, the concept could be used on an interim basis for the first few flights and updated later to a high performance configuration.

In addition to the options described above, alternate pump designs were considered. The choice of pump type and power source was not readily apparent from system considerations alone. Various design approaches ranging from high-speed, high flowrate designs to small pumps operating for relatively long durations were available. Also, alternate designs were available to reduce pump power requirements, thereby simplifying pump designs and/or reducing the APU interface complexity. This could be accomplished by either reducing the maximum accumulator pressure, or increasing accumulator capacity to lower pump flowrate requirements. As shown in Figure 3-23, by decreasing accumulator pressure from 600 to 400 psia, the pump horsepower required can be reduced from 127 hp to 70 hp, with an accumulator weight increase of 100 lb. Further reductions to 20 hp could be achieved by increasing accumulator capacity for an additional weight penalty of 300 lb. This latter change would have the additional benefit of reducing the number of pump/APU

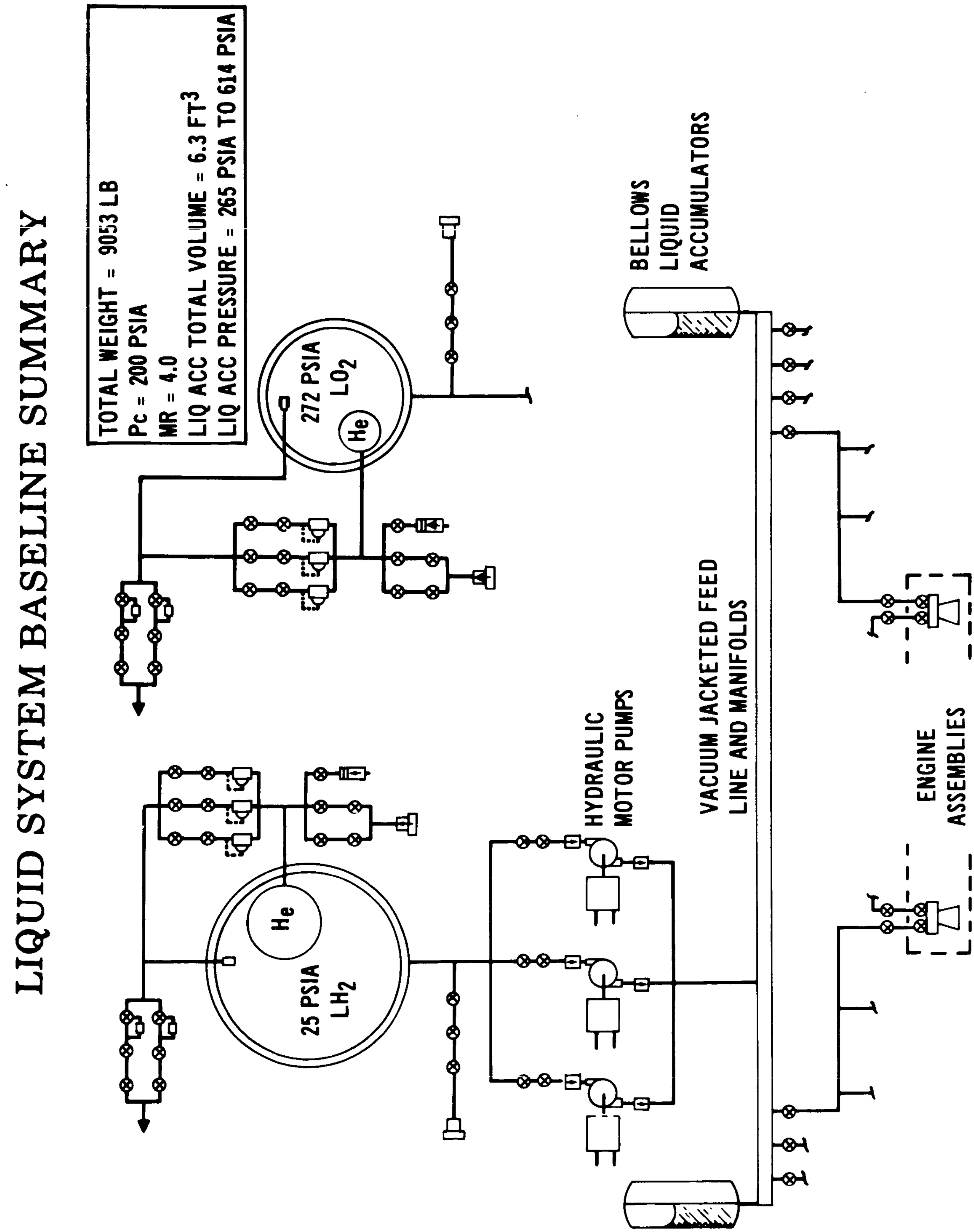

LIQUID SYSTEM BASELINE SUMMARY
TOTAL WEIGHT = 9053 LB
Pc = 200 PSIA
MR = 4.0
LIQ ACC TOTAL VOLUME = 6.3 FT^3
LIQ ACC PRESSURE = 265 PSIA TO 614 PSIA
272 PSIA
LO_2
He
25 PSIA
LH_2
He
HYDRAULIC
MOTOR PUMPS
VACUUM JACKETED FEED
LINE AND MANIFOLDS
BELLOWS
LIQUID
ACCUMULATORS
ENGINE
ASSEMBLIES

Figure 3-20

HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- ENGINE MR = 4.0
- CHAMBER PRESSURE = 200 PSIA
- STORAGE TANK PRESSURE, O_2 = 272 PSIA
  H_2 = 25 PSIA
- H_2 LIQ ACC, TEMP = 40°R
  PRESS = 265 TO 614 PSIA

COMPONENT	WEIGHT - LB	
	HYDROGEN	OXYGEN
PROPELLANT WEIGHT		
USABLE	1046	4185
RESIDUALS, LINES	22	351
TANKS	21	25
VENTED	194	13
TOTAL	1283	4574
PROPELLANT TANKAGE	381	315
PRESSURIZATION	97	262
MOTORS AND PUMPS	115	0
APU PROPELLANT	35	
FEED LINES AND INSULATION	230	230
COMPENSATORS	139	139
LIQUID ACCUMULATORS		
TANK	130	0
PRESSURIZATION	22	0
ISOLATION VALVES (28)	60	60
ENGINES (36)	981	
TOTAL SYSTEM WEIGHT	9,053	

Figure 3-21

COMPARISON OF ALTERNATE LIQUID SYSTEM CONCEPTS

SYSTEM	RELATIVE WT (LB)	APU INTERACTIONS	REMARKS
BASIC HYBRID	0	REQUIRES 40-50 APU CYLES, POWER SATISFIED BY ONE APU UNIT	HIGH PERFORMANCE SYSTEM, USING HYDRAULIC POWER. REQUIRES POSITIVE DISPLACEMENT PUMP DEVELOPMENT
HYBRID-NO VACUUM JACKET	-365	REQUIRES 40-50 APU CYCLES	VACUUM JACKET REMOVAL REDUCES WEIGHT BUT WOULD EXPOSE HPI TO HANDLING AND ATMOSPHERIC DAMAGE. FURTHER STUDY WARRANTED.
ALL HYDRAULIC PUMPED H_2 AND O_2	-355	REQUIRES 40-50 APU CYCLES BUT 2 APU UNITS REQUIRED TO FURNISH HORSEPOWER	COMPLEXITY ASSOCIATED WITH PUMPING OXYGEN IS NOT WARRANTED BY THE WEIGHT POTENTIAL INVOLVED.
HYBRID-O_2 PRESS IN H_2 TANK	-127	REQUIRES 40-50 APU CYCLES AND ONE APU UNIT	USE OF SINGLE H_e SYSTEM IN H_2 TANK WITH PASSIVE THERMAL CONDITIONER FOR O_2 APPEARS ATTRACTIVE.
FULLY PRESSURIZED (100 PSIA CHAMBER PRESSURE)	+1545	NONE	SIMPLE SYSTEM, BUT HEAVY FOR LARGE IMPULSE LEVELS. COULD BE ATTRACTIVE FOR INTERIM SYSTEM

Figure 3-22

TYPICAL HYBRID SYSTEM OPTIONS

P_C (PSIA)	$P_{MAX.}$ (PSIA)	PUMP HP	ACCUMULATOR CAPACITY (LB)	CYCLES	SYSTEM WEIGHT (LB)
200	600	127	17	40	9053
200	400	70	17	40	9153
200	400	20	60	8	9450
200	600	20	64	8	9320

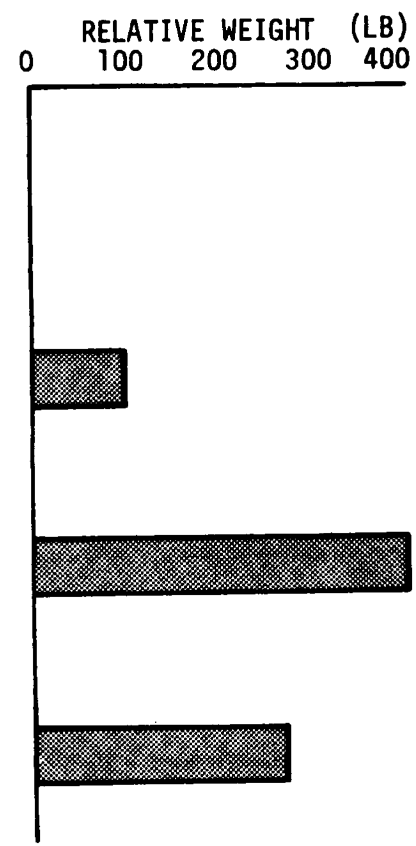


Figure 3-23

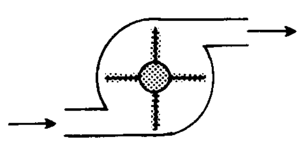
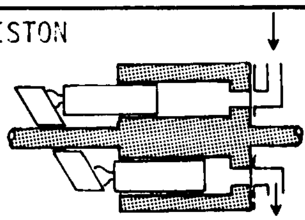
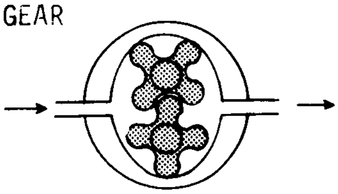
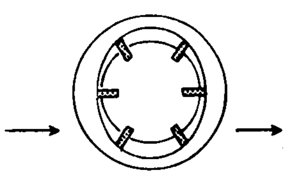
starts from 40 to 8 cycles per mission. Thus, considerable design flexibility is available since, for a small weight penalty, a large range of pump requirements can be accommodated. This could greatly relieve the pump design and development program by allowing relaxation of technology risk through an overall system design compromise. Since the weight difference between design points options was small, a separate subcontract was issued to Pesco Products to evaluate alternate pump designs. Both centrifugal and positive displacement pump designs were considered for outlet pressures from 400 to 600 psia and pump flowrates for thrust levels from 250 lb to 5750 lb. Also, both electric and hydraulic power sources were investigated.

Specific pump design data is presented in Appendix E and a comparison of pump types is shown in Figure 3-24. Centrifugal pumps were shown to be heavy and inefficient at the lower speeds associated with either hydraulic (10,000 rpm) or electric motors (22,000 rpm). To be acceptable, pump speeds near 100,000 rpm would be required. For this reason, centrifugal pumps for this application necessitate a hot gas turbine drive of the same general type used for the gaseous system. However, a liquid system turbopump would present considerably less technology risk. With liquids, the accumulator capacity can be increased for a small weight penalty, thereby allowing slower pump acceleration and increased bearing life. As a contrast, in a gaseous system, the 5 sec start time associated with state-of-the-art bearing acceleration limits in long life machines (25,000 to 30,000 rpm/sec) would result in a 1000 lb weight penalty. In addition, the liquid system can accommodate a wider range of pump performance, and it would be much easier to integrate the pump into the system than in the gaseous systems where pump performance can affect operation of the propellant heat exchangers. In fact, no significant technology problems are anticipated with liquid APS hydrogen turbopumps.

In addition to the centrifugal pumps described above, piston, gear,and vane type, positive displacement pumps were also evaluated. These pumps require slower operating speeds, on the order of 10,000 rpm, and are more adaptable to hydraulic and electric drives. The piston pumps were found to be heavy, due to the large flow capacities associated with low hydrogen density. This required large pistons and heavy rotors which limited pump operating speed. Both gear and vane pumps appear to be better suited for hydrogen pumping than piston pumps. These pumps are lighter and simpler than the piston type. The preferred pump type is a vane pump, similar to a hydrogen vane pump designed and demonstrated by Pesco Products. This work was part of a General Electric Company contract entitled, "Final Pumping

CANDIDATE LH_2 PUMP CONCEPTS

(PESCO PRODUCTS)

TYPE	WEIGHT LB	EFF %	COMMENTS
CENTRIFUGAL 	307	33	HEAVY AND INEFFICIENT AT LOW SPEEDS (20,000 TO 40,000 rpm) USE OF CENTRIFUGAL PUMPS WILL BE LIMITED TO HOT GAS TURBINE DRIVES
PISTON 	460	66	HEAVY DUE TO LARGE PISTONS AND ROTOR REQUIRED TO ACCOMMODATE HYDROGEN FLOW RATE. REQUIRES CASE DRAIN TO REMOVE PROPELLANT LEAKAGE PAST PISTON SEALS
GEAR 	194	60	LOW COST, SIMPLE DESIGN. HEAVIER THAN VANE PUMPS
VANE 	117	60	LIGHTWEIGHT, EFFICIENT DESIGN FEASIBILITY OF DESIGN FOR LH_2 HAS BEEN DEMONSTRATED

BASED ON 600 PSI HEAD RISE, 2.65 LB/SEC H_2 FLOW, HYDRAULIC DRIVE

Figure 3-24

System Liquid Hydrogen/Liquid Methane, J85 Control System", performed for NASA Lewis Research Center. Considerable material development was accomplished, and no critical, new technology effort is anticipated.

The effect of the power source on system weight is summarized in Figure 3-25. Except for the centrifugal turbopump concept, which is shown to be the lightest, the weights are for vane type positive displacement pumps. A hydraulic powered pump is weight competitive and could be provided for 150 to 300 lb, depending on pump outlet pressure. Electric motor concepts, limited to the APU electrical output of 15 KW (20 hp), would weigh 300 to 500 lb more than the turbopump system. Finally, a small DC motor operating from fuel cell power would result in a penalty of approximately 800 lb.

Based on these data, the turbopump system was the preferred concept. A schematic of the turbopump system is shown in Figure 3-26. This is the lightest and simplest system and, most importantly, it would operate independently of the APU. Thus, the pumps could be operated at any time, the APU could remain inactive throughout the orbital phase of the mission, and resizing of APU electric or hydraulic systems would not be required. The second choice would be the hydraulic motor operated, vane pump, which is attractive from a weight standpoint, but would interface significantly with the APU.

E243-131

EFFECT OF PUMP POWER SOURCE ON SYSTEM WEIGHT

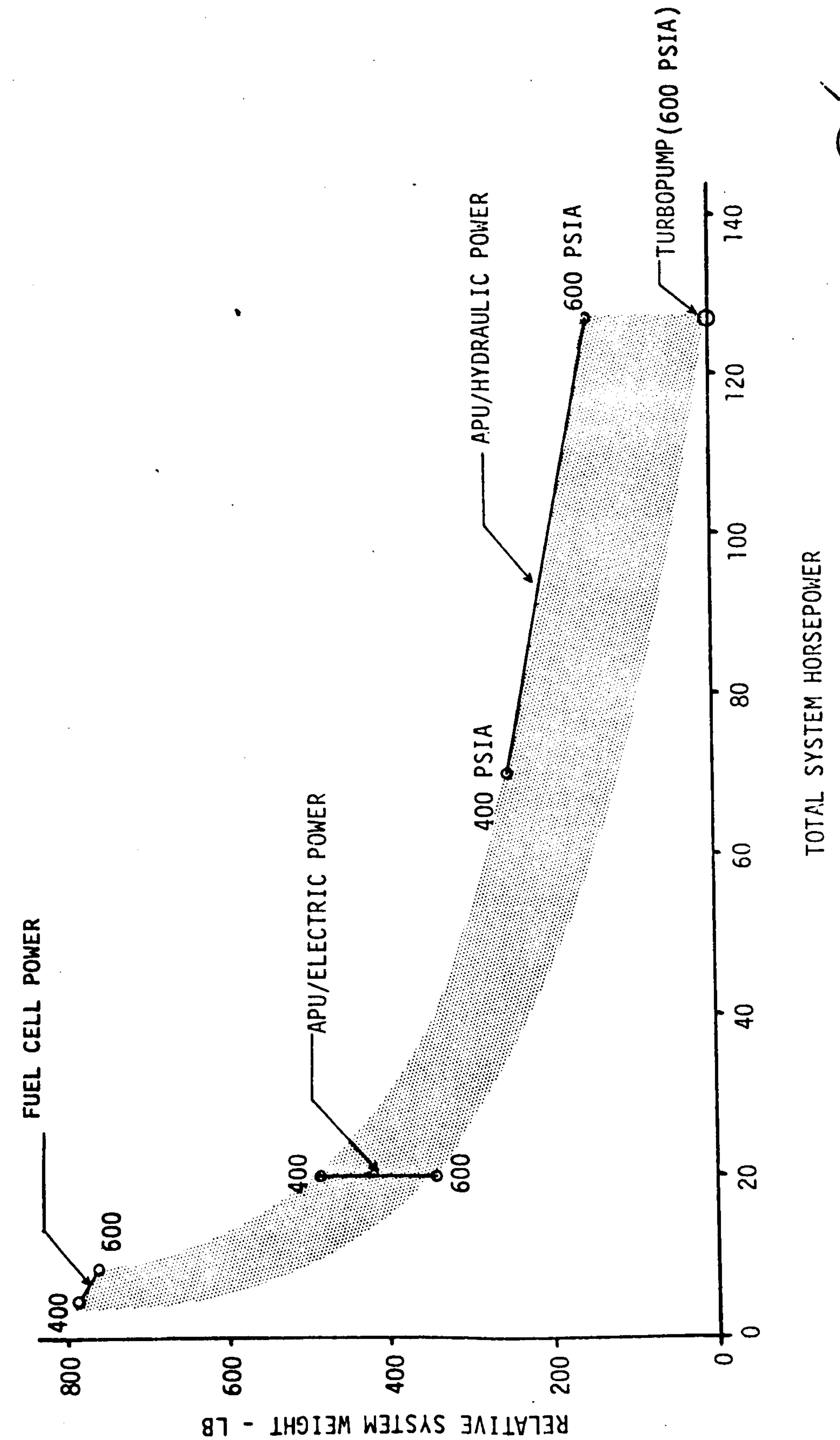


Figure 3-25

LIQUID HYDROGEN/LIQUID OXYGEN APS SCHEMATIC

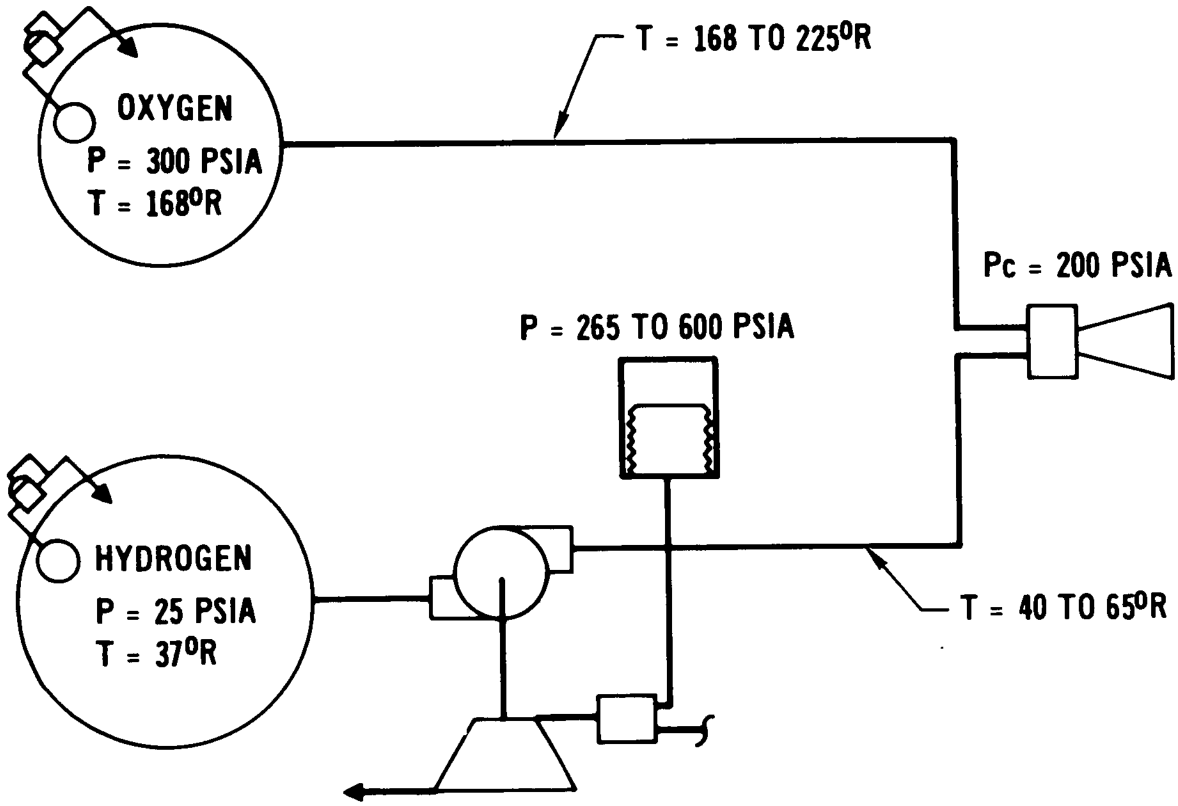


Figure 3-26

4. CONCLUSIONS

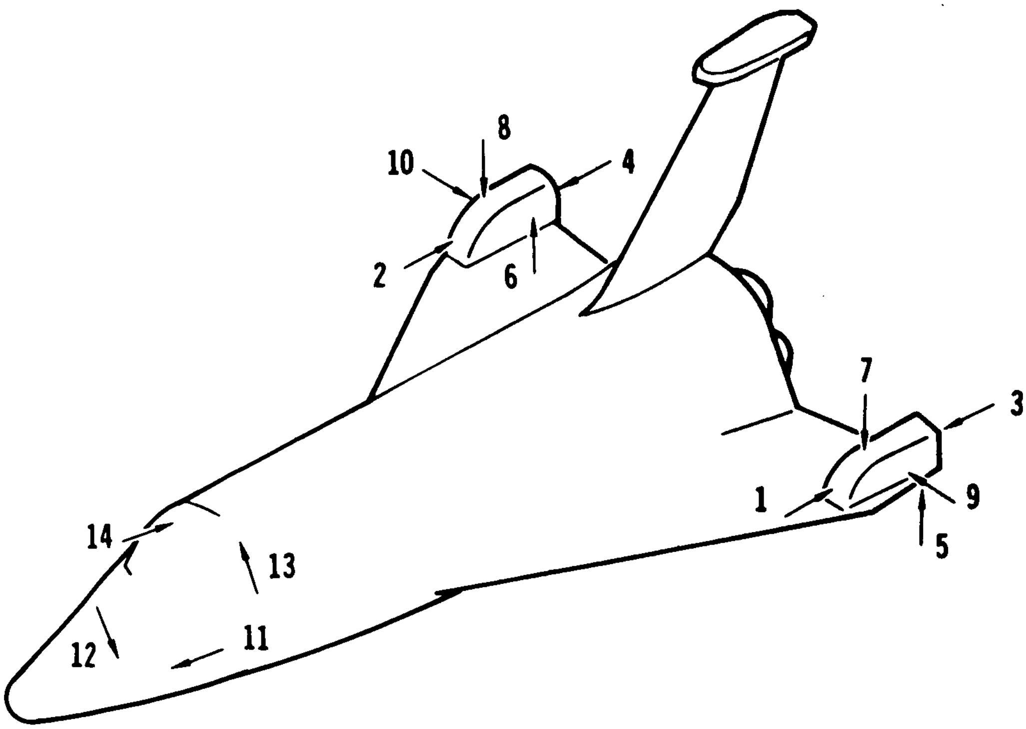
Gaseous RCS concepts using motor operated pumps which are weight competitive with a gaseous O_2/H_2 turbopump system can be developed. However, the most attractive concept investigated was a liquid O_2/H_2 system, using fully pressurized oxygen and a hydrogen turbopump. This concept provides a system weight advantage of 1500 lb over the parallel gas generator turbopump system (with perfect controls) described in Reference (h).

The effort described in the preceeding sections was based on a fully reusable orbiter with internal tankage. Study Phases C and E considered storable monopropellants and bipropellants for external tank orbiter vehicles. A comparison of the liquid O_2/H_2 concept with both monopropellant and bipropellant systems was made to determine their relative weights. Two external tank orbiter vehicles, corresponding to a Mark I - Mark II development approach as shown in Figure 4-1, were specified for storable system study. The impulse requirements for these vehicles range from 1.3 (10^6) lb-sec for the Mark I vehicle to 1.7 (10^6) lb-sec for the Mark II vehicle.

The results of this study suggest two approaches to the evolution of a high performance system for a Mark II vehicle. Figure 4-2 shows these approaches. One approach starts with a gaseous hydrogen-liquid oxygen system. The system uses liquid oxygen because the weight penalties associated with avoidance of oxygen pumps are small and distribution of liquid oxygen was found to be feasible. Gaseous hydrogen is used because the engine ignition requirements are state-of-the-art, similar to the Pratt and Whitney RL-10 engine. This gaseous system could be updated for a Mark II concept by either increasing the gas generator operating temperature (Reference (h)), thereby increasing system efficiency, or by utilizing a liquid turbopump system which eliminates the gas generator and gaseous accumulator. The decision as to which means of improvement was most attractive could be made on the basis of the relative status of technology demonstration programs in the areas of liquid ignition and high temperature heat exchangers.

The second approach shown in Figure 4-1 starts with a simple, fully pressurized liquid-liquid system for the Mark I vehicle. Later a turbopump and liquid accumulator could be added to the hydrogen side providing a high performance Mark II design.

BASELINE ORBITER FOR STORABLE APS STUDIES



	MSC 040A	MARK II
WEIGHT (INSERTION) (LBM)	207,200	228,700
PAYLOAD (LBM)	45,000	65,000
LENGTH (FT)	120.7	120.7
NUMBER OF THRUSTERS	34	42
THRUSTER THRUST (LB)	600	600
TOTAL IMPULSE (LB-SEC)	1.343×10^6	1.693×10^6

Figure 4-1

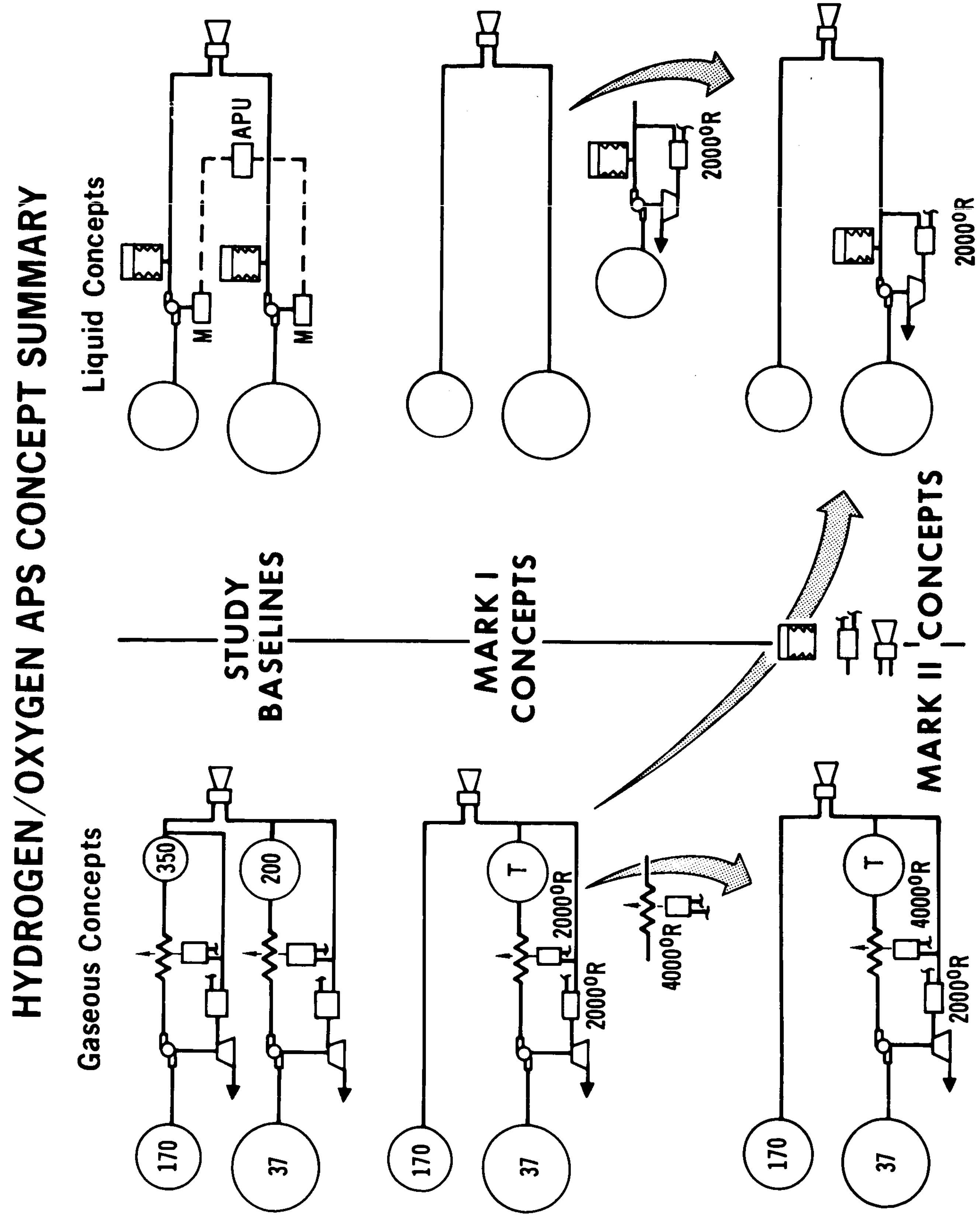

HYDROGEN/OXYGEN APS CONCEPT SUMMARY
Gaseous Concepts
Liquid Concepts
STUDY BASELINES
MARK I CONCEPTS
MARK II CONCEPTS
350
200
170
37
T
APU
M
2000°R
4000°R

Figure 4-2

A comparison of weights for systems integrally mounted in the vehicle is shown in Figure 4-3. Monopropellant hydrazine and storable bipropellant system weights were taken from the preliminary study Phase C effort (Reference (j)). The Mark I and Mark II total impulse values are noted. For the Mark I vehicle, use of fully pressurized liquid oxygen/liquid hydrogen would be slightly heavier than a hydrazine RCS system, but this system could be updated to a fully pumped liquid hydrogen/pressurized liquid oxygen system for the Mark II vehicle, which would be approximately 3,000 lb lighter than the hydrazine system. The comparison, shown in Figure 4-3, is for integrally mounted systems. However, the toxic hydrazine and storable bipropellant systems are designed for installation in removable pods to allow rapid removal of the propellants and transportation to a remote refurbishment site. Since the oxygen/hydrogen propellants are not toxic, the modular approach is unnecessary. A more valid comparison then would be between integrally installed cryogenic concepts and modular storable concepts. Weight comparisons on this basis are shown in Figure 4-4. It should be noted that the total impulse values for both Mark I and Mark II are higher for modular systems than for integral systems due to control cross coupling effects that result with modular system engine installations. Further penalties are assessed against the modular systems relative to the integral systems because of pod structure weights and the effect of increased pod structural weight on landed weight. Vehicle studies show that a weight penalty equal to 40 percent of the total inert system weight is required to account for resizing the aerodynamic surfaces, landing gear, etc. With these effects included,the fully pressurized liquid oxygen/liquid hydrogen system for the Mark I vehicle is shown in Figure 4-5 to weigh less than a hydrazine system and only 800 lbs more than a storable bipropellant system. Subsequent development of a pumped liquid hydrogen system for the Mark II vehicle would increase the weight savings to nearly 4500 lb. Thus, liquid oxygen/liquid hydrogen systems offer a very large potential weight savings.

COMPARISON OF INTEGRAL RCS CONCEPTS

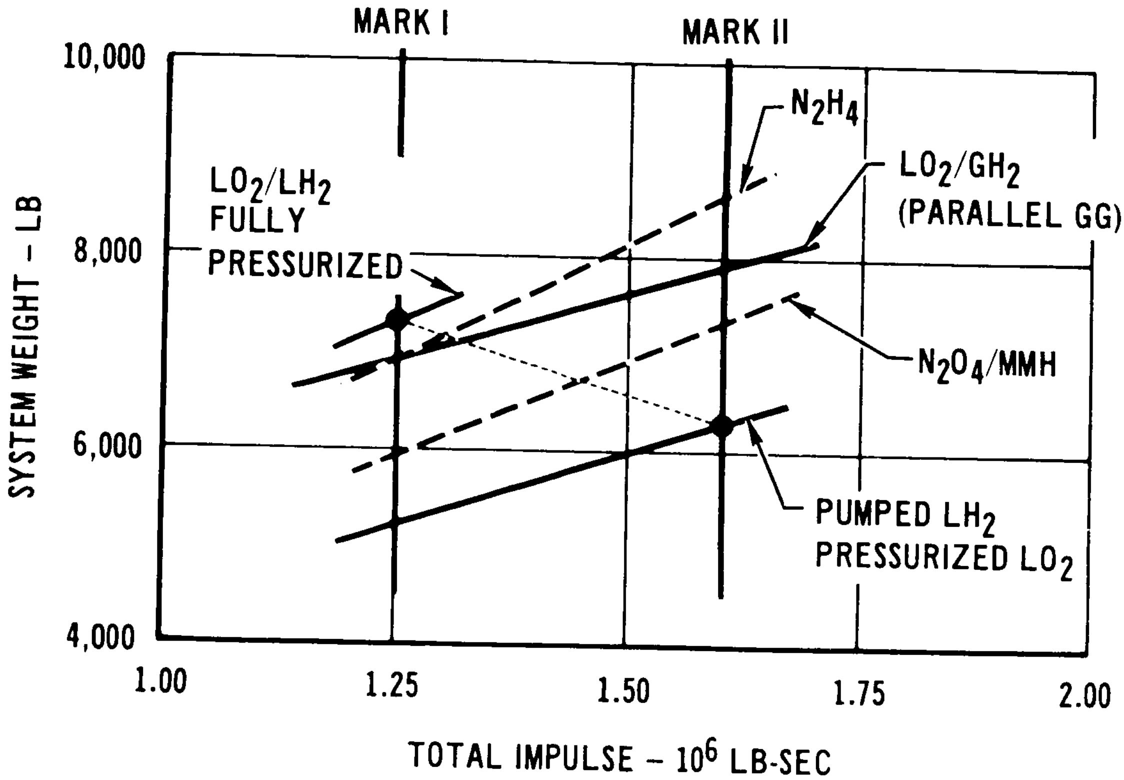


Figure 4-3

COMPARISON OF MODULAR STORABLE RCS CONCEPTS WITH INTEGRAL CRYOGENIC CONCEPTS

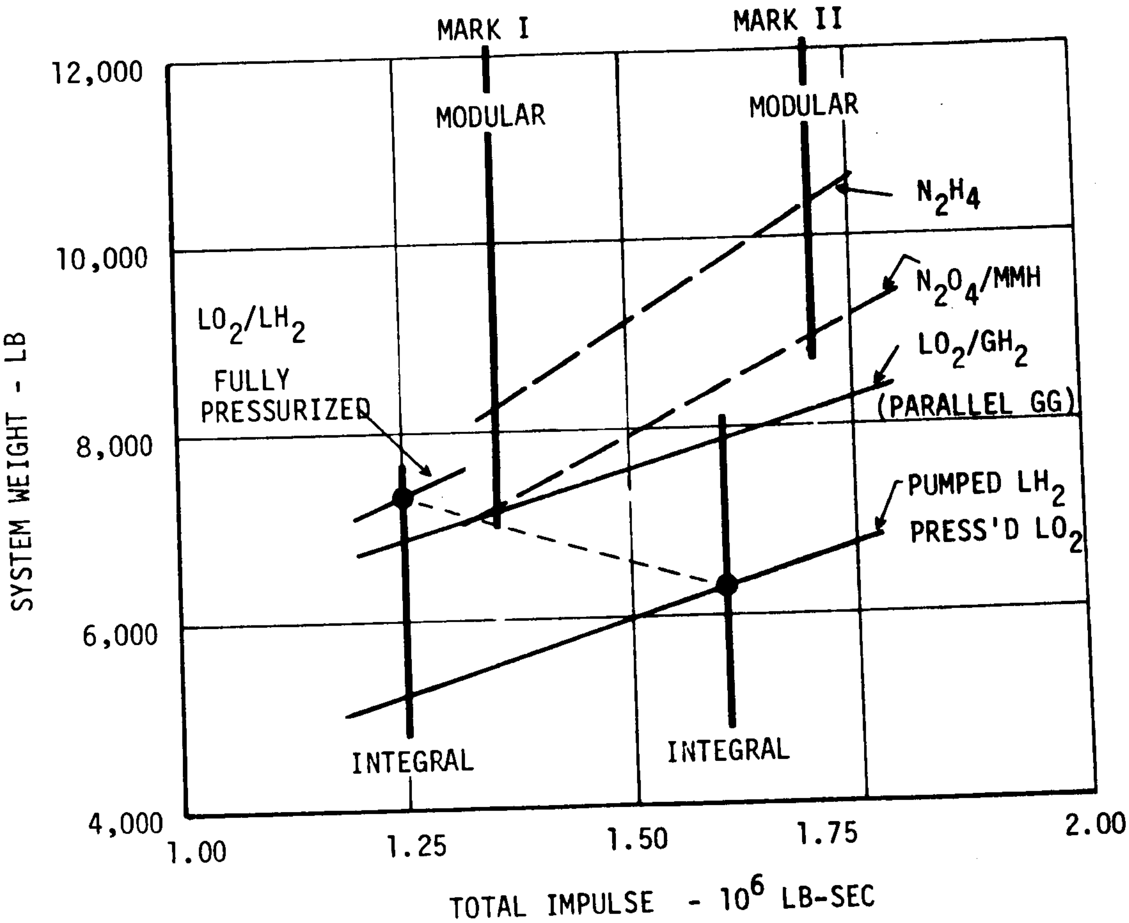


Figure 4-4

COMPARISON OF MODULAR STORABLE RCS CONCEPTS WITH INTEGRAL CRYOGENIC CONCEPTS

(INCLUDES 40% LANDED WEIGHT PENALTY)

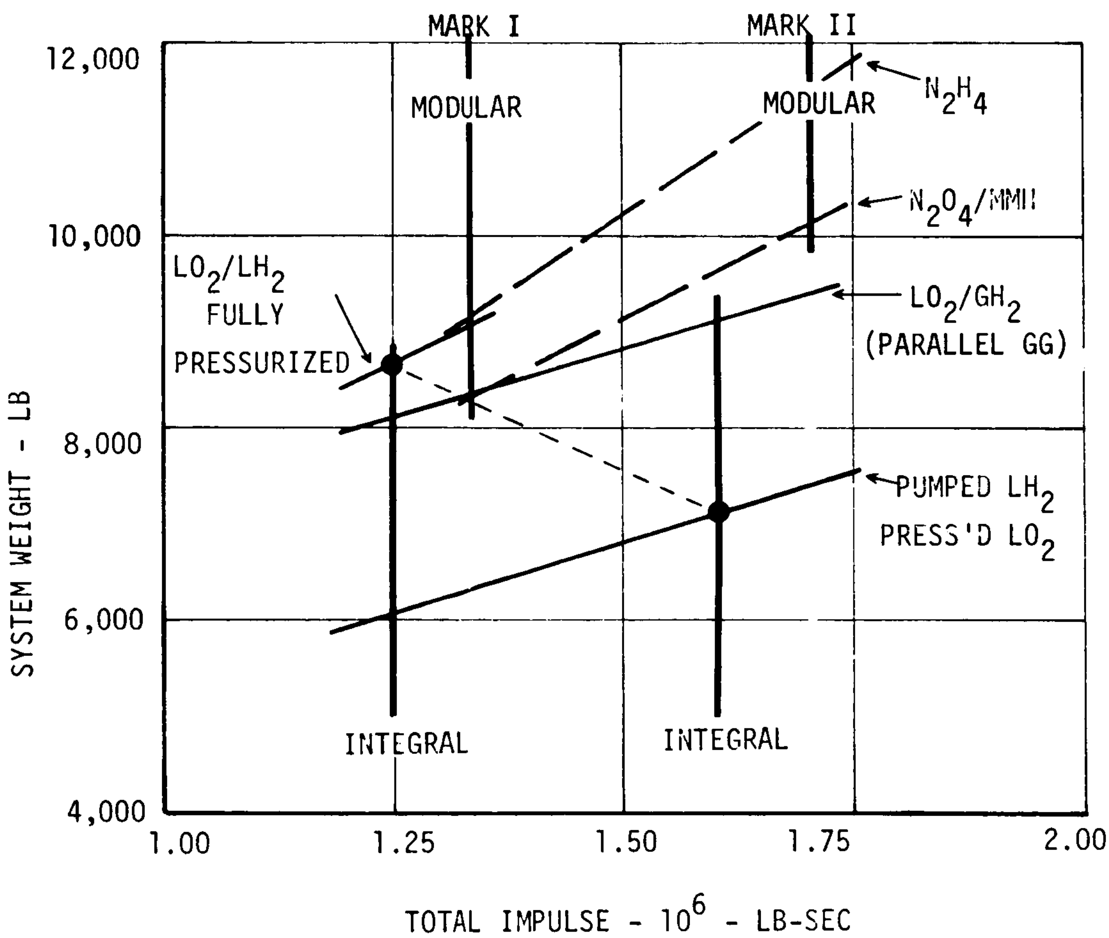


Figure 4-5

5. REFERENCES

(a) Kelly, P. J., and Regnier, W.W.,"Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition Study - Summary Report," McDonnell Douglas Report MDC E0299, 12 February 1971.

(b) Gaines, R. D., Goldford, A. I., and Kaemming, T. A., "Space Shuttle High Pressure Auxiliary Subsystem Definition Study - Subtask B Report," McDonnell Douglas Report No. MDC E0298, 12 February 1971.

(c) Green, W. M., and Patten, T.C., "Space Shuttle Low Pressure Auxiliary Propulsion Subsystem Definition - Subtask B Report," McDonnell Douglas Report No. MDC E0302, 29 January 1971.

(d) Benson, R. A., Shaffer, A., Burge, H. L., "Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition, Final Report," TRW Report 17611, 31 March 1971.

(e) Kelly, P.J., "Space Shuttle Auxiliary Propulsion System Design Study - Executive Summary," McDonnell Douglas Report No. MDC E0674, 29 December 1972.

(f) Kelly, P.J., "Space Shuttle Auxiliary Propulsion System Design Study - Program Plan," McDonnell Douglas Report No. MDC E0436, 15 July 1971, Revised 6 December 1971.

(g) Orton, G. F., and Schweickert, T. F., "Space Shuttle Auxiliary Propulsion System Design Study - Phase A, Requirements Definition," McDonnell Douglas Report No. MDC E0603, 15 February 1972.

(h) Orton, G. F., and Schweickert, T. F., "Space Shuttle Auxiliary Propulsion System Design Study - Phase B, Candidate RCS Concept Comparisons," McDonnell Douglas Report No. MDC E0567, 15 February 1972.

(i) Bruns, A. E., and Regnier, W. W., "Space Shuttle Auxiliary Propulsion System Design Study - Phase C, Oxygen-Hydrogen RCS/OMS Integration," McDonnell Douglas Report No. MDC E0346, 15 December 1971.

(j) Anglim, D. D., Bruns, A. E., Perryman, D. C., and Wieland, D. L., "Space Shuttle Auxiliary Propulsion System Design Study - Phase C, Earth Storable RCS/OMS/APU Integration and Phase E, System Performance Analysis," McDonnell Douglas Report No. MDC E0708, 29 December 1972.

(k) "Space Shuttle Phase B System Study Final Report, Part II, Technical Summary," McDonnell Douglas Report MDC E0308, 30 June 1971.

(l) Pearson, W. E., et al. "Shuttle APS Propellant Thermal Conditioner Study, Final Report," Bell Aerospace Company Report No. 8665-950002, December 1971.

(m) Robson, J. H., Cannon, W. A., and English, W. D., "Pressurization Systems Design Guide Volume III, Pressurant Gas Solubility in Liquid Propellants," McDonnell Douglas Report DAC-60510-F1, July 1968.

APPENDIX A
COMPONENT MODELS

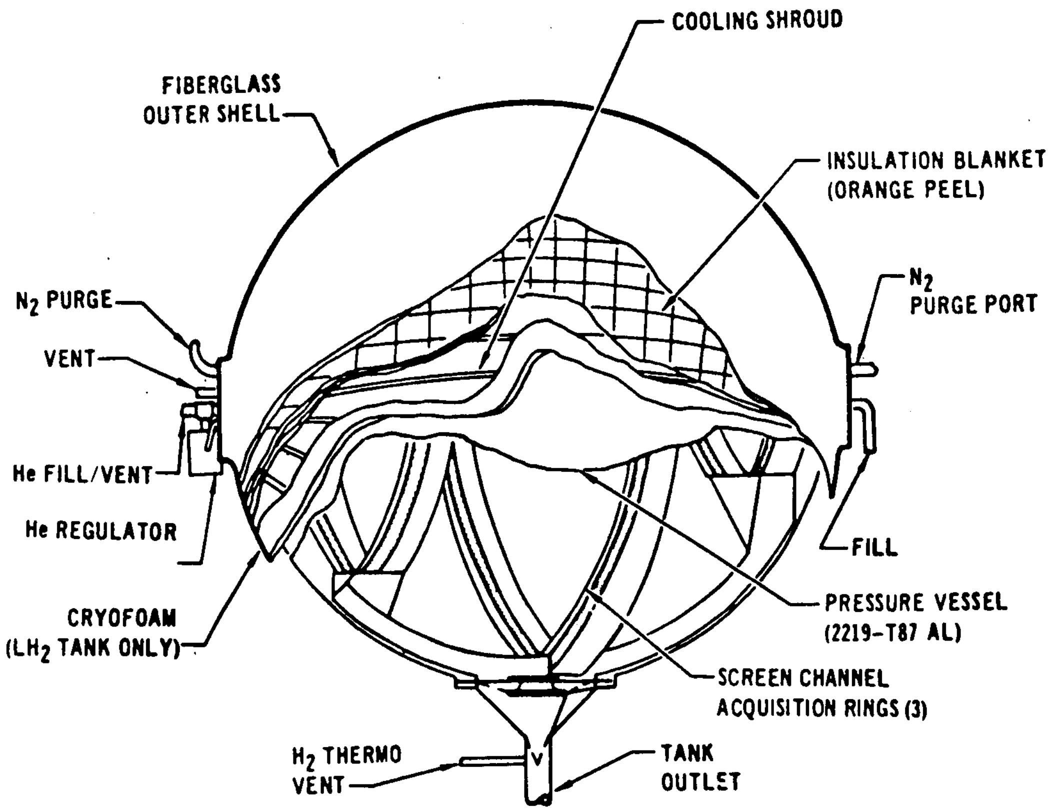
A1. Introduction - Component and subassembly weight and performance models were required to perform valid trade studies and to allow accurate system weight and performance comparisons. Extensive use was made of component models developed under the MDC Space Shuttle Auxiliary Propulsion Subsystem Definition Study (Contract NAS 8-26248), and those developed for Phase B effort in this study for gaseous oxygen hydrogen, turbopump RCS. Detailed discussions of the design and performance assumptions associated with these RCS component models are found in References (a), (b), and (h). The following paragraphs summarize these gaseous system component models used in this study and also describe the new component models developed for liquid accumulators, motor operated pumps, liquid distribution lines, and liquid oxygen-hydrogen thrusters.

A2. System Components -

A2.1 Main Propellant Tankage - The cryogenic propellants are stored in spherical, highly insulated propellant tanks. The propellant tankage assembly, shown in Figure A-1, consists of an aluminum pressure vessel, a screen channel propellant acquisition device, a vapor cooled shroud, multilayer insulation (MLI), a fiberglass protective cover, and required support structure.

The insulation is double aluminized mylar with dacron net separators. Insulation thickness was selected to provide minimum weight, considering insulation weight and propellant boiloff-vent losses. A vapor cooled shroud, consisting of a tubular heat exchanger mounted to a thin spherical aluminum shell, was used to reduce heat input. With this concept, subcooled liquid propellant is vaporized in the shroud tubing to remove all of the heat input. Propellant circulation fans are not required, and the main propellant bulk is not heated. The hydrogen tank also incorporates foam insulation of sufficient thickness to allow the use of a nitrogen purge during pre-launch operations. The fiberglass shell, used to protect the insulation from atmospheric or handling damage, is vented during ascent to evacuate the MLI and is repressurized during entry with residual helium, to avoid crushing loads.

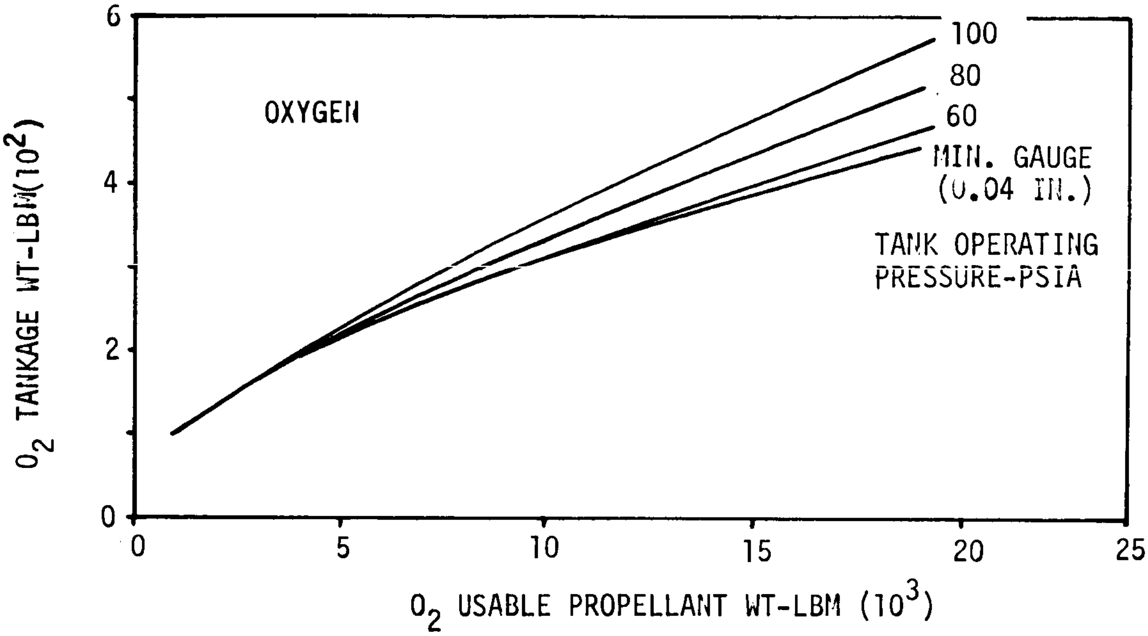
Parametric storage assembly weights for both the oxygen and hydrogen tankage, including boiloff and venting for ascent, 7 days on orbit, and descent losses, tankage non-optimum design factors, screen channels, and minimum gage effects, are shown in Figure A-2.



TANKAGE MODEL

Figure A-1

PROPELLANT TANKAGE WEIGHT



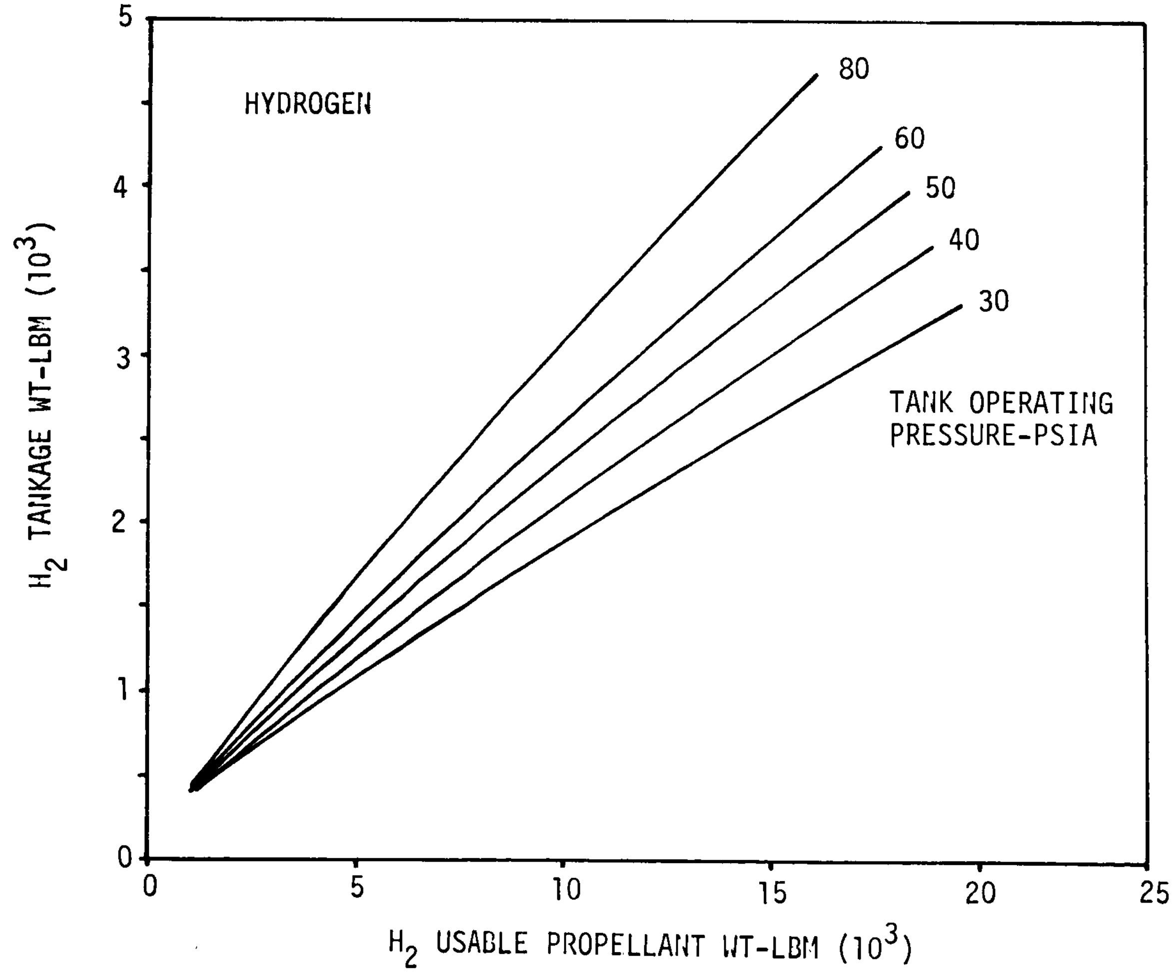


Figure A-2

A2.2 Liquid Propellant Accumulator Tankage - Accumulator tankage models were developed for both screen and bellows type propellant acquisition. The screen tank model was, in general, the same as that used for main propellant storage as described in Paragraph A2.1, above, with weights as shown in Figure A-3. Bellows accumulator tank designs were based on high cycle life, low-weight considerations. Bellows life characteristics, shown in Figure A-4, can be related either to pitch/span ratio, or for a given span, to extended/nested length ratio. With a typical span of 0.75 in., a pitch/span ratio of 0.5 would be achieved with length ratios of 10 to 20 for welded bellows. Formed bellows require higher nested pitch values to prevent excessive convolute stresses. This would decrease the allowable length ratio to 5 to 10 for formed bellows.

Formed bellows were selected for this study on the basis of fewer leakage paths than with welded bellows. No significant tank weight differences between welded and formed bellows tankage were noted. However, current bellows tank designs are heavy compared to non-bellows tanks. A large portion of the bellows tank weight is generally associated with a stainless steel shell. Figure A-5 shows that the bellows tank weights can be reduced nearly 50% by using a titanium outer shell. Propellant compatibility problems are not of concern since the propellants are stored within the bellows, which is fabricated of stainless steel. Tankage weight estimates, shown in Figure A-6, show that tank diameters of 20 to 25 in. will result in minimum tank weights for the range of volumes required. The tank length to diameter ratio varies from 1 to 2 for these conditions. Weights shown in Figure A-6 are low since they do not include weights for the movable bellows head. Complete bellows tank weight estimates, based on the Bell Aerosystems Minuteman III bellows tank with a titanium shell, are shown in Figure A-7.

A2.3 Pressurization - The main propellant storage tanks are pressurized from high pressure helium gas supplies. Regulated tank pressures of 25 psia (hydrogen) and 30 psia (oxygen) were used for the storage tanks when pumps were provided in the system design. In fully pressurized systems, the storage tanks were pressurized to full system pressure by regulation of the helium supply pressure. For the main storage tanks pressurization weights were based on isothermal expansion of the pressurant neglecting solubility effects.

For high pressure liquid accumulators, both regulated and blowdown pressurization was evaluated. Weights for regulated pressurization are shown in Figure A-8 for bellows tanks, where the propellant and pressurant are isolated. The weights shown do not apply to accumulators which use screen acquisition devices, in which

TANKAGE WEIGHT - SCREEN TYPE LIQUID ACCUMULATOR

WEIGHTS INCLUDE SUPPORTS,
FIBERGLASS SHELL, INSULATION
AND SCREEN ACQUISITION DEVICE

HYDROGEN

OXYGEN

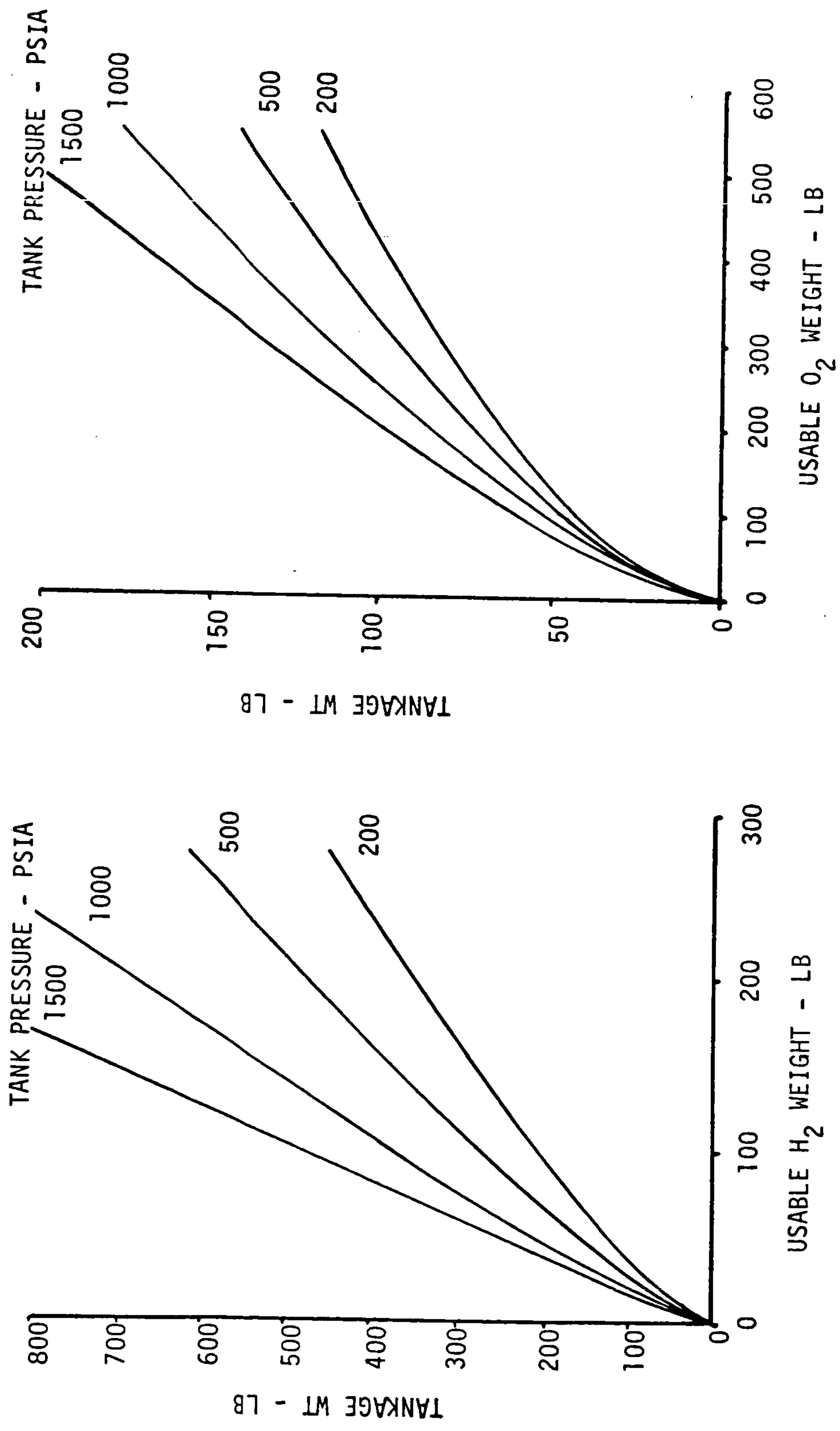
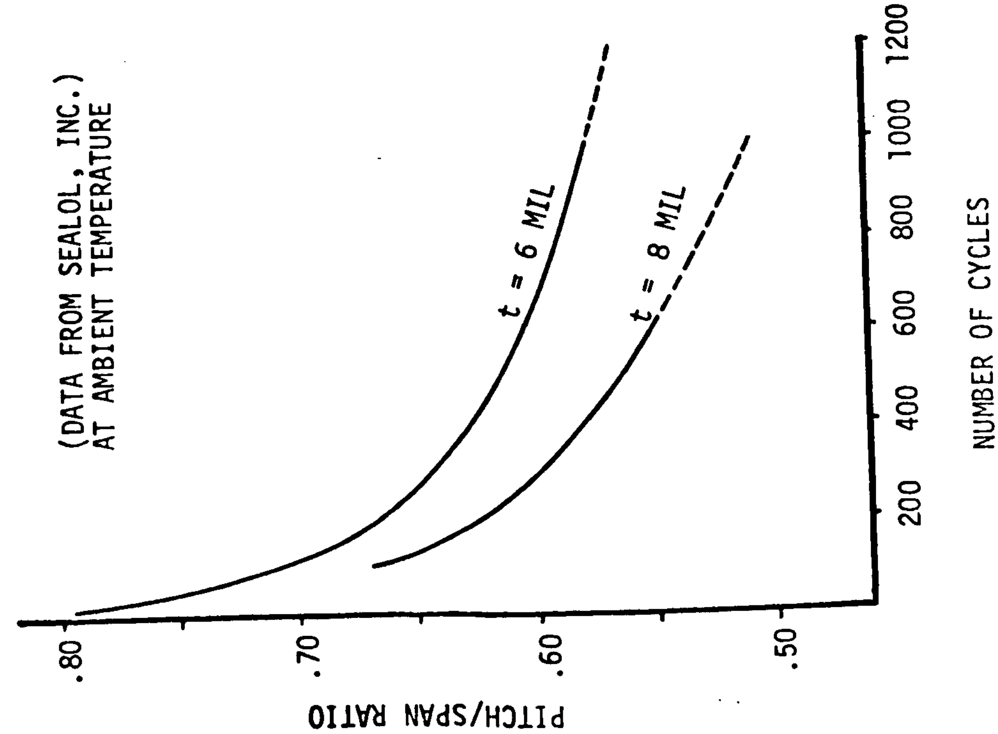


Figure A-3

EFFECT OF LIFE ON BELLOWS CHARACTERISTICS



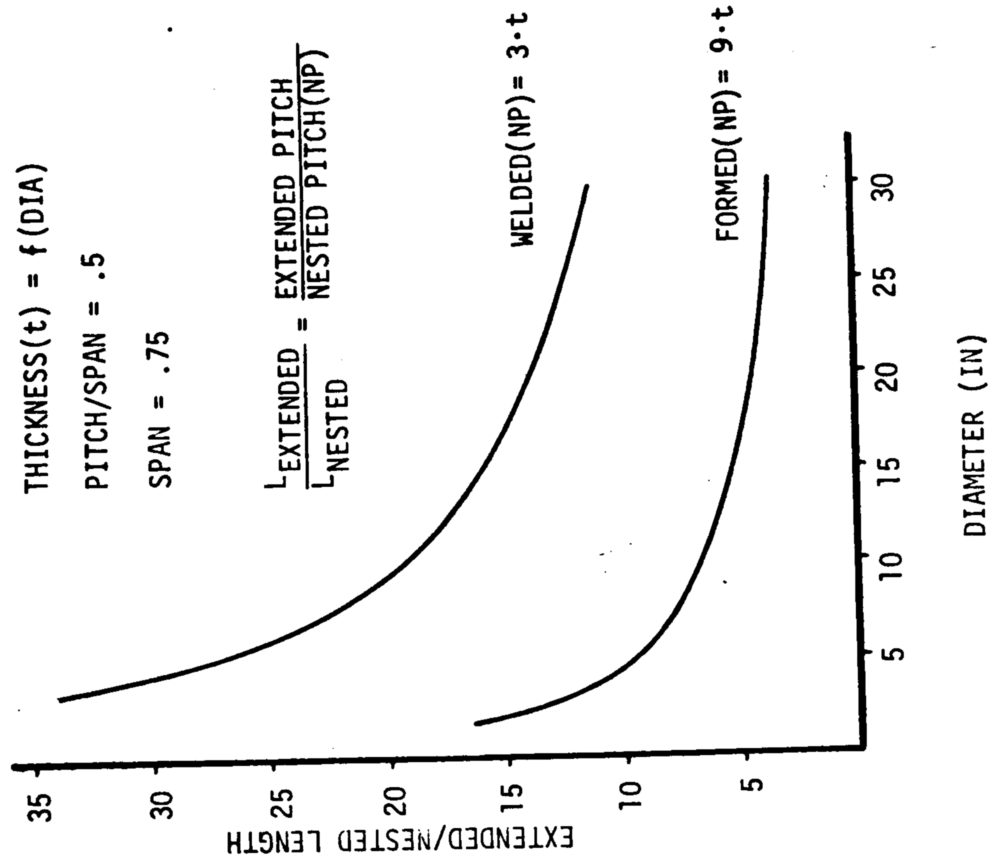


Figure A-4

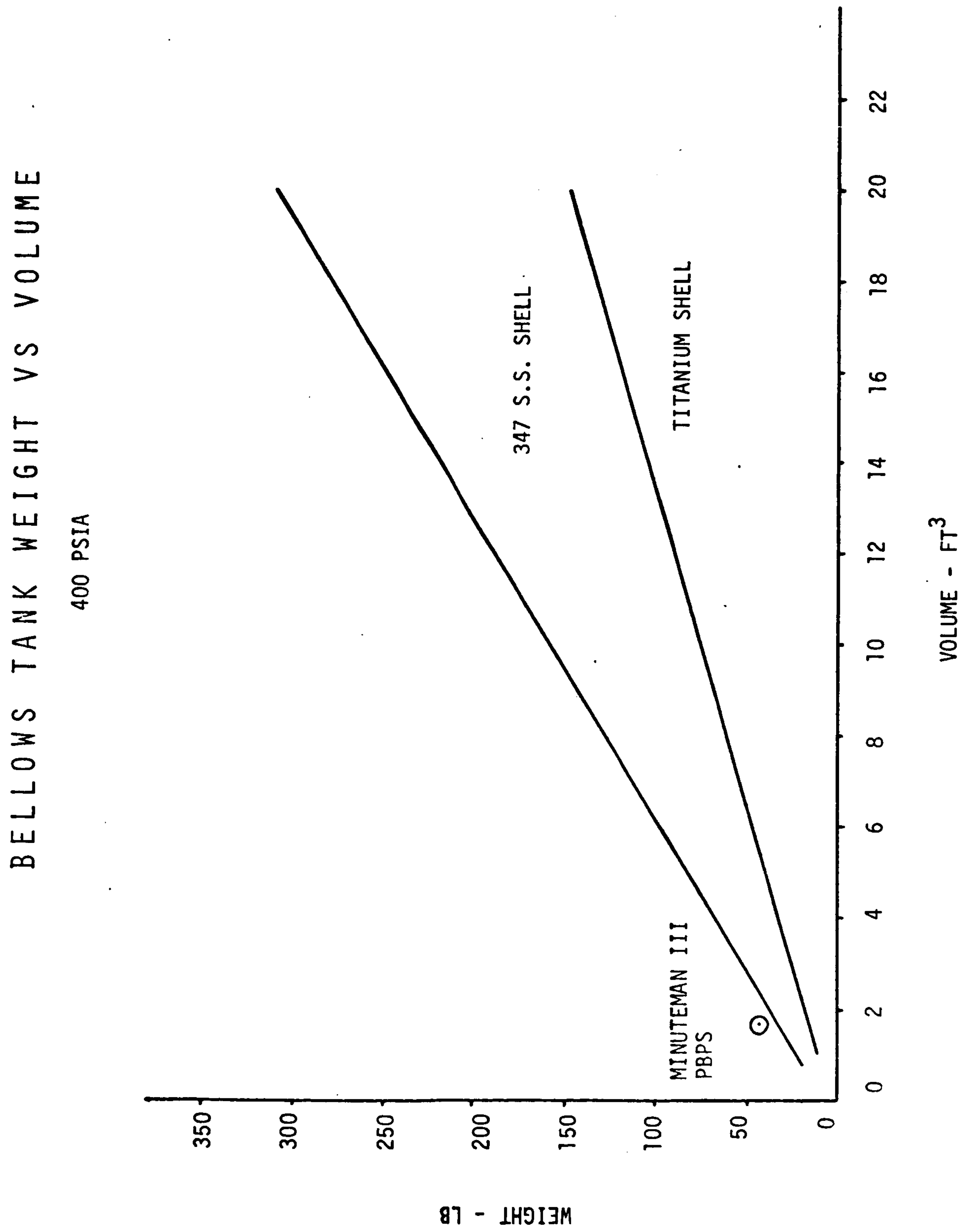

BELLOWS TANK WEIGHT VS VOLUME
400 PSIA
347 S.S. SHELL
TITANIUM SHELL
MINUTEMAN III
PBPS
WEIGHT - LB
0
50
100
150
200
250
300
350
VOLUME - FT^3
0
2
4
6
8
10
12
14
16
18
20
22

Figure A-5

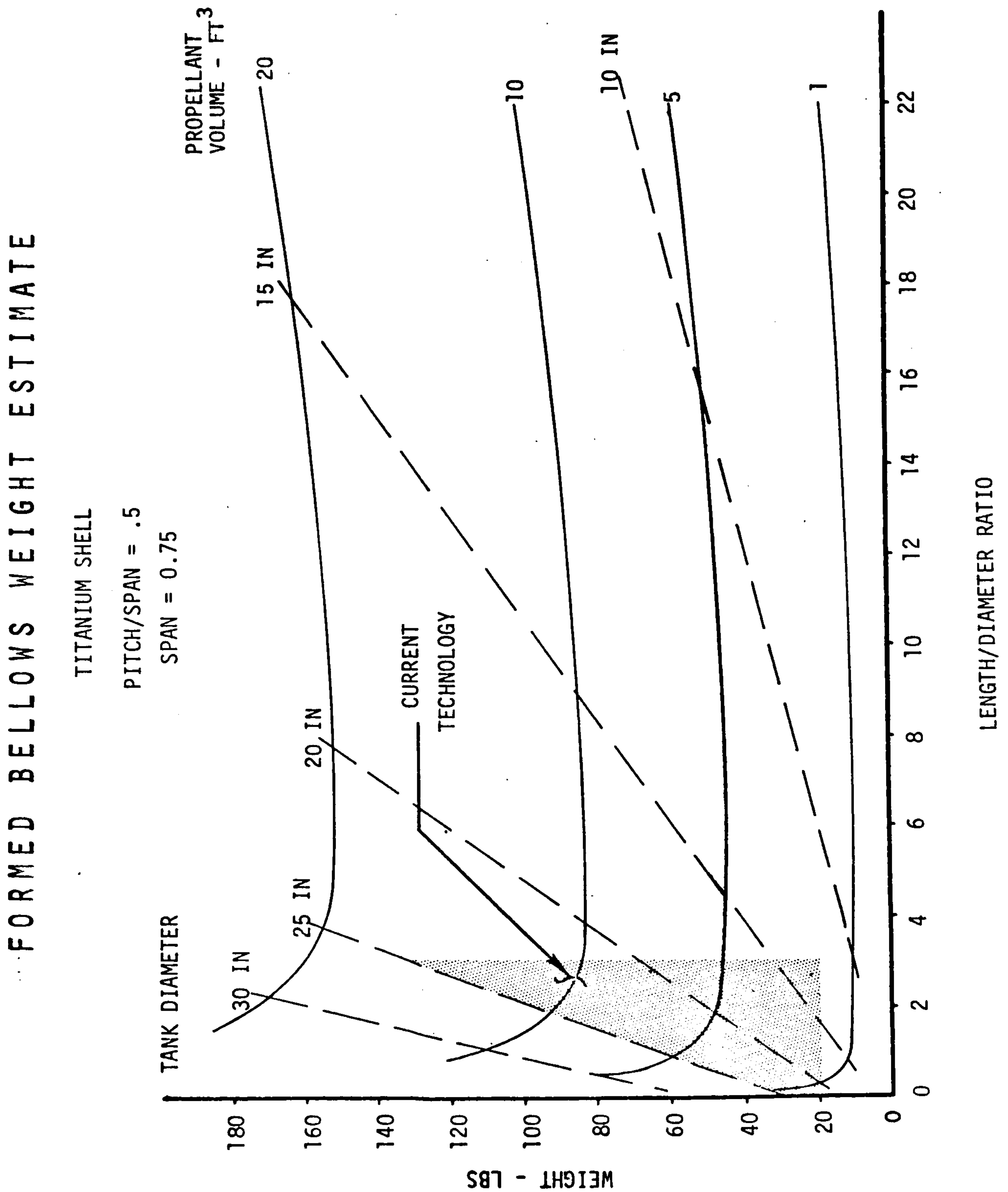

FORMED BELLOWS WEIGHT ESTIMATE
TITANIUM SHELL
PITCH/SPAN = .5
SPAN = 0.75
TANK DIAMETER
30 IN
25 IN
20 IN
15 IN
10 IN
CURRENT TECHNOLOGY
PROPELLANT VOLUME - FT^3
20
10
5
1
WEIGHT - LBS
0
20
40
60
80
100
120
140
160
180
LENGTH/DIAMETER RATIO
0
2
4
6
8
10
12
14
16
18
20
22

Figure A-6

HIGH PRESSURE BELLOWS ACCUMULATORS

Bell Aerospace Company

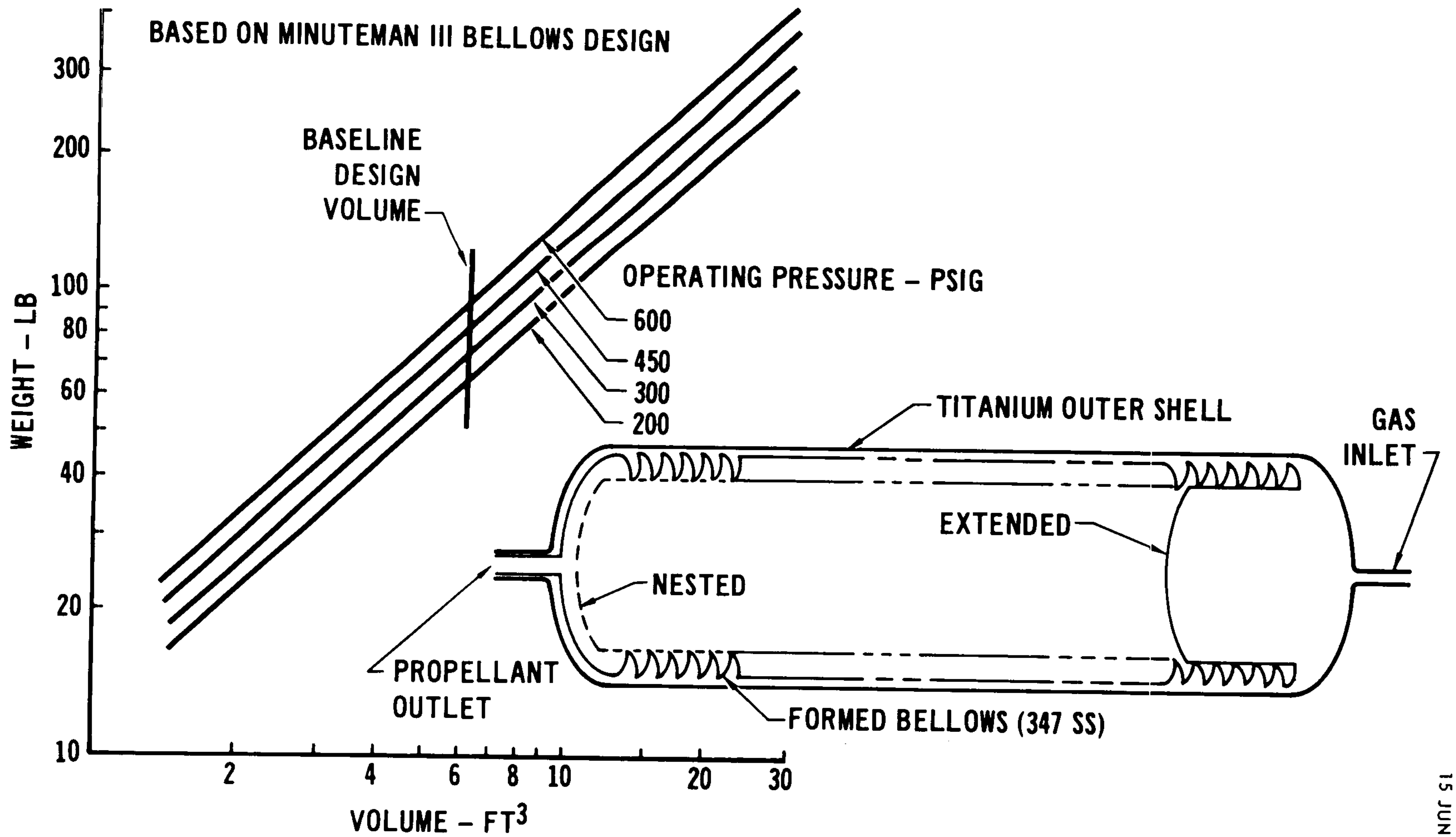


Figure A-7

REGULATED HELIUM PRESSURIZATION

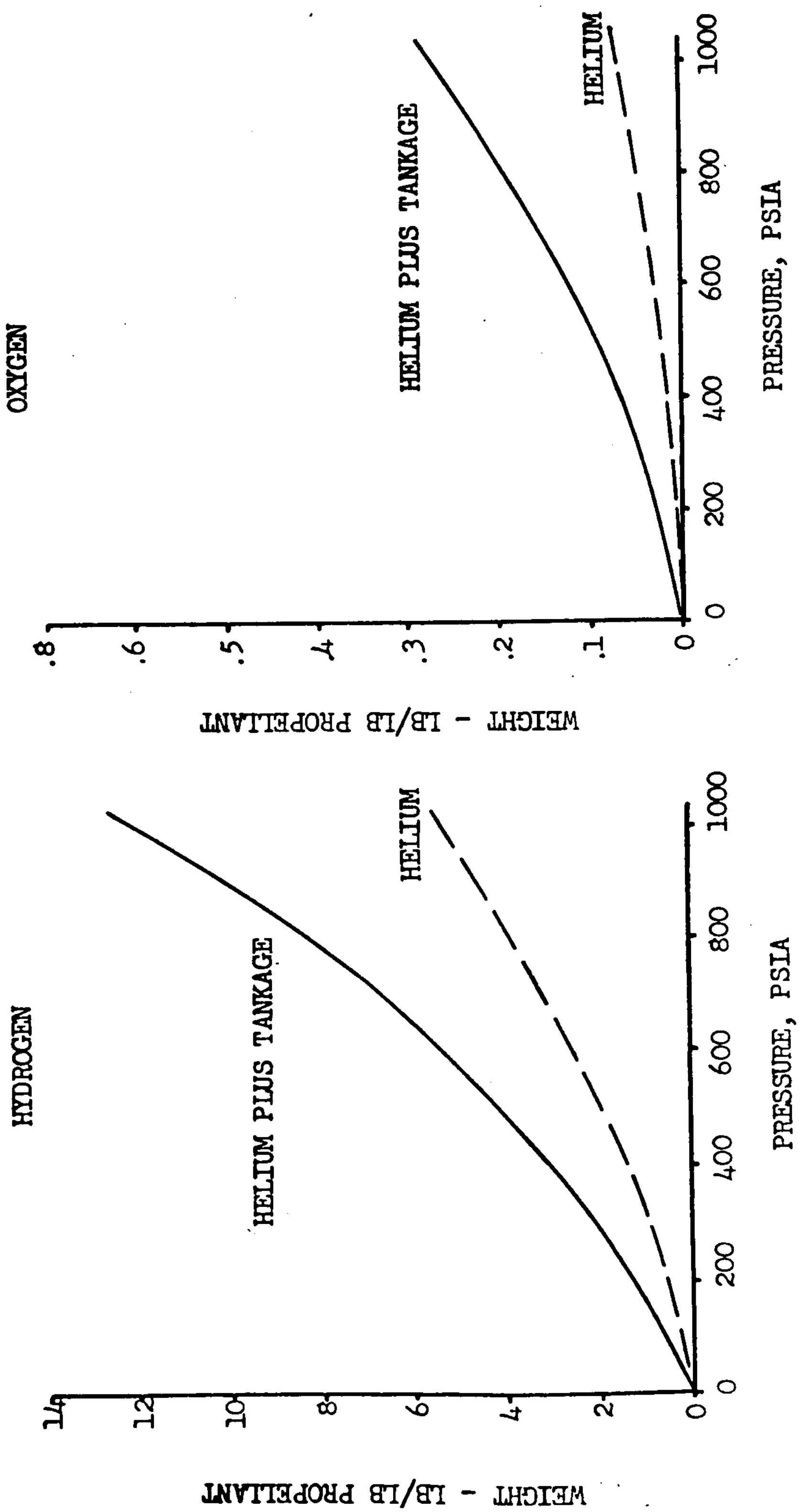


Figure A-8

case the pressurant gas is in direct contact with the liquid propellant and pressurant solubility effects must be considered. Pressurant solubility was determined from Reference (m). As shown in Figure A-9, the helium solubility in hydrogen is significant at high pressure and could result in an additional weight penalty on the order of 10% of the hydrogen propellant weight.

Blowdown pressurization of the liquid accumulators, where the accumulator pressure decays to a predetermined final value, was also evaluated. This concept is particularly attractive,since the accumulator pressurant need not be vented during refill as is the case with regulated pressurization. Instead, the pressurant is recompressed by pumped liquid when the accumulator is refilled. With this approach, maximum tank pressure is established by the ratio of gas volume to total tank volume. The gas volume ratio was selected on the basis of minimum tank weight. Weights for blowdown pressurization with screen acquisition (solubility effects included) are shown in Figure A-10. Minimum total weight occurs with gas volumes of 70 to 80% (hydrogen) and 60 to 70% (oxygen). Gas volumes in excess of these result in excessive tank pressure. The weights shown are for 100 lb hydrogen capacity and 400 lb oxygen capacity. However, the gas ratio selection was found to be valid for the range of accumulator volumes of interest.

The weights associated with blowdown pressurization can be significantly reduced by using bellows tankage,thereby avoiding the weight penalty associated with pressurant gas solubility which is especially critical at the high pressures required for blowdown pressurization. Blowdown pressurization weights with bellows tanks are shown in Figure A-11. With bellows, minimum tank weight occurs for gas volume ratios of 45% (hydrogen) and 33% (oxygen) of the total volume. These are lower than with screen acquisition and result in higher initial tank pressures for a given final pressure. The initial and final pressure levels for both screen and bellows accumulators are summarized in Figure A-12.

A comparison of regulated and blowdown pressurization with both screen and bellows tanks is shown in Figure A-13 for hydrogen and Figure A-14 for oxygen. The lightest concept is blowdown pressurization with bellows tankage. It can be seen that the use of either screen tankage or regulated pressurization would incur large weight penalties, primarily due to solubility and vent losses.

PRESSURIZATION GAS SOLUBILITY

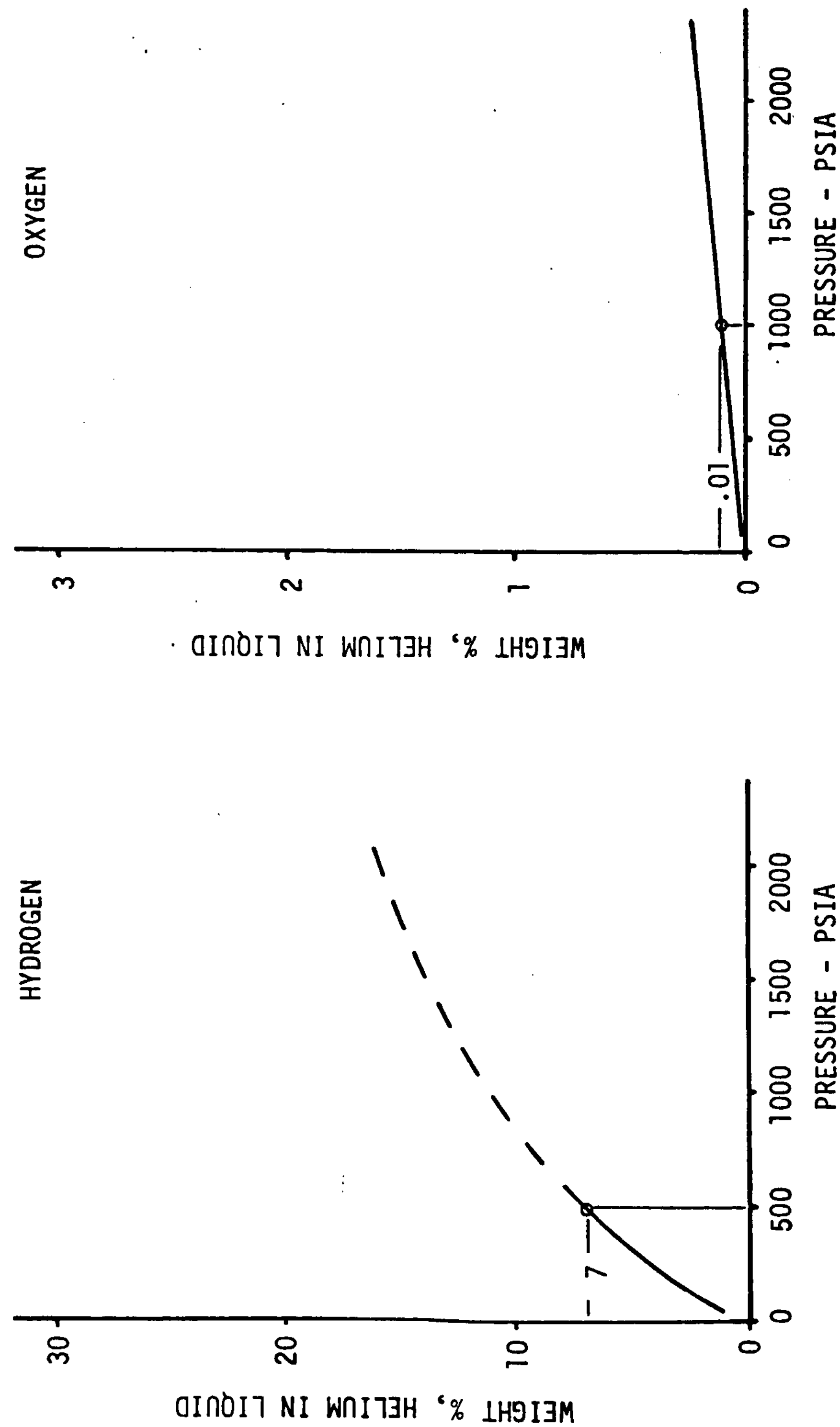


Figure A-9

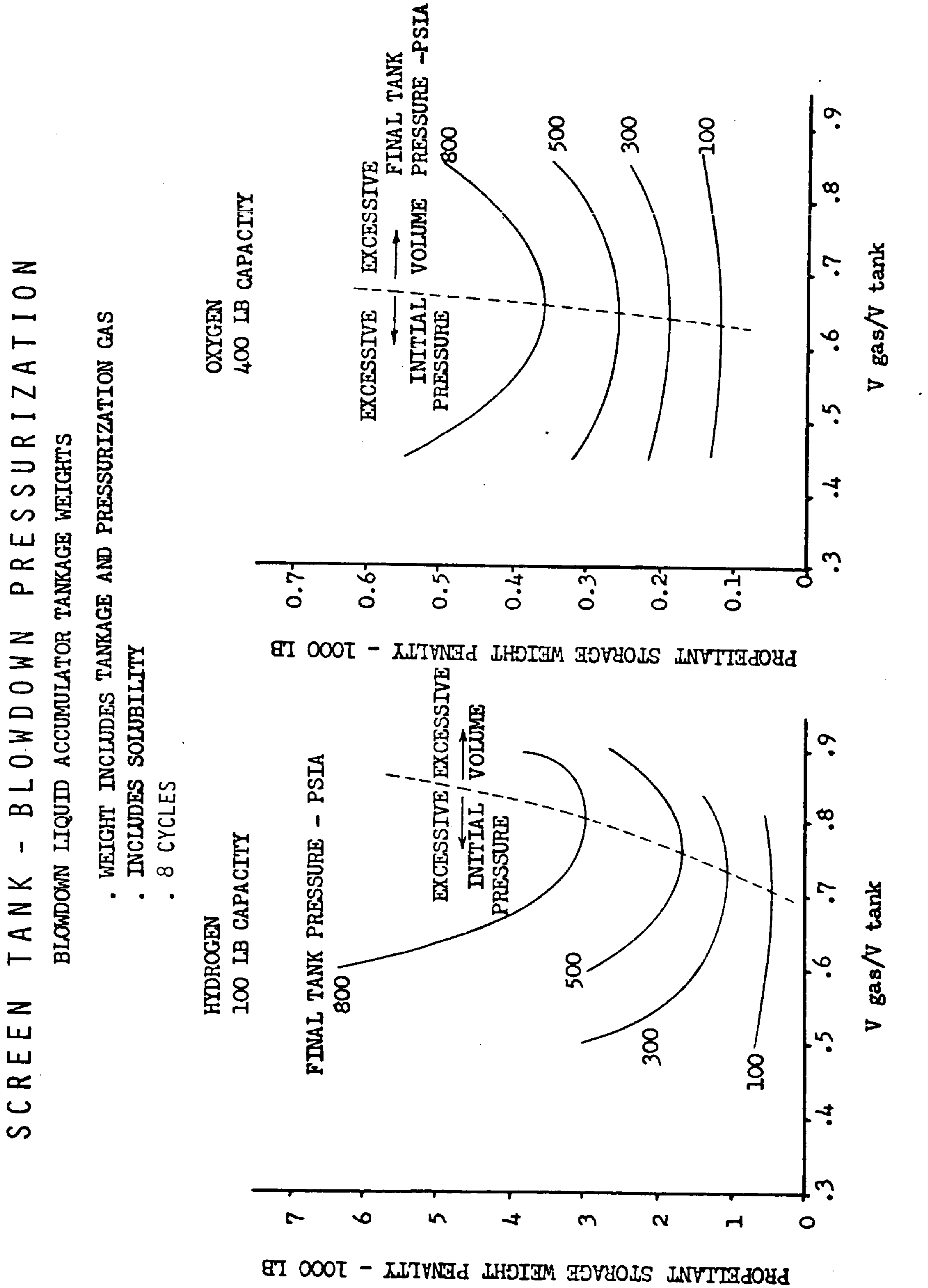

SCREEN TANK - BLOWDOWN PRESSURIZATION
BLOWDOWN LIQUID ACCUMULATOR TANKAGE WEIGHTS
. WEIGHT INCLUDES TANKAGE AND PRESSURIZATION GAS
. INCLUDES SOLUBILITY
. 8 CYCLES
HYDROGEN
100 LB CAPACITY
FINAL TANK PRESSURE - PSIA
800
500
300
100
EXCESSIVE INITIAL PRESSURE
EXCESSIVE VOLUME
PROPELLANT STORAGE WEIGHT PENALTY - 1000 LB
0
1
2
3
4
5
6
7
.3
.4
.5
.6
.7
.8
.9
V gas/V tank
OXYGEN
400 LB CAPACITY
FINAL TANK PRESSURE -PSIA
800
500
300
100
EXCESSIVE INITIAL PRESSURE
EXCESSIVE VOLUME
PROPELLANT STORAGE WEIGHT PENALTY - 1000 LB
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
.3
.4
.5
.6
.7
.8
.9
V gas/V tank

Figure A-10

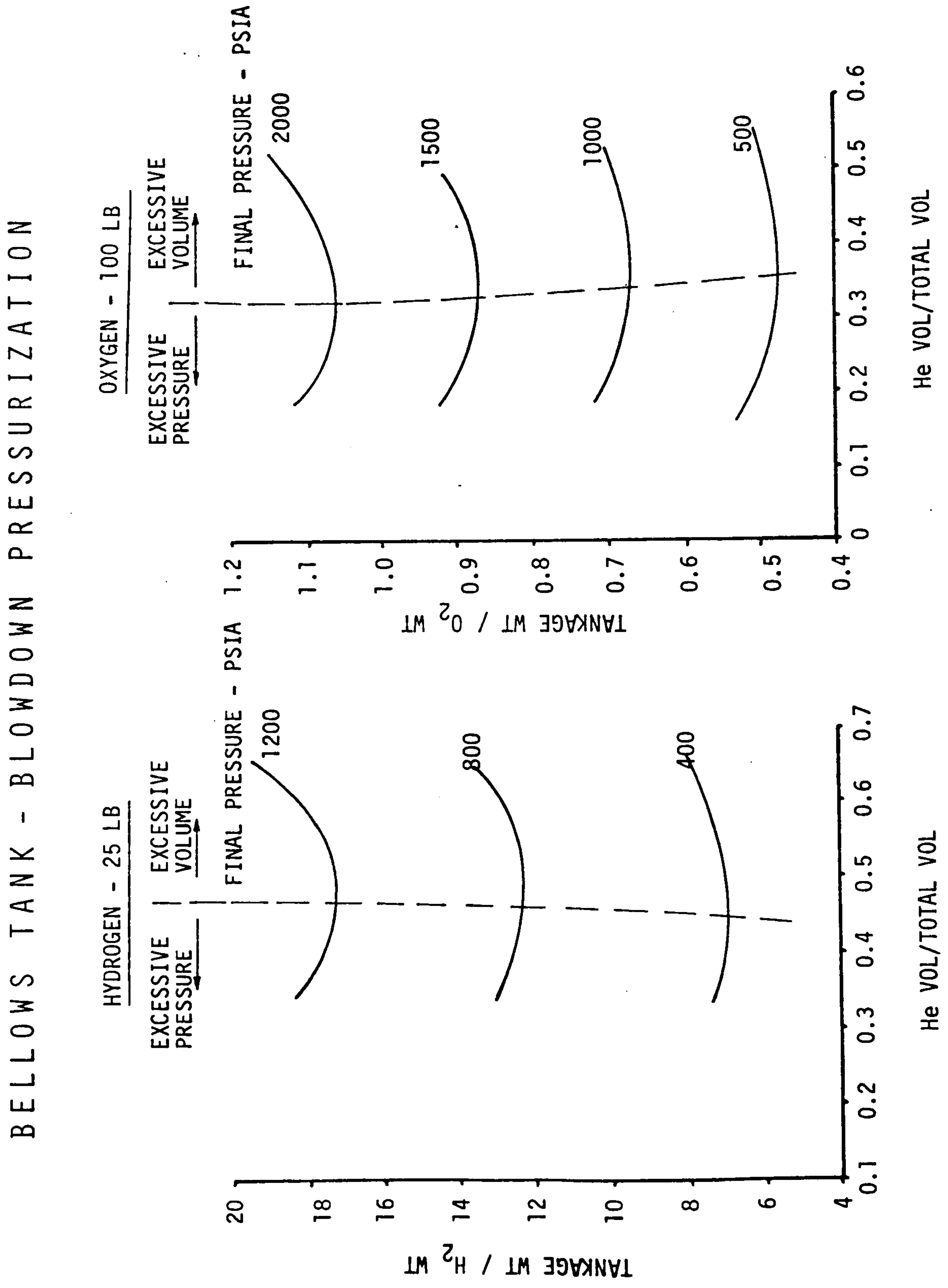

BELLOWS TANK - BLOWDOWN PRESSURIZATION
OXYGEN - 100 LB
EXCESSIVE PRESSURE
EXCESSIVE VOLUME
FINAL PRESSURE - PSIA
2000
1500
1000
500
TANKAGE WT / O_2 WT
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0
0.1
0.2
0.3
0.4
0.5
0.6
He VOL/TOTAL VOL
HYDROGEN - 25 LB
EXCESSIVE PRESSURE
EXCESSIVE VOLUME
FINAL PRESSURE - PSIA
1200
800
400
TANKAGE WT / H_2 WT
20
18
16
14
12
10
8
6
4
0.1
0.2
0.3
0.4
0.5
0.6
0.7
He VOL/TOTAL VOL

Figure A-11

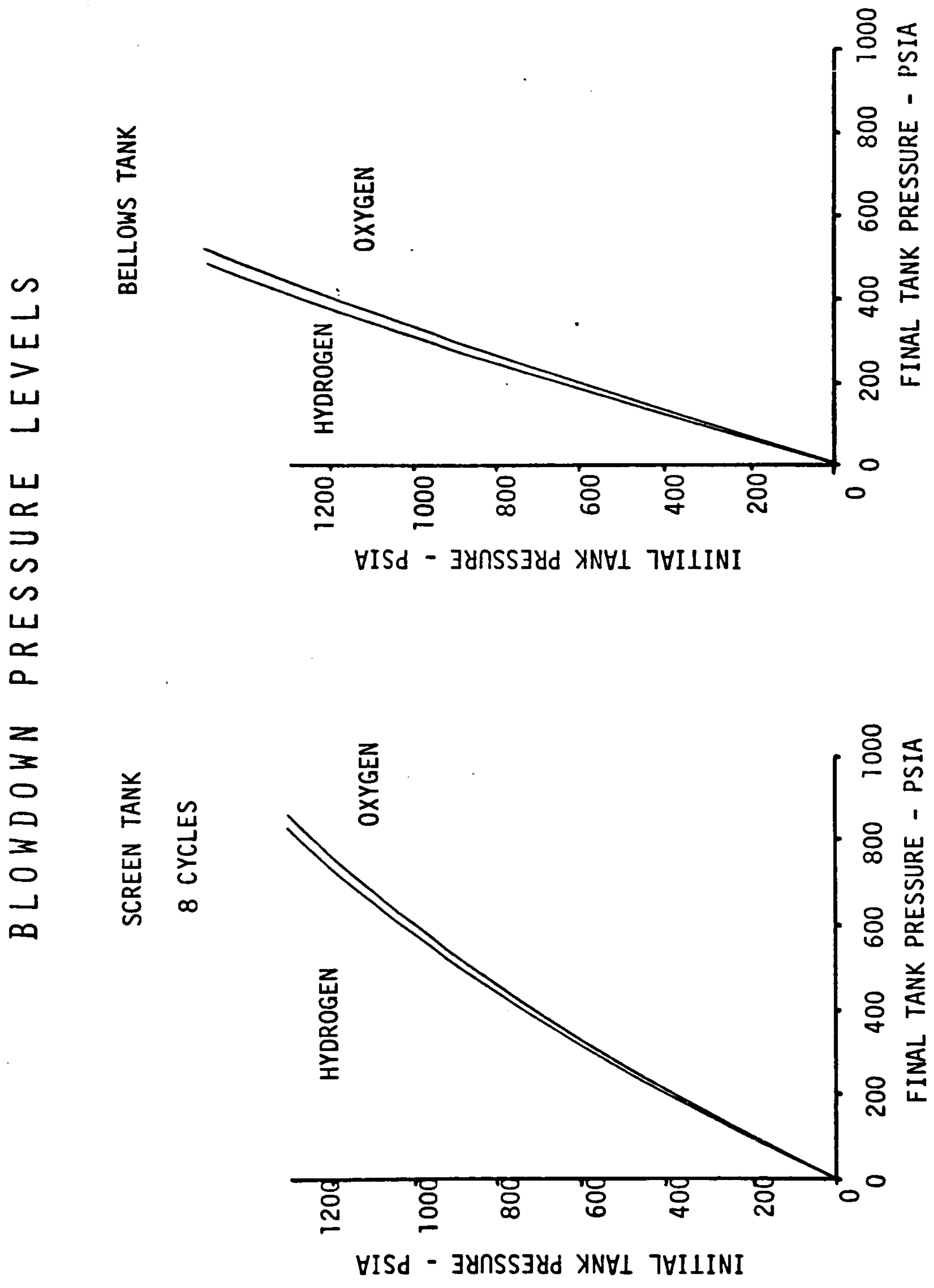

BLOWDOWN PRESSURE LEVELS
SCREEN TANK
8 CYCLES
HYDROGEN
OXYGEN
INITIAL TANK PRESSURE - PSIA
0
200
400
600
800
1000
1200
FINAL TANK PRESSURE - PSIA
0
200
400
600
800
1000
BELLOWS TANK
HYDROGEN
OXYGEN
INITIAL TANK PRESSURE - PSIA
0
200
400
600
800
1000
1200
FINAL TANK PRESSURE - PSIA
0
200
400
600
800
1000

Figure A-12

HYDROGEN PRESSURIZATION CONCEPT WEIGHT COMPARISON

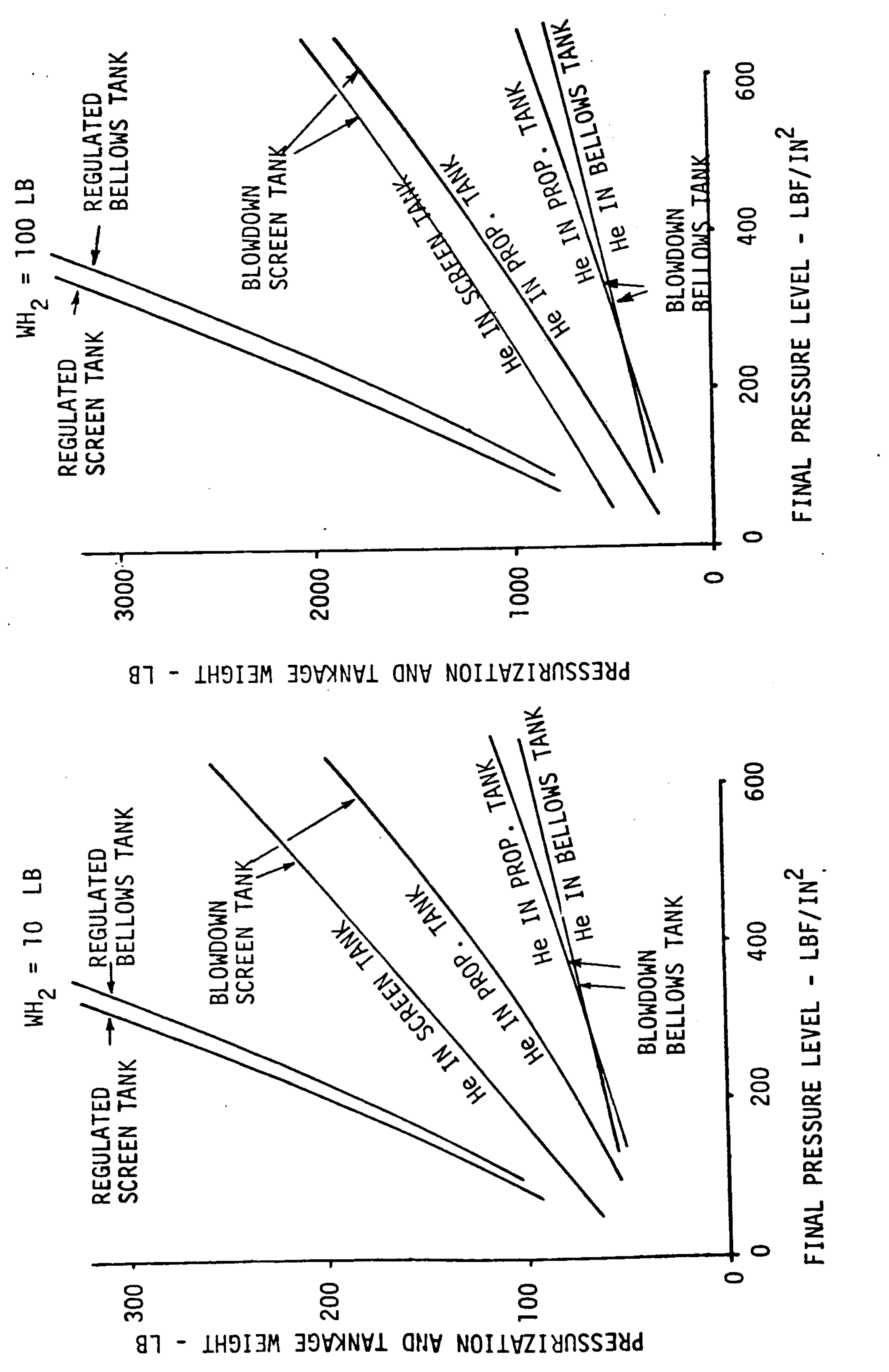


Figure A-13

OXYGEN PRESSURIZATION CONCEPT WEIGHT COMPARISON

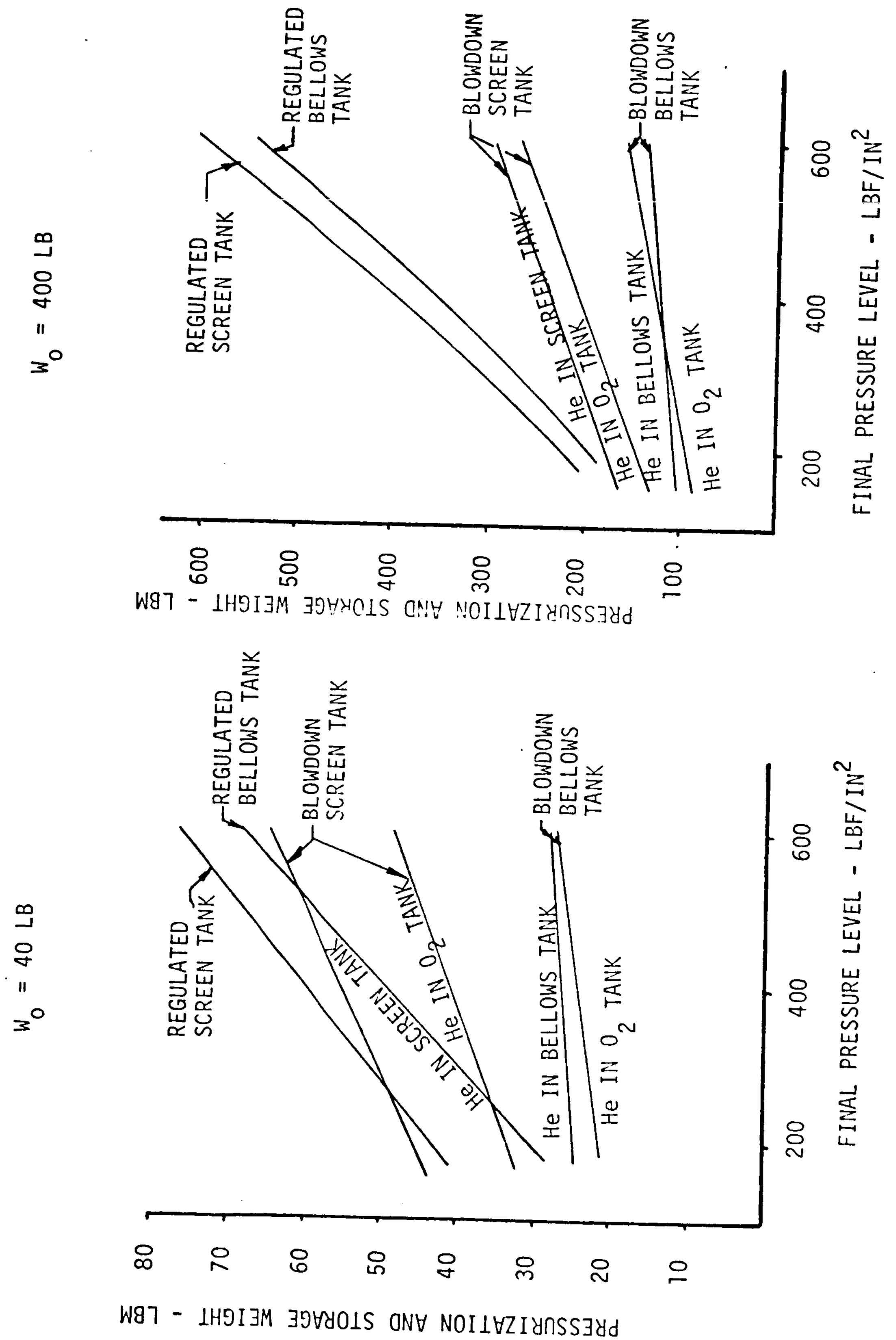


Figure A-14

A2.4 Motor Operated Pump - Pump characteristics in the initial study phases were based on centrifugal pump data provided by Pesco Products. This data base was subsequently expanded to cover alternate positive displacement pump concepts under a separate subcontract to Pesco Products. Work performed under this subcontract is presented in Appendix D.

The initial centrifugal pump data is shown in Figures A-15 and A-16 for oxygen and hydrogen respectively, over the range of flow and head rise values required. The oxygen pump design characteristics are less severe than those for hydrogen and a single stage pump design satisfies the range of requirements. Satisfactory oxygen pump specific speeds were possible in both the liquid and gaseous system types using a 400 cps power supply, consistent with Phase B Space Shuttle APU capability. The operating conditions of the hydrogen pumps in the two system types investigated had somewhat different power supply requirements. In the case of the liquid systems,pump head requirements at minimum system weight were considerably lower than those necessary to provide minimum weight in the gaseous systems because of the pressure amplification induced by the gaseous accumulators. Multiple stage hydrogen pumps were required in both system types but, even with multiple stages, satisfactory designs could not be achieved in the gaseous systems with a 400 cps power supply. This results from the relationship of speed, head rise and pump specific speed. In order to realize the desired specific speed, a high head rise required high pump speed,and while 400 cps power supplies were satisfactory for the liquid system hydrogen pumps,they were too low for the gaseous system hydrogen pumps. In the gaseous systems,higher frequency electrical supplies were required to operate at a speed sufficiently high to limit the number of hydrogen pump stages to 3 or less. The data shown in Figures A-15 and A-16 are for the hydrogen pump speed associated with an 800 cps power supply, twice that provided by the current shuttle APU design. Additional details concerning centrifugal pump design and performance are given in Appendix E.

A summary of the pump weights is shown in Figure A-17. These weights are relatively small, indicating that the other considerations will be more significant than pump weight. For example, the ideal pump horsepower requirements are shown in Figure A-18. The power requirements for hydrogen range from 20 to 200 hp and those for oxygen range from 4 to 40 hp, depending on pump flow and head rise. The significance of these requirements is shown in Figure A-19 for hydrogen pumps. The Phase B Shuttle APU provided 20 hp electrically and 275 hp hydraulically.

PESCO PRODUCTS HYDROGEN PUMP DESIGN DATA

HEAD RISE (PSI)		HYDROGEN FLOWRATE - LB/SEC .1	.9	1.7	2.5
50	WEIGHT (LB)	4.3	6.6	8.4	9.9
	NO. STAGES	2	1	1	1
	STG. SPECIFIC SPEED	938	1672	2301	2778
	SHAFT POWER (HP)	.5	4.1	7.5	10.9
200	WEIGHT (LB)	11.9	18.9	21.6	22.6
	NO. STAGES	4	2	2	2
	STG. SPECIFIC SPEED	557	993	1367	1650
	SHAFT POWER (HP)	2.5	18	32	45
350	WEIGHT (LB)	—	25.1	33.5	40
	NO. STAGES		4	2	2
	STG. SPECIFIC SPEED		1098	897	1083
	SHAFT POWER (HP)		31	60	84
500	WEIGHT (LB)	—	—	46.5	55.2
	NO. STAGES			4	2
	STG. SPECIFIC SPEED			1155	828
	SHAFT POWER (HP)			83	125

Figure A-15

PESCO PRODUCTS OXYGEN PUMP DESIGN DATA

- SINGLE STAGE PUMPS

HEAD RISE (PSI)		OXYGEN FLOWRATE - LB/SEC 4	6	8	10
50	WEIGHT (LB)	5.6	10.8	12.7	13.5
	STG. SPECIFIC SPEED	3537	2928	3379	3780
	SHAFT POWER (HP)	1.3	2.0	2.7	3.4
200	WEIGHT (LB)	6.4	7.8	9.	10.1
	STG. SPECIFIC SPEED	2500	3065	3537	3957
	SHAFT POWER (HP)	5.4	8.1	10.8	13.6
350	WEIGHT (LB)	11.0	12.3	13.8	14.6
	STG. SPECIFIC SPEED	1642	2013	2324	2593
	SHAFT POWER (HP)	9.8	14.5	19.1	23.6
500	WEIGHT (LB)	15.0	16.8	18.1	19.1
	STG. SPECIFIC SPEED	1255	1539	1777	1988
	SHAFT POWER (HP)	14.8	21.2	27.9	33.8

Figure A-16

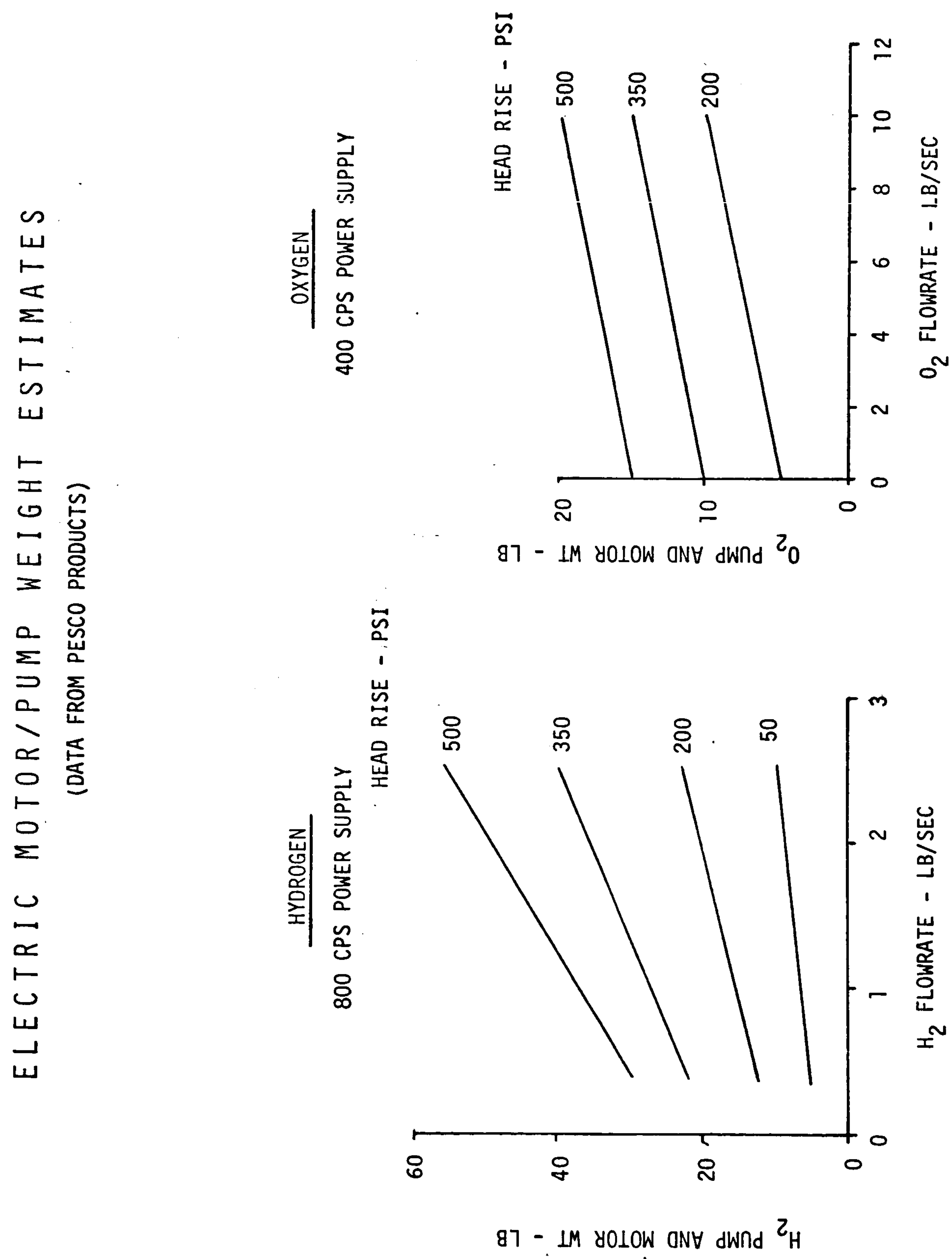

ELECTRIC MOTOR/PUMP WEIGHT ESTIMATES
(DATA FROM PESCO PRODUCTS)
HYDROGEN
800 CPS POWER SUPPLY
HEAD RISE - PSI
500
350
200
50
H_2 PUMP AND MOTOR WT - LB
0
20
40
60
H_2 FLOWRATE - LB/SEC
0
1
2
3
OXYGEN
400 CPS POWER SUPPLY
HEAD RISE - PSI
500
350
200
O_2 PUMP AND MOTOR WT - LB
0
10
20
O_2 FLOWRATE - LB/SEC
0
2
4
6
8
10
12

Figure A-17

PUMP FLUID POWER REQUIREMENTS

(EFFICIENCIES NOT INCLUDED)

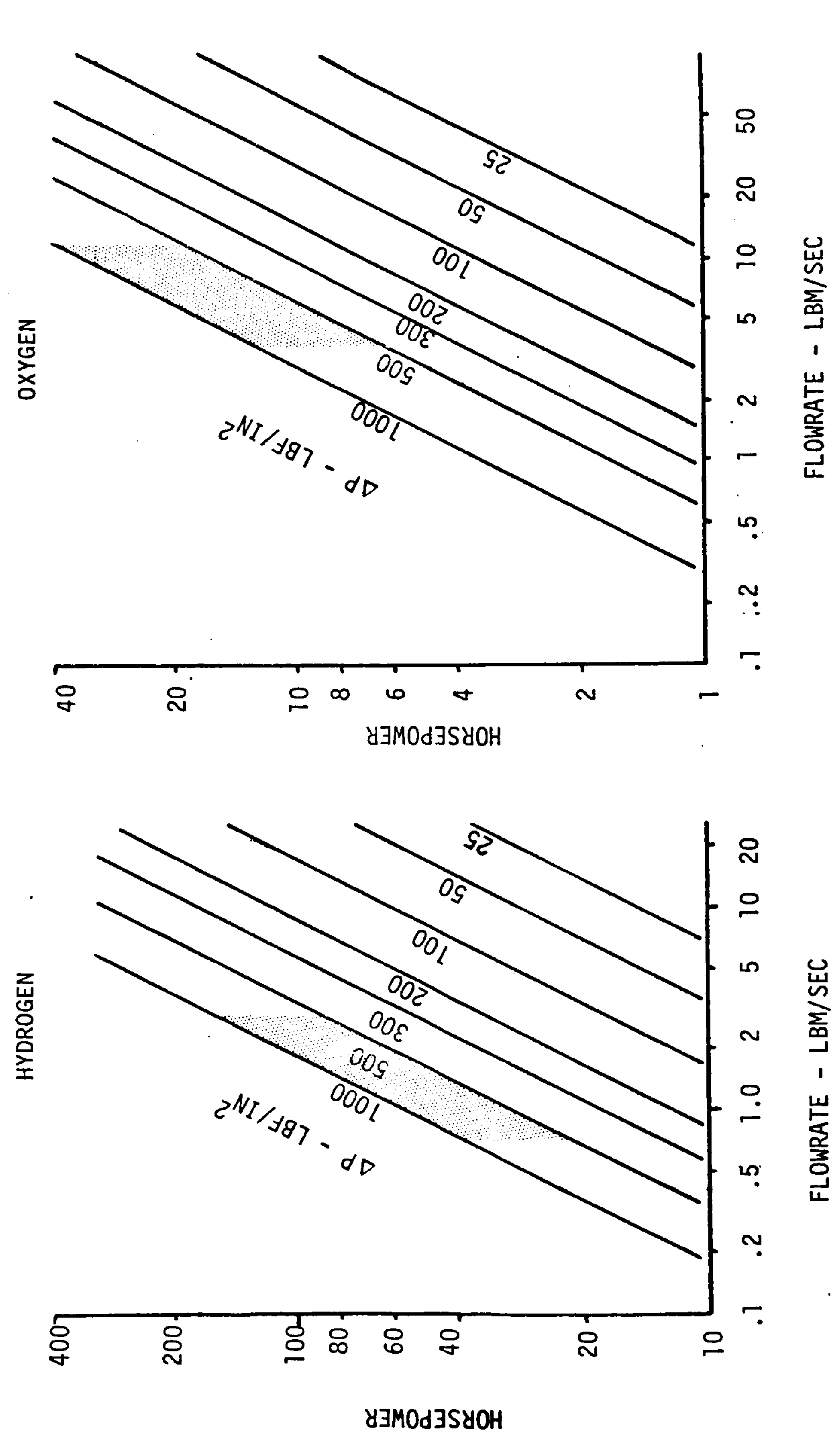


Figure A-18

PUMP AND DRIVE WEIGHT VS HORSEPOWER

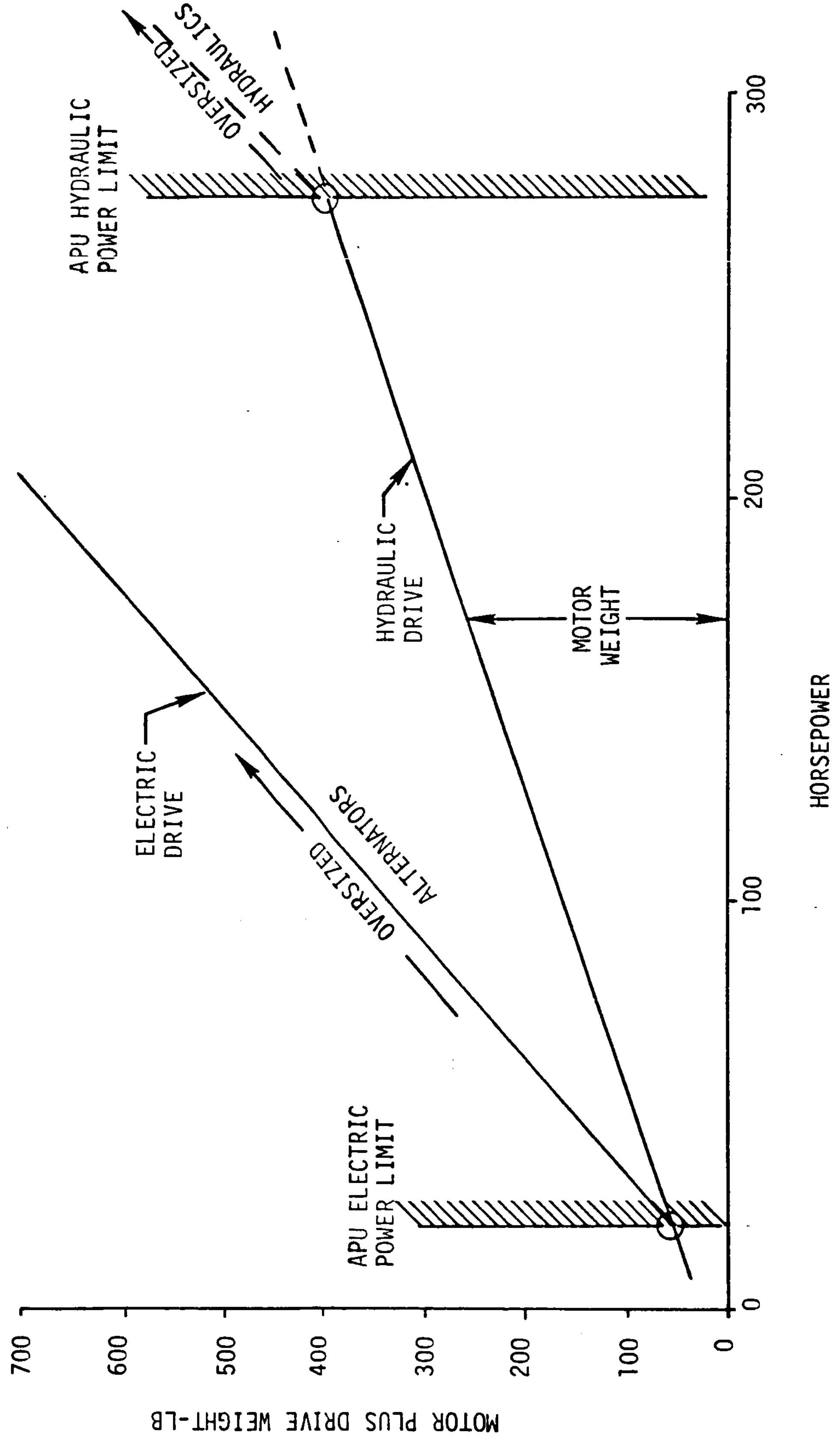


Figure A-19

The use of electric motors would result in an additional weight penalty, due to alternator resizing, for power requirements greater than 20 hp. Thus, electric motor pumps would be more attractive at low power requirements and hydraulic motor pumps would be attractive at high power requirements. Specific pump sizing requirements and effects are discussed under the gaseous and liquid system study results presented in Paragraph 3.0.

A2.5 APU Characteristics - The APU characteristics were based on the O_2/H_2 APU design from the Phase B Space Shuttle vehicle studies. The weight penalty required for power generation is shown in Figure A-20. The weight increment shown is for APU propellant, tankage and pressurization required for pump operation. The APU operating time required for delivery of the RCS total impulse (2.23×10^6 lb-sec) is fixed by the pump flow rate and this defines the equivalent pump thrust. Thus, the two parameters used in Figure A-20 relate the operating horsepower of the APU and operating time. For a given power requirement, low pump flow rates (low pump thrust) require longer operating times which would increase the APU propellant consumption as APU specific propellant consumption varies from 1.7 lb/hp-hr at 100% power to 2.05 lb/hp-hr at idle. The data of Figure A-20 indicate that high flow rates and low horsepower are most desirable because they incur a minimum propellant penalty.

A2.6 Gaseous RCS Propellant Distribution - Models for gaseous accumulators, gas generators, heat exchangers, turbopumps and associated lines and valves were based on the study effort described in Reference (h). The gas generators operate at 2000°R. Individual component models were grouped as composite assemblies for the gaseous system studies. For example, gaseous feed system weights are shown in Figure A-21 as a function of chamber pressure, number of accumulator cycles and maximum accumulator pressure. These data were used in system design studies to determine the chamber pressure and accumulator pressures resulting in minimum system weight (paragraph 3.0).

A2.7 Liquid RCS Distribution - Vacuum jacketed feed lines are required to minimize heat input when the propellants are distributed as liquids. The distribution system layout is shown in Figure A-22. Primary distribution lines feed propellants from the storage tanks to the manifolds, which are located forward and aft in the vehicle, and to the wing tip mounted engine groups. The manifold design is illustrated in Figure A-23.

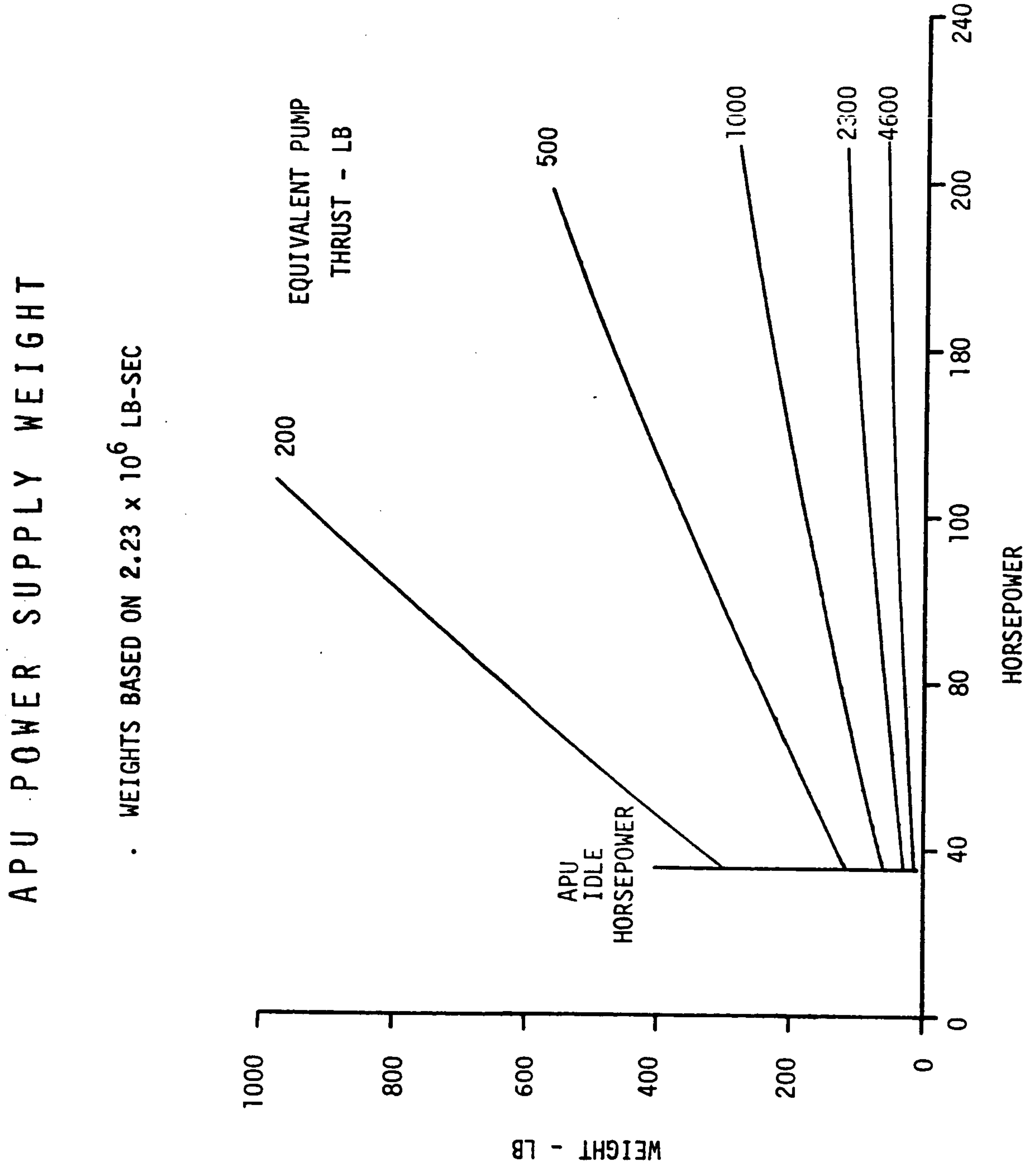

APU POWER SUPPLY WEIGHT
• WEIGHTS BASED ON 2.23 x 10^6 LB-SEC
EQUIVALENT PUMP THRUST - LB
200
500
1000
2300
4600
APU IDLE HORSEPOWER
WEIGHT - LB
0
200
400
600
800
1000
HORSEPOWER
0
40
80
100
180
200
240

Figure A-20

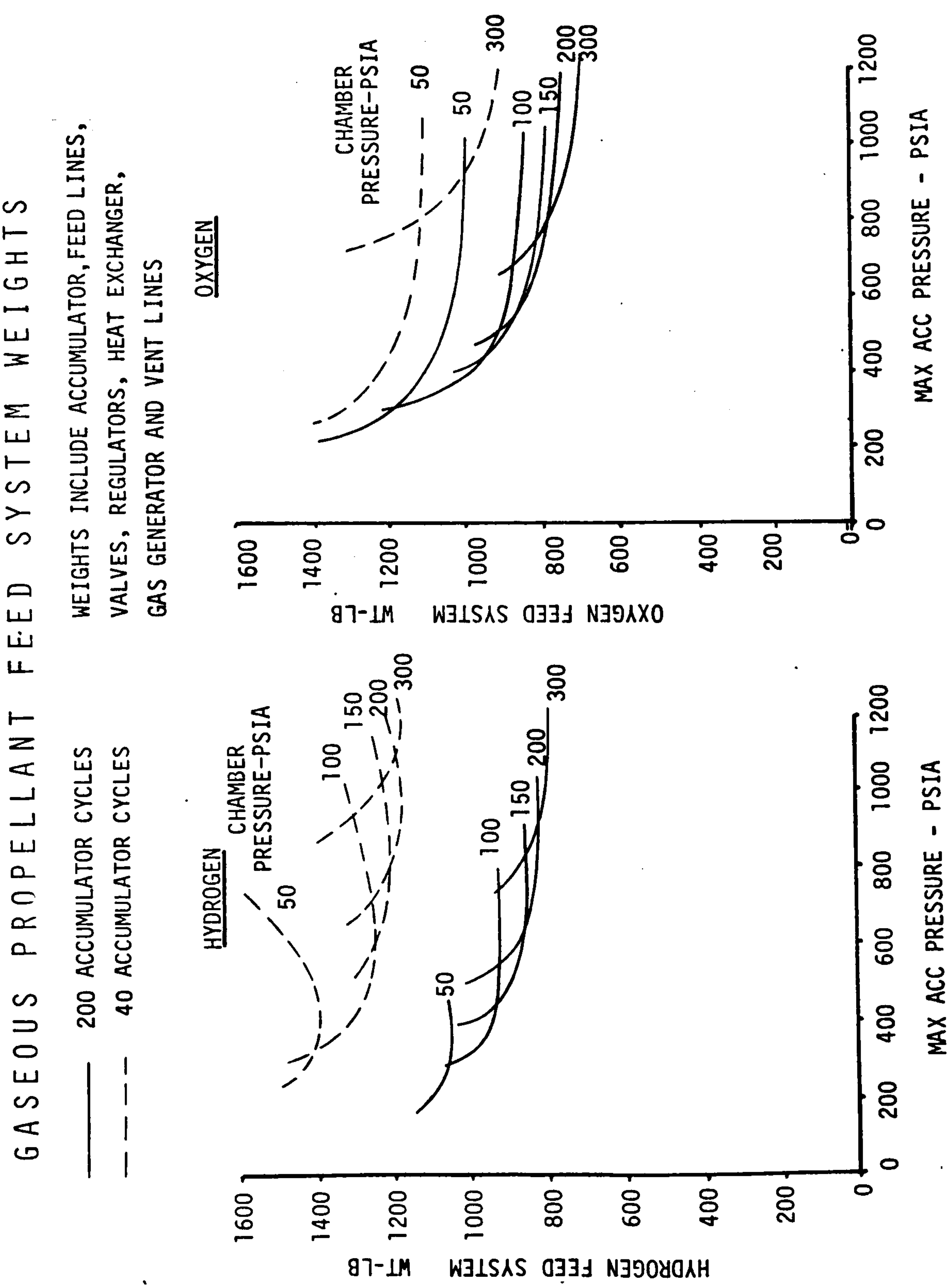

GASEOUS PROPELLANT FEED SYSTEM WEIGHTS
200 ACCUMULATOR CYCLES
40 ACCUMULATOR CYCLES
WEIGHTS INCLUDE ACCUMULATOR, FEED LINES, VALVES, REGULATORS, HEAT EXCHANGER, GAS GENERATOR AND VENT LINES
OXYGEN
CHAMBER PRESSURE-PSIA
50
100
150
200
300
OXYGEN FEED SYSTEM WT-LB
MAX ACC PRESSURE - PSIA
HYDROGEN
HYDROGEN FEED SYSTEM WT-LB
0
200
400
600
800
1000
1200
1400
1600

Figure A-21

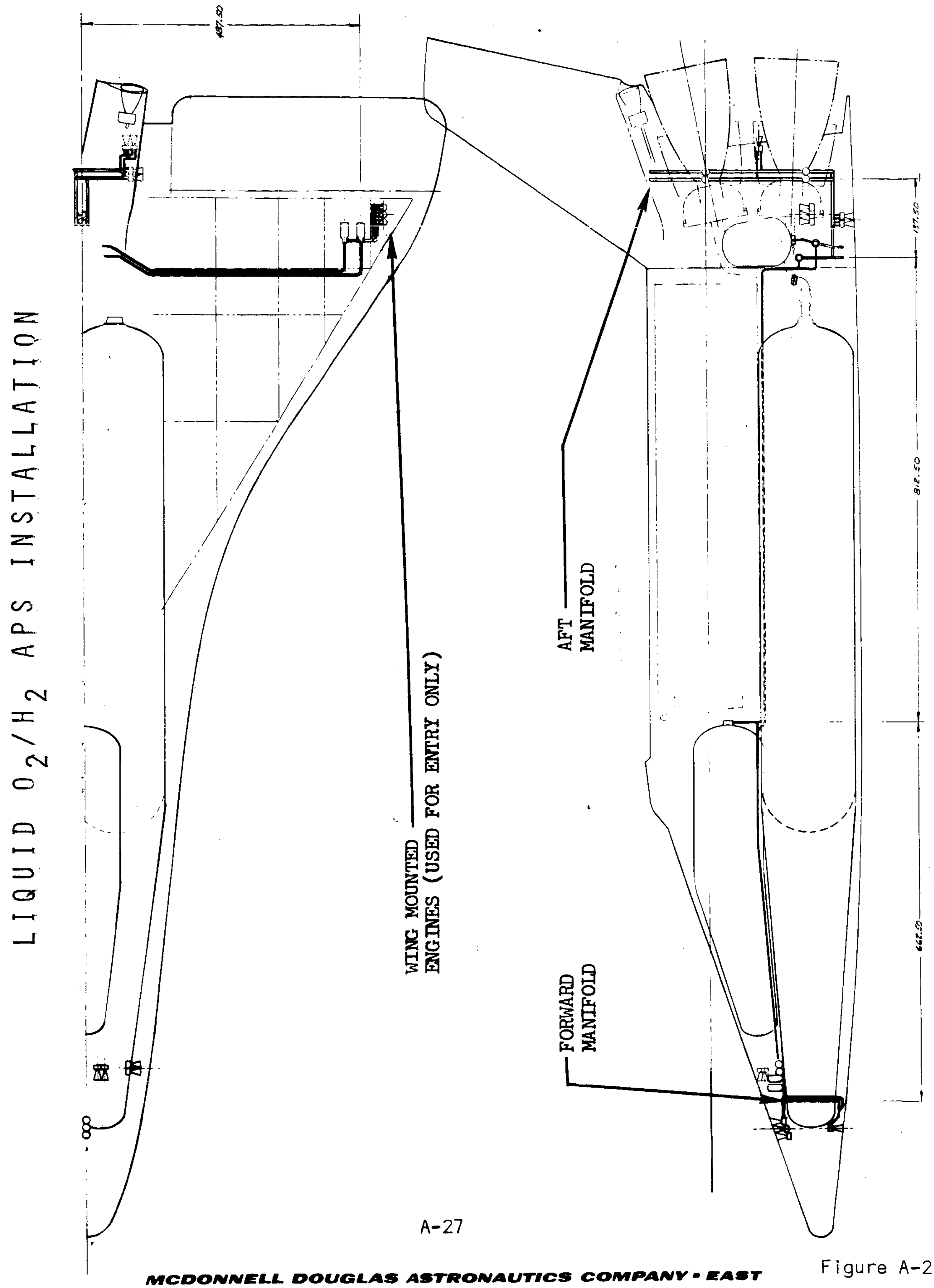

LIQUID O_2/H_2 APS INSTALLATION
WING MOUNTED ENGINES (USED FOR ENTRY ONLY)
AFT MANIFOLD
FORWARD MANIFOLD
487.50
137.50
812.50
662.50

Figure A-22

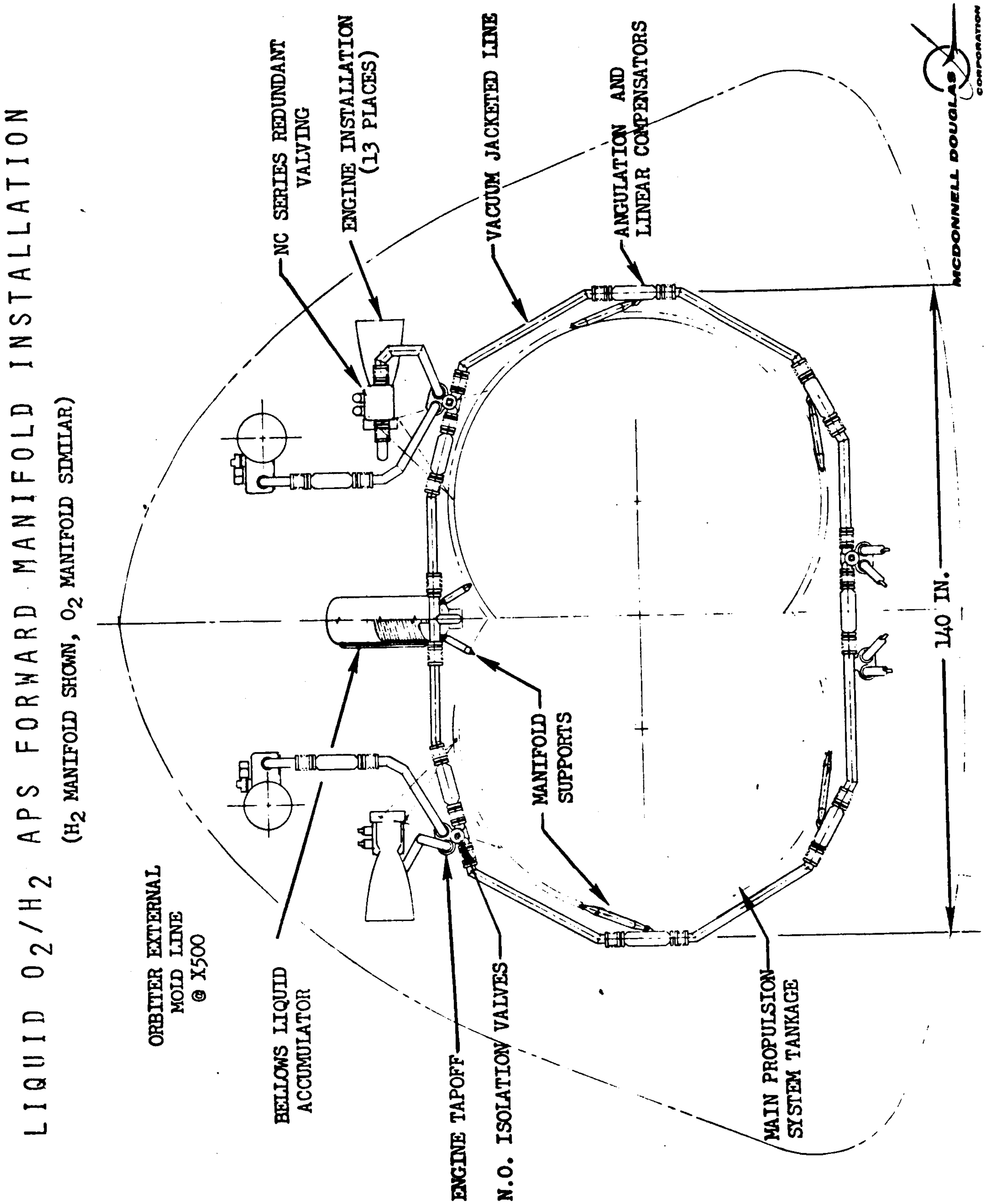

LIQUID O_2/H_2 APS FORWARD MANIFOLD INSTALLATION
(H_2 MANIFOLD SHOWN, O_2 MANIFOLD SIMILAR)
ORBITER EXTERNAL MOLD LINE @ X500
BELLOWS LIQUID ACCUMULATOR
ENGINE TAPOFF
N.O. ISOLATION VALVES
NC SERIES REDUNDANT VALVING
ENGINE INSTALLATION (13 PLACES)
VACUUM JACKETED LINE
ANGULATION AND LINEAR COMPENSATORS
MANIFOLD SUPPORTS
MAIN PROPULSION SYSTEM TANKAGE
140 IN.

Figure A-23

The use of ring shaped manifolds eliminates propellant traps and heating of stagnated propellant in manifold ends. Straight line segments were used throughout the manifold to facilitate installation. Compensators are provided to accommodate angular alignment and thermal contraction during line chilldown. The compensator design is shown in Figure A-24. The inner line assembly uses brazed line joints throughout and each compensator contains two angulation joints ($\pm 5°$) and a linear expansion junction (ΔT = 720°R). Manual couplings and bellows assemblies are provided in the outer jacket to allow inner line brazing during installation. Each compensator assembly weighs approximately 4 lb. Weights for the complete line assembly are shown in Figure A-25. It can be seen that the compensator weights are a significant portion of the total feed line weights.

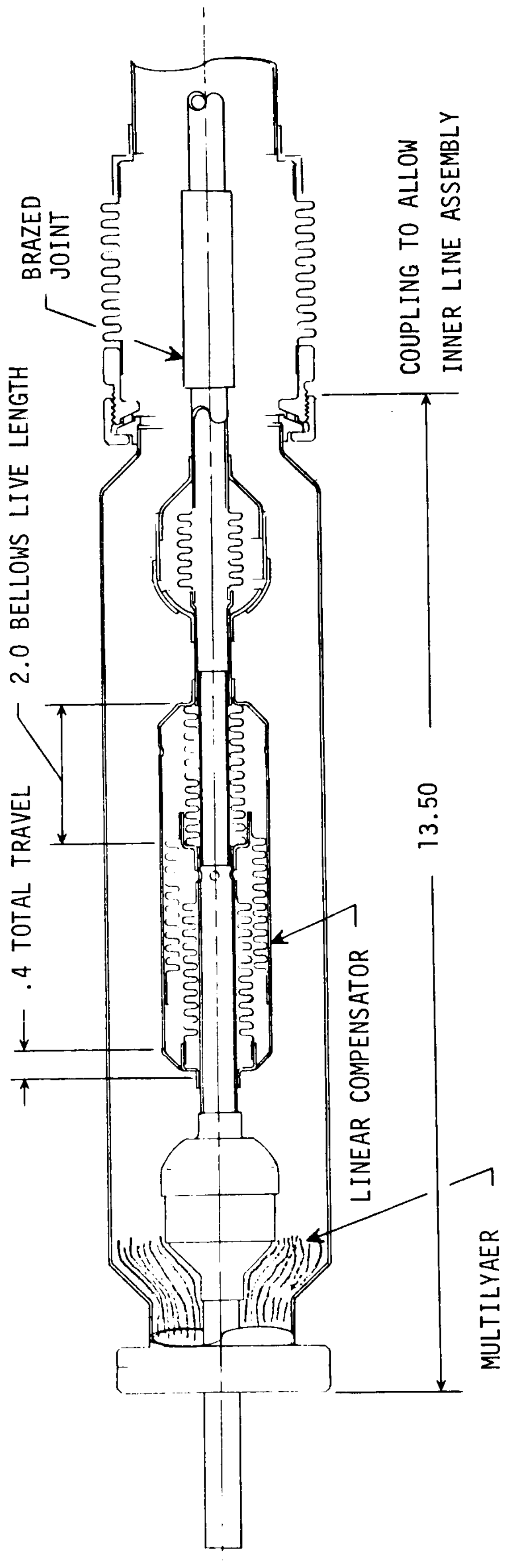
Line spacers are required between the inner line and the outer jacket to prevent crushing of the multilayer insulation. The spacer design used for weight and heat short estimates is shown in Figure A-26. The material was 0.060 in. thick fiberglass and resulted in a heat conduction path 2 in. long between the outer jacket and the inner line.

A2.8 Thruster Assemblies - The thruster data used for gaseous system design was taken from data generated by Aerojet Liquid Rocket Company under Contract No. NAS 8-26248 (Reference (a)). The gaseous engine is a ducted film cooled design, as shown in Figure A-27. Aerojet also provided preliminary design and performance estimates for a liquid O_2/H_2 thruster under this contract. The liquid O_2/H_2 thruster design, shown in Figure A-28 uses a combination of regenerative and film cooling. The engine, at the design point shown, weighed 14.5 lb (without valves) and provided a steady state specific impulse of 417.5 sec. Engine performance losses are shown in Figure A-29. This performance is less than that of the gaseous engine (436 sec) because the propellants are supplied as liquids (lower inlet enthalpy) and the nozzle contour was selected to minimize engine length and weight. The engine performance could be increased approximately 8 seconds with a high performance nozzle contour, however, the engine weight would increase approximately 2.5 lb per engine and this would effectively cancel the weight savings due to improved performance. This effect is shown in Figure A-30; the potential propellant weight savings of 110 lb associated with an Isp increase of 8 sec would be reduced to a net system weight savings of 20 lb by including the high performance nozzle weight penalties. The savings could not compensate for the increased installation difficulty due to larger nozzles.

ANGULATION AND LINEAR COMPENSATOR

DESIGN CRITERIA

DESIGN PRESS	500 PSIG
TEMP. RANGE	720°R
MATL.	2219 AL. ALY.
ANGULATION	±5° MIN. EA. JOINT.



INNER LINE DIA.	COMPENSATOR TOTAL WEIGHT
.5	3.7 LBS
1.0	3.8 LBS
1.5	4.0 LBS

Figure A-24

H_2 LINE WEIGHT SUMMARY

	LENGTH	WEIGHT	VOLUME
FWD. MANIFOLD	34 FT.	18 LB.	0.39 FT^3
COMPENSATORS		26 LB.	
H_2 TANK TO FWD. MANIFOLD	52 FT.	28 LB.	0.59 FT^3
COMPENSATOR		9 LB.	
H_2 TANK TO REAR MANIFOLD	68 FT.	37 LB.	0.77 FT^3
COMPENSATOR		8 LB.	
REAR MANIFOLD	80 FT.	43 LB.	0.90 FT^3
COMPENSATORS		65 LB.	
VEHICLE TO WING TIP			
RIGHT	41 FT.	22 LB.	0.44 FT^3
LEFT	41 FT.	22 LB.	0.44 FT^3
COMPENSATORS		15 LB.	
MANIFOLD TO ENGINES			
12 LINES	1-2 FT.	14 LB.	0.24 FT^3
9 LINES	2-3 FT.	15 LB.	0.30 FT^3
8 LINES	4-5 FT.	22 LB.	0.45 FT^3
COMPENSATORS		26 LB.	
TOTAL		370 LB.	4.52 FT^3

Figure A-25

VACUUM JACKETED LINE ASSEMBLY

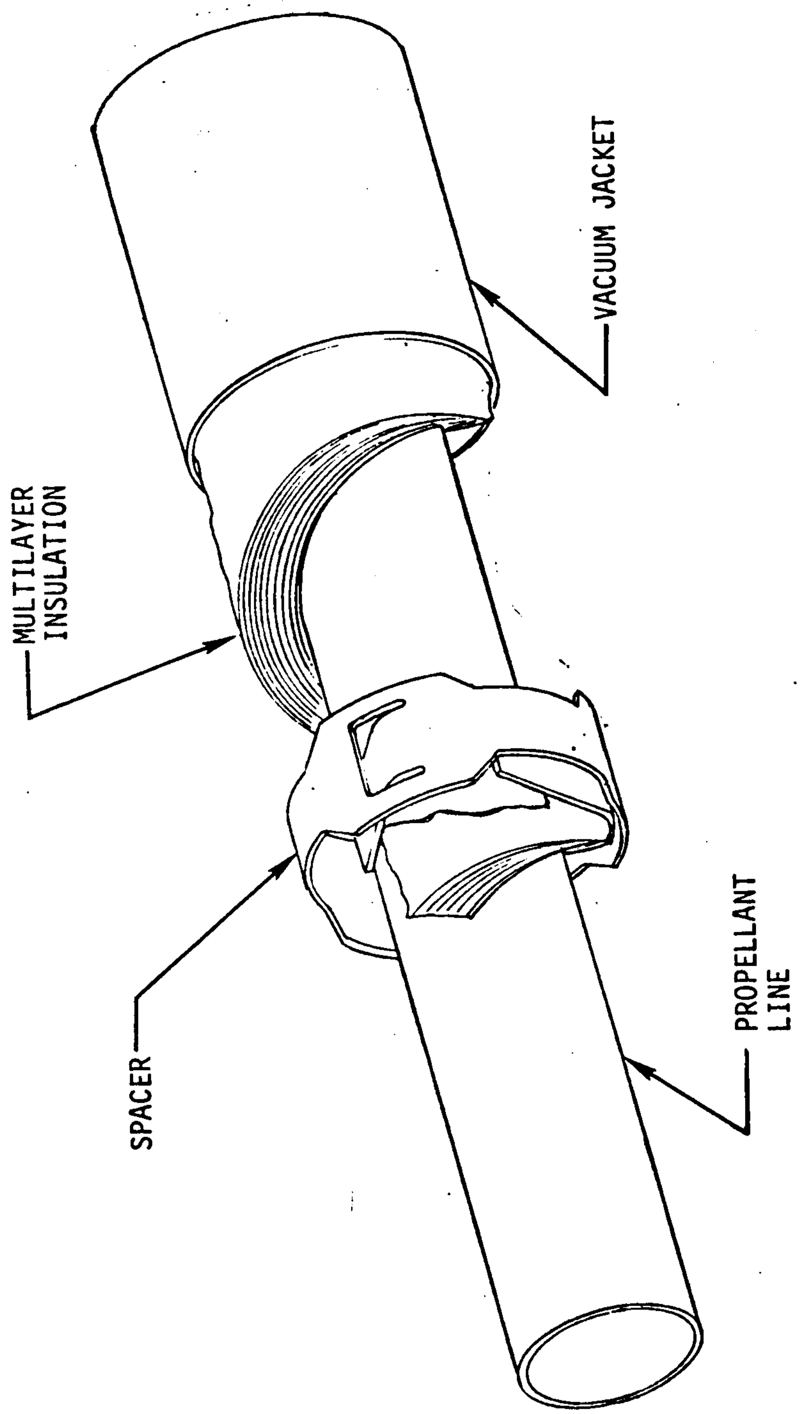
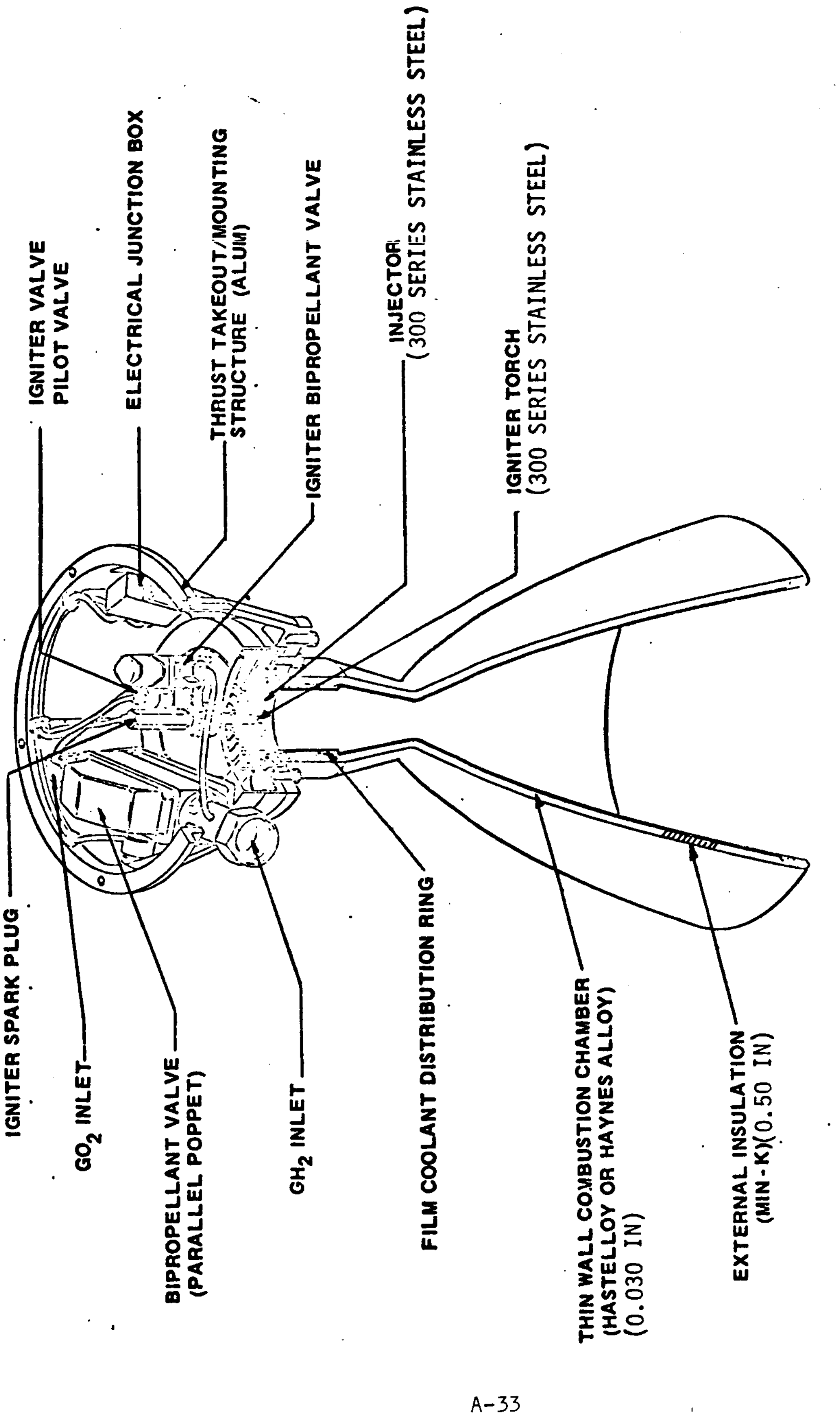


Figure A-26



HIGH PRESSURE APS THRUSTER DUCTED FILM COOLED

Figure A-27

AEROJET LIQUID O_2/H_2 THRUSTER WEIGHT MODEL

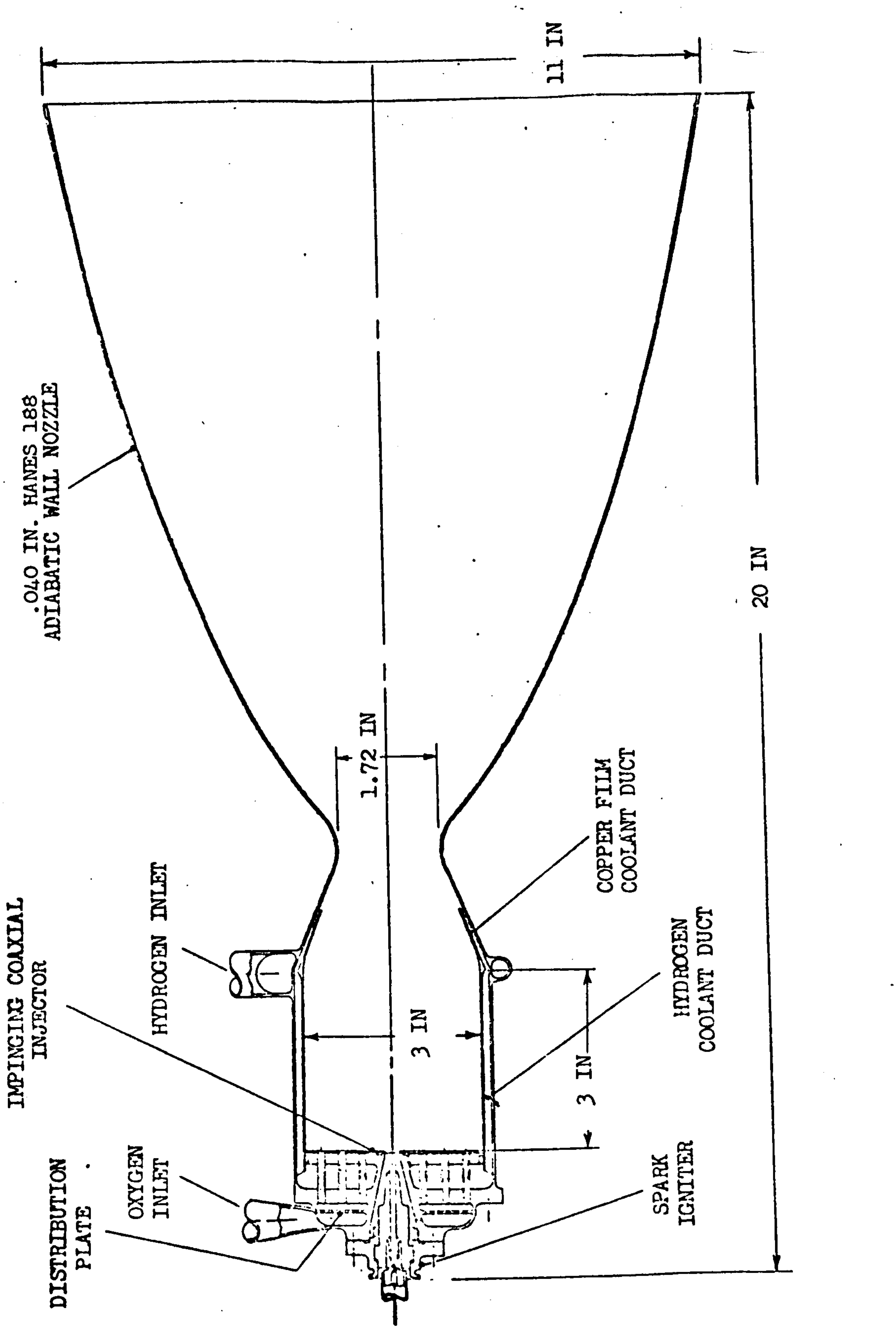


Figure A-28

LH_2/LO_2 ENGINE PERFORMANCE ESTIMATE

(AEROJET LIQUID ROCKET COMPANY)

THRUST	1150 LB
CHAMBER PRESSURE	300 PSIA
MIXTURE RATIO	4.0
AREA RATIO	40
OXYGEN TEMPERATURE	180°R
HYDROGEN TEMPERATURE	40°R
% FUEL FILM COOLING	16%
THEORETICAL VACUUM SPECIFIC IMPULSE	451.5 SEC
KINETICS LOSS	2.5 SEC
MIXTURE RATIO MALDISTRIBUTION	0.6 SEC
BOUNDARY LAYER LOSS	9.1 SEC
DIVERGENCE LOSS	9.5 SEC
FUEL FILM COOLING LOSS	5.8 SEC
ENERGY RELEASE LOSS	6.5 SEC
DELIVERED VACUUM SPECIFIC IMPULSE	417.5 SEC

Figure A-29

LO_2/LH_2 ENGINE WEIGHT AND PERFORMANCE SENSITIVITY

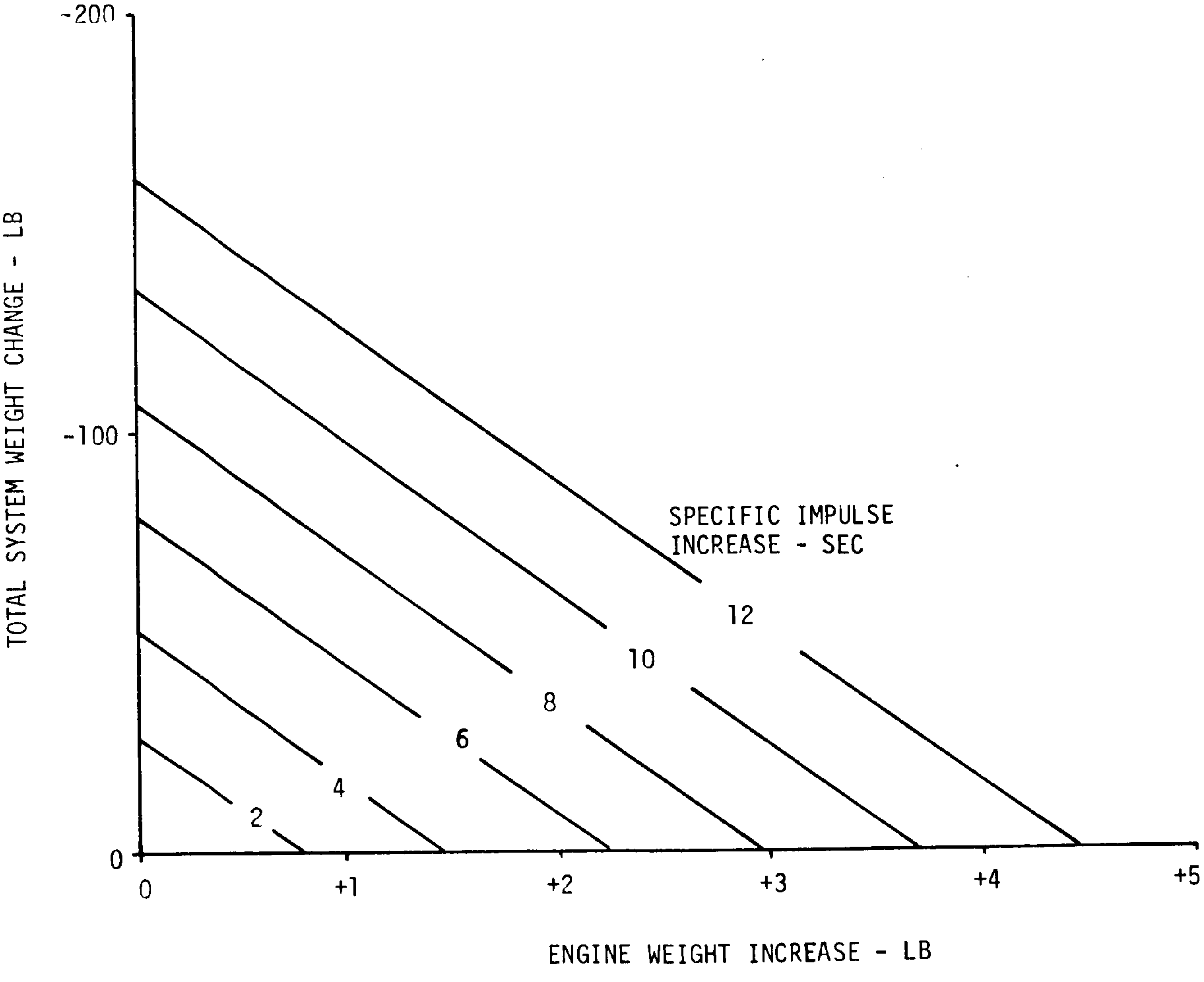


Figure A-30

APPENDIX B

THERMAL MODEL

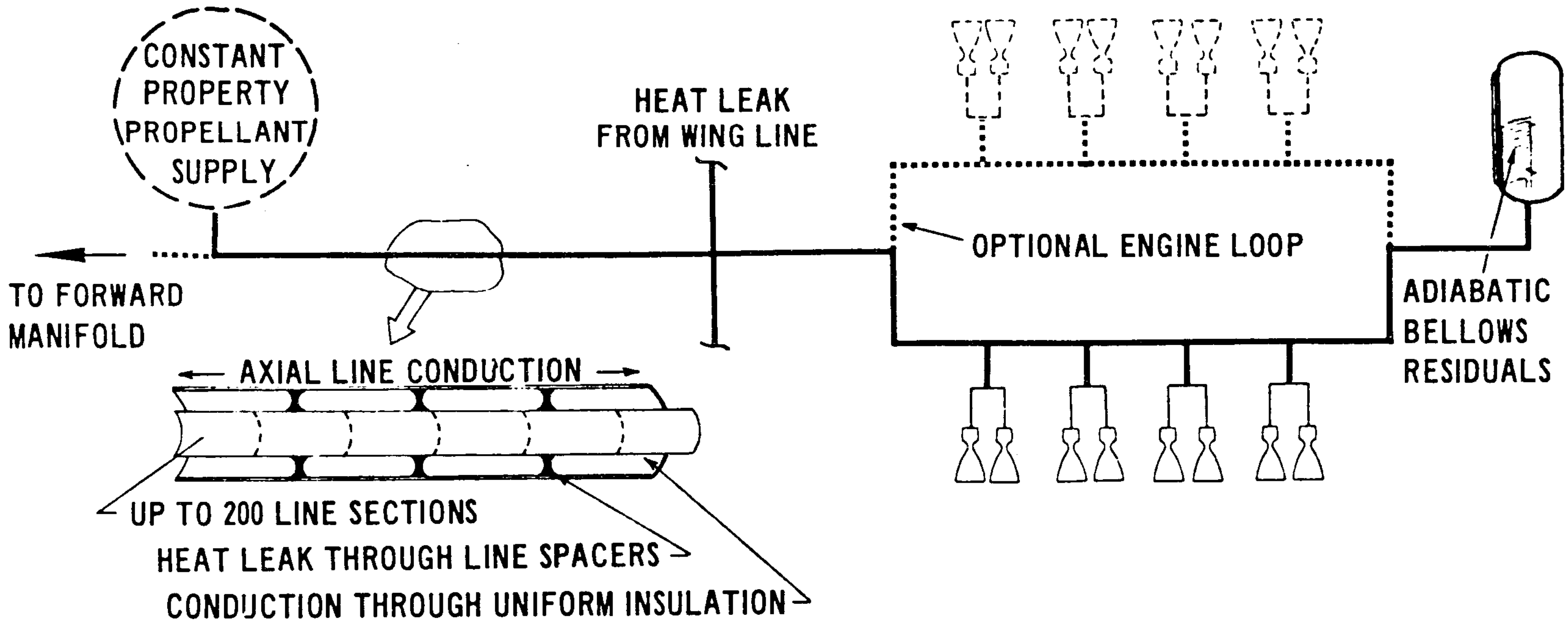
B1 Introduction - One of the major concerns associated with the use of liquid cryogenics is the temperature/density variation at the engine inlet resulting from heat input to the propellant during distribution. This appendix describes a thermal model developed to calculate transient thermal behavior within the liquid distribution system. The model incorporates heat conduction through the insulation, engine heat soakback, and major heat leaks associated with structural members such as line supports, inner line spacers, and manifold junctions. The model, which is described below, was used to verify that liquid cryogenics can be successfully distributed in the APS. A description and summary of the pertinent system thermal characteristics is given in paragraph 3.2.2 and the governing equations and assumptions used in the analyses are presented in this appendix.

B2 Thermal Model Design Criteria - The model was established for a system using vacuum jacketed feed lines with multilayer insulation (MLI) between the inner line and outer jacket. The model was tailored to the APS installation in the vehicle, consisting of a primary feed line located along the length of the vehicle, manifolds at the forward and aft end of the vehicle, lines from the main feed line to each wing tip,and engine feed lines from the manifolds to each engine. A schematic of the model is shown in Figure B-1. The lines are divided into small segments and the mass and energy equations are numerically integrated with time for each individual segment. Conduction through the insulation, conduction along the feed lines,and heat shorts to the inner line were accommodated. Fluid thermal expansion is accommodated by allowing propellant movement at constant pressure from the propellant tank to the accumulator. At the end of each time interval, fluid flow effects,both from thermal expansion and from engine firings, are calculated and used to update the fluid properties in each segment.

Feed line thermal conductivity and specific heat are evaluated at each time step for each segment. These data vary greatly with temperature and to maintain accuracy, real line material properties, shown in Figure B-2, were incorporated in the model. Real fluid properties were also used to account for compressibility effects.

B3 Thermal Analyses - The fluid composition within each segment was assumed homogeneous; enthalpy entering a segment through mass transfer was assumed to be distributed uniformly and new values of integrated properties were calculated at

FEED LINE THERMAL MODEL



THERMAL MODEL

- ISOBARIC HEATING
- SOLUTION OF MASS AND ENERGY CONSERVATION EQUATIONS FOR EACH LINE SECTION USING IMPLICIT DIFFERENCE TECHNIQUE
- REAL FLUID AND LINE PROPERTIES
- HEATING
  - a) UNIFORM INSULATION
  - b) LINE AXIAL CONDUCTION
  - c) LOCALIZED HEAT LEAKS

ENGINE FIRING CAPABILITY

- RESUPPLY FROM PROPELLANT TANK OR BELLOWS TANK IS AVAILABLE
- SINGLE OR MULTIPLE FIRING ON EITHER OR BOTH SIDES OF LOOP
- LINE-VAPOR THERMAL EQUALIZATION AFTER FIRING

Figure B-1

LINE THERMAL PROPERTIES

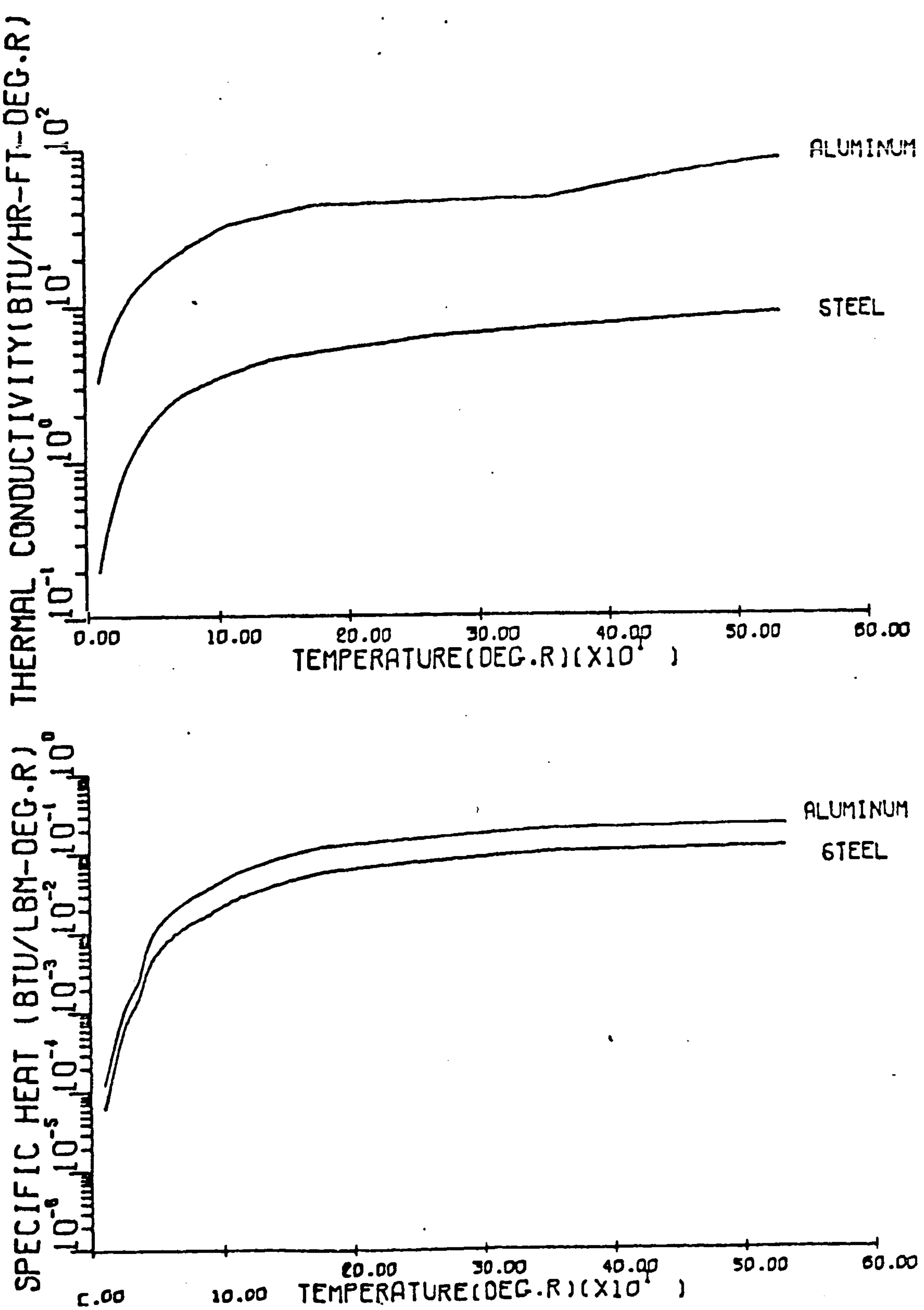


Figure B-2

each time increment. Thus, analysis accuracy was dependent on the number of line segements and the computational time increment. These were independently selected and up to 200 line segments could be accommodated.

Heat transfer occurs because of temperature differences through the line insulation, conduction from adjacent segments and conduction through thermal shorts. The governing equations for the j-th segment are: (See Figure B-3 for Symbol Definition):

ENERGY BALANCE:

$$dhj = \frac{M_{in_j}\ (h_{in_j} - h_j) + Q_j - \rho wCwVwdTw}{\rho jVj} \qquad (1)$$

MASS BALANCE:

$$d\rho_j = (M_{in_j} - M_{out_j})/V_j \qquad (2)$$

EQUATIONS OF STATE:

$$T_j = T_j\ (h_j, p) \qquad (3)$$

$$\rho_j = \rho_j\ (h_j, p) \qquad (4)$$

These equations were solved for the 4 unknowns, h_j, ρ_j, T_j, and M_{out_j}.

For the solution, an implicit energy equation was used to increase the maximum allowable step size. The implicit expression for the energy equation is:

$$dh_j = \emptyset *W_{in} * (h_{in} - h_j) + (1 - \emptyset) *W'_{in} *(h'_{in} - h'_j)$$

$$+ \dot{Q}_j * \Delta t - \rho wVwCw_j *dTw_j / \rho_j V_j \qquad (5)$$

where primed quantities represent values being calculated during the current time-step and unprimed variables represent either calculations of the previous timestep or contributions,such as the heat flux,which are based on gradients existing at the previous time step.

The heating rate ($\dot{Q}_j$) of each segment was calculated in three parts: heat through the multilayer insulation, heat conducted from the line, and direct heat shorts. The model assumed that the line was insulated uniformly with an insulation of sufficiently low thermal conductivity that the outer surface remained at the temperature of the ambient surroundings. The insulation thickness and an effective thermal conductivity were input and held constant throughout the calculation.

The heat leak through the insulation of each segment was evaluated as:

NOMENCLATURE

h	Fluid enthalpy (Btu/lbm)
M	Fluid mass entering or leaving section (lbm)
ρ	Fluid density (lbm/ft^3)
C	Line specific heat (Btu/lbm-°R)
V	Volume (of fluid or line section) (ft^3)
T	Temperature (°R)
P	Pressure ($lbf/in.^2$)
$\dot{Q}$	Heat transfer rate into fluid-line section (Btu/ lbm)
dt	Time increment (hr)
dT	Temperature increment (°R)
K	Thermal conductivity (Btu/hr-ft-°R)
O.D.	Outer Diameter of line
QPIPE	Thermal Conductivity of line
APIPE	Cross-section area of line (not flow area)
DELTAL	Section length (ft)
INSUL	Insulation Thickness

SUBSCRIPTS

j	Identified with Section j
in	Associated with mass coming into section
out	Associated with mass leaving section
w	Line wall property
amb	Ambient surrounding conditions

Figure B-3

$$\dot{Q}_{INSUL_j} = \frac{2\ K_{INSUL}\ (T_{AMB} - T_j)\ \Delta L}{(\ln\ (1+ \frac{2*\Delta INSUL}{O.D.}\ )}$$

Heat leaks due to structural supports, interline spacers, or other lines were included by identifying the location of the heat leak and the thermal resistance between the line and a thermal reservoir with a speicified heat capacity and initial temperature. More than one heat leak could be associated with a given segment.

Heat leaks at engine locations can employ a time dependent heat source temperature option to evaluate engine heat soakback effects. For this option, the temperature history of a line connecting an engine to the line manifold was approximated with a quadratic curve fit:

$$TDR\ (t_{last\ firing}) = AMINI\ (T_{amb},\ A_0+A_1*(t-t_{last\ firing}) + A_2*(t-t_{last\ firing})^2)$$

Coefficients were selected to approximate thruster valve temperature vs time with a thruster thermal standoff as described in Paragraph 3.2.2. Conduction between line segments was approximated using a two point gradient, viz:

$$\dot{Q}COND_j = QPIPE(T_j)*\ APIPE*\ (T_{j+1}-2*T_j+T_{j-1})/DELTAL$$

where QPIPE is the line thermal conductivity at temperature T_j, APIPE is the cross-section area of the line wall,and DELTAL is the section length.

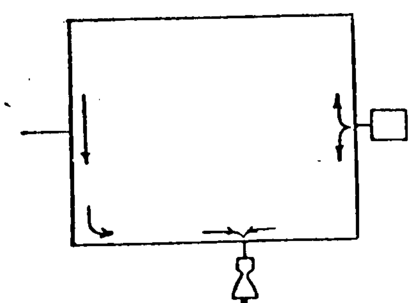
B4 <u>Fluid Transfer Options</u> - Fluid movement results from either thermal expansion or engine firings. Localized heat shorts lead to regions of warmer fluid along the line with a resultant local fluid expansion. In fact, the first heat short causes propellant heating and thermal expansion which pushes propellant downstream. This moves fluid away from other thermal shorts and tends to limit the amount of heat input to an incremental propellant volume. Similarly, engine usage shifts the propellant in the line, introducing fresh,cooler propellant which serves to chill and cool the line. To accommodate these effects, the contents of each section are shifted as required to satisfy propellant flow. Calculations are begun at the end of the line connected to the storage tank, where the mass flow into the first segment due to thermal expansion is necessarily zero. The calculated mass ejected from Section j, M_{out_j} obtained from the solution of equations (2) - (5) is then used as the known value of M_{in} for the $_{j+1}$ segment. This calculation is performed successively for each segment and provides instantaneous propellant properties along the entire line. For the last segment, M_{out} is the mass transferred by thermal expansion, which is not used in an engine. This propellant is displaced into the bellows reservoir or could be vented overboard. An energy and mass balance is maintained in the bellows reservoir by accumulating all of the mass exhausted by thermal expansion from the last section of the line and summing the total enthalpy of this mass. The instantaneous bellows-reservoir properties are then obtained using real fluid properties.

The fluid transfer associated with some of the available engine firing options are shown in Figure B-4. Fluid movement, when firing occurs at a single location, is shown in Figures B-4a and B-4b for the two options of resupply from the bellows reservoir and for the main propellant supply.

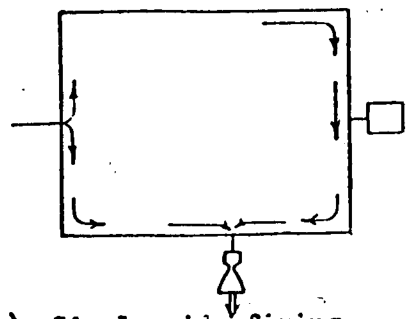
The flow in each side of the loop is inversely proportional to the loop length from the resupply to the firing location. Thus, if firing always occurred at the same individual site, fluid would eventually be cleared in the entire loop. When two engines or more are fired at a separated location on one side of the loop as shown by the solid engine diagrams in Figure B-4c and B-4d, the fluid transfer is approximated by assuming that the fluid leaves the line via the engine most remote from the resupply source. In most instances this approximation would not be expected to generate significant errors.

When engines on both sides of the loop are fired simultaneously, resupply is approximated by assuming that flow moves only between the resupply and the firing engines as shown in Figures B-4e and B-4f. No fluid motion at the opposite end of the loop is permitted.

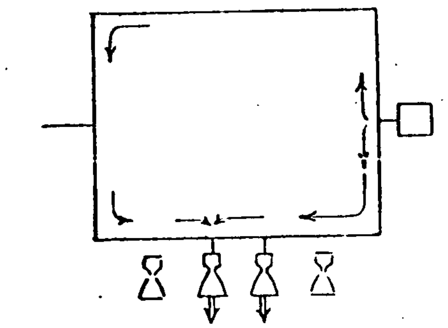
LINE FLUID TRANSFER FIRING OPTIONS



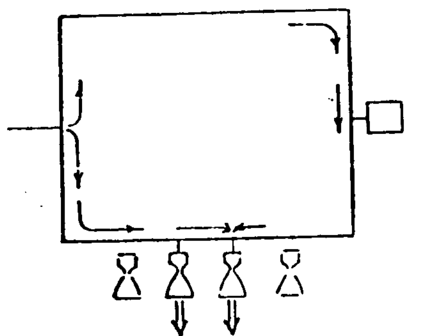
a) Single side firing
Resupply from bellows reservoir



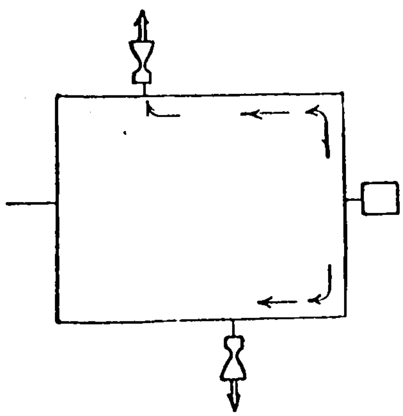
b) Single side firing
Resupply from propellant supply



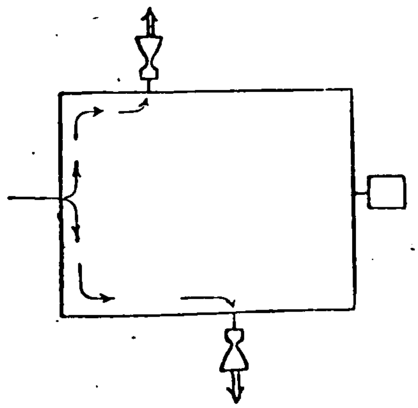
c) Multiple firing on side
Resupply from bellows reservoir



d) Multiple firing on side
Resupply from propellant supply



e) Both sides firing
Resupply from bellows reservoir



f) Both sides firing
Resupply from propellant supply

Figure B-4

APPENDIX C

Gaseous System Trades

C1 Introduction - The gaseous O_2/H_2 system data discussed in Paragraph 3.1 was based on detailed system design and sizing studies. The purpose of this appendix is to identify those trades performed and to show system weight sensitivity to chamber pressure, mixture ratio, etc. The concepts considered are shown in Figure C-1. The simplest system is a fully pressurized concept without pumps. This will be shown to result in large tankage and pressurization weights. Tankage and pressurization weights are reduced by using either electric or hydraulic powered pumps or by using a mechanical bellows pump concept. In addition, since the oxygen pressurization requirements are much less severe than those for hydrogen, a hybrid system using fully pressurized oxygen and pumped hydrogen is attractive. The effect of chamber pressure and mixture ratio on system weight was evaluated for each concept to define minimum weight designs. In addition, gaseous accumulator operating pressures and volumes were designed to yield minimum weight for each concept. The results of these analyses are described below.

C2 Reference Turbopump Concept - A reference system for comparison, was taken from the turbopump data reported in Reference (h). This system is a parallel gas generator concept, weighing 10,600 lb at a chamber pressure of 300 psia, as shown in Figure C-2. The gaseous accumulators were sized such that only 40 pump cycles would be required for pulsing RCS usage. These, together with 10 steady-state maneuver burns, result in 5000 total cycles for 100 missions. This was the estimated bearing life limit for rapid start turbopumps (0.5 sec). As shown in Figure C-2, the accumulators are heavy, 810 lb, and their weight would increase greatly for fewer pump cycles (larger accumulators).

C3 Fully Pressurized RCS - The fully pressurized system, in which the turbopump is removed, is shown in Figure C-3. The system is much heavier than that of the parallel GG system, weighing approximately 15,500 lb. However, without a turbopump, the system need not be constrained to 40 cycles and the system can be designed on the basis of heat exchanger life capability. Current NASA technology programs indicate that the heat exchangers are capable of 200 cycles per mission. Resizing the accumulators to 200 cycles reduces system weight to approximately 15,000 pounds.

C4 Motor Operated Pumps - Fully pressurized system weights can be greatly reduced by using an electric motor operated pump to reduce tankage and pressurization

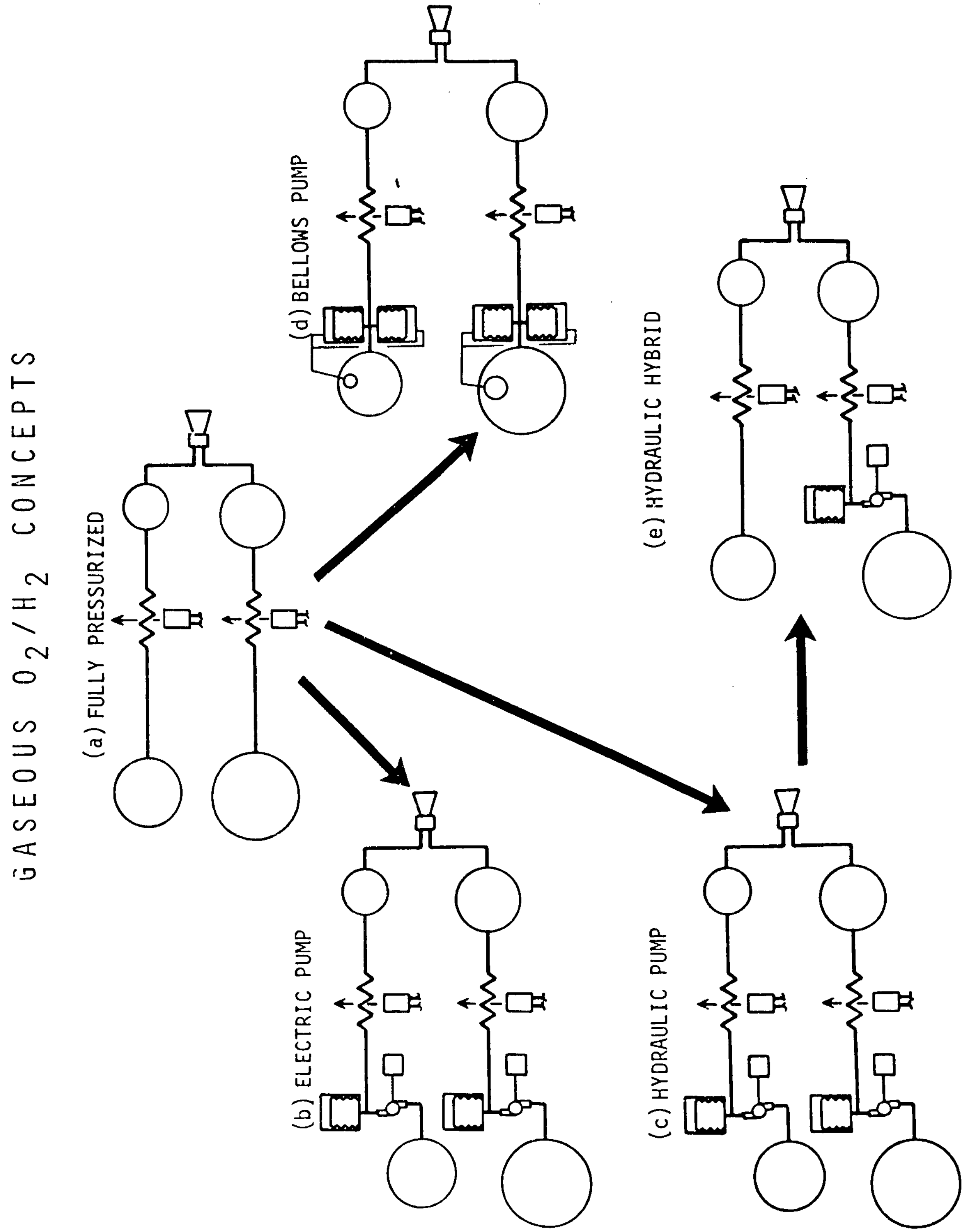

GASEOUS O_2/H_2 CONCEPTS
(a) FULLY PRESSURIZED
(b) ELECTRIC PUMP
(c) HYDRAULIC PUMP
(d) BELLOWS PUMP
(e) HYDRAULIC HYBRID

Figure C-1

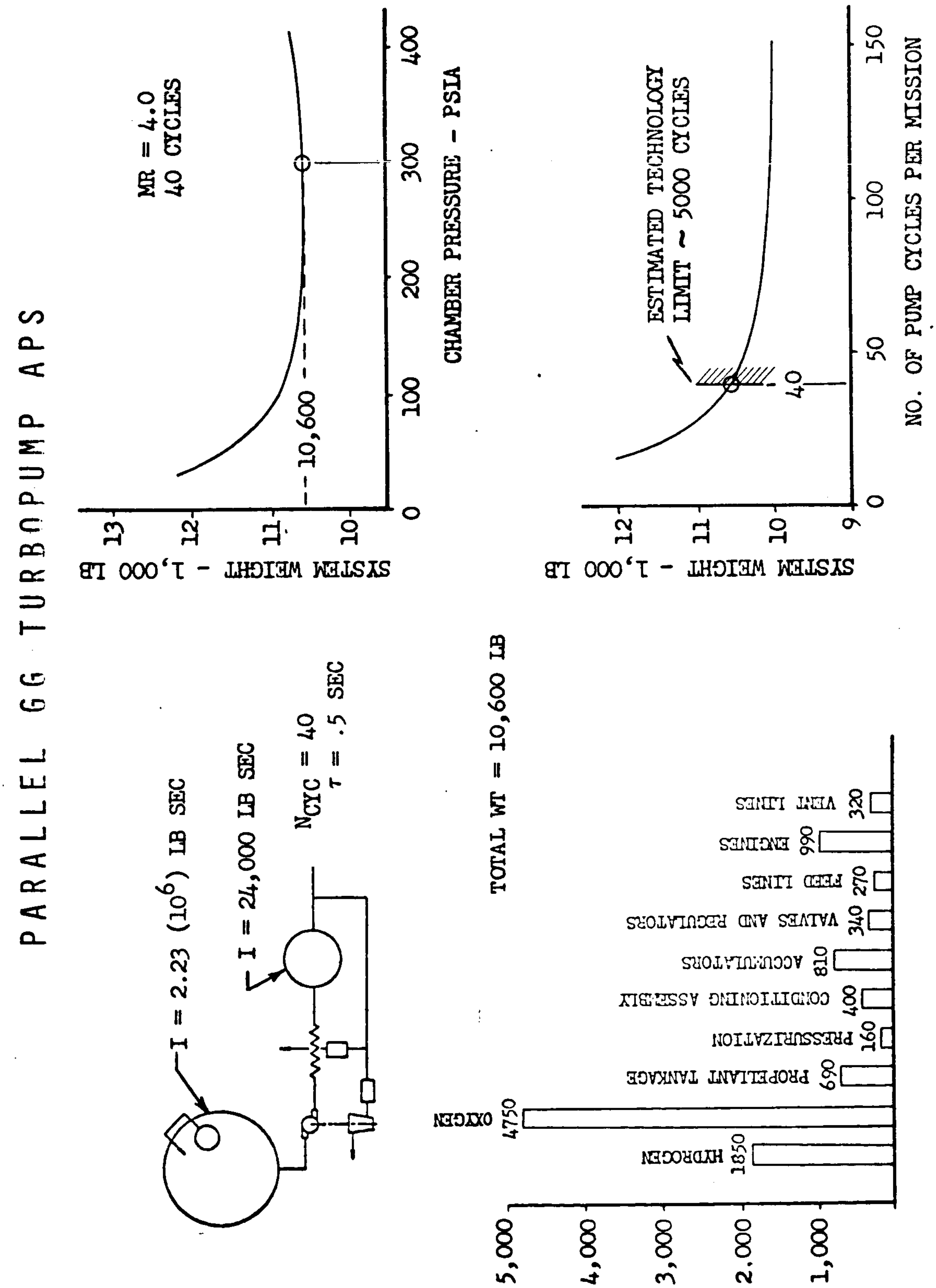

PARALLEL GG TURBOPUMP APS
MR = 4.0
40 CYCLES
SYSTEM WEIGHT - 1,000 LB
13
12
11
10
10,600
0
100
200
300
400
CHAMBER PRESSURE - PSIA
ESTIMATED TECHNOLOGY LIMIT ~ 5000 CYCLES
40
SYSTEM WEIGHT - 1,000 LB
12
11
10
9
0
50
100
150
NO. OF PUMP CYCLES PER MISSION
I = 2.23 (10^6) LB SEC
I = 24,000 LB SEC
N_CYC = 40
τ = .5 SEC
TOTAL WT = 10,600 LB
5,000
4,000
3,000
2,000
1,000
1850 HYDROGEN
4750 OXYGEN
690 PROPELLANT TANKAGE
160 PRESSURIZATION
400 CONDITIONING ASSEMBLY
810 ACCUMULATORS
340 VALVES AND REGULATORS
270 FEED LINES
990 ENGINES
320 VENT LINES

Figure C-2

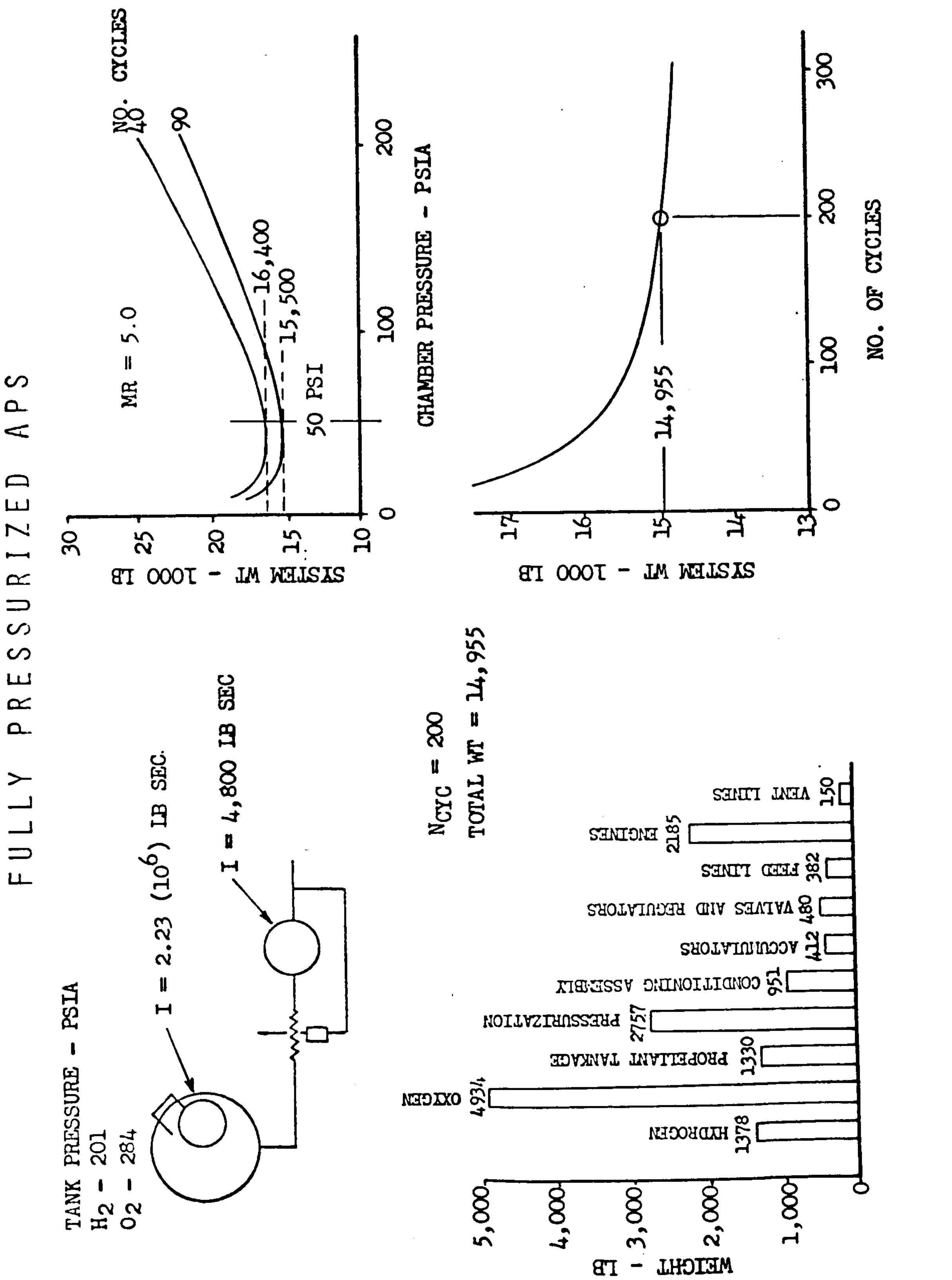

FULLY PRESSURIZED APS
TANK PRESSURE - PSIA
H_2 - 201
O_2 - 284
I = 2.23 (10^6) LB SEC.
I = 4,800 LB SEC
N_{CYC} = 200
TOTAL WT = 14,955
WEIGHT - LB
5,000
4,000
3,000
2,000
1,000
0
1378 HYDROGEN
4934 OXYGEN
1330 PROPELLANT TANKAGE
2757 PRESSURIZATION
951 CONDITIONING ASSEMBLY
412 ACCUMULATORS
480 VALVES AND REGULATORS
382 FEED LINES
2185 ENGINES
150 VENT LINES
MR = 5.0
NO. CYCLES
40
90
16,400
15,500
50 PSI
SYSTEM WT - 1000 LB
30
25
20
15
10
0
100
200
CHAMBER PRESSURE - PSIA
14,955
SYSTEM WT - 1000 LB
17
16
15
14
13
0
100
200
300
NO. OF CYCLES

Figure C-3

system weights. As shown in Figure C-4, the electric motor system weighs 11,183 lb at a chamber pressure of 300 psia. This system could be extended to 200 cycles by resizing the gaseous accumulators. The resultant system weight would be decreased to 10,400 lb which would be competitive with the turbopump APS. Operating with a liquid accumulator,as shown in Figure C-5, increases the tankage weight penalty, and minimum system weight results at a lower chamber pressure (150 psia). Pump flow requirements are reduced but system weight is increased approximately 100 lb with an accumulator. However, with the accumulator considerable design flexibility is provided as a wide range of pump design conditions can be accommodated for a small weight penalty.

The electric motor system described above includes weight penalties for APU resizing since the alternator requirements greatly exceed the APU capability(20 hp). Hydraulic motor concepts were evaluated since the power requirements more nearly match the APU hydraulic power capability(275 hp). The hydraulic concept, shown in Figure C-6, provides minimum weight at approximately 300 psia chamber pressure, however, the system was constrained to 200 psia to remain within the APU hydraulic power limit of 275 hp. At the design point, the hydrogen pump requires 227 hp, and the oxygen pump requires 48 hp. At a design life of 200 cycles, the hydraulic driven pump system would weigh 10,080 lb or 520 lb less than the parallel gas generator turbopump concept. Operation with a liquid accumulator, shown in Figure C-7,would increase system weight, as was the case with the electric motor concept. However, the weight increase is minimized by using bellows accumulators. Also, a higher chamber pressure can be accommodated with a liquid accumulator (due to pump flow rate reductions) without exceeding APU horsepower limits. Again design flexibility is available with the hydraulic powered system to accommodate changes in propellant pump characteristics or system pressure schedules.

A summary of electric and hydraulic system weights is given in Figure C-8. A single design point reflecting no liquid accumulators, 40 gaseous accumulator cycles, and a chamber pressure of 200 psia was selected for comparison. System weights using the electric motor driven pump are slightly higher than with hydraulic driven pumps (450 lb) due to alternator resizing. Increased APU power capability was not required since the APU furnishes 275 hydraulic hp.

C5 Bellows Pump Concept - A purely mechanical approach which avoids the APU interface is shown in Figure C-9. With this concept, two bellows tanks are alternately pressurized to furnish the high pressures required for system operation. As propellant is removed from one tank, the other tank is replenished. After one

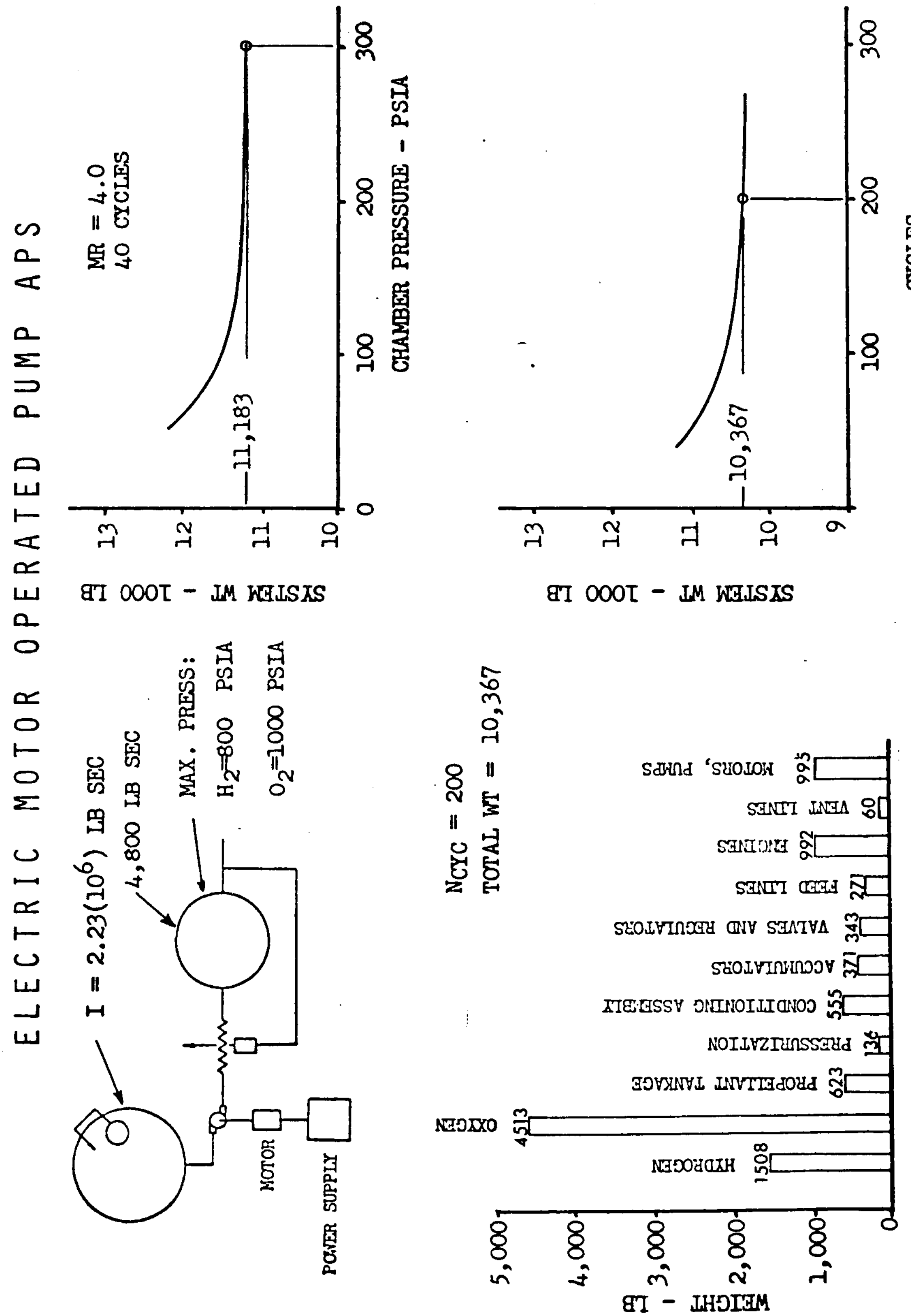

ELECTRIC MOTOR OPERATED PUMP APS
I = 2.23(10^6) LB SEC
4,800 LB SEC
MAX. PRESS:
H_2=800 PSIA
O_2=1000 PSIA
MOTOR
POWER SUPPLY
MR = 4.0
40 CYCLES
11,183
SYSTEM WT - 1000 LB
CHAMBER PRESSURE - PSIA
10,367
CYCLES
N_{CYC} = 200
TOTAL WT = 10,367
WEIGHT - LB
HYDROGEN 1508
OXYGEN 4513
PROPELLANT TANKAGE 623
PRESSURIZATION 136
CONDITIONING ASSEMBLY 555
ACCUMULATORS 371
VALVES AND REGULATORS 343
FEED LINES 271
ENGINES 992
VENT LINES 60
MOTORS, PUMPS 995

Figure C-4

ELECTRIC MOTOR OPERATED PUMP APS WITH LIQUID ACCUMULATORS

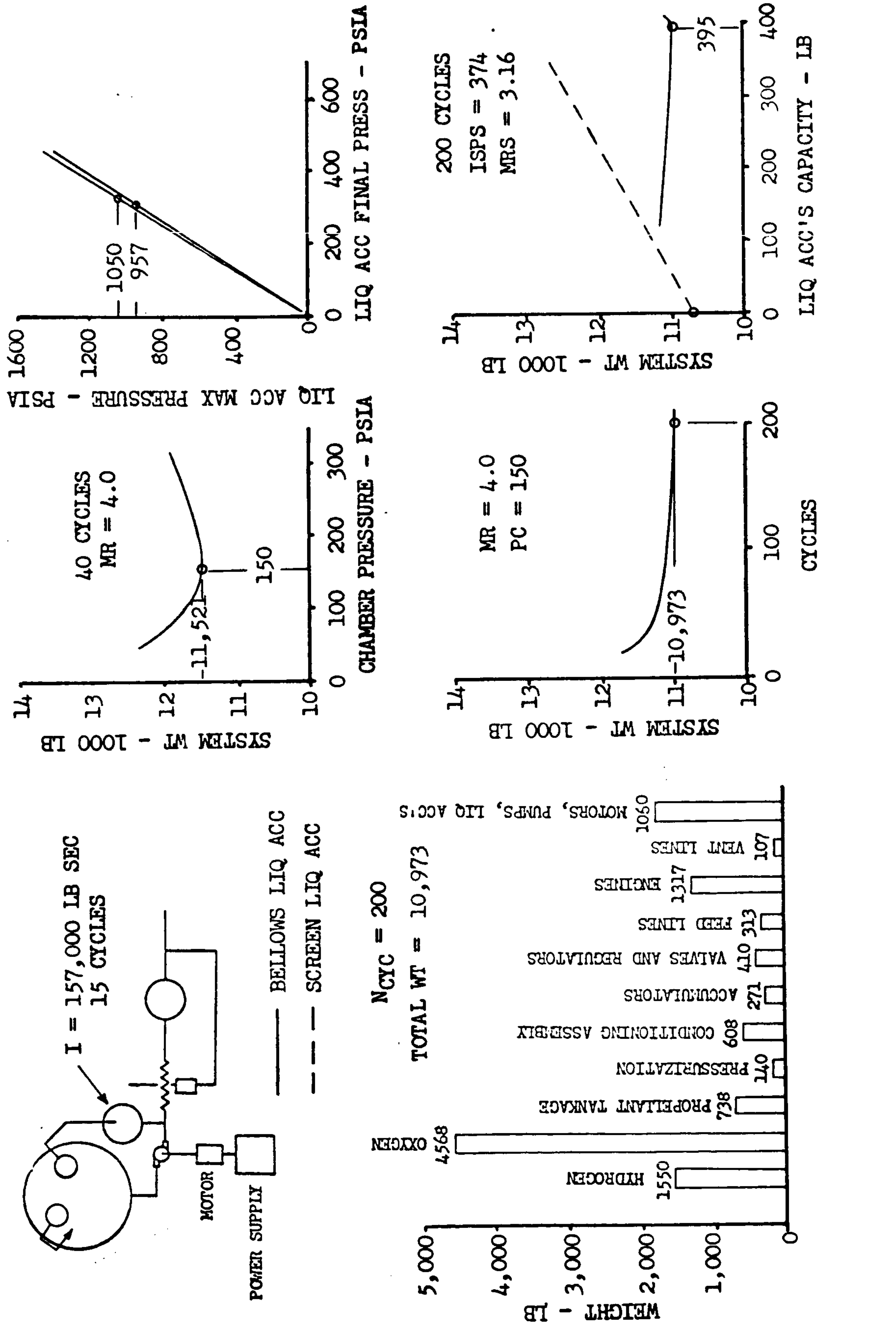


Figure C-5

HYDRAULIC MOTOR CONCEPT

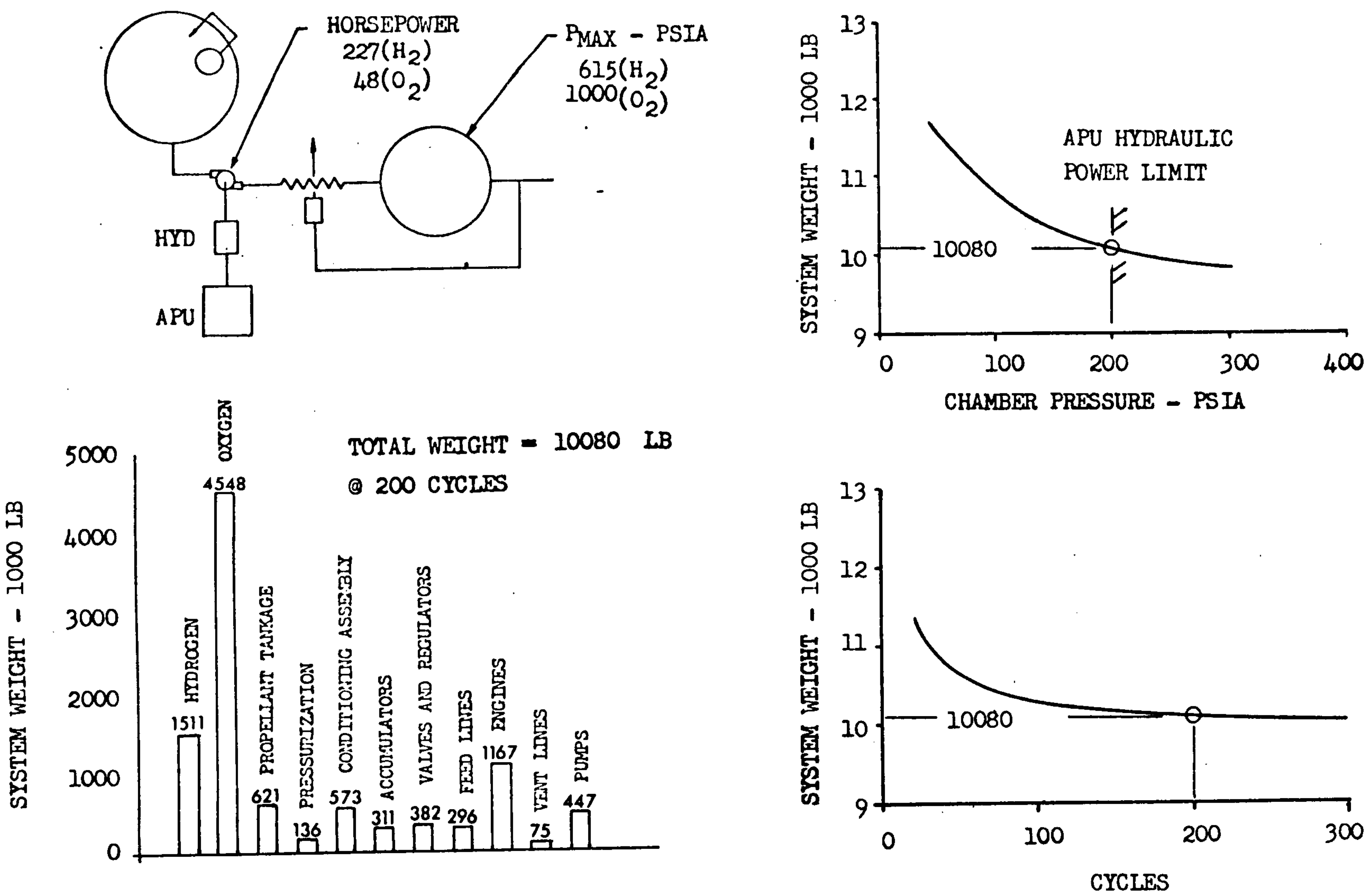


Figure C-6

HYDRAULIC MOTOR/PUMP APS WITH LIQUID ACCUMULATORS

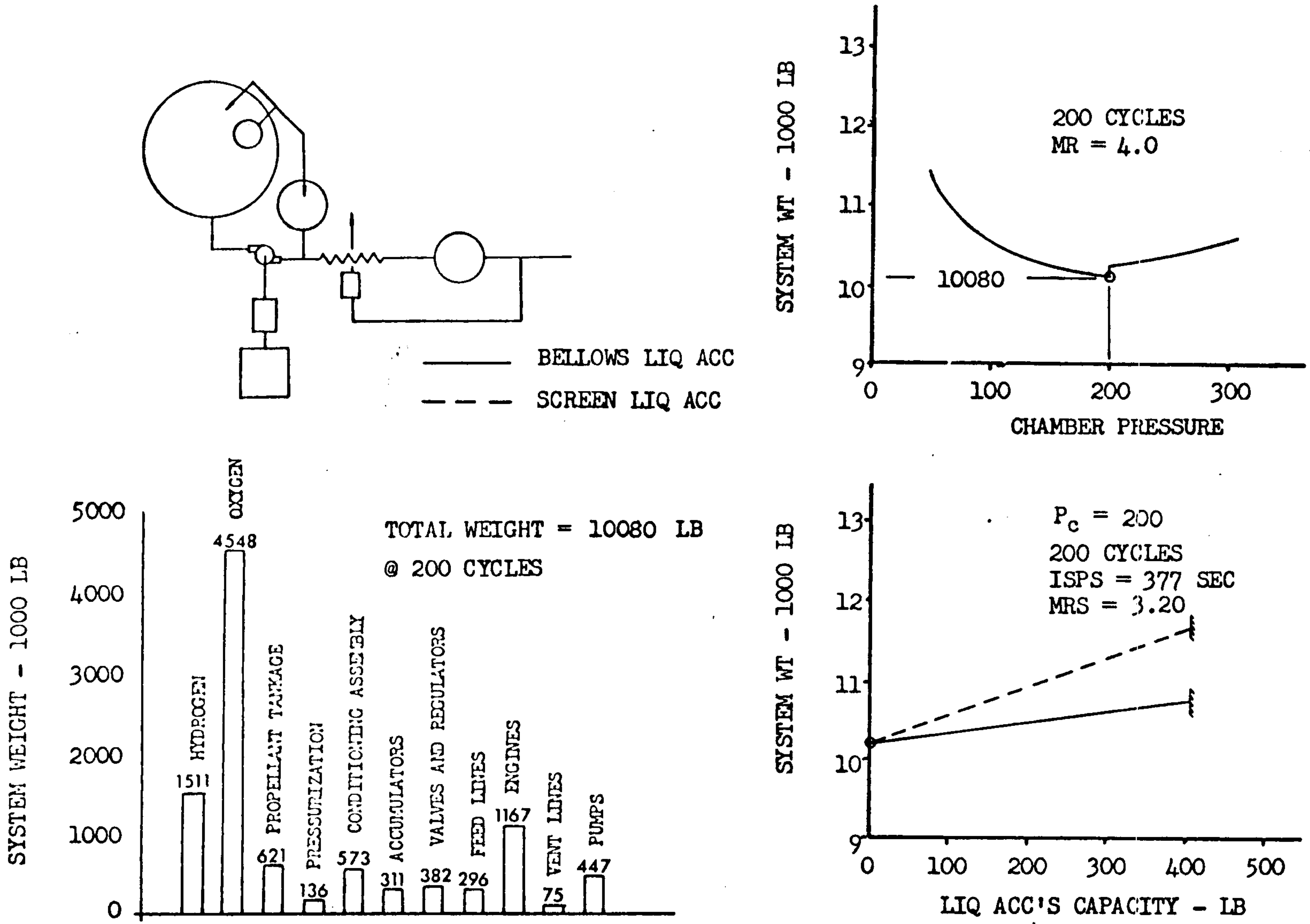


Figure C-7

WEIGHT COMPARISON OF ELECTRIC AND HYDRAULIC MOTOR CONCEPTS

NO LIQUID ACCUMULATORS
40 PUMP AND GAS ACCUMULATOR CYCLES

	ELECTRIC MOTOR CONCEPT	HYDRAULIC MOTOR CONCEPT
CHAMBER PRESSURE	200 PSI	200 PSI
PUMP PRESSURE (H_2)	615 PSI	615 PSI
(O_2)	1000 PSI	1000 PSI
HORSEPOWER (H_2)	227 HP	227 HP
(O_2)	48 HP	48 HP
SYSTEM WT LESS PUMPS, MOTORS	10393	10393
MOTOR AND PUMP WEIGHTS	358	280
ADDITIONAL ALTERNATOR WT	375	—
APU PROPELLANT	90	90
TOTAL SYSTEM WEIGHT	11216	10763
WEIGHT DIFFERENCE	453	—

Figure C-8

BELLOWS PUMP/COLD HELIUM PRESSURIZED

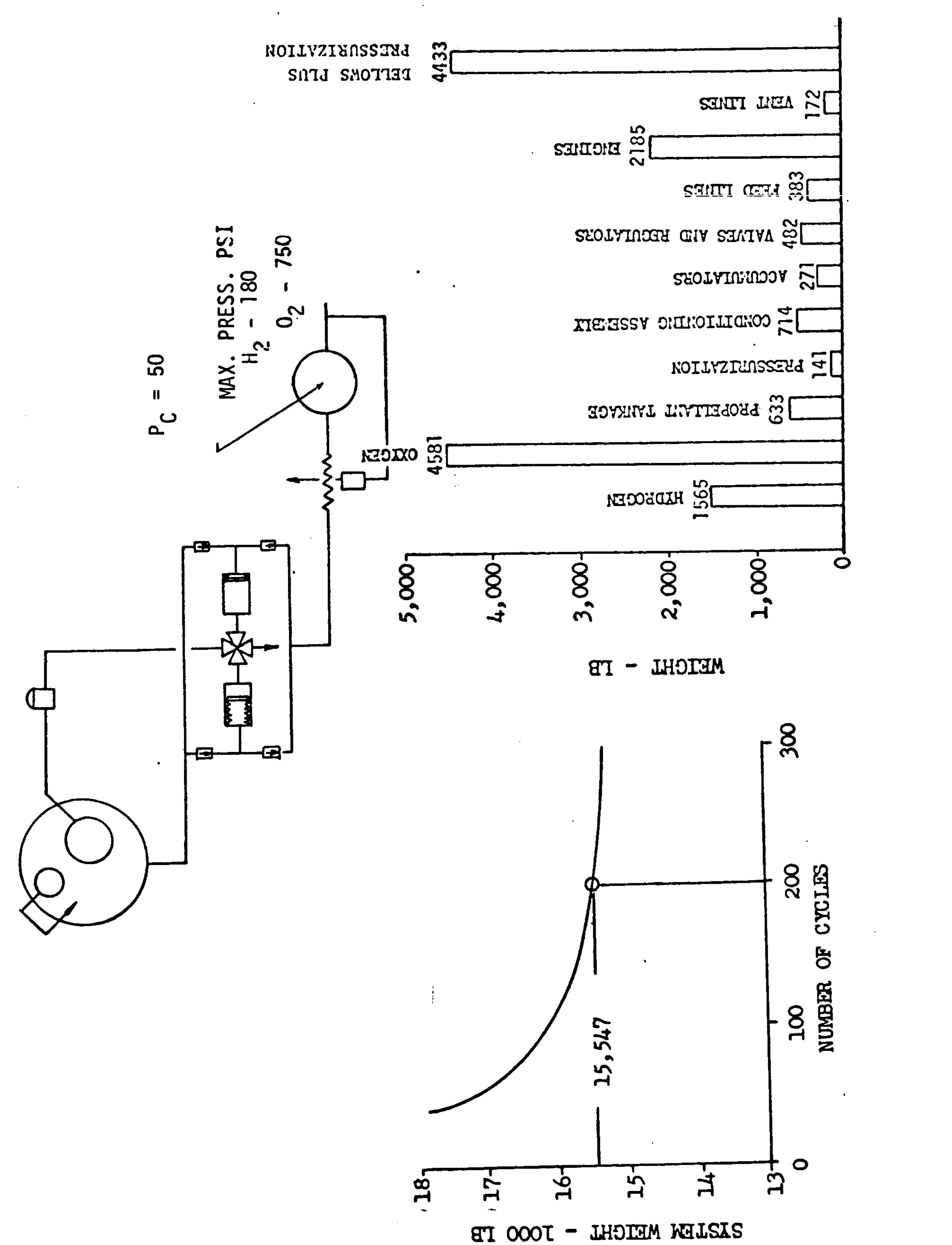


Figure C-9

tank is expended, propellant is used from the other full tank and the cycle is repeated. If cold helium is used as a pressurant the system is heavy. Venting the pressurant on each bellows cycle results in pressurant mass requirements the same as those for a fully pressurized system. System weight exceeds 15,000 lb. However, if warm helium is used as the pressurant the system, shown in Figure C-10, weighs 11,255 lb at a chamber pressure of 150 psia. The concept was based on the use of the primary heat exchanger to provide 500°R helium gas. Propellant heating from the warm gas would not significantly affect system operation, since the propellants are vaporized in heat exchangers immediately downstream of the pumps. Heated hydrogen gas was also investigated for the hydrogen bellows pump as shown in Figure C-11, but the system weight change was less than 25 lb, primarily due to the heavy, high pressure hydrogen supply tank required. The use of pressurized bellows pumps would result in a weight increase of 600 lb over the parallel gas generator turbopump. Although the bellows pump concept would be approximately 1000 lb heavier than the hydraulic system, it is an attractive concept since it requires no APU power.

C6 Hybrid Concept - Another attractive approach is a hybrid concept using pumped hydrogen and fully pressurized oxygen. In this approach the complexities associated with oxygen pumping are eliminated, yet the system weight increase would be minimal. The hybrid system, using hydraulic driven hydrogen pumps is shown in Figure C-12. Two sets of data are shown, for 40 and 200 gaseous hydrogen accumulator cycles. Hydrogen accumulators sized for 200 cycles resulted in lower maximum accumulator pressures for minimum weight, on the basis of accumulator pressure-volume sizing characteristics. Since the pressures were lower, higher chamber pressures could be accommodated within the 275 hp APU capability. Thus, without oxygen pumping, a chamber pressure of 275 psia can be obtained. The resulting system weight is 10,820 lb at 200 accumulator cycles, or only 220 lb heavier than the parallel gas generator turobpump concept. The above paragraphs show that many attractive gaseous systems without turbopumps are available which are weight competitive with the parallel gas generator system. A summary of the characteristics of each concept and their evaluation is given in Paragraph 3.1.

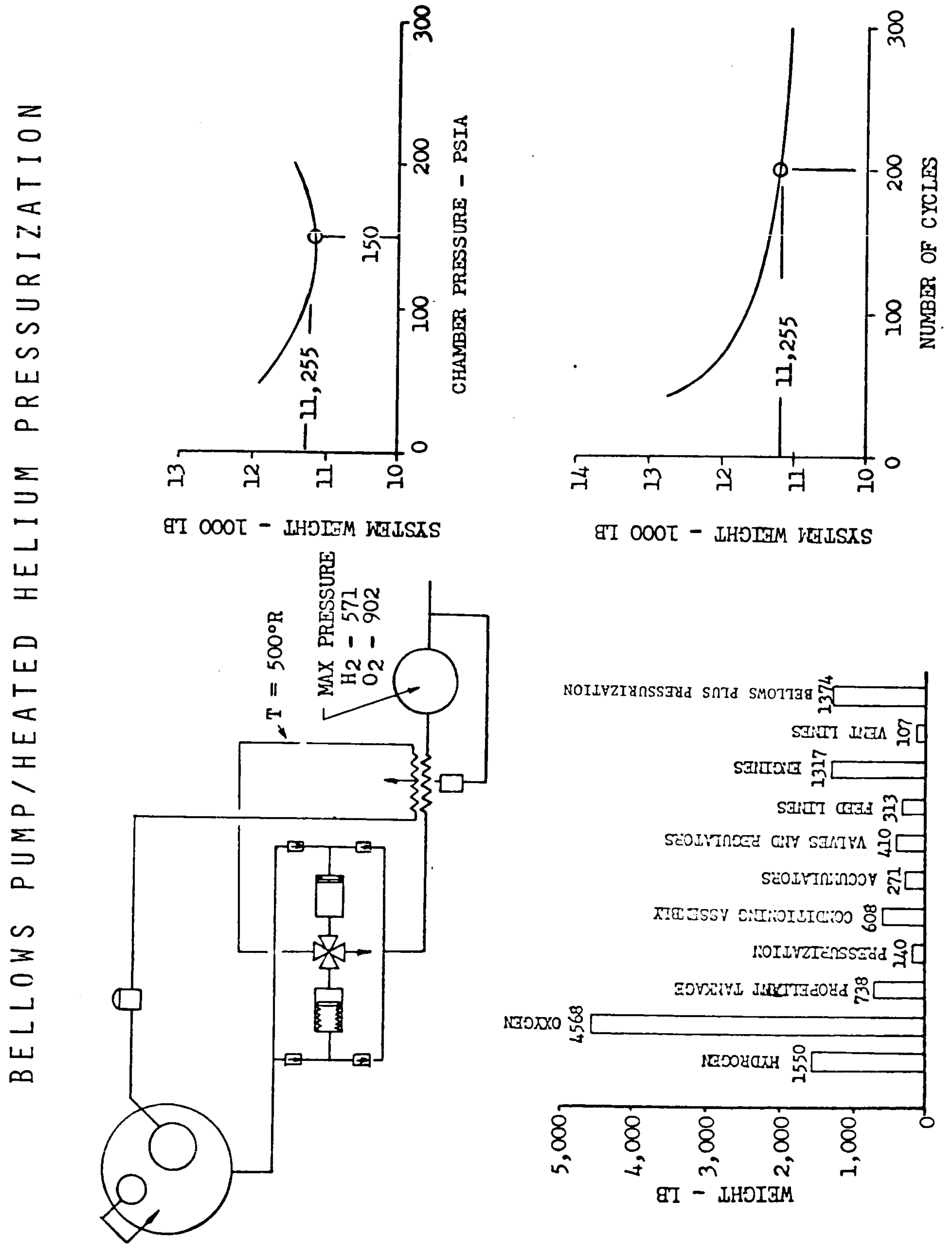
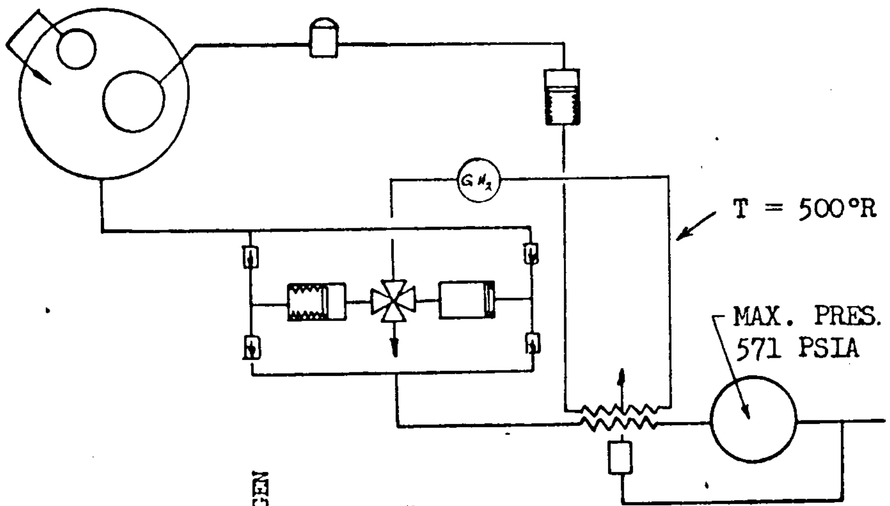
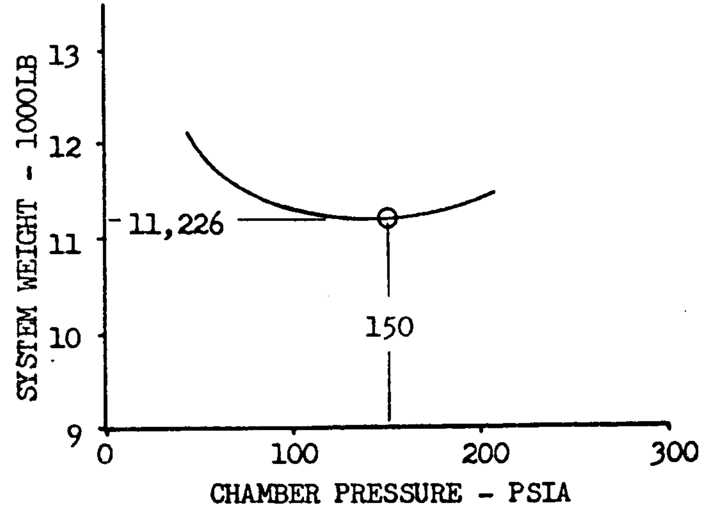

BELLOWS PUMP/HEATED HELIUM PRESSURIZATION
T = 500°R
MAX PRESSURE
H2 - 571
O2 - 902
SYSTEM WEIGHT - 1000 LB
11,255
CHAMBER PRESSURE - PSIA
NUMBER OF CYCLES
WEIGHT - LB
1550 HYDROGEN
4568 OXYGEN
738 PROPELLANT TANKAGE
140 PRESSURIZATION
608 CONDITIONING ASSEMBLY
271 ACCUMULATORS
410 VALVES AND REGULATORS
313 FEED LINES
1317 ENGINES
107 VENT LINES
1374 BELLOWS PLUS PRESSURIZATION

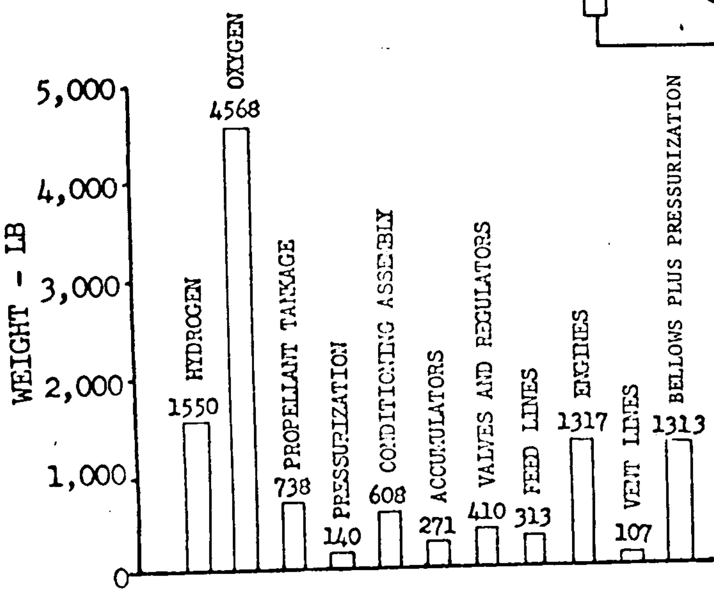
Figure C-10

BELLOWS PUMP - HEATED HYDROGEN PRESSURIZATION

(H_2 SIDE ONLY, HEATED HELIUM O_2 SIDE)







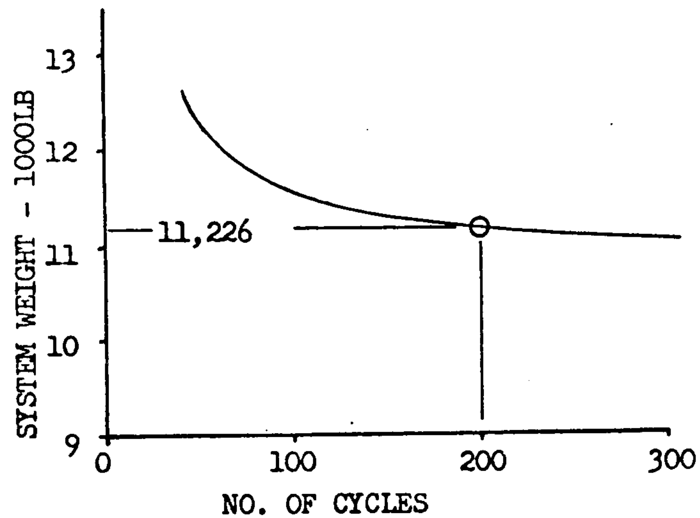


Figure C-11

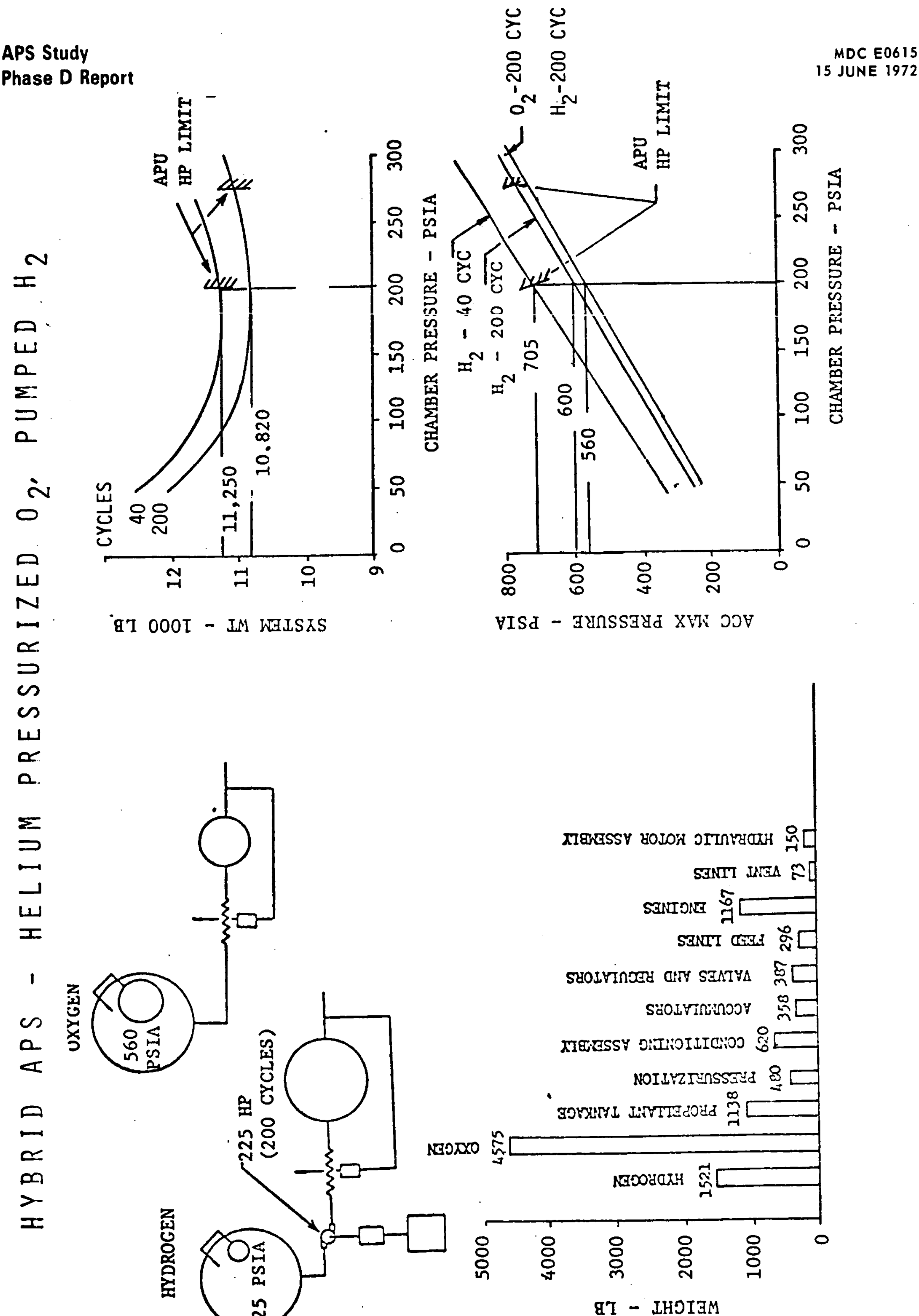

HYBRID APS - HELIUM PRESSURIZED O_2, PUMPED H_2
HYDROGEN
25 PSIA
OXYGEN
560 PSIA
225 HP (200 CYCLES)
SYSTEM WT - 1000 LB
CYCLES
40
200
11,250
10,820
APU HP LIMIT
CHAMBER PRESSURE - PSIA
ACC MAX PRESSURE - PSIA
O_2-200 CYC
H_2-200 CYC
H_2 - 40 CYC
H_2 - 200 CYC
705
600
560
WEIGHT - LB
HYDROGEN 1521
OXYGEN 4575
PROPELLANT TANKAGE 1138
PRESSURIZATION 480
CONDITIONING ASSEMBLY 620
ACCUMULATORS 358
VALVES AND REGULATORS 387
FEED LINES 296
ENGINES 1167
VENT LINES 73
HYDRAULIC MOTOR ASSEMBLY 150

Figure C-12

APPENDIX D
LIQUID RCS TRADES

D1 Baseline Definition - The baseline liquid system, shown schematically in Figure D-1, is a hybrid concept using pumped hydrogen and fully pressurized oxygen, and weighs 9053 lb. Minimum system weight, shown in Figure D-2, is provided at a chamber pressure of 200 psia and a mixture ratio of 4.0. The effect of line sizing on system weight is shown in Figure D-3. System weights are affected more by oxygen line diameter due to the relatively large weight of residual oxygen. An oxygen feed line diameter of 1.0 in. was selected on the basis of weight. Since the weight sensitivity to line diameter was less with hydrogen, a 1.5 in. diameter was selected on the basis of line heat transfer. This size provides additional heat sink capacity(larger liquid residuals) for a small weight penalty.

System weight as a function of pump horsepower and maximum accumulator pressure is shown in Figure D-4. The optimizations shown are for the design chamber pressure of 200 psia, and also for a fixed accumulator capacity which in turn fixes pump flow rate. With pump flow rate fixed, pump horsepower is a function of accumulator pressure only. The data presented in Figure D-4 reflect the effect of variations in accumulator pressure on system weight in terms of both the actual pressure (Figure D-4b) and the pump horsepower required to produce that pressure (Figure D-4a). A minimum in weight occurs because of liquid accumulator weight minimizing at a gas volume/tank volume of about 0.4 and increased pump weight and APU propellant at higher pressures (horsepower). As shown, minimum weight is achieved at 600 psia and the corresponding pump horsepower is 127. The same effects are shown in Figure D-5 for other chamber pressures (the 200 psia curve is a duplicate of that shown in Figure D-4). Again, as in Figure D-4, accumulator capacity and pump flow rate are fixed and the pump horsepower is related to accumulator pressure. Hence along each constant chamber pressure line,accumulator pressure is variable. The dotted line of Figure D-5 describes the locus of minimum system weight as a function of horsepower for the fixed accumulator capacity used. Similar weight trends, obtained for other accumulator sizes, are shown in Figure D-6. These data indicate that the liquid system has sufficient flexibility to accommodate a large range of design points, for a small weight penalty. For example, the design pump power can be reduced from 127 to 20 hp, by increasing accumulator capacity from 17 to 70 lb. This could be accomplished at minimum weight by lowering chamber pressure from 200 to 150 psia but this would also reduce the allowable heat transfer (lower heat

11-17

LIQUID SYSTEM BASELINE SUMMARY

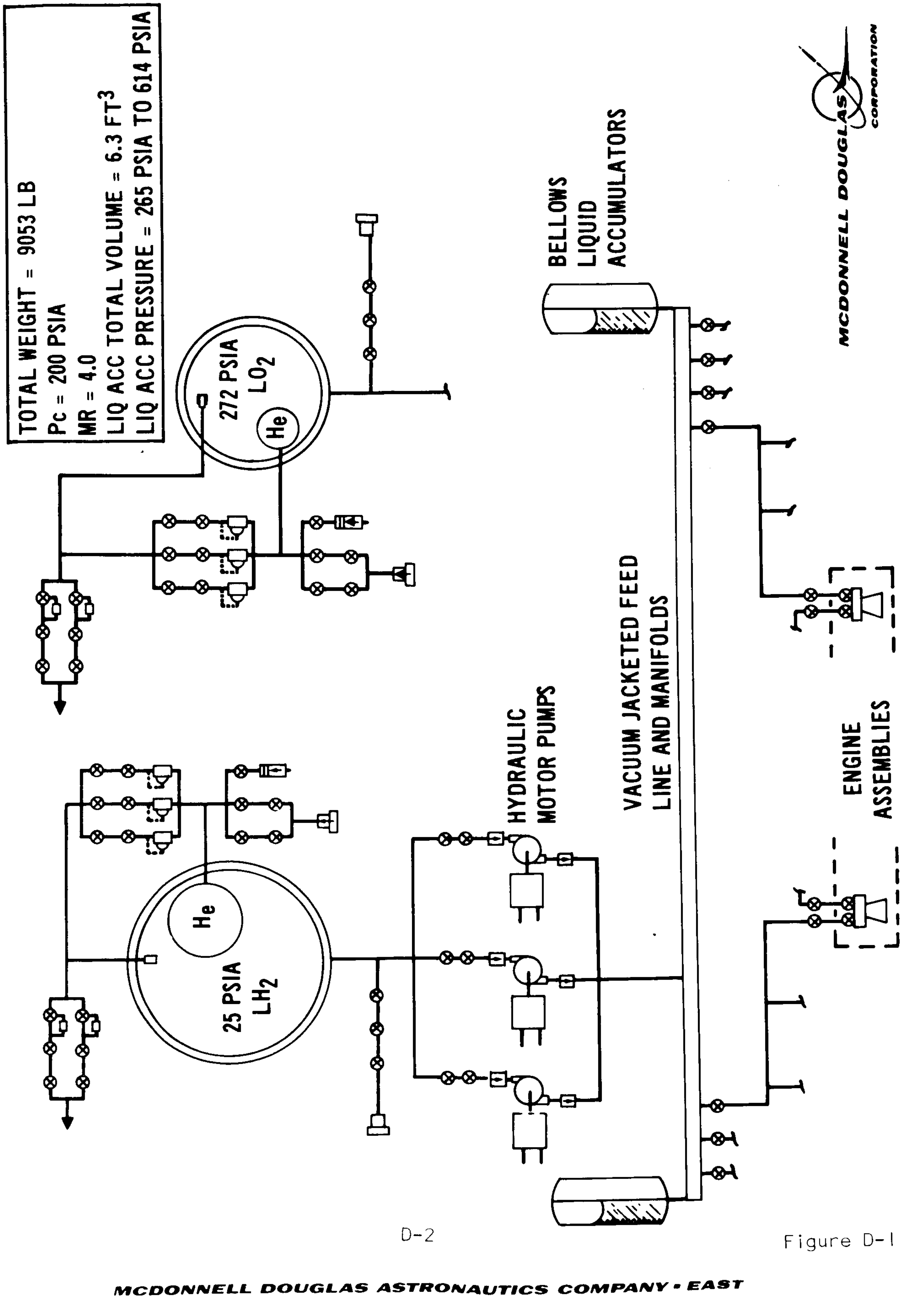


Figure D-1

HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

o He Press O_2, Hydraulic Pump H_2

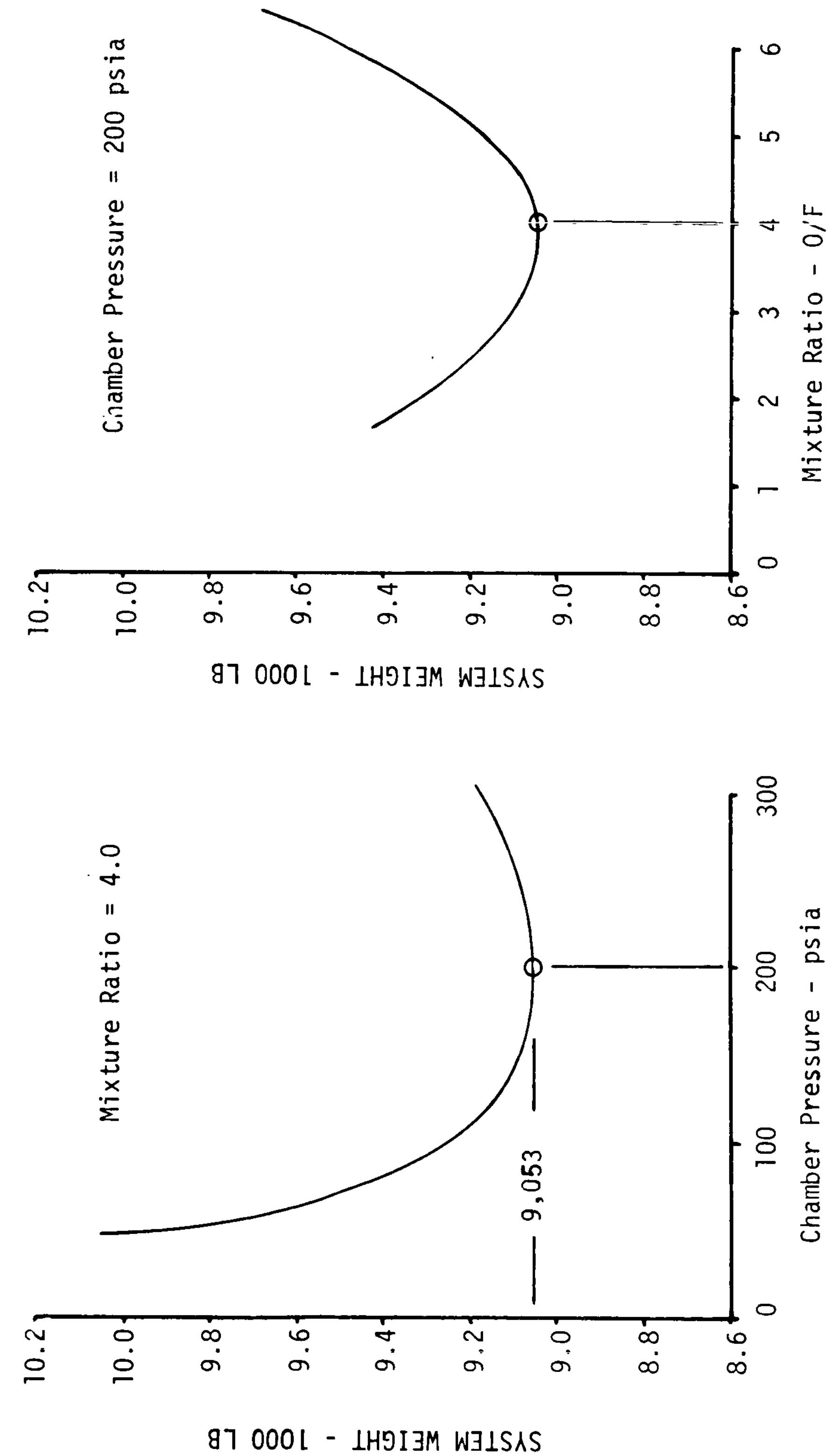


Figure D-2

WEIGHT EFFECT OF LINE SIZING
HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- H_e PRESS O_2, HYDRAULIC PUMP H_2
- P_c = 200 PSIA
- MR = 4.0

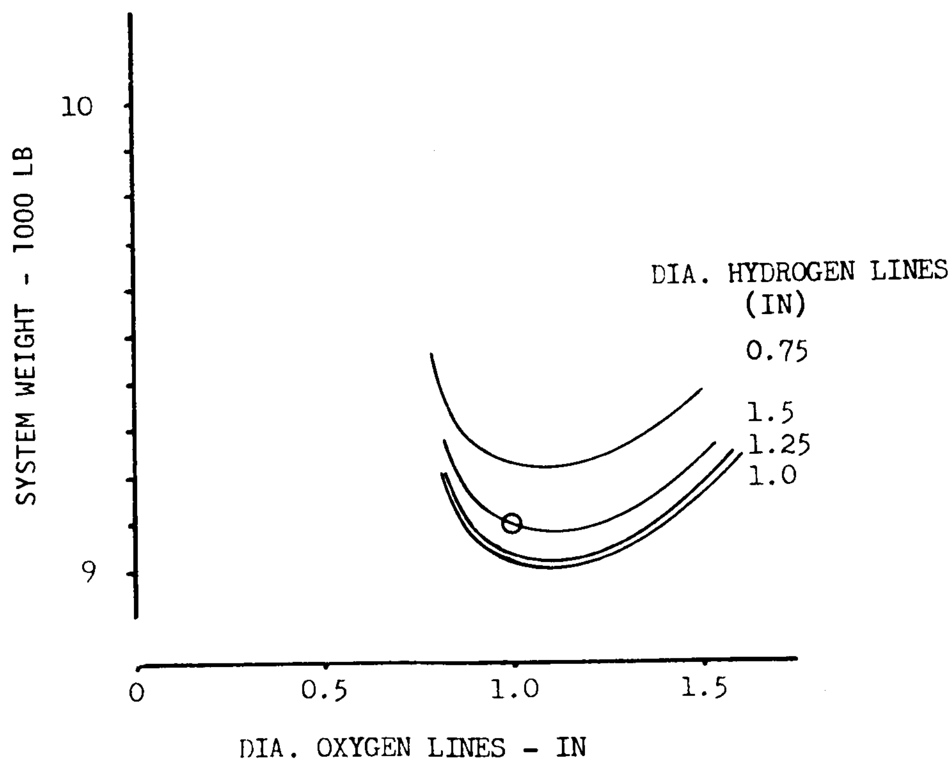


Figure D-3

HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- o He Press O_2, Hydraulic Pump H_2
- o Blowdown Liq Acc
- o MR = 4.0
- o Pc = 200 psia

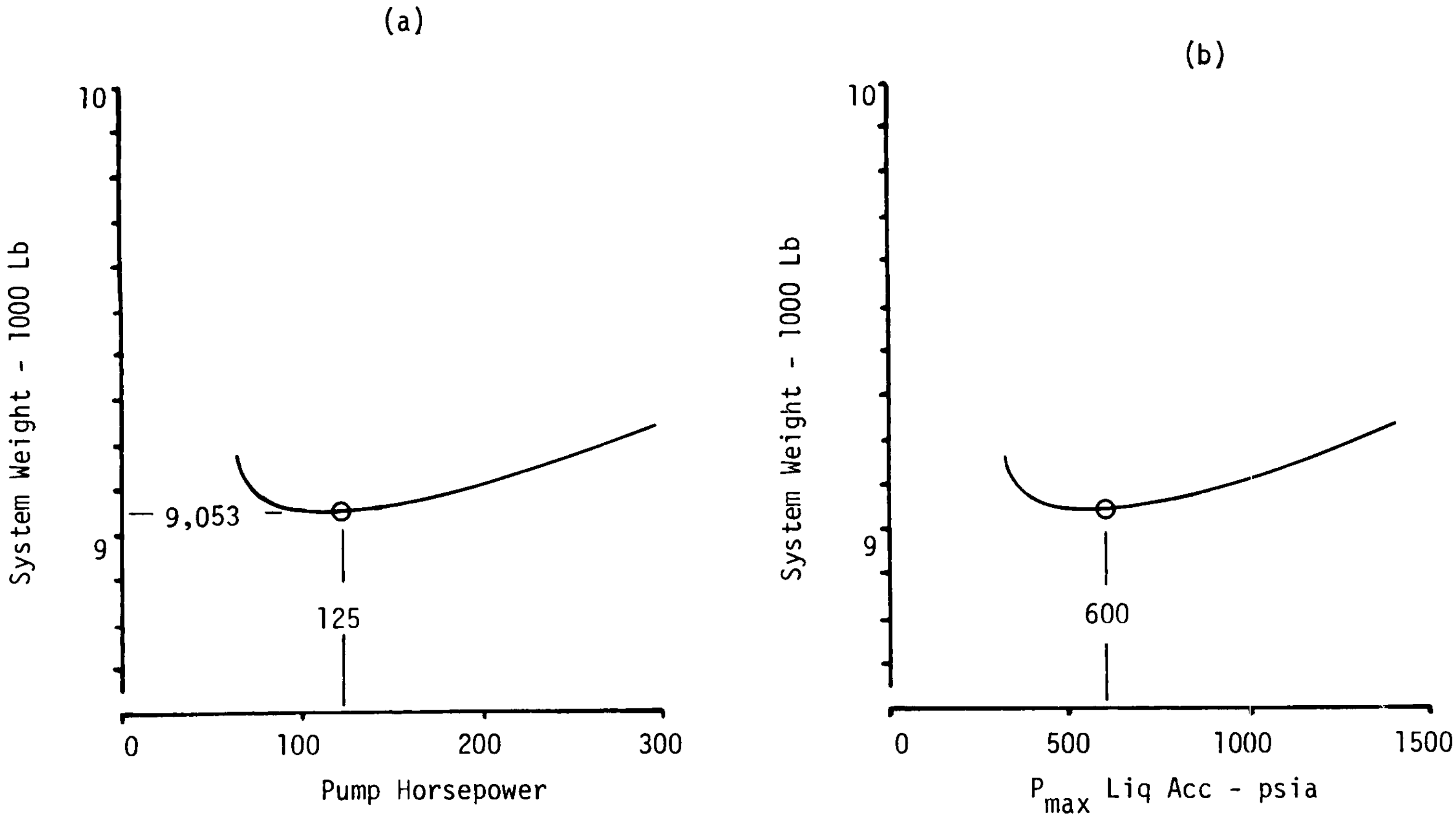


Figure D-4

EFFECT OF CHAMBER PRESSURE ON POWER REQUIREMENTS

- HYBRID APS
- BLOWDOWN ACC, NO VACUUM JACKET
- H_2 ACCUMULATOR CAPACITY - 17.4 LB (40 CYCLES)
- VARIABLE H_2 ACCUMULATOR PRESSURE

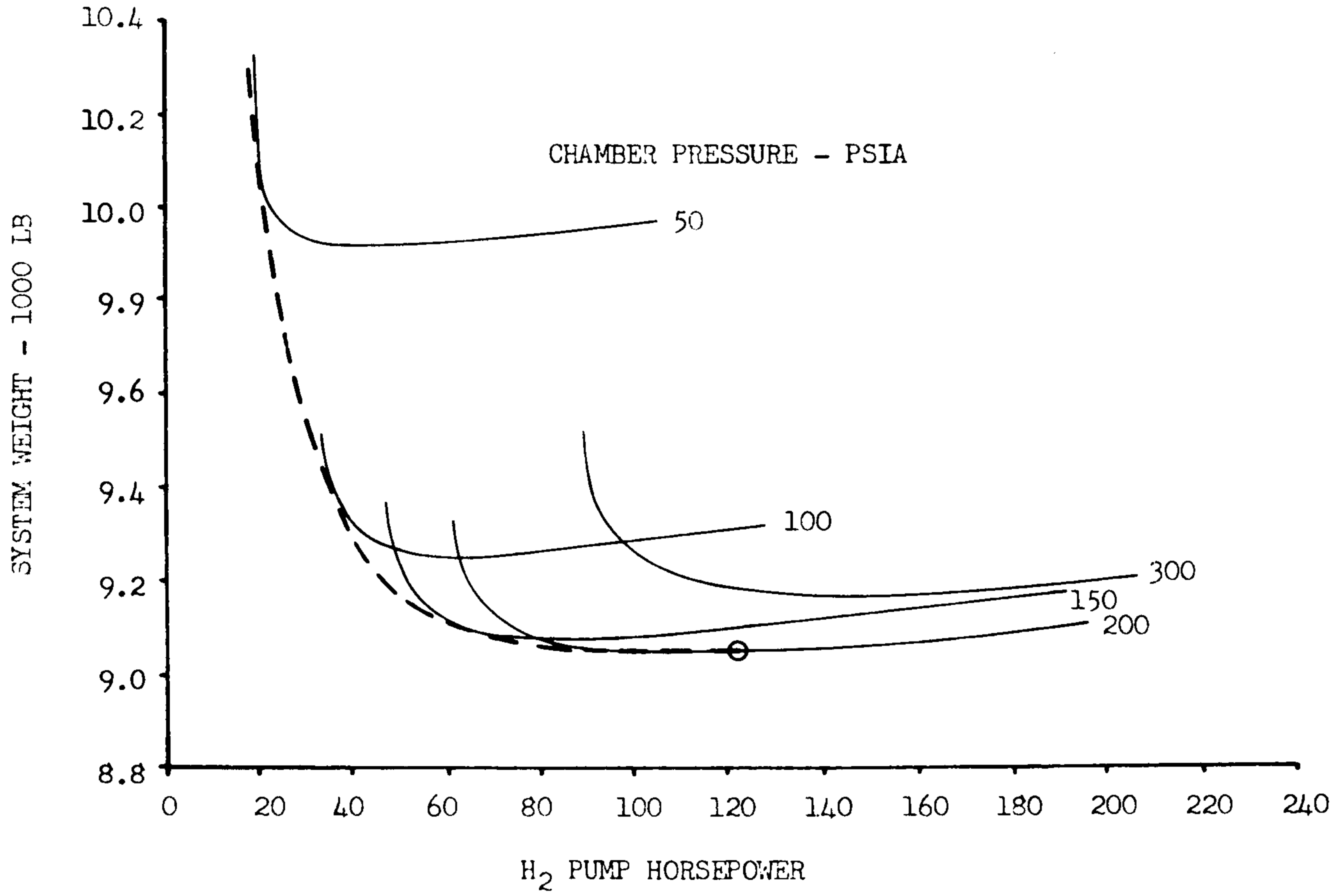


Figure D-5

EFFECT OF ACCUMULATOR CAPACITY AND PUMP POWER REQUIREMENTS ON SYSTEM WEIGHT

- o HYBRID APS
- o BLOWDOWN ACC

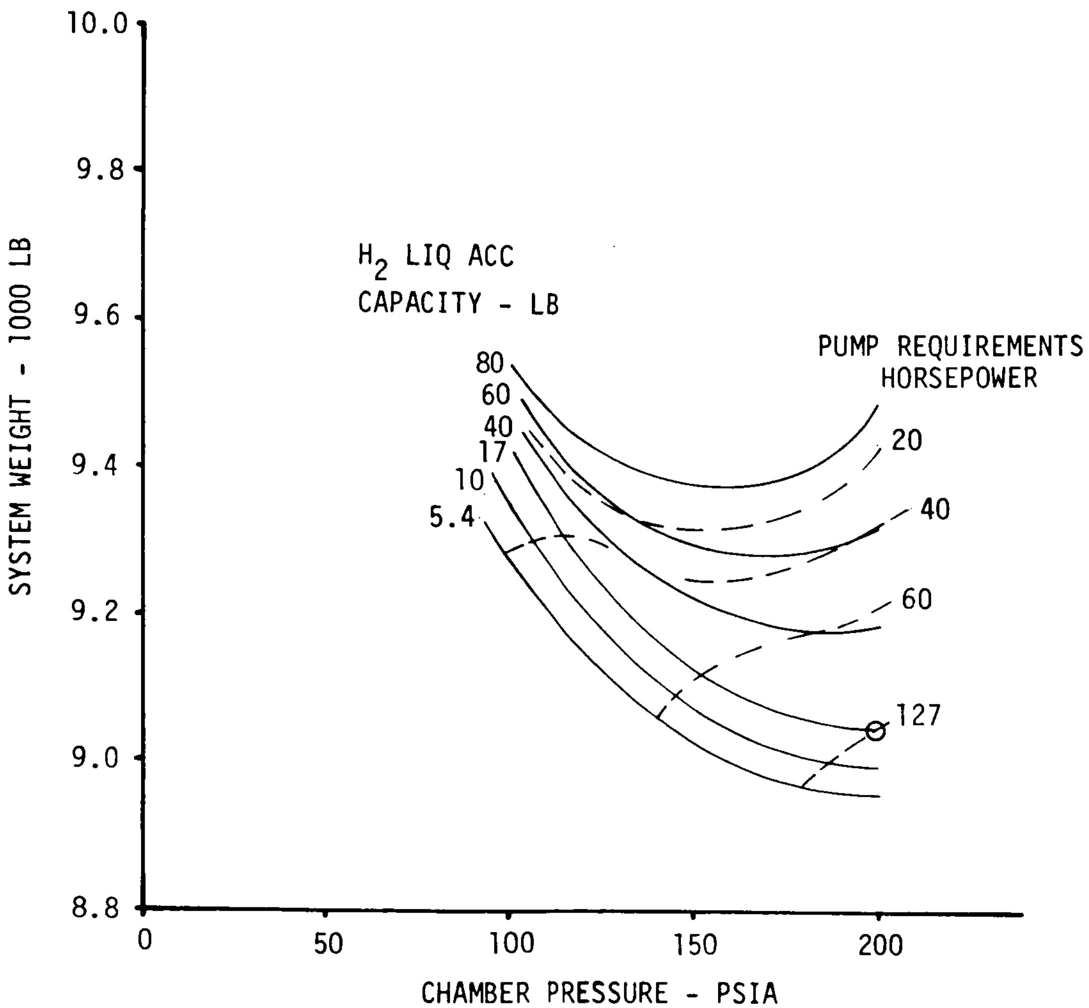


Figure D-6

capacity margin) and would not save sufficient weight to be recommended. Instead, the alternate design points shown in Figure D-7 are most attractive. These data show that reductions in accumulator pressure, from 600 to 400 psia, and increases in accumulator capacity, from 17 to 60 lb,can greatly reduce pump power requirements for a small weight penalty. Increases in accumulator capacity are particularly attractive in that the number of cycles required to provide the attitude control impulse requirements can be reduced from 40 to 8 cycles. This would reduce the total cycle life requirements, including 10 steady-state cycles, from 50 to 18, per mission, thereby simplifying the APU interface.

A weight breakdown of the baseline hybrid system is shown in Figure D-8. As shown, a large portion of the system weight is associated with the propellant feed lines (700 pounds), oxygen pressurization system, motors and pumps, and liquid accumulators. Design options to reduce the weight of each have been investigated. The following paragraphs describe the effects of non-vacuum jacketed lines, alternate oxygen pressurization schemes, and alternate pump options on system weight and design point selection.

D2 Non-Jacketed Feed Lines - The use of non-jacketed lines would significantly reduce system weight. Even though the HPI would be exposed to potential handling and atmospheric damage, the potential weight savings warrants further investigation. The system weight as a function of chamber pressure and mixture ratio is shown in Figure D-9. Minimum weight occurs at a chamber pressure of 200 psia and an engine mixture ratio of 4, the same as the baseline concept discussed above. The weight without vacuum jacketed lines is 8694 lb., giving a weight savings of 370 lb. The effects of pump power and accumulator pressure on system weight is shown in Figure D-10. As shown, the system optimizes for an accumulator pressure of 500 psia resulting in a pump horsepower of 100 hp. The effects of accumulator sizing and pressure on system weight and pump horsepower are shown in Figure D-11. As with the baseline, the horsepower requirements could be reduced to 20 hp for a system weight increase of approximately 400 lb.

D3 Oxygen Pressurization Options - The alternate oxygen pressurization concepts considered were the use of pumped oxygen and a fully pressurized concept with the pressurant stored in the hydrogen tankage. Both options would reduce the baseline oxygen pressurization weights. System weights with a hydraulic driven pump for the oxygen are shown in Figure D-12. The system weighs 8,707 lb at a mixture ratio of 4.0 and 300 psi chamber pressure. This could reduce the baseline system weight by 350 lb. The pump and accumulator pressure effects, shown in

TYPICAL HYBRID SYSTEM OPTIONS

P_c (PSIA)	$P_{MAX.}$ (PSIA)	PUMP HP	ACCUMULATOR CAPACITY (LB)	CYCLES	SYSTEM WEIGHT (LB)
200	600	127	17	40	9053
200	400	70	17	40	9153
200	400	20	60	8	9450
200	600	20	64	8	9320

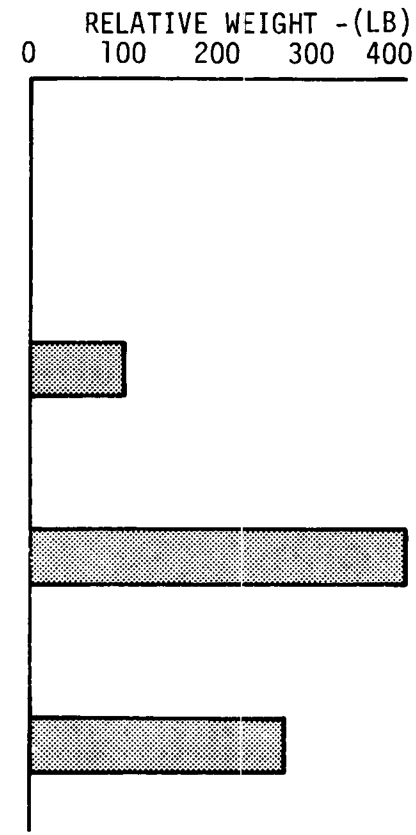


Figure D-7

HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- ENGINE MR = 4.0
- CHAMBER PRESSURE = 200 PSIA
- STORAGE TANK PRESSURE, O_2 = 272 PSIA
  H_2 = 25 PSIA
- H_2 LIQ ACC, TEMP = 40°R
  PRESS = 265 TO 614 PSIA

COMPONENT	WEIGHT - LB	
	HYDROGEN	OXYGEN
PROPELLANT WEIGHT		
USABLE	1046	4185
RESIDUALS, LINES	22	351
TANKS	21	25
VENTED	194	13
TOTAL	1283	4574
PROPELLANT TANKAGE	381	315
PRESSURIZATION	97	262
MOTORS AND PUMPS	115	0
APU PROPELLANT	35	
FEED LINES AND INSULATION	230	230
COMPENSATORS	139	139
LIQUID ACCUMULATORS		
TANK	130	0
PRESSURIZATION	22	0
ISOLATION VALVES (28)	60	60
ENGINES (36)	981	
TOTAL SYSTEM WEIGHT	9,053	

Figure D-8

NON-JACKETED FEEDLINES

HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

° H_e PRESS O_2, HYDRAULIC PUMP H_2

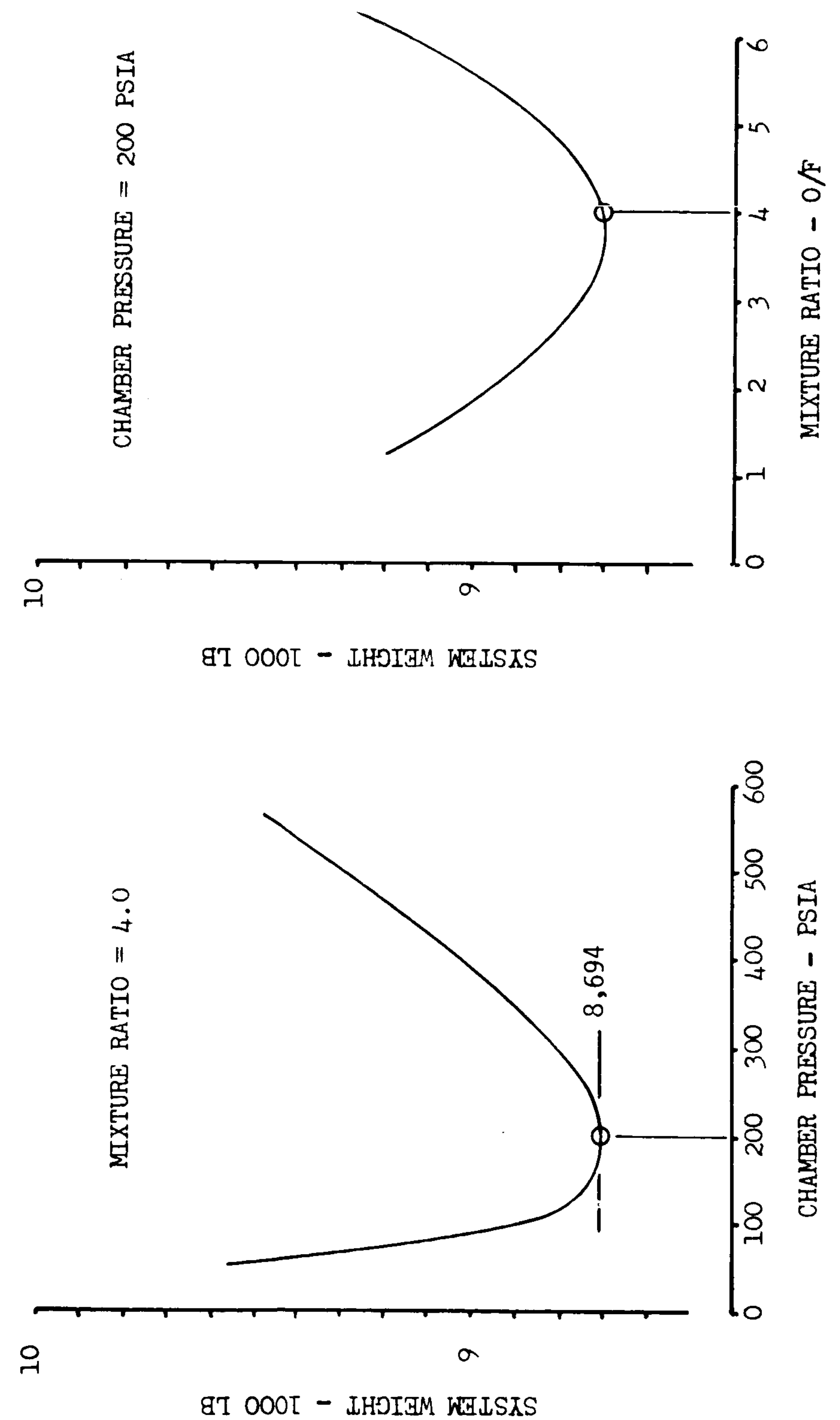


Figure D-9

NON-JACKETED FEEDLINES
HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- H_e PRESS O_2, HYDRAULIC PUMP H_2
- MR = 4.0
- P_c = 200 PSIA

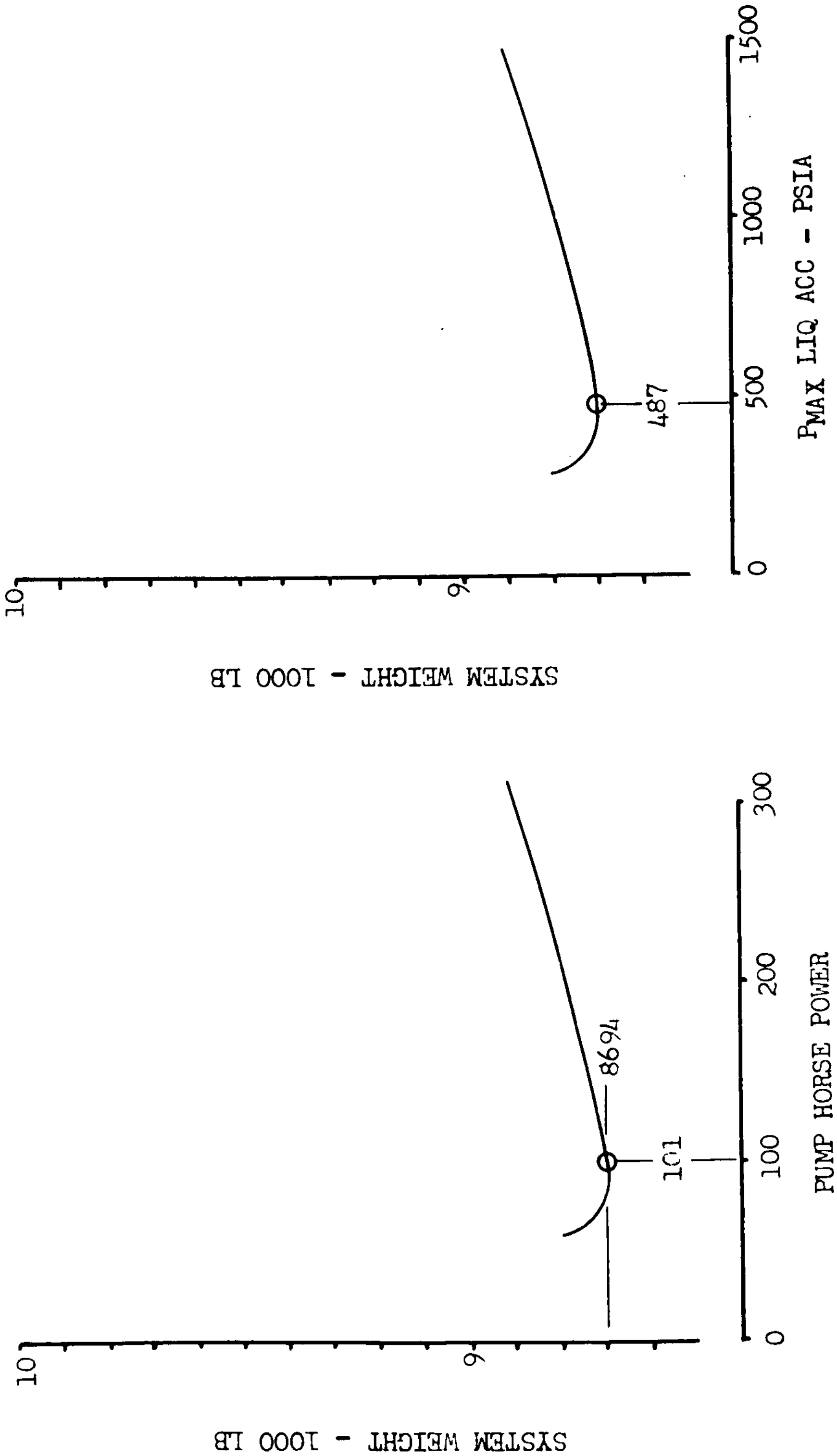


Figure D-10

NO LINE VACUUM JACKET

HYDRAULIC HYBRID LIQUID O_2/H_2 APS

- H_e PRESS O_2, HYDRAULIC PUMP H_2
- P_c = 200 PSIA
- MR = 4.0

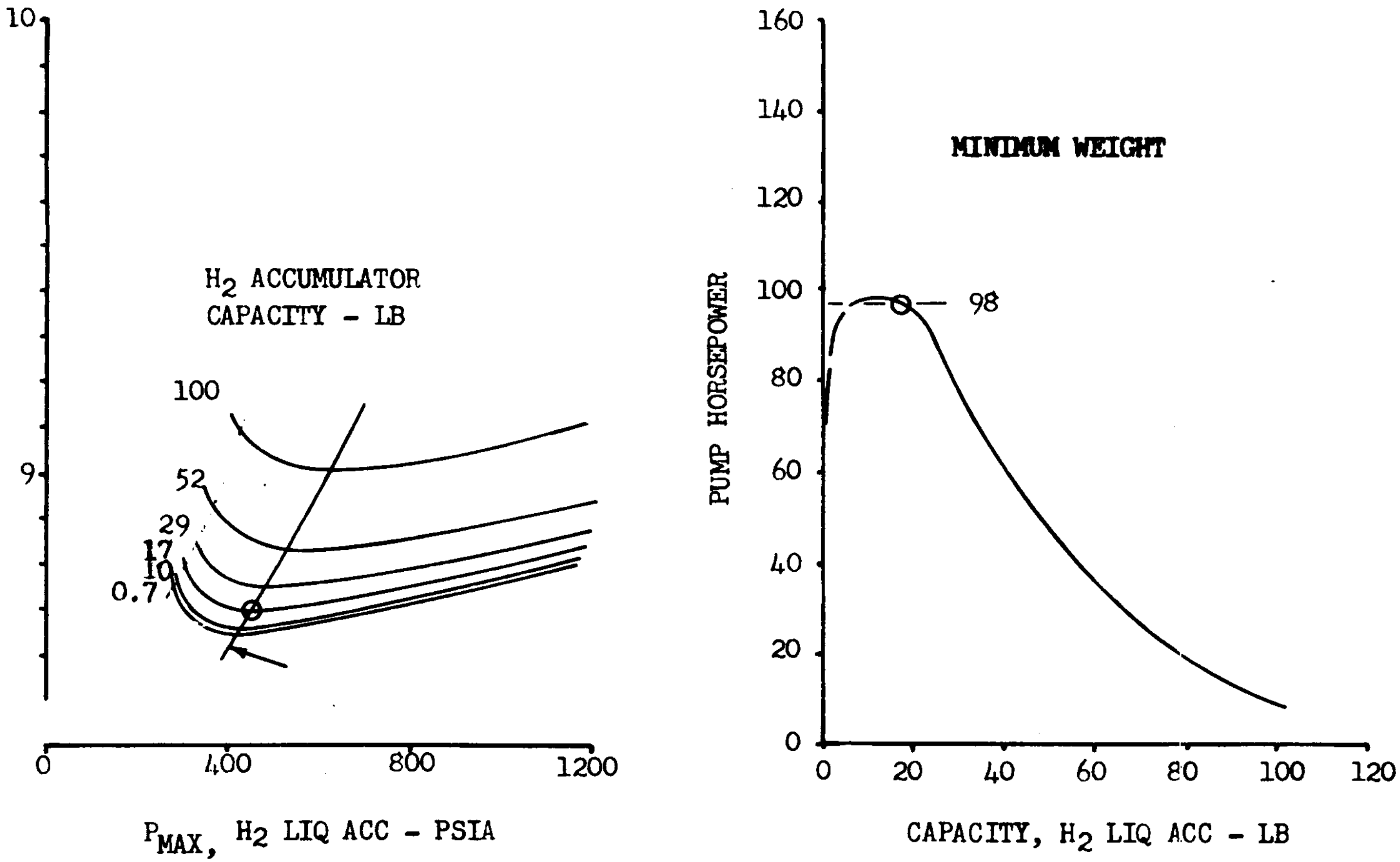


Figure D-11

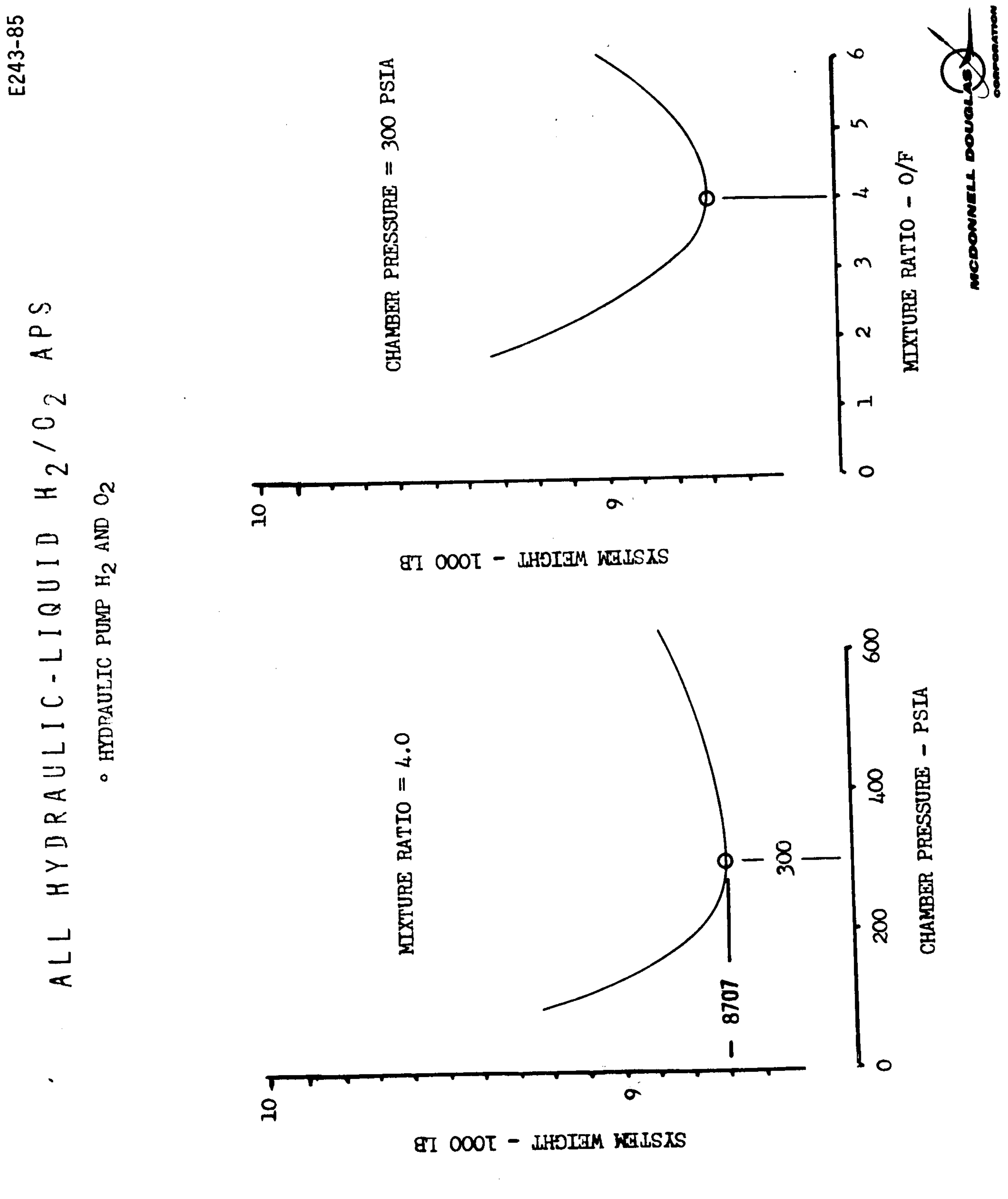

E243-85
ALL HYDRAULIC-LIQUID H_2/O_2 APS
° HYDRAULIC PUMP H_2 AND O_2
CHAMBER PRESSURE = 300 PSIA
MIXTURE RATIO - O/F
0
1
2
3
4
5
6
SYSTEM WEIGHT - 1000 LB
10
9
MIXTURE RATIO = 4.0
8707
300
CHAMBER PRESSURE - PSIA
0
200
400
600
SYSTEM WEIGHT - 1000 LB
10
9
MCDONNELL DOUGLAS CORPORATION

Figure D-12

Figure D-13, resulted in selection of 1,000 psi accumulator pressures and pump power requirements of 258 hp. Both pressure and horsepower are higher than that of the baseline due to the use of a higher chamber pressure (300 psi). The effect of accumulator sizing on power requirements and system weights is shown in Figure D-14. Substantial increases in system weight would be required in order to achieve a significant horsepower reduction. To reach the 20 horsepower level, a penalty of approximately 1500 lb would be incurred. Thus, although the system weight is reduced by pumping the oxygen, weight sensitivity to horsepower requirements is increased.

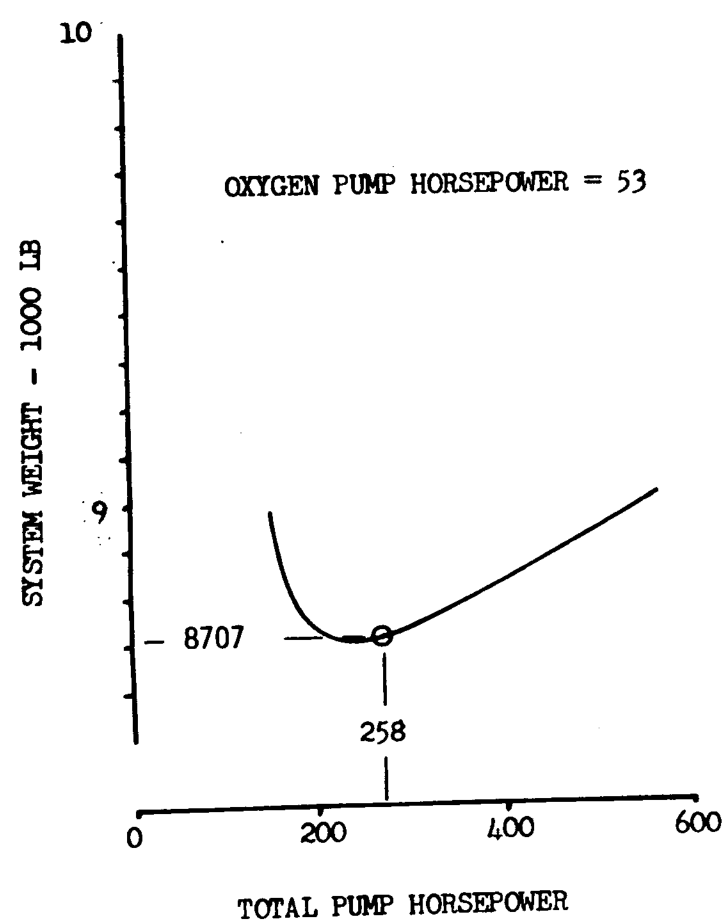
A simpler way to reduce the oxygen pressurization weights for the hybrid system would be to store the helium required for oxygen pressurization within the hydrogen tank. This would reduce the pressurant tank weight by nearly 50% as shown in Figure D-15. A system weight reduction of approximately 150 lb could be obtained. The helium would require heating prior to introduction into the oxygen tank. However, heat requirements are very small (100 BTU total) and could potentially be accommodated by a simple passive heat exchanger using vehicle structure. With this approach, the weight advantage of pumped oxygen would be reduced to only 200 lb.

D4 Liquid Turbopump RCS - Another attractive option was found to be replacement of the hydraulic pump with a hot gas driven turbopump. The turbopump system would operate independently of the APU and would eliminate the APU interface. In addition, the low system weight sensitivity to liquid accumulator sizing, noted above, would allow the use of slow startup turbopumps, avoiding the major technology consideration associated with the gaseous turbopumps, namely, bearing life under rapid startup conditions. Turbopump system weights are shown in Figure D-16. The system weighs essentially the same as the baseline and would operate at 200 psi chamber pressure and a mixture ratio of 4.0.

D5 Regulated Pressure Accumulators and Fully Pressurized RCS - Two remaining options, both resulting in large system weight penalties, were investigated. These were the use of regulated pressure liquid accumulators and a fully pressurized, non-pumped concept. Although the accumulator pressures are less with regulated pressurization than with blowdown, the accumulators would be vented during each refill, greatly increasing the total pressurant requirements. In fact, the pressurization requirements would be the same as for fully pressurized systems. Minimum system weight occurred at low chamber pressure, 100 psia, as shown in Figure D-17. This system weighs approximately 10,000 lb, nearly 1000 lb greater

ALL HYDRAULIC-LIQUID H_2/O_2 APS

- o HYDRAULIC PUMP H_2 AND O_2
- o MR = 4.0
- o PC = 300 PSIA



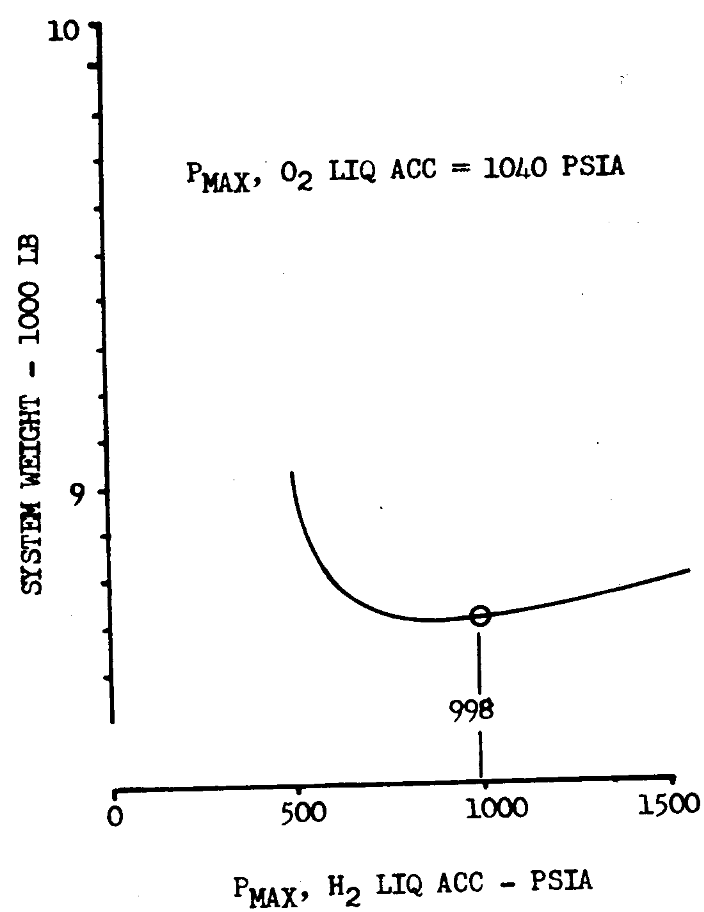


Figure D-13

ALL HYDRAULIC-LIQUID H_2/O_2 APS

- o HYDRAULIC PUMP H_2 AND O_2
- o MR = 4.0
- o PC = 300 PSIA

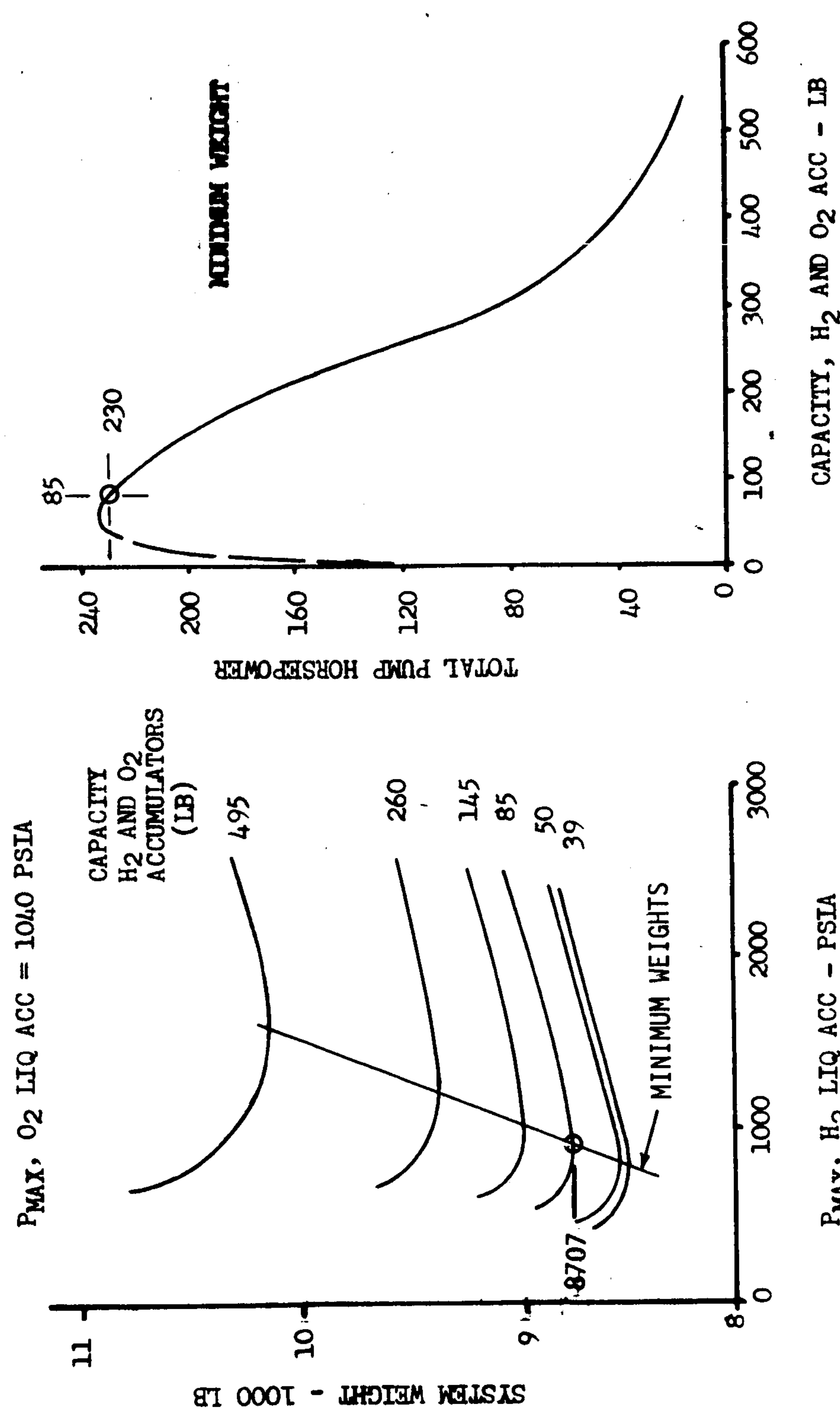
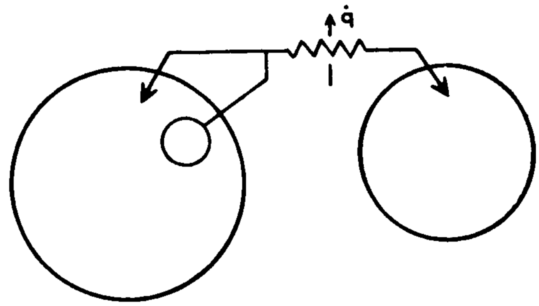


Figure D-14

EFFECT OF O_2 PRESSURIZATION STORAGE IN H_2 TANK



DESIGN CONDITION	He STORAGE TANK LOCATION	
	O_2 TANK	H_2 TANK
He TANK PRESSURE - PSIA	3000	3000
He TANK TEMPERATURE - °R	168	37
He DENSITY - LB/FT^3	5.15	12.4
He WEIGHT - LB	60	75
TANK WEIGHT - LB	348	183
TOTAL WEIGHT - LB	408	254

Figure D-15

TURBOPUMP HYBRID - LIQUID O_2/H_2 APS

- He PRESS O_2
- BLOWDOWN LIQ ACC

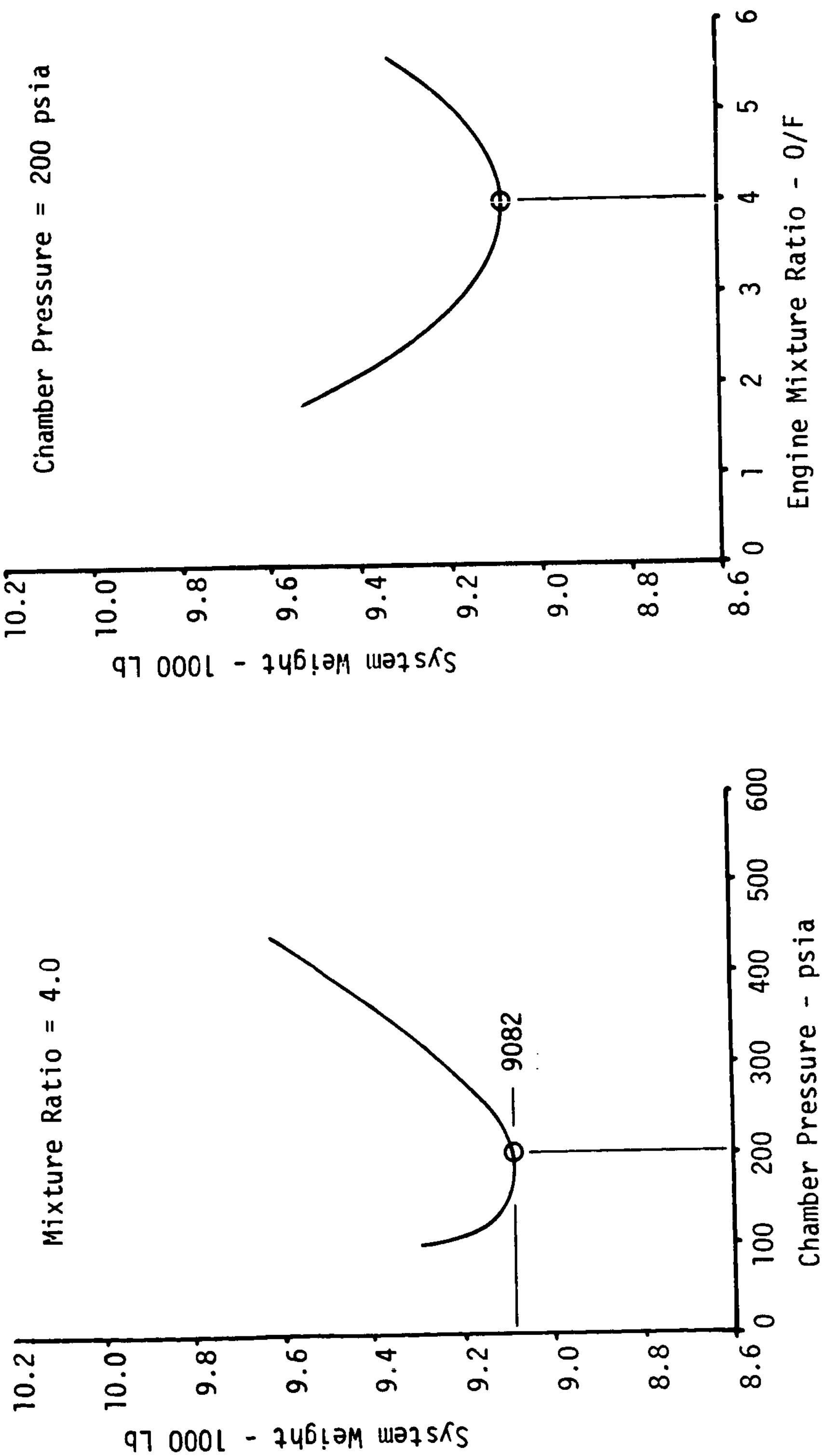


Figure D-16

REGULATED LIQUID ACCUMULATOR HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- H_e PRESS O_2, HYDRAULIC PUMP H_2
- MR = 4.0
- P_c = 100 PSIA

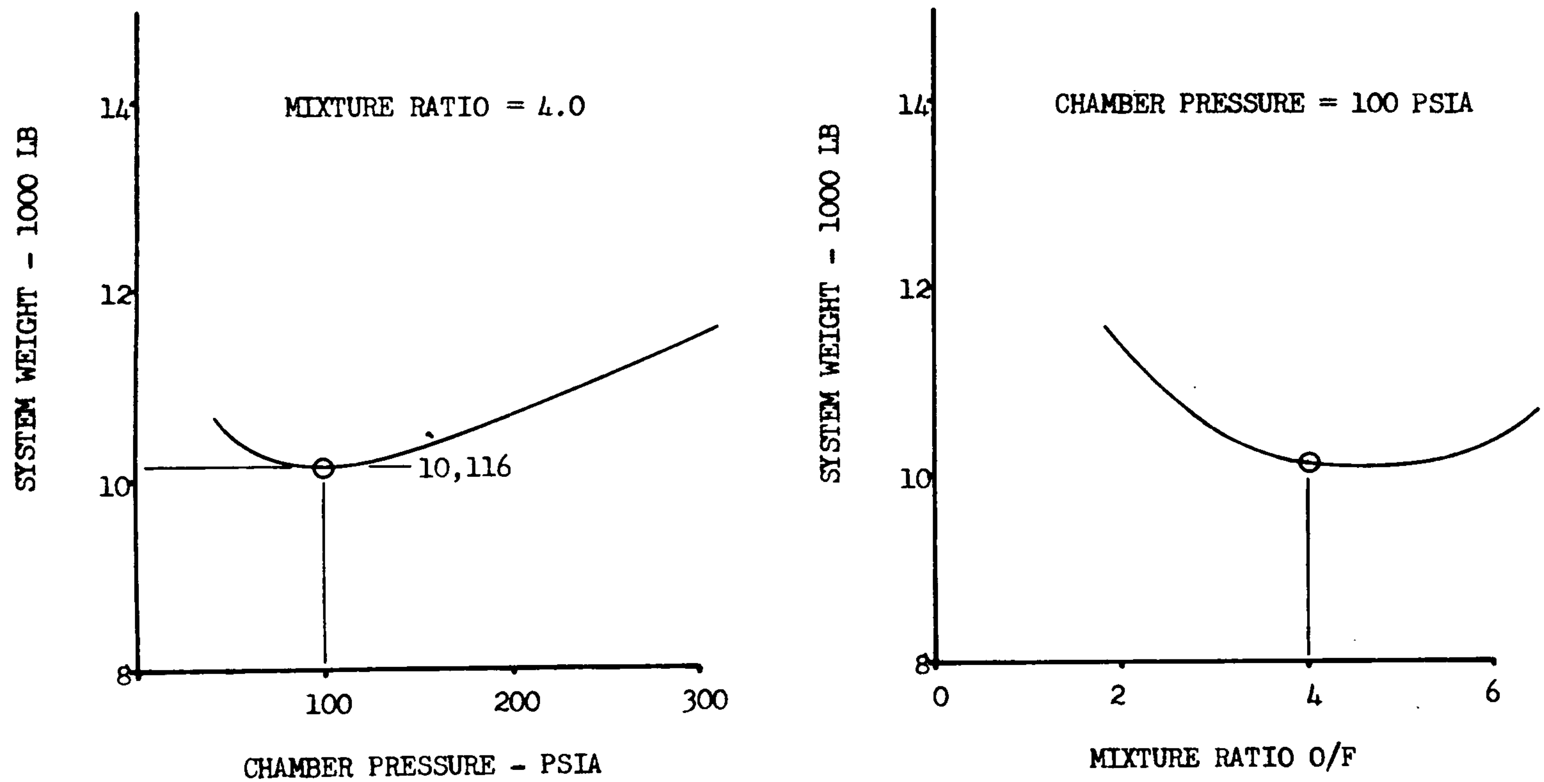


Figure D-17

than the baseline. The effect of accumulator capacity on horsepower and system weight is presented in Figure D-18. As shown, only 30 hp would be required, due to selection of 100 psia chamber pressure. The alternate, a fully pressurized system (Figure D-19) weighed 10,600 lb at 100 psi chamber pressure which is 600 lb more than the regulated accumulator concept. At 200 psi, the fully pressurized system would weigh 12,364 lb. These concepts are clearly not attractive for an operational system, but could be used for initial interim systems and updated later to a higher performance concept.

REGULATED LIQUID ACCUMULATOR
HYDRAULIC HYBRID - LIQUID O_2/H_2 APS

- H_e PRESS O_2, HYDRAULIC PUMP H_2
- MR = 4.0
- P_c = 100 PSIA

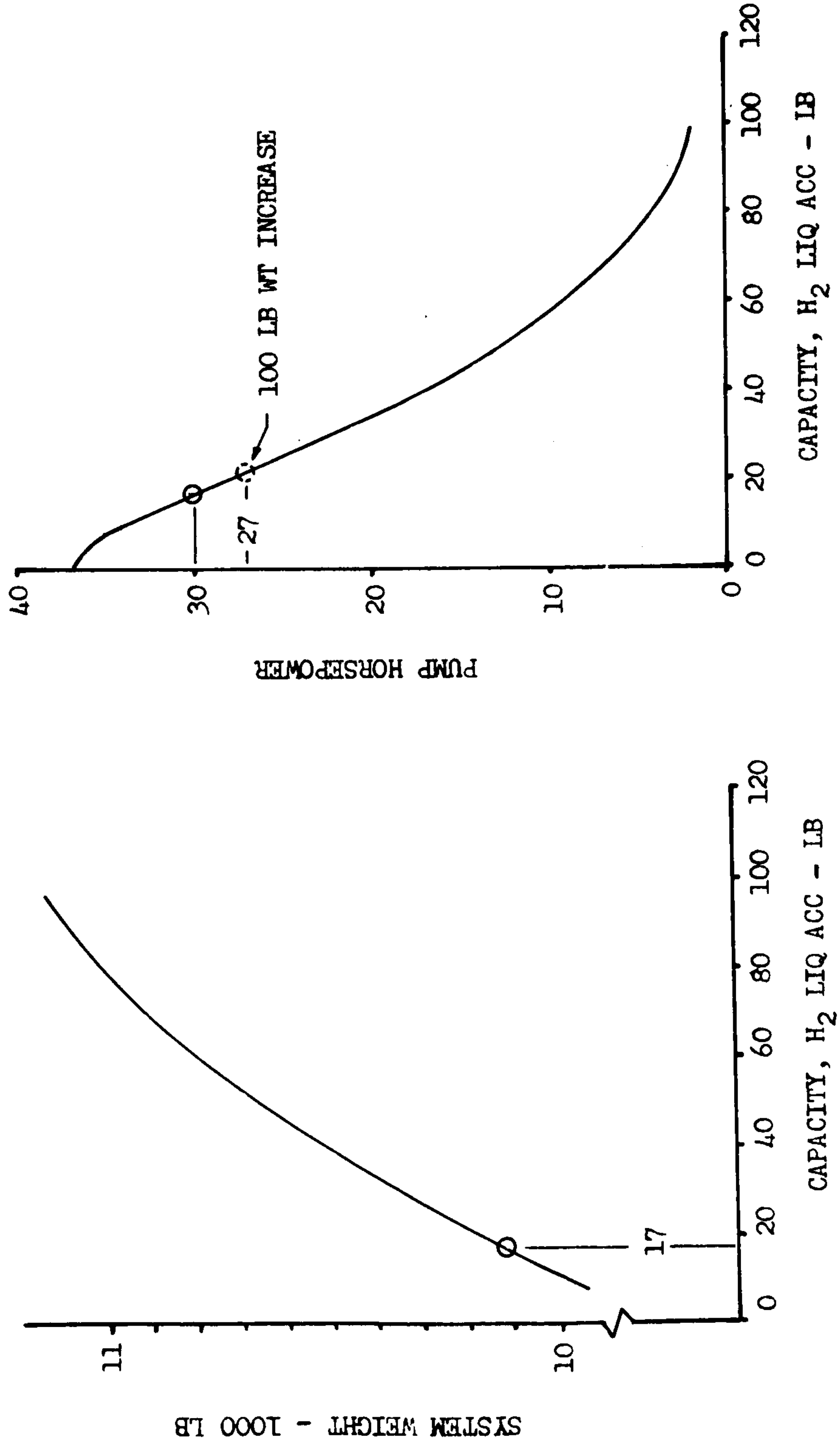


Figure D-18

FULLY PRESSURIZED - LIQUID O_2/H_2 APS

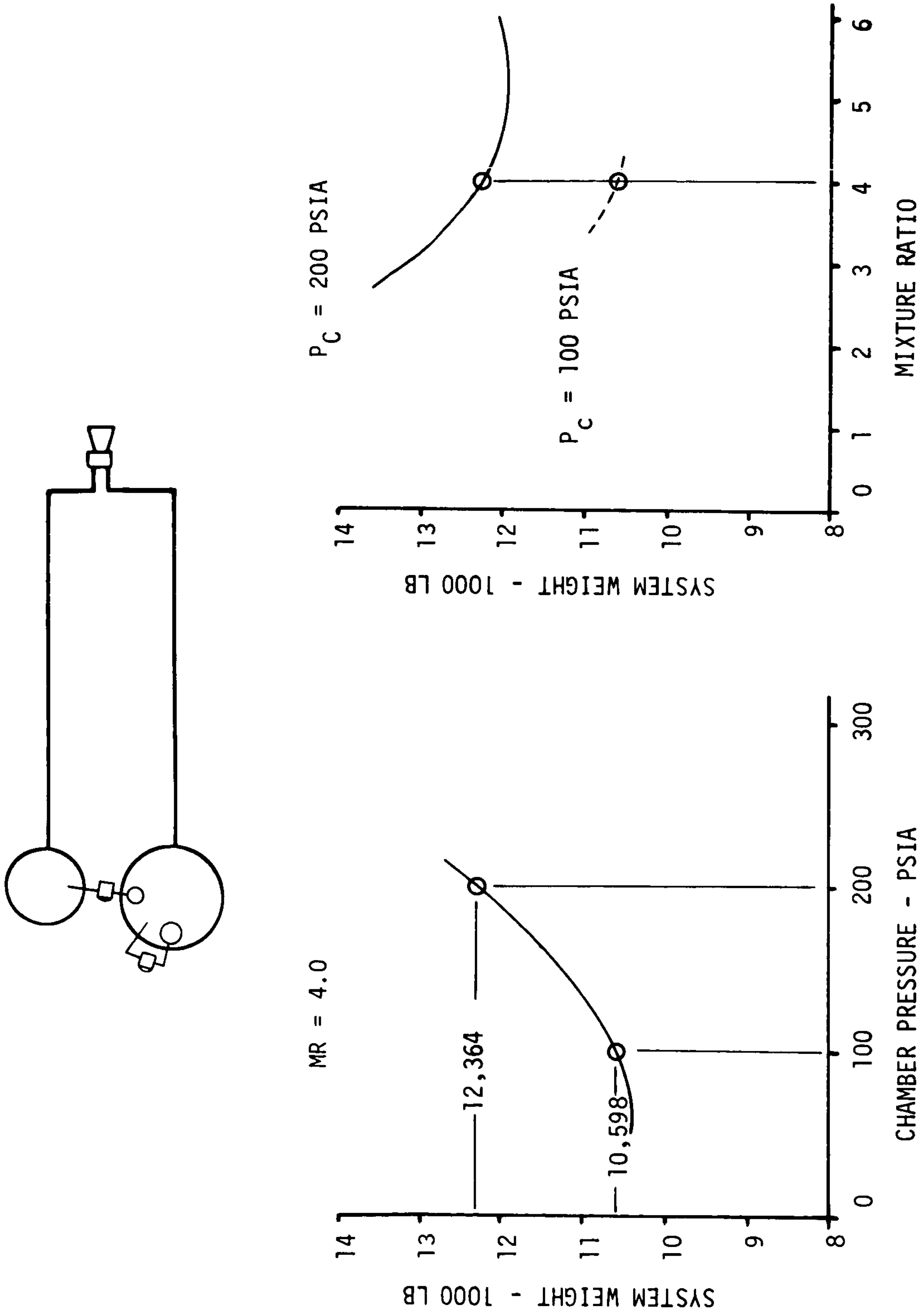


Figure D-19

APPENDIX E

LIQUID HYDROGEN MOTOR OPERATED PUMP STUDY

E1 Introduction - The best choice of pump and power source type was not apparent from liquid RCS system weight and design studies alone, as discussed in Paragraph 3.3. Weight differences between alternate pump types and power sources were small, indicating that pump design and technology factors could be the most significant criteria in pump and power source selection. In order to evaluate the available options in more detail, a subcontract was issued to Pesco Products, a division of Borg-Warner Corporation (now with Sunstrand Corporation). The subcontract was for an analytical study of hydrogen pumps to provide sufficient data for valid pump comparisons and selection, and for evaluation of pump design requirements resulting from system trade studies. Both centrifugal and positive displacement pumps were considered. The positive displacement pump effort included gear, vane and piston pumps. To provide sufficient data for system trades, both hydraulic and electric motor drives were considered over the range of head rise and flow rate requirements. Pump flow rate limits were established by maximum flow conditions without a liquid accumulator and the minimum flow rate with a large accumulator. Pump outlet pressures from 400 to 600 psia were based on system operating conditions that provided for a 100 lb increase in system weight referenced to the highest performance design point.

The following paragraphs summarize the study results and identify significant pump/motor operating characteristics. An evaluation of these options in conjunction with overall system design considerations is given in Paragraph 3.3.

E2 Centrifugal Pumps - The primary design problem with motor operated centrifugal pumps is matching of the pump and motor characteristics. High pump efficiency requires a high flow rate and a relatively high stage specific speed, as shown in Figure E-1. For the maximum hydrogen flow rate requirement of 280 gpm (2.65 lb/sec), the maximum pump efficiency would be 75% at a stage specific speed of 2000. However, specific speed is a function of stage head rise, flowrate and pump speed. In order to meet the high flow rate and low head rise requirements (280 gpm and 400 psi respectively), a single stage pump would require an operating speed of 150,000 rpm. Lower flow rates and/or higher pressures would require even higher speeds.

Hydraulic motors generally operate below 10,000 rpm and electric motors could provide 21,600 rpm, based on the Space Shuttle APU 400 cps power supply (see Appendix A). In order to satisfy pump requirements and still deliver a reasonable

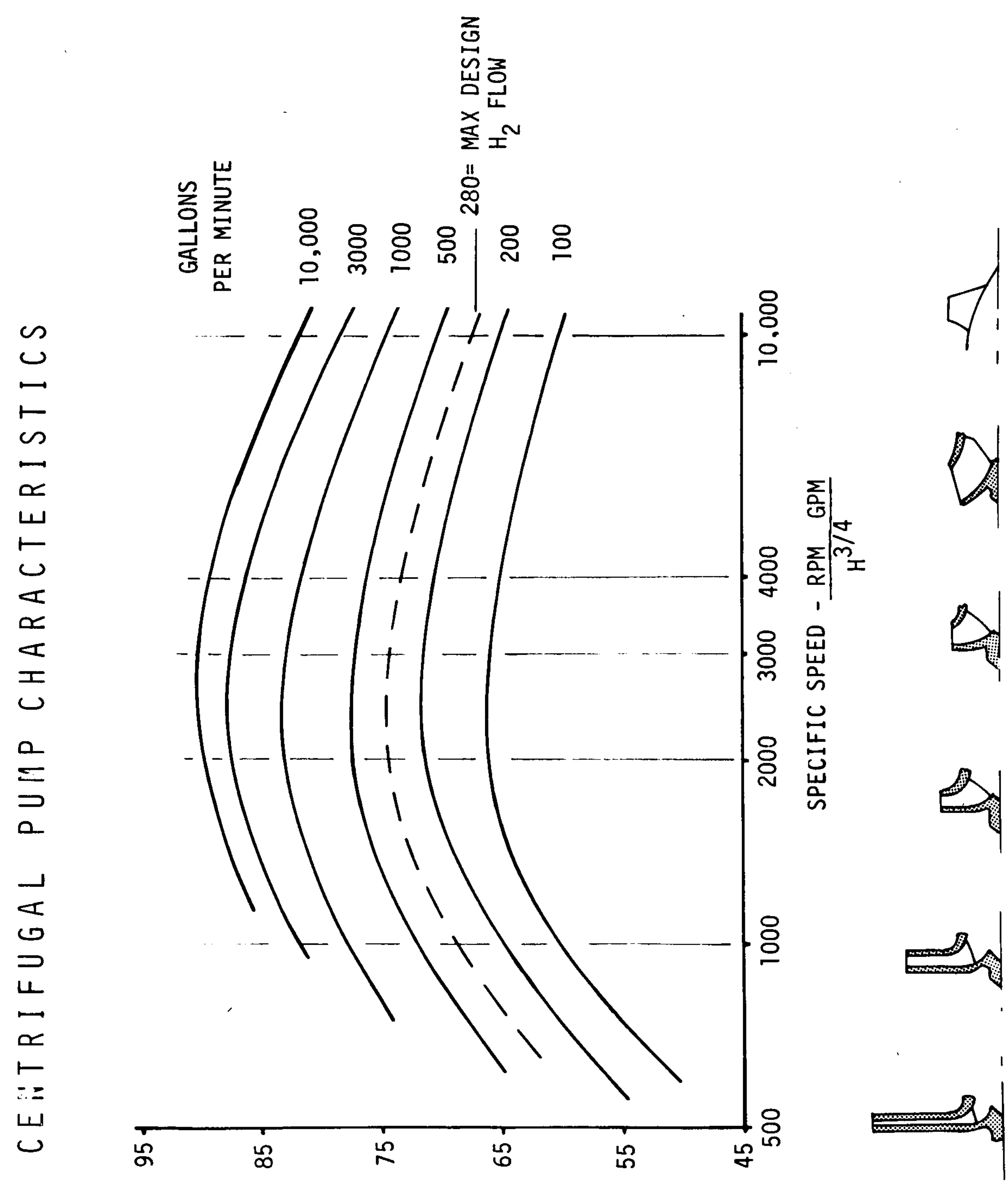

CENTRIFUGAL PUMP CHARACTERISTICS
EFFICIENCY - %
95
85
75
65
55
45
500
1000
2000
3000
4000
10,000
SPECIFIC SPEED - RPM √GPM / H^3/4
GALLONS PER MINUTE
10,000
3000
1000
500
280= MAX DESIGN H2 FLOW
200
100

Figure E-1

efficiency, additional pump stages will be required. This would decrease the head per stage thereby increasing the stage specific speed. The number of stages required for electric motor drive is shown in Figure E-2. No more than three pump stages were considered on the basis of pump complexity.

Based on the above data, the centrifugal pumps were limited to high flow rates only. The following pump characteristics were defined for 2.65 lb/sec hydrogen flow:

Outlet Pressure (psia)	No Stages	Power Source	Unit Weight (lb)	Overall Efficiency (%)
600	3	Electric	117	45
		Hydraulic	358	39
400	2	Electric	88	46
		Hydraulic	112	41

With centrifugal pumps, electric motors provide lighter weight and higher efficiency. In addition, a gear box is required with hydraulic drive, resulting in a more complex pump design.

E3 Positive Displacement Pumps - Positive displacement pumps are generally superior to centrifugal pumps for high head rise, low flow rate applications. These requirements correspond to pumps operating with large accumulators where low flow rates can be accommodated. The pump types considered are illustrated in Figure E-3. Piston, gear and vane pumps were evaluated.

Piston pumps operate as rotating barrels containing multiple pistons and cylinders. The cylinder barrel is mounted at an angle to the drive barrel, such that piston movement relative to the cylinder walls occurs when the barrels are rotated. This concept is shown in detail in Figure E-4. Pump speed is limited by centrifugal piston forces and by piston velocity in the cylinders. This is particularly significant for large flow rates, requiring large pistons and cylinders. As shown in Figure E-5, pump speeds would be reduced to nearly 1000 rpm at the maximum flow requirement. An additional problem would be the inherent leakage past piston rings and past the rotating valve plate used to duct flow between inlet and outlet ports. This leakage requires a case drain which could either vent the leakage overboard or back into the main propellant supply.

Gear pumps are simple, reliable, and inexpensive. They provide a continuous transfer of fluid from the inlet to the outlet port without reciprocating parts or valving. Normally, gear pumps are designed with pressure loaded side plates and physical wiping between the gears and gear case. This minimizes leakage past

LH_2 CENTRIFUGAL PUMP CHARACTERISTICS
PESCO PRODUCTS

SPEED = 21,600 RPM

SPECIFIC SPEED = 500

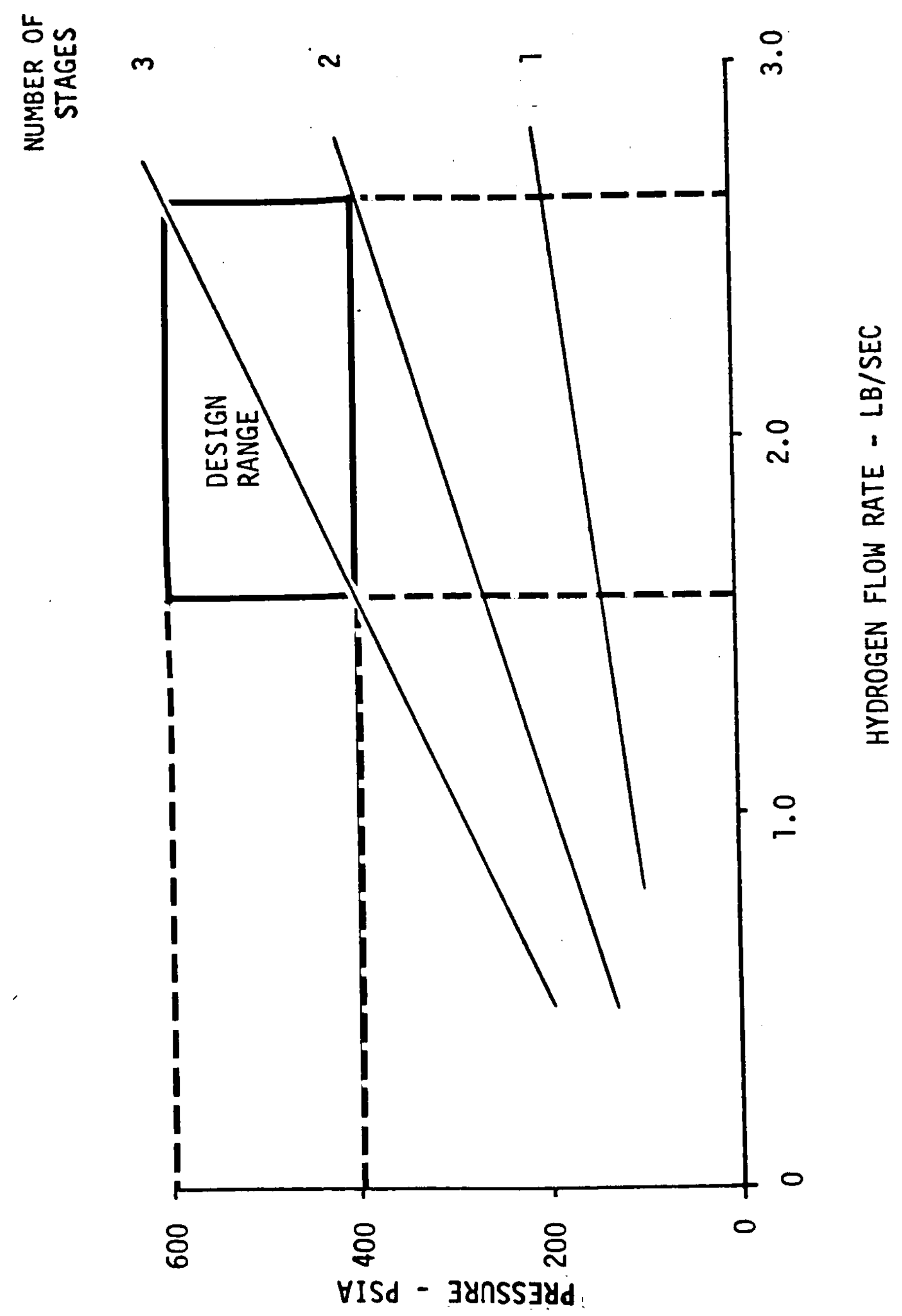
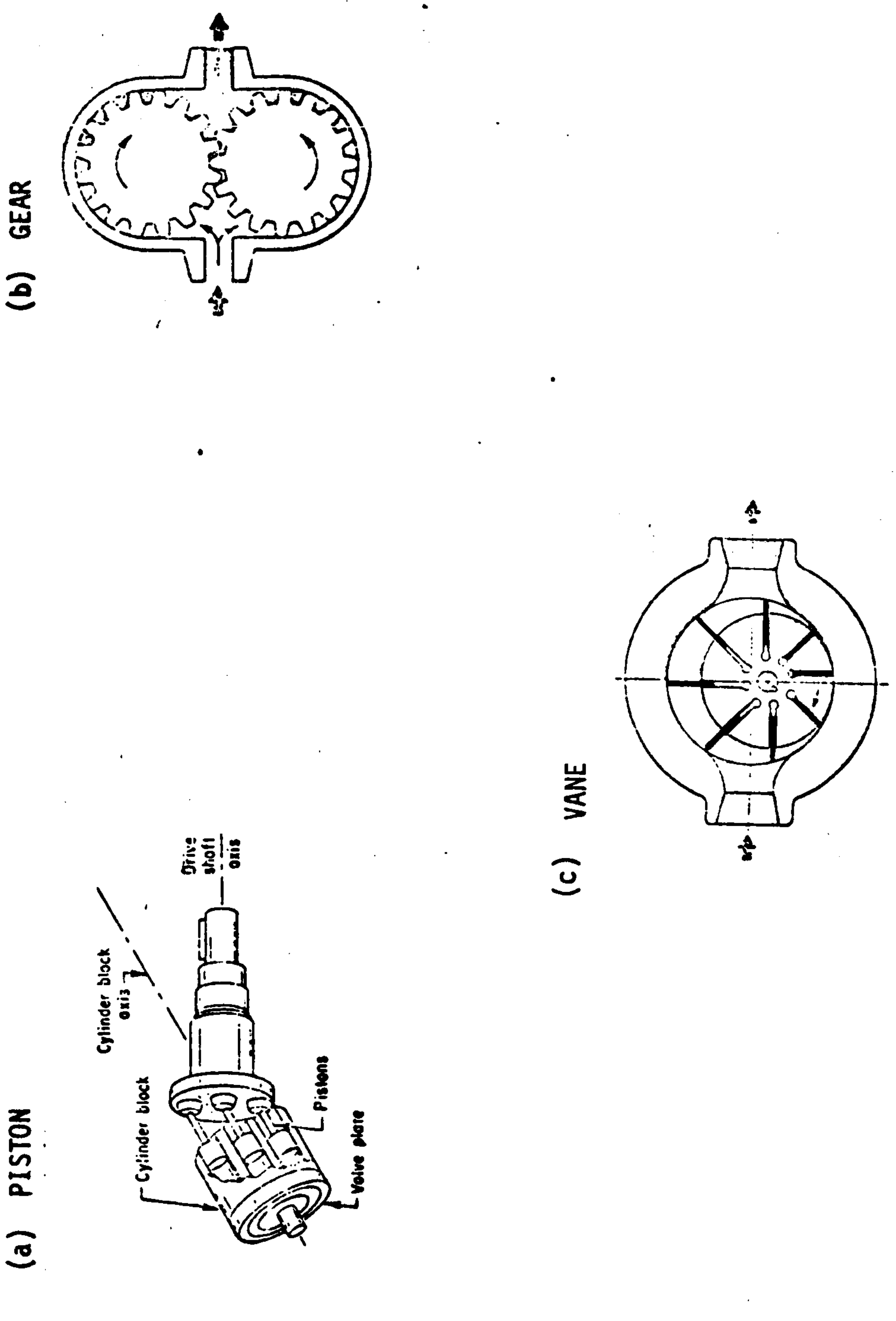


Figure E-2

POSITIVE DISPLACEMENT PUMP CONCEPTS



PISTON PUMP INTERNAL VIEW

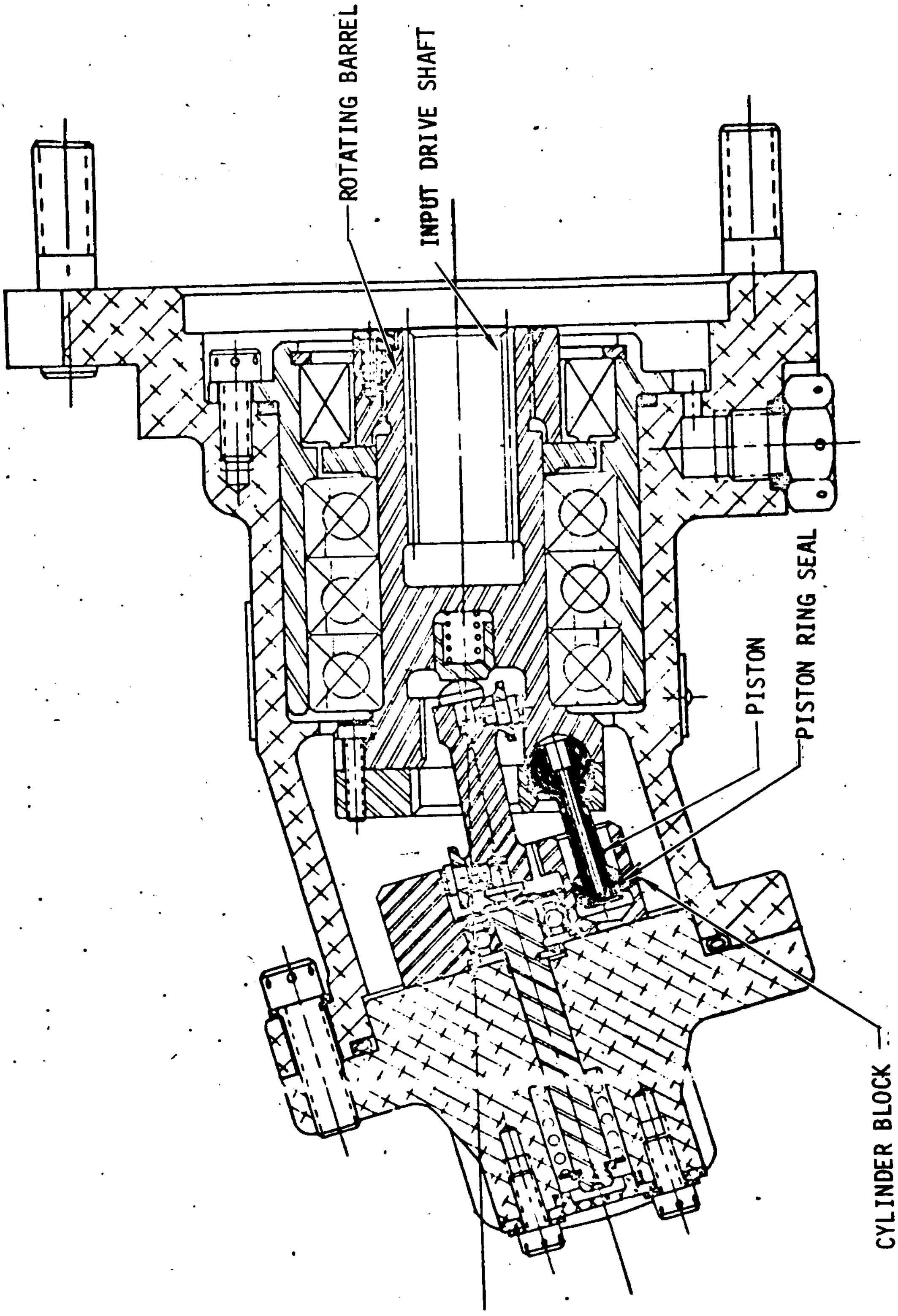


Figure E-4

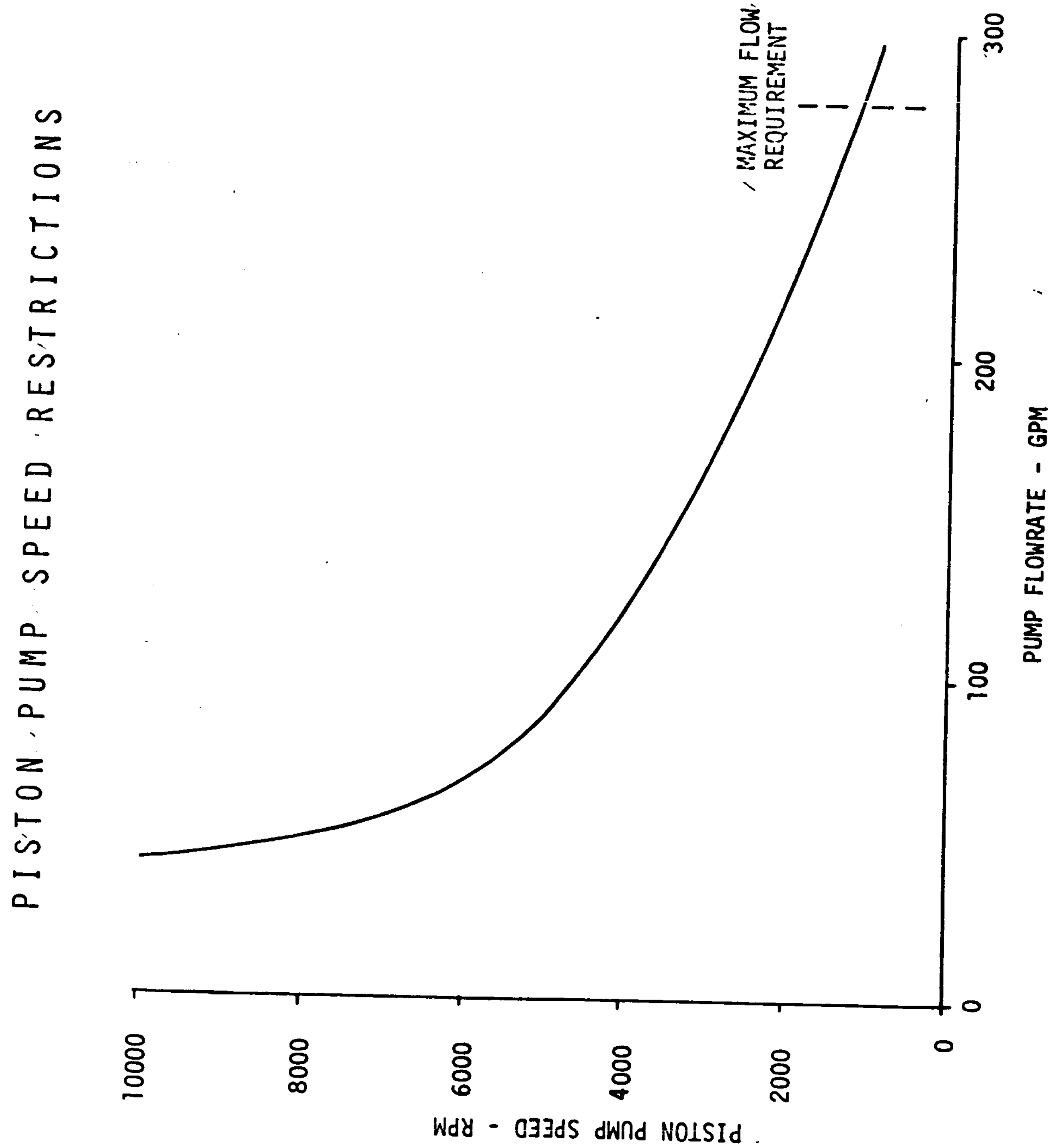

PISTON PUMP SPEED RESTRICTIONS
MAXIMUM FLOW REQUIREMENT
PISTON PUMP SPEED - RPM
0
2000
4000
6000
8000
10000
PUMP FLOWRATE - GPM
0
100
200
300

Figure E-5

the gear tips. However, because of low hydrogen lubricity, pressure loading and gear wiping features would not be employed. Fixed clearances would be used resulting in increased internal leakage. Since the leakage is carried through the pump, no external case drain would be required.

Vane pumps consist of a rotor with sliding vanes mounted within an elliptical cam ring. Side and radial inflow filling is used to keep inlet pressure losses to a minimum. Pump internal leakage is routed past the bearings for cooling, then returned to the vane pump inlet. This pump type has undergone considerable development effort at Pesco Products, including a detailed materials evaluation of sixty-five blade-cam material combinations to compare wear, friction, and run-in characteristics. A vane pump has successfully been used to pump liquid hydrogen at 35 gpm, 270 psia, and 4000 rpm. Disassembly of the unit after the tests indicated that there were no significant problem areas.

A summary of the pump characteristics is shown in Figures E-6 and E-7 for electric and hydraulic drives respectively. Weight data are shown in Figures E-8 and E-9. Piston pumps are heavier than either gear or vane pumps. Hydraulic powered pumps are lighter than electric motor operated pumps due to better speed matching between the hydraulic power and the pumps. Electric motors would require low speed operation,which increases weight and also reduces efficiency. On the basis of weight, efficiency,and development effort, a vane pump using hydraulic drive appears to be the best choice. A layout of this pump type is shown in Figure E-10. The design shown incorporates an inducer to provide low-NPSP operation and a thermal standoff required to prevent excessive cooling of the hydraulic drive fluid.

ELECTRIC MOTOR OPERATED POSITIVE DISPLACEMENT PUMPS

PUMP TYPE	FLOW (LB/SEC)	PRESS RISE (PSI)	PUMP RPM	UNIT WEIGHT (LB)	ENVELOPE		POWER INPUT KW	EFFICIENCY %
					DIAMETER IN	LENGTH IN		
PISTON	2.65	600	1200	2680	30	51	170	47
	2.00	600	2000	1017	25	33	110	50
	1.32	600	3600	266	15	24	68	54
	0.66	600	6400	55	7	19	30	61
	0.37	600	12000	15	4	12	15.6	64
	0.37	400	12000	13.2	4	12	12.7	52
	2.65	150	1200	1200	25	41	60	31
	2.00	150	2000	417	18	26	35	40
	1.32	150	3600	111	10	20	22	44
	0.66	150	6400	27	6	18	9	51
VANE	2.65	600	4000	315	15	32	160	49
	0.66	600	4000	96	22	29	40	49
	2.65	600	12000	136	21	26	144	38
	0.66	600	12000	49	17	20	36	38
	0.37	600	7200	33	8	17	20	50
	0.37	400	7200	30	7	16	15	45
GEAR	2.65	600	4000	392	15	35	160	49
	0.66	600	4000	135	10	30	40	49
	2.65	600	12000	200	10	27	144	38
	0.66	600	12000	74	7	21	36	38

Figure E-6

HYDRAULIC MOTOR OPERATED POSITIVE DISPLACEMENT PUMPS

PUMP TYPE	FLOW (LB/SEC)	PRESS RISE (PSI)	PUMP RPM	UNIT WEIGHT (LB)	ENVELOPE		POWER INPUT HP	EFFICIENCY %
					DIAMETER IN	LENGTH IN		
PISTON	2.65	600	1200	460	13	50	160	66
	2.00	600	2000	189	10	33	113	65
	1.32	600	3600	64	6	24	75	65
	0.66	600	6400	28	5	19	39	63
	2.65	400	1200	450	13	50	124	51
	2.65	150	1200	336	13	43	57	44
	2.00	150	2000	145	10	29	35	54
	1.32	150	3600	51	6	21	24	53
	0.66	150	6400	21	5	16	12	51
VANE	2.65	600	4000	121	12	36	177	60
	0.66	600	4000	47	12	30	45	58
	2.65	600	12000	87	12	30	206	36
	0.66	600	12000	42	11	23	53	35
	2.65	400	4000	116	10	35	128	53
GEAR	2.65	600	4000	198	12	39	177	60
	0.66	600	4000	86	10	31	45	58
	2.65	600	12000	151	10	31	206	36
	0.66	600	12000	67	7	25	53	35

Figure E-7

LH_2 POSITIVE DISPLACEMENT PUMP WEIGHTS

- o 600 PSIA PUMP HEAD RISE
- o ELECTRIC MOTOR OPERATED

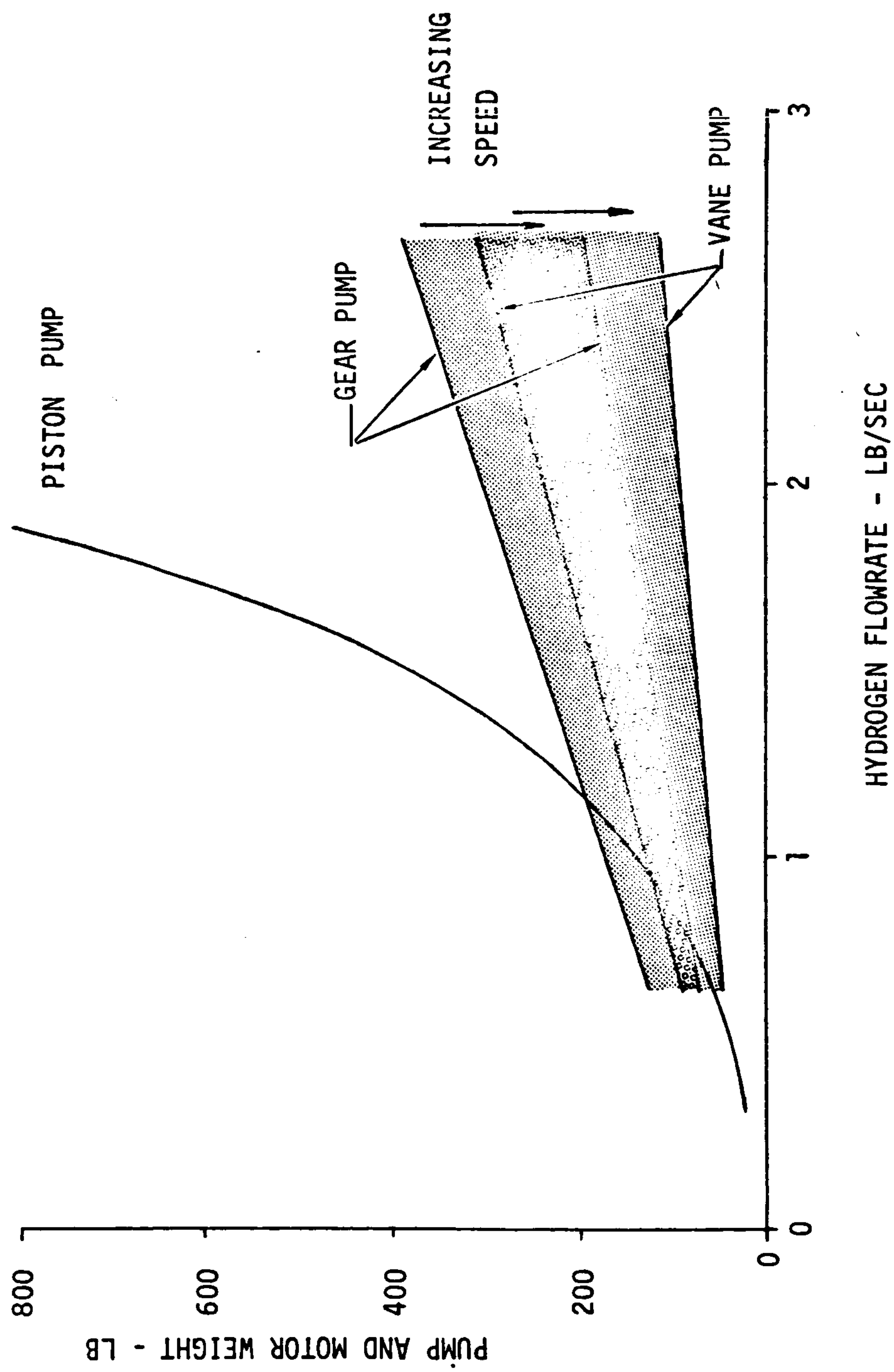
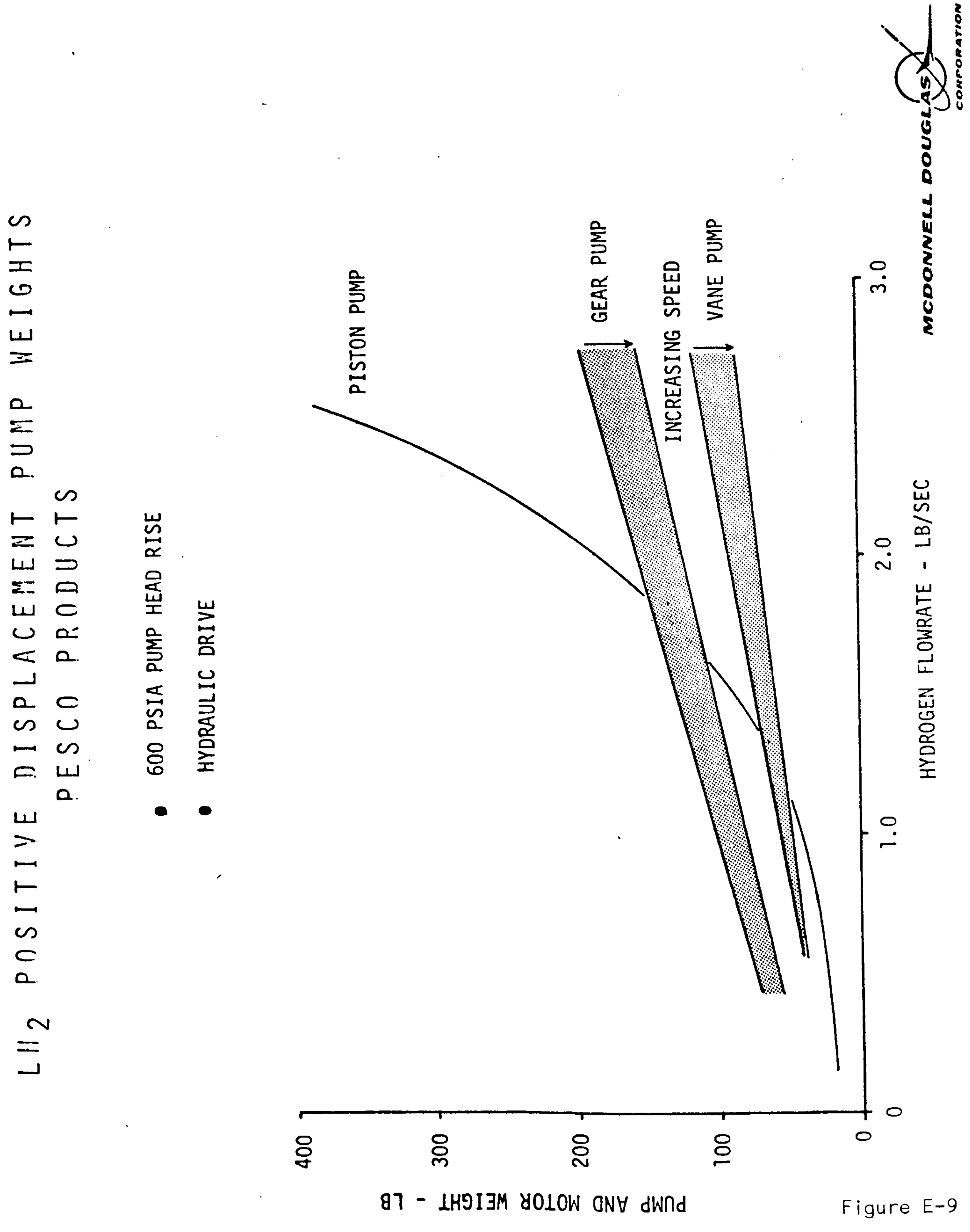


Figure E-8

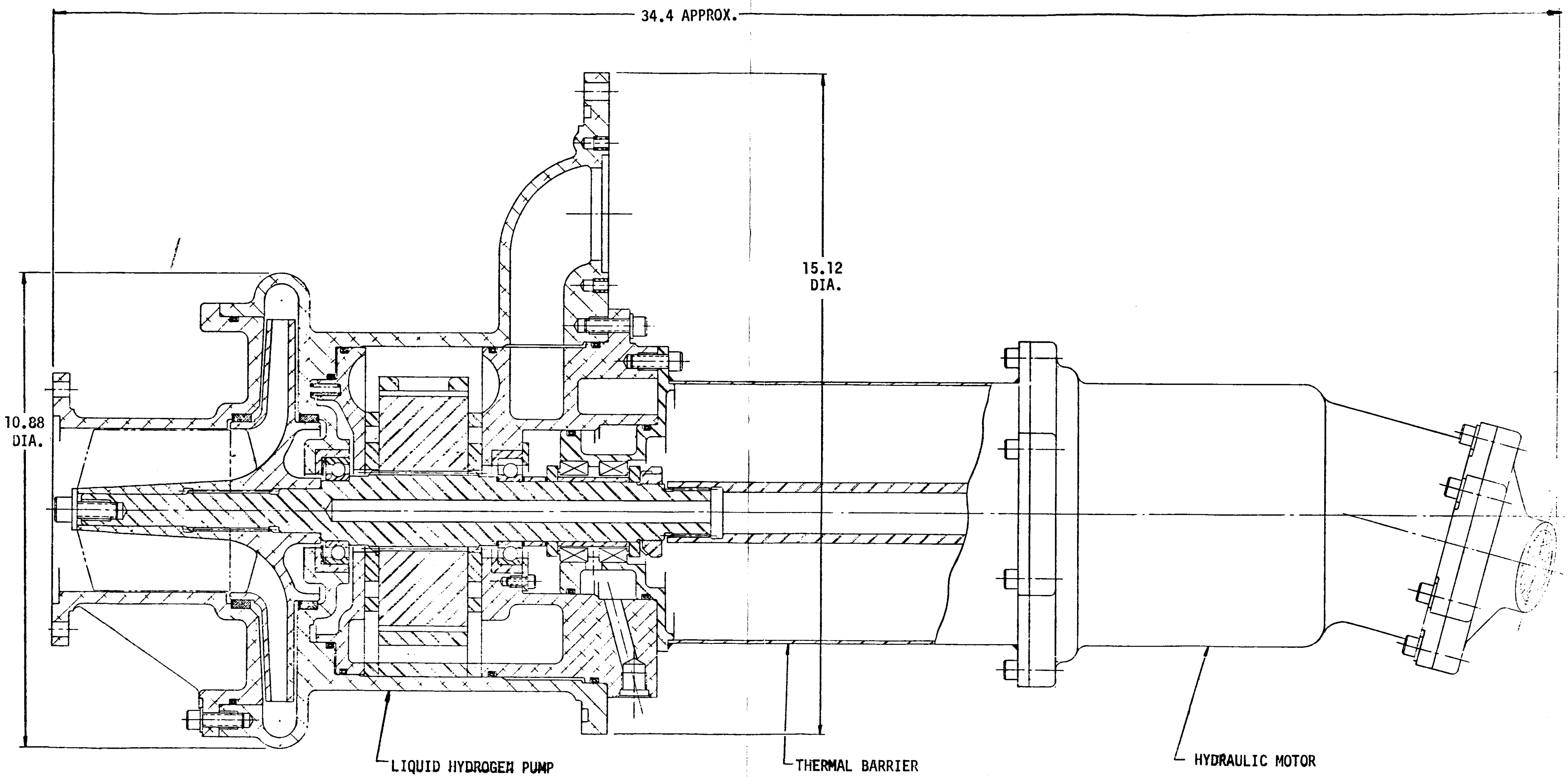
E243-115



MCDONNELL DOUGLAS CORPORATION

Figure E-9

VANE PUMP, HYDRAULIC DRIVE



PERFORMANCE: 2.65 LBS/SEC LH_2 @ 600 PSI RISE @ 4000 RPM

Figure E-10