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# FILLING OF ORBITAL FLUID MANAGEMENT SYSTEMS

**GENERAL DYNAMICS**  
*Convair Division*



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CASD-NAS-78-010

## FILLING OF ORBITAL FLUID MANAGEMENT SYSTEMS

July 1978

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16. Abstract  <p>A study was performed with three objectives: 1) analyze fluid management system fill under orbital conditions, 2) determine what experimentation is needed, and 3) develop an experimental program. The fluid management system was a 1.06m (41.7 in) diameter pressure vessel with screen channel device. Analyses were conducted using liquid hydrogen and N<sub>2</sub>O<sub>4</sub>. The influence of helium and autogenous pressurization systems was considered.</p> <p>Analyses showed that fluid management system fill will be more difficult with a cryogen than with an earth storable. The key to a successful fill with cryogen is in devising techniques for filling without venting liquid, and removing trapped vapor from the screen device at tank fill completion. This will be accomplished with prechill, fill, and vapor condensation processes. Refill will require a vent and purge process, to dilute the residual helium, prior to introducing liquid. Neither prechill, chill, nor purge processes will be required for earth storables.</p> <p>The prechill, fill and vapor condensation processes were selected for further evaluation and modelling. Ground test plans were prepared.</p>					
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## FOREWORD

The following final report summarizes the technical effort conducted under Contract NAS3-21021 by the General Dynamics Convair Division from September 2, 1977 to May 4, 1978. The contract was administered by the National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio.

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All data are presented with the International System of Units as the primary system and English Units as the secondary system. The English system was used for the basic calculations.

## TABLE OF CONTENTS

	<u>Page</u>
LIST OF FIGURES	ix
LIST OF TABLES	xiii
SUMMARY	xv
1 INTRODUCTION	1-1
1.1 FLUID MANAGEMENT SYSTEM CONFIGURATION	1-1
1.1.1 System Thermal Mass	1-1
1.2 DESIGN REQUIREMENTS FOR CRYOGENIC SYSTEM	1-2
1.2.1 Liquid Supply Requirements	1-3
1.2.2 Fluid Storage Conditions	1-3
1.2.3 Maximum Pressure Allowables	1-3
1.3 DESIGN REQUIREMENTS FOR EARTH STORABLE SYSTEM	1-3
1.3.1 Liquid Supply Requirements	1-3
1.3.2 Fluid Storage Conditions	1-4
1.3.3 Maximum Pressure Allowables	1-4
1.4 ANALYSIS APPROACH	1-4
2 FLUID MANAGEMENT SYSTEM CONCEPT SELECTION	2-1
2.1 SELECTION CRITERIA	2-1
2.2 CONCEPT RANKINGS	2-1
2.2.1 Feasibility	2-1
2.2.2 Adaptability of Existing Configurations	2-8
2.2.3 Operational Simplicity	2-8
2.2.4 Versatility	2-8
2.2.5 Gravity Insensitivity	2-9
2.2.6 Ground Testability	2-9
2.2.7 State of Development Development Costs	2-9
2.2.8 Recurring Costs	2-10
2.2.9 Hardware Weight	2-10
2.2.10 Fluid Weight Penalty	2-10
2.2.11 Reliability	2-11
2.3 CONCEPT SELECTION	2-11
3 FILLING ANALYSIS FOR SELECTED CONCEPT (CRYOGENS)	3-1
3.1 PRECHILL ANALYSIS	3-3
3.1.1 Prechill With Hydrogen Vapor	3-3
3.1.2 Prechill With Liquid Hydrogen	3-9
3.2 TANK CHILL ANALYSIS	3-10

TABLE OF CONTENTS, CONTINUED

	<u>Page</u>
3.2.1 Tank Chill Thermodynamics	3-10
3.3 TANK FILL ANALYSIS	3-15
3.3.1 Thermal Equilibrium Tank Fill	3-15
3.3.2 Fill Model for Liquid Spray Dominance	3-16
3.3.3 Fill Model for Vapor Bubble Dominance	3-17
3.4 VAPOR REMOVAL FROM SCREEN DEVICE	3-30
3.4.1 Bubble Condensation Model	3-31
3.4.2 Summary	3-33
3.5 TANK REFILL	3-33
3.5.1 Tank Refill (Autogenous)	3-33
3.5.2 Tank Refill (Helium Pressurant)	3-35
3.5.3 Refill of an Evacuated Tank	3-38
3.5.4 Vapor Removal From Screen Device Following Refill	3-39
3.5.5 Summary	3-39
4 FILLING ANALYSIS FOR SELECTED CONCEPT (EARTH STORABLES)	4-1
4.1 THERMAL EQUILIBRIUM TANK FILL	4-2
4.2 REFILL ANALYSIS	4-4
4.2.1 Selected Helium Vent Procedure	4-4
4.3 Vapor Removal From Screen Channel Device	4-7
4.4 SUMMARY	4-7
5 MODELLING OF PROPELLANT FILL PROCESSES	5-1
5.1 PROCESS SCREENING FOR MODELLING	5-2
5.1.1 Processes Subject to Rigorous Analysis	5-2
5.1.2 Processes Requiring Experimentation	5-3
5.1.3 Processes Not Adequately Defined	5-3
5.2 MODELLING OF SELECTED PROCESSES	5-3
5.2.1 Tank Prechill	5-3
5.2.2 Tank Fill	5-5
5.2.3 Vapor Condensation	5-9
6 TEST PLAN - FILLING OF ORBITAL FLUID MANAGEMENT SYSTEMS	6-1
6.1 GROUND BASED TESTING	6-2
6.1.1 Inflow Testing	6-2
6.1.2 Vapor Collapse Tests	6-8
6.2 SPACED BASED TESTING	6-12
7 REFERENCES	7-1
APPENDIX A HYPRS COMPUTER PROGRAM	A-1
APPENDIX B PRE-CHILL OF CONTAINER WITH COLD GASEOUS HYDROGEN (GH <sub>2</sub> )	B-1
APPENDIX C DISTRIBUTION LIST	C-1

## LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
1-1	Screen Channel Configuration	1-1
1-2	Pressure Vessel Girth Joint and Support Methods	1-2
3-1	Modified Fluid Management System With Screen Channel Configuration	3-1
3-2	Comparison of One-g and Low-g Storage Tank Fill Employing One-g Fill Techniques	3-2
3-3	Peak Pressure During Management System Chillover Versus Mass Addition and Entering Liquid Vapor Pressure	3-5
3-4	Maximum Pressure During Management System Chillover Versus Initial Tank Temperature and Entering Liquid Hydrogen Vapor Pressure	3-6
3-5	Hydrogen Vapor Mass Required to Prechill the Fluid Management System Tank	3-8
3-6	Time to Prechill the Fluid Management System With Hydrogen Vapor	3-8
3-7	Fluid Management System Tank Wall Temperature History During a Gaseous Hydrogen Prechill	3-9
3-8	Water Spray Particle Size Versus Pressure for Fulljet Nozzles	3-12
3-9	Liquid Hydrogen Droplet Diameter Versus Flowrate for Two Spray Nozzle Configurations	3-13
3-10	Maximum Pressure During Management System Tank Chill as a Function of Liquid Spray Parameters	3-14
3-11	Fluid Management System Final Tank Pressures for a Temperature Equilibrium Chill and Fill Process	3-17
3-12	Fluid Management System Pressure and Temperature Histories During Initial Ten Percent Period (Spray Flowrate = 0.91 kg/sec)	3-13
3-13	Fluid Management System Pressure and Temperature Histories During Initial Ten Percent Period (Spray Flowrate = 0.227 kg/sec)	3-18
3-14	Apparatus for Study of Vapor Dispersal in Liquid	3-19
3-15	Sauter Mean Diameter of Hydrogen Vapor Dispersed in Liquid Hydrogen Versus Input Fluid Power	3-22

LIST OF FIGURES, CONTINUED

<u>Figures</u>		<u>Page</u>
3-16	Mechanism for Entraining Vapor in Liquid During Tank Fill	3-22
3-17	Vapor Entrainment Rate Into Liquid Bulk Versus Liquid Hydrogen Momentum Inflow and Percent Liquid Fill of Fluid Management System Tank	3-24
3-18	Fractional Volumetric Hold-Up of Vapor in Liquid Hydrogen as a Function of Liquid Momentum Inflow and Bubble Stay-Time for the Fluid Management System Tank	3-26
3-19	Fractional Volumetric Hold-up of Vapor in Liquid Hydrogen Function of Liquid Momentum Inflow and Bubble Stay-Time Fluid Management System Tank	3-28
3-20	Heat Rate From Entrainment Vapor to Liquid Hydrogen	3-29
3-21	Entrained Vapor Total Heat to Liquid Hydrogen for Fluid Management System Tank	3-29
3-22	Vapor Energy Removal Required to Achieve Thermal Equilibrium With Liquid Hydrogen in Fluid Management System Tank	3-30
3-23	Heat Transfer Controlled Bubble Collapse	3-32
3-24	Collapse Time for Spherical Bubbles in Liquid Hydrogen	3-32
3-25	Entering Liquid Hydrogen Vapor Pressure Required to Maintain a Constant Liquid Vapor Pressure in Tank During Fill	3-35
3-26	Final Fluid Management System Pressure for Propellant Refill Without Prior Helium Vent	3-36
3-27	Hydrogen Vent Mass Required During System Blowdown to Provide Indicated Helium Partial Pressure Following Propellant Refill	3-36
3-28	Management System Pressure at End of Vent Required to Provide Indicated Helium Partial Pressure Following Propellant Refill	3-37
3-29	Fluid Management System Helium Residual Following $\text{GH}_2$ Purge	3-39
4-1	Maximum Management System Pressure During Fill With $\text{N}_2\text{O}_4$	4-2
4-2	Final Storage Tank Pressures for $\text{N}_2\text{O}_4$ Thermodynamic Equilibrium Fill Process	4-3
4-3	Management System Pressure at End of Vent Required to Provide Indicated Helium Pressure Following $\text{N}_2\text{O}_4$ Refill	4-5
4-4	$\text{N}_2\text{O}_4$ Vapor Vent Mass Required During System Blowdown to Provide Indicated Helium Partial Pressure During Propellant Refill	4-6



## LIST OF FIGURES, CONTINUED

<u>Figures</u>		<u>Page</u>
4-5	Collapse Time for Spherical Bubbles in Liquid $N_2O_4$	4-7
5-1	Predicted Forced Convection and Free Convection Heat Transfer Coefficients During Normal Gravity Test Tank Prechill Test	5-6
5-2	Expected Trend of Tank Pressure History During Fill Process	5-9
5-3	Percent of Energy Removed as a Function of Nondimensional Time	5-10
5-4	Vapor Collapse Process Within Screened Device	5-11
6-1	Scale Model Centaur OOS Test Article	6-3
6-2	Tank Prechill Test Set-up Schematic	6-4
6-3	Instrumentation	6-6
6-4	Tank Fill Test Set-up	6-9
6-5	Typical Vapor Collapse Test System	6-11
6-6	Refill Systems Test on Spacelab	6-13

LIST OF TABLES

<u>Table</u>		<u>Page</u>
1-1	System Weight Summary	1-2
2-1	Candidate Concepts for Capillary Device Low-Gravity Refilling	2-2
2-2	Definition of Comparison Criteria	2-6
6-1	GH <sub>2</sub> and GN <sub>2</sub> Prechill Test Matrix	6-5
6-2	Instrumentation Location	6-7
6-3	LH <sub>2</sub> Fill Test Matrix	6-10
6-4	Freon MF Fill Test Matrix	6-10
6-5	Proposed Test Matrix for Vapor Collapse Tests	6-10

## SUMMARY

The study was concerned with three main areas: 1) analyzing the process of filling a fluid management system under orbital conditions, 2) determining what portion of the filling process requires experimentation, and is subject to modelling, 3) developing an experimental program plan for evaluating management system concept hardware and operational procedures. The fluid management system groundruled for this study, and identified in Reference 1-1, is a pressure vessel, 1.06m (41.7 inches) in diameter with a screen channel device for providing 100 percent liquid flow. Analyses were conducted for a cryogenic and earth storable management system, where liquid hydrogen and  $N_2O_4$  were selected as representative propellants.

A propellant fill procedure was designed to meet two objectives. First, system pressures must be maintained within acceptable limits without expelling liquid propellants, and second, the effective operation of the screen channel device must not be impaired by vapor trapped within it during the fill process.

The case of filling with a cryogenic propellant presents the greatest difficulty in controlling tank pressure without venting liquid. An empty storage tank will reside at a substantially higher initial temperature than the cryogen prior to the initial fill, and the transfer of energy from the tank to the propellant during the fill process will result in venting to avoid exceeding the system structural limits. Because the liquid-vapor distribution is not well defined in a low-g condition, venting to relieve over-pressure may result in the expulsion of liquid.

Vapor will be trapped in the screen channel device in two ways. First, vapor will enter the device during the fill process. Second, if helium pressurant is present in the tank, it must be expelled prior to refill, and screen dry out with subsequent vapor penetration will occur during the helium vent.

The filling analysis showed that these problems can be circumvented by introducing the processes of tank prechill, fill and tank pressurization to collapse trapped vapor.

The prechill process is required to reduce tank temperature to an acceptably low level prior to initiating the tank fill process. Prechill will consist of a series of charge and vent cycles, where either liquid or vapor is introduced during the charge cycle. Vapor only will be expelled during the vent cycle because the elevated tank temperature will preclude the possibility that liquid will be present at vent initiation. A model, based on forced convection heat transfer, was developed to determine the number of charge and vent cycles required to achieve prechill.

At the completion of prechill, the tank is locked up and liquid introduced through one or more spray nozzles to accomplish tank fill. A fill condition of 90 percent or greater will be achieved without the need for venting if near-thermal equilibrium conditions are present. It was determined that sufficient bulk fluid agitation will be created by the entering liquid to provide near-thermal equilibrium during fill. Together, tank prechill and bulk fluid agitation guarantee a no-vent fill.

Liquid hydrogen system refill is similar to initial fill, once pre-conditioning is complete. For a system with helium pressurization, pre-conditioning requires that much of the helium be expelled before refill can be initiated. The selected procedure requires that all usable liquid be transferred to the propellant supply tank, prior to initiating vent and purge cycles designed to dilute tank helium mass to an acceptably low level.

Refill will be greatly simplified if helium pressurization is replaced with autogenous pressurization. Pre-conditioning is not required in this case. It is necessary only to introduce liquid at a sufficiently high flow rate and velocity to maintain thermal equilibrium during refill.

At the completion of initial fill or subsequent refills, the tank will be pressurized in order to condense the hydrogen vapor trapped within the screen channel device. It is estimated that all the trapped vapor will be condensed within approximately five minutes, assuring effective operation of the device.

Fill and refill procedures for an earth storable fluid management system will be less complicated than for a liquid hydrogen system. A prechill process will not be required because system and propellant temperatures will be approximately the same. Tank pressures during fill will remain well below the system vent pressure level. There will be no need to employ a purge procedure prior to initiating refill because  $N_2O_4$  will remain within the screen device during the helium expulsion process. Finally, condensation of  $N_2O_4$  vapor trapped within the screen channel device will take an order of magnitude less time to accomplish than the condensation of hydrogen vapor.

The processes selected for further evaluation and, therefore, modelling were prechill, fill, and vapor condensation. All other aspects of low-g fluid management system fill and refill were judged to be sufficiently well defined. Prechill and fill are similar in one important aspect; it is intended that heat and mass transfer be dominated by forced convection in order that these processes remain independent of acceleration environment. Consequently, a modelling analysis was performed to identify methods of verifying, or modifying, the empirical forced convection equations. This analysis showed that forced convection dominance would be maintained during a one-g prechill test and that the results would be directly applicable to a low-g environment. A determination was also made that rigorous modelling techniques cannot be employed to establish equivalence between normal gravity and low-g fill. It was shown, however,

that normal gravity tests will prove useful for evaluating and verifying low-g fill procedures because normal gravity is a more severe tank fill environment.

It is proposed that inflow tests be conducted in a scale model orbit-to-orbit shuttle (OOS) tank. The test tank will be insulated with multilayer insulation and mounted within a vacuum chamber in order to minimize external heat leaks. Freehill tests will be conducted with  $\text{GN}_2$  and  $\text{GH}_2$ . Fill tests will be conducted using liquid hydrogen and Freon MF, which has physical properties similar to those of  $\text{N}_2\text{O}_4$ .

For the vapor condensation process, the indicated times to achieve vapor collapse are short. The analytical model, which assumes that heat conduction will control the rate of condensation, requires verification. The model suggests that ground tests will be valid only if free convection effects are less significant than conduction. A screen device will not be included as part of the fill test apparatus since free convection effects cannot be eliminated. Instead, it is proposed that series of tests be conducted with liquid hydrogen and Freon MF using an apparatus that will provide for heat conduction dominance during vapor condensation. These tests will verify a vapor collapse analytical model for earth storable and cryogenic propellants.

# 1

## INTRODUCTION

The objectives of this study were; to analyze the process of filling a fluid management system under orbital conditions; to determine what portion of the filling process requires experimentation, and is subject to modelling; and to develop an experimental program plan for evaluating management system concept hardware and operational procedures.

### 1.1 FLUID MANAGEMENT SYSTEM CONFIGURATION

Basic to this study is the fluid management system identified in Reference 1-1. This system is a liquid hydrogen pressure vessel, 1.06 m (41.7 in) in diameter with a screen channel device (Figure 1-1) and helium pressurization system for fluid transfer. It contains a high performance thermal control system comprised of a vapor cooled shield thermodynamic vent system, multilayer insulation, and vacuum jacket.

1.1.1 SYSTEM THERMAL MASS. The total management system mass of 64.4 kg (142.0 lb<sub>m</sub>) is summarized in Table 1-1. However, an effective thermal mass of 13.6 kg (30 lb<sub>m</sub>) is achieved with the support method of Figure 1-2 which isolates the pressure vessel, screen device, and internal hardware from the remainder of the

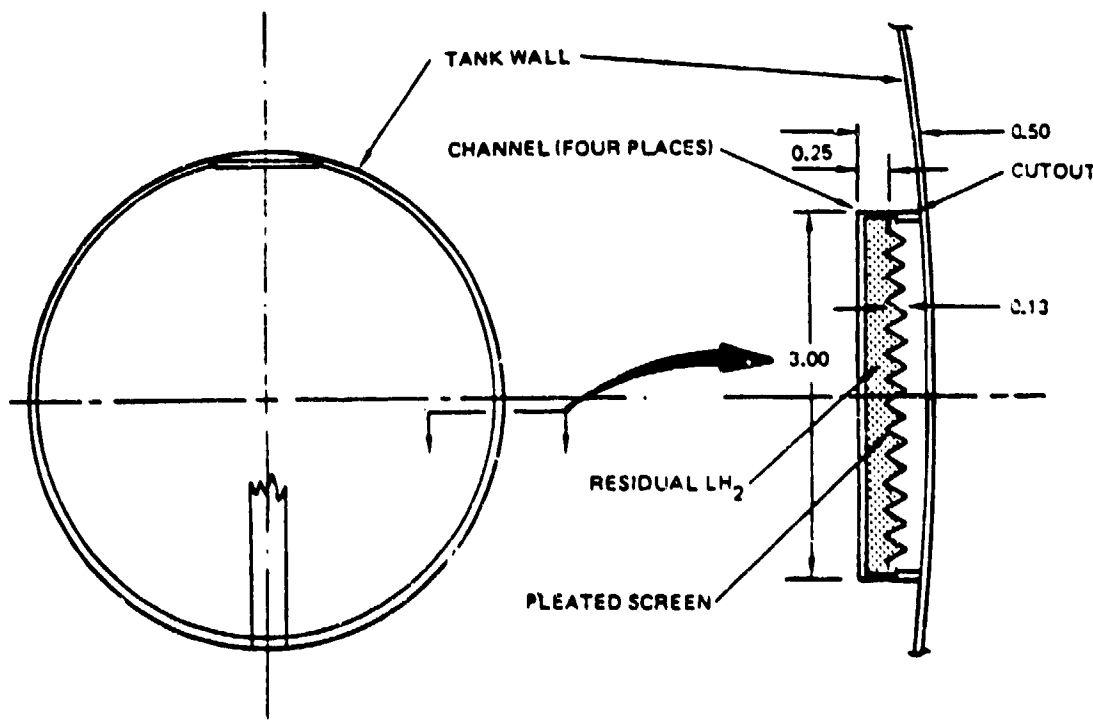


Figure 1-1. Screen Channel Configuration

1-1

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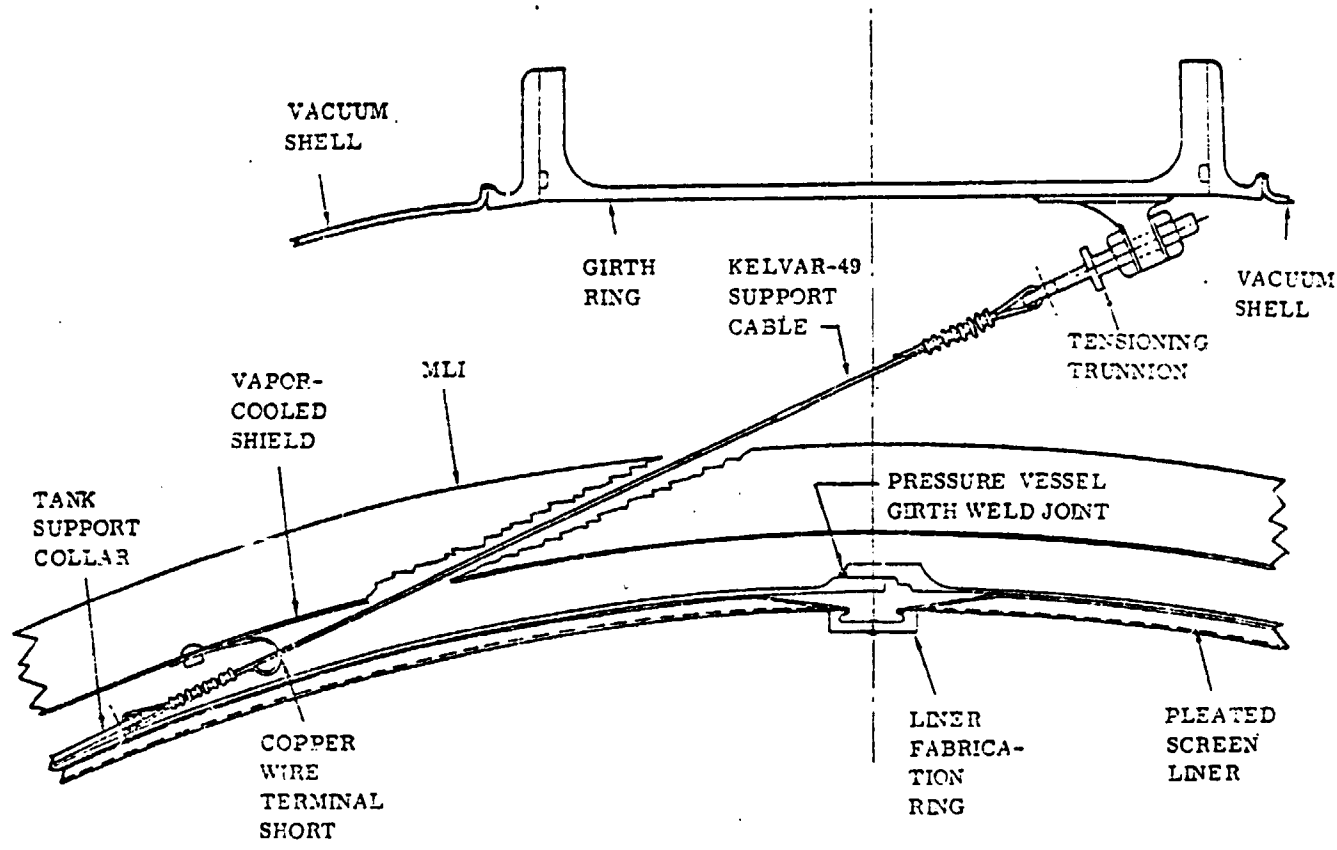


Figure 1-2. Pressure Vessel Girth Joint and Support Method

system. The thermal mass is defined as that portion of the total which is chilled to liquid temperatures during the propellant fill process. For this study, the effective mass was assumed to be 13.6 kg (30 lb<sub>m</sub>).

### 1.2 DESIGN REQUIREMENTS FOR CRYOGENIC SYSTEM

The fluid management system is designed for long duration storage of cryogenics (7 to 30 days), during which time fluid is supplied to other systems. Depending upon the application, flow demand can be variable, as with flow to reaction control system thrusters, or well-defined, as might occur for continuous flow to a fuel cell. Either liquid hydrogen or liquid oxygen will be stored in the management system.

Per agreement with the NASA/LeRC, liquid hydrogen was selected as the propellant for

Table 1-1. System Weight Summary

	kg	lb <sub>m</sub>
Pressure Vessel	8.6	19.0*
Vapor-Cooled Shield	4.4	9.6
Multilayer Insulation	4.9	10.9
Screen Device	2.5	5.5*
Collar Supports	1.0	2.3
Internal Hardware/ Plumbing	2.5	5.5*
Vacuum Shell	19.8	43.6
Girth Ring	<u>20.7</u>	<u>45.6</u>
Totals	64.4	142.0

\* Mass affected during tank prechill or chill.

analysis of the cryogenic management system. This selection was made because a preliminary evaluation showed that the problems associated with management system filling are more severe for liquid hydrogen. Furthermore, the Reference 1-1 design was based upon liquid hydrogen as the stored fluid. A system designed for liquid hydrogen will also be applicable to other fluids.

1.2.1 LIQUID SUPPLY REQUIREMENTS. Pure liquid must be available at the management system outlet at all times. This requirement is translated to mean that the screen channel device must be free of vapor once the system is filled and available for use. The system configuration given in Reference 1-1 was designed without consideration being given to orbital propellant fill capability. Consequently, the first task undertaken in this study was a selection process to identify those modifications needed to fill the channel device with liquid in orbit.

1.2.2 FLUID STORAGE CONDITIONS. Operating conditions require that liquid hydrogen be stored at a vapor pressure of  $344.5 \text{ kN/m}^2$  (50 psia) and that it be subcooled by  $68.9 \text{ kN/m}^2$  (10 psia) with helium partial pressure. There are applications when liquid subcooling is not required. For these cases autogeneous pressurant will replace helium. Liquid storage conditions will remain unchanged.

1.2.3 MAXIMUM PRESSURE ALLOWABLES. Pressure transient will be experienced by the pressure vessel propellant tank during the fill procedure. Per agreement with the NASA/LeRC, maximum allowable pressures cannot exceed  $689 \text{ kN/m}^2$  (100 psia) at any time during propellant tank fill.

### 1.3 DESIGN REQUIREMENTS FOR EARTH STORABLE SYSTEM

The fluid management system for earth storables will contain either nitrogen tetroxide ( $\text{N}_2\text{O}_4$ ) or monomethyl-hydrazine (MMH). This system will differ from that for cryogenics in three major aspects; temperature environment, pressurant condition, and material compatibility. System temperatures will not vary substantially from propellant temperatures at any time prior to or during a fill operation. This is in contrast to the large temperature excursion experienced during cryogen system chilldown.

System pressures are typically maintained above 300 psia with helium pressurant, which will maintain propellants in a subcooled state prior to and following tank fill. There are design problems peculiar to the storage of  $\text{N}_2\text{O}_4$  which favor titanium rather than aluminum as a tank material. But these problems, in general, are related to material compatibility and, therefore, are beyond the scope of this contract. Fortunately, the thermal mass of the cryogenic system aluminum tank is similar to that of a titanium tank that would be designed for earth storables. Because thermal mass is an important parameter associated with management system fill, any analysis conducted on the aluminum tank will also be applicable to a titanium tank. Consequently, this study employed the cryogenic fluid management system for the earth storable fill analyses.



All analyses are conducted for nitrogen tetroxide ( $N_2O_4$ ) because its higher vapor pressure will make propellant fill more difficult than if monomethylhydrazine (MMH) is employed.

1.3.1 LIQUID SUPPLY REQUIREMENTS. Pure liquid must be available at the management outlet at all times. As with the cryogenic system, this means that the screen channel device must be free of vapor upon completion of a propellant fill process.

1.3.2 FLUID STORAGE CONDITIONS. The following fluid operating conditions were selected for this study phase:

$N_2O_4$  vapor pressure =  $130.9 \text{ kN/m}^2$  (19 psia)  
Helium partial pressure =  $1936 \text{ kN/m}^2$  (281 psia)  
Total tank pressure =  $2067 \text{ kN/m}^2$  (300 psia)

1.3.3 MAXIMUM PRESSURE ALLOWABLES. The maximum allowable tank pressure for this study was selected to be  $2412 \text{ kN/m}^2$  (350 psia), which is representative of earth storable systems.

#### 1.4 ANALYSIS APPROACH

The direction of this study was influenced by attitudes held on manned activity in an orbital environment relative to a vehicle system or space station re-supply. These attitudes were translated into guidelines to perform filling operations in a timely manner and to establish a fill procedure that is relatively independent of acceleration environment.

Regarding the first guideline, that of expediting operations, it is reasonable to assume that a fill procedure should be completed within minutes rather than hours because system fill is one of many operations to be conducted. However, a reduction in time is generally gained at the expense of increased fluid losses, which is undesirable due to the high cost of transporting propellants into orbit. Since there was no direct method of determining the tradeoff between time and fluid expended, individual operations were restricted to approximately 10 minutes, unless the mass expended was considered to be excessive.

Regarding the second guideline, a fill procedure will become more and more independent of acceleration if sufficient fluid agitation is provided internal to the tank. That is, the heat and mass transfer process to occur during propellant fill will be dominated by forced convection processes, rather than acceleration, once fluid agitation has increased to the level of an inertia-dominated environment. The advantages of an inertia dominated environment are clear. First, propellant tank fill will be accomplished in a short time period. Second, heat and mass transfer is better defined under orbital conditions if internal tank thermodynamics is dominated by the forced convection process. Finally, a forced convection environment increases the likelihood that normal gravity tests can be scaled to low gravity.

# 2

## FLUID MANAGEMENT SYSTEM CONCEPT SELECTION

A comprehensive screening of orbital fluid management system concepts was conducted. The basic application used was the system described in Reference 2-1, employing helium pressurization and screened channels. Systems considered included the use of pressurant to condense vapor, valving arrangements, pumping, capillary pumping, using thermodynamic venting, vacuum refilling, inflow baffling, high pressure manifolding, shaping of channels and propellant depot refilling stations. These concepts were developed into the most reasonable, or most likely candidates for orbital refilling. Table 2-1 describes these candidates, their operation, advantages and disadvantages, and other comments. The descriptions and the comparisons made in the following paragraphs are applicable to a wide range of receiver applications ranging from small Spacelab spherical tanks to large vehicle tankage.

### 2.1 SELECTION CRITERIA

Each of the concepts shown in Table 2-1 was compared on the basis of the eleven criteria shown in Table 2-2. The comparisons are tabulated in Table 2-3. The evaluation was performed using hydrogen as the baseline fluid, however the results are generally applicable to other cryogenic fluids with both condensible and non-condensable pressurant. Results are also applicable to storable propellants with the exception of the concept using thermodynamic vent fluid or vapor cooled shield fluid (Concept 7) to remove vapor from the capillary device.

### 2.2 CONCEPT RANKINGS

Concepts were scored from 0 to 10 on each criterion, with 10 being the highest and 0 the lowest. A zero score denotes an unacceptable performance measure for that criterion. A score of ten on a criterion indicates that the concept will be totally acceptable with no uncertainty or potential technical problems. A zero score on any single criterion was sufficient cause to eliminate that concept. The total scores reflect a gross ranking of the systems considered. The following paragraphs refer to the rationale behind the rankings in Table 2-3. The concepts are discussed in the order they are ranked for that particular criteria.

**2.2.1 FEASIBILITY.** The systems (Concepts 2, 3, 4 and 8) using helium to condense any trapped vapor are technically feasible as evidenced by Reference 2-2 IRAD testing where a LN<sub>2</sub> start basket was filled using helium pressurant to condense LN<sub>2</sub> vapor. Using throttled fluid (Concept 12) is also a low risk approach since it too has been

Table 2-1. Candidate Concepts for Capillary Device Low-Gravity Refilling

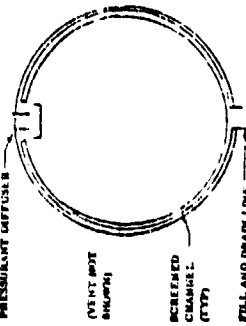
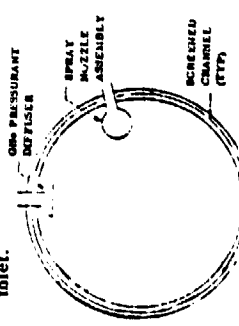
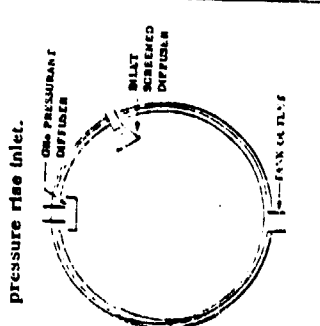
CANDIDATE CONCEPTS	METHOD OF OPERATION	ADVANTAGES	DISADVANTAGES	COMMENTS/ANALYSIS REQUIRED
<p>1. Standard one-g filling.</p>  <p>Labels: PRESSURANT DIFFUSER, SCREENED CHAMBER (STP), FILL AND DRAIN LINE.</p>	<p>Fill using normal ground filling inlet.</p>	<p>No additional hardware.</p>	<p>Liquid vapor interface unknown during filling. Vapor trapped in device.</p>	<p>Unacceptable because orbital propellant management system will not be filled.</p>
<p>2. Helium pressurant/liquid spray inlet.</p>  <p>Labels: GAS PRESSURANT DIFFUSER, SPRAY NOZZLE ASSEMBLY, SCREENED CHAMBER (STP).</p>	<p>Fill tank with liquid using spray nozzle to achieve thermodynamic equilibrium during chilldown and fill. Use helium to condense any vapor trapped in the capillary device during filling. Helium could be added just before transfer to avoid any possibility of forming vapor in the capillary device after pressurization.</p>	<p>Fairly simple hardware. Does not depend upon position of the low-g interface and therefore can be tested in normal gravity with assurance that it will operate in low gravity. Thermal equilibrium results in low pressure rise during fill. No capillary device hardware modification required. Could be retrofitted to existing systems.</p>	<p>High pressure rise will occur during chilldown which may require venting to occur. Use of helium makes filling a partially full tank more complicated because the helium will probably have to be vented off to keep final pressures within limits. Blowdown and purge required to replace GHe with GHe before fill of a tank containing only vapor.</p>	<p>Determine time required for collapsing vapor. Determine if the pressure rise will require venting for typical tanks during chilldown. (Helium venting is not peculiar to this system but to any system that uses helium pressurization). Would be most efficient of Concepts 2, 3 and 4 if venting is required during chill-down or filling.</p>
<p>3. Helium pressurant/low pressure rise inlet.</p>  <p>Labels: GAS PRESSURANT DIFFUSER, TANK OUTLET, MILIT SCREENED DIFFUSER.</p>	<p>Fill tank with diffuser to achieve low pressure rise during chilldown. Use helium to condense any vapor trapped in the screen device.</p>	<p>Fairly simple hardware. Low pressure rise and therefore best chance of not venting during chilldown. No capillary device hardware modification required. Could be "retrofitted" to existing systems.</p>	<p>Nonequilibrium conditions during fill, therefore high pressure rise. Possibility of vapor formation after helium addition if mixing and therefore pressure decay occurs. Some disadvantages as Concept 2 with respect to GHe removal.</p>	<p>How much extra helium <math>\Delta P</math> would be required to assure no vapor formation during mixing? Pressure history during chilldown and fill would have to be analyzed. Same comments as Concept 2 with respect to GHe removal.</p>

Table 2-1. Candidate Concepts for Capillary Device Low-Gravity Refilling (Continued)

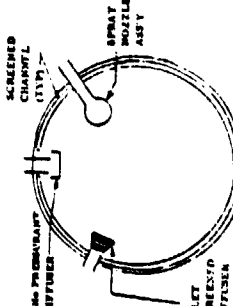
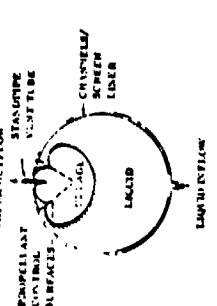
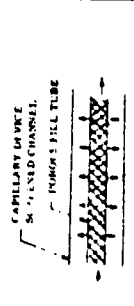
CANDIDATE CONCEPTS	METHOD OF OPERATION	ADVANTAGES	DISADVANTAGES	COMMENTS/ANALYSIS REQUIRED
<p>4. Helium pressurant/low pressure rise and high chill-down efficiency inlets.</p> 	<p>Combine Concepts 2 and 3; chill using a diffuser, mix with a mixer or spray nozzle during fill and use helium to condense any trapped vapor. The diffuser gives minimum pressure rise during chill and the nozzle gives minimum pressure rise during fill.</p>	<p>Configuration most likely to produce a no vent transfer. No capillary device hardware modifications required. Could be retrofitted to existing systems.</p>	<p>Chilldown configuration will be more gravity dependent than the spray configuration. More hardware would be required than for Concepts 2 or 3. Control system would be required to monitor temperatures to determine when to switch inlets. If calculations are wrong then we could wind up venting a low enthalpy fluid during chilldown. Same disadvantages as Concept 2 with respect to GHe removal.</p>	<p>Diffuser would have to be designed for all gravity conditions. Fluid and thermal analysis of diffuser operation would be required during chilldown and fill. Vapor collapse time would need to be analyzed. Same comment as Concept 2 with respect to GHe removal. Would only be advantageous to Concept 2 if a no vent transfer would be accomplished.</p>
<p>5. Propellant control surfaces/vent tube.</p> 	<p>Control surfaces are positioned to control liquid/vapor interface shapes and a tapered vent tube is used to vent vapor only from the tank. (No provisions are specified for filling the capillary device).</p>			<p>Method is not strictly applicable to capillary device refilling. Method is a passive venting method.</p>
<p>6. Piping inside channels to force vapor out of the channels (with high pressure source)</p> 	<p>Use a high pressure liquid source with a baffled or spray inlet tube to force liquid into the channel and force vapor and liquid out. The method requires continuous sheets of liquid around the flow tube. Perhaps a fine mesh variable micron rating screen can be used to control liquid flow into the capillary device.</p>	<p>Positive liquid interface control, does not require helium to condense vapor trapped, operationally simple. Operation should only be slightly g dependent. Could be used to drive pressurant out of the capillary device. (Tank venting and purging would not be required to remove helium from the capillary device).</p>	<p>Complex fabrication. Structural enhancement required to allow channels to withstand impingement loads, relatively heavy. Slower tank fill will probably result unless we raise supply tank filling pressure. Residuals will be higher because of volume of filling system.</p>	<p>Screen backup areas will probably increase, increasing the capillary device area. Initial work would center on feasibility of variable porosity material for flow and preliminary weight assessments. Flow analysis of the filling system required. Pressures in the system should be calculated and weights determined.</p>

Table 2-1. Candidate Concepts for Capillary Device Low-Gravity Refilling (Continued)

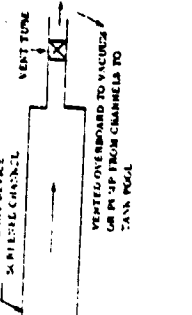
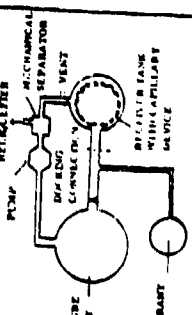
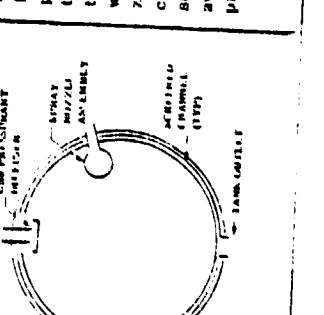
CANDIDATE CONCEPTS	METHOD OF OPERATION	ADVANTAGES	DISADVANTAGES	COMMENTS/ANALYSIS REQUIRED
<p>7. Thermodynamic vent or vapor cooled shield inlet to remove fluid from the channels.</p> 	<p>Pump liquid out of the capillary device. Use liquid (or vapor) being pumped to void capillary device of vapor. Once channels are completely filled the pumping is stopped. Channels are determined to be filled when a given quantity of fluid is pumped (or a given time elapses). Vent through the same path that normal outflow could take in order to remove any vapor that would normally flow out of the device.</p>	<p>Does not require use of helium, filling of a partially full tank is thus no problem (no more than normal). Could be easily integrated into existing thermal conditioning system. Could be used to remove helium as well as hydrogen vapor from the capillary device. (Tank venting and purging would not be required to remove helium from the capillary device).</p>	<p>Gravity dependent. Complex fluid dynamic design. Uses moving parts. May require control and valving to flow both high flow rate and filling and normal steady state boiloff flow. Excess fluid may be vented overboard. Filling time is constrained by pumping time.</p>	<p>A pocket of vapor could remain trapped in a corner of the system. Methods could be outlined to analyze drag of vapor bubbles and flow of liquid. Venting through the outlet area at a higher flow rate than would normally occur should assure that any vapor trapped during "venting" will remain trapped during outflow. "G" dependence of this statement should be studied.</p>
<p>8. Propellant depot approach using high pressure liquid and helium sources, mechanical separators, refrigeration systems, pumping and vent for filling and emptying primary tankage.</p> 	<p>Use elements of filling system #2, 3 or 4 but add systems for recovering, separating and reliquifying boiloff from the tank being filled. The system would have mechanical separators, refrigerators, pumps and auxiliary tanks for filling and emptying primary tankage. Hardware of this type would likely be normally present on a propellant depot.</p>	<p>Most efficient use of fluid. All liquid can be saved and possibly all vapor using mechanical separators to separate <math>H_2</math> from <math>GH_2/GHe</math> and using a reliquefier to liquify hydrogen and separate <math>GHe</math>. Filling can probably be accomplished very quickly.</p>	<p>Uses a lot of power and mechanical equipment. Expensive because of the machinery required. This solution is really not in the same time frame as the other concepts; it is more in the future.</p>	<p>This concept is worth considering and keeping in mind for future applications but the complexity and cost probably does not warrant selection at this time.</p>
<p>9. Autogenous pressurization to condense vapor.</p> 	<p>Use warm (autogenous) pressurant to condense any vapor trapped during filling. Use either inlets 2, 3 or 4 for filling the tank. Control tank pressure decay after pressurization to encourage surface evaporation at the screen and suppress bulk boiling within the screen device. Pressurization could be accomplished using cold hydrogen pressurant. Pressurization just before transfer will avoid mixing and pressure decay problems.</p>	<p>Does not require any helium. Filling of a partially full tank is simplified because receiver tank pressurant does not have to be vented to keep final pressures within limits. Hardware is independent of capillary device fabrication.</p>	<p>Thermal conditioning and propellant tank thermodynamics must be considered to prevent bulk boiling in the screen device during pressure decay when warm pressurant mixes with cold liquid. Would probably therefore require more analysis than concepts using helium. (This assumes mixing analysis is more complicated than vent down analysis). May be more "g" dependent than using helium pressurant because of mixing, etc.</p>	<p>Consider methods for controlling the pressure decay during mixing after pressurization, to prevent bulk boiling. Determine pressure decay for these methods and heat load to the capillary device the real conditioning surface.</p>

Table 2-1. Candidate Concepts for Capillary Device Low-Gravity Refilling (Continued)

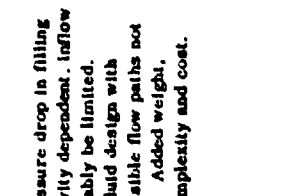
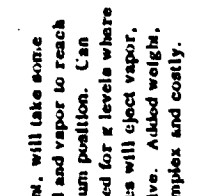
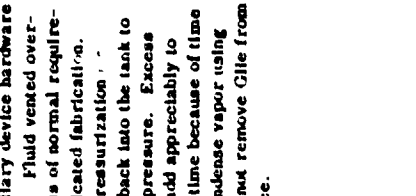
CANDIDATE CONCEPTS	METHOD OF OPERATION	ADVANTAGES	DISADVANTAGES	COMMENTS/ANALYSIS REQUIRED
<p>10. Internal vases to direct flow.</p> 	<p>Internal vases to direct flow into the screen device so that the device will fill without trapping vapor. Slow down liquid in the capillary device to prevent wetting during filling. (Direct liquid away from screen surfaces). Probably a series of perforated ballheads would keep liquid from squirting out ahead of the main liquid front to wet screens prematurely.</p>	<p>Operationally simple if flow rate modulation is not required. Pressurization not required to condense trapped vapor if system works. Helium initially in a dry tank and capillary device will be directly removed from the capillary device during filling.</p>	<p>Increased pressure drop in filling system. Gravity dependent. Inflow rate will probably be limited. Complicated fluid design with potentially feasible flow paths not yet identified. Added weight, fabrication complexity and cost.</p>	<p>Preliminary design study to identify potentially feasible configurations and/or operational procedures that would be required, followed by complicated flow analysis.</p>
<p>11. Shaping capillary devices to position and eject vapor.</p> 	<p>Shape capillary devices using tapered sections and other shapes to position vapor in area(s) of the capillary device so that it may be vented.</p>	<p>Operationally simple. Positive interface control. No pressurization required. Will remove trapped helium.</p>	<p>Time dependent. Will take some time for liquid and vapor to reach their equilibrium position. Can only be designed for g levels where capillary forces will eject vapor. Gravity sensitive. Added weight, fabrication complex and costly.</p>	<p>Study shape of device versus driving force versus anticipated g level. Determine positioning time versus g level.</p>
<p>12. Open loop refrigeration to condense vapor.</p> 	<p>Use throttled vent fluid to cool the capillary device and condense any liquid trapped in the screened volume. (Vent fluid overboard). Attach cooling tubes to the capillary device.</p>	<p>Does not use helium or other pressurant for filling. Could be used as a positive method of condensing vapor by overpowering the system with coolant flow. Relatively insensitive to gravity.</p>	<p>Requires capillary device hardware modifications. Fluid vented overboard in excess of normal requirement. Complicated fabrication. May require pressurization to pump fluid back into the tank to maintain tank pressure. Excess weight. May add appreciably to overall filling time because of time required to condense vapor using cooling. Will not remove Gille from capillary device.</p>	<p>Heat transfer analysis required to determine condensation rate. Throttling pressure, cooling flow rate, tube spacing, geometry. Thermodynamic analysis required to determine tank pressure history and if pressurization or pumping is required.</p>

Table 2-2. Definition of Comparison Criteria

Criteria	Definition
Technical Probability of Successful Operation	How likely is the concept to accomplish the desired filling of capillary devices under all applicable conditions?
Adaptability to Existing Configurations	How much modification is required to incorporate the concept in an existing capillary device design or configuration?
Operational Simplicity	How much actuation of valving, motors, pumps, etc. is required? (How much different would the filling process be, operationally, than normal gravity filling of an empty tank).
Versatility	How applicable is the concept to the full range of possible capillary device configurations, including channels, liners, start baskets and combinations of these devices?
Gravity Sensitivity	How sensitive is the process to low gravity?
Ground Testability	If the concept is gravity sensitive, can modelling techniques be used to successfully design and run ground tests to verify low gravity performance?
State of Development/Development Cost	How much technology work is required to prove the operation of the concept? Will major advances be required? What is the cost of this development work?
Cost Effectiveness/Recurring	What is the cost of building additional units, beyond the initial unit?
Weight/Hardware	What is the total weight of all components and device modifications required to implement the system?
Weight/Fluid	What is the weight penalty due to fluid vented or residual fluid that can be attributed to the concept?

Table 2-3. Ranking of Candidate Concepts

Key: Highest - 10 Lowest - 0	Concept and Number (From Table 1)											
	Liquid Spray Inlet/Helium Pressurant	Low Pressure Rise Inlet/Helium Pressurant	Low Pressure Rise and High Chilled Efficiency Inlets/ Helium Pressurant	Autogenous Pressurization to Condense Vapor	Piping Inside Channels to Force Vapor Out of the Channels	Propellant Depot Approach	Thermodynamic Vent or Vapor Cooled Shield Inlet to Remove Fluid From Channels	Open Loop Refrigeration to Condense Vapor	Internal Vanes to Direct Flow	Shaping Capillary Devices to Position and Eject Vapor	Standard One G Filling	Propellant Control Surfaces/ Vent Tube
Criteria	2	3	4	9	6	8	7	12	10	11	1	5
Technical Probability of Successful Operation	10	10	10	7	7	10	7	8	7	7	0	0
Adaptability to Existing Configurations	10	10	10	10	2	10	7	4	4	2		
Operational Simplicity	8	8	6	6	10	1	8	6	9	6		
Versatility	10	10	10	10	10	10	7	7	5	4		
Gravity Insensitivity	9	7	7	6	10	7	3	6	4	1		
Ground Testability	10	7	7	6	10	7	3	6	4	1		
State of Development	9	8	7	8	6	3	5	6	5	6		
Cost Effectiveness/ Recurring	10	9	9	8	5	9	8	6	7	6		
Weight/Hardware	9	8	7	9	2	1	7	4	5	5		
Weight/Vented Fluid	10	9	10	10	4	10	7	3	3	4		
Reliability/Safety	10	10	9	9	10	7	9	8	10	10		
Total	105	86	92	86	76	75	71	64	83	52	N/A	N/A

demonstrated on the ground. The additional uncertainty with using open loop refrigeration centers on the need to maintain the liquid in a subcooled condition. Forcing liquid into the capillary device through porous tubing (Concept 6) would be straight forward, however the difficulties of fabricating a tube of variable porosity caused this concept to be downgraded. For the thermodynamic vent/vapor cooled shield (Concept 7) the main difficulty would appear to be the possibility of trapping vapor in the channels where drag forces would not be able to extract the vapor. The uncertainty with the autogenous pressurization concept (Concept 9) would be in controlling mixing, pressure reduction and resultant boiling in the capillary device. Use of internal vanes to control flow (Concept 10) has not been demonstrated and would require flow analysis to determine what types and geometries of barrier would provide uniform flow and delay premature wetting. Residuals could also be substantially increased with this concept. The use of shaped channels (Concept 11) to eject vapor would impact the overall geometry and volumetric efficiency and would, additionally, be sensitive to the gravitational acceleration and disturbing accelerations that could adversely reposition liquid. Two of the concepts described in Table 2-1 were disqualified



because of their low probability of success. The normal gravity filling method (Concept 1) will not be acceptable because it would only work under chance circumstances. The passive vent tube (Concept 5) will not help to fill the capillary devices, thus it was eliminated.

2.2.2 ADAPTABILITY OF EXISTING CONFIGURATIONS. The devices using pressurant to collapse vapor (Concepts 2, 3, 4, 8 and 9) would require, at the most, minimum modifications to existing configurations since only spray nozzles and/or diffusers are added to the existing pressurization and fluid acquisition systems. (Only screen liner type configurations would require any modification at all, because access through them would have to be provided). Adding a thermodynamic vent inlet (Concept 7) is a minor revision that will require plumbing into a channel device but would be major for a screen liner in probably requiring the addition of internal vanes to direct liquid flow in order to entrain trapped vapor. Piping inside the channels (Concept 6) and shaped channels (Concept 11) are major reworks. Internal vanes to direct the flow (Concept 10) and open loop refrigeration (Concept 12) would require extensive design modifications to existing configurations.

2.2.3 OPERATIONAL SIMPLICITY. Operational simplicity of the piping device (Concept 6) is the highest since a direct flow path can be used with no flow modulation. Internal vanes (Concept 10) may also fall into this range, however it is possible that the flow will have to be reduced to achieve the desired filling pattern. The spray nozzle (Concept 2) and the diffusers (Concept 3) were downgraded because they require pressurization (with the associated plumbing). The thermodynamic vent (Concept 7) vapor cooled shield inlet will require operation of additional valving compared to Concept 6. The configuration with a diffuser and spray nozzle (Concept 4) is complicated by the fact that the inflow must be switched from the diffuser to the spray nozzle when the tank is chilled down; in order to provide a minimum tank pressure profile. The autogenous pressurization scheme (Concept 9) is complicated by potential operational requirements for controlled tank mixing after pressurization. Shaping the channels (Concept 11) will probably require control of the disturbances or attitude control accelerations between main outflow periods in order to prevent adverse liquid/vapor position prior to transfer. The open loop refrigeration system (Concept 12) is complicated by the use of valving that must be sequenced and the need for controlling tank pressure when substantial quantities of cooling fluid must be used. The propellant depot approach (Concept 8) uses significantly more equipment than the other concepts.

2.2.4 VERSATILITY. The versatility of the concepts using pressurization to condense vapor (Concepts 2, 3, 4, 8 and 9) are highest (for systems normally using pressurization) since they can be applied to any type of system including start basket systems. Similarly it appears that forcing liquid into the capillary device using piping inside the device (Concept 6) will be applicable to all conceivable configurations. Thermodynamic vent system flow (Concept 7) may not be useful in filling compact configurations such as start baskets. Open loop refrigeration systems (Concept 12) will not work for liner devices without systematically reducing tank pressure. Internal vane and shaped channel

concepts (Concepts 10 and 11) have significant influence on the external and internal envelope of the capillary device and therefore may not be directly applicable to some liner or start basket configurations.

2.2.5 GRAVITY INSENSITIVITY. The inflow with spray nozzles (Concepts 2, 4, 8 and 9) or porous tubing (Concept 6) is insensitive to gravity since flow will be inertia dominated. The heat transfer between liquid, vapor and tank wall during spraying and the condensation of vapor by pressurization will be slightly gravity dependent. The flow pattern from a diffuser (Concepts 3, 4 and 9) will also change as a function of gravity. Autogenous pressurization (Concept 9) may be additionally sensitive because of the importance of the mixing process. Low gravity heat transfer and tank thermodynamics are significant factors in the functioning of the open loop refrigeration system (Concept 12). The flow patterns in the internal vane system (Concept 10) used to direct flow will be strongly gravity dependent. Flow patterns, in the concept (Concept 7) using venting to "drag" any vapor out of the device will depend upon the balance between drag, gravity and surface forces. The shaped channels (Concept 11) depend directly on low gravity to operate and will be extremely sensitive to changes in gravity.

2.2.6 GROUND TESTABILITY. Ground testability will generally be a direct function of gravity sensitivity. Exceptions occur when a process will operate better in low gravity than in normal gravity but can still be demonstrated to operate in normal gravity. The concept (Concept 2) using spray nozzles is anticipated to be in this category.

2.2.7 STATE OF DEVELOPMENT DEVELOPMENT COSTS. The concept using spray nozzles (Concept 2) will require a minimum amount of analysis and testing. Diffusers and pressurization (Concepts 2, 3, 4, 8 and 9) add analysis requirements because of the gravity sensitivity of the diffuser flow and the nonequilibrium pressure rise calculations required. Hardware design and fabrication requirements will be minimal for the concepts using helium pressurization (Concepts 2, 3, 4 and 8). For the other concepts, hardware development requirements are more substantial. Development of the porous tube concept (Concept 6) will be a difficult fabrication task. No work has been done on this concept. Similarly incorporating shaped channels into a viable design (Concept 11) that gives low residuals will be a difficult design and fabricating task. Little work beyond a conceptual nature has been done on this type of design for other than spacecraft applications. For the open loop refrigeration system (Concept 12) development requirements will include evaluation of heat transfer between the cooling fluid, contained liquid and the tank contents and the resulting temperature and pressure history. The fabrication requirements of connecting the cooling tubes to the capillary device would necessitate a development effort to determine satisfactory attachment methods. The autogenous pressurization development requirements (Concept 9) are greater than that for helium pressurization because of the need for consideration of mixing and mixers. Thermodynamic vent system and vapor cooled shield development (Concept 7) would require extensive testing to assure that vapor is not trapped. No testing of this concept has been done. Internal vanes (Concept 10) would have to be developed without substantially increasing residuals. Shaped channels (Concept 11) of

light enough weight to be competitive would be a serious development difficulty to overcome. Developing this concept for typical orbital mission adverse acceleration requirements would be a substantial effort. The extensive hardware requirements for the propellant depot (Concept 8) approach would make this the lowest ranking concept for this criteria.

2.2.8 RECURRING COSTS. The recurring costs increase with the amount of hardware added to the tank and the complexity of that hardware. The spray nozzle approach (Concept 2 and 9) is simplest. The propellant depot approach (Concept 8) is complicated by the fittings required to attach the receiver to the depot. Using an inlet diffuser (Concepts 3 and 4) is slightly more complicated. Attaching a vent (thermodynamic vent to remove fluid, Concept 7) requires additional fittings and valving. Autogenous pressurization (Concept 9) may require hardware to control tank mixing after pressurization. The hardware requirements are greater for (Concept 10) the internal vanes inside the channels. Shaped tapered channels (Concept 11) and channels with tubing attached (Concept 12) will be more difficult to fabricate. The system with the highest recurring cost will be the system using porous tubing (Concept 6). The cost will increase due to the tubing and the increased structural requirements of beefing up the channels to handle the higher pressure and the impingement forces during filling.

2.2.9 HARDWARE WEIGHT. The spray nozzle concept (Concepts 2 and 9) will have the lowest weight, followed by the screened diffuser (Concept 3). The diffuser nozzle combination (Concept 4) will be slightly greater in weight. The weight of adding a vent to channels to remove fluid (Concept 7) is next in ranking. Internal vanes (Concept 10) and shaped tapered channels (Concept 11) will have increased weight because of the added material required to make the vanes and the tapered channel. Tubing weight (Concept 12) is the main penalty for the open loop refrigeration system. The weight penalty for the porous tubing (Concept 6) and channel structural supports penalize the system that forces liquid into the channels with variable porosity tubing. Concept 8, the propellant depot approach, has significantly more components than the other concepts.

2.2.10 FLUID WEIGHT PENALTY. In general, minimum fluid weight penalty will be due to the systems using screened diffusers for chilldown and spray nozzles for filling (Concept 3) if venting is not required during chilldown. If venting will be required during chilldown, the spray nozzle case (Concept 2) gives the lowest vented fluid weight penalty. Residuals will not be affected for any of the concepts using pressurization (Concepts 2, 3, 4, 8 and 9). For the propellant depot approach (Concept 8) vented fluid losses will be a minimum since all vented fluid will be reliquified. The system using thermodynamic vent flow (Concept 7) to "purge" the channels of vapor will likely have to vent fluid in addition to the normal requirement in order to remove all the vapor from the channels. Residuals will not be affected. Using cooling tubes (Concept 12) to condense the vapor in the channels will require substantially more vented fluid than the normal boiloff would require. Residuals will not be affected. Residuals will be increased for the concepts using porous tubing (Concept 6) shaped channels (Concept 11) and internal vanes (Concept 10).

2.2.11 RELIABILITY. Reliability is a direct function of the number of moving parts. The systems (Concepts 2, 3, 6, 10 and 11) rated with a score of 10 will have no more moving parts than a fluid management system that is not refilled in orbit. They have a fill line, pressurization line, outflow line and corresponding valving. The autogenous pressurization system (Concept 9) may require systems to control mixing. The concept using diffusers and spray nozzles (Concept 4) will have an additional valve (or a three way valve) and some controls. The thermodynamic vent system (Concept 7) may have an additional valve and controls for operating continuously until filling of the capillary device occurs. Open loop refrigeration (Concept 12) will require an additional valve and pressure regulator to cool the channels. The orbital propellant depot (Concept 8) has considerably more equipment than the other concepts but it is likely to be highly reliable and redundant.

## 2.5 CONCEPT SELECTION

Results of the systems comparison indicate that the concepts using pressurization for condensing any vapor trapped in the capillary device are the most promising (Concepts 2, 3, 4 and 9). However, Concept 3 will not be given further consideration because its primary advantage, that of providing low pressure rise during tank chill, will be minimized or nullified due to the introduction of a prechill phase preceding tank chill (see Section 3 for further discussion). Concept 4 is also eliminated from further consideration because it too includes a diffuser to provide low pressure rise during tank chill. Concept 9, which employs autogenous pressurization, is a preferred approach because the avoidance of helium can greatly simplify management system refill. There are a number of applications, however, which require helium pressurization to maintain a continuous supply of subcooled liquid from the fluid management system. Consequently, Concept 2 will also be evaluated in greater detail in Section 3.

# 3

## FILLING ANALYSIS FOR SELECTED CONCEPT (CRYOGENS)

Analyses were conducted for filling the system described in Figure 1-1 with liquid hydrogen but modified to allow liquid flow into the tank through one or more spray nozzles, and to allow venting through a tapered vent tube. A sketch of the modified fluid management system is given in Figure 3-1.

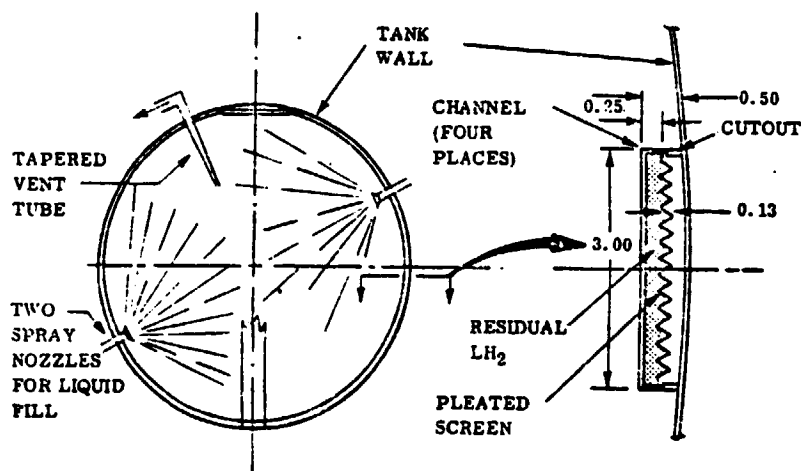


Figure 3-1. Modified Fluid Management System With Screen Channel Configuration

The filling of fluid management systems in orbit introduces complexities not experienced during one-g propellant fill. Filling a storage tank on the ground can be easily accomplished because vapor is readily expelled as liquid is introduced. Even cooling a storage tank prior to cryogenic fill presents no problem. Except in the case of complex, multiple compartment acquisition systems, the liquid vapor interface is usually well-defined during normal gravity filling. Vapor

will not be trapped within the screen acquisition device if splashing or wicking do not wet the screen in advance of the liquid/vapor interface. If vapor is trapped during normal gravity fill, its location will be known, and if necessary it can be vented using valving and vent lines.

However, low-g propellant fill cannot be satisfactorily accomplished with standard one-g techniques. This is due to the fact that the liquid-vapor distribution is not sufficiently well defined to enable pure vapor venting during the filling operation. The influence of g-environment upon storage tank fill is illustrated in Figure 3-2. The cost of transporting propellants into space for in-orbit propellant fill is sufficiently high that two phase venting is undesirable. Vapor trapped within the screen device is also unacceptable because pure liquid flow from the fluid management system is required, and cannot be guaranteed unless the screen device is free of vapor.

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Propellant fill is further complicated by two conditions that will exist: an empty storage tank will reside at a substantially higher initial temperature than the cryogen prior to initial fill, and a partially full tank requires that a major portion of the helium pressurant be vented before refill can be initiated. The question to be resolved for the

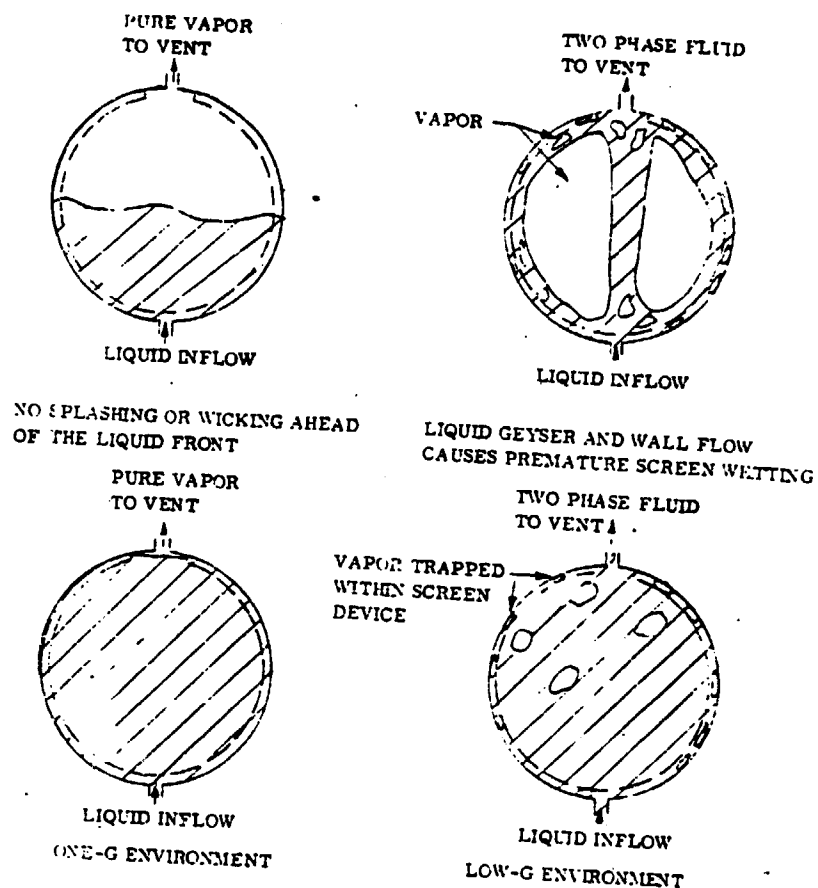


Figure 3-2. Comparison of One-g and Low-g Storage Tank Fill Employing One-g Fill Techniques

former condition is how best to fill the storage tank without expending excessive propellants in the process, and without exceeding structural allowable storage tank pressure. A solution requires that trades be conducted to explore various combinations of tank prechill, chill, and fill.

The second case, that of refilling a partially full storage tank, requires that a major portion of the helium pressurant be vented before propellant refill can be initiated. Liquid boiling will occur throughout this helium vent process which will increase the possibility of losing liquid overboard. Although this is undesirable, a greater concern is that screen dry-out will occur due to boiling, increasing the likelihood that helium will penetrate it.

In an effort to circumvent or minimize the possible adverse effects of the identified problem areas, detailed analyses were conducted. A subsequent procedure to fill the system under orbital conditions was devised, with consideration given to the following variables: liquid inflow rate and velocity, initial storage tank temperature, entering liquid vapor pressure, and liquid spray characteristics. Detailed analyses of each phase of fluid management system fill procedure are provided in the following subsections discussions.

### 3.1 PRECHILL ANALYSIS

Fluid management prechill is required whenever initial temperature is such that the stored tank energy will result in excessive pressure during the chilldown mode. Prechill is accomplished by introducing liquid or vapor into the propellant tank at velocity that provides good heat exchange between the high temperature walls and the cooling fluid. This procedure has the advantage of requiring little mass to effect tank cooling. Liquid hydrogen will be more readily available than hydrogen vapor in orbit, and is considered to be the fluid candidate for effecting prechill. It is possible, however, that the vapor normally generated in chilling the transfer lines to the pressure vessel can also be employed for prechill. A discussion is given in this section on prechill with both phases.

3.1.1 PRECHILL WITH HYDROGEN VAPOR. The primary requirement for system prechill is to reduce tank temperatures sufficiently that tank chill and fill will be accomplished without venting. It is implicit in this requirement that venting is unacceptable during the chill and fill mode because of the possibility that an unknown quantity of liquid will be lost overboard since propellant control cannot be maintained during this process. Venting is acceptable during prechill, however, because the elevated tank temperatures will quickly evaporate liquid during this phase.

3.1.1.1 Prechill Requirements. The primary factor in determining prechill requirements is the theoretical maximum pressure that can occur during tank chill as a function of initial tank temperature. Maximum tank pressure, for a given mass addition, occurs when the tank vapor and tank wall reside at the same temperature. The analytical development for theoretical maximum pressure is given below.

The First Law expression for introducing liquid into a container is

$$dE_g + dE_w = h_L dm_L \quad (3-1)$$

$$dE_g = (u_g m_g)_2 - (u_g m_g)_1 = \text{change in ullage energy}$$

If one assumes that the tank is initially evacuated

$$m_{g1} = 0 \text{ and } dE_g = u_{g2} m_{g2} \quad (3-2)$$

Also for an initially evacuated container,  $m_{g2} = dm_L$  (3-3)

$$dE_w = (u_w m_w)_2 - (u_w m_w)_1 = \text{change in tank wall energy}$$

Since tank mass is constant,  $dE_w = (u_{w2} - u_{w1}) m_w$  (3-4)

Combining Equations 3-1 through 3-4

$$u_{g2} m_{g2} + (u_{w2} - u_{w1}) m_w = h_L m_{g2} \quad (3-5)$$

$$(u_{g2} - h_L) m_{g2} = (u_{w1} - u_{w2}) m_w \quad (3-6)$$

Finally

$$m_{g2} = (u_{w1} - u_{w2}) m_w / (u_{g2} - h_L) \quad (3-7)$$

where

$u_{w2}$  and  $u_{g2}$  are evaluated at  $T_2$

$dE_g$  = change in tank wall internal energy

$h_L$  = enthalpy of liquid entering tank

$dm_L$  = differential liquid mass addition to tank

$u_g$  = internal energy of vapor in tank

$m_g$  = mass of vapor in tank

$u_w$  = internal energy of tank wall

$m_w$  = tank wall mass

$T$  = temperature

subscript

1 = conditions at beginning of interval

2 = conditions at end of interval

From the equation of state, gas pressure is

$$P_{g2} = \left( \frac{m ZRT}{V} \right)_{g2} \quad (3-8)$$

where

$Z$  = compressibility factor

$R$  = gas constant



$V$  = tank volume

$P_{g2}$  = gas pressure

Peak pressure during tank chill can be determined as a function of mass addition by a simultaneous solution of Equations 3-7 and 3-8. Results are given in Figure 3-3 for a range of incoming liquid vapor pressure and initial tank temperature conditions. Note

that the peak pressures of this figure are the theoretical maximums that can occur for the given initial tank temperature. A lesser or greater mass addition will result in a lower tank pressure than the theoretical maximum level. Note also that entering liquid vapor pressure will have a minor influence upon tank pressure.

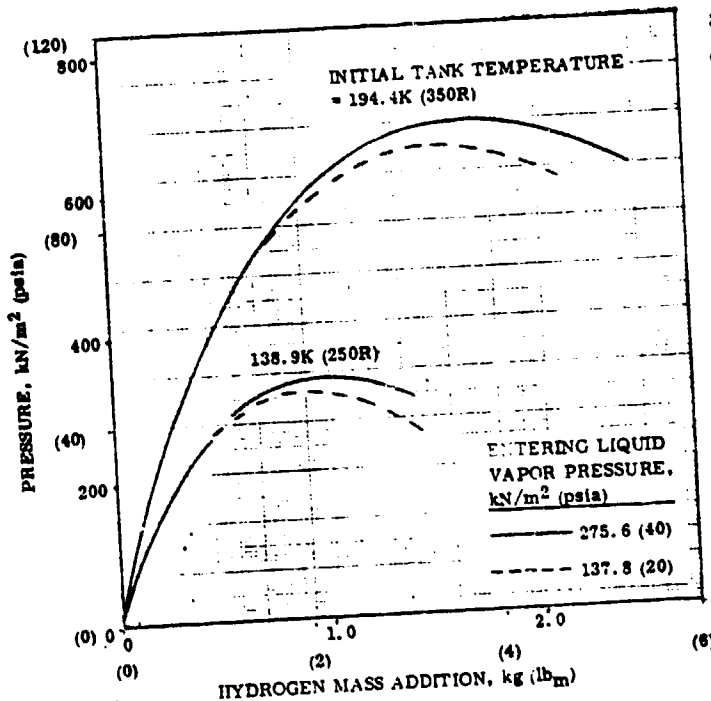


Figure 3-3. Peak Pressure During Management System Chillover Versus Mass Addition and Entering Liquid Vapor Pressure

The theoretical maximum pressures described by Figure 3-3 are plotted in Figure 3-4 as a function of initial tank temperature. Initial temperatures in excess of 194.4K (350R) were not considered because of the requirement to maintain system pressure below the maximum allowable level of 689 kN/m<sup>2</sup> (100 psia). This figure shows that maximum pressures will be 50 percent or less of the maximum allowable for initial tank temperatures lower than 138.9K (250R).

There is an obvious advantage to prechilling the pressure vessel to the lowest temperature possible because any concern relative to tank chill over-pressure can be eliminated. This advantage must be weighed against the complication incurred as more hydrogen vapor and time is required to prechill the system to lower and lower temperatures. An extreme condition to consider is the difficulty of prechilling to liquid hydrogen temperatures. Without analysis one can conclude that prechill mass requirements will be quite high at low tank temperatures because vapor vent temperatures and, therefore, vent enthalpy will be correspondingly low. Furthermore, the likelihood of liquid residing in the tank increases as it approaches liquid temperature, which increases the probability that liquid will be vented overboard during tank blowdown.

3.1.1.2 Tank Prechill Model. A prechill subroutine was introduced into the HYPRES computer program in order to adequately assess the variables of a prechill procedure.

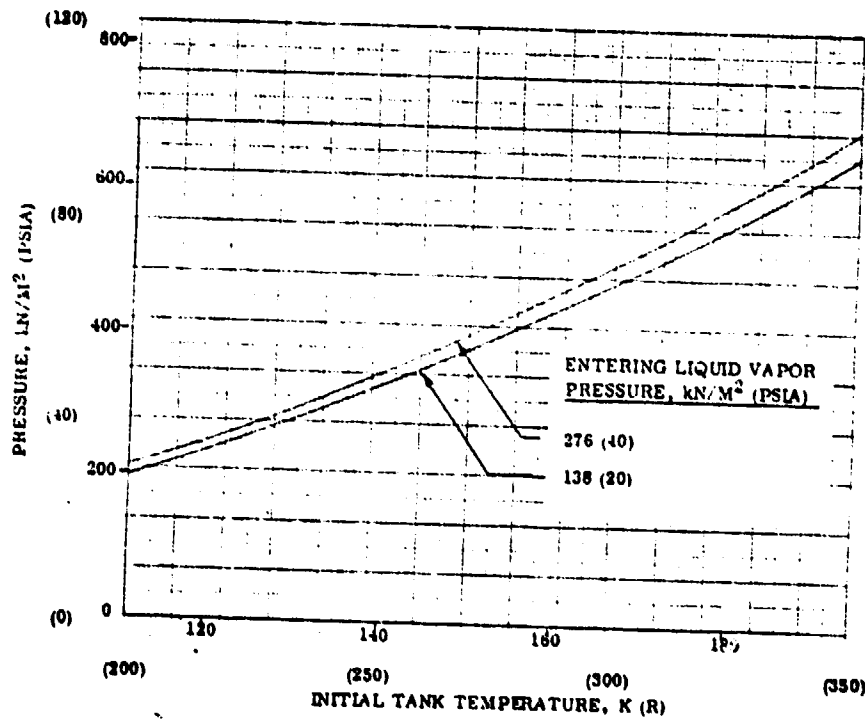


Figure 3-4. Maximum Pressure During Management System Chardown Versus Initial Tank Temperature and Entering Liquid Hydrogen Vapor Pressure

representative of the heat exchange mechanism that will occur when vapor is continuously introduced into the fluid management system tank. The heat transfer correlation and the required modifications are given as

$$\frac{h}{\rho c_p} \left( N_{PR} \right)^{2/3} = 0.13 \left( \frac{(P_i/V) \mu}{\rho^2} \right)^{1/4} \quad (\text{Reference 3-1}) \quad (3-9)$$

where

$h$  = heat transfer coefficient

$\rho$  = fluid density

$c_p$  = constant pressure heat capacity

$N_{PR}$  = Prandtl number

$P_i$  = mixer input power

$V$  = tank volume

$\mu$  = fluid viscosity

The subroutine is described in greater detail in Appendix B, whereas the model assumptions, results and conclusions are given in the following paragraphs.

The key to this analysis is in identifying the appropriate heat transfer coefficient between incoming vapor and the tank walls. Conventional forced convection expressions for flow over a flat plate or for jet impingement upon a surface, represent configurations that are too dissimilar to be applicable. Instead, a correlation developed for industrial mixing processes was selected as being

Equation 3-9 was developed for liquids contained in cylinders. These liquids were continuously agitated with a mixing unit. Mixer input power was responsible for fluid agitation and is one of the variables of Equation 3-9. For the prechill application, it is believed that fluid agitation will be the same whether a mixer or fluid inflow is responsible. It is important only to have equivalent power conditions. Power output rather than power input will influence fluid agitation. Consequently equivalence will be between fluid power input and mixer power output. This results in

$$P_i (\text{EFF}) = P_o = \dot{m}v^2 \quad (3-10)$$

where

$P_o$  = mixer power output

EFF = mixer efficiency (conservatively assumed as 40 percent for this study)

$\dot{m}$  = entering mass flow rate

$v$  = entering fluid velocity

$\dot{m}v^2$  = fluid power input

Substituting Equation 3-10 into 3-9 results in

$$\frac{h}{\rho c_p} (N_{PR})^{2/3} = 0.163 \left[ \frac{(\dot{m} v^2 / V) \mu}{\rho^2} \right]^{1/4} \quad (3-11)$$

Equation 3-11 indicates that heat transfer to the tank walls can be controlled by varying entering flowrate and velocity.

The following charge and vent procedure was selected for fluid management system prechill

1. Charge the tank at a known vapor flowrate until the difference between wall and gas temperature has reached a specified value.
2. Vent the tank to a pre-determined low pressure. A reasonable level was selected as 6.89 kN/m<sup>2</sup> (1.0 psia).
3. Charge and vent the tank as required to reduce tank temperature to the pre-determined level.

Figure 3-5 gives prechill mass as a function of final tank temperature. Note that prechill mass requirements increase as final tank temperature decreases. Also, prechill mass

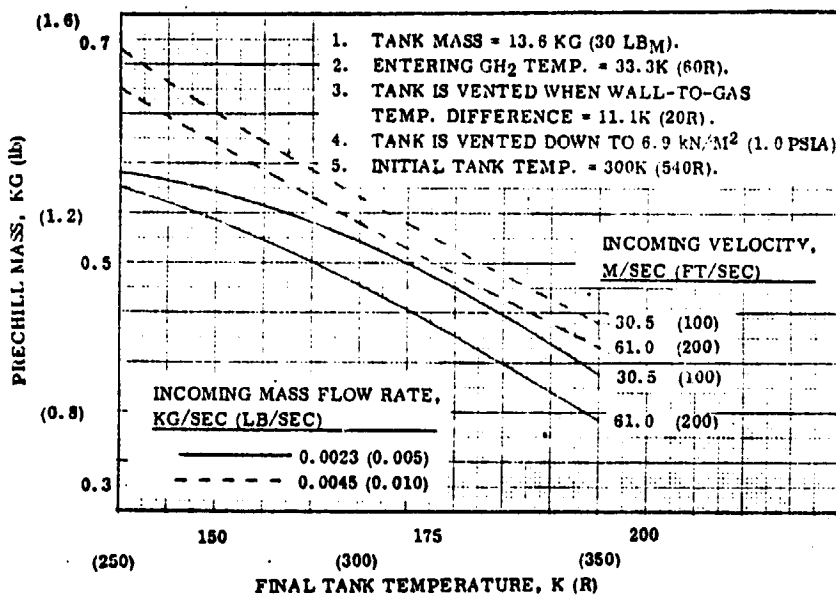


Figure 3-5. Hydrogen Vapor Mass Required to Prechill the Fluid Management System Tank

An evaluation of Figures 3-4 through 3-7 indicates that a final tank temperature of about 138.9K (250R) represents a reasonable trade between the advantage of low peak

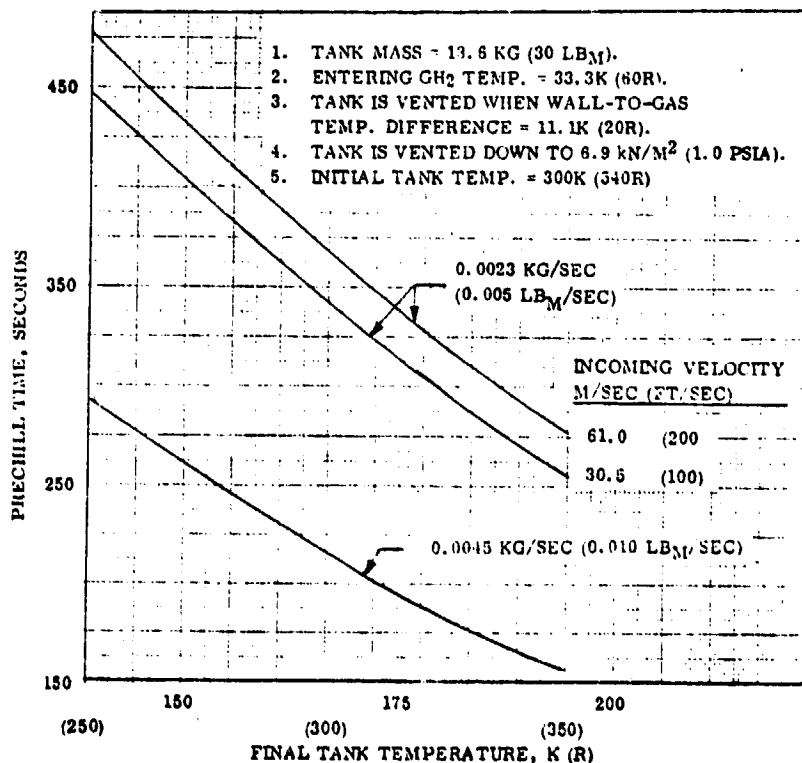


Figure 3-6. Time to Prechill the Fluid Management System With Hydrogen Vapor

can be reduced by decreasing charge flow rate or increasing vapor velocity. As expected increased prechill times are required to provide lower final temperatures, Figure 3-6. These times can be reduced by increasing charge flow rate or by decreasing vapor velocity. Finally, Figure 3-7 shows that the frequency of charge and vent cycles increases as final tank temperature decreases. This figure also shows that fewer cycles are needed for tank prechill as charge flow rate is increased.

pressures during tank chill and the disadvantages of increased hydrogen mass, time, and charge and vent cycles. These figures also indicate that (1) the influence of entering velocity is minimal, (2) prechill mass requirements are directly proportional to entering flowrate, and (3) prechill time and charge and vent cycles are inversely proportional to entering flowrate.

3.1.1.3 Summary. The above parametric data has resulted in the following recommendation for system prechill:

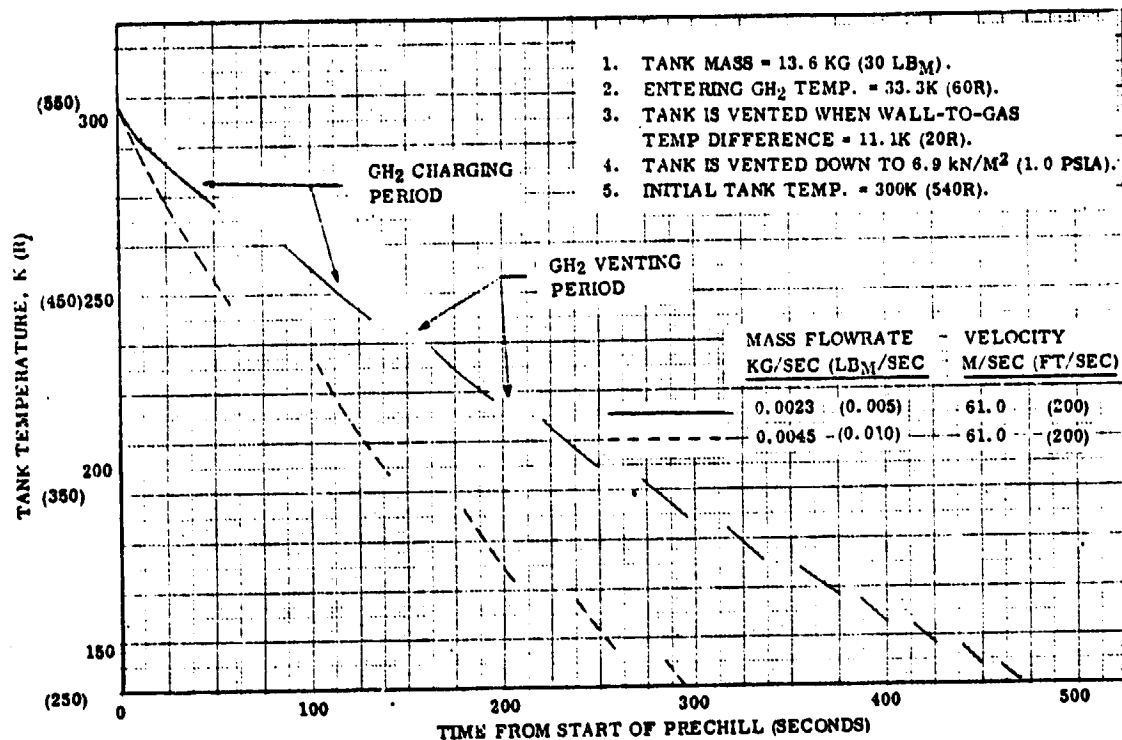


Figure 3-7. Fluid Management System Tank Wall Temperature History During a Gaseous Hydrogen Prechill

prechill flowrate = 0.0045 kg/sec (0.01 lb<sub>m</sub>/sec)

incoming velocity = 61 m/sec (200 ft/sec)

The tank will be prechilled to 138.9K (250R) and 0.66 kg (1.45 lb<sub>m</sub>) will be expended. GH<sub>2</sub> will be expended during a 295 second period. The higher flowrate was selected because it will reduce the required number of charge and vent cycles from nine, at a flowrate of 0.0023 kg/sec (0.005 lb<sub>m</sub>/sec), to five. The increased prechill mass resulting from the higher flowrate is acceptable, considering that the smaller number of charge and vent cycles will simplify the procedure.

3.1.2 PRECHILL WITH LIQUID HYDROGEN. It is expected that liquid hydrogen will be more readily available in orbit than hydrogen vapor. Thus it will be more convenient to initiate prechill with liquid. Thermodynamically and fluid dynamically there is little difference between a liquid and vapor prechill. Approximately 15 percent less hydrogen is needed for prechill when it enters as a liquid, due to the heat of evaporation which increases its energy absorbing capability. Heat transfer coefficients at a given fluid power input, will be the same as for gas phase flow, once the liquid is evaporated. The primary difference between liquid inflow and vapor inflow results from film boiling which occurs when liquid strikes the hot tank walls. But, since film boiling heat transfer rates will represent only approximately 15 percent of the fluid energy absorbing capability, total prechill time and the number of charge and vent cycles will not differ

significantly between liquid and vapor inflow. It is concluded, therefore, that the results for prechill with vapor inflow apply equally to a liquid inflow.

### 3.2 TANK CHILL ANALYSIS

The fluid management system tank temperature will not exceed 138.9K (250R) at the start of the chill process. Liquid hydrogen flow to the tank will be initiated once the system vent valve is closed. Film boiling will occur as liquid strikes the warm tank walls, and the resulting evaporation will increase absolute maximum pressure to 339 kN/m<sup>2</sup> (49.2 psia), as indicated by Figure 3-4. This pressure will be about 50 percent of the maximum allowable level. Neither tank chill nor the subsequent tank fill process will be influenced whether pressure peaks at 339 kN/m<sup>2</sup> (49.2 psia), or at a lower level. As a result, it will not be necessary to impose constraints on the flow process for purposes of controlling tank pressure and, it will be possible to perform tank chill with any constraints imposed by either prechill or tank fill. Thus it is seen that the selected prechill process has virtually eliminated all concerns relative to an in-orbit tank chill process.

**3.2.1 TANK CHILL THERMODYNAMICS.** Despite the knowledge that a no-vent tank chill can readily be accomplished, it is still necessary to evaluate the influence of inflow parameters upon tank pressure during this process. Tank pressure determination involves a complex interrelationship between the incoming liquid, ullage mass and tank mass. A computer program (HYPRES), which is described in Appendix A, has been written to analyze the tank chill process. Briefly, it is assumed that liquid enters the tank as a spray, and exchanges energy with vapor as it flows through the ullage. Heat exchange between the wall and ullage vapor is also permitted. Liquid droplet evaporation or condensation will occur, depending upon liquid spray characteristics and ullage-to-liquid temperature differences. Liquid-to-wall heat transfer is limited to film boiling. The nucleate boiling process is not included because the stored wall energy is insignificant in this temperature range.

Two variables requiring further analysis are the drop size determination and wall boiling during the chill process. Droplet size determination is required to verify that spray conditions reside within the acceptable analysis range. Wall boiling rates must be identified as a function of such variables as fluid properties, velocity, drop size and gravity.

**3.2.1.1 Liquid Droplet Size Determination.** Spray-nozzle drop size information is available from manufacturer's data for air-water combination, but rarely for other fluids. No all-purpose defining equation for the effect of physical properties exists. The dependence of drop size on geometry, velocity, and properties is complicated by changes in the nature of the break-up process as these variables change. The suggested predictive route starts with manufacturer's air-water data at the desired flow rate, and then drop size corrections are made for different physical properties. The recommended equation for drop correction is given in Reference 3-2 as

$$\frac{d_{vm}}{(d_{vm})_{\text{water}}} = \left[ \frac{\sigma}{73} \right]^{0.5} \left[ \frac{\mu_L}{1.0} \right]^{0.3} \left[ \frac{62.4}{\rho_L} \right]^{0.3} \quad (3-12)$$

where

$d_{vm}$  = volume median droplet diameter

$\sigma$  = surface tension, dynes/cm

$\mu_L$  = liquid viscosity, centipoise

$\rho_L$  = liquid density,  $\text{lb}_m/\text{ft}^3$

The exponential dependences in Equation 3-12 represent averages of values reported by a number of studies.

A more significant drop size than  $d_{vm}$  is the Sauter mean diameter,  $d$ , which has the same ratio of surface area to mass as the total drop population. This diameter was used in the study because a representative surface area is critical to the chill process. It is typically 70 to 90 percent of  $d_{vm}$ ; a value of 80% was selected for this study.

3.2.1.2 Spray Nozzle Selection. A full cone spray nozzle was selected for propellant tank chill and fill. Typical nozzle performance characteristics were obtained from Spraying Systems Co., including particle size as a function of nozzle inlet pressure for a variety of nozzles, Figure 3-8. Equation 3-12 was employed to determine drop size for the combination of liquid hydrogen flowing through gaseous hydrogen. Droplet diameter is plotted versus liquid flowrate for two spray nozzle configurations, Figure 3-9. These configurations were selected because of the large drop size created during the spray process. Note that the maximum hydrogen drop size is expected to be less than  $600 \mu$  (0.024 inches).

3.2.1.3 Liquid Splattering on a Hot Surface. The actual heat exchange process to occur during tank chill will be that of heat transfer between an individual droplet and the hot tank walls as contact occurs. This method of heat exchange is different from that of pool film boiling for one obvious physical reason; film boiling is based upon a hot surface immersed in a pool of liquid where a stable vapor film has developed, whereas heat transfer due to droplet splattering is of a more transient nature where a vapor film is created and destroyed with each impact. The following expression was developed in Reference 3-3 for the maximum heat transfer experienced by the impingement of a single droplet upon a hot surface

$$\frac{Q_{\max}}{\rho_L d^3 \lambda} = 8.44 \times 10^{-4} \left[ \frac{\rho_L^2 v^2 d}{\rho v_F} \right]^{0.341} \quad (3-13)$$

and

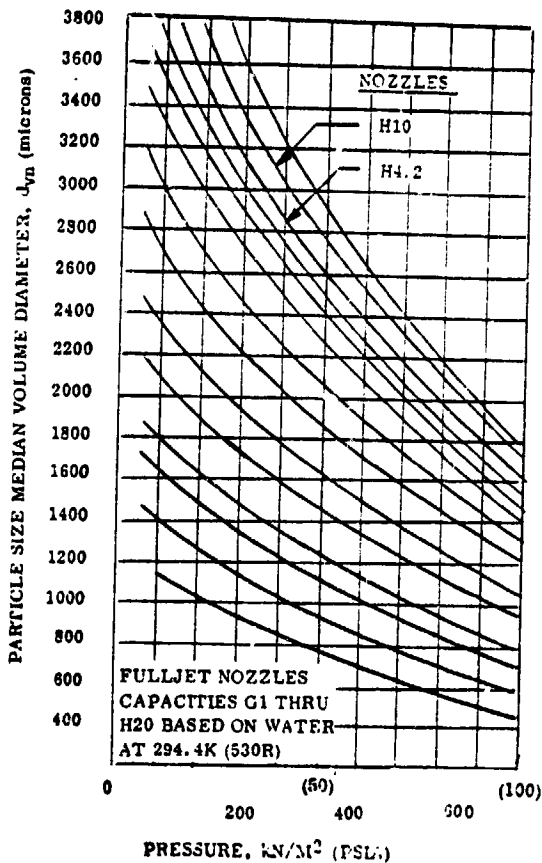


Figure 3-8. Water Spray Particle Size Versus Pressure For Fulljet Nozzles

$$T_F = (T_w + T_s) / 2, \text{ film temperature}$$

$$\sigma = \text{surface tension}$$

$$\lambda = h_{fg} + C_{pv} \frac{T_w - T_s}{2} \quad (3-14)$$

where

$Q_{max}$  = maximum heat transfer per drop

$\rho_L$  = liquid density

$d$  = drop diameter

$\lambda$  = modified heat of evaporation by Equation 3-14

$h_{fg}$  = enthalpy of evaporation

$C_{pv}$  = vapor specific heat at constant pressure

$T_w$  = wall temperature

$T_s$  = liquid saturation temperature

$V$  = drop velocity

$\rho_{vF}$  = vapor density evaluated at  $T_F$

Equation 3-13 was based upon results of tests with water, Acetone, and Ethanol. It was observed that the maximum heat transfer condition corresponded to a droplet impact angle of 27 degrees as measured from the normal to the hot wall. The equation is valid for values of the independent variable,  $\rho_L^2 v^2 d / \rho_{vF} \sigma$ , less than  $10^7$ . Furthermore, it was determined that the efficiency of the process, defined as the ratio of actual heat transfer per drop to total energy required to evaporate the drop, is of the order of 50 percent for a single collision. Subsequent secondary collisions of splattered particles with the hot surface tend to increase the overall efficiency.

Observations made during the Reference 3-3 experimental effort indicated that the heat transfer mechanism was that of conduction across a vapor film. Qualitatively, a droplet will begin spreading on impact with the hot surface and the combination of contact surface area,  $A$ , and film thickness,  $\delta$ , with time influences heat transfer rate. Both  $A$  and  $\delta$  are influenced by drop size, impact velocity and impact angle, although the exact relation is not known.



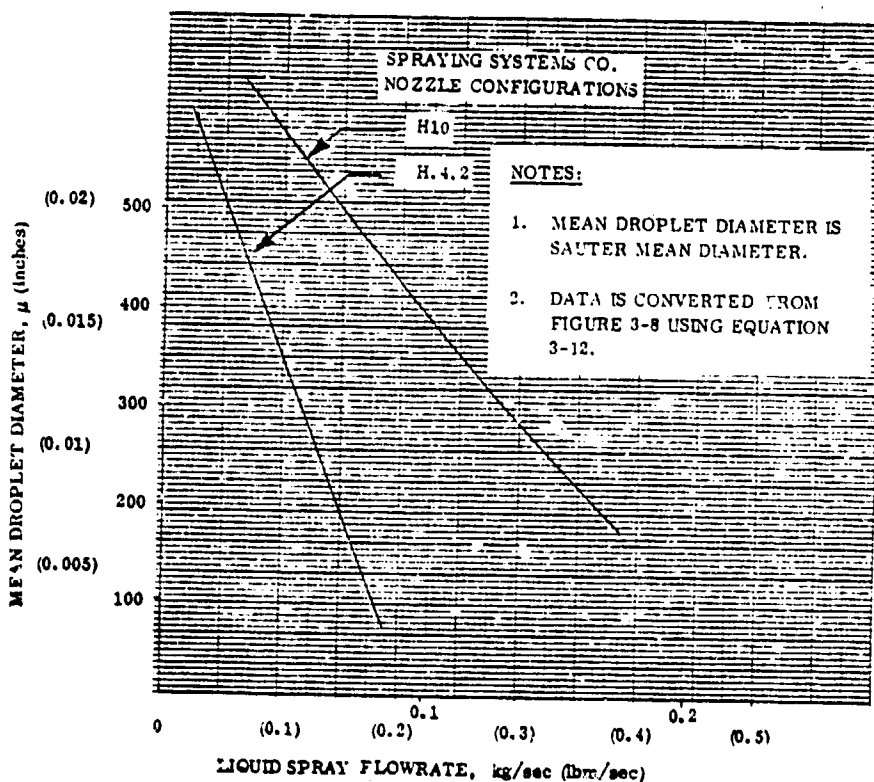


Figure 3-9. Liquid Hydrogen Droplet Diameter Versus Flowrate for Two Spray Nozzle Configurations

Maximum heating rate for a spray condition can be determined by combining Equation 3-13 with the following expressions

$$\dot{Q}_{\max} = \dot{n} \times Q_{\max} \quad (3-15)$$

$$\dot{m} = \dot{n} \times (\text{mass/drop}) = \dot{n} \left( \frac{\pi}{6} \rho_L d^3 \right) \quad (3-16)$$

and

$$\dot{Q}_{\max} = \frac{\dot{m} \times Q_{\max}}{\frac{\pi}{6} \rho_L d^3} = (\dot{m} \lambda) \times 8.44 \times 10^{-4} (\rho_L^2 V^2 d / \rho_{VF} \tau)^{0.341} \quad (3-17)$$

or

$$\frac{\dot{Q}_{\max}}{\dot{m}} = 4.42 \times 10^{-4} \lambda (\rho_L^2 V^2 d / \rho_{VF} \tau)^{0.341} \quad (3-18)$$

where

$\dot{Q}_{\max}$  = maximum heat transfer rate under spray conditions

$n$  = number of droplets per second striking a hot surface

$\dot{m}$  = mass flowrate of spray condition

Equation 3-18 has been introduced into the HYPRES computer program. The significance of this expression is that the splattering phenomenon appears to be independent of acceleration environment. Thus it is reasonable to expect that chill tests, conducted in a normal gravity environment, will be applicable to a near zero gravity environment. This is not to suggest that scaling or extrapolation problems may not exist. However, uncertainties in wall boiling heat transfer rate should not invalidate the liquid spray approach to tank chill. Furthermore, regardless of the uncertainties of Equation 3-18, peak pressure will not exceed  $339 \text{ kN/m}^2$  (49.2 psia).

3.2.1.4 Peak Tank Chill Pressures. Computer runs were conducted to obtain parametric data on the influence of liquid flow variables upon maximum pressure during tank chill. Figure 3-10 provides a summary of these computer output results. The significant results of this Figure are (1) peak pressures increase in magnitude at reduced inflow rates, and (2) peak pressures are insensitive to flowrate, droplet velocity and droplet diameter at flowrates exceeding  $0.18 \text{ kg/sec}$  ( $0.40 \text{ lb}_m/\text{sec}$ ). These observations can be explained as follows: maximum pressure during tank chill occurs when all the tank wall energy is convected to the ullage vapor. The other pressure extreme occurs when tank wall energy is absorbed by liquid. Thus, peak pressures occur at low flowrates because much of the liquid is evaporated before striking the

SYSTEM PARAMETERS

1. TANK VOLUME -  $0.62 \text{ M}^3$  (22  $\text{FT}^3$ )
2. ENTERING LIQUID VAPOR PRESSURE -  $137.8 \text{ KN/M}^2$  (20 PSIA)
3. TANK MASS -  $13.6 \text{ KG}$  (30  $\text{LB}_M$ ) ALUMINUM
4. INITIAL TANK TEMPERATURE -  $139.9\text{K}$  (2500)

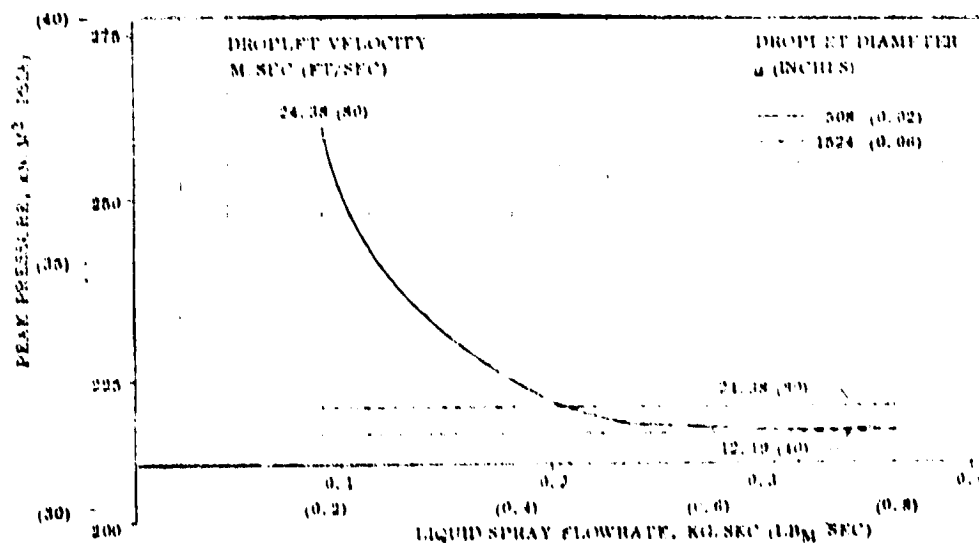


Figure 3-10. Maximum Pressure During Management System Tank Chill as a Function of Liquid Spray Parameters

tank wall with the result that considerable wall-to-ullage energy transfer occurs. Conversely, the high flowrate condition will maintain a liquid film on the tank walls and absorb all the wall energy.

3.2.1.5 Summary. Analysis shows that the selected prechill procedure has virtually eliminated all concern relative to a tank chill procedure. This is because the theoretical maximum pressure to occur during a no-vent tank chill will be about  $339 \text{ kN/m}^2$  (49.2 psia). Figure 3-10 indicates that the expected maximum pressure will be less than  $276 \text{ kN/m}^2$  (40 psia) if liquid is sprayed into the tank. There is no possibility that venting will be required since the maximum allowable pressure is  $689 \text{ kN/m}^2$  (100 psia). Furthermore, any uncertainty in film boiling heat transfer rate that may exist due to liquid impingement on a hot wall, cannot possibly result in a pressure greater than the theoretical maximum quoted above.

### 3.3 TANK FILL ANALYSIS

Propellant tank fill represents a continuation of liquid flow initiated for the tank chill process. Tank fill is defined as commencing at the instant that tank temperature drops to liquid temperature. This definition has been selected for the convenience of analysis since from this time on the fill process is not influenced by fluid management system configuration, other than volume. That is, propellant state conditions during fill are only a function of the thermodynamic interaction between liquid and vapor.

3.3.1 THERMAL EQUILIBRIUM TANK FILL. The single requirement for tank fill is to maintain acceptably low pressures during the process. Tank pressures will be at a minimum if thermal equilibrium conditions are maintained during fill. Thermal equilibrium will be approached as heat and mass exchange between the phases is increased, which can be achieved by creating a highly agitated fluid condition. Given the assumption of a thermal equilibrium tank fill one obtains a simple relationship between initial tank temperature at the start of no-vent chill and fill, incoming liquid vapor pressure, and final liquid vapor pressure. This relationship is derived below from the First Law of Thermodynamics for liquid flow into a closed container

$$dE_g + dE_L + dE_W = h_L dm_L \quad (3-19)$$

$$dE_g = (u_g m_g)_2 - (u_g m_g)_1 = \text{change in ullage energy} \quad (3-20)$$

$$dE_L = (u_L m_L)_2 - (u_L m_L)_1 = \text{change in liquid energy} \quad (3-21)$$

$$h_L dm_L = h_L (m_{L,2} - m_{L,1}) = \text{enthalpy change due to entering liquid} \quad (3-22)$$

$$dE_W = (u_W m_W)_2 - (u_W m_W)_1 = m_W (u_2 - u_1)_W \quad (3-23)$$

$$= m_W \Delta u_W = \text{change in tank wall energy}$$

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If one assumes that the tank is initially evacuated

$$m_{g1} = m_{L1} = 0$$

and

$$dE_g = u_{g2} m_{g2} \quad (3-24)$$

$$dE_L = u_{L2} m_{L2} \quad (3-25)$$

$$h_L dm_L = h_L m_{L2} \quad (3-26)$$

Combining Equations 3-19 through 3-26 we have

$$u_{g2} m_{g2} + u_{L2} m_{L2} + m_w \Delta u_w = h_L m_{L2} \quad (3-27)$$

Solving Equation 3-27 for  $h_L$ , results in

$$h_L = u_{g2} m_{g2}/m_{L2} + u_{L2} + \Delta u_w m_w/m_{L2} \quad (3-28)$$

At thermal equilibrium one can relate  $h_L$ ,  $u_{L2}$  and  $u_{g2}$  to liquid vapor pressure and temperature. Consequently,  $u_{L2}$  and  $u_{g2}$  are known once final vapor pressure is specified.  $\Delta u_w$  can also be determined since initial and final temperatures are given. Finally,  $h_L$  (which is a function of entering liquid vapor pressure) can be determined for a desired liquid fill condition.

Equation 3-28 is summarized in Figure 3-11 which gives entering liquid vapor pressure as a function of initial tank temperature and final tanked liquid vapor pressure for a 90 percent and 100 percent liquid fill condition. Note that final liquid fill condition has a minimal influence on the other variables. Note also that final vapor pressure is less than entering liquid vapor pressure. This difference is due to the combination of initial tank wall energy and the heat of compression which are released to the tank fluid during chill and fill.

**3.3.2 FILL MODEL FOR LIQUID SPRAY DOMINANCE.** The intent of the tank fill process will be to create conditions conducive to attaining near-thermal equilibrium. These conditions will be achieved by introducing liquid into the tank through spray nozzles, which resulting spray will create a large liquid-vapor surface area. The combination of large surface area and high droplet velocity will provide the high heat transfer rates needed to attain near-thermal equilibrium conditions. As tank fill continues, the internal tank fluid environment will change from one where liquid droplets reside within the ullage volume to one where vapor bubbles are entrained within a liquid bulk. The transition from heat transfer dominated by liquid droplets to heat transfer dominated by vapor bubbles cannot be determined precisely, but it is expected to occur in the range of 10 percent to 20 percent liquid fill.

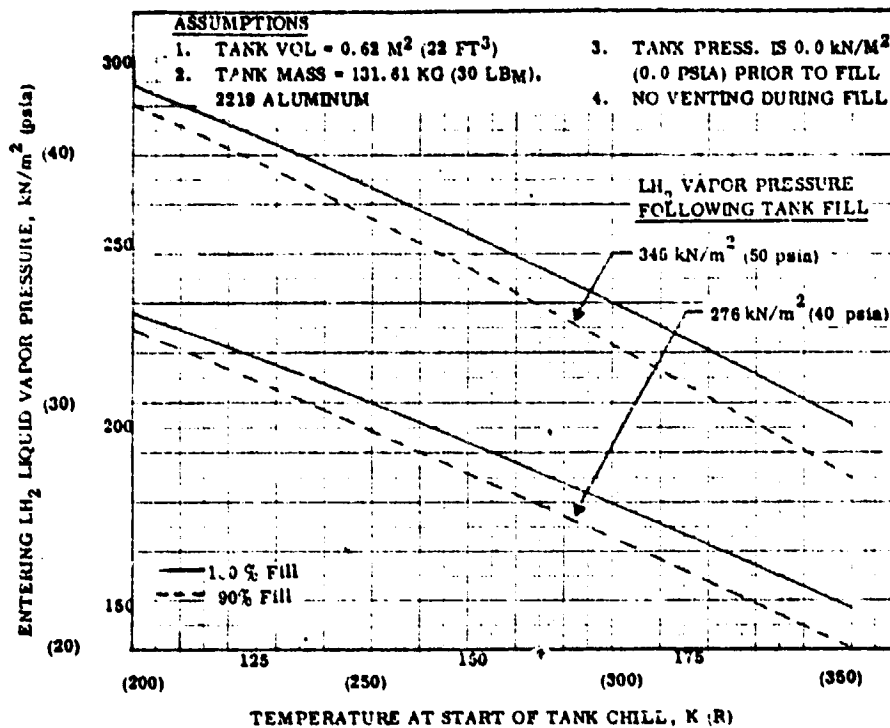


Figure 3-11. Fluid Management System Final Tank Pressures for a Thermodynamic Equilibrium Chill and Fill Process

3.3.2.1 Initial Fill Pressures. Equations describing heat and mass exchange with the ullage are those given in Section 3.2, and contained in HYPRES, for the tank chill process.

Propellant tank pressure and hydrogen temperature histories have been determined using the HYPRES computer program and results are given in Figure 3-12 and 3-13 for inflow rates of 0.091 kg/sec (0.2 lb<sub>m</sub>/sec) and 0.227 kg/sec (0.5 lb<sub>m</sub>/sec), respectively. Note that there is a gradual pressure decay rate once peak pressure has occurred. This is due to the influence of incoming liquid as it continues to chill the ullage. Also, it is seen that vapor temperature approaches liquid temperature within seconds from the start of liquid inflow. The temperature histories show that near-thermal equilibrium conditions are achieved early in the tank fill process.

3.3.3 FILL MODEL FOR VAPOR BUBBLE DOMINANCE. As the propellant tank begins to fill with liquid, the mechanism of liquid spray in a vapor environment will change to that of vapor bubble entrainment and dispersal within the liquid volume. Entrainment will occur as a result of liquid impingement with the liquid bulk. Dispersal will be caused by agitation created by liquid sprayed into the tank. It is estimated that this mechanism will be the dominant mode of heat and mass exchange throughout most of the tank fill process.

SYSTEM PARAMETERS AND FLOW CONDITIONS

1. TANK VOLUME = 0.62 M<sup>3</sup> (22 FT<sup>3</sup>)
2. TANK MASS = 13.6 KG (30 LB<sub>M</sub>) ALUMINUM
3. INITIAL TANK TEMPERATURE = 138.9 K (250R)
4. ENTERING LIQUID VAPOR PRESSURE = 137.8 kN/M<sup>2</sup> (20 PSIA)
5. LIQUID SPRAY FLOWRATE = 0.091 KG/SEC (0.2 LB<sub>M</sub>/SEC)
6. DROPLET DIAMETER = 762 μ (0.06 IN)
7. DROPLET VELOCITY = 24.38 M/SEC (80 FPS)

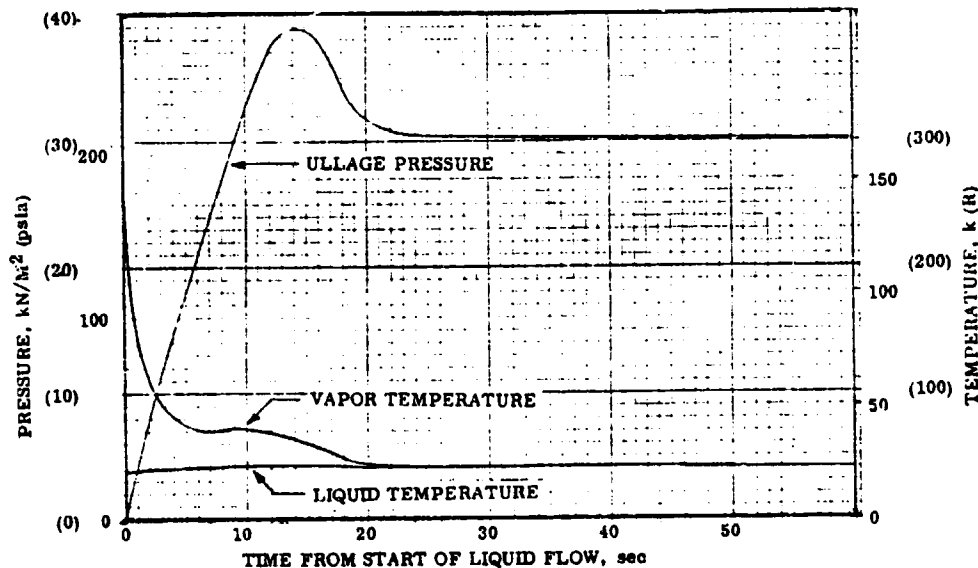


Figure 3-12. Fluid Management System Pressure and Temperature Histories During Initial Ten Percent Fill Period (Low Flowrate)

SYSTEM PARAMETERS AND FLOW CONDITIONS

1. TANK VOLUME = 0.62 M<sup>3</sup> (22 FT<sup>3</sup>)
2. TANK MASS = 13.6 KG (30 LB<sub>M</sub>) ALUMINUM
3. INITIAL TANK TEMPERATURE = 138.9K (250R)
4. ENTERING LIQUID VAPOR PRESSURE = 137.8 kN/M<sup>2</sup> (20 PSIA)
5. LIQUID SPRAY FLOWRATE = 0.227 KG/SEC (0.5 LB<sub>M</sub>/SEC)
6. DROPLET DIAMETER = 762 μ (0.06 IN)
7. DROPLET VELOCITY = 24.38 M/SEC (80 FPS)

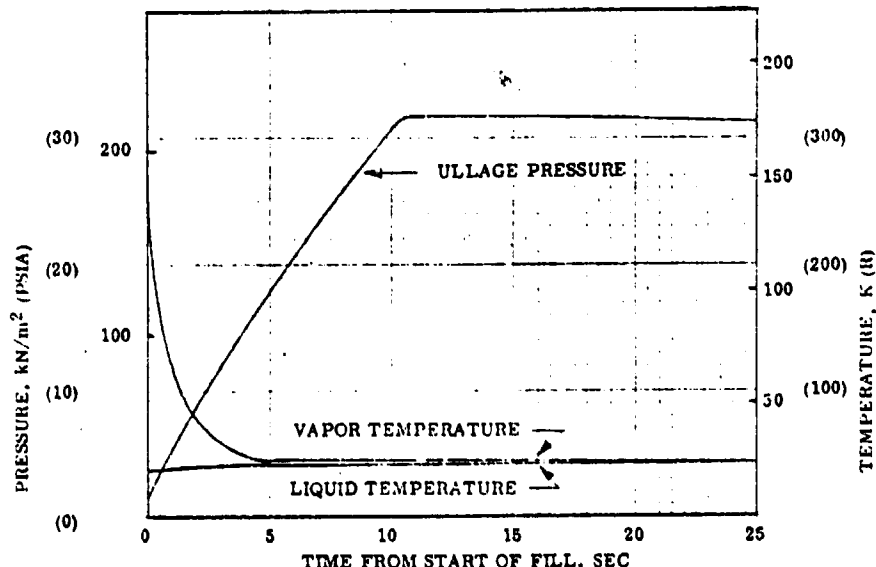


Figure 3-13. Fluid Management System Pressure and Temperature Histories During Initial Ten Percent Fill Period (High Flowrate)

3.3.3.1 Vapor Dispersal Mechanism. There are many industrial applications for which vapor dispersal in liquid is essential to achieving a high rate of heat and mass transfer. This subject has been extensively studied and much of the work has been collected and summarized in Reference 3-1. The dispersal of vapor in liquid is brought about by fluid dynamic forces which have to overcome the static forces of surface tension. Such surface forces resist dispersion by attempting to retain bubble sphericity and prevent gross distortion leading to break-up. The dynamic forces which bring about dispersion may be due to bouyancy or induced fluid flow creating viscous or inertial force which, if they do not act equally over the surface of a bubble, may cause it to deform and eventually break-up. Mechanical agitation is employed in mixing vessels to create shear stress by means of turbulence, which in turn causes bubble break-up. The empirical equations describing bubble diameter, percent vapor volume entrained beneath the liquid surface (vapor hold-up) and heat and mass transfer coefficients are applicable to a configuration similar to that shown in Figure 3-14. The figure describes a cylindrical tank filled with liquid, within which is immersed a mechanical device for mixing the liquid. Vapor is introduced into the tank through a manifold located near the bottom of the tank. The large vapor bubbles that form near the manifold are broken up into numerous small bubbles as a result of liquid shearing forces counteracting and overcoming surface forces of the individual bubbles. This bubble breakup mechanism is a Weber number phenomenon.

The equation for determining bubble diameter is given in Reference 3-2, Equation 18-138 as

$$d = \frac{0.0279 \sigma^{0.6} \epsilon^{0.5}}{\gamma^{0.2} (P_1/V_L)^{0.4}} + 0.09 \quad (3-29)$$

where

$d$  = mean sauter diameter, cm

$\sigma$  = surface tension, dyne/cm

$\epsilon$  =  $V_g/(V_g+V_L)$ , fractional volumetric holdup of the dispersed phase

$V_g$  = vapor volume entrained beneath the liquid surface, ft<sup>3</sup>

$V_L$  = liquid volume, ft<sup>3</sup>

$\gamma$  =  $(\rho_L/\rho_{H2O})$

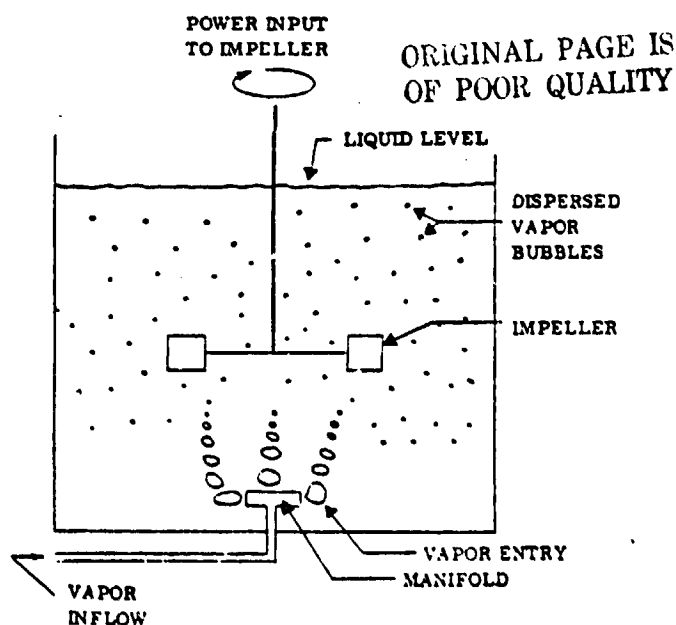


Figure 3-14. Apparatus for Study of Vapor Dispersal in Liquid

$\rho_L$  = liquid density

$\rho_{H2O}$  = density of water

$P_i$  = input horsepower to the mixing device, hp

By expressing  $p$  in watts and  $V_L$  in  $m^3$ , bubble diameter can be expressed as

$$d = \frac{1.636 \sigma^{0.6} \epsilon^{0.5}}{\gamma^{0.2} (P_i/V_L)^{0.4}} + .09 \quad (3-30)$$

3.3.3.2 Heat Exchange of the Dispersed Phase With Liquid. When the particles in a mixing vessel are just completely suspended, turbulence forces balance those due to gravity, and the heat and mass transfer rates are the same as for particles moving freely under gravity. At higher agitation levels, turbulence forces are greater than those due to gravity and can no longer be equated with them so a separate treatment, described below, must be applied. This approach, which treats turbulent forces as those which determine heat and mass transfer rates, has also been applied to fixed bodies submerged in mixing vessels. In the case of gas-liquid dispersions it is impracticable to exceed gravitational forces by mechanically induced turbulence since agitators operate poorly in gas-liquid dispersions.

Turbulence in the immediate vicinity of a liquid or particle in a dispersion affects heat and mass transfer rates between the particle and the fluid. The theory of local isotropy may be used to give information on the turbulent intensity in the small fluid volume around the particle, which results in the following expression for turbulence Reynolds number

$$N_{Re} = \rho_L^{1/3} (P_i/V_L)^{1/6} d^{2/3} / \mu_L^{1/2} \quad (3-31)$$

where

$\mu_L$  = liquid viscosity

By employing the usual functional relationship between Reynolds number and Nusselt number for heat transfer, experimenters have obtained the following expression (Equation 183, Reference 3-1)

$$\frac{h}{C_p \rho_L} (N_{Pr})^{2/3} = 0.13 \left[ \frac{(P_i/V_L) \mu_L}{\rho_L^2} \right]^{1/4} \quad (3-32)$$



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where  $N_{Pr}$  = Prandtl number, and units for the variables  $h/(C_p \rho_L)$  and  $[(\rho_L/V_L)\mu_L]/\rho_L^2)^{1/4}$  are given in m/sec. Equation 3-32 is applicable over the range  $10^{-4}$  to  $10^3$  for the parameter  $[(\rho_L/V_L)\mu_L]/\rho_L^2)^{1/4} / N_{Pr}^{2/3}$ . This equation is identical to Equation 3-9 except that liquid is used.

3.3.3.3 Bubble Diameter During Tank Fill. Equation 3-29 cannot be directly employed to determine bubble diameter during the tank fill process because the expression includes power input to a mixing device which agitates the fluid. However, substitution of Equation 3-10 into 3-29, and once again assuming that mixer efficiency is 40 percent, results in

$$d = \frac{1.134 \sigma^{0.6} \epsilon^{0.5}}{\gamma^{0.2} (\dot{m}v^2/V_L)^{0.4}} + 0.09 \quad (3-33)$$

Bubble diameter is plotted versus  $\epsilon$  and  $\dot{m}v^2$  in Figure 3-15 for saturated liquid hydrogen properties evaluated at 276 kN/m<sup>2</sup> (40 psia). The decrease in bubble diameter as  $\dot{m}v^2$  increases is expected because the increased turbulent forces will be balanced by the higher surface tension forces of smaller bubbles. Bubble diameter will increase as  $\epsilon$  increases due to the increased rate of bubble coalescence as the volume percent of entrained vapor is increased.

3.3.3.4 Vapor Entrainment During Tank Fill. Vapor holdup,  $\epsilon$  must be known before Equation 3-33 can be employed to determine bubble diameter. For a typical industrial application, the vapor inflow rate is an input variable. During tank fill, however, vapor entrainment will be caused by the interaction of liquid jet with liquid bulk as one penetrates the other. The following simplifying assumptions have been made in order to determine the quantity of entrained vapor in liquid:

1. Acceleration environment results in a nearly flat liquid-vapor interface.
2. Mixing between incoming fluid and liquid bulk is for a circular jet.
3. Mixing between re-entering jet and liquid bulk is for a circular jet. This allows for a rapid jet velocity decay and, therefore, decreased vapor penetration depth.
4. Entrained vapor flows at the average jet velocity,  $v_{LX}$ .
5. Vapor penetrates liquid to a depth,  $X$ , at which time  $v_{LX} = 10 v_b$ , where  $v_b$  = bubble rise velocity.
6. Bubble stay-time,  $t = X/v_b$ .

Figure 3-16 illustrates a mechanism for entraining vapor during the fill process. A liquid jet is introduced into the tank and entrains additional liquid as it flows through

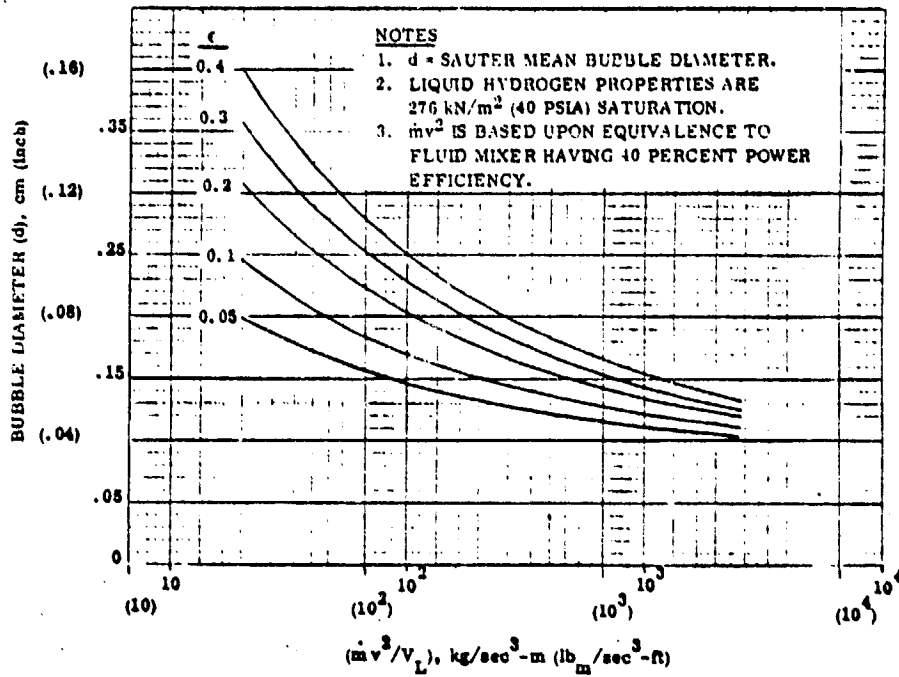


Figure 3-15. Sauter Mean Diameter of Hydrogen Vapor Dispersed in Liquid Hydrogen Versus Input Fluid Power

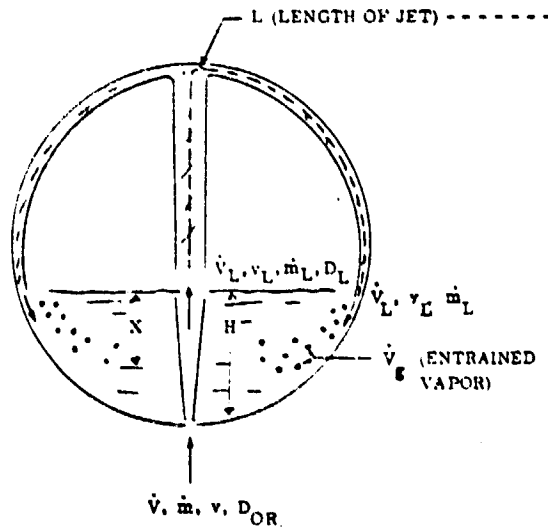


Figure 3-16. Mechanism for Entraining Vapor in Liquid During Tank Fill

the bulk toward the ullage. The jet, which has increased in size, rises about the liquid surface, traverses the ullage space, strikes the opposite end of the tank and is deflected back to the liquid. Vapor is entrained into the liquid bulk as a result of jet penetration.

Vapor entrainment in liquid can be estimated as

$$V_g = \dot{V}_g t \quad (3-34)$$

The volume rate of vapor entrainment can be approximated from the following empirical expression (Equation 18-133, Reference 3-2)

$$\dot{V}_g = 0.0316 \left( \frac{\rho_L L}{\sigma} \right)^{0.5} \left( \dot{V}_L * v_L \right) \quad (3-35)$$

$$= 7.1 \sqrt{L} \left( \dot{V}_L * v_L \right) \quad (3-36)$$

(for LH<sub>2</sub> properties evaluated at 276 kN/m<sup>2</sup> (40 psia))

where

$\dot{V}_g$  = volume rate of vapor entrainment into liquid pool

L = length of liquid jet (defined by Figure 3-16)

$\dot{V}_L$  = volume flowrate of jet re-entering bulk

$v_L$  = velocity of jet re-entering bulk

$\sigma$  = surface tension

It is reasonable to assume that  $\dot{V}_L$  and  $v_L$  remain constant as the jet travels through the ullage before returning. It is now necessary to define  $\dot{V}_L$  and  $v_L$  as a function of the entering jet properties and tank geometry.

Jet entrainment can be described as a constant momentum process. Thus  $\dot{m}v$  (entering jet momentum) =  $\dot{m}_L v_L$  (exiting jet momentum). Furthermore,  $\dot{m}v = \rho \dot{V} * v = \rho \dot{V}_L * v_L$ . Or

$$\dot{V}_L * v_L = \dot{m}v / \rho \quad (3-37)$$

Substituting Equation 3-37 into Equation 3-36

$$\dot{V}_g = 7.1 \sqrt{L} (\dot{m} * v / \rho) \quad (3-38)$$

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where  $\dot{m}$  and  $v$  are, respectively, the entering jet mass flow rate and velocity.

Equation 3-38 shows that vapor entrainment will be a function of the incoming jet momentum and the length of liquid jet. The entrainment flow rate is plotted in Figure 3-17 for a range of entering jet momentum and liquid fill conditions of the fluid management system. It is seen that the entrainment rate of vapor can be quite high.

Bubble stay-time,  $t$ , must be estimated before Equation 3-34 can be solved for  $V_g$ . This requires identifying bubble rise velocity and vapor penetration depth. The rise velocity of a spherical bubble is determined by equating bubble buoyancy to bubble drag. Or,

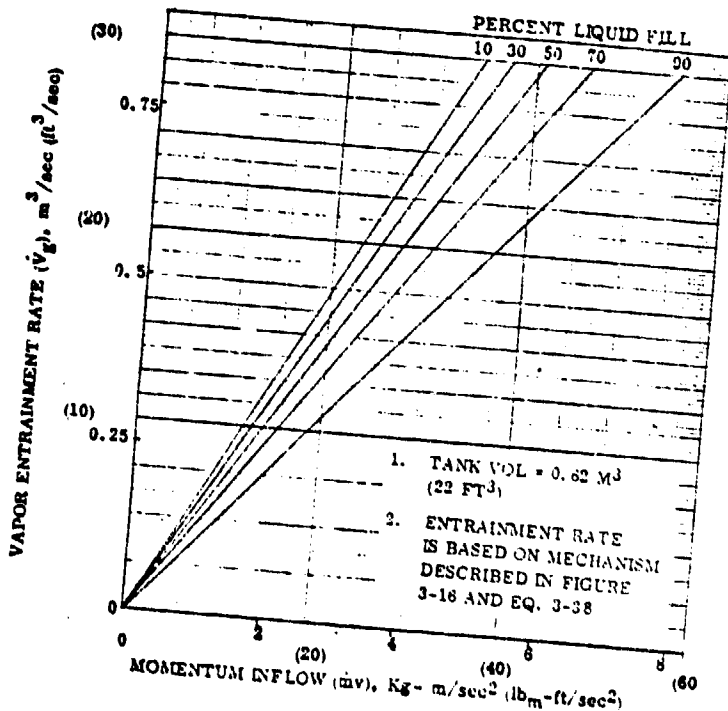
$$(\rho_L - \rho_g) V_b g = C_D A_b \rho_L \frac{v_b^2}{2} \quad (3-39)$$

but

$$V_b = \frac{4}{3} \pi \left( \frac{d}{2} \right)^3 = \frac{\pi}{6} d^3 \quad (\text{bubble volume}) \quad (3-40)$$

and

$$A_b = \frac{\pi}{4} d^2 \quad (\text{bubble cross-sectional area}) \quad (3-41)$$



Substituting (3-40) and (3-41) into (3-39) and recognizing that  $\rho_L \gg \rho_v$ , bubble rise velocity is determined as

$$v_b = \sqrt{(4/3) dg / C_D} \quad (3-42)$$

Assuming that bubble Reynolds number is greater than  $10^4$ , for which  $C_D = 0.44$ ,

$$v_b = 1.74 \sqrt{dg} \quad (3-43)$$

where

$\rho_v$  = vapor density

$g$  = gravity environment

$C_D$  = bubble drag coefficient

$v_b$  = bubble rise velocity

Figure 3-17. Vapor Entrainment Rate Into Liquid Bulk Versus Liquid Hydrogen Momentum Inflow and Percent Liquid Fill of Fluid Management System Tank

Bubble penetration depth can be determined as a function of incoming fluid conditions using the following expressions for jet velocity decay (Reference 3-4)

$$v_2/v_1 = 3.09 D_1/\Delta X \quad (3-44)$$

$$D_2 = 0.324 \Delta X \quad (3-45)$$

where

$v_1$  and  $D_1$  = jet source velocity and diameter

$v_2$  and  $D_2$  = average jet velocity and diameter at a distance,  $\Delta X$ , downstream of source

Referring to Figure 3-16, jet conditions at the liquid surface relative to the source are

$$v_L = 3.09 v D_{OR}/H \quad (3-46)$$

and

$$D_L = 0.324 H \quad (3-47)$$

Also, jet conditions at  $X$  relative to re-entering conditions at the liquid surface are

$$v_{LX} = 3.09 v_L D_L/X \quad (3-48)$$

Substituting (3-46) and (3-47) into (3-48) gives

$$v_{LX} = 3.09 v D_{OR}/X \quad (3-49)$$

Solving (3-49) for  $X$  results in

$$X = 3.09 v D_{OR}/v_{LX} \quad (3-50)$$

From assumption 6, substituting  $10 v_b = v_{LX}$  into (3-50) gives

$$X = 0.309 v D_{OR}/v_b \quad (3-51)$$

Finally, bubble stay-time can be determined by substituting (3-43) and (3-51) into  $t = X/v_b$ , which results in

$$t = 0.102 D_{OR} v (dg) \quad (3-52)$$

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Bubble stay-time is plotted in Figure 3-18 as a function of momentum inflow and acceleration environment. Note that an acceleration environment of  $10^{-1}$  g's will

- NOTES: 1. Vapor entrainment model from Figure 3-16 is assumed.  
 2. Bubble stay-time,  $t = .102 D_{OR} v/(dg)$   
 3. Bubble diameter = .41 cm (0.16 inches), selected because it represents maximum size (see Figure 3-15).

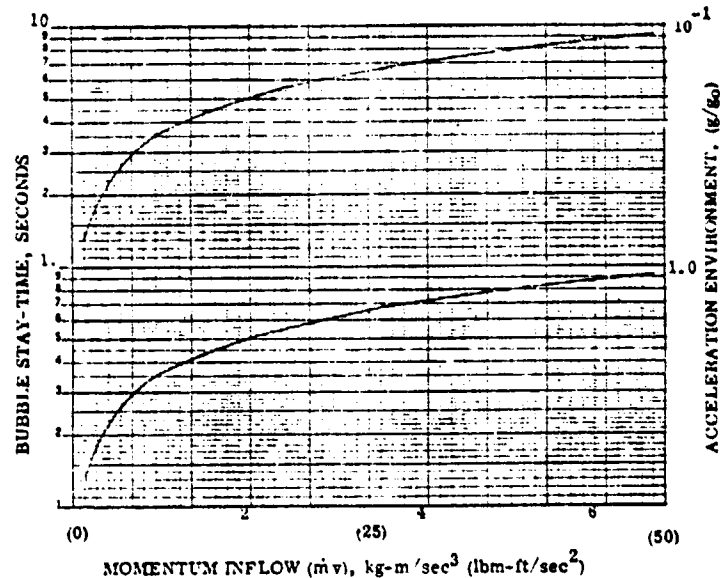


Figure 3-18. Fractional Volumetric Hold-Up of Vapor in Liquid Hydrogen as a Function of Liquid Momentum Inflow and Bubble Stay-Time for the Fluid Management System Tank

result in bubble stay-times of five to nine seconds. Figure 3-19 provides plots of  $\epsilon$  versus momentum inflow for a 50 percent liquid fill, and bubble stay-times of one, two and three seconds. These results should be used with caution, however, because the empirical data upon which Equation 3-35 is based was limited to  $\dot{V}_g/\dot{V}_L \leq 20$ . Thus an uncertainty exists for ratios in excess of 20. Nevertheless, it is obvious that a mechanism does exist for the entrainment of large quantities of vapor during the tank fill process. The ensuing discussion will show that large quantities of entrained vapor are not needed to provide a near thermal equilibrium environment during fill.

3.3.3.5 Bubble Heat Transfer During Tank Fill. Equation 3-11 can be used to determine heat transfer from the dispersed bubbles to the liquid bulk during fill by introducing the appropriate liquid properties. This equation becomes

$$\left( \frac{h}{C_P \rho_L} \right) \left( N_{Pr} \right)^{2/3} = 0.163 \left[ \frac{(\dot{m} v^2 / V_L) \mu_L}{\rho_L} \right]^{1/4} \quad (3-53)$$

where  $h$  is now the heat transfer coefficient between bubble and liquid, and  $V_L$  is the liquid volume rather than tank volume.

Introducing liquid hydrogen properties at 273 kN/m<sup>2</sup> (40 psia) saturation, and solving for h, we have

$$h = 812.1 \left( \frac{\dot{m} v^2}{V_L} \right)^{1/4}, \text{ watt/m}^2\text{-}^\circ\text{K} \quad (3-54)$$

Now, the heat transfer rate for each bubble can be expressed as

$$\frac{\dot{Q}_B}{V_B} = \frac{h A_B (T_B - T_L)}{V_B} = \frac{\pi d^2}{(\pi/6) d^3} h (T_B - T_L) = \frac{6h (T_B - T_L)}{d} \quad (3-55)$$

where

$\dot{Q}_B$  = heat transfer rate from each bubble

$V_B$  = bubble volume

$T_B$  = bubble vapor temperature

$T_L$  = liquid bulk temperature

Total heat transfer rate from the total dispersed vapor volume can be determined from Equation 3-55 by introducing the total number of vapor bubbles, n,

$$\frac{\dot{Q}_T}{V_g} = \frac{n \dot{Q}_B}{n V_B} = \frac{\dot{Q}_B}{V_B} = \frac{6h (T_B - T_L)}{d} \quad (3-56)$$

Combining Equations 3-54 and 3-56 gives

$$\frac{\dot{Q}_T}{V_g} = 4872 (\dot{m} v^2 / V_L)^{1/4} (T_B - T_L) / d \quad (3-57)$$

Finally,

$$\dot{Q}_T = 4872 V_g (\dot{m} v^2 / V_L)^{1/4} (T_B - T_L) / d \quad (3-58)$$

where

$\dot{Q}_T$  = total heat transfer rate from the entrained vapor

$V_g$  = total vapor volume entrained in liquid





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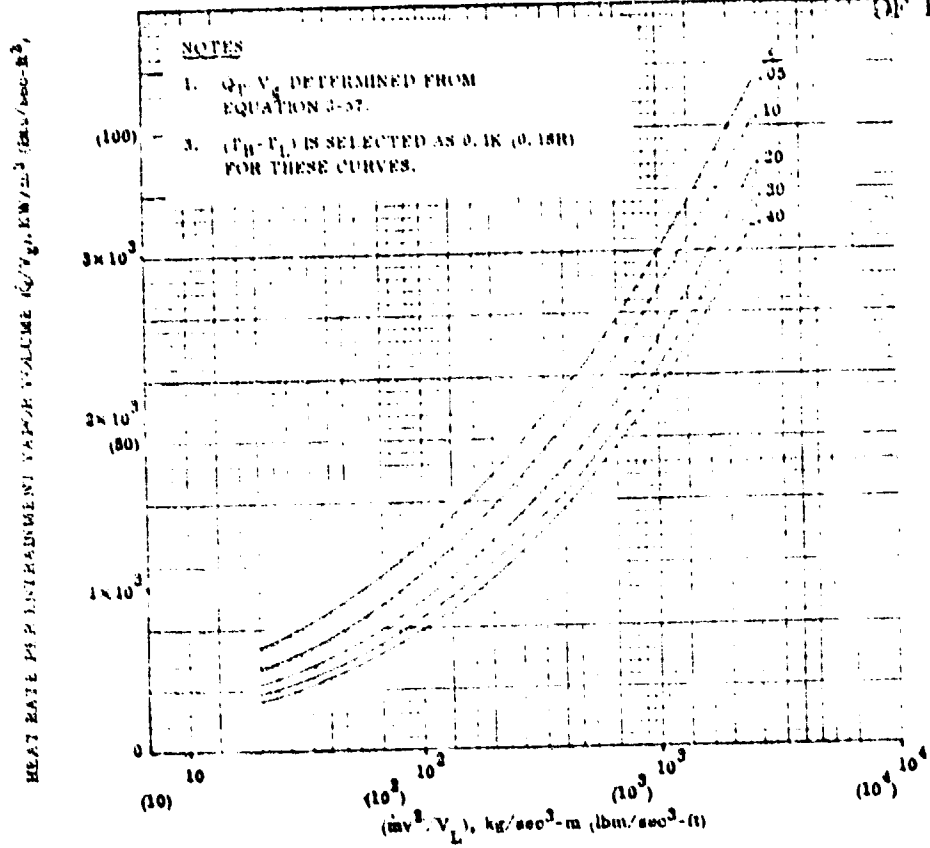


Figure 3-20. Heat Rate From Entrainment Vapor to Liquid Hydrogen

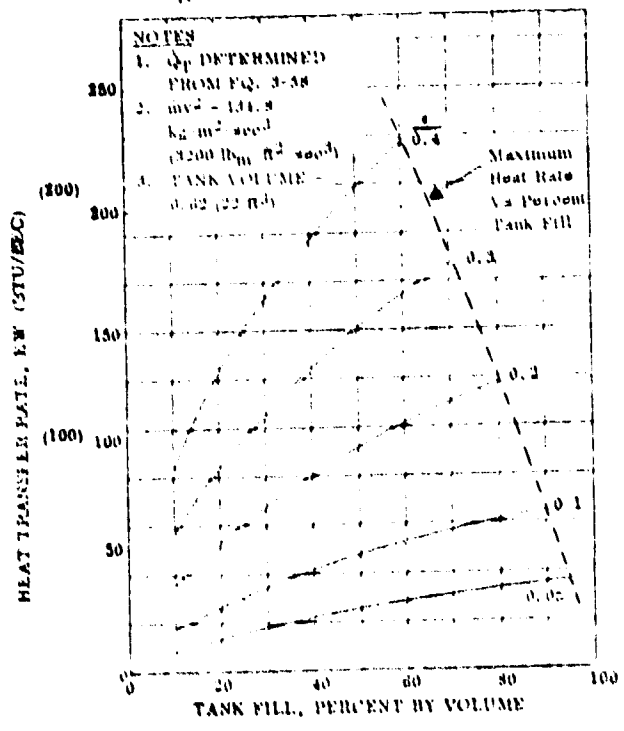


Figure 3-21. Entrained Vapor Total Heat to Liquid Hydrogen for Fluid Management System Tank

the fluid phase temperature difference. Consequently, more time is required to achieve equilibrium between saturated vapor bubbles and liquid than between superheated vapor bubbles (at the same pressure) and liquid. It is, therefore, more conservative to assume that the vapor is initially saturated. The curves of Figure 3-22 are solutions of Equation 3-59 for  $\dot{Q}$  versus percent tank fill and  $(T_B - T_L)$ , where  $T_B$  is vapor temperature saturated at ullage pressure. A comparison of Figures 3-21 and 3-22 indicates that equilibrium can be achieved in less than one second.

3.3.3.7 Summary. Development of the equations describing vapor dispersal has shown that vapor holdup is a key variable in determining propellant thermodynamic conditions during tank fill. A simplified model for calculating vapor holdup has been described which results indicate that

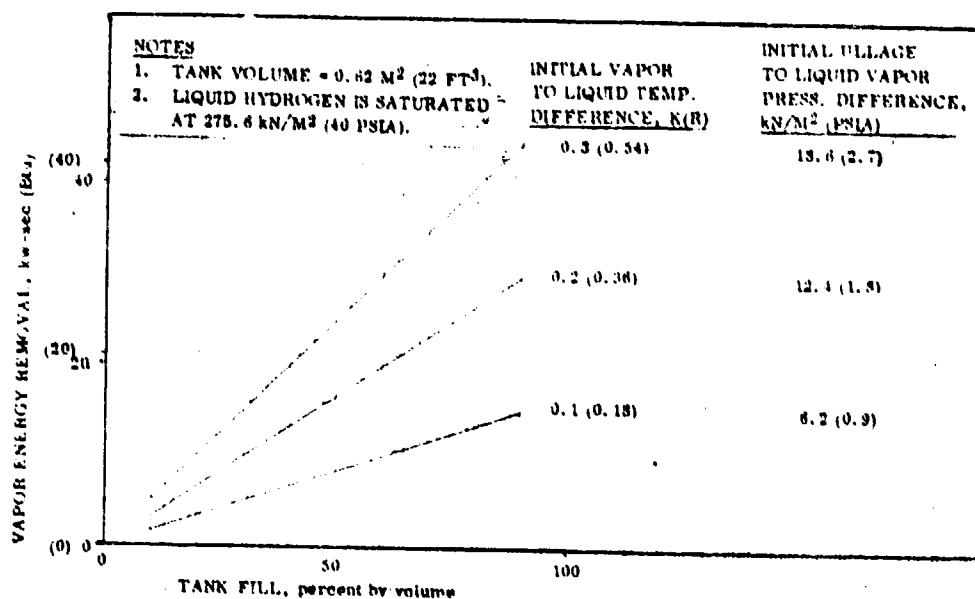


Figure 3-22. Vapor Energy Removal Required to Achieve Thermal Equilibrium With Liquid Hydrogen in Fluid Management System Tank

considerable vapor entrainment will occur during fill. In spite of the fact that the model is more applicable to a one-g environment than a very low-g environment, the belief is that considerable vapor entrainment in liquid will occur as a result of liquid inflow conditions. Furthermore, the role of vapor buoyancy is such that bubble stay-times will be substantially greater in zero-g than in one-g.

This leads to the conclusion that thermal equilibrium is more likely to occur in zero-g than at normal gravity for the same propellant inflow conditions.

Near-thermal equilibrium conditions should be experienced during low-g tank fill for an acceptable range of liquid inflow velocity and flowrate. The results of Figure 3-21 show that vapor to liquid heat transfer rates in excess of 63.3 kW (60 Btu/sec) can be expected to occur for inflow conditions of 0.23 kg/sec (0.5 lb/sec) and 24.4 m/sec (80 ft/sec), which is equivalent to a fluid power input of 134.3 kg-m<sup>2</sup>/sec<sup>3</sup> (3200 lb<sub>m</sub>-ft<sup>2</sup>/sec<sup>3</sup>). This ullage chilling condition is sufficient to achieve thermal equilibrium in a fraction of a second. Although specific calculations have not been made for lower inflow conditions, a review of Figure 3-20 indicates that an incoming flowrate of 0.045 kg/sec (0.1 lb<sub>m</sub>/sec) will still provide near-thermal equilibrium fill conditions.

### 3.4 VAPOR REMOVAL FROM SCREEN DEVICE

At the completion of propellant fill, vapor will be trapped within the screen device because procedures to prevent this are not known. Vapor can, however, be easily removed from the channels. A condensation process will be employed for vapor removal.

Vapor condensation begins when subcooling is initiated, which will occur when the tank is pressurized with helium from 344.5 kN/m<sup>2</sup> (50 psia) to 413.4 kN/m<sup>2</sup> (60 psia). There is no question that the trapped vapor will eventually condense. The only uncertainty is the time required to complete the process.

3.4.1 BUBBLE CONDENSATION MODEL. An analysis was conducted to determine the time required for hydrogen bubbles to collapse.

To determine bubble condensation times, a bubble collapse program was written for the HP 9830 computer. The program calculates the collapse time of bubbles of different initial radii for various tank pressure differentials. The equations used in the program are based on the theoretical investigation of Florschuetz and Chao, presented in Reference 3-5. The analysis is applicable to any pressurant as long as noncondensable gases do not exist in the vapor bubble. The collapse rate is controlled, primarily, by heat transfer rather than by liquid inertia or surface tension. The analysis for heat transfer controlled bubble-collapse is based on calculating the conduction from the bubble surface into the liquid. The general equation is

$$\gamma = 1 - \sqrt{\tau_H} \quad (3-61)$$

where

$$\gamma \text{ (nondimensional radius)} = r/r_0$$

$$\tau_H \text{ (nondimensional time)} = \frac{4}{\pi} \text{Ja}^2 \frac{\alpha t}{r_0^2}$$

$$\text{Ja} = \text{Jakob number, } \rho_L C_L \Delta T / (\rho_V h_{fg})$$

$$\alpha = \text{thermal diffusivity of liquid}$$

$$t = \text{time}$$

$$r = \text{bubble radius}$$

$$r_0 = \text{initial bubble radius at } t = 0$$

$$\rho_L = \text{liquid density}$$

$$\rho_V = \text{vapor density}$$

$$\Delta T = \text{saturation temperature at final system pressure minus liquid temperature}$$

$$C_L = \text{liquid specific heat capacity}$$

$$h_{fg} = \text{heat of evaporation}$$

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The numerical solution of the equation is presented in Figure 3-23, where nondimensional radius is plotted versus nondimensional time.

Results of Equation 3-61 for hydrogen are presented in Figure 3-24. The curves plotted in Figure 3-24 are the times required for complete bubble collapse ( $\gamma = 0$ ) for a given

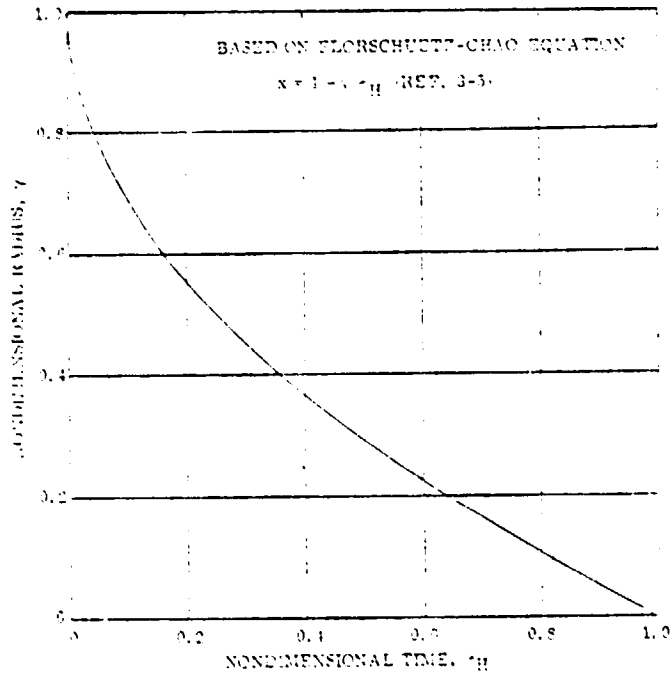


Figure 3-23. Heat Transfer Controlled Bubble Collapse

initial tank pressure, bubble radius, and tank pressure differential. It is noted that a bubble of 2.54 cm (1.0 inch) initial radius will collapse within 210 seconds. This dimension is substantially greater than that of the 0.64 cm (0.25 in) screen channel width, which is believed to be the dimension for heat transfer controlled bubble collapse. The condensation of vapor contained within the screen channels will be completed within minutes after the system is pressurized. It is unlikely that this time to complete vapor condensation will be critical to system performance. Guidelines have not been established to show when pure liquid must be available within the screen channel for propellant transfer, nor has this study identified a need for early availability of pure liquid.

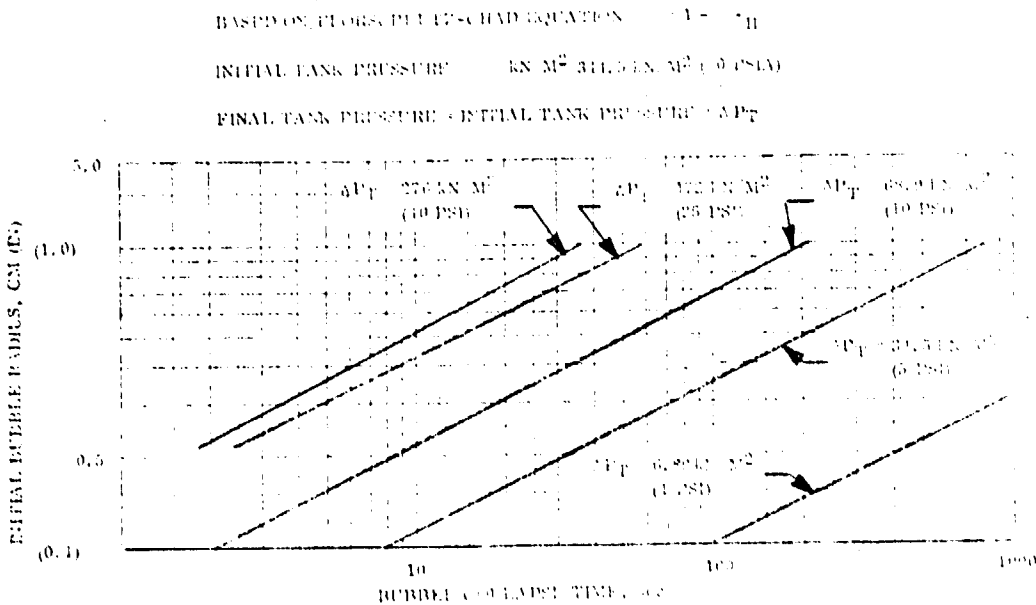


Figure 3-24. Collapse Time for Spherical Bubbles in Liquid Hydrogen

3.4.2 SUMMARY. Pressurization is an acceptable means of removing vapor from the screen channels, if the resultant condensation process is completed within approximately four to five minutes.

### 3.5 TANK REFILL

Fluid management systems will require propellant refill capability in order to extend their useful life in space. The refill procedure must be capable of handling all operations from topping-off a tank with liquid, to a complete refill. Such capability is mandatory not only because propellant resupply can be a costly operation, but also because flexibility with orbital procedures must be maintained.

There are similarities between the initial fill of a fluid management system, and subsequent refills. Each requires pre-conditioning before liquid flow to the tank is initiated. Once pre-conditioning is complete, propellant refill will be identical to initial propellant fill, which is discussed in detail in Section 3.3. This section will deal only with the specific operations needed to pre-condition the fluid management system.

Pre-conditioning will depend solely on the use of autogenous pressurization and helium pressurization. A helium pressurization system will require a pre-conditioning procedure because much of the helium must be vented prior to beginning liquid flow. The procedure may be complicated if helium enters the screen volume. Potential problems and solutions are discussed in Section 3.5.2.

Pre-conditioning will not be required for an autogenous pressurization system. Refill can be successfully implemented, as will be shown in Section 3.5.1, by flowing liquid hydrogen into the tank.

3.5.1 TANK REFILL (AUTOGENOUS). Refill of a system containing liquid and its own vapor as a pressurant is a straightforward operation. Liquid must be introduced at the correct vapor pressure, and fluid inflow power must be sufficiently high to assure near-thermodynamic equilibrium conditions. A determination has already been made that near-thermal equilibrium conditions will exist during fill. To be considered is the relationship between initial and final tank fluid conditions, and entering liquid vapor pressure for a thermal equilibrium process. The inlet vapor pressure can be determined on the basis of the following First Law analysis

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$$dE_L + dE_g = hdm \quad (3-62)$$

$$dE_L = (u_L m_L)_2 - (u_L m_L)_1 = \text{change in liquid energy} \quad (3-63)$$

$$dE_g = (u_g m_g)_2 - (u_g m_g)_1 = \text{change in vapor energy} \quad (3-64)$$

$$hdm = h (m_{T2} - m_{T1}) = \text{total energy of entering liquid} \quad (3-65)$$

$$m_T = m_L + m_g$$

Making the assumptions that initial and final liquid temperatures in the tank are equal, and that phase equilibrium exists,  $u_{L2} = u_{L1} = u_L$  and  $u_{g2} = u_{g1} = u_g$ . Combining these assumptions with Equations 3-62 through 3-65 results in

$$u_L (m_{L2} - m_{L1}) + u_g (m_{g2} - m_{g1}) = h (m_{T2} - m_{T1}) \quad (3-66)$$

Now, total mass within the tank can be expressed as

$$m_T = V_T \rho_L - (\alpha - 1) m_g \quad (3-67)$$

and

$$m_{T2} - m_{T1} = (\alpha - 1) (m_{g1} - m_{g2}) \quad (3-68)$$

where

$$m_T = m_L + m_g = \text{total mass of fluid in tank}$$

$$\rho_L = \text{liquid density}$$

$$\rho_g = \text{vapor density}$$

$$\alpha = \rho_L / \rho_g$$

Combining Equations 3-66 through 3-68 we find that

$$u_L (m_{L2} - m_{L1}) + u_g (m_{g2} - m_{g1}) = h (\alpha - 1) (m_{g1} - m_{g2}) \quad (3-69)$$

Also, by adding and subtracting  $u_L m_{g1}$  and  $u_L m_{g2}$  to the left side of Equation 3-69

$$u_L (m_{T2} - m_{T1}) + u_{ev} (m_{g2} - m_{g1}) = h (\alpha - 1) (m_{g1} - m_{g2}) \quad (3-70)$$

where

$$u_{ev} = (u_g - u_L) = \text{internal energy of evaporation}$$

Combining Equations 3-68 and 3-70

$$u_L (\alpha - 1) (m_{g1} - m_{g2}) + u_{ev} (m_{g2} - m_{g1}) = h (\alpha - 1) (m_{g2} - m_{g1}) \quad (3-71)$$

Solving for the entering liquid enthalpy,  $h$

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$$h = u_L - \frac{u_{ev}}{\alpha - 1}$$

(3-72)

Results are plotted for liquid hydrogen in Figure 3-25 as incoming liquid vapor pressure versus final liquid vapor pressure in the tank. It is interesting to note that this vapor pressure relationship is independent of liquid mass in the tank at the start of refill.

The above results will simplify refill because the same entering liquid vapor pressure will be required regardless of initial liquid fill. For example, an entering liquid vapor pressure of  $235.6 \text{ kN/m}^2$  (34.2 psia) will be needed to maintain the tank propellant vapor pressure constant at  $344.5 \text{ kN/m}^2$  (50 psia).

**3.5.2 TANK REFILL (HELIUM PRESSURANT).** Refill is complicated by the need to vent helium before propellants can be introduced. If helium is not vented, its partial pressure will become excessive as a liquid fill approaches 100 percent. This is illustrated in Figure 3-26, which shows the influence of initial liquid volume on final tank pressure, following refill to a 90 percent liquid volume. An initial helium partial pressure of  $68.9 \text{ kN/m}^2$  (10 psia) will increase tank

pressure to  $964.6 \text{ kN/m}^2$  (140 psia) as liquid is tanked from 10 percent fill to a 90 percent fill condition.

CONDITIONS

1. TANK VOLUME =  $0.02 \text{ m}^3$  (0.71  $\text{ft}^3$ ).
2. LIQUID VAPOR PRESSURE IN TANK REMAINS CONSTANT.
3. THERMODYNAMIC EQUILIBRIUM CONDITIONS EXIST AT  $\text{LH}_2$  TEMPERATURE.
4. NO HELIUM PRESSURANT.
5. DATA IS APPLICABLE TO ANY INITIAL LIQUID FILL VOLUME.

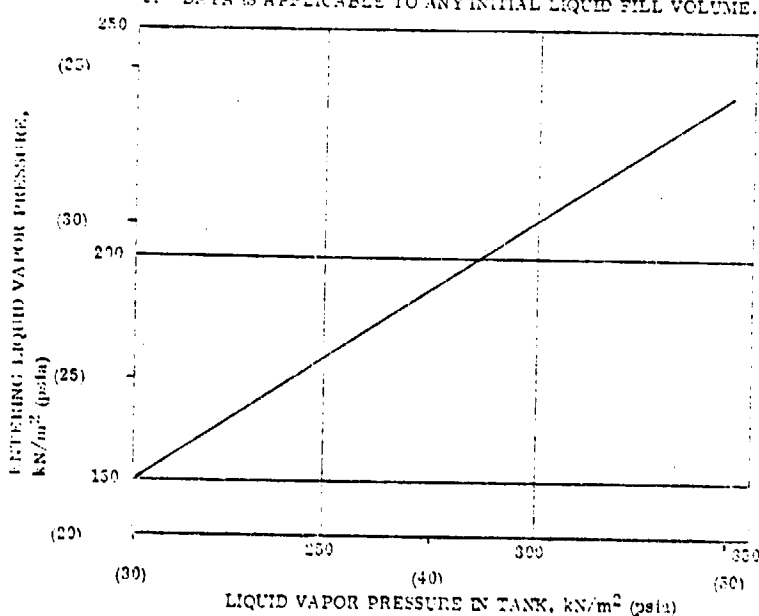
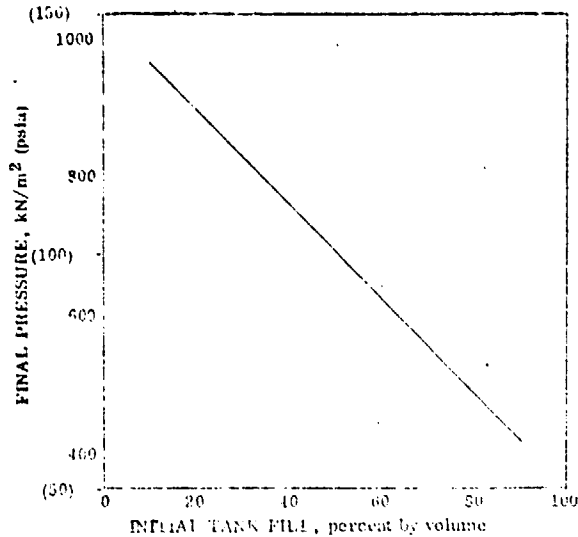


Figure 3-25. Entering Liquid Hydrogen Vapor Pressure Required to Maintain a Constant Liquid Vapor Pressure in Tank During Fill

**3.5.2.1 Tank Helium Vent.** There are two methods for venting helium from the tank. The first is the transfer of liquid to a storage tank through the acquisition device until unusable residuals remain, and then venting the ullage until pressure has been reduced to a predetermined level. The second method simply initiates ullage venting without first expelling liquid. The mass of vapor vented, to achieve a given helium pressure following refill, can be determined from Figure 3-27. It can be seen that vent mass increases with a decrease in liquid volume

**CONDITIONS**

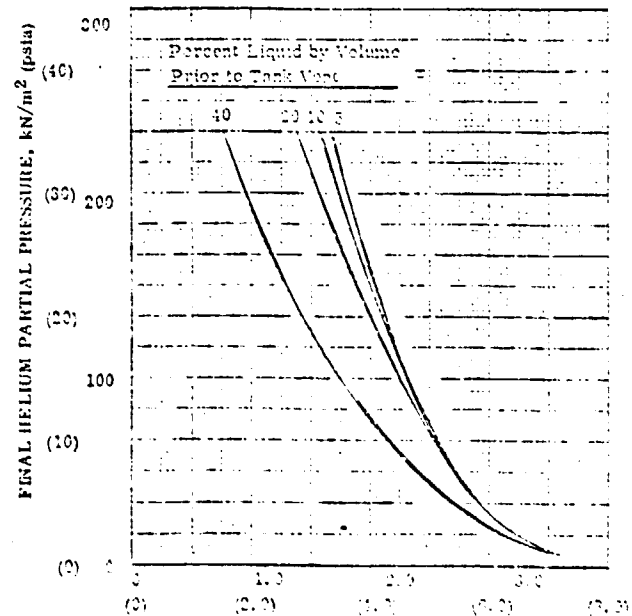
1. TANK VOLUME = 0.62 M<sup>3</sup> (22 FT<sup>3</sup>)
2. INITIAL LIQUID VAPOR PRESSURE = 344.5 KN/M<sup>2</sup> (50 PSIA)
3. FINAL LIQUID VAPOR PRESSURE = 311.5 KN/M<sup>2</sup> (45 PSIA)
4. INITIAL HELIUM PARTIAL PRESSURE = 68.9 KN/M<sup>2</sup> (10 PSIA)
5. FINAL HELIUM VOLUME = 90% BY TANK VOLUME.
6. NO HELIUM IS VENTED DURING TANK FILL.



**Figure 3-26. Final Fluid Management System Pressure for Propellant Refill Without Prior Helium Vent**

**CONDITIONS**

1. TANK VOLUME = 0.62 M<sup>3</sup> (22 FT<sup>3</sup>).
2. INITIAL LIQUID VAPOR PRESSURE = 344.5 KN/M<sup>2</sup> (50 PSIA).
3. INITIAL HELIUM PARTIAL PRESSURE = 68.9 KN/M<sup>2</sup> (10 PSIA).
4. FINAL HELIUM PARTIAL PRESSURE IS BASED UPON 1% RESIDUE VOLUME FOLLOWING TANK REFILL.
5. VAPOR ONLY IS VENTED.



**Figure 3-27. Hydrogen Vent Mass Required During System Blowdown to Provide Indicated Helium Partial Pressure Following Propellant Refill**

at the start of venting. Figure 3-28 gives the tank pressure at the end of venting as a function of helium pressure following refill. This figure shows that the tank must be vented to lower pressure levels as the initial liquid volume is reduced.

The data of Figures 3-27 and 3-28 appear to indicate that liquid should not be transferred before venting. These results, however, ignore the possibility of two-phase venting in a low or zero-g environment. The likelihood of venting liquid can be reduced by decreasing vent flowrate. It is judged that an excessively long vent duration is required before venting can be conducted in a surface tension dominated environment.

The preferred method for venting helium is to first transfer the propellant to a storage tank. Two-phase venting, for this condition, is minimized because any propellant mass lost overboard will be insignificant.

The most convenient method for transferring propellant appears to be the use of the supply tank, which will service the system, as a fluid receiver. The following



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CONDITIONS

1. TANK VOLUME =  $0.62 \text{ M}^3$  ( $22 \text{ FT}^3$ ).
2. INITIAL LIQUID VAPOR PRESSURE =  $344.5 \text{ kN/M}^2$  (50 PSIA).
3. INITIAL HELIUM PARTIAL PRESSURE =  $68.9 \text{ kN/M}^2$  (10 PSIA).
4. FINAL HELIUM PRESSURE IS BASED UPON 10% ULLAGE VOLUME FOLLOWING TANK REFILL.
5. VAPOR ONLY IS VENTED.

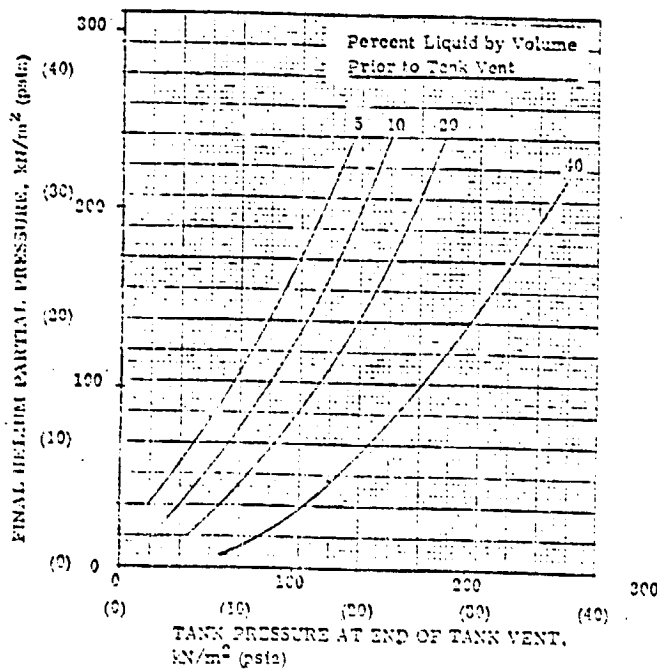


Figure 3-28. Management System Pressure at End of Vent Required to Provide Indicated Helium Partial Pressure Following Propellant Refill

below  $68.9 \text{ kN/m}^2$  (10 psia), which satisfies refill criteria. Unfortunately, this procedure does not address the potential problem of helium entry to the screen channel device, which would violate the primary system requirement of providing pure liquid flow to other systems.

There is a possibility that liquid hydrogen evaporation during tank blowdown will cause helium penetration of the screen volume. All precautions must be taken to prevent helium entering because it cannot be expelled. Thus, the primary requirement, of providing pure liquid flow to other systems, could be violated. The following describes the most likely mechanism for the flow of helium into the screen device:

1. During tank blowdown to  $41.3 \text{ kN/m}^2$  (6.0 psia), the residual liquid will boil as its vapor pressure decays from  $344.5 \text{ kN/m}^2$  (50 psia) to  $34.4 \text{ kN/m}^2$  (5 psia).
2. The boiling process will result in considerable vapor generation within the screen volume, which will expel much of the contained liquid.

procedure will be used:

1. Chardown the propellant transfer line connecting the system and supply propellant tank.
2. Transfer all usable propellant from the system to the supply propellant tank. Liquid residual will be less than five percent by volume.

The system will be ready for helium expulsion at the completion of Step 2. Helium will be expelled by performing a tank blowdown to reduced pressures.

The above will not overly complicate a refill procedure and is preferable to introducing an extended vent duration coupled with the uncertainty of two-phase venting.

Figure 3-28 indicates that system pre-conditioning will be satisfied once a tank blowdown is performed to reduce pressure below  $41.3 \text{ kN/m}^2$  (6.0 psia). According to the figure, refill to a 90 percent level will maintain final helium partial pressure

3. If the liquid film on the screen evaporates at a rate greater than can be made up by wicking from the bulk liquid, screen dry-out will occur.
4. Loss of the protective film will increase the possibility of helium diffusion into the pure hydrogen vapor environment within the screen volume.

The potentially adverse effects of helium within the screen device demand that helium entry be prevented. The most direct method of preventing this is to expel all available helium prior to initiating refill. Two approaches considered were: system evacuation to a vacuum and, a tank purge. System evacuation was eliminated from detailed consideration because of propellant freezing. Liquid hydrogen will freeze at pressures below  $6.89 \text{ kN/m}^2$  (1.0 psia). The potential for liquid freezing within the vent system cannot be ignored as tank pressure is reduced to a vacuum. Consequently, a tank purge procedure was adopted.

3.5.2.2 Helium Dilution by Tank Charge and Vent. Helium can be diluted to concentrations which will not interfere with the screen device's functioning by repeated charge and vent cycles with gaseous propellant. A helium bubble, no more than 0.25 cm (0.1 in) long, in each of the four channels represents a total volume that should not interfere with the effective functioning of the device. This is equivalent to a total helium mass of 0.0013 kg (0.0029  $\text{lb}_m$ ). The amounts of gaseous hydrogen expended to achieve this total helium mass, was based on the following procedure:

1. After transfer line chilldown and liquid transferral, the tank will be vented to  $6.89 \text{ kN/m}^2$  (1.0 psia) and 13.9K (25R). Under these conditions 0.008 kg (0.0183  $\text{lb}_m$ ) of helium remains in the tank.
2. The tank will be charged with gaseous hydrogen to a predetermined pressure, then vented back down to  $6.89 \text{ kN/m}^2$  (1.0 psia).
3. Step 2 will be repeated until no more than 0.0013 kg (0.0029  $\text{lb}_m$ ) of helium remains.

Figure 3-29 shows the results of charging to  $68.9 \text{ kN/m}^2$  (10 psia) and to  $137.9 \text{ kN/m}^2$  (20 psia). Helium mass is reduced to an acceptable level with two charge and vent cycles, and a consumption of 0.25 kg (0.56  $\text{lb}_m$ ) and 0.30 kg (0.66  $\text{lb}_m$ ) respectively. This figure indicates that sufficient helium dilution can be achieved in a relatively short period of time, with only a small loss of gaseous hydrogen. The cycle that charges the tank to  $68.9 \text{ kN/m}^2$  (10 psia) is selected because the helium dilution process is more effective.

3.5.3 REFILL OF AN EVACUATED TANK. Propellant tank refill will be initiated upon completion of the tank charge and vent procedure. Liquid will be introduced at a relatively high vapor pressure due to initial conditions of an evacuated tank at liquid hydrogen temperature. Figure 3-11 shows that the entering liquid vapor pressure must exceed  $289 \text{ kN/m}^2$  (42 psia) if a final vapor pressure of  $344.5 \text{ kN/m}^2$  (50 psia) is to be

- NOTES: 1. INITIAL CONDITIONS ARE: 63.9 kN/m<sup>2</sup> (10 psia), 13.9 K (25F) AND 0.009 KG (0.0183 LBM) HELIUM  
 2. MAXIMUM ALLOWABLE HELIUM RESIDUAL IS 0.0013 KG (0.0029 LBM)  
 3. CURVE 1 - CHARGE TO 63.9 kN/m<sup>2</sup> (10 psia), VENT TO 6.39 kN/m<sup>2</sup> (1.0 psia), GH<sub>2</sub> ENTERS AT 76.9 K (137R)  
 4. CURVE 2 - CHARGE TO 137.9 kN/m<sup>2</sup> (20 psia), VENT TO 6.39 kN/m<sup>2</sup> (1.0 psia), GN<sub>2</sub> ENTERS AT 21.4 K (170R)

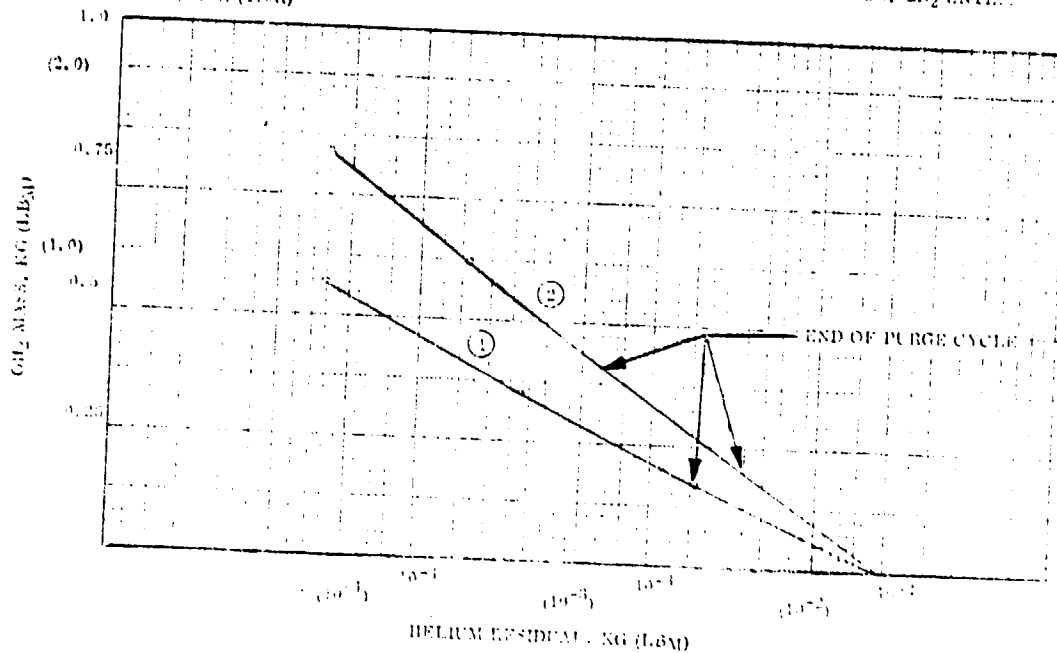


Figure 3-29. Fluid Management System Helium Residual Following GH<sub>2</sub> Purge

attained. As discussed in Section 3.3 near-thermal equilibrium conditions will be experienced during refill for an acceptable range of liquid inflow velocity and mass flowrate. The discussion of Section 3.3 further indicates that entering flowrates as low as 0.045 kg/sec (0.1 lb<sub>m</sub>/sec) will provide near-equilibrium conditions during refill.

3.5.4 VAPOR REMOVAL FROM SCREEN DEVICE FOLLOWING REFILL. At the completion of propellant refill, vapor will be trapped within the screen device. The process of vapor removal following propellant tank fill was discussed in Section 3.4, and the selected procedure is equally applicable to conditions following refill. The procedure will be to pressurize the propellant tank to 413.4 kN/m<sup>2</sup> (60 psia) with helium, which will subcool the liquid by 63.9 kN/m<sup>2</sup> (10 psid). It was indicated in Section 3.4 that trapped vapor will be condensed in approximately 210 seconds. This same time is applicable to the post-refill condensation period.

3.5.5 SUMMARY. Fluid management system refill is identical to the initial fill once pre-conditioning is complete. For the case of initial tank fill, pre-conditioning includes the prechill and chill process. For a system with autogenous pressurization no pre-conditioning will be required, and it will not be necessary to transfer propellant to a resupply tank prior to refill. Pre-conditioning will be relatively complicated for

a helium pressurization system, however, because much of the helium must be vented before refill is begun. The selected procedure for this case will be:

1. Transfer all usable propellant to the resupply tank.
2. Vent the tank down to  $6.89 \text{ kN/m}^2$  (1.0 psia).
3. Conduct two cycles where the tank is charged with hydrogen vapor to  $68.9 \text{ kN/m}^2$  (10 psia) and vented down to  $6.89 \text{ kN/m}^2$  (1.0 psia). A total of 0.25 kg (0.56  $\text{lb}_m$ ) hydrogen vapor will be consumed in the process of diluting the helium to an acceptably low level.

The tank is now ready for liquid refill.

Liquid refill for the system with autogenous pressurization will be conducted with liquid entering at a vapor pressure of  $235.6 \text{ kN/m}^2$  (34.2 psia). This inflow condition will provide a final vapor pressure of  $344.5 \text{ kN/m}^2$  (50 psia). It was estimated in Section 3.3, that near-thermal equilibrium condition will be maintained during initial fill, at liquid flowrates as low as  $0.045 \text{ kg/sec}$  ( $0.1 \text{ lb}_m/\text{sec}$ ). Thermal equilibrium for liquid refill is expected at the same flowrate condition.

Liquid refill conditions for the system with helium pressurization will be identical to those described above, except that entering liquid vapor pressure will be approximately  $289 \text{ kN/m}^2$  (42 psia). Vapor will be trapped within the screen device at the completion of refill. This vapor will be condensed within approximately 210 seconds after the system is pressurized to its operational level of  $413.4 \text{ kN/m}^2$  (60 psia).

# 4

## FILLING ANALYSIS FOR SELECTED CONCEPT (EARTH STORABLES)

The same fluid management system described in Section 3 was evaluated for filling with  $N_2O_4$  under orbital conditions. This system configuration reflects design requirements imposed by the problems of liquid hydrogen storage in orbit for long durations. There are design problems peculiar to the storage of  $N_2O_4$ . But these problems, in general, are related to material compatibility and, therefore, are beyond the scope of this contract.

System fill with  $N_2O_4$  in an orbital environment poses fewer problems than liquid hydrogen because of differences between an earth storable and cryogen system; temperature environment and operating pressure level. System temperatures will not vary significantly from  $N_2O_4$  temperatures at any time during a fill or refill operation. Consequently, neither prechill nor chill will be required to precede tank fill.

The management system operating pressure of  $2067 \text{ kN/m}^2$  (300 psia) is considerably higher than for liquid hydrogen. This higher pressure level will simplify tank fill because fill pressures will be maintained below the normal operating levels without difficulty. This is illustrated in Figure 4-1 which gives the maximum tank pressure as a function of percent liquid fill. Isentropic compression of the ullage is assumed, which means that heat exchange with the liquid and tank walls is assumed to be zero. Note that 90 percent fill can be effected without exceeding  $1378 \text{ kN/m}^2$  (200 psia) pressure. Unlike cryogenics this extremely conservative approach can be used because results are acceptably low.

One problem that cryogenics and earth storables have in common is that an undefined (or poorly defined) liquid-vapor distribution will exist during low-gravity tank fill. This inability to determine vapor location will make it difficult both to assure that no liquid is vented during tank fill, and to assure that vapor is not trapped within the screen channel device.

A second problem common to an earth storable and cryogenic management system is the need to vent helium before propellant refill can be initiated. Helium venting must be conducted with care under orbital conditions because liquid may also be vented. In addition, helium may enter the screen device during this period, unless precautions are taken.

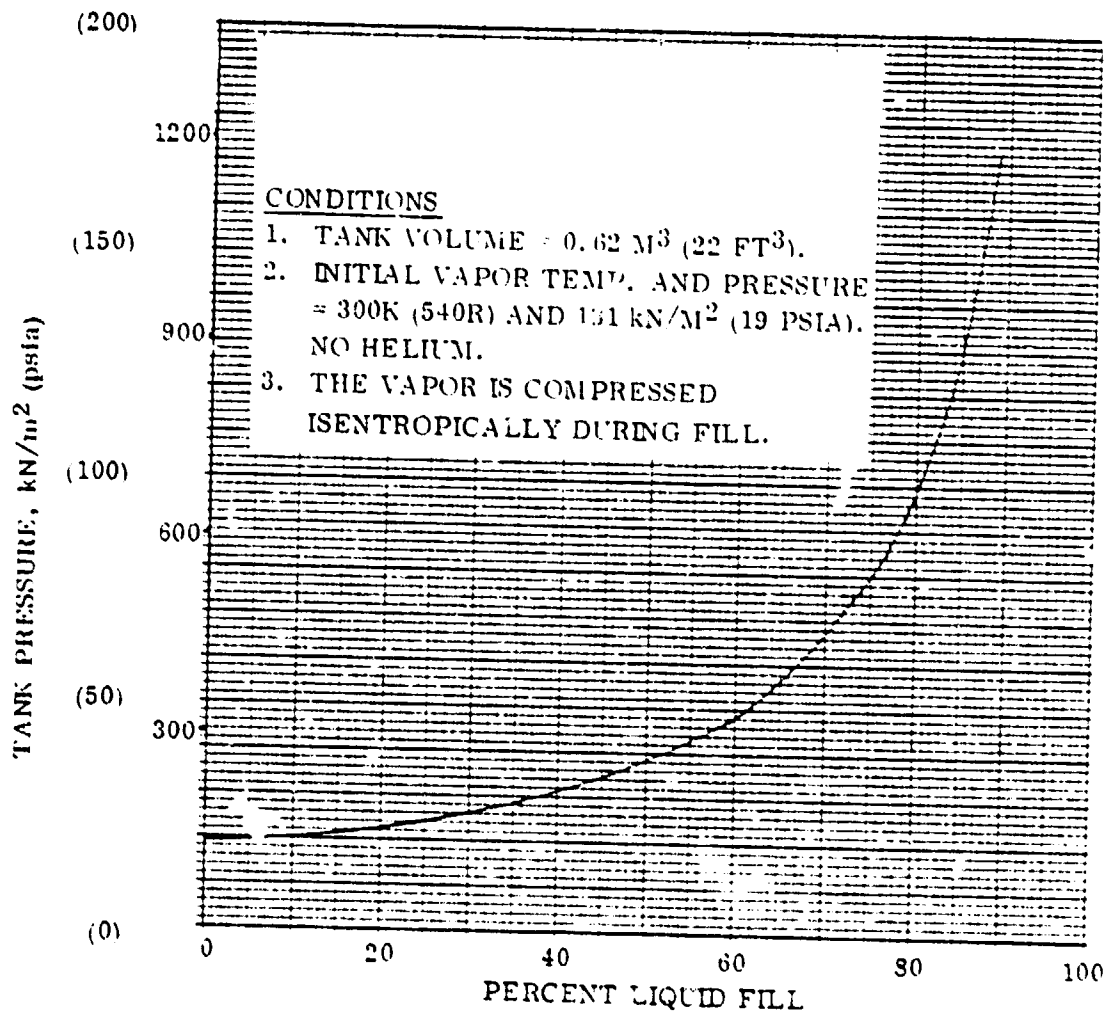


Figure 4-1. Maximum Management System Pressure During Fill With N<sub>2</sub>O<sub>4</sub>

#### 4.1 THERMAL EQUILIBRIUM TANK FILL

The single requirement for tank fill is that acceptably low pressures be maintained during the process. Thermal equilibrium fill represents the minimum propellant tank pressure condition and, therefore, should be maintained during fill. The thermal equilibrium tank fill relationship derived for liquid hydrogen (Equation 3-28) applies as well to N<sub>2</sub>O<sub>4</sub>. This equation is given below.

$$h_L = u_{g2} m_{g2} + u_{L2} m_{L2} + \Delta u_w m_{in} m_{L2} \quad (4-1)$$

where

- $h_L$  = enthalpy of entering liquid final vapor
- $u_{g2} m_{g2}$  = total internal energy
- $m_{L2}$  = final liquid mass
- $\Delta u_W m_W$  = management system energy change during tank fill
- $u_{L2}$  = final liquid internal energy

It is possible to relate  $h_L$ ,  $u_{L2}$  and  $u_{g2}$  to liquid vapor pressure and temperature under thermal equilibrium conditions. Thus  $u_{L2}$  and  $u_{g2}$  are known once final vapor pressure is specified.  $\Delta u_W$  can also be determined, when initial and final temperatures are given. Finally,  $h_L$  (which is a function of entering liquid vapor pressure) can be determined for a desired liquid fill condition.

Equation 4-1 is summarized for  $N_2O_4$  in Figure 4-2 which gives entering liquid vapor pressure as a function of initial tank temperature and final tanked liquid vapor pressure. It is interesting to note that the effect of initial wall temperature upon final tank pressure is negligible. This data illustrates that the  $N_2O_4$  thermal mass will so dominate tank

CONDITIONS

1. TANK VOLUME = 0.62 m<sup>3</sup> (22 ft<sup>3</sup>)
2. TANK MASS = 13.61 kg (30 lb<sub>m</sub>), 2219 ALUMINUM
3. TANK PRESSURE IS 0.0 kN/m<sup>2</sup> (0.0 PSIA) PRIOR TO FILL
4. NO VENTING DURING FILL

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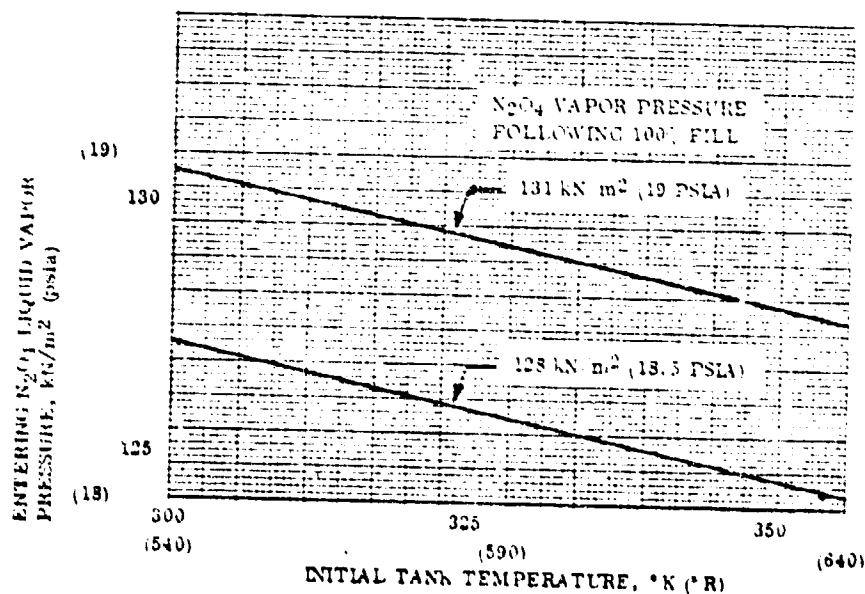


Figure 4-2. Final Storage Tank Pressures for  $N_2O_4$  Thermodynamic Equilibrium Fill Process

fill that the tendency will be for final tank pressures to approach that of the incoming liquid vapor pressure.

The same fill technique, that of liquid spray into the tank, will be employed for  $N_2O_4$  as for liquid hydrogen. This approach will assure that  $N_2O_4$  will be at least as close to thermal equilibrium as liquid hydrogen, at the same power inflow conditions. This assessment is made on the basis that the liquid-to-vapor thermal mass ratio is six times greater for  $N_2O_4$  than for hydrogen. Thermal equilibrium is more readily achieved for fluids having high thermal mass ratios.

#### 4.2 REFILL ANALYSIS

There are two requirements that must be satisfied during system refill. One is that helium must not enter the screen channel device at any time. The second requirement is that propellant tank pressure remain below the maximum allowable of  $2412 \text{ kN/m}^2$  (350 psia).

It is assumed that tank pressure will reside at  $2067 \text{ kN/m}^2$  (300 psia) prior to initiating a refill procedure. Helium partial pressure will be approximately  $1936 \text{ kN/m}^2$  (281 psia), and  $N_2O_4$  partial pressure will be  $131 \text{ kN/m}^2$  (19 psia). Should refill be attempted without venting helium, its partial pressure will be increased to a level well in excess of  $2067 \text{ kN/m}^2$  (300 psia) during liquid fill. Thus sufficient helium must be vented to maintain tank pressure within acceptable limits.

Tank venting must guarantee that the screened volume will not be contaminated by helium entry. Helium cannot penetrate the device while it remains filled with propellant. However, once vapor resides within the device, helium entry can occur as a result of molecular diffusion or forced convection flow.

There are two methods available for venting helium while maintaining the integrity of the screen device. The first method requires that the tank be vented to a vacuum prior to each refill. All usable propellant will be transferred to a storage tank prior to evacuating the tank, as will be employed for cryogenics. The second method initiates ullage venting without first expelling liquid. Method one can guarantee no helium will reside within the screen device because the tank has been evacuated. The disadvantage of this approach is that a near-vacuum condition will result only after a lengthy vent period. It may even be necessary to purge the tank contents with warm vapor in order to evaporate liquid or sublime frozen propellant.

4.2.1 SELECTED HELIUM VENT PROCEDURE. The second and preferred method for venting helium is to initiate ullage venting while a pre-determined liquid mass resides within the tank. This procedure will also guarantee that helium will not enter the screen device because the device will remain filled with liquid throughout the propellant tank vent process. The following steps describe the vent procedure:



1. The partial pressure of helium and  $N_2O_4$  vapor will decrease as tank pressure is decreased.  $N_2O_4$  evaporation will begin at the screen surface once liquid vapor pressure exceeds the partial pressure of  $N_2O_4$  vapor.
2. Liquid will be drawn into the capillary device to replace the liquid lost due to surface evaporation at the screen.
3. The liquid bulk cannot boil while pressure exceeds liquid vapor pressure. Consequently, tank vent will be terminated at a pressure greater than  $131 \text{ kN/m}^2$  ( $19 \text{ psia}$ ) to assure that the capillary device remains filled.

Note: The only evaporation (or boiling) that can occur is at a liquid-vapor interface, and only if vapor pressure at the surface exceeds partial pressure above the surface.

A determination has been made of the amount of venting required prior to tank refill. Figure 4-3 shows the final helium pressure following refill as a function of tank pressure at the end of vent, and the percent liquid in the tank prior to venting. It is noted that a final helium pressure less than  $1936 \text{ kN/m}^2$  ( $281 \text{ psia}$ ) will be realized without initially venting the tank below the  $N_2O_4$  boiling pressure. Figure 4-4 indicates that approximately  $3.2 \text{ kg}$  ( $7 \text{ lb}_m$ ) to  $4.5 \text{ kg}$  ( $10 \text{ lb}_m$ ) of  $N_2O_4$  vapor will be vented during this process.

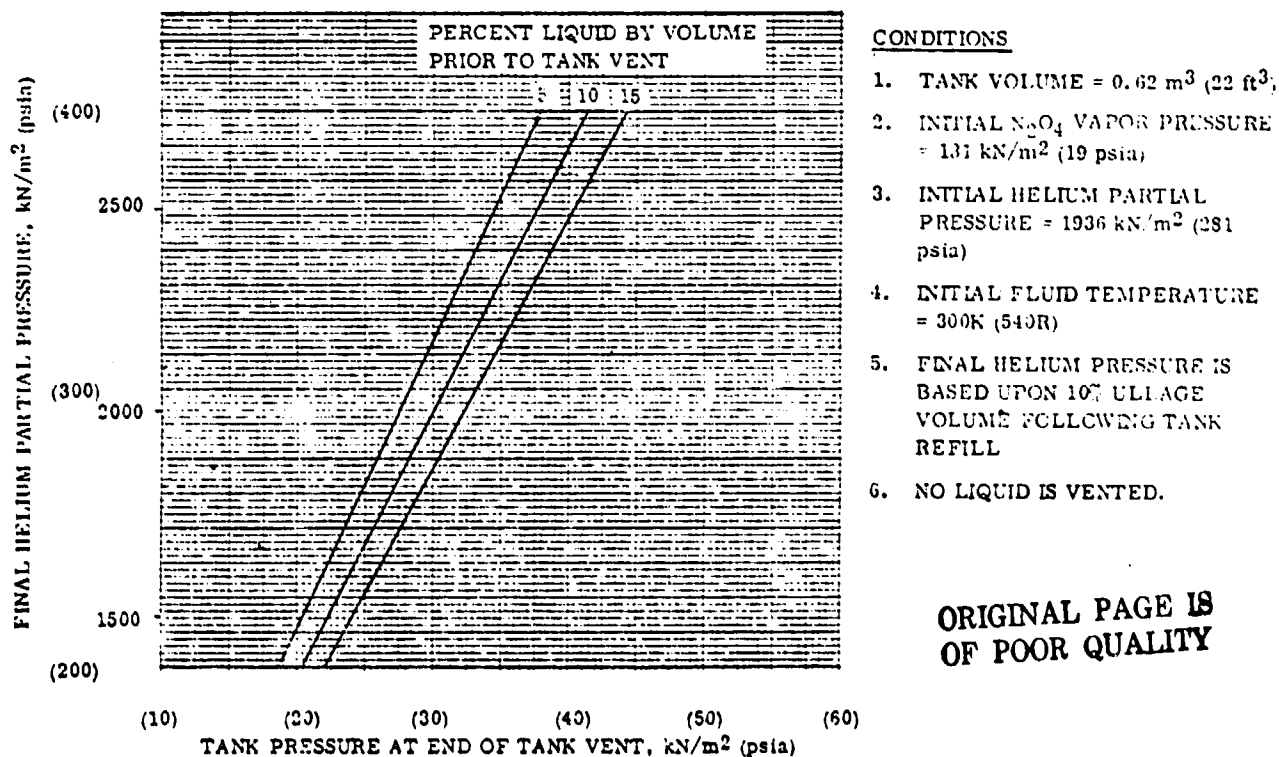
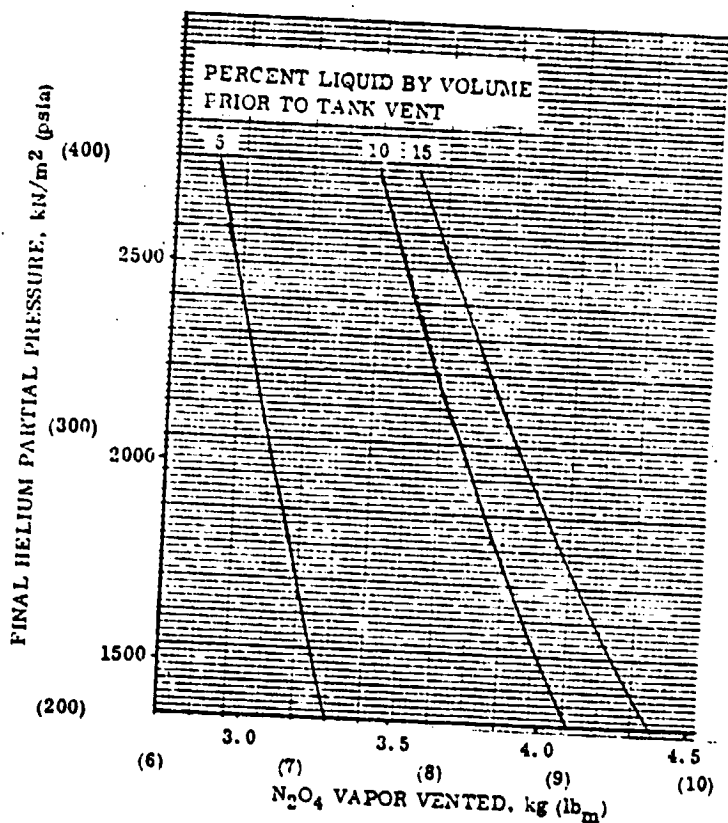


Figure 4-3. Management System Pressure at End of Vent Required to Provide Indicated Helium Pressure Following  $N_2O_4$  Refill



#### CONDITIONS

1. TANK VOLUME =  $0.62 \text{ m}^3$  ( $22 \text{ ft}^3$ )
2. INITIAL  $\text{N}_2\text{O}_4$  VAPOR PRESSURE =  $131 \text{ kN/m}^2$  ( $19 \text{ psia}$ )
3. INITIAL HELIUM PARTIAL PRESSURE =  $1936 \text{ kN/m}^2$  ( $281 \text{ psia}$ )
4. INITIAL FLUID TEMPERATURE =  $300\text{K}$  ( $540\text{R}$ )
5. FINAL HELIUM PRESSURE IS BASED UPON 10% ULLAGE VOLUME FOLLOWING TANK REFILL
6. NO LIQUID IS VENTED.

Figure 4-4.  $\text{N}_2\text{O}_4$  Vapor Vent Mass Required During System Blowdown to Provide Indicated Helium Partial Pressure During Propellant Refill

There is a question regarding the quantity of liquid that should be in the tank during helium venting. A case can be made for reducing the initial mass to a minimum. This condition has the advantage of minimizing the amount of vapor to be vented, as indicated by Figure 4-4. More importantly, liquid venting is less likely to occur when smaller quantities are present. A requirement exists, however, to maintain communications between the liquid bulk and the screen channel liquid. This communication is needed to assure that the bulk replaces losses due to surface evaporation at the screen. Sufficient liquid must be present in the tank to guarantee that communications will be maintained. It was judged that a ten percent liquid residual at the start of helium vent represents an acceptable compromise to this conflict.

4.2.1.1 Propellants Consumed During Helium Vent. System refill will be accomplished at the loss of some propellant and helium pressurant. Figure 4-4 shows that approximately  $3.8 \text{ kg}$  ( $8.3 \text{ lb}_m$ ) of  $\text{N}_2\text{O}_4$  vapor will be expelled during each refill procedure;  $1.7 \text{ kg}$  ( $3.7 \text{ lb}_m$ ) of helium will also be vented. These fluid quantities will be lost regardless of the selected refill procedure. The fact is that the helium must be vented before refill can be initiated and, a mixture of helium and propellant vapor will be vented in the process.

### 4.3 VAPOR REMOVAL FROM SCREEN CHANNEL DEVICE

Upon completion of propellant fill or refill, vapor will be trapped within the screen device, because procedures to prevent this are not known. Vapor can, however, be easily removed from the channels by condensation. The same condensation process will be employed for earth storables as for cryogenics (see Section 3.5).

Vapor removal will be accomplished when the system is pressurized to its operating level of 2067 kN/m<sup>2</sup> (300 psia). The trapped vapor will condense due to being immersed, or surrounded by, subcooled propellant. The general equation describing the heat transfer controlled bubble collapse process is given by Equation 3-61. That equation is also applicable to the condensation of N<sub>2</sub>O<sub>4</sub> vapor. Results of Equation 3-61 for N<sub>2</sub>O<sub>4</sub> properties are given in Figure 4-5. Bubble collapse times are considerably shorter than for liquid hydrogen. This is seen by comparing bubbles of 2.54 cm (1.0 in) initial radius. An N<sub>2</sub>O<sub>4</sub> vapor bubble will collapse in less than 10 seconds, when immersed in liquid subcooled by 276 kN/m<sup>2</sup> (40 psid); whereas a hydrogen bubble will collapse in 210 seconds under its system operating pressure. As discussed in Section 3.5, the time to collapse vapor trapped within the screen device will not exceed the time for a 2.54 cm (1.0 in) radius bubble to collapse.

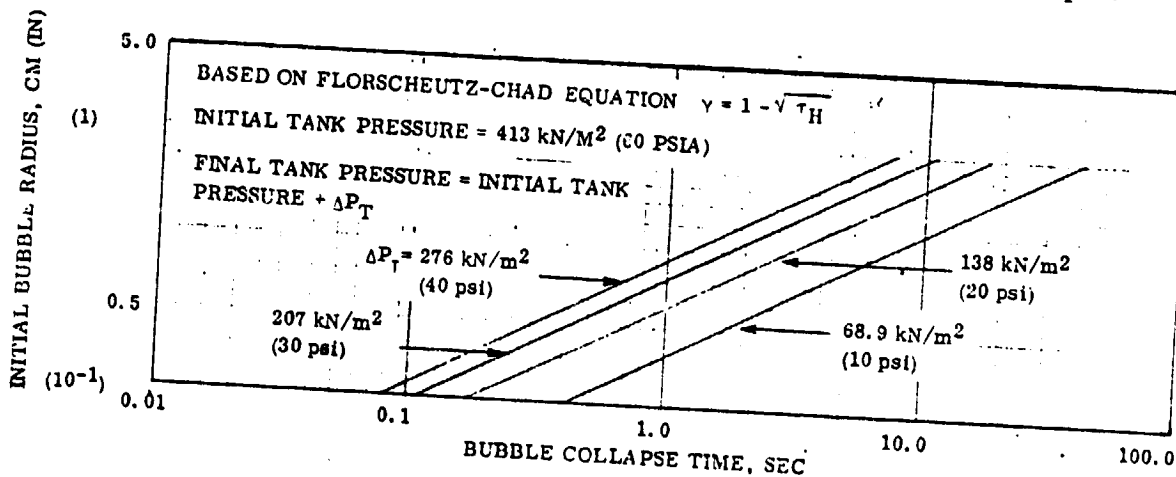


Figure 4-5. Collapse Time for Spherical Bubbles in Liquid N<sub>2</sub>O<sub>4</sub>

### 4.4 SUMMARY

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Fluid management system refill will be identical to the initial fill once pre-conditioning has been completed. For the case of initial tank fill, no pre-conditioning will be required. Liquid will simply be introduced into the tank at a sufficiently high flowrate and velocity to maintain near-thermal equilibrium conditions. Although specific analyses were not conducted for N<sub>2</sub>O<sub>4</sub>, it is estimated that liquid flowrates as low as 0.315 kg/sec (0.7 lb<sub>m</sub>/sec) will maintain acceptably low tank pressures during the fill process. Venting will not be required during tank fill.

Although no precautions will be taken to keep vapor from entering the screen device, the collapse time is so rapid for  $N_2O_4$  bubbles, that the device is expected to be filled with liquid at the completion of tank fill. This condition will exist because fill will require several minutes to complete, during which time the propellant will be subcooled by more than  $276 \text{ kN/m}^2$  (40 psi). As previously indicated, vapor collapse times for this subcooled condition will be substantially lower than the time to complete tank fill.

Pre-conditioning is required for system refill because much of the helium must be vented before refill is begun. The selected procedure for this case will be:

1. Transfer liquid to or from the tank in order to provide a ten percent fill condition.
2. Vent the tank down to  $206.7 \text{ kN/m}^2$  (30 psia).

The screen channel device will remain filled with liquid during this time, as all evaporation at the screen will be replaced by the tank liquid. A total of 3.8 kg (8.3 lb<sub>m</sub>) of  $N_2O_4$  and 1.7 kg (3.7 lb<sub>m</sub>) of helium will be vented during each refill. These fluid quantities will be vented regardless of the selected vent procedure.

Propellant flow conditions for tank refill will be the same as those identified above for the initial fill condition. Tank pressures will be maintained acceptably low during this process so that venting will not be required. Furthermore, the screen device will remain filled with liquid throughout this time period.

# 5

## MODELLING OF PROPELLANT FILL PROCESSES

Fluid management system fill and refill is examined in this section to determine what experimentation is required to demonstrate this capability in a low g environment. Emphasis is placed on identifying normal gravity tests that can be conducted to model low gravity filling behavior. This approach was taken because of the ease with which ground based tests can be conducted compared to the more complex, and considerably more expensive, space based tests. The modelling and scaling techniques employed, however, are also applicable to space based tests.

Based on the analysis performed in Sections 3, 4 and 5 the important phases in filling cryogenic and noncryogenic tanks containing "total control" type acquisition devices have been identified. Refilling cryogenic tanks containing capillary devices that are partially full demonstrates the complete range of possible phenomena that must be handled. Removal of the liquid from the tank must be accomplished prior to the removal of helium from the tank. As discussed, removing the helium by purging and venting will be one of the most difficult phases of the refilling process. Once the helium is effectively removed, refilling is similar to the initial filling of the tank.

Prechill of the tank is accomplished by inflowing liquid or cold vapor to cool the tank to the desired level. When the fluid in the tank has reached predetermined pressure and temperature conditions the tank is vented. Venting only occurs when vapor exists in the tank. The prechill/vent sequence cools the tank sufficiently to allow the tank chill process to subsequently be carried out with a locked up tank. This eliminates the need to control the liquid/vapor interface for venting purposes.

After tank chill has occurred, the filling process is identical for cryogenic and non-cryogenic tanks and efficient filling will be accomplished by maintaining thermodynamic equilibrium. Spraying liquid into the tank at a high flowrate and velocity, provides the fluid agitation to accomplish a low pressure rise condition, and maximize the amount of liquid that can be put in the tank without exceeding a given tank pressure.

At this point the tank is full but the capillary device is likely to contain trapped vapor. This vapor will be condensible, if the helium removal has been successful. The vapor will be condensed by subcooling the trapped vapor with pressurant. In cases where warm vapor is used for pressurization, the pressurization should occur just prior to the end-use propellant outflow from the tank in order to minimize any mixing between the liquid and vapor that could cause tank pressure reduction and subsequent bulk boiling in the capillary device.

## 5.1 PROCESS SCREENING FOR MODELLING

A thermodynamic model was developed in Sections 3 and 4 for each of the processes identified above. These analytical models were combined to provide a complete description of system fill and refill, and serve as the primary basis for identifying problem areas and developing procedures for low-g fill. In several instances the models provide an accurate representation of the physical process. Experimentation will not be required for these processes. Additional data will be required for the other processes, however, before sufficient confidence can be placed in procedures for low-g fill and refill.

The first task, therefore, is to conduct a screening evaluation for placing processes in the following categories:

1. Process is subject to rigorous analysis.
2. Process is adequately defined, but uncertainties exist regarding analytical models or the ability to extrapolate to low-g.
3. Process is not sufficiently well known and, therefore, extrapolation to a low-g environment is unacceptable.

Processes that belong in the first category will not be considered for experimentation because the analytical models are sufficiently accurate to enable evaluation. Similarly, the third category of processes will not be subject to experimentation because insufficient information is available to identify an acceptable test plan. Only processes in the second category will be evaluated to determine modelling and scaling equations from which normal gravity tests can be conducted and results extrapolated to a low-g environment.

5.1.1 PROCESSES SUBJECT TO RIGOROUS ANALYSIS. The only processes in this category are tank vent and purge for a system which must expel helium.

5.1.1.1 Purge of Cryogenic System Prior to Refill. The ullage vapor of a nearly empty propellant tank will contain hydrogen vapor at  $344.5 \text{ kN/m}^2$  (50 psia) and helium at  $68.9 \text{ kN/m}^2$  (10 psia). The tank will be vented down to  $6.89 \text{ kN/m}^2$  (1.0 psia) preparatory to a purge that will dilute helium concentration to an acceptable level. Two charge and vent cycles will be conducted using warm hydrogen vapor; the charge period will increase tank pressure to  $63.9 \text{ kN/m}^2$  (10 psia) and the vent period will reduce pressure to  $6.89 \text{ kN/m}^2$  (1.0 psia).

Analyses performed to describe the above are sufficiently accurate to obviate the need for experimentation. There is nothing in the charge and vent phenomena that is influenced by gravity environment. It is likely that any purge tests conducted will be useful in establishing a more effective procedure; but, such data is not considered to

be technology oriented and is, therefore, not applicable to a test plan.

5.1.1.2 Earth Storable System Vent Preceding Refill. It will be necessary to expel much of the helium contained within the tank before  $N_2O_4$  refill can be implemented. The analysis of Section 4.2 indicated that sufficient helium will be expelled in the process of venting the tank from its initial pressure of  $2067 \text{ kN/m}^2$  (300 psia) to  $206.7 \text{ kN/m}^2$  (30 psia). Bulk boiling of  $N_2O_4$  will not occur during the vent process because tank pressure will remain above the  $N_2O_4$  vapor pressure of  $131 \text{ kN/m}^2$  (19 psia). The small liquid quantity contained within the tank will remain relatively free of disturbances as long as tank vent is conducted slowly.

5.1.2 PROCESSES REQUIRING EXPERIMENTATION. The three processes requiring experimentation are tank prechill, tank fill, and vapor bubble condensation. Each was analytically modelled (see Sections 3 and 4) in an effort to identify acceptable system fill and refill procedures. The models do require verification, however, because the equations used represent extrapolations of existing empirical data. These processes will be discussed in greater detail in Section 5.2.

5.1.3 PROCESSES NOT ADEQUATELY DEFINED. One process that may be in this category is tank chill. Boiling will occur during tank chill as liquid spray impacts the hot tank walls. This boiling phenomenon is rather complex due to the transient nature of a vapor film being created and destroyed with each droplet impact. Equation 3-13 describes the phenomenon of liquid splattering on a hot surface, and indicates that heat transfer will be independent of acceleration environment. However, the equation is considered to be unsubstantiated because there is no variable acceleration data available to confirm this independence.

Rather than grapple with the boiling phenomenon uncertainty, the preferred alternative was to identify a procedural change to circumvent any uncertainty. In this case the decision was made to establish a prechill process to eliminate any concern with excessive pressures occurring due to boiling during tank chill. This was done by allowing prechill to remove sufficient tank energy that peak pressure would be acceptably low during tank chill.

## 5.2 MODELLING OF SELECTED PROCESSES

Modelling or scaling analyses will be used to determine the feasibility, and subsequently the conditions and configurations, of ground based model testing for providing data useful to determining the filling characteristics of an earth storable and cryogenic liquid fluid management system.

5.2.1 TANK PRECHILL. The initial management system fill procedure in low-g will require a prechill process to reduce tank temperature to an acceptably low level. Tank prechill will be accomplished by introducing low temperature hydrogen vapor at a sufficiently high velocity that forced convection heat transfer will dominate the heat

exchange process. The primary advantage of maintaining a forced convection environment is that heat transfer will be independent of acceleration environment. This provides a dual benefit: first tank prechill can be effected within a relatively short period, which is highly desirable; more importantly, it will be possible to conduct normal gravity prechill experiments that will be directly applicable to a low-g environment.

The prechill subroutine, Appendix B, was used to select a management system prechill procedure for a low-g environment. The heat transfer correlation selected as being representative of the heat exchange mechanism is given by Equation 3-11, and repeated below for convenience.

$$\frac{h}{\rho C_p} (N_{PR})^n = A \left[ \frac{(\dot{m}v^2/V) \mu}{P^2} \right]^m \quad (5-1)$$

where

$h$  = heat transfer coefficient

$(\dot{m}v^2/V)$  = fluid power input per unit volume

$P$ ,  $C_p$ ,  $N_{PR}$  and  $\mu$  are fluid properties

$n$ ,  $m$ ,  $A$  are empirical coefficients where

$$n = 2/3$$

$$m = 1/4$$

$$A = 0.163$$

Equation 5-1 was obtained from a correlation developed for industrial mixing processes, where tests were conducted with liquids and fixed bodies in cylindrical tanks. Furthermore, fluid agitation was provided by a mixer. Differences that exist between the industrial mixing process and the intended space application are such that verification, or modification, of Equation 5-1 will be necessary. These differences include fluid phase, tank geometry, and method of fluid agitation. Of the three, the influence of fluid phase is probably trivial, because fluid property effects are included in the equation. It is expected that tank geometry will have a small influence upon the final form of Equation 5-1. This assertion is based on the fact that empirical data upon which the equation is based, includes a variety of cylindrical containers.

The validity of Equation 5-1 for system prechill will be influenced primarily by the method of fluid agitation. There would appear to be a significant difference between agitating a fluid with a mixing device, and agitating by means of fluid input power. However, investigations indicate that heat transfer is a function of the fluid turbulence intensity, which is a function of  $\dot{m}v^2/V$ . It is expected that the only variation to



Equation 5-1 will be in the magnitude of the empirical coefficients. The coefficient most likely to change is A; coefficients n and m are not expected to change.

Ground-based tests for determining the validity of Equation 5-1 will be applicable only if free convection heat transfer is dominated by a forced convection process. It is seen that the forced convection heat transfer coefficient, h, can be improved by increasing the fluid input power term,  $\dot{m}v^2$ . Thus, a series of prefill tests will be conducted, for a range of  $\dot{m}v^2$  that will provide forced convection dominance. The feasibility of this approach has been demonstrated with the aid of the Appendix B subroutine where prefill of the Centaur-OSS model test tank was modelled. Analysis results are plotted in Figure 5-1. It is noted that forced convection coefficients will be about three times greater than free convection at entering fluid conditions of 0.0009 kg/sec (0.002 lb<sub>m</sub>/sec) and 152 m/sec (500 ft/sec). This inflow condition is feasible for ground test purposes.

5.2.1.1 Test Parameters. Tests will be conducted for a range of fluid power input values,  $\dot{m}v^2$ , using gaseous hydrogen and nitrogen flow into the Centaur-OSS test tank. Heat transfer rates to the vapor from the hot tank will be determined using gas temperature and pressure, and tank temperature. The resultant heat transfer coefficients will be calculated. Empirical coefficients will be selected which provides the best match for Equation 5-1 with test data.

5.2.2 TANK FILL. The intent of the tank fill process will be to create conditions conducive to attaining near-thermal equilibrium. These conditions will be achieved by introducing liquid into the tank through spray nozzles, which resulting spray will provide the high heat transfer rates needed to attain near-thermal equilibrium conditions. As tank fill continues, the internal tank fluid environment will change from one where liquid droplets reside within the ullage volume to one where vapor bubbles are entrained within a liquid bulk. The transition from heat transfer dominated by liquid droplets to heat transfer dominated by vapor bubbles is expected to occur in the range of 10 percent to 20 percent liquid fill. As propellant tank fill continues, the mechanism of liquid spray in a vapor environment will change to that of vapor bubble entrainment and dispersal within the liquid volume. Entrainment will occur as a result of liquid impingement with the liquid bulk. Dispersal will be caused by agitation created by liquid sprayed into the tank. This mechanism will be the dominant mode of heat and mass exchange throughout most of the tank fill process, and is the only mechanism that will influence tank pressures toward the completion of tank fill. Consequently, this mechanism will be analyzed to determine what modelling and scaling techniques will be applied toward the development of a test plan.

**NOTES:**

1. TANK VOLUME = 0.29 M<sup>3</sup> (10.3 FT<sup>3</sup>)
2. TANK MASS = 39.5 KG (87 LBM), CRES
3. INITIAL TANK TEMP. = 300K (540R)
4. ENTERING CH<sub>2</sub> FLOWRATE = 0.0009 KG/SEC (0.002 LB<sub>M</sub>/SEC)
5. ENTERING CH<sub>2</sub> TEMP. = 252K (140R)

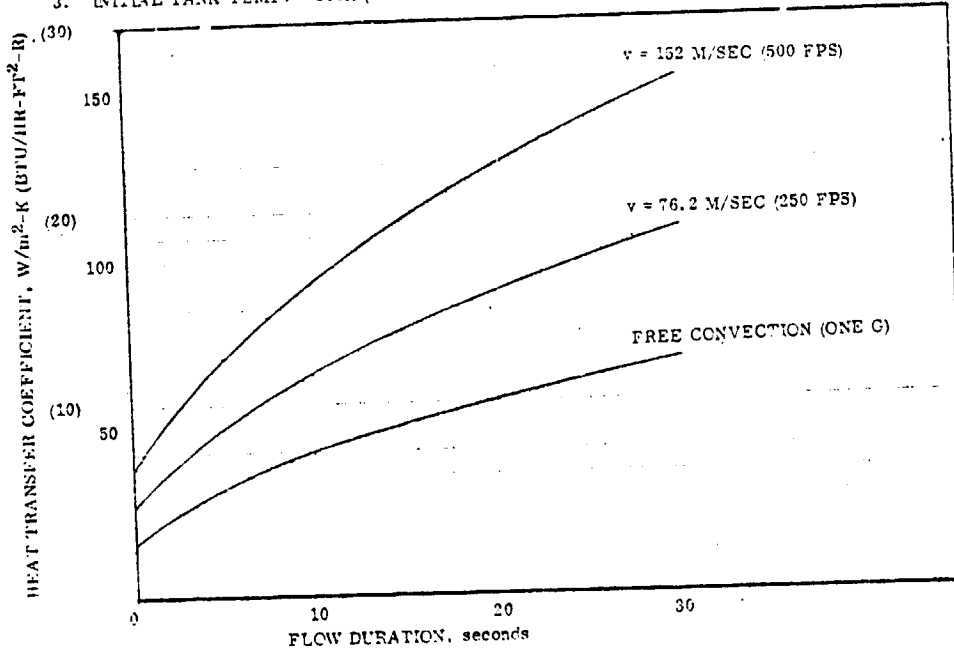


Figure 5-1. Predicted Forced Convection and Free Convection Heat Transfer Coefficients During Normal Gravity Test Tank Prechill Test

The applicable expressions for determining heat transfer between the dispersed vapor phase and its surrounding liquid were developed in Section 3. These equations are given below.

$$\dot{Q}_T = 4872 V_g (\dot{m}v^2/V_L)^{1/4} (T_B - T_L)/d \quad \text{(from Equation 3-58)} \quad (5-2)$$

$$d = \frac{1.134 \sigma^{0.6} \epsilon^{0.5}}{\gamma^{0.2} (\dot{m}v^2/V_L)^{0.4}} + 0.09 \quad \text{(from Equation 3-33)} \quad (5-3)$$

$$V_g = \dot{V}_g t \quad (5-4)$$

$$\dot{V}_g \propto (\dot{m}v/\rho_L) \quad \text{(from Equation 3-38)} \quad (5-5)$$

$$t \propto v (D_{or}/d_g) \quad \text{(from Equation 3-52)} \quad (5-6)$$

$$\epsilon = V_g/(V_g + V_L) \quad (5-7)$$

Combining Equations 5-4 through 5-6

$$V_g \propto \dot{m} v^2 (D_{or} / d_g \rho_L) \quad (5-8)$$

All variables influencing the thermodynamics of propellant tank fill are contained in Equations 4-2, 5-3, 5-7 and 5-8. The task now is to establish a relationship. An inspection of these equations indicates that rigorous modelling techniques cannot be employed to establish equivalence between normal gravity and low-g fill. This is due to the unknown influence of acceleration environment upon vapor entrainment, and because vapor entrainment is an important variable in the determination of  $\dot{Q}_T$  and  $d$ . The following discussion will show, however, that normal gravity tests will provide data useful for evaluating and verifying low-g fill procedures.

The first step in this modelling assessment is to modify the above equations to reflect the ratio of conditions between model tests and orbital application. Let

$$\dot{Q}_T^* = \dot{Q}_{TM} / \dot{Q}_{TP}, \quad d^* = d_m / d_p, \quad V_g^* = V_{gm} / V_{gp}, \quad \text{etc.}$$

where

m = one-g model test condition

p = low-g prototype test condition

Thus, the equations become

$$\dot{Q}_T^* = V_g^* (\overline{\dot{m} v^2 / V})^{* 1/4} (T_B - T_L)^* / d^* \quad (5-9)$$

$$d^* = f^* \epsilon^{* 0.5} / (\overline{\dot{m} v^2 / V})^{* 0.4} \quad (5-10)$$

$$V_g^* = \overline{\dot{m} v^2}^* (D_{or}^* / d^* g^*) \quad (5-11)$$

$$\epsilon^* = V_g^* / (V_g + V_L)^* \quad (5-12)$$

Fluid properties do not appear in these equations because it is assumed that the same fluid is employed for both the model and prototype test conditions.

Equation 5-11 is the key to this assessment because its influence is present in the remaining equations. More important, it is subject to the greatest uncertainty because of the acceleration influence,  $g^*$ . This equation indicates that  $V_g^*$  will decrease as  $g^*$  increases. Furthermore,  $g^* \gg 1$  when a one-g environment is compared to an orbital environment. Because the influence of the other variables of Equation 5-11 will obviously be small compared to  $g^*$ ,  $V_g^* < 1.0$ . If this is so, then it follows that,

$$\epsilon^* < 1.0$$

$$d^* < 1.0$$

$$\dot{Q}_T^* < 1.0$$

A physical description of the above is given by describing changes to the internal tank fluid behavior for a test where all parameters are kept constant with the exception of acceleration environment, which is gradually being decreased.

1. A reduction in acceleration environment will increase the stay-time of entrained vapor due to a reduction in bubble rise velocity.
2. This increased stay-time will result in a greater steady state volume of entrained vapor, because the rate at which vapor is entrained into the liquid does not change.
3. The greater volume of entrained vapor will increase liquid-vapor surface.
4. Finally, the combination of increased heat transfer area and time will result in an increased heat transfer rate between liquid and entrained vapor.

It is concluded that the influence of a low-g environment will be to increase the likelihood that thermal equilibrium conditions will exist during the tank fill process. It can, therefore, be stated that fill tests conducted in normal gravity represent a more severe environment for maintaining acceptably low tank pressures.

It will still be extremely useful to conduct normal gravity tests if the results are that near-equilibrium conditions will be maintained during fill. Such test results will be interpreted as reflecting a greater deviation from thermal equilibrium than will occur in a low-g environment. With this limitation on the interpretation of test results, a flow test parameter can be selected. Equations 5-9 and 5-10 show that the only other parameter to influence heat transfer of the dispersed vapor phase during tank fill is  $\rho v^2 V$ , the fluid power input per unit volume. This fluid power input parameter affects heat transfer coefficient as well as vapor bubble diameter. It is recommended that  $\rho v^2 V$  be employed for scaling ground test results to a low-g environment.

Figure 5-2 illustrates expected results for a series of tank fill tests. It is reasonable to assume that tank pressure will approach equilibrium pressure as the fluid power input parameter is increased. As indicated by Figure 5-2 a critical parameter value exists for which equilibrium will be maintained during tank fill. Any additional increase in the parameter value will not further decrease propellant tank pressure. The test objective will be to obtain a relationship between the fluid power input parameter and tank pressure, which is a measure of fluid equilibrium conditions. Of special interest will be identification of the minimum parameter value for which equilibrium will occur. Tests will be conducted using liquid hydrogen for the cryogen

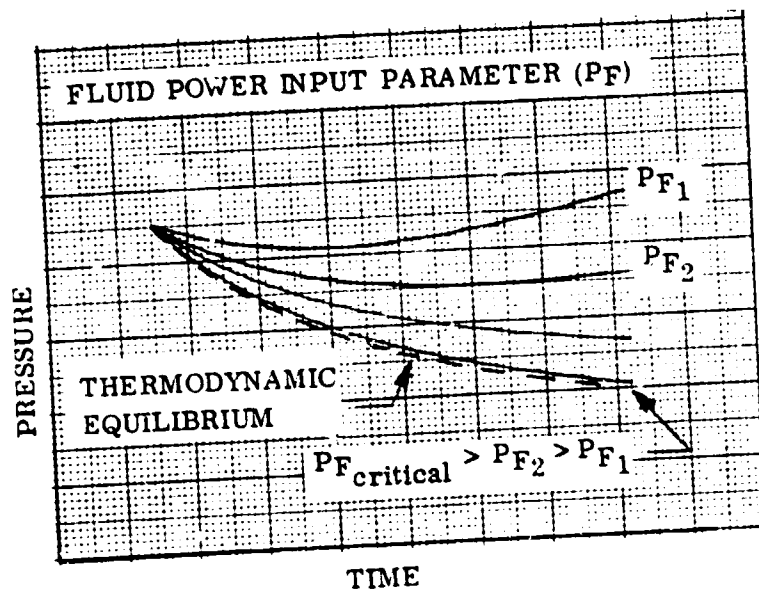


Figure 5-2. Expected Trend of Tank Pressure History During Fill Process

system tests, and Freon MF for the earth storable system tests.

5.2.3 VAPOR CONDENSATION. At the completion of propellant fill or refill, vapor will be trapped within the screen channel device. The fluid management system has a requirement to provide vapor-free liquid to other systems. Consequently, vapor must be removed from within the screen device.

A condensation process will be employed for vapor removal. Vapor condensation begins when liquid is subcooled, which will occur when the tank is pressurized. The condensation model used in Sections 3 and 4, was based upon conduction heat transfer controlled bubble collapse and is expressed as

$$v = 1 - \frac{r}{H} \quad (\text{from Equation 3-61}) \quad (5-31)$$

where

$$v = \frac{r}{r_0}$$

$$\tau_H \text{ (nondimensional time)} = \frac{4}{\pi} J_a^2 \frac{\gamma t}{r_0^2}$$

$J_a$  = Jakob number  $f$  (fluid properties)

$t$  = time

$r$  = bubble radius at  $t$

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$r_0$  = initial bubble radius

$\alpha t / r_0^2$  = Fourier Modulus

There is a parallel between Equation 5-13 and unsteady state heat conduction to or from a constant diameter, or solid, sphere.

First, each expression is a function of the Fourier Modulus,  $\alpha t / r_0^2$ , which is non-dimensional time for a constant diameter sphere. There is also a surprising similarity in the curves for heat removal from a collapsing bubble and a solid sphere when plotted as a function of non-dimensional time. A comparison is shown in Figure 5-3.

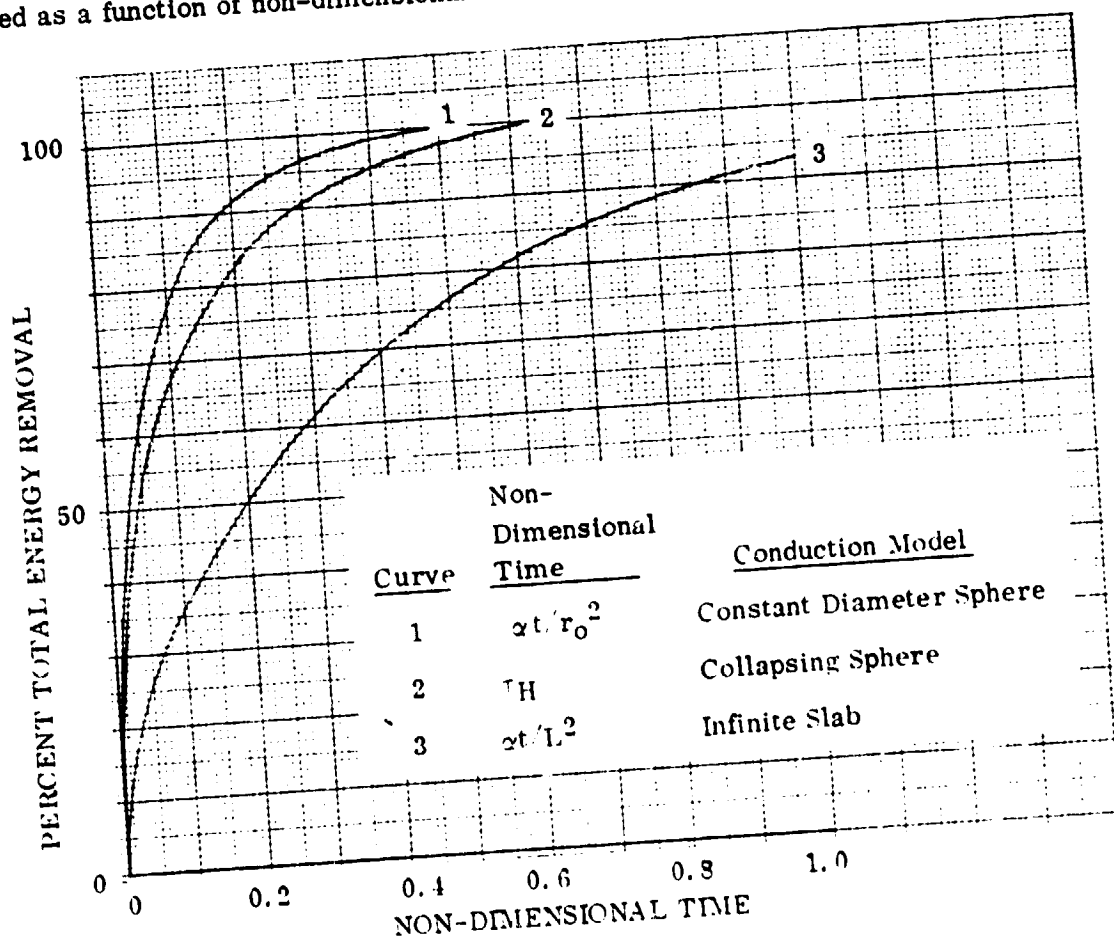


Figure 5-3. Percent of Energy Removed as a Function of Nondimensional Time

Although it is expected that vapor in bubble form will reside within the screen acquisition device, the possibility exists that larger quantities of vapor may be trapped and, in the extreme, the device could conceivably be filled with vapor. Because one cannot identify a realistic vapor entrapment condition for the acquisition device, it will be necessary to assume the worst case condition.

Figure 5-4 illustrates the process of vapor collapse due to condensation. It is assumed that conduction of heat from the vapor interface and from the interface to the liquid bulk will control the rate of condensation. Liquid will enter the screened channels as the vapor volume is reduced by condensation. Although flow across the screen will be uniformly distributed, the channel shape dictates that the liquid will collect at ends of the channel. Thus the liquid-gas interfacial surface area will be reduced as vapor collapse progresses while the vapor width remains constant.

There is no available analytical model for vapor collapse of the indicated condition. However, it is reasonable to assume that the Fourier Modulus,  $\alpha t/L^2$ , is the key parameter in any evaluation, where  $2L$  is the channel width. Because the model indicates that vapor width will remain constant, it is likely that the relationship of energy removed to dimensionless time will be like that for a slab. This relationship is given in Figure 5-4.

This model suggests that ground tests must be performed in such a manner that free convection effects are less significant than conduction. Since free convection effects cannot be eliminated from the propellant fill tests, a screened device will not be included as part of the test apparatus. Instead, a series of tests will be conducted with liquid hydrogen and Freon MF which should be adequate to verify an analytical model both for earth-storable and cryogenic propellants. An evaluation of test results will show whether the model will be more similar to unsteady state conduction heat transfer in a slab, or to a collapsing volume model.

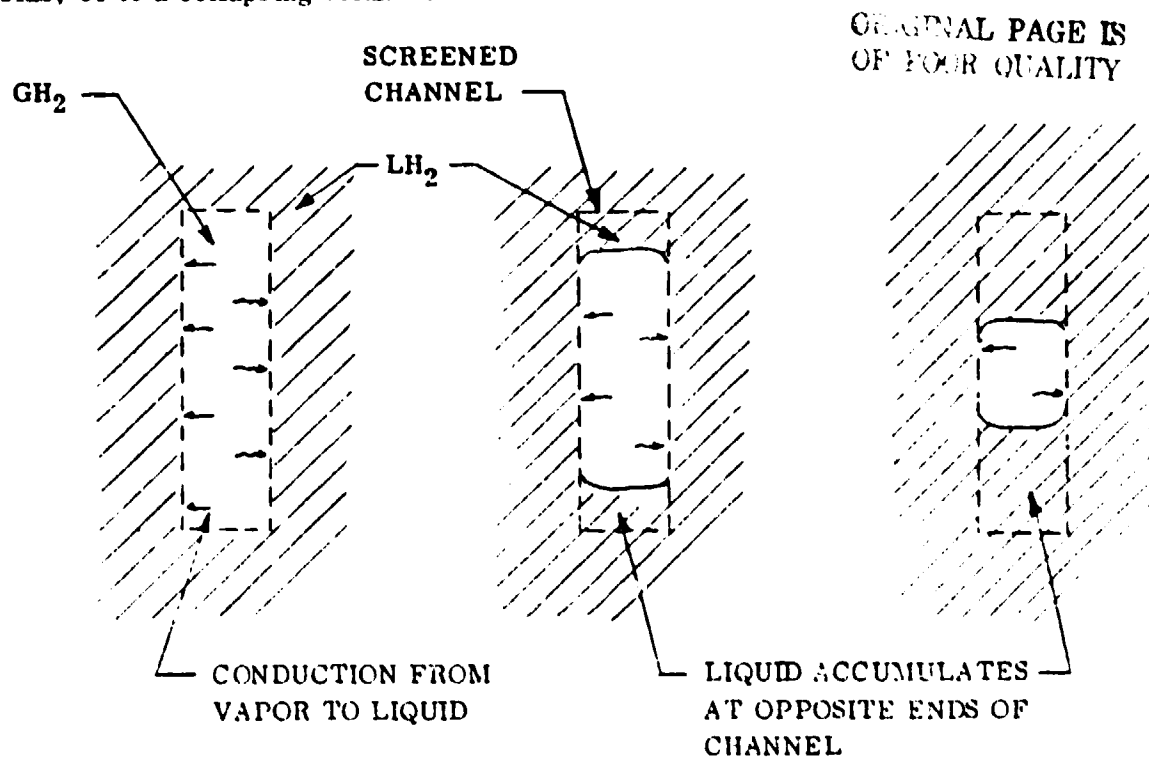


Figure 5-4. Vapor Collapse Process Within Screened Device

# 6

## TEST PLAN - FILLING OF ORBITAL FLUID MANAGEMENT SYSTEMS

The discussion of Section 5 identifies three processes of fluid management system fill and refill, which require experimentation to verify the analytical techniques developed in Sections 3 and 4. These processes are; tank prechill, tank fill, and condensation of trapped vapor. A modelling analysis has identified the dominant heat transfer and fluid mechanic parameter which controls the thermodynamic rate processes during prechill and fill. This parameter is the fluid power input per unit volume or,  $\dot{m}v^2/V$ , and it is a measure of the intensity of fluid agitation within the propellant tank. A determination was made that the vapor condensation process will be controlled by conduction heat transfer.

The parameter,  $\dot{m}v^2/V$ , will be used to determine forced convection heat transfer between the entering propellant vapor and hot tank walls during prechill. The basic form of the applicable heat transfer coefficient expression is given by Equation 3-11. Its empirical coefficients are uncertain because the expression was developed for a somewhat different set of conditions than will be experienced during prechill. Consequently, experiments will be conducted to identify a set of coefficients applicable to the prechill process.

The parameter,  $\dot{m}v^2/V$ , is the dominant variable in identifying the internal tank fluid condition during tank fill/refill. It determines the volume fraction of vapor entrained in liquid, and the bubble size population. Thus,  $\dot{m}v^2/V$  is the key to determining heat transfer rates between entrained vapor and liquid during tank fill. Experiments will be conducted to identify the threshold value of  $\dot{m}v^2/V$  for which thermal equilibrium will occur.

The condensation rate of vapor trapped within the screen device will be a function of vapor volume and shape. It is obvious that a small bubble will collapse more rapidly than a large bubble. Furthermore, a spherical bubble will collapse more rapidly than a bubble of another shape due to a greater surface area-to-volume ratio. Trapped vapor will be spherical in shape for all diameters less than 0.64 cm (0.25 in), the channel width dimension. All bubbles of a greater volume will be constrained by the channel width and, in the extreme, will acquire the channel shape. Vapor bubble condensation time can be determined from Equation 3-61. No comparable equation has been identified for determining collapse time for a slab or block of vapor.

The test plan outlined in the following paragraphs is intended to provide experimental data to model the critical elements described above. This test plan considered both ground based testing and space based testing. For ground based testing, candidates for test types were bench tests, drop tower tests and aircraft tests. Spacelab testing was considered for space-based testing.

Initially, the ground based testing will be described.



## 6.1 GROUND BASED TESTING

The ground based testing will consist of filling of a scale model test tank in order to run prechill and fill tests. Vapor collapse tests will be run using a transparent dewar.

Testing of the vent/purge procedure to remove condensible pressurant from the tank was considered as a candidate for testing. The decision was not to run tests since the recommended purging and venting operations are well understood and do not require any additional technology information. The vent pressure, number of purge cycles and mechanics of venting and purging can all be determined analytically.

For tank chilldown, testing is not required because prechill will lower tank temperature to the point where excessive pressures will not occur. Tank chilldown will essentially be replaced by the prechill period.

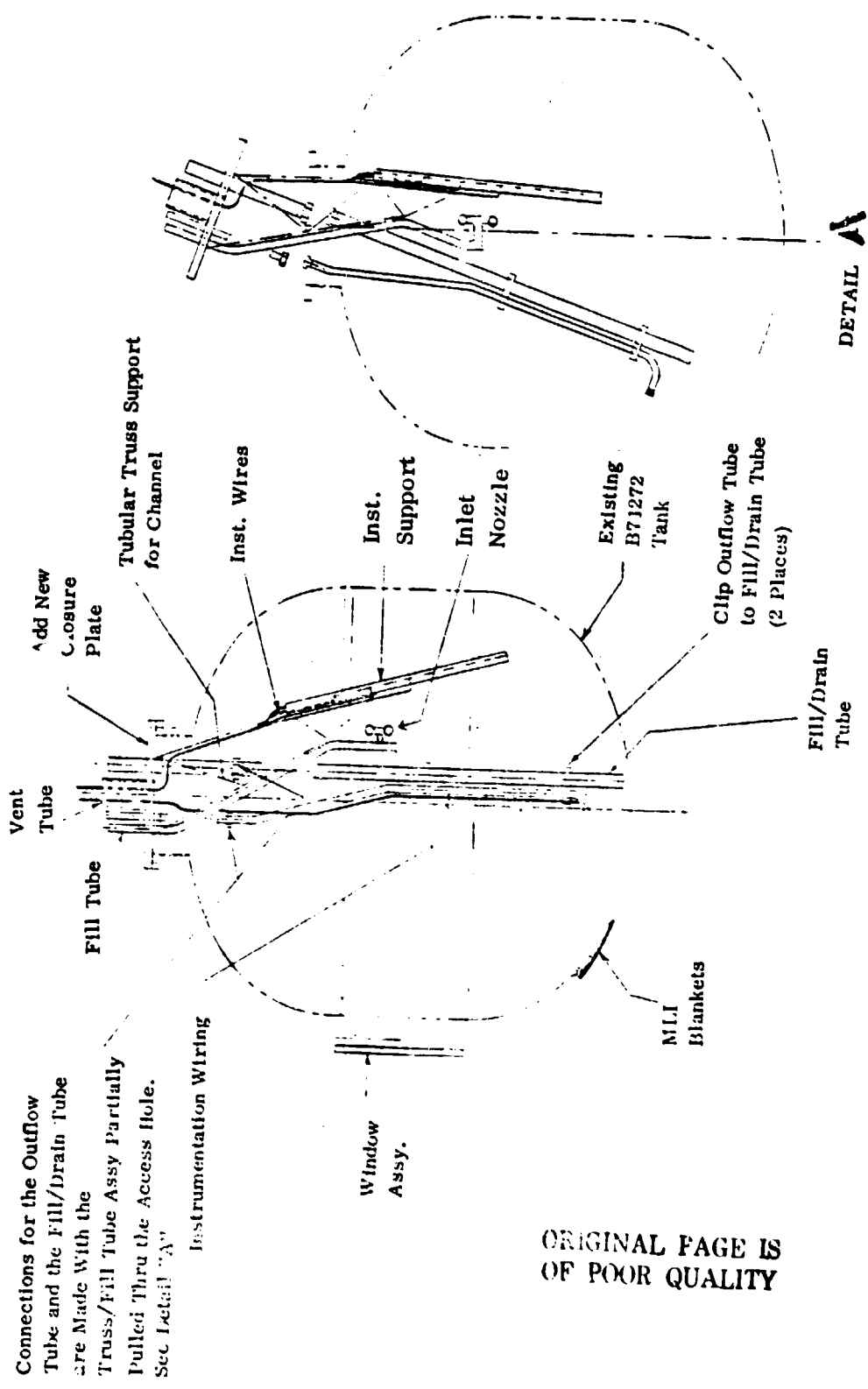
Drop tower and aircraft facilities were evaluated for providing required refilling data. These facilities were judged to be inadequate due mainly to the short low gravity test period obtainable with either method. Some prechill information could be obtained using drop towers however the thermodynamic analysis and modelling techniques indicate that normal gravity testing will provide sufficient information to verify the analytical models.

6.1.1 INFLOW TESTING. Inflow testing, consisting of prechill and fill testing will be conducted in a scale model orbit to orbit shuttle (OOS) tank shown in Figure 6-1. This tank is 74 cm (29 in) in diameter with a midsection length of 23.3 cm (9.2 in). The tank mass is 39.5 kg (87 lb<sub>m</sub>) and the tank volume is 0.29 m<sup>3</sup> (10.3 ft<sup>3</sup>). This tank was selected over a smaller spherical tank because of its lower tank mass to volume ratio. The OOS tank has a window in the cylindrical portion of the tank that can be used for viewing the experiment.

6.1.1.1 Prechill Tests. The objective of these tests is to verify the suitability of using Equation 3-11 to model tank prechill. Tests will be run with cold GN<sub>2</sub> and GH<sub>2</sub> into the test tank that is initially at room temperature. The prechill test set up is shown in Figure 6-2. The test tank will be insulated with multilayer insulation (MLI) and mounted within a 91 cm (36 in) diameter vacuum chamber in order to minimize external heat leak during the test period.

Prechill of the tank will be used to eliminate the need for venting a receiver tank containing liquid and vapor in low gravity. Prechill can be accomplished by introducing liquid or vapor into the propellant tank. For the application studied the vapor normally generated in chilling the transfer lines to the tank will also be employed to chill the tank. Consequently, the prechill tests will be run with low temperature vapor.

It is anticipated that system prechill will be conducted in the following manner: cold fluid will be introduced through a spray nozzle for a fixed time period. Once flow is terminated heat exchange will occur between the tank wall and fluid until temperature equilibrium



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Figure 6-1. Scale Model Centaur/OOS Test Article

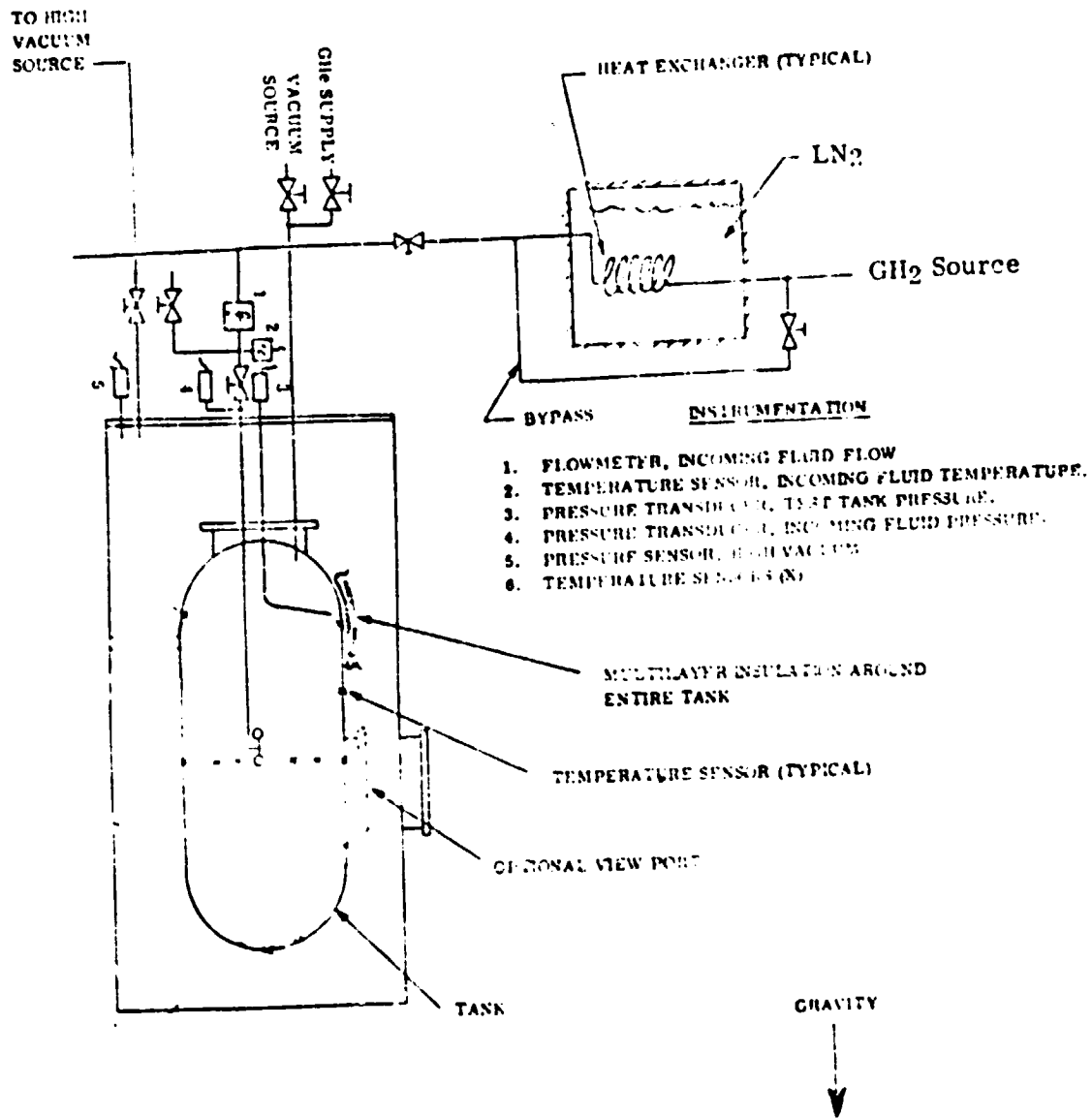


Figure 6-2. Tank Prechill Test Set-up Schematic

occurs. The tank will be vented down to near zero pressure in preparation for a second period of vapor inflow. The process will be repeated as many times as required to reduce tank temperature to an acceptable level for no-vent chill and fill.

Ground tests can be conducted to obtain prechill data applicable to a low gravity or zero gravity environment only if one-g free convection heat transfer effects become negligible. This will be achieved with the proposed test program by providing a high inflow vapor velocity to create forced convection heat transfer as the dominant heat exchange mode with the tank wall. (Only a small percentage of the total heat removal occurs during the process of liquid evaporation).

A test matrix for the GH<sub>2</sub> and GN<sub>2</sub> tests is given in Table 6-1. This test series will explore the variables of inflow configuration, vapor flowrate, initial tank temperature and, of course, fluid properties. The primary inlet configuration, nozzle #1 in Table 6-1, will be a spray nozzle configuration. The high degree of agitation, mixing and liquid-to-vapor heat transfer with a spray nozzle should make it an ideal inlet configuration for both the prechill and fill periods. Alternately, a bare pipe will be used as nozzle #2. Both configurations will also be tested in the fill tests. Predictions indicate that the bare pipe may provide the required chilldown efficiency and agitation to permit the simpler bare pipe configuration to be used.

Instrumentation to be used in the proposed testing are listed in Table 6-2. Figure 6-3 schematically illustrates the instrumentation locations. Temperature measurements will be made with both thermocouples (copper/constantan or chromel/constantan) and platinum resistance thermometers (Rosemont Model 134L). The platinum thermometers will also serve as liquid gas detectors by increasing the excitation current. All pressure measurements will be made with full bridge strain gage type pressure transducers (Statham Model 350). Flow measurements may be made with turbine

Table 6-1. GH<sub>2</sub> and GN<sub>2</sub> Prechill Test Matrix

CONTROLLED PARAMETER	RUN NUMBER									
	1	2	3	4	5	6	7	8	9	10
Inlet Configuration	1	1	1	1	1	2	2	1	1	1
Vapor Flowrate*	100%	90%	50%	20%	90%	90%	20%	100%	60%	20%
Test Fluid	Gaseous Hydrogen							Gaseous Nitrogen		
Test Tank Initial Temperature = 294.4K (530R).										
Test Tank Initial Pressure = 6.9 kN/m <sup>2</sup> (1.0 psia).										
Entering Vapor Temperature = 77.7K (140R)										
Terminate Test at 413 kN/m <sup>2</sup> (60 psia) Tank Pressure.										
* 100% flowrate condition of 0.004 kg/sec (0.009 lb <sub>m</sub> /sec) for GH <sub>2</sub> and 0.024 kg/sec (0.054 lb <sub>m</sub> /sec) for GN <sub>2</sub>										

(See Table 6-2 for key to instrumentation type.)

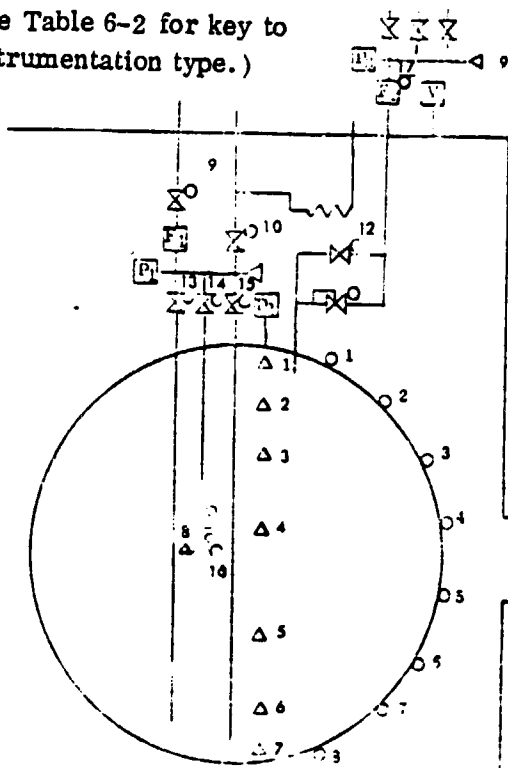


Figure 6-3. Instrumentation

increased between the phases. This condition will be promoted by creating a highly agitated, high velocity condition within the tank during fill. The high velocity condition will serve two important purposes. First, a finer spray (i. e. smaller drops) will occur as liquid velocity is increased. The reduced drop sizes will enhance heat and mass exchange with the ullage. Second, an increased liquid velocity will further improve heat and mass exchange with the ullage. Of course, fill tests conducted on the ground will become more representative of filling in space at higher liquid velocities as free convection effects become less significant.

In order to promote agitation in the tank, two inlets are used to direct flow towards the ends of the tanks. Calculations indicate that if the upward facing inlet is placed at 30% of the total height of the tank below the tank top the spray nozzle should effectively agitate the fluid to allow mixing until the tank is essentially full. One of the primary objectives of the testing will be to evaluate the degree of ullage agitation as the liquid level is increased above the upper facing jet. The experimental data on ullage temperature and pressure versus liquid level and spray nozzle outflow rate will be correlated with the analyses presented in Section 2.3.

The fill tests are applicable to both cryogenic and noncryogenic refilling. The scale model Centaur-OOS tank shown in Figure 6-1 will also be employed for fill testing.

type volumetric meters (Flow Technology Omniflow Model FTM-5-LB), orifice and pressure transducer combination head meters, and/or hot film anemometers (Thermo Systems Inc. Model 1053). A magnetic tape digital recorder (Dymec Model 150) or a digital punch tape recorder (Dymec Model 2400) will be the primary recorder for initial and final steady state data and the slowly varying test run data. An analog tape recorder (Ampex Model 1200 or 1300) will be used to record any higher frequency transient data required during the test runs.

6.1.1.2 Tank Fill Tests. Propellant tank fill should be conducted in such a manner that near-thermodynamic equilibrium will exist. This represents the condition of minimum tank pressure. Equilibrium between the liquid and vapor will be approached as heat and mass exchange is

Table 6-2. Instrumentation Location

Symbol	Proposed Instrumentation Type	Location
O 1	Cu/Con Thermocouple	Test Tank 100% Level
O 2		" " 90% "
O 3		" " 80% "
O 4		" " 60% "
O 5		" " 40% "
O 6		" " 20% "
O 7		" " 10% "
O 8		" " 0% "
O 9		Body-Valve 18
O 10		" " 19
O 11		" " 20
O 12		" " 21
O 13		" " 22
O 14		" " 23
O 15		" " 24
O 16	Spray Nozzle or Bare Pipe	
O 17	Gas Flow Meter Temperature	
Δ 1	Platinum Resist Thermo.	Test Tank 98% Level
Δ 2		" " 90% "
Δ 3		" " 75% "
Δ 4		" " 60% "
Δ 5		" " 40% "
Δ 6		" " 10% "
Δ 7		" " 2% "
Δ 8	Platinum Resist Thermo.	Liquid Inflow Temperature
Δ 9	Platinum Resist Thermo.	Gas Outflow Temperature
V	Ionization Vacuum Gage	Vacuum Chamber
F <sub>1</sub>	Turbine Flowmeter	Liquid In/Out Flow
F <sub>2</sub>	Hot Film Anemometer	Gas, Vapor Outflow
P <sub>1</sub>	Strain Gage Pressure Trans.	Liquid In/Out Flow
P <sub>2</sub>	Strain Gage Pressure Trans.	Test Tank Ullage
P <sub>3</sub>	Strain Gage Pressure Trans.	Gas Vapor Outflow

The test set-up is similar to that employed for prechill tests, but with the addition of a vacuum jacketed supply tank, Figure 6-4, that can provide liquid saturated between 101.3 kN/m<sup>2</sup> (14.7 psia) to 172.3 kN/m<sup>2</sup> (25.0 psia). Supply tank pressure can be in excess of 689 kN/m<sup>2</sup> (100 psia), which is more than enough to provide the maximum anticipated flowrate of 0.18 kg/sec (0.4 lb<sub>m</sub>/sec).

The LH<sub>2</sub> fill test matrix is shown on Table 6-3. Test variables include inlet configuration, LH<sub>2</sub> flowrate and velocity, entering liquid vapor pressure and test tank initial pressure. As for the prechill tests, the spray nozzle will be the primary inlet configuration and the bare pipe will be the alternate configuration. Two inlets are used for both alternates with flow directed from the middle towards the ends of the tank.

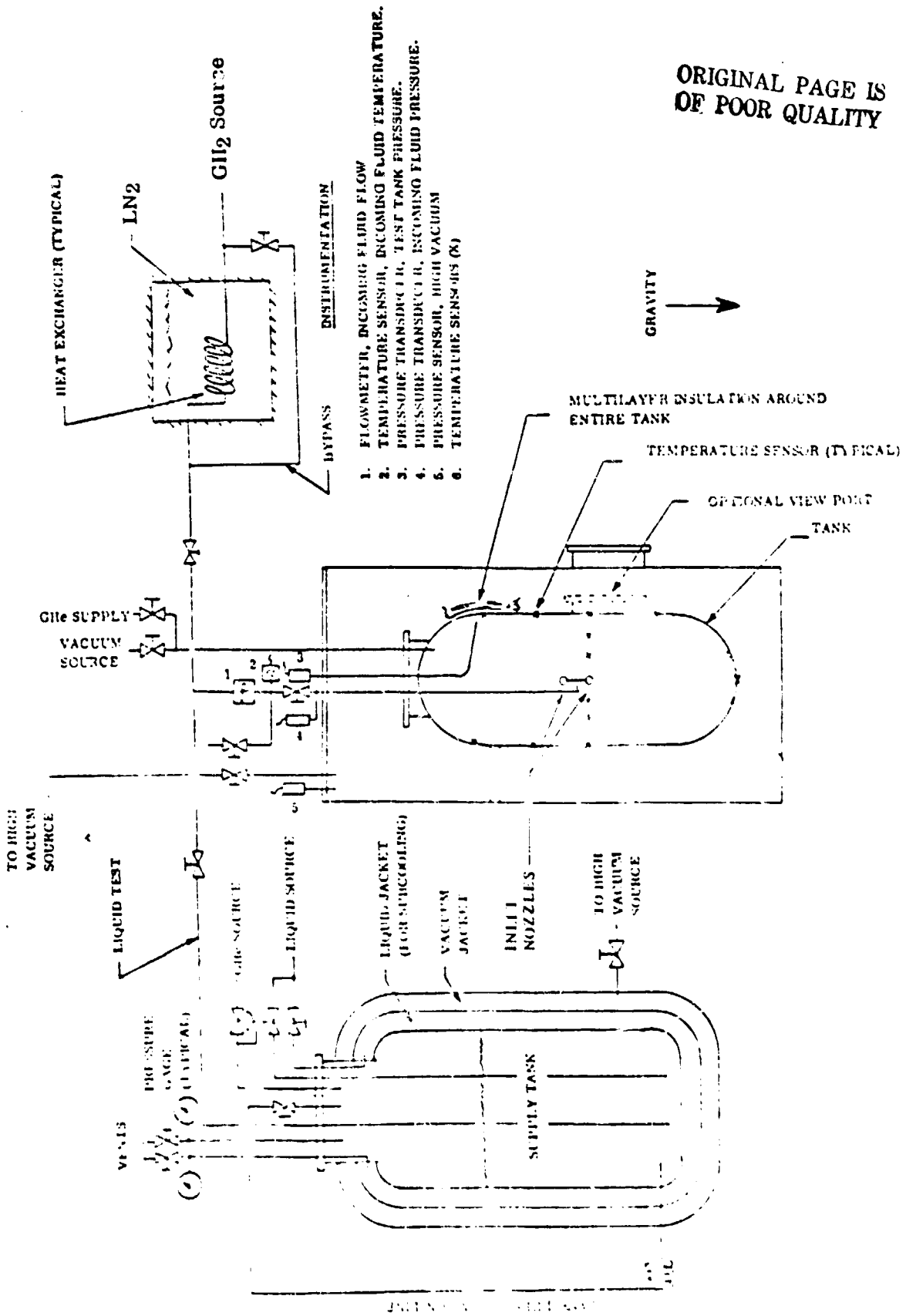
Tests to simulate tank fill with N<sub>2</sub>O<sub>4</sub> will be conducted by employing Freon MF as the simulant fluid. Freon MF was selected because its pertinent physical properties were similar to those of N<sub>2</sub>O<sub>4</sub>, especially liquid vapor pressure which is about 124 kN/m<sup>2</sup> (18 psia) at room temperature. A test matrix, given in Table 6-4, is similar to that for LH<sub>2</sub> except that initial tank temperature is not a test variable.

**6.1.2 VAPOR COLLAPSE TESTS.** Vapor will be trapped in the acquisition device during the propellant fill process. Once tanking is completed, pressurant will be introduced to condense this trapped vapor. As long as no noncondensable vapor is trapped within the acquisition device condensation will occur because the propellant vapor cannot remain at the elevated pressure while surrounded by subcooled liquid. The only uncertainty with this method is the time required for vapor to condense. Consequently a model was formulated in Section 3.4. For ground tests to be successful convection effects must be less significant than conduction. Since free convection cannot be eliminated from the propellant fill tests, vapor collapse tests cannot be appended to the prechill and fill test setup. Instead, the following test configuration is recommended.

A glass dewar as shown in Figure 6-5 will be used to permit viewing of the liquid vapor interface while minimizing conduction from the dewar walls. Heat transfer is allowed to occur at the liquid/vapor interface. The volume contained within the dewar represents the vapor trapped within the capillary device. The fluid/vapor contact area is known because of the relatively flat shape produced by gravity. This test should be a worst case because there will be negligible cooling of the vapor from the basket walls compared to a screen acquisition device where the screened walls provide more vapor/liquid contact.

Tests will be performed with both liquid hydrogen and Freon MF in order to simulate cryogenic and storable propellant vapor collapse. Testing will be performed within an air conditioned blockhouse in order to closely control environmental temperatures.

A suggested matrix of tests is presented in Table 6-5. A typical test sequence will be as follows:



1. FLOWMETER, INCOMING FLUID FLOW
2. TEMPERATURE SENSOR, INCOMING FLUID TEMPERATURE.
3. PRESSURE TRANSDUCER, TEST TANK PRESSURE.
4. PRESSURE TRANSDUCER, INCOMING FLUID PRESSURE.
5. PRESSURE SENSOR, HIGH VACUUM
6. TEMPERATURE SENSORS (X)

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Figure 6-4. Tank Fill Test Set-up



Table 6-3. LH<sub>2</sub> Fill Test Matrix

CONTROLLED PARAMETER	RUN NUMBER									
	1	2	3	4	5	6	7	8	9	10
NOZZLE	1	1	1	1	1	1	1	2	2	2
LH <sub>2</sub> FLOWRATE*	100%	90%	80%	70%	TBD	90%	80%	100%	90%	80%
ENTERING LIQUID VAPOR PRESSURE, KN/m <sup>2</sup> (PSIA)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	172.3 (25.0)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	172.3 (25.0)
TEST TANK INITIAL PRESSURE, KN/m <sup>2</sup> (PSIA)	413 (60)	413 (60)	413 (60)	413 (60)	413 (60)	413 (60)	276 (40)	413 (60)	413 (60)	413 (60)

INITIAL TEST TANK TEMPERATURE IS LH<sub>2</sub> TEMPERATURE  
 TEST TERMINATED AT APPROXIMATELY 90 PERCENT FILL

\* 100% FLOWRATE CONDITION IS .18 kg/sec (0.4 lb<sub>m</sub>/sec)

Table 6-4. Freon MF Fill Test Matrix

CONTROLLED PARAMETER	RUN NUMBER						
	1	2	3	4	5	6	7
NOZZLE	1	1	1	1	1	1	2
FLOWRATE*	100%	90%	80%	70%	TBD	90%	80%
ENTERING LIQUID VAPOR PRESSURE, KN/m <sup>2</sup> (PSIA)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	172.3 (25.0)	101.3 (14.7)

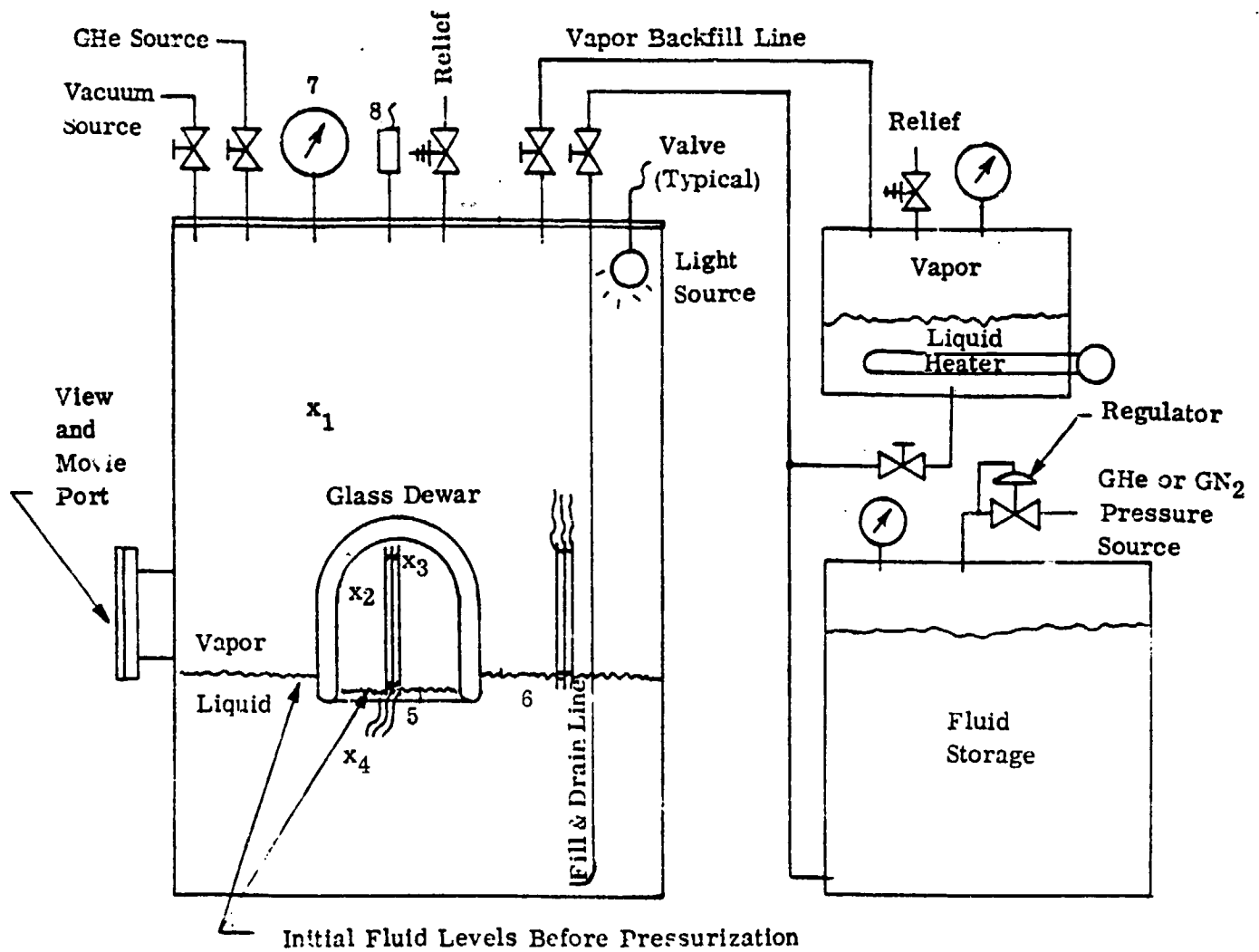
TEST TANK INITIAL PRESSURE IS 101.3 KN/m<sup>2</sup> (14.7 PSIA)

TEST TANK INITIAL TEMPERATURE IS 298.9K (520R) TO 305.6K (550R)

\* 100% FLOWRATE CONDITION IS 1.13 kg/sec (2.5 lb<sub>m</sub>/sec)

Table 6-5. Proposed Test Matrix for Vapor Collapse Tests

CONTROLLED PARAMETER	1	2	3	4	5	7	9
TEST FLUID	LH <sub>2</sub>	LH <sub>2</sub>	LH <sub>2</sub>	LH <sub>2</sub>	FREON MF	FREON MF	FREON MF
INITIAL LIQUID VAPOR PRESSURE, KN/m <sup>2</sup> (psia)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)	172.3 (25)	101.3 (14.7)	101.3 (14.7)	101.3 (14.7)
TANK ΔP INCREASE, KN/m <sup>2</sup> (psia)	34.5 (5)	68.9 (10)	137.9 (20)	68.9 (10)	34.5 (5)	68.9 (10)	137.9 (20)



### INSTRUMENTATION

<u>No.</u>	<u>Description</u>
1	Temperature Sensor, Tank Ullage
2	Temperature Sensor, Dewar Inner Wall
3	Temperature Sensor, Inside Dewar
4	Temperature Sensor, Test Fluid
5	Capacitor Type Liquid Level Sensor, Inside Dewar
6	Capacitor Type Liquid Level Sensor, Outside Dewar
7	Pressure Gage (Typical)
8	Pressure Transducer, Tank Ullage Pressure

Figure 6-5. Typical Vapor Collapse Test System

1. Evacuate the test tank and backfill with GHe to remove moisture and air.
2. Evacuate and backfill with test fluid vapors to remove GHe.
3. Fill the tank with test fluid to the initial level.
4. Allow temperature stabilization.
5. Pressurize the tank to the desired pressure with GHe.
6. Monitor vapor collapse inside glass dewar visually and record liquid level, pressure, and temperature data versus time.

## 6.2 SPACED BASED TESTING

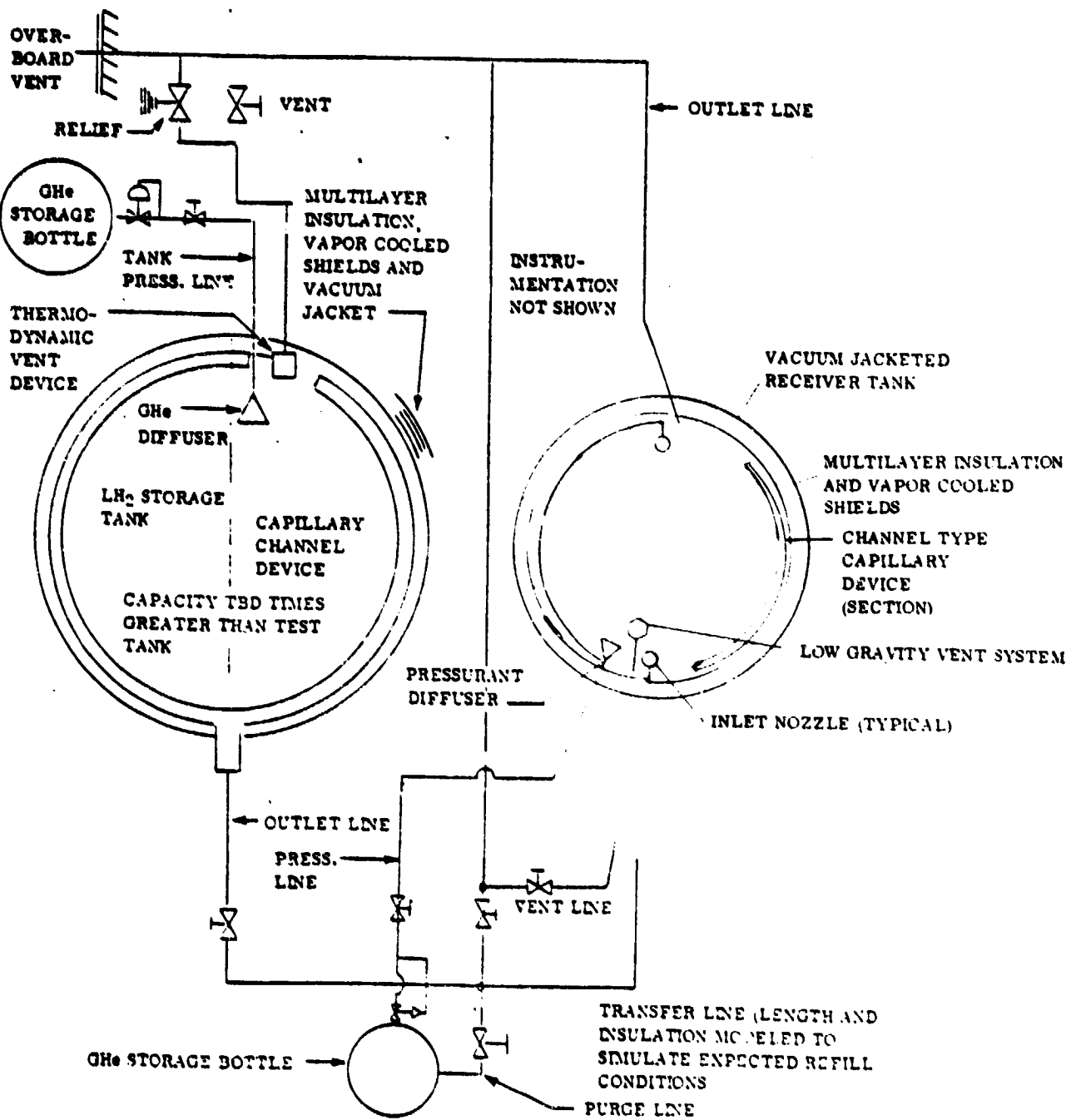
One of the main features of the orbital refilling approach advocated in this study is that the principal features of the refilling process can be verified with ground testing. Thus the benefit of space based testing would be a combination of; demonstration of the complete operation of the system and verifying that the gravity dependence of the processes involved does not effect overall system operation. A spacelab experiment would be ideally suited for obtaining this information. Several receiver tank shapes and sizes should be studied in order to cover the full range of possible orbital refilling applications.

Spacelab refill tests will include the major processes of draining a partially full tank, purging and venting to remove the noncondensable pressurant, prechill or chardown, filling, and vapor collapse using pressurant. Liquid hydrogen is suggested as the test fluid for reasons similar to those given for its selection in Reference 6-1 . The test system would be designed to fit on a single spacelab pallet. Essential elements of the system are shown in Figure 6-6.

The system will consist of a supply tank with a channel type capillary device for acquisition of liquid in low gravity, several receiver tanks with partial (start baskets) or total control (channel) type start baskets, and pressurant storage bottles. Plumbing, instrumentation and telemetry hardware will also be required. The test will be designed as a complete package however, much of the instrumentation such as data acquisition and transmitting systems would be usable for other spacelab experiments. A portion of the experiment package showing only one of the potential receiver tanks is shown in Figure 6-6.

Instrumentation should include:

1. Temperature sensors at selected locations on the test tank walls, in the tank interior, in the inlet line, outlet line, vent line, and screen acquisition device.



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Figure 6-6. Refill Systems Test on Spacelab

2. Flow measurement devices and liquid/vapor sensors in the inlet, outlet, and vent lines.
3. Devices to monitor liquid position in the tank interior.
4. Pressure measurement devices in the test tank, screen acquisition device, inlet line, and outlet line.
5. Acceleration measurement devices on the test tank in three axis.
6. Data acquisition system, preferably with capability of transferring data to a receiving station on earth for analysis between test runs.

Inflow testing would consist of chillover and fill testing for the selected inlet configuration at several flow rates. Pressure, liquid position and temperatures would be measured. If venting is required, vent flow rate and inlet and exit quality from the vent system would be monitored. Each of these tests would be followed by a vapor collapse test conducted by injecting pressurant into the receiver to condense vapor trapped in the start basket. Tests to determine whether helium could be removed from a partially full tank would be conducted with tanks less than 10 percent full. After removing the residual liquid, purging and venting will be undertaken in order to remove the helium.

A complete test series would consist, typically, of draining, purging and venting, chilling, filling, and condensing trapped vapor by pressurizing.

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## APPENDIX A

### HYPRS COMPUTER PROGRAM

The HYPRS computer program was developed with IRAD funds. It is summarized herein to provide a more complete document of the study results.

The program models the filling of a tank with a liquid propellant which enters as a uniform spray. Heat and mass exchange between liquid and vapor and between the propellant and the tank walls, and the resulting temperatures and pressures are calculated. The model can also include a fixed mass of helium pressurant.

The program is diagrammed in Figure A-1. The main program, HYPRS, reads data and initializes variables. DIFE3 calls DERIV, which contains the computational routines, and CONTROL, which controls output; it also integrates the time dependent variables, using values passed from DERIV. In addition, propellant properties are accessed as Block Data.

A more detailed flowchart of subroutine DERIV is shown in Figure A-2. Subroutine VAPOR initializes mass, pressure, and temperature, and PROP computes vapor and liquid properties. The other routines handle some aspect of the system heat and mass transfer. The model can be broken into four major segments, roughly chronological

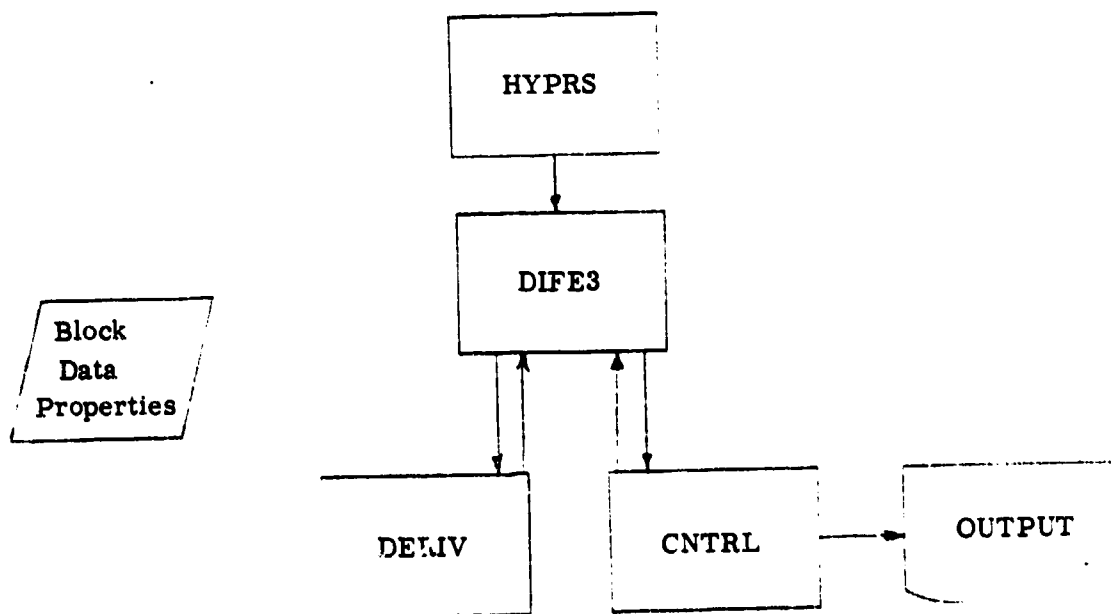
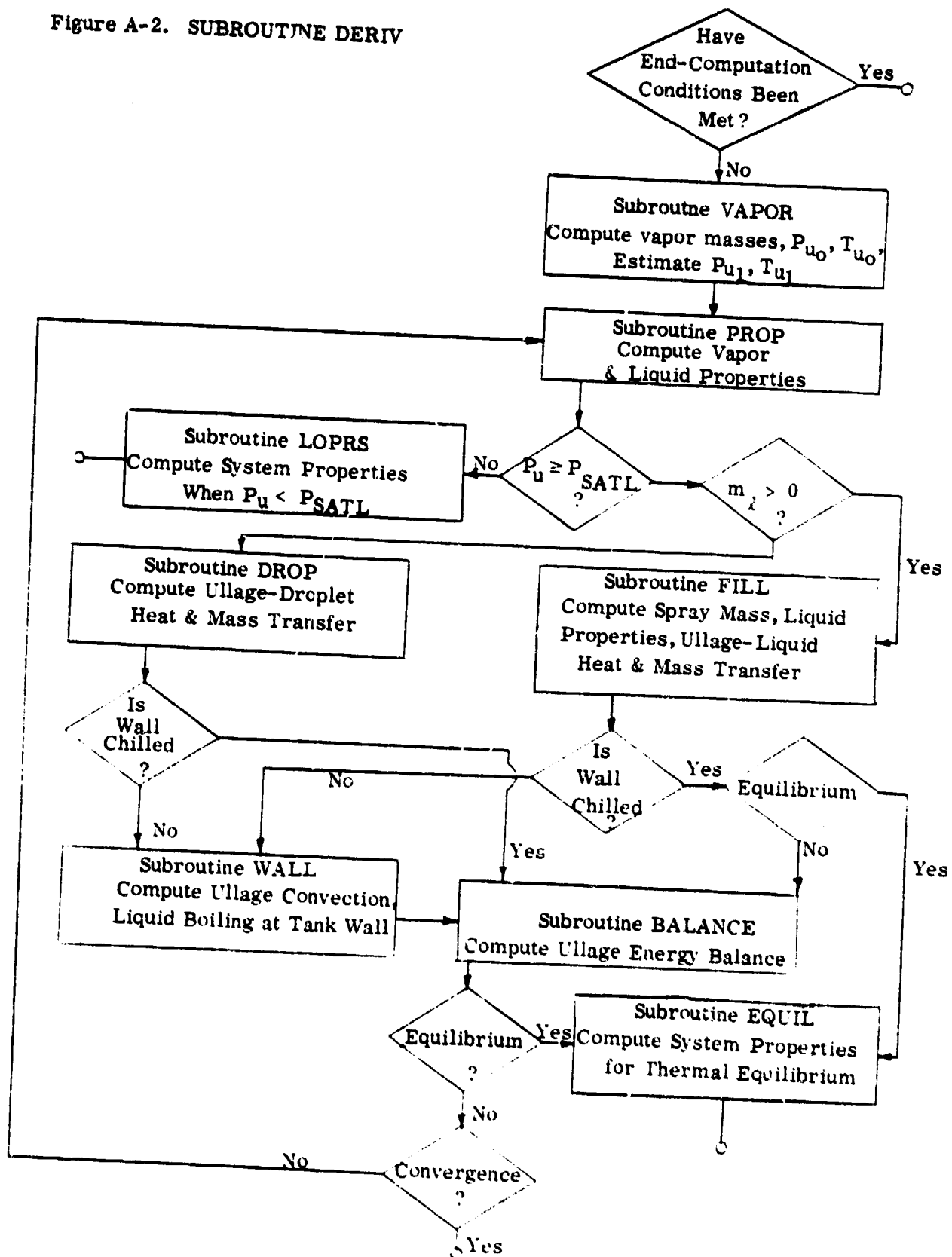


Figure A-1. Computer Program HYPRS

Figure A-2. SUBROUTINE DERIV





(corresponding to) subroutines LOPRS, DROP, FILL, and EQUIL. Subroutine LOPRS is called when ullage pressure is less than incoming liquid vapor pressure. This would correspond to liquid being sprayed into an almost-evacuated tank. In this case it is assumed that all incoming liquid is evaporated, and that all heat for evaporation comes from the walls. When ullage pressure is greater than or equal to incoming liquid vapor pressure, but no liquid has accumulated, subroutine DROP is called. The routine uses incoming liquid mass, velocity and properties, and computes forced convection from the ullage to uniform, spherical droplets. When liquid begins to accumulate, subroutine FILL is called. This subroutine uses the same equations to model vapor-liquid heat and mass transfer, however liquid properties are assumed to be the result of perfect mixing between the incoming spray and the liquid residing in the tank, and the volume of liquid which resides as a spray is assumed to be limited to a percentage of the ullage volume. During these two phases of the model, two other subroutines, WALL and BALANCE, may be called. If the wall temperature is above liquid saturation temperature, subroutine WALL is called. This routine determines the area of dry and wetted wall, and computes convective heat transfer from the dry wall area to the ullage, and boiling heat transfer across the wetted area. Once the wall has reached liquid saturation temperature, it is assumed that virtually all the heat has been extracted from the wall, and the subroutine is henceforth by-passed. If the system has not reached thermodynamic equilibrium, the energy balance is made by calling subroutine BALANCE. This subroutine computes the ullage energy and mass, and compares the pressure and temperature derived from these quantities with the pressure and temperature initially estimated and used throughout the calculations. If the comparison is not acceptably close, the pressure and temperature estimates are refined, and subroutine PROP is called again, to recompute properties, and start the computational cycle over again. The fourth major model segment is contained in subroutine EQUIL, when the system is judged to have reached thermal equilibrium. This would correspond to the phase of tank fill during which vapor is entrained in the liquid, and incoming spray keeps the system well mixed.

This subroutine supersedes all previous ones; it computes vapor-liquid heat and mass transfer, ullage energy balance, and checks for convergence. In the following sections, the four major model segments and subroutines WALL and BALANCE are presented in greater detail.

#### A.1 SUBROUTINE LOPRS

If liquid vapor pressure exceeds ullage pressure during the initial period of liquid flow into the tank, subroutine LOPRS is called. This subroutine assumes that a small quantity of liquid will instantaneously evaporate as drops pass through the ullage, and the remaining liquid will be chilled in the process. This subcooled liquid will strike the tank and evaporate as a result of heat exchange with the walls.

This process has been simplified by assuming that all liquid evaporated is a result of heat exchange with the tank walls and, the evaporated liquid enters the ullage as a saturated vapor. For these assumptions, the rate at which heat is extracted from the wall is:

$$\dot{Q}_{\text{wall}} = \dot{m}_l \times h_{fg} \quad (\text{A-1})$$

where:

$\dot{Q}_{\text{wall}}$  = rate of heat transfer from the wall

$\dot{m}_l$  = liquid mass inflow rate

$h_{fg}$  = latent heat of evaporation, at incoming liquid conditions

Because all liquid is assumed to be evaporated, the rate at which liquid is added to the ullage is equal to the liquid inflow rate, that is

$$\dot{m}_v = \dot{m}_l \quad (\text{A-2})$$

and the rate at which energy is added to the ullage is then

$$\dot{E}_{\text{ull}} = \dot{m}_l \times h_{v\text{-sat}} \quad (\text{A-3})$$

where:

$\dot{E}_{\text{ull}}$  = rate of energy addition to the ullage

$h_{v\text{-sat}}$  = enthalpy of saturated vapor

An ullage pressure,  $P_u$ , is estimated, and the ullage temperature,  $T_u$  is looked up as a function of the estimated pressures and computed energy.  $P_u$  is then compared to  $P_u'$ , ullage pressure computed using the equation of state

$$P_u' = (RT_u \dot{m}_v) / V_u \quad (\text{A-4})$$

where:

- R = gas constant
- $m_v$  = vapor mass
- $V_u$  = ullage volume

If the two pressures are not acceptably close, the pressure estimate is revised and the process re-iterated until convergence is achieved.

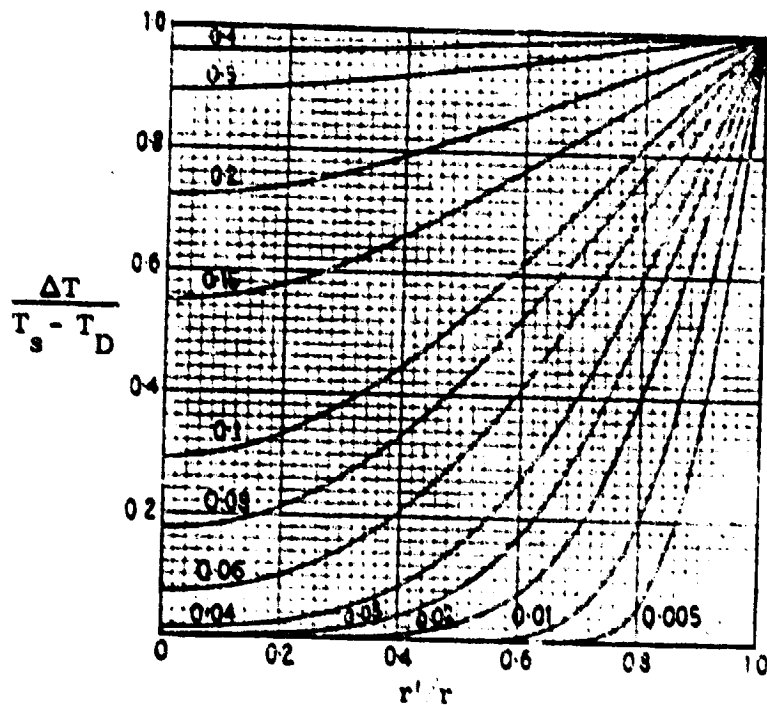
## A.2 SUBROUTINE DROP

When ullage pressure is at least as great as liquid vapor pressure, calculations are made on the basis of a typical or average spherical droplet. Subroutine DROP incorporates this model for the case of no accumulated liquid. The typical droplet is assumed to be uniformly at temperature  $T_D$ , except for a surface layer of negligible thickness at temperature  $T_s$ , liquid saturation temperature corresponding to ullage pressure. Further, the droplet is assumed to move at constant velocity,  $v$ , the liquid inflow velocity. Heat transfer is modelled as forced convection from the ullage to the droplet, and conduction from the surface to the interior of the droplet. The convective heat transfer coefficient is given by

$$\bar{h}_D = \left( \frac{k_v}{D} \right) \left( 2.0 + 0.6 \left( \frac{Dv}{u_v \nu_v} \right)^{1/2} \right) \left( \frac{Cp_v u_v}{k_v} \right)^{1/3} \quad (\text{Ref A-1}) \quad (\text{A-5})$$

where:

- $k_v$  = thermal conductivity
- D = average droplet diameter
- v = incoming liquid velocity
- $u_v$  = specific volume of vapor
- $\nu_v$  = viscosity of vapor
- $Cp_v$  = specific heat at constant pressure of vapor



Numbers on curves  
are values of  $x$ .  $r'$   
is distance from  
droplet center

Figure A-3. Temperature Distribution within the Droplet.

The total rate at which heat is convected to the droplets is then

$$\dot{Q}_D = \bar{h}_D \times A_D \times (T_n - T_s) \times n \quad (A-6)$$

where:

$A_D$  = average droplet surface area

$n$  = number of droplets in the tank during the computation interval.

The average temperature change within the droplet due to conduction from the surface to the interior of the droplet is modeled as a function of the dimensionless parameter

$$x = \frac{k_l \times \Delta t \times v_l}{c_l \times r^2} \quad (A-7)$$

where:

$k_l$  = thermal conductivity of liquid

$\Delta t$  = computation time interval

$v_l$  = specific volume of liquid

$c_l$  = specific heat of droplet

$r$  = average droplet radius

This function, represented here as  $f(x)$ , is incorporated into the program as a polynomial curve fit to the curves derived in Reference A-2, and shown in Figure A-3.

The average droplet temperature change is given by

$$\Delta \bar{T} = (T_g - T_D) \times f(x), \quad (A-8)$$

and the total rate at which heat is conducted within the droplets is:

$$\dot{Q}_{KD} = \dot{m}_l \times c_l \times \Delta \bar{T} \quad (A-9)$$

Evaporation is assumed to occur when  $\dot{Q}_D$  exceeds  $\dot{Q}_{KD}$ ; in this case the evaporation rate,  $\dot{m}_D$  is:

$$\dot{m}_D = \frac{\dot{Q}_D - \dot{Q}_{KD}}{(h_{fg} + h_s - h_D)} \quad (A-10)$$

where:

$h_{fg}$  = latent heat of evaporation at ullage pressure

$h_s$  = enthalpy of liquid saturated at ullage pressure

$h_D$  = incoming droplet enthalpy

If the droplets are not totally evaporated, the change in energy of the remaining liquid is given by

$$\Delta u_D = \frac{\dot{Q}_{KD} - (\dot{Q}_{KD} - \dot{Q}_D) \left( \frac{h_s - h_D}{h_{fg}} \right)}{(\dot{m}_l - \dot{m}_D)} \quad (A-11)$$

If, on the other hand,  $\dot{Q}_D$  is less than or equal to  $\dot{Q}_{KD}$ , condensation occurs, and the condensation rate,  $\dot{m}_D$ , is expressed as

$$\dot{m}_D = \frac{\dot{Q}_D - \dot{Q}_{KD}}{h_{fg}}, \quad (A-12)$$

while the change in droplet energy is

$$\Delta u_D = \frac{\dot{Q}_D + (\dot{Q}_{KD} - \dot{Q}_D) \left( \frac{h_s - h_o}{h_{fg}} \right)}{(\dot{m}_l - \dot{m}_D)} \quad (A-13)$$

### A.3 SUBROUTINE WALL

Any liquid not evaporated in the ullage then strikes the tank wall. If the wall temperature is greater than liquid saturation temperature, the droplets will splatter and boil. If the droplet mass is not sufficient to completely cover the wall, convection from the walls to the ullage will take place over the unwetted area. This is modelled in subroutine WALL. The rate of boiling heat transfer,  $\dot{Q}_W$ , is given by the equation.

$$\dot{Q}_W = 7.345 \times 10^{-5} \left( h_{fg} + C_l \left( \frac{T_w - T_s}{2} \right) \right) \left( \frac{r \times u_v \times V^2}{\sigma_l \times v_l^2} \right)^{.341} (\dot{m}_l - \dot{m}_D) \quad (A-14)$$

where:  $T_w$  = wall temperature  
 $\sigma_l$  = liquid surface tension  
 $v_l$  = specific volume of liquid

The convective heat transfer coefficient, wall to ullage is

$$\bar{h}_w = \left( \frac{.13 C_p}{u_v} \right) \left( \frac{\dot{m}_l \mu_v (v_v V)^{1.2}}{V_T \times \text{eff}} \right)^{1.4} / \left( \frac{C_p u_v}{k_T} \right) \quad (A-15)$$

where:

$V_T$  = tank volume

$\text{eff}$  = is defined by equation 3-10, and is assumed to be 0.40 for this study

resulting in a heat transfer rate from wall to ullage of

$$\dot{Q}_{WG} = \bar{h}_w \times (T_w - T_u) \times A_w \quad (\text{A-16})$$

where:

$A_w$  = area of dry wall

The total rate at which heat is transferred from the wall,  $\dot{Q}_{WT}$  is the sum of  $\dot{Q}_W$  and  $\dot{Q}_{WG}$ , and the rate of evaporation at the wall,  $\dot{m}_w$ , is given as

$$\frac{\dot{Q}_w}{h_f - h_s + C_l (T_s - T_D)} \quad (\text{A-17})$$

where:

$h_f$  = enthalpy of liquid evaluated at film temperature,  $T_f$

$T_f = (T_w + T_s)/2$

By the time the wall temperature reaches liquid saturation temperature, virtually all the energy has been extracted from it, so after that point, the wall calculations are bypassed.

#### A.4 SUBROUTINE FILL

When liquid begins to accumulate in the tank subroutine FILL is called and the heat and mass transfer between liquid droplets and the ullage is calculated in the same way, but the liquid properties are assumed to be the result of perfect mixing between incoming liquid, and liquid residing in the tank. Further, the mass of droplets is no longer computed as the mass of liquid entering during the computation

interval, but rather as a function of the total liquid mass in the tank and the ullage volume. It is assumed that the only significant heat and mass transfer between vapor and liquid is that which occurs between the ullage and that liquid which exists as a spray. The internal energy of the mixed liquid,  $u_l$ , is

$$u_l = u_l + (\dot{m}_l \Delta t (h_D - u_l) / m_l) \quad (A-18)$$

where:

$u_l$  = internal energy of liquid before this time step

$m_l$  = total mass of liquid - that which resided in the tank

at the previous time step, plus the current inflow.

The mass of liquid existing as a spray,  $m_s$ , is defined as

$$m_s = \left[ \left( \frac{p}{1-p} \right) \times V_n \right] / v_l \quad (A-19)$$

where:

$V_n$  = ullage volume

$p$  = droplet packing factor-that percentage of tank volume not occupied by bulk liquid which can be occupied by liquid spray. For simulations run for this study a  $p$  of 0.02 was assumed.

Using liquid properties based on  $u_l$ , and the spray mass,  $m_s$ , heat transfer rates, liquid evaporation or condensation rates, and the resulting liquid temperature and energy can be computed. If liquid temperature and energy exceed or are equal to that of liquid saturated at tank pressure, it is assumed that the tank contents have reached thermal equilibrium. If this is not the case, a different path is followed, subroutine BALANCE is called and the ullage energy balance is made using the mass and energy terms previously calculated.

#### A.5 SUBROUTINE BALANCE

When equilibrium has not been reached the change in ullage energy is expressed as the sum of the heat transferred to or from it, the product of the mass condensed or evaporated times the appropriate enthalpy, plus the change in ullage volume times pressure. That is,

$$\Delta E_u = \Sigma \dot{Q} \Delta t + \Sigma \dot{m} h \Delta t + P \Delta V_u \quad (A-20)$$



where  $h$  is equal to the enthalpy of saturated liquid if condensation occurs and the enthalpy of saturated vapor if evaporation occurs. The change in ullage volume is given by

$$\Delta V_u = (\dot{m}_l - \dot{m}_D - \dot{m}_w) \times \Delta t \times v_l \quad (\text{A-21})$$

and the new ullage mass,  $m_{u1}$ , is given by

$$m_{u1} = m_u + (\dot{m}_D + \dot{m}_u) \Delta t \quad (\text{A-22})$$

where:

$m_u$  = ullage mass at the previous time step

Using these quantities a check is made to see if they are in agreement with the estimated values of pressure and temperature used in the computations. If the agreement is acceptable, end-of-time step values are tabulated and printed, otherwise, the estimates of pressure and temperature are refined and the process repeated.

#### A.6 SUBROUTINE EQUIL

When thermal equilibrium has been reached, calculations of mass and energy balance made in subroutine EQUIL supersede previous calculations. Total mass,  $m_T$ , and energy,  $U_T$ , are computed

$$m_T = m_l + m_v \quad (\text{A-23})$$

$$U_T = (m_l \times u_l) + (m_v \times u_v) + \dot{Q}_{WT} \Delta T \quad (\text{A-24})$$

where  $m_l$ , the liquid mass includes both liquid inflow, and that mass residing in the tank at the beginning of the time step, and  $u_l$ , the specific energy, is the result of perfect mixing. The vapor internal energy,  $u_v$ , is that corresponding to the beginning of the time step. A system temperature is estimated, and pressure, energy, and system quality—and, hence, masses — are computed. If the energy and mass do not match  $U_T$  and  $m_T$ , the temperature estimate is refined and the process is repeated until convergence is achieved.

The program user can specify a number of different termination criteria —percentage fill, liquid mass, wall chill-down, or time—and can also specify through input the tank size and material, and the specific propellant of interest.

A-1 Bird, R. B., Stewart, W. E., and Lightfoot, E. N., Transport Phenomenon, John Wiley and Sons, Inc., New York, 1960.

A-2 Carslow, H. S. and Jaeger, J. C., Conduction of Heat in Solids, Oxford University Press, Oxford, 1947.

## APPENDIX B

### PRE-CHILL OF CONTAINER WITH COLD GASEOUS HYDROGEN (GH<sub>2</sub>)

A schematic of prechill with GH<sub>2</sub> is shown in Figure B-1. Cold hydrogen will flow in at constant total enthalpy (constant temperature and pressure). Heat transfer from the tank wall to the GH<sub>2</sub> will cool the tank wall and heat up the GH<sub>2</sub>. When the wall temperature approaches GH<sub>2</sub> temperature, tank venting will occur to expel the high vapor. This fill and vent process will be repeated until the tank wall is chilled to the desired temperature. The time rate of change of energy being stored in the GH<sub>2</sub> is equal to the time rate of energy being input during inflow or taken out during venting.

$$\frac{mV^2}{2gJ} = \dot{H} + \dot{Q} - \dot{U} \quad (B-1)$$

$$\dot{Q} = HT \cdot A_W \cdot (T_W - T_G) \quad (B-2)$$

$$W_W C_W \dot{T}_W = -\dot{Q} \quad (B-3)$$

$$H V > 0$$

$$HT = C_1 \cdot \rho \cdot C_p \left[ \left( (mV^2 \text{ vol} \cdot u \cdot \rho^2) \right)^{1/4} \right] N_{PR}^{2/3} \quad (B-4)$$

$$H V = 0$$

$$HT = 2 \cdot CL \cdot K \cdot (AL)^{1/3} \cdot X \quad (B-4a)$$

$$AL = G \cdot X^3 \cdot \rho^2 \cdot C_p \cdot (T_W - T_G) - (T_G \cdot u \cdot K)$$

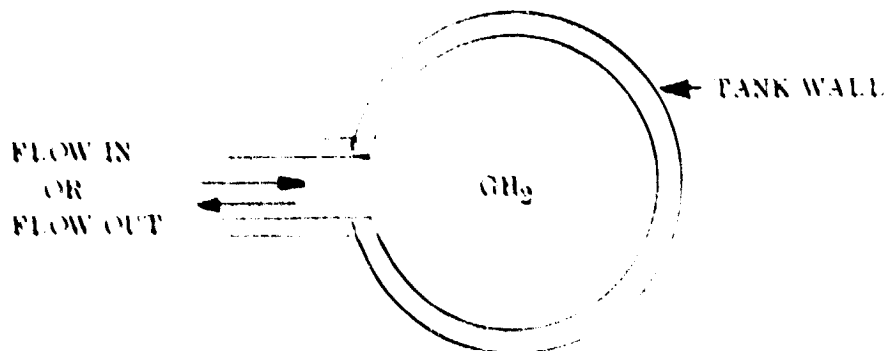


Figure B-1. Prechill Schematic

$$\text{If } AL < 1.E09 \text{ CL} = .55$$

$$\text{If } AL > 1.E09 \text{ CL} = .13$$

$$P = f(\rho_1 T) \quad (B-5)$$

$$\dot{H} = \frac{d(mh)}{dt} = m\dot{h} + h\dot{m} \quad (B-6)$$

$$\dot{U} = m\dot{u} + u\dot{m} \quad (B-7)$$

$$\dot{H}_{\text{total}} = \frac{\dot{m}V^2}{2gJ} + \dot{H} \quad (B-8)$$

where

$\dot{m}$  = mass flow rate

$V$  = flow velocity

$g$  = standard sea level acceleration

$J$  = mechanical equivalent of heat

$m$  = mass

$h$  = specific enthalpy

$\dot{h}$  = time rate of change of  $h$

$u$  = internal energy

$\dot{u}$  = time rate of change of  $u$

$\dot{Q}$  = heating rate

$P$  = gas pressure

$VOL$  = volume of container

$T_g$  = container wall temperature

$\rho$  = gas density

$A_w$  = heat transfer surface

$W_w$  = mass of container

$T_w$  = container wall temperature

$\dot{T}_w$  = time rate of change of  $T_w$

$N_{PR}$  = Prandtl number

$\mu$  = viscosity

$C_p$  = constant pressure specific heat of  $\text{GH}_2$

$C_w$  = specific heat of container

$G$  = acceleration

$X$  = characteristic dimension

$K$  = thermal conductivity

Equations B-1 through B-8 were used to determine the time to prechill a container to a specific temperature using cold  $\text{GH}_2$ . The simultaneous solution was performed using a digital computer program. The program was written in a series of subroutines.

Subroutine "PRESUR" is the principal subroutine to be called by the main program "HYPRES." Figure B-2 is a flow chart for "PRESUR."

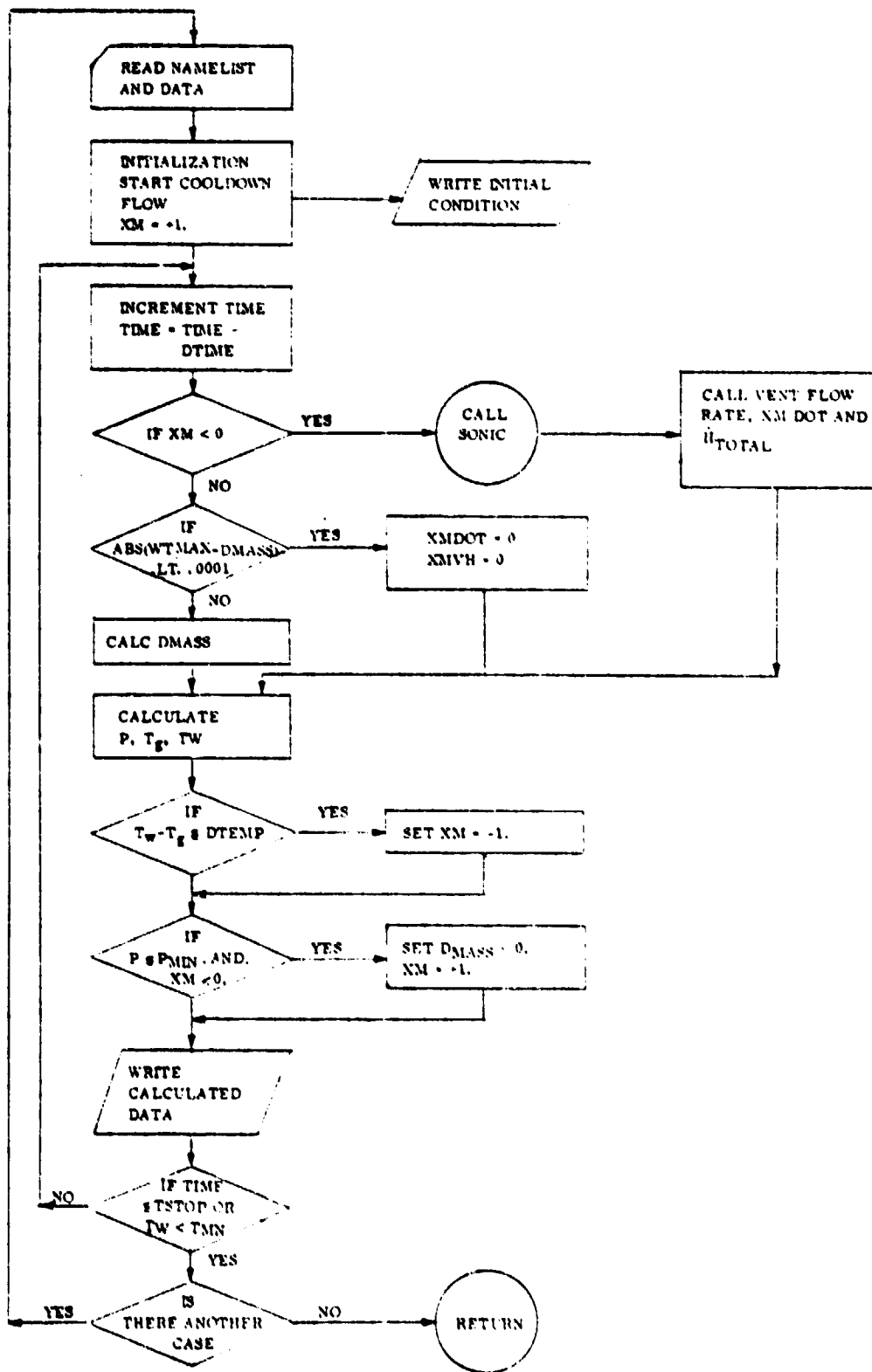


Figure B-2. Flow Chart for Subroutine PRESUR