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CENTAUR PROPELLANT ACQUISITION SYSTEM STUDY

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M. D. Walter

Prepared for
National Aeronautics and Space Administration
LEWIS RESEARCH CENTER
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16. Abstract

A study was performed to determine the desirability of replacing the hydrogen peroxide settling system on the Centaur D-1S with a capillary acquisition system. A comprehensive screening was performed to select the most promising capillary device fluid acquisition, thermal conditioning, and fabrication techniques. Refillable start baskets and bypass feed start tanks were selected for detailed design. Critical analysis areas were settling and refilling, start sequence development with an initially dry boost pump, and cooling the fluid delivered to the boost pump in order to provide necessary net positive suction head (NPSH). Design drawings were prepared for the start basket and start tank concepts for both LO_2 and LH_2 tanks. System comparisons indicated that the start baskets using wicking for thermal conditioning, and thermal subcooling for boost pump NPSH, are the most desirable systems for future development.

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PREFACE

This report was prepared by the Convair Division of General Dynamics Corporation under Contract NAS3-17802. The contract was administered by the National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio. The NASA Project Manager for the contract was Mr. John Aydelott. This is the final report on the contract, summarizing the technical effort from January 9, 1974 to May 25, 1975. Convair program manager was M. H. Blatt.

The contributions of the following individuals are gratefully acknowledged:

R. J. Conway	- Pressurization system analysis and capillary device thermal conditioning system screening
R. N. Ford	- Boost pump and propellant duct thermal conditioning analysis
E. Makela	- Reliability analysis
R. L. Pleasant	- Wicking, vibrations and start tank thermal conditioning analysis
M. D. Walter	- System design and weight estimates

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SUMMARY

Capillary acquisition systems were investigated for replacing the hydrogen peroxide propellant settling system on the Centaur. The Centaur D-1S, a Centaur designed to be compatible with Space Shuttle, was used as the baseline vehicle.

The study defined candidate integrated capillary acquisition and vent systems for the Centaur D-1S liquid hydrogen and liquid oxygen tanks. Detailed designs were prepared for the most promising systems. These designs were used for performance comparisons with the baseline peroxide settling system. A development plan was then prepared for the recommended capillary systems.

Initially, candidate concepts were investigated for accomplishing fluid acquisition, capillary device thermal conditioning and fabrication, boost pump thermal conditioning, and propellant duct thermal conditioning. These candidates were tabulated and compared and the most promising candidate recommended for additional study. A decision tree was formulated to rank the possible system combinations and to determine the critical decisions to be answered in determining the optimum system approach. The most desirable system was determined to be a start basket refilled by settled propellant using uncooled boost pumps and propellant ducts, with thermal subcooling for providing boost pump NPSH.

Work was then directed toward answering the critical questions: Can capillary device refilling be accomplished with settled propellant? Can a successful start sequence be accomplished with uncooled boost pumps? Can thermal subcooling (removing heat from the fluid flowing to the boost pump) be used to provide boost pump NPSH? These questions were successfully answered, allowing selection of the refillable start basket for detailed design. The other system selected was a bypass feed start tank.

Capillary device sizing was based upon volumetric requirements for the start sequence, propellant settling, thermal conditioning flow, initial ullage for the start tank, liquid required to suppress pullthrough (residuals), trapped vapor during refilling, and channel volume. Analyses were performed to determine the effect of vibrations, feedline startup and shutdown transients, filling, abort, propellant utilization, and draining on capillary device design. Wicking was briefly investigated for providing liquid flow for capillary acquisition device passive thermal conditioning.

Thermal subcooler sizing was performed using throttled tank fluid to cool the hot side fluid flowing to the boost pumps. Thermal conditioning of the start baskets was analyzed using an active cooling system employing cooling coils containing throttled vent fluid. Non-vented start tanks were devised using fiberglass honeycomb insulation to control pressure rise and cold helium pressurization to suppress boiling.

Eighteen design drawings and one isometric sketch were prepared for the LO_2 start basket, thermal subcooler and start tank, and LH_2 start basket, thermal subcooler and start tank. The drawings illustrate the locations of components, device contours, support arrangements, attachment points, and assembly requirements.

Comparisons were made between the baseline Centaur D-1S settling system and seven capillary device system options on the basis of reliability, hardware weight, payload penalty, cost, power requirements and flight profile flexibility. The most desirable capillary acquisition systems were determined to be passively thermal conditioned start baskets using thermal subcooling to provide boost pump NPSH. A development plan was prepared for passively cooled start baskets encompassing technology development, hardware fabrication, flight qualification, testing on a future Centaur flight.

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INTRODUCTION

The objective of this study was to define candidate integrated capillary acquisition and thermodynamic vent systems for the Centaur D-1S liquid hydrogen and liquid oxygen tanks. Detailed designs of selected acquisition systems were made to compare performance with the baseline system. The desirability of additional capillary device development was determined, and the scope of this development program was defined.

During low gravity coast, vehicle drag and disturbing acceleration may position propellant away from the tank outlet. Engine start under these conditions will cause vapor to enter the pumps, producing cavitation, poor engine operation, and possible feed system failure. To eliminate these undesirable occurrences, means must be provided to position liquid in the feedlines and over the tank outlet. The method currently used on Centaur is to settle the propellants by using small thrusters to apply a linear acceleration to the vehicle. This method, while well proven, imposes mission constraints in waiting for propellant to be settled and weight penalties which are a function of the number of engine burns. The use of a capillary or surface tension device to trap propellants over the outlet in low gravity is a more advanced but less proven technique. Weight penalty for the surface tension device is less sensitive to number of engine burns and provides added mission flexibility in allowing quick engine startup.

The capillary devices must perform the function of retaining propellants over the tank outlet for boost pump and engine startup. This study examined both the requirements of a cryogenic capillary acquisition system in performing this function and the interaction of the acquisition system with related vehicle systems.

The systems interacting with the acquisition system are shown in Figure 1-1. These systems are the pressurization system, vent system, propellant gaging system, main engines, boost pumps and propellant ducts.

Capillary acquisition systems fall into two main classes: partial acquisition devices such as start baskets or start tanks that rely upon fluid settling for refill; and "total" acquisition concepts such as liners or channels that cover a substantial portion of tank area and maintain continuous contact with the main liquid pool. A partial acquisition concept operates by maintaining liquid over the outlet in sufficient quantity to allow the main liquid pool to be settled. The settled liquid refills the acquisition device. During engine firing, but prior to main liquid pool settling, vapor enters the acquisition device volume. Capillary device geometry must be designed so that the entering vapor does not create adverse liquid spilling from the basket away from the engine outlet or cause difficulties in refilling the device with liquid. Total control devices are either maintained full of liquid during main engine burns or refilled between burns by capillary pumping, venting or mechanical pumping.

Thermal conditioning of the capillary device is a major design consideration. To maintain liquid over the tank outlet, propellant vaporization and bulk boiling within the acquisition device must be prevented. Vaporization can be caused by incident heating through the tank walls, heating from the engines, boost pumps, and feedlines, and by pressure fluctuations in the tank due to venting or pressurant cooling.

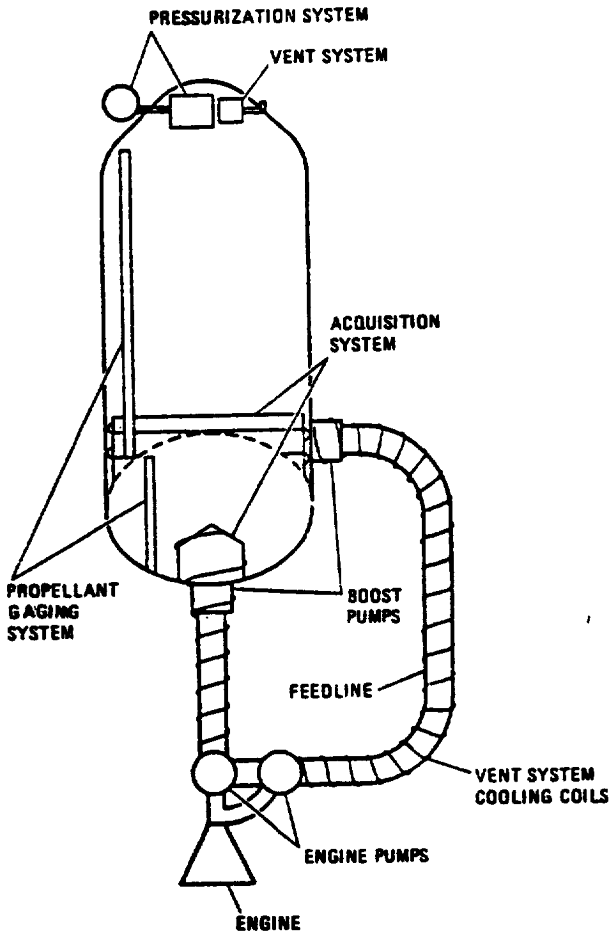


Figure 1-1. Centaur D-1S Acquisition System Interfaces

The pressurization system has a major interaction with the acquisition device. Since pressurization will be accomplished when the liquid is unsettled, the use of warm pressurant will cause rapid ullage pressure decay when the cold liquid is "settled" through the pressurant. Cold pressurant should be used in lieu of warm pressurant to alleviate this problem.

The vent system influences the acquisition system design by causing forced convection heat transfer to occur at the basket surface and by causing tank pressure reductions that could drop the saturation temperature of the tank below the acquisition device surface temperature. The primary candidate for thermally conditioning the capillary device is to use throttled vent fluid for cooling. In order to have sufficient cooling capacity to thermally condition the contained fluid, liquid is supplied from the capillary device to the inlet of the vent system cooling loop.

Propellant utilization systems such as the capacitance gaging technique used on the Centaur D-1S cannot sense any liquid trapped in the capillary device above the settled liquid. Means must be provided for either separately sensing this trapped liquid or empirically verifing analytical predictions of the trapped liquid quantity.

A primary consideration of the study was the interaction of the boost pump and propellant ducts with the capillary device. The method of thermal conditioning the boost pumps and ducts directly affects feed system chilldown and capillary device volumetric requirements. Methods of supplying boost pump NPSH were a major concern in studying pressurization system alternatives. Feed system startup and shutdown transients may influence acquisition system retention requirements.

Engine soakback heating contributes to feed system chilldown requirements. Engine vibrations may induce capillary device vibrations that cause loss of retention capability.

Since the acquisition system interacts with many other systems in the vehicle, comparison of acquisition systems cannot be done by merely looking at the acquisition device alone. Considerations must be given to all changes to the vehicle caused by the particular acquisition system being implemented.

1.1 GROUND RULES

The baseline vehicle configuration for this study is the Centaur D-1S as defined in Contract NAS3-16786 and reported in NASA CR-134488, (Reference 1-1). The Centaur D-1S is a minimum change D-1T configuration, modified to be compatible with the Space Shuttle interface, operations and safety requirements. Approximately 95% of the existing D-1T components remain unchanged for the D-1S. Figure 1-2 illustrates modifications made to the existing D-1T to evolve to D-1S, D-1S(R) (reusable Centaur D-1S) and RLTC (Reusable Large Tank Centaur) configurations. (These were the advanced Shuttle integrated Centaur versions existing at the initiation of the study). Several of the changes to the D-1T, as noted below, may affect the deployment of a capillary acquisition system on the D-1S.

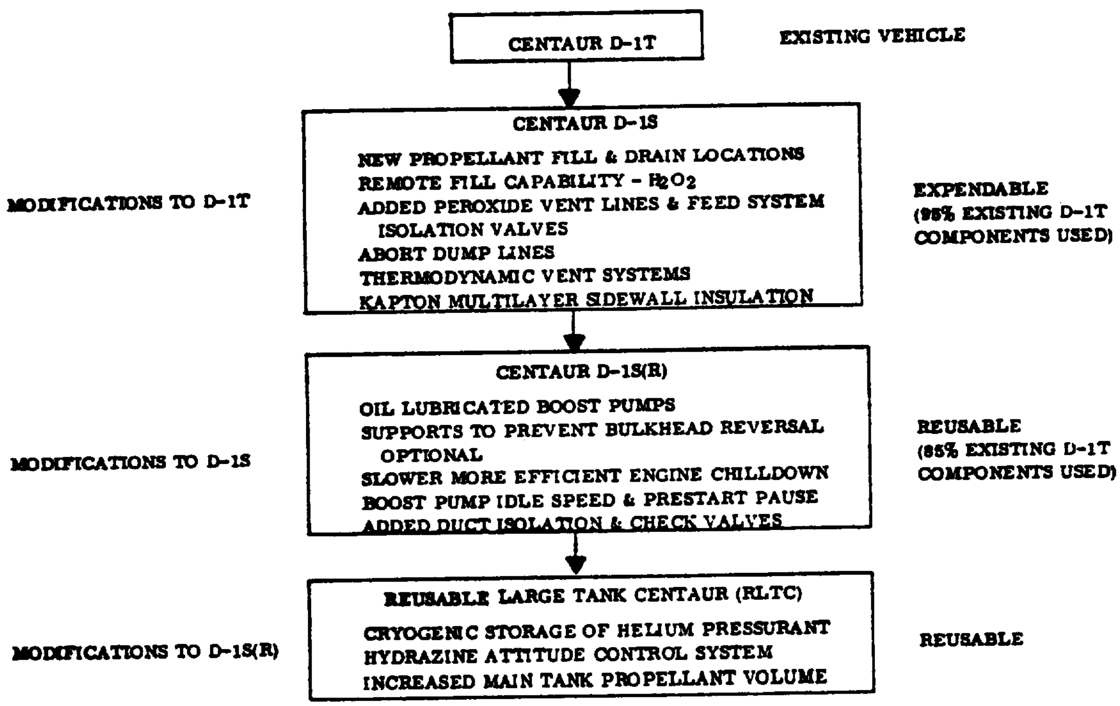
The propellant fill and drain system revision is necessitated by relocation of the fill and drain disconnects from the tank skin line to the aft umbilical panel. Line sizes were increased to 3.5 inches (8.89 cm) for LO_2 and 4.25 inches (10.8 cm) for LH_2 in order to accommodate the 300 second abort dump requirement. A zero g vent system was incorporated in each propellant tank due to the Centaur requirement for tank venting in low-g while in the Shuttle payload bay. The sidewall insulation system consists of two layers of double-aluminized Kapton with a fiberglass scrim spacer. Isolation valves have been added to the peroxide vent and feed system lines for system safing in the orbiter. Remote peroxide fill capability was also added.

Mission profiles for the study were the planetary, synchronous equatorial and low earth orbit flight profiles of NAS3-16786, (Ref. 1-1), as given in Tables 1-1, 1-2 and 1-3.

Heating rates, nominal tank pressure levels, and other mission conditions were obtained from NASA-CR-134488 (Ref. 1-1). Parameters not specified were generated using analytical or empirical techniques consistent with the design of the Centaur D-1T and D-1S.

The baseline D-1S thermodynamic vent systems consist of coiled tube heat exchangers, pump/mixers, shutoff valves, regulators, and filters. The study evaluated modifications to this system required to thermally condition the contained fluid. This entailed additional cooling loops in parallel with the main bulk heat exchanger.

The integrated propellant acquisition and thermodynamic vent systems designed in this study provide pressure control, both within the Orbiter payload bay and during the flight mission profiles, for the same environmental conditions used in Contract NAS3-16786, Ref. 1-1. The propellant acquisition and thermodynamic vent systems designed in this study neither impose constraints on the operation of the Shuttle nor affect the Centaur/Shuttle abort compatibility.



OTHER MODIFICATIONS CONSIDERED WERE ELECTRIC DRIVEN BOOST PUMPS FOR D-1S(R) AND RLTC AND THE POSSIBILITY OF USING PPO FOAM INSULATION TO REDUCE HEAT INPUT AT THE INTERMEDIATE BULKHEAD/LH_2 SIDEWALL INTERSECTION.

Figure 1-2. Evolution of Centaur D-1T to Future Centaur/Shuttle Vehicles

Table 1-1. Planetary Mission Profile

Event/Time (min.)	Initial Mass lb_m (kg_m)	Burn Time, sec	Propellant Burned, lb_m (kg_m)	Final Mass lb_m (kg_m)	Initial Percent Full *	Initial Acceleration, g
Loading	25,450 (11554) LO_2					
(T = 0)	5,279 (2397) LH_2					
MES1	25,304 (11488) LO_2	441.4	24,894 (11302) LO_2	410 (186) LO_2	96	0.61
(T = 67)	5,164 (2344) LH_2		4,941 (2243) LH_2	223 (101) LH_2	95	
	49,413 (22434) Vehicle			19,578 (8888) Vehicle		

Main engine thrust 30,000 lb_f (13620 kg_f). *Assumes 26,313 lb_m (11946 kg_m). LO_2, 5,459 lb_m (2478 kg_m). LH_2 for full tank, Maximum ACS thrust = 24 lb_f (10.9 kg_f). - Maximum ACS acceleration before last burn = 4.86×10^{-4} g's, Main engine flow rates - LO_2 = 56.4 lb/sec (25.6 kg/sec), LH_2 = 11.2 lb/sec (5.1 kg/sec), ISP = 443.82 sec, Payload = 14,465 lb_m (6567 kg_m). Dry weight = 4,439 lb_m (2015 kg_m). Burnout acceleration = 1.53 g's.

Table 1-2. Synchronous Equatorial Mission Profile

Event/Time (min.)	Initial Mass, lb_m (kg_m)	Burn Time, sec	Propellant Burned, lb_m (kg_m)	Final Mass lb_m (kg_m)	Initial Percent Full*	Initial Acceleration, g
Loading	25,450 (11554) LO_2					
(T = 0)	5,279 (2397) LH_2					
MES1	25,304 (11488) LO_2	305.4	17,299 (7854) LO_2	8,005 (3634) LO_2	96	0.63
(T = 67)	5,164 (2344) LH_2		3,360 (1525) LH_2	1,804 (819) LH_2	96	
	47,447 (21541) Vehicle			26,653 (12100) Vehicle		
MES2	7,975 (3621) LO_2	132.3	7,496 (3403) LO_2	419 (190) LO_2	30.3	1.12
(T = 384)	1,723 (782) LH_2		1,465 (665) LH_2	258 (117) LH_2	31.6	
	26,788 (12162) Vehicle			17,525 (7956) Vehicle		

Main engine thrust 30,000 lb_f (13620 kg_f). *Assumes 26,313 lb_m (11946 kg_m). LO_2, 5,459 lb_m (2478 kg_m). LH_2 for full tank, Maximum ACS thrust = 24 lb_f (10.9 kg_f). Maximum ACS acceleration before last burn = 8.96×10^{-4} g's, Mixture ratio = 5.0, Main engine flow rates - LO_2 = 56.65 lb/sec (25.7 kg/sec), LH_2 = 11.03 lb/sec (5.01 kg/sec), ISP = 443.35 sec, Payload = 12,199 lb_m (5538 kg_m). Dry weight = 4,604 lb_m (2090 kg_m). Burnout acceleration = 1.71 g's.

Table 1-3. Low Earth Orbit Mission Profile

Event/Time (min.)	Initial Mass lb_m (kg_m)	Burn Time, sec	Propellant Burned, lb_m (kg_m)	Final Mass lb_m (kg_m)	Initial Percent Full*	Initial Accel. g
Loading	25,450 (11554) LO_2					
(T = 0)	5,279 (2397) LH_2					
MES1	25,304 (11488) LO_2	88.6	5,052 (2294) LO_2	20,252 (9194) LO_2	96	0.71
(T = 67)	5,164 (2344) LH_2		955 (434) LH_2	4,209 (1911) LH_2	95	
	42,048 (19090) Vehicle			36,042 (16363) Vehicle		
MES2	20,165 (9155) LO_2	191.32	10,915 (4955)	9,250 (4200) LO_2	77	0.84
(T = 118)	4,153 (1885) LH_2		2,060 (935) LH_2	2,093 (950) LH_2	76	
	35,824 (16264) Vehicle			22,849 (10373) Vehicle		
MES3	9,167 (4162) LO_2	120.51	6,875 (3121) LO_2	2,286 (1038) LO_2	35	1.33
(T = 408)	2,010 (913) LH_2		1,294 (587) LH_2	716 (325) LH_2	37	
	22,568 (10246) Vehicle			14,397 (6536) Vehicle		
MES4	2,198 (998) LO_2	18.90	1,078 (489) LO_2	1,121 (509) LO_2	8.4	2.11
(T = 459)	650 (295) LH_2		204 (93) LH_2	456 (207) LH_2	11.91	
	14,192 (6443) Vehicle			12,910 (5861) Vehicle		
MES5	1,031 (468) LO_2	10.8	614 (279) LO_2	417 (189) LO_2	3.9	2.36
(T = 553)	393 (178) LH_2		116 (53) LH_2	277 (126) LH_2	7.2	
	12,698 (5765) Vehicle			11,967 (5433) Vehicle		

Main engine thrust = 30,000 lb_f (13620 kg_f). *Assumes 26,313 lb_m (11946 kg_m), LO_2, 5,459 lb_m (2478 kg_m), LH_2 for full tank. Maximum ACS thrust = 24 lb_f (10.9 kg_f), Maximum ACS acceleration before 5th burn = 1.89×10^{-3} g's, Mixture ratio = 5.298, Main engine flow rates - LO_2 = 57.05 lb/sec (25.9 kg/sec), LH_2 = 10.77 lb/sec (4.89 kg/sec), ISP = 443.8 sec, Payload = 6260 lb_m (2842 kg_m), Dry weight = 4901 lb_m (2225 kg_m), Burnout acceleration = 2.51 g's.

2

TASK I, CANDIDATE SYSTEMS

The objective of the screening of candidate systems was to identify possible methods of accomplishing capillary propellant acquisition for the Centaur D-1S and to evaluate these systems based on weight, feasibility, and operational advantages to determine which candidates compare favorably to the baseline hydrogen peroxide system.

In determining candidate systems for acquisition, concepts for capillary device fluid containment, pressurization, thermal conditioning, structure and assembly, and boost pump and feedline thermal conditioning were considered separately. Initially, all possible means of satisfying mission and vehicle requirements were identified for each of the concept categories. Each fluid acquisition system candidate was conceptually designed to meet Centaur D-1S mission requirements and was then evaluated based on approximate system weight and operational advantages compared to the existing peroxide system. Candidates were screened only to the point at which they could be logically rejected. For example, if a system could not be conceptually designed to meet Centaur D-1S requirements, it was eliminated without determining system weight. Further, if system weight exceeded existing system weight by more than 20%, the concept was rejected. If the concept still remained as a candidate, then operational advantages or disadvantageous compared to the existing system and to other candidate acquisition systems were assessed.

Thermal conditioning and pressurization candidates were compared based on relative advantages and disadvantages, complexity, and weight. Promising fabrication alternatives were determined for screen joining, screen-to-backup material joining, backup material selection, barrier material selection, load support and cooling tube attachment.

Fluid acquisition, thermal conditioning, pressurization, and fabrication candidates were combined into three system candidates requiring additional study. These candidates were; refillable start baskets, bypass feed start tanks and channels refilled by pumping. The work required to assess the feasibility of these candidates was mapped on a decision tree in order to most efficiently allocate program resources. The decision tree indicated that work in Tasks II, III and IV should concentrate on refillable start baskets and bypass feed start tanks.

2.1 BASELINE SYSTEM

In order to make weight comparisons, a preliminary assessment of the baseline Centaur D-1S system equivalent payload weight penalty was made. The payload sensitivity factors given in Table 2-1 were used to generate the equivalent payload weight shown in Table 2-2 for the low altitude five burn mission. The weight penalty includes propellant settling, feedline conditioning, main tank venting and tank pressurization.

Propellant settling is accomplished by the H_2O_2 auxiliary propulsion system. The settling force is provided by the four axial H_2O_2 engines which are used only for settling. Settling imposes a large demand on auxiliary propulsion system weight for the multiburn missions. In addition to settling, the auxiliary propulsion system provides attitude control and drives the boost pump turbines. The total auxiliary propulsion system hardware weight is 99 pounds (44.95 kg) for the planetary (one-burn) mission, 153 pounds (69.5 kg) for the geosynchronous (two-burn) mission, and 207 pounds (44 kg) for the low-altitude (five-burn) mission. The total fluid weights, for all peroxide usage, are 130 pounds (59 kg), 256 pounds (116 kg) and 472 pounds (214 kg) for the respective missions.

Propellant settling thrust is also provided during inflight engine chilldown. Prior to engine start, liquid propellant is driven by the boost pump through the feedlines and turbopumps. The resulting cooldown prevents excessive propellant heating (and resulting vaporization) during the pump acceleration transients.

Table 2-1. Centaur D-1S Payload Sensitivity Factors

Criteria		Mission		
		Planetary	Sync. Equatorial	Low Altitude
Jettison Weight		-1.000	-1.000	-1.000
Propellant Weight		+0.646	+0.587	+0.420
LH_2 and LO_2 Loss Before Burn (Without I_{SP} Effect)	No. 1	-0.646	-0.587	-0.420
	No. 2		-0.946	-0.510
	No. 3			-0.771
	No. 4			-1.215
	No. 5			-1.337
LH_2 and LO_2 Lost After Last Burn (Residual or RFP)		-1.646	-1.587	-1.419
Auxiliary Propellant Used Prior to Burn	No. 1	0	0	0
	No. 2		-0.459	-0.072
	No. 3			-0.352
	No. 4			-0.796
	No. 5			-0.917
Auxiliary Propellant Used After Last Burn (Residual)		-1.000	-1.000	-1.000

Table 2-2. Baseline Acquisition System Weight Penalties (Low Earth Orbit Five Burn Mission)

Weight Penalty Element	Actual Weight lbs (kg)		Equivalent Payload Weight lbs (kg)	
Peroxide System Hardware Weight				
Settling Motors	23.0	(10.44)	23.0	(10.44)
H_2O_2 Bottle	54.0	(24.52)	54.0	(24.52)
Peroxide System Fluid Weight	214.0	(97.2)	75.8	(34.41)
Pressurant for Peroxide	0.421	(0.2)	0.3	(0.14)
Residual Peroxide	12.0	(5.45)	12.0	(5.45)
Peroxide System Total			165.1	(75)
Vent System Hardware Weight	62.0	(28.1)	62.0	(28.1)
Vented Propellant				
LO_2	44.0	(20)	18.5	(8.4)
LH_2	44.4	(20)	26.8	(12.2)
Vent System Weight Penalty			107.3	(48.7)
Pressurization System Hardware Weight	417.0	(189.3)	417.0	(189.3)
Pressurant Used for Main Engine Burns	13.54	(6.15)	13.54	(6.13)
Pressurization System Weight Penalty			430.54	(195.45)
Total Existing System Payload Penalty			703.0	(319.2)

Sufficiently subcooled liquid is provided at the boost pumps during this period by a 3 psi (20.7 kN/m^2) pressurization of the propellant tanks. This pressurization sufficiently condenses bubbles formed in the boost pump sumps such that boost pump cavitation is prevented.

Pressurization system weight is therefore directly affected by start system design. The D-1S pressurization system consists of the helium pressurant, stored in bottles at high pressure and ambient temperature, along with valves, regulators, plumbing, sensors, and harnesses. Pressurization system dry weights are 257 pounds (117 kg) for the planetary mission, 300 pounds (136 kg) for the geosynchronous mission and 417 pounds (189 kg) for the low earth orbit mission. For the worst case five burn low earth orbit mission, the helium requirement is 14 lbm (6.4 kg).

For the baseline system, the propellant settling function accounts for approximately 77 pounds (35 kg) of hardware weight. This includes settling motors, associated portions of the cluster assembly, H_2O_2 bottle supports, pressurization line and propellant lines. In addition to the H_2O_2 system weight, 214 pounds (97 kg) of H_2O_2 and 0.3 pounds (0.14 kg) of helium are expended during the five-burn mission and 12 pounds (5.4 kg) of residual peroxide are loaded into the third bottle (added to the acquisition function).

The baseline D-1S configuration also includes the thermodynamic vent systems which maintain thermally destratified propellant tanks. This system is required for all missions because of the extreme pressure rise rates experienced in small stratified ullages such as exist during the initial coast. Additionally, it is difficult to position a small ullage bubble over the vent when the tanks are 95% full or greater. Also means must be provided for venting the Centaur while in the cargo bay of Shuttle without disturbing the Shuttle. LO_2 vent system weight and power are 29 pounds (13 kg) and 80 watts. LH_2 vent system weight and power are 33 pounds (15 kg) and 12 watts.

Baseline system weights were determined for the auxiliary propulsion acquisition function, the LO_2 and LH_2 tank vent systems, and the pressurization system requirements both for the auxiliary propulsion and main propulsion systems. Hardware weight and fluid expended were translated into equivalent payload penalty using the payload sensitivity factors shown in Table 2-1. Since outflow requirements and resulting pressurization system weights are greater for the five burn, low earth orbit mission, this mission is used for establishing maximum acquisition system payload penalties.

Payload sensitivity factors for pressurant are considered to be one. This assumes that no pressurant is vented during the mission.

The Centaur D-1S baseline acquisition and vent system equivalent payload weight penalties are shown in Table 2-2.

2.2 FLUID CONDITIONING CANDIDATES

Possible capillary acquisition device candidates for satisfying Centaur D-1S requirements were conceptualized. These concepts were compared based on their advantages and disadvantages as shown in Table 2-3. Adoption or rejection recommendations are given for each of the concepts considered. Characteristics that could be advantageous to the system are: refillable between burns, low development requirements, ground testability, and low weight (including pressurization). Conversely, characteristics that could be disadvantageous to the system are: high weight, need for orbital experiment to prove concept, reliance upon settling for refill, requirement for separate pressurization system, difficulty in ground checkout and requirement for moving parts.

As indicated in Section 2.0, concepts were retained for additional study if they did not exceed the existing system weight of 703 lbs by more than 20%. The lowest weight system using a pressurization system was the bypass feed start tank with a weight of 709 lb_m (321.9 kg) The primary reason for the low weight of this system was the lower pressurization system weight of the start tank system compared to the baseline system and the other capillary acquisition systems. Other systems near the acceptable weight range were the basic start basket 867 lbs (393.6 kg) and the channels refilled by pumping concept 930 lbs (422.22 kg). Even though these systems were slightly outside the weight limit they had the potential for using thermal subcooling for providing pressurization system weight reduction compared to using cold pressurant. (Thermal subcooling is discussed in Section 2.3). They were therefore retained for additional study. The weight comparisons show that pressurization system weight was a significant factor in total system weight.

Caution should be used in making weight comparisons between systems listed in table 2-3 since some systems have LO_2 vent systems and others use LH_2 boiloff to cool the LO_2 tank. The type of vent system used is noted in each weight breakdown. The integrated LH_2/LO_2 vent system (No LO_2 venting) has a weight savings of 51.5 lbs (23.4 kg) (107.3 lbs (48.7 kg) to 55.8 lbs (25.3 kg)) compared to using separate LO_2 and LH_2 vent systems.

Data used for developing the system weights are capillary device retention, outflow, and thermal conditioning requirements determined in a manner similar to that in NAS8-21465 (Ref. 2-1 and 2-2). Equations 2-1 and 2-2, generated for a maximum g level of 2.52 g's at the end of burn 5 of the low earth orbit mission with a safety factor of 2, were used to determine retention requirements for LH_2 and LO_2.

$$LO_2\ ,\ h = \frac{1.86}{D_{BP}}\ ,\ \text{feet (2-1)},\quad LH_2,\ h = \frac{4.85}{D_{BP}}\ ,\ \text{feet (2-2)}$$

Table 2-3. Conceptual Fluid Containment Candidates

Candidate Concept	Operations & Design Features	Characteristics	Advantages/Disadvantages, Recommendations
1. Start baskets - refilled by settled fluid	Screened enclosures on LO_2 and LH_2 outlet, refilled through side screens. LH_2 device channels and compartments for delivering thermal conditioning flow to chilldown the engines and condition the feedlines and capillary devices, LO_2 basket has screen liner for this purpose. Relies upon fluid settling to refill the basket (channels remain filled during the entire mission) using a standpipe to minimize trapped vapor during refilling. Device is directly influenced by main tank pressurization. Devices are spaced off the wall to eliminate direct heat input from the wall to the capillary device. This allows liquid to spill out of the basket when screen retention limits are exceeded during main engine burns. Cryogenic helium pressurant is needed to pressurize unsettled propellant.	Completely passive. Thermodynamic vent system cooling used to condition capillary device and feedlines. 200×600 mesh pleated pullthrough suppression screens. Equivalent payload weight penalty, using LO_2 and LH_2 venting, is 867.1 lbm (393.7 kg). Using LH_2 vent fluid to thermally condition the LO_2 tank reduces the payload penalty to 815.6 lbm. LH_2 device is 30 ft^3 (0.849 m^3) and 31.25 in. (0.79 m) high. LO_2 device is 6.56 ft^3 (0.186 m^3) with the top 8 in. (0.203 m) above the top of the sump.	System weight penalty is higher than baseline system weight penalty of 703 lbm (318 kg). Baseline system with no LO_2 venting would have an equivalent payload penalty of 651.5 lbm (296.8 kg). System relies upon fluid settling for refill. Passive. Compact. Relatively easy to assemble. Can be used with thermal subcooling to potentially reduce pressurization system weight by 540 lbs. Recommended.
2. Start basket - nonrefillable	Similar to above only it contains sufficient propellant for all main-engine burns plus thermal conditioning. Full screen liner used to contain liquid within the basket for outflow and thermal conditioning. Liquid cannot spill from device because multiple screens are utilized to provide retention during main engine burns. Device, thus, does not have to be refilled by settling.	LO_2 volume is 27.5 cu ft (0.78 m^3) LH_2 volume is 150 cu ft (4.25 m^3) LH_2 device ht = 46 in. (1.68 m) LO_2 device ht = 33 in. (0.84 m) Multiple screens and increased volume increase payload penalty to: Acquisition system, 379 lb (171.9 kg) LH_2 vent system wt 55.8 lb (25.3 kg) (No LO_2 venting) Pressurization system weight, 599.8 lbm (272.3 kg) Total payload weight penalty, 1034.6 lbm (469.7 kg)	System does not have to be refilled. System weight greatly exceeds existing system weight. Multiple screen liner technique is not well proven. Rejected.
3. Start basket - removable	Device is designed to be small enough to allow installation and removal from the tank through normal tank access provisions.	Not applicable to LH_2 tank because tank must be built around the device. For the LO_2 tank with a sump, the device could be made small enough to fit, however the resulting device height for the 5.5 cu ft (0.16 m^3) volume would exceed the liquid level in the tank during the final burns making it impossible to refill.	Operationally unfeasible. Rejected.
4. Start basket - with refill valve	To prevent vapor from being trapped by wetted screens during device refill, refill valves can be employed to vent vapor and allow quicker and more efficient refill. Refill valves are opened during engine outflow and closed before outflow ceases.	Valves weigh 10 to 15 lb (4.54 to 6.80 kg) per tank in addition to Concept 1 weights. Main engine burns are sufficiently long and have high enough accelerations to minimize trapped vapor by using small standpipes.	Nonpassive. Does not provide significant operational advantage while weighing 20 to 30 lb (9 to 13.6 kg) more than Concept 1. Rejected.

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Table 2-3. Conceptual Fluid Containment Candidates (Continued)

Candidate Concept	Operations & Design Features	Characteristics	Advantages/Disadvantages, Recommendations
5. Start basket - multiple compartment	Compartments are added to Concept 1 to contain all the propellant in the tank after the first three engine burns on the five-burn mission. The intermediate bulkhead is used as a wall for the LH_2 capillary device to prevent spilling during the last two burns.	Acquisition system weight LH_2 160.0 lb (72.6 kg) LO_2 60.0 lb (27.2 kg) Vent system penalty 55.8 lb (25.3 kg) Pressurization penalty 599.8 lb (272.1 kg) Total payload penalty 875.6 lb (397.5 kg)	Reduces dependence upon settling a small amount of propellant with high accelerations. Weight and difficulty of assembly are higher than Concept 1. Rejected.
6. Screen liners for total control 	Liners, covering the entire tank surface with screen material, remain completely full during the entire mission by employing multiple screen barriers. Liner must operate at up to 2.5 g's over essentially full tank head. Thus many layers of screen are required to obtain sufficient retention. LH_2 - 42 layers, LO_2 - 38 layers. Single screen liner that might be utilized for venting between burns, weighs 96 lbm (44 kg) for LO_2 and 284 lbm (129 kg) for LH_2 making it too heavy to be practical.	LH_2 acq. sys. wt. 11658.0 lb (5288.0 kg) LO_2 acq. sys. wt. 3352.0 lb (1611.0 kg) Vent system payload penalty (LH_2 only) 55.8 lb (25.3 kg) Pressurization payload penalty 599.8 lb (272.1 kg) 15665.6 lb (7196.7 kg)	Theoretically can maintain liquid outflow during any portion of each mission. Difficulties in preventing gas from entering outlet from between the wetted screens. Excessive weight. Rejected.
7. Channels for total control	Similar to Concept 6 only uses channels that cover one quarter of the tank area. Concept is easier to thermally condition but is not as good fluid dynamically as Concept 6. Screen surface area is lower, thus weight will be lower than Concept 6.	LH_2 acq. sys. wt. 4805.0 lb (2179.5 kg) LO_2 acq. sys. wt. 1463.0 lb (663.6 kg) Vent system - payload penalty (LH_2 only) 55.8 lb (25.3 kg) Pressurization payload penalty 599.8 lb (272.1 kg) 6923.6 lb (3140.6 kg)	Same as 6. Rejected.
8. Start basket (or start tank) - multiple screen layer	Similar to Concept 1, instead of small compartments with channels providing thermal conditioning flow, a liner is used extending to the top of the basket. The liner remains full during main-engine burns. A multiple liner is required for the LH_2 tank. The LO_2 configuration is identical to Concept 1. Configurations are designed to eliminate internal compartment.	LH_2 weight, for three screen layers required, increases by 88 lb (40 kg) compared to Candidate Concept 1 making total payload penalty equal to 899.6 lb (408.1 kg)	Does not have spilling problem which affects Concept 1, but conversely is more difficult to refill. Multiple liner concept, relies upon multiple liquid/vapor interface formation with resultant possible gas passage during outflow. System weight exceeds start basket weight. Rejected.
9. Capillary sponge	Concept uses fine mesh porous material to fill the tank creating a true total control capillary device. Acceptable pore size is around 0.25 μ or lower. Acceptable density is about 0.25 to 0.50 lb/cu ft (4 to 8 kg/m³). Supplier survey made to determine if material is available. Alternate concept considered use of thin sheets of porous material of 0.25μ bubble point in place of fine mesh screen.	No material is light enough with a fine enough structure. Pressure drops are also too high for thin sheets of material to operate properly.	Has potential of providing retention equivalent to multiple screen liner. Operationally unfeasible with existing materials. Rejected.

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Table 2-3. Conceptual Fluid Containment Candidates (Continued)

Candidate Concept	Operations & Design Features	Characteristics	Advantages/Disadvantages, Recommendations
10. Capillary device - refillable between burns by capillary pumping	Uses capillary pumping between main engine burns to refill capillary device such as start basket. Concept requires a large tapered capillary collector, check valves to prevent liquid from being displaced by disturbing accelerations and long periods with no disturbing accelerations (2000-3000 seconds) to accomplish refilling.	Weight exceeds existing system weight. Disturbing accelerations from ACS cannot be preprogrammed to allow long refilling period.	Operational uncertainties. High weight. Does not rely upon settling. Rejected.
11. Capillary device - refillable by thermodynamic venting	Concept uses a thermodynamic vent system attached to a screen liner or channel device (single screen layer). During main-engine firing most of the liquid spills from the screen device because retention is exceeded. After waiting in low gravity for wicking and wetting to reseal the screens with liquid, the thermodynamic vent device is actuated and the capillary device volume is vented overboard until liquid replaces all vapor in the device.	Based on venting one capillary device volume between burns, calculations indicate that excessive fluid venting would be required by this concept. Vented fluid weight penalties could exceed 1000 lb (454 kg) if liquid in the device is around 50% of the total volume. Hardware weight penalty is near the existing system weight. Total system payload penalty is between 1700 lb (771.1 kg) and 2800 lb (1270 kg).	Does not rely upon fluid settling for refilling. High weight penalty due to fluid vented. Rejected.
12. Capillary device - refillable by pumping 	As an alternate to Concept 11, rather than venting the capillary device volume, a small pump is installed to pump the fluid back into the main pool of fluid. Both channel devices with a start basket and channel devices alone were examined. Channels can be made small enough so that volume remaining in tank during last burn is sufficient to refill device.	Vented mass penalty of Concept 11 is eliminated. System payload penalty using a channel and basket with venting of both tanks is 1081 lbs (491 kg). Equivalent weight penalty for the minimum weight system employing only LH_2 venting and no baskets (only channels) is 878lb (398kg).	Does not rely upon fluid settling for refilling. Requires additional flow analysis. Competitive weight (Thermal subcooling can reduce weight to below baseline system.) Recommended.
13. Start tank a. Through feed system 	This concept uses a separate tank to contain propellants for restarting the main engines. Channels within the tanks provide continuous liquid outflow for thermal conditioning. Start tank is pressurized prior to main engine burn, engine outflow valve is opened and flow to engines commences. Flow to engines provides thrust, settling propellants. When main tank propellants are settled, main tank is pressurized and valve between main tank and start tank is opened. Start tank vent valve is opened for refilling.	Concept requires several additional valves compared to the start basket of Concept 1. Requires main tank and start tank pressurization. Press. sys. wt. penalty 1057.0 lb (479.9 kg) Vent sys. wt. penalty (LH_2 venting only) 55.8 lb (25.3 kg) LH_2 acquisition device 74.0 lb (33.6 kg) LO_2 acquisition device 40.0 lb (18.1 kg) LH_2 start tank 60.0 lb (27.2 kg) LO_2 start tank 30.0 lb (13.6 kg) Valves 50.0 lb (22.7 kg) 1366.8 lb (620.5kg)	Not completely passive. Relies on fluid settling for refill. Too heavy. Rejected.

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Table 2-3. Conceptual Fluid Containment Candidates (Continued)

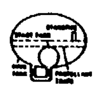
Candidate Concept	Operations & Design Features	Characteristics	Advantages/Disadvantages, Recommendations
13. b. Bypass feed system 	Same as a. except after propellant is settled, main engine flow bypasses the start tank through valve 1 (and valve 2 is closed). Valve 3 is opened for start tank refilling after start tank is vented below main tank pressure (through valve 4). Alternate concept does not vent start tank but refills tank from high pressure fluid downstream of boost. This would require additional plumbing but vent valve and start tank venting could be eliminated. Boost pump NPSH for main tank flow is provided by the head of the settled propellant.	Requires an additional valve (or a three-way valve instead of a two-way) but eliminates need for main tank pressurization. Start tank venting required for basic concept. Acq. device, main tank 259.8 lb (117.9 kg) Vent sys., start tanks (same as 13a) Valving 74.0 lb (33.6 kg) Press. penalty 324.0 lb (147.1 kg) 657.8 lb (298.7 kg) Bypass feed start tank weight with an LO_2 vent system is 709.3 lb (322.0 kg).	Lower weight penalty than existing system when no LO_2 venting is employed. Nonpassive-valving required. Relies upon fluid settling for refill. Possible flow transient problem in cycling between valves 1 and 2. Recommended.
14. Start tank with vacuum refill 	Similar to Concept 13b, except start tank is vented to vacuum between burns and refilled from main tank propellants positioned near the start tank. Requires additional acquisition provisions. This concept is useful for missions and vehicles where many short burns occur depleting the start tank without refilling. Essentially a start basket and a start tank are used in series.	Additional acquisition system weight for main tank. Vented propellant penalty. Unnecessary liquid venting occurs.	Low gravity refill. Additional venting required compared to 13b. Acquisition system required near start tank in the main tank. Weight penalty will be high. Rejected.
15. Start tank with capillary refill	Similar to start basket with capillary refill (Concept 10). Can reduce start tank volume.	Added weight, complexity and operational deficiencies (see Concept 10).	Rejected.
16. Start basket - channels with vacuum refill	Similar to Concept 11 used to clear vapor from the device between burns.	Vented fluid penalty could be high (see Concept 11).	Rejected.
17. Start tanks - pumping	Using a mechanical pump to remove vapor from the channels inside a start tank could have some use for missions where short burns do not refill the device. This would allow internal device to be refilled in low gravity from pool in start tank. Channels to refill the start tank itself, appear to be too heavy.	Additional pumps required. Operational flexibility added. Capability is not really needed for D-1S but may be worthwhile for advanced vehicles.	Added capability compared to 13b. Low weight (higher than 13b). Pumps required. Concept recommended for study with Concept 13b for advanced mission requirements. Rejected for present D-1S requirements.

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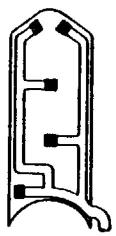
Table 2-3. Conceptual Fluid Containment Candidates (Continued)

Candidate Concept	Operations & Design Features	Characteristics	Advantages/Disadvantages, Recommendations
18. Capillary pickup heads 	Use solid thin-walled tubes with multiple screen filter elements placed in strategic locations within the tank. The tubes and filter elements remain filled with liquid during the entire mission. Screen area can be reduced considerably from the multiple liner and channels of Concepts 6 and 7 although this results in some increase in pool residuals. An evaluation was made of screen pressure drop that could be tolerated to determine the screen area that allows a minimum weight configuration to be designed.	LH_2 acq. sys. wt. 245.0 lbs (111.0 kg) LO_2 acq. sys. wt. 300.0 lbs (136.1 kg) (Filter Elements) Tubing weight 12.0 lbs (5.4 kg) Vent system weight 55.8 lbs (25.3 kg) Pressurization sys. wt. 599.8 lbs (272.1 kg) 1212.6 lbs (550.0 kg)	Excessive weight. Basic concept is rejected, however single layer pickup heads will be considered for 'refilling by pumping concept.' Rejected.

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$$LO_2,\ h = \frac{0.567}{D_{BP}},\ \text{meters} \quad LH_2,\ h = \frac{1.48}{D_{BP}},\ \text{meters}$$

where:

h = retained head

D_{BP} = screen bubble point in microns

Aluminum screens with aluminum backup plate were used to the limit of aluminum screen retention (14 microns for 200 × 1400 mesh screen). Below 14 microns, aluminum screen is unobtainable. Between 14 microns and 10 microns the lowest weight and least expensive alternative is to use 325 × 2300 stainless steel screen and aluminum backup. Since 325 × 2300 is the practical lower limit for screen retention, requirements below 10 microns require multiple screen layers.

Weight characteristics of screen devices as presented in Table 2-4 are based on data generated in NAS8-21465 (Ref. 2-1 and 2-2). These preliminary weights were used only as rules of thumb in determining whether the candidate acquisition devices would exceed the existing system weight. Weights for each candidate system were generated as well as accompanying vent system and pressurization system weights. System weights were handled in a manner identical to that shown in the Section 2.0 for the existing system. Payload sensitivity factors were applied to each weight item to determine the equivalent penalty of the capillary acquisition system and other vehicle system changes caused by the presence of that capillary acquisition system.

Table 2-4. Screen Device Weight Characteristics

Screen Device	Weight lb/sq ft (kg/m²)	
	Aluminum	Stainless Steel
Start basket (perforated plate backup)	1.23 (6.0)	2.5 (12.2)
Start basket (open isogrid backup)	0.90 (4.39)	NA
Channels (perforated plate backup)	0.56 (2.73)	1.12 (5.47)
Liner (perforated plate backup)	0.32 (1.56)	0.65 (3.17)

A more detailed breakdown of the weight of the LO_2 and LH_2 basic start basket concepts are given below in order to aid in understanding the weight comparisons given in Table 2-3.

Weights are generated for both hydrogen and oxygen tank capillary device configurations. The hardware weight of the hydrogen system is

Capillary Device and Conditioning Coils	118.0 lb	(53.6 kg)
Feedline Conditioning Hardware	5.2 lb	(2.36 kg)
Thermal Conditioning Hardware	3.2 lb	(1.45 kg)
	126.4 lb	(57.4 kg)

Oxygen system hardware weight is

Capillary Device and Conditioning Coils	22.3 lb	(10.12 kg)
Feedline Conditioning Hardware	4.5 lb	(2.04 kg)
Pressure Control and Thermal Conditioning Hardware	6.8 lb	(3.09 kg)
	33.6 lb	(15.25 kg)

Acquisition System Hardware Weight = 126.4 (57.38 kg) + 33.6 (15.25 kg) = 160 lb (72.6 kg)

Weight penalties for these devices must also include any weight increments resulting from integration of the acquisition device with other vehicle subsystems.

Equivalent Payload Penalties

LO_2 Vent System Hardware Weight	33.0 lb	(15 kg)
LO_2 Propellant Vented	18.5 lb	(8.4 kg)
LH_2 Vent System Hardware Weight	29.0 lb	(13.2 kg)
LH_2 Propellant Vented	26.8 lb	(12.2 kg)
Total Vent System Weight	107.3 lb	(48.8 kg)
Pressurization System Hardware Weight	487.0 lb	(221 kg)
Pressurant Weight (Cold Helium)	112.8 lb	(51 kg)
Total Pressurization System Weight	599.8 lb	(272 kg)

Assume that all helium remains in tank until the end of the mission.

Total System Weight = 599.8 lb (212 kg) + 107.3 lb (48.8 kg) + 160 lb (72.6 kg) = 867.1 lb (394 kg)

These were preliminary weight estimates. More detailed design and analysis was performed in Task IV (Section 5).

2.3 ACQUISITION SYSTEM THERMAL CONDITIONING CANDIDATES

Of primary importance in the screening of propellant acquisition concepts or devices is the adaptability for thermal conditioning. Thermal conditioning, whether it be an actively controlled system or a passive system, is necessary for the maintenance and control of liquid within the device. The formation of vapor and the resulting displacement of liquid is caused by either heat transfer to the device or a reduction in pressure within the device below the liquid vapor pressure. Prevention of vapor formation and possible screen dryout caused for either reason is the purpose of the thermal conditioning system; it therefore, includes both heat transfer and pressure control.

Thermal conditioning alternatives available for propellant acquisition device concepts are as follows:

1. Acquisition device conditioning

   Cooling coils on device - outside
   Cooling coils on device - inside
   Cooling coils inside device

2. Entire tank conditioning

   Cooling coils on tank wall - outside
   Cooling coils on tank wall - inside
   Cooling coils on tank shield

3. Pressure conditioning

   Subcooling by tank pressurization
   Subcooling by start tank pressurization
   Subcooling by cooling the fluid before it reaches the boost pump

4. Intertank conditioning

   Separate vent systems in each tank
   LH_2 boiloff used to cool LO_2 tank

These concepts were evaluated based on functional performance, system weight, ease of fabrication and installation, development requirements, safety, testability, reliability and cost.

The baseline D-1S vehicle incorporates a thermodynamic vent system that controls tank pressure and destratifies propellant temperature. Throttled vent fluid is a source of coolant flow for thermal conditioning as long as liquid exists at the vent system inlet. In this event, tank venting will provide a means of cooling the acquisition device. However, the vent requirements of the main tanks are not sufficient to provide adequate thermal conditioning of the acquisition device. For example, if perfect mixing of the LH_2 tank, exclusive of the acquisition device, were to occur during the 4.22 hour coast between the third and fourth burns of the five burn low earth orbit mission, no venting would be initiated and no thermal conditioning of the acquisition device would normally occur. Yet, during that period, sufficient heat may enter the tank in the area of the LH_2 acquisition device to vaporize 36 pounds (16 kg) of LH_2.

It follows, then, that thermal conditioning by liquid venting through cooling coils should be actively controlled by acquisition device thermal demand or by imposing a preprogrammed steady-state or variable rate sufficient to handle worst case predicted heating. Any additional propellant vented (above normal tanking venting quantities) will be assessed against the acquisition device.

Table 2-5 discusses the operation, design features, advantages, disadvantages and recommendations for acquisition device thermal conditioning concepts.

2.3.1 <u>ACQUISITION DEVICE THERMAL CONDITIONING.</u> Thermal conditioning can be accomplished by cooling the acquisition device itself or the liquid within the device, thereby maintaining the required screen wetting. In this case, start liquid should be self-conditioned, operating independently from the baseline D-1S vent system configuration. This approach is more desirable than using the thermodynamic vent system fluid for cooling because its performance does not depend on main propellant orientation or main tank pressure excursions.

Relative to entire tank cooling, acquisition device cooling has the obvious advantage of lesser size, complexity and weight. An additional advantage of acquisition device cooling lies in the potential ability to subcool the start liquid by a desired amount. Should boost pump cooling prove incomplete or unfeasible, the subcooling necessary for the required effective boost pump NPSH (net positive suction head) can be provided by cooling the start liquid to the desired level. This represents a significant reduction in the pressurization system requirements. It also represents a desirable alternative to boost pump conditioning.

Acquisition device cooling can be accomplished in three ways; cooling coils attached outside the device walls, cooling coils attached to the inside walls, and cooling coils located inside the device but not in thermal contact with the device walls. Direct acquisition device cooling is recommended over entire tank conditioning and intertank conditioning. The recommended concept is to use cooling coils outside the device on the device walls (Table 2-5, Item 1.1).

Table 2-5. Thermal Conditioning System Screening

Concept	Operation and Design Features	Advantages	Disadvantages	Recommendations
1.0 Acquisition Device Cooling	Maintain a liquid state within the acquisition device by transferring heat from the acquired liquid to throttled vent fluid. System is activated to control liquid temperature below saturation.	Compact design. Operates independently from other systems. Permits use of start liquid thermal subcooling to eliminate prestart pressurization.	Active system. Requires computer control for tank pressures below vent pressure.	Recommended for start basket or start tank.
1.1 Cooling Coils Outside Device Walls	Cooling coils containing throttled vent fluid thermally bonded to outer surface of capillary device or start tank wall.	Easier to fabricate and service than inner surface bonding.	Increases outer surface area of device and therefore external heating.	Inferior thermally to 1.2 but recommended for further study due to fluid and fabrication advantages.
1.2 Cooling Coils Inside Device Walls	Cooling coils thermally bonded to inner surface of capillary device or start tank wall.	Coils in contact with start liquid enhance subcooling capability. Smooth outer surface of device reduces heat transfer from tank fluid.	May be difficult to attach and connect cooling tubes on the inside of the device. May increase liner pressure drop due to coils obstructing flow.	Recommended for further study.
1.3 Cooling Coils Inside Device Not on Walls	Cooling coils are inside the device but do not conform to surface. May be attached to device or supported independently.	Fabricated and maintained independently from device. Removable. Greater design tolerances.	Large temp. gradients and local vaporization may occur inside device. May require mixing. Relies on low-g heat transfer to transfer heat from the contained liquid to the cooling coils.	Rejected as inferior to 1.2

Table 2-5. Thermal Conditioning System Screening (Cont.)

Concept	Operation and Design Features	Advantages	Disadvantages	Recommendations
2.0 Entire Tank Cooling	Control tank pressure and maintain equilibrium liquid/vapor state by intercepting all tank heating. Uses cooling coils wrapped externally to the tank wall.	Eliminate need for mixer and thermodynamic vent system.	Large complex design. Impacts design of many structures. Extensive use of tubing bonded thermally constitutes weight, design, handling and maintenance problems. Vapor may form in device during coast outflow.	Rejected as inferior to Concept 1.0.
2.1 Cooling Coils on Tank Walls	Heat is intercepted at tank surface by cooling coils attached to tank structure and supports.	Lower weight than 2.2. Uses tank wall as conductor.	Significant problem in thermally bonding tubing to thin wall structure. May require redesign of intermediate bulkhead. Sets up thermal gradients in tank structure.	Rejected.
2.2 Cooling Coils on Shield Outside Tank Walls	Heat is intercepted outside tank surface by cooling coils attached to a conducting shield.	Good ground testability. Does not require tube bonding to thin wall tank structure. Does not impact tank design.	Poor response characteristics may require pre-programmed vent rate and, therefore, excessive vent mass. Does not accommodate internal heat sources such as intermediate bulkhead. Weight penalty for structural support to tubing and integrated conducting shield.	Rejected.

Table 2-5. Thermal Conditioning System Screening (Cont'd)

Concept	Operation and Design Features	Advantages	Disadvantages	Recommendations
3.0 Intertank Conditioning	Use vent fluid from liquid hydrogen tank to thermally condition liquid oxygen tank.	Potentially eliminates LO_2 vent system and mixer. Eliminates oxygen vent losses.	Safety/reliability risk prevents acquisition device LH_2 cooling coils inside LO_2 tank. Requires concepts 2.0 and 4.0 for LO_2 tank.	Rejected as inferior to Concept 1.0 for capillary device thermal conditioning.
4.0 Pressure Conditioning for cooling and NPSH	Subcool liquid when necessary by pressurizing with helium.	Positive suction pressure can be provided in a very short time and accurately regulated. Currently used for Centaur prestart to provide boost pump NPSH.	High system weight. Zero-g pressurization of unsettled liquid requires greater pressurant quantities than current system. Not practical for an open device such as start basket during coast.	Recommended for systems requiring pressurant for liquid expulsion. Because of its relatively small volume, a start tank can efficiently use cold gas pressurization.
4.1 Thermal Subcooling for cooling and NPSH	Subcool liquid within acquisition device or while flowing to the boost pump.	Systems utilizing concept 1.0 can achieve the required positive suction pressure without tank pressurization. Significant pressurization system weight savings are possible.	A new subcooling system must be developed. Extra vent fluid is required to subcool liquid in acquisition device.	Recommended for start basket.

2.3.2 ENTIRE TANK CONDITIONING. The start liquid as well as all the liquid in the tank can be maintained at a saturated state (slightly subcooled in the presence of helium) if all external heating is intercepted at the tank boundaries. This can be accomplished by placing cooling coils over the entire tank wall surface as well as between the tanks. The primary advantage of cooling the entire tank is that it results in total tank pressure control, thereby eliminating the need for a separate thermodynamic vent system and mixer. However, the installation of cooling coils around the entire tank structure has a major impact on the design, structural integrity, maintenance and handling procedures of the Centaur propellant tanks. Approximately 5000 ft (1524 m) of tubing are required in addition to flow balancing manifolds, valves and sensors. The hardware tradeoff is likely to result in a high weight penalty.

Another disadvantage of entire tank cooling is that the start liquid can only be maintained at a state very near saturation. Minor heat leakage resulting from system inadequacy can result in vaporization within the capillary device adjacent to the tank walls.

2.3.3 INTERTANK CONDITIONING. An additional consideration for thermal conditioning using a propellant liquid vapor cycle is intertank conditioning. That is, using hydrogen gas coolant from the LH_2 tank thermal conditioning system to condition the LO_2 tank or LO_2 propellant acquisition device. Hydrogen gas exiting the LH_2 tank coils at near LH_2 temperature has an additional heat capacity of 350 Btu/lb (8.2×10^5 j/kg) when elevated to LO_2 temperatures. This exceeds the oxygen net heating rates which are 85%, 25%, and 26% of hydrogen tank heating for the open payload bay, low altitude orbit and synchronous orbit heating environments, respectively. This concept has the advantage of eliminating the thermodynamic vent system and mixer from the LO_2 tank. Its application is felt to be limited to entire LO_2 tank conditioning due to the combustion hazard of routing hydrogen through tubes inside the LO_2 tank. Since heat currently flows from the LO_2 tank to the LH_2 tank across the intermediate bulkhead, only the aft bulkhead requires cooling tubes. However, the aft bulkhead structure is complicated by the attachment of many supports and brackets. Heat penetration interception would thus require special attention.

Intertank conditioning is undesirable for acquisition device cooling because of the complexity of attaching cooling coils over the entire aft bulkhead and the uncertainty in the ability of the system to control thermal stratification in the tank. Even though this system can save 51.5 lbs (23.4 kg) over using separate vent systems, system adoption is based purely on vent system tradeoffs. This tradeoff would apply whether or not an acquisition system is employed.

2.3.4 PRESSURIZATION SYSTEM. The baseline D-1S pressurization system uses ambient temperature helium injected directly into the LH_2 tank through a diffuser. In the LO_2 tank, helium is bubbled through the liquid to partially displace ullage volume. Direct injection of helium pressurant into the LH_2 tank ullage is relatively efficient. The diffuser limits gas velocity, reduces heat transfer to the liquid, and allows a high ullage temperature gradient to exist. Total helium usage for the five-burn mission is supplied by three large bottles, resulting in the baseline total system hardware weight of 417 pounds (189.3 kg) with 14 pounds (6.4 kg) of helium expended.

Pressurization with a propellant acquisition system in the tank differs from the above method because of the zero-g environment. With unsettled propellant, liquid orientation can be detrimental to the interfacial heat and mass transfer such that direct pressurization methods are not applicable. Pressurization system weights shown in Table 2-6 are based on pessimistic assumptions regarding interfacial behavior. The pressurant was assumed to reach liquid temperature during pressurization. The resulting helium quantity makes ambient storage impractical and suggests that helium be stored at LH_2 temperature within the hydrogen tank for main tank pressurization. Ambient helium is used for auxiliary propulsion system (peroxide) pressurization.

Also, the advantages of bubbling helium through the LO_2 are no longer available at zero-g because buoyancy forces are not present for bubble propagation and gas-liquid mixing.

The pressurant quantities given in Table 2-6 for the acquisition system configurations, are the helium required to provide an additional partial pressure equal to the ΔP's shown. The pressurization results in equivalent liquid subcooling. Three psi (20.7 kN/m^2) is the amount of subcooling currently needed to satisfy boost pump NPSH under low-g conditions. Start tank pressurant weight is greater than the baseline D-1S pressurant weight because the greater density of cold pressurant overrides the reduced volume to be pressurized. For baseline system pressurization, total main LH_2 tank volume is 36 m^3 (1270 ft^3) and helium is injected into the main tank at approximately 0.04 kg m^3 (0.0025 lb/ft^3). For the LH_2 start tank, volume is approximately 1 m^3 (35 ft^3) and helium is injected into the start tank at approximately 3.52 kg/m^3 (0.22 lb/ft^3).

Table 2-6. D-1S Pressurization System Weights (Low Earth Orbit Five Burn Mission)

Configuration	Helium Pressurant		Storage Bottles		Total Weight lbm (kg)	
	ΔP psi (kN/M^2)	Mass lbm (kg)	Cryogenic	Ambient	Hardware +Residual lbm(kg)	Fluid Expended lbm (kg)
Baseline D-1S	3 (20.7)	10 (4.5)		3 large	417 (189.2)	14 (6.4)
Start Basket	3 (20.7)	109 (49.4)	2 large	1 small	487 (220.9)	113 (51.3)
Start Tank (Through Feed)						
Start Tank	3 (20.7)	27 (12.3)	1 small			
Main Tank	6 (41.4)	218 (98.9)	4 large	1 small	806 (366.5)	249 (112.9)
Start Tank (Bypass Feed)						
Start Tank	3 (20.7)	27 (12.3)	1 small	1 small	294 (133.4)	30 (13.6)
Main Tank	0	0				
Cooled Boost Pump and Sump (Thermal Subcooling)	0	0		1 small	56 (25.4)	4 (1.9)

2.3.4.1 Start Basket Pressurization. With a start basket, pressurization of the entire tank by 3 psi (20.7 kN/m^2) will properly condition the liquid for successful boost pump operation. The use of cold pressurant alleviates the danger of localized screen drying and pressure collapse due to chilling. An additional 70 pounds (31.8 kg) of system weight and 99 pounds (44.9 kg) of pressurant weight must be carried for this configuration.

2.3.4.2 Start Tank Pressurization. Pressurization system requirements for two start tank configurations (Concepts 13a and 13b of Table 2-3) are as shown in Table 2-6. The through-feed start tank requires a main tank over-pressurization for refilling the start tank during main engine firing. Main tank pressurization must be initiated prior to engine start in order to provide the necessary ΔP in time for the refill valves to open. All pressurization for the start tank configuration is, therefore, accomplished using helium stored cryogenically. The single, small bottle, ambiently stored, (conservatively estimated at 4 pounds (1.8 kg))satisfies the functions of engine valve operation, reaction control propellant pressurization, bleeds, purges, and leakage.

Start tank pressurization requirements shown in Table 2-6 were based on preliminary conservative estimates of chilldown flow plus settling times equal to five times free fall with main engine settling thrust. These values are shown in Table 2-7. The hardware and residual weights shown in Table 2-6 for ambient bottle storage are based on Centaur pressurization system values for existing hardware. These are titanium bottles 7365 cubic inches (0.12 m^3) and 4650 cubic inches (0.08 m^3) in volume. The respective weights for the large and small bottles including brackets, support line fittings and residual helium are 81 pounds (36.8 kg) and 56 pounds (25.4 kg). Hardware weight estimates for the cryogenically stored bottles are based on LN_2 formed stainless steel. Since specific bottle sizes are not currently available, system weight partials were developed from preliminary estimates made by Centaur fluid systems design. The bottle hardware weight of 2.346 pounds (1.07 kg) per pounds of helium required corresponds to a 100 pound (45.4 kg) bottle of 7365 cubic inches (0.12 (m^3) volume with a maximum pressure of 2600 psia (17914 kN/m^2). A maximum final pressure of 50 psia (345 kN/m^2) gives 47.8 pounds (21.7 kg) of deliverable helium with a 2.2 pound (1 kg) residual. Ten pounds (4.54 kg) of supports and 1 pound (.454 kg) of lines and fittings were assumed giving a hardware plus residual weight of 113 pounds (51.3 kg) for every 47.8 pounds (21.7 kg) of deliverable helium.

The bypass feed start tank (concept 13b, (Table 2-3) does not require main tank pressurization. After engine start, when propellants are settled under full thrust, main tank outflow bypasses the start tank and is pressurized under its own head. At this time the start tank outflow is terminated and the tank is vented below main tank pressure for start tank refill.

2.3.4.3 Start Basket Self Pressurization. For the start basket configuration, three start methods are available wherein tank pressure alone is sufficient to feed the boost pumps for successful start. One is to precool the boost pumps and sumps to liquid

Table 2-7. Start Tank Outflow Volume

Criteria	Propellant	Engine Start 1		Engine Start 2		Engine Start 3		Engine Start 4		Engine Start 5	
Settling Time, sec	LH_2	2.37		3.33		3.78		3.45		3.40	
	LO_2	1.39		2.09		2.23		2.10		2.05	
Settling Mass, lb (kg)	LH_2	25.6	(11.6)	36.0	(16.3)	40.8	(18.5)	37.3	(16.9)	36.7	(16.7)
	LO_2	79.4	(36.0)	119	(54)	127	(57.7)	120	(54.5)	117	(53.1)
Settling + Childown Mass lb (kg)	LH_2	91	(41.3)	101	(45.9)	106	(48.1)	102	(46.3)	102	(46.3)
	LO_2	150	(68.1)	190	(86.3)	187	(84.9)	191	(86.7)	188	(85.3)
Outflow ΔV, cu ft (m^3)	LH_2	20.8	(.59)	23.2	(.66)	24.3	(.69)	23.5	(.67)	23.4	(.06)
	LO_2	2.18	(.062)	2.75	(.078)	2.87	(.081)	2.77	(0.78)	2.72	(.077)

temperature prior to start. Another is to subcool the liquid in the start basket thermally by cooling to vapor pressure 3 psi (20.7 kN/m^2) below tank pressure. The third method is to remove the heat, necessary to subcool the liquid, while the liquid is flowing to the boost pump. If any of these cases can be feasibly implemented, the main tank pressurization systems can be eliminated. Table 2-6 gives pressurization system weights for this start basket approach. Only the helium required for other systems (e.g-attitude control) must be provided.

2.4 BOOST PUMP THERMAL CONDITIONING

In order to minimize line chilldown problems and to provide quick engine startup the baseline feed system conditioning concept considered maintaining the boost pump and propellant duct filled with liquid between burns. The main problem was to maintain the boost pump filled with liquid between burns. Sixteen potential methods were employed to cool the boost pump. These methods are identified in Table 2-8 with a description of the advantages, disadvantages and recommendation for each candidate.

Selected candidates were: wrapping the drive shaft area near the pump with cooling coils and purging the turbine rotor with cold helium. Wrapping the drive shaft area with cooling coils was selected for analysis because it was the method least likely to freeze the lubricants (alternative low temperature lubricants were identified in the event this could not be prevented). Purging the turbine rotor with cold helium was considered because it did not require pump modification and it cooled the hottest area of the power package. A combination of the two methods was also investigated. Other concepts identified were rejected prior to analysis because they would require excessive boost pump modification or were not as thermally efficient as the methods considered.

Methods that were too complex to be considered were purging the gearbox with helium (because of grease in the gearbox) and drilling purge holes in the drive shaft.

Table 2-8. LH_2 Boost Pump Cooling Candidates

Candidate	Advantages	Disadvantages	Comments/Recommendations
1. Wrap the drive shaft area near the pump with cooling coils.	1. Least likely to freeze the lubricant. 2. Easiest to analyze and control. 3. Does not require additional helium. 4. No turbine hardware modifications.	1. Removal of heat could cause the gearbox and turbine lubricant to freeze or to increase the lubricant viscosity to such an extent that, at turbine startup, torque loads on the drive shaft would exceed design loads. This will require use of an alternative low temperature lubricant. 2. Manufacturing problems with installing cooling coils in the region in question. 3. High heat fluxes that are expected for this area could cause oversizing of the thermodynamic vent, thereby drawing off large amounts of propellants to cool the boost pump.	Potentially provides positive heat flux interception at the pump interface so that cooling fluid is not lost in cooling areas that would benefit by being warm (such as lubricated areas). Recommended.
2. Purge turbine rotor with cold helium.	1. No pump modifications required.	1. Removal of heat is likely to cause freezing of turbine and gearbox lubricant. 2. Demands more helium to be used. To cool the helium, a heat exchanger would need to be designed for the helium supply. 3. The removal of heat could influence the power requirement of the heater installed on the H_2O_2 catalyst bed. 4. Installation of purge connections on turbine. 5. Ground evaluation testing would be needed.	Requires a helium supply, but it cools the hottest area of the power package and it does not require pump modification. Recommended.
3. Purge gearbox with cold helium.	1. Removes pump heating more directly than 2.	1. - 5. Same method as 2. 6. Uncertainties involved with heat removal from a grease filled gearbox. 7. Inflowing helium may blow on the grease and create bare spots on the gears.	Same function as Candidate 2. except that it cools the gearbox area. Uncertainty of heat removal due to complexity of gearbox . Rejected.
4. Purge both the gearbox and turbine with cold helium.	1. More uniform removal of heat.	1. - 7. Same method as 3.	Combination of the second and third candidates. Rejected for same reason as 3.
5. Purge the gearbox and wrap cooling coils around the drive shaft.	1. Provide design flexibility in removing heat by two methods.	1. - 3. Same as method 1. 4. - 10. Same as method 3.	Gearbox purging used to achieve cooling which has been deemed undesirable in terms of reliability of heat removal. Rejected.
6. Purge the turbine and wrap cooling coils around the drive shaft.	1. Provide design flexibility in removing heat by two methods.	1. - 3. Same as method 1. 4. - 8. Same as method 2.	Combination of already selected cooling techniques used for uniform cooling of the pump plus the added advantage of a dual system. Recommended.

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Table 2-8. LH_2 Boost Pump Cooling Candidates (Continued)

Candidate	Advantages	Disadvantages	Comments/Recommendations
7. Purge both the gearbox and turbine and wrap cooling coils around the drive shaft.	1. Provides design flexibility in removing heat by two methods.	1. - 10. Same as methods 1, 2, and 3.	Gearbox purging has been eliminated as a viable candidate for heat interception. Rejected.
8. Replace the existing LH_2 boost pump drive shaft with matl. that has a higher thermal resistance and put radiation blocks inside the hollow core to prevent rad. tunnelling.	1. Could prevent turbine gearbox lubricant from freezing.	1. Evaluation testing on the new drive shaft would be needed. 2. Material compatibility with rest of pump/gearbox unit would be necessary. 3. Radiation block would need to withstand high centrifugal forces (shaft speed - 35,000 rpm (3640 rad/sec). 4. Design of new drive shaft may not reduce heat flux to desirable levels, analysis would need to be done to verify this. 5. Require active cooling of pump.	Logical solution to the problem of heat conduction down the drive shaft but represents a major design change to the pump/power package. Also possible strength/thermal resistance materials problems. Rejected.
9. Install a purge bag over the entire turbine/gearbox assembly and purge with cold helium.	1. Fairly uniform removal of heat. 2. No external connections required in turbine/gearbox or pump.	1. Evaluation testing would be needed. 2. Manufacturing program to design a purge bag would be fairly extensive. 3. Freezing or sluggishness of the lubricant in the gearbox/turbine assembly would result when using cold (LH_2 temperature) helium. 4. Large quantities of helium required.	Bag requires a design capable of withstanding the environment of the boost pump. Large amounts of helium required to convectively cool the exterior of the power package. Rejected.
10. Improve insulation on outside of turbine/gearbox assembly in addition to active cooling concepts listed.	1. Reduces incident heating to the boost pump and sump.	1. Will not reduce the heat flux to the LH_2 pump due to operation of the turbine/gearbox. 2. May not reduce the heat flux due to solar heating to desirable levels, thermal analysis will need to be done.	Decreases the influence of solar and incident heating upon the turbine/gearbox. The weight of the insulation system should be compared with the weight of the vent fluid needed to intercept the radiant heat flux. Recommended.
11. Enclose the turbine/gearbox support structure on the pump & purge that area with vent fluid or cold helium.	1. Positive removal of heat from convective cooling of drive shaft.	1. - 4. Same as method 9.	Unreliability of heat removal by convective means at low temperatures with a reasonable helium purge rate. Rejected.

Table 2-8. LH_2 Boost Pump Cooling Candidates (Continued)

Candidate	Advantages	Disadvantages	Comments/Recommendations
12. Drill "purge" holes at either end of the drive shaft & purge GH_2 down the shaft into the gearbox and vent to space.	1. Does not require cold helium.	1. LH_2 would leak into holes and vaporize inside the shaft and vent back into the pump. 2. Unacceptable leakage may result from the tank to the gearbox. 3. LH_2 will freeze the gearbox lubricant. 4. Requires major design changes to boost pump.	Low reliability of this system during different phases of boost pump performance. Rejected.
13. Replace the turbine/gearbox with an electric driven motor and install the entire assembly in the LH_2 sump.	1. Eliminates the lubricant freezing problem. 2. Eliminates need for H_2O_2 supply for boost pump.	1. Major design changes and reliability test program required. 2. Pumps would be very expensive. 3. More electric power may need to be carried on-board.	Expensive, requires major design changes. Rejected.
14. Magnetically couple the turbine drive and the pump so as to remove the need for direct mech. coupling.	1. Increases resistance to heat flow into the pump. Reduces pump heating. 2. Keeps lubricant warmer.	1. Design changes necessary in area of pump/gearbox connection. 2. Reliability of magnetic coupling would need to be proven for design shaft loads and speeds. 3. High weight.	Lower reliability, expensive and requires major design changes. Rejected.
15. Disengage the drive shaft from the pump when in a non-operative phase of boost pump operation.	1. Increases resistance to heat flow into the pump. Reduces pump heating. 2. Keeps lubricant warmer.	1. Large design and reliability task to prove the performance of such a system. 2. Evaluation testing would be necessary. 3. High weight.	High weight, low reliability and major design changes. Rejected.
16. Subcool the sump area around the boost pump so the vaporized fluid is recondensed in the sump.	1. No pump modifications required. 2. Could reduce pressurization system requirements.	1. May not in itself prevent vapor formation at pump since the fluid would be static and there would be little mixing of warm fluid with the cold liquid.	Could eliminate much of the pressurization system and could thus present a major weight savings. No pump modifications. High vent fluid penalty may occur. Recommended.

The initial effort in Task III (Thermal Analysis) was to identify sources of heat input to the fluid stored in the boost pump. Heat input entered mainly from the turbine and gearbox driving the boost pump. Heat input also entered from the surroundings. A thermal analysis of the LH_2 boost pump (described in more detail in Section 4.5) was performed using both the existing boost pump thermal model and a model developed specifically for determing the heat input to the liquid contained in the pump. Pump heating rates were a maximum of 20 Btu/hr (5.86 watts). Inspection of the configuration and the model revealed that the heat entering the pump along the drive shaft could probably not be removed with the recommended concept. Maintaining liquid in the boost pumps between burns requires cooling modifications that would probably disqualify the system because of complexity. For this reason, effort was redirected in determining methods of using an uncooled boost pump and propellant duct.

2.5 PROPELLANT DUCT THERMAL CONDITIONING

Propellant duct thermal conditioning concepts were devised to intercept heat input due to; solar radiation, albedo, heating by warm components such as the engine, peroxide bottles and electrical boxes, and heating of the LH_2 ducts by the LO_2 tank. Table 2-9 lists the nine thermal conditioning methods that were initially considered. Methods that maintained wet ducts between burns employed cooling coils, cooled shields or cold helium purge to condition the liquid. Methods employing a dry duct were similar to the existing Centaur procedure of flushing the lines with liquid prior to main engine start.

The need for maintaining wet ducts between burns was obviated when methods for maintaining a wet boost pump became too complex for consideration. Also the need for providing a quick engine start up, which is the main advantage of a wet propellant duct, could not be identified as an important advantage for Shuttle based Centaur missions. Other pluses for a dry system are the lower complexity and cost and lower thermal conditioning fluid requirement when the ducts are not maintained at cryogenic temperature.

2.6 FABRICATION CANDIDATES

Many potential methods exist for fabricating capillary acquisition devices. Areas of major importance are; selecting barrier material and barrier backup material, attaching barrier material to backup material, attaching cooling tubes to device surfaces and supporting the acquisition device within the tanks.

2.6.1 BARRIER MATERIALS. Barrier materials normally used are screens or perforated plates. For Centaur applications screens are preferred because of their lighter weight, higher strength at small pore diameter and their potential wicking capability. For the finest meshes (smallest pore size-lowest micron ratings) screens are available only in stainless steel (304). Lighter weight aluminum (5056) is preferred to stainless steel and will be used for the coarser mesh applications

Table 2-9. Propellant Ducting Thermal Conditioning Candidates

Candidate	Advantages	Disadvantages	Comments/Recommendations
1. Wrap duct length with cooling coils sized for the expected coolant flowrate using throttled propellant needed to intercept the expected heat fluxes.	1. Line will always be at liquid temperature. 2. Centaur propellant bleed lines could be discarded. 3. Line chilldown period of engine prestart could be abolished or reduced.	1. Detailed thermal analysis needed to size the cooling coils for the heat fluxes expected. 2. If high heat fluxes are modeled, large amounts of coolant fluid could be needed. Could have a higher vented fluid penalty than chilling down the line between burns. 3. Cooling coils could change the structural dynamics of the ducting and an analysis would be needed to determine if a POGO problem exists. 4. Thru-tank fittings would be needed for passing vent fluid from tank to lines. 5. Requires liquid to be available to the vent system inlet. 6. For high heat flux and correspondingly large number of cooling coil windings system weight could be quite high.	Discarding of bleed lines, and reduction of line chilldown time before engine prestart signal. Recommended for further study.
2. Utilizes a foam or non-metallic system of channels to perform the function of cooling coils.	1. Line will always be at liquid temperature. 2. Centaur propellant bleed lines could be discarded. 3. Line chilldown period of engine prestart could be abolished or reduced. 4. System weight could be lighter than Candidate 1.	1. Detailed thermal analysis needed to size the cooling coils for the heat fluxes expected. 2. If high heat fluxes are modeled, large amounts of coolant fluid could be needed. Could have a higher vented fluid penalty than chilling down the line between burns. 3. Cooling coils could change the structural dynamics of the ducting and an analysis would be needed to determine if a POGO problem exists. 4. Thru-tank fittings would be needed for passing vent fluid from tank to lines. 5. Requires liquid to be available to the vent system inlet. 6. Containment material may leak fluid so that excessive coolant would need to be used for cooldown. 7. Material may cryopump material on to duct surface which is undesirable for heat transfer. 8. Possible compatibility and deterioration of materials with cryogens for extended periods (especially, LO_2).	High leakage rates expected from a foam system. Material thermal compatibility problems with the propellant. System rejected for further study.
3. Flush the propellant lines at either a low continuous rate or at a high rate at selected times. Flushing lines just before MES is same as standard Centaur practice.	1. Lines will be wet at engine prestart signal.	1. Requires liquid to be available to the pump. 2. Flushing requires amounts of hydrogen peroxide to be used in running the boost pumps.	For high heat flux rates to lines, this system would use the least amount of cooldown fluid. Lines will be insured "wet" at prestart signal. Recommended.

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Table 2-9. Propellant Ducting Thermal Conditioning Candidates (Continued)

Candidate	Advantages	Disadvantages	Comments/Recommendations
4. Install purge bag arrangement for missions with high heat fluxes and purge with cold vaporized propellant or cold helium.	1. Lines will be chilled prior to boost pump startup.	1. High system weight. 2. Requires large amounts of propellants or helium. 3. Complex manufacturing program to develop such a system.	Heavy complex system design. Rejected.
5. Install all components on aft bulkhead on an equipment module and "hide" the propellant lines underneath this shield. Purge this area if heat flux is still a problem.	1. Reduces heat flux to propellant lines due to radiation.	1. Major vehicle design change. 2. May cause a undesired reduction in heat flux to LOX tank and LOX freezing may result. 3. This method alone does not ensure line chilldown.	Major vehicle design change. May cause LOX freezing. Does not ensure propellant line chilldown. Rejected.
6. Improve insulation on the propellant feedlines.	1. Reduces heat flux to lines.	1. May not reduce heat flux to desired levels to maintain lines at liquid propellant temperature. 2. Will not reduce heat flux due to support conduction.	Heat flux reduction to the lines may warrant the use of high performance insulation. Recommended.
7. Keep ducts out of solar radiation.	1. Reduces heat flux to lines.	1. Too restrictive a measure for cooling lines. 2. Will not reduce heat flux due to support conduction.	Much too restrictive for Centaur vehicle performance. Rejected.
8. Cool the conductive supports of propellant lines.	1. Reduces heat flux to lines.	1. Will have only a limited affect on radiation heat transfer. 2. This method alone does not ensure line chilldown.	Reduces heat to lines. Recommended.
9. Direct any ACS exhaust impingement away from the propellant ducting.	1. Reduces heat flux to lines.	1. Involves a design change to either shield the propellant ducts or a major design change to reposition the ACS motors.	Major design change. Rejected.

where it is available (aluminum screen as fine as 200 × 1400 has been woven on special order).

Wicking screen is generally preferred over non-wicking screen because of its ability to remain wetted when subjected to incident heat flux. Wicking, however, can retard refilling by wetting the screen around a device before the vapor inside the device can be replaced with liquid.

Pleated screen may be required in applications where additional surface area is required to reduce pressure drop such as in channels or screened tubes within a screened enclosure. The basic structure and cooling tube attachment methods are more complex for pleated screens. Cooling the pleated screen is more difficult because a good path does not exist between the cooling coils and extremities of the screen pleat.

2.6.2 BACKUP MATERIALS. Fluid flow across the screens and the attendant pressure drop will cause a deflection of the screens. During periods such as propellant settling, certain design configurations may result in severe screen deflection, and backup of the screen may be required. The candidates for screen backup are described in the following paragraphs.

2.6.2.1 Open Isogrid. This is a structure machined from solid plate by numerical control machining. The plate is usually machined flat and bent to shape after machining although it can be machined in the final form. It derives its main strength and stiffness from the I-beam or "flanged" ribs as shown in Figure 2-1 and is the most efficient, light-weight, load-bearing structure to date. The amount of "open area" of this backup plate approaches the maximum attainable. Material is usually limited to aluminum due to machinability. Since this structure is efficient and no longer prohibitively expensive with N/C milling, it is a possible candidate for areas requiring strong structural capability. Closed isogrid is a strong candidate for resisting the crushing pressures imposed upon the start tank walls.

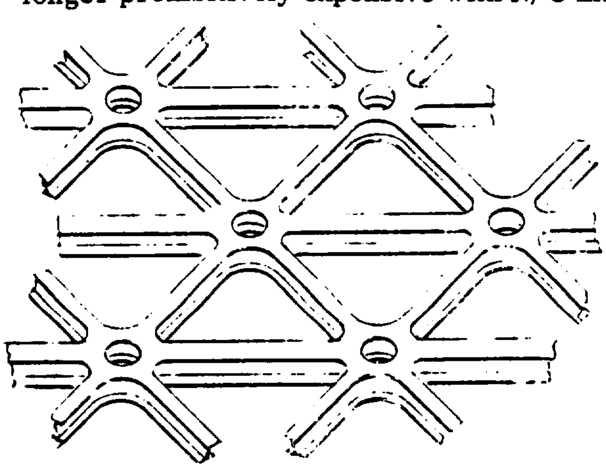


Figure 2-1. Open Isogrid Configuration

2.6.2.2 Perforated Plate. For relatively small areas of screen, such as between structural framework, backup can be provided by perforated sheet as shown in Figure 2-2. The main advantage of this material is that it is readily obtainable in a variety of materials, gauges, hole sizes and

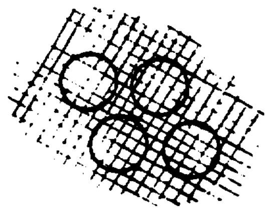


Figure 2-2. Screen Backed Up with Perforated Plate

open area. It is relatively easy to work, although it does not provide the rigidity of the isogrid without an excessive weight penalty. The open area will be less than that of the isogrid, creating additional pressure drop and deflection. Since it is sheet stock, however, it can be rigidized by various methods, such as forming stiffening flanges around the holes as the perforations are made or forming stiffening flanges at the edges where two panels are joined.

Due to its simplicity, ease of construction, and adaptability to different configurations, the perforated sheet is the primary candiate for use as screen backup.

2.6.2.3 Honeycomb Composite. It is possible to build up a structure that has a large amount of stiffness per unit weight by using honeycomb core with perforated face sheets. For use in LO_2 or LH_2, an all-aluminum structure or an all-CRES or combination of CRES and aluminum can be built up by brazing. The inherent disadvantage of this structure is that two perforated sheets are involved, and the flow restriction will be higher than for the other structures investigated. Also, the procurement of an acceptable honeycomb may be an LO_2 compatibility problem since most are bonded. If the structure is brazed, the bonding is destroyed and the honeycomb is not effective. The disadvantages of this concept outweigh its advantages. It does not appear to be an attractive candidate for screen backing.

2.6.2.4 Coarse Screen. A large mesh wire screen can also provide stiffness for screen backup as shown in Figure 2-3. A typical 2 × 2 square mesh screen with a 0.08 inch (0.20 cm) wire diameter will have a 70% open area and will weigh about 0.83 lb/sq ft (4 kg/m^2) for CRES or 0.28 lb/sq ft (1.3 kg/m^2) for aluminum. An equivalent weight in perforated sheet will have a gauge of 0.04 inch (0.10 cm) and only a 50% open area. Deflection tests were run on samples of 1×1 mesh screen and 0.040" (0.016 cm) gage perforated sheet with 3/8" (0.95 cm) diameter holes. Results indicated that for applications where stiffness is of overriding importance, coarse screens should be considered as alternatives to perforated plate. The perforated plate is slightly stronger in tension and should be considered for applications where tensile strength and/or ease of fabrication is most important.

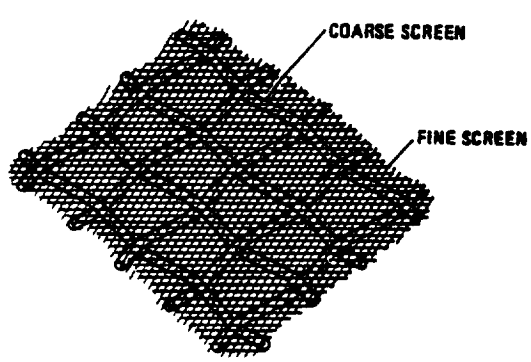


Figure 2-3. Coarse Screen Backup

2.6.3 ATTACHMENT OF BARRIER MATERIAL TO BACKUP. Candidate methods of attaching screen to backup materials are welding, brazing, soldering, riveting, bolting, and bonding. These methods are discussed in the following paragraphs.

2.6.3.1 Welding. Welding provides good thermal contact between the barrier and backup material. It is a permanent method of attachment and has a high degree of reliability. Resistance welding is easy and inexpensive but it leaves faying surfaces.

Fusion welding provides less control of heat and is thus not practical for fine mesh screen. It is most practical for perforated sheet since the weld is flat and produces no faying surfaces.

Electron beam welding minimizes the heat affected zone but it is an expensive method. It requires a vacuum chamber which limits the size of the work.

Ultrasonic welding is an experimental process most suited for welding foil or fine mesh screen to heavy gauge material. No vacuum chamber is required. The method is much more expensive than resistance welding and therefore less desirable.

2.6.3.2 Brazing. Brazing provides good thermal contact and is structurally strong. It can also be used to join different kinds of material. The main disadvantages are wicking of braze material into the screen and difficulty of cleaning flux out of the braze joints. The technique doesn't appear to be feasible in joining screens, but for special applications such as fastening cooling tubes to feedlines or attaching a weld flange to cooling tubes for use inside the propellant tanks, brazing is a likely candidate.

2.6.3.3 Soldering. This is essentially the same as brazing, but does not provide as strong a joint. It requires less heat for joining. Aluminum material would probably require copper plating. LO_2 soldering requires compatibility testing. Soldering does not appear to be a likely candidate for screen attachment.

2.6.3.4 Riveting. Riveting provides a positive means of fastening the screens to structure. It may cause screen distortion due to pressures required to set rivets. This is a more complex and heavier method than welding and should not be used unless welding cannot be accomplished.

2.6.3.5 Bolting. Bolts are a less positive method than rivets but they do provide a means of screen removal. Bolts will weigh even more than rivets. This method should be used only in areas where the screen is to be removable.

2.6.3.6 Adhesive Bonding. Adhesive bonding does not require excessive heat for application and is relatively easy to apply. It does not provide good thermal contact between bonded parts. Similar to brazing and soldering, it results in excessive screen blockage for screen-to-screen joining. There are very few LO_2 compatible adhesives. These would have to be tested in a typical application before bonding could be considered as a serious candidate for screen attachment.

Attachment of screen to the backup material will generally be done with resistance welding. Bolting should be considered when the screen is required to be removable. Resistance welding has been successfully demonstrated by GDC in previous fabrication programs of capillary acquisition devices. These demonstrations revealed that CRES Dutch twill and CRES and aluminum square weave screen can be seam welded and spot welded. Screens can be welded together or to aluminum or CRES sheets although welding CRES screens to aluminum sheet is not a structural weld.

2.6.4 ATTACHMENT OF COOLING TUBE TO DEVICE. The primary method for integrating the acquisition device and thermodynamic vent system is to use vent cooling coils in parallel with the bulk heat exchanger system used as the baseline Centaur D-1S vent system. This parallel system uses cooling coils attached to the acquisition device. The cooling tubes may be attached by welding, brazing, bonding, soldering or bolting. The method used must provide good thermal contact between the tube and the screen/plate material. Bolting does not provide uniform contact. Bonding is undesirable because it introduces a low conductivity material between the tube and device surface. Dip brazing is a primary candidate for small size acquisition devices. Tooling should be used to provide good fit between the backup material and cooling coils. In order to prevent screen clogging by the braze material the screen should be attached to the backup material after the tube attachment is made. Soldering could be used but it is generally more time consuming and less controllable than dip brazing or resistance welding. The primary candidate for large devices is resistance welding. Webbed extruded tubes seam or overlap spot welded to the device is the recommended tube attachment method for devices with dimensions greater than about 30 inches (0.76 m).

Testing was performed to determine if CRES screen can be sandwiched between aluminum sheets with a seam or spot weld through to tie the aluminum sheets together. It was found that a successful weld could be made using square weave screen of 150 mesh or coarser (so that the aluminum could flow through the screen during welding). The technique was unsuccessful with the very fine or closely woven twill screen. An alternative attachment method is to weld the tube to the backup material before attaching the screen to the backup material.

2.6.5 ACQUISITION SYSTEM SUPPORTS. The acquisition system attachment to the vehicle depends to a large extent upon the concept selected. For the LO_2 system device, a start basket or start tank fits inside the barrel and attaches directly to the tank by means of a flange welded to the tank aft bulkhead (The thrust barrel is a cylindrical structure inside the LO_2 tank designed to distribute engine loads into the tank structure.) Additional support can be provided by struts attached to the thrust barrel. The flange welded to the tank will be CRES. For access into the main body of the tank and for an aluminum to-CRES transition (if aluminum is used in the structure of the start basket or tank), the start basket or start tank will bolt to this flange. If the acquisition system is of the channel design, bolts will provide excessive weight due to the large area involved, and other means such as a bimetallic strip (e.g., Detacouple by DuPont) will be used to attach the aluminum structure to the CRES tank.

The LH_2 acquisition device attachment would be similar to the LO_2 system in the case of a channel device. However, the start tank or start basket would be mounted differently. In this case, the only dissimilar metal connection will be a transition section at the device outlet. This transition will be CRES and can be mechanically attached to the acquisition system and welded to the tank outlet.

In order to prevent damage to the intermediate bulkhead, the LH_2 start basket or start tank would be attached to the tank sidewall by struts. Clearance will be provided between the acquisition device and the bulkhead to allow for expansion of the bulkhead. Since access to the LH_2 tank is at the forward end of the tank, access panels through the acquisition device are not required. Inspection panels may be provided in a start basket or start tank for cleaning and inspection.

2.7 RECOMMENDED ACQUISITION SYSTEM CANDIDATES

Tables 2-10 and 2-11 summarize the results of Task I. Recommended acquisition device fluid conditioning candidates and thermal conditioning candidates for the capillary device, boost pump and propellant duct are described in Table 2-10. Recommended fabrication candidates are listed in Table 2-11.

In order to use program resources most efficiently, the recommended systems in Table 2-10 were analyzed to determine which system combinations were most desirable. These system combinations were then focused upon for Tasks II and III.

The process of discriminating between these systems has been formulated into a decision tree shown in Table 2-12. Decisions have been structured so that answering a question affirmatively allows adoption of a less complex, lighter and less costly system (on the left) while answering negatively forces adoption of the more complex, heavier and more costly system (on the right). The main design drivers, considering the Centaur D-1S, D-1S(R) RLTC and other advanced versions of Centaur, are cost, complexity and weight; with complexity and cost appearing to be the most important.

The first decision to be made is whether settling can be used to successfully refill the capillary device. If the answer is positive, a start basket or start tank system can be used. If settling will not refill the capillary devices, channels refilled by pumping will be studied. The system using channels refilled by pumping is heavier than the start basket and start tank and is more complex because it has a lower state of development, requires rotating machinery and will probably require an orbital test to prove out its operation. Looking at the left side of the tree, the next decision to be made is whether thermal subcooling can be used to provide NPSH for the contained fluid and thus eliminate the need for main tank pressurization. If this is answered affirmatively, the lighter weight, lower cost, refillable start basket system will be utilized. If thermal subcooling will not successfully provide NPSH requirements, the start tank system will be chosen to minimize main tank pressurization system requirements. Going down the tree, the subsequent decisions affect

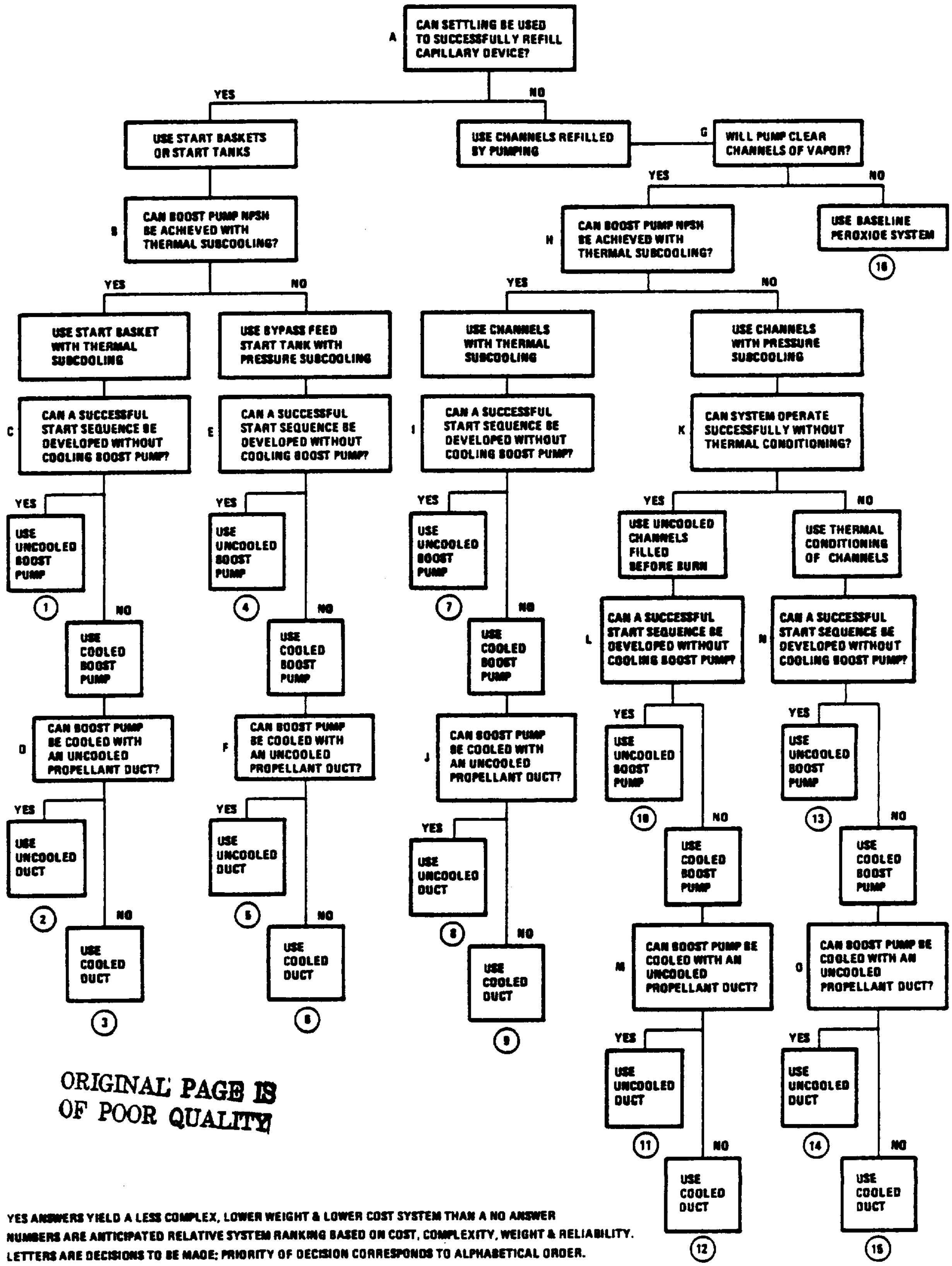
Table 2-10. Recommended Acquisition System Candidates

Fluid Conditioning Candidates	Capillary Device Thermal Conditioning Candidates	Boost Pump Thermal Conditioning Candidates	Propellant Ducting Thermal Conditioning Candidates
1. Start Basket. Screen device over the outlet and sump. Provides liquid for thermal conditioning requirements between burns from a liner or channel sized to remain full during the entire mission. Propellant duct cooling and or childown as well as capillary device cooling requirements are supplied from this liner. The basket is sized to provide liquid outflow to the main engines during fluid settling and collection and capillary device refill. Screened compartments are required in the LH_2 tank start basket to maintain liner flow between burns and liquid over the outlet. Lightest weight and least complex fluid containment concept. If main tank pressurization is required, however, cold gas pressurization requirements severely penalize this system. 2. Start Tank - Bypass Feed Device. Separate tank, approximately the same volume as the start basket is contained near the sump of the main tank. Outflow requirements, inner screened compartments, and lines and channels are similar to the start basket. Valves are used to control outflow and refilling as described in Table 2-3, Concept 13 b. Only start tank must be pressurized for main engine start. 3. Channels Refilled by Pumping. This concept does not rely upon fluid settling for refill. Uses channels that are wetted after a main engine burn by capillary action. When channels are completely wetted, a pump connected between the tank contents and a channel manifold begins pumping. Channels maintain contact with the liquid pool in the tank so that liquid circulates through the channels as the fluid in the channels is pumped back into the tank. Surface tension retention capability of the wetted channel keeps vapor from flowing into the channels during pumping. System operation produces full channels just before burn. During the main engine burn most of the liquid spills from the channel towards the outlet because retention capability is exceeded. Acquisition device weight is higher than the other two recommended candidates. Requires unique flow analysis compared to other two candidates (settling and spilling, wicking, wetting and internal vapor flow). Will probably require orbital test before system is adopted. (Concepts 1 and 2 will probably not require an orbital test).	1. Acquisition Device Cooling. Uses active cooling coils wrapped around the device. Cooling coils are fed throttled vent fluid from compartments inside the acquisition device. The primary cooling mode is continuous flow although intermittent flow designs will be considered. Cools contained fluid sufficiently to prevent screen drying. 2. Thermal Subcooling. Uses active cooling coils similar to concept 1 only subcools contained fluid sufficiently to provide boost pump NPSH. This concept could eliminate the main tank pressurization system potentially reducing system weight by 540 lbs. Useful for start basket and channels refilled by pumping concepts. 3. Pressure Conditioning. Uses cold helium to suppress vaporization in the contained liquid. Because of large pressurization system weight penalties this system is only applicable for the start tank. Since system 1 is lighter, pressure subcooling will be used as a backup should system 1 be too complex or difficult to apply to the start basket. Useful for the start tank in providing boost pump NPSH and suppressing boiling between burns.	1. Wrap Drive Shaft Area Near the Pump With Cooling Coils. Heat is taken out near the contained fluid. Heat can be removed readily from the drive shaft housing but heat removal from the drive shaft and impeller between burns is an extremely difficult problem. 2. Purge the Turbine With Cold Helium. This system requires a cold helium purge. Drive shaft cooling is difficult. No pump modifications are required and thus is least complex. Removes some of the heat directly. 3. Purge the Turbine With Cold Helium and Use Cooling Coils to Intercept Incident Heating. More uniform cooling and design flexibility in removing heat by two methods. Pump cooling from sources other than the turbine is handled by the cooling coils. If Candidates 1 to 3 are not satisfactory in eliminating vaporization in the contained boost pump fluid between burns, the more complex candidates employing gearbox purging or drive shaft purging will be required. Modifications required to implement these more complex cooling schemes are of sufficient complexity to severely jeopardize their adoption.	1. Wrap Duct Length With Cooling Coils. Throttle vent fluid and wrap cooling coils around duct. (Consider the use of hydrogen to cool the LO_2 duct). May require large amounts of cooling fluid compared to flushing and chilling down the lines before each burn. This could cause a significant increase in acquisition system volumetric requirements. Advantages are elimination of; line chilldown, time in start sequence required for engine chilldown, and propellant bleed lines. Also might be easier to cool the boost pump if duct is cool, simplifying the engine start sequence. Adds to system complexity because of cooling coils wrapped around the duct. 2. Flush Propellant Lines at Either a Low Continuous Rate or at a High Rate for a Specified Time Period Just Before MES. This eliminates the complexity of wrapping cooling coils around line. Flushing may require running the boost pumps. Lower cooling quantities used than Concept 1. May not be completely compatible with cooled boost pump option. (A portion of line just downstream of the boost pump may be required to be cooled in order to efficiently keep the boost pump cooled).

Table 2-11. Fabrication Candidates

Component or Process	Fabrication Alternatives
Screen Material	1. Aluminum screen where available 2. CRES screen for low micron ratings where aluminum is not available
Screen Mesh	1. Dutch twill screen for wicking applications 2. Square weave screen where refilling is of overriding importance.
Screen Pleating	1. Non-pleated screens are the baseline 2. Pleated screens where fabrication is not a problem and surface area requirements are high.
Screen Backup	1. Perforated aluminum plate is the baseline 2. Coarse screen should be used if extra stiffness is important. 3. Open isogrid offers increased strength.
Screen Attachment	1. Resistance welding is the baseline method 2. Bolting should be used where the screen is required to be removable.
Cooling Tube Attachment	1. Dip brazing for small devices 2. Resistance welding of extruded webbed tubes for large devices.

Table 2-12. Centaur Propellant Acquisition System Schematic of Decision Making Process (Decision Tree)



YES ANSWERS YIELD A LESS COMPLEX, LOWER WEIGHT & LOWER COST SYSTEM THAN A NO ANSWER
NUMBERS ARE ANTICIPATED RELATIVE SYSTEM RANKING BASED ON COST, COMPLEXITY, WEIGHT & RELIABILITY.
LETTERS ARE DECISIONS TO BE MADE; PRIORITY OF DECISION CORRESPONDS TO ALPHABETICAL ORDER.

feed system complexity. It is desirable to make no changes to the existing boost pumps and propellant ducts. As shown on the tree, the next question in the left most branch will be whether a start sequence can be developed without cooling the boost pump. If boost pump cooling is required then the need for feedline cooling will be assessed.

On the right half of the tree, the concept using channels refilled by pumping is evaluated. The first question is whether the system will successfully clear vapor from the channels between burns. If this cannot be accomplished, none of the recommended capillary systems will be satisfactory and the baseline peroxide system will win by default. If the system will clear vapor, the next question is whether thermal subcooling can be achieved. If this cannot be done, the need for thermal conditioning will then be determined. Since the channels could be pumped full just prior to the start sequence, it is possible that active thermal conditioning (other than the channel pumping) would not be required. Other decisions are similar to those discussed in the left hand branch of the tree.

Each of the small circles at the end of each branch denotes a system. Numbers represent the preliminary ranking of the systems in terms of desirability. The most desirable system, for example, is (1) with the start basket using thermal subcooling and no boost pump or propellant duct cooling. This system appears to be several hundred pounds lighter than the baseline system. It will be similar in complexity to the baseline system because addition of the acquisition system will be offset by elimination of the main tank pressurization system. System (2) also would be potentially several hundred pounds lighter than the baseline system but would be more complex since cooling coils and purging would probably be required to cool the boost pump. System (3) is more complex due to cooling coils required for the duct. Systems (4), (5), and (6) are heavier than systems (1), (2) and (3) due to pressurization system requirements. In terms of complexity, the start tank of system (4) has an extra tank, three or four valves and a start tank pressurization system compared to the boost pump and feedline cooling of system (3). System (4) is thus at least as complex as system (3) and is heavier in weight. Similar arguments can be made for the other relative rankings given.

Priority was given in Tasks II and III to answering the critical questions represented in the decision tree; A. Can settling be used to successfully refill the capillary device? B. Can boost pump NPSH be achieved with thermal subcooling? C. Can a successful start sequence be developed without cooling the boost pump? These questions were answered affirmatively and system (1) using a start basket with thermal subcooling and an uncooled boost pump was selected as one of the systems to be designed. In order to have two distinctly different systems for design and comparison, the other system eventually selected was system (4), using a bypass feed start tank and an uncooled boost pump.

SECTION 3

TASK II, FLUID ANALYSIS

Start tank and start basket fluid analyses were performed in order to determine capillary acquisition volumetric requirements and performance. Initially the critical questions in Table 2-12 were addressed; Can a successful start sequence be achieved without cooling the boost pump? Can settling be used to successfully refill the capillary device? A successful start sequence was developed and a conservative analysis affirming successful refilling with settled fluid was performed. Fluid analysis then was continued by determining the effect of start transients and vibrations on capillary device liquid retention. Start basket and start tank sizing was then performed, based on start sequence, thermal conditioning, residual, and channel volume requirements. Wicking to provide flow for maintaining wet start basket screens was analyzed. Problems of filling on the ground and possible abort of Centaur while in the cargo bay of Shuttle were addressed. The interaction of the propellant utilization system with the start basket was considered.

3.1 START SEQUENCE

Several start sequences were evaluated for an initially dry sump, pump and propellant duct. Both the existing start sequence on the Centaur D-1T (identical to the baseline Centaur D-1S start sequence) and concepts being considered for advanced versions of Shuttle-based Centaur were evaluated. The baseline Centaur D-1S start sequence entails turning on the boost pumps after the propellant is settled and opening the engine shutoff valves when the boost pumps are up to speed. Chilldown of both engines with both LH_2 and LO_2 occurs for a preset time determined prior to the mission.

The advanced concepts considered included the use of trickle chilldown, dual speed boost pumps, additional vent valves, split chilldown, preprogrammed chilldown time, and chilldown time controlled by temperature sensors. The starting sequence selected was based on modifications that would add minimum additional complexity to the existing Centaur and would not require requalification of the engines.

The trickle chilldown option flows through the engines at a low flow rate with boost pumps not operating to maximize heat transfer between the fluid and the engines. This saves propellant and reduces capillary device volume but it is not required for capillary device operation. Since it would require engine requalification, trickle chilldown was eliminated from consideration.

Dual speed boost pumps can allow low chilldown flow rates that make the chilldown process more efficient. This concept has already been qualified and would save a small amount of engine chilldown propellants [30 lb (136 kg)] for RLTC. This is not enough to warrant the increase in complexity compared to the baseline D-1S start sequence.

Split chilldown offers the option of shutting off the LO_2 or LH_2 being used for chilling if chilldown does not occur simultaneously. This option would save about 6 lb (2.7 kg) of payload for the five burn mission of D-1S (plus about one cubic foot of capillary device volume). This savings does not warrant the added complexity and possibility of engine requalification.

Temperature controlled chilldown involves sensing component temperatures in order to determine when to close or open valves. The termination of sump and pump chilldown will be temperature sensed in order to save propellant. This change is relatively simple to implement.

Any engine testing required due to the recommended start sequence can be accomplished within the scope of the anticipated D-1S engine. The anticipated testing is to check out engine performance under the wider range of engine temperature conditions to be experienced with the Shuttle-integrated Centaur.

The recommended start sequence using start baskets and propellant ducts is:

1. Open the fuel and oxidizer shutoff valves (upstream of engines) and flow through the system until the pump and sump are chilled and filled.

2. Close the fuel and oxidizer shutoff valve (optional).

3. Start the boost pump and chill down the lines through the recirculation system. (If fuel and oxidizer valves remain open, this fluid is dumped overboard).

4. When the boost pump is up to speed, open the shutoff valves and use a normal chilldown sequence for the engine.

The start sequence selected resembles the existing Centaur start sequence as closely as possible. The main difference lies in the fact that the existing start sequence settles the propellant prior to start and therefore has the boost pump full. The capillary device systems have a dry boost pump upon start sequence initiation.

In order to chill the boost pump and sump and fill it with liquid, the engine shutoff valve is opened to "vent" the feed system to vacuum providing the necessary driving pressure. After the boost pump is filled, the start sequence proceeds identically as is currently employed.

Two options exist in the current start sequence. One option uses the recirculation system to return flow into the tank during boost pump start-up. The other option dumps this fluid directly overboard through the engines by keeping the fuel and oxidizer shutoff valves open during boost pump start-up. When fluid is dumped overboard during the entire start sequence, engine chilldown occurs at a more efficient flow rate than for the sequence where boost pump start-up and line chilldown fluid is recirculated. This would cause engine chilldown losses to be lower for direct dumping. For the present analysis, the compensating effects of efficient engine chilldown and propellant dumped overboard are assumed to be offsetting.

The capillary device must supply all liquid required during the start sequence before the main liquid pool is settled. Fluid requirements during steps 1, 2, 3 and 4 were evaluated both for cargo bay heating conditions and orbital heating conditions, as shown in Tables 3-1 and 3-2. The volumetric requirements are consistent with the start sequence thrust profiles shown in Figures 3-1 and 3-2.

Table 3-1. Engine Start Sequence Capillary Device Requirements (Maximum Cargo Bay Heating Conditions)

	Mass Required, lb_m (kg)			
	LO_2		LH_2	
Sump and Pump Chilldown and Vent	7.10	(3.2)	18.9	(8.6)
Sump and Pump Fill	105.00	(47.7)	9.4	(4.3)
Boost Pump Start Up	107.60	(48.9)	24.2	(11.0)
Engine Chilldown	71.00	(32.2)	65.0	(29.5)
	290.70 lb_m	(132.0 kg)	117.5 lb_m	(53.4 kg)
	4.24 ft^3	(0.12 m^3)	27.2 ft^3	(0.77 m^3)

Chilldown requirements for the pump and sump were based on saturated liquid entering the sump area and saturated vapor leaving. A payload penalty of 44 lb (20 kg) results for the 5-burn mission compared to the existing start sequence where the boost pump and sump chilldown fluid is recirculated. When fluid is dumped through the engines, additional heat capacity of the vapor can be used to

Table 3-2. Engine Start Sequence Capillary Device Requirements (Nominal Orbital Heating Conditions)

	Mass Required, lb_m (kg)			
	LO_2		LH_2	
Sump and Pump Childown and Vent	1.6	(0.7)	7.1	(3.2)
Sump and Pump Fill	105.0	(47.7)	9.4	(4.3)
Boost Pump Start Up	107.6	(48.8)	24.2	(11.0)
Engine Chilldown	57.0	(25.9)	48.0	(21.8)
	271.2 lb_m	(123.1 kg)	88.7 lb_m	(40.3 kg)
	3.96 ft^3	(0.11 m^3)	20.53 ft^3	(0.58 m^3)

chilldown the engines and reduce this chilldown fluid equivalent payload penalty by 10 lb (4.5 kg) for the 5-burn low earth orbit mission. (Some of the cooling capacity is used to chilldown the lines, but since this fluid can be recirculated, no weight savings results from this chilldown.) If all the cooling capacity of the chilldown fluid can be used, and the fluid leaves at the pump temperature, a minimum payload penalty of 18 lb_m (8.2 kg) results. (No fluid is "wasted" in chilling down the lines.)

Flow rates for the boost pump not operating are based on RL10 engine data for retro-maneuver blowdown. Engine chilldown requirements were obtained from D-1S mission profiles, taking into account that ground chill of the engines with helium would not be used.

The work documented in this section indicated that a successful start sequence could be developed for an initially dry boost pump and sump. This affirmatively answered question C on the decision tree (Table 2-12). Tables 3-1 and 3-2 were used with Figures 3-1 and 3-2 in order to determine the capillary device outflow volume required to settle the propellants. This is described in Section 3.2. Capillary device refilling was then examined (question A on Table 2-12) by using capillary device preliminary volumes and allowable refilling time (refilling time allowable = total burn time - settling time).

3.2 SETTLING

Examination was made of existing methods of predicting propellant settling time in order to determine their applicability to Centaur D-1S settling with acquisition device outflow. For the existing peroxide settling system, the settling process occurs at 24 pounds (106.8 N) of thrust. For the acquisition system, settling occurs during the

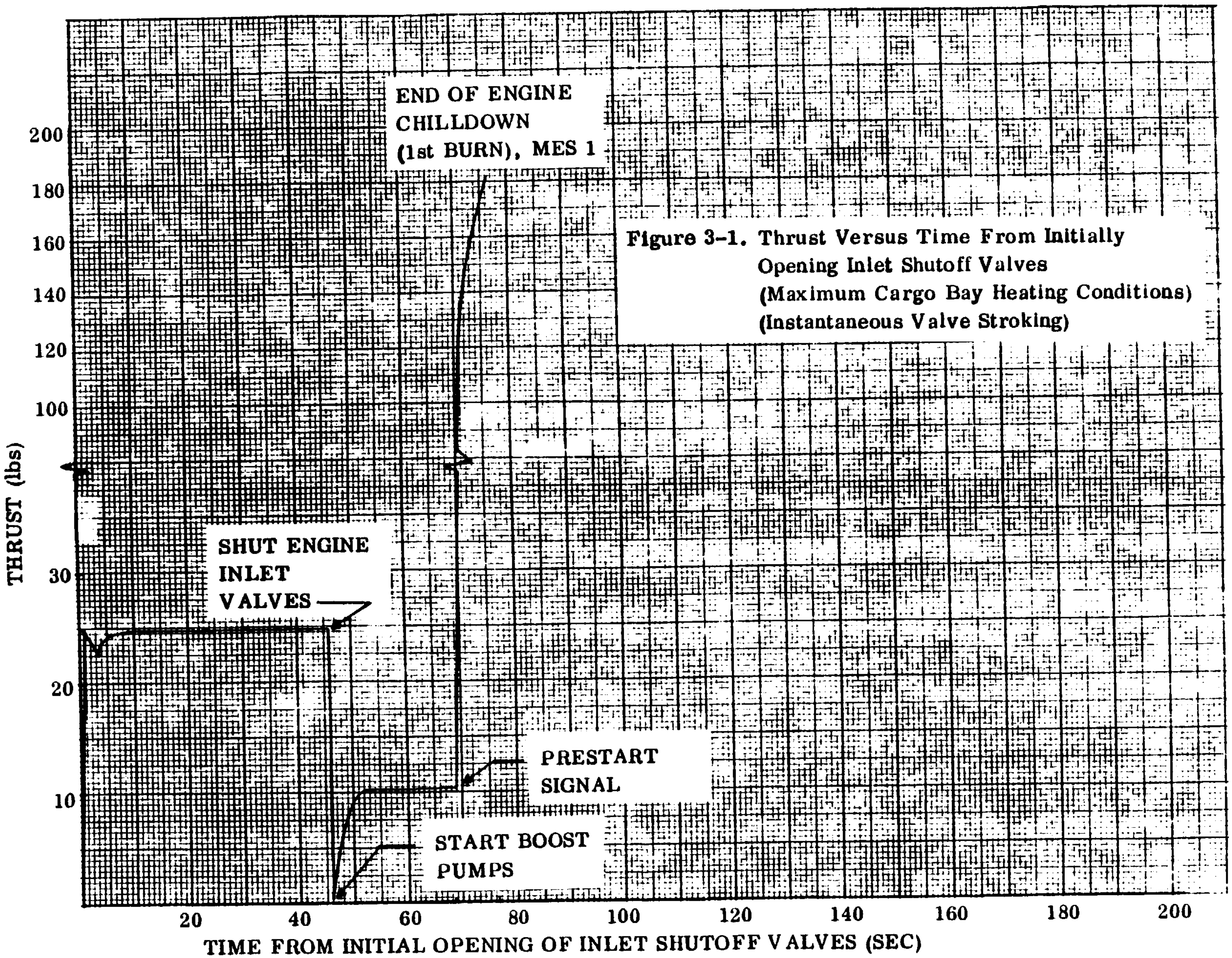


Figure 3-1. Thrust Versus Time From Initially Opening Inlet Shutoff Valves (Maximum Cargo Bay Heating Conditions) (Instantaneous Valve Stroking)

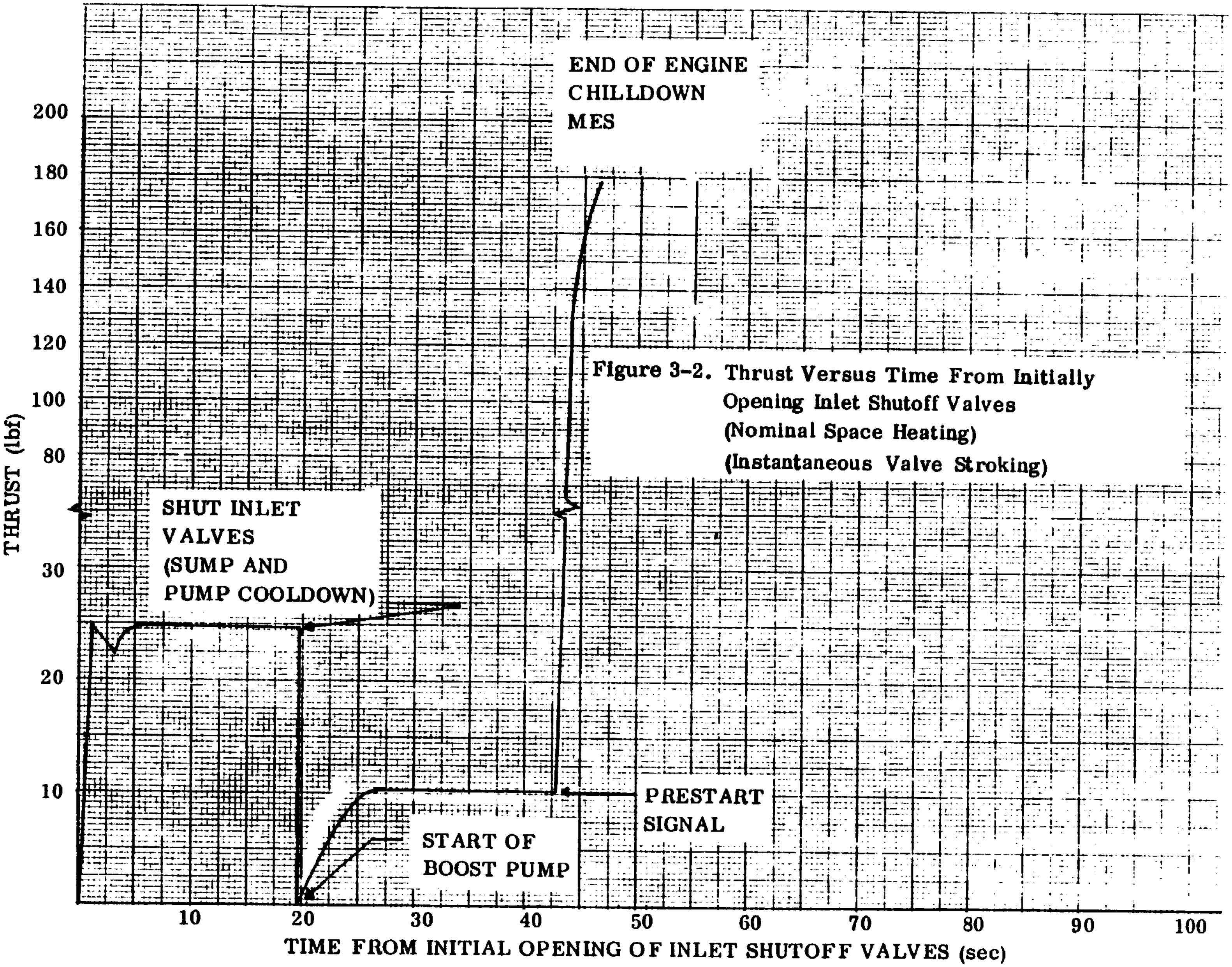


Figure 3-2. Thrust Versus Time From Initially Opening Inlet Shutoff Valves (Nominal Space Heating) (Instantaneous Valve Stroking)

start sequence with thrust levels as shown in Figures 3-1 and 3-2. Thrust builds up to a maximum of 30,000 pounds (133.5 kN) during the final stages of settling.

Capillary device volume is a function of the amount of time required to settle the propellant since flow must be provided directly from the capillary device until settled fluid begins to refill the capillary devices.

Propellant settling was examined for each Centaur D-1S mission and engine firing. Several correlations were used. Initially, NASA/LeRC drop tower correlations were utilized (Ref. 3-1 and 3-2) to determine liquid motion down the tank side wall and collection in the outlet area. These correlations were found to be applicable to low Weber number flow regimes and for settling for providing liquid-free venting rather than for providing vapor-free liquid outflow for engine restart. Similar conclusions were reached for the results of LMSC drop tower test correlations (Ref. 3-3) and Centaur settling predictions (Ref. 3-4).

McDonnell Douglas normal gravity test correlations (Ref. 3-5) break the settling phenomena into several time intervals: the time to impact the aft bulkhead, turbulent and laminar dissipation time, slosh decay time, and bubble rise time. Test results, obtained by stretching diaphragms over the liquid positioned in the forward end of the tank and then visually observing fluid settling motion when the diaphragm was pierced, were presented in the form of slosh decay, turbulent dissipation, and laminar dissipation factors. Tests were run for simple cylindrical tank geometries. Two problems exist in applying these results: the tank geometries for both tanks are more complex than the test tank geometries and low gravity interface shapes and initial surface perturbations that could cause Taylor instabilities are difficult to control using diaphragms to position liquid. Another problem is that values of the semiempirical coefficients, to be used in the expressions quantifying the time intervals, cannot be determined from the information presented. Either more test data, better correlation, or a more lucid rationale for computing the required coefficients is required.

The simplified Marker and Cell Technique, SMAC (Reference 3-6), is a technique that is applicable for evaluating point designs. This technique embodies a finite difference solution to the Navier-Stokes equations and is particularly useful in the high Bond number and Weber number regimes where geysering and recirculation become dominant. Due to its running time and complexity, the SMAC model has limited predesign value.

Another method of computing settling time is an extremely crude approximation sometimes used for predesign calculations. This method merely multiplies the free-fall time (the time between initiation of thrust and liquid impingement on the aft bulkhead) by some constant, as high as five, in order to account for liquid geysering and energy dissipation after liquid impingement on the aft bulkhead. The justification in using an approximation of this type is that the constant can be chosen to yield a conservative settling time value and that no better simple method is available at this time.

Settling time was predicted for the thrust levels in Figure 3-1 and 3-2 using five times the free fall time. Each thrust period was treated separately with a computation made to determine the free fall distance travelled. When 25 times the initial distance between the liquid positioned in the forward end of the tank and the aft bulkhead was travelled (25 X_L) the settling process was considered complete. (After settling was completed in the LO_2 tank, thrust barrel refilling times were computed.) The settling time was found by accumulating the distance travelled under free fall:

$$X_L = \frac{1}{2} at^2$$

where,

X_L = the distance travelled by the fluid

a = the vehicle acceleration

t = the duration of the acceleration

3.2.1 SETTLING CALCULATIONS. Calculations were performed for each burn of the three missions. For the start sequence, the average thrust profiles were assumed as shown in Tables 3-3 and 3-4.

Table 3-3. Cargo Bay Heating-Start Sequence

Period	ΔTime	Thrust
Sump and Pump Chilldown	46 seconds	25 lb_f (111.25N)
Boost Pump Startup	20 seconds	10 lb_f (44.5N)
Engine Chilldown	30.5 seconds	130 lb_f (578.5N)

Table 3-4. Nominal Orbital Heating-Start Sequence

Period	ΔTime	Thrust
Sump and Pump Chilldown	19 seconds	25 lb_f (111.25N)
Boost Pump Startup	20 seconds	10 lb_f (44.5N)
Engine Chilldown	26.5 seconds	130 lb_f (578.5N)

Settling distances for the start sequence were compared to 25 X_L. If settling was not completed during the start sequence, main engine thrust at 30,000 lb_f (133.5 kN) was used to make up the remaining settled distance. The worst case for the LH_2 tank proved to be burn 2 of the synchronous equatorial, 2 burn mission. For this burn an additional volume of 9.54 ft^3 (0.27 m^3) was required in addition to the start sequence volume of Table 3-2. This corresponds to 3.83 seconds additional main engine burn time. For the LO_2 tank, the worst case settling time also occurred for the second burn of the two-burn mission. Additional volume of 1.41 ft^3 (0.04 m^3) was required at main engine thrust (1.70 seconds of full main engine flow) to settle the LO_2 in addition to that indicated in Table 3-2. These volumes were used for sizing the start baskets and start tanks. (For cargo bay heating, start sequence thrust time is longer due to the higher initial component temperatures and resultant longer chilldown times. This longer thrust time is sufficient to settle propellants prior to initiation of full main engine thrust).

3.2.2 <u>THRUST BARREL REFILLING.</u> Calculations were performed to determine the time required to fill the LO_2 thrust barrel. Calculations were initially performed for both low thrust levels and main engine thrust levels.

The thrust barrel for the baseline Centaur D-1T and D-1S is a cylindrical shell, 24.73" (0.63 m) radius and 16" (0.41 m) high, placed symmetrically over the outlet to distribute the load from the thrust structure. On the top surface of the thrust barrel are 1-1/2" (3.81 cm) and 4" (10.2 cm) diameter holes with a total flow area of 1.18 ft^2 (0.11 m^2). On the side of the thrust barrel, near the bottom are nineteen 2.4" (6.1 cm) diameter holes and sixty-six 0.5" (1.27 cm) diameter holes with a total flow area of 0.69 ft^2 (0.064 m^2). This is shown schematically in Figure 3-3.

An analysis was performed for both stable (Bo < 0.84) and unstable (Bo > 0.84) holes on the top of the thrust barrel. For stable holes, surface tension will resist the passage of vapor out of the thrust barrel and retard the refilling process.

The analysis assumed that liquid covered the thrust barrel completely before refilling commenced. The hydrostatic head must drive the liquid into the basket while permitting an equal volume of vapor to be ejected.

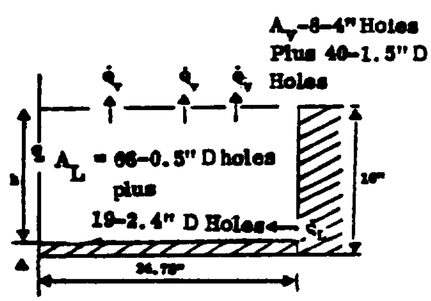


Figure 3-3. Thrust Barrel Refilling

$$Q_L = CA_L V_L = CA_v V_v$$

$$\Delta P_L = \frac{\rho_L V_L^2}{2g_c}, \quad \Delta P_v = \frac{\rho_v V_v^2}{2g_c}$$

$$V_L = \sqrt{\frac{2g_c \Delta P_L}{\rho_L}} \therefore Q_L = A_L \sqrt{\frac{2g_c \Delta P_L}{\rho_L}}$$

For Bo > 0.84, ΔP_σ across the top is zero.

$$\Delta P_L = \frac{\rho_L gh}{g_c} - \Delta P_v, \quad \frac{\Delta P_v}{\Delta P_L} = \frac{\rho_v V_v^2}{\rho_L V_L^2} = \frac{\rho_v}{A_v^2} \frac{A_L^2}{\rho_L} \quad \therefore \quad \frac{\Delta P_v}{\Delta P_L} \sim \frac{1}{45} \text{ and}$$

the vapor pressure drop can be neglected.

$$Q_L = C\,A_L\,\sqrt{2gh}$$

$$Q_L = \frac{dV}{d\theta},\quad dV = A_c\,dH$$

$$\therefore\quad \dot{Q}_L = \frac{A_c\,dH}{d\theta} = C\,A_L\,\sqrt{2gh}$$

$$H = 16 - h \quad\therefore\quad \frac{dH}{d\theta} = -\frac{dh}{d\theta}$$

$$\therefore\quad -\frac{A_c\,dh}{d\theta} = C\,A_L\,\sqrt{2gh}$$

$$d\theta = -\frac{A_c}{C\,A_L\,\sqrt{2g}}\;\frac{dh}{\sqrt{h}}$$

Integrating over the thrust barrel height yields

$$\Delta\theta = \frac{2\,A_c}{C\,A_L\,\sqrt{2g}}\;h^{1/2}\Big]_{h_i}^{h_f} \tag{3-1}$$

For stable Bond numbers at the top holes ($Bo < 0.84$),

$$\Delta P_L = \rho\frac{g}{g_c}\,h - \Delta P_\sigma,\ \text{where}\ \Delta P_\sigma = \rho\frac{g}{g_c}\,h_\sigma$$

$$-\frac{A_c\,dh}{d\theta} = C\,A_L\,\sqrt{2g\,(h - h_\sigma)}$$

$$d\theta = -\frac{A_c}{C\,A_L\,\sqrt{2g}}\;\frac{dh}{\sqrt{h - h_\sigma}},\quad \Delta\theta = -\frac{2\,A_c}{C\,A_L\,\sqrt{2g}}\;\Big[(h - h_\sigma)^{1/2}\Big]_{h_i}^{h_f} \tag{3-2}$$

where,

Q_v, Q_L = vapor and liquid volume flow rate

A_v, A_L = vapor and liquid flow area

C = flow coefficient

V_v, V_L = vapor and liquid velocity

i, f = initial and final

h = head

H = height of liquid in the thrust barrel

A_c = thrust barrel cross sectional area

ΔP_L, ΔP_v, ΔP_σ = liquid, vapor and surface tension pressure drop

ρ_v, ρ_L = vapor and liquid density

g_c = dimensional constant

g = acceleration

θ = time

For cases of interest to the Centaur D-1S, Equation 3-1 applies. The equation was solved for thrust barrel refilling time under main engine thrust. Thrust barrel refilling times proved to be too long. For example, complete thrust barrel refilling time was found to be 6.4 seconds for the fourth burn of the 5-burn mission. Time to refill to the level of the top of the start basket was found to range from 3 to 6 sec for the burn conditions of interest. For a 6-second refill time, outflow volume was equivalent to approximately 5 ft^3 (0.14 m^3) of additional LO_2 capillary device volume. This increase in volume would be detrimental to refilling for several reasons. First, the device height would have to be increased, making it difficult to submerge the device in liquid when the tank is relatively empty on the fourth burn of the five-burn mission. Second, the volume would increase approximately 75%, increasing device refilling time accordingly. Third, the available time for device refilling would be reduced because of the engine burn time taken up by the thrust barrel refilling. (For the fourth burn of the low earth orbit mission, total engine burn time is 18.9 seconds.)

For these reasons, thought was given to reducing thrust barrel refilling time by increasing hole sizes on the top and sides of the thrust barrel. Side holes were increased by an area ratio of six to 4.12 ft^2 (0.38 m^2) in order to get refilling time down to about one second under main engine thrust. Using this side area for liquid flow, the top area was increased to 3.48 ft^2 (0.32 m^2) in order to maintain vapor pressure drop at one-tenth the liquid pressure drop.

The new holes were analyzed structurally (Reference 3-7) to determine modifications required. At this time calculations were performed to determine if main engine settling loads would require beefing up of the LO_2 and LH_2 tank structure. No beefing up was required to the tankage. For the thrust barrel, stiffeners are required on the sides of the thrust structure and increased thickness is required for the ring and membrane at the top of the thrust structure. Total weight increase was 11.2 lb (5.08 kg) for the thrust barrel modifications.

Thrust barrel refilling calculations were performed for the new geometry, assuming that vapor pressure drop is one-tenth the liquid pressure drop. Comparison of capillary device outflow requirements for settling and thrust barrel refilling for each burn for the three missions indicated that the worst case was the second burn of the two-burn synchronous equatorial mission. Main engine settling time for this burn will be 1.70 seconds, requiring additional capillary device volume of 1.41 ft^3 (0.04 m^3) at main engine flow. Thrust barrel refilling volume is 1.28 ft^3 (0.036 m^3) for this burn, corresponding to 1.54 seconds of main engine flow.

3.3 CAPILLARY DEVICE REFILLING

Settling calculations discussed in Section 3.2.1 were used to compute the time available for refilling. For each burn requiring refilling, settling time was subtracted from total burn time to determine available time for refilling. The fourth burn on the five-burn mission was found to have minimum refilling time for both LO_2 and LH_2 tanks. For the LO_2 tank, refilling time available was 18.90 seconds — 1.54 seconds for thrust barrel refilling - 1.70 seconds for settling = 15.66 seconds. For the LH_2 tank, refilling time available was 18.90 seconds — 3.83 seconds for settling = 15.07 seconds.

Refilling calculations were performed for the start basket. Only hydrostatic pressure was assumed as the driving pressure, no dynamic refilling was assumed. Refilling was assumed not to start until settling was complete. Screen wetting was assumed to exist during the entire refilling period. The screen retention pressure thus inhibits refilling during the entire period. Capillary device refilling was computed based on pressure differences between the inside and outside of the capillary device. Outflow to the boost pumps was assumed to exist due to pressure differences in the feed system. Thus, calculations are based only on enough flow to refill the capillary device.

3.3.1 <u>LO_2 BASKET REFILLING</u>. For the LO_2 start basket, calculations were carried out incrementally. As shown in Figure 3-4, the LO_2 basket was broken down into three regions: a sump region, a cylindrical region and a conical region. Equations were formulated and solved for each region as a function of screen area. Screen area will be reduced to approximately 50% with the use of perforated plate for backup material. Additional reductions in open area will be principally due to attachment of cooling tubes.

For the sump region, V = 1.76 ft^3 (0.05 m^3), flow for refilling occurs through the cylindrical and the conical screens. 50 × 50 mesh was used as the basket screen material. The channels, composed of 325 × 2300 screen, are designed to remain full during the entire mission and thus do not have to be refilled by settling.

For the cylinder, $d A_s = \pi D \, dh$

$$h_L = h - h_\sigma - h_v$$

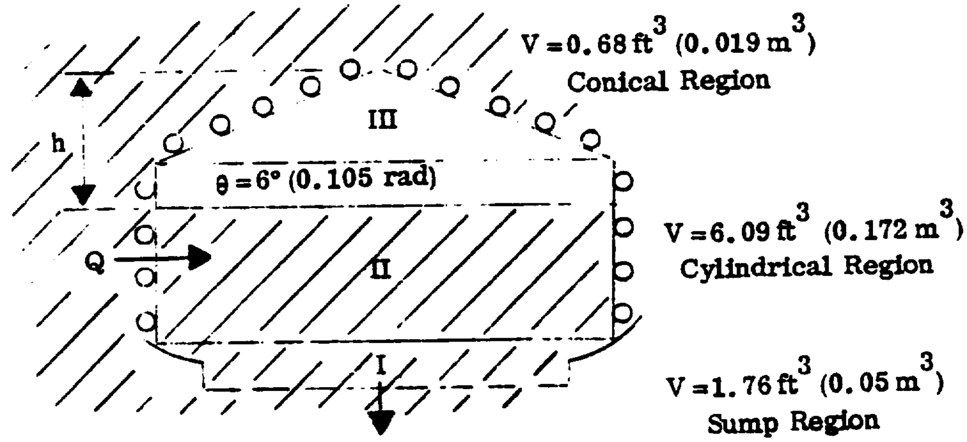


Figure 3-4. LO_2 Start Basket Refilling

for 50 × 250 screen, $h_L = 0.0408\ V_L + 0.207\ V_L^2$

$$h_v = 1.62 \times 10^{-3}\ V_v + 9.1 \times 10^{-4}\ V_v^2$$

$$h_\sigma = 2.17'' \text{ or } 0.18' \ (5.51 \text{ cm})$$

for the worst case at the end of the first burn of the five burn mission, where $\frac{g}{g_o} = 0.83$.

where,

A_s = surface area

h = head

D = basket diameter

h_L, h_σ, h_v = liquid flow pressure loss, surface tension retention head and vapor flow pressure loss

V = volume

V_v, V_L = vapor and liquid velocity

In order to minimize the trapped vapor volume, a 2.17" (5.51 cm) standpipe is added to the apex of the cone. h_σ can be ignored in the flow equations if the standpipe height is not added to the hydrostatic head.

$$h_L = h - h_v, \quad h = h_L + h_v$$

$$\therefore\ h_L + h_v = 0.0424\ V_L + 0.208\ V_L^2$$

Solve for V_L in terms of h with the quadratic formula and substitute into

$$Q = \int_0^{h_{f_1}} V_L \, \pi D \, dh, \quad V_L = -0.102 + \sqrt{0.0104 + 4.81\,h}$$

(Where h_{f_1} is the height of the cylindrical section.)

$$Q = \int_0^{h_{f_1}} -0.102 + \sqrt{0.0104 + 4.81\,(2.11)h} \quad 11.52\,dh \tag{3-3}$$

where, Q = volume flow rate

V_L = liquid velocity

r = cone radius

θ = cone angle

For the conical section, a similar procedure was used to compute sump filling.

A_s = Cone area = 55.1 dh for a 6 degree (0.105 radian) cone

$Q = V_L A_s$

and

$$Q = \int_{h_{f_1}}^{h_{f_2}} (0.102 + \sqrt{0.0104 + 4.81\,(2.11)h}) \quad 55.1\,dh \tag{3-4}$$

where h_{f_1} and h_{f_2} are the bottom and top of the conical section. Integrating equations 3-3 and 3-4 provides a total flow rate for filling the sump.

Sump filling time was then computed from $\Delta t_{sump} = \frac{\text{sump volume}}{Q}$. Refilling time varies linearly with the inverse of the flow rate which varies directly and linearly with the screen open area ratio.

Filling the cylindrical region involves a double integration over the cylinder section because the driving head is changing as a function of time as the cylinder is being filled.

For the same expression for Q_{sump} as used before;

$$Q = \int_0^{h_{f_1}} V_L \pi D_{cyl}\, dh + \int_{h_{f_1}}^{h_{f_2}} \frac{V_L \pi r}{\sin\theta}\, dh \text{ is set equal to } Q = \frac{dV_L}{dt} \text{ where } V_L = \pi A_c\, dh$$

thus, $\pi A_c \frac{dh}{dt} = \int_0^{h_{f_1}} V_L \pi\, dh + Q$ cone is solved incrementally as a function of time.

Refilling time $= \frac{\Delta \text{Volume}}{Q}$.

For the conical section, a similar double integration is required.

$$A_c \frac{dh}{dt} = \int_{h_{f_1}}^{h_{f_2}} \frac{V_L \pi r}{\sin\theta}\, dh$$

Solutions were performed as a function of screen open area for the three regions considered. Results are tabulated as a function of screen open area in Table 3-5.

Table 3-5. LO_2 Start Basket Refilling Time

Screen Open Area (%)	Refilling Time (seconds)			
	Sump	Cylinder	Cone	Total
12.5	0.608	3.856	9.0	13.464
25	0.304	1.928	4.5	6.732
50	0.152	0.964	2.25	3.366
100	0.076	0.482	1.125	1.683

System design calculations indicate that screen open area will be 32%. Thus, refilling will take place satisfactorily. For the fourth burn of the low earth orbit mission, complete refilling will not take place because the liquid level will not cover the basket at the end of the burn. The basket will be filled with 7.9 ft^3 (0.22 m^3) of liquid which will be more than sufficient to provide for thermal conditioning between burns four and five and start sequence and settling requirements for burn 5.

The outer screen finally selected for the LO_2 start basket was 20 × 300 mesh. Calculations were not repeated over those described above because screen flow pressure drop is lower for 20 × 300 mesh compared to 50 × 250 mesh and therefore refilling will be accomplished in a shorter time. The only change made was a reduction in standpipe height to 1.56 inches (3.96 cm) from 2.11 inches (5.36 cm) because of the lower retention capability of 20 × 300 screen.

3.3.2 LH_2 START BASKET REFILLING. An analysis was performed using 50 × 250 screen and assumptions similar to the LO_2 start basket analysis in order to determine refilling time as a function of screen open area. Screened compartments were initially not included in the analysis. A single step procedure was used to compute refilling time. A 5.58" (14.2 cm) standpipe was used to minimize trapped vapor volume.

The LH_2 basket was assumed to have a triangular vertical cross section. The section was a right 45° (0.78 rad) triangle with a leg of 26.8" (0.68 m). Surface area was computed as a function of height. Cross sectional area was computed as a function of height. Flow equations were formulated.

$$Q = V_L \, dA_s$$

$$Q = \frac{dVolume}{dt}$$, where volume as a function of height was

plotted and time was incremented by integrating $Q = V_L dA_s$ in small increments of h. Results obtained are tabulated in Table 3-6.

Table 3-6. LH_2 Start Basket Refilling Time

Screen Open Area (%)	Refilling Time (seconds)
12.5	10.9
25	5.45
50	2.72
100	1.36

The screen separating the bottom compartment from the top compartment (14 mesh) will trap 0.28 ft^3 (0.008 m^3) in the bottom compartment under worst case refilling conditions. This will impede refilling slightly over that indicated by Table 3-6, but not significantly enough to prevent refilling.

Screen open area is anticipated to be 29% based on calculations in the System Design task. Thus, refilling should take place satisfactorily. The outer screens finally selected for the basket surfaces were 40 × 200 and 50 × 250 mesh. The 40 × 200 mesh screen on the top compartment will allow slightly better refilling than predicted in Table 3-6. Use of 40 × 200 screen allows a reduction in standpipe height to 4.32" (10.97 cm)

3.3.3 START TANK REFILLING. Several options were considered for refilling the start tanks including hydrostatic refilling, and pumping the fluid from downstream of the boost pumps back into the start tank. Hydrostatic refilling without venting the start tank to below the main tank pressure could not be accomplished in the required refilling time. Venting the start tank to below the main tank provided sufficient pressure head for accomplishing refilling and was a simpler, less complex solution than pumping the fluid from downstream of the boost pump into the start tank.

Start tank refilling is accomplished by venting the start tanks to 5 psi (34.45 kN/m^2) below the main pressure prior to refilling. The pressure differential between the start tank and main tank is maintained by venting during refilling. For the LO_2 tank refilling should occur in less than 13.5 seconds using the valves shown in Section 5.5. For the LH_2 tank refilling should occur in less than 3 seconds using the valves shown in Section 5.6.

3.4 FEEDLINE TRANSIENTS

During the operational duty cycle of an acquisition device, it will be expected to supply liquid to the boost pump during the main engine start sequence and to contain the liquid at main engine shutdown. Pressure surges in the system caused by valve or pump opening or closing must be analyzed to determine if capillary device retention will be degraded.

Initially a complex start transient analysis was envisioned using the computer models, MAIN (Reference 3-8) and HAMMER (Reference 3-9). The program MAIN is applicable to start transients and uses frictional pressure drop in the lines and flow acceleration pressure drop to generate pressure histories in the feed system. The program HAMMER analyzes transient shutdown by integrating the equations of one-dimensional unsteady flow of a compressible liquid using the two step Lax-Wendroff finite-difference technique.

A simpler model was also identified that modelled pressure changes by considering a compressible fluid travelling in an elastic pipe.

Equations were formulated for "slow" and "fast" opening valves in Reference 3-9. The limitation of this analysis was the lack of correction for the pressure loss attenuation of friction within the pipe, bends, reducers or turbomachinery. This factor will smooth start and shutdown pressure surges.

As a first cut at the transient flow problem, pressure changes in the feed system due to flow acceleration were examined for the start and shutdown sequences. The pressure change due to acceleration;

$$\Delta P_a = \frac{1}{g_c} \frac{\ell}{A} \frac{d\dot{M}}{dt} \sim \frac{1}{g_c} \frac{\ell}{A} \frac{\Delta \dot{M}}{\Delta t} \qquad (3\text{-}5)$$

where,

ℓ = the length of duct having cross sectional flow area, A

$\frac{d\dot{M}}{dt}$ = the rate of change of flow rate

g_c = a dimensional constant

$$32.2 \frac{lb_m \, ft}{lb_f \, sec^2} = 9.81 \frac{kg_m}{kg_f} \frac{m}{sec^2}$$

Maximum anticipated pressure transients were computed using Equation 3-5 for each of the start sequence flow periods. Pressure changes due to acceleration are given in Table 3-7.

Table 3-7. Start Transient Pressure Changes

Event	ΔP - psf (N/m^2)			
	Fuel		Oxidizer	
Open engine shutoff valve to chill and fill sump	1.67×10^{-3}	(0.08)	6.4×10^{-3}	(0.31)
Boost pump startup	1.46	(69.8)	3.0×10^{-1}	(14.35)
Engine turbopump startup	2.38×10^{-1}	(11.4)	3.18	(152.2)

For the LO_2 and LH_2 channels, pressure retention capability is 76.2 psf (3.64 kN/m^2) and 11.4 psf (0.545 kN/m^2) respectively. These are well in excess of the anticipated start transient pressure changes in Table 3-7. Retention requirements during the start sequence are relatively insignificant compared to the steady state period (channels are sized for full main engine flow pressure drop under full main engine thrust). The start basket screens do not have to retain liquid during the start sequence; liquid will spill over the outlet under thrust. Thus, the retention requirements of the start baskets and channels will not be affected by pressure changes due to acceleration.

For shutdown, maximum pressure changes due to deceleration will be approximately 32 psf (1.53 kN/m^2) for the LH_2 system and 18 psf (0.86 kN/m^2) for the LO_2 system. This could cause some backflow of liquid into the basket particularly for the LH_2 system. (Average pressure changes due to deceleration during the shutdown period will be 5.3 psf (0.25 kN/m^2) for LH_2 and 28 psf (0.13 kN/m^2) for LO_2.) The lines should be filled with liquid during this period so the flow through the system should be liquid. Also, the recirculation system, subcoolers and other obstructions should attenuate these pressure surges considerably.

The problem of pressure surges during line chilldown is a possible significant problem when flowing subcooled liquid into an initially warm duct. The magnitude of analysis required to address this problem was beyond the scope of this study. Work required in this area is briefly discussed in Section 6.7.2.

3.5 VIBRATIONS

The acquisition device will experience vibrational loading due to main engine firing, attitude control system firing, and due to functioning of auxiliary equipment (boost pumps, Shuttle equipment, etc.). The vibrations can impose accelerations on the acquisition device or induce resonance at the acquisition device natural frequency that can degrade screen retention capability.

An analysis was performed to determine the vibrational spectrum in the region of the acquisition device and to compare it to the natural frequencies of capillary device sections. Vibrational acceleration was added to acceleration due to thrusting in order to determine total imposed acceleration on the acquisition device.

During main engine firing periods, only the LO_2 and LH_2 channels must retain liquid and prevent gas ingestion. During other periods, both the start baskets and the channels must maintain their retention capability.

For the screen to prevent gas ingestion, the total pressure difference across the screen, including vibration acceleration effects, must be less than the screen bubble point.

$$\frac{\Delta P_t}{BP} < 1.0$$

and

$$\Delta P_t = \Delta P_i + \rho\,(g_o + g\,(rms))\,h$$

where

ΔP_t = total pressure differential across screen

BP = screen retention capability (bubble point)

ΔP_i = vapor pressure differential across screen

ρ = liquid density

g_o = gravity field

g(rms) = root mean square of input vibration acceleration

h = hydrostatic head

Vibration measurements in the vicinity of the acquisition device environment were taken on the Centaur H_2O_2 bottle, a main engine gimbal mount, and the LO_2 boost pump flange during main engine firing and boost pump operation. At selected time periods, this flight data was analyzed for power spectral density (g^2/frequency vs frequency) of random vibration, sinusoidal vibration peaks, g (rms), and overall vibration levels, g (rms), (Reference 3-10). However, the analyzed data cannot be used directly as the acquisition device vibrational environment because vibration response is very location-sensitive and no measurements were taken at the location of start basket attachment to the propellant tanks. The closest measurement was made at the main engine gimbal mount (approximately 6 inches (15.24 cm) from LO_2 start basket attachment) where overall vibration levels were measured as high as 6.9g (rms). These vibrations attenuate considerably with distance from the gimbal mounts. It would be overly conservative and beyond the capability of the retention device to try to design the screens to operate in this environment. To compute vibration levels at the screens, an analytical model should be developed to include both the acquisition device and the path from gimbal mount measurement location to the device attachment location. This type of analysis is beyond the scope of the current study.

A comparison was made between the computed natural frequency of an acquisition device screen/perforated plate and measured sinusoidal vibrations occurring during main engine firing, to check for possible resonance. Lack of analyzed flight data during Centaur attitude control system operation limited the comparison to main engine firing periods.

Recent testing showed that the natural frequency of a screen/perforated plate with liquid on one side can be computed by the equation for a simply supported thin rectangular perforated plate with 1/4 the total liquid mass acting as an effective point mass at the midpoint (Reference 3-11). The 1/4 factor was based on the kinetic energy imparted to the liquid when the screen/plate was deflected. The expression for the natural frequency of the first mode of the screen/plate without the liquid was given by

$$f = \frac{\pi}{2} \sqrt{\frac{g D^*}{\gamma_p t_p + \gamma_s t_s}} \left[\frac{1}{a^2} + \frac{1}{b^2}\right] \qquad (3\text{-}6)$$

and

$$D^* = \frac{E^* t^3}{12 \left(1 - \nu^{*2}\right)}$$

where

g = acceleration

t_p = thickness of the perforated plate

t_s = thickness of the screen

a = width of the screen/plate

b = length of the screen/plate

γ_p = density of the plate

γ_s = density of the screen

D^* = flexural rigidity of the perforated plate

$E^* = 0.265\ E$

E = Young's modulus

$\nu^* = 0.37\nu$

ν = Poisson's ratio

The acceleration term, g, in the above equation is the sum of the acceleration field and either a pure sinusoidal vibration (as during testing) or the sinusoidal part of sine-random vibration (as during actual operation).

Maximum Centaur acceleration, occurring at a time when the acquisition device (channels) must function, takes place at the end of the next to last burn. According to the Centaur mission profiles of Tables 1-1, 1-2 and 1-3, this maximum acceleration is approximately 2.37 g's at the end of the 4th burn of the 5-burn low earth orbit mission. Equation 3-6 can be used to compute the natural frequency of the screen/perforated plate in the LO_2 and LH_2 channels as a function of sinusoidal vibration levels at the most critical flight time (maximum acceleration field) by assuming that

$$g = 2.37 + g\ (\text{rms})$$

The first mode of natural frequency of the LO_2 channel was calculated (without considering liquid in the channel) as a function of sinusoidal g (rms) and is shown as the left-hand curve of Figure 3-5. Considering liquid effects would lower the natural frequency and move the curve further to the left.

The sinusoidal part of sine-random vibratory excitations measured in the LO_2 tank aft bulkhead region, as tabulated in Reference 3-10, are shown as plotted points on the right-hand side of Figure 3-5. The separation between the LO_2 channel natural frequency curve and the measured environmental values indicates that resonance is not likely to occur. However, a more extensive natural frequency analysis in which the total device is modeled should eventually be performed for assurance against premature screen failure due to resonance.

If vibrational analysis had shown that a problem existed with either hydrostatic head pressure or natural frequencies, then relatively simple modifications could be made

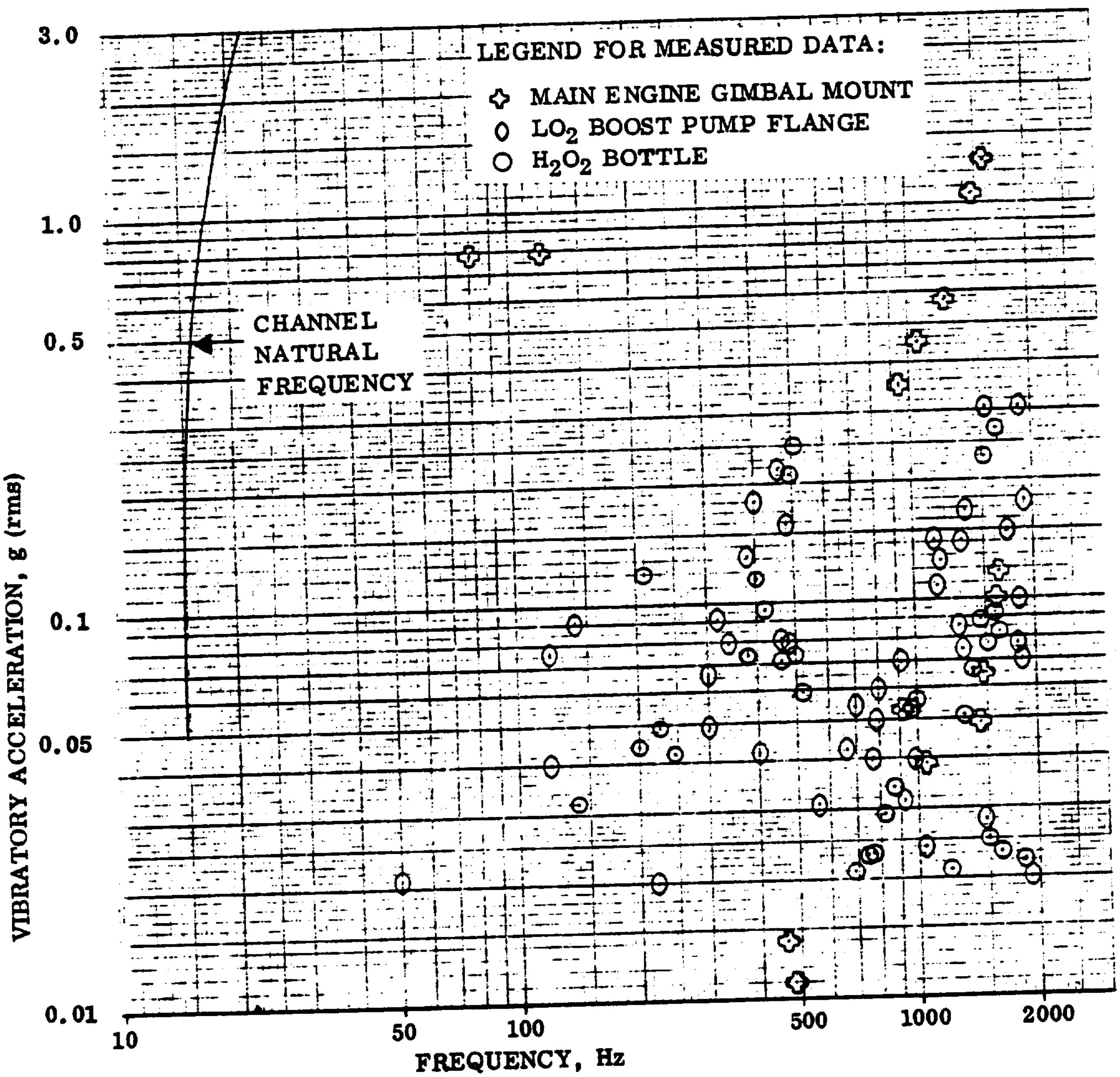


Figure 3-5. Sinusoidal Part of Sine-Random Excitation — for LO_2 Tank Channel Natural Frequency at End of 4th Burn — Measured Values During Centaur Main Engine Firing.

to the acquisition device design to avoid the problem. Softening of the attachment to the propellant tanks by springs, bellows, etc., would reduce screen vibration levels. Other alterations might include changing the size of the unsupported screen/plate area, the plate density (perforations), the flexural rigidity (material), or the thickness.

3.6 CAPILLARY DEVICE SIZING

Capillary device volume must be sufficient to contain all propellants required for; the start sequence and propellant settling, thermal conditioning between burns, and residual fluid to prevent vapor pullthrough. Within the capillary device there will be channels or screened tubes that will stay full of liquid during the entire mission. These channels will be designed to prevent any vapor that may be present in the capillary device from entering the subcooler and sump region.

3.6.1 START BASKET SIZING — LO_2. Table 3-8 shows volumetric requirements for the LO_2 and LH_2 start baskets. Start sequence volume was determined by feed system childown and propellant settling requirements. Thermal conditioning volumetric requirements are discussed in Section 4.2. The following paragraphs discuss configuration requirements and the determination of channel and residual volume.

Table 3-8. Start Basket Volumetric Requirements

Requirement	LH_2, ft^3	(m^3)	LO_2, ft^3	(m^3)
Start Sequence				
Sump and Pump Chill and Vent	1.64	(0.046)	0.02	(0.006)
Sump and Pump Fill	2.18	(0.062)	1.53	(0.043)
Boost Pump Startup	5.60	(0.158)	1.57	(0.044)
Engine Chilldown	11.11	(0.314)	0.83	(0.023)
Settling (Main Engine)	9.54	(0.27)	1.41	(0.04)
Thrust Barrel Filling (Main Engine)			1.28	(0.036)
	30.07	(0.85)	6.64	(0.19)
Thermal Conditioning				
Subcooling Flow	1.53	(0.043)	0.30	(0.0084)
Conditioning Flow	13.6	(0.384)	1.29	(0.037)
Channel Volume	2.17	(0.061)	0.18	(0.0051)
Residual Volume	0.97	(0.027)	0.12	(0.0034)
Trapped Vapor (Bottom Compartment)	0.28	(0.008)		
Total	48.62	(1.37)	8.53	(0.24)

The LO_2 basket is constrained by the thrust structure with a radius of 24.73" (0.63 m). The basket radius is thus 22" (0.56 m) in order to permit installation within the thrust barrel and over the thermal subcooler. The thermal subcooler, a heat exchanger in the LO_2 sump, is discussed in Section 4.1. Basket height is held to a minimum in order to permit refilling on the fourth burn of the five-burn mission. In order to efficiently resist settling loads, the top of the start basket is configured as a six-degree (0.104 rad) cone. The basket thus consists of a cylindrical section topped by a cone, as shown in Figure 3-4.

An iterative procedure was used to size the basket since channel dimensions are dependent upon start basket geometry and start basket volume is dependent upon channel volume.

Before settling thrust is initiated, approximately 1.3 ft^3 (0.036 m^3) of the start basket will contain vapor. The channels have to maintain contact with the liquid in the basket during this time in order to provide low g outflow. Thus the channels must extend up into the basket away from the outlet to maintain sufficient liquid contact area to permit liquid outflow without ingesting vapor. Upon initiation of thrust, liquid in the basket will begin to be settled. Several worst case conditions were assumed in sizing the channels. One condition assumed that the maximum unsupported head existed, with ullage initially positioned over the outlet. The other startup condition assumed that all the flow is through one channel with no unsupported head. Four channels were assumed of 325 × 2300 screen. A high retention capability screen is required to minimize residuals during the final stages of draining.

Channel sizes were evaluated parametrically for both cases, computing flow pressure drop as a function of channel to tank surface area ratio. Channel height was based on the competing factors of increasing the height to minimize screen flow pressure drop and reducing the height to minimize hydrostatic pressure. Pressure loss analysis was also performed for the period just before start basket refilling commences, and the final draining period (in order to determine residuals). The pressure loss analysis compared the screen retention pressure to system pressure losses, where

$$\Delta P_\sigma > \Delta P_h + \Delta P_s + \Delta P_c + \Delta P_b + \Delta P_e$$

ΔP_σ, the screen retention pressure capability $= \dfrac{\phi \sigma}{D_{BP}}$

where,

ϕ = a dimensionless constant depending upon the individual screen and fluid being used

σ = the surface tension

D_{BP} = the bubble point diameter of the screen (for 325 × 2300, $D_{BP} = 10\mu$)

ΔP_h, the hydrostatic pressure difference $= \rho \dfrac{g}{g_c} h$

where,

ρ = the fluid density

h = the differential head supported by the screened channel

g = the acceleration

g_c = a dimensional constant

ΔP_s, the screen pressure loss = $A \mu V_s + B \rho V_s^2$

where,

A and B = viscous and inertial constants determined in Reference 3-12

μ = the fluid viscosity

ρ = the fluid density

V_s = the free stream velocity of the fluid approaching the screen

ΔP_c, the channel pressure loss = $\frac{fL}{D_H} \frac{\rho V_c^2}{2 g_c}$

where,

f = the friction factor for the screened channel, determined from Reference 3-12

L = the length of fluid travel in the channel

D_H = the hydraulic diameter of the flow cross section

V_c = the channel fluid velocity

g_c = a dimensional constant

ΔP_b, the pressure loss due to turning into and out of the channel $\frac{NKEC \rho V_c^2}{2 g_c}$

where,

K, E, C are determined from graphs similar to those in Reference 3-13

K is a pressure loss coefficient depending upon the bend radius

E is an aspect ratio factor depending upon the width and height of the channel

C is a correction factor based on the turn angle

N is the number of bends

ΔP_e, the expansion pressure loss from the channels into the subcooler = $K_e \dfrac{\rho V_c^2}{2 g_c}$

where,

$$K_e = \left(1 - \frac{A_c}{A_{sc}}\right)^2$$

A_c = the channel cross section area

A_{sc} = the subcooler cross sectional area

The LO_2 start basket installation is discussed in Section 5.1. Screen open area of 50% was assumed due to perforated plate area blockage. Channel cross section was selected to be 14.12" (0.36 m) × 1/2" (1.27 cm). Four channels are required. The channels run parallel to the aft bulkhead out to a radius of 18" (0.46 m), then they turn vertically and stop at Sta. 2204.24. Vertical height is approximately 4.35" (11.04 cm). The subcooler in the LO_2 sump and bottom of the tank reduces liquid volume in the basket considerably in this region. Residual calculations performed are cited in Section 6.2. Channel residuals are 0.124 ft^3 (0.0035 m^3) and pool residuals are 0.32 ft^3 (0.009 m^3). Overall residuals depend upon pullthrough suppression in the thermal subcooler. The tank and channel residuals given in Table 3-8 are worst case residuals at burnout. Start basket sizing would not require this much residual volume because the other sizing periods have more severe requirements for thermal conditioning and settling. Thus, using worst case residuals, thermal conditioning and settling volumetric requirements are conservative since all these requirements do not occur during the same time period.

Other screen meshes were examined for the screened channels, such as 200 × 600 and 165 × 800. The crucial factor in selecting the higher retention capability 325 × 2300 screen compared to these lower flow pressure loss candidates was the support requirement for maintaining the channel region extending vertically into the tank full of liquid during the final stages of draining. The 325 × 2300 screen can hold approximately 2.3" (5.84 cm) of unsupported head (with a safety factor of 2) while the other screens will hold less than an inch of unsupported head under full thrust during the fifth burn. The lower retention capability screens cannot be designed both to extend up into the basket to maintain contact with the liquid during thermal conditioning, and to retain sufficient head under main engine acceleration to yield low residuals.

The start basket outer screen must resist the loads indicated in Table 3-9. Using a safety factor of 2, a 69 micron screen will be required should the start basket be completely surrounded by vapor during OMS thrusting. A 50 × 250 (1 WP) mesh plain dutch weave screen, D_{BP} = 65 microns, will be used for this start basket outer screen (1 WP indicates that each shute wire passes alternately over and under 1 warp wire).

Table 3-9. Centaur D-1S Accelerations Affecting Acquisition System Design

Thruster	Thrust		Vehicle Wt. Extremes		Acceleration Limits
	lb_f	(N)	lb_m	(kg)	g's
Shuttle Orbiter OMS	2 @ 6000 = 12,000[1]	(53,400)	210,700[2] 218,100[3]	(95,600) (98,930)	5.70×10^{-2} 5.50×10^{-2}
Shuttle Orbiter RCS	2 @ 900 = 1800[1]	(8,010)	210,700 218,100	(95,574) (98,930)	8.54×10^{-3} 8.25×10^{-3}
Centaur APS	24	(107)	12,698[4] 49,413[5]	(5,760) (22,414)	1.89×10^{-3} 4.86×10^{-4}
Centaur Main Engine	30,000	(133,500)	11,967[6] 36,042[7]	(5,428) (16,363)	2.51 0.83

1. "Orbiter 101, Preliminary Design Review (PDR), Introduction, Volume 1," Rockwell International Space Division - Downey, NAS9-14000, 4 February 1974.

2. Based on OMS deployment weight, from SD72-SH-0120-17, "Space Shuttle Mass Properties Status Report," 2 February 1974, and telecon with Tom Edmunds, Rockwell International Space Division, 5-1-74, using Centaur D-1S weight of 42,000 lb (19051 kg).

3. Based on OMS deployment weight, from SD72-SH-0120-17, "Space Shuttle Mass Properties Status Report," 2 February 1974, and telecon with Tom Edmunds, Rockwell International Space Division, 5-1-74, using Centaur D-1S weight of 49,400 lb (22,408 kg).

4. Centaur D-1S weight before fifth burn of low earth orbit mission. (Minimum vehicle weight during APS thrusting.)

5. Centaur D-1S weight before first burn of planetary mission.

6. Centaur D-1S weight after fifth burn of low earth orbit mission. (Minimum vehicle weight during main engine thrusting.)

7. Centaur D-1S weight after first burn of low earth orbit mission (maximum vehicle weight at the end of a burn for refilling calculations).

This is a conservative selection since OMS accelerations will tend to settle the main tank liquid over the outlet. The tank will be full at this time except for the initial ullage of 4 to 5% used for the missions of interest. For reference, sizing to shuttle ACS conditions would require a bubble point of 152 microns for the LO_2 start basket, based on lateral loads. (The lateral dimension is 44" (1.12 m) compared to the vertical dimension of 14.5" (0.37 m).)

3.6.2 START BASKET SIZING — LH_2. The LH_2 start basket (design details are discussed in Section 5.3) is placed over the LH_2 sump in the bottom of the tank. The basket extends circumferentially around the tank between the intermediate bulkhead and the LH_2 tank side wall. A gap of approximately 30° (0.52 rad) in the basket structure is required to allow entrance of the fill and drain line into the tank.

Within the start basket is a channel of 325 × 2300 Dutch twill weave stainless steel screen designed to remain full of liquid during the entire mission. The channel feeds a thermal subcooler, a heat exchanger designed to provide boost pump NPSH, which is described in Section 4.1. Basket height is held to a minimum to keep retention requirements down, as well as to aid in refilling when only a small amount of fluid remains in the tank. Basket height was determined to be slightly greater than 30 inches (0.76 m).

The LH_2 basket screen selection and internal configuration was driven by the dual requirements of providing sufficient wetted screen area of the channel to permit initial start sequence flow to occur without ingesting vapor into the channel and minimum channel height so that channel retention requirements during main engine thrust can be minimized. To illustrate, attempting to size a single channel to maintain contact with the liquid pool (under worst case conditions of thermal conditioning usage), and the subcooler inlet would result in a channel retention requirement of about 5 microns. This would require the use of multiple layer screens.

A simpler solution is to divide the start basket into screened compartments. An upper compartment allows vapor to enter to replace liquid used for thermal conditioning. A lower compartment, with a greater retention capability than the upper compartment, does not allow vapor to enter from outside the basket. The lower compartment will be maintained full of liquid provided that the upper compartment contains sufficient volume for thermal conditioning plus the liquid required to maintain sufficient wetted area between the two compartments. The channel can be sized to minimize residuals since it will be surrounded by liquid during the initial start sequence.

The outer basket top screen was selected to be 40 × 200 (1 WP) mesh screen (single warp) with a bubble point of 84 microns. This dictates a standpipe height of 4.32" (0.11 m) to minimize trapped vapor during refilling. The outer basket bottom screen will be 50 × 250 (1 WP) with a bubble point of 65 microns. The screen separating the two compartments will be 14 × 14 mesh. Except for the 325 × 2300 stainless steel channel screen, all screens used will be aluminum.

The 14 × 14 mesh screen between the two compartments is configured into a flat membrane with circular screen tubes extending into the upper compartment every 30 degrees. These tubes maintain sufficient flow area so that the total pressure drop of vapor entering the 40 × 200 mesh screen plus the liquid pressure drop in flowing across the 14 × 14 mesh screen from the top to the bottom compartment does not exceed the retention capability of the 50 × 250 mesh bottom compartment screen. The 14 × 14 mesh screen is sufficient to resist Centaur attitude control system accelerations.

Channel sizing was based upon minimizing residuals during burnout. Initially 200 × 600 screen was considered for the channel because of its low pressure drop vs retention characteristics. The severe retention requirements using 2.5 g's at burnout, however, dictated the use of the higher retention capability 325 × 2300 screen. A comparison of the two channels indicated channel surface area would be greater for the 325 × 2300 channel. Thus, channel residuals will be greater for the 325 × 2300 screen but this will be more than offset by lower pool residuals for this screen compared to 200 × 600 screen. Calculations using pressure loss expressions described in Section 3.6.1 indicated the use of an 8 ft (2.43 m) long × 3 in. (7.62 cm) wide × 13 in. (0.33 m) high channel feeding directly into the subcooler. Channel residuals will be 2.17 ft^3 (0.061 m^3) and pool residuals will be 0.97 ft^3 (0.027 m^3).

3.6.3 START TANK SIZING — LO_2 AND LH_2. Start tank venting will not be required between main engine burns (see analysis, Section 4.4). Thus, start tank volumetric requirements are the sum of: start sequence volume, main tank settling volume, screened channel volume, liquid volume required to prevent vapor entering the screened tubes (channels) during the mission, and ullage volume requirements based on anticipated pressure rise rates. LO_2 and LH_2 start tank volumes are 8.45 ft^3 (0.24 m^3) and 36.84 ft^3 (1.05 m^3), respectively.

At the initiation of the start sequence, some ullage volume will exist in both the LO_2 and LH_2 start tanks. For the LO_2 start tank, this is because pressure rise will exceed allowable limits for small ullage volumes, since pressure rise increases rapidly as the ullage goes to zero. For LH_2, this will also be a factor but an additional factor is that total heating between burns will vary and initial ullage volume will have to be sized based on worst case conditions. Thus, even if the LH_2 start tank can be self pressurized to a full condition, this cannot be relied upon unless complicated sensing and mixing equipment are used. For these reasons, screen elements are required in the start tanks to assure that liquid outflow will exist during the start sequence and main tank settling. Pleated screen elements were used similar to those used in Reference 3-14. A matrix of lengths and tube diameters were examined. Pleated screen filter elements were selected with three times the surface area of cylindrical tubes. Tubes chosen were evaluated for pressure loss based on bend losses, channel pressure losses, hydrostatic head and screen pressure losses. A primary selection criteria was that the pressure loss in the channel when fully submerged be less than one-half the total screen retention capability.

For LH_2, two tubes were analyzed extending outward from either side of the outlet. Tube configurations evaluated were diameters of 3" (7.6 cm) 4" (10.2 cm), and 5" (12.7 cm); and tube lengths of 1 ft (0.3 m), 2 ft (0.6 m), 3 ft (0.91 m) and 4 ft (1.22 m). Tubes were placed horizontally in the start tank for sizing purposes. The selected screened tubes for LH_2 were two 5 in. (12.7 cm) I.D. by 2 ft (0.61 m) long tubes.

Four LO_2 configurations were evaluated consisting of four tubes extending out from the outlet in a horizontal position spaced at 90 degrees. Tube sizes considered were 1 ft (0.3 m), 2 ft (0.6 m), 3 ft (0.91 m) and 4 ft (1.22 m) long, and 3 in. (7.6 cm), 4 in. (10.2 cm) and 5 in. (12.7 cm) in diameter. The selected screened tubes for LO_2, based on pressure loss comparisons were four 3 in. (7.6 cm) I.D. by 1 ft (0.3 m) long pleated screen tubes.

Residuals were computed for each configuration. For LH_2, the channel residuals were found to be 0.55 ft^3 (0.016 m^3). Pullthrough at final draining was predicted to occur 1.18" (3.0 cm) above the bottom of the screened tubes. For the LO_2 pleated screens, the channel residuals are 0.2 ft^3 (0.0057 m^3) and pullthrough height was predicted to be 0.6" (1.52 cm) at the end of the fifth burn of the low earth orbit mission. Pullthrough heights were based on horizontal placement of the pleated screen elements. In practice, as indicated in Sections 5.5 and 5.6, the screened tubes had to be canted to fit inside the start tanks and maintain clearance with the start tank wall. Based on this revised placement and calculations approximating the pullthrough height and residuals (described briefly in Section 6.2) overall residuals were determined.

3.7 WICKING

Wicking screens were selected for start basket outer barriers. Work done in Reference 3-15 demonstrated that screen wetting and retention when subjected to evaporation could be maintained more readily with Dutch weave screens (wicking) than with square weave screens (nonwicking). This attribute of wicking screens overrides the possibility that refilling can be retarded by premature screen wetting caused by wicking.

Incident heating that causes screen dryout when using passive thermal conditioning (wicking flow) could occur due to convection or conduction heat transfer around the start basket surfaces when surrounded by vapor. Anticipated maximum heat transfer coefficients of 0.6 Btu/hr-ft^2-°R (3.41 watts/m^2K) for both GO_2 and GH_2 will occur due to forced convection heat transfer during mixing of stratified fluid. Maximum wicking distances from the liquid pool will be 6 and 1.5 feet (1.8 and 0.5 m) for LH_2 and LO_2 respectively.

Equations derived in References 3-16 and 3-17 were used for predicting the distributed heat flux that can be intercepted by wicking flow in a screen. Using an assumed ΔT of 1°R (0.55K), a uniform heat transfer coefficient of 0.6 Btu/hr-ft^2-°R (3.41 watts/m^2K) would cause screen dryout at a distance from the liquid pool of 0.4 feet

(11.4 cm) for LH_2 and 0.8 ft (25 cm) for LH_2. Thus, wicking flow provided by screen alone is unacceptable for capillary device thermal conditioning.

Wicking using screens and perforated plates was found, in Reference 3-18, to give order of magnitude increases over wicking by screen alone. The equation formulated in Reference 3-18

$$Q = \frac{\rho g_c h_{f_g} \sigma K (a + b)}{L^2 \mu}$$

was used to compute the heat flux that can be intercepted by wicking flow with a screen/plate in zero gravity. Where,

Q = incoming heat flux, Btu/sec-ft^2

ρ = liquid density, lb_m/ft^3

g_c = gravitational constant, 32.2 lb_m ft/lbf sec^2 (9.81 kg_m/kg_f-m/sec^2)

h_{fg} = heat of vaporization, Btu/lb_m

σ = liquid surface tension, lb_f/ft

K = screen/plate wicking constant, ft [typically 5.0×10^{-5} ft (15 microns)]

a = screen depth, ft; b = separation of plate and screen, ft
[a + b typically = 6.6×10^{-4} ft (201 microns)]

L = plate/screen distance which will be wetted by wicking, ft

μ = liquid viscosity, lb_m/ft-sec

For ΔT's of 1°R (0.55 K) heat transfer coefficients of 45 and 167 Btu/hr-ft^2-°R (255 and 948 watts/ m^{2}K) were found for LH_2 and LO_2 respectively at a distance of 1 ft (0.30 m) from the liquid pool. These calculations are encouraging but more data is required for determining low gravity wicking rates for non-zero gravity. (Data from Reference 3-18 was for screens and non-perforated plates.) An assessment should also be made of the local heating rates that could exist in areas where temperature differences could be greater than 1°R (0.55K).

3.8 FILLING

Filling of the start baskets and start tanks on the ground is most easily accomplished by backfilling through the outflow line and outflowing and inflowing until the system is full. This can be done for the LO_2 start basket and start tank since the fill and drain line is located in the sump.

For the LH_2 tank, with the tank fill and drain line entering the tank adjacent to the capillary device, this is not possible. For the LH_2 tank a slow fill should be employed so that the channels can be filled prior to being externally covered with liquid. The fill rate can then be increased. A similar procedure could be used in the LH_2 start tank with the refill valve and vent valves open.

The start tanks could use helium pressurant to condense trapped vapor in the channels if the preferred procedure (mentioned above) is unsuccessful. The cooling coils could be used to purge the channels for the start baskets employing active cooling.

3.9 ABORT

Abort considerations discussed in Reference 3-19 indicated that the baseline abort mode for the Centaur D-1S is to dump both propellant tanks and land empty. The minimum abort dump time will be approximately 260 seconds.

Any pressurant required for abort will be carried in the payload bay of Shuttle. Because of this pressurization requirement, the existing helium diffuser will be retained in the LH_2 tank. For the LO_2 tank, a simplified LO_2 bubbler would be employed. These would be required both for the start tank and the start basket. The LO_2 abort dump line is a 3.5" (8.89 cm) I.D. line located in the LO_2 sump. The LH_2 abort dump line is a 4.25" (10.8 cm) I.D. line located on the side of the LH_2 tank away from the sump. LH_2 capillary devices will not affect the abort dump draining since the start tank and the start basket are configured to not interfere with the fill and drain line.

The LO_2 start basket and start tank are between the fill and drain outlet and the tank propellant. According to the calculations performed for Reference 3-19 abort line sizing, the capillary device pressure loss during abort could increase the overall dump line system pressure by 3 or 4 psi (20.7 or 27.6 kN/m^2) with no line size increase required. Pressure loss will be well below this limit for the start tank. For the start basket, calculations were performed to compute the pressure loss in the start basket,channel and subcooler. At a pressure loss of 4 psi (27.6 kN/m^2), only 82 lb/sec (37 kg/sec) could be passed, compared to a requirement of approximately 100 lb/sec (45.4 kg/sec).

For these flow conditions, a 200 × 600 pleated screen is required to seal between the channel and subcooler. This screen (with a flow area to projected area ratio of 3 to 1) supports the channels in a full condition between burns while resisting the penetration of vapor formed in the subcooler.

As a supplement to the flow through the start basket and subcooler, the bypass line that vents the sump to the tank was used. Several line sizes were examined for an 18" (0.46 m) long line with a 90° (1.56 rad) single bend. A line size of 1.5" (3.81 cm) I.D. was selected. This can pass a flow of 35 lb/sec (15.89 kg/sec) at the maximum pressure drop of 4 psi (27.6 kN/m^2). At the required total flow rate of 100 lb/sec

(45.4 kg/sec) the additional pressure loss upstream of the dump system will be 3 psi (20.7 kN/m^2). Approximately 70 lb/sec (31.8 kg/sec) will pass through the start basket and subcooler and 30 lb/sec (13.6 kg/sec) through the bypass line. Thus, no changes will be required in the abort system with the start baskets or start tanks.

3.10 PROPELLANT UTILIZATION

The propellant utilization (PU) system uses capacitance probes in the LH_2 and LO_2 tanks on Centaur in order to sense the amount of mass in each tank. This information is used to automatically adjust the mixture ratio to the engine.

The capacitance probes are functional only during main engine thrust. Between burns in low gravity the capacitance probes,consisting of concentric aluminum cylinders,are filled with liquid due to the dominance of surface forces with the wetting cryogens. (The LH_2 probe consists of a 2" (5.08 cm) diameter cylinder inside a 3" (7.62 cm) diameter cylinder and the LO_2 probe consists of an outer 2-1/2" (6.35 cm) diameter aluminum support tube with a 2" (5.08 cm) diameter and 1-1/2" (3.81 cm) diameter inner aluminum cylinder. Both probes extend over nearly the entire length of each tank. Normally the PU probes are not activated until five seconds of main engine burn time has elapsed. This is to allow the liquid level in the probe to translate from the low gravity full state to the liquid level in the tank. The TC-2 flight indicated that this time period should be closer to ten seconds.

Start basket operation results in vapor entering the start basket between burns. During a start sequence liquid will spill from the baskets. Subsequent liquid collection during settling may wet the start basket causing vapor to be temporarily trapped until refilling can take place. As indicated in Section 3.3, LO_2 refilling will be accomplished in 5.8 seconds and LH_2 refilling will be accomplished in 5 seconds for the worst case refilling burn. For the refilling that takes place at the lowest g level (after the first burn) refilling should be accomplished in less than 10 seconds. The small vapor volume in the standpipes is the only volume trapped in the start baskets after refilling. Thus, if the use of the PU system is delayed until 10 seconds after the main tank fluid is settled, the start basket will have no impact on the normal operation of the PU system.

For purposes of checking out start basket performance during flight operation, it will be advantageous to use liquid level sensors in the start baskets. These capacitance probes should be used on the first few Centaur flights incorporating the start baskets, until their performance is well documented. These capacitance probes will be largely for data taking purposes. On a flight test they could be used to monitor whether the start baskets will be able to successfully perform the next start sequence. The probes will not be tied into the PU system.

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SECTION 4

TASK III, THERMAL ANALYSIS

Thermal analysis was performed in the areas of thermal subcooling, start basket and start tank thermal conditioning, tank pressure control, and boost pump thermal conditioning. Major emphasis was placed on the critical areas of; thermal subcooling to provide boost pump NPSH and start basket thermal conditioning to prevent screen dryout.

4.1 THERMAL SUBCOOLING

As indicated in the decision tree of Table 2-12 and the corresponding discussion, the use of thermal subcooling is critical to the utilization of start baskets because of the high weight penalty of the cold helium pressurization system otherwise required.

In order to provide satisfactory boost pump operation, sufficient subcooling must be supplied to prevent cavitation. The subcooling must be sufficient to intercept heat input to the fluid entering the boost pump as well as to provide boost pump NPSH. These requirements are 4 Btu/sec (4.2 kW) and 0.12 psi (0.9 kN/m^2) for the LH_2 boost pump and 4 Btu/sec (4.2 kW) and 0.72 psi (4.96 kN/m^2) for the LO_2 boost pump. In the existing Centaur, pressurant is used to subcool the liquid flowing to the pumps and suppress boiling. For the start basket application throttled fluid is used to remove heat from this fluid to achieve subcooling.

4.1.1 START BASKET SUBCOOLING. Several thermal subcooling schemes were considered that operated by cooling the capillary device contents before an engine burn in order to achieve boost pump NPSH requirements. This type of system consists of a start basket wrapped with cooling coils whose function is to subcool the liquid contained in the start basket sufficiently to provide boost pump NPSH. For this type of system, any trapped vapor in the contained volume of the start basket causes saturated liquid to be present in the basket. (The amount of saturated and subcooled liquid depends upon the mixing occurring in the basket.) Since delivery of saturated liquid to the boost pump is unacceptable, means were explored for eliminating the presence of saturated vapor and liquid from the basket prior to the subcooling period.

Vapor can be present in the basket from the following sources: vapor entering to replace liquid used for thermal conditioning, vapor trapped during refilling, and vapor entering during spilling. Saturated liquid can enter the basket during engine firing from the following sources: collected liquid impinging on the capillary device and spilled liquid warmed by coming in contact with the tank wall. Means were explored for eliminating each source of saturated vapor and liquid from the start basket.

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Thermal conditioning fluid removal can be isolated from the remainder of the start basket by impervious walls. This would essentially leave one part of the start basket to feed the outlet and the other to feed the thermal conditioning cooling coils with no communication between the two compartments.

Spilling could be prevented, for the start sequence described in Section 3.1, during the low thrust periods by restricting the vapor flow into the top of the start basket. Spilling and vapor ingestion prevention are discussed in Reference 4-1, Section 2.2.2. Vapor ingestion through the side screens should be prevented if liquid in the main liquid pool is to remain subcooled. Preventing vapor ingestion requires the side screens to have surface tension retention capability in excess of the hydrostatic head of the contained fluid and forces the use of multiple barrier, fine mesh screens. This requirement, and the difficulties that would occur in hydrostatically refilling the device, make spilling and vapor ingestion prevention undesirable.

Preventing vapor from being trapped during refilling depends upon the screen retention requirement between settling burns. The top screen on the start basket should be sized to resist the worst combination of contained fluid height and disturbing acceleration. If this retention requirement is low enough, the Bond number of the screen will be unstable under high "g" refilling and no vapor will be trapped. Unfortunately this will not be the case for the Centaur D-1S since the vehicle must withstand Shuttle OMS firing accelerations, as shown in Table 3-9. These accelerations dictate screen retention requirements of approximately 84 microns for LH_2 and 68 microns for LO_2. This is well within the stability limit (Bo = 0.45) of 1850 μ for LH_2 and 1200 μ for LO_2 at 0.83 g's. Thus, unless a refill valve is used, vapor will be trapped within the device during refilling.

Condensation of trapped vapor might be accomplished with vent fluid cooling. Since the basket will generally be surrounded by vapor, this would be a difficult and lengthy process. Condensation of trapped vapor would periodically cause a vapor pressure reduction in the start basket that would result in warm vapor entering the basket from the surroundings.

Impingement of warm fluid on the basket during the settling process is one of the main disadvantages of thermally subcooling the start basket contents. During collection, warm liquid will come into contact with the start basket. This liquid will enter the basket and be delivered to the engines. If the start basket contents are overcooled below the NPSH requirements, then mixing of this warm fluid with the "overcooled" fluid could still satisfy NPSH requirements. This would require a significant increase in cooling requirements in addition to an increase in capillary device volume to assure that subcooled liquid has been collected by engine thrust before the capillary device is depleted.

A combination of the difficulties encountered due to spilling, liquid impingement during settling,and vapor entrapment during refilling, caused consideration of a thermal subcooling scheme that subcools the liquid as it flows into the sump.

4.1.2 HEAT EXCHANGERS FOR SUBCOOLING (THERMAL SUBCOOLERS). Heat exchangers were analyzed for supplying boost pump NPSH by cooling the liquid flowing to the boost pump. This thermal subcooling concept eliminates main tank pressurization and requires pressurization only for auxiliary systems such as the attitude control system. This option utilizes a heat exchanger to deliver subcooled liquid to the boost pump. The heat exchanger concept, shown schematically in Figure 4-1, uses throttled vent fluid, as shown thermodynamically in Figure 4-2, to cool the hot side fluid flowing to the boost pumps.

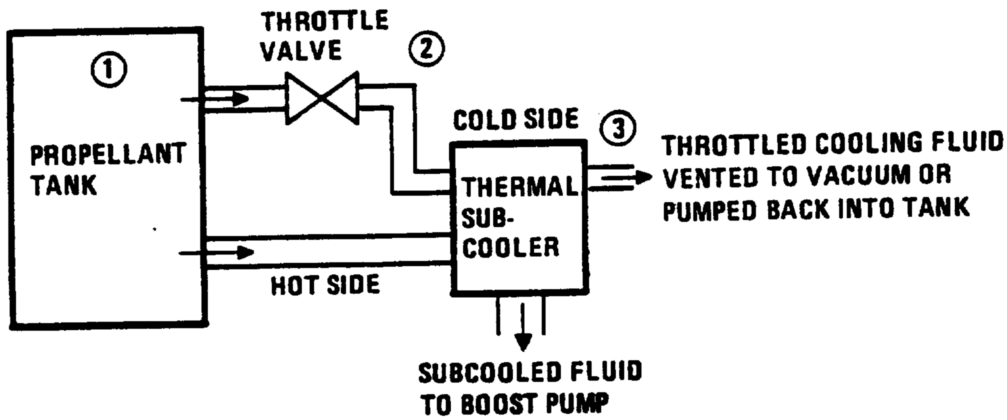

Figure 4-1. Schematic of Thermal Subcooling

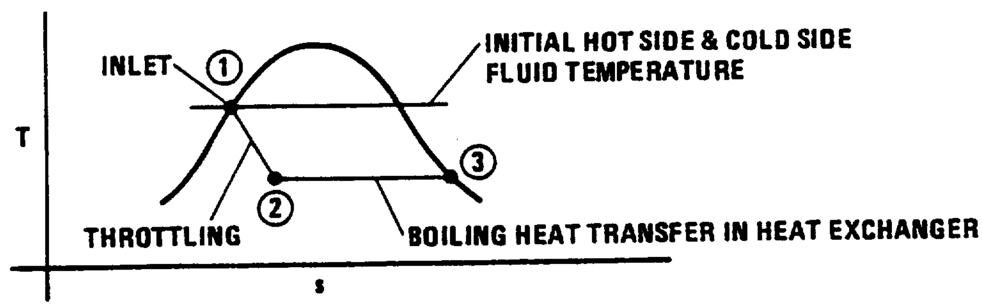

Figure 4-2. Cooling Fluid Thermodynamic States

Sufficient heat must be transferred to remove any heat transferred to the hot side fluid from the warm boost pump and bearings, provide boost pump NPSH, and counteract any pressure drop caused by the thermal subcooler itself.

Screened channels provide liquid flow to the hot side of the exchanger. The cold side fluid is also extracted from the screened channels and throttled to a lower pressure and temperature before entering the subcooler. Multipass parallel flow heat exchangers were utilized. Several configurations were examined for both the LO_2 and LH_2 subcoolers. The objective in designing the heat exchanger surface was to provide high heat exchanger effectiveness coupled with a low pressure drop. The heat exchanger surfaces provide sufficient heat transfer for overcoming pump heating and NPSH requirements as well as system pressure drop. NPSH requirements are 0.72 psi (4.96 kN/m^2) for the LO_2 boost pump and 0.13 psi (0.90 kN/m^2) for the LH_2 boost pump. Heating was assumed to be 4.0 Btu/sec (4.22 kw) from each pump.

Heat exchanger sizing was based upon cooling at both the minimum (boost pump and sump chilldown) and maximum (main engine) flow rates using cooling fluid throttled to five psia (34.5 kN/m^2). This throttle pressure was chosen to give a high ΔT between the hot and cold side as well as to provide sufficient ΔP for driving the cold side flow through the heat exchanger. For the LO_2 subcooler, the inlet conditions are 5 psia (34.45 kN/m^2) and 145.8 R (80.9K) for the cold side fluid, and 31 psi (213.6 kN/m^2) and 176 R (97.6K) for the hot side fluid. For the LH_2 subcooler, the inlet conditions are 5 psia (34.45 kN/m^2) and 30.8 R (17.1K) for the cold side fluid and 20 psia (137.8 kN/m^2) and 38 R (21K) for the hot side fluid.

Heat Exchanger Sizing

Heat exchanger heat transfer requirements were determined from analyzing boost pump NPSH, incident heating to the hot side fluid, and subcooler hot side fluid pressure loss requirements.

The cooling requirement, for NPSH only, is $\dot{Q} = \dot{m}\, C_p\, (\Delta T/\Delta P)$ (NPSH); where $\dot{m}$ is the flow rate, C_p is the liquid specific heat, $\Delta T/\Delta P$ is the slope of the vapor pressure curve between the conditions of interest and $\dot{Q}$ is the heat rate to be removed from the fluid flow to the boost pump. At main engine steady state flow rate conditions, LH_2 boost pump NPSH of 0.13 psi (0.9 kN/m^2) is equivalent to heat removal of 1.38 Btu/sec (1.46 kW). LO_2 boost pump NPSH of 0.72 psi (4.96 kN/m^2) is equivalent to heat removal of 12 Btu/sec (12.7 kW). Heat input to the fluid directly from the boost pump due to bearing heating was conservatively assumed to be 4.0 Btu/sec (4.22 kW) for both LH_2 and LO_2. To obtain the total cooling requirement, any pressure drop in the subcooler and duct, ΔP_L, must be included plus any heat input to the fluid, $\dot{Q}_{in}$, after it leaves the subcooler and before it enters the engine turbopump. The total heat input removed in the subcooler should thus be

$$\dot{Q}_{removed} = \dot{m}\, C_p \frac{\Delta T}{\Delta P} \left(NPSH + \Delta P_L\right) + \dot{Q}_{in} \qquad (4\text{-}1)$$

Which, for LO_2, for example is $\dot{Q}_{removed} = 16.7\ (\Delta P_L)$ + Btu/sec where ΔP_L is in psi and $\dot{Q}_{removed}$ is in Btu/sec.

The heat exchanger to remove this heat input was determined by iteration since the requirement of heat transfer is dependent upon the pressure drop which is dependent upon the hot side flow path and fluid velocity. Thus for each condition, a set of hot side flow areas or plate spacings were considered in order to determine the total pressure drop and total heat transfer area as a function of hot side flow area. Minimum heat exchanger surface area was determined by examining the plotted results. For the LO_2 subcooler, for example, the heat exchanger configuration was divided up into converging and diverging flow paths between parallel plates. Fins were placed between the plates to increase the heat transfer area. Pressure loss and heat transfer calculations were done incrementally by breaking each flow path into at least three sections, computing the flow velocity, heat transfer coefficient and friction factor and applying these over the section length and area to obtain the total heat transfer and pressure loss per flow path.

The total number of paths, n, was then found from

$$\dot{Q}_{removed} = \dot{m}\, C_p \frac{\Delta T}{\Delta P}\left(NPSH + + \Delta P_{\substack{screen+\\ duct}} + n\Delta P_{(subcooler\ per\ pass)}\right)$$

$$+ \dot{Q}_{in} = n\dot{Q}_p \tag{4-2}$$

where $\dot{Q}_p$ is the average heat transferred per pass. An average $\dot{Q}_p$ was used because the heat transfer was slightly different in the converging and diverging passages. The exchanger with the minimum number of passes was generally the optimal exchanger for the geometric constraints (fitting into the LO_2 sump or LH_2 sump region) imposed.

Another way of expressing Equation 4-2 is to convert the ΔP per section into a heat transfer rate;

$$\dot{Q}_{removed} = \dot{m}\, C_p \frac{\Delta T}{\Delta P}\left(NPSH + \Delta P \begin{array}{c} screen \\ + duct \end{array}\right) + \dot{Q}_{in} = n\, \dot{Q}_{np}$$

where $\dot{Q}_{np}$ is the net heat transferred (subtracting the heat transfer required to overcome the pressure loss) per hot side flow passage.

4.1.2.1 Thermal Analysis. The thermal analysis is a heat balance between the hot and cold sides of the heat exchanger. On the hot side, forced convection laminar and turbulent heat transfer equations for flow over a flat plate were used. For forced convection, laminar flow, $N_{Pr} \geq 0.6$,

$$N_{Nu_L} = 0.664\, N_{Re_L}^{\ 1/2}\, N_{Pr}^{\ 1/3}$$ was used, Reference 4-2.

For forced convection turbulent flow, the correlation used was

$$N_{Nu_L} = 0.036\, (N_{Pr})^{1/3}\left[N_{Re_L}^{\ 0.8} - N_{Re_{CR}}^{\ 0.8} + 18.44\, (N_{Re_{CR}})^{1/2}\right]$$ Reference 4-2.

where

N_{Nu_L} is the Nusselt number, hL/k

N_{Re_L} is the Reynolds number, $\rho VL/\mu$

N_{Pr_L} is the Prandtl number, $\mu C_p/k$

$N_{Re_{CR}}$ is the transition Reynolds number (400,000 or 500,000)

h is the heat transfer coefficient

L is the characteristic length

V is the fluid velocity

μ is the viscosity

k is the thermal conductivity

C_p is the specific heat at constant pressure

All properties are evaluated at the mean film temperature.

On the cold side, for quality less than 0.9, Kutateladze nucleate boiling heat transfer coefficients were assumed (Reference 4-3). The cold side heat transfer was determined by calculating the heat transfer rate per unit surface area and wall to fluid temperature difference:

$$\frac{\dot{Q}}{A_{SC}\,[0.555\,(\Delta T_{WC})]^{2.5}} = 1.547 \times 10^{-7} \left[\frac{112.3\, C_{P\ell C}}{(h_{sV} - H_{s\ell})\rho_{VC}}\right]^{1.5} \times$$

$$\left[\frac{0.0173\, k_{\ell C}\,(0.01603\,\rho_{\ell C})^{1.282}\,(6.894 \times 10^{4}\, P_{Ci})^{1.75}}{(\sigma_{\ell C})^{0.906}\,(14.88\,\mu_{\ell C})^{0.626}}\right]$$

In the foregoing equation, the following units apply:

$\dot{Q}$, Btu/hr	$C_{P\ell C}$, Btu/lb-°R	P_{Ci}, lb/in.^{2}	$\mu_{\ell C}$, lb/ft-sec
A_{SC}, ft^2	$k_{\ell C}$, Btu/hr-ft-°R	$(h_{sV} - h_{s\ell})$, Btu/lb	
ΔT_{WC}, °R	$\rho_{\ell C}$, lb/ft^3	$\sigma_{\ell C}$, dynes/cm	

A_{SC} = total cold side heat transfer surface area

$C_{P\ell C}$ = cold side liquid specific heat

$\dot{Q}$ = heat transfer rate

$k_{\ell C}$ = cold side liquid thermal conductivity

P_{Ci} = cold side inlet pressure

ΔT_{WC} = temperature difference between wall and cold side fluid

h_{sV} = specific enthalpy of saturated vapor on cold side

$h_{s\ell}$ = specific enthalpy of saturated liquid on cold side

$\rho_{\ell C}$ = cold side liquid density

$\sigma_{\ell C}$ = surface tension of cold side liquid

$\mu_{\ell C}$ = cold side liquid viscosity

Heat transfer across primary surfaces (across plates separating hot side from cold side fluid) was determined by heat balance:

$$\dot{Q}_{transferred} = h_H A (T_H - T_W) = h_C A (T_W - T_C)$$

where,

$T_H - T_C$ is known and $(T_H - T_W) + (T_W - T_C) = T_H - T_C$

T_H is the hot side temperature

T_C is the cold side temperature

h_H is the hot side heat transfer coefficient

h_C is the cold side heat transfer coefficient

A is the heat transfer area

T_W is the wall or plate temperature

Hot side and cold side heat transfer were cross plotted to find T_W and thus $\dot{Q}$ transferred. A typical cross plot for LO_2 is shown in Figure 4-3.

In order to increase heat exchanger efficiency, fins were used on the hot side between the primary surfaces to increase the heat transfer area. Heat transfer to the finned surfaces were lower than to the primary surfaces because the fin temperatures were higher than the primary coolant surface temperatures.

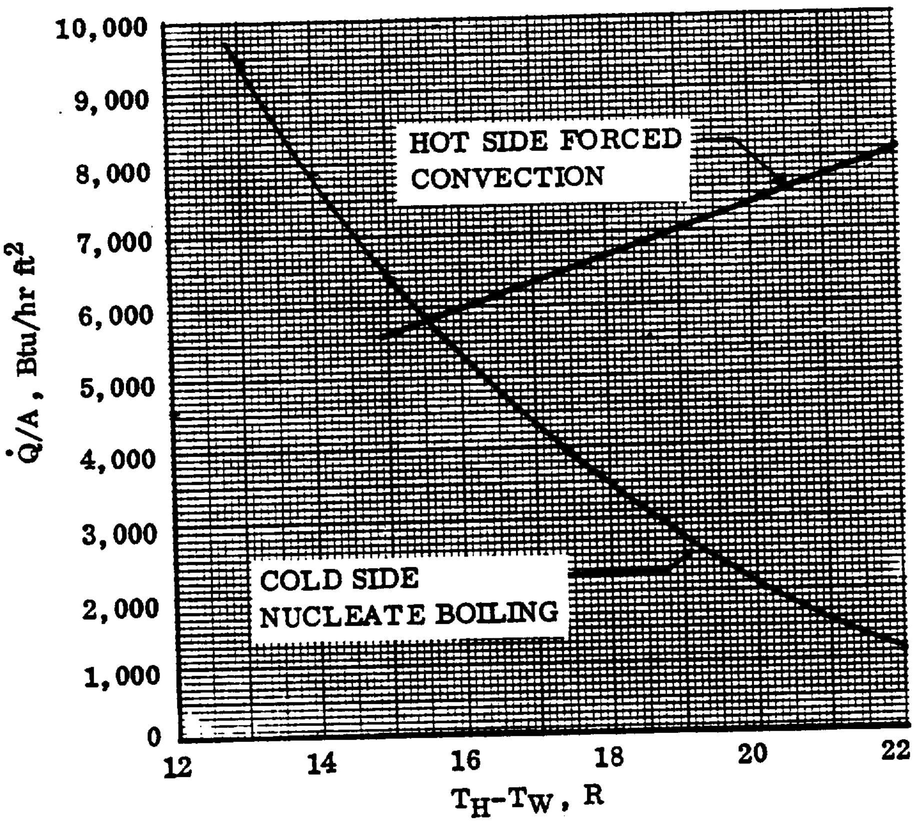

Figure 4-3. Typical LO_2 Heat Balance.

Heat transfer to the fin was determined with a heat balance computing the incident heat input from the hot side fluid to the cold side fluid as a function of the temperature of the fin at the midpoint between the primary surfaces, the hot and cold side heat transfer coefficients and the conduction along the fin. Fin heat transfer was found to be between 25 and 30% of primary surface heat transfer on a unit area basis.

4.1.2.2 Pressure Loss Analysis. Pressure losses in the thermal subcoolers were determined from existing correlations. For pressure loss in screens at the inlet to the subcooler, Ref. 4-4 was used. For frictional pressure loss, expansion, contraction and bend losses, equations and graphs similar to those in Ref. 4-5 were used.

Screen pressure loss was determined by

$$\Delta P_s = A\mu V + B\rho V^2$$

where

A and B are empirical constants

μ is the fluid viscosity

ρ is the fluid density

V is the freestream velocity upstream of the screen

Pressure loss in bends were found from

$$\Delta P_b = KEC\ (\rho V^2/2\,g_c)$$

where

K is a pressure loss coefficient depending upon the radius ratio of the bend and the aspect ratio of the duct or passage cross section.

E is an aspect ratio factor also depending upon the aspect ratio.

C is a correction factor for other than 90° angle turns.

V is the velocity in the duct or passage.

Frictional pressure loss was found from

$$\Delta P_f = (fL/D_H)\ (\rho)\ (V^2/2\,g_c)$$

where

L is the length of the section.

D_H is the hydraulic diameter of the section.

f is the friction factor determined from a Moody diagram, such as found in Reference 4-6.

Expansion losses were found from

$$\Delta P_e = K_e\ (\rho V_1^{\ 2}/2\,g_c)$$

where

V_1 is the velocity before the expansion and $K_e = [1 - (A_1/A_2)]^2$

where A_1 and A_2 are the areas before and after the expansion, respectively.

Contraction losses were found from

$$\Delta P_c = K_c C_c\ (\rho V_2^{\ 2}/2g_c)$$

where

V_2 is the exit velocity

K_c is a function of the area ratio A_1/A_2 between the entrance and exit

C_c is a function of the entrance rounding.

On the cold side of the heat exchanger, pressure loss was determined using a method developed by Martinelli and Nelson, Reference 4-7. This method described in detail in Appendix D-1, Reference 4-3, computes the pressure loss for two-phase turbulent flow using experimentally derived parameters dependent upon the fluid vapor to liquid density ratio, liquid to vapor viscosity and fluid quality. The experimental coefficients are used to convert the single-phase pressure loss with either liquid or vapor to the two-phase pressure loss. A computer program, written for the HP 9100 calculator, was used to compute pressure loss for both LH_2 and LO_2 configurations.

4.1.2.3 Heat Exchanger Sizing. Total subcooler pressure drop as a function of the number of heat exchanger passes was determined. Pressure drop at the inlet and exit of the subcooler due to screens, expansions, and contractions ($\Delta P_{duct+screen}$) was determined. Heat exchanger per pass was determined for each configuration examined.

The number of heat exchanger passes required was found from

$$n = \frac{\dot{m}C_p \frac{\Delta T}{\Delta P} (NPSH + \Delta P_{duct+screen}) + \dot{Q}_{in}}{\dot{Q}_p - \dot{m}\, C_P \frac{\Delta T}{\Delta P} (\Delta P_{per\ pass})} \tag{4-3}$$

The heat exchanger configuration yielding the minimum heat exchanger surface area (or minimum number of heat exchanger passes) was selected. For the LO_2 subcooler this configuration consisted of four passes on the hot side with 0.5 inch (1.27 cm) plate spacing. In order to minimize hot side surface area, five cold side passages are used. The LO_2 subcooler was placed in the LO_2 sump. Hot side pressure loss was approximately 1.4 psi (9.65 kN/m^2).

Attempts to design a heat exchanger to fit into the LH_2 sump were unsuccessful due to the limited space in the sump compared to the required heat transfer area. The subcooler was therefore designed to fit in the bottom of the tank, inside the capillary device. A single hot side passage with fins was used. In order to minimize hot size surface area, the hot side was completely surrounded by cold side fluid with a two pass, parallel flow/counter flow cold side arrangement. An equation similar to Equation 4-3 was solved for heat exchanger length as a function of plate spacing and hot side geometry. Calculations indicated that a single pass, 13 in. (33.02 cm) high × 6 ft (1.82 m) in circumference, with 1 in. (2.54 cm) plate spacing will be adequate. Fins run circumferentially along the exchanger, spaced 1 in. (2.54 cm) apart. Hot side pressure drop is 1.2 psi (8.4 kN/m^2). Cold side configuration consists of two passes of 1 in. (2.54 cm) × 13 in. (33.02 cm) ducting along the outer walls of the subcooler. Insulation may be required to limit the heat transfer from the cold side fluid to the surroundings on the cold side surfaces away from the subcooler. Cold side configuration consists of a series of vanes spaced 4 in. (10.16 cm) apart directing the flow, maintaining an annular two-phase flow pattern.

Hot side flow rates at steady state engine flow are 57 and 11 lb/sec (25.9 and 5 kg/sec) for LO_2 and LH_2 respectively. Subcooler cold side flow rates were determined from

$$\dot{m}_C \Delta h_C = \dot{m}_H C_p \Delta T_H$$

where,

$\dot{m}_H$ is the steady state engine flow rate

C_p is the liquid heat capacity

ΔT_H is the amount of subcooling produced in the subcooler plus the temperature equivalent of the subcooler system pressure loss

Δh_C is the enthalpy available for cooling (in the nucleate boiling regime) using two-phase throttled fluid

$\dot{m}_C$ is the cold side flow rate

Subcooler cold side flow rates are 752 lb/hr (341 kg/hr) and 240 lb/hr (109 kg/hr) for LO_2 and LH_2 respectively. Cold side pressure loss was determined using these flow rates, the Martinelli-Nelson two phase pressure loss correlations and single phase flow pressure loss correlations similar to those used on the hot side. Vanes are used on the cold side to induce annular flow patterns keeping liquid on the wall in order to promote nucleate boiling. For the LO_2 subcooler, cold side pressure loss will be approximately 1 psi (6.89 kN/m^2). For the LH_2 configuration cold side pressure loss will be approximately 1/4 psi (1.72 kN/m^2).

The subcooling obtained from the total tank head under main engine thrust conditions is insufficient to provide boost pump NPSH plus subcooler pressure loss. The subcoolers should thus be operative during all boost pump operating periods in order to assure that boost pump NPSH requirements are met.

Between burns no attempt will be made to keep the subcoolers full. An inlet screen between the subcooler and the screened channels prevents vapor from flowing into the channels. Prior to initiation of a main engine start sequence, subcooler cold side flow will commence. This will chilldown the subcooler hot side surfaces and permit the subcooler to provide adequate boost pump NPSH during the entire start sequence.

Successful conceptual design of the thermal subcooler completed the affirmative resolution of the three major decisions in the decision tree in Table 2-12. The primary system studied was a start basket, with a dry pump and propellant duct, using thermal subcooling for providing boost pump NPSH. The alternate configuration, chosen for preliminary design, in addition to the thermally subcooled start basket, was chosen to be the bypass feed start tank. This was based on selecting the next best system (Table 2-12) that would be significantly different from a start basket.

In order to reduce subcooler system weight penalty, and to make the system insensitive to the number of main engine burns, a vacuum pumping system for returning the subcooler fluid back to the tank can be used. This system, conceived principally for the start basket thermal conditioning fluid, uses a surge tank and vacuum pump described in Section 4.3.

4.2 TANK PRESSURE CONTROL WITH THERMAL SUBCOOLING

The use of thermal subcoolers to replace tank pressurization means that tank pressure profiles will be altered from the baseline Centaur D-1S pressure profiles. The principal thermal subcooling option is to dump the subcooler flow overboard. With this option, no fluid is added to the main propellant tank after launch. Analyses were performed to evaluate the reduced tank pressures that would occur during the five-burn mission to determine if tank pressures would be above allowable limits for main engine restart.

The PRISM program was used to determine the pressure history in the LO_2 tank during the five-burn low earth orbit mission. HYPRS was used for the LH_2 tank. (Both programs are used to predict pressure history during operational Centaur flights. Neither program has been formally documented.) The objective of the study was to determine what the minimum tank pressures would be in both tanks when using thermal subcooling to replace the pressurization system in supplying boost pump NPSH. For this purpose, minimum heating rates from Reference 4-8 were assumed and homogeneous thermodynamic conditions were used for both tanks. (With the assumption of thermodynamic equilibrium in the tank, no venting would be required since no net evaporation would be occuring at the screen surface.) A no-vent condition also corresponds to the passively cooled start baskets discussed in Section 3.7. Pressure histories for this case, shown in Figure 4-4, indicate that the tank pressure will drop below the tank pressure (19 psia (130.9 kN/m^2 for LH_2 and 29 psia (200 kN/m^2 for LO_2) normally used for starting the boost pumps. Boost pumps have been successfully operated during engine burns at tank pressures as low as 13 psia (90 kN/m^2) for the LH_2 tank and 24 psia (165 kN/m^2) for the LO_2 tank. The tendency for cavitation increases as the tank pressure decreases but this can be accounted for in proper thermal subcooler design. (In addition, it should be noted here that the subcooler designs generated to date have used the nominal tank saturation temperature in performing heat transfer calculations. The reduced saturation temperature of the inlet

Low Earth Orbit - 5 Burn Mission - Homogeneous Conditions-Pressure History - Thermal Subcooling - No Venting Between Burns

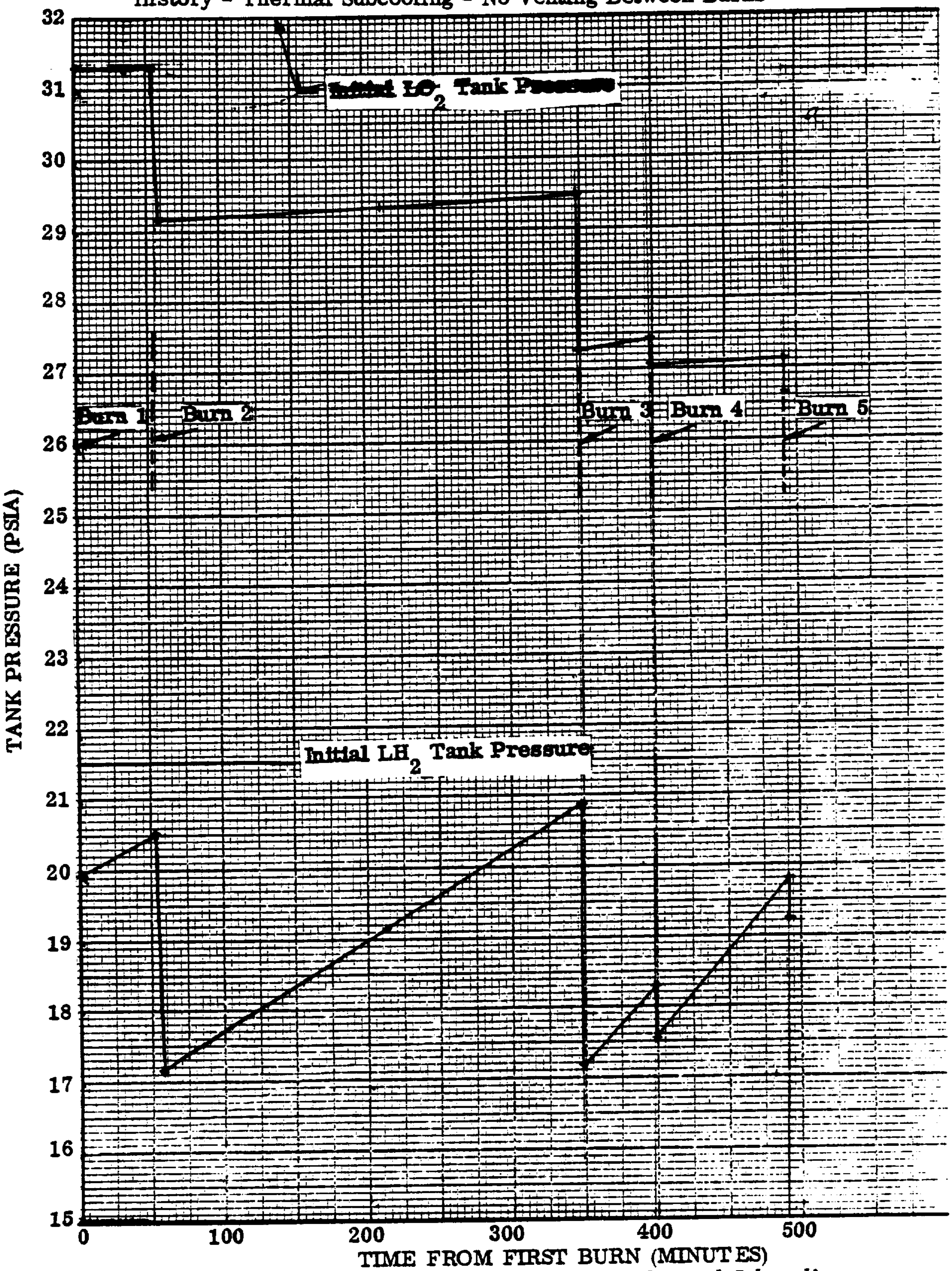

Figure 4-4. Typical Pressure Histories with Thermal Subcooling

fluid below these nominal conditions, as shown in Figure 4-4, will result in reduced temperature differences between the subcooler hot and cold side fluid and thus increase subcooler heat transfer area requirements.) In summary, the tank pressure conditions required for utilizing thermal subcooling appear to be acceptable for boost pump start-up but they are below the range encountered in Centaur flight experience. Future thermal subcooler calculations should incorporate reduced hot side inlet pressure into the heat transfer requirements.

For conditions of nonhomogeneous tank thermodynamic conditions between burns, pressure rise will be higher than shown in Figure 4-4 for the no-vent condition between burns. For venting, as would occur with an active thermal conditioning system, tank pressure reductions would be experienced between burns because of excess fluid vented due to condensation heat loads on the start basket (see Section 4.3). This might create unacceptably low pressure conditions for engine restart. Pumping fluid back into the tank rather than dumping overboard will eliminate these unacceptable conditions. Pumping start basket thermal conditioning fluid back into the tank will result in pressure profiles similar to those shown in Figure 4-4. (They may be slightly higher due to additional vacuum pump power added to the tanks.) If thermal subcooling fluid is pumped back into the tanks, the steep pressure decline during engine firing will be reduced somewhat. The effect will not cause a net pressure increase during outflow however, since, for LH_2, the engine outflow rate is 57 lb/sec (29 kg/sec) and the subcooler outflow rate is 240 lb/hr (109 kg/hr). This is equivalent to a main engine volume outflow rate of 2.5 ft^3/sec (0.07 m^3/sec) for LH_2, and a pumped coolant volume inflow rate of 2.10 ft^3/sec (0.06 m^3/sec) for GH_2.

4.3 START BASKET THERMAL CONDITIONING

The objectives of start basket thermal conditioning are to prevent dryout of the start basket outer screens and to prevent vapor formation in the screened channels feeding the subcoolers. Dryout of the screens must be prevented because capillary devices for wetting fluids operate by keeping vapor out of the contained liquid space. If screens dry out, vapor can enter freely, allowing the wetting fluid to migrate from the screened enclosure. Vapor formation in the start basket will occur due to pressure changes, or incident heating or fluid removal. Screened channels within the start basket prevent vapor from entering the subcoolers and capillary device thermal conditioning system. In order to obtain satisfactory subcooler and capillary device thermal conditioning, the channels must be maintained full at all times. In order to prevent heat input to the channels from causing vaporization in the channels, the capillary device cooling system is designed to maintain the screens slightly below saturation temperature.

Several methods exist for thermal conditioning the capillary device as described in Section 2.3. Pressure conditioning for cooling was determined to have too great a weight penalty because of the requirement for cold gas pressurization with an unsettled propellant. Total tank conditioning (vapor cooled shields) was found to be incompatible with internal heat sources such as intermediate bulkhead heat transfer. The primary concept selected for capillary device thermal conditioning was the use of cooling coils attached to the start basket screened surfaces containing throttled vent fluid. Channels inside the start basket provide coolant liquid to a throttle valve where it is throttled to a lower pressure and temperature. The two-phase mixture is then passed, in cooling coils, around the periphery of the start basket, acting as a heat sink in removing the heat incident on the start basket surfaces. This concept was studied in detail with design drawings presented, for LO_2 and LH_2 capillary device active thermal conditioning systems, in Sections 5.1 and 5.3. The high vent fluid and cooling coil weight penalty of this method caused passive thermal conditioning to be considered. A brief analysis of wicking for preventing screen dryout, as described in Section 3.7, appeared promising for saving both vent fluid and cooling coil weight.

Conditioning of the basket must be accomplished during the entire mission; ground hold, launch, cargo bay orbital coast, Centaur main engine burns and Centaur orbital coast. During periods of high acceleration, when liquid is bottomed in the tanks, start basket screen drying is not a problem since liquid will cover the outlet. Any vapor that forms in the basket during these periods will be vented out through the top of the basket. The exception could be vapor trapped below the channels feeding the subcoolers. This vapor could cause vaporization in the channels which would be unacceptable. Thus, while the start basket screens only need to be conditioned under low gravity (unsettled) conditions, the area adjacent to the channels needs to be conditioned at all times.

The start basket thermal conditioning analysis consisted of examining the possible heat transfer modes that could exist around the start basket. Forced convection due to mixing, free convection (both for liquid and vapor surrounding the basket), and condensation were examined for both LO_2 and LH_2 over the range of possible acceleration conditions.

Condensation heat transfer coefficients were computed from Reference 4-9,

$$h_m = 0.943 \left(\frac{k^3 \rho^2 g \lambda}{x \mu \Delta T} \right)^{1/4} \quad \text{for laminar condensation on a vertical wall.}$$

Where,

h_m is the heat transfer coefficient

g is the acceleration

k is the liquid thermal conductivity

ρ is the liquid density

λ is the heat of vaporization

μ is the liquid viscosity

x is the distance from the leading edge to the location of interest

ΔT is the temperature difference between the saturated liquid and the start basket surface

Free convection heat transfer coefficients for vertical planes were computed from

$$\frac{h_m x}{k_f} = \frac{0.68 \left(Pr_f^{1/2} \, Gr_f^{1/4}\right)}{0.952 + Pr_f}, \text{ Reference 4-9,}$$

where

subscript f is the fluid of interest (liquid or vapor)

Pr_f is the Prandtl number, $\frac{\mu_f C_{pf}}{k_f}$,

where C_{pf} is the specific heat at constant pressure

Gr_f is the Grashoff number, $\frac{x^3 \rho_f^2 g \beta_f \Delta T}{\mu_f^2}$

where β_f is the coefficient of volumetric expansion

ΔT is the temperature difference between the fluid and the surface. Other properties are as defined in the condensation heat transfer expression.

Forced convection heat transfer coefficients were determined from the laminar flow heat transfer relationship

$$\frac{h_m x}{k} = 0.664 \, (Pr)^{1/3} \, (Re_x)^{1/2}, \text{ Reference 4-2.}$$

where

Re_x is the Reynolds number, $\frac{\rho V x}{\mu}$

where V is the fluid velocity.

Other quantities are as defined above. The fluid velocity was based on mixer flow velocities determined for the baseline Centaur D-1S thermodynamic vent systems.

As shown in Table 4-1, condensation heat transfer during Shuttle payload bay conditions (worst case unsettled accelerations) will provide maximum heat load to the start basket screens. Screen drying will not be caused directly by condensation since this will tend to deposit liquid on the screen. The thermal conditioning system, consisting of coiled tubes containing throttled vent fluid wrapped around the start baskets, will receive the greatest heat input during condensation. It is possible that small areas of superheated vapor could exist around the basket while condensing heat transfer is dissipating the thermal conditioning cooling capacity over the remainder of the basket. Thus, superheated vapor could cause screen drying if thermal conditioning cooling capacity has been exhausted by condensation. In order to be conservative, the cooling coils must be designed to handle condensation over the entire basket.

Table 4-1. Start Basket Heat Transfer Coefficients — Shuttle Payload Bay Conditions

Fluid	Heat Transfer Mode	h_f, Btu/hr-ft^2- R	(watt/m^2-K)
GO_2	Free Convection	0.3	(1.70)
↓	Forced Convection (V = 0.21 ft/sec (0.064 m/sec))	0.6	(3.41)
	Condensation, Vertical Wall, Laminar Flow	384	(2180)
LO_2	Free Convection	14.2	(80.6)
↓	Forced Convection (V = 0.21 ft/sec (0.064 m/sec))	34.2	(194)
GH_2	Free Convection	0.4	(2.3)
↓	Forced Convection (V = 0.11 ft/sec (0.033 m/sec))	0.6	(3.41)
	Condensation	158	(897)
LH_2	Free Convection	7.3	(41.4)
↓	Forced Convection (V = 0.11 ft/sec (0.033 m/sec))	7.5	(42.6)

To prevent start basket screen drying, the thermal design criteria was to cool the screens to below the saturation temperature corresponding to the minimum tank pressure between burns. The thermodynamic vent system for the baseline Centaur D-1S vehicle was designed for a 1 psi (6.89 kN/m^2) pressure band. For hydrogen this corresponds to a temperature band of 0.37 R (0.21K) and for oxygen a temperature band of 0.73 R (0.41K). The hydrogen screen temperature will be maintained at 0.5 R (0.28K) below the maximum hydrogen saturation temperature while oxygen screen temperature will be maintained 1.0 R (0.55K) below the maximum oxygen saturation temperature. Cooling coil spacing and incident heat transfer to the basket were determined using fin equations (Reference 4-1) for continuous cooling coil attachment. The major geometric variables used in Equations 4-4 and 4-5 are illustrated in Figure 4-5.

$$T_{(a/2)} = T_H - \frac{(T_H - T_C)}{\cosh(N\ a/2)} \qquad (4\text{-}4)$$

$$\frac{\dot{Q}}{A_b\ (T_H - T_C)} = \frac{K_w\ t_e\ N\ \mathrm{Tanh}(N\ a/2)}{a/2} \qquad (4\text{-}5$$

where:

$N = \sqrt{\bar{h}_f/(K_w t_e)}$

K_w = effective conductivity

t_e = conduction thickness of the structure to which cooling coils are attached

$\bar{h}_f$ = average incident heat transfer coefficient

A_b = total surface area of basket

T_C = coolant temperature

$a/2$ = half the distance between the coils

$\dot{Q}$ = total heat input to cooling fluid

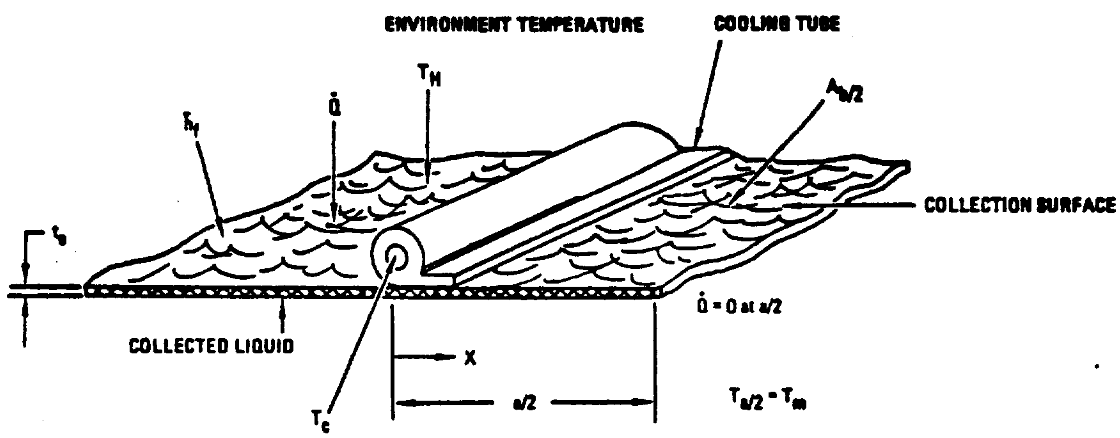

Figure 4-5. Continuous Cooling Configuration

The effective conductivity for the screen/plate combinations used as the start basket capillary barriers were taken as the sum of the plate and screen conductivities in the direction of the heat flow path. Conductivity testing of Dutch weave screen materials on a Convair IRAD program, Reference 4-10 was extrapolated to the screens and temperatures of interest. A conductivity of 35% of the conductivity of a solid metal sheet of identical thickness was conservatively assumed to be the screen thermal conductivity. The perforated plate, with 50% open area was taken to have a thermal conductivity of 50% of the solid material conductivity.

Local heat sources due to penetrations, sump heating, and intermediate bulkhead-tank wall intersection heat shorts were studied to determine if any additional cooling provisions were required. Calculations indicated that under worst case conditions, condensation heat loads would exceed conduction heat loads from warm vapor that could exist in the sump or intermediate bulkhead-tank wall area. Preliminary calculations of conduction along struts and supports indicated that no problem should exist that could not be easily solved by wrapping a cooling coil around the strut or by altering the thermal resistance between the penetration and tank wall. (Reducing the conductivity or increasing the attachment point area.) These observations indicated that condensation heat loads could safely be used for the level of detail called for in this study. Additional work may be required to refine support configurations for manufacturing level drawings.

Equations 4-4 and 4-5 were used with condensation heat transfer coefficients for boost, cargo bay and orbital heating. (Thermal conditioning during boost conditions is only required below the channels where vapor formation in the start basket could potentially cause vaporization inside the screened channels.)

In order to maintain vented propellants within realistic limits, a control system was used to control the vent flow rate in accordance with the "g" level conditions. This would require an adjustable shutoff valve and throttle valve that would adjust the coolant flow rate and coolant temperature to provide cooling for handling condensation heat loads at the existing acceleration. Even with this type of system, calculations indicated that the high heat load caused by condensation resulted in an excessive weight penalty if this fluid was dumped directly overboard (as much as 809 pounds (367 kg) for the five-burn mission, as shown in Section 6.3).

Several solutions were examined for reducing the payload penalty due to excess venting as well as for reducing the capillary device volumetric requirements since all cooling fluid is taken from the start basket. Complete mixing of the tank contents would prevent superheated vapor from existing in the tank and would minimize capillary device thermal conditioning system requirements. Prevention of superheated vapor by complete mixing is unrealistic for the LH_2 tank because of the narrow spacing between the capillary device and intermediate bulkhead and the high heat flux into the tank in this area. Also continuous complete tank mixing during relatively high acceleration conditions is impractical because of excessively high mixer weight and power requirements.

Configuring the channels so that bubbles formed during boost would not come into contact with the channels would allow boost heating conditions to be neglected. This is potentially possible because liquid surrounds the basket during boost thus preventing start basket screen drying. Heat input to the basket will form vapor in the basket but buoyancy forces will cause this vapor to be vented out through the top screens. Vapor could, however, be trapped under the channels used to feed the subcooler and supply thermal conditioning system fluid. This vapor could cause vaporization in the channel that could not be vented due to the high retention capability of the channel. In order to prevent this, channels could be configured so as not to trap vapor beneath them. This appears likely to increase residuals. Since only a small portion of the basket surface must be protected during boost conditions, a separate cooling loop for this region would be preferable to altering the channel configuration.

The most promising active cooling method considered relies upon condensed vapor surrounding the basket to replace thermal conditioning flow exiting from the basket. The conditioning flow is then pumped back into the tank. Using condensate to fill the basket between burns prevents capillary device volumetric increases over that given in Table 3-7 if proper means of sensing the cooling capacity of the thermal conditioning fluid can be utilized. If fluid is vented at a rate corresponding to handling condensation heat transfer and condensation does not occur or occurs over only a small fraction of the basket, insufficient liquid may surround the basket to replace the liquid used for thermal conditioning. A method of sensing the temperature of the conditioning fluid after it exits from the area around the basket should be used to control the cooling flow rate. This method can employ a known heat source to superheat the vapor a calibrated amount and sense the coolant temperature to increase or decrease the flow rate accordingly. Flow rate is thus adjusted automatically to provide the required cooling. This minimizes vented fluid weight penalty for an open loop system as well as minimizing capillary device volume (capillary device volume will still have to contain sufficient additional liquid to supply cooling liquid for the longest period between burns when subjected to the highest incident heat transfer, other than condensation). This corresponds to forced convection heat transfer between burns 1 and 2 for the 2-burn synchronous equatorial mission. Corresponding start basket volumes for these conditions are 9 ft^3 (0.25 m^2) for LH_2 and 0.6 ft^3 (0.017 m^3) for LO_2. Table 3-6 shows start basket volumes for thermal conditioning of 13.6 ft^3 (0.384 m^2) for LH_2 and 1.29 ft^3 (0.037 m^3) for LO_2. Since the designs for the start baskets were nearly completed when this calculation was done, the start basket volumetric requirements were not reduced. Thus, 4.6 ft^3 (0.13 m^3) and 0.69 ft^3 (0.02 m^3) of extra cooling capacity exists in the LH_2 and LO_2 baskets for possible inefficiencies in the cooling flow rate feedback control systems.

The feedback control system adjusting the flow rate to correspond to outlet temperature reduces basket volumetric requirements but does not reduce fluid penalty since potential worst case conditions of condensation must be assumed. In order to reduce the weight penalty due to venting conditioning fluid, a closed system was studied using a pump to circulate the conditioning fluid back into the tank. This type of system also simplifies the conditioning on the ground and during boost periods when vacuum conditions do not exist around the vehicle for conveniently venting the tanks. A small evacuated surge tank would be used to start the system.

Calculations were performed to size the pumps required for this purpose. Pump weight, battery weight to supply pump power, tube weight and added boiloff due to pump power were considered in the parametric analysis to determine optimum tube spacing and flow rate for the LO_2 and LH_2 capillary devices. Pump weight was found to be relatively invariant compared to tube weight over the range of variables considered. Pump sizing was based on pumping coolant flow rates (over each mission period of interest) back across the throttling pressure range used. Pump power requirements were determined using equations from Reference 4-11 for pump and motor efficiency vs fluid power/total input power for LH_2. The overall pump efficiency equation recommended in Reference 4-11 for pumps between one and 20 watts power output was $\eta_t = 0.155\ (P_o)^{1/3}$, where P_o is the liquid power output. For LH_2 electrical power input, $P_i = \frac{P_o}{\eta_t}$. For LO_2 pumps, motor efficiencies may be reduced due to the use of canned stators. A curve of power input for a canned stator versus power input for an uncanned stator, from Reference, 4-12, was used to determine LO_2 pump input power. The vacuum pumps will have relative specific speed of less than 500.

Pumping coolant flow back into the tank makes the active thermal conditioning system relatively insensitive to total mission time. Only boiloff and battery power due to pump operation, which are a relatively small penalty (see Section 6.3), would increase with increased mission time.

Maximum allowable tube spacing was computed for worst case heating conditions. For regions directly under the channels, these tube spacings corresponded to condensation heat transfer coefficients at boost "g" levels. For other areas, the Shuttle OMS acceleration levels were used. Equations 4-4 and 4-5 were used with the ΔT between the saturated hot side fluid and the midpoint on the screen between adjacent cooling tubes taken to be 1.0 R (0.55 K) for LO2 and 0.5 R (0.28 K) for LH_2. Coolant flow rates were determined from

$$\dot{m}_c = \dot{Q} / \Delta h \qquad (4\text{-}6)$$

where Δh is the enthalpy change of the coolant in the tubes attached to the start basket. Flow rates are shown in Table 4-2.

Table 4-2. Start Basket Cooling Flow Rates

Fluid	Location	Ground and Boost		Cargo Bay		Orbit	
LH_2	Basket Bottom	*562	(255)	106	(48)	35	(16)
	Basket Sides	-	-	320	(145)	76	(34)
	Total	562	(255)	426	(193)	111	(50)
LO_2	Tank Bottom	394	(179)	60	(27)	18	(8)
	Basket	-	-	152	(69)	46	(21)
	Total	394	(179)	212	(96)	64	(29)

*Flow rate in lb/hr (kg/hr)

On the side of the LH_2 basket, adjacent to the channels, the tubes are 0.75" (1.91 cm) I.D., spaced at 1.44" (3.66 cm). The remainder of the basket has 0.75" (1.91 cm)

I. D. tubes, spaced at 2.4" (14.63 cm). In order to maintain low LH_2 cooling tube pressure drop, parallel flow paths were used.

Separate loops are used for the LO_2 basket screened surfaces and the tank wall forming the bottom of the start basket. The screened surfaces have tubes of 5/8" (1.59 cm) I. D., spaced at 1.2" (3.05 cm); while the tubes on the tank wall are 3/8" (0.95 cm), spaced at 1.2" (3.05 cm). Cooling coil pressure losses were determined using the Martinelli Nelson two-phase flow pressure loss analysis (Reference 4-7) cited in Section 4.1.2.3.

For cooling requirements subsequent to boost, reduced cooling flow rates will be obtained by adjusting cooling system shutoff valving as a function of exit temperature and "g" level. The throttle temperature would also be reduced to prevent overcooling. These calculations were performed and are reflected in the flow rates of Table 4-2. The throttle temperature was adjusted to satisfy Equation 4-4 for reduced heat transfer coefficient. The heat input was then computed according to Equation 4-5 and converted to a flow rate using Equation 4-6. Boiloff and battery weight due to pump power requirements were based on using a pump designed for boost conditions and operated at off-design conditions during non-boost periods. Pump flow requirements are based on flow rates given in Table 4-2. For slightly more pump boiloff and battery weight, a constant flow rate cooling and pumping system could be designed. Weight penalties for dumping fluid overboard were also based on using these coolant flow rates.

4.4 START TANK THERMAL CONDITIONING

In order to simplify thermal conditioning for the LO_2 and LH_2 start tanks, attempts were made to keep the start tanks locked up between burns. After refilling the start tanks from main tank fluid, the start tanks will be burped 3 psi (20.7 kN/m^2) with cold helium in order to suppress boiling in the screened enclosures between burns. An analysis was performed to determine the maximum total heat input into the start tanks prior to each engine burn and the corresponding pressure increase in order to establish start tank thermal conditioning and venting requirements.

Worst case heat transfer to the start tank was assumed. Incident heating in the LO_2 start tank aft bulkhead area was determined by examining the detailed calculations used in References 4-8 and 4-13 for the condition where Centaur is in the Shuttle cargo bay with the doors closed. Assumptions were made to adjust these heating rates based on the position of the start tank in the sump and alterations in the duct and valve configuration. Within the main tank, the main tank contents are initially warmer than the start tank and heat transfer to the start tank was assumed to be by condensation. The worst case heat transfer conditions occur when the Centaur is in the Shuttle cargo bay. An average g level of 5.25×10^{-4} g's was calculated for this period using Shuttle aerodynamic drag and a duty cycle assuming RCS thrusters were operative 5% of the time between OMS burns. Using main tank fluid to start tank surface temperature differences of 2 R (1.11K) the condensation heat rates into uninsulated LH_2 and LO_2 start tanks were computed to be 6650 and 1606 Btu/hr (1.95 and 0.47 kW) respectively. Pressure rise rates calculated from the empirical correlations of Reference 4-14 indicated these heat rates were too high. The LO_2 start tank uses a portion of the main tank aft bulkhead and is therefore limited to a maximum pressure of 48 psia (331 kN/m^2). The LH_2 start tank is designed by settling loads and can take 37 psi (255 kN/m^2).

Pressure profiles in the start tank will be: refilling at 5 psia (34.5 kN/m^2) below main tank pressure (26 psia (17.9 kN/m^2) for LO_2 and 15 psia (103 kN/m^2) for LH_2), burp by 3 psi (20.7 kN/m^2) (29 psia (200 kN/m^2) for LO_2 and 18 psia (124 kN/m^2) for LH_2), pressure rise between burns (48 psi (331 kN/m^2) max for LO_2 and 37 psia (255 kN/m^2) max for LO_2), and vent down again to 5 psia (34.5 kN/m^2) below the main tank after main tank liquid is settled and the start tank outflow valve is closed.

In order to reduce start tank heating, insulation was added to the start tank. A 1/4-in. (0.64 cm) depth flex-core fiberglass honeycomb insulation layer outside the LH_2 start tank and inside the LO_2 isogrid dome was selected for analysis. A stainless steel skin over the exposed honeycomb surface in each tank (similar to the Centaur intermediate bulkhead design) will prevent leakage of liquid into the insulation. The honeycomb in each tank would be evacuated to space. Insulation weights will be approximately 61 lbs (27.7 kg) for the LH_2 tank (including 49 lbs (22.2 kg) for the steel skin) and 15 lbs (6.8 kg) for the LO_2 tank (including 10 lbs (4.54 kg) for the steel skin). With this insulation, condensation heat rates, for temperature difference of 5 R (2.78K), were found to be 74 Btu/hr (21.7 watts) into the LH_2 start tank and 11 Btu/hr (3.22 watts) into the LO_2 start tank. Heat is also transferred into the LO_2 start tank across the common aft bulkhead. Individual areas considered in the heating rate calculations were: supply duct and valve heating, sump heating, line and wire support heating, and the effect of opening the cargo bay doors. The analysis, based on Reference 4-13, deleted penetration heating from the thrust structure since the start tank lies inboard of the thrust barrel. Heating from a pneumatics panel located between the sump and thrust barrel was eliminated by moving the panel outboard of the thrust barrel. The resulting heat rate across the aft bulkhead was 141 Btu/hr (41.3 watts) for a total average heat rate in the cargo bay of 152 Btu/hr (44.5 watts).

Pressure rise rates were computed from Reference 4-14 equations for low gravity conditions:

For LH_2 $$\frac{\Delta P}{\Delta t} = 81\ \dot{Q}/MS$$

For LO_2 $$\frac{\Delta P}{\Delta t} = 1450\ (\dot{Q}/MS)^{1.14}$$

where

$\frac{\Delta P}{\Delta t}$ is the pressure rise rate in psi/hr

$\dot{Q}$ is the incident heat rate on the tank in Btu/hr

M is the total mass in the tank

S is the percent ullage volume

For an initial ullage of 5%,pressure rise will be 10 psi (68.9 kN/m^2) for LH_2. This is well below the allowable ΔP of 17 psi (117 kN/m^2). For LO_2 pressure rise rates are more severe and initial ullage volume of 16% was selected to keep the maximum tank pressure below 48 psi (331 kN/m^2). The calculations indicated that nonvented start tanks were practical for both the LH_2 and LO_2 tanks.

4.5 BOOST PUMP THERMAL CONDITIONING

The initial baseline boost pump thermal conditioning concept was to keep the boost pump and sump filled with liquid between burns. Methods identified for this purpose, discussed in Section 2.3, were wrapping the drive shaft area with cooling coils and purging the turbine rotor with cold helium. Effort was expended in identifying the sources of heat to the fluid contained in the boost pump and sump. In addition to the heat input, a major area of interest was the temperature of the gearbox lubricant when the pump is static and filled with cryogen. Sources of low temperature lubricant were identified for possible lubricant replacement in the event lubricant temperatures fell below allowable limits of -25 F (-32C).

Only the LH_2 boost pump was analyzed. This was done because an existing thermal model was already set up for the LH_2 boost pump and the LH_2 boost pump was felt to be more difficult to cool than the LO_2 boost pump.

The existing thermal model (Reference 4-15) was used on the Convair thermal analyzer program (Reference 4-16) for the configuration shown in Figure 4-6. The objective was to run a complete mission profile for the low earth orbit mission. After several runs it became obvious that the running time of this model would be much longer than the resources allotted to this subtask would permit. For this reason, a simplified boost pump thermal model was developed to permit complete mission simulations. The model combined some of the nodes of the existing thermal model to simplify the representation of the turbine, gearbox and boost pump. A run was made to simulate a warm gearbox and turbine with a cooled pump volute in order to obtain gearbox lubricant temperatures and pump heating rates. Figure 4-7 gives the results for the five-burn mission. Pump heating rates were a maximum of 20 Btu/hr (5.8 watts) and minimum lubricant temperature was 9 F (-13C) which is well above the minimum of -25 F (-32 C).

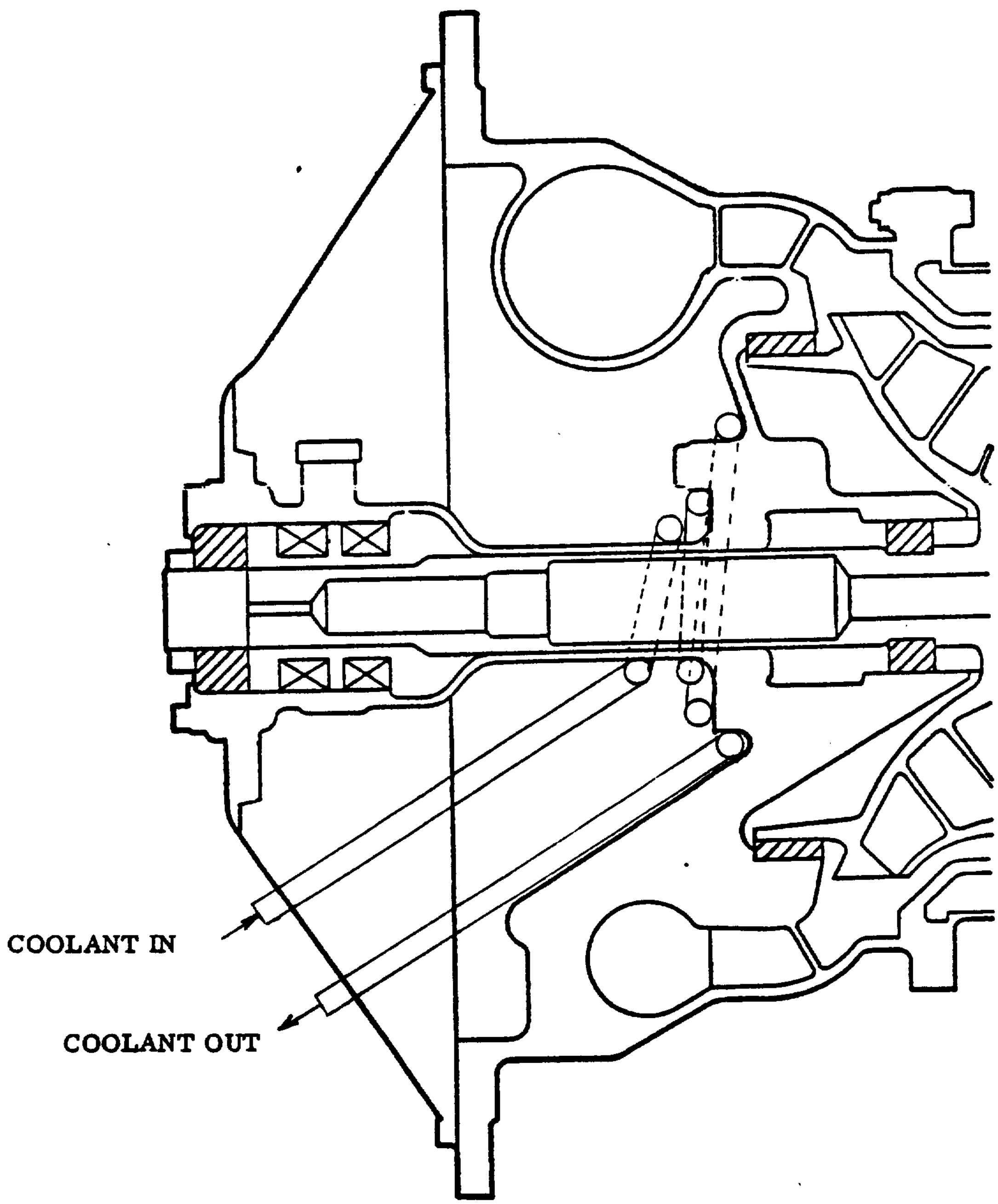

Figure 4-6. LH_2 Boost Pump Showing Possible Location of Cooling Coils

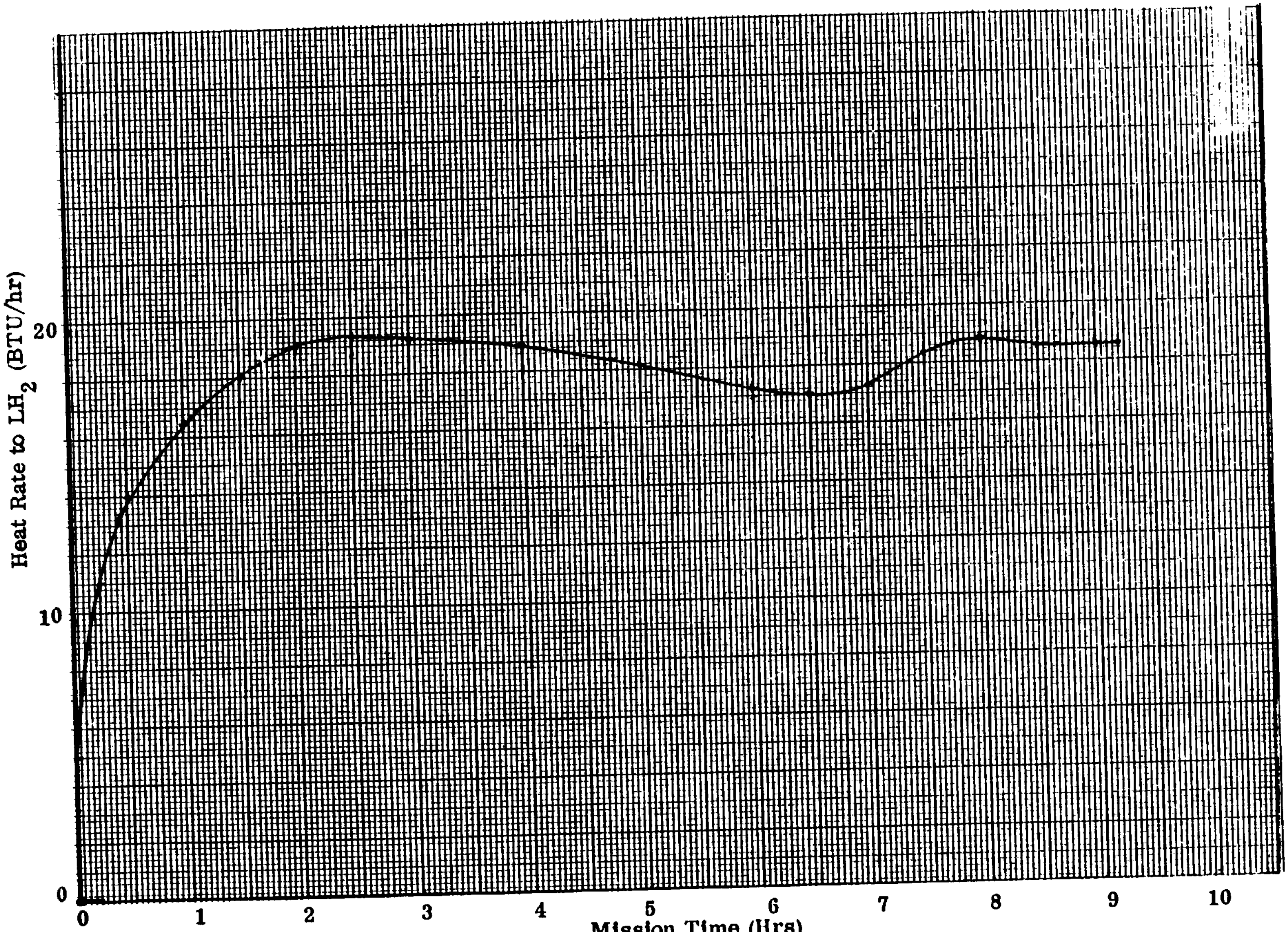

Figure 4-7. Boost Pump Heating — Heat Rate to Contained LH_2 (Simplified Boost Pump Model/5-Burn Mission)

The analysis revealed some shortcomings in the recommended cooling schemes for the boost pump. Heat could enter the contained liquid along the rotating shaft using the cooling coils or turbine purging methods. In order to assure that no heat would enter the boost pump along the shaft, either gearbox purging or driveshaft purging (see Table 2-8) would be required. Both these concepts involve complex changes to the boost pump that would probably disqualify the acquisition system on the basis of complexity. For this reason, acquisition concepts employing uncooled boost pumps were explored in more detail.

In the course of the boost pump thermal conditioning study, several low temperature lubricants were identified for replacing the gearbox lubricant. For future reference, these were low temperature solid film lubricants including Electrofilm 2006, 2306, and 2396 (Reference 4-17), and Everlube 811 (Reference 4-18).

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SECTION 5

TASK IV, SYSTEMS DESIGN

Preliminary designs were made of both a start tank and a start basket for both the LO_2 tank and the LH_2 tank.

The start baskets for both fluids are basically similar in that they have an outer screen cooled by liquid from screened channels inside the start basket. Also, each has an internal subcooler fed from the same screened channels.

The start tanks are not cooled but are insulated to prevent excessive heat input and pressure rise. Pleated screen channels are provided at the tank outlet to prevent vapor outflow and to reduce residuals.

In the start basket configurations, all fluid for the engines passes through the basket and subcooler throughout engine operation. While in the start tank, bypass valves are employed so that only the initial starting fluid is provided by the start tank.

5.1 LO_2 START BASKET

The LO_2 start basket is shown in Figure 5-1 as installed in the Centaur LO_2 tank. Minor modifications to the engine thrust cylinder are required in the form of a support ring between the upper plate and the cylinder sidewalls to provide a support for the start basket.

The cylinder and cone construction is basically an outer shell of perforated aluminum sheet with .625" OD by .035" wall (1.59 cm × .089 cm wall) aluminum tubing brazed to the outer surface. Brazing is the preferred method of attachment of the tubing to the perforated shell if the assembly is small enough to be dip brazed since it gives less distortion and greater contact area than welding. Aluminum screen, 50 x 250 mesh (1 WP), is spotwelded to the inner surface of the perforated shell (see Figure 5-2, view A-A)

Since the normal flow through the screen is from the outside in, the optimum placement of the screen is on the outside surface of the perforated sheet. The close spacing of the cooling tubes on the outside surface of the perforated sheet, however, leaves no practical way to attach the screen on the outer surface and still properly seal the basket. Thus the screen is inside the basket spotwelded to the perforated shell. A base ring (Figure 5-2, Sec. G-G) attaches the top cone section to the cylinder sidewall and also provides strut attachment. The cooling tube configuration is shown in Figures 5-1 and 5-2.

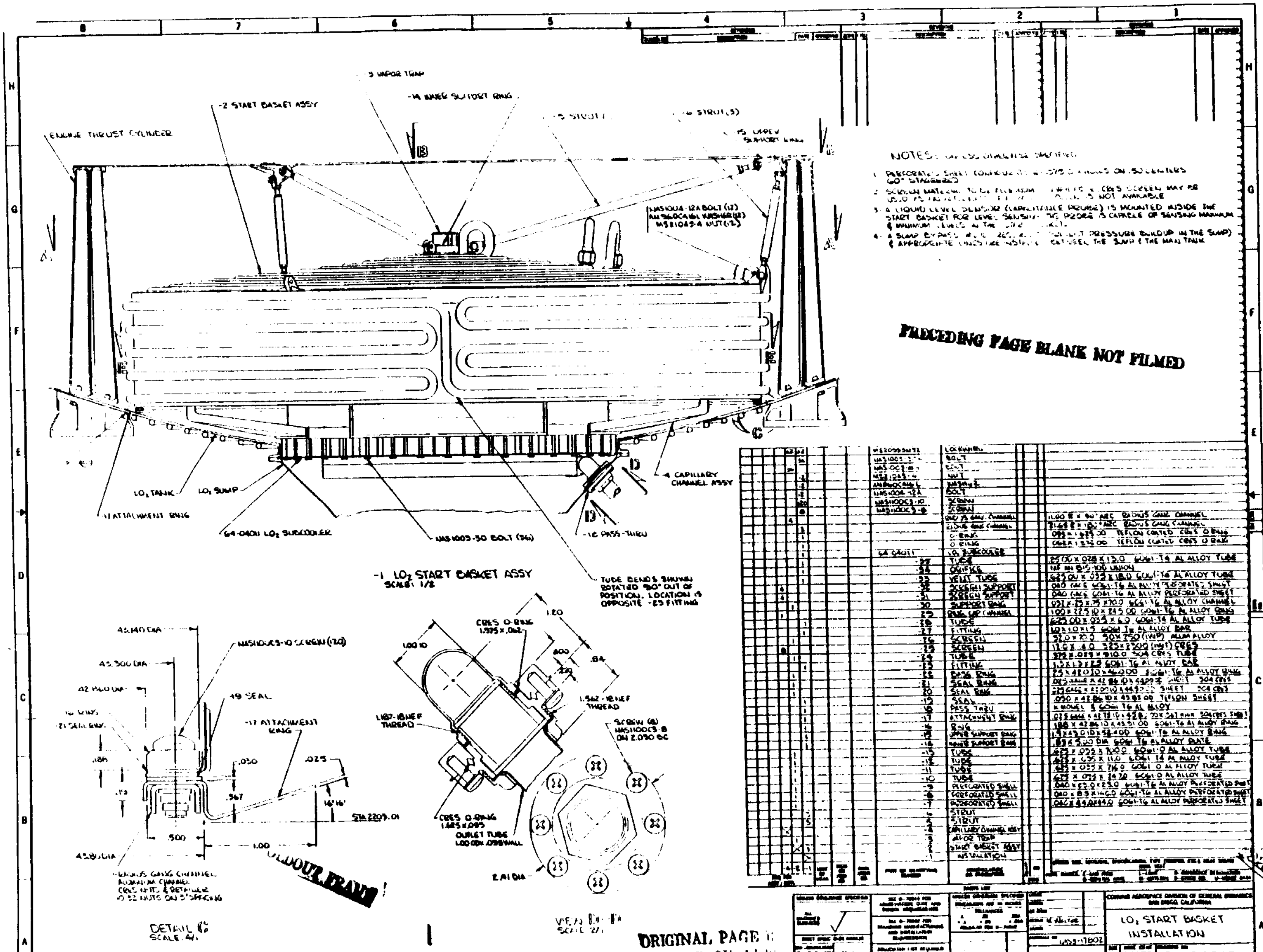

FIGURE 5-1. LO2 START BASKET INSTALLATION

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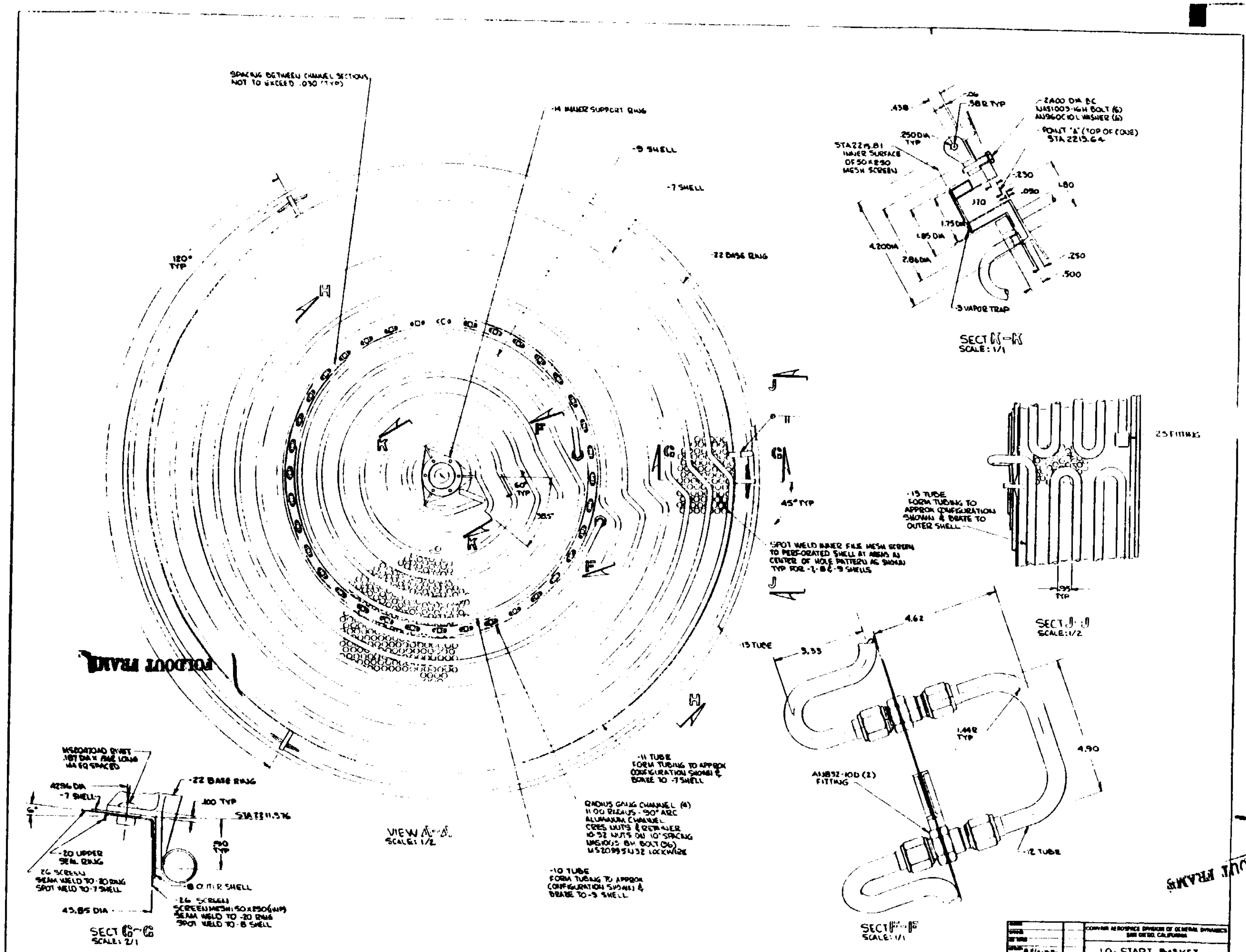


FIGURE 5-2. LO_2 START BASKET INSTALLATION

The start basket is made up of three basic sections; 1) attachment struts and inner support ring, 2) cylinder and outer cone assembly and 3) inner cone assembly and vapor trap. An attachment ring is welded to the LO_2 tank (Detail C, Figure 5-1) with the start basket bolted to this attachment ring. A Teflon seal is used between the attachment ring and the basket to prevent leakage. The inner cone assembly is attached to the outer cone and cylinder assembly by screws with the screen used as the seal between the two cone sections. Screws are used so the basket can be disassembled in the LO_2 tank for access into the main part of the tank. Access is accomplished by removal of the sump and subcooler (the screened channels remain in place) and removal of the inner cone which allows access to the attachment struts. The three long -5 struts and this inner support ring can then be removed to allow access to the thrust cylinder cover. Assembly is in the reverse order. The cooling tube spacing is 1.20 inches (3.04 cm) between edges of the brazed joints (Figure 5-3, Sect. H-H). The tubing around the cylinder section does not wind continuously around the perforated cylinder but is in two sections (Figures 5-1 and 5-2, Sect. J-J) of 180 degrees (π radians). This allows for differential expansion during brazing of the tubing to the cylinder and prevents gaps between the tube and the perforated sheet. The top cone inner and outer sections are connected by screws and a removable section of tubing (Figure 5-2, Sect. F-F). Cooling coils of .375" OD by .025" wall (.95 cm OD by .064 cm wall) CRES tubing are attached to the exterior of the LO_2 tank below the basket (Figure 5-3, Sect. H-H) to prevent heat input through the tank wall. The tube spacing is the same as the larger interior tubing, 1.20 inches (3.04 cm) between brazed joints.

The cone inner section has a standpipe to minimize trapped vapor at the top. The standpipe has a 50×250(1 WP) screen as its top surface (Sect. K-K, Figure 5-2) and is cooled by fluid from the capillary channels prior to the fluid cooling the basket exterior (Figure 5-3, Sect. H-H).

A thermal subcooler and screened channels (Figure 5-3, Sect. H-H) are installed inside the basket. The screened channels supply fluid for cooling both the start basket and the LO_2 tank skin below the basket. The capillary channel construction is opposite that of the basket in that the screen is outside the perforated sheet. In this case there are no cooling coils to prevent placing the screen/backing plate combination in the normal order for flow from the outside to the inside of the channels. The 325×2300 CRES screen is not spotwelded to the perforated sheet. The screen is attached at the edges by seam welding to the solid aluminum sheet welded to the edges of the perforated sheet (Figure 5-3 Sect. E-E and Sect. N-N). The upper edge of each screened channel has a solid sheet for joining but is designed to minimize vapor entrapment.

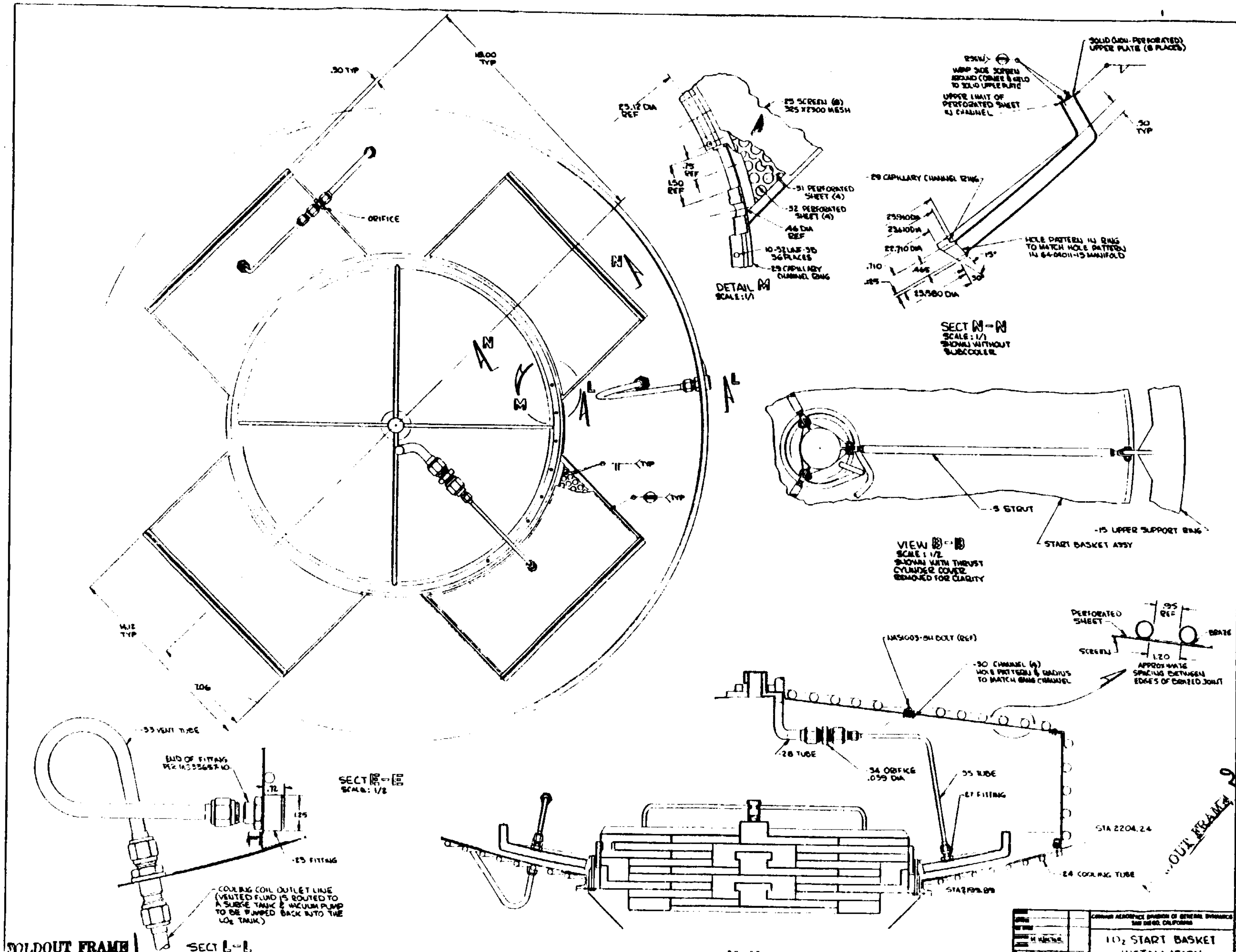


FIGURE 5-3. LO_2 START BASKET INSTALLATION

Each screened channel is attached to a common ring which is attached to the subcooler by bolts. A seal is provided by a matching taper between the subcooler and the channel ring. Slots are provided in the channel ring for flow from the channels to the subcooler.

In order to prevent pressure build-up in the sump during no-flow conditions, a bypass line and valve are provided between the sump and the main tank. This line will prevent screen breakdown in the inlet channels. This line and valve are not shown on the drawings.

5.2 LO_2 THERMAL SUBCOOLER

The subcooler used in the start basket is shown isometrically in Figure 5-4. A conventional top and sideview are shown in Figure 5-5. Operation of the subcooler is as follows: Hot side fluid enters through the slots in the tapered ring into a manifold and thence into the finned area between the -2 and -3 plates. A screen (200 × 600 mesh) is provided between the manifold and the flow area between the plates (Section M-M, Figure 5-6). When the fluid reaches the center of the finned area, it passes through holes in the plate (Figure 5-7, Section B-B) to the area between plates -4 and -5. The fluid then moves to the outside edge of the plate and passes through holes (Figure 5-7, Section D-D) to the area between plates -6 and -7. It then moves to the center area to holes (Figure 5-7, Section F-F) and down to the area between plates -8 and -9. From here it passes to the outer edge and out into the sump.

Each of the areas that hot side fluid passes through on its way through the subcooler is a finned area in which heat is being extracted.

Fluid for subcooling is provided from the manifold. Four tubes feed the volume above the -12 orifice. The pressure drop across the orifice is 26 psid (179 kN/m^2) at a mass flow rate of 752 lb/hr (341 kg/hr). This provides the cold side fluid for subcooling. In this flow path, the cold fluid enters at the top in the center (through the -17 tube). Baffles on the -2 plate force the fluid to move over the surface of the plate in a spiral path. Holes at the outer edge (Figure 5-6, Section A-A) allow the cold side fluid to pass to the level between plates -3 and -4. On this level, the fluid is spiraled in an opposite direction from the outside edge to the center (Figure 5-6, Section C-C) where it passes through channels and a central hole in an area between plates -5 and -6. At this level the flow is similar to the first area between plates -11 and -2, moving toward the outside edge. At the outside edge, it passes through channels (Figure 5-6, Section E-E) to an outer manifold and down to an area between plates -7 and -8. This

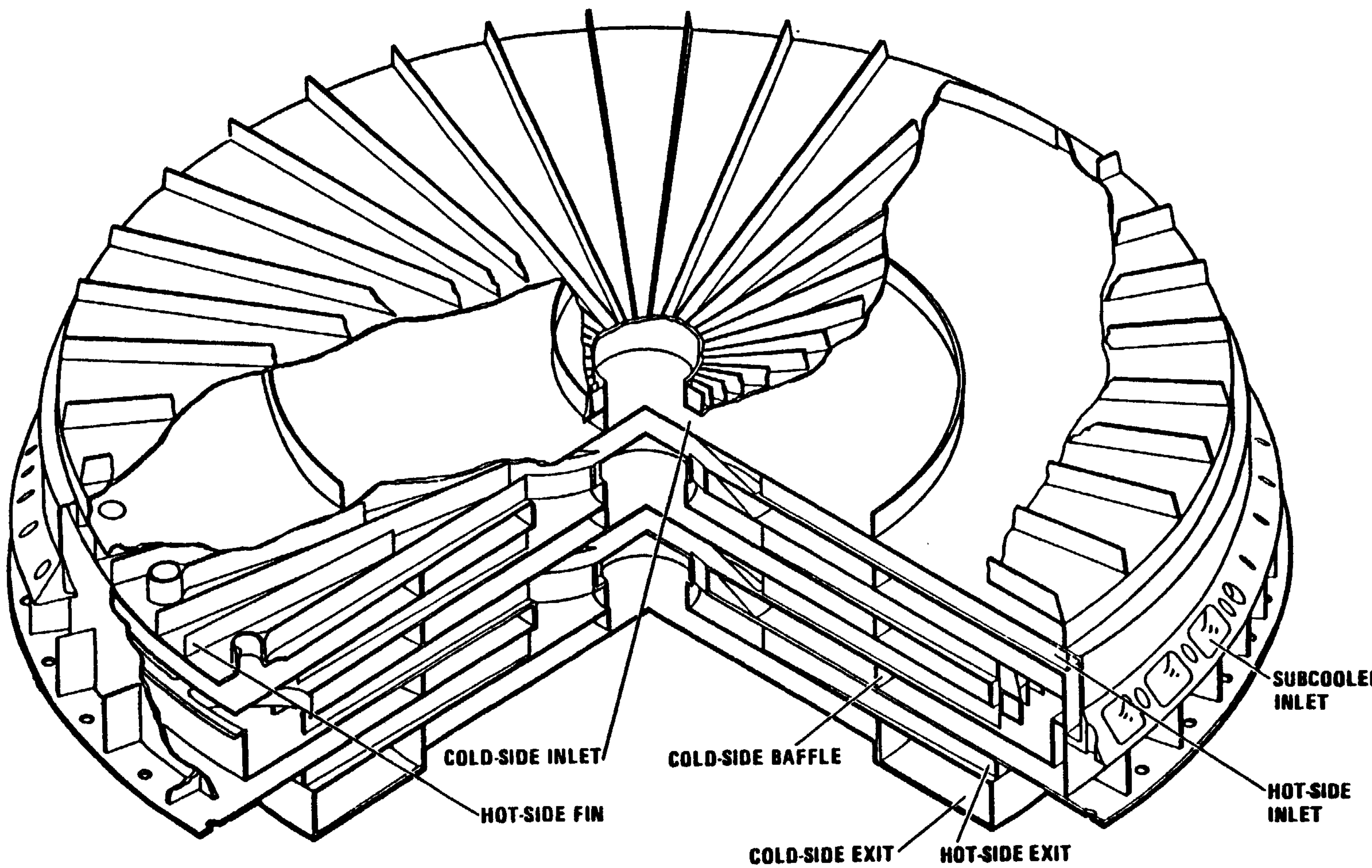
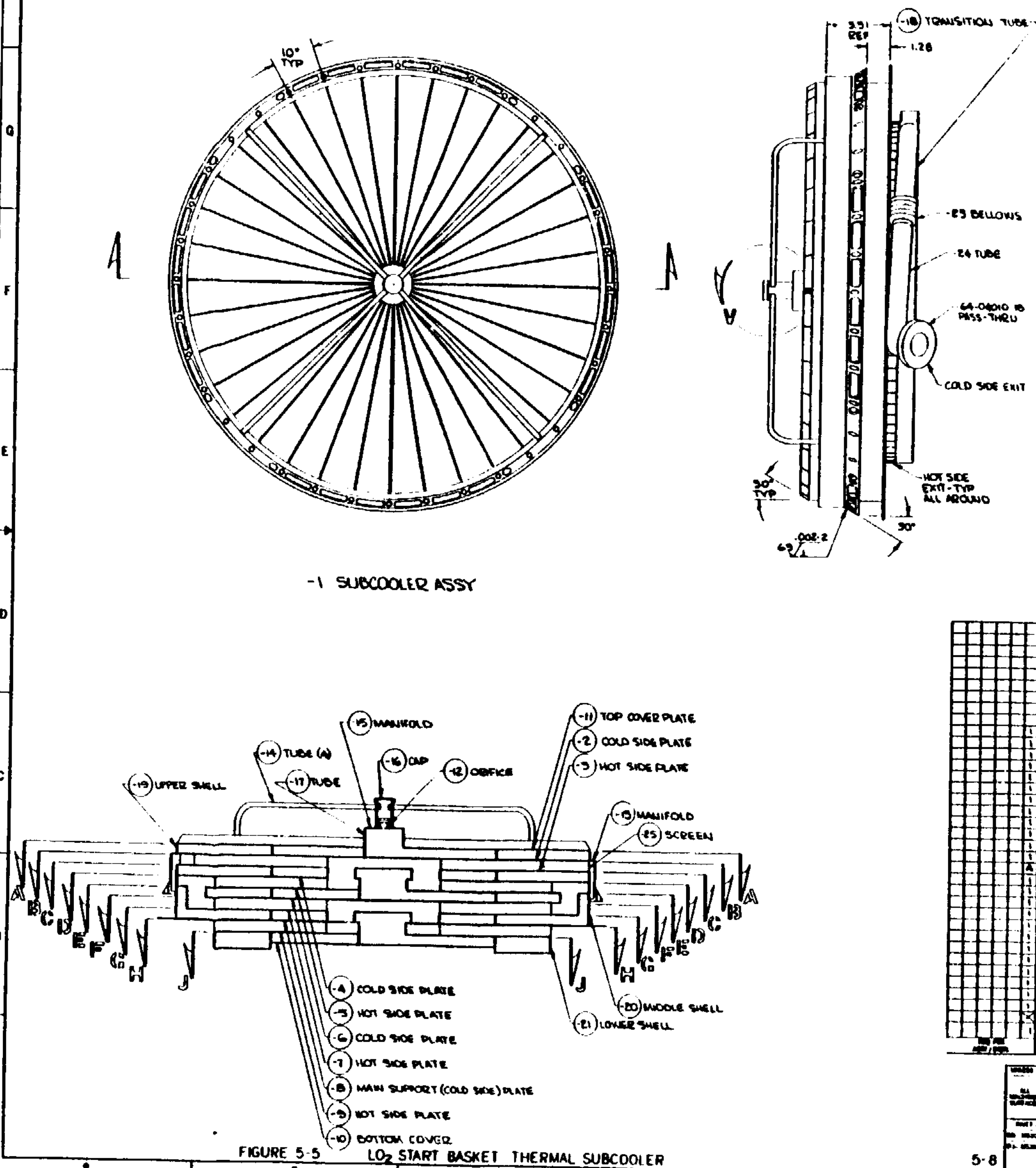


Figure 5-4. LO_2 Thermal Subcooler Isometric



NOTES: UNLESS OTHERWISE SPECIFIED

1- ALL MATERIAL 6061 AL ALLOY
2- INDIVIDUAL PLATES TO BE NC MACHINED FROM PLATE STOCK. ASSEMBLY TO BE DIP BRAZED TO ASSURE CONTINUOUS INTERNAL CONTACT
3- LEAKAGE OF COLD SIDE OF ASSY BETWEEN ORIFICE & EXIT NOT TO EXCEED 10^{-7} atm HE/SEC

PART OR IDENTIFYING NUMBER		NOMENCLATURE OR DESCRIPTION	MATERIAL, SPECIFICATION
	-25	SCREEN	7.5 X 10 200 X 600 MESH ALUMINUM SCREEN
64-04010-16		PASS-THRU	
	-24	TUBE	1.25 OD X .040 WALL X 6.0 6061-T4 AL ALLOY TUBE
	-23	BELLOWS	6061-T6 AL ALLOY
	-22	FLOW DIVIDER	.040 X 1.0 X 40.0 6061 AL ALLOY SHEET
	-21	SHELL-LOWER	.040 X 6 X 70.0 6061 AL ALLOY SHEET
	-20	SHELL-MIDDLE	.040 X 3.0 X 70.0 6061 AL ALLOY SHEET
	-19	SHELL-UPPER	.040 X 7 X 72.0 6061 AL ALLOY SHEET
	-18	TRANSITION TUBE	1.25 OD X .040 WALL X 6.0 6061-T4 AL ALLOY TUBE
	-17	TUBE	2.0 OD X .035 WALL 6061-T6 AL ALLOY TUBE
	-16	CAP	1.25 DIA X .5 6061-T6 AL ALLOY BAR
	-15	MANIFOLD	2.3 DIA X 1.8 6061-T6 AL ALLOY BAR
	-14	TUBE	.25 OD X .028 WALL X 14.0 6061-T4 AL ALLOY TUBE
	-13	MANIFOLD	21.8 ID X 24.0 OD X .25 RING 6061 AL ALLOY
	-12	ORIFICE	.75 OD X .5 6061-T6 AL ALLOY BAR
	-11	TOP COVER PLATE	.50 X 22.5 DIA 6061 AL ALLOY PLATE
	-10	BOTTOM COVER	.060 X 19.0 DIA 6061 AL ALLOY SHEET
	-9	PLATE	.62 X 19.0 DIA 6061 AL ALLOY PLATE
	-8	PLATE, MAIN SPRT	.62 X 24.5 DIA
	-7	PLATE	.62 X 21.5 DIA
	-6	PLATE	.62 X 22.5 DIA
	-5	PLATE	.62 X 22.5 DIA
	-4	PLATE	.62 X 22.5 DIA
	-3	PLATE	.62 X 22.5 DIA
	-2	PLATE	.50 X 22.5 DIA 6061 AL ALLOY PLATE
	-1	SUBCOOLER ASSY	

CONVAIR AEROSPACE DIVISION OF GENERAL DYNAMICS
SAN DIEGO, CALIFORNIA

SUBCOOLER -ASSY-
LO_2 START BASKET

E 14170 64-04011

SCALE 1/2

FIGURE 5-5 LO_2 START BASKET THERMAL SUBCOOLER

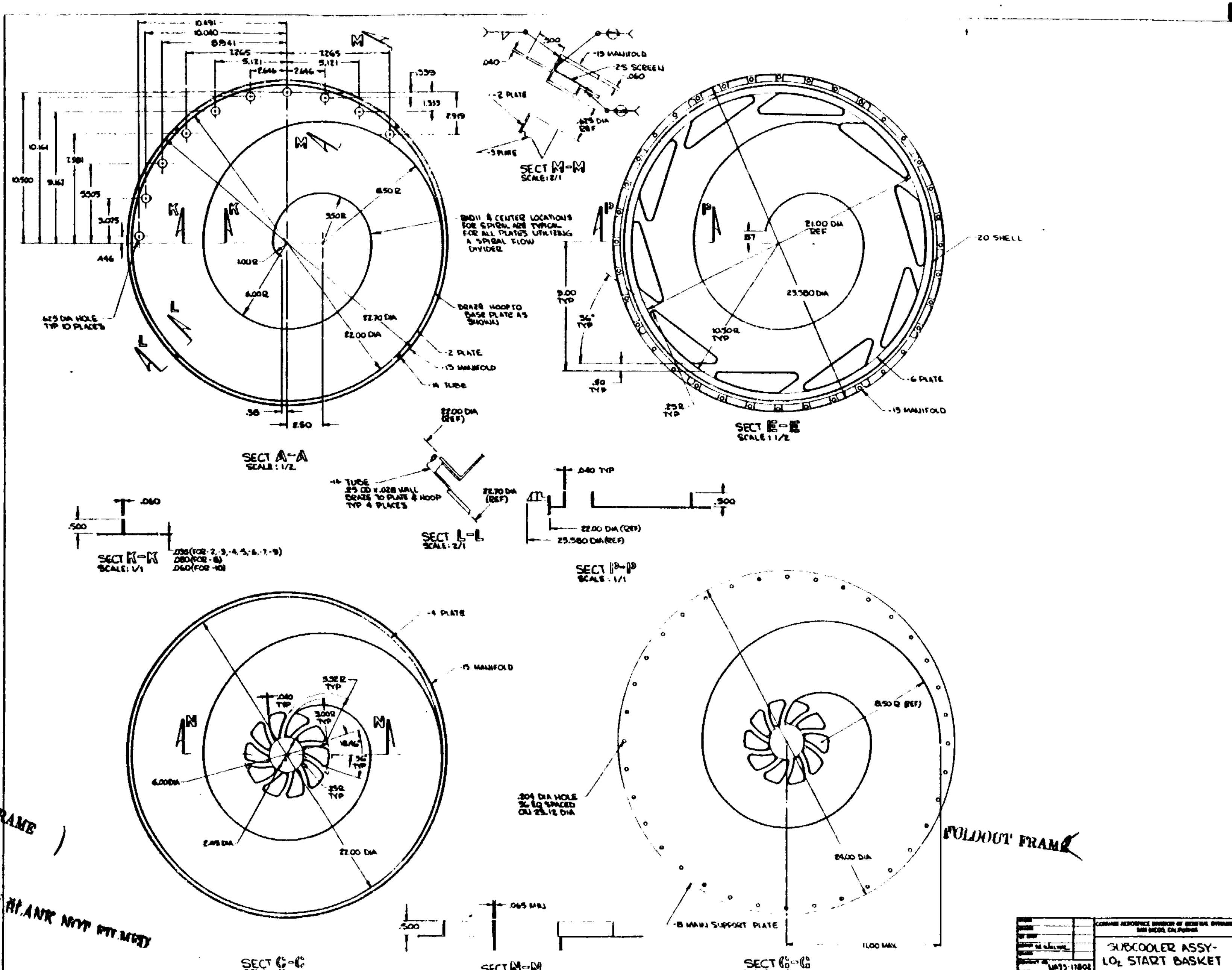


FIGURE 5-6. LO_2 START BASKET THERMAL SUBCOOLER

FOLDOUT FRAME

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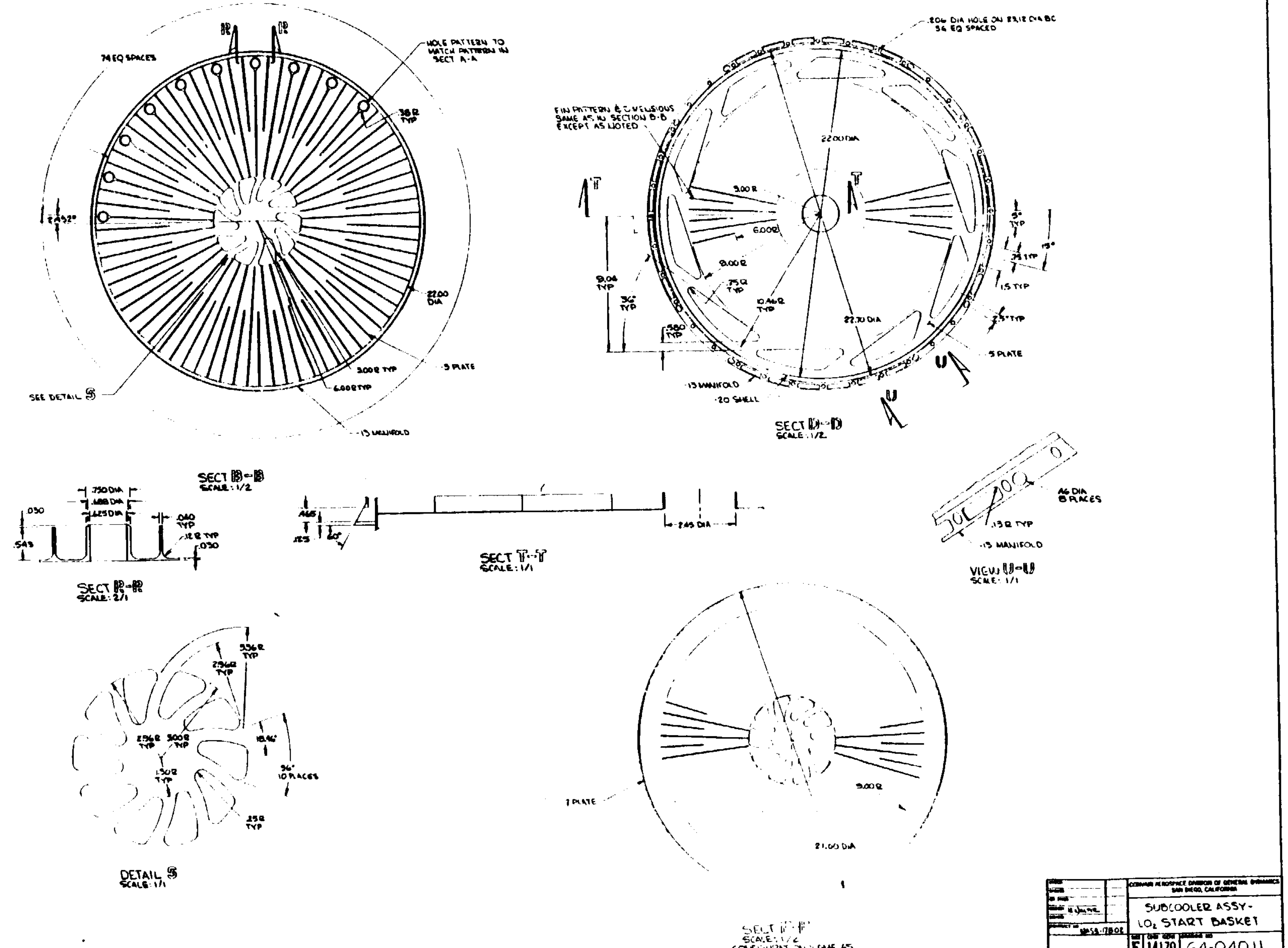


FIGURE 5-7. LO_2 START BASKET THERMAL SUBCOOLER

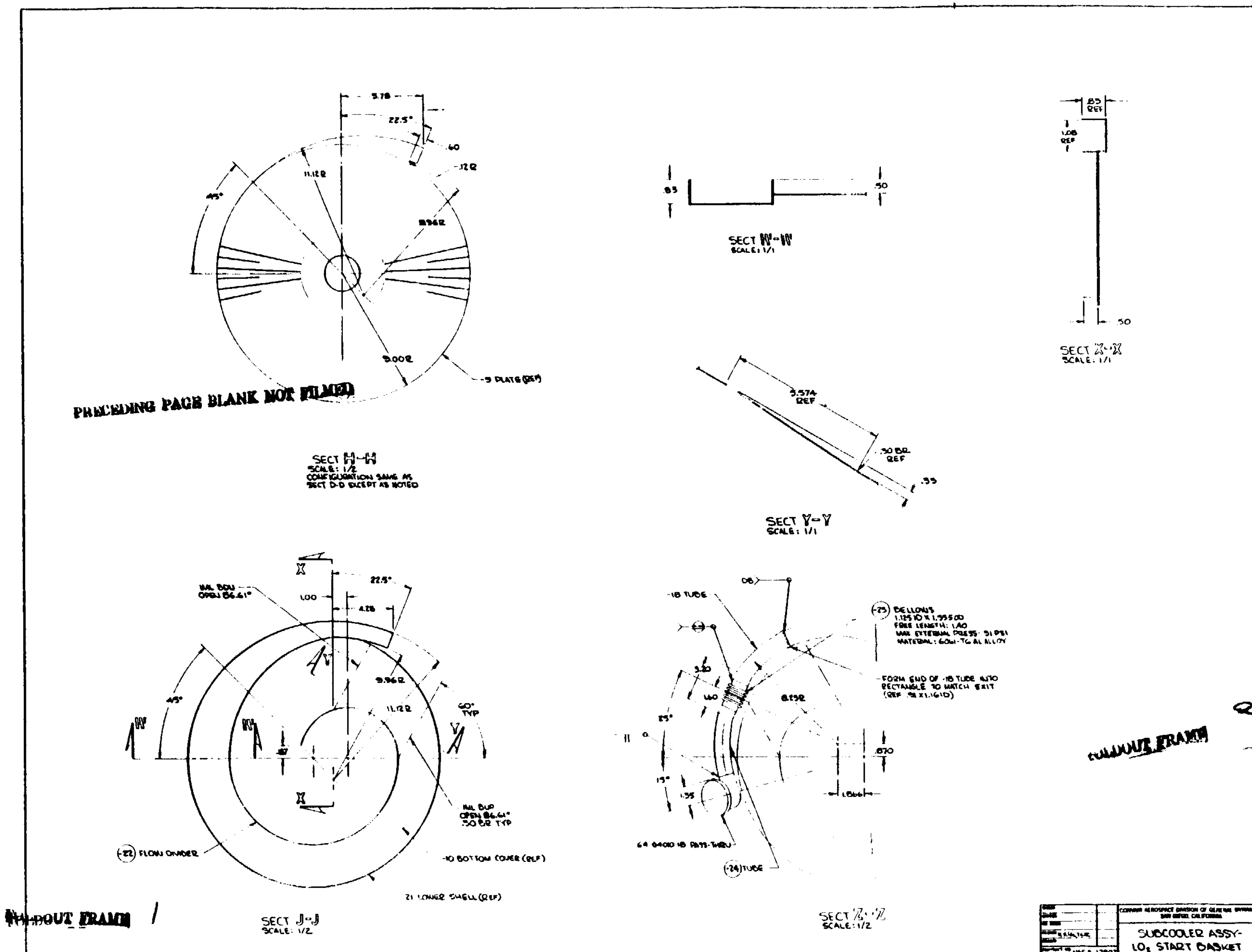


FIGURE 5-8. LO_2 START BASKET THERMAL SUBCOOLER

process is repeated until the fluid reaches the bottom area. Here, it is funneled into an outlet tube, through a sump pass-through and then either dumped overboard or pumped back into the main LO_2 tank.

Construction of the subcooler is intended to be by dip brazing. The individual plates are most efficiently made by numerical control milling. This allows the individual plates to be stacked together and the entire assembly to be dipped. The -13 manifold and -25 inlet screen are added later by welding. The exit tube has a bellows welded in to allow for movement between the sump wall and the subcooler. A pass-through (Figure 5-1, View D-D) allows the sump to be installed with the subcooler in place and the pass-through to be connected after sump installation. The pass-through seals can also be changed without removing the sump or subcooler.

The top surface of this subcooler has fins for structural support since this plate will have approximately 26 psid (179 kN/m^2) across the surface.

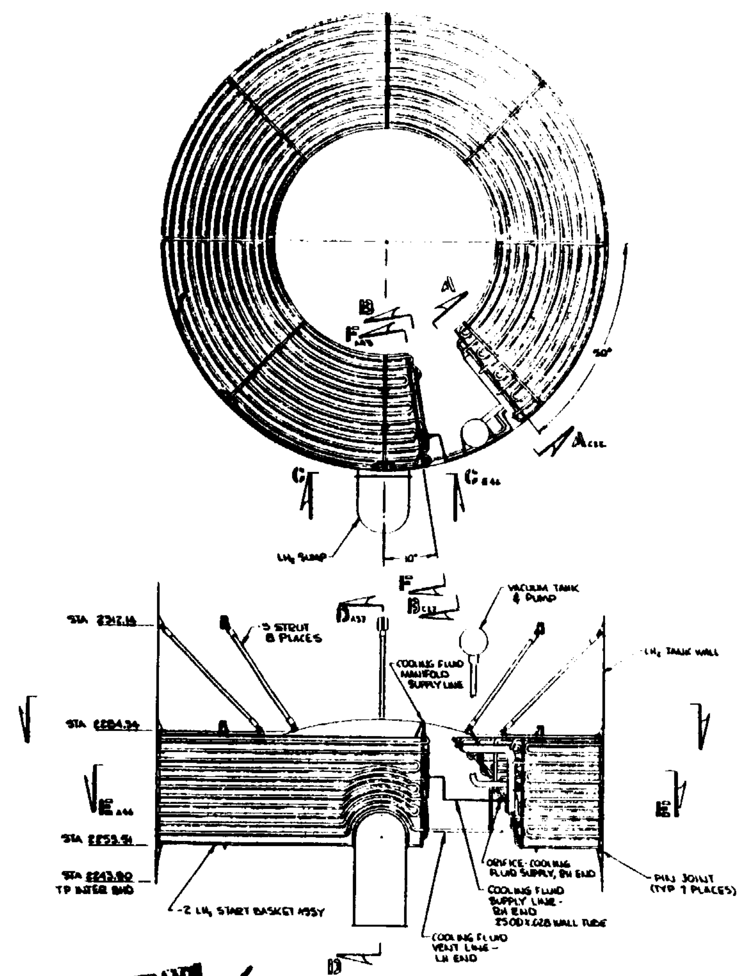
The overall dimensions of the subcooler are: 24.00 in(0.61 m) dia by 8.00 in (0.2 m) deep.

5.3 LH_2 START BASKET

LH_2 start basket details are shown in Figures 5-9 thru 5-11. The basket is designed to fit in the bottom of the LH_2 tank in the area above the intermediate bulkhead (Figure 5-11 Sect. D-D) and extend circumferentially around the outer portion of the tank for 330 degrees (5.76 radians). A gap is left for the fill and drain line entrance into the tank.

The basket assembly is attached only to the tank sidewall. There are no connections between the basket and the intermediate bulkhead. Space between the basket and the bulkhead allows for expansion of the bulkhead and for movement of the basket (due to thermal contractions and expansions, acceleration forces, etc.) There are eight struts to hold the inner edge of the basket, eight brackets at the basket outer top edge and seven pin joints at the outer lower edge of the basket plus a transition connector at the basket outlet to hold the basket in place.

The basket construction is similar to the LO_2 start basket in that the screen surface is covered by a perforated sheet and cooling coils are attached to the perforated sheet. In this case, however, the cooling coils are extruded with attaching flanges (Figure 5-11, Detail V) and are spot welded to the perforated sheet (Figure 5-10, View L-L). Spot welding is used because large panel size makes dip brazing unfeasible. Furnace brazing can be accomplished on large pieces, however, the different shapes, tubing versus perforated sheet, will expand at slightly different rates and cause gaps in the brazed joint. The distortion with spot welding should be minimal.



NOTES: UNLESS OTHERWISE SPECIFIED

1- TUBE MATERIAL: EXTRUDED - 6063 AL ALLOY ; DRAWN - 6061 AL ALLOY
2- PERFORATED SHEET: .040 GAGE 6061 AL ALLOY WITH .375 DIA HOLES ON .50 CENTERS
3- SCREEN SIZE: EXTERNAL SCREEN - TOP SECTION, 40 X 200 (TWP)
   EXTERNAL SCREEN - BOTTOM SECTION, 50 X 250 (TWP)
   INTERNAL SCREEN - DIVIDER, 14 X 14 SQUARE MESH X .020 WIRE
   SUBCOOLER INLET, 325 X 2300 (TWT)
4- SCREEN MATERIAL TO BE ALUMINUM IF AVAILABLE. CRES SCREEN MAY BE SUBSTITUTED IF ALUMINUM SCREEN IS NOT AVAILABLE
5- IN ORDER TO MINIMIZE TRAPPED VAPOR DURING REFILLING, A 4.5 INCH HIGH STANDPIPE OF 40 X 200 MESH SCREEN WILL BE INCORPORATED IN THE TOP BASKET SURFACE.
6- A LIQUID LEVEL SENSOR (CAPACITANCE PROBE) IS MOUNTED INSIDE THE START BASKET FOR LEVEL SENSING. THE PROBE IS CAPABLE OF SENSING MAXIMUM & MINIMUM LEVELS IN THE BASKET
7- THE MAIN TANK LEVEL SENSING PROBE IS RELOCATED TO THE OPEN AREA NEAR THE TANK FILL & DRAIN LINE
8- A SUMP BYPASS VALVE (REQUIRED TO PREVENT PRESSURE BUILDUP IN THE SUMP & SUBSEQUENT BACK FLOW INTO THE SUBCOOLER) & APPROPRIATE LINES ARE INSTALLED BETWEEN THE SUMP & THE MAIN TANK

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LH2 START BASKET INSTALLATION

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FIGURE 5-9. LH2 START BASKET

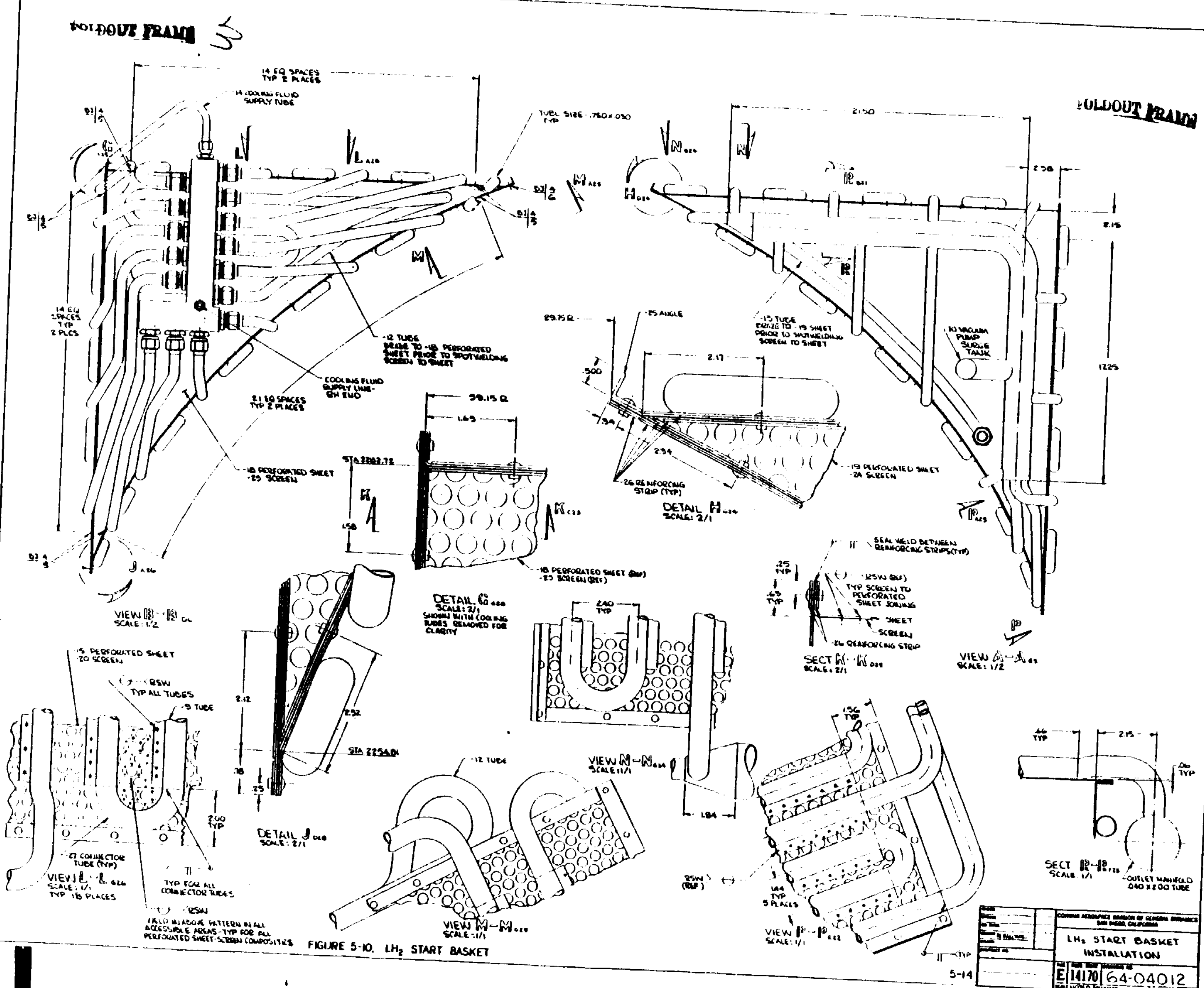

FIGURE 5-10. LH$_2$ START BASKET

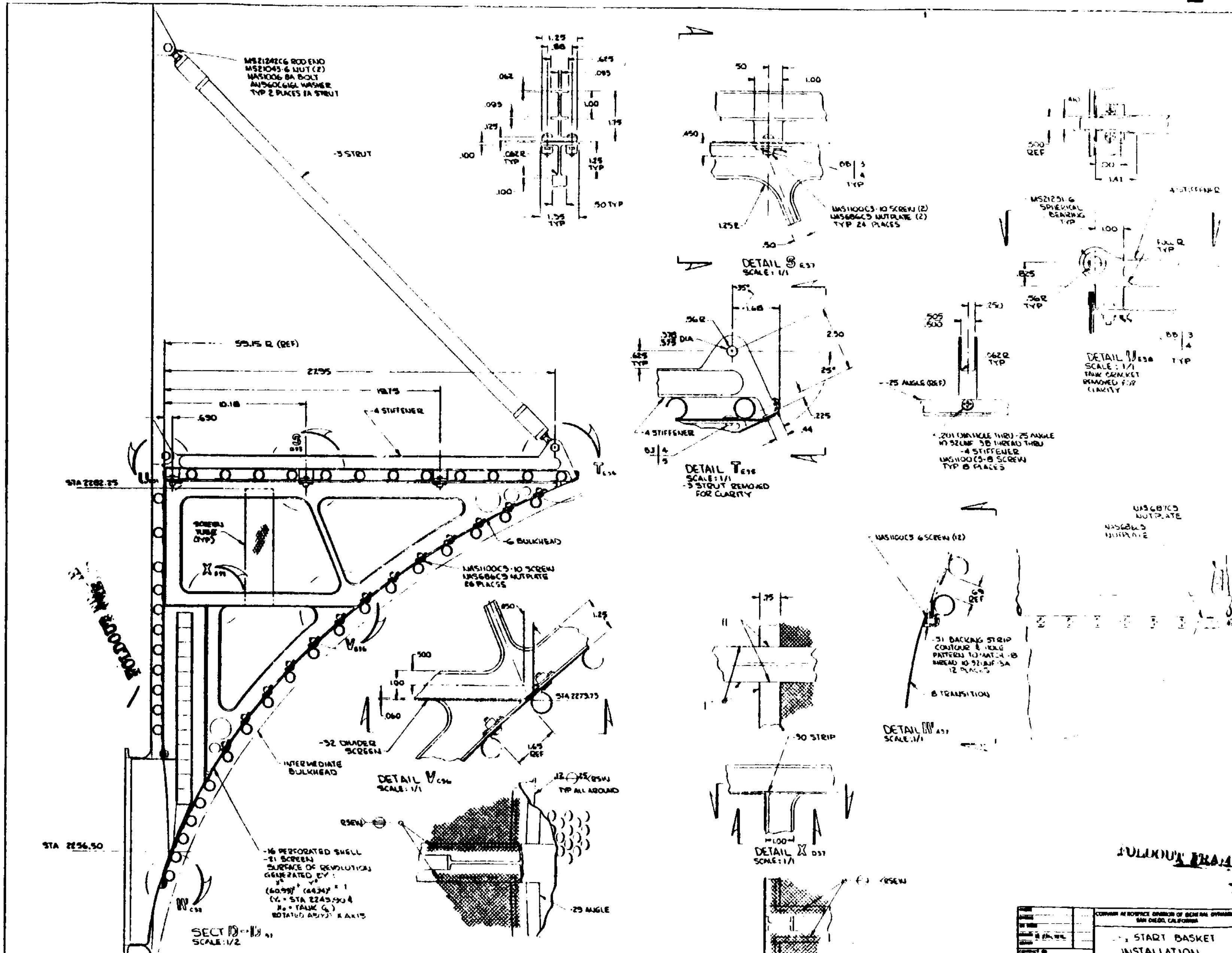

FIGURE 5-11. LH_2 START BASKET

The spacing between the cooling tubes is .69 inches (1.75 cm) in the area of the sub-cooler and channel (Figure 5-11, detail W) and 1.65 inches (4.19 cm) in other areas. The tubing size is .75 inch OD by .030 inch wall (1.91 cm OD by 0.076 cm) with a .75 inch (1.91 cm) wide flange. Tube spacing between tube roots is given in Section 4.3.

The curved surface of the basket (over the intermediate bulkhead) is held in shape by the use of a formed bulkhead (Figure 5-11, Sect. D-D) at each of the strut locations. The lower basket panel is bolted to each of these formed bulkheads. A stiffener, mounted on the basket top, transfers loads from the formed bulkheads into the tank brackets and struts. A coarse divider screen is installed inside the basket approximately 9 inches (0.23 M) below the top surface to make the basket into two compartments (Figure 5-11, Details V, X). Aluminum 14 mesh square weave screen is used. Screen tubes, of the same material, are located in the divider to keep the lower compartment filled with liquid (Figure 5-11, Sect. D-D, Figure 5-12, Sect. E-E).

The basket is designed to be made up in panels (top, outside, lower, right end and left end) and riveted and welded together. The riveting is for strength and the welding seals the joints.

Cooling fluid is brought from the screened channels to a manifold on the lefthand end panel (Figure 5-10, View B-B) where, in order to prevent excessive pressure drop through the cooling loops without having overly large cooling tubes, fourteen cooling loops are used to cool the top, outside and lower panels. The vent manifold is mounted on the right hand end panel. Each cooling loop requires the fluid to make three passes along the basket surface; from the supply manifold to the right hand end, back to the left hand end and back again to the right hand end to the vent manifold. Each cooling loop has its own individual orifice since the line lengths are not equal between loops. Separate cooling loops are provided to cool each end panel (Figure 5-9, 5-10, Sect. B-B, Sect A-A).

The vent fluid will either by plumbed to a vacuum surge tank and vacuum pump and back into the LH_2 tank or dumped directly overboard downstream of the start basket.

A stainless steel transition section is welded to the tank outlet to provide a transition between the aluminum basket and the steel tank. The transition section is bolted to the basket outlet. A modification to the fuel sump is required in that the pump bypass will be rerouted straight up from the top of the sump to a point above the basket before it enters the main tank. Since this can be accomplished with the same elevation on the bypass outlet and practically the same line length with only a minor change in bends, the pump should not require requalification.

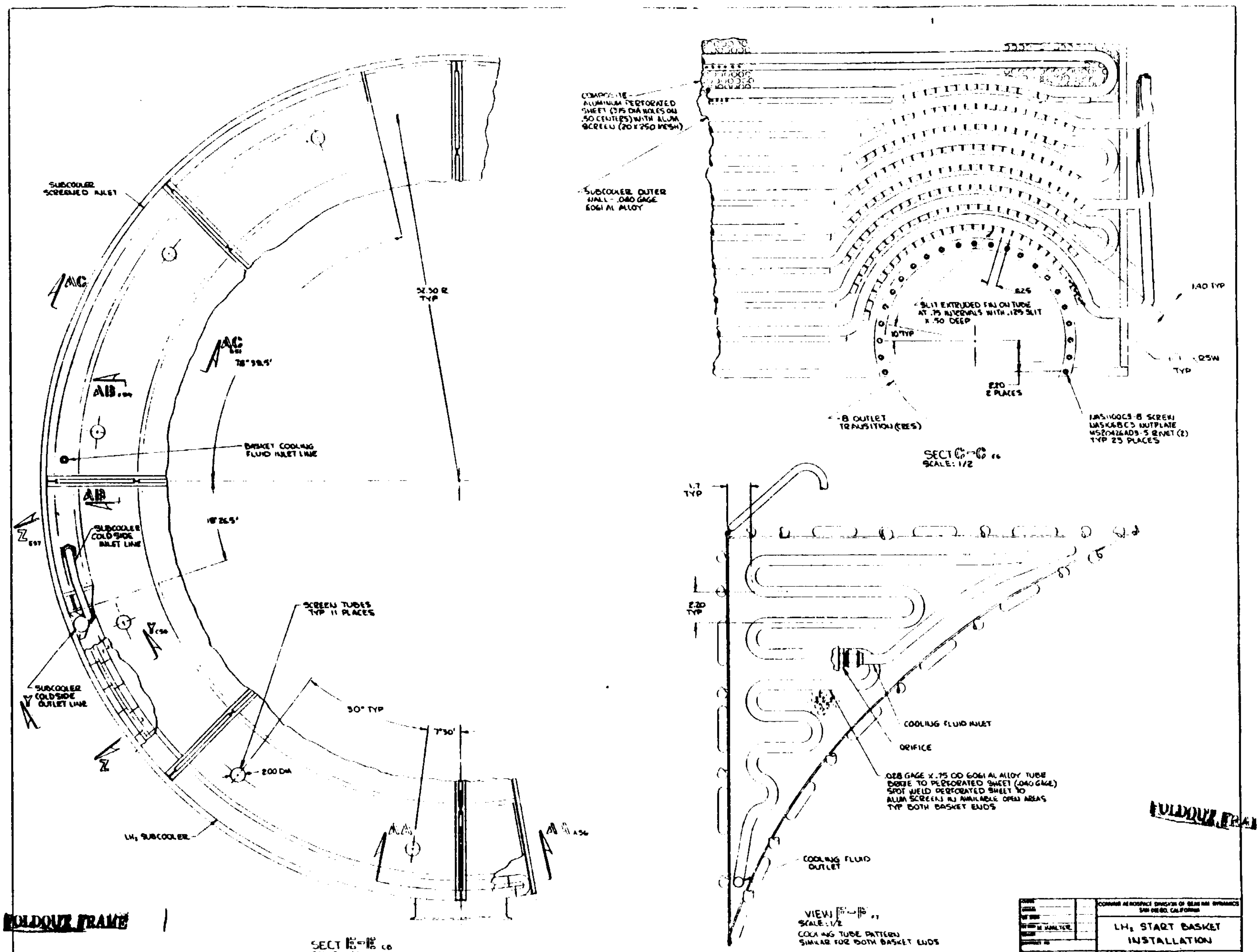

FIGURE 5-12. LH_2 START BASKET

In order to prevent pressure buildup in the sump and subcooler and subsequent screen breakdown and drying out, a sump bypass line with a shutoff valve is installed between the sump and the main tank. The valve can be opened to relieve pressure in the sump during non-operating times and closed when start basket operation is desired. The line and valve are not shown in the drawings.

5.4 LH_2 THERMAL SUBCOOLER

The LH_2 flow from the start basket is passed through a thermal subcooler prior to entering the basket outlet transition section (Figure 5-12, Sect. E-E). The subcooler consists of a hot side finned flow passage, a cold side flow passage with baffles and a screened inlet. Flow for the subcooler cold side is drawn from the screened inlet and expanded through an orifice (Figure 5-13, Sect. Z-Z). Baffles then help the fluid to cool the entire surface of the hot side flow passage (Figure 5-13, Sect. Y-Y). The cold fluid passes down the entire inside length of the subcooler, around the end and back to the inlet before it is vented out or pumped back into the main tank.

The screened channel is separated from the subcooler by a screen so that when there is no flow, the screened inlet will not empty. During fluid flow, the fluid passes through the inlet screen into the hot side of the subcooler. The hot side is sectioned into thirteen sections by fins which help transfer heat from the hot fluid to the cold side. The hot fluid only makes one pass along the length of the subcooler. At the outlet end the fins direct the flow downward toward the outlet (Figure 5-13, Sect. AA-AA).

The subcooler is made of both solid aluminum sheet and milled aluminum plate stock. The entire assembly is welded closed. The screened channel is of an angle framework construction with screen backing of perforated sheet (Figure 5-13, Sect. AC-AC, Detail AE). Angle bracing in the center prevents the sides from collapsing during fuel flow. The screened channel has a clearance of 1.00 inch (2.54 cm) all around to allow flow from all directions to enter the inlet.

The screened channel to the subcooler is 96 inches (2.44 m) long. The subcooler is 72 inches (1.83 m) long from the screened inlet to the center of the fluid outlet. The overall length of the subcooler is 82 inches (2.08 m)

5.5 LO_2 START TANK

The LO_2 start tank is basically a spherical dome mounted in the base of the LO_2 tank with a lower cover plate to isolate the LO_2 sump from the start tank. (Figure 5-14, Sect. A-A). A feed line with four screened inlet tubes provides flow during start

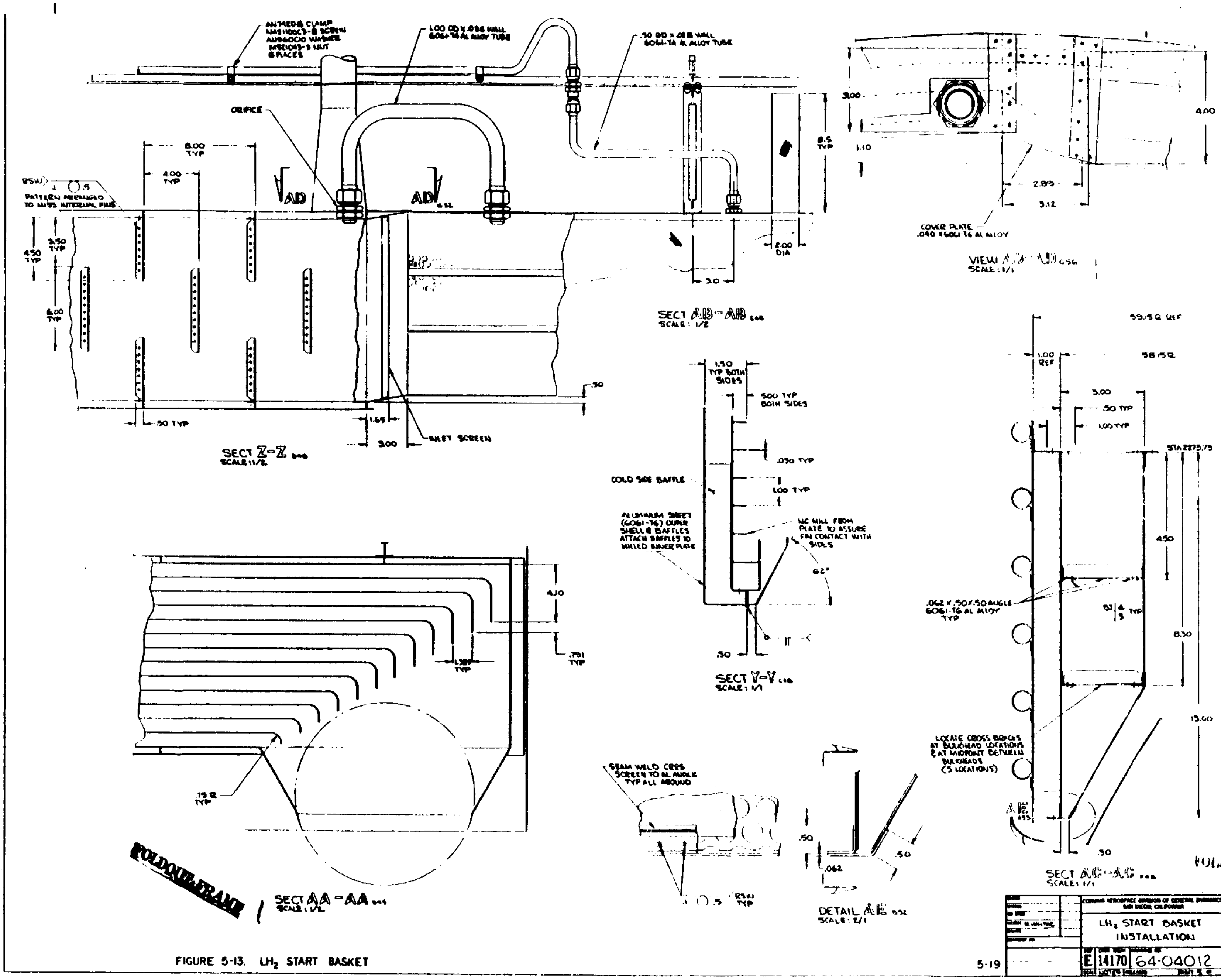

FIGURE 5-13. LH2 START BASKET

NOTES: UNLESS OTHERWISE SPECIFIED

1. A LIQUID LEVEL SENSOR (CAPACITANCE PROBE) IS MOUNTED INSIDE THE START TANK FOR LEVEL SENSING. THE PROBE IS CAPABLE OF SENSING MAXIMUM & MINIMUM LEVELS IN THE START TANK
2. THE EXTERNAL VALVES (VENT, REFILL, FEED) ARE STANDARD FLIGHT HARDWARE

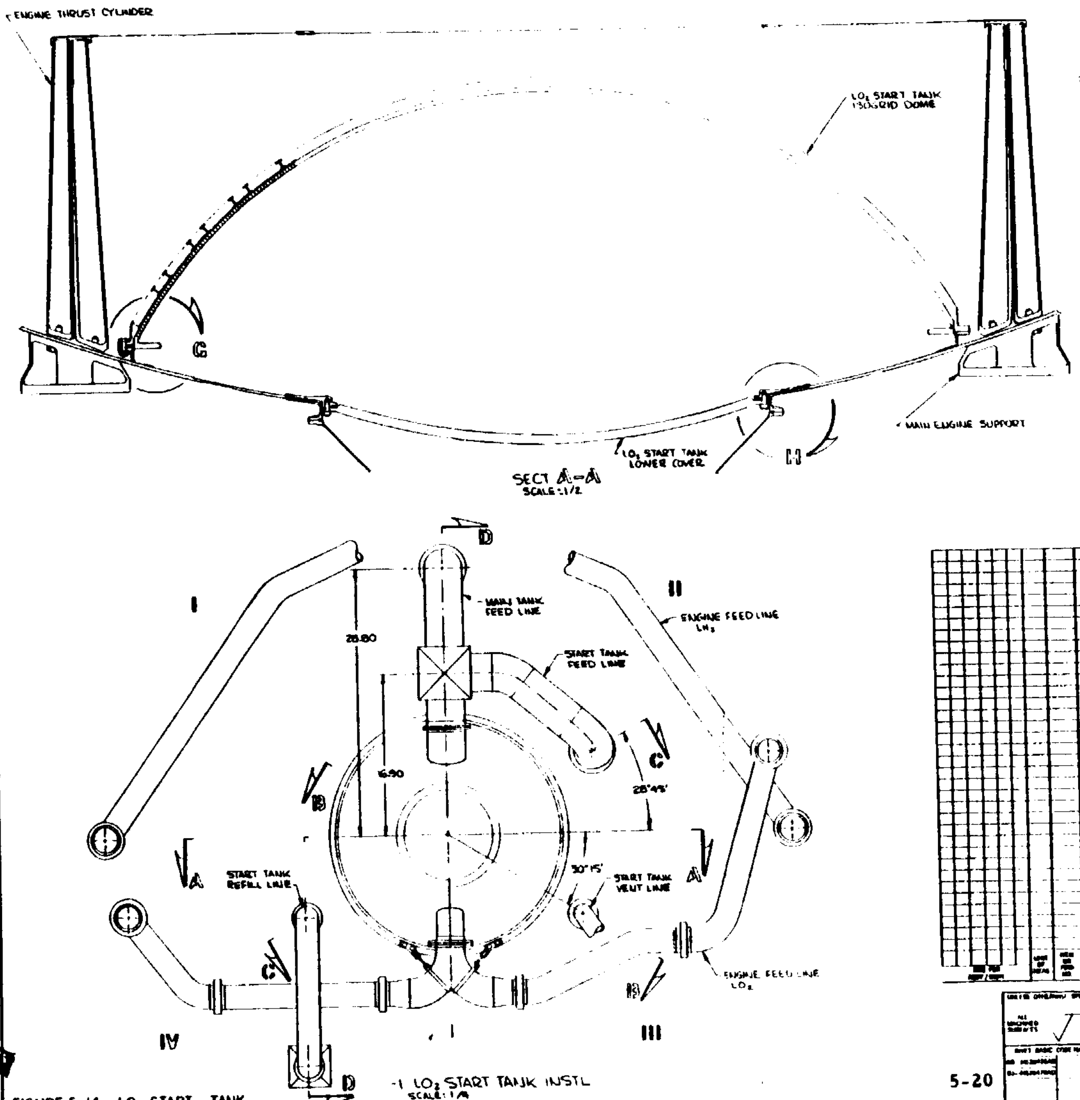

FIGURE 5-14. LO_2 START TANK

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LO_2 START TANK INSTALLATION

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tank through a three-way feed valve into the LO_2 sump. A main tank feedline (Figure 5-14, 5-15 Sect. D-D) provides flow for the engines after engine start. A start tank refill line (Figure 5-15 Sect. D-D) and a vent line (Figure 5-16, Sect. B-B) provide means of refilling the start tank when required. Flow through these lines is controlled by valves which are mounted outside the LO_2 tank for safety. The refill valve mounted outside the tank causes an additional line weight penalty compared to mounting the valve inside the tank.

The LO_2 tank spherical dome is of aluminum and is an isogrid design to minimize weight and maximize strength. It is designed to withstand 10 psi (68.9 kN/m^2) external pressure and 150 psi (1034 kN/m^2) internal pressure. The aluminum isogrid dome is attached to the CRES tank by means of bolts and a seal (Figure 5-15, Sect. G-G).

The start tank is insulated from the main tank by a fiberglass honeycomb (Flexcore) and an inner spring bulkhead liner (CRES). A vacuum line to the space between the inner liner and the isogrid dome is used to evacuate the honeycomb (Figure 5-16, Sect. C-C). The inner liner is designed with a spring ring so that internal pressure will hold it against the honeycomb. The pressure load is transferred through the honeycomb into the isogrid dome.

The cover between the start tank and the sump is also an isogrid dome. It is similar to the upper isogrid dome except it will not withstand as high an external pressure and it is not insulated (Figure 5-15, Sect E-E). It is bolted to an attachment ring at the tank outlet (Figure 5-15, Sect H-H).

The start tank feedline is shown in Figure 5-16, Sect C-C & Figure 5-14. Four pleated screen tubes are attached to an inlet manifold on the inlet end of the feedline to minimize the residuals left in the tank at tank depletion. The feedline is a CRES tube welded into the tank. The ends of the inlet screens are bolted to the tank wall to provide stability for the entire structure. The feed tube is routed to a three-way valve outside the LO_2 tank which allows flow from either the start tank or the main tank to flow through to the sump.

The start tank vent line (Figure 5-16, Sect. B-B) is designed to vent the start tank only when there is thrust on the vehicle. A perforated sheet across the top of the vent tube and a positive purge on the tube keeps liquid out of the vent tube when the tank is in low gravity. An identical approach is used on the existing Centaur LO_2 tank vent tube to prevent liquid loss during venting.

The start tank refill line, also designed for use where there is thrust on the vehicle, is basically a line from the bottom of the main tank to the inside of the start tank with a shutoff valve to close off the line. (Figure 5-15, Sect. D-D). A baffled outlet prevents

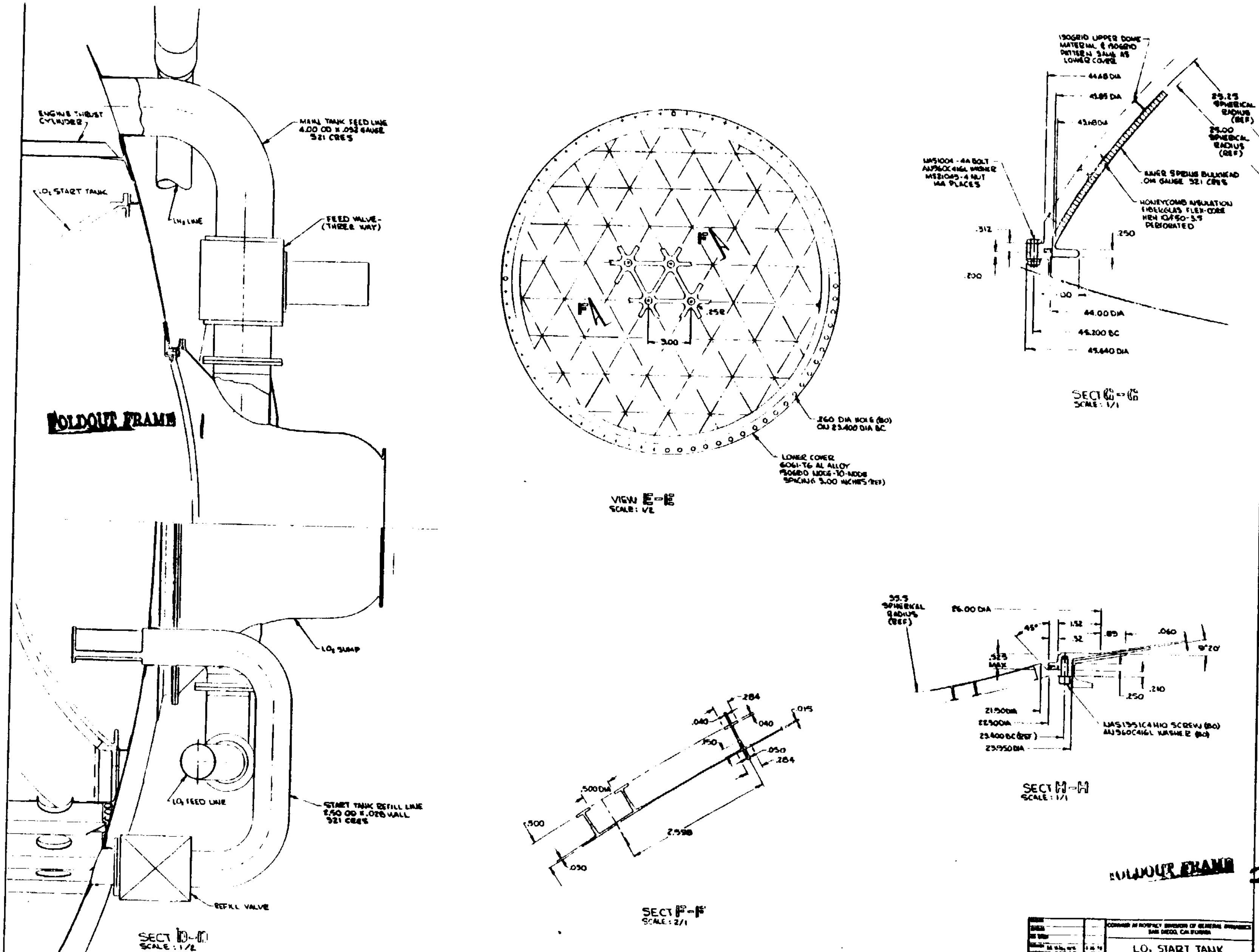


FIGURE 5-15. LO_2 START TANK

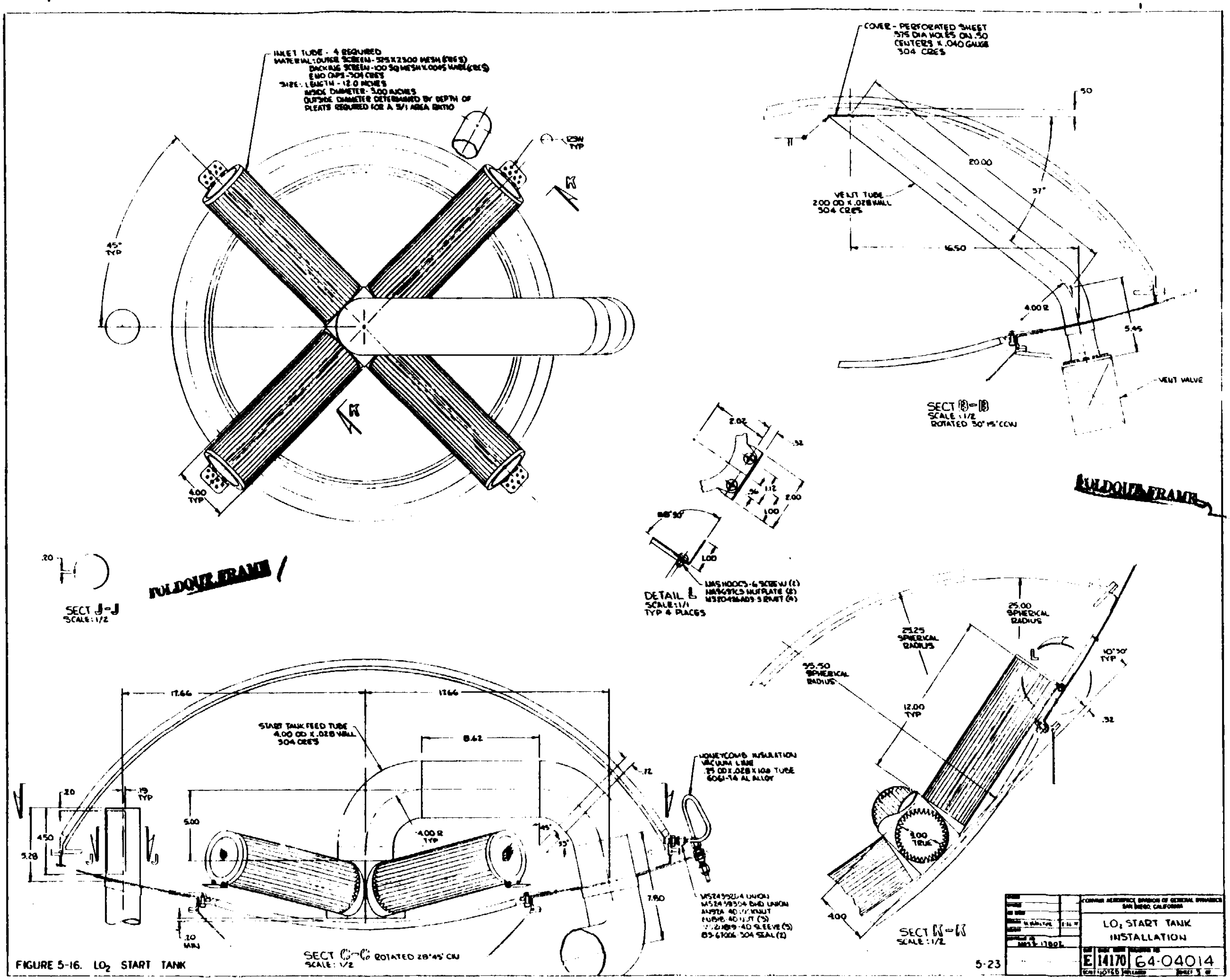


FIGURE 5-16. LO_2 START TANK

refill liquid from flowing over the vent inlet so that refill can be accomplished with the vent valve open.

A liquid level sensor, mounted inside the start tank, is used to determine valve operating times to prevent overflowing during refill or discharging gas during engine operation. This sensor will likely be a capacitance probe rather than point sensors to give better readout of levels during operation.

5.6 LH_2 START TANK

The LH_2 start tank is a toroidal tank located in the area above the intermediate bulkhead between the LO_2 and LH_2 tanks. (Figure 5-17). The tank is slanted on a 5° (0.087 rad) angle to promote drainage during outflow. The tank has an outflow line with a feed valve, a refill line with a refill valve and a vent line with a vent valve. All of the valves except the vent valve are mounted inside the LH_2 tank for optimum line orientation. The tank is held in place by six sets of struts (Figure 5-19, Sect. C-C) which will allow some movement as the tanks expand and contract. The feedline attachment is a hard line attached directly to the fuel sump. The vent line is the only other line attached directly to the main tank wall. This line has a bellows incorporated in it to allow relative movement between the two tanks at this point.

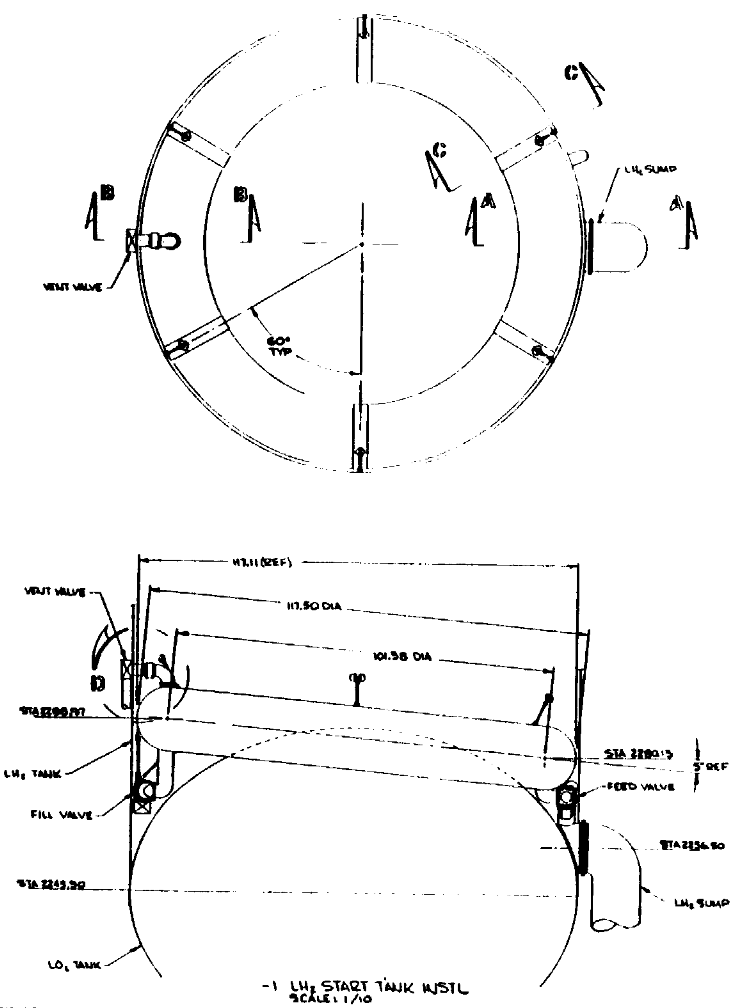
The start tank toroidal shell was designed for an external pressure of 5 psid (34.5 kN/m^2). Four different materials were investigated, aluminum, titanium, CRES and Inconel. The tank gauge was calculated per Reference 5-1 (Figure 5-20) and the corresponding shell weight calculated for each material. The aluminum, at 98 pounds (44.5 kg) was considerably lighter than any of the other three. Thus, with its ease of construction, it was chosen as the tank material. A fiberglass honeycomb with an aluminum vacuum barrier is installed on the outside of the tank. The honeycomb is .25 inches (.64 cm) thick and the aluminum skin is .020 inches (0.051 cm). The aluminum outer skin is sealed at all penetrations into the start tank and a vacuum line is provided to evacuate the honeycomb.

The start tank feed valve is a two-way valve that either feeds the fuel sump from the start tank or from the main tank. (Figure 5-18, Sect. A-A, Detail G). A special high flow, low pressure drop valve is required for this and a typical design is shown. The valve shown is basically a ball-sleeve combination that allows flow from both tanks while in an intermediate position. This prevents a momentary flow stoppage during valve actuation.

The start tank feedline has two inlet tubes consisting of pleated screens. Each screen tube is 5 inches (12.7 cm) inside diameter by 24 inches (0.61 m) long. The screened inlets are CRES and are connected to the aluminum inlet manifold by clamps (Figure 5-18, View F-F).

NOTES: UNLESS OTHERWISE SPECIFIED

1- HELIUM BOTTLES FOR START TANK PRESSURIZATION ARE MOUNTED INSIDE THE MAIN LH_2 TANK NEAR THE FWD TANK RING. BOTTLE REQUIREMENTS ARE:
   SIZE - 15.7 INCH DIA
   VOLUME - 1.07 FT^3 (26 LB HELIUM AT LH_2 TEMP)
   WEIGHT - 61 LB (INCLUDES SUPPORTS, FITTINGS & LINES)
   MATERIAL - CRES
   NUMBER OF BOTTLES - 1
2- LIQUID LEVEL SENSORS (CAPACITANCE PROBES) ARE MOUNTED INSIDE THE START TANK FOR LEVEL SENSING. TWO SENSORS ARE USED - ONE AT THE OUTLET (FEED VALVE) FOR MINIMUM LEVEL SENSING DURING OUTFLOW & ONE AT THE VENT OUTLET FOR MAXIMUM LEVEL SENSING DURING REFILL
3- TANK INSULATION CONSISTS OF .85 THICK FIBERGLASS PERFORATED HONEYCOMB CORE (3.5 #/FT^3 FLEX-CORE) WITH A .020 GAUGE ALUM OUTER SHELL VACUUM BARRIER. THE CORE AREA IS VENTED OUTSIDE THE MAIN TANK TO PROVIDE A VACUUM IN THE HONEYCOMB IN SPACE. THE OUTER SHELL IS SEALED AT ALL TANK PENETRATIONS
4 THE MAIN TANK LEVEL SENSOR WILL BE RECONFIGURED & RELOCATED TO BYPASS THE START TANK



FOLDOUT FRAME

FIGURE 5-17. LH_2 START TANK

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LH_2 START TANK INSTALLATION

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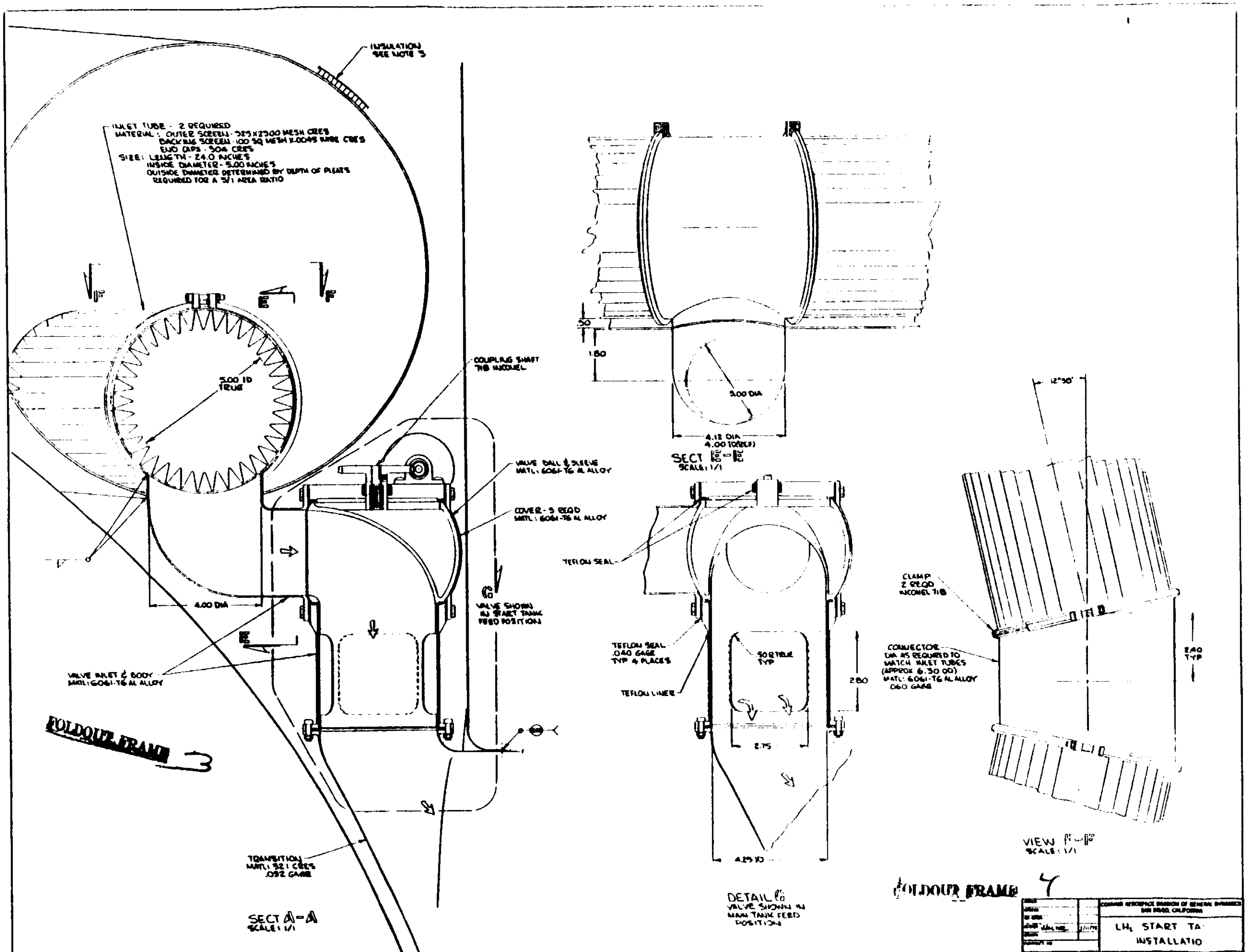

FIGURE 5-18. LH2 START TANK

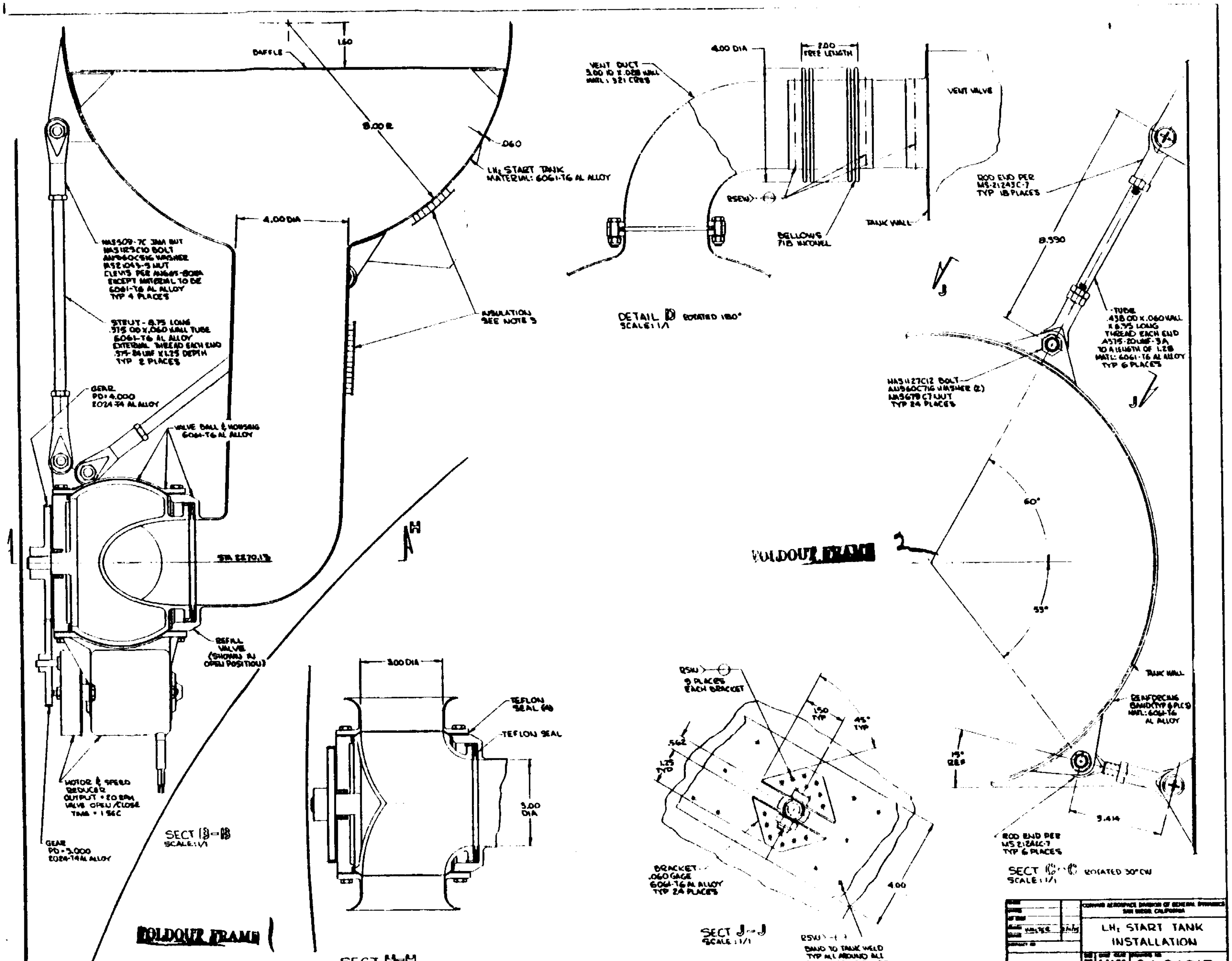

FIGURE 5-19. LH$_2$ START TANK

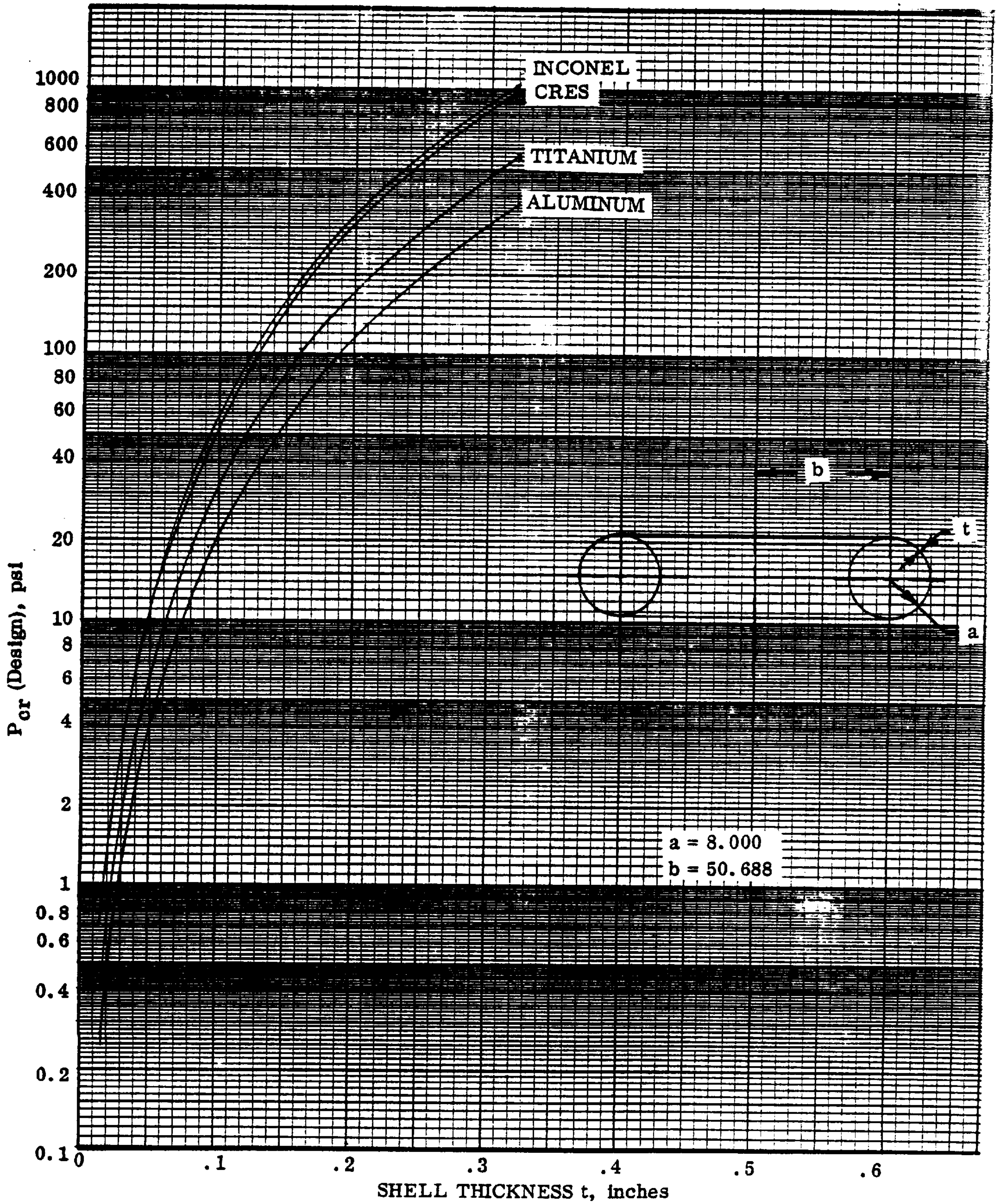

Figure 5-20. External Buckling Pressure - Toroidal Shell

The refill valve is a double opening ball valve to allow fluid to enter from both directions around the intermediate bulkhead. It is situated as far down in the corner of the tank as possible to reduce residuals. A baffle is installed in the start tank just above the refill line inlet to prevent flow from entering the vent line when both the refill line and the vent line are open.

Liquid level sensors are required in the start tank to determine when to close the vent and refill valves during refill and when to switch the feed valve to prevent gas discharge into the sump on engine start. Thus, one sensor is located near the vent or highest part of the tank and another sensor is located near the outflow tubes or the lowest part of the tank. These sensors will be capacitance probes rather than point sensors.

5.7 WEIGHT ESTIMATES

An estimate of the weights of the components in each configuration was made by calculating the material volumes and multiplying by the specific volumes of the material. Since the designs have not been fully analyzed for structural requirements, the actual, as built, weight will likely vary somewhat from those preliminary figures. There will also be a weight variation due to dimensional tolerances. As an example of the weight estimates performed, start basket and start tank hardware weights are given in Tables 5-1, 5-2, 5-3 and 5-4.

Table 5-1. LO_2 Start Basket Hardware Weight Estimates

Component	Hardware Weight - Lbs	(kg)
Start basket shell	11.9	(5.4)
Standpipe	2.2	(1)
Base ring	2.9	(1.3)
Attachment ring	2.1	(1)
Seal	0.2	(0.1)
Upper support ring	4.4	(2)
Struts	1.8	(0.8)
Top ring	0.6	(0.3)
Start basket screen	3.9	(1.8)
Channel assembly	8.8	(4.0)
Basket total	38.8	(17.6)
Capacitance probe	2.0	(0.9)
Bypass line	3.5	(1.6)
Assorted hardware	5.5	(2.5)
Cooling Coils	18.8	(8.5)
Thermal Subcooler	23.8	(10.8)
Vacuum pumping system-capillary device		
Battery	1.9	(0.9
Pump	5	(2.3)
Surge Tank	3	(1.4)
Pump System Total	9.9	(14.5)

Table 5-2. LH_2 Start Basket Hardware Weight Estimate

Component	Hardware Weight -lbs	(kg)
Start basket shell	58.9	(26.7)
Backing strips	4.1	(1.9)
Dividers	38.7	(17.6)
Nut plates	3.6	(1.6)
Transition to tank outlet	5.0	(3.9)
Frame	3.0	(1.4)
Strips	0.8	(0.4)
Screen 14 x 14 mesh	4.3	(2.0)
" 40 x 200 mesh	11.6	(5.3)
" 50 x 250 mesh	6.3	(2.9)
Inlet channel	2.0	(.9)
Basket total	138.3	(62.8)
Capacitance probe	2.0	(0.9)
Bypass line	3.5	(1.6)
Assorted hardware	5.5	(2.5)
Cooling coils	131.6	(59.8)
Thermal Subcooler	18.8	(8.5)
Vacuum pumping system-capillary device		
Battery	9.4	(4.3)
Pump	3	(1.4)
Surge tank	3	(1.4)
Pump System Total	15.4	(7.0)

Table 5-3. LO_2 Start Tank Hardware Weight Estimates

Component	Hardware Weight - lbs	(kg)
Insulation	10.8	(4.9)
Upper dome	25.8	(11.7)
Lower cover	5.2	(2.4)
Screens	2.9	(1.3)
Screen outlet	0.3	(0.1)
Outlet tube	4.7	(2.1)
Vent tube	1.3	(0.6)
Refill line	2.0	(0.9)
Main feed line	2.0	(1.0)
Flanges (line)	3.2	(1.5)
Attachment ring	9.8	(4.4)
Valves	24.5	(11.1)
Bolts and base	9.2	(15.3)
Miscellaneous	3	(1.4)
Start tank total	104.7	(47.5)
Capacitance probes	2.5	(1.1)

Table 5-4. LH_2 Start Tank Hardware Weight Estimates

Component	Hardware Weight lbm	(kg)
Start tank	97.7	(44.4)
Bands	9.6	(4.4)
Struts	7.5	(3.4)
Insulation	61.3	(27.8)
Inlet screen	5.3	(2.4)
Brackets	1.6	(0.7)
Transitor	5.0	(2.3)
Baffle	1.5	(0.7)
Feed Valve	7.8	(3.5)
Inlet valve	8.0	(3.6)
Refill line	1.0	(0.5)
Struts	0.4	(0.2)
Bellows	0.5	(0.2)
Flange	0.7	(0.3)
Miscellaneous	13.6	(6.2)
Start tank total	221.5	(100.6)
Capacitance probes	2.5	(1.1)

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SECTION 6

TASK V, SYSTEM COMPARISON

Comparisons were made between the capillary acquisition systems designed in Task IV and the baseline peroxide settling system and warm helium pressurization system. In addition to the actively cooled start baskets, passively cooled start baskets using capillary pumping to replace the cooling coils were considered. The system options compared were:

Option 1 - Baseline system - pressurization system plus settling system.

Option 2 - Start baskets using passive capillary device cooling (wicking) and subcoolers for providing boost pump NPSH with subcooler coolant flow dumped overboard.

Option 3 - Start baskets using passive cooling and subcoolers for NPSH with coolant flow pumped back into the tank.

Option 4 - Start baskets using cooling coils for capillary device cooling and subcoolers for NPSH with all coolant flow dumped overboard.

Option 5 - Start baskets using cooling coils for capillary device cooling and subcoolers for NPSH with all coolant flow pumped back into the tank.

Option 6 - Start baskets using cooling coils for capillary device cooling and subcoolers for NPSH with cooling coil flow dumped overboard and subcooler flow pumped back into the tank.

Option 7 - Start baskets using cooling coils for capillary device cooling and subcoolers for NPSH with cooling coil flow pumped back into the tank and subcooler flow dumped overboard.

Option 8 - Bypass feed start tanks with cold helium pressurization.

System options are described in more detail in Appendix A. These descriptions were used to obtain relative reliability comparisons for the eight system options. Comparisons were also made on the basis of hardware weight, payload penalty, recurring costs, power requirements and flight profile flexibility. The following sections describe the individual comparisons performed for the three missions of interest. The comparisons include all subsystems and processes that were affected by acquisition system selection (See Appendix A for example).

6.1 RELIABILITY

Reliability values were obtained for the eight system options for the synchronous equatorial and low earth orbit mission profiles shown in Tables 1-1, 1-2 and 1-3. Individual components required for each option were tabulated and reliability for each system was assessed using estimated failures per million hours of operation. Identical prediction techniques as used for the Centaur D-1S in Reference 6-1 were employed. The failure rate model follows a negative exponential function. The exponent of the function is the sum of the failure rates multiplied by an environmental factor (boost, coast, main engine burn) and the mission time. Table 6-1 gives the relative reliability for each option for each of the three missions examined. Mean missions between failures are also given. The baseline system is the most reliable mainly because the baseline pressurization system is largely redundant and therefore does not degrade system reliability appreciably. Bypass valves degrade the capillary system reliability.

Table 6-1. Relative Reliability

Option	Five Burn		Two Burn		One Burn	
	R[xx]	MMBF[x]	R	MMBF	R	MMBF
1	0.999522	2092	0.999585	2409	0.999721	3584
2	0.999411	1698	0.999488	1953	0.999659	2933
3	0.999214	1272	0.999317	1464	0.999541	2179
8	0.998440	641	0.998644	737	0.999089	1098
4	0.997655	426	0.997962	491	0.998655	743
5	0.997497	399	0.997824	459	0.998538	684
6	0.997497	399	0.997824	459	0.998538	684
7	0.997497	399	0.997824	459	0.998538	684

(x) MMBF = Mean missions between failures, which is another measure of reliability.

(xx) R = Reliability.

6.2 HARDWARE WEIGHT

Hardware weights determined for each of the eight system options for each of the three missions are shown in Table 6-2. Hardware weights for the capillary systems were determined from weight analysis of the drawings and descriptions of Section 5. The weight of the passively cooled start basket was taken to be the weight of the start baskets (using active conditioning) with cooling coil weight deleted. Assorted hardware weight includes bypass lines and valves, and liquid level sensors. The vacuum system weight includes batteries, pumps, lines and surge tanks. The thrust cylinder weight penalty is caused by increased settling loads during main engine thrust settling. The settling system hardware weight penalty consists of one peroxide bottle with bladder and fittings, and four settling rockets with fittings and lines. The pressurization system hardware weight penalty consists of pressure bottles, solenoid valves and lines.

As previously stated, components that are identical for all eight options are not included in the comparison. Table 6-2 indicates that the start basket options have lower hardware weight for the five burn mission than the baseline system. The start tank hardware weight is quite high due to tankage and valve requirements. Options 2 and 3, the passively cooled start baskets are also lighter than the baseline system for the two burn and one burn missions.

6.3 PAYLOAD WEIGHT PENALTY.

The most meaningful weight comparison that can be made is to compare the equivalent payload weight of each option for the three missions of interest. This is done in Tables 6-3, 6-4 and 6-5 using payload sensitivity factors given in Table 2-1. Hardware weight penalties are included as well as liquid residuals, chilldown penalty, boiloff penalty for pumping fluid back into the tank, pressurant usage, peroxide usage, start tank venting penalty and the penalty for occupying tank volume that would otherwise be available for containing propellant. Two additional system options were considered. For the two options that dump capillary device coolant flow overboard, weighted average accelerations are used for computing heat transfer coefficients. These options, 4a and 6a, adjust the flow rate using a sensing system, to satisfy incident heating requirements during the missions. Options 4 and 6,using maximum disturbing accelerations to determine heat transfer coefficients and vent rates, vent at a constant flow rate in orbit.

Liquid residuals for the baseline Centaur D-1S system were determined using burnout accelerations for each mission. For LH_2, curves of LH_2 mass in the tank versus burnout acceleration from Reference 6-2 were used. For LO_2,residuals were based upon the head required to give the proper boost pump NPSH at burnout.

For the start baskets, residuals were computed for the tank and channels. These residuals were equal to the liquid in the tank at the time of penetration of vapor .

Table 6-2. Hardware Weight Penalty lb_m (kg_m)

	5 Burn Mission								2 Burn Mission								1 Burn mission							
	Option 2	Option 3	Option 4	Option 6	Option 7	Option 5	Option 1	Option 8	Option 2	Option 3	Option 1	Option 4	Option 6	Option 7	Option 5	Option 8	Option 2	Option 3	Option 1	Option 4	Option 6	Option 7	Option 5	Option 8
Capillary Device	177.1	177.1	177.1	177.1	177.1	177.1	-	326.2	177.1	177.1	-	177.1	177.1	177.1	177.1	326.2	177.1	177.1	-	177.1	177.1	177.1	177.1	326.2
Cooling Coils	-	-	150.4	150.4	150.4	150.4	-		-	-	-	150.4	150.4	150.4	150.4		-	-	-	150.4	150.4	150.4	150.4	
Assorted Hardware	11	11	11	11	11	11	-	5	11	11	-	11	11	11	11	5	11	11	-	11	11	11	11	5
Subcooler	42.6	42.6	42.6	42.6	42.6	42.6	-		42.6	42.6	-	42.6	42.6	42.6	42.6	-	42.6	42.6	-	42.6	42.6	42.6	42.6	
Vacuum System	-	15.9	-	15.9	25.3	28.2	-		-	15.7	-	-	15.7	22.8	25.5		-	15.6	-	-	15.6	18.2	20.8	
Thrust Barrel	11.2	11.2	11.2	11.2	11.2	11.2	-	11.2	11.2	11.2	-	11.2	11.2	11.2	11.2	11.2	11.2	11.2	-	11.2	11.2	11.2	11.2	11.2
Settling System	-	-	-	-	-	-	77	-	-	-	77	-	-	-	-	-	-	-	77	-	-	-	-	-
Pressurization System	60	60	60	60	60	60	417	348	60	60	300	60	60	60	60	304	60	60	256.7	60	60	60	60	288
TOTAL	301.9	317.8	452.3	468.2	477.6	480.5	494	690.[illegible]	301.9	317.6	377	452.3	468	475.1	477.8	646.4	301.9	317.5	333.7	452.3	467.9	470.5	473.1	630.4
	(137)	(144)	(205)	(212)	(217)	(218)	(224)	(313)	(137)	(144)	(171)	(205)	(212)	(215)	(217)	(293)	(137)	(144)	(152)	(205)	(212)	(213)	(215)	(286)

Options:
1. Baseline D-1S
2. Start basket, passive cooling, subcooler dumped overboard.
3. Start basket, passive cooling, pumped subcooler.
4. Start basket, thermal conditioning (TC) and subcooler dumped, active cooling.
5. Start basket, TC and subcooler pumped back into the tank, active cooling.
6. Start basket, TC dumped, subcooler pumped, active cooling.
7. Start basket, TC pumped, subcooler dumped, active cooling.
8. Start tank bypass feed.

Table 6-3. Acquisition System Equivalent Payload Weight Comparison lb_m (kg_m)

5 Burn (Low Earth Orbit Mission Profile)

Weight Penalty Element		Option 3	2	5	1	7	8	6a	4a	6	4
Capillary Device	LH_2 LO_2	138.3 38.8	138.3 38.8	138.3 38.8	- -	138.3 38.8	221.5 104.7	138.3 38.8	138.3 38.8	138.3 38.8	138.3 38.8
Cooling Coils	LH_2 LO_2	-	-	131.6 18.8	-	131.6 18.8	-	131.6 18.8	131.6 18.8	131.6 18.8	131.6 18.8
Assorted Hardware		11	11	11	-	11	5	11	11	11	11
Subcooler	LH_2 LO_2	18.8 23.8	18.8 23.8	18.8 23.8	-	18.8 23.8	-	18.8 23.8	18.8 23.8	18.8 23.8	18.8 23.8
Dumped Fluid, Subcooler	LH_2 LO_2	-	38 119	-	-	38 119	-	-	38 119	-	38 119
Dumped Fluid, Capillary Device Cooling	LH_2 LO_2	-	-	-	-	-	-	263.4 154.9	263.4 154.9	415 394	415 394
Dumped Fluid, Start Tank	LH_2 LO_2	-	-	-	-	-	12.4 21.3	-	-	-	-
Vac. System, Subcooler (Pumping Coolant Back to the Tank) Includes Boiloff + Battery + Hardware	LH_2 LO_2	5.9 11	-	-	-	-	-	5.9 11	-	5.9 11	-
Vac. System, Capillary Device Cooling	LH_2 LO_2	-	-	-	-	22.9 12.6	-	-	-	-	-
Vac. System, Subcooler and Capillary Device	LH_2 LO_2	-	-	23.8 15.6	-	-	-	-	-	-	-
Vol. Penalty Due to Not Loading Fluid Because of Vol. Displaced by Added Hardware		30.2	30.2	51.7		51.2	22.7	51.4	50.3	51.4	50.3
Residual Payload Penalty**	LH_2 LO_2	21.3 114.5	21.3 114.5	21.3 114.5	80 84.5	21.3 114.5	70 132.5	21.3 114.5	21.3 114.5	21.3 114.5	21.3 114.5
Δ Childown Penalty due to Start Sequence		1014.1	1014.1	1014.1	979.8	1014.1	1014.1	1014.1	1014.1	1014.1	1014.1
Thrust Barrel Revisions		11.2	11.2	11.2		11.2	11.2	11.2	11.2	11.2	11.2
Settling System Including Peroxide Payload Penalty		-	-	-	165.2	-	-	-	-	-	-
Pressurization System		64	64	64	431	64	378	64	64	64	64
Total		1502.9 (683) 1406.6* (639)	1643.0 (746) 1546.7* (702)	1697.3 (771)	1740.5 (1084)	1849.9 (840)	1993.4 (905)	2092.8 (950)	2231.8 (1013)	2483.5 (1128)	2622.5 (1191)

* Assumes that subcooler retards pull through.

** Worst case assumptions (subcooler does not retard pullthrough).

Options:
1. Baseline D-1S
2. Start basket, passive cooling, subcooler dumped overboard.
3. Start basket, passive cooling, pumped subcooler.
4. Start basket, thermal conditioning (TC) and subcooler dumped, active cooling.

4a. Uses vent rate adjusted to suit "g" level, active cooling.

5. Start basket, TC and subcooler pumped back into the tank, active cooling.
6. Start basket, TC dumped, subcooler pumped, active cooling.

6a. Uses vent rate adjusted to suit "g" level, active cooling.

7. Start basket, TC pumped, subcooler dumped, active cooling.
8. Start tank bypass feed.

Table 6-4. Acquisition System Equivalent Payload Weight Comparison lb_m (kg_m)

2 Burn (Synchronous Equatorial Mission Profile)

Weight Penalty Element		Option									
		3	1	2	5	7	8	6a	4a	6	4
Capillary Device	LH_2	138.3	-	138.3	138.3	138.3	221.5	138.3	138.3	138.3	138.3
	LO_2	38.8		38.8	38.8	38.8	104.7	38.8	38.8	38.8	38.8
Cooling Coils	LH_2	-	-	-	131.6	131.6	-	131.6	131.6	131.6	131.6
	LO_2				18.8	18.8		18.8	18.8	18.8	18.8
Assorted Hardware		11	-	11	11	11	5	11	11	11	11
Subcooler	LH_2	18.8	-	18.8	18.8	18.8	-	18.8	18.8	18.8	18.8
	LO_2	23.8		23.8	23.8	23.8		23.8	23.8	23.8	23.8
Dumped Fluid, Subcooler		-	-	28.4	-	28.4	-	-	28.4	-	28.4
				88.9		88.9			88.9		88.9
Dumped Fluid, Capillary Device Cooling	LH_2	-	-	-	-	-	-	194.4	194.4	313.5	313.5
	LO_2							114.2	114.2	305.2	305.2
Dumped Fluid, Start Tank	LH_2	-	-	-	-	-	3.0	-	-	-	-
	LO_2						5.3				
Vac. System, Subcooler (Pumping Coolant Back to the Tank) Includes Boiloff + Battery + Hardware	LH_2	5.7	-	-	-	-	-	5.7	-	5.7	-
	LO_2	10.7						10.7		10.7	
Vac. System, Capillary Device Cooling	LH_2	-	-	-	-	18.2	-	-	-	-	-
	LO_2					11.6					
Vac. System, Subcooler and Capillary Device	LH_2	-	-	-	14.3	-	-	-	-		-
	LO_2				18.9						
Vol. Penalty Due to Not Loading Fluid Because of Vol. Displaced by Added Hardware		42.3		42.3	72.1	71.5	31.7	71.7	70.2	71.7	70.2
Residual Payload Penalty**	LH_2	22.7	96.5	22.7	22.7	22.7	69.8	22.7	22.7	22.7	22.7
	LO_2	115	90	115	115	115	127.4	115	115	115	115
ΔChilldown Penalty due to Start Sequence		399.4	381.2	399.4	399.4	399.4	399.4	399.4	399.4	399.4	399.4
Thrust Barrel Revisions to Reduce Refilling Time		11.2		11.2	11.2	11.2	11.2	11.2	11.2	11.2	11.2
Settling System Including Peroxide Payload Penalty		-	108.5	-	-	-	-	-	-	-	-
Pressurization System		64	308	64	64	64	319	64	64	64	64
Total		901.7	984.2	1002.6	1098.7	1212.0	1298.0	1390.1	1489.5	1700.2	1799.6
		(409)	(447)	(455)	(499)	(550)	(589.4)	(631)	(676)	(771)	(817)

** Worst case assumptions (subcooler does not retard pullthrough).

Options:
1. Baseline D-1S
2. Start basket, passive cooling, subcooler dumped overboard.
3. Start basket, passive cooling, pumped subcooler.
4. Start basket, thermal conditioning (TC) and subcooler dumped, active cooling.
4a. Uses vent rate adjusted to suit "g" level, active cooling.
5. Start basket, TC and subcooler pumped back into the tank, active cooling.
6. Start basket, TC dumped, subcooler pumped, active cooling.
6a. Uses vent rate adjusted to suit "g" level, active cooling.
7. Start basket, TC pumped, subcooler dumped, active cooling.
8. Start tank bypass feed.

Table 6-5. Acquisition System Equivalent Payload Weight Comparison lb_m (kg_m)

1 Burn (Planetary Mission Profile)

Weight Penalty Element		Option 3	1	2	5	6a	7	4a	6	8	4
Capillary Device	LH_2	138.3	-	138.3	138.3	138.3	138.3	138.3	138.3	221.5	138.3
	LO_2	38.8		38.8	38.8	38.8	38.8	38.8	38.8	104.7	38.8
Cooling Coils	LH_2	-	-	-	131.6	131.6	131.6	131.6	131.6	-	131.6
	LO_2				18.8	18.8	18.8	18.8	18.8	-	18.8
Assorted Hardware		11	-	11	11	11	11	11	11	5	11
Subcooler	LH_2	18.8	-	18.8	18.8	18.8	18.8	18.8	18.8	-	18.8
	LO_2	23.8		23.8	23.8	23.8	23.8	23.8	23.8		23.8
Dumped Fluid, Subcooler		-	-	23.1	-	-	23.1	23.1	-	-	23.1
				72.5			72.5	72.5			72.5
Dumped Fluid, Capillary Device Cooling	LH_2	-	-	-	-	29.8	-	29.8	70	-	70
	LO_2					17.5		17.5	81		81
Dumped Fluid, Start Tank	LH_2	-	-	-	-	-	-	-	-	-	-
	LO_2										
Vac. System, Subcooler (Pumping Coolant Back to the Tank) Includes Boiloff + Battery + Hardware	LH_2	5.6	-	-	-	5.6	-	-	5.6	-	-
	LO_2	10.6				10.6			10.6		
Vac. System, Capillary Device Cooling	LH_2	-	-	-	-	-	11.1	-	-	-	-
	LO_2						10				
Vac. System, Subcooler and Capillary Device	LH_2	-	-	-	11.7	-	-	-	-	-	-
	LO_2				12.6						
Vol. Penalty Due to Not Loading Fluid Because of Vol. Displaced by Added Hardware		46.5		46.5	79.4	79	78.7	77.3	79	34.9	77.3
Residual Payload Penalty**	LH_2	23	101	23	23	23	23	23	23	69.7	23
	LO_2	115.9	92	115.9	115.9	115.9	115.9	115.9	115.9	126.3	115.9
Δ Childown Penalty due to Start Sequence		186	173	186	186	186	186	186	186	186	186
Thrust Barrel Revisions to Reduce Refilling Time		11.2		11.2	11.2	11.2	11.2	11.2	11.2	11.2	11.2
Settling System Including Peroxide Payload Penalty		-	89.1	-	-	-	-	-	-	-	-
Pressurization System		64	262.2	64	64	64	64	64	64	297	64
Total		693.5 (315)	717.3 (326)	772.9 (351)	884.9 (402)	923.7 (419)	976.6 (443)	1001.4 (455)	1027.4 (466)	1056.3 (480)	1106.1 (502)

** Worst case assumptions (subcooler does not retard pullthrough).

Options:
1. Baseline D-1S
2. Start basket, passive cooling, subcooler dumped overboard.
3. Start basket, passive cooling, pumped subcooler.
4. Start basket, thermal conditioning (TC) and subcooler dumped, active cooling.

4a. Uses vent rate adjusted to suit "g" level, active cooling.

5. Start basket, TC and subcooler pumped back into the tank, active cooling.
6. Start basket, TC dumped, subcooler pumped, active cooling

6a. Uses vent rate adjusted to suit "g" level, active cooling.

7. Start basket, TC pumped, subcooler dumped, active cooling.
8. Start tank bypass feed.

into the screened channels feeding the subcoolers. After vapor ingestion into the channel, the worst case assumption is that this vapor will instantaneously enter the boost pump. For the start baskets the residuals shown are the channel volume, subcooler volume, sump volume and tank pool residuals. For the LO_2 start basket, since baseline residuals are determined by NPSH, the maximum residuals will not exceed the baseline residuals plus the tank residuals at pullthrough into the screened channel. Minimum residual quantities were also computed for the start baskets assuming that the vapor ingested into the channels is retarded in the subcoolers and that draining proceeds below the channels and subcoolers. Actual residuals will be bracketed by the "worst case" and "minimum" residual numbers, probably closer to the "worst case." The minimum residuals for LH_2 correspond to the tank residuals plus the sump volume. For LO_2 the minimum residuals are the tank residuals alone. LO_2 sump residuals will be zero, using the baseline method of residual calculation, since NPSH will be supplied by the subcooler.

The start tank residuals were computed in a manner similar to the start basket residuals. Liquid level in the start tanks at vapor ingestion into the screened tubes was determined for both LO_2 and LH_2 tanks considering the screened tube area and the angle of the screened tube with respect to the tank bottom. Liquid levels were converted into tank residuals. Total start tank residuals were the sum of the start tank pool residuals, screened tube residuals, refill line residuals, sump residuals, main tank outlet line residuals, start tank outlet line residuals and main tank pool residuals.

Chilldown penalties are based upon the data quoted in Table 3-1 for chilldown penalty. Chilldown penalty is higher for the first burn of any mission because of the higher Centaur D-1S propellant feed system temperature experienced in the cargo bay of Shuttle. The chilldown penalty for this burn for the baseline Centaur D-1S will be 179 lb_m (81 kg) of LO_2 and 89 lb_m (40 kg) of LH_2. For orbital heating conditions, the chilldown fluid used for the baseline Centaur D-1S is 165 lb_m (75 kg) of LO_2 and 72 lb_m (33 kg) of LH_2. These chilldown penalties assume a start sequence where the propellant is settled to fill and chilldown the sump and boost pump. Equivalent payload weights were determined for each of the three missions.

For the start baskets and start tanks, engine shutoff valves have to be opened to chilldown and fill the pump and sump. Chilldown fluid for the pump and sump must be assessed as extra weight penalty for the capillary devices. For the cargo bay conditions this will be 7 lb_m (3 kg) of LO_2 and 14 lb_m (6.4 kg) of LH_2. For orbital conditions this will be 2 lb_m (0.9 kg) of LO_2 and 5 lb_m (2.3 kg) of LH_2. These weight penalties were added to the baseline system chilldown weight and converted to payload weight for the three missions.

The boiloff penalty for pumping fluid back into the tank was determined from the pump fluid power requirements for pumping subcooler flow and or capillary device cooling flow back into the tank. Payload penalty was determined by converting the power input to the boiloff rate and applying the appropriate time period (time between burns for capillary device cooling, burn time for subcooler operation) and payload sensitivity factor.

Pressurant usage was based on the data presented in Table 2-6 for the five burn mission. Additional weight not included in Table 2-6 is included for pressurant diffusers for the cooled sump, and cryogenic valves and other equipment for the start tank. Data for the other missions was generated using scaling factors developed for evaluating Shuttle based Centaur pressurization systems as used in Reference 6-3.

Settling system weight penalty is based on requiring three peroxide bottles and 493 lbs (224 kg) of peroxide for the 5 burn mission. Removing the settling function permits one bottle to be removed from the vehicle for each of the three missions. The second bottle is offloaded in varying degrees for the three missions. For example, for the five burn mission 493 lb_m (224 kg) of peroxide is required including twelve pounds of residual per bottle. Each bottle contains 218.5 lb_m (99 kg) of peroxide including 12 lb_m (5.4 kg) of residual. For the 5 burn mission, approximately 214 lb_m (97 kg) of peroxide will be required for settling. The third bottle, containing 56 lb_m (25.4 kg) of fluid [44 lb_m (20 kg) usable] is chargeable to settling plus 170 lb_m (77 kg) in the second bottle.

The start tank venting penalty is based on venting the start tank down from maximum pressure to refilling pressure and maintaining at the refilling pressure until the start tank is full. Calculations assumed venting vapor only while in the settled condition in reducing LO_2 tank pressure from 40 to 26 psia (276 to 179 kN/m^2) and LH_2 tank pressure from 17.6 to 15 psi (121 to 103 kN/m^2). A penalty equivalent to the mass of one start tank volume of vapor was taken for both the LO_2 and LH_2 tank for the refilling period.

Hardware occupying tank volume reduces the amount of fluid that can be carried in the tanks. Small changes in mixture ratio created by changes in the relative quantities of LO_2 and LH_2 loaded can easily be accommodated by the propellant utilization system. Thus excess volume penalty in the LH_2 tank will not be accompanied by a corresponding offloading in the LO_2 tank. The volume of each component in each tank was determined by assessing the volume penalty. The start tank volume penalty included the volume taken up by the solid elements (screen, plates, etc.) of the start basket and subcooler and the total volume of the cooling coils, and pumping components (for the LH_2 tank). No volume penalty was taken for pressurant diffusers and bubblers since all configurations will require some type of pressurant fittings for abort. The start tank volume penalty included the solid volume of the start tank insulation, tankage, screens, valves and lines and the total volume of the helium bottle stored in the LH_2 tank. Volumes for both tanks were multiplied by the liquid density to obtain the total payload weight penalty for volume displaced by each system option.

6.4 RECURRING COST

Recurring costs were determined for each system option using existing Centaur system costs as references. These costs, shown in Table 6-6, include production, installation and inspection. (They do not include research and development, qualification or flight verification costs. These costs are included in the development plan of Section 6.7). Cost estimates indicate that the start baskets not requiring pumping of coolant back into the tank will be lower in cost than the baseline system.

6.5 POWER REQUIREMENTS

Power requirements for each of the options requiring power are shown in Table 6-7. Power for valve actuation and feedback sensor control is neglected. Power requirements are due to pump operation for pumping coolant fluid back into the tank.

6.6 FLIGHT PROFILE FLEXIBILITY

Flight profile flexibility was assessed for the options considered. Start sequence time is shorter for the capillary devices. Also weight comparisons appear to increasingly favor the passively cooled start baskets for missions having a higher number of burns. Conversely, for the baseline system, shorter main engine burns can be achieved compared to the capillary device because of the need to settle and refill the capillary device during main engine firing. (Short engine burn times are likely to be impractical because of excess chilldown propellant lost).

Overall, the capillary devices can provide greater flight profile flexibility because of the shorter start sequence time and their greater applicability to multiburn missions (greater than five burns). Potentially, start sequence time can be reduced to the engine chilldown time with capillary devices and subcoolers if the boost pumps can be removed and the propellant ducts can be maintained full of liquid up to the engine shutoff valves.

6.7 RECOMMENDED DEVELOPMENT PLAN

This section presents technology, hardware, flight qualification and flight test preparation programs for the most promising capillary acquisition systems.

6.7.1 PROMISING CAPILLARY ACQUISITION SYSTEMS. Comparisons of the eight acquisition system options indicated that capillary acquisition systems offer greater potential flight profile flexibility than the baseline settling system. The passively cooled start baskets (Options 2 and 3) compared favorably to the baseline system on the basis of hardware weight and equivalent payload weight for the five burn mission. The weight advantage will increase for missions with greater than five burns.

Table 6-6. Relative Recurring Costs*

Cost Element in Thousands of Dollars		Option 2	Option 1 1 Burn Mission	Option 1 2 Burn Mission	Option 4	Option 1 5 Burn Mission	Option 8	Option 3	Option 5	Option 7	Option 6
Basket	LH_2	40	-	-	40	-	-	40	40	40	40
	LO_2	12	-	-	12	-	-	12	12	12	12
Channels	LH_2	4	-	-	4	-	1.5	4	4	4	4
	LO_2	3	-	-	3	-	1.5	3	3	3	3
Subcooler	LH_2	6	-	-	6	-	-	6	6	6	6
	LO_2	30	-	-	30	-	-	30	30	30	30
Cooling Coils + Sensing	LH_2	-	-	-	20	-	-	-	20	20	20
	LO_2	-	-	-	10	-	-	-	10	10	10
Wicking Provisions	LH_2	8	-	-	-	-	-	8	-	-	-
	LO_2	6	-	-	-	-	-	6	-	-	-
Passthroughs and Bypass Lines	LH_2	4	-	-	4	-	-	4	4	4	4
	LO_2	4	-	-	4	-	-	4	4	4	4
PU Probes	LH_2	5	-	-	5	-	5	5	5	5	5
	LO_2	5	-	-	5	-	5	5	5	5	5
Vacuum Pumping System	LH_2	-	-	-	-	-	-	25	25	25	30
	LO_2	-	-	-	-	-	-	35	35	35	40
Valves for Start Tank	LH_2	-	-	-	-	-	40	-	-	-	-
	LO_2	-	-	-	-	-	6	-	-	-	-
Start Tanks	LH_2	-	-	-	-	-	24	-	-	-	-
	LO_2	-	-	-	-	-	10	-	-	-	-
Pressurization System		25	110	110	25	130	110	25	25	25	25
Settling System		-	51	51	-	51	-	-	-	-	-
Total Cost		152	161	161	168	181	203	212	228	228	238

Options:
1. Baseline D-1S.
2. Start basket, passive cooling, subcooler dumped overboard.
3. Start basket, passive cooling, pumped subcooler.
4. Start basket, thermal conditioning (TC) and subcooler dumped, active cooling.
4a. Uses vent rate adjusted to suit "g" level, active cooling.
5. Start basket, TC and subcooler pumped back into the tank, active cooling.
6. Start basket, TC dumped, subcooler pumped, active cooling.
6a. Uses vent rate adjusted to suit "g" level, active cooling.
7. Start basket, TC pumped, subcooler dumped, active cooling.
8. Start tank bypass feed.

* Costs include production, installation, and inspection.

Table 6-7. Power Requirements

Power requirements in watt-hrs

		OPTION			
		3	5	6	7
5 burn mission	LH_2	26	477	26	450
	LO_2	19	108	19	89
Total		45	585	45	539
2 burn mission	LH_2	20	368	20	349
	LO_2	14	88	14	74
Total		34	456	34	423
1 burn mission	LH_2	18	175	18	157
	LO_2	13	58	13	44
Total		31	233	31	201

All other options do not require electrical power. Power requirements are included in Tables 6-2, 6-3, 6-4 and 6-5 according to one lb_m of battery weight per 48 watt-hours. Power for valve actuation and feedback sensor control is neglected.

Capillary device system reliability is slightly less than the baseline system reliability. Start baskets not using the pumped coolant system are estimated to have lower cost than the baseline settling system. The potential flight profile flexibility, weight and cost advantages of Option 2 (the passively cooled start basket with coolant dumped overboard) make this configuration worthy of additional attention.

Option 3 offers greatest potential payload weight advantage. This option consists of passively cooled start baskets using vacuum pumping systems to return subcooler coolant to the tank. A major advantage of this configuration is its insensitivity to the number of burns required. This is because the subcooler operates over the entire mission burn time regardless of the number of burns. (The only variable affecting subcooler coolant vacuum pump operation is the start sequence time which will increase as the number of burns increases. This will increase pump induced boil-off and battery weight). These advantages are somewhat offset by reduced reliability, increased cost and increased power requirements compared to Option 2.

Both passively cooled start basket configurations (Options 2 and 3) have advantages over the baseline Centaur D-1S hydrogen peroxide settling system which may be of potential benefit for multiburn advanced mission requirements. The potential benefits of these two options warrant additional investigation. The following sections describe the development program recommended for these start basket configurations.

6.7.2 <u>TECHNOLOGY STUDIES.</u> A review was made of the recent literature to gather recommended technology programs for developing capillary acquisition devices. In addition to these reports (References 6-4 to 6-10), recommended work solving technology gaps uncovered in Tasks II & III was included in the development plan. The preliminary recommendations of NAS3-17814 "Low G Fluid Transfer Technology," were scanned to assure that all required technology programs were included. Table 6-8 gives a brief description of each technology program, the source of the recommendation, the application of the program to the passively cooled start baskets for Centaur D-1S, a classification of the program into analysis, design, fabrication or testing and a classification of the importance of funding the program. The importance of each program is classified into one of the following five categories:

1. This is a critical program. Centaur D-1S passively cooled start baskets could not be employed without this program.

2. Substantial design, fabrication or testing improvements are possible if this program is implemented.

3. A program currently in progress may close this technology gap.

4. Some design, fabrication or testing improvement is possible if the program is implemented.

Table 6-8. Capillary Acquisition Device Technology Requirements

Potential Technology Program	Source of Recomm-endation	Category	Application to Advanced Versions of Centaur	Program Requirements
Passive cooling - Use capillary pumping to thermally condition screen surfaces to prevent dryout.	NAS3-17802	1*	Capillary device screen cooling.	Analysis, ground test, low-g prototype testing.
Thermal subcooling - Use a heat exchanger with throttled tank fluid as the coolant to subcool the main engine flow.	NAS3-17802	1	Replacement of the Centaur pressurization (and boost pump) system.	Analysis, design, and ground demonstration cryogenic testing.
Vapor flow across a wetted screen - Determine the capability of capillary barriers to maintain retention while subjected to vapor flow.	NAS3-17802	1	Capillary device screen cooling. Vapor must enter the basket to replace evaporated liquid.	Analysis and ground testing. Drop tower tests for selected configuration and flow rates.
Line chilldown pressure transients - Determine the effect of flowing subcooled fluid into an initially warm propellant duct.	NAS3-17802	2	Start sequence with an initially warm duct.	Analysis, ground test and development of techniques to minimize pressure surges.
Draining and pullthrough with screens and subcoolers. Analyze draining in channels and subcoolers. Use techniques developed in NAS8-21465 for analyzing screen pullthrough suppression capability. Empirically evaluate capillary device draining and adjust the analysis based on the data obtained.	NAS3-17802 NAS8-21465	2	Capillary device draining. Interaction of the subcoolers and screened channels.	Analysis, ground testing and drop tower testing.
Settling and Refilling - Determine settling and collection phenomena with capillary device start sequence thrust. Determine refilling time for capillary devices.	NAS3-17802 NAS8-27685 (Ref. 6-5) NAS8-21465 (Ref. 6-4)	2	Settling predictions under main engine thrust levels. Refilling time predictions for capillary devices.	Analysis and ground testing. Low g testing for verification.
Vibration - Determine the vibrational spectrum to be experienced at the capillary device during actual mission conditions. Determine if any design modifications are required to minimize vibrational problems.	NAS3-17802 NAS8-27685 (Ref. 6-6)	2	Capillary device structural design and mounting.	Instrumentation of a Centaur flight vehicle. Analysis of Shuttle based Centaur flight profile. Vibrational analysis of the capillary devices. Redesign of supports and mounting provisions, if required.
Effect of pressure changes and heat transfer variations on screen wetting - Determine the effect of changes in tank pressure and incident heating on capillary device screen dryout. Include the effect of mixing and disturbances.	NAS3-17802 NAS7-200 (Ref. 6-7) NAS8-21465	2	Capillary device screen cooling. Interaction between the vent system and capillary device.	Analysis, ground testing. Low g testing for verification.
Vacuum pump system for saving subcooler cold side fluid (only required if Option 3 is selected).	NAS3-17802	2	Saving subcooler fluid from being dumped overboard.	Analysis, design, fabrication, ground testing.
Screen repair techniques - Develop repair techniques that are compatible with oxidizers.	NAS3-17802	2	Screen repair without having to disassemble sections of the capillary device.	Manufacturing development and materials research study.
Spacing of screen layers	NAS3-17802	2	May be useful for providing capillary pumping for screen cooling.	Analysis and testing of passive cooling. Manufacturing development.
Screen impact loads due to settling	NAS7-200 (Ref. 6-7)	2	Structural design of the start baskets and the LO_2 thrust barrel are directly affected.	Analysis and ground testing. Low g testing for verification. Conduct this program in conjunction with the settling and refilling study. Analytical, drop tower testing, low g aircraft or orbital tests.

* See text, Section 6.7.2 for explanation of categories.

Table 6-8. Capillary Acquisition Device Technology Requirements (Continued)

Potential Technology Program	Source of Recommendation	Category	Application to Advanced Versions of Centaur	Program Requirements
Film bubble point technique.	NAS9-12182 (Ref. 6-9)	2	Would allow bubble point testing of a complete screen assembly, simplifying final checkout procedures.	Bench tests.
Heating of the capillary device by warm pressurant gas - Determine the retention degradation of screens subjected to warm pressurant.	NAS8-27685 (Ref. 6-6)	3	Cold pressurant or thermal subcooling should be used with cryogenic capillary devices.	Analysis, ground tests.
Tank thermodynamics and thermal control for screen liners integrated with a thermodynamic vent system.	NAS3-15846 (Ref. 6-8)	3	Might provide some ideas for start basket thermal control using passive thermal conditioning.	Analytical.
Thermal expansion - Determine the effect of thermal expansion and contraction on the clearances between the capillary device and tank walls.	NAS3-17802	4	Capillary device/tank wall clearances.	Fabricate a capillary device, install in the Centaur LH_2 tank, load with LH_2 and measure clearances under the entire range of possible pressures and temperatures.
Pressure drop effects in a channel due to mass injection.	NAS3-17802 Some work done in NAS8-27685	4	Capillary device draining.	Analysis, ground testing.
Feed system flow dynamics - Determine the effect of shutdown and startup pressure transients, caused by valves and pumps, on system retention capability.	NAS8-27685	4	Pressure transients due to start up and shutdown transients do not appear to be a problem.	Analysis, bench tests
Dissimilar metal joining.	NAS9-12182 (Ref. 6-9)	4	Joining stainless steel screens to aluminum backup materials and aluminum backup to steel tank walls.	Manufacturing development.
Screen Joining - Joining techniques including brazing, welding and diffusion bonding techniques for aluminum and titanium.	NAS9-12182 (Ref. 6-9)	4	Screen to backup plate joining.	Manufacturing development.
Thermal conductivity - Determine the thermal conductivity of screens and screen/perforated plate assemblies.	NAS3-17802 NAS8-21465 (Ref. 6-4)	5	None. Only required for active cooling schemes.	Bench tests.
Low gravity (filling and) refilling	NAS7-200 (Ref. 6-7)	5	Would only be useful if the Centaur was refueled in flight.	Analysis drop tower testing, low g aircraft or orbital tests.
Screen Forming - Double curvature screen forming technique development.	NAS9-12182 (Ref. 6-9)	5	None	Manufacturing development.
Long term storage - Determine the effect of long term storage on surface tension.	Heller and Cadwallader (Ref. 6-10)	5	None. Long term storage is not required.	Ground testing, materials

5. This program would be potentially useful for capillary devices but does not apply to passively cooled start baskets or the Centaur D-1S type application.

Programs fitting into categories (1) and (2) were selected as recommended technology programs. Cost and schedule estimates were made for each of the programs. Several of the programs were combined to reduced overall costs. The schedules and costs are shown in Table 6-9.

The passive cooling program consists of analysis and bench testing of capillary pumping configurations for maintaining wetted capillary device screens. The configurations evaluated will be screens, screens and plates and open channels.

Thermal subcooling evaluation consists, initially, of analytical evaluation of boost pump replacement. Prototype testing will then be conducted to verify system performance. A LO_2 subcooler will be fabricated.

In conjunction with the subcooler program, a prototype channel and start basket configuration will be fabricated and bubble point tested using a film bubble point procedure. The capillary device and subcooler will then be assembled and installed in the cryogenic test facility. Testing will proceed with subcooler thermal and pressure loss testing. Capillary device outflow testing will then be performed. Thermal testing will be conducted to determine the passive cooling capacity of the start basket capillary pumping configuration as a function of incident heating and pressure fluctuations caused by mixing and venting.

The settling and refilling task will include analysis of settling patterns, interaction of liquid jets and collected liquid on screened barriers and subsequent screen device refilling, and impact loads on tankage, thrust structure and capillary devices during main engine settling. Tests will be conducted primarily with ground experimentation using stretched diaphragms to hold the main tank storable fluid prior to settling over a scale model start basket. Some drop tower tests will be required to verify scaling parameters.

Line childown pressure transients should be analyzed with a nonequilibrium transient thermodynamic model. A typical propellant duct configuration should be designed, fabricated, and cryogenically tested.

Vapor flow across a wetted screen should be analyzed to determine, for particular screens and fluids, the vapor velocity at which the screen retention capability is lost. Bench tests should be run for a variety of screens and fluids.

Vibration test data should be obtained by installing accelerometers on the outside of the LH_2 and LO_2 tanks adjacent to the proposed start basket attachment points. The cost of this program is dependent upon the use of spare telemetry packages that should

Table 6-9. Passively Cooled Start Basket Development Plan

DEVELOPMENT PROGRAMS	(No. Req'd)	ROUGH ORDER OF MAGNITUDE COSTS	
TECHNOLOGY PROGRAMS			
Passive Cooling - Analysis and Bench Testing		50K	
Thermal Subcooling -			
Analysis		29K	
Design, Fabrication of Prototype Subcooler		30K	
Draining and Pullthrough With Screens & Subcoolers - Analysis		15K	
Prototype Channel and Start Basket Fabrication		15K	
Film Bubble Point Checkout		12K	
Functional Testing		4K	
Assembly of Capillary Device, Subcooler		35K	
Facility Buildup		20K	
Subcooler Testing		15K	
Outflow Testing		20K	
Thermal Testing - Incident Heating, Pressure Decay, Mixing and Venting			
Settling Refilling & Impact Loads Analysis and Model Tests		75K	
Line Chilldown Pressure Transient - Analysis, Design, Fab. & Cryo Test With Prototype Duct (*Model of Duct)		90K*	125K
Vapor Flow Across a Wetted Screen - Analysis and Bench Testing		45K	
Vibration Testing at the Container Start Basket Attachment Points - Obtain and Analyze Flight Data		70K	
Vacuum Pumping System - Design, Fab., Assemble and Cryogenically Test		(75K)	Only reqd for pumped coolant options.
Screen Repair Techniques - LO_2 and LH_2		15K	
Screen Spacing - Controlled Wicking Paths		15K	
Total Technology Program Reporting		30K	
Technology Program Requirements		555K + (75K) + 30K Reporting Total Technology Program Costs	
HARDWARE PROGRAMS			
Capillary Device, 2 LH_2, 1 LO_2	(3)	60K	
Subcooler, 2 LH_2, 1 LO_2	(3)	35K	
Vacuum Pumping System, 1 LH_2, 1 LO_2	(2)	(35K)	Only reqd for pumped coolant options.
Bypass Lines, Valves, 1 LH_2, 2 LO_2	(3)	30K	
Capacitance Probes, Pass Thrus, 1 LH_2, 2 LO_2	(3)	15K	
Total Hardware Program Reporting			10K Reporting
Hardware Program Requirements		140K + (35K) + 10K Reporting Total Production Costs	
QUALIFICATION PROGRAMS			
Capillary Device, LO_2 and LH_2		110K	
Subcooler, LO_2 and LH_2		80K	
Vacuum Pumping System, LO_2 Only		(110K)	Only reqd for pumped coolant options.
Bypass Lines and Valves, LO_2 Only		50K	
Passthrough, LO_2 Only		25K	
Capacitance Probe, LO_2 Only		40K	
Total Qualification Program Reporting			10K Reporting
Qualification Program Requirements		305K + (110K) + 10K Reporting Total Qualification Costs	
FLIGHT TEST PROGRAM PREPARATION			
Installation in the Tanks		125K	
Final Assembly, Inspection and Checkout			
Reporting			10K Reporting
Flight Test Program Preparation Rqmts		125K + 10K Reporting Total Installation and Checkout Costs	
Total Program Costs		1.73M	Including 220K for vacuum pumping system, 60K for reporting and 204K for studies that are currently under contract to NASA

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be available from Titan/Centaur backup units. The program cost includes sensor calibration, installation and data reduction. The data will then be used to determine if retention or natural frequency problems exist due to vibration. Design modifications to the start basket supports or structure will be recommended as required.

The vacuum pumping system used to return coolant to the tank will be designed, fabricated and cryogenically tested to determine the operating characteristics of the system.

Screen repair techniques that are compatible with LH_2 and LO_2 will be developed. Methods evaluated will include metal patches, adhesives, brazing and soldering.

Fabrication techniques for closely controlled screen layer spacings will be developed in order to permit controlled wicking paths for passive thermal conditioning.

Total costs for developing a passively cooled start basket with thermal subcoolers will be $555,000 plus $30,000 for reporting, for Option 2. For Option 3, using a vacuum pumping system the cost will be increased by $75,000. Some of these program costs have already been committed. Passive cooling and thermal subcooler evaluations will be performed on NAS3-16963, "Centaur Propellant Thermal Con-Conditioning Study." Line chilldown pressure transients will be studied on a current NASA/LeRC procurement. These procurements will reduce the unfunded portion of the development program to approximately $421,000 for Option 2 and $496,000 for Option 3 (including reporting).

The critical programs in the number one category of Table 6-7 were passive cooling, thermal subcooling and vapor flow across wetted screens. In order to complete these programs approximately $200,000 will be required of which approximately 125,000 has yet to be committed.

6.7.3 HARDWARE PROGRAMS. In addition to the hardware requirements of the technology programs, hardware must be fabricated for flight qualification testing and flight testing. Two separate sets of hardware will be used for this purpose. Units used for flight qualification testing will not be flown.

Capillary devices and subcoolers will be fabricated for both the LH_2 and LO_2 tanks. Two will be required for the LH_2 tank qualification and flight test programs. One will be required for the LO_2 tank qualification program. The LO_2 basket and subcooler used for technology development testing will be used for flight testing.

One LO_2 vacuum pumping system will be required for qualification testing. The LH_2 pumping system will not have to be qualified since it will be similar to the LO_2 system. One LH_2 pumping system consisting principally of a sump and vacuum pump will be fabricated. The LO_2 system fabricated for the technology program testing will also be used for flight testing.

An LO_2 bypass line, capacitance probe, passthrough, and shutoff valve will be subjected to qualification testing. Additional LO_2 and LH_2 units will be fabricated for flight testing.

Total program funding for hardware programs will be $140,000 for Option 2. Option 3 will require an additional $45,000. Reporting for production of flight system and qualification hardware will cost $10,000.

6.7.4 QUALIFICATION PROGRAM. A matrix of qualification testing requirements was prepared for the components to be qualified. This matrix, shown in Table 6-10 was used to determination qualification testing costs and schedules. Components that were similar for the LO_2 and LH_2 applications; pumping systems, bypass lines, passthrough, capacitances probes and shutoff valve will only be qualified in their LO_2 configuration. Table 6-10 shows the cryogenic and noncryogenic vibration, structural, electrical, shock, flow, temperature shock, performance, acceleration, leakage, and life testing to be performed. Estimates were made for each applicable test based on Centaur IUS development cost estimates (Reference 6-11). Included in the costs are writing of specifications for the test, test planning and test coordination. Qualification of the capillary devices, subcoolers and the vacuum pump system will commence after successful development testing of these components.

Qualification costs will be $305,000 for Option 2 plus $10,000 for reporting. Qualification costs for Option 3 will be an additional $110,000.

6.7.5 FLIGHT TEST PROGRAM PREPARATION. After successful qualification of the flight components, the components fabricated for flight test will be assembled on the Centaur as it is being fabricated and assembled. The time schedule shown has the normal test tank build up of approximately two months. During this time the capillary acquisition system components will be assembled and installed in the tanks. After the completion of tank fabrication, the Centaur will be pressure tested and sent to the final assembly area. The inspection and checkout of the capillary acquisition system will be completed in the final assembly area.

The capillary acquisition system flight test configuration will consist of the baseline Centaur settling and pressurization systems plus the recommended passively cooled start baskets and subcoolers. In order to make the flight test on a minimum interference basis, a single burn Centaur D-1T mission with no zero-g coast should be used. This mission should be flown and completed with sufficient propellant remaining to allow a functional test of the acquisition device. After the mission, start basket retention and thermal conditioning capability will be tested during a low-g coast period. Then an engine start sequence will be demonstrated with flow from the start baskets and subcoolers. Settling and refilling time will be measured during the start sequence and main engine thrust settling. Additional thermal conditioning, start sequence, subcooling and refilling demonstrations will be made if sufficient residual quantities are present.

Vehicle costs are not included since a dedicated flight is not required. A normal mission can be performed with the flight test configuration.

Table 6-10. Qualification Test Matrix

Component and Description	Approximate Size	Approx. Weight Lb	(kg)	Pressure*	Vibration Testing	Structural Testing	Electrical Testing	Flow-Pressure Drop Testing	Shock Testing	Temperature Shock Testing	Performance Testing	Acceleration Testing	Leakage Testing	Life Testing	Duration - Operative Life-Requirement	
Capillary Devices - Sheet Metal Structure Covered by Screen	10' (3.04m) D× 3' (0.9m) high	138	(63)	20 psia (138kN/m^2), ΔP = 1 psi (6.9 kN/m^2)	C	X		X				X	C			LH_2
	4' (1.22m) D× 1' (.305m) high	38	(17.3)	30 psia (207 kN/m^2), ΔP = 5 psi (34.5 kN/m^2)	C	X		X				X	C			LO_2
Thermal Subcooler - Plate Fin Heat Exchanger	3.5'(1.07m)D× 13"(.33m)× 3"(7.5cm)	19	(8.6)	20 psia (138 kN/m^2) ΔP = 1.2 psi (8.3 kN/m^2)	C	X		X		C	C	X				LH_2
	22"(0.56m) D× 6" (0.15m) H	24	(10.9)	30 psia (207 kN/m^2), ΔP = 1.4 psi (9.6 kN/m^2)	C	X		X		C	C	X				LO_2
Pumping System - Surge Tank and Vacuum Pump With Lines	2 ft^3 (0.056 m^3)	10	(4.5)	1 psia (6.9 kN/m^2) to 20 psia (138 kN/m^2)	C	X	X	X	X	C	C	X	C	C	5 cycles	LH_2
	2 ft^3 (0.056 m^3)	15	(6.8)	1 psia (6.9 kN/m^2) to 30 psia (207 kN/m^2)	C	X	X	X	X	C	C	X	C	C	5 cycles	LO_2*
Bypass Line - Line Between Sump and Tank Containing a Shutoff Valve	1.5" (3.8m) D × 3' (0.9m) L	5	(2.3)	20 psia (138 kN/m^2)	C	X		X					C			LH_2
	1.5" (3.8m) D × 1.5' (0.46m) L	5	(2.3)	30 psia (207 kN/m^2)	C	X		X					C			LO_2*
Passthroughs - Seals to Prevent Leakage Where Lines Exit the Sump Area		.5	(0.22)	20 psia (138 kN/m^2) to vacuum	C				X	C		X	C			LH_2
		1	(0.45)	20 psia (138 kN/m^2) to vacuum	C				X	C		X	C			LO_2*
Capacitance Probes - to Sense Capillary Device Liquid Level	3' (0.9m) × 1" (2.54m) O.D.	2	(0.9)	20 psia (138 kN/m^2)	C	X	X		X		C	X				LH_2
	1' (0.3m) × 1" (2.54m) O.D.	2	(0.9)	30 psia (207 kN/m^2)	C	X	X		X		C	X				LO_2*

Key: C - Cryogenic
X - Noncryogenic

* Only the LO_2 components for these items will be qualified. LH_2 components will be qualified by similarity.

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7

CONCLUSIONS AND RECOMMENDATIONS

7.1 CONCLUSIONS

The passively cooled start baskets appear to be the most promising capillary acquisition systems. Cooling coils required for active cooling resulted in an excessive weight penalty when designed for worst case condensation heat transfer.

Thermal subcoolers appear to have substantial hardware weight and equivalent payload weight advantage compared to helium pressurization. Thermal subcoolers (heat exchangers using throttled cooling fluid to cool the fluid flowing to the boost pump) were the most promising new subsystem analyzed in this study.

A pumping system for returning coolant to the tanks, while adding complexity, offers the advantage of reduced weight and sharply reduced payload penalty sensitivity to the number of engine burns and total mission time.

7.2 RECOMMENDATIONS

Development of capillary devices, particularly passively cooled start baskets, should be pursued.

Programs receiving primary attention for passively cooled start basket development should be:

1. Use of thermal subcoolers to provide propellant feed system NPSH.
2. Use of capillary pumping (wicking) for passive thermal conditioning to prevent screen dryout.
3. Determination of screen wetting limits when subjected to vapor flow representative of start basket thermal conditioning between burns.

Other recommended development programs, including technology, hardware, flight qualification, and flight test have been identified in Tables 6-7 and 6-8.

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APPENDIX A

ACQUISITION SYSTEM OPTIONS

This section expands upon the components and operations required to operate each of the systems listed in Section 6.

<u>Option 1.</u> The baseline Centaur D-1S system, including the warm helium pressurization system, settling motors and the hydrogen peroxide used for settling.

The pressurization system consists of three large ambiently stored bottles of helium. The main tank pressurization system loop is redundant with pairs of solenoid valves, orifices and check valves provided for feeding both the LO_2 and LH_2 tanks. A bubbler is provided in the LO_2 tank for injecting helium through the liquid. The LH_2 tank contains a helium energy dissipator. Three hydrogen peroxide bottles containing silastic rubber bladders are used for attitude control, settling and boost pump operation (turbine driven). Four hydrogen peroxide settling rockets are used to settle propellants prior to engine restart.

<u>Option 2.</u> Passively cooled start baskets using capillary pumping (wicking) for capillary device thermal conditioning and subcoolers (heat exchangers) for providing boost pump NPSH.

Subcooler coolant flow is dumped overboard. Shutoff valves are required for the subcooler cold side flow. The start baskets consist of wicking barriers to promote capillary pumping and fine mesh screened channels to permit all liquid flow to the subcooler. Both cold and hot side flow is delivered to the subcooler by the channels. The cold side flow is throttled through an orifice between the channel outlet and subcooler inlet. All capillary devices (both start tanks and start baskets) have the four settling rockets removed and require only two hydrogen peroxide bottles for attitude control and boost pump operation (for the five-burn mission). Bypass feedlines with shutoff valves are required for all start basket concepts in order to vent the sump area back into the tank preventing pressures in the sump and subcooler area from forcing vapor into the screened channels. Liquid vapor sensors will be used in all capillary devices to sense liquid level. Heat exchangers using throttled tank fluid coolant, provide boost pump NPSH for all start baskets options (2 to 7). One small ambient helium pressure bottle is required for engine actuation and attitude control system pressurization. Fittings for main tank pressurization are required for a possible abort.

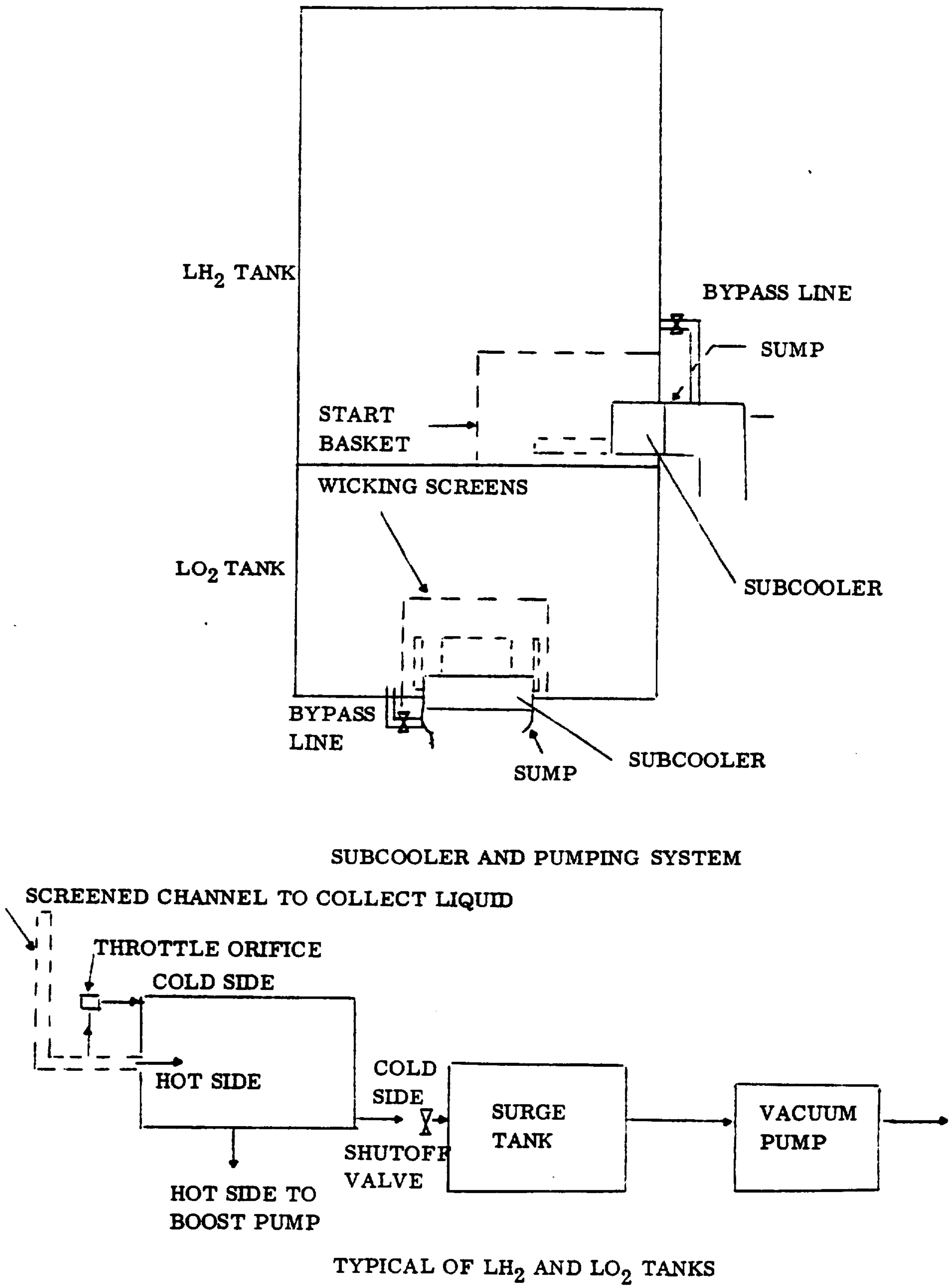

Figure A-1. Schematic of Passively Cooled Start Basket With Pumped Subcooler Flow (Option 3)

Option 3. Passively cooled start baskets using capillary pumping for capillary device thermal conditioning and subcoolers for providing boost pump NPSH. Subcooler coolant flow is pumped back into the tank using a surge tank and vacuum pump.

Components are similar to Option 2 with the addition of a pumping system for returning subcooler flow to the tank. Downstream of the cold side shutoff valve is a surge tank which is evacuated to less than 5 psi (34.5 kN/m^2) with a vacuum pump that pumps (or compresses) the vaporized coolant back into the tank. A typical passively cooled start basket for Option 3 is shown in Figure A-1.

Option 4. Actively cooled start baskets using cooling coils for capillary device thermal conditioning and subcoolers for providing boost pump NPSH. Cooling coil flow and subcooler coolant flow are dumped overboard.

This option uses cooling coils fed from the screened channels within the screened start basket. Throttling valves are used to obtain the required coolant temperature difference. Shutoff valves are required for the cooling loop for each tank. Feedback sensors are required to control the coolant flow rate as a function of the outlet temperature. All other attributes are similar to Option 2. Option 4 is the design described in the start basket drawings of Sections 5.1 and 5.3. A schematic of this system is shown in Figure A-2.

Option 5. Actively cooled start baskets using cooling coils for capillary device thermal conditioning and subcoolers for providing boost pump NPSH. Cooling coil flow and subcooler coolant flow is pumped back into the tank using a surge tank and vacuum pump.

Option 5 is similar to Option 4 plus a surge tank and pumping system for pumping both the LO_2 and LH_2 capillary device and subcooler coolant flow back into the tank. One surge tank and vacuum pump handles the capillary device and subcooler flow for each tank. Separate shutoff valving is required for each of the four coolant flow lines entering the surge tanks.

Option 6. Actively cooled start baskets using cooling coils for capillary device thermal conditioning and subcoolers for providing boost pump NPSH. Cooling coil flow will be dumped overboard and subcooler coolant flow will be pumped back into the tank using a surge tank and vacuum pump.

This option has lower flow and power requirements than Option 5. System components are similar to Option 5, with the lines from the cooling coil shutoff valves to the surge tanks deleted.

Option 7. Actively cooled start baskets using cooling coils for capillary device thermal conditioning and subcoolers to provide boost pump NPSH. Subcooler

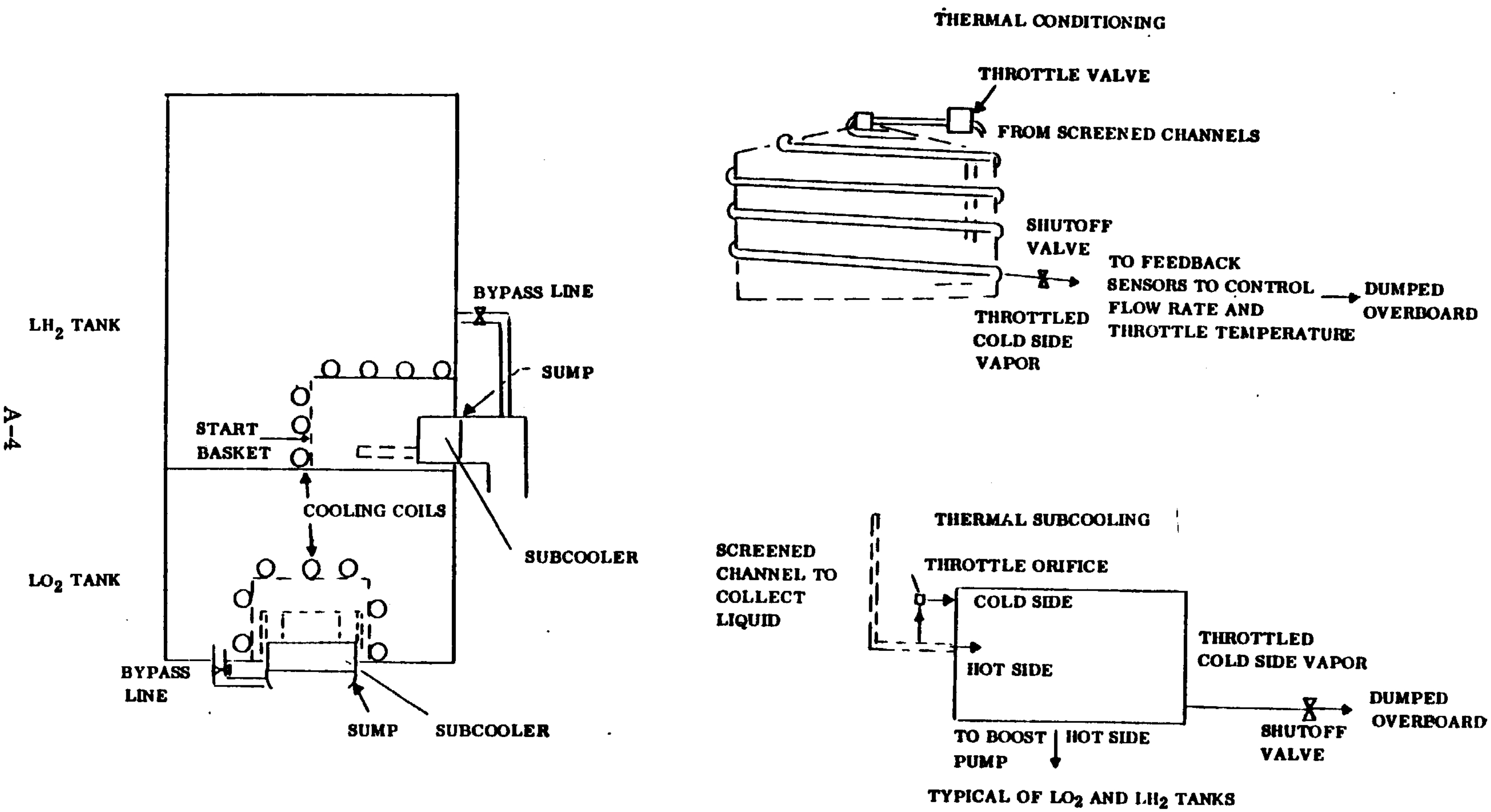

Figure A-2. Schematic of Actively Cooled Start Basket With Dumped Capillary Device and Subcooler Flow (Option 4) coolant flow will be dumped overboard and cooling coil flow will be pumped back into the tank using a surge tank and vacuum pump.

The system components for this device are similar to Option 5, with the lines from the subcooler shutoff valves to the surge tanks deleted. This option has slightly lower coolant pumping flow and power requirements compared to Option 5.

Option 8. Bypass feed start tanks with cold helium pressurization.

This option is shown schematically in Figure A-3. The system operates by pressurizing the start tank through valve 4. The engine shutoff valves are opened to vent the boost pumps and sumps to vacuum. The start sequence is then initiated from the start tank flow by opening valve 1A and the main engines are chilled down and fired. The propellant in the main tanks is settled and valve 1 is switched from position 1A to position 1B to shut off start tank flow and initiate flow from the main tank. While outflowing from the main tank the settled start tank is vented to below the main tank pressure through valve 2 and the refill valve (valve 3) is opened. The start tanks are vented during refilling to minimize refilling time. The refill valves are closed when the start tanks reach their desired level. Liquid vapor sensors (capacitance probes) are used for this purpose to minimize the vented propellant. Between burns, no start tank venting is required. The pressurization system for the start tank consists of one helium bottle stored in the LH_2 tank. No main tank pressurization is required, since all main tank outflow occurs with main engine thrust providing the necessary boost pump NPSH. Fittings would be required for main tank pressurization during a possible abort. Abort helium is charged to the Shuttle payload bay.

The system comparisons were made considering all subsystems and processes that were affected by capillary device deployment. For example, the pressurization system for the start tanks use cold helium pressurization (stored at LH_2 temperature); the baseline Centaur D-1S system uses warm helium pressurization stored at ambient conditions [400R (222K)]; and the start baskets use subcoolers (heat exchangers that remove heat from the fluid entering the sump) to provide boost pump NPSH. Additional capacitance probe elements will be used to sense liquid in the start baskets and to control refill and vent valve operation with the start tanks in addition to the baseline propellant utilization system. Bypass lines will be required for the start basket options in order to vent the sump region back into the tank, preventing pressure buildup between burns from forcing vapor into the screened channels feeding the subcooler. The LO_2 start basket will also use the bypass line to provide flow during a possible abort dump (see Section 3.9). For the LO_2 start tank and start basket, thrust barrel refilling will be enhanced by increasing the open area on both the sides and top of the thrust barrel. This change, and the increased settling loads due to main engine thrust settling, will require the thrust barrel stiffeners and forward ring to be structurally beefed up. The thermodynamic vent systems used on the baseline Centaur D-1S will also be required for the start baskets and start tank and thus are not included in the comparison. All subsystems that are identical for the baseline Centaur D-1S and the start basket and start tank configurations are similarly excluded from the comparison.

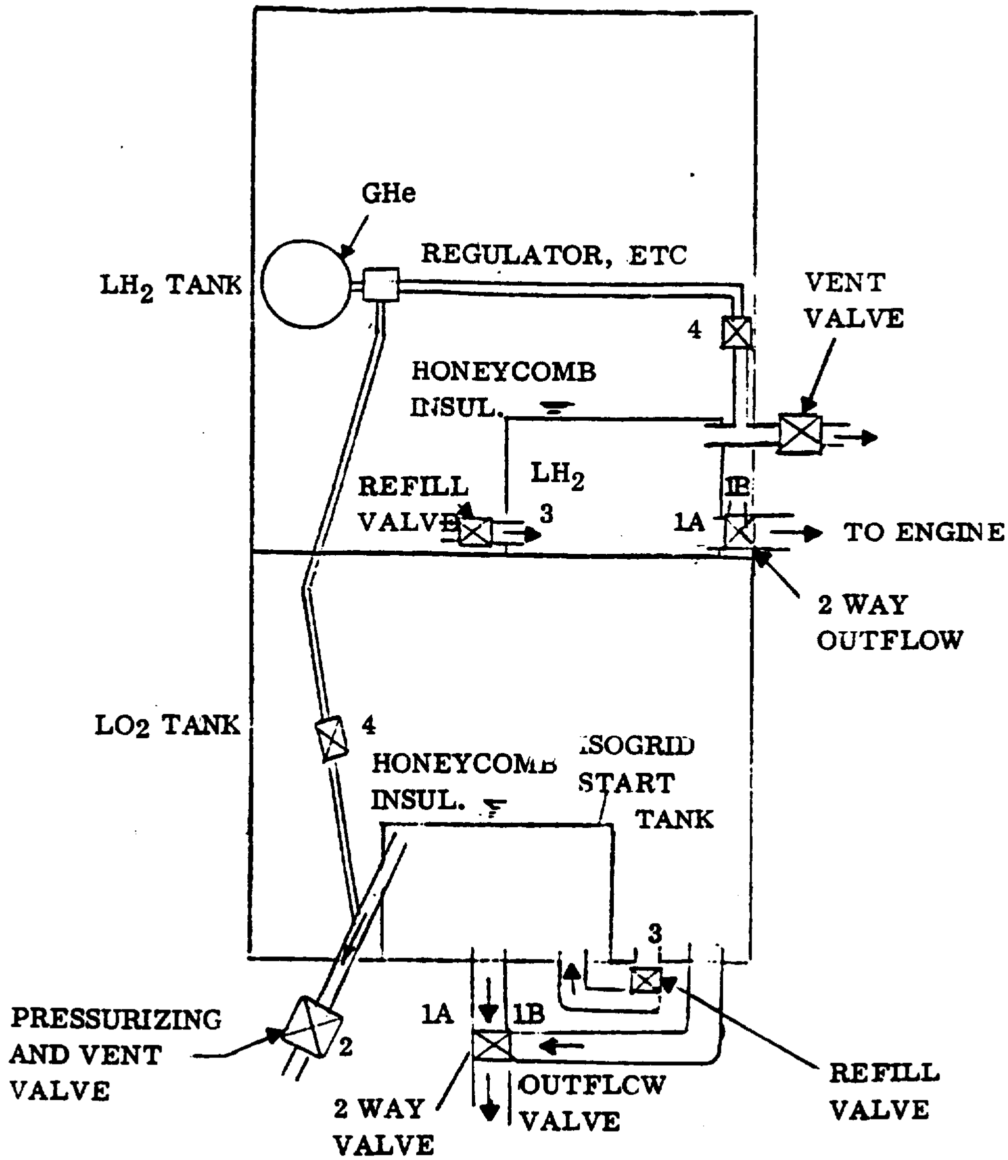

Figure A-3. Schematic of Bypass Feed Start Tank (Option 8)